



1. NASA
CR
128498

01-128498

3.

GRUMMAN AEROSPACE CORP.



N72-30902

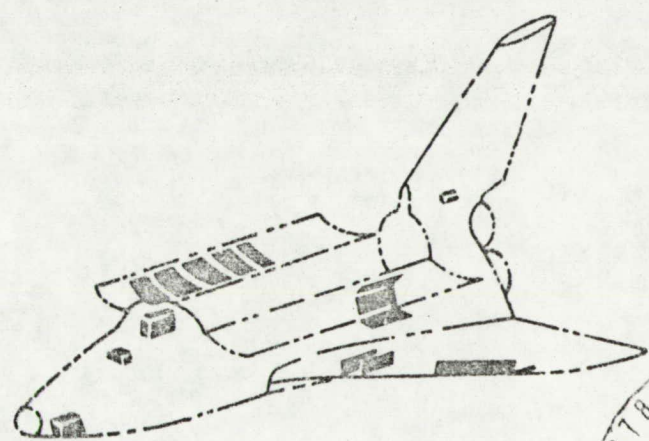
Unclass
39545

4.

SPACE SHUTTLE ORBITER HEAT PIPE APPLICATIONS

VOLUME 2 FINAL REPORT

(NASA-CR-128498) SPACE SHUTTLE ORBITER
HEAT PIPE APPLICATIONS. VOLUME 2: FINAL
REPORT. FINAL REPORT J.F. Alario, et al
(Grumman Aerospace Corp.) Apr. 1972. 381 p.
CSCL 22E G3/31



**SPACE SHUTTLE ORBITER
HEAT PIPE APPLICATIONS**

**VOLUME 2
FINAL REPORT**

Prepared By:

J.P. Alario
R.C. Prager

Approved By:

M.N. Tawil
Grumman Aerospace Corp.
R. Bullock
NASA/MS

Contract **NAS 9-12034**

Prepared For

National Aeronautics and Space Administration
Manned Spacecraft Center
Houston, Texas 77058

By

Grumman Aerospace Corporation
Bethpage, New York 11714

April 1972

SPR - 14

PRECEDING PAGE BLANK NOT FILMED

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.

Major contributors to the study were:

Thermal Analysis: J. Alario
P. Dominguez
D. Lehrfeld
R. Prager
M. Tawil

Weights Analysis: J. Sims

Structural Design: J. Fiorello
J. Valentine

PRECEDING PAGE BLANK NOT FILMED

CONTENTS

<u>Section</u>	<u>Page</u>
1 ABSTRACT	1-1
2 INTRODUCTION	2-1
3 SUMMARY	3-1
4 IDENTIFICATION/PRELIMINARY EVALUATION	4-5
5 PRELIMINARY DESIGN STUDIES	5-1
6 STUDY PLANS	6-1
7 GENERAL DESIGN CONCEPT	7-1
8 FINAL EVALUATION	8-1
9 DETAILED DESIGN & ANALYSIS	9-1
9-1 HEAT PIPE AUGMENTED COLD RAIL	9-3
9-2 AVIONICS HEAT PIPE CIRCUIT	9-23
9-3 HEAT PIPE/PHASE CHANGE MATERIAL MODULAR SINK	9-43
9-4 AIR-TO-HEAT-PIPE HEAT EXCHANGER	9-77
9-5 HEAT PIPE RADIATOR FOR COMPARTMENT TEMPERATURE CONTROL	9-93
10 CONCLUSIONS	10-1
11 RECOMMENDATIONS	11-1
12 REFERENCE	12-1

PENDICES

	<u>Page</u>
-1 IDENTIFICATION SHEETS FOR PROPOSED APPLICATIONS	A-1
-2 LEADING EDGE HEAT PIPE APPLICATIONS	A-35
-3 SLA BACKFACE TEMPERATURE CONTROL USING HEAT PIPES	A-39
-4 HEAT PIPE APPLICATIONS FOR HYPERGOLIC FUELED APU	A-45
-5 HEAT PIPE APPLICATIONS DESCRIPTIONS	A-51
B-1 HEAT PIPE JOINING TECHNIQUES	B-1
B-2 DESIGN CURVES FOR SELF-PRIMING HIGH CAPACITY HEAT PIPES	B-11
C-1 HEAT PIPE APPLICATIONS EVALUATION SHEETS	C-1
C-2 HEAT PIPE RADIATOR FOR WASTE HEAT REJECTION	C-13
D-1 FLANGE CONDUCTANCE STUDY FOR HP AUGMENTED COLD RAIL	D-1
D-2 COMPUTER SUBROUTINE, HP AUGMENTED COLD RAIL	D-4
D-3 HEAT TRANSFER FILM COEFFICIENTS	D-7
E AVIONICS HEAT PIPE CIRCUIT ANALYSIS DETAIL	E-1
F-1 SYSTEM HEAT LEAKS	F-1
F-2 PCM CONTAINER ANALYSIS	F-5
G DETAILED DESIGN CALCULATIONS FOR THE AVIONICS RACK HEAT EXCHANGER	G-1



ILLUSTRATIONS

FIGURE NO.		PAGE
4-1	INERTIA FORCE VARIATION DURING SHUTTLE MISSION	4-6
5-1	TPS LEADING EDGE (SPL-114)	5-6
5-2	HP RADIATOR APPLICATION, MAIN LANDING GEAR COMPARTMENT (SPL-103)	5-14
5-3	HEAT PIPE/HEAT TRANSPORT SYSTEM (SPL-102A) HEAT PIPE CIRCUIT (SPL-102B)	5-19 5-20
5-4	HP/ABSORBER MODULAR SINK (SPL-112)	5-22
5-5	AIR-TO-HEAT PIPE HEAT EXCHANGER (SPL-105)	5-27
5-6	FLIGHT VOICE/DATA RECORDERS (SPL-111)	5-31
5-7	HEAT PIPE RADIATORS, PAYLOAD DOORS (SPL-101)	5-37
5-8	HEAT PIPE RADIATOR WITH HP FLUID HEADER (SPL-113)	5-44
5-9	HIGH CAPACITY COLD RAIL (SPL-104)	5-50
5-10	FUEL CELL RADIATORS (SPL-110)	5-52
5-11	HEAT PIPE RADIATORS, AIR BREATHING ENGINE COMPARTMENT	5-58
7-1	AMMONIA/FREON TWO-FLUID WICK	7-8
7-2	WATER/FREON TWO-FLUID WICK	7-9
8-1	EVALUATION MATRIX	8-1
9-1.1	FLUID FLOW NETWORK	9-4
9-1.2	CROSS SECTION OF COLD RAIL THERMAL MODEL	9-6
9-1.3	TEMPERATURE DISTRIBUTION ALONG COLD RAIL	9-10
9-1.4	TEMPERATURE DISTRIBUTION ALONG HP AUGMENTED COLD RAIL	9-10
9-1.5	TEST SCHEMATIC	9-14
9-1.6	INSTRUMENTATION	9-15
9-1.7	BASILINE TEST SETUP	9-18
9-1.8	AUGMENTED COLD RAIL, DETAILED DESIGN DRAWING (SPL-104)	9-21
9-2.1	HEAT PIPE AVIONICS CIRCUIT SCHEMATIC	9-26
9-2.2	SYSTEM CAPACITY VS. FLOW RATE	9-27
9-2.3	LOCAL MAX LOAD VS. TOTAL LOAD ON A COLD RAIL	9-28
9-2.4	HP CIRCUIT LAYOUT WITH GRAVITY VECTORS	9-33
9-2.5	HP CIRCUIT TEST, ANGLE CONFIGURATION	9-34
9-2.6	BENCH TEST ARRANGEMENT - HEADER PIPE	9-36
9-2.7	BENCH TEST ARRANGEMENT - COLD RAIL PIPE	9-36
9-2.8	HEAT PIPE CIRCUIT, DETAILED DESIGN DRAWING (SPL-102B)	9-41
9-3.1	MODULAR HEAT SINK	9-44
9-3.2	TEMPERATURE RESPONSE OF STRUCTURE	9-46
9-3.3	TRANSPORT HEAT PIPE CAPACITY	9-51
9-3.4	DIODE HEAT PIPE CAPACITY	9-52
9-3.5	HEAT PIPE COLD PLATE MODEL	9-55
9-3.6	EFFECTIVE CONDUCTANCE	9-58
9-3.7	WEIGHTS - PCM & FINS	9-59
9-3.8	CONDUCTANCE PER UNIT WEIGHT	9-60
9-3.9	WEIGHTS - DIRECT PCM ATTACHMENT TO PLATE	9-61

ILLUSTRATIONS
(CONTINUED)

FIGURE NO.		PAGE
9-3.10	TOTAL CONDUCTANCE	9-62
9-3.11	HONEYCOMB CORE/PCM PERFORMANCE	9-63
9-3.12	MODULAR HEAT SINK PERFORMANCE	9-65
9-3.13	TEST CONFIGURATION	9-70
9-3.14	MODULAR HEAT SINK, DETAILED DESIGN DRAWING (SPL-111)	9-75
9-4.1	AVIONICS RACK SCHEMATIC	9-78
9-4.2	HEAT EXCHANGER SCHEMATIC	9-79
9-4.3	HEAT PIPE ΔT VS. NUMBER OF PIPES AND LENGTH	9-83
9-4.4	EVAPORATOR HEAT FLUX VS. NUMBER OF PIPES AND LENGTH	9-83
9-4.5	AVIONICS RACK HP HEAT EXCHANGER	9-86
9-4.6	SQUARE HEAT PIPE FORMING	9-87
9-4.7	ARTERY DETAIL	9-87
9-4.8	HEAT PIPE EXCHANGER FOR AIR COOLED EQUIPMENT, DESIGN LAYOUT (SPL-105)	9-90
9-5.1	MAIN LANDING GEAR COMPARTMENT RADIATOR SYSTEM	9-95
9-5.2	MLG COMPARTMENT IDEALIZATION	9-97
9-5.3	ORBITAL ENVIRONMENT	9-99
9-5.4	AVERAGE LANDING TEMPERATURE VS. INSULATION THICKNESS AND COMPARTMENT RADIATOR HEAT INPUT	9-100
9-5.5	LANDING GEAR TEMPERATURE VS. RADIATOR SURFACE HEAT INPUT (SOLAR INERTIAL ORBIT)	9-101
9-5.6	LANDING GEAR TEMPERATURE VS. RADIATOR SURFACE HEAT INPUT (EARTH ORIENTED ORBIT)	9-102
9-5.7	HP RADIATOR SYSTEM SCHEMATIC	9-103
9-5.8	LANDING GEAR COMPARTMENT EFFECTIVE SINK TEMPERATURE	9-107
9-5.9	RADIATOR SYSTEM ANALYSIS PROCEDURE	9-108
9-5.10	RADIATOR FIN EFFECTIVENESS	9-109
9-5.11	COMPARTMENT RADIATOR FIN EFFECTIVENESS	9-111
9-5.12	HEAT PIPE TRANSPORT CAPACITIES	9-112
9-5.13	HEAT EXCHANGER PERFORMANCE CURVES	9-114
9-5.14	HP RADIATOR SYSTEM PERFORMANCE	9-115
9-5.15	HEAT PIPE RADIATOR FOR MAIN LANDING GEAR, DESIGN LAYOUT DRAWING (SPL-103)	9-124

TABLES

<u>No.</u>		<u>Page</u>
3-1	SHUTTLE HEAT PIPE APPLICATIONS EVALUATION	3-3
4-1	GAC SUBSYSTEM DEFINITIONS	4-2
4-2	TYPICAL HEAT SOURCES ON THE SPACE SHUTTLE	4-4
4-3	TYPICAL SPACE SHUTTLE HEAT SINKS	4-5
4-4	SHUTTLE HEAT PIPE APPLICATIONS CANDIDATES	4-8
5-1	HIGH TEMPERATURE HEAT PIPE PERFORMANCE DATA	5-5
5-2	HEAT PIPE ROSTER - MAIN LANDING GEAR WHEEL WELL SYSTEM	5-13
5-3	HEAT PIPE ROSTER - HEAT PIPE HEAT TRANSPORT CIRCUIT	5-18
5-7	HEAT PIPE ROSTER FOR RADIATOR	5-36
5-8	ROSTER FOR RADIATOR WITH HP/FLUID HEADER	5-43
5-11	HEAT PIPE ROSTER - AIR BREATHING ENGINE COMPARTMENT	5-57
7-1	HEAT PIPE ROSTER (BY SHAPE)	7-2
7-2	SUBASSEMBLY LISTING FOR TABLE 7-1	7-5
7-3	HEAT PIPE WORKING FLUID/WICK DESIGN INTERCHANGEABLE	7-7
8-1	SCORING OF SAFETY FACTORS	8-3
8-2	SHUTTLE HEAT PIPE APPLICATIONS - SUMMARY	8-9
8-3	SHUTTLE HEAT PIPE APPLICATIONS FOR DETAILED DESIGN AND ANALYSIS	8-10
9-1.1	AUGMENTED COLD RAIL RACK WEIGHTS	9-11
9-1.2	TEST SCHEDULE	9-17
9-2.1	HEAT PIPE CIRCUIT AVIONICS EQUIPMENT RACK WEIGHTS	9-30
9-2.2	TEST LOADING CONDITIONS	9-38
9-3.1	DESIGN PARAMETERS, MODULAR HEAT SINK	9-48
9-3.2	PHASE CHANGE MATERIALS (PCM)	9-50
9-3.3	MODULAR HEAT SINK FLIGHT VOICE RECORDER APPLICATION WEIGHTS	9-66
9-3.4	TEST PLAN	9-72
9-4.1	HEAT PIPE HEAT EXCHANGER WEIGHTS	9-89
9-5.1	NOMENCLATURE, COMPARTMENT RADIATOR SYSTEM	9-104
9-5.2	WEIGHTS: HEAT PIPE RADIATOR APPLICATION, MAIN LANDING GEAR COMPARTMENT	9-117

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 radiating panel 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.

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 which 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 systems.

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 temperatures 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.

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

Table 3-1 Shuttle Heat Pipe Applications Evaluation

<u>Original 27 Candidates</u>	<u>11 Prelim. Design Study</u>	<u>6 Detail Design</u>
<ol style="list-style-type: none"> 1. TPS leading edge 2. Landing gear 3. Avionics HP circuit 4. Modular sinks 5. ATR equipment 6. Flight and voice recorders 7. HP radiator W/HP header 8. HP radiator with integral HP/ fluid header 9. ECS cold rail 10. HP radiator for fuel cell 11. Air breathing engine compartment 12. OMS LH₂ boiloff 13. High intensity lights 14. Battery 15. Tracking radar 16. Fluid evaporator 17. Fuselage TJS, interference heating 18. TPS panel 19. Control surface pivots 20. OMS LO₂ boiloff 21. Main LO₂ tank boiloff 22. C-Band directional antenna 23. Electrical wiring 24. Hydraulic actuators 25. APU 26. LO₂ natural recirculating system 27. Water chiller 	<p>TPS leading edge Landing gear Avionics HP circuit Modular sinks ATR equipment Flight and voice recorders HP radiator with HP header HP radiator with integral HP/Fluid header HP Augmented cold rail (High capacity) HP radiator for fuel cell Air breathing engine compartment</p>	<p>HP radiator for compartments HP radiator, waste heat** HP circuit, avionics* HP augmented cold rail* Modular sink (voice rec) Air cooled equip rack</p>
		<p>*Selected for prototype fab dwgs and test plans</p>
		<p>**Selected for study under separate contract</p>

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 40°F and 117°F with a heat exchanger flow rate of 150 lb/hr, or only 30% of the maximum available rate. The system weighs 0.74 pounds per ft² of radiator surface vs. .78 lb/ft² for a conventional fluid radiator. The weight penalty for a system which uses electrical heaters would be 1.61 lb/ft².

SECTION 4

IDENTIFICATION/PRELIMINARY EVALUATION

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 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 serves 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 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 Fig. 4-1, the direction of these forces can vary through 180 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

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 (IO₂, OMS, ARPS)

2. Aero Surfaces

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

3. External LH₂ 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

TABLE 4-1 (Continued)

- 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	
<ul style="list-style-type: none"> o Avionics <ul style="list-style-type: none"> - "Black boxes" <ul style="list-style-type: none"> - Aircraft avionics - Spacecraft electronics - Radar antennae - High power wiring & connectors o Electrical power equipment <ul style="list-style-type: none"> - APU - Fuel cells - Batteries o Hydraulic equipment <ul style="list-style-type: none"> - Pumps - Hydraulic lines & control valving - Actuators 	<ul style="list-style-type: none"> o Environmental Control Equipment <ul style="list-style-type: none"> - Radiators - Heat exchangers o Main propulsion equipment <ul style="list-style-type: none"> - Gimbal rings - Heat exchangers o Air breathing engine equipment <ul style="list-style-type: none"> - Lubricants - Propellant o Structure <ul style="list-style-type: none"> - Landing gear - Engine compartments - Wheel wells - Pivots and attachments
External	
<ul style="list-style-type: none"> o TPS <ul style="list-style-type: none"> - Aerothermodynamic heating - Post-flight soakback - Plume impingement 	<ul style="list-style-type: none"> o Environmental <ul style="list-style-type: none"> - Solar radiation - Earth radiation - Direct and reflected radiation from other space vehicles

TABLE 4-3 - TYPICAL SPACE SHUTTLE HEAT SINKS

Internal	External
<ul style="list-style-type: none"> o Structure o Cryogenic tankage and piping o Cryogenic boiloff o Water boilers o Water sublimators o Freon boilers o Fluid loop elements o Cold plates, cold rails, heat exchangers o Air cycle equipment o Propellants 	<ul style="list-style-type: none"> o Space o Deployable space radiators o Fixed space radiators o Other space station modules or vehicles o Ground support equipment o Air conditioning, inert gas purge

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:

<u>Phase</u>	<u>Description</u>	<u>Duration (Hr)</u>
Launch	Prior to lift off	2
Boost	Ascent to orbit	2
Cn-orbit	270 n mi, $i = 55^\circ$	164
Entry	De-orbit to sea level	2
Landing	Up to GSE hook-up	1/2

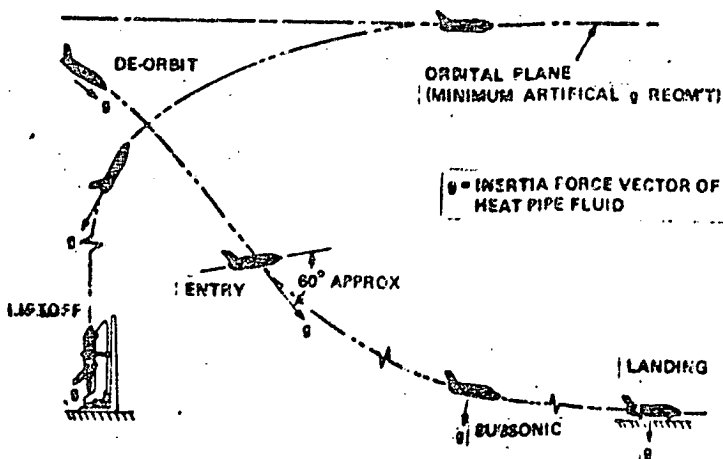


FIGURE 4-1 - INERTIA FORCE VARIATION DURING SHUTTLE MISSION

Identification sheets for each proposed application are contained in Appendix A-1. Each sheet gives a description of the application, its requirements and its advantages and disadvantages.

They have been evaluated 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. A description of each application is given in Appendix A-5.

Additional work was done in areas related to applications previously identified, in response to changing Shuttle definitions. These discussions for

- (1) coupling the wing leading edge to the upper wing surfaces to equalize temperatures,
 - (2) lowering the backface temperatures of superlight ablator panels,
- and
- (3) using heat pipes in a lube oil/hydraulic fluid heat exchanger for a hypergolic fueled APU,

are contained in Appendices A-2, A-3, and A-4, respectively. None of them were evaluated as prime contenders.

TABLE 4-4 - SHUTTLE HEAT PIPE APPLICATIONS CANDIDATES

Prime Contenders (Rating = 2)

<u>Title</u>	<u>Subsystem</u>
1. TPS Leading Edge	Structure
2. Landing Gear	Structure
3. Avionics HP Circuit	Avionics
4. Modular Sinks	Avionics
5. ATR 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

Possible Contender (Rating = 1)

12. OMS LH ₂ Boiloff	Structure
13. High Intensity Lights	Avionics
14. Battery	Avionics
15. Tracking Radar	Avionics
16. Fluid Evaporator	ECS

Rejected (Rating = 0)

17. Fuselage TPS, Interference Heating	Structure
18. TPS Panel	Structure
19. Control Surface Pivots	Structure
20. OMS IO ₂ Boiloff	Structure
21. Main IO ₂ Tank Boiloff	Structure
22. C-Band Directional Antenna	Avionics
23. Electrical Wiring	Avionics
24. Hydraulic Actuators	Power
25. APU	Power
26. IO ₂ Natural Recirculating System	Propulsion
27. Water Chiller	ECS

SECTION 5

PRELIMINARY DESIGN STUDIES

Eleven prime contenders for heat pipe applications on the space shuttle have been identified in the previous section. They represent applications offering tangible benefits over their counterpart baseline thermal control systems and a realistic chance of being implemented. This section gives the preliminary design studies of these prime contenders in eleven self-contained write-ups. Each write-up includes a written discussion of the application, supporting drawings showing the heat pipe systems and shuttle interfaces, and preliminary heat pipe design details including capacity requirements, working fluids, wick design, pipe lengths and diameters. The write-ups are presented in the following order:

<u>Section</u>	<u>Title</u>
5-1	TPS Leading Edge (SPL-114)
5-2	Landing Gear (SPL-103)
5-3	Avionics HP Circuit (SPL-102)
5-4	Modular Sinks (SPL-112)
5-5	ATR Equipment (SPL-105)
5-6	Flight and Voice Recorders (SPL-111)
5-7	HP Radiator with HP Header (SPL-101)
5-8	HP Radiator with Integral HP/Fluid Header (SPL-113)
5-9	HP Augmented Cold Rail (SPL-104)
5-10	HP Radiator for Fuel Cell (SPL-110)
5-11	Air Breathing Engine Compartment (SPL-109)

5-1 - TPS LEADING EDGE (SPL-114)

The baseline thermal protection system for the leading edges of the shuttle wings and vertical fin is an oxidation resistant carbon-carbon reinforced pyrolyzed plastic (RPP) composite structure. This class of materials, although still under development, has shown the potential for long term exposure in an oxidizing atmosphere at temperatures exceeding the limits of coated metals. The substrate materials, comprised of graphite cloth or filaments, offer low density (~ 2 lbs/ft² of surface area) with good strength at elevated temperatures. The development work is being done for NASA by Vought Missiles & Space Co., Dallas, Texas. Their phase I evaluation is given in VMSC Final Report T143-5R-00044 (MSC-02557) and covers the performance period through February 1971. Two diffusion coating systems are candidates for the oxidation resistant coatings: siliconized RPP and zirconium-boron-silicon coated RPP.

The report indicates that siliconized RPP is projected to meet a 100 mission life capability at 3500°F radiation equilibrium temperature and a 10 mission life capability at 3710°F. The former temperature exceeds the 2900°F which was previously considered the limit for a 100 mission life. However, it must be emphasized that the 3500°F limit is projected, it does not exist now.

The report also points out the importance of minimizing thermal stresses in the carbon - carbon material. It is the most significant design parameters for coated RPP with a high elastic modulus but somewhat less important for base and coated RPP with a low elastic modulus. The baseline approach for reducing circumferential temperature gradients is to increase material thickness. But the RPP's low thermal conductivity (2 - 7 Btu/Hr-Ft-°F) makes this prohibitive from a weight viewpoint when applied to areas of severe temperature gradients. Gradients were reduced only 7.5% when the thickness was increased from .20 to .50 inches.

High temperature liquid metal heat pipes can be applied to the leading edges in areas of high aero-heating to reduce the adverse circumferential temperature gradients in the carbon-carbon. Sheet 2 of Figure SPL-114 shows

the radiation equilibrium temperature distribution at the wing leading edge during peak heating (400 seconds after re-entry). It includes the equalizing effects of internal reradiation between the backface surfaces. Accounting for this cross-radiation reduces the maximum surface temperature at the stagnation point by 370°F . The corresponding thermal gradient across the .20 inch thickness is 278°F .

It should be noted that failsafe considerations demand that the minimum material thickness be determined by the single mission capability of uncoated RPP which has a recession rate of 4 to 5 (10^{-4}) in/sec. This means a minimum thickness on the order of .20 inches.

A preliminary evaluation of the use of high temperature heat pipes was done by D. Ernst, P. Shefsiek and J. Davis of the Thermo Electron Corp., Waltham, Mass. under an informal agreement with Grumman Aerospace. The subsequent application is based upon their contributions for the design of the heat pipes.

Three configurations for the application of heat pipes to the leading edge are proposed. They are all based on isothermalizing the surfaces to which they are attached and all of them use lithium as the working fluid since the operating temperature range is on the order of 3000°F .

The first configuration, shown on sheets 1, 2 and 3 of SPL-114, involves isothermalizing the entire wing leading edge. Heat pipes located along the element lines of the airfoil (see sheet 3) carry heat to a secondary HP header which follows the circumferential contour of the foil section (see sheet 2). Each secondary header, in turn, feeds the primary HP header which runs parallel to the leading edge at about the 7% chord line. The theoretical effect of this application would be a uniform temperature of about 1450°F along the entire leading edge.

Preliminary evaluations have raised objections to the number of HP to HP joints that are required (excessive ΔT 's), the long length of primary header called for, and the obvious weight penalty incurred by these many pipes.

Configuration II (sheet 4) involves isothermalizing only a small section of the leading edge by attaching heat pipes which follow the circumferential contour of the edge. Heat Pipes (1" wide by 1/4" deep) are placed side by side, as necessary, to equalize the temperatures in high heating rate areas. Application of an isothermalizer heat pipe to the section of sheet 2 would result in a theoretical uniform temperature of 2630°F, a reduction of 490°F from the maximum stagnation line temperature. It would require a heat pipe capable of transmitting 26 watts/cm² of surface area, which is low compared to heat pipes which have been built and tested as seen in Table 5-1.

The heat pipe envelope could be made from one of the materials shown in Table I on sheet 4. One of the areas requiring further investigation is the bonding mechanism of the HP envelope to the RFP backface. Materials compatible with lithium might also be a problem although Thermo Electron has successfully run a TZM-lithium heat pipe at 3300°F and 190 watts/cm² for 4600 hours. It used a 60/60 Molybdenum screen mesh for its wick. They have also demonstrated a tungsten-lithium pipe using 50/50 tungsten mesh at 3730°F and 100 watts/cm². The demonstrated operating temperatures and power densities are well within the requirement for this application (2600 - 2800°F and 26 watts/cm²).

An approach to overcome the attachment bonding problem is to form a leading edge structure that contains enclosed internal channels which act as heat pipe envelopes (see Configuration III). Thus, the heat pipes are integral to the RFP structure and also contribute to the structural stiffness of the panel. About 50% of the heat pipe weight can then be credited to structure, thereby decreasing the weight penalty associated with using the heat pipes.

TABLE 5-1 - HIGH TEMPERATURE HEAT PIPE PERFORMANCE DATA*

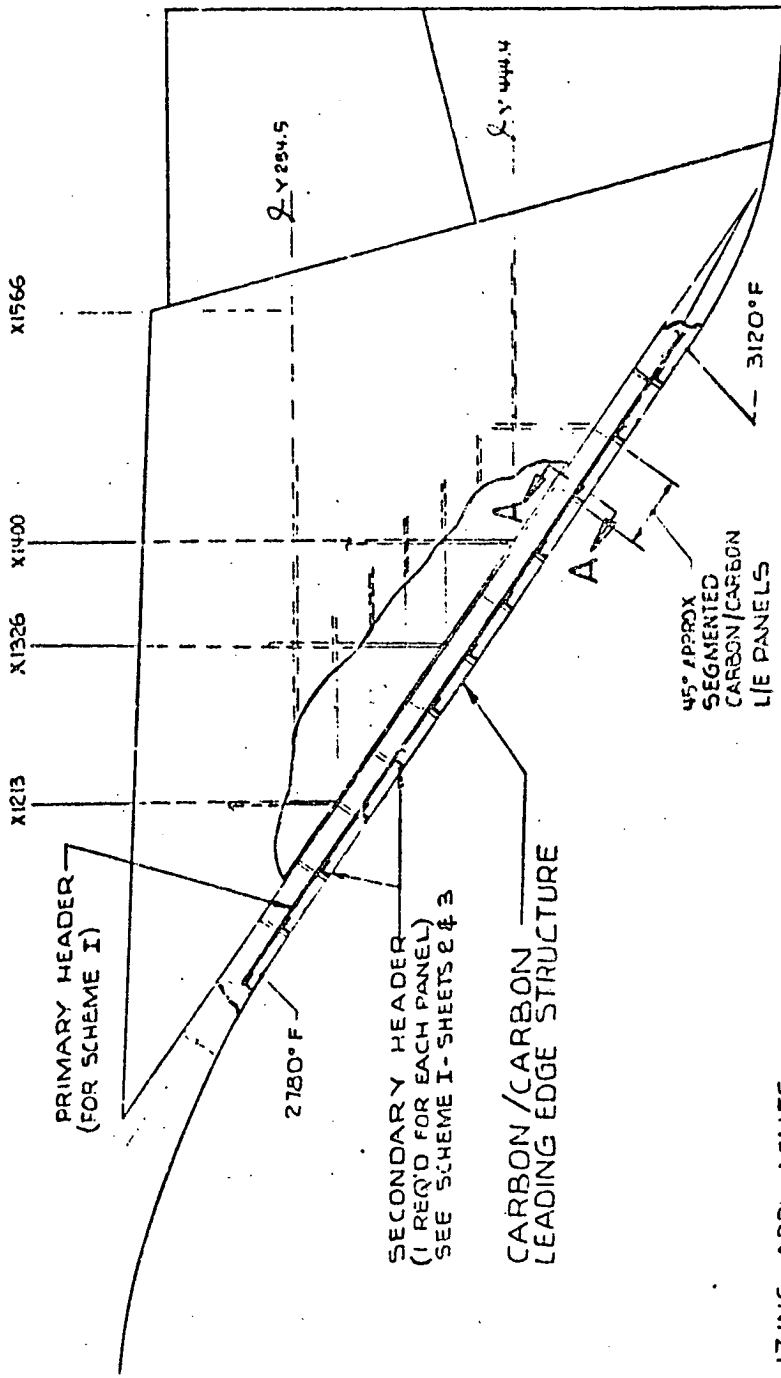
Heat Pipe	Temperature	Heat Fluxes			Operational Length Against Gravity	Liquid Flow in Terms of Heat Flux per Cross-sectional Area of 60 x 60 Mesh
		Evaporator	Condenser	Vapor		
TZM-L	1560°C	190 w/cm ²	14-42 w/cm ²	2.3 kw/cm ²	12"	4.1 kw/cm ²
W-L	1100°C	17.5 w/cm ²	31 w/cm ²	1.4 kw/cm ²	7"	2.5 kw/cm ²
W-L	1500°C	35 w/cm ²	62 w/cm ²	2.8 kw/cm ²	7"	5 kw/cm ²
W-L	1800°C	56 w/cm ²	100 w/cm ²	4.5 kw/cm ²	7"	8 kw/cm ²
Proposal	1500°C	26 w/cm ²	26 w/cm ²	0.3 kw/cm ²	**	0.2 kw/cm ²

* Test data provided by Thermo Electron Corp., Waltham, Mass.

** Total Length 12" Evaporator at mid-point.
 Total Height 8"



31



56

ISOTHERMALIZING APPROACHES
 TO CARBON CARBON LEADING EDGE
 SCHEME I — SHEETS 1, 2 & 3
 SCHEME II — SHEET 4
 SCHEME III — SHEET 5

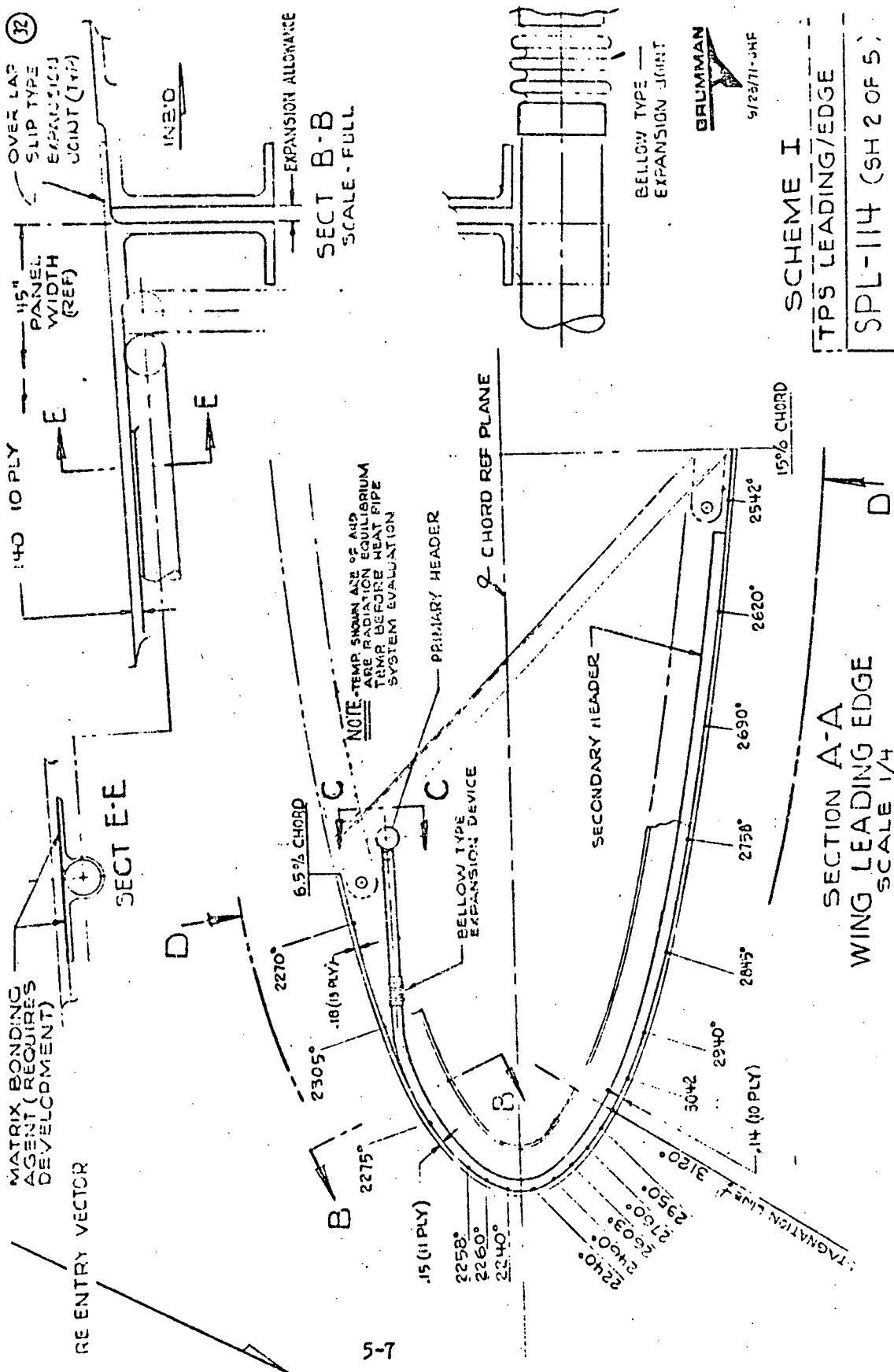
VIEW LOOKING DN ON WING*
 (SCALE 1/75)

* SCHEMES SHOWN ARE SIMILARLY APPLICABLE
 TO OTHER STRUCTURAL COMPONENTS USING
 CARBON/CARBON AS TPS (IE, VERT FIN)



SPL-114 (SH 1 OF 5)
 TPS LEADING EDGE
 (CARBON CARBON MATERIAL)

FIG. 5-1



OVER LAP SLIP TYPE EXPANSION JOINT (177)

115" PANEL WIDTH (REF)

140 10 PLY

SECTION E-E

MATRIX BONDING AGENT (REQUIRES DEVELOPMENT)

RE ENTRY VECTOR

D

6.5% CHORD

C

NOTE - TEMP. SHOWN ARE OF AND ARE RADIATION EQUILIBRIUM TEMP. BEFORE HEAT PIPE SYSTEM EVALUATION

PRIMARY HEADER

BELLOW TYPE EXPANSION DEVICE

2% CHORD REF PLANE

SECONDARY HEADER

C

3042

2940°

2845°

2758°

2690°

2620°

2542°

15% CHORD

D

SECTION A-A
WING LEADING EDGE
SCALE 1/4

SECTION B-B
SCALE - FULL

EXPANSION ALLOWANCE

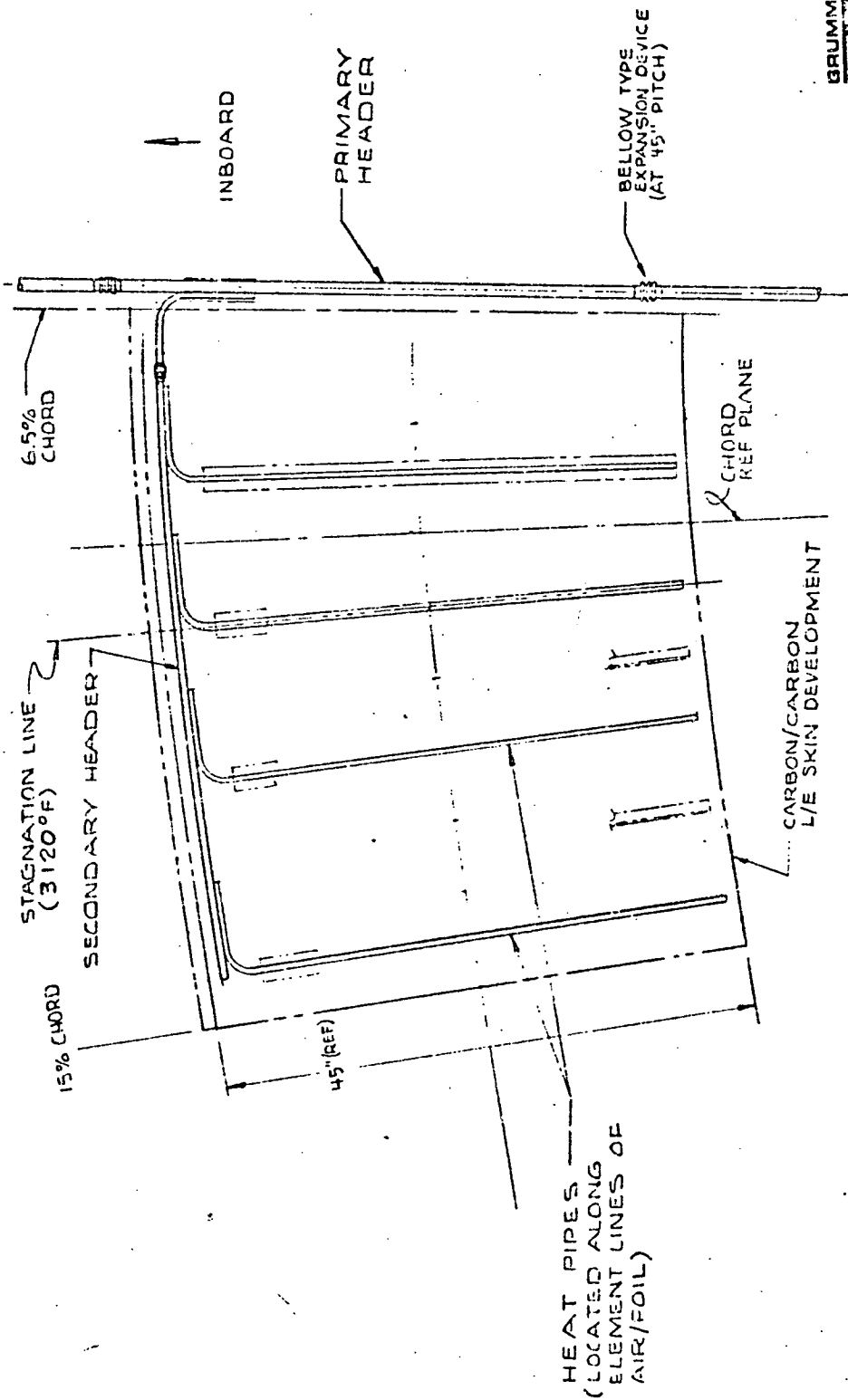
BELLOW TYPE EXPANSION JOINT

BRUMMAN
9/25/71-JHF

SCHEMATIC I
TPS LEADING/EDGE

SPL-114 (SH 2 OF 5)

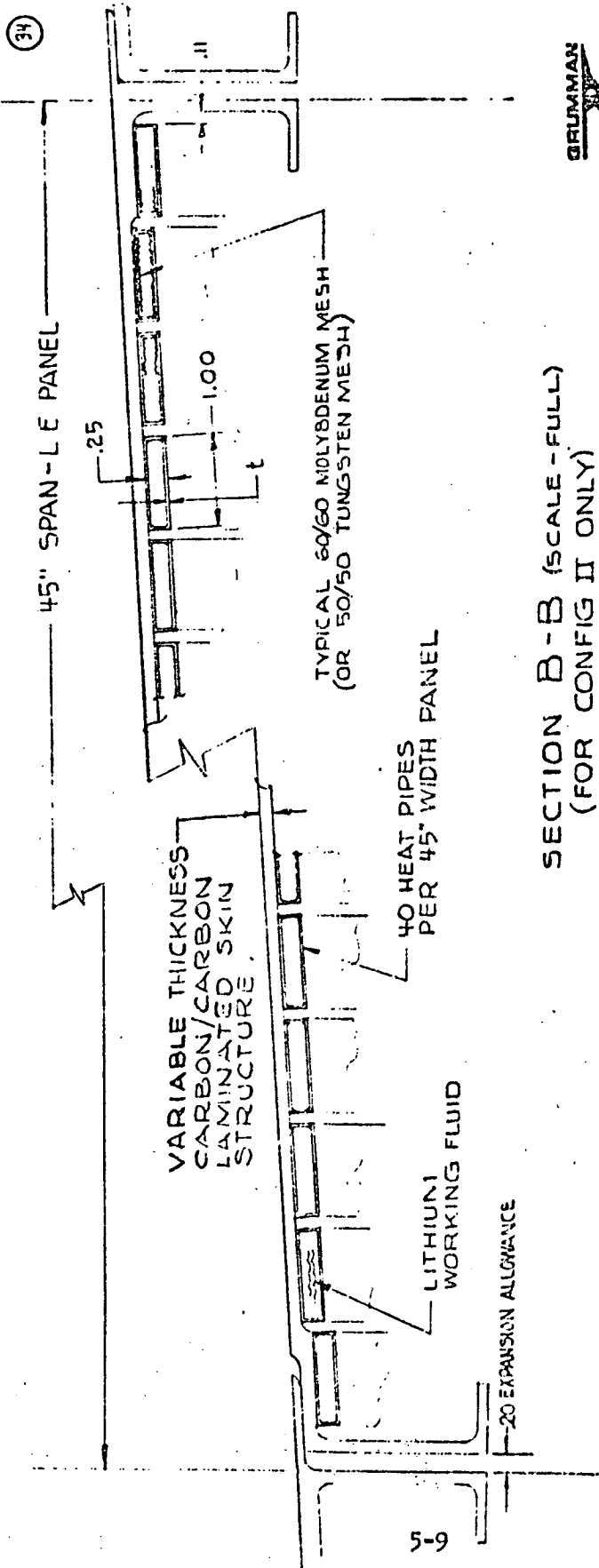
33



97-5/71-JHF

SCHEME I
 TPS LEADING/EDGE
 SPL-114 (SH 3 OF 5)

VIEW D-D
 LEADING EDGE SHOWN
 UNFOLDED TO SHOW TYPICAL
 HEAT PIPE CIRCUIT WITHIN
 INSIDE SURFACE OF PANEL
 (APPROX SCALE 1/8)



SECTION B-B (SCALE - FULL)
(FOR CONFIG II ONLY)



CONFIG II - SOME PROBLEM AREAS

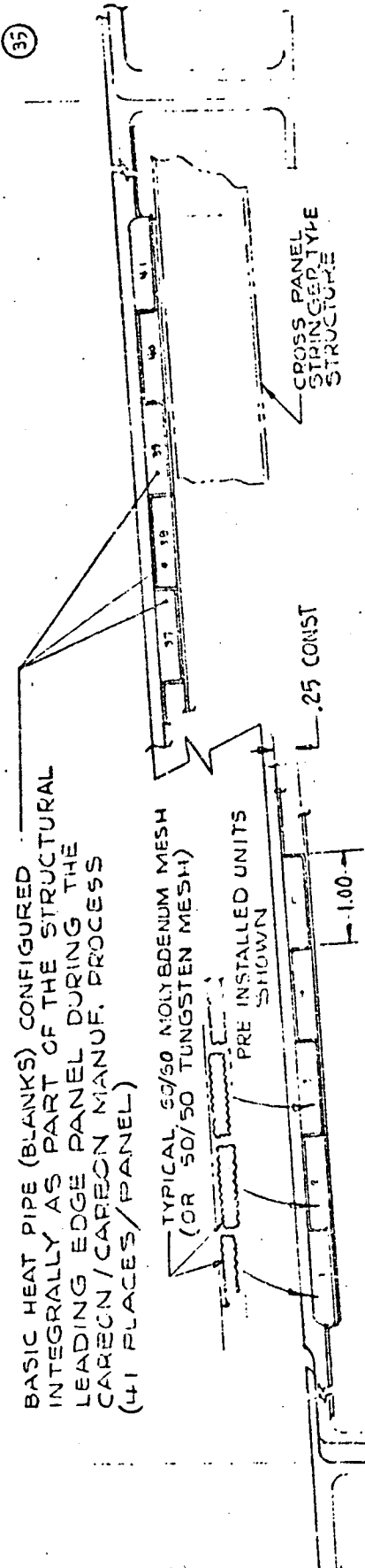
- 1- MATRIX BONDING AGENT FOR TEMPERATURE RANGE INDICATED REQUIRES DEVELOPMENT
- 2- REDUNDANT TOOLING REQUIRED TO MATCH HEAT PIPE SURFACE TO INTERNAL (LAMINATED) SURFACE OF CARBON/CARBON LEADING-EDGE SKIN.
- 3- HEAT PIPES ALONG THE INTERNAL SURFACE OF THE L/E AIRFOIL PREVENTS THE INCLUSION OF CROSS TYPE STRUCTURE, SUCH AS STRINGERS, INTERCOSTALS ETC.

* MOST FAVORABLE

SPL-114 (SH 4 OF 5)

TABLE I - MATERIAL SELECTION DATA (WITHOUT MANUF INPUT)

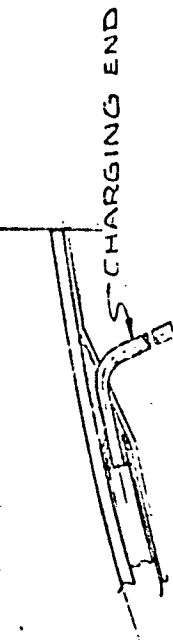
MAT'L	MELTING PT (°F)	DENSITY (G/CM ³)	THERMAL COND AT 20°C (W/CM ² °C)	MECH PROP AT 20°C (TENS)	WT-PER INCH PER 40 HPI/SQ INEL
ALUMINUM	9425	.60	31.5	13 KSI	1.8"
TUNGSTEN	6170	.70	96.6	45 KSI	2.1"
NIOBIUM	5760	.76	43.7	66 KSI	2.26
MOLYBDENUM	4730	.87	84.5	16 KSI	1.11"
50-50 Ti	4730	.87	84.5	18 KSI	1.11"
50-50 Ti	4730	.87	84.5	54 KSI	< 1.11"



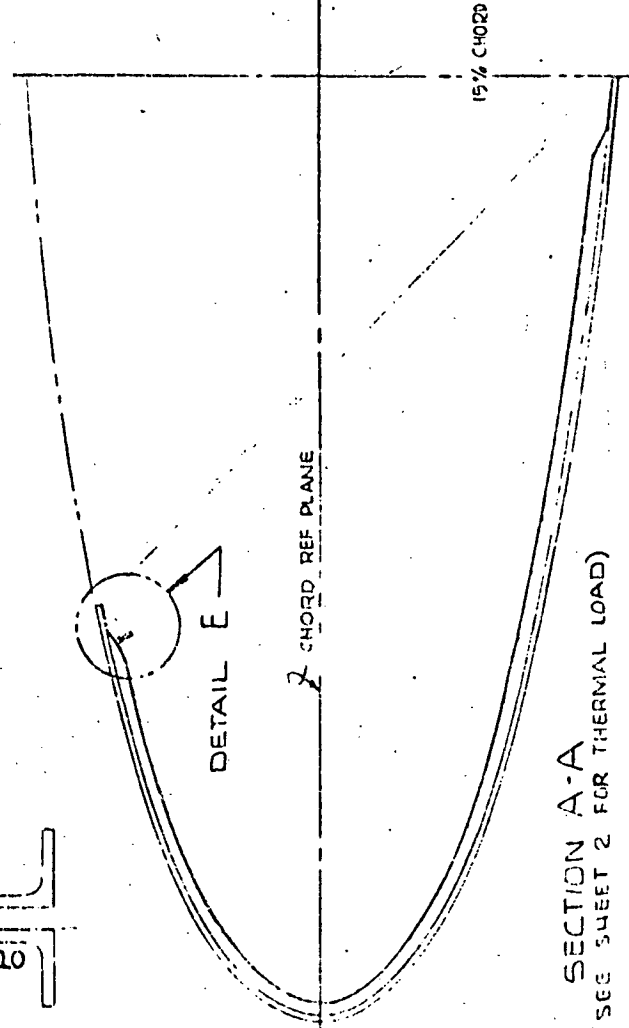
35

SECTION B-B (SCALE - FULL)

5-10



DETAIL E
SCALE - FULL



SECTION A-A
(SEE SHEET 2 FOR THERMAL LOAD)

GRIMMAN
9/28/71-JHF

SCHEME III
TPS LEADING/EDGE
SPL-114 (SHEET 5 OF 5)

5-2 - WHEEL WELL RADIATOR - ON-ORBIT MINIMUM
TEMPERATURE CONTROL (SPL-103)

A system for heating the wheel assembly while in orbit is necessary, as rubber tires cannot survive temperatures below -65°F , and should not be used for landings below -50°F .

Under normal operating conditions waste heat is brought from the low-temperature cabin electronics back to the area of fuselage station X1212, where it is carried outboard and then up to the rearmost low temperature radiator panel. At this point, some heat could be diverted to panels in the main landing gear wells as shown in SPL-103.

It was decided to use the heat obtained from the circuit to heat two radiator panels in each wheel well, one on each side facing a tire. In this configuration, the energy emitted by the radiator would maintain an acceptable tire temperature.

In choosing the capacities required in the heat pipes to be used, the heat loss to the skin at -180°F from an assumed wheel well wall temperature of -20°F was calculated to be 300 watts per wheel well. It was then assumed that this amount of energy must be supplied to the radiators.

The wheel well radiator headers are connected to the main heat transport circuit by a diode heat pipe to each well. A diode heat pipe, permitting flow only toward the wheel well, was chosen to prevent leakage of heat back into the heat transport circuit during re-entry.

This diode could be made of a half-inch diameter heat pipe, which, in the required dimensions, has a capacity of 544 watts and a maximum loading requirement of 300 watts.

From each of these diode heat pipes, there are two half-inch diameter headers supplying each radiator panel with 150 watts. If the headers are filled with ammonia the capacity of each is close to 1.4 kilowatts; if filled with Freon-21, 285 watts.

From this header, there are six feeder pipes into the radiator, each of which must supply 25 watts. If ammonia-filled feeder heat pipes are used, the capacity of each will be in excess of 2.5 kilowatts; if Freon-21, the capacity of each will be about 500 watts.

TABLE 5-2 - HEAT PIPE ROSTER - MAIN LANDING GEAR WHEEL WELL SYSTEM

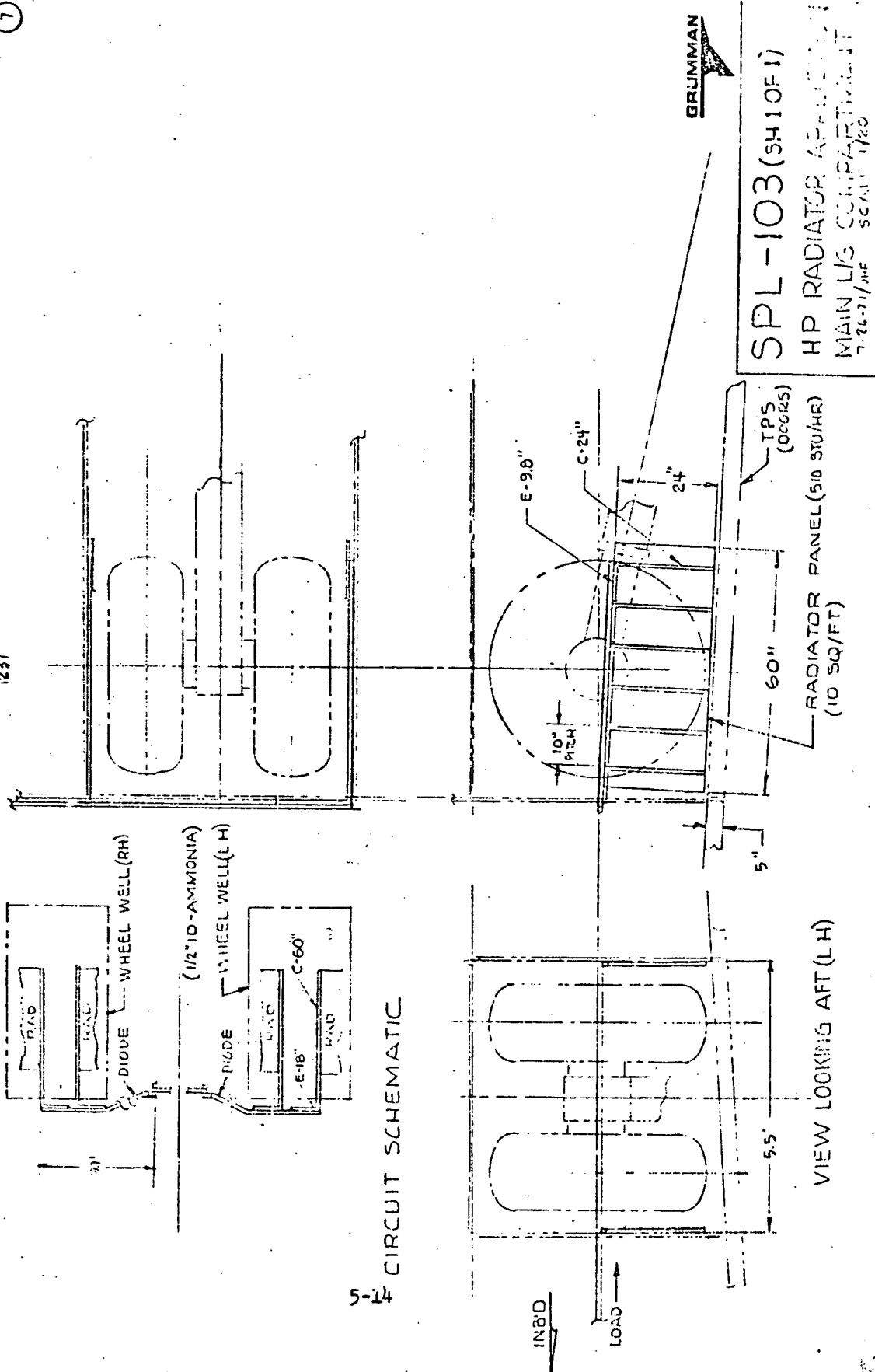
Heat Pipe Name	Dimensions	Fluid	Wick	Number Required and Placement	Computed Max. Capacity (Watts)	Required Capacity (Watts)	ΔT at Req'd. Capacity
Diode from Header Leader to Wheel Well Radiator Header	$\frac{1}{2}$ " ID, $L_{EVAP} = 48$ " $L_{COND} = 42$ " , $L_{TRANS} = 69$ "	NH ₃	GAC Spiral Artery/Tunnel	2 - 1 each main landing gear well	544	300	3.0°
	OR 1" ID, $L_{EVAP} = 48$ " $L_{COND} = 42$ " , $L_{TRANS} = 69$ "	Freon-21					
Wheel Well Radiator Header	$\frac{1}{2}$ " ID, $L_{EVAP} = 18$ " $L_{COND} = 60$ " , $L_{TRANS} = 0$ "	NH ₃	GAC Spiral Artery/Tunnel	4 - 1 to each panel, 2 panels to each main landing gear wheel well	1,317	150	3.2°
		OR Freon-21					
Wheel Well Radiator Feeders	$\frac{1}{2}$ " ID, $L_{EVAP} = 9.8$ " $L_{COND} = 24$ " , $L_{TRANS} = 0$ "	NH ₃	GAC Spiral Artery/Tunnel	24 - 6 to each panel, 2 panels to each main landing gear wheel well	2,517	25	1.0°
		OR Freon-21					
					515	25	2.0°

5-13

*Recent tests indicate an improvement of (at least) a factor of two over these figures; i.e., one-half the ΔT .

7

1237



5-44 CIRCUIT SCHEMATIC

VIEW LOOKING AFT (LH)

FIG. 5-2

5-3 - HEAT PIPE HEAT TRANSPORT SYSTEM (a), AVIONICS HEAT PIPE
CIRCUIT (b) (SPL-102)

(a) HP Heat Transport System

When the vehicle is in orbit, much of the internal load is generated by electronics in and around the cabin area and by the fuel cells which are forward of the cabin area. This heat is to be dissipated by a group of radiator panels on the inside of the cargo bay doors. It is necessary to transport this heat energy some distance through the ship and the heat pipe circuit described herein has been designed for this purpose.

The electronics gear in the cabin area may be divided into two categories: "high-temperature" electronics (operating at around 90°F) and "low temperature" electronics (operating at around 40°F). These high and low-temperature sources make it thermally more efficient to use separate "circuits" for each, to transport and reject the waste heat load.

In designing such heat pipe circuits, there is a trade-off involved - the longer the transport section, the less the maximum heat pipe capacity, in an almost hyperbolic relationship. Yet the designer would like to minimize the number of series connected heat pipes between points in order to minimize contact conductance temperature drops between them. In the present case, with loads on the order of 5 kilowatts, it is necessary to do so.

It was decided to use a series - parallel design (as shown in SPL-102) in order to maintain the necessary heat transfer capacity in the event of failure of one heat pipe. Large capacity pipes (1" ID, ammonia-filled) are specified, in view of the large capacities and comparatively long effective lengths required.

The "hot" circuit consists of two parallel links of three heat pipes in series, connected fore and aft by isothermalizing "crossover" heat pipes. These pipes in series each have a capacity of over 4.9 kilowatts, and, at maximum heat production by the fuel cells, are required to carry only 2,110 watts. In the event of a complete failure of one of the series links, the other is capable of carrying the full load.

When this heat load reaches fuselage station X 918, it is carried to the radiator header by two "leader" heat pipes, one on each side, with each having a capacity of 4,164 watts and a load-carrying requirement of 2,110 watts.

The "cold" circuit is similar, but, due to the larger maximum load produced by the cabin area electronics (6,780 watts), it was necessary to go to four parallel links of three heat pipes each. This circuit is shown as originating in the center of the cabin area, as few details are available concerning specific placement of equipment.

Each of the four parallel links has a capacity of 4,064 watts and is required to carry a maximum load of 1,690 watts. If a heat pipe in one link failed, the other links would then have to carry a maximum of 2,260 watts, still well within the predicted capacity of the heat pipe. If a less critical margin of safety is acceptable the system needs only half the number of pipes.

At fuselage station X1110, heat flow splits between the two forward "cold" radiator panels (with a capacity of 3,917 watts and a requirement of 2,260 watts) and a continuation of two of the parallel links by two "terminal" heat pipes (with a capacity of 5,397 watts and a requirement of 1,130 watts), attached at fuselage station X1212 to the headers of the aft radiator panel on each side.

At this point, some of the heat energy is taken from the area of this aft leader/header interface to the radiator panels in the main landing gear wheel well (see Section 5-2).

(b) Avionics HP Circuit (SPL-102B)

While the foregoing discussion dealt with an all heat pipe heat transport system, it is possible (and perhaps advisable) to use a conventional fluid loop as the transport system between smaller heat pipe circuits and an ultimate sink (e.g., radiator, fluid boiler). This would allow the newer heat pipe technology to be gradually integrated with more common systems and controls. The avionics subsystem is an area where such a hybrid system might be particularly advantageous.

Standard avionics equipment which has been designed for fluid cold rail or cold plate cooling will instead be attached to all heat pipe rails or plates. The mounting interfaces would be unchanged. The "feeder" heat pipes from the cold rails will then be joined to intermediate heat pipe headers which, in turn, will be connected to the main fluid header. Figure SPL-102B shows one such arrangement. Its main advantages are more flexibility and improved maintainability.

The absence of fluid paths within an equipment rack eliminates flow splits and flow balancing problems - especially when requirements change and items are modified and relocated. The complete lack of fluid couplings permits entire racks to be removed without contaminating the heat transport fluid system and requiring endless cycles of purging and recharging - a quite common occurrence on previous spacecraft fluid loop systems.

One-half inch diameter heat pipes will be used in the cold rails, with water as the working fluid for applications inside the pressure shell and Freon-21 used outside. The heat transport capacity for a typical $\frac{1}{2}$ inch water pipe is 850 watts at 100°F and 250 watts at 40°F.

TABLE 5-3 - HEAT PIPE ROSTER - HEAT PIPE HEAT TRANSPORT CIRCUIT

Heat Pipe Name	Dimensions	Fluid	Wick	Number Required and Placement	Computed Max Capacity (Watts)	Required Capacity (Watts)	ΔT : Required Capacity
Basic $\frac{1}{8}$ " Member - "Hot" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 62.5"	RH ₃	GAC Spiral Artery/Tunnel	6 - 2 Parallel Links of 3 Heat Pipes in Series	4,978	2,110	9.3°
Basic $\frac{1}{8}$ " Member - "Cold" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 96"	RH ₃	GAC Spiral Artery/Tunnel	12 - 4 Parallel Links of 3 Heat Pipes in Series	4,064	1,690	7.9°
Terminal Art $\frac{1}{8}$ " Member - "Cold" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 51.6"	RH ₃	GAC Spiral Artery/Tunnel	2 - 2 Parallel Installations	5,397	1,130	5.3°
Leader to Radiator Header - "Cold" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 101"	RH ₃	GAC Spiral Artery/Tunnel	4 - 1 each to Left and Right Radiator Headers - Stations X1110 X1212	3,917 3,917	2,260 1,130	10.9° 5.3°
Leader to Radiator Header - "Hot" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 89"	RH ₃	GAC Spiral Artery/Tunnel	2 - 1 each to Left and Right Radiator Headers - Station X918	4,164	2,110	9.8°
Crossover Unit - "Hot" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 48"	RH ₃	GAC Spiral Artery/Tunnel	2 - 1 each at Fore and Aft of "Hot" circuit Connecting Right and Left Sides	5,486	0 to 2,110	0° to 9.8°
Crossover Unit - "Cold" Circuit	1" ID, L _{EVAP} = 48" L _{COND} = 48", L _{TRANS} = 24"	RH ₃	GAC Spiral Artery/Tunnel	2 - 1 each at Stations X1110 and X1212 Connecting Right and Left Sides of "Cold" Circuit	6,787	0 to 3,390	0° to 14°

*Percent tests indicate an improvement of (at least) a factor of two in these figures, i.e., half the ΔT .

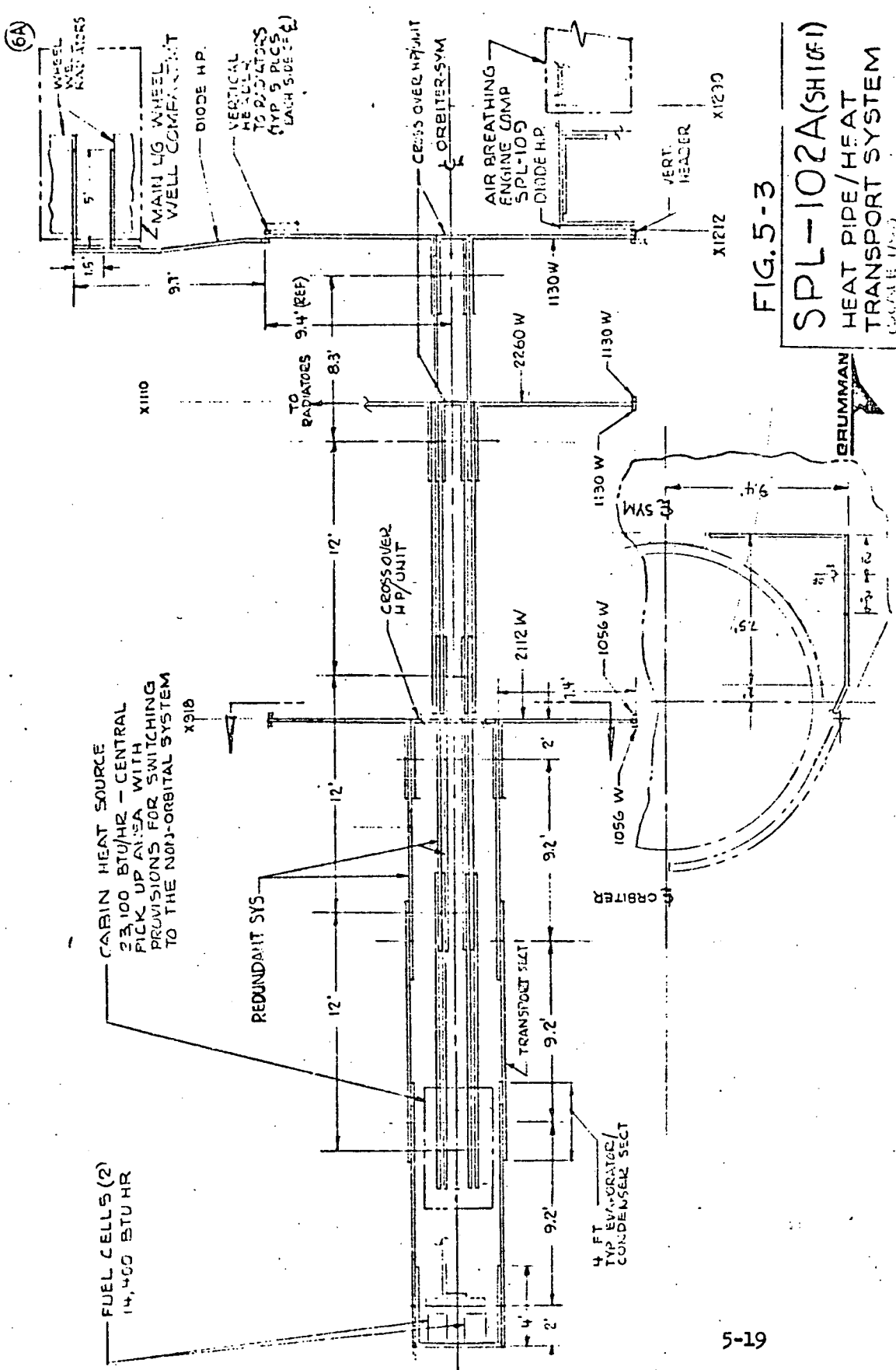
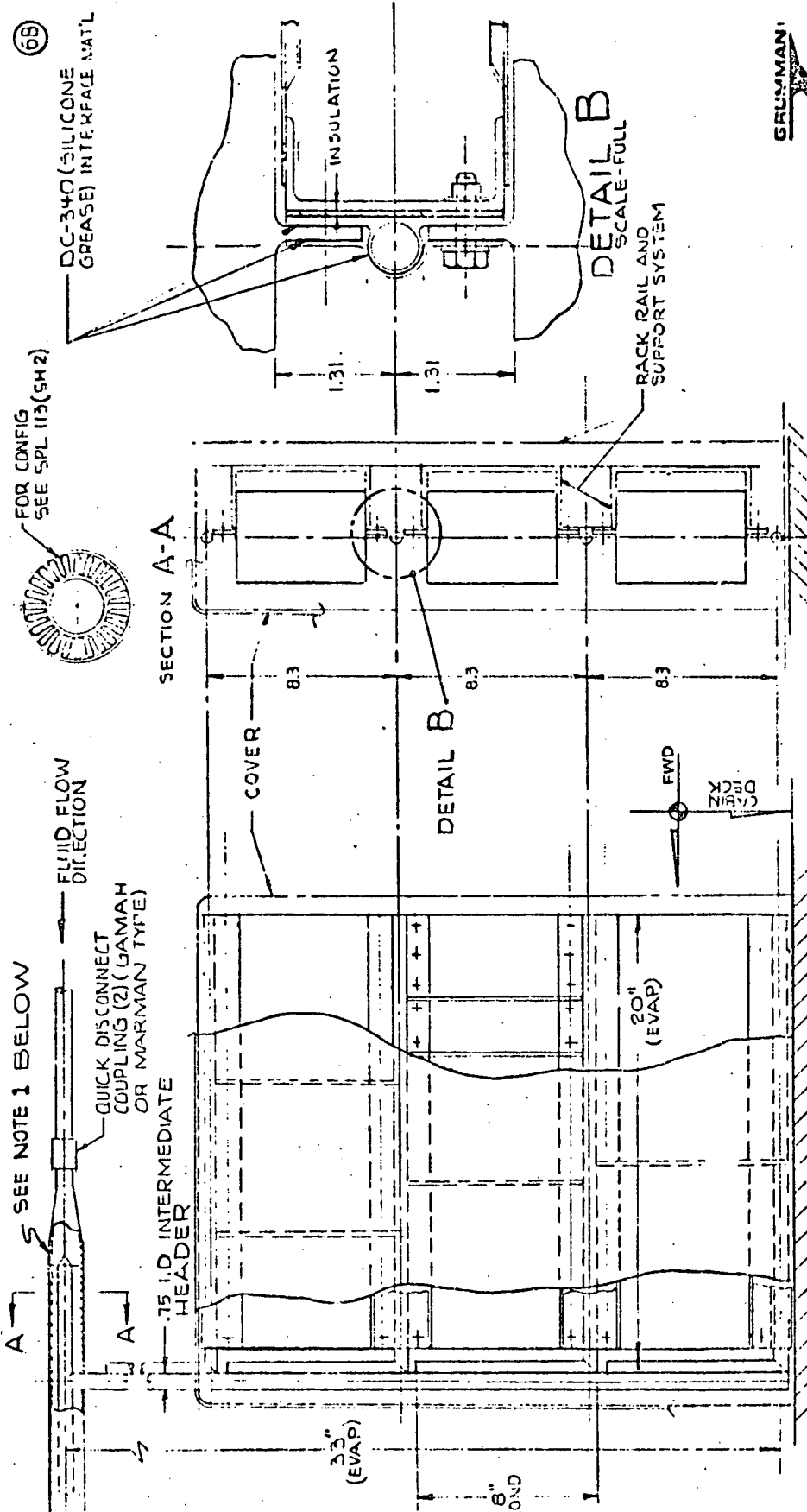


FIG. 5-3
SPL-102A(Shiff I)
 HEAT PIPE/HEAT
 TRANSPORT SYSTEM
(SCALE 1:10)



SPL-102-B (SH 1 OF 1)
AVIONIC HEAT PIPE CIRCUIT
 SCALE - NOTED

NOTES
 1- CONVENTIONAL HEAT PIPE HEADER CAN BE SUBSTITUTED FOR THE FLUID TYPE HEADER SHOWN ABOVE

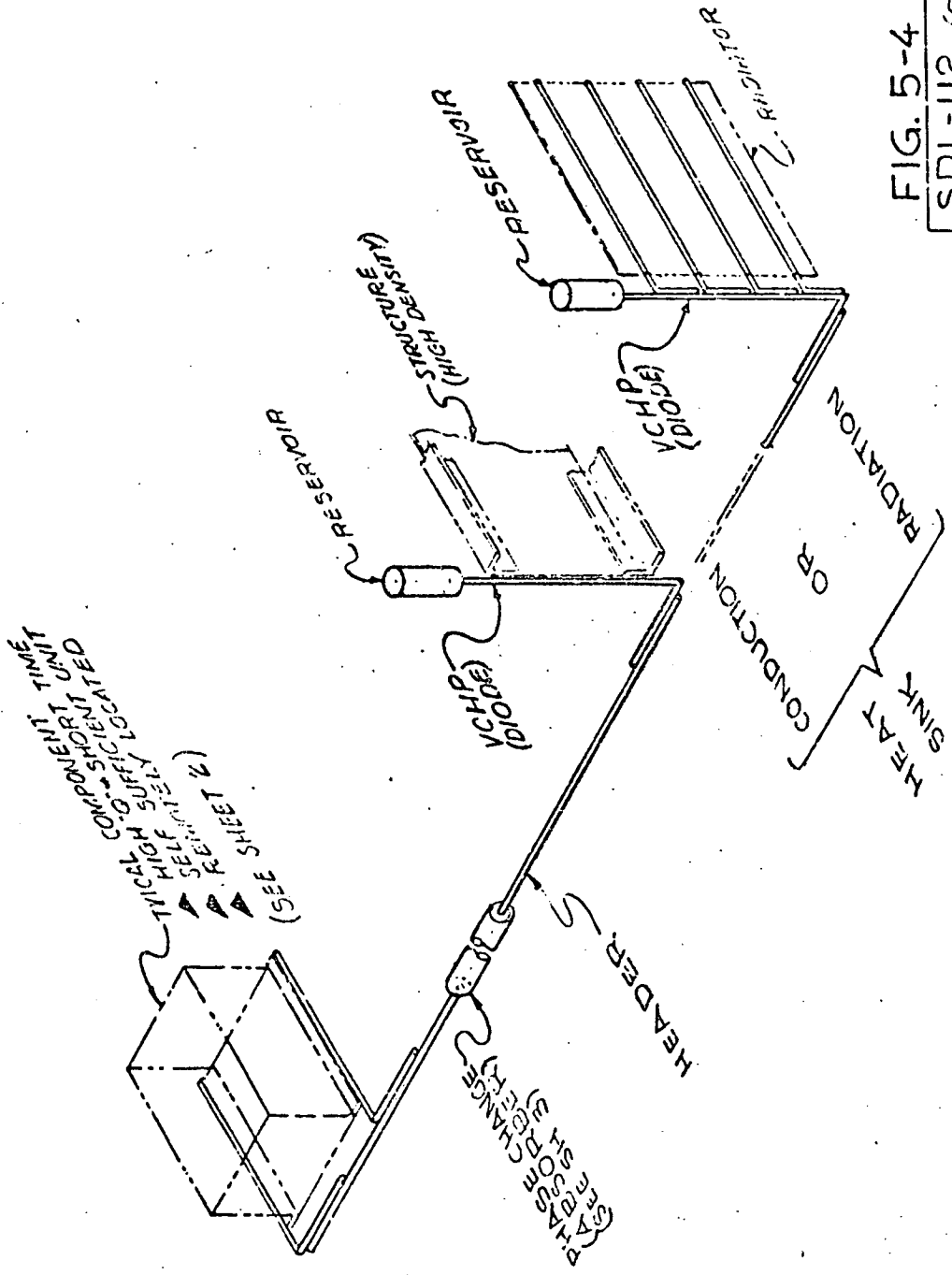
MODULAR EQUIP RACK (SCALE: 1/4)
 ADAPTED FOR 100% HEAT PIPE/
 COLD RAIL INTERNAL CONFIG.

5-4 - MODULAR SINKS (SPL-112)

Sheet 1 of SPL-112 illustrates this concept. It would be simpler to cool a remotely located, low power output (except for short periods of high power dissipation) component by some local means rather than extending a pumped fluid coolant loop to these remote locations. This concept describes one means of accomplishing this.

Heat could be removed from the component by means of heat pipes (sheet 2), in a manner similar to that used to cool electronic components with cold rails. This heat would be carried by a header running through the modular sink to a diode heat pipe. The latter is in turn attached to a sink (structure or a small radiator) capable of dissipating the small steady load produced by the component.

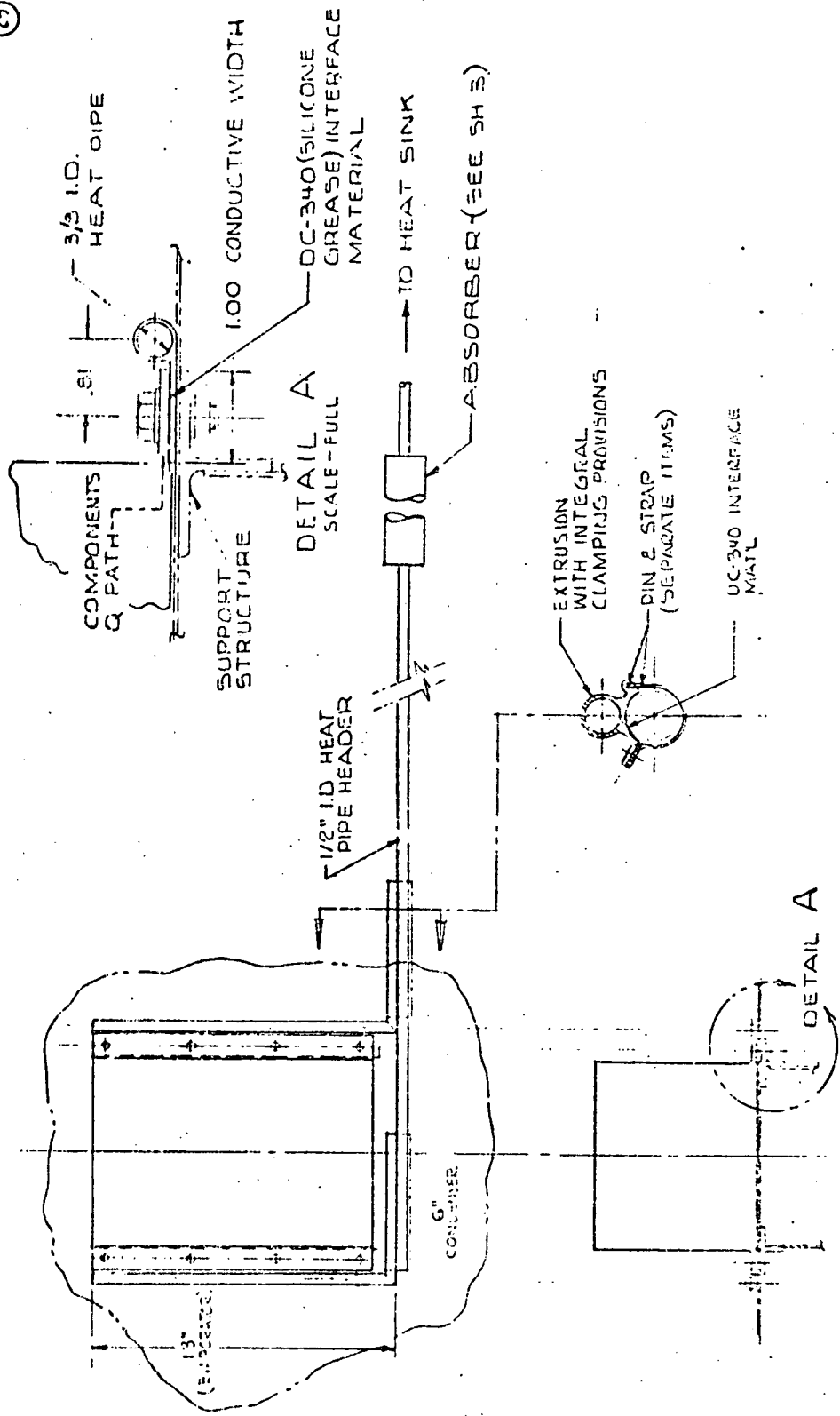
Under low-power conditions, the modular sink would be unaffected. When the component switches to its high power mode, the conventional sink would be unable to absorb the extra energy. As the sink temperature increased, the heat pipe vapor temperature would increase until it reached the melting point of the phase change material. The absorber would then retain the extra heat and hold the vapor temperature in the heat pipe steady. When the component returned to its low power mode, the phase change material would resolidify, giving up the heat it had absorbed to the sink, thus resetting the device for the next cycle.



5-22

THIS DRAWING IS UNCLASSIFIED DATE 05/08/2001 BY 60322 UCBAW/SJS/STP

25



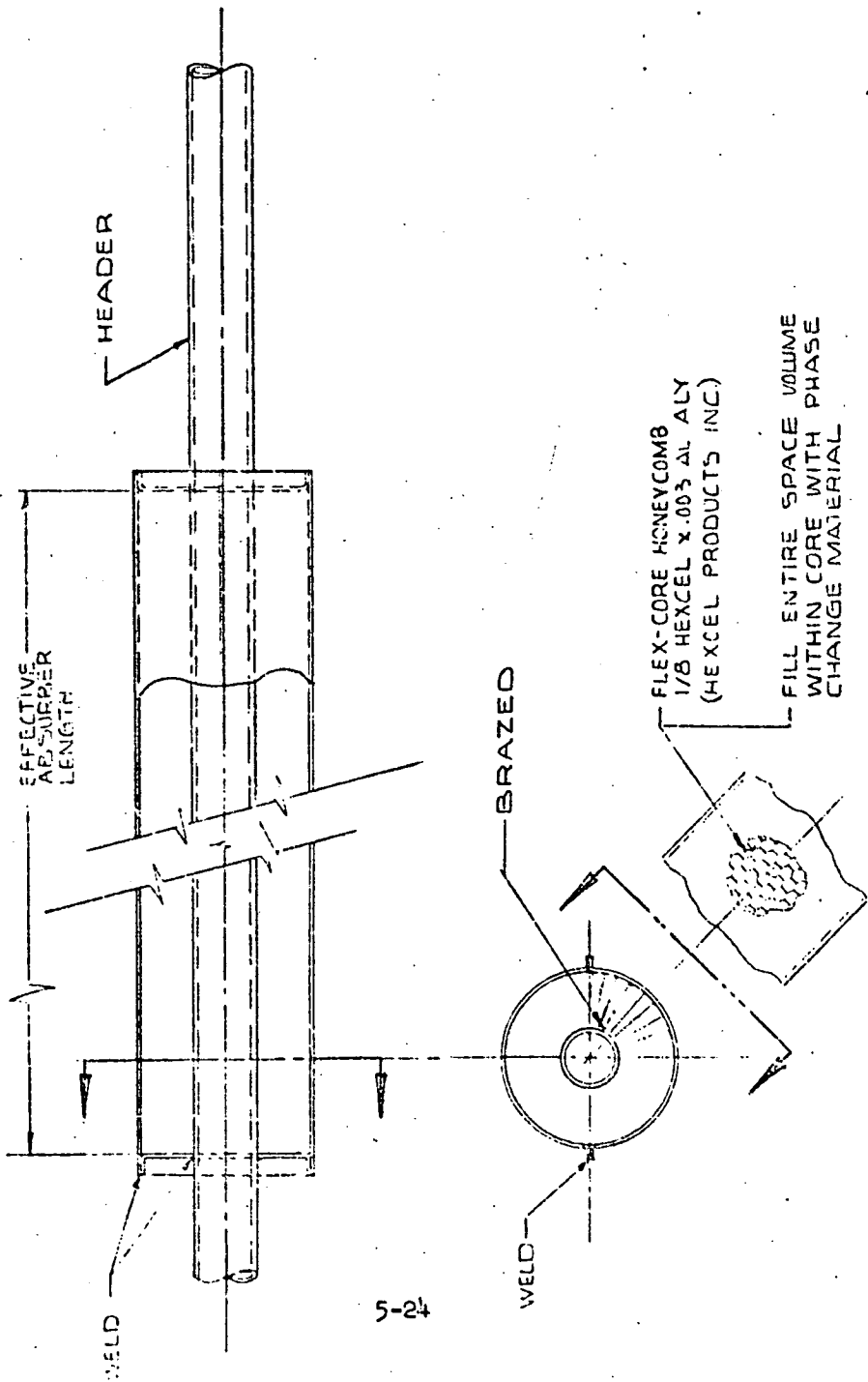
HEAT ABSORBER - MODULAR SINK

SPL - 112 SH 2 OF 3

SCALE 1/4" = 1"



912771-JHF



5-24

HP/ABSORBER ~ MODULAR SINK
 SPL-112 (SH 3 GF 5)
 SCALE - FULL

GRUMMAN
 31/271-JWF

5-5 ADAPTATION OF MILITARY AND COMMERCIAL AIR AVIONICS
TO THE SHUTTLE (SPL-105) - AIR TRANSPORT
RACK EQUIPMENT

Extensive use of available commercial airline and military aircraft avionics is planned for the Shuttle because they are about one-fifth to one-tenth the cost of "space-tailored" equipment. However, much of this equipment has been designed for standard mounting fixtures (racks) and requires a convective atmosphere for proper temperature control. Some means for marrying these requirements, without alteration, is necessary to take full advantage of the potential cost savings.

Modular designed heat pipe-to-air heat exchanger elements in conjunction with an air circulating enclosure can provide a standard approach for the adaptation of aircraft avionics to the shuttle orbiter. A typical enclosure for cabin located equipment would consist of suitable shelving or bracketing for mounting standard military air packaging or stacked printed circuit cards. Forced air circulation would be provided by a centrally located fan or blower. The blower air is diverted from a supply plenum to parallel passages where it passes up, through and over the equipment.

Current ATR mounting configurations (Sheet 1 of SPL-105) feature passages which lose a portion of their air to an adjacent passage. The cooling air which passes up and through the equipment comes from the adjoining passage, through holes in the mounting plates. Air which passes over the tops of the equipment comes directly from the supply plenum. The projected ATR mounting configuration (sheet 2) calls for completely separate, non-communicating, parallel passages where all the flow passes up, through and over the equipment. Each charge of air makes only one pass.

In both configurations the heated air from each channel flows into a return plenum where it mixes and is directed to the lower intake. Immediately before reaching the intake, the air flows through the finned heat pipe heat exchanger modules where it rejects its heat. The heat load picked up by the heat pipes is then transferred to either a heat pipe header or fluid header for eventual transport to a final sink (e.g., radiator, expendable fluid).

The ATR enclosure would be designed to hold sufficient internal pressure to survive temporary loss of crew compartment air. Adequate vibration and shock attenuation would be designed into the enclosure to eliminate shock mounts on each individual component.

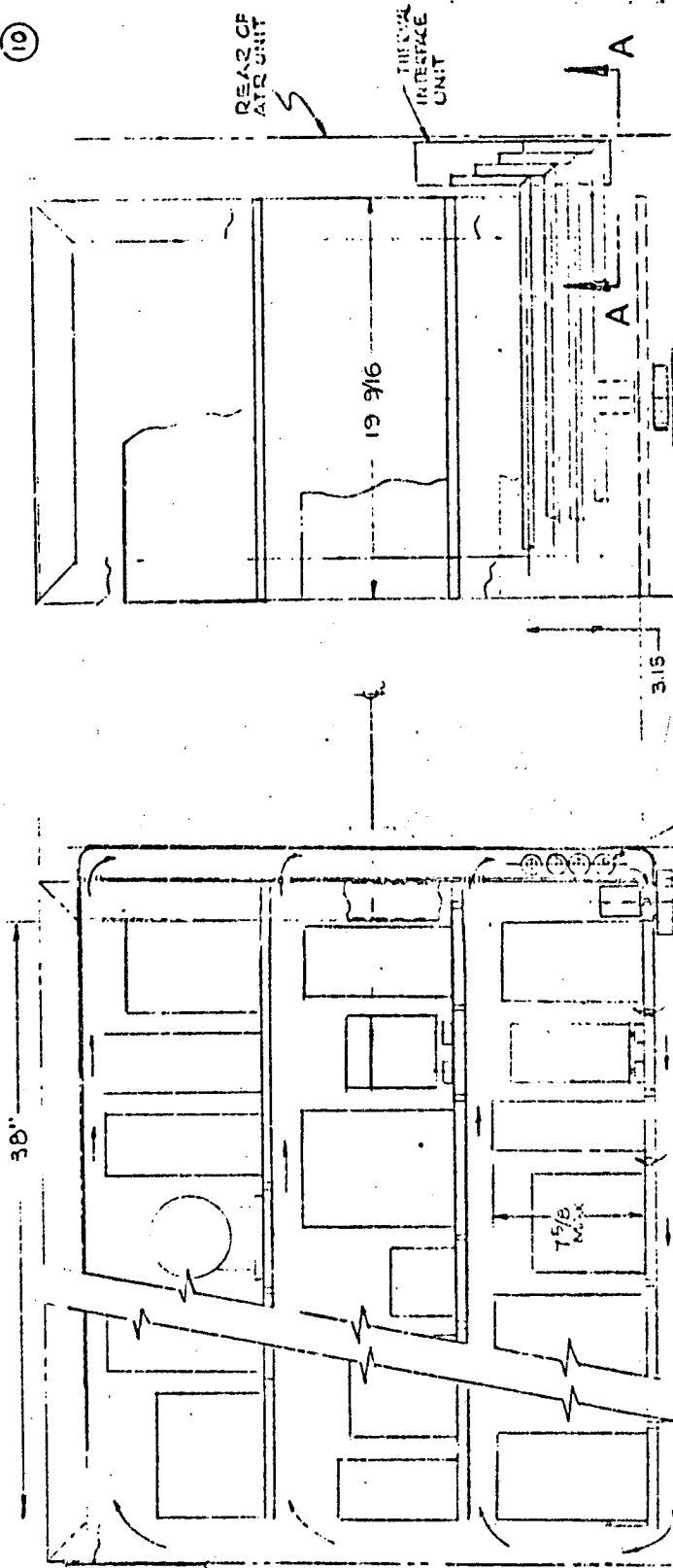
The proposed heat pipe designs would be more desirable than similar coolant system liquid-to-air exchangers in that they would provide more flexibility and eliminate flow splits, pressure drop or fluid connection problems associated with the liquid loop systems.

There are additional benefits in that some degree of inherent control (for example, the use of a VCEP) could be passively involved, giving better temperature regulation and thus more reliable electronics performance.

Improved temperature regulation and forced air circulation patterns would also preclude the need for additional thermal testing beyond the requirements of the governing specifications.

The use of heat pipes would also improve maintainability and reliability. The repair or removal of an ATR enclosure would be simplified since it would not require the opening or resealing of fluid lines. Also, a heat pipe puncture or leak would not endanger the electronics as much as a similar leak in a pumped loop heat exchanger, since it contains a limited amount of non-conductive distilled water as compared with the entire volume of relatively impure (pump contamination) ECS fluid.

10



REAR OF ATR UNIT

THERMAL INTERFACE UNIT

19 9/16

3.15

FINNED HEAT PIPE (4)
.500 INSIDE PIPE DIA
1.00 O.D. OF FIN

MAIN BLOWER

STRUCTURE

CONNECTOR AREA

REAR OF ATR UNIT

THERMAL INTERFACE UNIT
APPROX 14.52 WATTS (GAIN)

SECT A-A
(ROTATED 90° C'WISE)

VOL ELECTRONICS = 12 FT3
TOTAL Q_{LOSS} = 1452 WATTS

CC CONFIG I ~ CURRENT
ATR FORM FACTOR

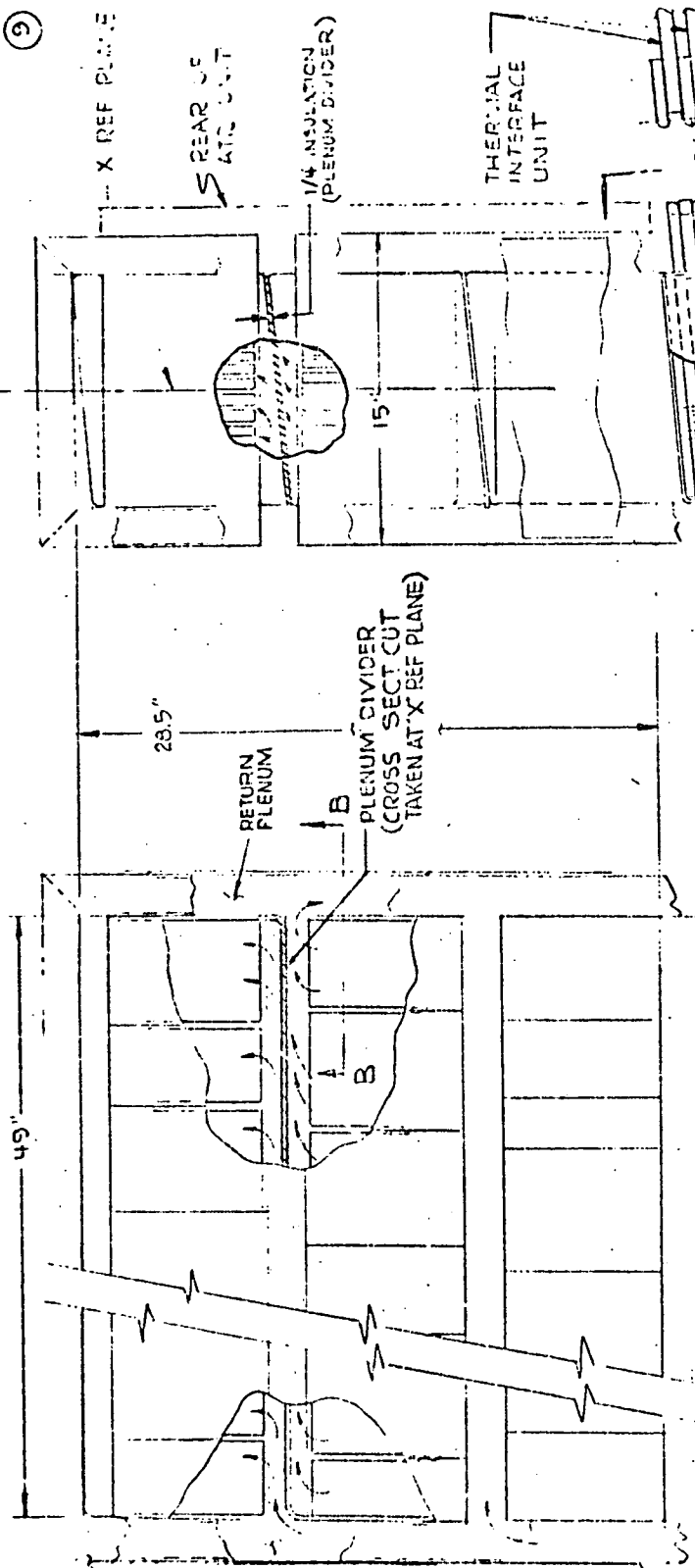
SPL-105 (SH 1 GF2)

SCALE 1/2C

FIG. 5-5



9



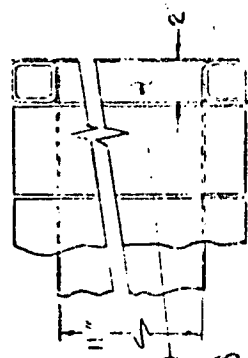
HEAT PIPE HEAT EXCHANGER MODULE

FINNED HEAT PIPE (3)
 .500" INSIDE DIA (PIPE)
 1.00" OUTSIDE DIA (FIN)

VOL ELECTRONICS = 12 FT³
 TOTAL Q_{DISS} = 1800 WATTS

CONFIG II - PROJECTED
 ADVANCED ATR FORM FACTOR

SPL-105(SH2 CF2)
 AIR/HEAT PIPE IN
 CONJUNCTION WITH ATR
 (SCALE 1/25)



RETURN PLENUM (RH SIDE)
 SECT B-B

5-6 - FLIGHT VOICE RECORDER/DATA RECORDER TEMPERATURE
CONTROL (SPL-111)

The flight data and voice recorders for the Shuttle are FAA Type III (ejectable, unrestricted location). They are currently located in the lower aft portion of the vertical fin as shown on SPL-111, sheet 1. The standard recorders are required to function in an ambient thermal environment from -68°F to $+160^{\circ}\text{F}$ and must not be adversely affected by exposure from -86°F to $+160^{\circ}\text{F}$. The published electrical power profiles (Grumman Shuttle Memo B35-160MO-61) call for continuous operation of both recorders (100% duty cycle) throughout the 7 day earth orbital mission (168 hrs.). The flight data recorder dissipates 21 watts and the voice recorder 20 watts.

The temperature extremes of the surrounding fin structure are:

100°F	:	Launch & Boost
-40°F to 110°F	:	On-Orbit
600°F Max	:	Re-entry

Thus, the surrounding structure can be used as a heat sink during on-orbit operations only.

Using conventional circulating fluid techniques (i.e., ECS loop) to control the temperature of the equipment would require long, out-of-the-way runs for the coolant lines. This creates additional pressure losses and flow balancing problems in the fluid loop system in the event that the recorders are relocated.

Heat pipes internal to a common cold plate type mounting surface can transfer heat from both recorders to either an intermediate sink or the local fin structure, as the situation demands. See sheet 2 for a typical installation. Heat flows from the recorders through the mounting plate to the internal heat pipes. These heat pipes, in turn, couple the mounting plate to diode heat pipes and also to the intermediate sink. During cold conditions, when the surrounding structure is cooler than the temperature of the equipment, the diode pipes complete the connection to structure, bypassing the intermediate sink, and creating a direct path from recorders to the structural sink. When the

structure gets warmer than the equipment the diode reverses, breaking the heat flow path to structure. Now the heat is transferred to the intermediate sink which can be either a phase change material (used if the recorders are on intermittently) or an expendable fluid such as water (used when operation is continuous). In either case, the intermittent sink and the recorders are well insulated from the structure to prevent their absorbing heat during extreme heating conditions.

The recorders are mounted so that the inertia vector assists the return of fluid from the condenser sections to the evaporator sections (mounting plate) of the heat pipes during boost and reentry mission phases. Sheet 2 shows a configuration using a phase change material as the intermediate sink. The configuration shown on sheet 3 uses an expendable fluid heat sink. This would be the only suitable intermediate sink for a 100 percent duty cycle and would require up to 21.4 pounds of water, depending upon how long the fin structure can be used as a heat sink.

The 3/8 inch I.D. heat pipes would employ a spiral artery wick using Freon-21 as the working fluid. Each pipe has a capacity of 200 watts, more than enough to satisfy operational requirements while also providing a good measure of redundancy in case one of the pipes malfunctions.

21

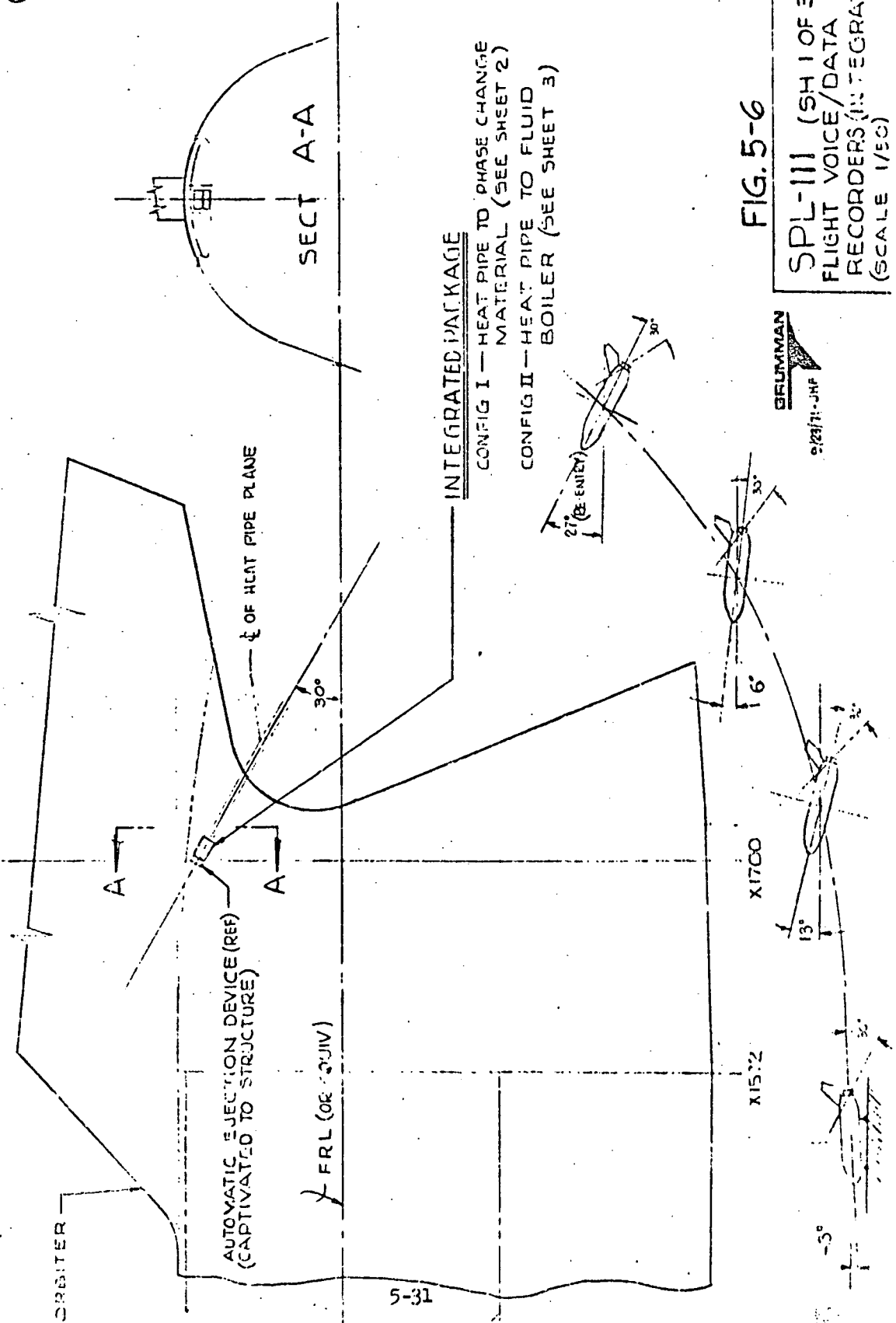


FIG. 5-6

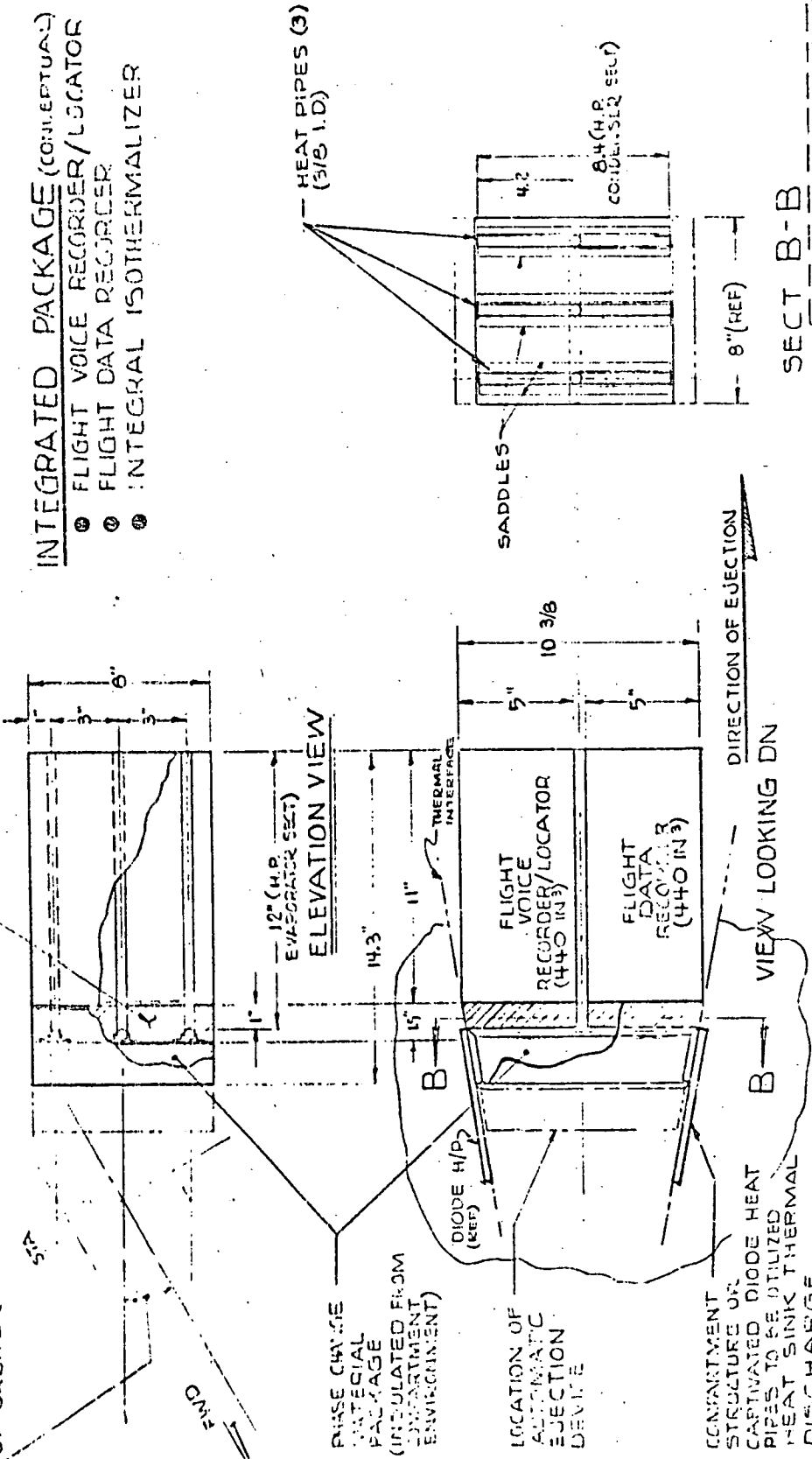
SPL-III (SH 1 OF 3)
 FLIGHT VOICE/DATA
 RECORDERS (INTEGRATED)
 (SCALE 1/50)

INSULATION (SHADED AREA)

NOTE: PLANE OF HEAT PIPES TO BE 30° TO FUSelage REF LINE (FRL) AS SHOWN WHEN INSTALLED

INTEGRATED PACKAGE (CONCEPTUAL)

- ① FLIGHT VOICE RECORDER/LOCATOR
- ② FLIGHT DATA RECORDER
- ③ INTEGRAL ISOTHERMALIZER



HEAT PIPES (3)
(3/8 I.D.)

4.2
8.4 (H.P. CO-IDE. SEC. 1)

SADDLES

8" (REF)

SECT B-B

FLIGHT VOICE/DATA RECORDER

SPL-111 (SH 2 CF 5)
(SCALE 1/4)

CONFIG I - H/P TO PHASE CHANGE MAT'L

9/23/71-JHIF

GRUMMAN

PHASE CHANGE MATERIAL PACKAGE (INSULATED FROM ENVIRONMENT)

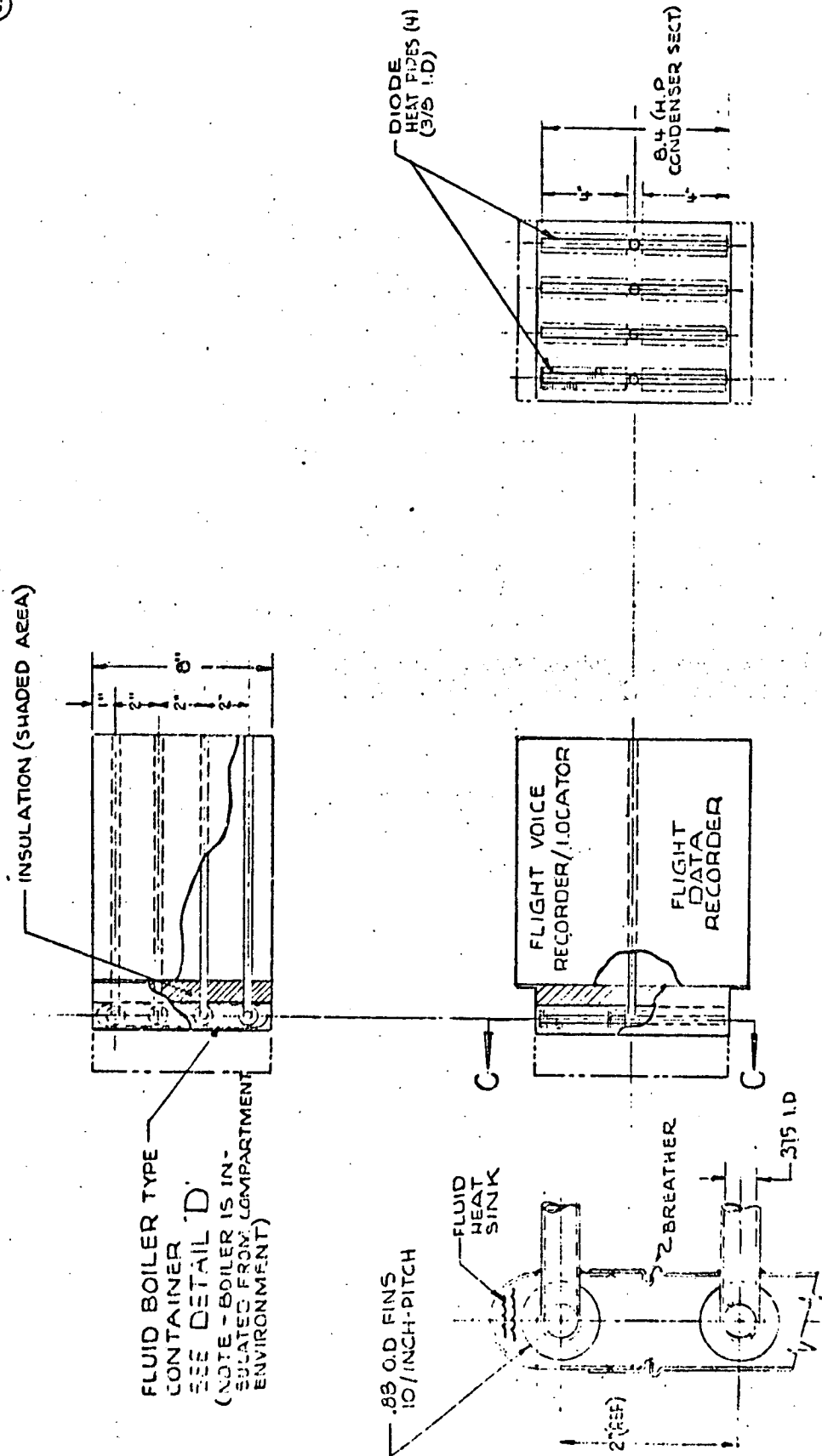
LOCATION OF AUTOMATIC EJECTION DEVICE

COMPARTMENT STRUCTURED OF CAPTIVATED DIODE HEAT PIPES TO BE UTILIZED IN HEAT SINK THERMAL DISCHARGE

DIRECTION OF EJECTION

VIEW LOOKING DN

23



SECT C-C
 FLIGHT VOICE/DATA RECORDED
 SPL-III (SH 3 OF 3)
 (SCALE 1/4 & NOTED)

GRUMMAN
 9/23/71-JHF

CONFIG II-H/P TO FLUID BOILER
 (SAME AS CONFIG I-EXCEPT AS SHN)

DETAIL D
 (SCALE-FULL)

5-7 - HEAT PIPE RADIATOR SYSTEM (SPL-101)

The baseline concept uses fluid-loop radiators in orbit to dissipate the heat generated by the fuel cells and various cabin and remote electronic systems. Alternatively, the radiators (and related headers) could be made using heat pipes.

The fuel cells and "high temperature electronics" must be maintained at temperatures in the 85 - 90°F range; the "low-temperature" cabin electronics must be maintained at around 4°C. Although it is possible to operate all the radiator panels at the same temperature, it is more efficient to segregate the load into two radiator systems: a high-temperature system and a low-temperature system.

The baseline radiator area of 700 ft² is divided among 10 panels, 5 on each cargo bay door, the forward two panels on each side (the high-temperature panels) being 9' 10" by 6' 5", and the three aft panels (the low-temperature panels) 11' 1" by 6' 5".

Heat is brought from the internal heat transport system to the radiators by means of 1" diameter variable conductance heat pipe headers, and transferred throughout the panel by means of small ($\frac{1}{2}$ " diameter) feeder heat pipes.

The six "cold" (~ 60°F) radiator panels must dissipate a load of 32,500 Btu/Hr from the low temperature electronics. This is 3,750 Btu/hr or 1,130 watts per panel (and therefore per header). Computer studies indicate that an ammonia-filled heat pipe, with an eleven-foot condenser, a twenty-five inch transport section, and a four-foot evaporator has a capacity of over two kilowatts.

Sheet 2 of SPL-101 shows one possible configuration for the twenty-five inch transport section. It assumes heat is brought to the evaporator of the header at the shuttle wall, then carried around a flexible transport section consisting of a wick inside flexible hosing, and finally given up to the feeder pipes in the radiator panel by the condenser section.

Several alternate headers are being considered, with rotating conductive couplings between heat pipes (rather than direct transport within a heat pipe) at the hinge line.

One such coupling is shown on sheets 3 and 4 of SPL-101. The radiator header would have its condenser section unchanged, would require no transport section, and would have an evaporator section shaped like a trapezoidal flat plate rather than a cylinder. It would be placed on the hinge line, and on either side of it, in a "sandwich-like" configuration, would be placed identically shaped condenser sections of two other heat pipes fixed in position relative to the shuttle wall and carrying heat to the hinge line from the internal heat transfer circuit. Heat would be transferred across the hinge line by conduction from these condenser to the evaporator of the header. (The requirement that there be no projections through the skin when the cargo bay doors are closed dictates the shape of the evaporator and condensers involved.)

A variable conductance heat pipe header was chosen to provide control. As the heat production of internal systems decreases, the operating temperature of the header drops, causing the non-condensable gas in the reservoir to expand, blocking off a part of the condenser section. This, in effect, cuts off part of the radiator area and so prevents the working temperature from dropping too low, acting as a control device on the radiator. By properly designing the header, it would be possible to cut the radiator off entirely in no-load conditions.

From the condenser section of the header, heat is picked up by a large number of small feeder pipes extending out into the radiator panel, spaced at six-inch intervals. As shown, each feeder pipe has an evaporator length of $5 \frac{3}{4}$ " and a condenser length of 6' 5". Computer studies show that such a pipe, with an ammonia fill, is capable of transporting over 600 watts; the load it would be required to carry is 51.3 watts (1,130 watts divided by 22, the number of feeder pipes per "cold" radiator).

TABLE 5-7 HEAT PIPE POSTER FOR RADIATOR

Heat Pipe Name	Dimensions	Fluid	Wick	Number Required and Placement	Computed Max Capacity (watts)	Required Capacity (watts)	ΔT at Required Capacity
Header (Sheet 2) - Flexible Transport Section, to "Cold" Panel	1" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$, $L_{COND} = 133"$ OR	NH ₃	GAC Spiral Artery/Tunnel	6 - 1 per each "Cold" Radiator Panel	3,581	1,130	4.5°
	3/4" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$, $L_{COND} = 133"$						
- Flexible Transport Section, to "Hot" Panel	1" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$, $L_{COND} = 133"$ OR	NH ₃	GAC Spiral Artery/Tunnel	4 - 1 per each "Hot" Radiator Panel	3,659	1,056	4.3°
	3/4" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$, $L_{COND} = 118"$						
ALTERNATE							
Header - Conductive Coupling Movable Header, to "Cold" Panel (Sheet 3)	1" ID, $L_{COND} = 133"$ $L_{EVAP} = 0"$ $L_{TRANS} = 2$: trapezoidal Flat Section	NH ₃	GAC Spiral Artery/Tunnel	4 - 1 per each "Cold" Radiator Panel	TBD	1,130	6.2°
	1" ID, $L_{COND} = 118"$ $L_{EVAP} = 0"$ $L_{TRANS} = 2$: trapezoidal Flat Section						
- Conductive Coupling Movable Header, to "Hot" Panel (Sheet 3)	1" ID, $L_{COND} = 118"$ $L_{EVAP} = 0"$ $L_{TRANS} = 2$: trapezoidal Flat Section	NH ₃	GAC Spiral Artery/Tunnel	4 - 1 per each "Hot" Radiator Panel	TBD	1,056	5.8°
	1" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$ $L_{COND} = 2$: trapezoidal Flat Section						
- Conductive Coupling Fixed Header (Sheet 4)	1" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$ $L_{COND} = 2$: trapezoidal Flat Section	NH ₃	GAC Spiral Artery/Tunnel	20 - 2 per each Movable Header	TBD	565 (to "Cold" Panel)	4.2°
	1" ID, $L_{EVAP} = 48"$ $L_{TRANS} = 89"$ $L_{COND} = 2$: trapezoidal Flat Section						
Feeder	1" ID, $L_{EVAP} = 5.75"$ $L_{TRANS} = 0"$, $L_{COND} = 77"$ OR Freon-21	NH ₃	GAC Spiral Artery/Tunnel	132 - 22 per "Cold" Radiator Panel	1,061	51.3	3.2°
				80 - 20 per "Hot" Radiator Panel	1,061	52.8	3.3°
				132 - 22 per "Cold" Radiator Panel	435	51.3	7.4°
				80 - 20 per "Hot" Radiator Panel	433	52.8	7.6°

*Recent data indicates an improvement of (at least) a factor of two in these figures, i.e., one-half the temperature drop.

2

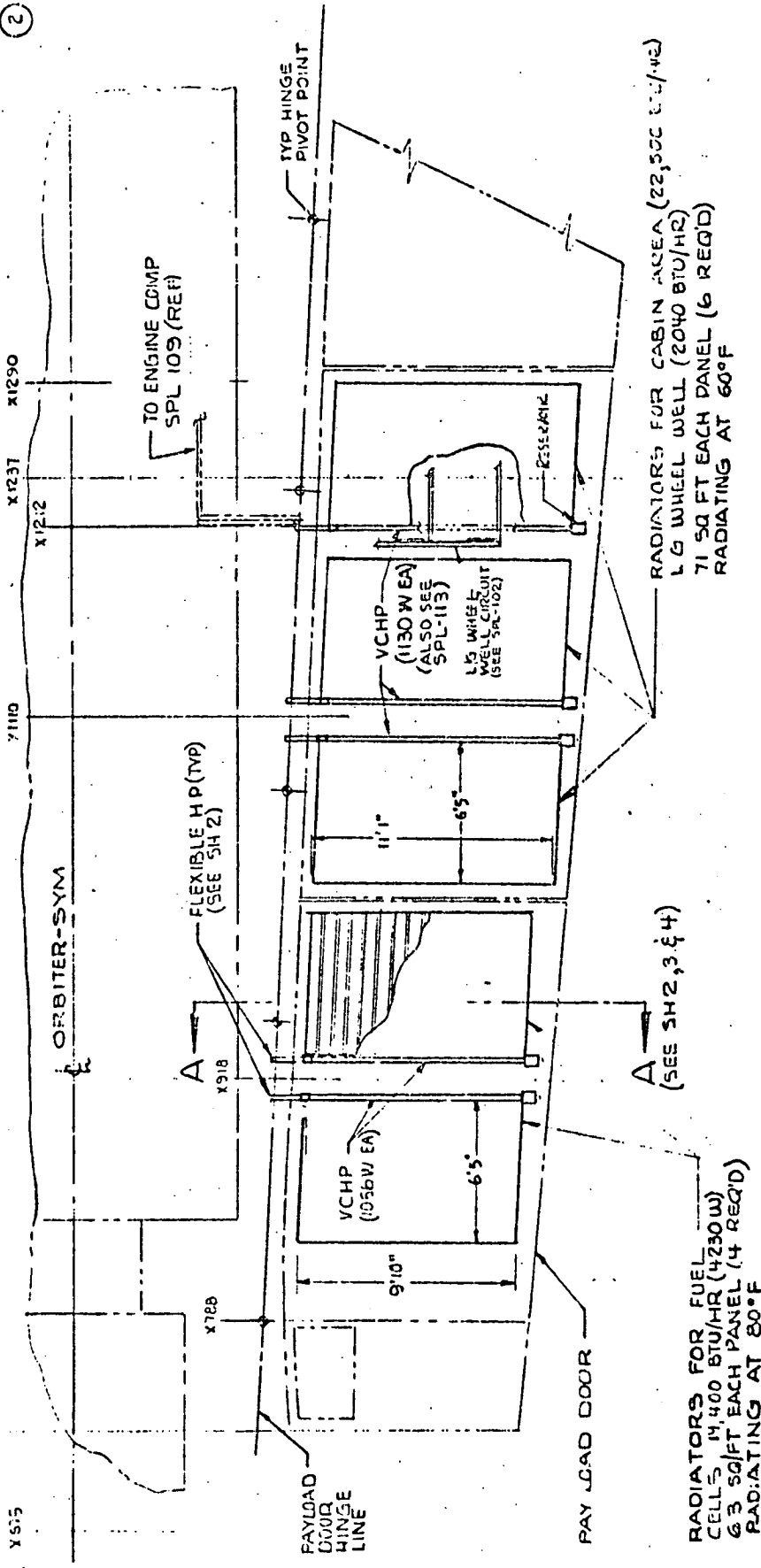


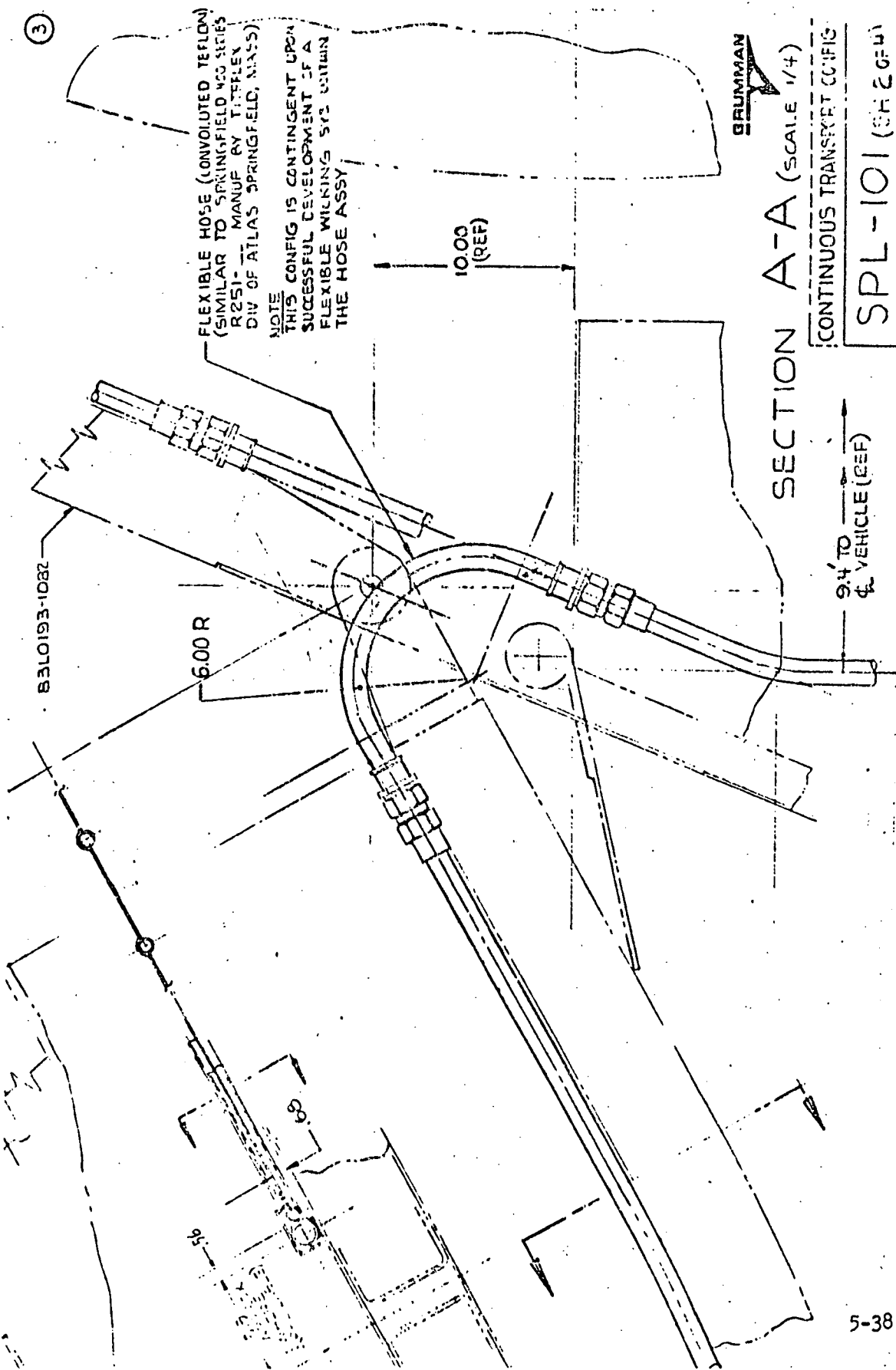
FIG. 5-7

SPL-101 (SH 1 OF 4)
 HEAT PIPE RADIATORS
 ON PAYLOAD DOORS
 (SCALE 1/50)



VIEW LOOKING DN-LH SIDE

3



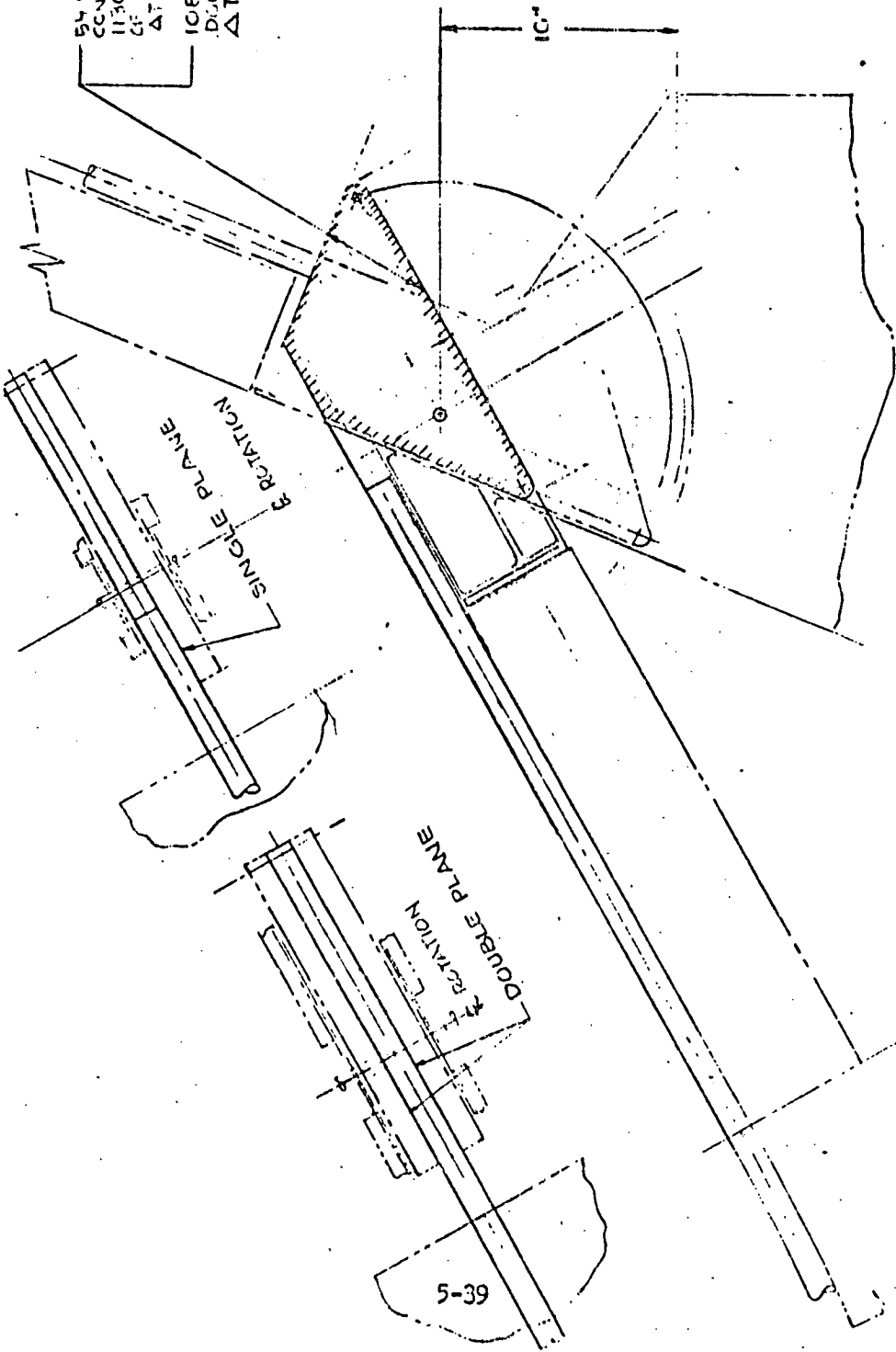
FLEXIBLE HOSE (CONVOLUTED TEFLON)
 (SIMILAR TO SPRINGFIELD HQ3 SERIES
 R251- MANUP BY TITFLEX
 DIV OF ATLAS SPRINGFIELD, MASS)

NOTE
 THIS CONFIG IS CONTINGENT UPON
 SUCCESSFUL DEVELOPMENT OF A
 FLEXIBLE WICKING SYS WITHIN
 THE HOSE ASSY

4

54 SO INCHES (SINGLE PLANE)
CONDUCTIVE SURFACE
1130 VAC'S APPROXIMATE
CF 1520 BTU/HR/FT²
AT 3.68°F

108 SO INCHES
DOUBLE PLANE
AT 3.45°F



5-39



CONDUCTIVE TRANSPORT CONFIG

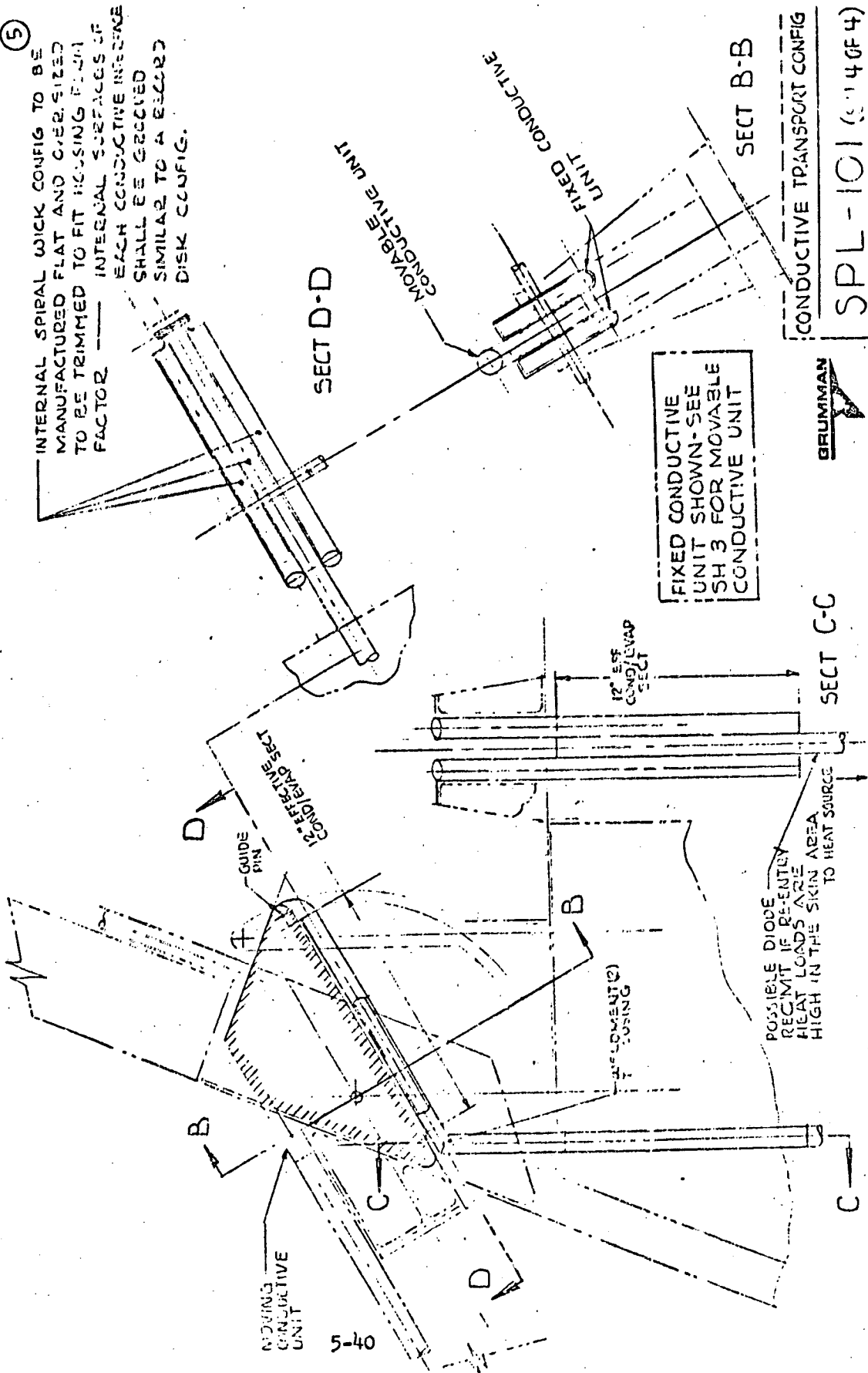
SPL-101 (SH 3CF4)

SECTION A-A (SCALE 1/4)

MOVABLE CONDUCTIVE UNIT SHOWN - SEE SH 4 FOR FIXED CONDUCTIVE UNIT

5

INTERNAL SPIRAL WICK CONFIG TO BE MANUFACTURED FLAT AND OVER-SIZED TO BE TRIMMED TO FIT HOUSING DESIGN FACTOR — INTERNAL SURFACES OF EACH CONDUCTIVE INTERFACE SHALL BE GROOVED SIMILAR TO A RECORD DISK CONFIG.



FIXED CONDUCTIVE UNIT SHOWN - SEE SH 3 FOR MOVABLE CONDUCTIVE UNIT



SPL-101 (REV 4 OF 4)

5-8 - HEAT PIPE RADIATOR WITH INTEGRAL H/P
FLUID HEADER (SPL-113)

An alternate approach to the heat pipe radiator is shown in drawing SPL-113. This configuration assumes that a pumped fluid loop system has been chosen as the basic heat transport mechanism from heat sources to the radiators, and it is desired to connect a variable conductance heat pipe-controlled radiator to this system as the heat rejection device.

The use of VCHP control has advantages over conventional fluid loop bypass and regenerative systems in that it is self-controlled - there is no need for external monitoring or control devices. Reliability should be very high, as there are no moving parts.

A proposed configuration for the VCHP header/fluid header interface is shown in sheet 2 of SPL-113. The evaporator section of the heat pipe header is immersed in the fluid and held in place by means of many small fins which are brazed to the outside of the heat pipe to increase the heat transfer. The condenser section of the heat pipe header lies above the fluid manifold, with an insulating block between them. Small feeder heat pipes extend perpendicularly from the heat pipe header into the radiator panel, spaced at six-inch intervals.

There are two panel configurations to be studied. On a 132 inch by 77 inch radiator panel, with the short dimension parallel to the cargo bay door hinge line, the header could be placed either along the short dimension (parallel to the hinge line, as shown on sheet 2) or along the long dimension (perpendicular to the hinge line, as shown on sheet 3). The former, having a shorter exposed fluid header, requires less meteoroid protection and therefore is lighter; the latter, though it exposes 264 inch of the pumped fluid line to space (fluid header and return), gives better thermal performance because of the increased area of fluid-to-heat pipe contact.

In either configuration, the ammonia-filled heat pipe header can easily carry the required load of 1200 watts per panel: the long header has a capacity of 4,300 watts and the short header of 6,450 watts.

In both cases, the feeder pipes have an evaporator length of 8.75 inches; the feeder pipes used with the long header have a condenser length of 77 inches, and those used with the short header have a condenser length of 132 inches. Of these two feeders, the longer would be required to carry 92.4 watts and has a calculated capacity of 815 or 330 watts (with ammonia and Freon-21 fills, respectively), and the shorter has a required load of 54.6 watts and a calculated capacity of 1,260 watts or 460 watts (with respective ammonia and Freon-21 fills).

Use of a variable conductance heat pipe header for thermal control requires the heat pipe operating temperature, and therefore the entire radiator panel, to be below the fluid outlet temperature. This results in a less thermally efficient radiator panel than a pumped fluid loop panel, part of which would be operating at a relatively high temperature. If this low operating temperature requirement should make the shuttle radiator area insufficient to dissipate the required heat energy, an alternate design not having this restriction, consisting of a fluid header (requiring conventional controls) and heat pipe feeders, will be investigated.

Two such configurations are shown on sheet 4, having possible advantages over the baseline system. As meteoroid bumpers are unnecessary except for the fluid header, the configurations shown may be lighter than the baseline; fluid pressure drops in these systems must be compared to those in the baseline system. By employing a hybrid system, where only some of the feeders are VCHP's, much of the requirement for conventional-type thermal control may be eliminated. Another approach would rely on load segregation or series panel arrangements resulting in each panel radiating at a different temperature. For example, one panel would have a 100°F fluid inlet with a 90°F outlet, the next panel a 90°F and etc.

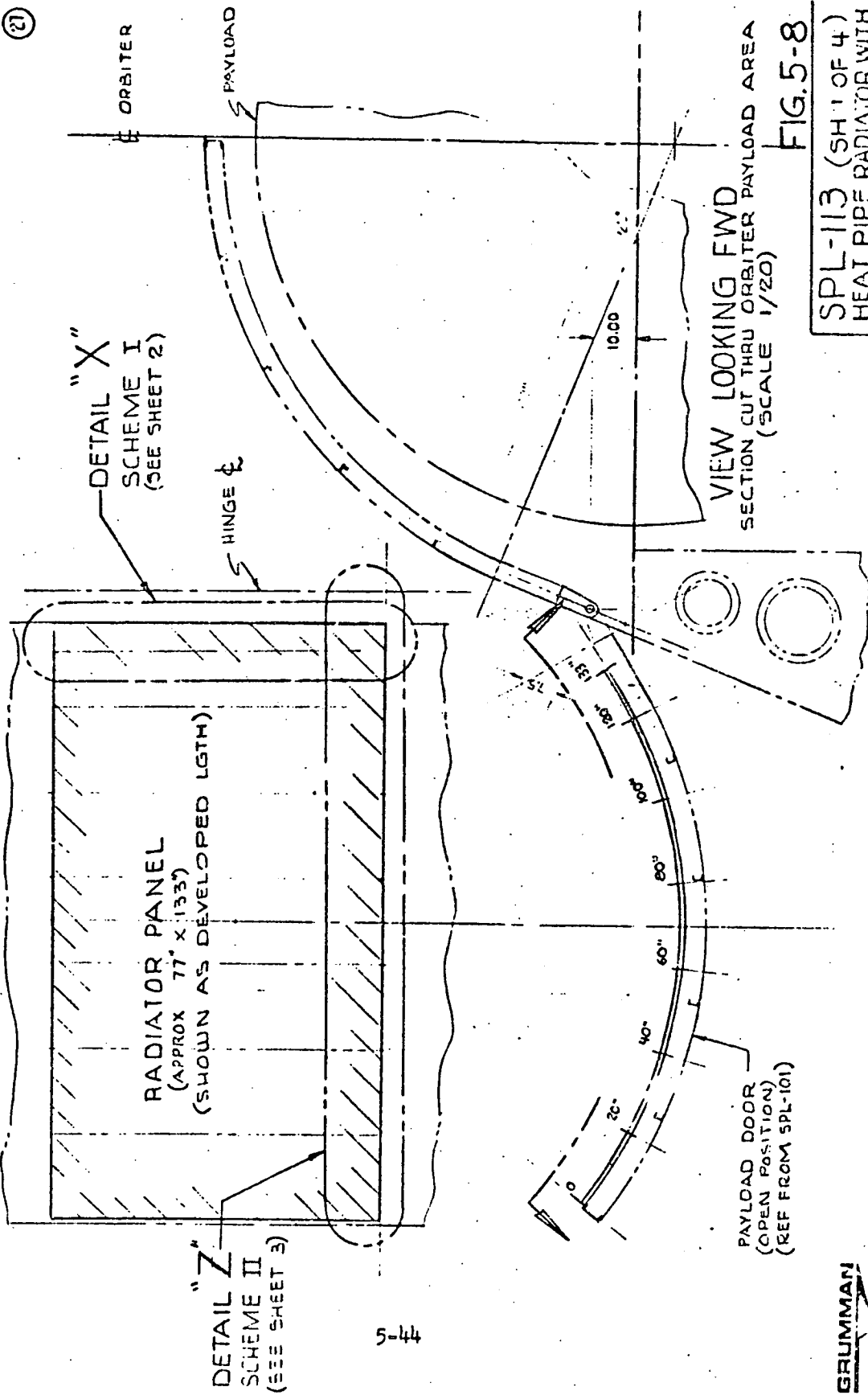
Any of these alternate systems would use feeder heat pipes similar to those of the fluid/heat pipe header design, so capacities and requirements would be similar.

TABLE 5-8: ROSTER FOR RADIATOR WITH HP/FLUID HEADER

Heat Pipe Name	Dimensions	Fluid	Wick	Number Required and Placement	Computed Max Capacity (Watts)	Required Capacity (Watts)	ΔT A Requir Capac
Long Header	1" ID, 132" Evaporator, 132" Condenser	NH ₃	GAC Spiral Artery/Tunnel	10 - 1 per Radiator Panel	4,300	1,200	2.1
Short Header	1" ID, 77" Evaporator, 77" Condenser	NH ₃	GAC Spiral Artery/Tunnel	10 - 1 per Radiator Panel	6,450	1,200	3.6
Short Feeder	1/2" ID, 5.75" Evaporator, 77" Condenser	NH ₃ —OR Freon-21	GAC Spiral Artery/Tunnel	220 - 22 per Radiator Panel with Long Header	1,260 460	54.6	0.6 1.0
Long Feeder	1/2" ID, 5.75" Evaporator, 132" Condenser	NH ₃ —OR Freon-21	GAC Spiral Artery/Tunnel	130 - 13 per Radiator Panel with Short Header	815 330	92.4	0.7 1.1

*Recent data shows (at least) a factor of two improvement, i.e., one-half the ΔT .

21



DETAIL "X"
SCHEME I
(SEE SHEET 2)

DETAIL "Z"
SCHEME II
(SEE SHEET 3)

RADIATOR PANEL
(APPROX 77" X 133")
(SHOWN AS DEVELOPED LGTH)

HINGE

ORBITER

PAYLOAD

VIEW LOOKING FWD
SECTION CUT THRU ORBITER PAYLOAD AREA
(SCALE 1/20)

FIG. 5-8

SPL-113 (SH 1 OF 4)
HEAT PIPE RADIATOR WITH
HEAT PIPE FLUID HEADER

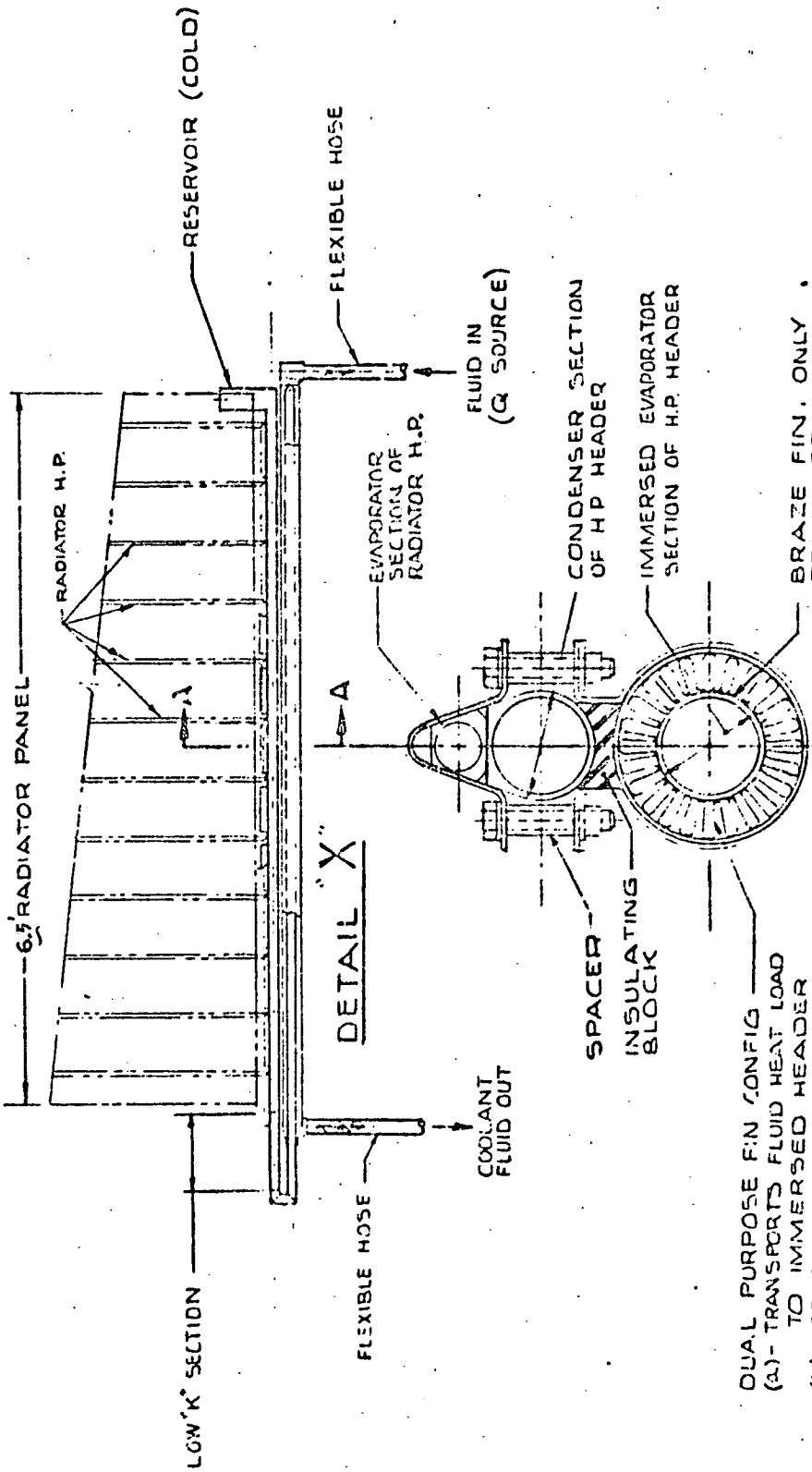
PAYLOAD DOOR
(OPEN POSITION)
(REF FROM SPL-101)

5-44



9/23/71-JHF

(29)



QUAL PURPOSE FIN CONFIG
 (A) - TRANSPORTS FLUID HEAT LOAD
 TO IMMERSED HEADER
 (B) - CONCENTRICALLY SUPPORTS
 IMMERSSED HEADER WITHIN
 FLUID LINE

SECTION A-A
 SCALE - FULL

GRJMMAN

9/29/71-JHF

SCHEME I

HP/RAD WITH INTGL HP/FLUID HCR

SPL-113 (SH 2 OF 4)

25

HINGE

DETAIL Z
(DEVELOPED)

RESERVOIR
FLEXIBLE HOSE (2)
FLUID RETURN
FLUID SUPPLY

RADIATOR
RADIATOR FIN PORTION
EVAPORATOR
CONDENSER
HEAT PIPE HEADERS
FLUID SUPPLY
FLUID RETURN

SECT C-C

SCHEME II

HP/RAD WITH INTEL HP FLUID HDR

SECTION B-B (ROTATED CW 90°)
(SAME AS SECT A-A EXCEPT AS SHN)

SPL-113 (SH 3 OF 4)



BUMPER

B

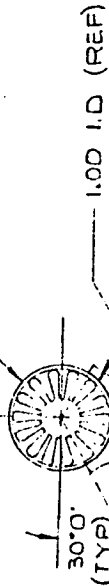
B

SCHEME III -- FLUID HEADERS

30

INTERFACED DIRECTLY TO RADIATOR HEAT PIPES

FLUID HEADER - DUAL



CONDUCTIVE SURFACE AREA EQUIV TO 2 SQ/IN PER INCH OF LENGTH
BOND DUAL FLUID HEADER TO SADDLE SHAPED HEAT PIPE WITH HYSOL EPOXY (SILVER FILL)

SADDLE SHAPED HEAT PIPE (EVAPORATOR SECTION)

SECTION D-D (SCALE - FULL)

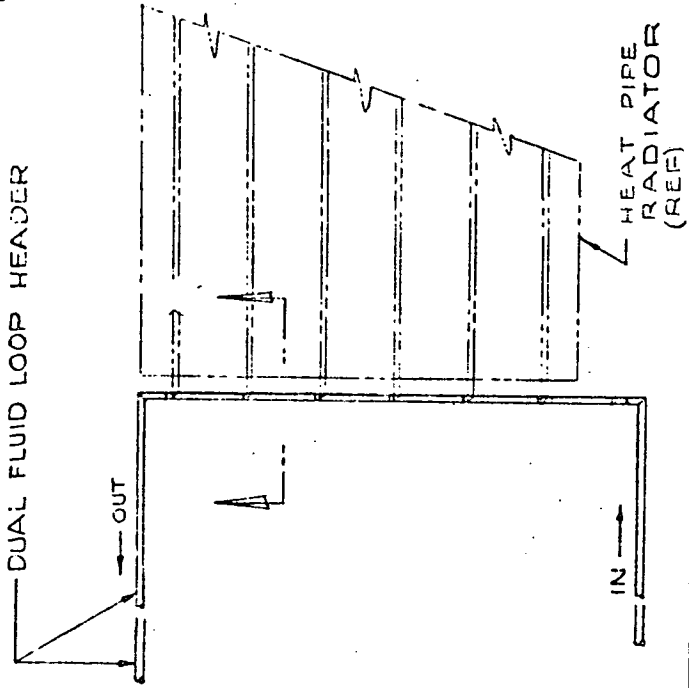
FLUID HEADER SHAPE (EXTRUDED SHAPE WITH INTERNAL FIN CONFIG. BEZEL TO ONE SURFACE OR INTEGRALLY EXTRUDED)

MARMAN TYPE CLAMP FINN - BEZEL TO THE FLAT INSIDE SURFACE

WIDE 120° TAPER INTERFACE PERMITS SELF ALIGNMENT, GREATER SURFACE CONTACT AREA AND NUT SIZE TYPE INSTL FOR EASY REMOVAL

EVAPORATOR SECTION TO BE MANUF FROM EXTRUDED SHAPE

RADIATOR HEAT PIPE HAS THE ADVANTAGE OF QUICK AND EASY REPLACEMENT



GRUMMAN

9/12/71-JMF

SCHEME III

HP/RAD WITH INTGL HP/FLUID HDR

SPL-115 (SH 4 CF 4)

SECTION D-D (ALTERNATE)
(SCALE - FULL)

5-9 - HEAT PIPE AUGMENTED COLD RAIL

Examination of the Power Conditioning Electronics proposed for the Space Shuttle Orbiter depicts dissipation levels of some of the flange-mounted modules in the range of 40 watts per linear inch per side. This value is approximately 16 times greater than the maximum design values reflected in the Lunar Module cold rail design and exceeds the capability of simple fluid cold rails. If not thermally corrected, this condition would necessitate a less efficient (in terms of weight and volume) redesign of electronics packages.

An evaluation of the entire power inverter assembly, which contains the high dissipation component, showed total dissipation for its 20-inch length to be 260 watts. This can be further translated to an approximate requirement of 200 watts for the entire 20-inch rail. This range (200 w/ 20 in., or 10 w/in.) is not far from present cold rail capacities considering the use of both coolant passages simultaneously. Evaluation of cold rail designs showed the limiting parameter to be the resistance between the short length of heat input section on the rail and the wetted coolant interface.

An improvement would be achieved by spreading or diffusing heat to increase the effective coolant wetted area. An increase in rail material thickness is simple but would cause an undesirable weight penalty. Using a heat pipe to provide high-capacity longitudinal isothermalizing is simple and light. Figure SPL-104 depicts the proposed configuration. The heat pipe is an integral part of the extruded rail and it serves to distribute a localized high heat load over the length of the rail.

All dimensions for coolant passages and rail thicknesses are sized for approximate capacity. An optimization of capacity, width, and weight trade-off will be performed during detailed design.

The capacity of the rail is to be defined by two parameters:

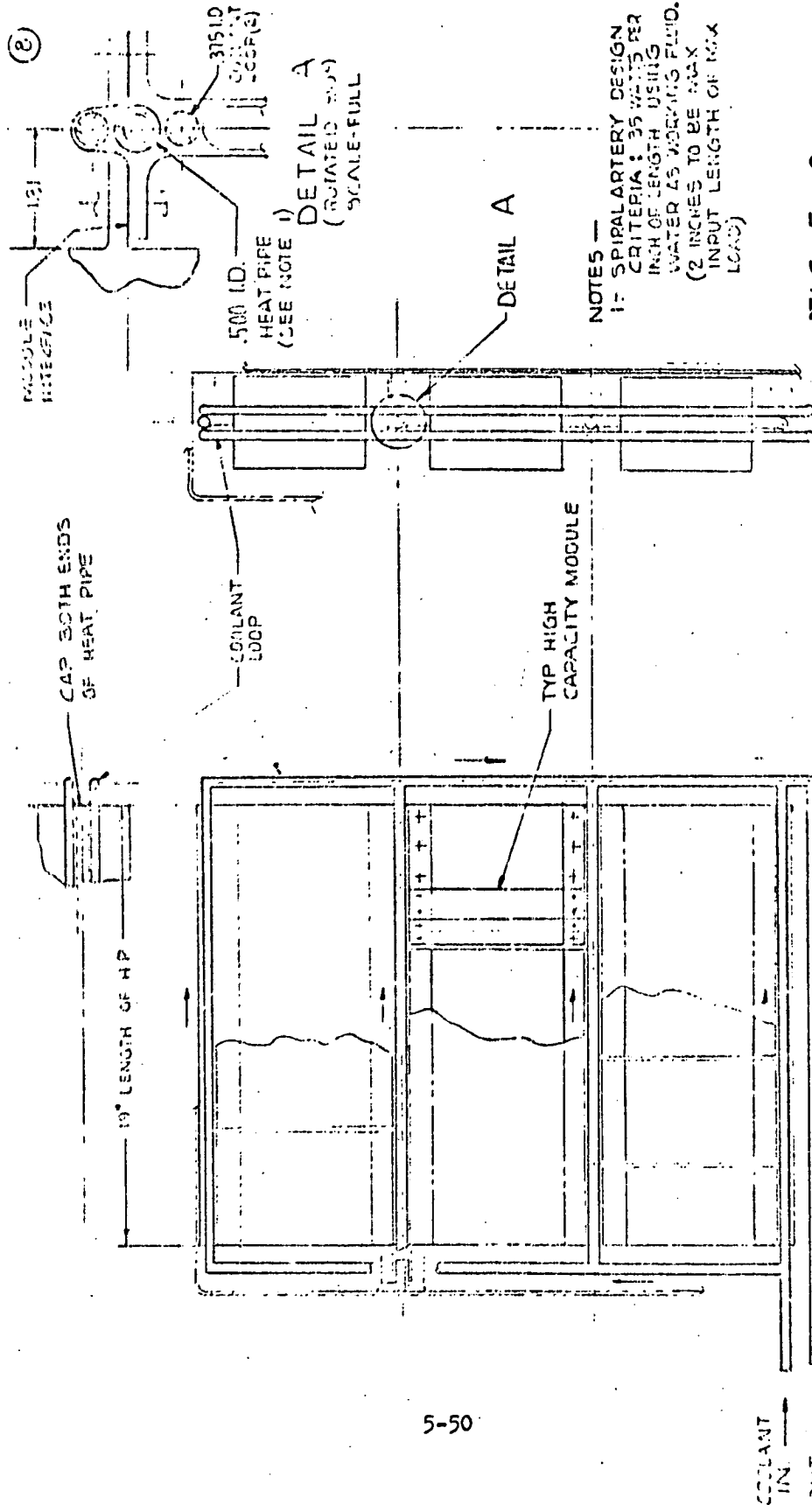
- (1) Peak localized input, limited to any two-inch long section of the rail's mounting flange, and

- (2) The total maximum average heat absorption rate applicable to the entire rail length.

Both parameters must be satisfied for adequate rail performance. The overall rail capacity is also a function of the mean coolant temperature.

Heat Pipe Augmented Cold Rail Summary

Length:	20 inches
Coolant Passages:	(2)--3/8", I.D. extended fin
Heat Pipe:	1/2" diameter Spiral Artery Heat Pipe--Water fill
Total Load Capacity:	200 W
Max Local Dissipation Density:	40 W/inch for 2" maximum
Operating Temperature	40 to 90°F



NOTES —
 1- SPIRAL ARTERY DESIGN
 CRITERIA: 35 WATTS PER
 INCH OF LENGTH USING
 WATER AS WORKING FLUID.
 (2 INCHES TO BE MAX
 INPUT LENGTH OF MAX
 LOAD)

FIG. 5-9

SPL-104 (SH 1 CF 1)
 HIGH CAPACITY COLD RAIL
 SCALE: 1/4" = 1"



MODULE EQUIPMENT RACK (SCALE: 1/4")
 APPLICABLE TO
 ITEM 7 - POWER DISTRIBUTION & CONTROL
 ITEM 3 - POWER CONDITIONING (ELECTRONICS)

COOLANT
 IN
 OUT

5-10 - HP RADIATOR FOR FUEL CELL (SPL-110)

The fuel cells, as provided by the vendor, have an internally pumped fluid loop cooling system using FC-75 as the coolant, which is connected through a heat exchanger interface to the user's heat rejection system.

An all heat pipe system for transporting the waste heat away from the exchanger to high-temperature radiators is discussed in Section 5-3, and the high-temperature radiator itself is described in Section 5-2.

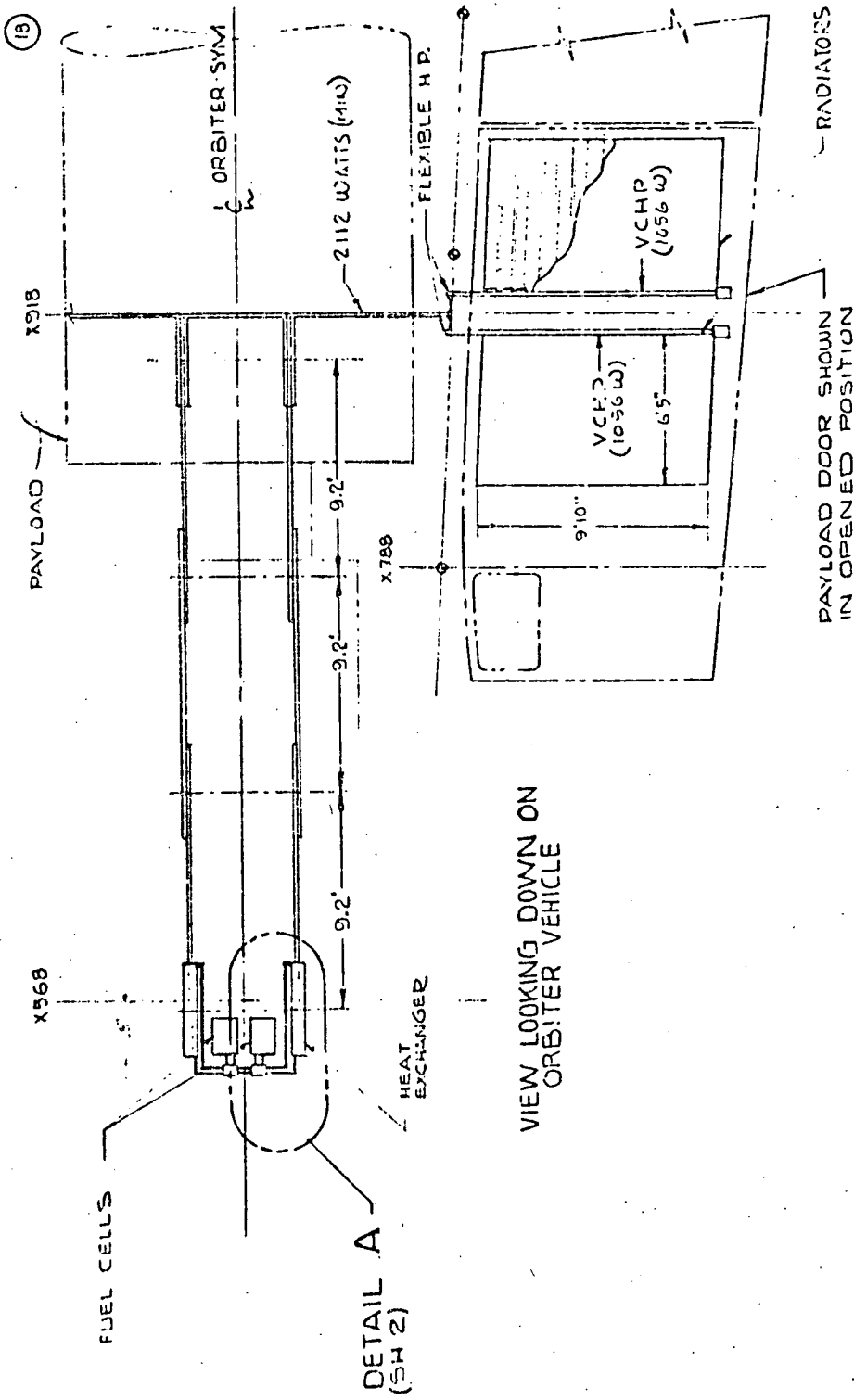
The total system is shown on sheet 1 of SPL-110. Sheet 2 is a more detailed view of the fuel cell and heat exchanger arrangement. Hot FC-75 is pumped out of the fuel cell by an internal pump, through a servo valve, and then through a heat exchanger where the fluid is cooled by contact with the finned evaporator section of a heat pipe. The cooled FC-75 then returns to the fuel cell.

Sheet 3 shows two sample approaches for design of the FC-75/heat pipe heat exchanger. The basic design concept encloses the heat pipe evaporator in a baffled chamber through which the hot FC-75 flows.

The structure of the heat exchanger on the left is like that of a conventional shell-and-tube heat exchanger with segmental baffles. In addition to causing the fluid to pass repeatedly over the heat pipe, the baffles, if brazed or welded to the pipe, would increase efficiency by acting as fins.

An alternate approach, shown on the right of sheet 3, consists of a concentric arrangement of a heat pipe and a cylindrical manifold, with a spiral-like fin around the heat pipe. This fin would act as an extended surface and, by making the FC-75 flow in a spiral about the heat pipe instead of along the tube in the axial direction, increase the length of the fluid path. Both effects tend to increase the amount of heat transferred.

The choice of these or other configurations depends upon the as yet unanalyzed thermal performance of each of them, and the pressure drop in the FC-75 across each of them.

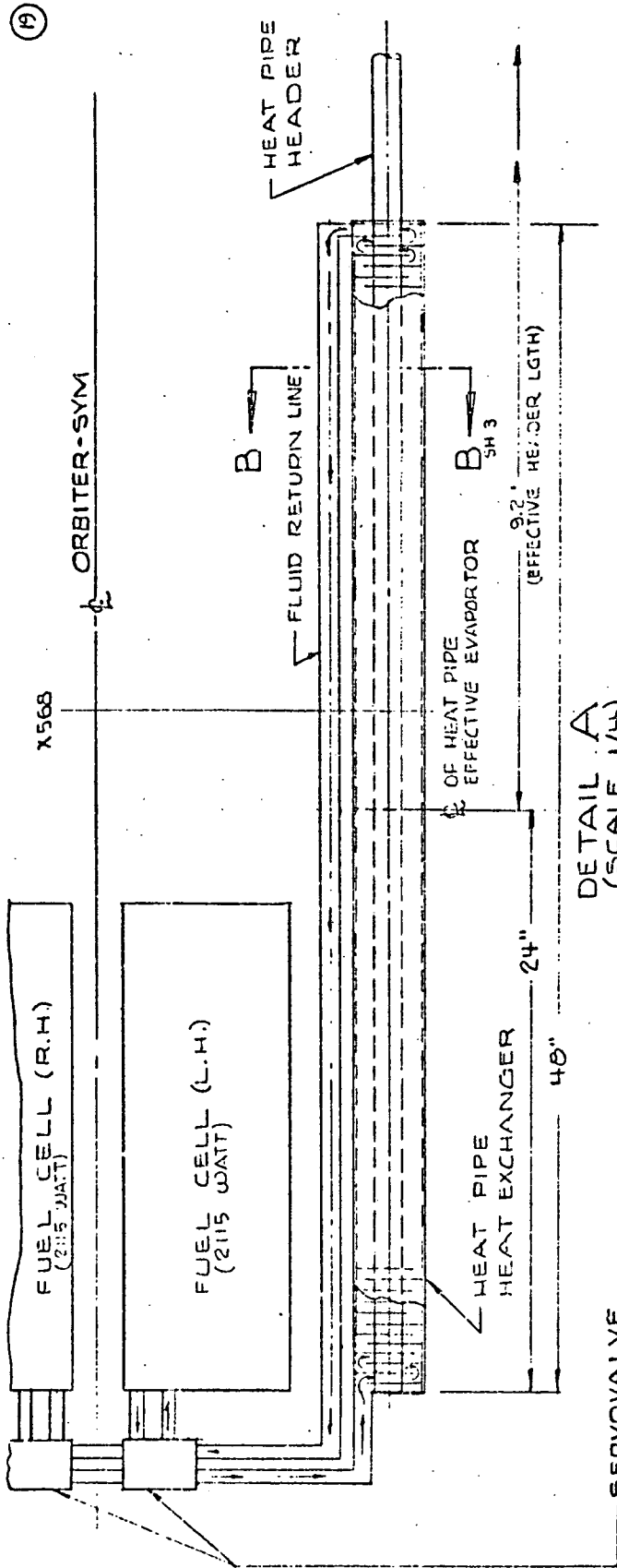


VIEW LOOKING DOWN ON ORBITER VEHICLE

FIG. 5-10

SPL-110 (SH10F3)
 HEAT PIPE / HEAT EXCHANGER
 CIRCUIT FROM FUEL CELLS
 TO RADIATORS (SCALE 1/50)





SERVOVALVE
 (FOR COOLANT FLUID CONTROL WITH
 ADDITIONAL SWITCH-OVER FEATURES)

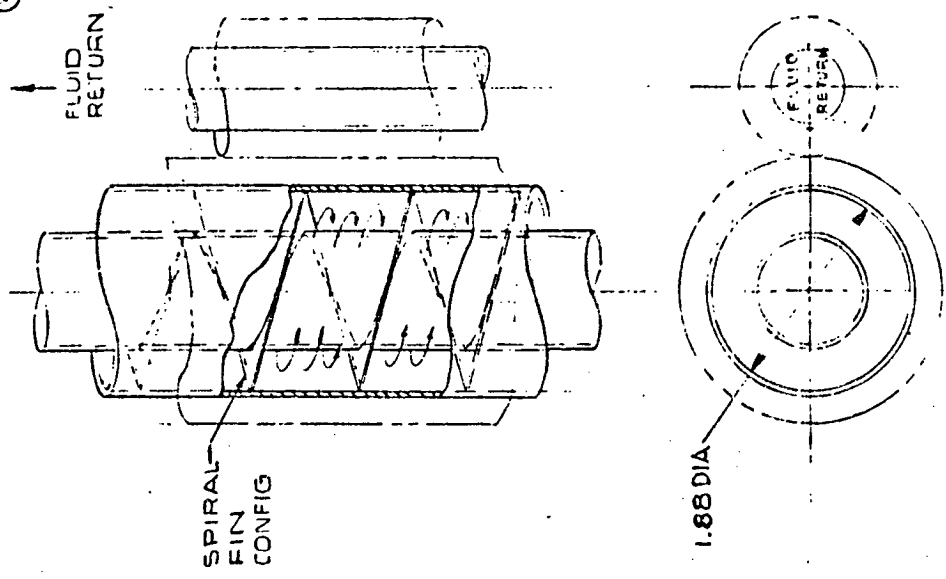
DETAIL A
 (SCALE 1/4)

FUEL CELLS HP/HX CIRCUIT

GRUMMAN
 3/24/71-JHF

SPL-110 (SH 20F3)

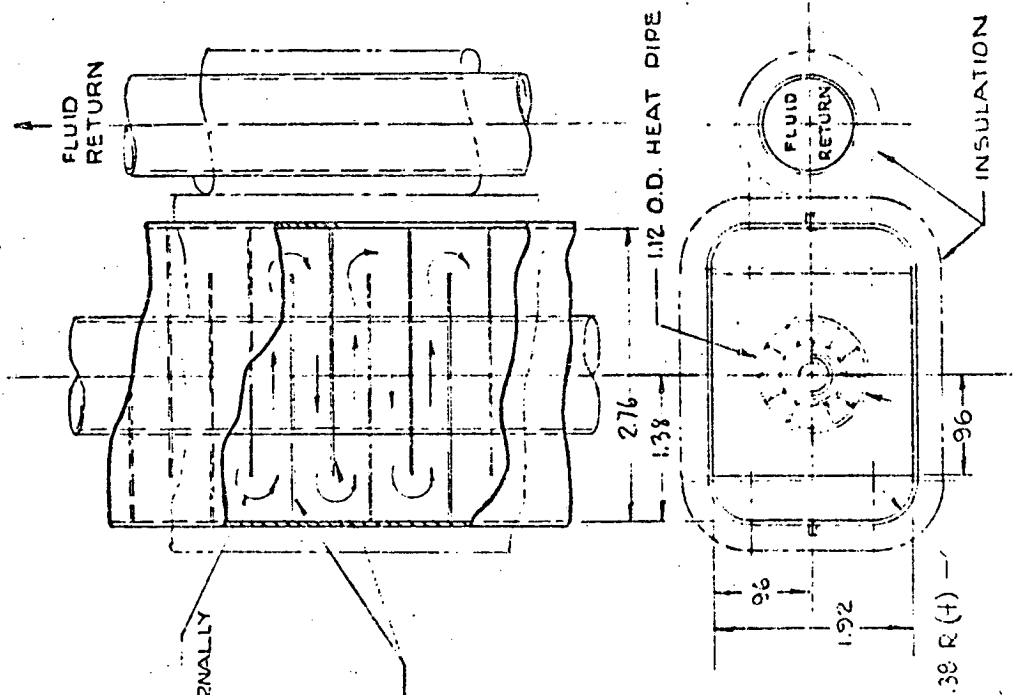
20



ALTERNATE CONFIGURATION

FUEL CELLS HP/HX CIRCUIT

SPL-110 (SH 30F 3)



SECTION B-B
(SCALE - FULL)

5-11 - AIR-BREATHING ENGINE COMPARTMENT - ON-ORBIT
MINIMUM TEMPERATURE CONTROL (SPL-109)

It is desirable to use as many currently existing engine components as possible in order to minimize cost. In order to do this, an acceptable non-operating environment must be provided during the applicable mission phases. Most engine components are designed to meet present MIL specs: when non-operational, they must be able to survive and hence must be kept within the -65°F to $+160^{\circ}\text{F}$ temperature range.

To estimate the required survival heat input, the heat loss by conduction through the wall of the compartment at an assumed temperature of -20°F to the skin at -180°F was calculated and found to be 300 watts per engine compartment.

Heat from the fuel cells and cabin area electronics is normally brought back to the radiator panels. Some of the heat that would have gone to the rear-most radiator panel could be used to heat radiator panels in the air-breathing engine compartment.

Drawing SPL-109 shows a typical configuration for heat pipe radiators in this compartment. Three panels are shown for each compartment: one on the inboard wall, one above the engines (in the stowed position) and one below. The positions of these radiators relative to those of the stowed engines are shown in section B-B, sheet 1 of drawing SPL-109.

Header D brings heat from the vertical header leading to the rear-most radiator panel, to header A, which supplies the engine compartment panels. It should be a diode pipe, to prevent heat from leaking back into the environmental control system during re-entry. If a $3/5$ inch I.D. Freon-21 heat pipe were used, it would have a capacity of more than a kilowatt, while its load would be only 300 watts, thus providing a suitable safety margin.

Header A takes the heat from header D and distributes it to panels 1 and 2. For parts of its length, it must carry 300 watts. A 1 inch I.D. Freon-21 heat pipe can be used; its capacity would be 543 watts. Header C

takes some of the heat load from A and carries it to header B, which feeds panel 3. Based on panel area, the load in both C and B would be 86 watts. A one-half inch I.D. Freon-21 heat pipe can be used with a capacity of 250 watts for B and 427 watts for C.

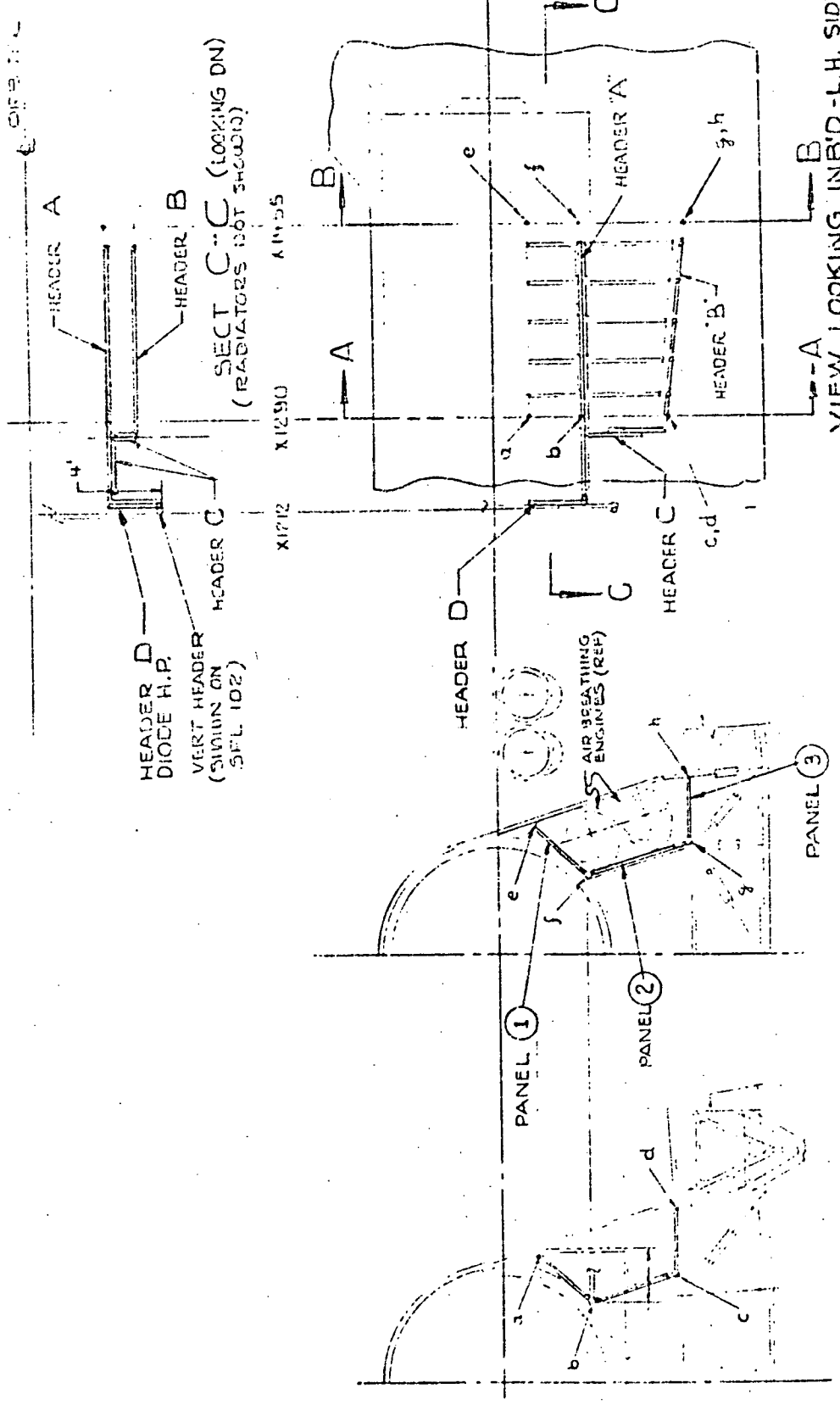
All feeder pipes could be made as $\frac{1}{2}$ " I.D. Freon-21 heat pipes. Those on panels 1 and 3 would have capacities of 460 watts but need only be capable of carrying 17.2 watts. Those on panel 2 have capacities of 400 or 425 watts, depending on condenser length; their maximum required capacity is 25.8 watts. These pipes are oversized and some additional effort can be extended to provide smaller lighter designs.

Heat Pipe Name	Dimensions	Fluid	Wick	Number Required and Placement	Computed Max Capacity (Watts)	Required Capacity (Watts)	ΔT At* Required Capacity
Header A	1" ID, 48" Evaporator, 60" Transport, 156" Condenser	Freon-21	G/C Spiral Artery/Tunnel	2 - 1 to each Compartment	543	300	3.0°
Header B	$\frac{1}{2}$ " ID, 48" Evaporator, 18" Transport, 156" Condenser	Freon-21	G/C Spiral Artery/Tunnel	2 - 1 to each Compartment	251	86	1.5°
Header C	$\frac{1}{2}$ " ID, 48" Evaporator, 12" Transport, 48" Condenser	Freon-21	G/C Spiral Artery/Tunnel	2 - 1 to each Compartment	427	86	1.9°
Header D	$\frac{3}{4}$ " ID, 48" Evaporator, 0" Transport, 48" Condenser	Freon-21	G/C Spiral Artery/Tunnel	2 - 1 to each Compartment	1100	300	3.6°
Feeder, Panels 1,3	$\frac{1}{2}$ " ID, 30" Evaporator, 54" Condenser	Freon-21	G/C Spiral Artery/Tunnel	20 - 5 on each panel 1, 5 on each panel 3	460	17.2	.2°
Feeder, Panel 2	$\frac{1}{2}$ " ID, 30" Evaporator, 72" Condenser	Freon-21	G/C Spiral Artery/Tunnel	5 - 3 on each panel 2	400	25.8	.3°
Feeder, Panel 2	$\frac{1}{2}$ " ID, 30" Evaporator, 63" Condenser	Freon-21	G/C Spiral Artery/Tunnel	4 - 2 on each panel 2	425	25.8	.3°

*Recent data indicates an improvement (at least) of a factor of two in these figures, i.e., one-half of the temperature difference.

TABLE 5-11 - HEAT PIPE ROASTER - AIR BREATHING ENGINE COMPARTMENT

(16)



SPL-109 (SH 1CF?)
 HEAT PIPE RADIATORS
 AIR BREATHING ENGINE
 (SCALE 1/75)

FIG. 5-11

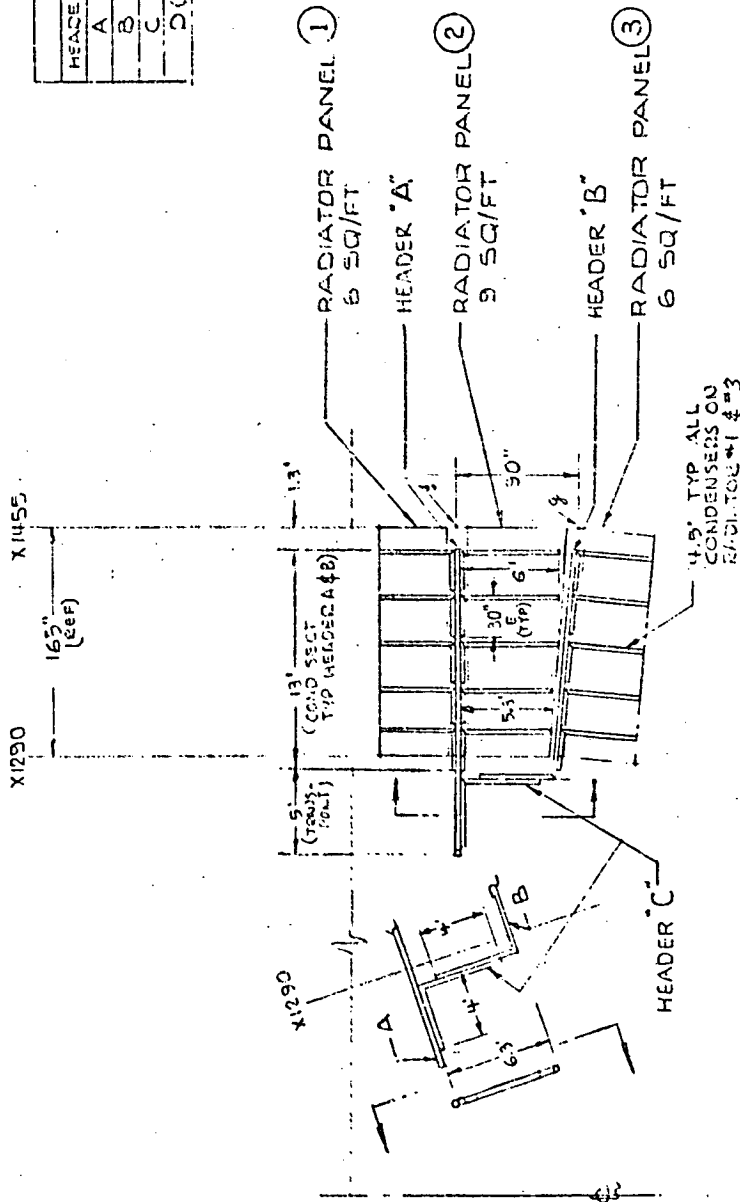


SECT B-B

SECT A A

TABLE I

HEADERS DIMENSIONS IN FT			
HEADER	EVAPORATOR	TRANSIT	CONDENSER
A	4.0	5	13.0
B	4.0	1.5	13.0
C	4.0	1	4.0
D (total)	4.0		4.0



VIEW SHOWING UNFOLDED
RADIATOR PANELS WITHIN
ENGINE COMPARTMENT

GRUMMAN

9/22/41-JHF

AIR BREATHING ENGINE COMP
SPL-109 (REV 2009)

SECTION 6

STUDY PLANS

Early in the program study plans were formulated which outlined procedures for the hardware development and test verification of promising heat pipe applications in three related areas:

- Space Station
- Space Shuttle
- Common Space Shuttle/Station applications and space radiators

The purpose of the study plans are to provide a timely and comprehensive assessment of the resources which would be required to effectively demonstrate the most practicable heat pipe systems.

The following information was provided for each heat pipe application cited in the study plans in order to present a clear picture of what is to be done, why, and how to do it.

- A description of each task proposed and the technical approach to accomplish it
- A scheduling chart with appropriate manhour estimates
- The type and scope of analyses needed for hardware design and performance verification
- An estimate of the design work required to support manufacturing
- The steps needed to manufacture the hardware
- A manufacturing schedule
- A detailed plan for verification testing of the HP hardware including test setup, instrumentation, data acquisition, procedures, etc.
- A test schedule
- A list of the facilities needed to analyze, manufacture and test the hardware

Study Plan

Space Station Heat Pipe Hardware Fabrication and Testing

The primary inputs to this plan were the results of the Space Station thermal control concept study which Grumman recently completed (Reference 4). It provided a wealth of well-understood, specific heat pipe applications information which was available for immediate evaluation.

The following heat pipe systems for the Space Station were recommended for hardware development.

1. A multiple heat pipe circuit to demonstrate operation of a system for transporting thermal loads from a number of sources to a single sink.
2. A high-capacity heat pipe header to verify the operation of a "random" multiple input pipe capable of transporting large loads over long lengths.
3. An air-to-heat pipe heat exchanger module capable of functioning in a life support system as a self-controlled atmospheric temperature regulator.

Study Plan - Space Shuttle Heat Pipe Hardware Fabrication and Testing

The following heat pipe systems for the Shuttle were recommended for hardware development.

1. An avionics heat pipe circuit for transporting thermal loads from a number of equipment sources to either a pumped loop or heat pipe sink.
2. A heat pipe/phase change material modular heat sink to demonstrate the operational feasibility of a self-sufficient modular thermal control system which could be used to regulate the temperature of remotely located power dissipating equipment.

3. A heat pipe augmented cold rail capable of cooling high thermal load density electronics packages which cannot be accommodated by conventional cold rail systems.
4. A heat pipe radiator system for compartment temperature control.
5. An air-to-heat pipe-to-fluid heat exchanger to control the thermal environment of air cooled electronics.

Study Plan - Common Space Station/Shuttle Applications and Radiator Heat Pipe
Hardware Fabrication and Testing

This study plan concerned only heat pipe radiator applications which were found common to both the Space Station and Shuttle, since all of the major Space Station applications previously cited were also common to the Shuttle and need not be repeated.

The study plan concentrated on the development of the components needed to construct a heat pipe radiator system for waste heat rejection. These components, which were detailed as separate tasks, consisted of the following items.

1. A flight weight variable conductance heat pipe header (VCHP) to serve as the distribution and control device for a heat pipe radiator panel.
2. A heat pipe/fluid header to enable a pumped loop heat transfer system to reject waste heat via a heat pipe radiator.
3. A combined VCHP and fluid header which would provide the required interface between a pumped loop and a heat pipe radiator as well as provide automatic load control and distribution.
4. Simple heat pipes which can serve as load distribution elements (feeder heat pipes) on a radiator panel.

5. A heat pipe radiator panel to demonstrate the feasibility of rejecting heat to space using heat pipes as the primary control and distributed elements.

SECTION 7

GENERAL DESIGN CONCEPT

A significant reduction in the number of different heat pipes required for the Shuttle 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 are proposed: the self-contained modular design and the modular sub-assembly design.


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 performance in excess of a single unit's capability is needed. These modules are placed in parallel for larger capacities and in series where long lengths are required.

In the modular sub-assembly design, heat pipes are configured by combining standard lengths of major subassemblies. The basic subassemblies are the pipe tube, wick, reservoir, low-k section, mitered joint and flexible joint. These pieces are joined together to create the desired heat pipe configuration.

For either concept to be used in any Shuttle location without restriction, it is also necessary that they function satisfactorily with at least two working fluids. This allows the same heat pipe hardware to be used both inside the pressure shell, where low toxicity is important, and outside, where thermal transport properties might be paramount. Thus, a standardized wick compatible with either modular scheme and operating with either of two working fluids is central to a truly general design concept.

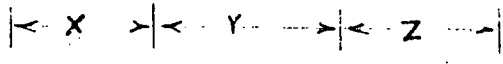
Table 7-1 categorizes the heat pipe requirements for all 11 of the candidate Shuttle HP applications of section 5 of this report. They are classified, in order, according to general shape, pipe I.D., evaporator length, condenser length and transport length. The number of pipes and applicatic reference are also indicated. By reviewing this listing any

TABLE 7-1
HEAT PIPE ROSTER (EY SHAPE)

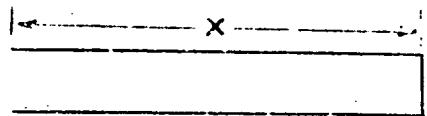
Heat Pipes Shaped Like: 

Drawing Number	Heat Pipe Name	X	Y	Pipe ID	Fluid	# Required
SPL-101	Radiator Feeder	5.75"	77"	0.5"	NH ₃ or F-21	212
SPL-103	Wheel Well Header	18"	60"	0.5"	NH ₃ or F-21	4
SPL-103	Wheel Well Feeder	9.8"	24"	0.5"	NH ₃ or F-21	24
SPL-105*	Finned Heat Pipe	3.2"	20"	0.5"	H ₂ O	4/unit
SPL-109	Header A	4'	18'	1"	F-21	2
SPL-109	Header B	4.5'	13'	0.5"	F-21	2
SPL-109	Header C	4'	5'	0.5"	F-21	2
SPL-109	Header D	4'	4'	0.75"	F-21	2
SPL-109	Panel 1/3 Feeder	30"	54"	0.5"	F-21	20
SPL-109	Short Panel 2 Feeder	30"	62"	0.5"	F-21	6
SPL-109	Long Panel 2 Feeder	30"	72"	0.5"	F-21	4

*Modified - large number of fins added to evaporator.

Heat Pipes Shaped Like: 

Drawing Number	Heat Pipe Name	X	Y	Z	Pipe ID	Fluid	# Required
SPL-102	Basic Member(hot)	48"	62.5"	48"	1.0"	NH ₃	6
SPL-102	Basic Member(cold)	48"	96"	48"	1.0"	NH ₃	12
SPL-102	Terminal Member(cold)	48"	51.6"	48"	1.0"	NH ₃	2

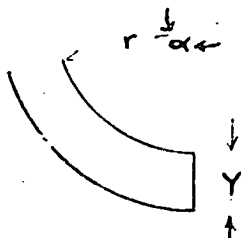
Heat Pipes Shaped Like: 

Drawing Number	Heat Pipe Name	X	Z	Pipe ID	Fluid	# Required
SPL-102	Crossover Unit (hot)	48"	48"	1.0"	NH ₃	2
SPL-102	Crossover Unit (cold)	48"	24"	1.0"	NH ₃	2
SPL-113	Radiator Header	6.4"	2"	1.0"	NH ₃	10



TABLE 7-1 (Continued)

Heat Pipes Shaped Like:



<u>Drawing Number</u>	<u>Heat Pipe Name</u>	<u>α</u>	<u>r</u>	<u>Y</u>	<u>Pipe ID</u>	<u>Fluid</u>	<u># Required</u>
SPL-113	Radiator Header	79°	96"	2"	1.0"	NH ₃	10

Note: Radiator headers of this type and those mentioned immediately above are alternate systems; only one would be used.

heat pipe designs common to one or more applications are plainly apparent.

After careful examination, it was determined that self-contained modular heat pipes would not satisfy enough applications to warrant further serious consideration as a design approach. There are four basic configurations, each with significantly different requirements for its evaporator, transport and condenser sections - a total of 17 different sizes. Satisfying the requirements of these varied applications with self-contained modules would mean many series/parallel circuits with many pipe to pipe conductive attachments. This would result in heat pipe installations which are inherently heavier and less efficient (large temperature drops) than customized designs. Appendix B-1 contains several examples of the heat pipe joining techniques which were investigated.

While the self-contained modular approach is not feasible, 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. Table 7-2 describes the basic modular subassemblies that can be used to satisfy the assorted requirements of the Shuttle HP applications. They can be machined and fabricated in large lots beforehand, drawn from stock, cut to length, and assembled when required. The modular subassembly concepts offers manufacturing, assembly, scheduling, maintainability and reliability 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. The first step in the process of choosing such wicks must be to determine which fluids are to be used.

Because it is non-toxic, 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, each fluid is incompatible with the materials

TABLE 7-2

Heat Pipes of Table 1 could be made of the following subassemblies:

	<u>Threaded Lengths</u>	<u>Unthreaded (transport)</u>	<u>rt Δ bends</u>
<u>One-inch ID:</u>	50 - 48"	10 - 2"	30
	20 - 77"	2 - 24"	
	2 - 216"	2 - 48"	
		2 - 51.6"	
		6 - 62.5"	
		12 - 96"	
<u>Three-Quarter Inch ID:</u>	4 - 48"		2
<u>One-Half Inch ID:</u>	212 - 5.75"		274
	24 - 9.8"		
	4 - 18"		
	24 - 24"		
	30 - 30"		
	2 - 48"		
	22 - 54"		
	6 - 60"		
	6 - 62"		
	4 - 72"		
	212 - 77"		
2 - 156"			

*Neglecting the finned heat pipe of SPL-105 and the curved heat pipe header of SPL-113.

usually used to construct heat pipes for the other: water reacts unfavorably with aluminum and stainless steel, and ammonia with monel and copper.

Freon-21 can be used as a back-up fluid for both water and ammonia because of its low freezing point and because it is relatively non-toxic. It presents no materials compatibility problems with monel, aluminum, copper or stainless steel. Table 7-3 summarizes the compatibility problems resulting from using one working fluid in a wick which has been designed for another.

Due to the compatibility problem, there can be no one multi-fluid wick used on the shuttle with these fluids, and two separate two-fluid wicks must be manufactured. Since it is necessary to make two separate wicks, it is more efficient to maximize performance of the wick for the two fluids involved than to try to produce one wick design which could be produced of two different materials and used by all three fluids.

Outside the pressure shell, the wicks developed for Freon-21 can be used for both ammonia and Freon. Analytical studies show that ammonia, when used in a heat pipe with these wicks, shows little loss of transport capacity compared to its performance in its own wicks (see Figure 7-1).

Inside the pressure shell, wicks similar to those developed for water, but with a small enough gap spacing to allow a Freon-21 - filled heat pipe to self-prime under a one-g gravity field may be used without severe performance penalty (see Figure 7-2).

CONCLUSIONS

If it is desired to use a design employing two-fluid heat pipes, the use of two different wicks is recommended: one which permits the use of water or Freon-21 inside the pressure shell, and one which allows the use of ammonia or Freon-21 outside.

Of the two general design concepts which were evaluated, the modular subassembly approach is recommended over the self-contained modular design. The latter is heavier and less thermally efficient than customized designs, while the modular subassembly concept gives heat pipe systems nearly as efficient as the customized systems and with improved fabricability.

TABLE 7-3
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

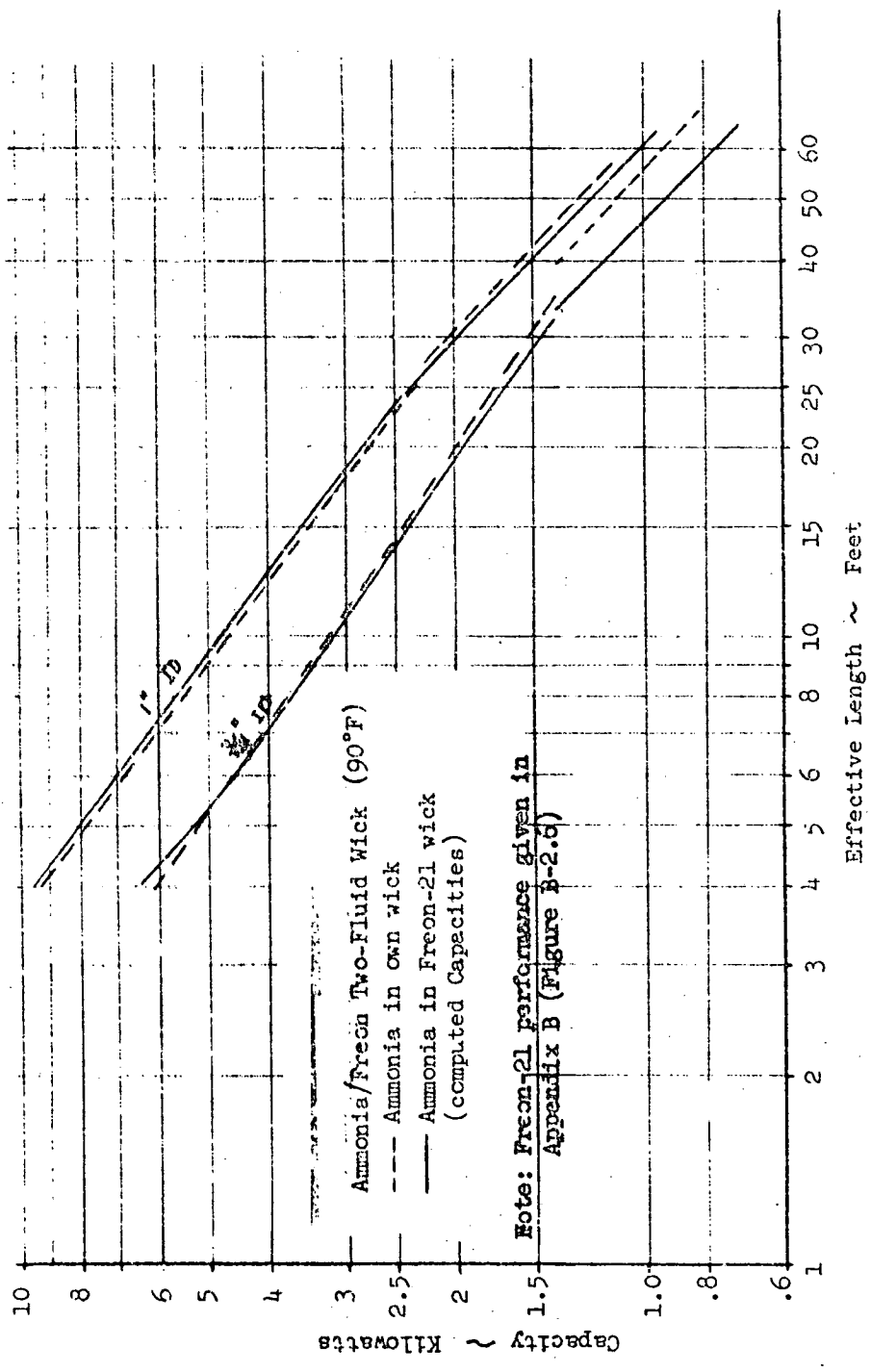


FIGURE 7-1

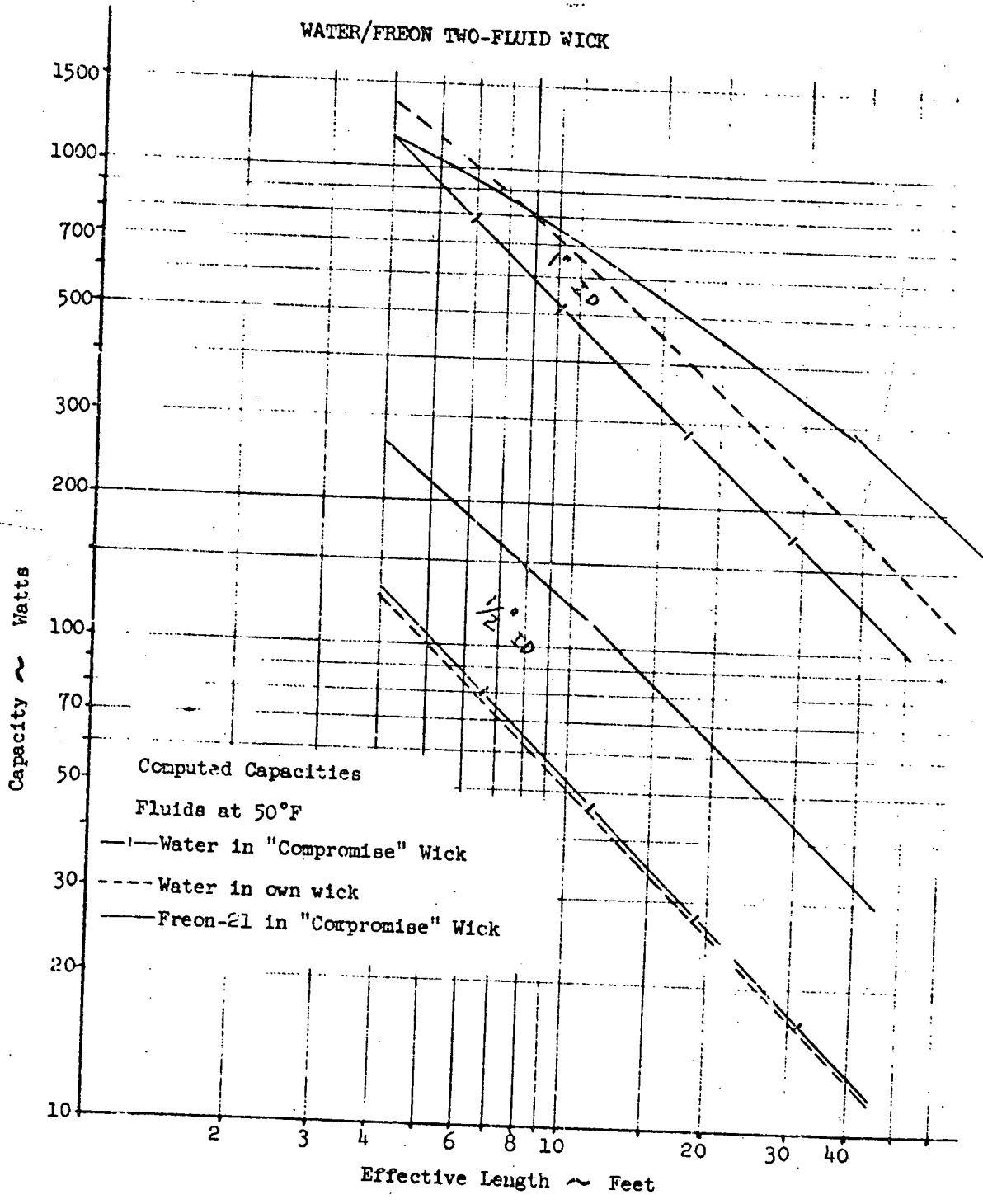


FIGURE 7-2



Before actually using a general 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 sub-assemblies 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.

If there are enough heat pipe applications to make the use of modular subassemblies practical, the following subassemblies would be manufactured using the appropriate materials:

$\frac{1}{2}$ " , $\frac{3}{4}$ " , and 1" ID threaded sections in 12' lengths

1" unthreaded sections in 12' lengths

$\frac{1}{2}$ " , $\frac{3}{4}$ " , and 1" ID mitered right angle joints

$\frac{1}{2}$ " , $\frac{3}{4}$ " , and 1" Freon wicks (for external Freon and ammonia pipes)

$\frac{1}{2}$ " and 1" compromise wicks (for internal water and Freon pipes)

SECTION 8
FINAL EVALUATION

The feasible Shuttle heat pipe applications have been identified and categorized in section 4 of this report as prime contenders (11), possible contenders (5) and rejects (11). This section establishes a logical evaluation procedure to choose the most promising applications (at least six) for detailed design and analysis. The selection is based on how well they compare with their counterpart baseline thermal control systems. In other words, the highest ranking applications are those most likely to be implemented on the Shuttle - and these are the ones that present the most improvement over corresponding baseline systems.

The comparisons are made on the basis of performance, weight and safety as gauged by six criteria: temperature gradient, capacity margin, power requirement, control requirement, weight and safety. These parameters were chosen because they are meaningful measures which can be readily defined and understood on an objective basis. Some obvious parameters such as cost, maintainability, reliability, development risk, and durability are only evaluated on a secondary basis since there is not enough factual data to make an unbiased comparison between heat pipe and baseline systems. This keeps subjectivity in the initial definition stages of the evaluation to a minimum.

RATING COMPARED TO BASELINE

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

Figure 8-1 - Evaluation Matrix

The basic evaluation procedure is a better than/worse than comparison with the baseline system for each of the six criteria. Figure 8-1 shows the evaluation matrix used. Definitions of the parameters are listed below:

1. 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.
2. 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.
3. Power Requirement - The amount of electrical power required to operate the heat transport device; it includes power for control systems.
4. 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.
5. Weight - The total weight of the heat transport device including its basic components and any special attachments, fittings and control elements.
6. 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 5. As shown in Table 8-1 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	High-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

TABLE 8-1 SCORING OF SAFETY FACTORS

For purposes of this safety evaluation, high pressure fluids are defined as fluids operating at pressures greater than 100 psia. Potential hazards are those situations or hardware to which exposure for periods greater than one hour could cause death or injury. This category also includes substances which are themselves not dangerous but which could react with other substances and either be hazardous or cause further reactions such as corrosion.

This type of evaluation assumes that all criteria are equally important, i.e., they have equal weights. The procedure can be made more discriminating by assigning relative weights to the evaluating parameters as in Reference 5. However, this would be of limited value considering the level of detail in this preliminary evaluation. This effort is regarded as a filtering process to reduce the number of application for detailed design to a manageable level.

The individual and collective engineering judgment of the Grumman evaluating team was used to evaluate each of the prime contenders according to the criteria and rating sheets. The completed evaluation sheets are contained in Appendix C-1. The best possible score for a given application is six "better than" ratings - one for each of the evaluation criterion. The worst possible score is six "worse than" ratings when compared to the baseline system.

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 8-2 contains a summary of the evaluation sheets for the contending applications along with their net numerical ratings. The highest rated application scored +4 and there were even 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 Section (5-2) and air breathing engine compartment Section (5-11) were considered part of the heat pipe radiator, compartment temperature control category. In the same fashion applications 5-7, 5-8 and 5-10 were considered heat pipe radiators for waste heat rejection. Table 8-3 summarizes the six generic categories which resulted. The advantages and disadvantages of both the accepted and rejected applications are discussed below:

ACCEPTED APPLICATIONS:

5-2/5-11 - Heat Pipe Radiator, Compartment Temperature Control

Both of these applications involve on-orbit temperature control of structural compartments. The baseline system uses Freon-21 (F-21) radiator panels and the HP system uses F-21 heat pipes. Thermal gradients in both systems are about the same since they both employ brazed tube-to-sheet joints. The heat pipe panels are capable of rejecting more heat to the compartment, if required, since the feeder pipes have excess capacity. The heat pipe system is self-controlling due to the built-in diode which prevents heat transfer to the ECS loop when the temperatures of the compartment are higher than the source temperature e.g., during entry.

Also, the heat pipe radiator panel can continue to function even if several of its panel heat pipes have been accidentally punctured. This is not the case with the conventional fluid radiator where at worst, the entire ECS loop can be drained of fluid, and at best, it requires involved leak detection systems with shut-off controls.

7-7/5-8/5-10 - Heat Pipe Radiator, Waste Heat Rejection

Several variations to the basic heat pipe radiator panel will be investigated to determine the most viable HP radiator system. They will involve three different supply header configurations:

1. A VCHP header coupled to an all heat pipe transport loop.
2. A hybrid VCHP/Fluid header serving as a heat exchanger interface between the heat pipe radiator panel and a fluid heat transport loop.
3. A fluid supply header coupled directly to the individual feeder heat pipes on the panel.

These systems will show various degrees of self-regulation, depending on the utilization of the VCHP. All of the HP radiator systems will eliminate flow splits and flow balancing problems and provide a high degree of reliability by minimizing the effect of single point failures.

5-3a - Heat Pipe Circuit

The heat pipe circuit consists of avionics equipment racks utilizing all heat pipe cold rails which are then manifolded to a heat pipe header. The heat pipe header is coupled to the main ECS fluid header of the heat transport system. The heat pipe provides greater cooling capacity than the cold rails both on a power density and total load per rail basis. Since the pipes are isothermal they provide flexibility for box relocation 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.

5-4 - Heat Pipe Augmented Cold Rail

The heat pipe augmented cold rail enables present fluid cold rail configurations to be used with the current generation of high power density avionics. Standard fluid cold rails, which are adequate for other types of Shuttle equipment, cannot handle the higher power densities of the power conditioning and distribution electronics. Without heat pipe augmentation, the previous generation electronics would have to be substituted for the newer and more compact equipment. This would result in heavier avionics and fewer components mounted per rail. More cold rail would be required, resulting in increased weight and flow pressure losses.

5-5 - Flight Data and Voice Recorder - Modular Heat Sink

Temperature control of the remotely located (vertical fin) flight and voice recorders would be provided by a self contained system which couples these remote heat sources to a structural or phase change 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 type of a thermal control system is self-regulating, requires no pumping power and eliminates long runs of fluid coolant line which would be needed in the baseline system. The absence of fluid connections also lessens that chance of leakage and equipment contamination.

5-6 - Air-Cooled Equipment

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.

This would also eliminate the elaborate fluid loop fill and purge procedures that would have been necessary whenever these fluid couplings were disconnected.

Rejected Applications

5-1 - Leading Edge TPS

The application of heat pipes of the leading edge TPS would require extensive development effort in the area of heat pipe to carbon/carbon bonding. The alternate scheme, where the heat pipe envelop is made of carbon/carbon and formed as part of the leading edge structure, would also require development to solve heat pipe end capping problems.

This application is not state-of-the-art. While the necessary heat pipe technology exists, using it in an area which itself is still under development would be a very questionable commitment of Shuttle resources.

A successful carbon/carbon development program would eliminate the need for high temperature heat pipes in this area. If leading edges capable of withstanding a 3500°F radiator equilibrium temperature for 100 missions are producible, standing a 3500°F radiation equilibrium temperature for 100 missions are producible, it would obviate the need for reducing the leading edge temperature ($\sim 3100^\circ\text{F max}$) to lower levels more conducive to multi-mission use. Since the Shuttle TPS design must be failsafe in any case, using supplementary heat pipes represents an unnecessary weight penalty. This runs contrary to the minimum risk, low cost philosophy which is prevalent in current Shuttle concepts.

5-3 - Heat Transport System

The complete heat pipe heat transport system is an all heat pipe system which transfers heat from internal equipment sources to a heat pipe radiator sink. On the Shuttle, this means transport lengths on the order of 100 feet requiring many heat pipe couplings and splices. The excessive transport lengths and the great number of heat pipe interfaces presently result in large temperature drops which would demand increased radiator area (above the baseline requirements) to reject the same heat loads at the new lower temperature levels.

5-4 - Heat Pipe/Phase Change Remote Sink

The cyclic heat loads from a remotely located source are transferred, via heat pipes, to a phase change material. When the equipment is off, the liquified phase change material is allowed to solidify by rejecting its heat to local structure. Although this is a feasible system, no specific applications areas have been found on the Shuttle at this time. It is expected that applications will arise as more shuttle definition becomes available. A similar scheme for thermal control of the flight data/voice recorders is covered in application 5-5.

The accepted selections are considered reasonable from the standpoint of all engineering feasibility. They were approved by NASA, with the request that work on the waste heat rejection radiator be suspended, since such studies should be advanced under a separate effort. The analysis of the radiator done at this point is presented in Appendix C-2.

The remaining five applications 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. These three were chosen based on the tangible benefits they offer and the likelihood that they might be implemented on the shuttle after successful demonstration testing.

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

TABLE 8-2 - SHUTTLE HEAT PIPE APPLICATION - SUMMARY

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

TABLE 8-3 - SHUTTLE HEAT PIPE APPLICATIONS FOR DETAILED

DESIGN AND ANALYSIS

1. Heat Pipe Radiator, Compartment Temperature Control	5-2 - Landing gear wheel wells	+4
	5-11 - Air breathing engine compartment	+4
2. Heat Pipe Radiator, Waste Heat Rejection	5-7 - HP Radiator with HP header	+4
	5-8 - HP Radiator with HP/Fluid header	+4
	5-10 - HP Radiator for Fuel Cells	+4
3. Heat Pipe Circuit	5-3a - Avionics heat pipe circuit	+4
4. Heat Pipe Augmented Cold Rail	5-9 - Heat pipe augmented cold rail	+4
5. Modular Heat Sink	5-6 - Flight Data & Voice Recorders	+3
6. Air Cooled Equipment	5-5 - ATR Equipment	+2

*Maximum rating is +6

SECTION 9

DETAILED DESIGN AND ANALYSIS

The following sections contain 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 Radiating Panel for Compartment Temperature Control

Each section is structured as follows:

- Introduction
- Analysis-contains supporting analysis including system requirements, assumptions, thermal models, and performance predictions.
- Design-contains detailed description and design drawings including necessary specifications
- Baseline-includes comparison with baseline thermal control systems, if applicable
- Test Plans-contains test plans and procedures for development of prototype hardware (first three applications only)
- Conclusions

MISSING PAGE BLANK NOT FILMED
SECTION 9-1

HEAT PIPE AUGMENTED COLD RAIL

9-1.1 INTRODUCTION

The rack concept for ECS loop cooling of electronic equipment was employed in the Apollo program. Equipment to be cooled is packaged in module boxes with flanges on two opposing sides, and the flanges are bolted to the rack's structural rails which are internally cooled by pumping cold fluid through two passages in the "cold" rail. The present system has added a heat pipe to the center of each cold rail to isothermalize the rail and thereby make the system capable of accepting the high watt densities imposed by thin module boxes dissipating large amounts of heat.

The basic configuration of the rail, as shown in Drawing SPL-104, is suggested by reference 6. The rail contains two fluid passages, both of which (for high power density equipment mounting) are assumed to be active. Redundancy requirements would be met by installing duplicate equipment racks.

The rail itself would be made of 6101 aluminum alloy, chosen for its reasonable strength and high thermal conductivity. The heat pipe in the rail center has water as a working fluid (to satisfy safety requirements for location inside the pressure shell), and has an envelope of copper or monel. Although GAC has had good results in compatibility tests using these materials with water, the use of a non-toxic Freon* would be possible if problems were to develop.

9-1.2 ANALYSIS

At an early stage in the analysis, it became evident that a flange thickness of 0.125 inches would be inadequate to maintain the temperature drop between the box flange and the cold rail at an acceptable level for high power density equipment. (See Appendix D-1)

*Freon-114 and Freon-12, in Underwriters Laboratories Report MH-2375, were classed in toxicity group 6. Toxicity group 6 contains those "... gases or vapors which in concentrations up to at least about 20 per cent by volume for durations of exposure of the order of 2 hours do not appear to produce injury."

To eliminate this problem, a thicker cold rail flange (0.25 inches) is used at those sites where high power density equipment modules could be mounted.

As the system to be modelled involved fluid flow and heat pipe operation, steady-state temperatures were determined by allowing a transient model to reach equilibrium rather than by adapting a steady-state method to the system.

The Abbreviated General Thermal Analysis Program described in reference 7 was used. It is a typical lumped parameter approximation, finite difference iterative heat transfer program. Heat pipe operation and fluid flow are modelled by inserting a subroutine to be executed prior to each network iteration. The subroutine used is listed in Appendix D-2.

The first half of this subroutine simulates the operation of a heat pipe by assuming that the heat pipe vapor temperature is constant along the heat pipe, and that it can be considered a fixed temperature within each iteration. The 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.

The fluid modelling system is taken directly from reference 8. A diagram of a typical section of fluid is shown in Figure 9-1.1.

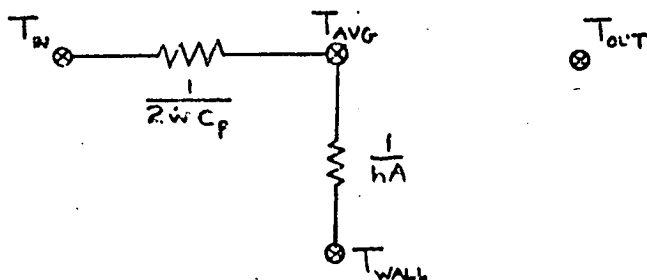


FIGURE 9-1.1 FLUID FLOW NETWORK



The outlet temperature, which is the inlet temperature for the next section, is defined by

$$T_{\text{out}} = 2 T_{\text{avg}} - T_{\text{in}}$$

A nodal analysis of the above network gives

$$2 \dot{w} c_p (T_{\text{in}} - T_{\text{avg}}) + hA (T_{\text{wall}} - T_{\text{avg}}) = 0$$

and, after manipulation,

$$\dot{w} c_p (T_{\text{out}} - T_{\text{in}}) = hA \left(T_{\text{wall}} - \frac{T_{\text{in}} + T_{\text{out}}}{2} \right)$$

a steady state representation of the fluid node. The second half of the subroutine defines and calculates the needed fluid temperatures.

For thermal modelling purposes, the rail, 21.6 inches long, was broken into 12 sections 1.8 inches in length. Locations of nodes and conductances in a typical cross-section are shown in Figure 9-1.2. Heat conduction along the rail was included by defining conductances between the four nodes shown within the cold rail and the corresponding nodes in adjacent sections.

In order to check the validity of the conductances assigned in Figure 9-1.2, two finer networks were constructed, one detailing the box to cold rail flange root conductance at one station (46 nodes), and the other conductances between rail flanges, fluid passages, and the heat pipe at one station of the rail (44 nodes). Both agreed with the results of the larger model which included all stations of the entire rail.

The analytical case studied includes a typical high power density module, i.e., the power switch module, 1.8 inches wide and generating 140 thermal watts (reference 6), mounted on the cold rail. Both sides of the rest of the rail were loaded to several watt-density levels.

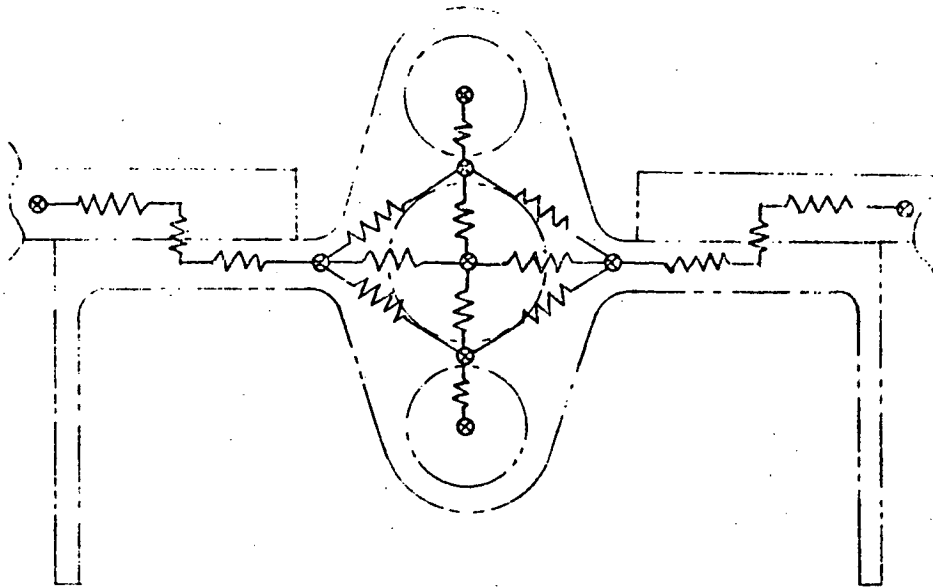


FIGURE 9-1.2 CROSS SECTION OF COLD RAIL THERMAL MODEL

Flow rates were established using the Apollo-type limitation of 4 watts per pound of coolant per hour. This flow was divided evenly between the two 3/8 inch diameter passages in the rail which include Lytron-type internal fins. Film coefficient data for such tubes containing 60/40 glycol/water mixtures was obtained, and corrected by means of the Sieder-Tate Relation (See Appendix D-3) to account for the use of distilled water, which is the shuttle baseline coolant.

Grumman has recorded film coefficients for water heat pipe evaporators in excess of $3,000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$, so comparatively conservative values of 2,000 and 2,500 were used for heat pipe evaporators and condensers, respectively.

Studies using the thermal model described on the previous pages produced the following conclusions:

- 1) The heat pipe operates by isothermalizing the cold rail, thus lowering rail temperatures at high watt density equipment mounting sites;

- 2) the power switch module of reference 6, dissipating 70 watts per side, may be mounted on the proposed heat pipe augmented cold rail. In addition, other equipment with an average power dissipation of 0.83 watts per inch may be mounted at all the other locations of the rail;
- 3) the power switch module cannot be mounted on a conventional cold rail since the latter cannot remove enough heat to prevent the module box temperature from rising above 140°F;
- 4) 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 9-1.3 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 the other 23 mounting sites. As employed here, the heat pipe lowers the peak flange temperature by 22°F, which 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 the 70 watt source at station 3. The box flange root temperature dropped only 3°F to 159°F, still unacceptably high. The power

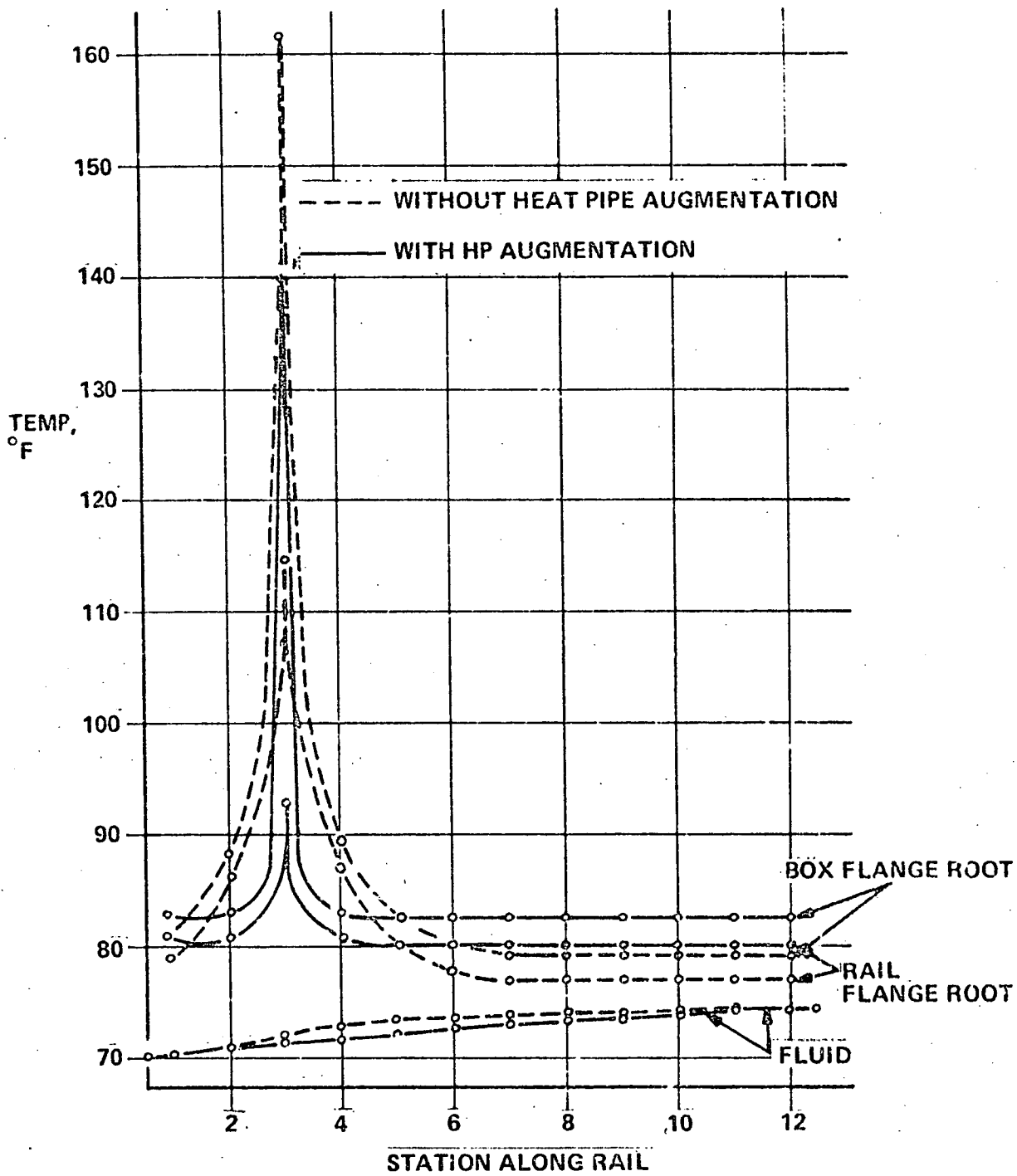


FIGURE 9-1.3 TEMPERATURE DISTRIBUTION ALONG COLD RAIL



switch module of reference 6 cannot be mounted on a standard cold rail.

Figure 9-1.4 shows the temperature distribution in the augmented rail with the same thermal input used in Figure 9-1.3, 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.

DESIGN

The detailed design, including design specifications, is shown in Drawing SPL-104. Detailed weights are given in Table 9-1.1.

The rail is made of an extrusion of 6101-T6 aluminum alloy (chosen for its high thermal conductivity), whose dimensions are defined in a detail of drawing SPL-104, and shall be manufactured in accordance with materials specification QQ-A-200/8. The costs of having such an extrusion made include a die making charge of \$1500, a set up charge of \$200, and an extrusion cost of \$2 per foot.

The tops of the rail flanges, as shown in detail A, will be finished to a 64 micro-inch RMS surface to enhance thermal contact with the equipment. At those sites where high power density modules will not be mounted, the flanges (as shown in the extrusion detail) will be machined down (from the bottom) to .125 inches, in order to minimize weight. Plate nut fittings are used at the mounting sites for box attachment.

As shown in section C-C, the fluid passages will be internally finned (to increase heat transfer into the fluid) using Lytron-type radial fins around an internal tube of .156 inch diameter. Gamah couplings will be welded or swaged to short tubes which will then be inserted and welded into the entrances and exits of the fluid passages in the cold rail to provide a maintainable loop coupling.

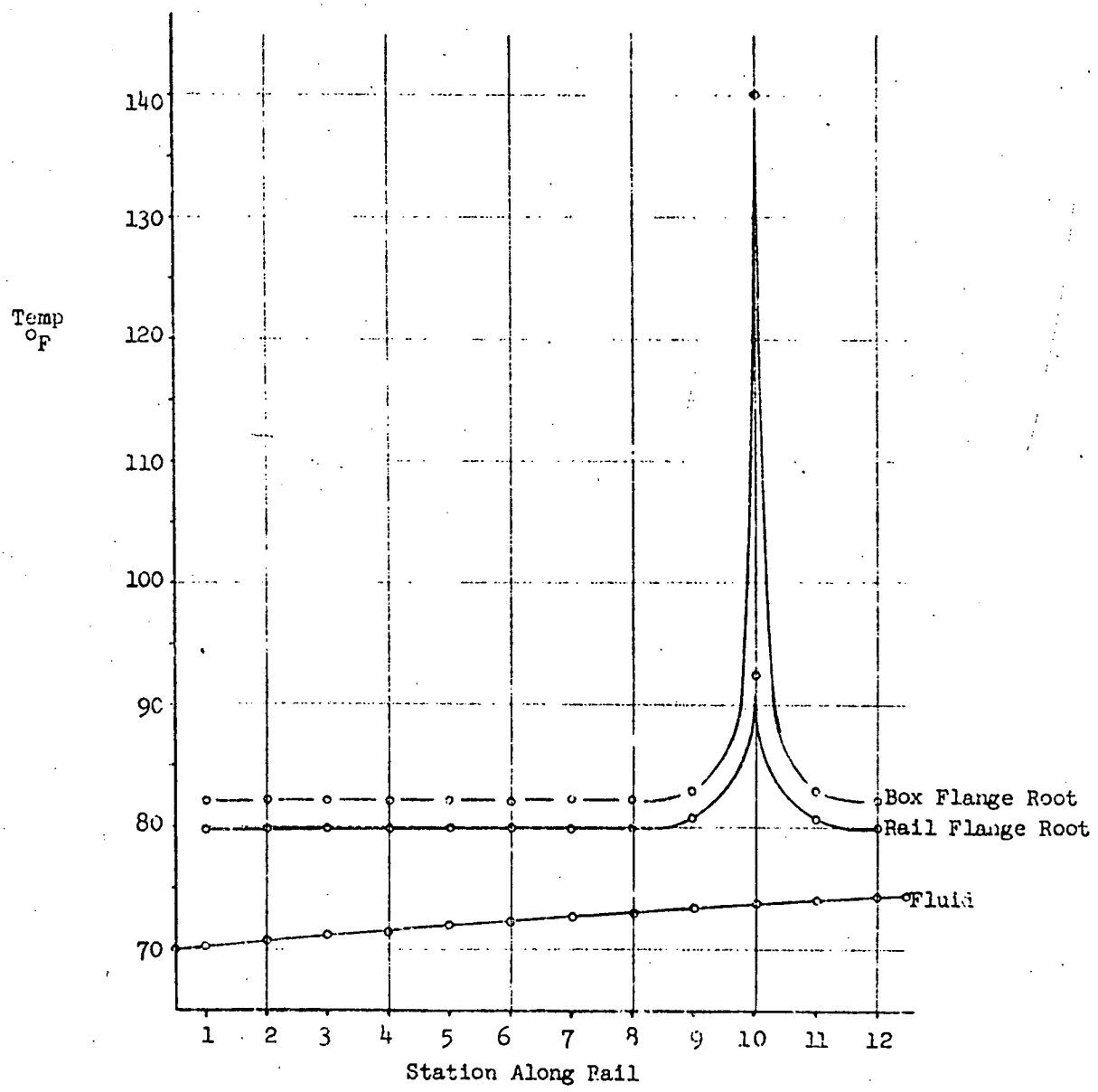


FIGURE 9-1.1 TEMPERATURE DISTRIBUTION ALONG HF AUGMENTED COLD RAIL



ITEM	QTY	UNIT WEIGHT LB.	TOTAL WEIGHT LB.	DATA/COMMENTS
Cold Rail Extrusions	8		(17.00)	
Low Q Load		1.00 lb/ft	7.50	Al
High Q Load		1.25 lb/ft	9.50	Al
Heat Pipe	8	(.230) lb/ft	3.60	
Fluid - H ₂ O		.017 lb/ft		Distilled Water
Spinal Artery		.030 lb/ft		Hollow core dia..06"
Retainer		.020 lb/ft		Socket dia. .20"
Tube		.163 lb/ft		.5" OD x 0.28" monel
Coolant Tubes	8	.035 lb/ft	1.08	
External Film		.023 lb/ft		
Inner Tube		.012 lb/ft		
Coolant Line		.005	.08	Al 0.28"
Cold Rail - Coupling	16	.032	.51	Al
Gaman Type Coupling	16			
		Total Weight	<u>22.27</u>	
		lb.		
				or 2.78 lb/rail

TABLE 9-1.1



The heat pipe shown has an envelope of 0.5 inch O.D. nominal 101(OFHC) copper tubing (ASTM Specification B68.75.280) or 0.5 inch O.D. Monel Alloy 400 tubing (MIL-T-1363), internally threaded to 100 threads/inch. These materials were chosen to eliminate potential gas generation problems with distilled water. This tubing will be installed within the cold rail by reaming the extruded hole to allow for a push fit for the full length (21.6 inches) plus a trim allowance. The rail/tube interface will be bonded with silver-filled Hysol epoxy K8-4232 (Grumman instructions for application of Hysol specified in SD 252-I-73 from OAO program) to increase heat transfer across the interface. This cylindrical interface will be sealed at both ends to prevent exposure to air by applying sealant GSS-4100 (MIL-S-7502).

The structure and orientation of the wick are shown in section C-C. It will be supported by four retainer legs whose orientation must be in the direction specified in this cross-section 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 capacity of such a heat pipe, with the hollow core unprimed and a 10-inch evaporator and 10-inch condenser section 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.

Grumman tests have shown that a heat pipe using distilled water as a working fluid can be made of copper screen and tubing and coated with Ebonol C to increase the wetting properties of water. This design alternative may prove superior to an all-monel heat pipe due to manufacturing considerations.

Typical module equipment racks based on reference 6 are shown on the drawing. Configuration A shows typical load control center components mounted on an avionics equipment rack employing heat pipe augmented cold rails, and configuration "B" shows typical power conditioning components mounted on a similar rack.

Module (and rails) are to be prepared for mounting by applying silicone grease (Dow Corning 340) to the flange surfaces to be mated, then wiping the grease from the surface. This procedure leaves a thin film of grease which increases the interface conductance. This grease has been approved for use on the Lunar Module (5 psia, 100% O₂), as referenced in Grumman Specification LSM-14-6006 dated 19 July 1966.

The only additional failure mode introduced by augmenting the cold rail with a heat pipe is a possible leak in the heat pipe, which contains only small amounts of distilled water (~ 20 grams), a non-toxic substance. Pressures inside the heat pipe are negligible: at 100°F, internal pressure is about one psia, indicating no danger of bursting.

TEST PLAN

This section describes the test program designed to provide a thermal performance map for the heat pipe augmented cold rail depicted in drawing SPL-104. It calls for a series of parametric runs to determine response to varying fluid inlet temperatures, average load density, and high load density with and without heat pipe augmentation.

TEST CONFIGURATION

A schematic of the test arrangement is shown in figure 9-1.5. Provision is made for controlling the inlet temperature and flow rate of the coolant and for shutting off the flow in one or both passages to simulate partial failures. The cold rail itself is completely instrumented with suitable provision for readout (Figure 9-1.6). Simulated electronics heat loads are provided by strip heaters attached to both mounting flanges and controlled with Variac type devices.

The facility requirements are summarized below:

Power:	Four variac channels 2 at 200 watts each 2 at 100 watts each
Instrumentation:	Readout and recording for 40 Cu-Cn thermocouples
Coolant:	distilled water at 0 to 120 lbs/hr inlet temperature control from 50° to 120°F

GRUMMAN

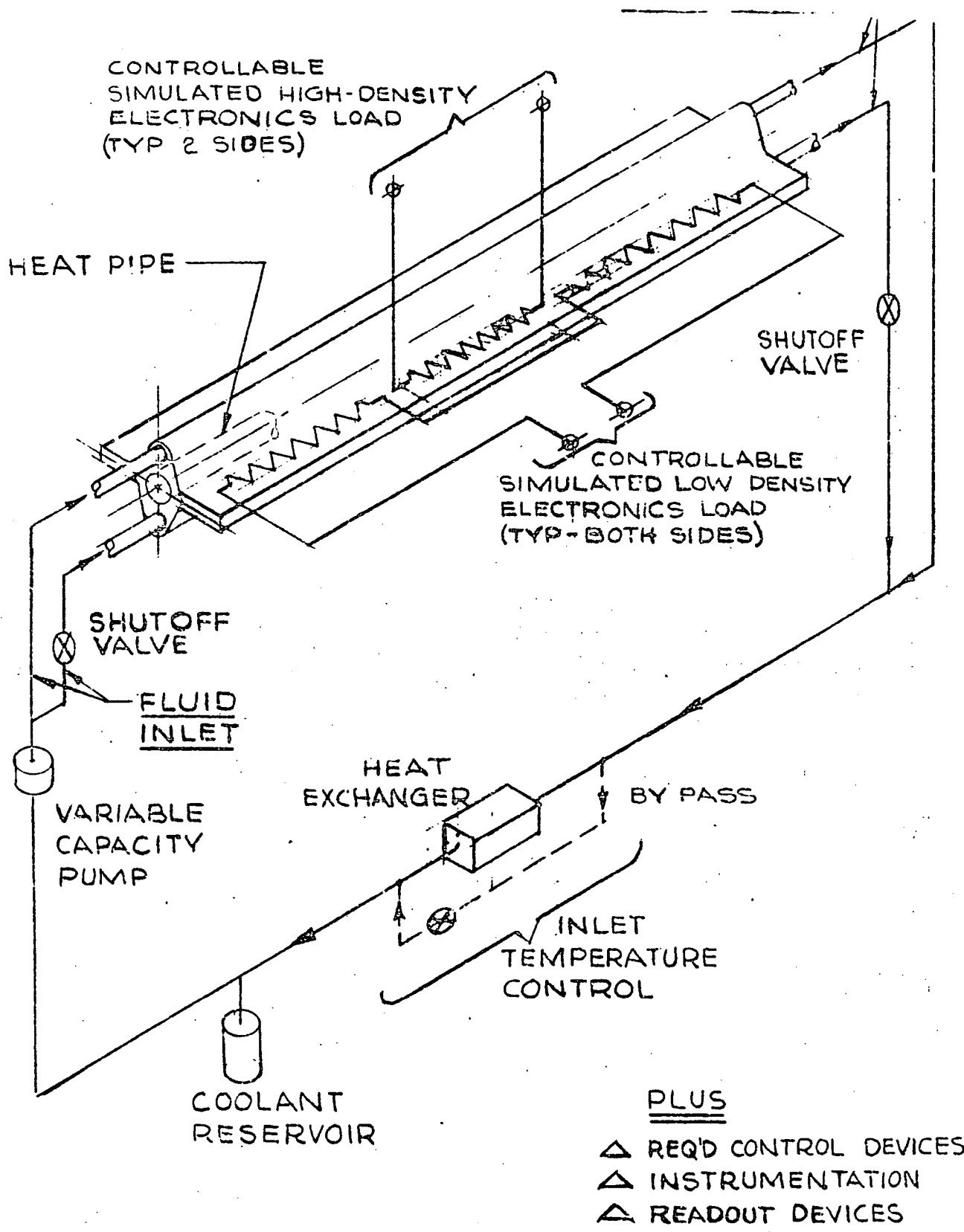


FIG 9-1.5 TEST SCHEMATIC



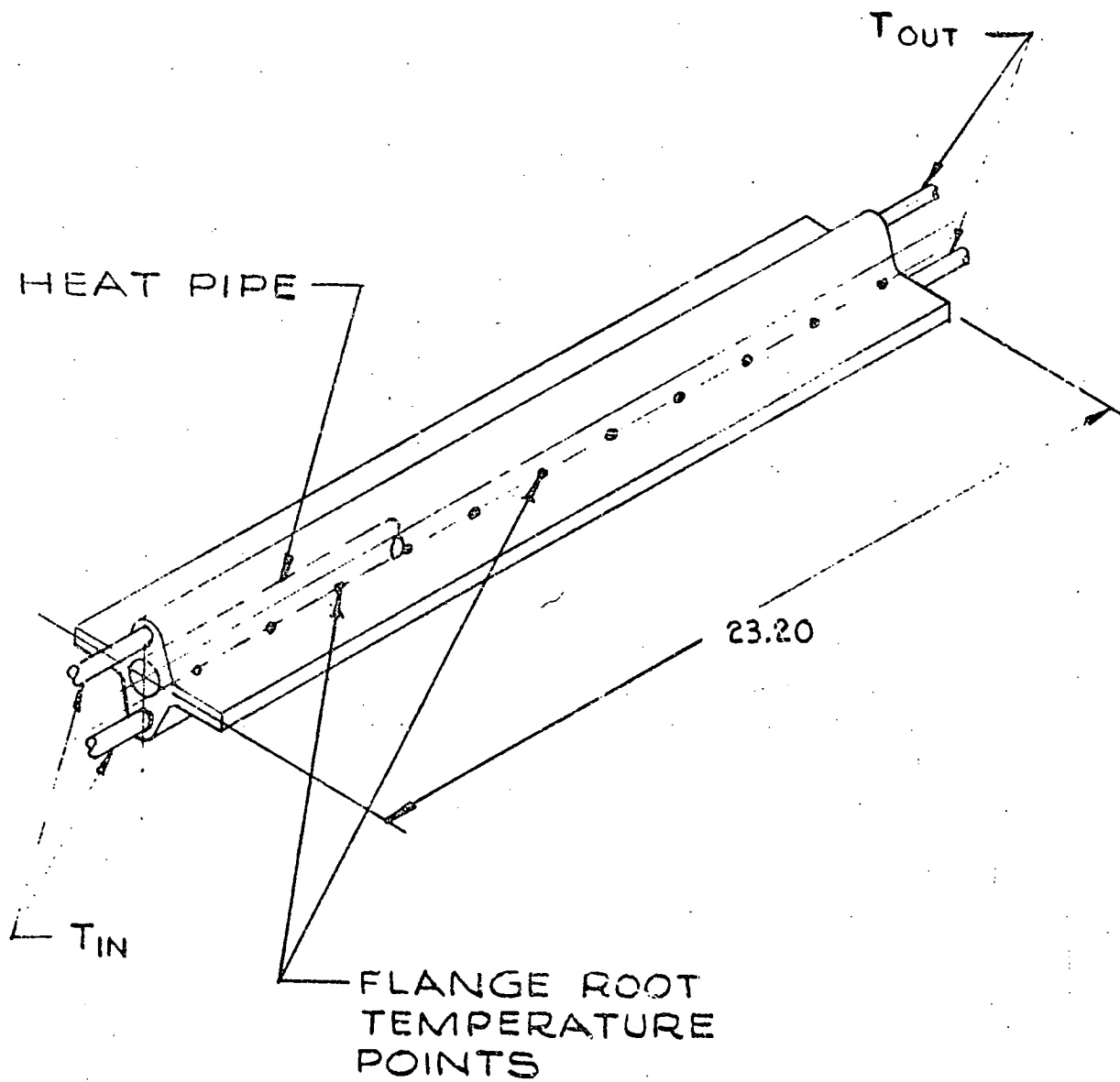


FIG. 9-1.6-INSTRUMENTATION

TEST PROCEDURE

The test procedures are designed to provide six specific types of information.

- (1) Bench or baseline HP data
- (2) Maximum load vs. flange root temperature
- (3) Average load vs. flange root temperature
- (4) Maximum load vs. average load
- (5) Load vs. fluid inlet temperatures
- (6) Load capacity without heat pipe

The test conditions are summarized in Table 9-1.2. The bench test (1 and 2) data will determine the HP capacity and temperature response. It will be conducted on a bare HP without the cold rail and measure both longitudinal and lateral conduction. This baseline data will be obtained from the test setup shown in Figure 9-1.7. Electrical heaters and spray bath cooling will be employed.

Tests 3 through 14 follow the same basic pattern in which a constant input (average Q/L) is provided to most of the cold rail flange. In addition a two inch length towards the inlet end of the rail is loaded independently to simulate a high density electronics package. The load on each section is independently controlled to meet test requirements. The test program provides a complete performance map of the HP augmented cold rail. Tests 15, 16 and 17 will evaluate response under partial failure conditions, i.e., only on coolant passage operating. The remaining two tests (18 and 19) will provide baseline data on the cold rail performance without an operative heat pipe.

CONCLUSIONS:

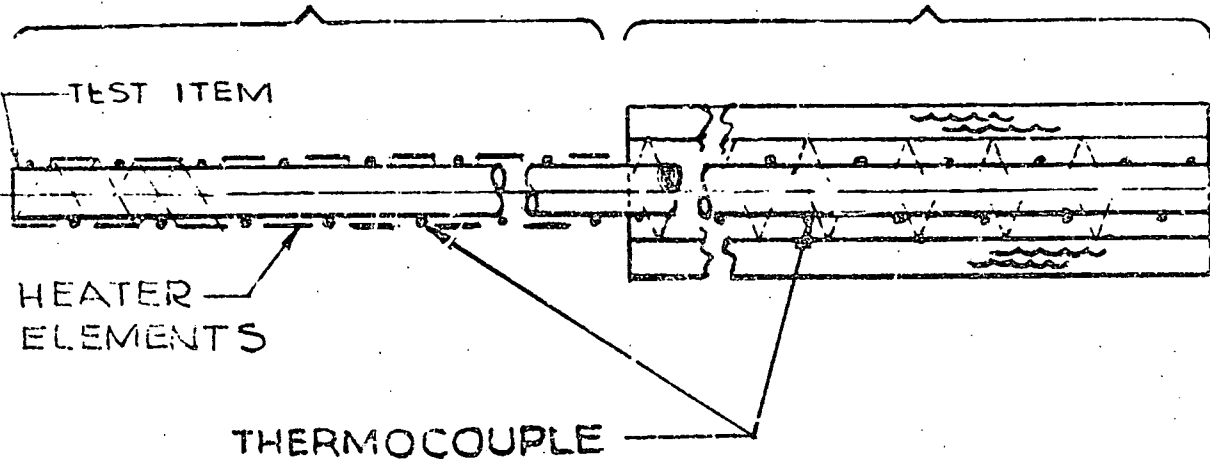
By placing a heat pipe in a cold rail, the effective conductances between the rail flange roots and the fluid passages are increase at mounting sites of high power density equipment. The heat pipe has no effect on the total heat load the rail is able to dissipate, but makes very high watt densities permissible if the load on the rest of the rail is low enough. By acting as an isothermalizer in the center of the rail, the heat pipe

TABLE 9-1.2

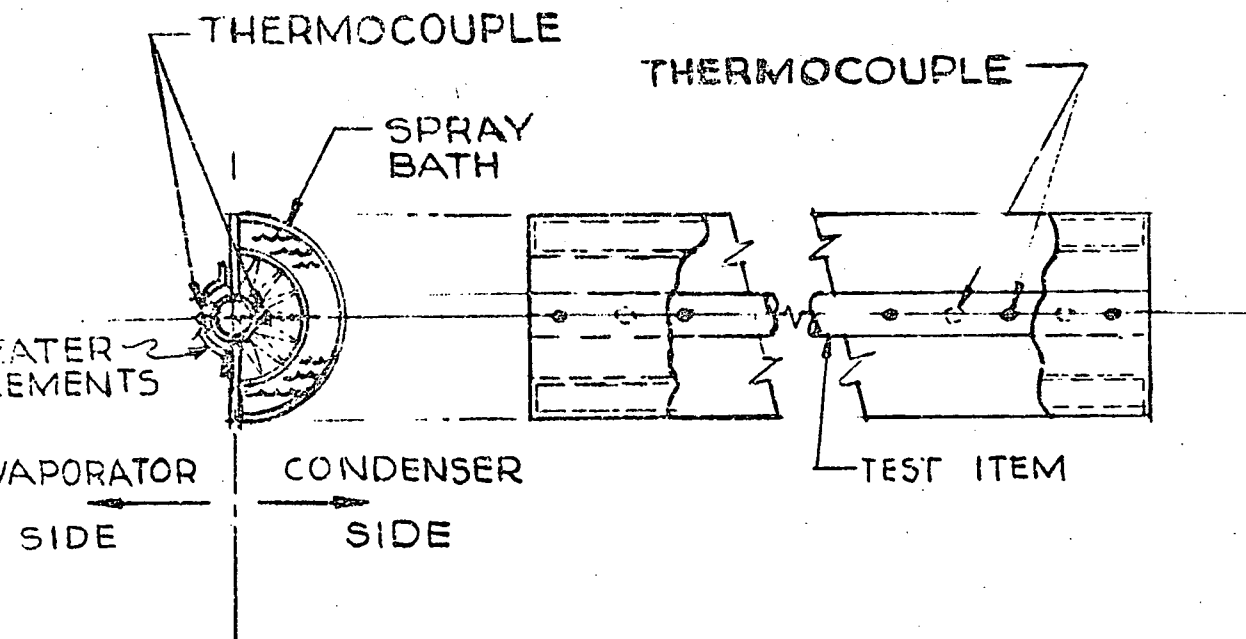
Test No.	Coolant Inlet Temp. °F	Average Q/L watts/inch/side	Max Q/L watts/inch/side	Active Coolant Passage	Comments
1	—	—	to burnout	—	o baseline data-longitudinal
2	—	—	to burnout	—	o baseline data-lateral
3	70	0	0 to burnout	2	o sensitivity to variations in average and maximum loads
4		2	2 to burnout	2	
5		4	4 to burnout	2	
6		6	6 to burnout	2	
7	80	0	0 to burnout	2	o sensitivity to coolant inlet temperature variations
8		2	2 to burnout	2	
9		4	4 to burnout	2	
10	90	0	0 to burnout	2	o coolant temperature variation
11		2	2 to burnout	2	
12		4	4 to burnout	2	
13	100	0	0 to burnout	2	o coolant temperature variation
14		2	2 to burnout	2	
15	70	0	0 to burnout	1	o single coolant passage data simulating partial failure
16		2	2 to burnout	1	
17		4	4 to burnout		
18	70	2	2 to burnout	2	o dump HP charge, evaluate cold rail w/o HP
19	70	4	4 to burnout	2	

ELECTRICAL HEATER
ON EVAPORATOR

SPRAY BATH
ON CONDENSER



(a) LONGITUDINAL CONDUCTIVITY



(b) LATERAL CONDUCTIVITY

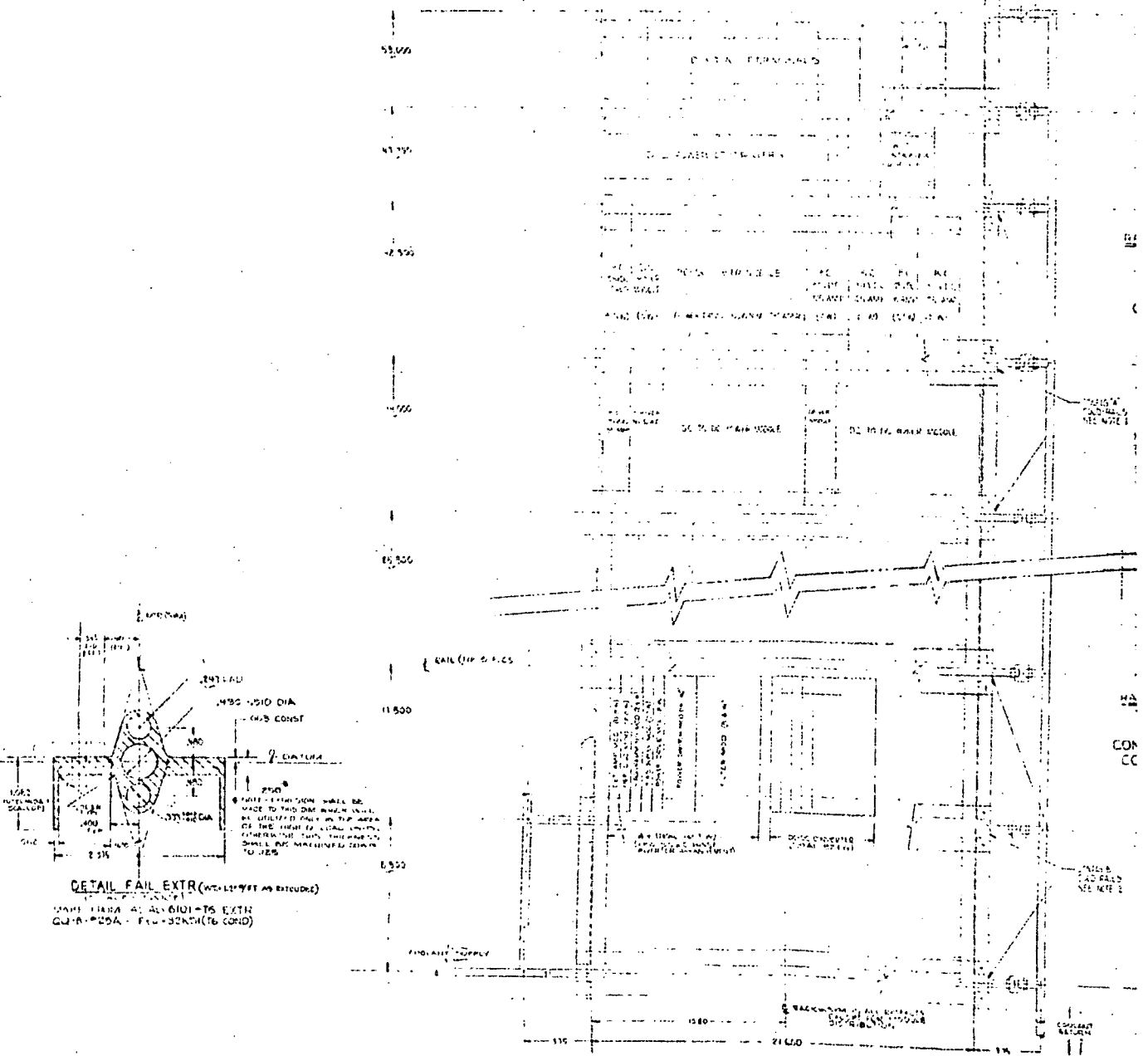
FIG 91.7-BASELINE TEST SETUP

makes all flange mounting locations equivalent, eliminating one variable from the designer's consideration.

Fabrication of a heat pipe augmented cold rail should present no major manufacturing problems, and can be completed at reasonable cost using state of the art techniques.

EOLDOUT PLAN

PRECEDING PAGE BLANK NOT FILMED



FOLDOUT ERAML 2

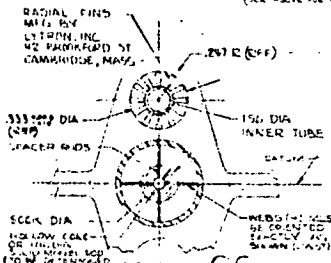
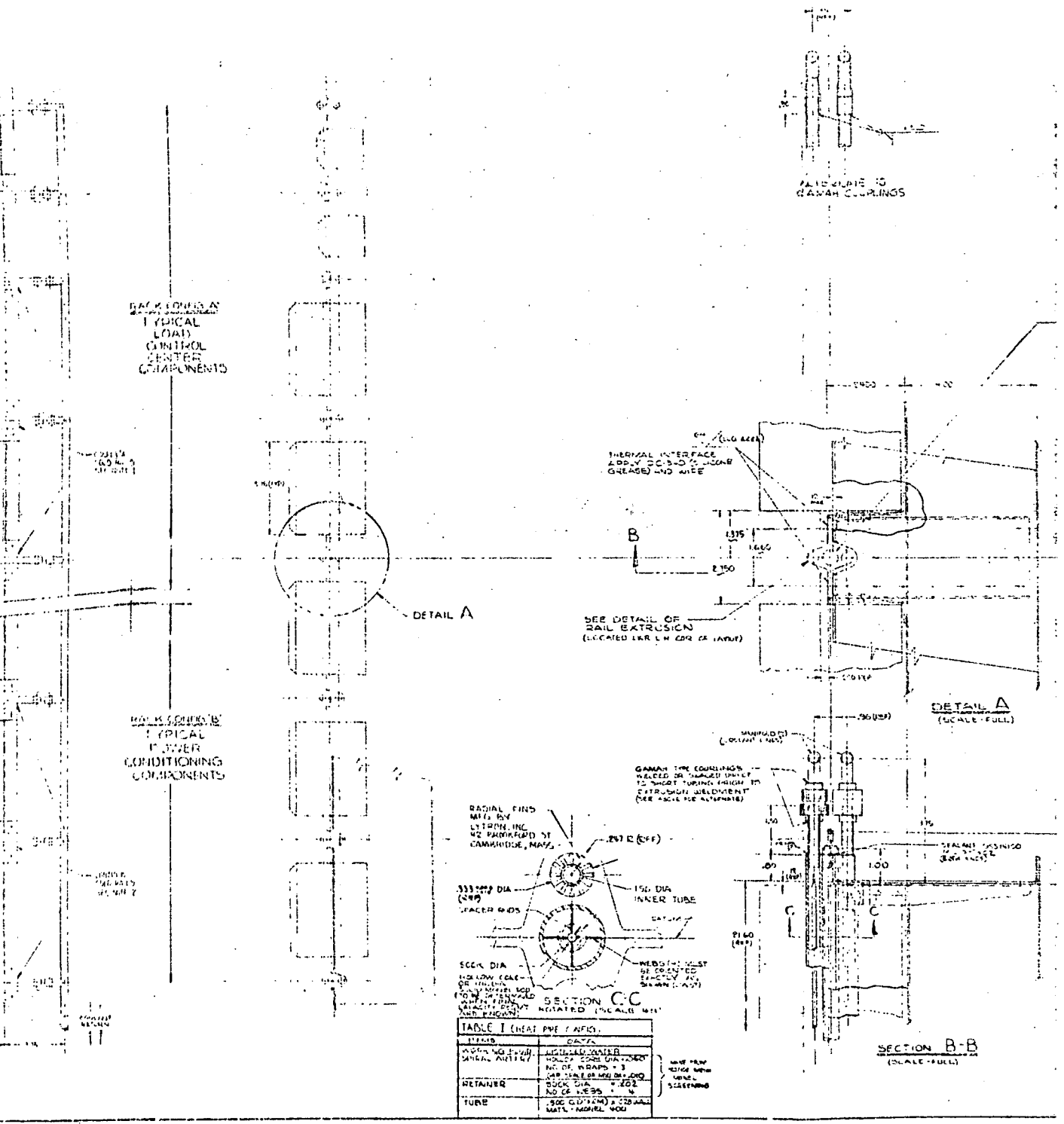


TABLE I (HEAT PIPE CONFIG.)

ITEMS	DATA
WATER	WATER
WATER NO. 1-1/2 IN	WATER NO. 1-1/2 IN
SHEATH MATERIAL	WATER NO. 1-1/2 IN
RETAINER	WATER NO. 1-1/2 IN
TUBE	WATER NO. 1-1/2 IN

NOTE FROM
NOTICE FROM
SCREENING

SECTION B-B
(SCALE FULL)

EQLDOUT FRAME 3



NOTES: (UNLESS OTHERWISE SPECIFIED)

WEIGHT	1.00
...	...
TOTAL	2.00

KEEPER HEAT SHIELD IS PLACED
 MATERIAL: ALUMINUM ALLOY 7050
 THICKNESS: 0.063
 GROOVE INSIDE DIA: 1.00
 INSTALL TUBING WITHIN GROOVE
 IN ALL SECTIONS. THE
 GROOVE SHALL BE MADE TO
 PROVIDE A PUSH FIT FOR THE
 TUBING (BOTH 4 TERN ALLOWANCE)
 THE COMPRESSIVE STRESS SHALL
 BE BONDING WITH INSUL. GRAY
 SILVER FILL BY K-2000 (REF. INSTRUCTION
 PER DD 252-1-13 FOR S-1000)

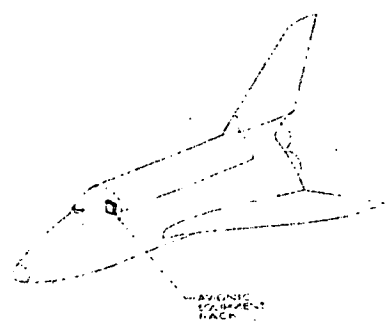
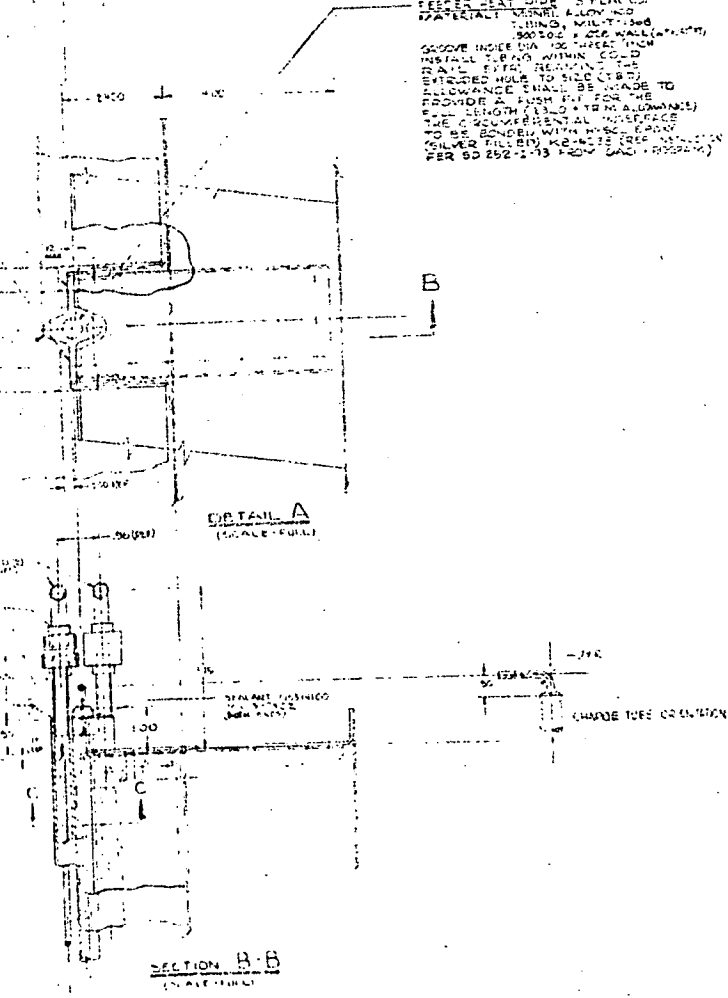


FIG. 9-1.8

DRAWING NO. 9-12034		CHANGING ROOMS COMPARTMENT	
DATE: 11/15/54		REVISED BY: R. H. L. H.	
PROJECT: AUGMENTED COLD RAIL RACK CONFIG 'A' & 'B' (LAYOUT)		PART NO. 5PL-104	
REV. NO. 26512		REV. NO. 104	

AVIONICS HEAT PIPE CIRCUIT

9-2.1 INTRODUCTION

The heat pipe avionics circuit is a cold rail-type electronic box cooling system using heat pipes within the cold rails as heat transfer devices and the pumped coolant as an ultimate sink, rather than the conventional system which pumps the fluid through the cold rails themselves.

The concept of a heat pipe avionics circuit follows logically from that of a heat pipe augmented cold rail. The heat pipe in the center of the cold rail is connected to a heat pipe header, which in turn is connected to the pumped liquid coolant loop. Instead of being used solely to isothermalize the cold rail, the heat pipe is the sink within the rail for heat generated by mounted electronics boxes, serving the function of the two fluid passages in the augmented rail. Like the heat pipe augmented cold rail, this system is not sensitive to the position of a module box on the rail, and can accept local concentrated loads higher than those within the capability of a conventional rail. Since the capacity of a rail in this system is not limited by a fluid film coefficient, it can be designed to be higher than in an equivalent fluid-cooled cold rail, whether or not the latter is heat pipe augmented. Further, a fluid-cooled cold rail requires four fluid connections per rail, 28 in a seven-rail system, while this all-heat pipe seven-rail circuit requires only two. The failure of a heat pipe has much less effect on the cabin environment than the failure of a line or connection in a fluid loop, as the heat pipe contains less than 50 grams of water at less than two psia., which is the most that can leak in the event of a failure of a single heat pipe.

9-2.2 ANALYSIS

Drawing SPL-102B shows a typical system, an equipment rack consisting of seven heat pipe cold rails connected to a heat pipe header, which interfaces with the pumped coolant fluid loop by means of a heat exchanger. The length of the rails shown in the drawing corresponds to that of the longest rails on the IM-vehicle, 44 inches. The maximum load on one rail, 200 watts, is derived using the same maximum load of 2.25 watts/linear inch/

side used in the design of Apollo vehicles. The maximum load on the system was taken to be 700 watts, half of the heat load obtained if all the rails were loaded to capacity, as experience has shown that cold rails are rarely fully-loaded. In addition to these heat loads, there are other constraints on the system. The temperature at any module box flange root must not exceed 140°F, and the fluid inlet and outlet temperatures are specified for maximum load conditions at 68.3°F and 96.4°F, respectively, implying a flow rate of 85 lb/hr of distilled water as coolant through the heat exchanger for a 700 watt load. The system is designed for use inside the pressure shell, so all heat pipes use water as a working fluid to avoid toxic or high pressure fluids within the life support area. The evaporator film coefficient used is 2,000 Btu/ft²-hr-°F and the condenser coefficient assumed is 2,500 Btu/hr-ft²-°F. These values are conservative, since higher film coefficients are being obtained in current heat pipe performance testing at Grumman. Thermal analysis of the circuit was conducted as outlined below:

- (1) Based on preliminary sketches, box flange to feeder heat pipe and feeder heat pipe to header heat pipe conductances were defined;
- (2) The feeder heat pipe vapor temperature was determined by calculating the temperature drop from the 140°F module box using the heat flux (200 watts) and the conductance;
- (3) The header heat pipe vapor temperature was determined by calculating the temperature drop from the feeder heat pipe vapor temperature using the heat flux (200 watts) and the feeder-to-header conductance;
- (4) The required "UA product" of the heat exchanger was calculated using the logarithmic mean temperature difference between the header heat pipe vapor and the fluid;
- (5) A heat exchanger which met this requirement and which caused a minimal pressure drop in the coolant loop was designed.

Details of the calculations performed in this analysis are given in Appendix E.

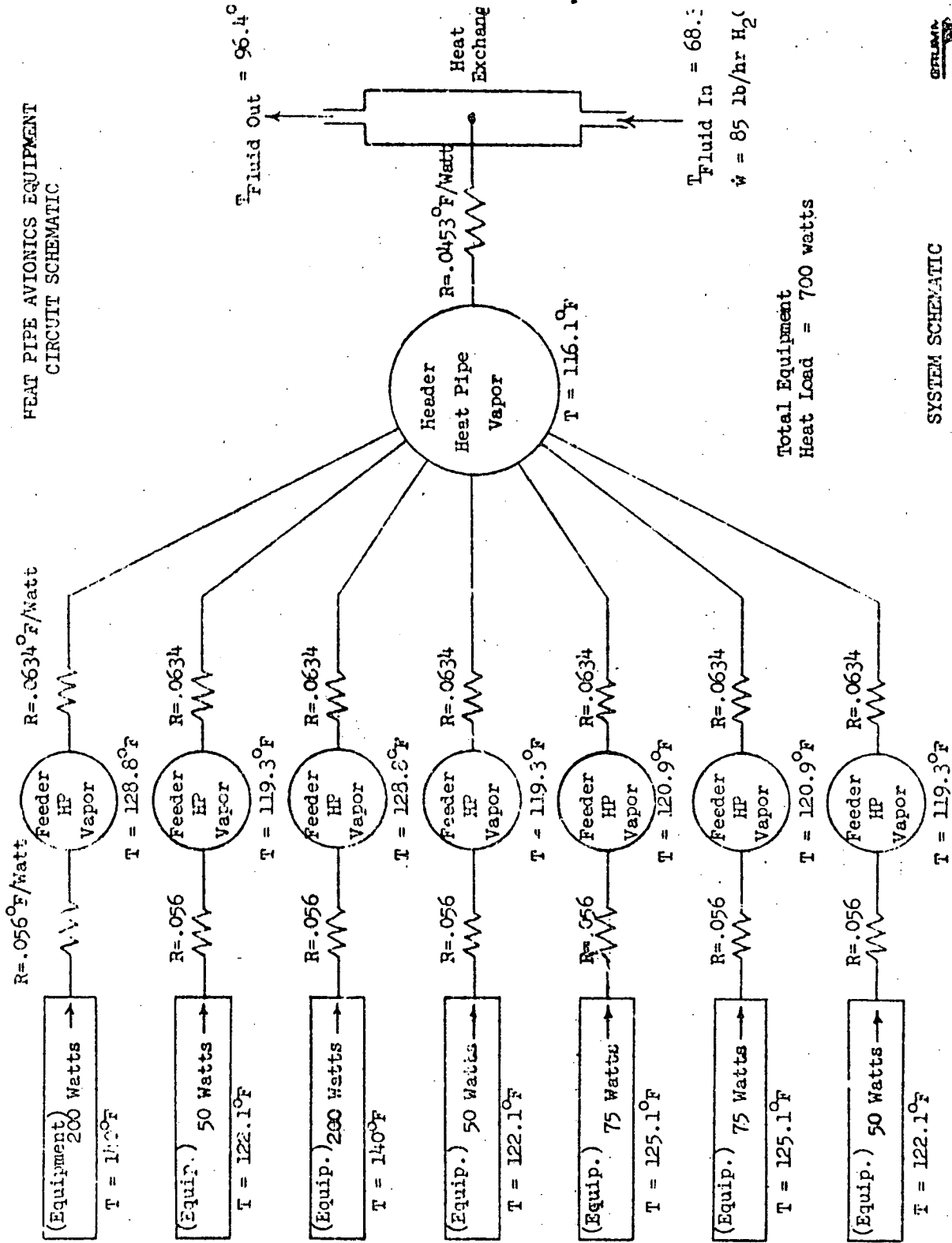
Conductances and temperatures in the system at design conditions are shown in schematic form in Figure 9-2.1. The equipment heat loads indicated are uniformly distributed over the two flange surfaces of the rails. As indicated by the resulting temperatures, the design goals are met.

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 9-2.2. 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°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°F . The temperature drop from a 140°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 9-2.3. 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 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

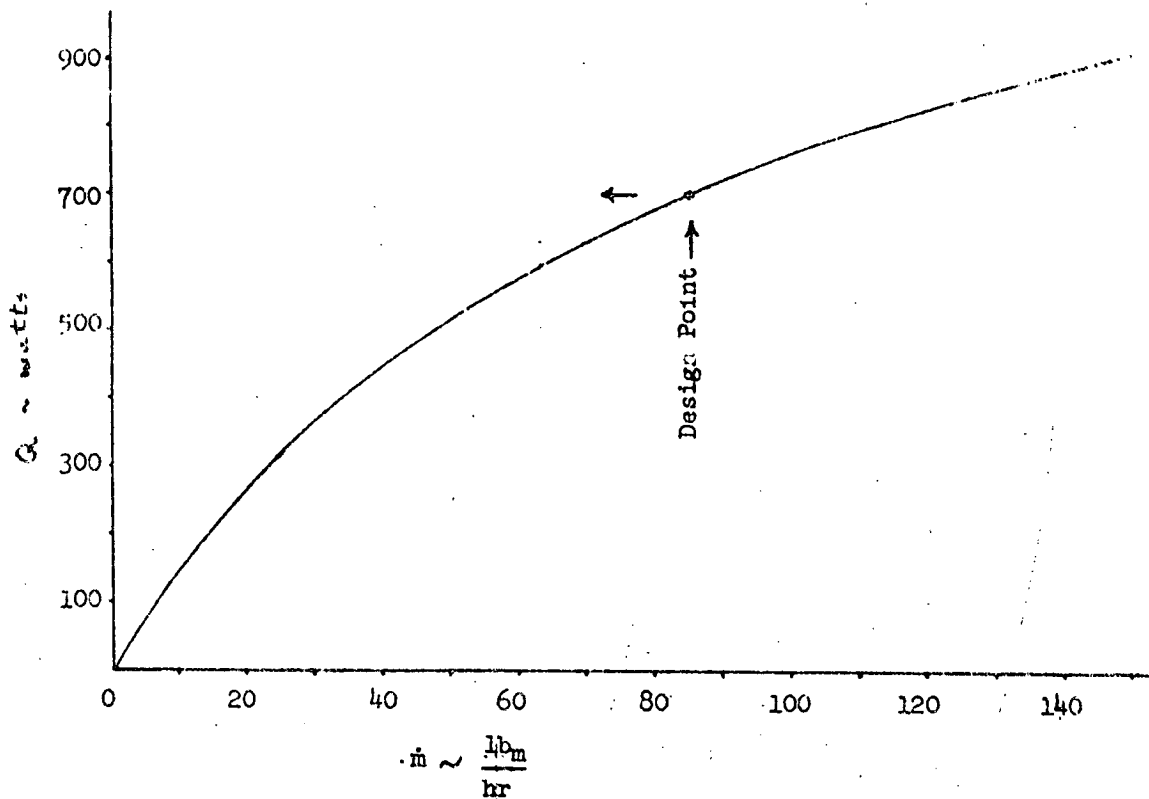
HEAT PIPE AVIONICS EQUIPMENT
CIRCUIT SCHEMATIC



SYSTEM SCHEMATIC

FIGURE 9-2.1

FIGURE 9-2.2
SYSTEM CAPACITY AS A FUNCTION OF COOLANT FLOW RATE



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

$$Q \text{ into coolant} = \dot{m} C_p (t_{\text{Fluid Out}} - t_{\text{Fluid In}})$$

By equating heat fluxes, the equation of the curve above is derived:

$$Q = \dot{m} C_p (T_{\text{HP Header}} - T_{\text{Fluid In}} - \frac{T_{\text{HP Header}} - T_{\text{Fluid In}}}{\ln^{-1} \left(\frac{U_o A_o}{\dot{m} C_p} \right)})$$

where

$$T_{\text{HP}} = 116.1^\circ\text{F}$$

$$T_{\text{In}} = 68.3^\circ\text{F}$$

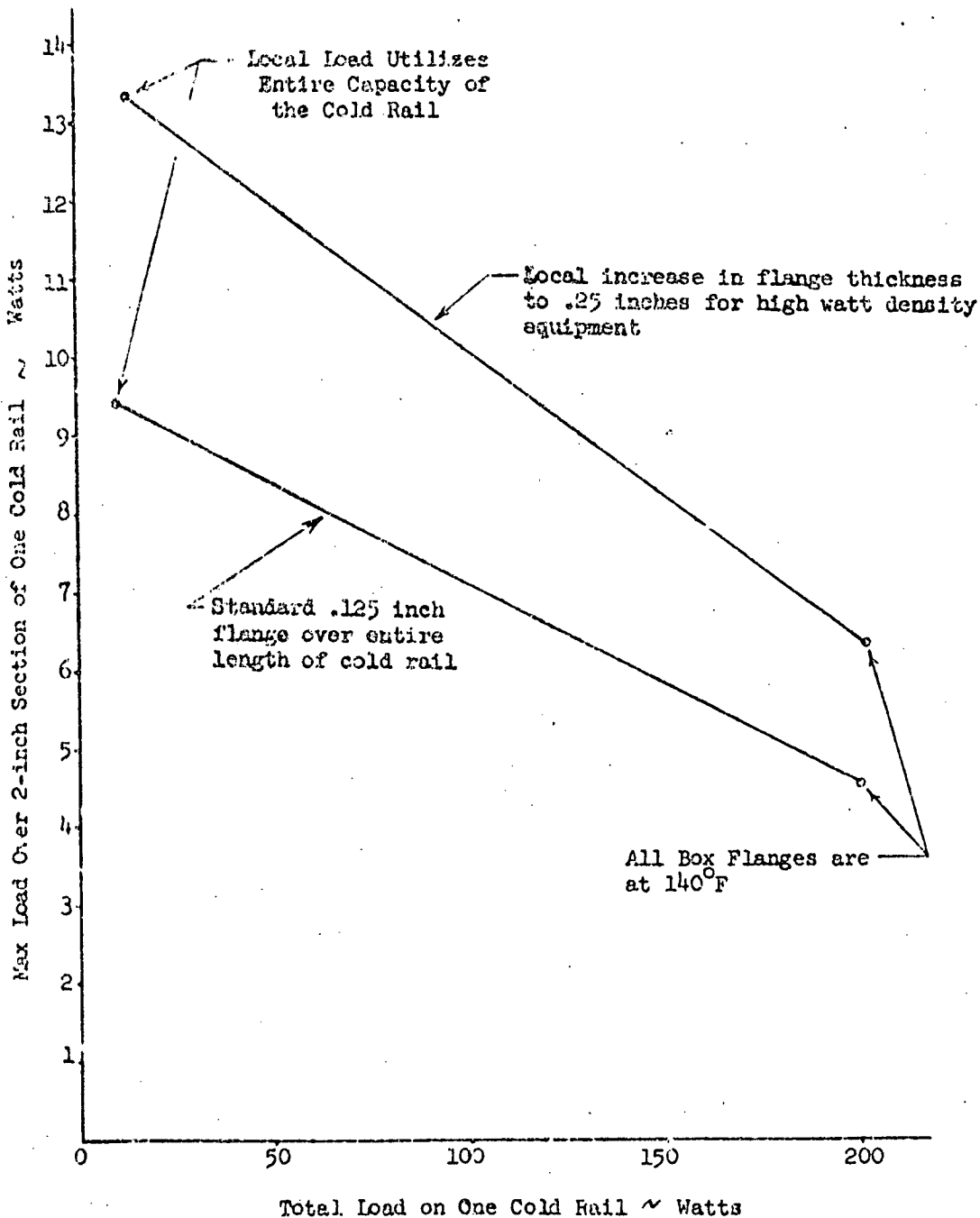
$$C_p = 1, \text{ as coolant used is water}$$

$$U_o A_o \text{ is a function of flow rate and exchanger design}$$

FIGURE 9-2.3

Local Max Load vs. Total Load on a Cold Rail

- o System Load is 700 watts
- o Coolant Flow Rate is 85 lb/hr
- o Heat Pipe Header is at 116.1°F



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 9-2.3. Other design capacities can easily be obtained with simple system changes.

9-2.3 DESIGN

The detailed design, including design specifications, is shown in drawing SPL-102B. Detailed weights are summarized in Table 9-2.1. The rails are formed of the same extrusion used to form the augmented cold rail, with the two fluid passages milled away as shown in Detail P of the drawing. This extrusion is of 6101 aluminum alloy, chosen for its high thermal conductivity. Material specifications are given in Section 9-1 of this report.

Since the working fluid of all heat pipes in the system is water, aluminum is unacceptable as a heat pipe wall material because non-condensable gas will be generated, effectively destroying the ability of the heat pipes to function in the system. To avoid this problem, a 101 (OFHC) copper tube (ASTM specification B68.75.280) will be inserted into the aluminum extrusion, which will be reamed to provide a push fit. Before installation, silicon grease (DC-340) will be applied (Grumman specification LSM-14-6006 dated 19 July 1966), then wiped away to leave a thin film. A ball or "bullet" will then be drawn through the copper tube expanding it, to insure good thermal contact with the aluminum.

The copper tube will then be grooved to 100 threads per inch, and the wick shown in Detail N will be inserted. (Internal configuration of the heat pipe and weights per foot of the cold rail assembly are given in Table 1 of the drawing.)

HEAT PIPE CIRCUIT AVIONICS EQUIPMENT RACK SPL-102B

ITEM	QTY	UNIT WEIGHT LB.	TOTAL WEIGHT LB.	DATA/COMMENTS
Heat Pipe	7	(.228) lb/ft	6.70	Unit length 50" rail not included
Fluid H ₂ O		.017 lb/ft		Distilled water
Spiral Artery		.030 lb/ft		Hollow core Dia. .06"
Retainer		.020 lb/ft		Sock dia. .260"
Tube		.161 lb/ft		0.5" OD X 0.28 copper
Elbow Transition	7	.029	.20	Monel
Condenser Section	7	.161	1.12	1/2" OD X 0.28" monel (.26 lb/ft) config. A (config. B 0.93 lb)
Header		.330	1.49	.028" monel .42" R SS artery .05 lb/ft config. A (config. B 1.40 lb) .032" Al.
Header Supports	21	.011	.23	Al.
Radius Blocks	42	.004	.17	Al.
Supports	14	.05	.70	.032" Al.
Straps	14	.008	.11	Fiberglass epoxy 07 lb/in ³
Blocks-Insulation	14	.023	.32	
Header/Fluid Xchr Assy			(.86)	
HP Header Assy		.380 lb/ft	.52	Copper 101 (tubing & screening)
Charge Tube		.033	.03	Copper (CDA #145)
Fin Corrugation		.015 lb/ft	.02	.010" Al.
Outer Jacket		.104 lb/ft	.12	.020 Al. aly 6061-T6
Flange (Machined)		.069	.07	Al. Aly
Flange (Weldment)		.048	.05	Al. Aly
Flange		.050	.05	Al. Aly
Total Weight			<u>11.90</u>	
Config. A				
Config. B			11.62	

TABLE 9-2.1



An elbow joint is machined from CDA #145 copper rod, as shown in Detail D, and tungsten arc welded (in accordance with Grumman specification Q.C.P.6.001) to the feeder heat pipe evaporator section installed within the extrusion. The wick of the condenser section (shown in Detail N) is attached next, by means of an elbow splice section created by electrodepositing copper on models of the wicks. The 101 (OFHC) copper condenser envelope, grooved (100 threads per inch) and then flattened on one side (as shown in Section F-F), is tungsten arc welded (as shown) to the elbow joint. The heat pipe walls and wick are then coated with Ebonol C. Such a coating increases the ability of water, the working fluid, to wet the surfaces of the heat pipe, enhancing pipe performance.

The end cap/charge tube assembly on the evaporator end of the feeder heat pipes is shown in Detail L. The assembly would be made from 0.5 inch diameter CDA #145 copper rod, pressed into the copper heat pipe envelope, and soft soldered in accordance with specification MIL-S-6872 using 60/40 tin/lead solders. The end caps at the condenser end (not shown) are similar, but the charge tube is not included.

Although dimensions are different - the header's diameter is larger and its evaporator has the flattened side - the construction of the heat pipe header is similar to that of the feeder heat pipes. The one significant difference is that a fluid heat exchanger is built around the header's condenser section so that, when the system is in operation, the condenser is immersed in coolant from the fluid loop.

After grooves have been cut in the condenser section, the outside is tin-plated, and the corrugated fin (Section J-J) and end plates, both 6061 aluminum alloy, are brazed to it. The aluminum outer jacket is then slipped over the assembly and resistance welded to the end plates in accordance with specification MIL-W-6860. The coolant inlet and outlet ports, part of the

machined end caps of the heat exchanger, are shown in Section G-G.

The rack assembly will then be laid out, and the header soldered to the condensers of the feeder heat pipes, in accordance with specification MIL-S-6872. Soldering provides excellent thermal contact, but to achieve the required structural rigidity, straps and radius blocks are necessary, as shown in Sections C-C and E-E. The straps are to be cut from type 304 stainless steel sheet (material specification QQ-S-7660, cond A), and the radius blocks machined from 2024-T62 aluminum alloy (material specification QQ-A-250/5) and given an alodine surface finish in accordance with specification MIL-C-5541.

As shown in Figure 9-2.4, a system orientation diagram, the heat pipes in the circuit are never required to work against the g-vector (i.e., evaporator end "up").

9-2.4 TEST PLAN

The testing described in this section is intended to provide verification of the thermal performance of the HP circuit. Two categories of testing are required, Component Qualification and System Performance. The former are designed to show acceptable operation for each HP prior to its inclusion in the system. This will minimize the chances for rework after assembly as well as provide information for possible system anomaly investigations. The system tests are designed to completely characterize the circuit's operation.

9-2.4.1 CONFIGURATION

The actual test article must deviate from the configuration shown in Figure 9-2.4 to permit proper ground testing. Figure 9-2.5 defines the test geometry which provides for both "zero-g" (on-orbit) and reflux boiler (entry) simulation. The test geometry differs from that of the analytical model in that the header heat pipe is straight rather than "L"-shaped.

Thermal input will be simulated with electrical heaters on each rail and cooling by the water flow through the jacket. The entire circuit will

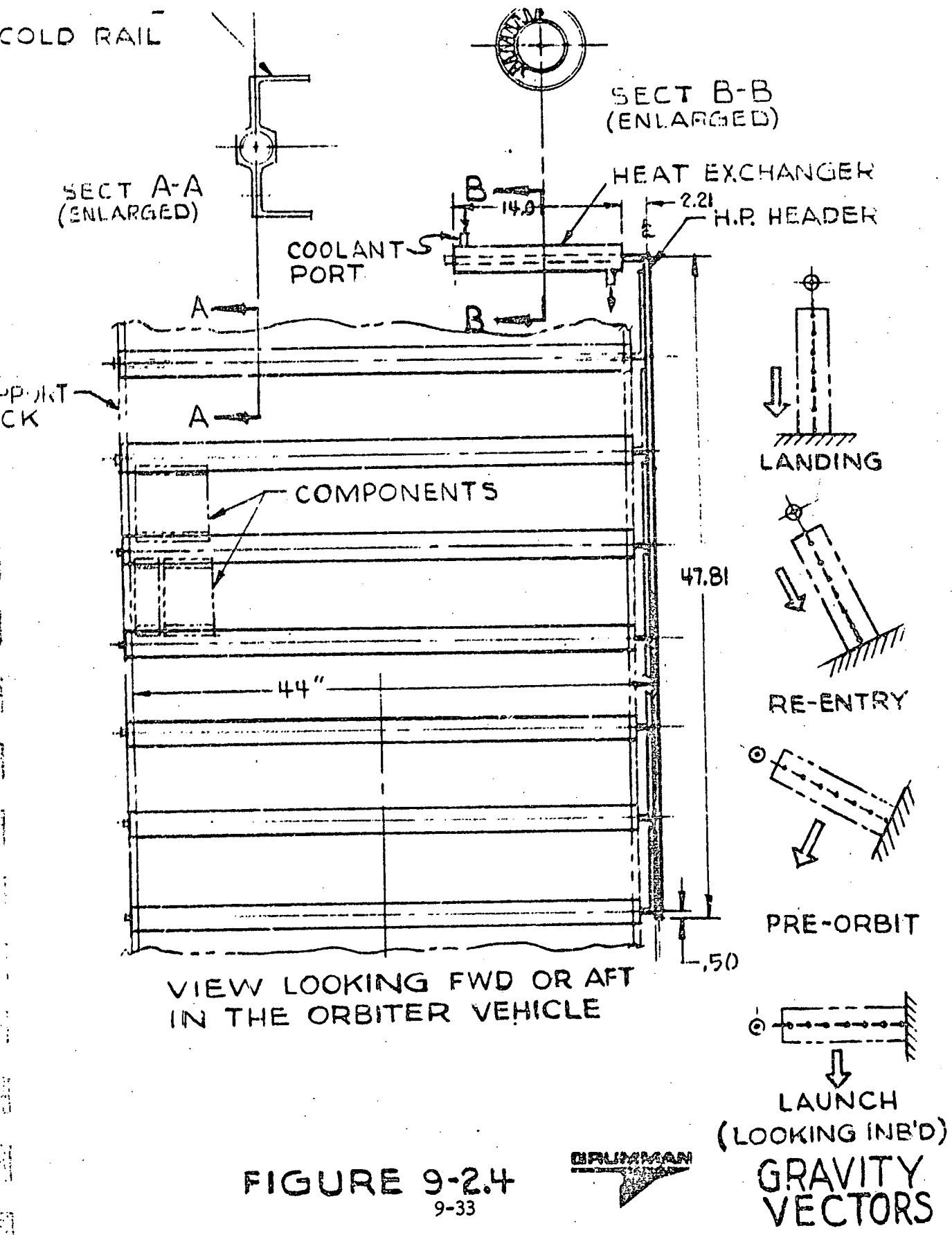


FIGURE 9-2.4
9-33

HEAT PIPE CIRCUIT
(SHOWN IN THIS VIEW
PRIOR TO COMPOUND
TILT)

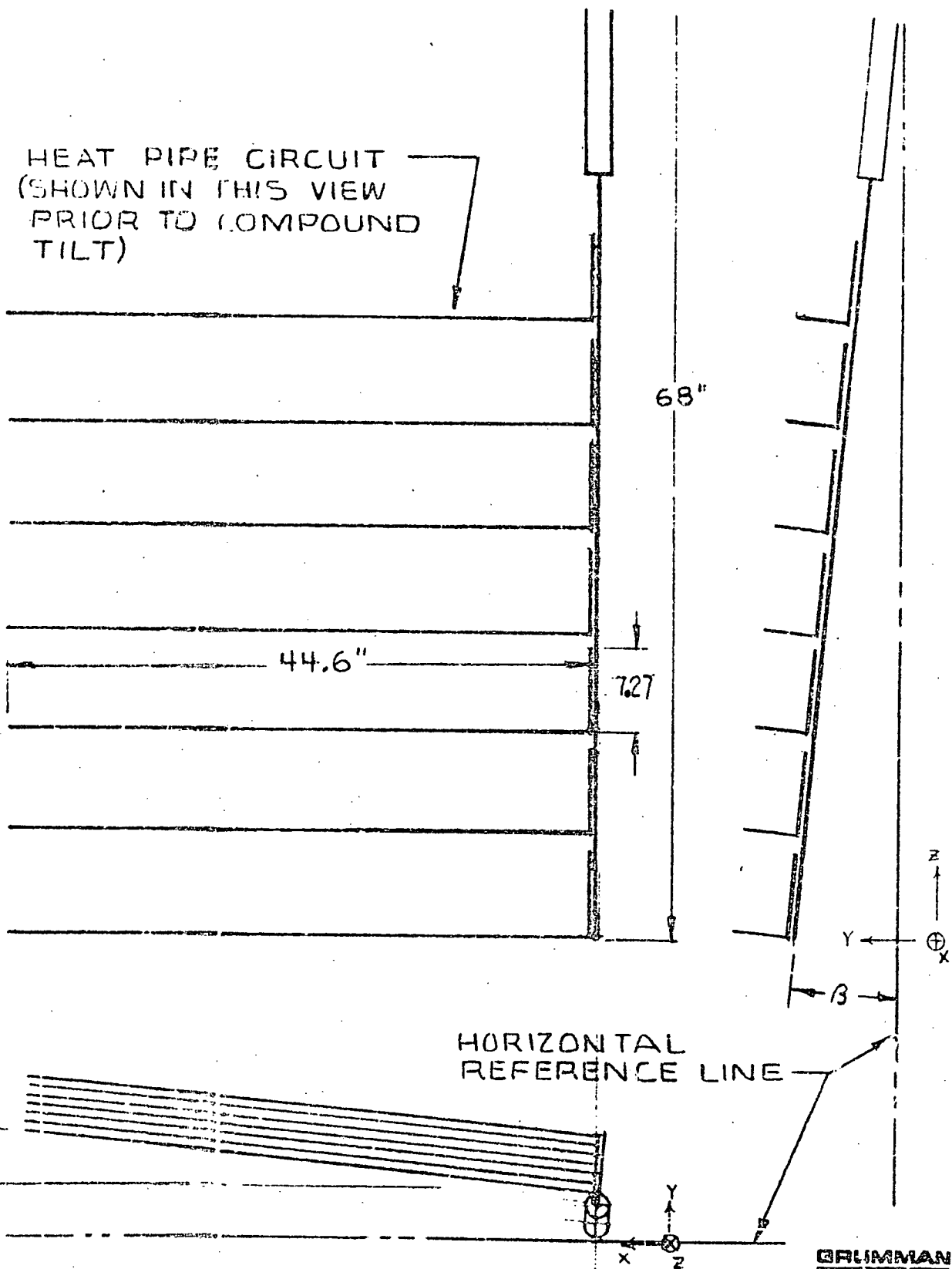


FIGURE 9-2.5



be mounted to a rigid test frame which can be levelled by use of screw jacks. This frame will also be used for handling and shipping.

External hookups for power, temperature instrumentation and fluid flow are provided for interfacing to existing facilities. The facility requirements are summarized below:

Power: 14 variac channels
120 volts, 2 amps each

Instrumentation: Readout provisions (strip charts) for
80 Cu-Cn thermocouples

Coolant: 150 lbs/hr of water with inlet temperature
control 40 to 120°F

Work area: 4' by 3' work surface with 4' minimum
clearance to ceiling

9-2.4.2 TEST PROCEDURE

Component Qualification: Each HP (8 total) will undergo qualification testing prior to acceptance. Each pipe will be equipped with a ribbon heater over its evaporator section and will be fully instrumented to obtain temperature data. Each pipe, in turn, will be installed in the bench test rig shown in Figure 9-2.6 and they will then be tested to verify that the horizontal performance listed below can be attained:

Cold rail pipe: 200 watts with 10°F delta-T

Header pipe: 700 watts with 10°F delta-T

The cold rail HP will be cooled with a spray bath (Figure 9-2.6) and the final check of the header will be performed with its cooling jacket installed (Figure 9-2.7). However the header performance will initially be verified using the spray bath.

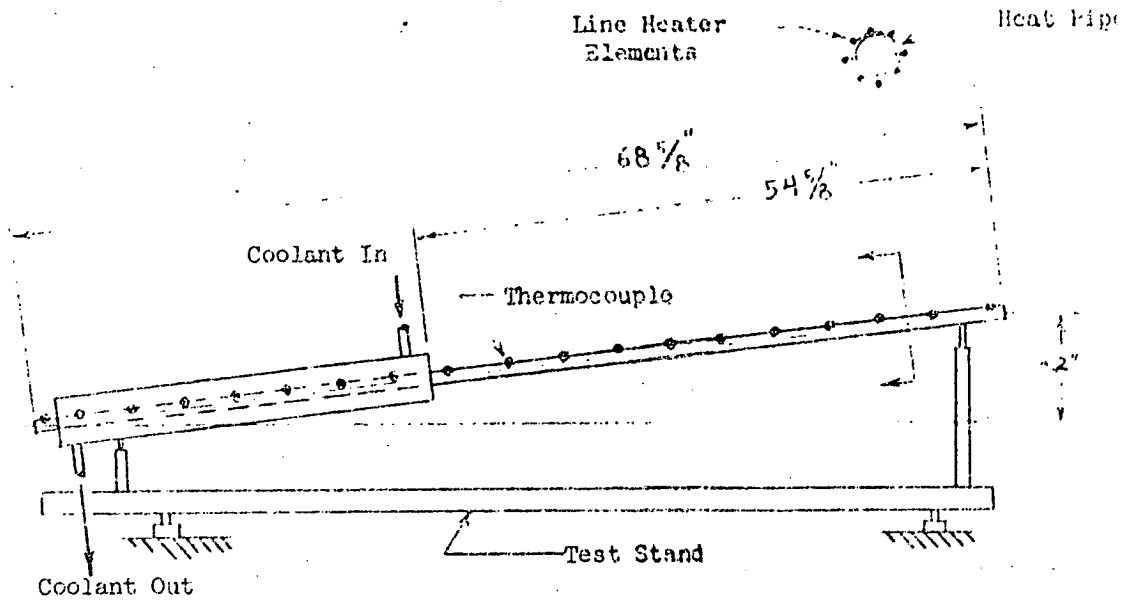


FIGURE 9-2.6 - BENCH TEST ARRANGEMENT - HEADER PIPE

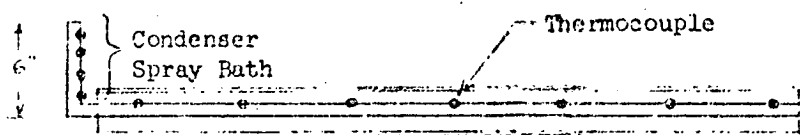
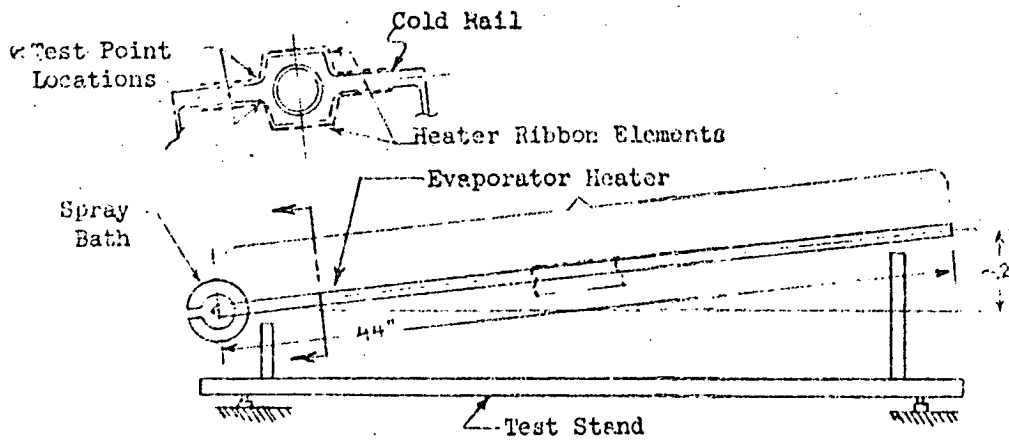


FIGURE 9-2.7 - BENCH TEST ARRANGEMENT - COLD RAIL PIPE

System Performance: After all components have been checked they will be assembled to form the circuit. Heater ribbon will be mounted on the rail flanges to simulate module box thermal input. The entire circuit will be held in a test fixture to maintain the geometry shown in Figure 9-2.5.

The first series of tests will be performed with the Z-X plane horizontal thus simulating "zero-g" operation. The emphasis during testing will be on understanding the interaction between loads on the circuit as they cycle on and off, at different power levels. The heater ribbons on opposite flanges on the same rail will be independent, so loads can be different.

Table 9-2.2 presents a summary of the test program. Test #1 is a "bench-mark" test and serves to demonstrate basic system performance. During this test, the two boxes (simulated by two strip heaters) connected to cold rail #7 (see Figure 9-2.5) can either be operating at the same or different power levels. Test #2, with identical heat loads as in test #1 imposed on rail #1, compares the sensitivity of the system to loads on rails at different locations in the circuit. The inputs will be stepped to 200 watts in 50 watt increments in order to determine the operating characteristics over the entire range.

Having demonstrated the ability of the system to accept 200 watts at one location, tests 3 through 6 will demonstrate that the system can accept multiple heat inputs over the entire heater evaporator length. For test 3, rail 1 would remain at a dissipation level of 200 watts, rail 2 would be brought to this same level, and rails 5 and 7 would be sequentially step powered to 50 watts. Subsequently, rails 4 and 6 would be step powered to 75 watts, and rail 3 to 50 watts to complete test 4. These tests will verify that the heat pipe header will perform satisfactorily (i.e., not "starve") as loads are added downstream of a large heat source. This series has been designed to show system performance while transporting a maximum load over the length of the intermediate header. Tests 5 and 6 have been

Table 9-2.2 - Test Loading Conditions

Test #	Rail #							Comments
	1	2	3	4	5	6	7	
1	-	-	-	-	-	-	200	"Benchmark" Test
2	200	-	-	-	-	-	-	
3	200	200	-	-	50	-	50	High Upstream Load
4	200	200	50	75	50	75	50	
5	50	-	50	-	-	200	200	High Downstream Load
6	50	75	50	75	50	200	200	
7	(*)	200	50	75	50	75	50	
8	(*)	200	-	-	50	-	50	Local Load
9	50	75	50	75	50	200	(*)	Test
10	50	-	50	-	-	200	(*)	

(*) The indicated rail will be broken into two zones: a two-inch section on one flange, which will be subjected to a high thermal load and the lightly loaded remainder of the rail. Local high density load capability will be explored.

included to demonstrate that the header pipe will be able to service additional loads upstream of a large heat input. These tests will indicate whether it is a better design practice to position smaller loads downstream or upstream of a large source, or whether it matters.

Tests 7 through 10 determine high watt density load characteristics of the circuit by loading one of the rails with a high density source on two inches, and plotting performance curves of total load on that rail (with a low load on the rest of the rail) against the high density load, keeping the calculated box flange root temperature at 140°F.

9-2.5 CONCLUSIONS

This circuit was designed to meet a specific set of design requirements, so no attempt was made to maximize heat pipe or system capacities. Larger local load capabilities, rail capacities, and total system capacities can be achieved if necessary by increasing the flow rate, lowering the coolant inlet temperature, or increasing the "UA product" of the heat exchanger. The real limits of a heat pipe circuit such as this one are the transport capacities and the maximum allowable input load densities of the heat pipes employed. The present design is well below these limits.

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 per rack, as opposed to four per conventional rail, 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.

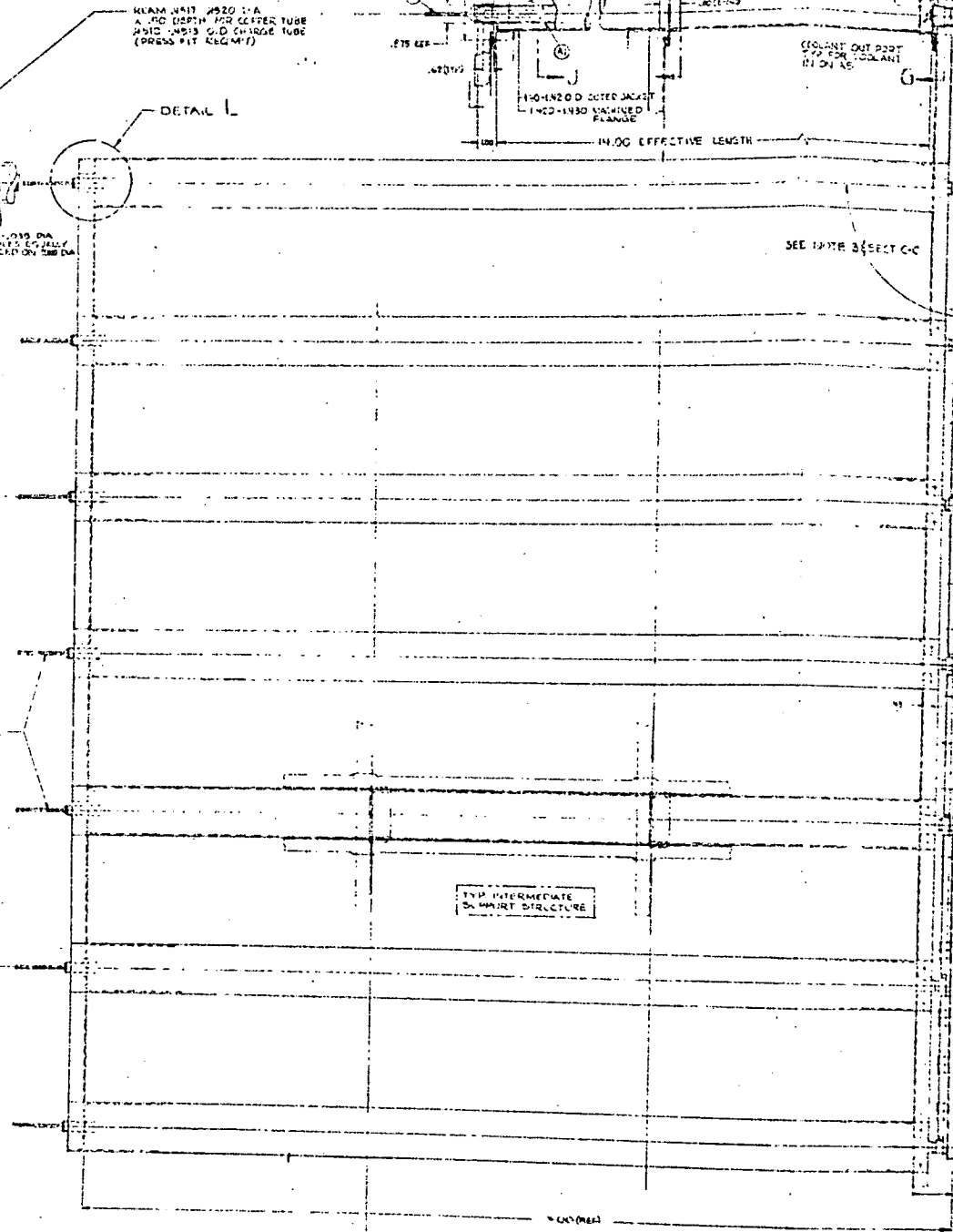
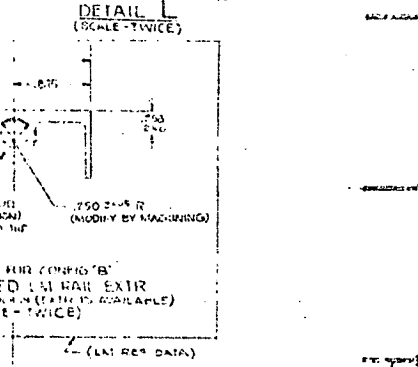
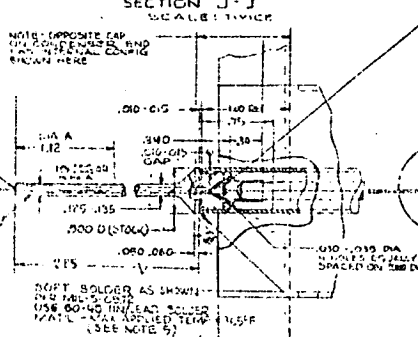
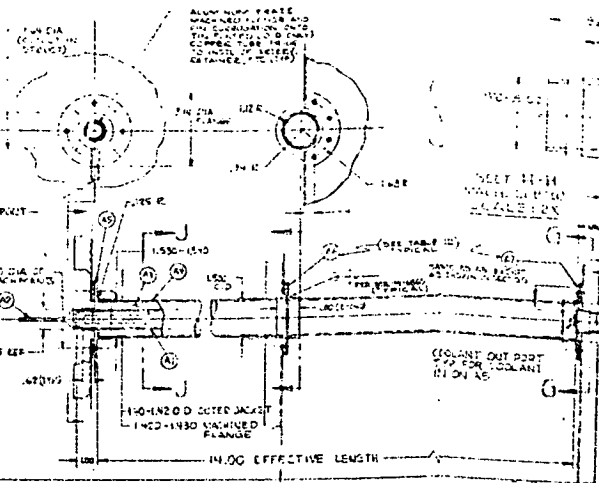
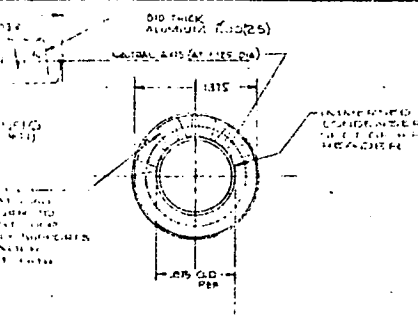
Such units also present advantages in the cooling of development flight instrumentation, carried only during early flights because installation of such a rack disturbs the already-installed fluid loop only minimally.

FOLDOUT FRAME 2

HEADER/FLUID HEAT EXCHANGER ASSY

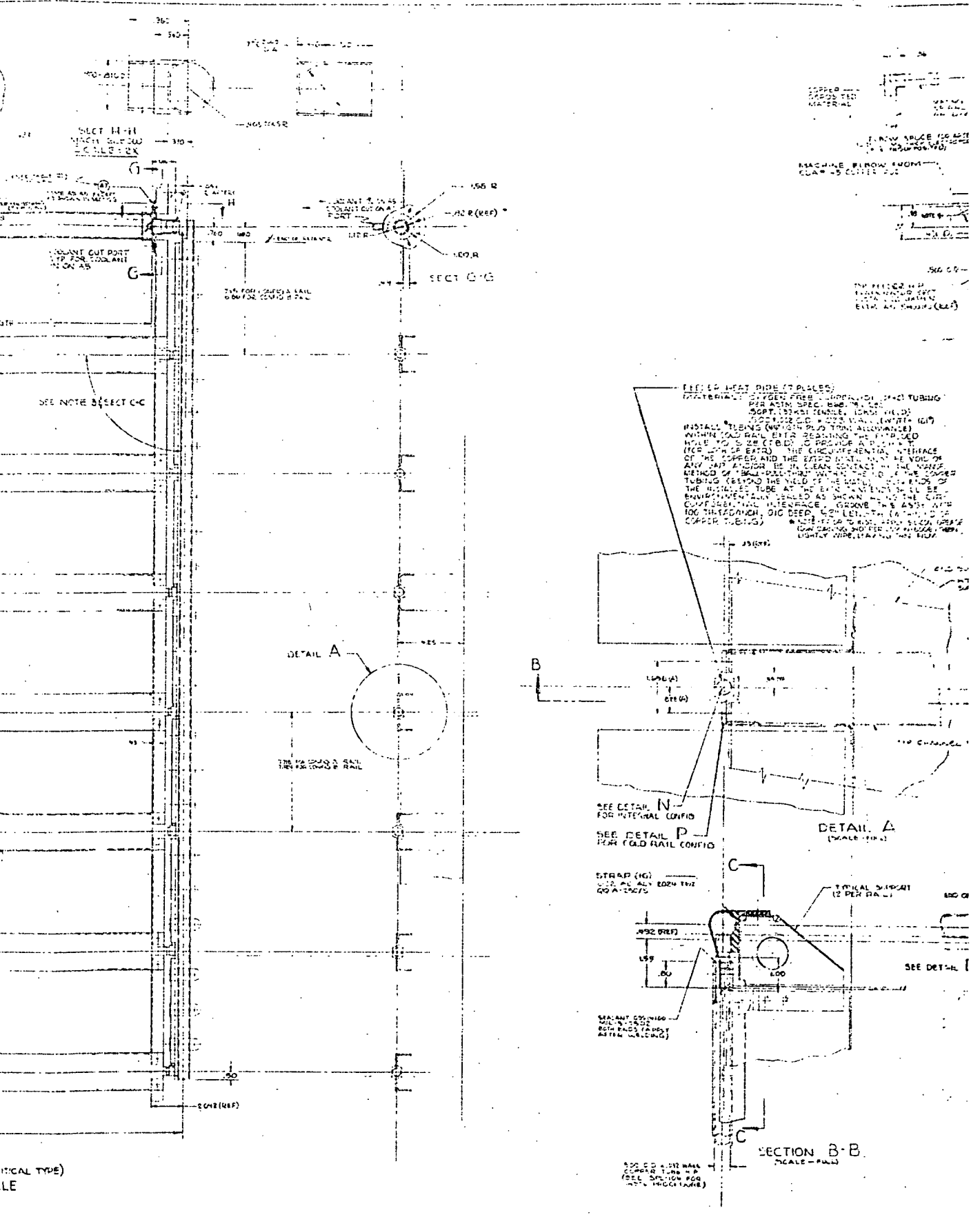
CODE	QTY	DESCRIPTION	UNIT	QTY
A1	1	HEAT EXCHANGER	UNIT	1
A2	1	FLANGE	UNIT	1
A3	1	FLANGE	UNIT	1
A4	1	FLANGE	UNIT	1
A5	1	FLANGE	UNIT	1
A6	1	FLANGE	UNIT	1
A7	1	FLANGE	UNIT	1
TOTAL				7

TABLE II



EQUIPMENT RACK FOR SELECTED COMPONENTS (MEDIUM-CRITICAL TYPE)
VIEW LOOKING FWD OR AFT IN THE ORBITER VEHICLE
(SCALE - HALF)

FOLDOUT FRAME 3



SECTION H-H
FRONT ELEVATION
SCALE 1/2"

SECTION C-C

SECTION B-B
SCALE 1/2"

FIELD HEAT SHIELD (TP PULLED)
MATERIAL: OXYGEN FREE ALUMINUM 304 TUBING
PER ASTM SPEC. 304, 1.5" OD
304 STAINLESS STEEL TUBING (1.315" O.D.)
INSTALL TUBING (NOTING 90° TURN ALLOWANCE)
WITHIN FOLD RAIL WITH BEARING SURFACE TO BE USED
W/ HOLE TO 1/2" DIA. TO PROVIDE A BEARING SURFACE
(FOR 1/2" DIA. STRAP). THE CIRCUMFERENTIAL JOINTS
OF THE STRAPS AND THE FIELD HEAT SHIELD SHALL BE VOID OF
ANY GAP AND/OR BE IN CLEAN CONTACT. THE WELDING
METHOD OF SEAL SHALL BE TIG WELDING. THE WELD OF THE COVERED
TUBING (BEYOND THE FIELD OF THE HEAT SHIELD) SHALL BE
THE UNCOVERED TUBE AT THE END AND SHALL BE
ENVIRONMENTALLY SEALED AS SHOWN. WELD THE CIRCUMFERENTIAL
JOINTS AS SHOWN. THE ASST. W/ 100 MIL/ MIN. DIA. DEEP. 45° BEVEL. (AT THE END OF
COVERED TUBING). *REPAIR TO WELD SHALL BE AVOIDED
(REPAIRING W/ PIPER OR ANOTHER WELD)
LIGHTLY WIRE BRUSH THE WELD.

DETAIL A

DETAIL A
(SCALE 1/2")

SEE DETAIL N
FOR INTERNAL CONFIG

SEE DETAIL P
FOR FOLD RAIL CONFIG

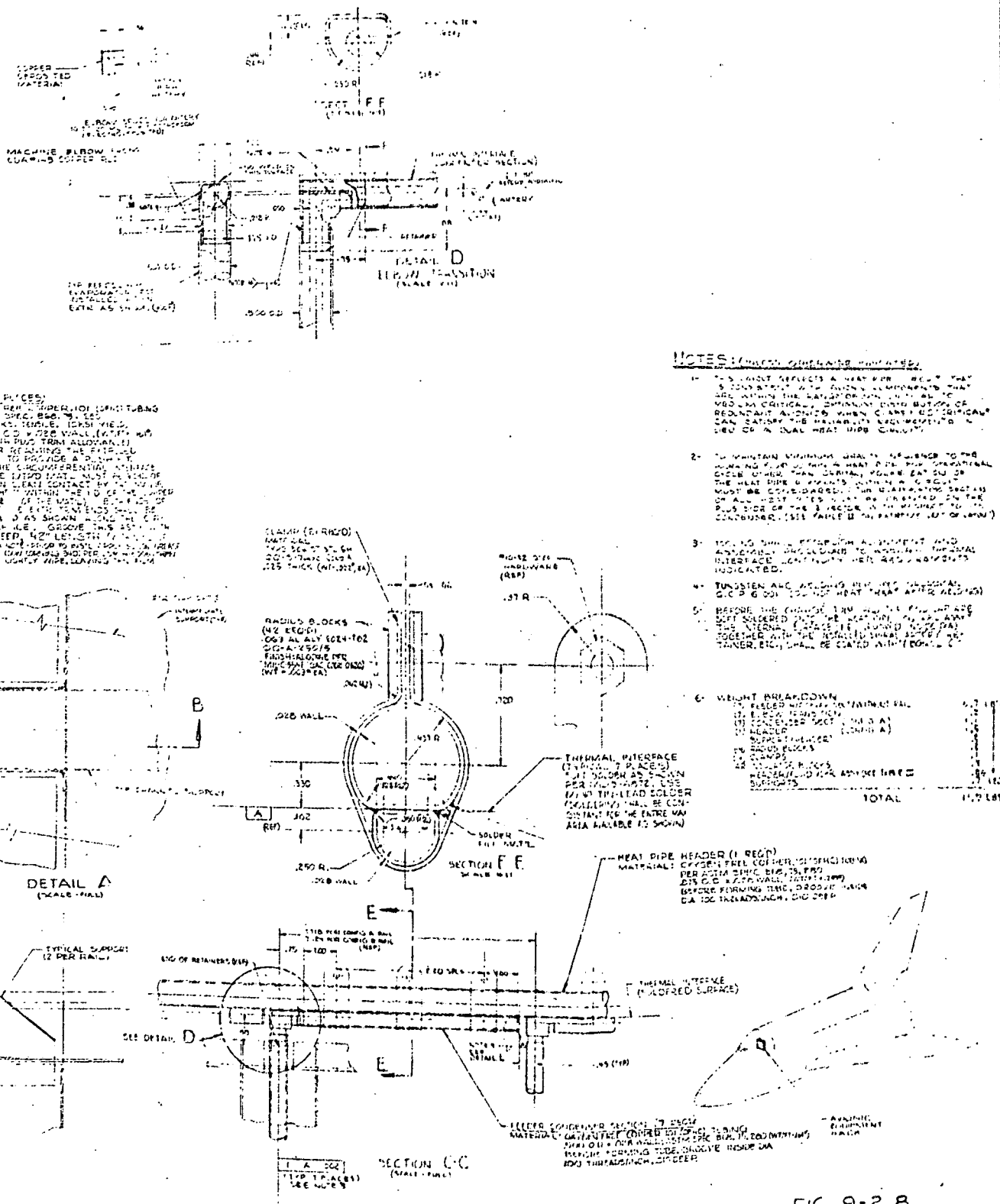
STRAP (HG)
3/16" DIA. 4130 STEEL
GOA-1262/5

TYPICAL SUPPORT
(2 PER DIA.)

SEALANT COVERING
MILS. 3-502
REINFORCED (APPY
ASTM. 1262/5)

304 STAINLESS STEEL TUBING
(SEE SPEC. FOR
WELDING PROCEDURE)

BOLDOUT FRAME



- NOTES:**
1. THIS DRAWING REFLECTS A HEAT PIPE HEAD THAT IS ONLY BEING USED TO ILLUSTRATE THE CONCEPTS THAT ARE WITHIN THE DRAWING. THE HEAD IS NOT BEING CRITICAL. DIMENSIONS AND TOLERANCES OF HEAD ARE NOT CRITICAL. WHEN CANNOT BE CRITICAL CAN SATISFY THE REQUIREMENTS OF HEAD OF A QUAL HEAT PIPE CIRCUIT.
 2. TO MAINTAIN MAXIMUM HEAT PIPE EFFICIENCY TO THE HEAT PIPE HEAD, THE HEAD IS TO BE OPERATIONAL OVER OTHER THAN DURING POWER OFF OF THE HEAT PIPE CIRCUIT. THE HEAD IS TO BE MUST BE CONSIDERED. THE HEAT PIPE HEAD IS TO BE OPERATIONAL OVER OTHER THAN DURING POWER OFF OF THE HEAT PIPE CIRCUIT.
 3. THE HEAT PIPE HEAD IS TO BE OPERATIONAL OVER OTHER THAN DURING POWER OFF OF THE HEAT PIPE CIRCUIT.
 4. TUNGSTEN AND WOLFRAM ARE TO BE OPERATIONAL OVER OTHER THAN DURING POWER OFF OF THE HEAT PIPE CIRCUIT.
 5. BEFORE THE CHANGE IN THE HEAT PIPE HEAD IS TO BE OPERATIONAL OVER OTHER THAN DURING POWER OFF OF THE HEAT PIPE CIRCUIT.

FIG. 9-2.8

NAS 9-1203H	HEAT PIPE CIRCUIT AVIONICS EQUIP HACK
76512	SPL-102B

PRECEDING PAGE BLANK NOT FILMED

9-3 HEAT PIPE/PHASE CHANGE MATERIAL MODULAR HEAT SINK

9-3.1 INTRODUCTION

The electronics for the flight data and voice recorders are integrated in a single package (4" x 7" x 6") and located in the aft fuselage of the shuttle, near the recorder (tapes and drives) assembly.

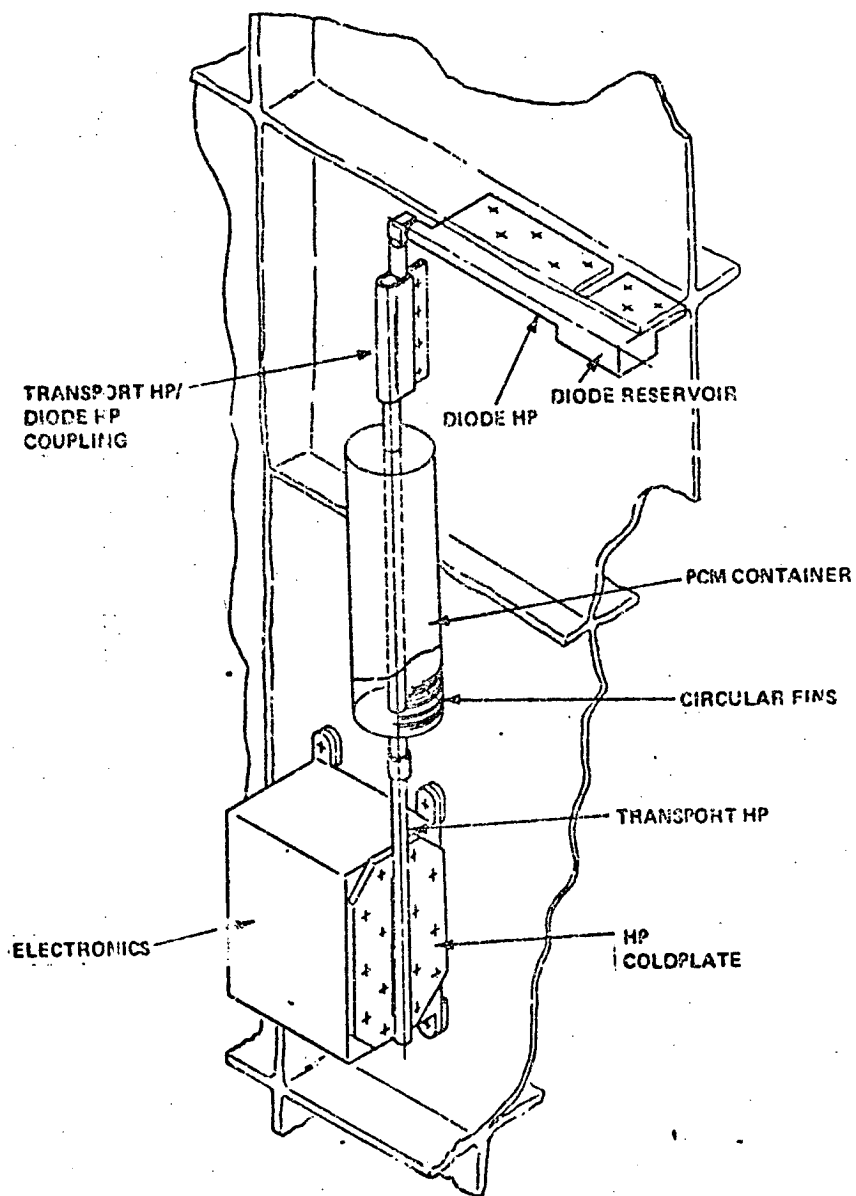
The modular heat sink thermal control concept, as designed for these recorder electronics, integrates heat pipes, a phase change material (PCM) and a structural bulkhead. The self-sufficient system couples the electronics base plate (heat source), via heat pipes, to either the structural or PCM heat sink, as required by the mission.

As illustrated in Figure 9-3.1, it consists of a cold plate which interfaces with the electronics base plate; a simple heat pipe (transport heat pipe) whose evaporator is integral with the cold plate; a PCM container attached to the middle of the transport heat pipe; and a diode heat pipe which connects the other end of the transport heat pipe to the structural bulkhead.

During boost and the entire on-orbit operation the base plate is coupled directly to the structural sink (average bulkhead temperature -40°C to $+110^{\circ}\text{F}$) through the transport and diode heat pipe connection. The intermediate PCM sink is bypassed. During entry the structure becomes too hot to function as an acceptable sink and the diode reverses, decoupling the structure from the transport heat pipe. This creates an isolated system consisting of the base plate, transport heat pipe and PCM container with the heat flux 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.

The PCM sink must be sized to absorb the entire electronics heat load plus the ambient heat leaks for the time that the structure cannot function as the heat sink. In addition, its melting temperature and conductance coupling to the transport heat pipe must be such that the temperature of the base plate does not exceed the maximum permissible design value of 140°F .

FIGURE 9-3.1 - MODULAR HEAT SINK



9-3.2 ANALYSIS

The following paragraphs present the results of the analysis done in support of the detailed design for the modular heat sink thermal concept.

- Requirements and Assumptions

- Electronics

The recorder electronics are designed to function in an ambient thermal environment from -68°F to $+160^{\circ}\text{F}$. The electrical power profiles (Reference 9) call for continuous operation throughout the seven day earth orbital mission (168 hours). Electrical power dissipation is 35 watts with a temperature difference of 20°F between the electronics and the base plate. Therefore, the maximum base plate design temperature is 140°F .

- Environment

Recent shuttle studies predict the following temperature extremes for the aft fuselage structure:

100°F	:	Launch & Boost
-40°F to 110°F	:	On-Orbit
350°F	:	Entry

Thus, the structure can be used as the heat sink for all except the entry mission phase.

Use of the PCM will be required for the following entry phases:

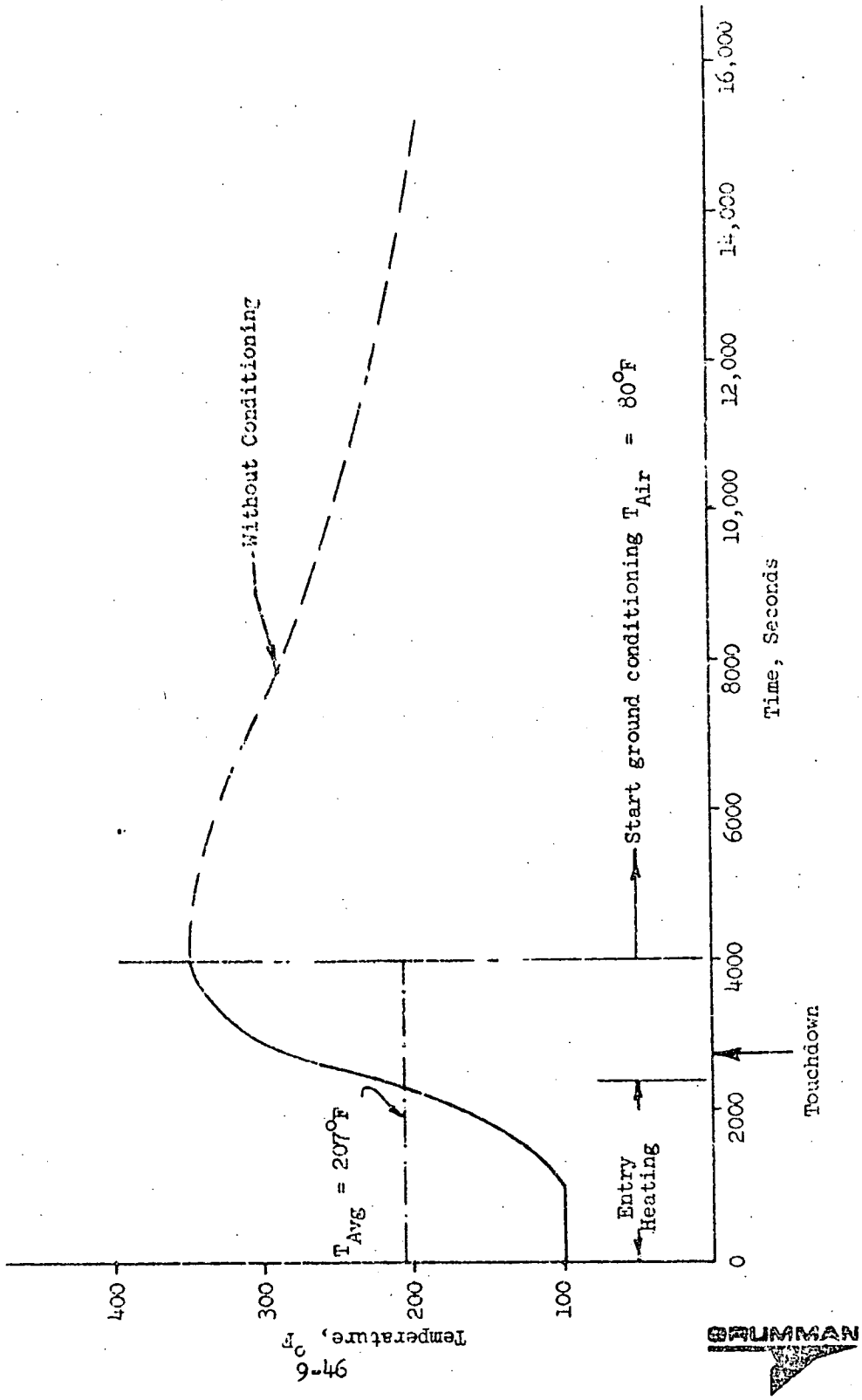
	Time (Min.)
De-orbit	: 21
Entry	: 33
Approach/Landing	: 10
Total	<u>64 Min.</u> = 1.07 Hr.

Figure 9-3.2 contains the temperature response of backface structure representative of the upper aft fuselage area during the above entry phases. Ground conditioning with air at 80°F is assumed after 4000 seconds. The average backface temperature up to ground conditioning is calculated from the response curve as 207°F .

FIG. 9-3.2 TEMPERATURE RESPONSE OF STRUCTURE

o Total Heat = 10,000 Btu/Ft²

c Data Taken from Reference 16



The internal atmosphere due to infiltration during entry results in an estimated average convective film coefficient of .1 Btu/Hr Ft² °F and a bulk atmosphere temperature approximately equal to the average structure.

- Design Parameters

The physical properties of the materials used in the modular heat sink concept are contained in Table 9-3.1. They include properties for the heat pipe working fluids, phase change materials, and insulation.

A contact thermal conductance of 1000 Btu/Hr Ft² °F was used between mechanically bonded aluminum surfaces. It assumes a 64 μ-inch surface finish and the use of wiped silicone grease between the mating surfaces. The value is consistent with the data of Reference 10 and Grumman's own in-house test results which have demonstrated values of 2000 Btu/Hr Ft² °F.

A fin effectiveness of .67 was used when calculating the heat transferred at the heat pipe surfaces. This was verified by calculations for a .500 ID aluminum pipe with .028 inch wall, an inside film coefficient of 2000 Btu/Hr Ft² °F, a uniform heat input over one half the circumference and insulation over the other half.

- System Design and Analysis

Seven basic steps were followed in the system design and analysis:

1. Establish theoretical weight of PCM required
2. Estimate system heat leaks
3. Calculate total PCM weight requirement (theoretical & losses)
4. Establish ΔT budget
5. Determine conductance requirements of components
 - cold plate
 - transport heat pipe
 - diode heat pipe
 - PCM container
6. Establish component designs to obtain required conductance
7. Establish system model and verify performance

TABLE 9-3.1 - Design Parameters

o Working Fluids

Property	Ammonia @ Temperature (°F)					Freon-21 @ Temperature (°F)				
	-40	0	40	90	125	-40	0	40	90	14
	Liquid Density, lb _m /ft ³	43.08	41.34	39.49	36.95	34.96	94.53	91.49	88.34	84.18
Liquid Viscosity, lb _m /hr-ft	.66	.617	.572	.511	.49	1.5	1.17	.945	.800	.63
Vapor Density, lb _m /ft ³	.040	.1098	.252	.601	1.03	.0312	.0966	.243	.6174	1.3
Vapor Viscosity, lb _m /hr-ft	.0162	.0206	.0225	.0299	.0261	.0227	.0242	.0257	.0282	.030
Latent Heat, BTU/lb _m	597.6	568.9	536.2	488.5	448.9	115.0	109.9	105.0	98.55	91.2
Surface Tension, lb _f /ft	.00245	.0022	.00172	.00126	.00099	.00194	.00165	.00144	.00117	.0009
Molecular Wt.	17.03					102.93				

o Inner Wall Film Coefficients * (BTU/Hr-Ft²-°F)

Heat Pipe	Evaporator	Condenser
Ammonia	2700	3500
Freon-21	1500	2000



* Grumman Test Data

o TG-15000 Insulation

Cp = .20 BTU/lb °F, ρ = 3.5 lb/ft³, k ≈ .003 BTU/Hr Ft °F (0-ATM)

The PCM must have a melting point between 110°F (the maximum on-orbit structural sink temperature) and 140°F (maximum permissible equipment base plate temperature) and it must also have a reasonably high heat of fusion. The three PCM's which best satisfied these two requirements were selected from the data provided in References 11 and 12 and are given in Table 9-3.2. Hexacosane is the preferred choice since it has the highest 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 candidate PCM's were non-toxic and showed good compatibility with aluminum, the container and heat pipe material.

o Theoretical PCM Weight (Hexacosane)

$$W = \frac{Q_c}{H_f} = 1.16 \text{ lbs}$$

o PCM Requirements Due to Heat Leaks

Appendix F contains the heat leak calculations consistent with the basic design assumptions. The losses are rounded off to 20% of the theoretical PCM requirement, or .24 lbs.

o Total PCM Weight

$$W_{\text{PCM}} = \text{Theoretical \& Losses} = 1.4 \text{ lbs.}$$

This actually includes a 33% safety factor since the PCM won't be needed during the de-orbit phase of entry (approximately 33% of the entry timeline of 1.07 hours).

Note: The maximum energy storage capacity of a PCM includes the specific heat capacity of the liquid phase and the filler material.

$$E_{\text{MAX}} = W_{\text{PCM}} H_f + \frac{1}{2} (T_{\text{PLATE}} - T_{\text{MELT}}) \left[W_F C_{\text{PF}} + W_{\text{PCM}} C_{\text{PLIQ}} \right]$$

These specific heat contributions are neglected in the design analyses.

TABLE 9-3.2

Phase Change Materials (PCM)

Name	Type	Formula	Melting Point (°F)	Heat of Fusion (BTU/lb)	Density lb/ft ³		Specific Heat BTU/lb-°F	
					Liquid	Solid	Liquid	Solid
*n-Hexacosane	Paraffin	C ₂₆ H ₅₄	133.0	110.0	48.0	53.4	.48	.53
n-Tricosane	Paraffin	C ₂₃ H ₄₈	117.5	100.0	47.7	53.0	.47	.52
Tristearin	Non-Paraffin Organic	(C ₁₇ H ₃₅ COO) ₃ C ₃ H ₅	133.0	82.1	53.8	-	-	-

*Selected Material

Thermal Conductivity(k) = .0865 BTU/Hr-Ft-°F

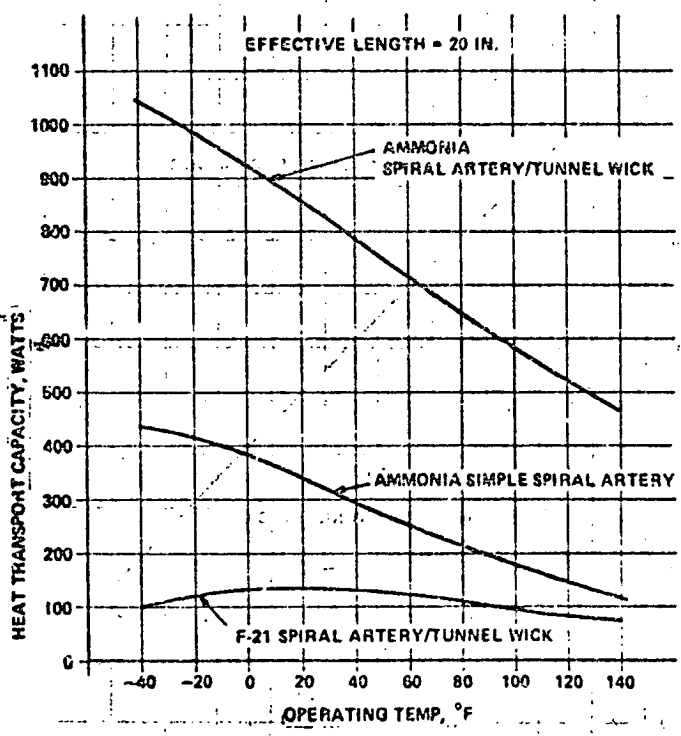


FIGURE 9-3.3. - TRANSPORT HEAT PIPE CAPACITY



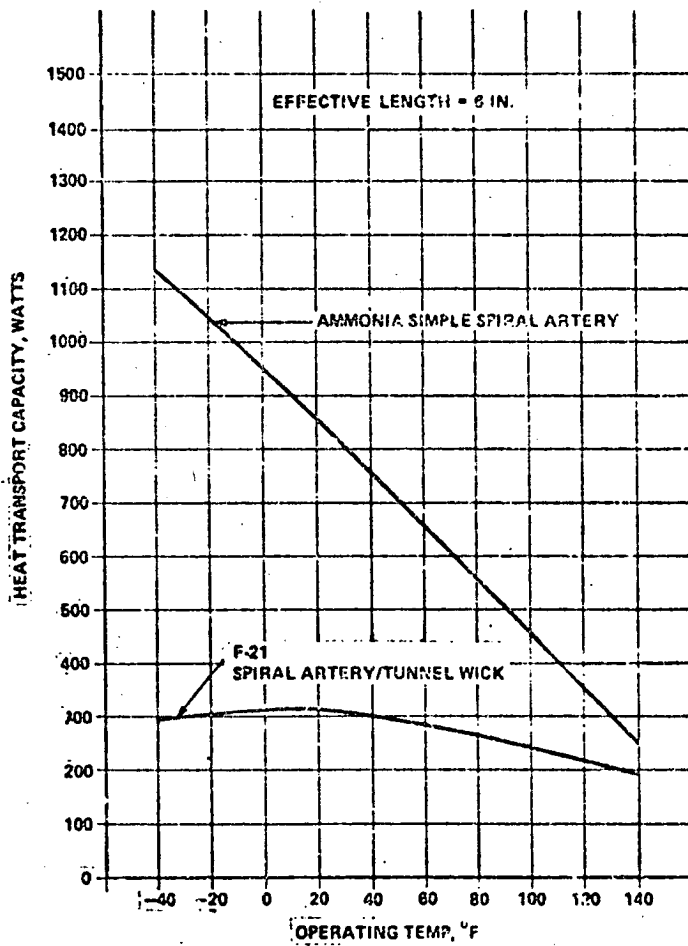


FIGURE 9-3.4 - DIODE HEAT PIPE CAPACITY

o Heat Pipe Analyses

Although several working fluids could be used for the temperature range involved (-40°F to +140°F), the selection was narrowed to ammonia and Freon-21. Using fine circumferentially grooved pipes with ammonia and Freon-21 as the working fluids, Grumman has successfully and repeatably demonstrated high evaporating and condensing film coefficients.

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.

One-half inch ID heat pipes were selected since they provide reasonable circumferential heat transfer area without excessive overcapacity. The designs of the arteries were optimized for the geometries involved and the results in the form of capacity versus operating temperature are presented in Figures 9-3.3 and 9-3.4 for the transport heat pipe and diode heat pipe, respectively. Reference 13 was used for the analysis. Based on these results ammonia is selected as the working fluid with a simple spiral artery wick for the diode heat pipe and a spiral artery/tunnel wick for the transport heat pipe.

The diode heat pipe operates on the liquid blockage principle; 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 the 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 .06 inch diameter for a 2.12 inch length.

$$d = \frac{4 \sigma \cos \theta}{\rho L}$$

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 net result is the same no matter what the cause for the reversal.

o Cold Plate Conductance

Figure 9-3.5 describes the nodal model used to determine the conductance of the heat pipe cold plate. For 6101-T6 aluminum and a .07 inch plate the effective cold plate conductance between the interface plane (excluding any contact conductance) and the heat pipe vapor is 74 Btu/Hr °F with Freon-21 as the working fluid and 130 Btu/Hr °F with ammonia. The evaporator film conductance (K_3) is the controlling influence in both cases.

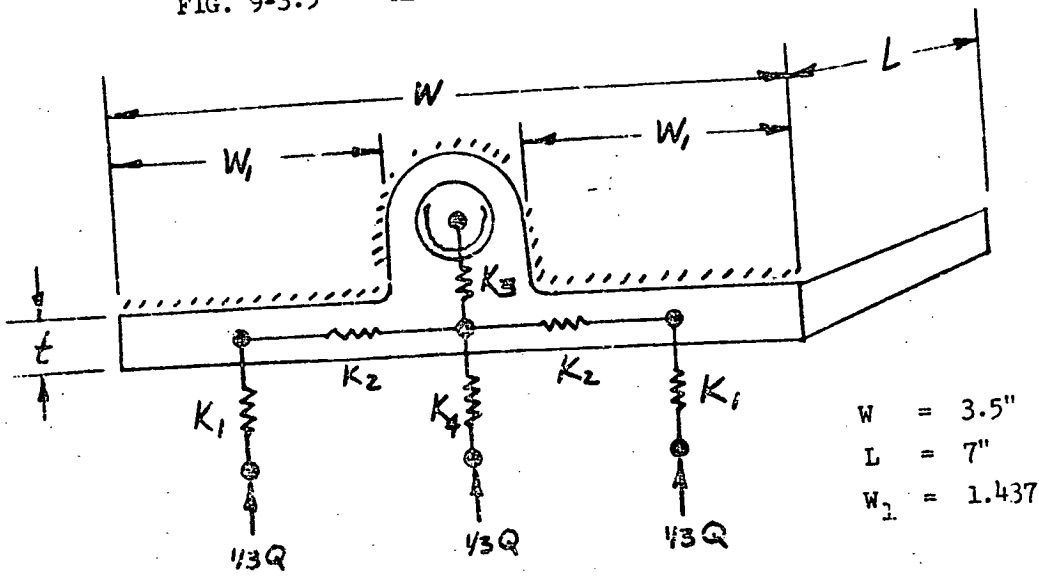
o PCM Container

This is the most critical and sensitive component of the entire system. It is not enough to provide the proper amount of PCM for the desired heat sink capacity - that's simple and straightforward. The biggest obstacle to be overcome is insuring uniform heat transfer within the PCM. The phase change materials generally have very low thermal conductivities which result in very steep temperature gradients required to transfer heat to the PCM. This gradient can result in an excessively high equipment temperature during the melting process. The integration of metallic filler materials into the PCM package will improve the thermal conductivity through the PCM. They provide low thermal resistance paths thus reducing the temperature gradient necessary to dissipate the required heat flux.

The actual design of the package becomes 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.

The two most effective filler materials reported in the literature (Refs. 11 and 12) are honeycomb matrices and fins. The honeycomb is commercially available and the fins can be easily fabricated in-house. In both cases, however, a good metal to metal contact between filler and the heat input surface is paramount. This implies simple geometries that are easy to assemble by brazing or soldering to provide good thermal contact.

FIG. 9-3.5 - HEAT PIPE COLD PLATE MODEL



$W = 3.5''$
 $L = 7''$
 $W_1 = 1.437$

$$K_1 = \frac{k W_1 L}{\frac{t}{2}} \quad K_2 = \frac{k t L}{\frac{W_1 + W_2}{2}} \quad K_3 = h_{EVAP} \eta_i \pi D_i L$$

$$K_4 = \frac{k W_2 L}{\frac{t}{2}} \quad \eta_i = \frac{1}{2} (1 + \eta_F) = .67$$

$$K_{\text{plate-Vapor}} = \frac{K_3 \left[\frac{2 K_1 K_2}{K_1 + K_2} \right] + K_4}{K_3 + \frac{2 K_1 K_2}{K_1 + K_2} + K_4}$$



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

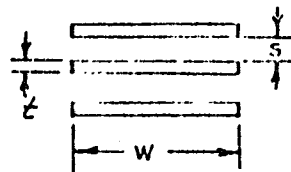
1. Cylindrical Container with circular (circumferential) fins
2. Cylindrical Container with straight (longitudinal) fins
3. Rectangular Container with straight fins mounted directly on the heat pipe cold plate.
4. Rectangular Container with honeycomb mounted directly on the heat pipe cold plate.

The cylindrical container is separated from the cold plate but connected to it by means of the transport heat pipe. Its outside diameter was fixed at 3.12 inches. The rectangular container receives its heat input directly from the cold plate and therefore does not require a transport heat pipe. Only one heat pipe, the diode, is necessary for this latter configuration.

Aluminum is the choice for the filler material since it best satisfies the basic requirements of high thermal conductivity, low weight, good corrosion resistance and low cost.

The thickness for the fins was determined by using the expression developed for optimum thickness rectangular fins.

$$\frac{t}{2} = \left[\frac{w^2 t^2 h}{4 \lambda^2 k_{FIN}} \right]^{1/3}$$



where $\lambda = 1.419$ and h is approximately $\frac{k_{PCM}}{(s/2)}$

For $w = 1.25$ inches

$$t^2 - \frac{t}{f} - 2.14 (10^{-3}) = 0$$

where f is the number of fins per inch. Preliminary estimates indicated $f \approx 10$ which gives a $t_{OPT} = .018"$. The nearest gauge and the one selected for all fins was $.016"$.

Appendix F-2 summarizes the expressions for effective conductance and weight which were derived for each of the four candidate PCM containers. Figures 9-3.6 to 9-3.8 present the results for the cylindrical container with circular fins. Figure 9-3.7 shows that the required PCM weight of 1.4 lbs. can be accommodated in a 10 inch length with up to 19 fins ($.016"$) per inch. Figure 9-3.6 shows that the required PCM conductance, for an ammonia heat pipe heat source in a 10 inch length is $2.5 \text{ Btu/Hr-in-}^{\circ}\text{F}$. Selecting 10 fins per inch for the design gives a value of $9 \text{ Btu/Hr-in-}^{\circ}\text{F}$ or slightly more than three times what is required. At 10 fins per inch for 10 inches, the maximum possible PCM weight would be 1.7 lbs. with a corresponding total weight (PCM + fins) of 2.9 lbs. Using only the required 1.4 lbs. of PCM leaves a void volume of 17.6% to accommodate PCM expansion and entrapment of gases. The total weight of PCM and fins would then be 2.6 lbs.

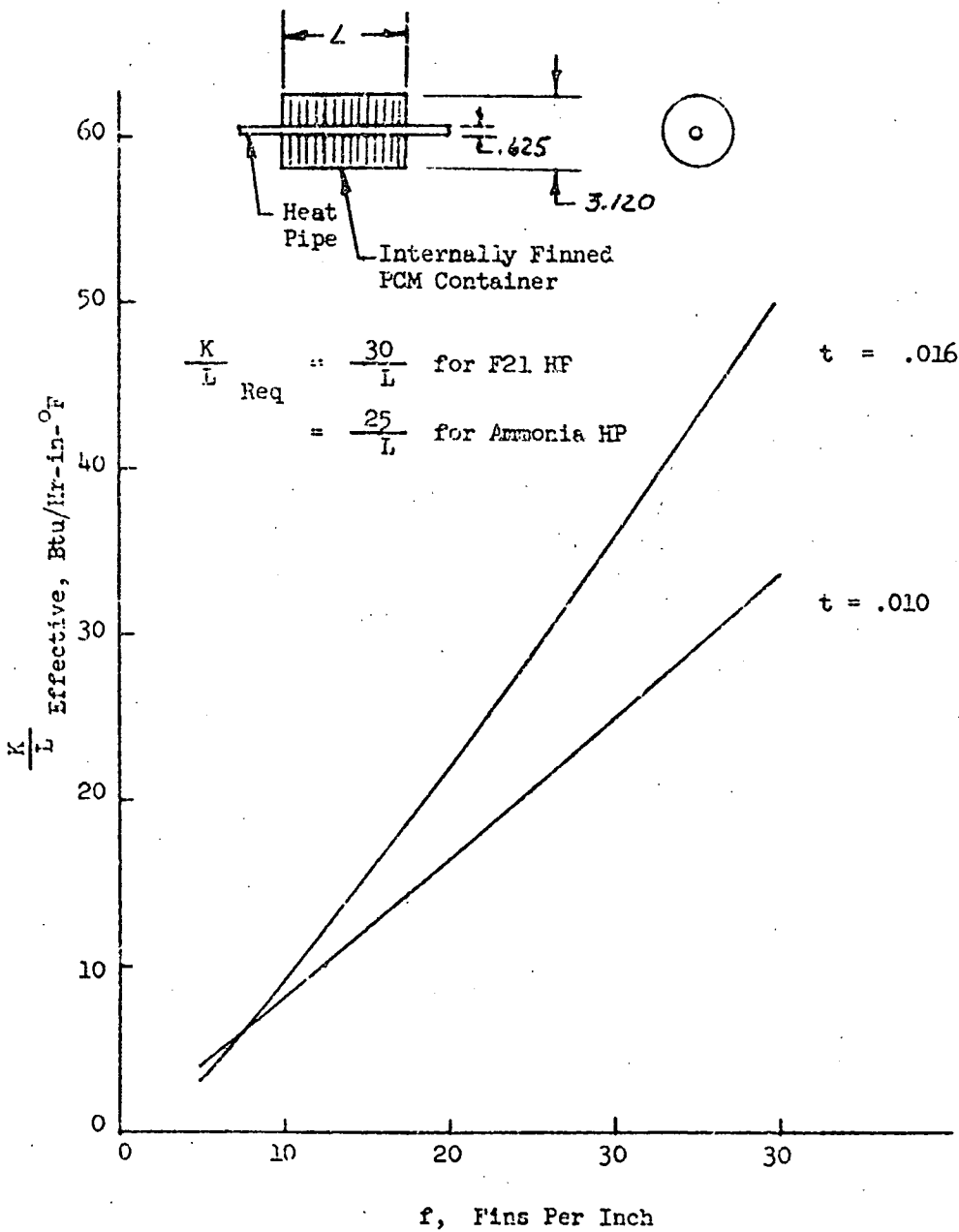
The performance of the straight finned container is extremely poor and is indicated in Appendix F-2. The configuration cannot provide the minimum required PCM conductance of $2.5 \text{ Btu/Hr-in-}^{\circ}\text{F}$; the best it can do is around .55.

With direct PCM attachment to the cold plate the required overall PCM conductance is reduced from $25 \text{ Btu/Hr-}^{\circ}\text{F}$ needed for the cylindrical container to $20 \text{ Btu/Hr-}^{\circ}\text{F}$. This is due to the more direct heat transfer path from source to sink. As indicated on Figure 9-3.10, for the straight finned configuration a design point of 10 fins per inch and a height of 2.5 inches would provide $23 \text{ Btu/Hr-}^{\circ}\text{F}$ or slightly more than the theoretical requirement. Figure 9-3.9 shows that the configuration would barely accommodate the required 1.4 lbs. of PCM. The resulting total weight is 2.35 lbs., a little lighter than the cylindrical container.

The results of the honeycomb flat plate PCM container (Figure 9-3.11) show that the conductance and PCM weight requirements cannot be reconciled. The height needed to accommodate the required PCM increases the resistance

FIG. 9-3.6 - EFFECTIVE CONDUCTANCE

- o HP Wall to PCM
- o Circular (Circumferential) Fins
- o t = Fin Thickness, in.



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

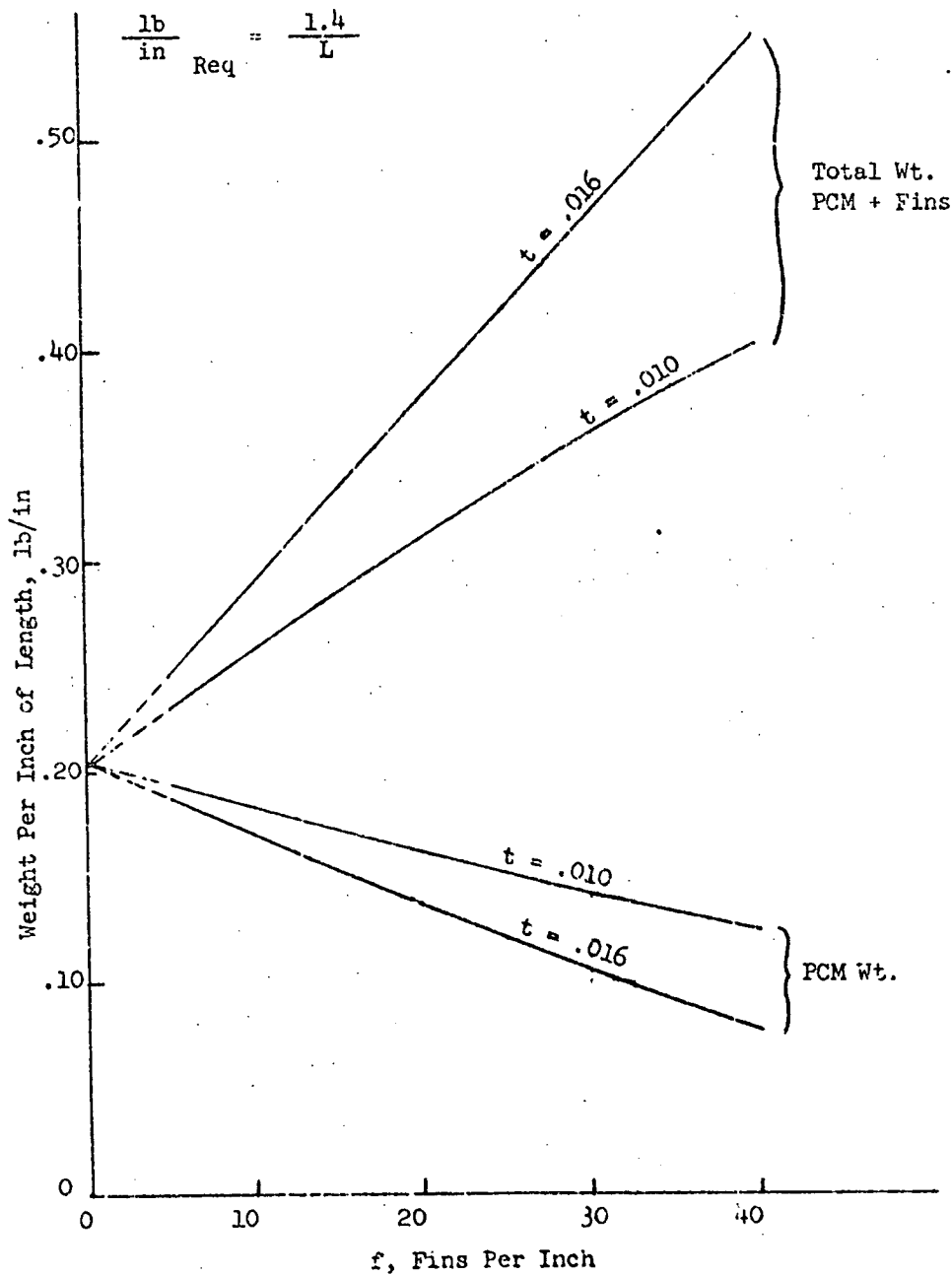


FIG. 9-3.8 - CONDUCTANCE PER UNIT WEIGHT
 o Circular (Circumferential) Fins
 o t = Fin Thickness, in.

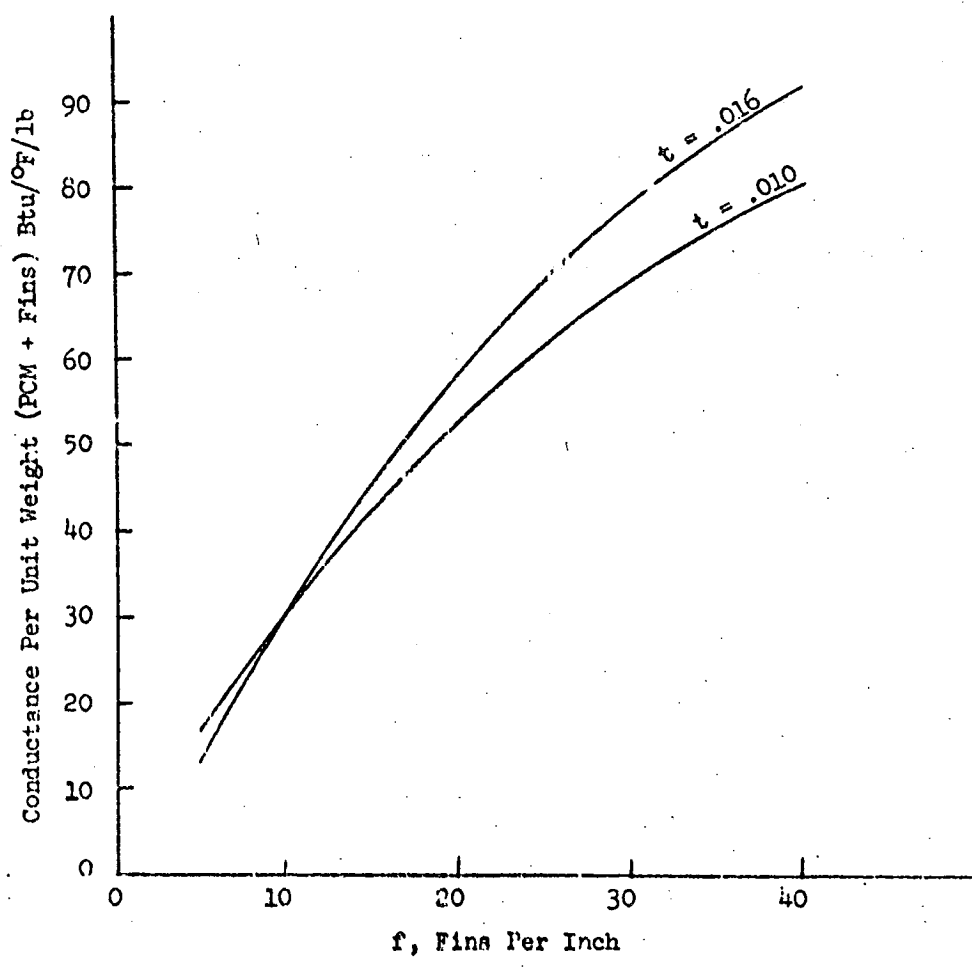


FIG. 9-3.9 - WEIGHTS - DIRECT PCM ATTACHMENT TO PLATE

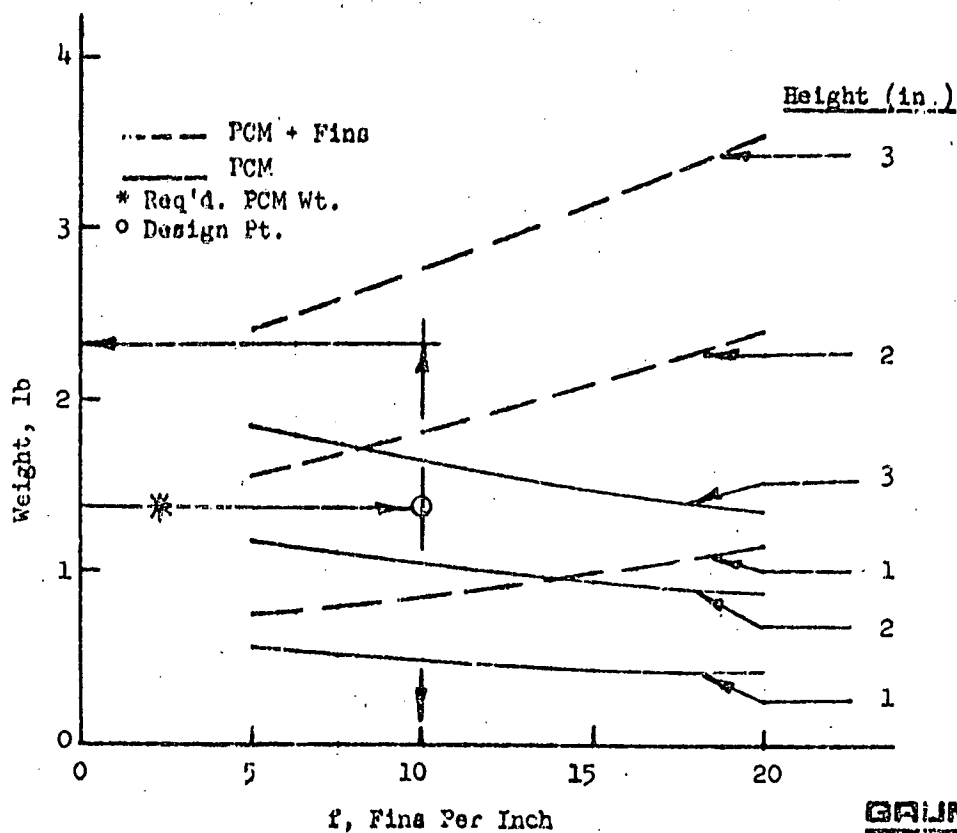
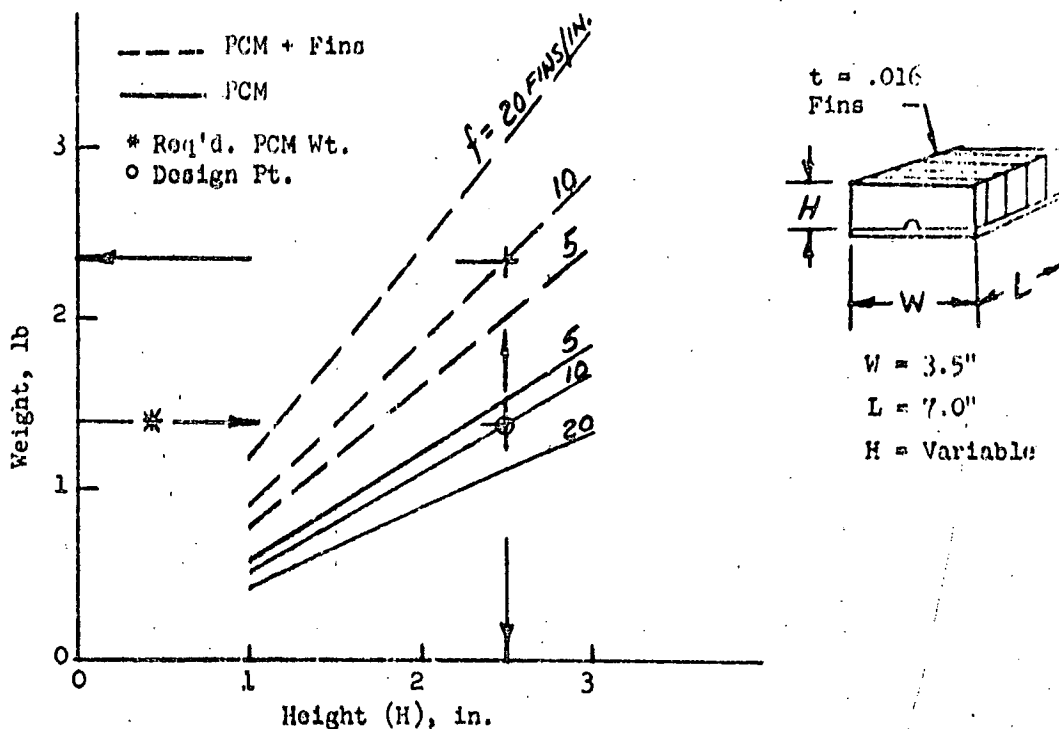


FIG. 9-3.10 - TOTAL CONDUCTANCE

Direct PCM Attachment to Cold Plate

◦ Design Pt.

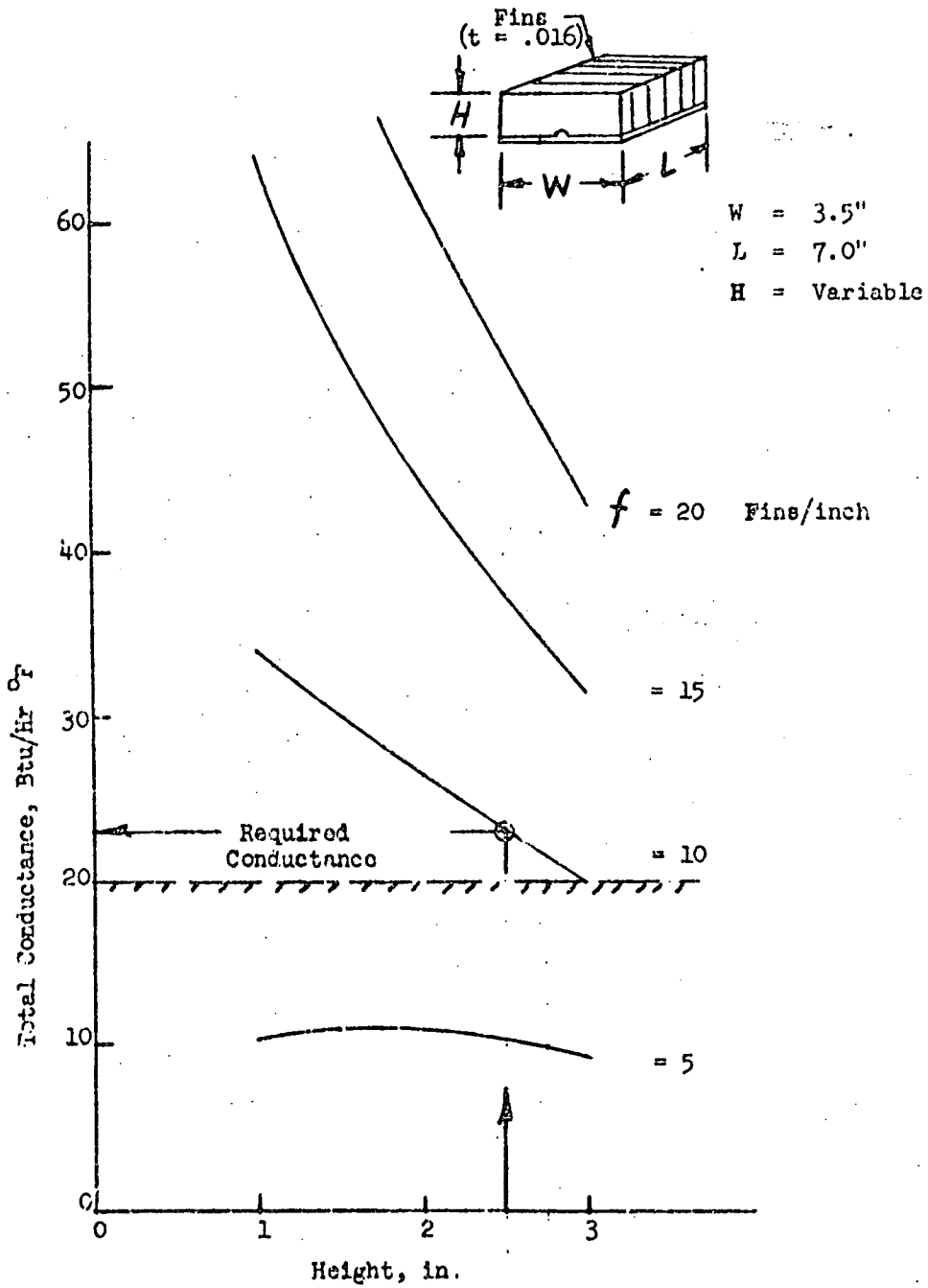
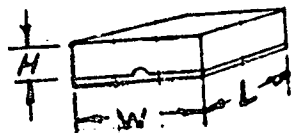
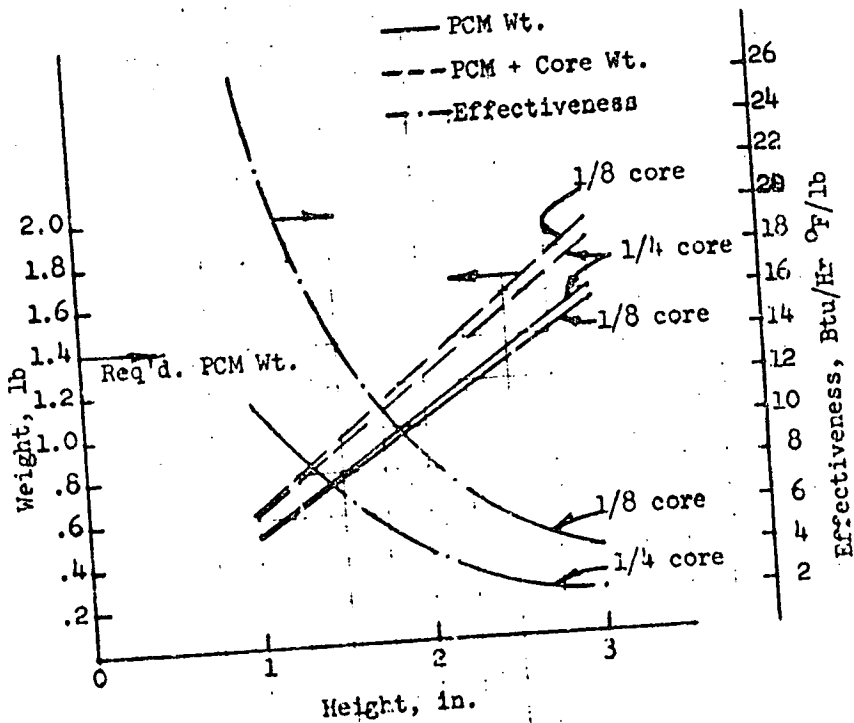
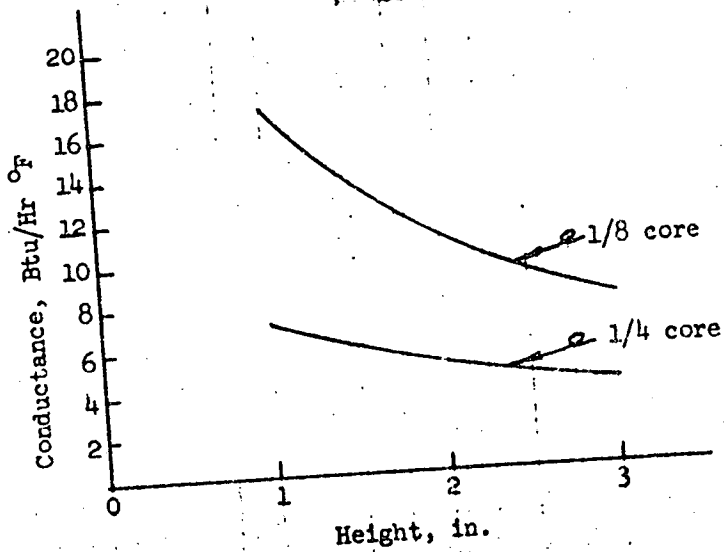


FIG. 9-3.11 - HONEYCOMB CORE/PCM PERFORMANCE



W = 3.5"
 L = 7.0"
 H = Variable



to heat flow thereby reducing the effective conductance below the required value.

On a conductance per pound (PCM + fins) basis, the circularly finned cylindrical container is the overwhelming winner. For this reason and for its inherently greater design flexibility it is the selected PCM container configuration.

o System Performance

Figure 9-3.12 presents the electronics baseplate temperature as a function of the structural sink temperature. The base plate varies from -20°F to $+130^{\circ}\text{F}$ (within the design requirement) for a sink variation of -40°F to $+110^{\circ}\text{F}$.

The entire modular heat sink system, as shown on drawing SPL-111, weighs 4.45 pounds. The component weights are detailed in Table 9-3.3.

9-3.3 DESIGN

The detailed design, including necessary design specifications, for the modular heat sink is shown in drawing SPL-111. The electronics box is attached to the structure by four No. 10 steel bolts using .300 inch thick glass cloth epoxy resin insulating washers between the box and structure. The steel bolts and insulating washers provide high thermal resistance paths which minimize the conductive heat transfer to structure. In addition, a $\frac{1}{4}$ inch thick fibrous insulation blanket (TG-15000 or Micro quartz) completely encases the entire system, including the electronics, cold plate, heat pipes and PCM container. A single wrap of oxidized nickel foil ($\frac{1}{2}$ mil) is wrapped around the insulated system to create a thermal radiation barrier.

The mechanically bonded interfaces which require good heat transfer (heat pipe cold plate, transport heat pipe condenser, diode heat pipe evaporator and condenser) are finished to 64 micro-inch RMS. Before mating, silicon grease (Dow Corning 340) is applied to and then wiped from the contact surfaces. This leaves a thin film of grease which increases the inter-

FIG. 9-3.12 - MODULAR HEAT SINK PERFORMANCE

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

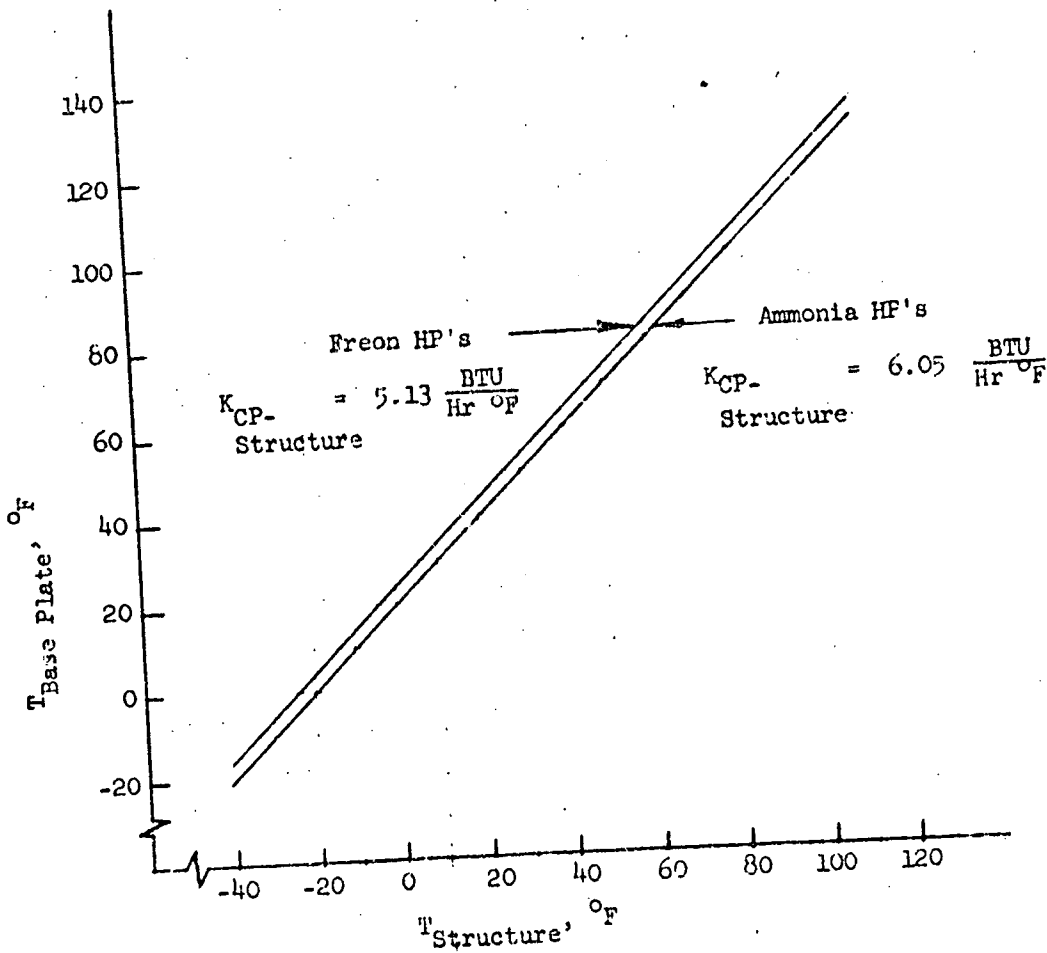
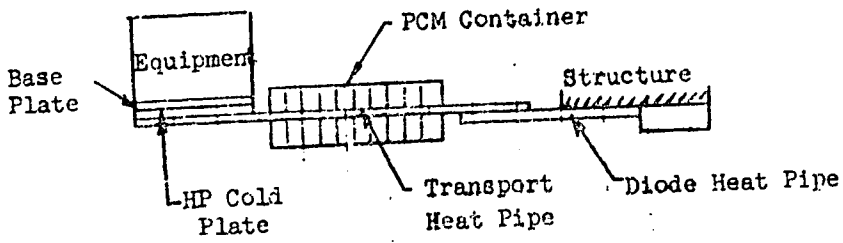


TABLE 2-3.3 - MODULAR HEAT SINK FLIGHT VOICE RECORDER APPLICATION (SPL-111)

ITEM	QTY	UNIT WEIGHT LB.	TOTAL WEIGHT LB.	DATA/COMMENTS
H.P. Weld Assy			(.50)	
Charge Tube			.01	Al
Evaporator (flanged)		.157 lb/ft*	.28	*Flange additional 7.7" long Al
Condenser		.157 lb/ft	.20	*Flange additional 13.6" long Al
End Cap			.01	Al
Dodge H.P. Weld Assy			(.37)	Al
Charge Tube			.01	
Evaporator		.157 lb/ft	.06	4.5 in
Elbow			.03	
Condenser		.157 lb/ft	.10	4.1 in
Reservoir			.17	
Cannister Weld Assy			(1.19)	Al
Tubing		.362 lb/ft	.31	.032 thick Al. 3 1/8" Dia.
End Baffles	2	.144	.29	.063 Al
Stiffeners	2	.075	.15	.032 Al
Fins			.44	.016 Al
9-6			2.10	48 lb/ft ³ 10" long 3 1/8" dia
9-6			.10	.25" t
FX - Hexacosane			.10	
Insulations			.07	
Clamp			.07	.06 lb/cu in
Insulator	8	.001	.01	Al
Radius Blocks	4	.010	.04	
Plate Nuts				
Total Weight			4.45	
		lb.		



face contact conductance. The procedure has been approved for the Lunar Module as reference in Grumman Specification LSM-14-6006 dated July 19, 1966.

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 (Aluminum 6101-T6) which contains the evaporator section of the transport heat pipe as an integral part. The heat pipe section is $\frac{1}{2}$ inch I.D. (.045 wall) with 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 .070 inches thick with a contact surface area of 24.5 in². It is joined to the base plate by 10 - 32 steel fasteners.

The transport heat pipe connects the cold plate flange to the PCM and to the diode heat pipe. At the PCM interface, near the midpoint of the heat pipe, circular aluminum fins (6101-T6) are brazed to the .590 outside diameter of the pipe only (10 fins per inch for a length of ten inches). The fins are .016 inches thick; each contains several notches and holes which act as flow passages to permit uniform distribution of PCM during filling. The condenser section of the heat pipe mates with the evaporator of the diode heat pipe at a milled-flat interface. They are mechanically joined by a clamp assembly with four 10 - 32 fasteners. The transport heat pipe is charged with ammonia after the PCM container is attached and filled.

- Design Details

Pipe:	Aluminum 6101-T6
Grooves:	Circumferential
Diameter:	.500 in I.D., .590 in O.D.
Working Fluid:	Ammonia (Charge = 11.93 gms)
Wick:	Spiral Artery/Tunnel Wick (.062" core)
Evaporator Length:	7 inches
Condenser Length:	4 inches
Overall Length:	22.73 inches

The diode heat pipe forms the on/off coupling between the transport heat pipe and the bulkhead heat sink. It operates on the liquid blockage principle wherein normal heat pipe operation is stopped when the evaporator vapor space is flooded with excess working fluid. During normal operation the excess fluid is held in a fluid reservoir which is welded to the diode's condenser. The reservoir consists of an array of fine capillary holes drilled into an aluminum block. The wick in the condenser does not communicate with the excess fluid in the reservoir. During entry, when the inertia force exceeds 1.25 'g', the capillaries in the reservoir can no longer retain the excess fluid and it is driven into the diode's evaporator. Both the condenser and the reservoir are flanged and attached to the bulkhead by 10 - 32 steel fasteners. An 'L' shaped charge tube is at the end of the evaporator.

- Design Details

Diode

Pipe:	Aluminum 6101-T6
Grooves:	Circumferential, .500 in I.D., .590 in O.D.
Working Fluid:	Ammonia (Charge = 13.97 gms)
Wick:	Simple Spiral Artery
Evaporator Length:	4 inches
Condenser Length:	4.1 inches
Overall length:	12.9 inches (including reservoir)

Reservoir

Material:	Aluminum 6101-T6
Overall Length:	3 inches
Capillaries:	.060" Dia. x 2.12 inches long
- number:	125
- volume:	.384 in ³ /in

The PCM container houses the phase change material (hexacosane, C₂₆H₅₄) and the conductive filler material (circular aluminum fins) and completely surrounds a centrally positioned ten inch segment of the transport heat pipe. It is a completely welded aluminum assembly whose major components are a 10 inch long, 3.125 I.D. tube (.023" wall) with a fill port, end baffles or discs which seal off the ends of the tube and support the central heat pipe, and the circular aluminum fins which are brazed to the heat pipe surface. The end baffles are welded to the tube and the heat pipe.



Prior to filling, the cannister and PCM are stabilized at a temperature of 230°F, then the container is evacuated and vacuum filled with liquid PCM.

9-3.4 BASELINE

The shuttle baseline thermal control system for remote electronics calls for cold plates and cold rails using Freon-21 as the coolant. Branch flow rates of 300 lbs/hr and tube insulation thicknesses of .40 inches are required to maintain temperature drops below 20°F (Reference 14). A cold plate for the recorder electronics requires a (UA) = 2.22 Btu/Hr-°F. An unfinned hollow plate ($\frac{1}{4}$ inch opening) provides only 1.31 Btu/Hr-°F. Thus, internal finning is required. Choosing surface designation 11.1 from reference 15, provides a (UA) of 18.9 Btu/Hr-°F. The resulting pressure drop, considering an overall line run of 40 feet, is .402 inches of water, which results in a rather minor power penalty of 1.28 (10^{-3}) watts. The total fluid line weight for the 3/8" ID, .020 wall tubing is .133 lb/ft; the cold plate weighs .437 lbs. The total system weight is 5.76 lb without the insulation and 6.74 lb with it.

9-3.5 TEST PLAN

This section describes the thermal testing which will provide a performance map of the modular heat sink shown in Figure 9-3.13.

It calls for bench testing to establish the operating characteristics of the component heat pipes, followed by parametric system tests to determine system performance under different heat loads and sink temperatures and during a simulated 3-g entry condition.

- Configuration

In order to facilitate 1-g and 3-g testing, the actual system tested will use a straight diode heat pipe rather than an L-shaped one. The thermal load on the transport heat pipe evaporator, and diode condenser and reservoir (when needed to simulate entry heating of structure) will be applied by using heater ribbon and Variac type control devices. When required, the diode condenser and reservoir will be cooled by a spray bath.

SPRAY HEATING OPERATION

⊗ THERMOCOUPLE LOCATIONS

TEMPERATURE CONTROLLABLE METHANOL SPRAY BATH (INACTIVE DURING HEATER OPERATION)

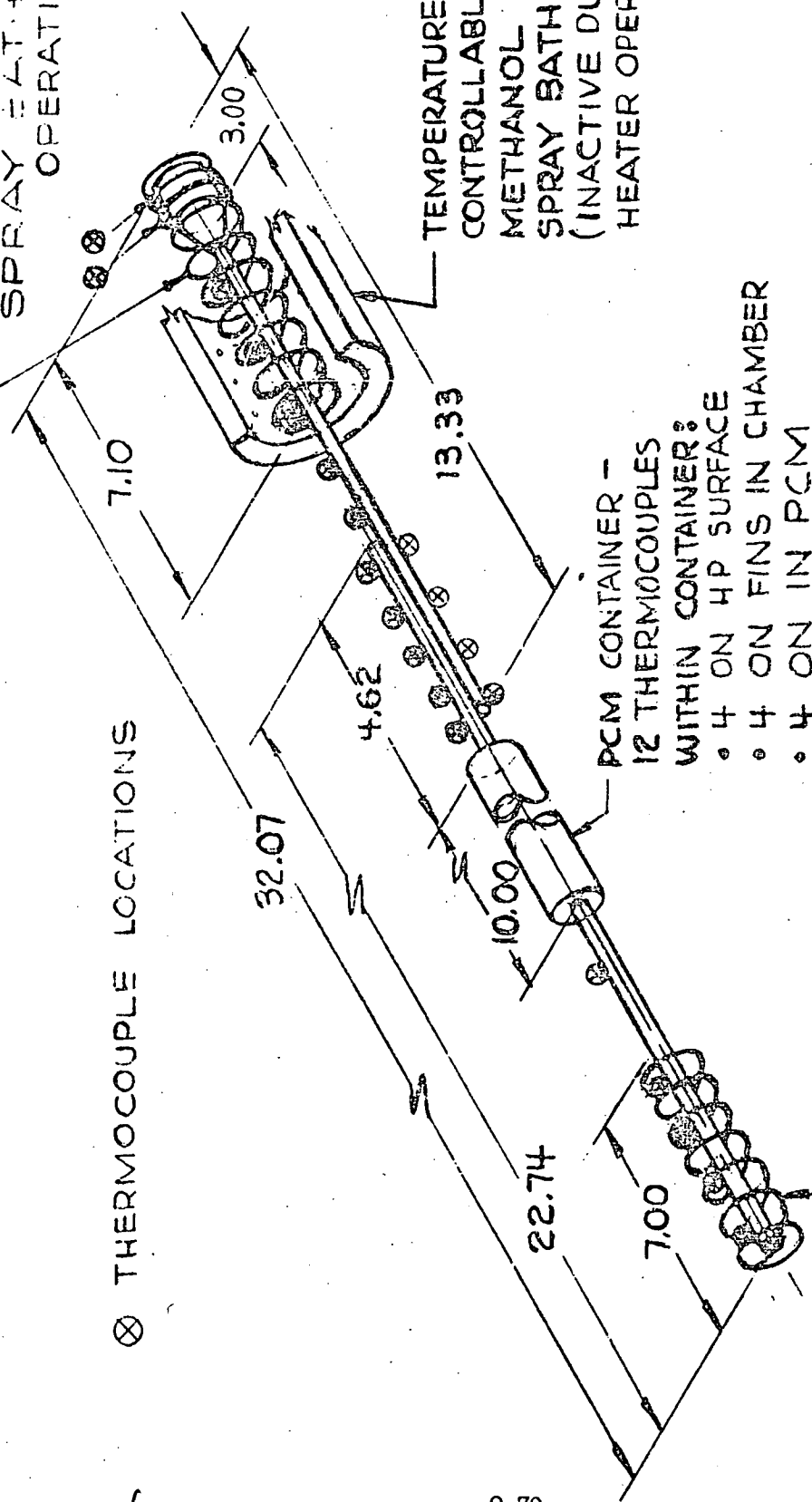
PCM CONTAINER - 12 THERMOCOUPLES WITHIN CONTAINER:
• 4 ON HP SURFACE
• 4 ON FINS IN CHAMBER
• 4 ON IN PCM

HEATING ELEMENTS

FIG 9-3.13 - TEST CONFIG.



C f



As indicated in Figure 9-3.13, thermocouples will be placed along the outside of each heat pipe, and at various locations in the phase change material cannister, both in the material itself and on the internal fins. By this means, the thermal performance of the system tested can be closely monitored.

The facility requirements are summarized below:

Power:	Two variac channels, 100 watts each
Instrumentation:	Readout and recording provisions (strip charts) for 40 Cu-Cn thermocouples
Coolant:	Methanol spray bath, with inlet temperature controllable between -50°F and 120°F
Centrifuge:	Up to 3-g, with provisions for the thermocouple and variac channels.

- Test Procedure

Table 9-3.4 presents a summary of the test program. Tests #1 and #2, which may be conducted concurrently, provide performance data for the system components. Test #1 is the bench test of the diode, and will provide capacity data in the forward mode, leakage in the reverse mode, and information about diode shutoff/startup characteristics. Test #2 is the bench test of the transport heat pipe/PCM container assembly; thermal capacity testing of the heat pipe will be performed first, and then the PCM will be evaluated.

After the two assemblies are joined to form the modular sink system, subsequent tests (#3 through #20) will be conducted to determine system performance under realistic mission conditions. Tests #3 through #11 will indicate the temperatures the transport heat pipe evaporator will attain under several different sink conditions. These tests, as well as tests #12 through #14, will be conducted with a slight adverse tilt (i.e., condenser below evaporator) to eliminate puddling contributions. During the latter three tests, the diode sink temperature will be varied to duplicate a typical entry heating and cooling pattern. Tests #15 through #17 duplicate the loads of a typical equipment dissipation cycle with fixed diode sink temperatures, and test #18 combines this typical dissipation cycle heat load with an entry heating and cooling pattern imposed on the diode.

TABLE 9-3.4 - TEST PLAN

<u>Test #</u>	<u>Q_{Equipment}</u>	<u>T_{Sink}</u>	<u>Comments</u>
1	-	-	Bench test of diode
2	-	-	Bench test of transport HP/PCM assembly
3	5 watts	-40°F	
4	15 watts	-40°F	
5	40 watts	-40°F	
6	5 watts	30°F	Steady State
7	15 watts	30°F	Performance
8	40 watts	30°F	
9	5 watts	110°F	
10	15 watts	110°F	
11	40 watts	110°F	
12	5 watts		entry heating & cooling pattern
13	15 watts		entry heating & cooling pattern
14	40 watts		entry heating & cooling pattern
15	typical equipment dissipation cycle	-40°F	
16	typical equipment dissipation cycle	30°F	
17	typical equipment dissipation cycle	110°F	
18	typical equipment dissipation cycle - entry heating and cooling pattern		
19	typical equipment dissipation cycle - entry heating and cooling pattern		
20	typical equipment dissipation cycle - entry heating and cooling pattern		

Test #19 duplicates the loads and temperatures of test #18 with the system mounted vertically so that the heat pipes are operating in the reflux boiler mode (i.e., heat source down and heat sink up). This test simulates the g-vector orientation of the system during entry and post-landing ground standby.

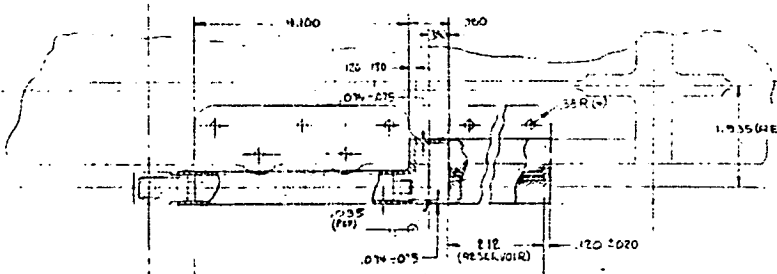
For Test #20, the system will be mounted on a centrifuge so that the effective g-vector will have the same orientation as in the reflux boiler test, and, at 3-g, the load of test #18 will be imposed to establish the system's ability to function under entry conditions.

9-3.6 CONCLUSIONS

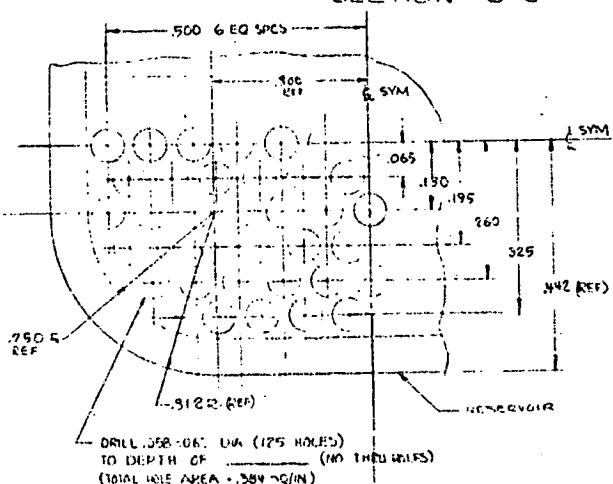
For this application, the modular sink weighs 4.45 pounds vs. the baseline weight of 6.74 pounds. Of greater, more universal importance, the modular sink provides a self-contained thermal control system for remote components and thereby eliminates multiple, complicated flow paths with their inherent balancing and leak problems. Additional applications to the shuttle include: rate gyros and gyrobatteries located between the LH and LOX tanks of the H-O tank assembly, and actuator electronics associated with engine gimbaling and landing gear operation.

The modular heat sink thermal control concept has broad applicability to remotely located heat sources, and can be manufactured with current technology and methods.

FOLDOUT FRAME



SECTION C-C



DETAIL D
SCALE 10:1

FLIGHT RECORDER ELECTRONICS
(WT 7 LBS)

INTERNAL GROOVES
100 PBR INCH X .010 DEEP

RETAINER WEBS
& RETAIN D. 50P DITCH

30CK DIA

MAKE RETAINER
AND ARTERY
FROM 100/100 X .005 WIRE DIA
MESH SCREENING
504 S STL
RE-W-160

TITANIUM (SAINV)
ATTACHMENTS (4)

30CK DIA
324 S STL

HEAT PIPE CROSS SECT
(SCALE 10:1)

TABLE I
(SEE NOTE 2)

LOADING AND LIFT-OFF	PCM	PHASE CHANGE MATERIAL (PCM) UNLOADS Q TO HIGH DENSITY STRUCTURE AS IT COOLS
RE-ENTRY	DIODE PCM	END OF THE HEAT PIPE STRUCTURE FROM EXCESSIVE Q LOAD FROM RE-ENTRY (ARRANGEMENT)
ON ORBIT		NORMAL Q LOAD TO HIGH DENSITY STRUCTURE
PRE-ORBIT		NORMAL Q LOAD TO HIGH DENSITY STRUCTURE
LIFTOFF	STRUCTURE	NORMAL Q LOAD TO HIGH DENSITY STRUCTURE
DUTY CYCLE	TEMPERATURE ORIENTATION	HEAT SINK DESIGNATED

TABLE II - HEAT PIPE CONFIG

ITEMS AND DATA	WEIGHT
WORKING FLUID ANTIMONY (100)	.01867 FT
SPIRAL ARTERY SOLID CORE DIA = .012 NO OF WRAPS = 14 GAP SIZE OR ROD DIA = .010 NO OF RODS = 13	.0472 FT
RETAINER 30CK DIA (O.D.) = .231 NO OF WEBS = 6	.0234 FT
TUBE AL FLY 6101 1651 QQ-B-525A	.0114 FT
TOTAL	.1667 FT

INSULATOR
MATERIAL: NYLON 66 (G181600)
FILL: BUBBLE 100 (G88)
(CLASS 1001 FIRST OF 10)
(TYP FOR ALL INSULATORS)

1/4" MINIMUM BLANKET
INTERNAL FIRMS MATL
WITH EXTERNAL COVER OF
OXIDIZED NICKEL POW (0099)
(TYP FOR ENTIRE SYSTEM
EXCEPT AS SHOWN)

DIODE HEAT PIPE WELD ASY
D1 CHANGE TUBE (20)
D2 EVAPORATOR
D3 ELBOW
D4 CONDENSER (ANNEA)
D5 HEATER (FLANGED)
MACHINE ABOVE ITEMS FROM
AL ALY 6101 WITH COVER 515A
(USE IN AS WELDED COND)

OPTIONAL D
DEPTH AS SHOWN
2.50
1.119

100 (REF)

550 (REF)

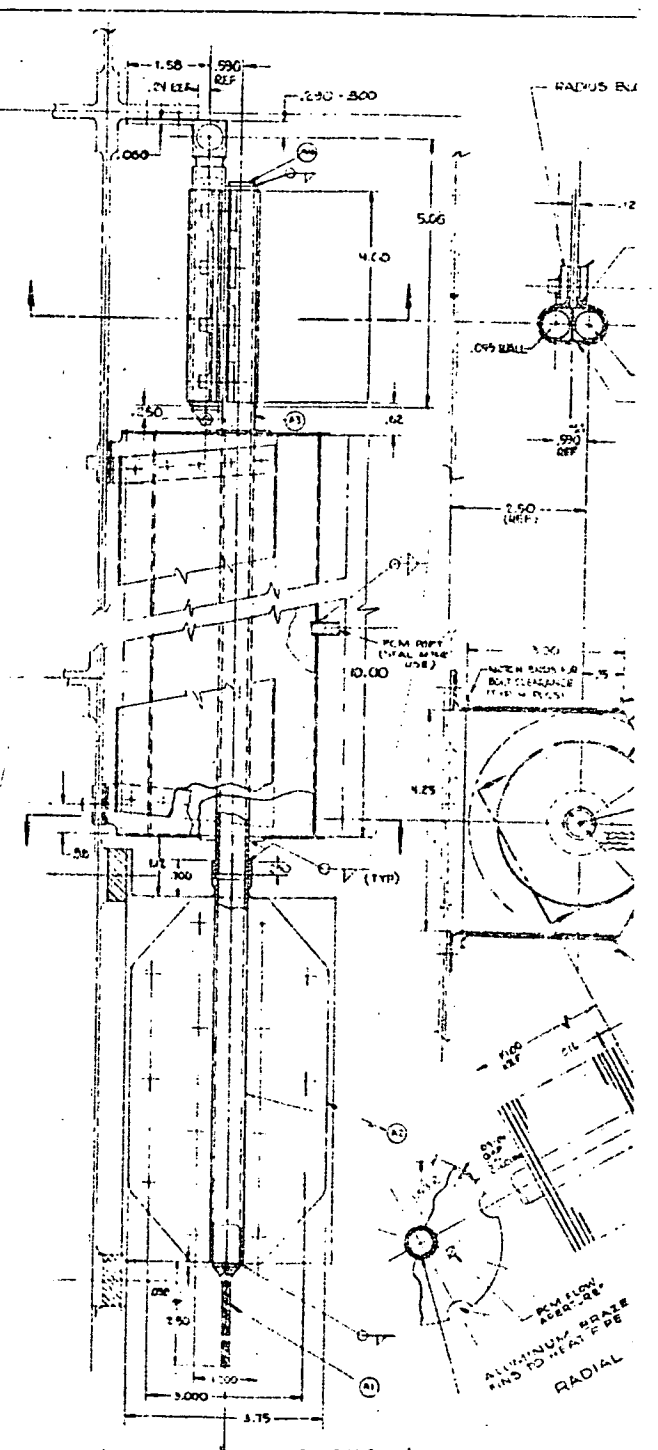
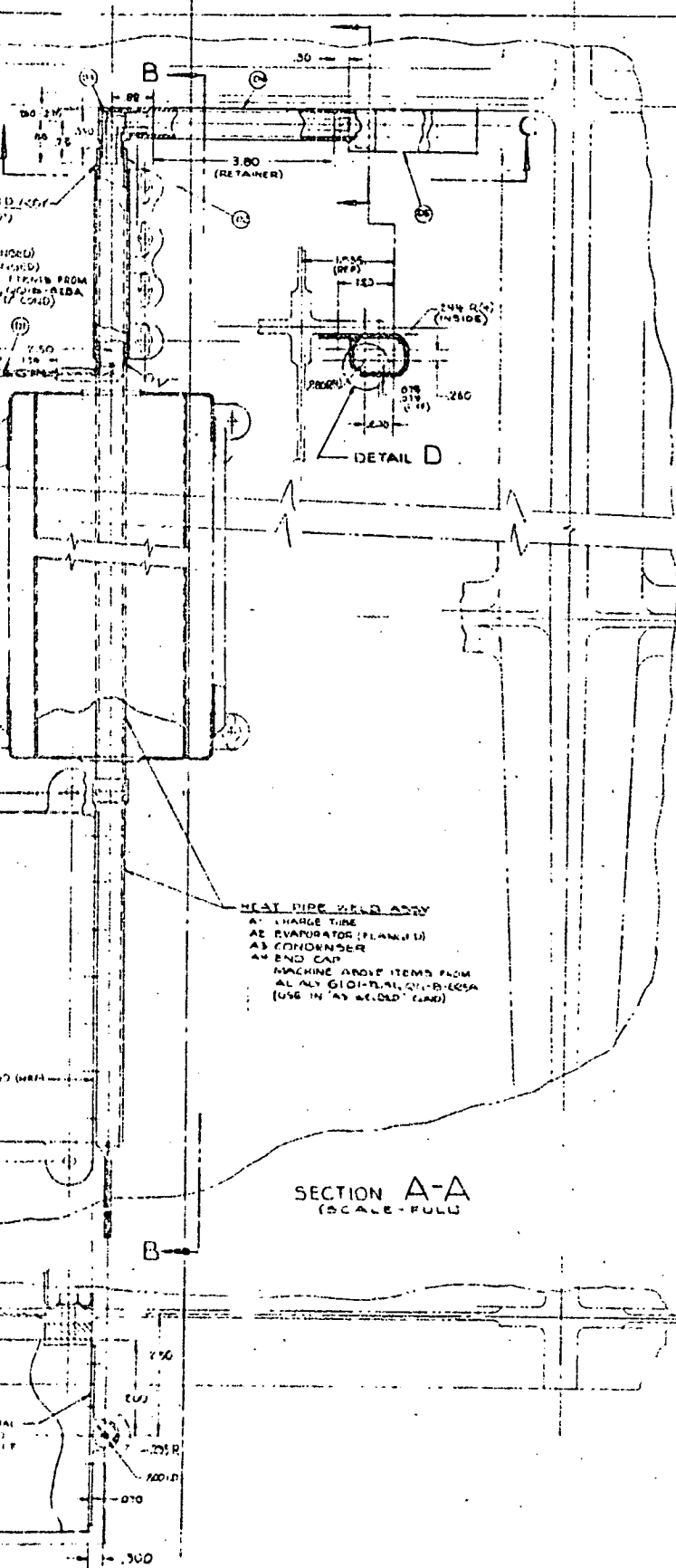
50 (REF)

350

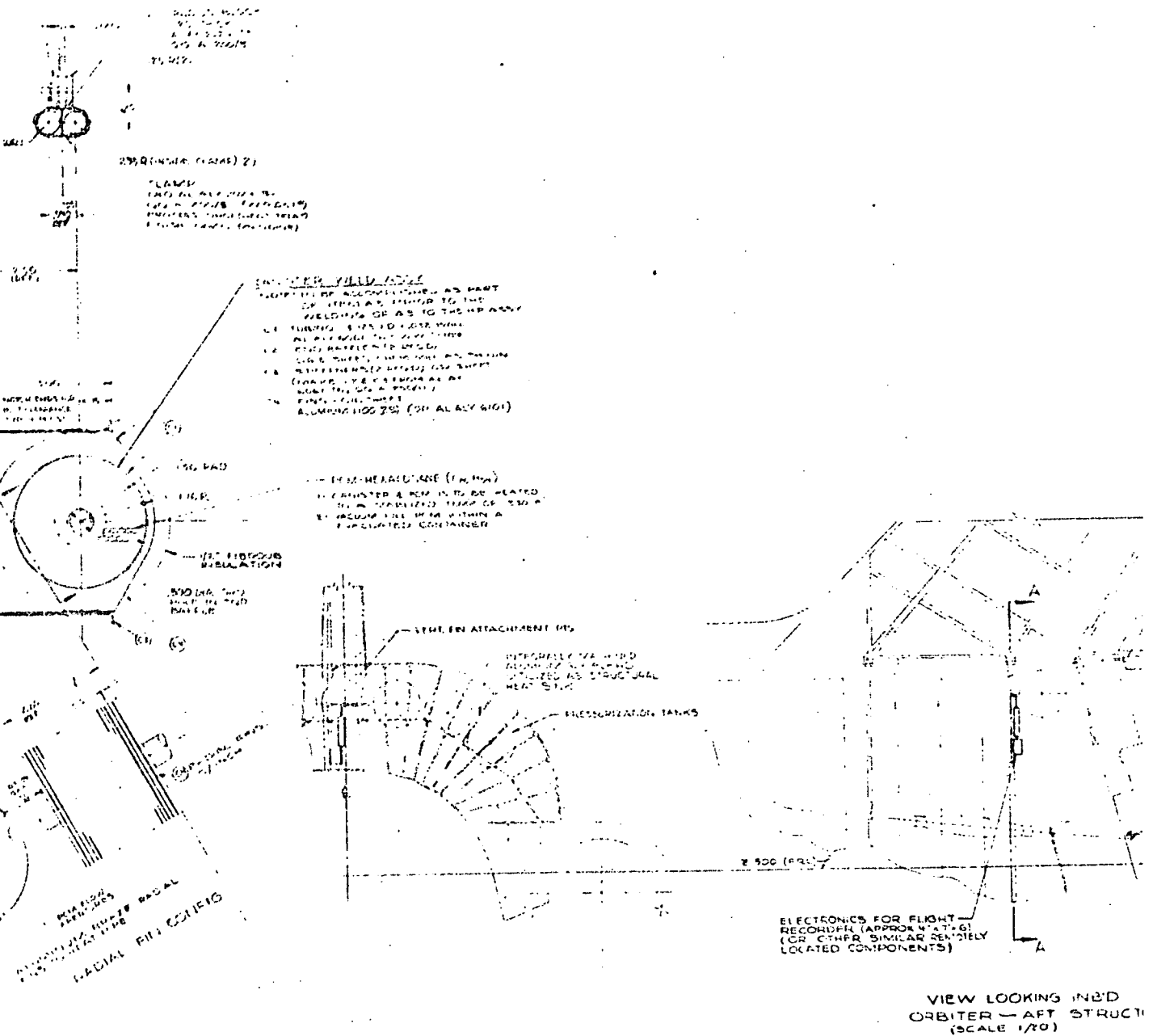
INTERNAL THERMAL
LOAD DIRECTED TO
THIS INTERFACE

100 (REF)

EOLDOUT FRAME 2



FOLDOUT FRAME 3



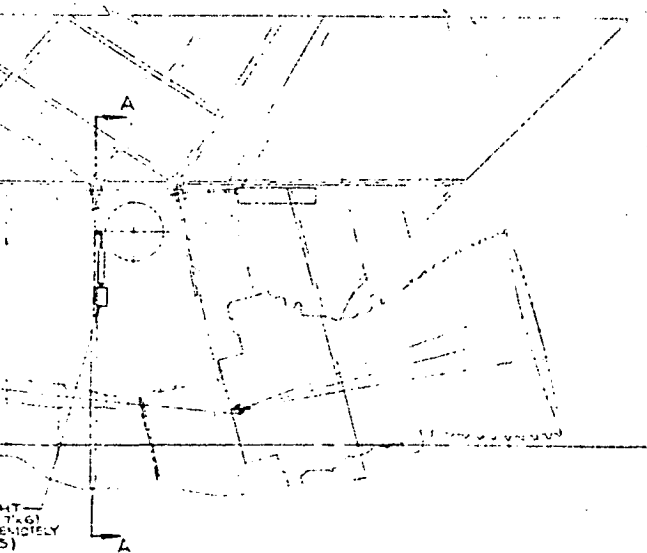
FOLDOUT FRAME 4

PRECEDING PAGE BLANK NOT FILMED

NOTES: (UNLESS OTHERWISE INDICATED)

1. THE INTENT OF THE MODULAR HEAT SINK SHOWN (IE. NORMAL LOAD TO STRUCTURE EXCEPT ON REENTRY Q LOAD GOES TO THE ABSORBER (PCM)) IS SPECIFICALLY APPLICABLE TO A "ELECTRONIC COMPONENT (EXAMPLE SHOWN IS THE FLIGHT RECORDER ELECTRONICS) WHICH IS REMOTELY LOCATED FROM THE WRITER'S FLUID COOLANT SYSTEM
2. THE MODULAR HEAT SINK CIRCUIT WITH RESPECT TO ITS OPERATIONAL SEQUENCE DURING THE FULL DUTY CYCLE IS DESCRIBED IN TABLE I (SEE LOWER LHM CORNER OF LAYOUT)
3. WEIGHT BREAKDOWN (WITHOUT COMPONENT)

HEAT PIPE WELD ASSY	0.7 LB
DRUM HEAT PIPE WELD ASSY	0.6
CANISTER WELD ASSY	1.2
PCM - H2A COSMINE	1.4
INSULATION	0.1
MISC	0.2 LB
TOTAL	4.2 LB



NEW LOOKING IN-B'D
WRITER - AFT STRUCTURE
(SCALE 1/20)

FIG. 9-3.14

DESIGN OFFICE: 303	CONTRACT NO: NAS 9-12034	OF 3000 RESEARCH CORPORATION -TOPAGE, NEW YORK 11744
DESIGNER: J. FORELLI	DATE: 11/27/57	MODULAR HEAT SINK FLIGHT VOICE RECORDER APPLICATION (LAYOUT)
PROJ. ENG: J. FORELLI	DATE: 11/27/57	
PROJ. ENG: J. FORELLI	DATE: 11/27/57	REV. NO. 26.12
PROJ. ENG: J. FORELLI	DATE: 11/27/57	SPL-111
PROJ. ENG: J. FORELLI	DATE: 11/27/57	
PROJ. ENG: J. FORELLI	DATE: 11/27/57	

PRECEDING PAGE BLANK NOT FILMED

9-4 AIR-TO-HEAT PIPE HEAT EXCHANGER

9-4.1 INTRODUCTION

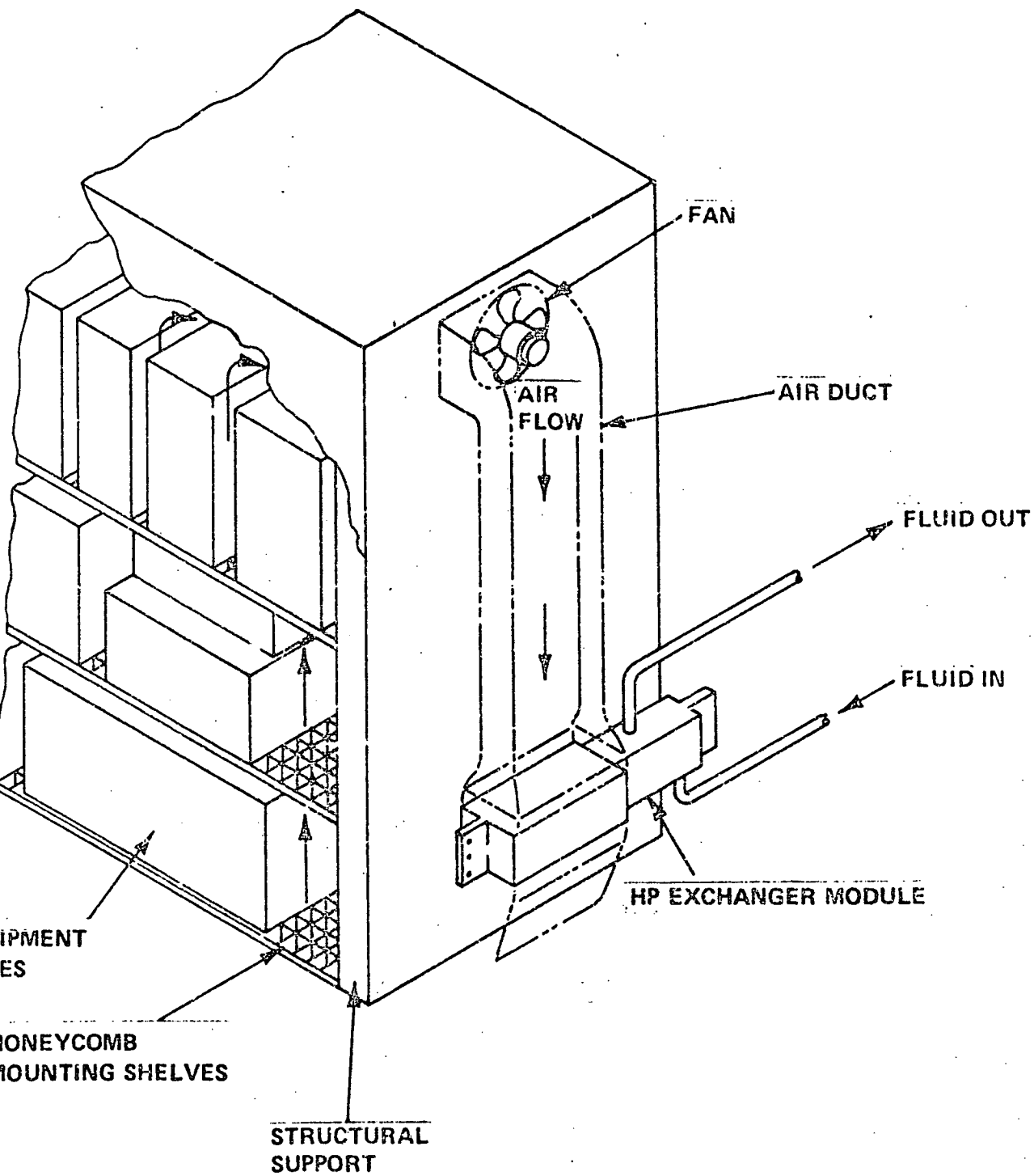
Shuttle plans call for the utilization of qualified military and commercial avionics equipment where possible as a cost effectiveness measure. Such equipment is designed to be cooled by free air convection with limitations on temperature and pressure. They will, therefore, be housed in sealed racks (see figure 9-4.1) with cooling provided by self-contained air circulation systems (sealed and self-contained to retard air loss in the event of temporary losses in cabin pressure.)

The heat exchanger discussed in this section transfers the heat picked up by the rack's air circulation system to the heat transport section of the vehicle's Environmental Control System. A unique feature of this design is that the possibility of a water system leak discharging fluid into the air stream (a possibility with any direct air/water exchanger) has been totally eliminated by using heat pipes to separate the air and water sides of the exchanger.

The unit has been sized to handle an avionics heat load of 1800 watts plus a 10% allowance for the fan assembly load. 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.

9-4.2 ANALYSIS

As shown in Figure 9-4.2, air at temperature T_1 , 10 psia (design condition), and a flow rate of \dot{V}_{air} , enters the avionics rack and is heated at the rate of 1800 watts to temperature T_2 (131°F, design value). The air moves past the fan assembly and is heated to temperature T_3 by the fan motor's heat load. The air then enters the air side of the exchanger wherein it loses its heat to the evaporator ends of a series of isothermalizer heat pipes, exiting at temperature T_1 .

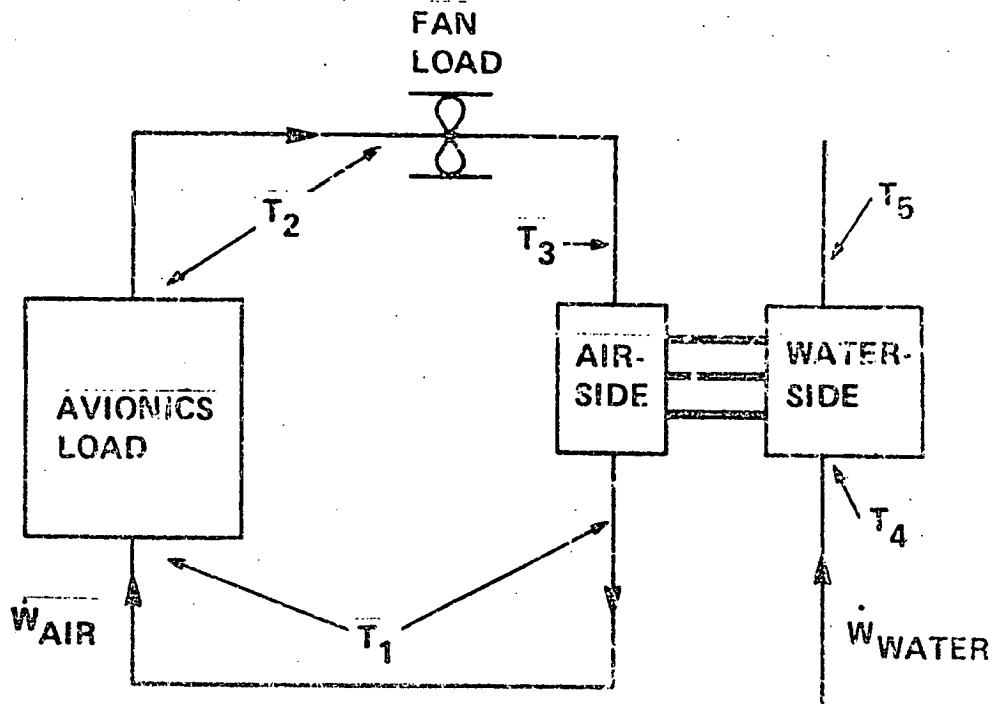


AVIONICS RACK SCHEMATIC

FIGURE 9-4.1

9-78

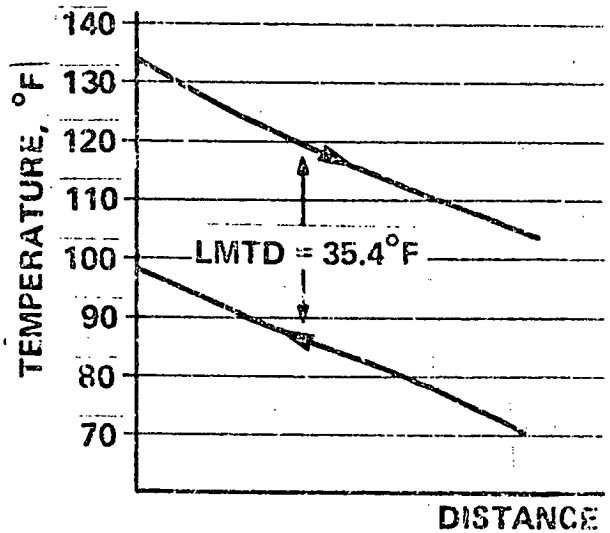




DESIGN DATA

- 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
- $T_1 = 105.4^\circ F$
- $T_2 = 131.0^\circ F$
- $T_3 = 133.6^\circ F$
- $T_4 = 70^\circ F$
- $T_5 = 98.2^\circ F$

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



HEAT EXCHANGER SCHEMATIC
FIGURE 9-4.2



The heat absorbed at the evaporator ends of the heat pipes is transmitted to the condenser ends where it is dissipated to the fluid in the water side of the exchanger. Water enters the exchanger at temperature T_4 (70°F, design condition) and a flow rate of \dot{w}_{water} (reflecting a flow budget of 1 lb. per hour per 8.25 watts of heat), is heated by the heat rejected from the heat pipes, and exits at temperature T_5 .

As can be seen in figure 9-4.2, the fluid streams are arranged in counter-flow. An engineering judgement was made that maximum design efficiency would result from a counter-flow configuration in which the heat capacities of the two fluid streams were made equal (i.e., $(\dot{w} C_p)_{\text{air}} = (\dot{w} C_p)_{\text{water}}$). By so doing, the temperature rise in the water would be exactly equal to the temperature drop in the air stream. This, in turn, would mean equal heat load to each of the several heat pipes since the temperature difference between the air and water streams at any station along the flow path would be the same. This last greatly facilitates the analysis and optimization of the system.

A further decision had to be made concerning the fan assembly requirements. Since fan power requirements are directly dependent on the air system pressure drop (and fan efficiency), one or the other must be specified. Qualitatively, a smaller allowable pressure drop means less fan power required but a larger, more complex heat exchanger. In most high performance aircraft applications, heat exchanger volume and weight are generally conserved at the expense of power since the availability of the latter is rarely limited. This philosophy cannot be adopted on Shuttle since the Shuttle must function as a spacecraft as well as an aircraft. The power and radiator system weight penalties incurred for each watt delivered by the fuel cell system while in orbit are substantial (0.485 lb/watt). While one would therefore tend to require as low a fan load as possible, there is a practical limit below which the required air passages and extended surfaces become impractical from a manufacturing viewpoint.

A decision was made to allow a fan power budget of 10% of the avionics heat load, and design the exchanger for the corresponding pressure drop. This value, though ultimately met by the final design, proved to be quite

stringent and was the pacing requirement in the design. Though initially a "guesstimate", the 10% figure has proven to be a valid lower limit for a practical design.

The above restrictions completely define the design envelope, and permit calculation of the system temperatures, flow rates, and required UA_{Total} . These are:

$$\dot{w}_{water} = 240 \text{ lbs/hr}$$

$$\dot{w}_{air} = 1000 \text{ lbs/hr}$$

$$T_1 = 105.4^\circ\text{F}$$

$$T_2 = 131^\circ\text{F}$$

$$T_3 = 133.6^\circ\text{F}$$

$$T_4 = 70^\circ\text{F}$$

$$T_5 = 98.2^\circ\text{F}$$

$$UA_{Total} = \frac{Q}{IMTD} \quad \text{where } Q = 1980 \text{ watts and } IMTD = 35.4^\circ\text{F}$$

$$UA_{Total} = 191.0 \text{ Btu/hr } ^\circ\text{F}$$

Since the flow of heat is serial through the air-side, heat pipes, and water side, the total thermal resistance may be expressed as the algebraic sum of the resistances through each element

$$\frac{1}{UA_{Total}} = \frac{1}{UA_{Air}} + \frac{1}{UA_{HP}} + \frac{1}{UA_{Water}}$$

The rationale governing the selection of the size, type, shape, number and relative condenser/evaporator lengths of the heat pipes required is briefly outlined below.

A maximum overall exchanger (and thus heat pipe) length was specified at 15 inches.

Water was selected as the working fluid to be used in the pipes based on its relatively high coefficient of performance and its unquestioned acceptance in a limited toxicity application.

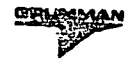
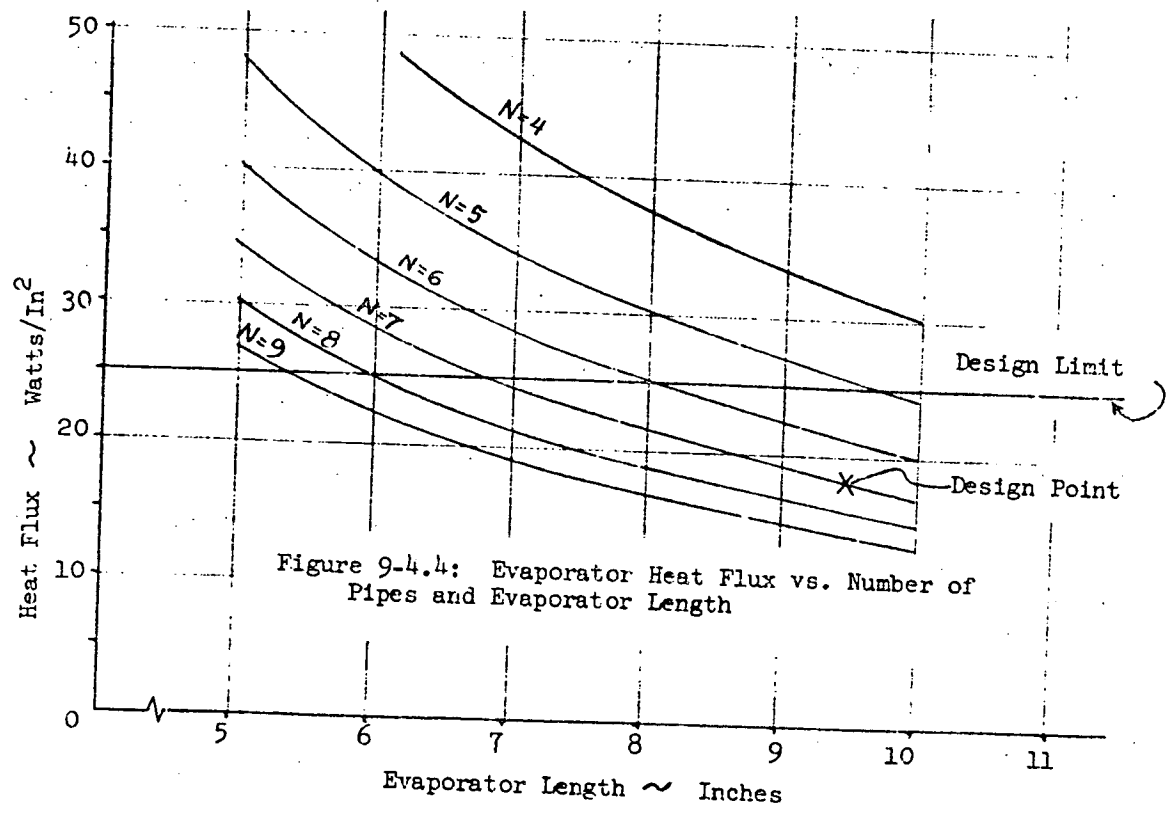
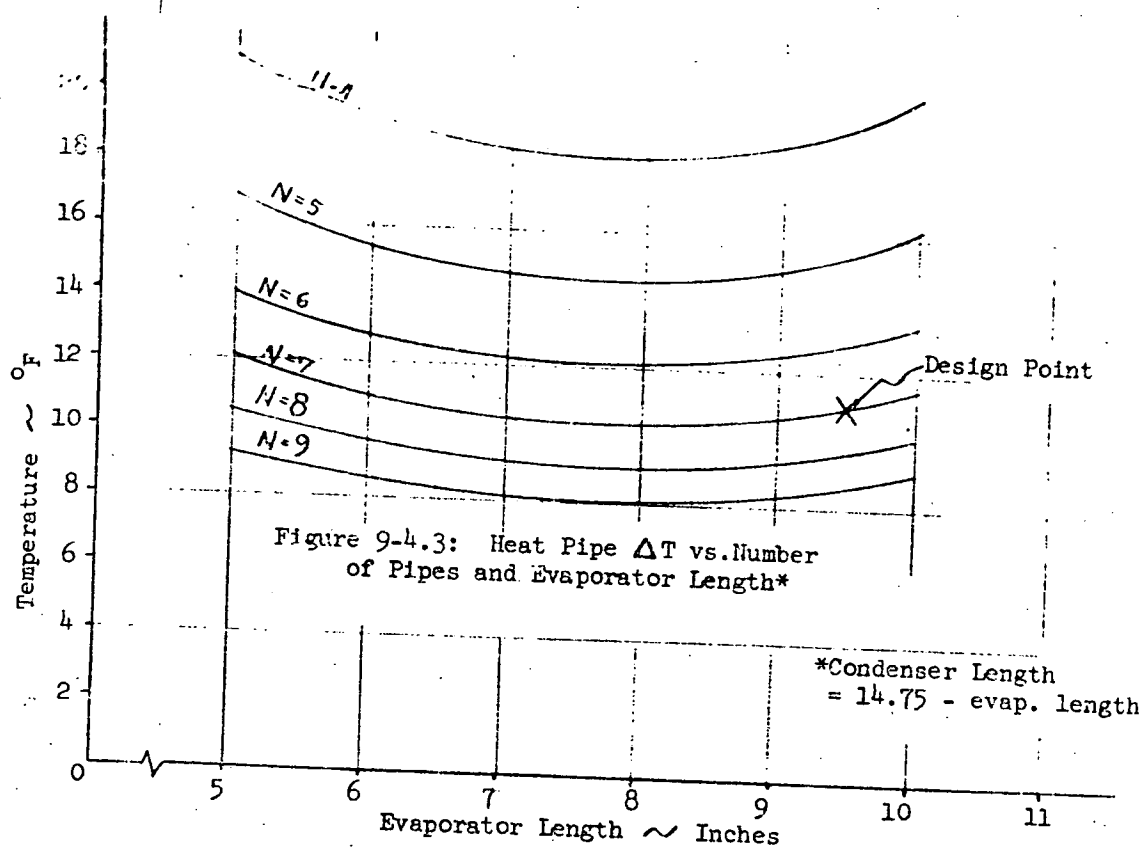
Pipes of square cross-section were selected to facilitate assembly into a heat exchanger of prismatic geometry. The planar outer surfaces of these pipes could readily be bonded to the rectangular plate-fin type cores to be used for the fluid passages.

Copper was the material of choice for two reasons. First, it has a proven compatibility with water. Second, its high thermal conductivity is needed to increase the fin conduction 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 (see Appendix G). Additionally, the evaporator section unit heat flux was determined as a function of the same variables. Experience within Grumman has indicated that evaporator fluxes should be limited to less than 25 watts/in² to preclude the possibility of local dry-out. Figure 9-4.3 and 9-4.4 show these relationships plotted.

On the basis of this analysis, seven pipes were selected as optimum. Less result in too large a ΔT , while more are of ever 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 10.9°F and an evaporator flux of 19.0 watts/in². While an 8 inch evaporator would have resulted in 10.4°F ΔT , it was felt the additional 1.5 inches (19% greater) length on the air side would be desirable at the expense of a 5% higher ΔT .



Eight pipes were included in the final design to provide redundancy in the event of the loss of any single pipe and to facilitate packaging.

With seven pipes, each will carry a load of 283 watts over an effective length of 7 5/8 inches. Analyses performed on 1/2 inch water pipes have shown an optimally designed pipe can exhibit transport capacities well in excess of this requirement.

Knowing the ΔT and heat load, $\frac{1}{UA_{HP}}$ was calculated and subtracted from

$$\frac{1}{UA_{Total}} \text{ to yield } \frac{1}{UA_{Air}} + \frac{1}{UA_{Water}} = 0.00362 \text{ Btu/hr } ^\circ\text{F.}$$

The major portion of the allowable resistance was allocated to the air-side, and split as follows

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

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

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^oF (see Appendix G).

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 (reference 17). The method uses a mathematical correlation of heat transfer data for the flow of air in rectangular ducts (reference 15). In particular, it is found that both the product of friction factor and Reynolds number, as well as Stanton number, Prandtl number (to a power) and Reynolds number, are constant in the laminar region. By specifying allowable UA's, pressure drops, flow length and geometric core arrangement, one may determine the size and number of core layers required as well as the numbers of fins per inch required in the extended surface.

Figure 9-4.5 details the final exchanger design. The pipes are arranged in two rows of four each. On the air-side, a 3-layer core is arranged on either side of each row of pipes. On the water side, only a single layer core is needed on either side of each row of tubes.

Though not shown in figure 9-4.5, provisions for a secondary (backup) water loop can be provided by adding a second layer of water-side core to each of the layers shown. These additional layers would provide the flow path for the secondary loop. For the core sizes shown, a 13°F higher rack air temperature would result when supplying this secondary water path with fluid at 70°F. This can be compensated for by either supplying secondary system water at a lower temperature, or alternatively by re-optimizing the basic water core design to reduce this 13°F penalty (i.e., use thicker fin material).

9-4.3 DESIGN

Since the configuration of this system is dictated by the thermal requirements and has been detailed above, the only remaining design task is to specify the fabrication methods to be employed.

As shown in figure 9-4.5, the design consists of two conventional heat exchangers connected by 8 water heat pipes of square cross-section. Grumman has a proven in-house technique for forming square pipes. Basically, grooves are first cut in a circular section and then upset formed into a square section (see figure 9-4.6). It was decided to select one-half inch (nominal) as the basic internal dimension with a 0.028 inch wall. A copper spiral artery wicking system has been designed for these pipes. The artery has a core diameter of 0.110 inches. The retainer has an outer diameter of 0.250 inches. (See Figure 9-4.7.) The inside of the heat pipe will be coated with Ebonol C prior to soldering of end caps, charging, and sealing. This coating increases the ability of the water working fluid to wet the wicking surfaces.

Both heat exchanger fin configurations are constructed of aluminum fins (alloy 1100), .008 inches thick, which are brazed to .010" thick face sheets (aluminum 6061-T6). The heat exchanger cores are then brazed or soldered to the copper heat pipes after appropriate surface treatment.

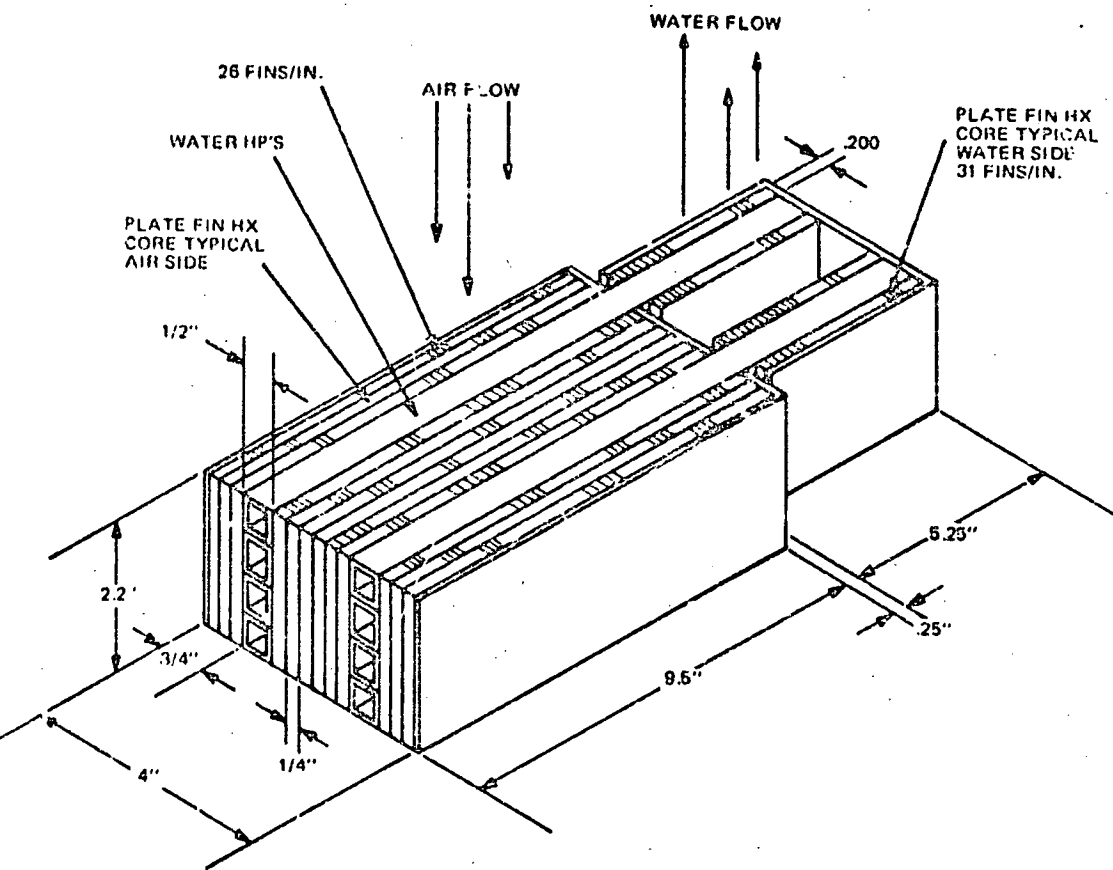


Figure 5-4.5 Avionics Rack HP Heat Exchanger

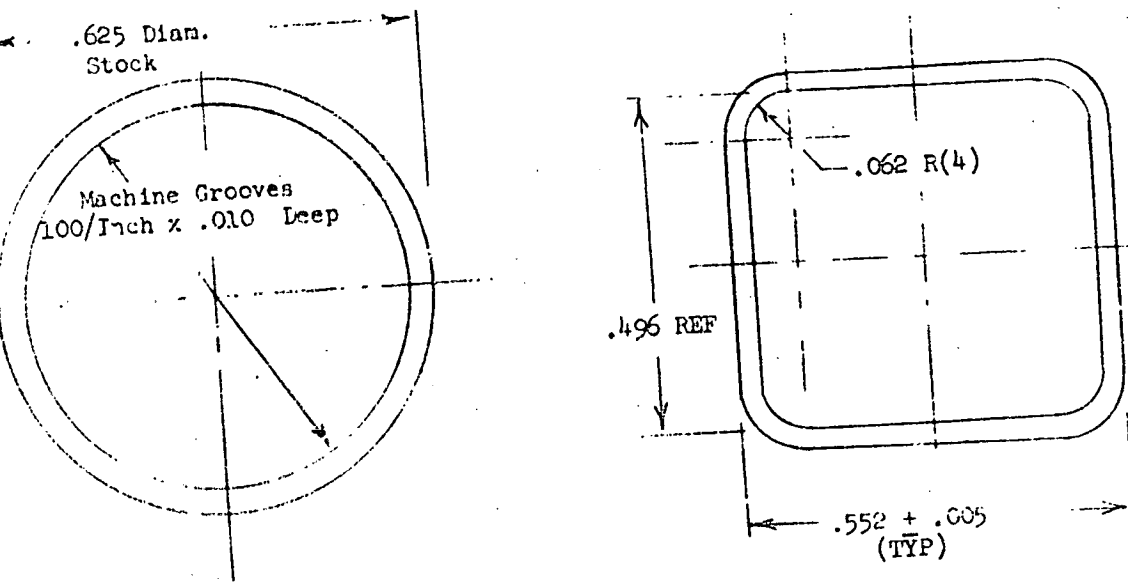
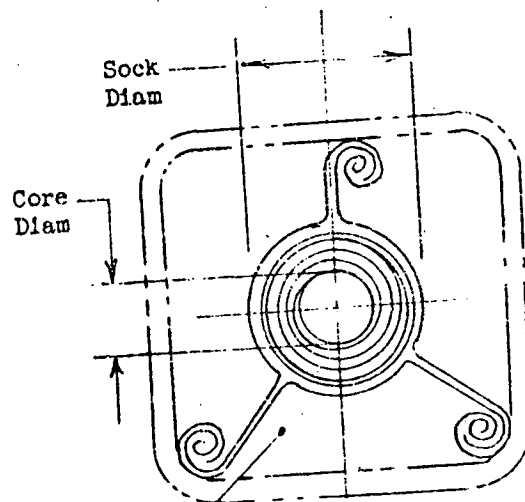


Figure 9-4.6 - Square Pipe Forming

HEAT PIPE INTERNAL CONFIGURATION

Items & Data

- Working Fluid
Distilled Water
- Spiral Artery
Hollow Core Dia. = .110
- Retainer
Sock Dia (C.D.) = .250
- Tube
Copper 101 (OFHC)
(ASTM Spec B68,75,280)



All internal surfaces and parts coated with Ebonol C.

Figure 9-4.7 - Artery Detail



The results of a detailed weight analysis, done in support of the design effort, are given in Table 9-4.1.

9-4.4 CONCLUSIONS

The baseline system consists of an enclosed rack, similar to that discussed above, with a heat exchanger that transfers heat directly from the circulating air in the rack to the pumped coolant (water) loop. This places the fluid side of the exchanger within the equipment enclosure. With the heat pipe system, because the coolant lines never enter the circulating air enclosure, there is no risk of electronic equipment damage due to fluid leakage in the heat exchanger.

TABLE 9-4.1

HEAT PIPE/HEAT EXCHANGER AIR COOLED EQUIPMENT SFJ-105

ITEM	QTY	UNIT WEIGHT LB.	TOTAL WEIGHT LB.	DATA/COMMENTS
Heat Pipe	8	(.240) lb/ft	2.40	Total length 10 ft (15" unit length)
Fluid - H2O		.024 lb/ft		Distilled water
Spiral Artery		.023 lb/ft		Hollow core dia. = .110
Retainer		.020 lb/ft		Sock Dia. (OD) .25"
Tube		.173 lb/ft		Copper 101 (CFHC)
Air Fins			(2.00)	
Fins		.17 lb/ft	1.62	26.3 fins/in .008" thick Al 1100
Face Sheets		.04 lb/ft	.38	01" except outer skin (.02") Al
Fluid Fins			(.36)	
Fins		.15 lb/ft	.26	31.3 fins/in .007" thick Al 1100
Face Sheets		.06 lb/ft	.10	.01" Al
End Closing Member				
- Air Flow			.30	.125" Al
- Fluid Flow			.31	.125" Al
Mid Member			.35	.125" Al
Total Weight			5.72	

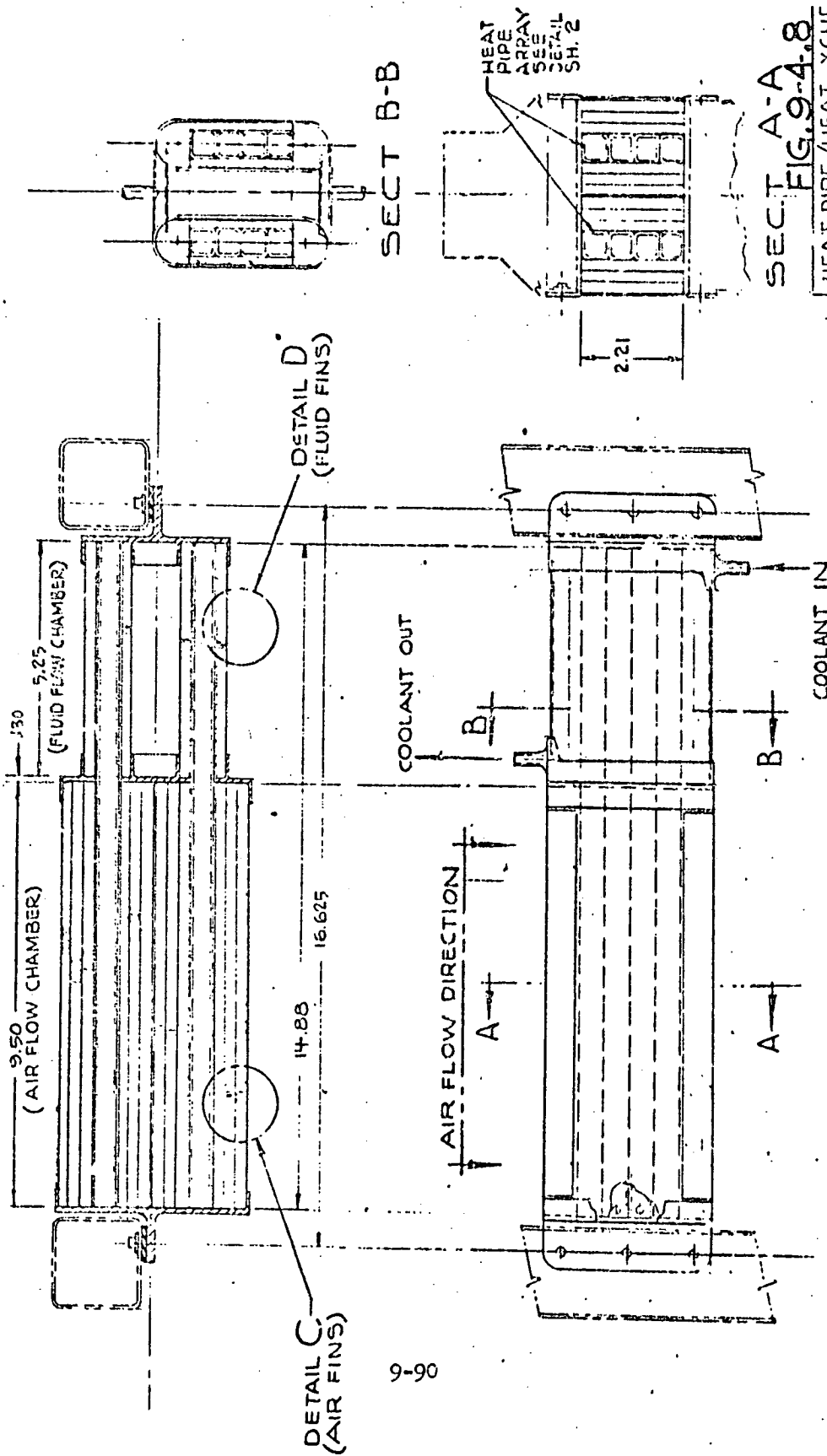
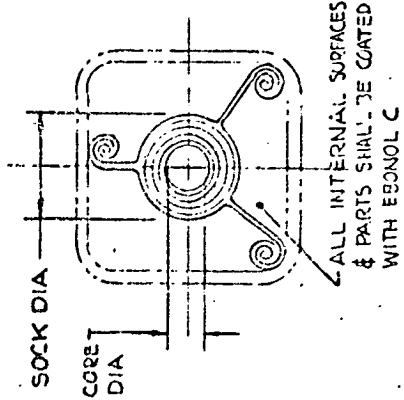
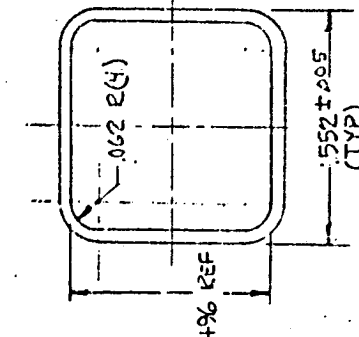
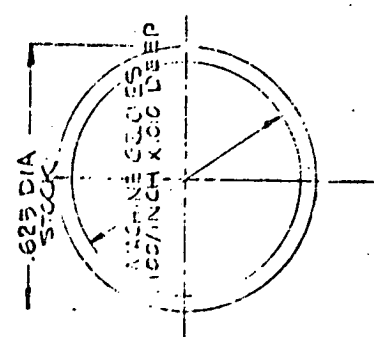


FIGURE 9-4.8

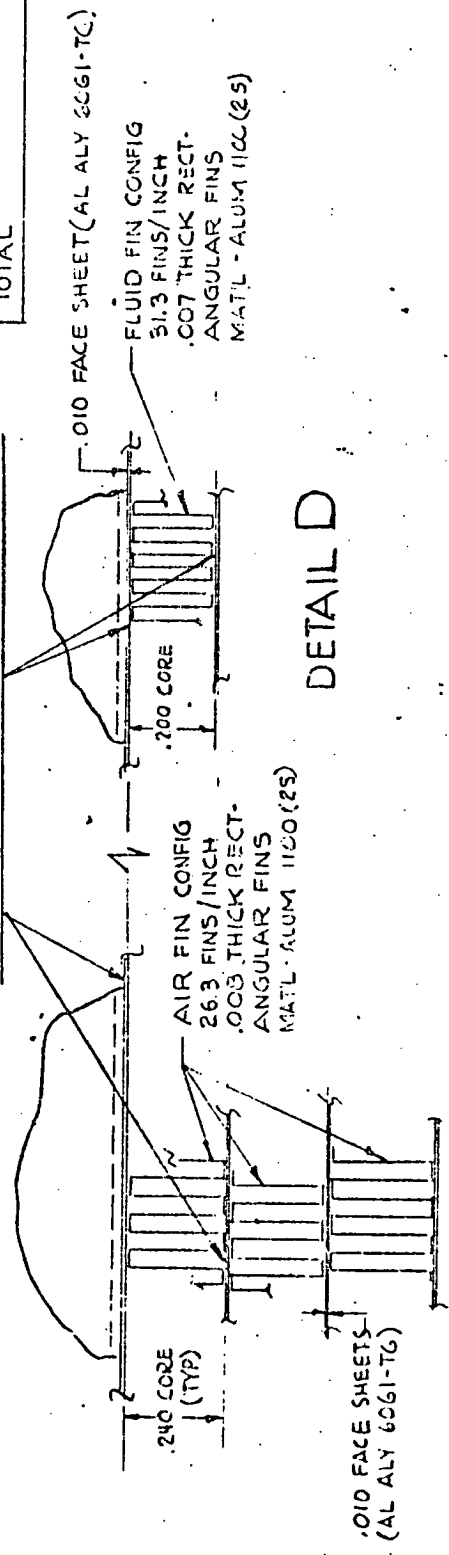
SECT A-A
 FIG. 9-4.8
 HEAT PIPE/HEAT XCHF
 AIR COOLED EQUIP
 SPL-105
 SHEET 1 OF 2





ALL INTERNAL SURFACES & PARTS SHALL BE COATED WITH EBONOL C

ALUMINUM BRAZE HEAT EXCHANGER ASSY. EXTERNAL SURFACE OF COPPER TUBES TO BE TIN PLATED PRIOR TO THIS OPERATION



DETAIL D

DETAIL C

TABLE I
INTERNAL HEAT PIPE CONFIGURATION

ITEMS & DATA	WT #/FT
WORKING FLUID DIJILLED WATER	.024
SPIRAL ACTERY HOLLOW CORE DIA = .110 NO OF WRAPS = 3 GAP SPACE (OR 2.0 DIA) = .010 NO OF RODS = 12	.023
RETAINER SOCK DIA (O.D) .250 NO. OF WEBS 3	.020 (EST)
TUBE COPPER 101 (OFHC) (ASTM SPEC B36, T5, 222)	.173
TOTAL	.24 #/FT

SCALE 4:1
SPL-105
SHEET 2 OF 2



PRECEDING PAGE BLANK NOT FILMED

9-5 HEAT PIPE RADIATOR FOR COMPARTMENT TEMPERATURE CONTROL

9-5.1 INTRODUCTION

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, and air breathing engine compartments. They all share the basic thermal requirement of maintaining their contents between prescribed temperature limits for all mission phases. This is generally accomplished by both insulating the compartment walls and providing a means of controlled heat input. The 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, in the form of heater ribbon, are the simplest to install since they can be attached to the inner walls of the compartment and require no supporting structure of their own. However, electrical power and thermostatic controls must be provided and the additional electrical power is a concern because of the weight penalty it presents in terms of extra fuel cell capacity and reactant requirements.

The fluid radiator concept utilizes the sensible heat of the Freon-21 in the orbiter's heat rejection system. The Freon-21 is tapped from the fuel cell coolant outlet, the warmest point in the heat rejection loop, and circulated through aluminum panels which line the compartment walls; heat is radiated from the panels to the surroundings. The fluid radiators need supporting structure, by-pass and shut-off controls and protection from tube puncture when such a hazard exists. The fluid controls are needed to prevent reverse heat transfer during entry.

The application of a heat pipe radiator system for compartment temperature control can have 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.

9-5.2 ANALYSIS

As shown in Figure 9-5.1 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 heat pipe radiator panels, are located on each side of the compartment facing a tire sidewall.

The heat exchangers are placed in parallel with the fluid flow. Thus, the maximum flow rate available to each exchanger is in proportion to the number of HP/fluid exchangers on the same fluid circuit. 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. 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.

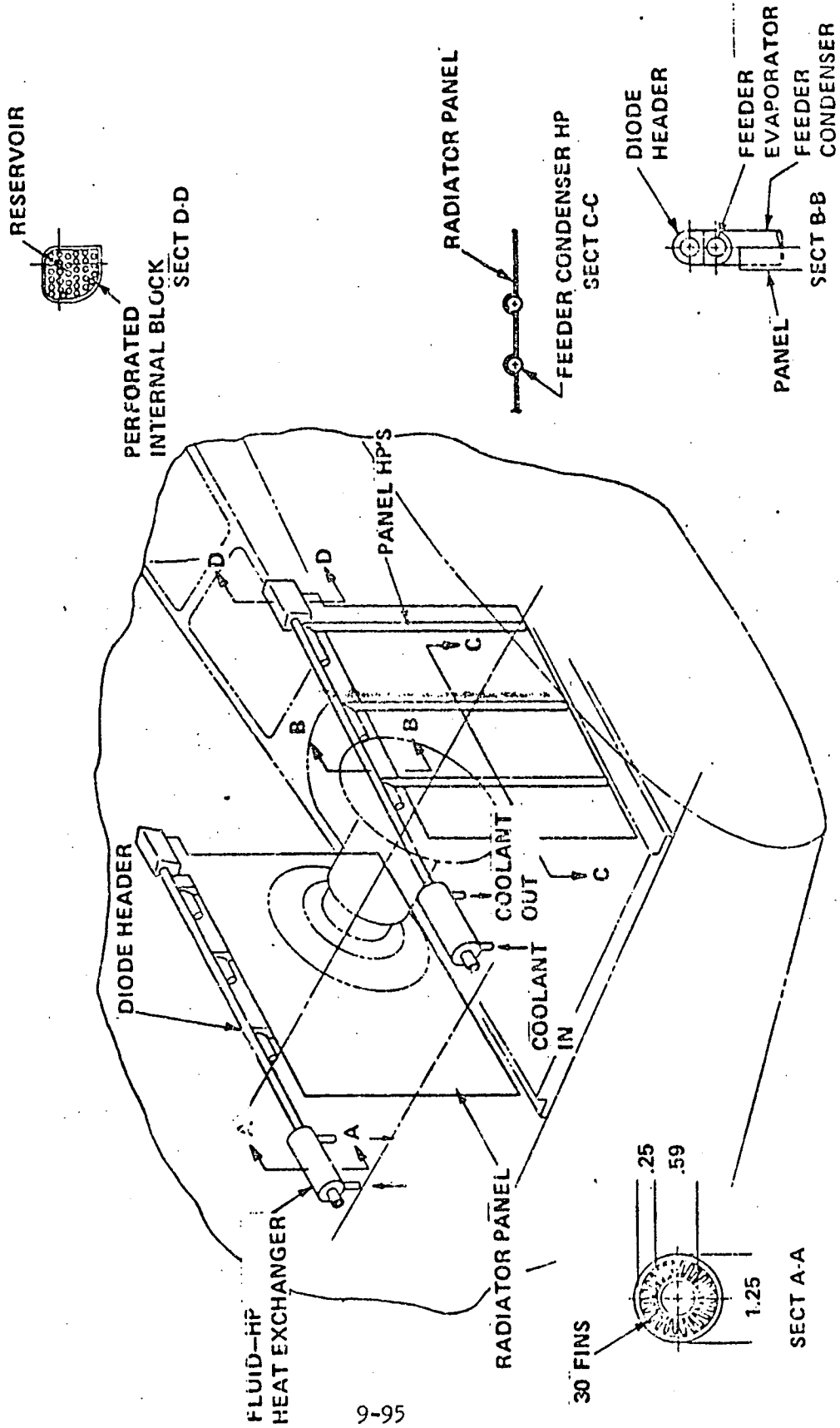
Energy is transferred from the fluid to the evaporator section of the diode heat pipe by means of the annular HP/fluid HX. The diode, functioning as a normal heat pipe, then transmits the heat to its condenser where it is transferred to the attached evaporator sections of the radiator panel feeder heat pipes. The energy is ultimately transferred to the radiator surface from the condenser sections of the feeder heat pipes which are joined to the panel. Whenever the panel temperature exceeds that of the diode evaporator the diode reverses, breaking the heat transfer path between the fluid and the radiator panel.

Design Requirements and Assumptions

o Temperature Constraints

	<u>Desired Temperature Range</u>		<u>Limiting Temperature Range</u>	
	<u>Low</u>	<u>High</u>	<u>Low</u>	<u>High</u>
Hydraulic Fluid:	-20°F	400°F	-65°F	630°F (transient)
Rubber Tires :	-50°F	200°F	-65°F	270°F

FIGURE 9-5.1 - MAIN LANDING GEAR COMPARTMENT RADIATOR SYSTEM



o Design Point: Landing gear temperature range = 0°F to 115°F

o Waste Heat Source

- Fluid : Freon-21 ($C_p = .25 \text{ Btu/lb}_m \text{ } ^\circ\text{F}$)
- Flow rate: 500 lb/hr per diode/fluid heat exchanger
- Inlet temperature range: 85°F to 117°F

o Design Parameters

The physical properties of the heat pipe working fluids and thermal insulation, and heat pipe film coefficients are given in Table 1.

A thermal contact conductance of 1000 Btu/Hr-Ft²-°F was used between mechanically bonded aluminum surfaces. It assumes a 64//-inch surface finish and the use of wiped silicone grease between the mating surfaces.

Fin effectiveness values used in the heat transfer calculations for the ammonia and Freon heat pipes are given below. They were calculated for a uniform heat input over the indicated portion of the circumference.

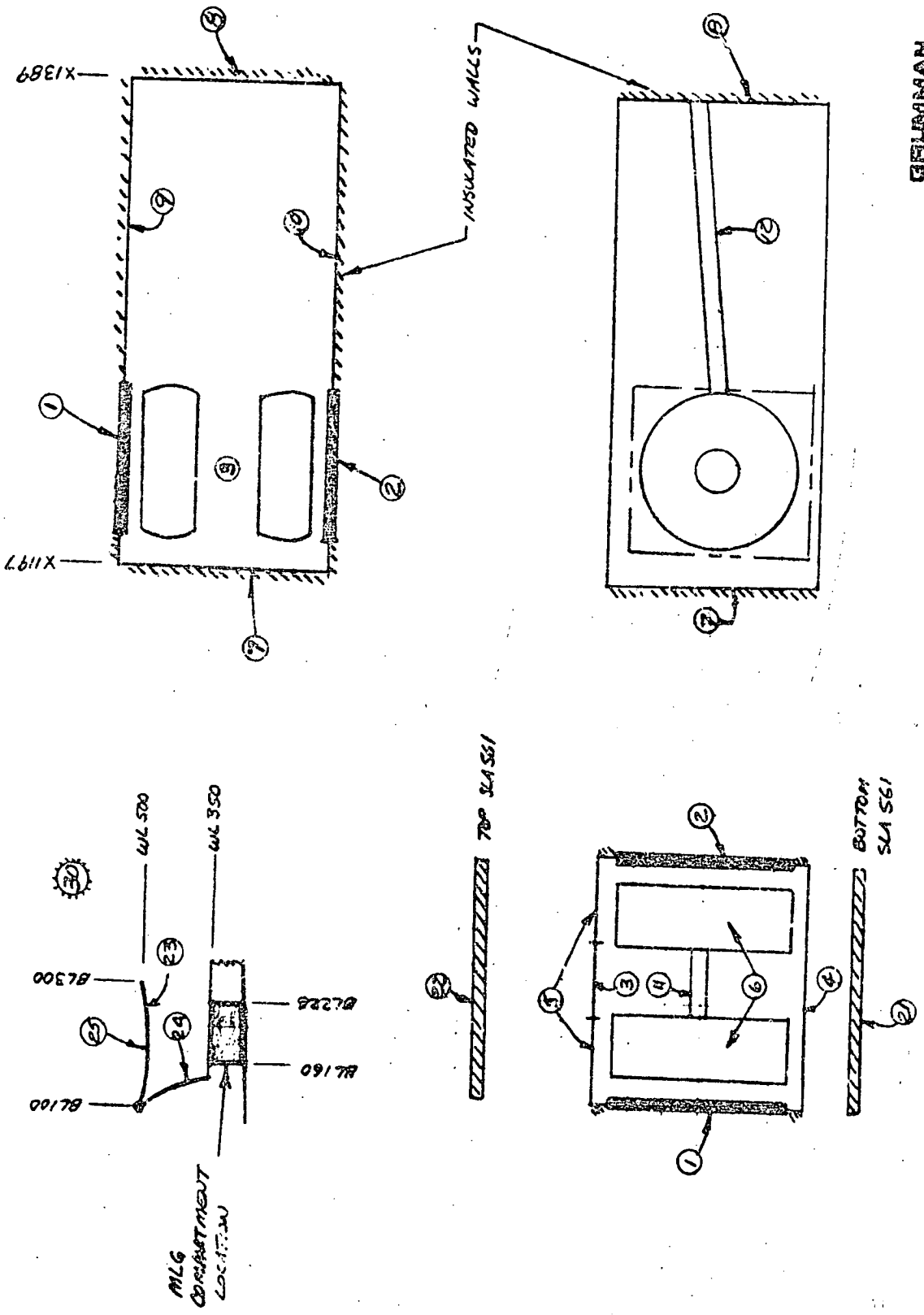
	η_F	Uniform Heat Transfer Segment
Diode Condenser	.446	90°
Feeder HP Evaporator	.545	90°
Feeder HP Condenser	.670	180°

o Orbital Environment

An eighteen node thermal model of the main landing gear compartment was used to establish compartment temperatures as a function of radiator panel heat input, insulation thickness, compartment structure emittance and orbital flight mode. Figure 9-5.2 shows the idealization, which assumed ablative top and bottom surfaces (SIA 561), a negligible heat leak through the compartment sidewalls, and 32 ft² of compartment radiator area (16 ft² per panel). The upper compartment surface (node 22) was assumed to be completely shadowed from direct solar inputs by the opened cargo bay door. The space radiator panels (node 25), which are mounted to the door, were fixed at an average temperature of 55°F. Optical properties for the external surfaces were assumed to be those of a flat absorber ($\alpha/\epsilon = .9/.9$).

MLG COMPARTMENT IDEALIZATION

FIG. 9-5.2



Based on Reference 1 (Section 8.3.5.4.1), orbital average inputs for an earth oriented mode (orbit plane parallel to ecliptic plane) were used as the "hot case" inputs, and orbital average inputs for a solar inertial mode (polar orbit) were used as the "cold case" inputs. These are summarized in Figure 9-5.3.

Average landing gear temperature as a function of insulation thickness, structural emittance and compartment radiator heat input are given in Figure 9-5.4. An emittance of .90 was used for the radiator panels. Emittance of .20 and .85 were picked as reasonable bounding values for the surrounding compartment structure, although the higher, .85, number would more accurately represent actual flight structure which has been subjected to the rigors of use. Based on these curves the compartment was assumed to be insulated with a one inch thickness of TG-15000 since this results in reasonable on-orbit landing gear temperatures (0 to 100°F) at moderate radiator panel heat inputs (10 to 15 Btu/Hr-Ft²) over the selected orbital extremes. No attempt was made to weight optimize the system.

System Design and Analysis

Figures 9-5.5 and 9-5.6 show the variation in landing gear and radiator panel temperature as a function of radiator heat input for the solar inertial and earth oriented flight modes, respectively. From Figure 9-5.5, assuming a structure emittance of .85, the minimum on-orbit landing gear temperature can be limited to 0°F by supplying 320 Btu/Hr to the compartment (10 Btu/Hr-Ft² of radiator). This same heat input during an earth oriented flight mode (Figure 9-5.6) will result in an upper temperature limit of about 50°F.

Generalized System Equations and Analysis Procedure System Equations

The following relationships define the temperatures throughout the heat pipe radiator system as represented in Figure 9-5.7. Refer to Table 9-5.1 for symbol definitions.

1. Heat transferred from fluid (Q):

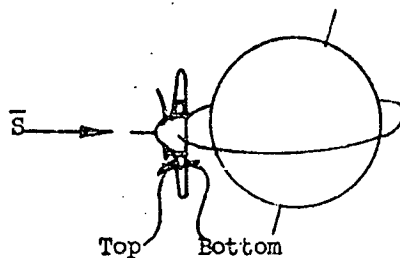
$$Q = mC_p (T_{in} - T_{out})$$

FIG. 9-5.3 - ORBITAL ENVIRONMENT

o Earth Oriented Mode

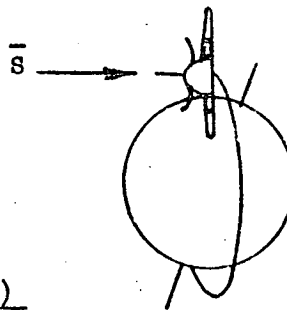
- Orbit Plane Parallel to Ecliptic Plane

	<u>Incident Energy (Btu/Hr-Ft²)</u>		
	<u>Solar Albedo IR</u>		
Top Surface :	140	0	0
Bottom Surface :	11	41	61



o Solar Inertial Mode (Polar Orbit)

	<u>Incident Energy (Btu/Hr-Ft²)</u>		
	<u>Solar Albedo IR</u>		
Top Surface :	442	4	19
Bottom Surface :	0	0	19

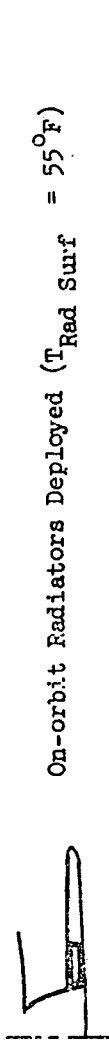


o Worst Case Orbital Environment (Btu/Hr-Ft² - Absorbed)

	<u>Top Surface *</u>		<u>Bottom Surface</u>	
	<u>(S + a)</u>	<u>IR</u>	<u>(S + a)</u>	<u>IR</u>
Hot Case :	0	0	52	61
Cold Case :	4	19	0	19

*Includes blockage by cargo bay doors

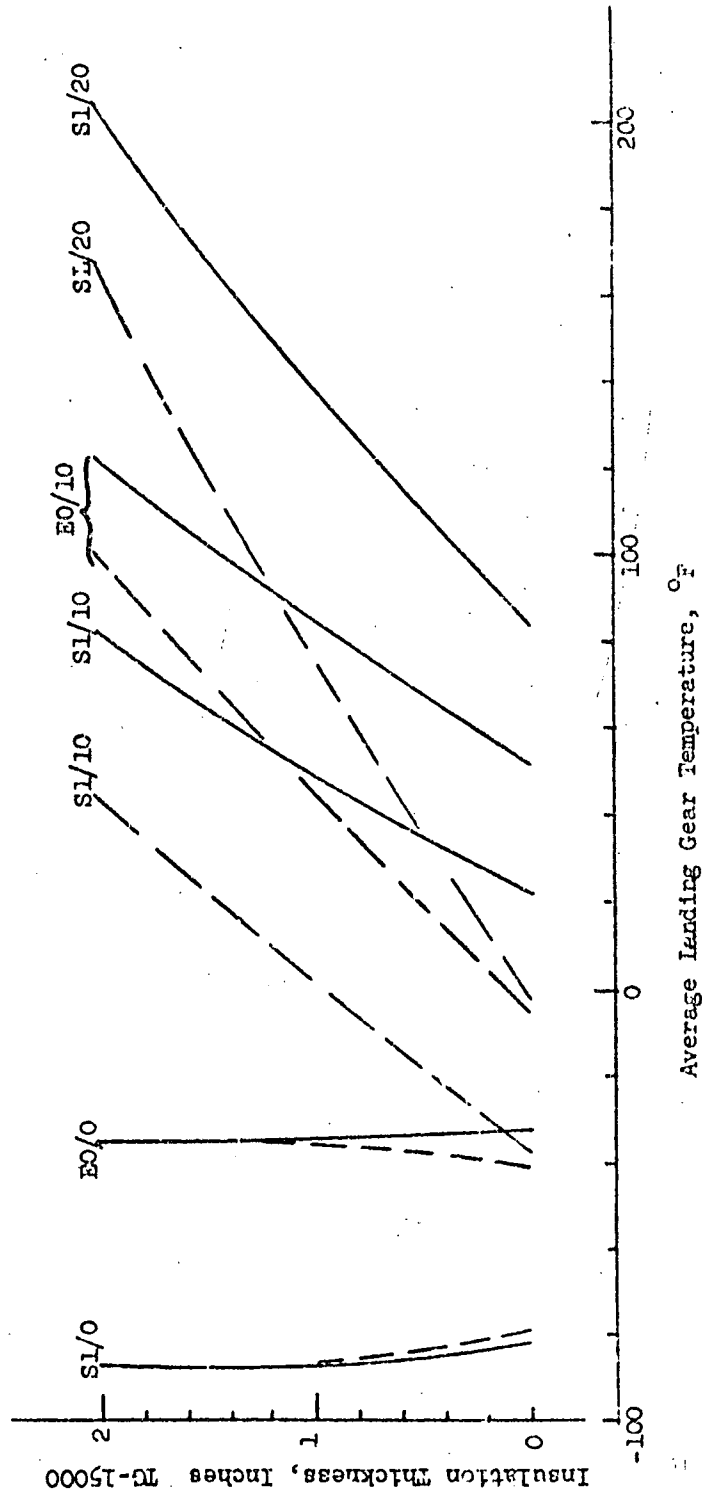
FIG. 9-5.4 - AVERAGE LANDING GEAR TEMPERATURE vs. INSULATION THICKNESS & COMPARTMENT RADIATOR HEAT INPUT



On-orbit Radiators Deployed ($T_{Rad Surf} = 55^{\circ}F$)

Sl/No. = Solar Inertial Orbit/Radiator Surface Heat Input (Btu/Hr-Ft²)
 Eo/No. = Earth Oriented Orbit/Radiator Surface Heat Input (Btu/Hr-Ft²)

— $\epsilon = .20$
 --- $\epsilon = .85$



BRUNN

FIG. 9-5.5 - LANDING GEAR TEMPERATURE vs. RADIATOR SURFACE HEAT INPUT

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

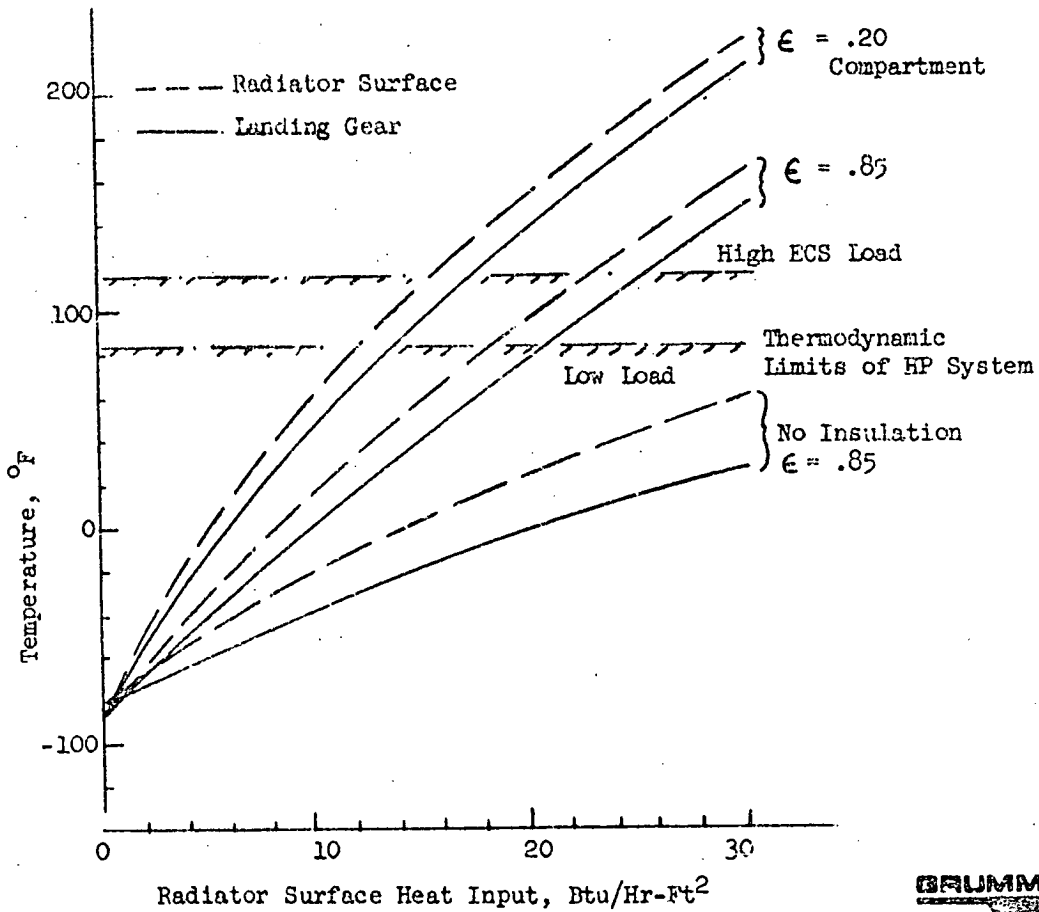
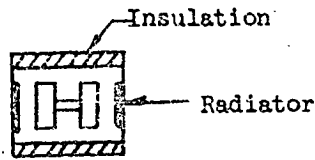


FIG. 9-5.6 - LANDING GEAR TEMPERATURE VS. RADIATOR SURFACE HEAT INPUT

- o Earth Oriented Orbit
- o Insulation Thickness = 1 inch (TG-15000)

Insulation



Radiator

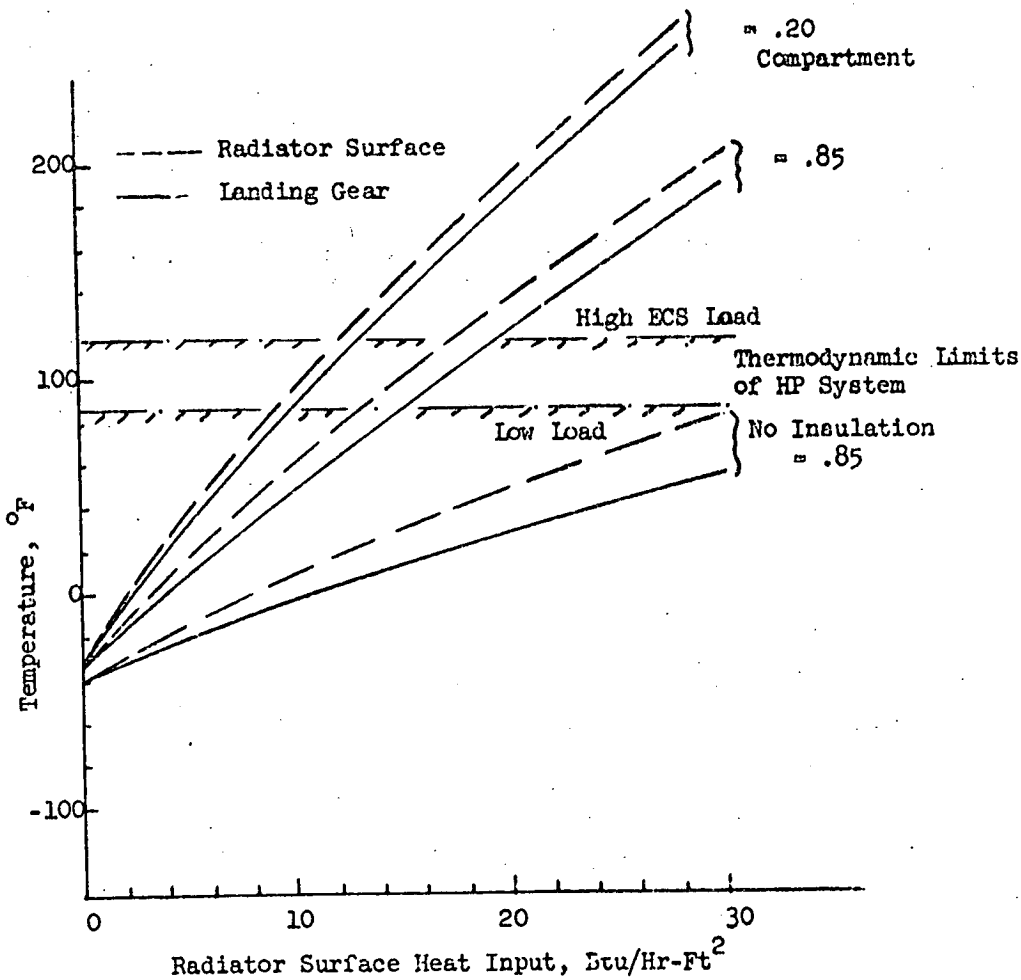
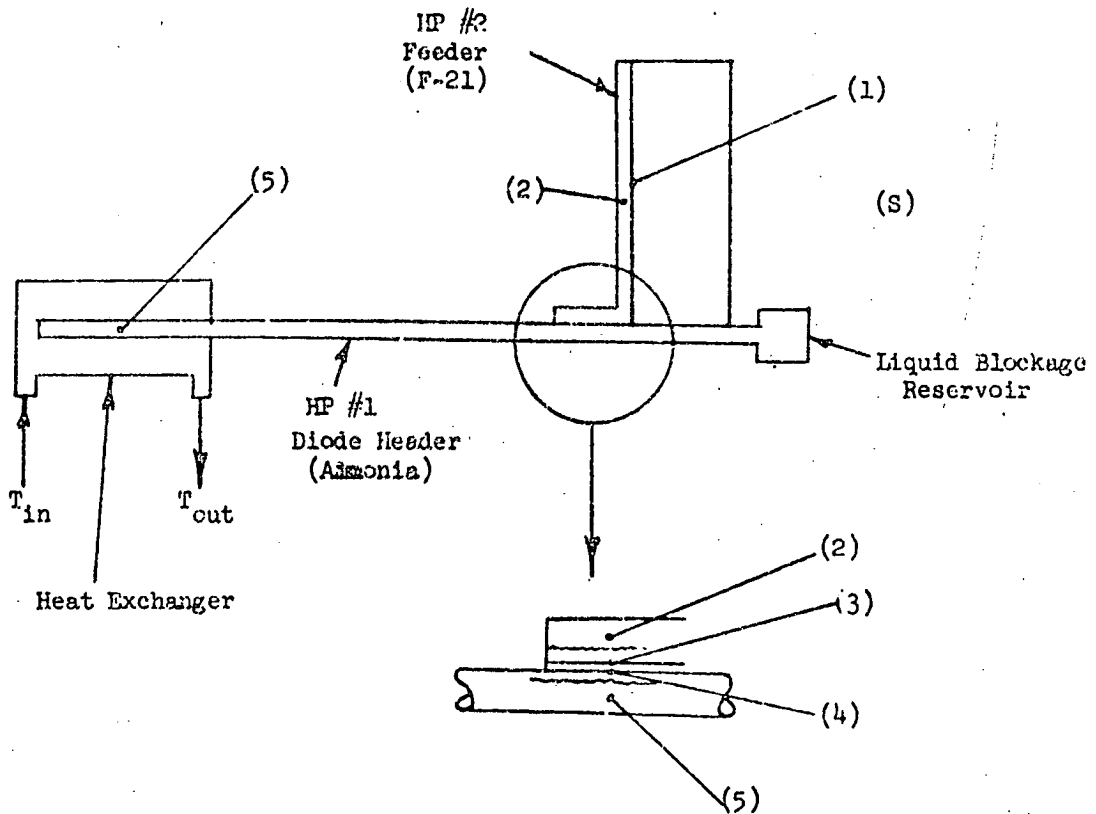


FIG. 9-5.7 - 4P RADIATOR SYSTEM SCHEMATIC



- (1) - Fin Root
- (2) - Feeder HP Vapor
- (3) - Feeder HP Evaporator Wall
- (4) - Header Condenser Wall
- (5) - Header Vapor
- (8) - Effective Compartment Sink

TABLE 9-5.1 - NOMENCLATURE

Q	=	Total radiator panel heat rejection
N	=	number of heat pipes on radiator panel
(mCp)	=	mass flow x specific heat for Freon-21
ϵ	=	heat exchanger effectiveness
N_R	=	Reynolds Number
N_{PR}	=	Prandtl Number
η	=	radiator panel fin effectiveness
η_F	=	fin efficiency
T	=	temperature
ϵ_R	=	surface emittance
A_R	=	radiator panel area
A_C	=	contact surface area feeder HP to header
A	=	HP surface area, internal
h	=	heat transfer film coefficient
h_c	=	interface contact conductance = 1000 Btu/Hr-ft ² -°F (377 psi needed, 700 psi used)

Subscripts

R	=	radiator surface
1	=	radiator panel fin root (feeder HP outside condenser wall)
2	=	feeder HP vapor
3	=	feeder HP evaporator outer wall
4	=	header HP condenser outer wall
5	=	header HP vapor
C1	=	condenser HP #1 (diode header)
C2	=	condenser HP #2 (feeder HP)
e2	=	evaporator HP #2 (feeder HP)
S	=	sink

2a. Diode header vapor temperature (T_5):

$$T_5 = T_{in} - \frac{(T_{in} - T_{out})}{\epsilon}$$

2b. Heat exchanger effectiveness (ϵ):

$$\epsilon = 1 - e^{-\frac{UA}{mC_p}}$$

3. Diode header condenser wall (T_4):

$$T_4 = T_5 - \frac{Q/N}{\eta_{FC1} h_{C1} A_{C1}}$$

4. Feeder HP evaporator wall (T_3):

$$T_3 = T_4 - \frac{Q/N}{h_c A_c}$$

5. Feeder HP vapor (T_2):

$$T_2 = T_3 - \frac{Q/N}{\eta_{Fe2} h_{e2} A_{e2}}$$

6. Radiator Panel Root Temperature (T_1):

$$T_1 = T_2 - \frac{Q/N}{\eta_{FC2} h_{C2} A_{C2}}$$

Equations 1 thru 6 can be combined to express the panel root temperature in terms of the fluid inlet temperature and a series of temperature drops throughout the system. This is done in equation 8.

$$8. T_1 = T_{in} - \frac{Q/mC_p}{\epsilon} - \frac{Q}{N} \left[\frac{1}{\eta_{FC1} h_{C1} A_{C1}} + \frac{1}{h_c A_c} + \frac{1}{\eta_{Fe2} h_{e2} A_{e2}} + \frac{1}{\eta_{FC2} h_{C2} A_{C2}} \right]$$

or

$$T_1 = T_{in} - \Delta T_{in-5} - \Delta T_{5-4} - \Delta T_{4-3} - \Delta T_{3-2} - \Delta T_{2-1}$$

The radiator panel root temperature and surface temperature can also be determined by considering the overall heat balance between the radiator panels and an effective compartment sink temperature (T_S) as defined in Figure 9-5.8, which was constructed from the results of the network analyses.

9. Radiator Panel Root Temperature based on compartment heat balance:

$$\sigma T_1^4 = \frac{(Q/A_R)}{\eta \epsilon_R} + \sigma T_S^4$$

10. Radiator Panel Root Temperature in terms of surface temperature:

$$T_R^4 = \eta T_1^4 + (1 - \eta) T_S^4$$

or

$$T_1^4 = \frac{1}{\eta} [T_R^4 - (1 - \eta) T_S^4]$$

Analysis Procedure

To match the system design with system requirements, equations 8 and 10 must be simultaneously satisfied for a given panel heat dissipation. The procedure is, of necessity, an iterative one that can be somewhat simplified by graphing the expressions. Figure 9-5.9 is a flow chart that shows the required input variables and major steps to be followed in arriving at a matched design.

o Radiator Panel

The basic radiator panel is made of 20 mil aluminum and has 16 ft² of surface area with a high emittance coating ($\epsilon = .9$). The required number of heat pipes per panel (feeder heat pipes) is determined by the required panel fin effectiveness.

Figure 9-5.10 is a general curve which gives the fin effectiveness of a rectangular panel as a function of material, root temperature, panel thickness and fin length. It assumes a sink temperature approximately equal

FIG. 9-5.8 - LANDING GEAR COMPARTMENT EFFECTIVE SINK TEMPERATURE

o Includes 'Weighted' Radiant Energy Contributions from Landing Gear & Compartment Walls

$$\sigma \frac{P}{S} = \sum_{i=1}^N \sigma T_i^4 \frac{\overline{F}_{R-i}}{\epsilon_R}$$

R = Radiator Surface

i = Enclosure Surfaces

S = Effective Sink

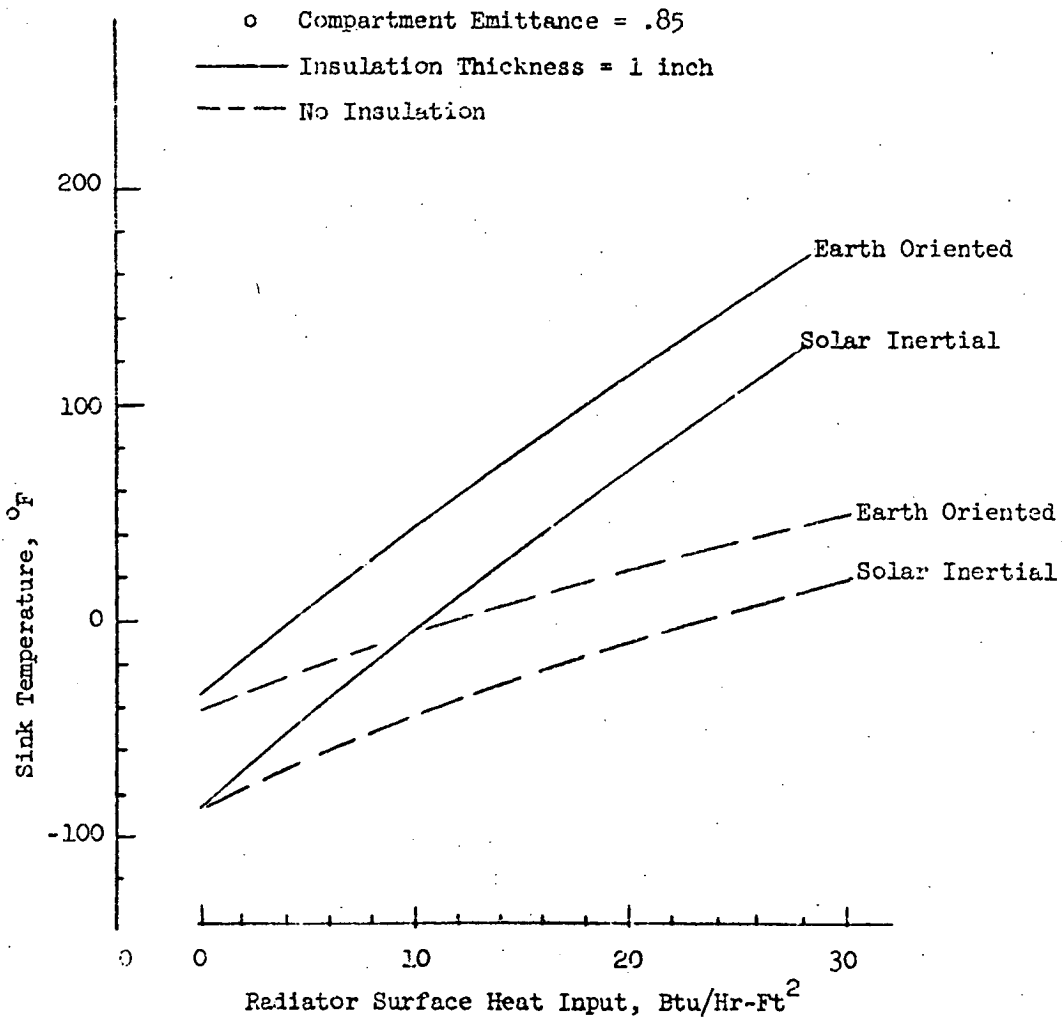


FIG. 9-5.9 - RADIATOR SYSTEM ANALYSIS PROCEDURE

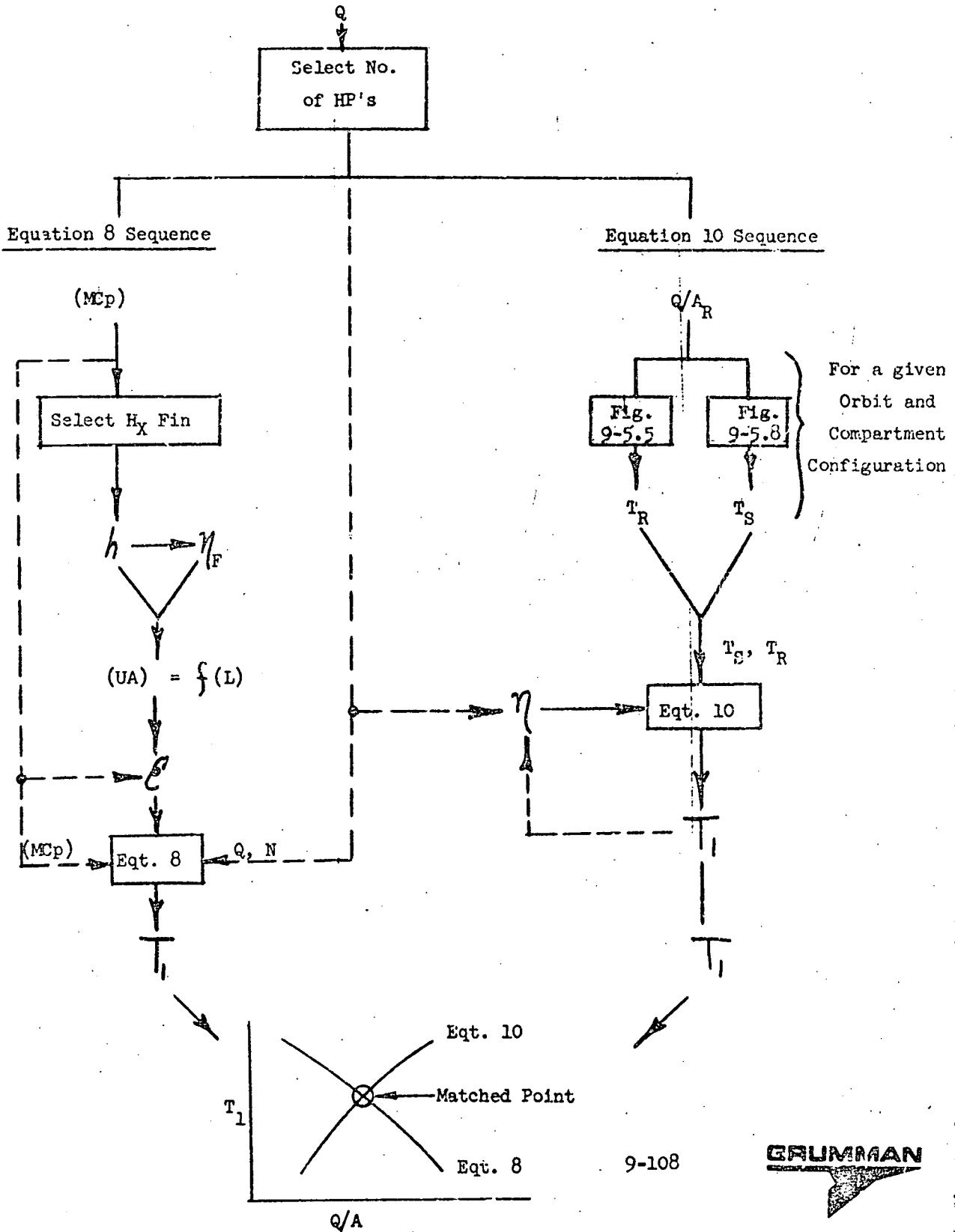


FIG 9-5.10 - RADIATOR FIN EFFECTIVENESS

η vs λ

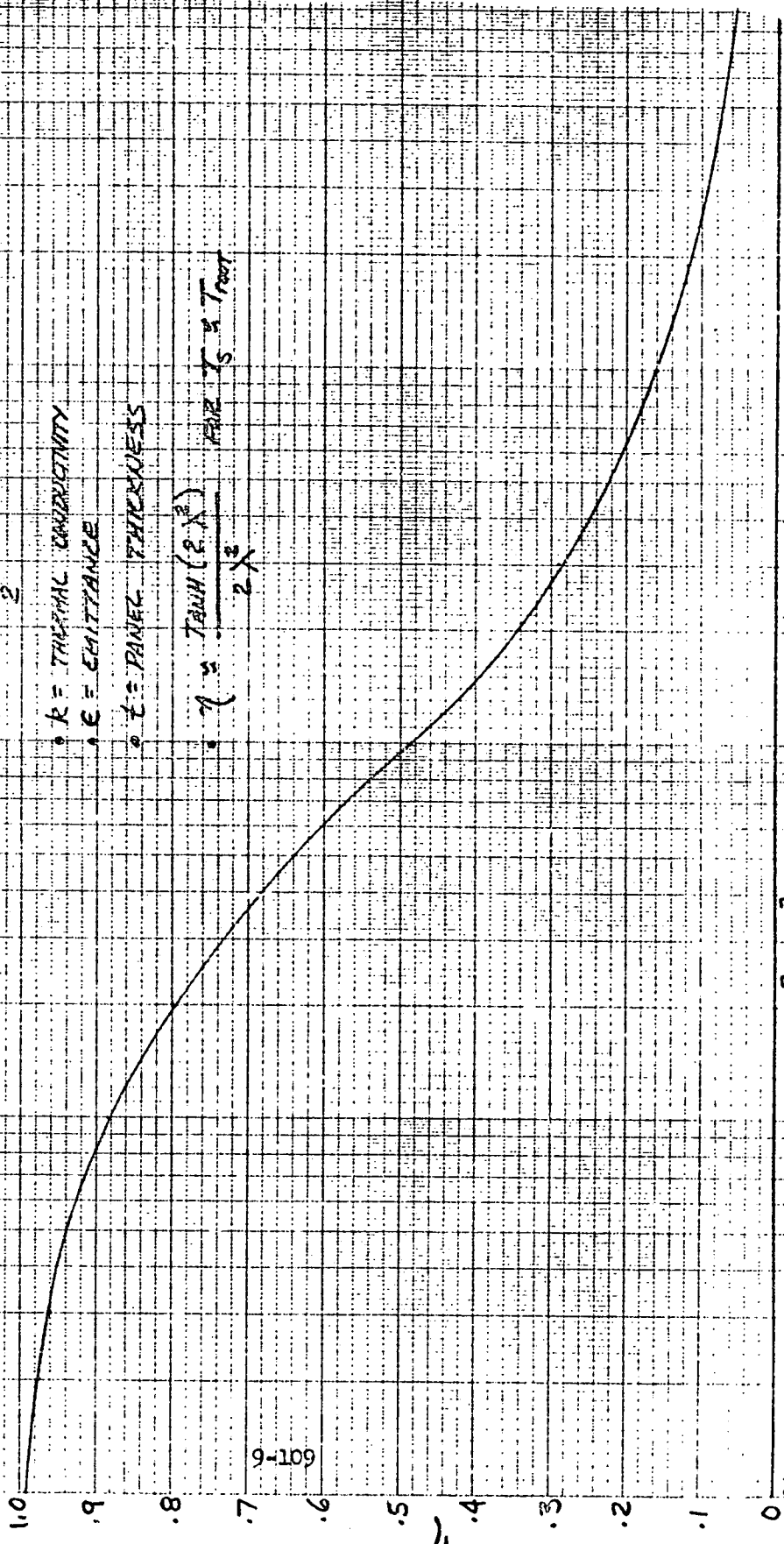
- $T_{\text{base}} = T_{\text{root}}$
- RECTANGULAR PANEL

$$\lambda = \frac{\text{Pitch} - \text{DD}}{2}$$

- k = THERMAL CONDUCTIVITY
- e = CHITTANCE
- t = PANEL THICKNESS

$$\eta = \frac{\text{TANH}(\lambda \sqrt{2k/e})}{\lambda \sqrt{2k/e}}$$

NOTE IS λ Troot



607-6

$$\lambda = \frac{e e L^2 T_{\text{root}}}{k t}$$

10.

to the root temperature, a situation which is conservative and typical of a compartment radiator application. This information was used to generate Figure 9-5.11, which gives the fin effectiveness as a function of root temperature for a 20 mil aluminum panel containing three heat pipes.

o Heat Pipes

Preliminary analyses showed that three heat pipes per panel would provide adequate thermal performance and sufficient margin to insure minimum landing gear temperatures of at least 0°F. Increasing the number of pipes for the sake of obtaining a higher panel fin effectiveness would increase weight while providing only nominal increases in system performance. For instance, as shown in Figure 9-5.14 using eight heat pipes per panel while maintaining 30% of full flow through the heat exchanger, increases panel heat dissipation by 1.2 Btu/Hr-Ft² (to 15.2 Btu/Hr-Ft²) and results in a landing gear temperature of 42°F as opposed to 32°F obtained with the three pipe panel. The eight heat pipe panel would weigh about 1 lb/ft² compared with .74 lb/ft² for the three heat pipe panel.

Each heat pipe in the system was designed to provide at least five times the required design point capacity to accommodate any changes in load without impacting the number or size of the heat pipes. At the 0°F (landing gear temperature) design point each panel must carry 47 watts, or about 18 watts per feeder heat pipe; the diode heat pipe must carry the entire 47 watt load. One-half inch ID pipes were selected since they give reasonable circumferential heat transfer area without excessive over capacity and they are easy to work with.

The artery designs were optimized, using Reference 13, for the specified lengths and working fluids (Freon-21 for the feeder heat pipes and ammonia for the diode) and resulted in spiral artery/tunnel wick designs being selected for both cases. The results, in the form of capacity versus operating temperature are presented in Figure 9-5.12. At 40°F a feeder heat pipe can transport 95 watts and the diode, 740 watts.

FIG. 9-5911 - COMPARTMENT RADIATOR FIN EFFECTIVENESS

- o $T_S/T_R \approx 1$
- o Aluminum Panel
- o Thickness = .020"
- o Surface Emittance = .90
- o Fin Length = 8.1"

3 Heat Pipes/Radiator

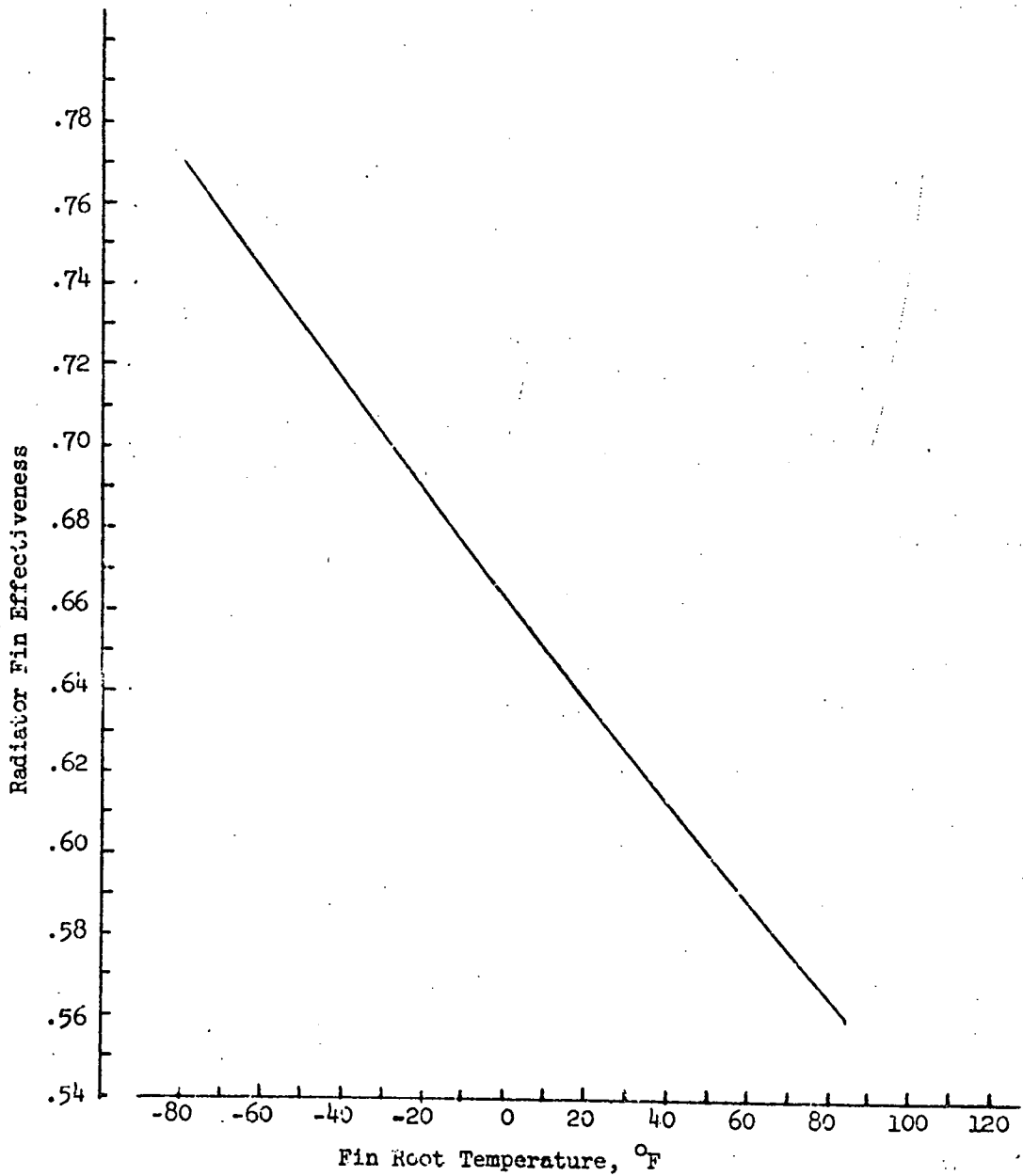
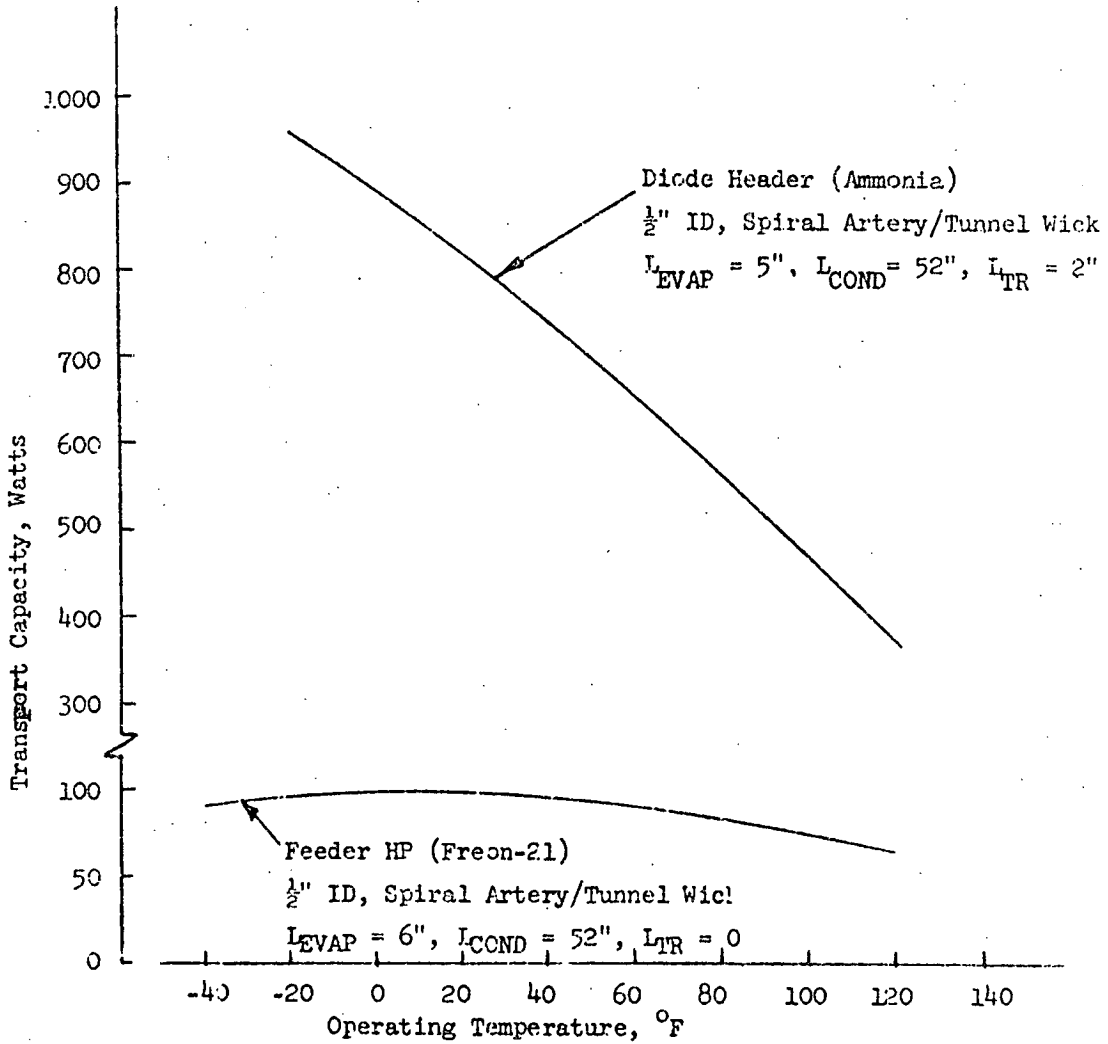


FIG. 9-5.12 - HEAT PIPE TRANSPORT CAPACITIES

- o Diode Header
- o Panel Feeder Pipe



o Heat Exchanger

The heat pipe-to-fluid heat exchanger consists of a finned annular passage which completely surrounds the centrally positioned evaporator of the diode. The methods and procedures in Reference 15 were used for the design and analysis, with surface designation 11.1 (11.1 fins per inch, $\frac{1}{4}$ inch high) selected as the fin configuration. This resulted in an exchanger annulus with fins, $.714 \text{ in}^2$ of free flow area, and a hydraulic diameter of $.1685 \text{ in}$. Its performance characteristics as a function of flow rate are presented in Figure 9-5.13.

o System Performance

To determine system performance, the analysis procedure previously cited was followed for a system using three heat pipes per panel and a five inch long heat exchanger. For a general system with N number of feeder pipes and a heat exchanger length, L, equation 8 becomes:

$$T_1 = T_{in} - \frac{(mC_p)}{Q} - \frac{Q}{N} \left[\frac{.0588}{L} + .084 \right]$$

which simplifies to the following equation for N = 3, L = 5 in.:

$$\frac{T_1 - T_{in}}{Q} = \frac{1}{Q(mC_p)} + .032$$

The above expression and equation 10 are plotted as a function of heat dissipation (Q/A) in Figure 9-5.14, the intercept points define the system performance at the indicated flow rates. As shown, a heat exchanger flow rate of only 3% of full flow will provide the necessary 10 Btu/Hr-Ft² needed to maintain a 0°F landing gear temperature (refer to Figure 9-5.5). However, by increasing the flow to just 30% of full flow the system will provide 14 Btu/Hr-Ft² and a minimum landing gear temperature of 32°F. Thus, a 30% flow rate was specified for the exchanger.

FIG. 9-5.13 - HEAT EXCHANGER PERFORMANCE CURVES

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

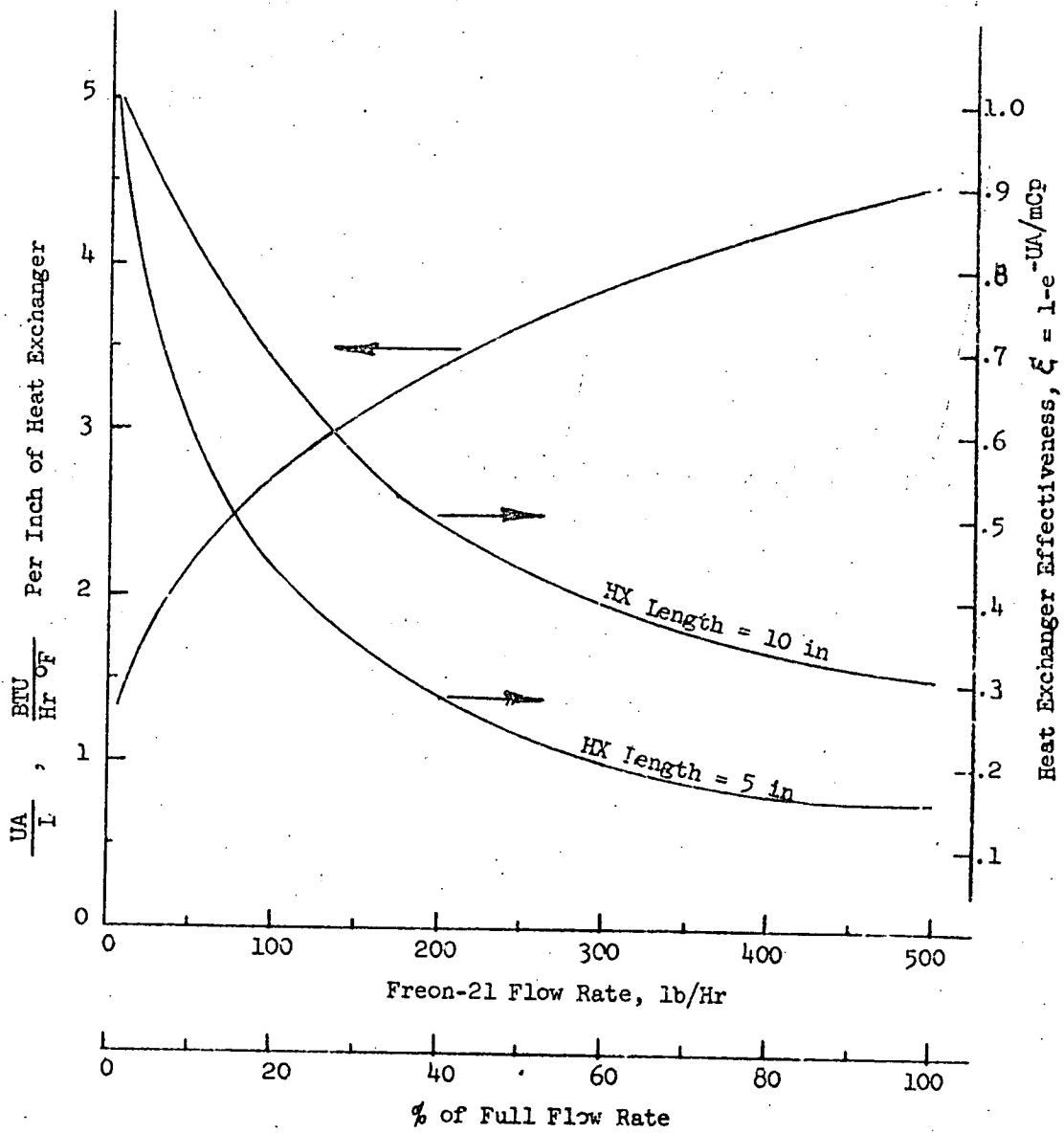
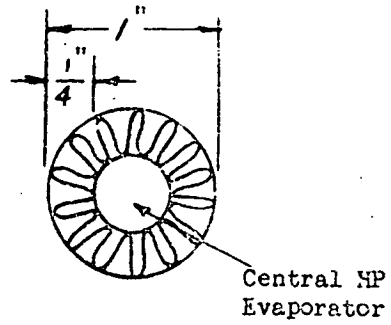
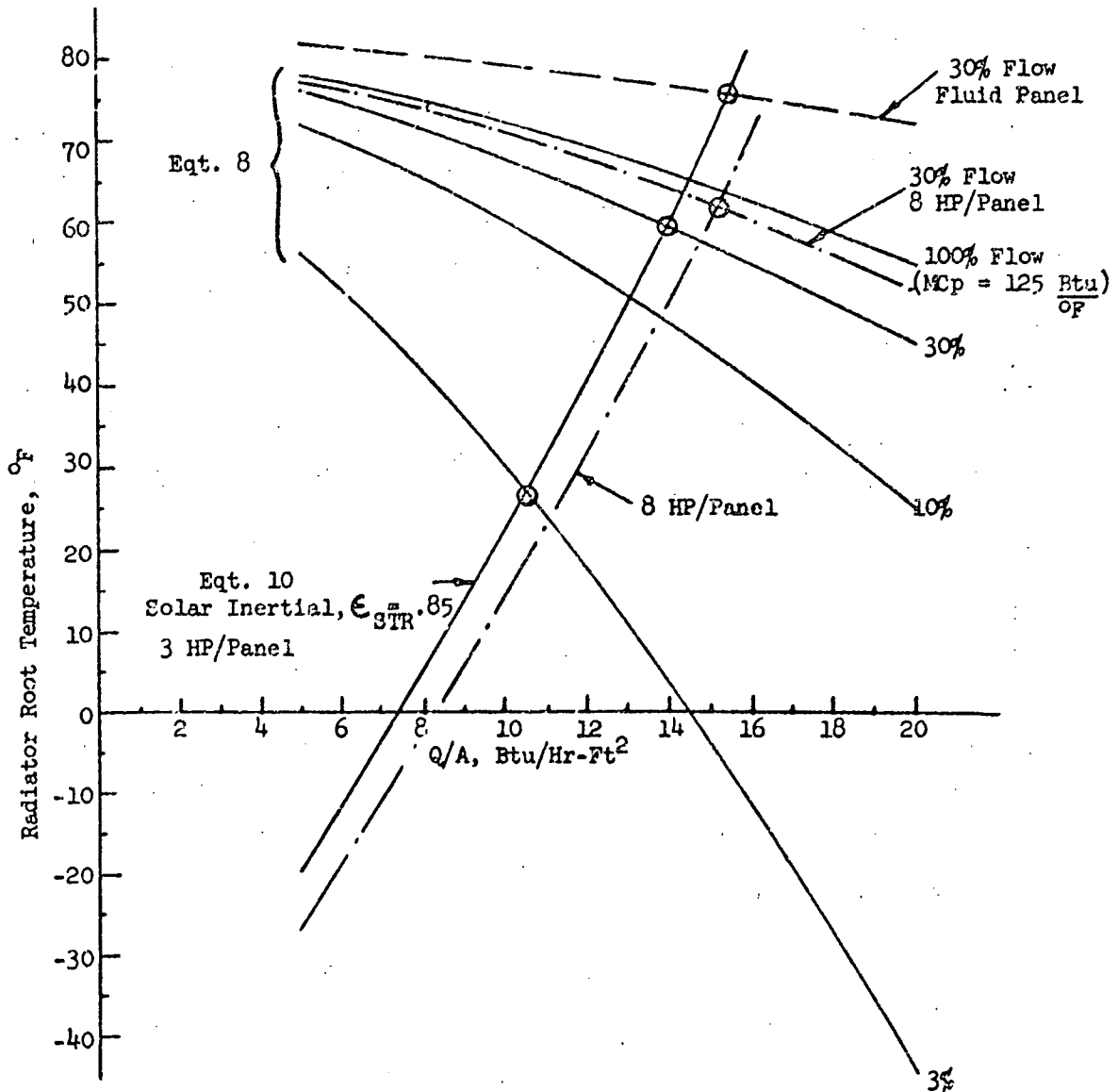


FIG. 9-5.14 - HP RADIATOR SYSTEM PERFORMANCE

- ○ 3 Heat Pipes/Panel
- Heat Exchanger Length = 5 in.
- - - ○ Series Pass Three Tube Fluid Radiator
- Note: Refer. to Fig. 9-5.5 to get minimum landing gear temperature vs. Q/A
- · - · - ○ 8 HP/Panel



The pressure drop through the exchanger can be expressed as (see Reference 15):

$$\Delta P = \frac{\left(\frac{\dot{m}}{A}\right)^2}{2 g_c \rho} \left[f \left(\frac{A_{HT}}{A}\right) + K_{Losses} \right]$$

where f is the friction factor, $\frac{A_{HT}}{A}$ is the ratio of heat transfer to free flow area $\left(\frac{A_{HT}}{A} = \frac{4L}{D_H}\right)$ and K_{Losses} = both core and exchanger

entrance and exit losses. For the 30% flow condition, $f = .015$ and $K_{Losses} \approx 2.42$ and the pressure drop (in psi) as a function of exchanger length (L) becomes:

$$\Delta P = (10^{-3}) \left[.435 L + 2.42 \right]$$

The ΔP thru a 5 inch long exchanger is $4.6 (10^{-3})$ psi. This can be stated in terms of a power penalty as

$$W = \left(\frac{\Delta P}{\rho}\right) \dot{m} = .446 (10^{-3}) \text{ watts}$$

which is of negligible impact on the system weight.

The system weight (wet) as given in Table 9-5.2, is .74 lbs/ft² of radiator area, or 23.6 lbs. total per compartment.

9-5.3 DESIGN

The design details of the heat pipe radiator system for the main landing gear compartment are shown in drawing SPL-103. Each compartment radiator panel has 16 ft² of surface area coated with a high emittance paint ($\epsilon = .90$), and three 1/2 inch I.D. "L" shaped aluminum heat pipes, approximately 6 inches by 52 inches. The long leg (condenser section) of the heat pipes are bonded (brazed or welded) to the 20 mil thick aluminum (6061-T6) panel every 16.8 inches, providing a radiator fin effectiveness which varies from 71% to 54% over the operating range of 40 to 115°F. The evaporator sections (short leg) are clamped to the condenser of the diode

ITEM	QTY	UNIT WEIGHT		TOTAL WEIGHT		DATA/COMMENTS
		LB.	LB./ft	LB.	LB.	
Surface Fla Panel	3	(.181)	lb/ft	4.51		Al Aly 6061-T6 .020T x 48" x 49.6"
Feeder H.P.		.031	lb/ft	2.53		Hollow core dia. = .15 no. of wraps 2
Fluid -freon 21		.008	lb/ft			No. of webs 9 stl
Artery		.031	lb/ft			.5" ID x 032 Al. Aly
Retainer		.111	lb/ft			020" Al 54 lb long typ. width 1.75"
Tube		.042	lb/ft	2.27		
Stiffeners				(.54)		
Feeder H/P to Diode	3	.071		.22		032" Al Aly
-Header Attachment	12	.007		.08		.125" Al Aly
Clamps	3	.037		.11		.032" Al. Aly 5.13" lg
Radius Block	12	.011		.13		#10.32 1 3/8" lg stl
Spacer Channels				.70		Ammonia
Bolts				(.17)		Corrug. Fin .010" Al .5" long
Diode Header	2	.410	lb/ft	.07		032 Al Aly
9 - Ammonia		.15	lb/ft	10		Al Aly
Heat Exchanger		.05		1.07		
Outer Jacket				11.79		0.74 lb/sq. ft.
End Closing Member						
10% Allowance Supt						
Edvre, Backup Strut	4		Total/Panel			
			Total for			
			Orbiter	47.2 lb.		



heat pipe with about 400 psi contact pressure. The attachment interfaces are milled flat surface (.45 inches wide, 64μ inch RMS) which can be either wiped with silicon grease or soft soldered to increase contact conductance. The soft solder would provide much better thermal conductance but at a sacrifice in joint maintainability; therefore, a wiped silicone grease interface was assumed for design and analysis purposes.

At the design point each panel heat pipe is required to carry about 23 watts; 68 watts are needed per panel. The artery design was optimized for the specified dimensions and Freon-21 as the working fluid. This resulted in a spiral artery/tunnel wick having a 95 watt capacity at 40°F. Details of the feeder heat pipes follow.

o Panel Feeder Heat Pipe (3 per panel)

Pipe:	Aluminum, 6061-T6
Grooves:	Circumferential
Diameter:	.500 in I.D. .564 in O.D.
Working Fluid:	Freon-21 (Charge = 52.6 gms)
Wick:	Spiral Artery/Tunnel Wick (.062" core)
Evaporator Length:	6 inches
Condenser Length:	52 inches
Overall Length:	58 inches (L-shaped)

The diode heat pipe shown in the drawing operates on the liquid blockage principle. When the temperature of the condenser exceeds that of the evaporator, the excess fluid which is held in a reservoir at the condenser end vaporizes. This excess fluid vapor then travels to the now cooler evaporator where it condenses and fills the vapor space, thereby "choking" the heat pipe.

The diode heat pipe is required to carry the entire 68 watt load of a radiator panel. It is a 1/2 inch I.D. aluminum pipe with a 5 inch evaporator, 40 inch condenser and 2 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. The liquid

blockage reservoir contains .9 in³ of ammonia, or enough to flood the evaporator and the transport section in the reverse mode. Details of the diode heat pipe follow.

o Diode

Pipe:	Aluminum, 6061-T6
Grooves:	Circumferential
Diameter:	.500 in I.D. .590 in O.D.
Working Fluid:	Ammonia, Charge = 21.6 gms (HP) + 9.4 gms (Reservoir)
Wick:	Spiral Artery/Tunnel Wick (.062" core)
Evaporator Length:	5 inches
Condenser Length:	39.6 inches
Overall Length:	49.4 inches (including reservoir)

o Reservoir (Liquid Blockage)

Material:	Aluminum, 6061-T6
Overall Length:	2.8 inches
Capillaries:	.060" dia. x 2.34 inches long
- number:	125
- volume:	.384 in ³ /in

To overcome difficulties associated with priming the vapor space in the blocked sections of the diode during ground testing in the shutoff mode, an alternate diode concept could be employed - the liquid trap diode. The liquid trap diode design uses a wick-lined reservoir located at the evaporator section of the diode. The trap wick is separate from, and does not communicate with, the wick in the operating portion of the diode.

In the normal mode of operation, the trap is dry, and the diode operates as an ordinary heat pipe, with the correct amount of working fluid for the wick design employed. When the liquid-trap end becomes the cold end of the pipe, condensation begins to occur on the liquid trap surfaces, as well as on the evaporator end internal surfaces. As liquid accumulates in the trap, the main heat pipe wick becomes underfilled causing a fairly rapid reduction in pumping capacity. Eventually, the main wick dries out completely and all the liquid is in the trap.

When the trap end again becomes the warm end, the trap acts as an evaporator until all the liquid is expelled. The trap volume requirement is based on the amount of liquid in the main heat pipe wick, and would appear to be most attractive for wicks having a relatively small volume. In contrast with the liquid blockage technique, it has very little dependence on vapor-space volume but takes longer to shutoff.

The trap reservoir is made by filling an aluminum cylinder with 200 x 200 aluminum mesh entrapping a void volume sufficient to hold the entire operating fluid charge. Details of a liquid trap reservoir designed for this application follow.

o Liquid Trap Reservoir

Material:	Aluminum, 6061-T6
Cylinder:	1.00 inch O.D., .035 wall
Mesh:	200 x 200 spiral wrap
Void Volume:	.64 in ³ /in
Required Length:	5.2 inches

Each heat pipe-to-fluid heat exchanger consists of a finned annular passage which completely surrounds the centrally positioned evaporator of the diode heat pipe. There are 30 aluminum fins in the annulus; each .010 inch thick and .250 inch high. They are aluminum brazed to the outer surface of the evaporator. The heat transfer length in the exchanger is 5 inches; a length consistent with the limiting design heat flux density of 25 watts/in² of heat pipe surface area, and the design point diode capacity. The finned passage of the heat exchanger is the same no matter what diode design is used. However, for the liquid trap diode, the exchanger would be modified slightly to surround the trap reservoir.

When installed in a compartment, the entire system, except for the radiating surface of the panel, would be encased in a fibrous insulation blanket to minimize heat losses and to insure predictable performance.

9-5.4 BASELINE

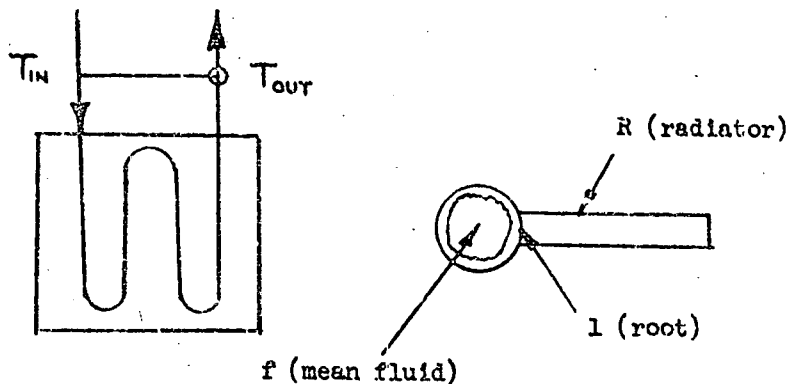
- Electrical Heaters -

For a system whose electrical power is not eventually picked up by the shuttle ECS, the weight penalty for each required watt is given as:

$$\frac{\text{lb}}{\text{watt}} = .200 + .235 (\% \text{ duty cycle})$$

Providing 14 Btu/Hr-Ft² with an 82% duty cycle results in a weight penalty of 1.61 lb/ft² of radiator.

- Fluid Panel -



The following relationships define the temperature throughout the fluid radiator system.

1. Heat transferred from fluid (Q):

$$Q = mC_p (T_{in} - T_{out})$$

- 2a. Fin root temperature (T₁):

$$T_1 = T_{in} - \frac{(T_{in} - T_{out})}{\mathcal{E}}$$

- 2b. Tube effectiveness (ε):

$$\mathcal{E} = 1 - e^{-\frac{\gamma_F hA}{mC_p}}$$

3. Mean fluid temperature (T_f):

$$T_f = T_1 + \Delta T_{IM} = T_1 + \frac{T_{in} - T_{out}}{\ln \left(\frac{T_{in} - T_1}{T_{out} - T_1} \right)}$$

4. Flow Equations:

Laminar

$$\frac{h D_H}{k} = 3.66$$

$$f = \frac{16}{N_R}$$

Turbulent

$$\frac{h}{C_p \dot{m}/A} = \frac{.023}{(N_R)^2} (N_{PR})^{-2/3}$$

$$f = \frac{8 (.023)}{(N_R)^2}$$

5. Radiator root temperature based on compartment heat balance:

$$\sigma T_1^4 = \frac{(Q/A)}{\eta \epsilon_R} + \sigma T_3^4$$

6. Radiator surface temperature (T_R)

$$T_R^4 = \eta T_1^4 + (1 - \eta) T_S^4$$

Assuming a series pass three-tube panel and 30% of full flow, the fin root temperature can be expressed as

$$T_1 = T_{in} - .0342 Q$$

This expression is plotted in Figure 9-5.14 as a dashed line, and can be compared with the heat pipe system. It shows a heat delivery capability of 15.5 Btu/Hr-Ft² which results in a minimum landing gear temperature, from Figure 9-5.5, of about 44°F and an average panel surface temperature of 62°F.

The pressure drop, expressed in general as:

$$\Delta P = \frac{\left(\frac{\dot{m}}{A}\right)^2}{2 g_c \rho} f \left(\frac{L}{D} + \frac{L}{D}_{Losses} \right)$$

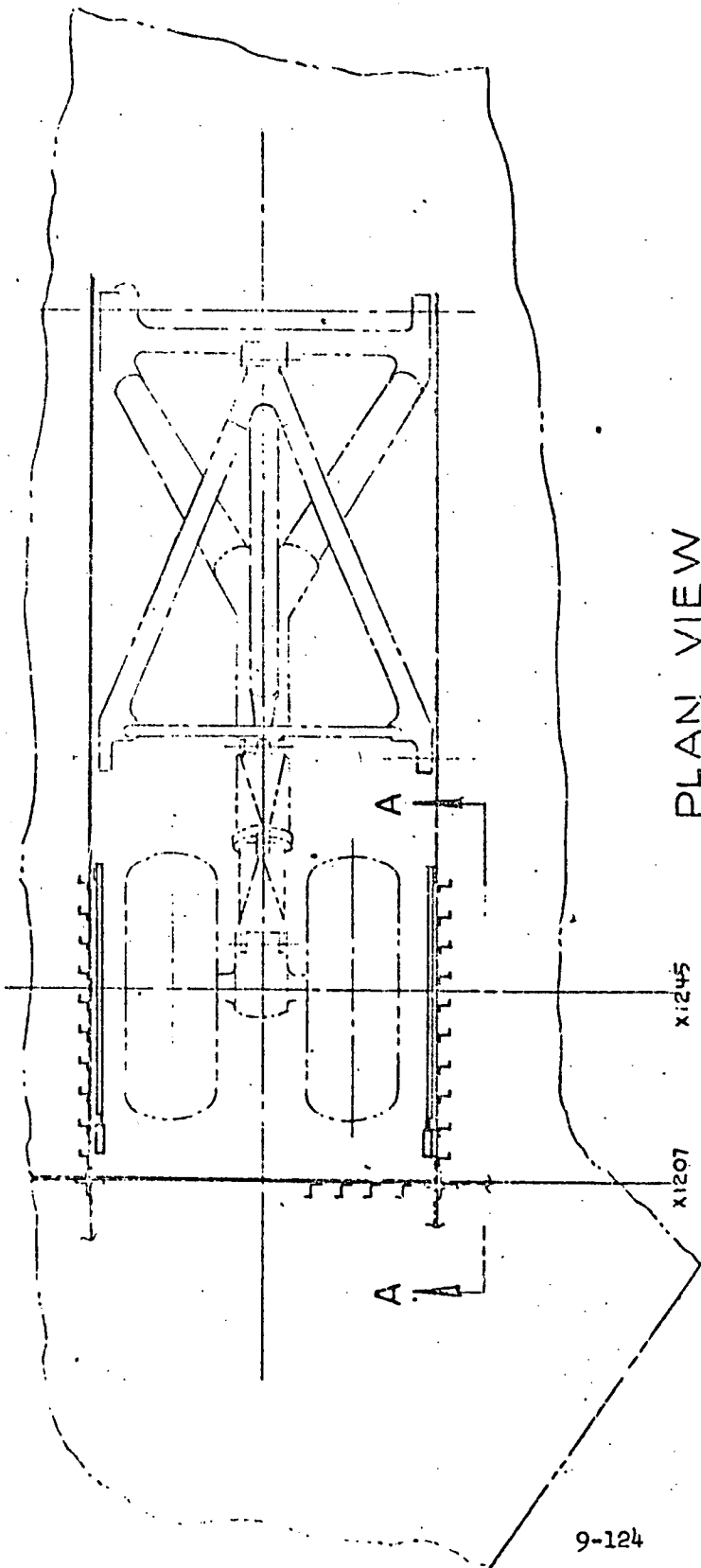
is calculated to be .0162 psi per panel, which results in a power penalty of .0016 watts/panel.

The panel weights are estimated from the design drawing (SPL-103) as follows:

$$\begin{aligned} WT &= \text{Panel} + \text{Stiffeners} + \text{Support Hardware} + \text{Tubes} + \text{Fluid} + \text{Bumpers} \\ &= 4.52 + 2.27 + 1 + 2.54 + .765 \\ &= 11.095 + 1/2 \text{ lb (for fittings)} \\ &= 11.095 \text{ lb} + 1 \text{ lb (for by-pass sensor and control)} \\ WT &= 12.095 \text{ lb or } .78 \text{ lb/ft}^2 \text{ of panel} \end{aligned}$$

9-5.5 CONCLUSIONS

The heat pipe radiator system, as described, is capable of maintaining the on-orbit main landing gear temperatures between 32°F and 117°F with a heat exchanger flow rate of 150 lb/hr, or only 30% of the maximum available rate. The system weighs .74 pounds per ft² of radiator surface vs. .78 lbs/ft² for a conventional fluid radiator, while the weight penalty for a system which uses electrical heaters would be 1.61 lb/ft². In addition to the power and weight advantages over the baseline systems, the heat pipe system is simpler and more reliable due to the inherent self-controlling features of its diode heater.

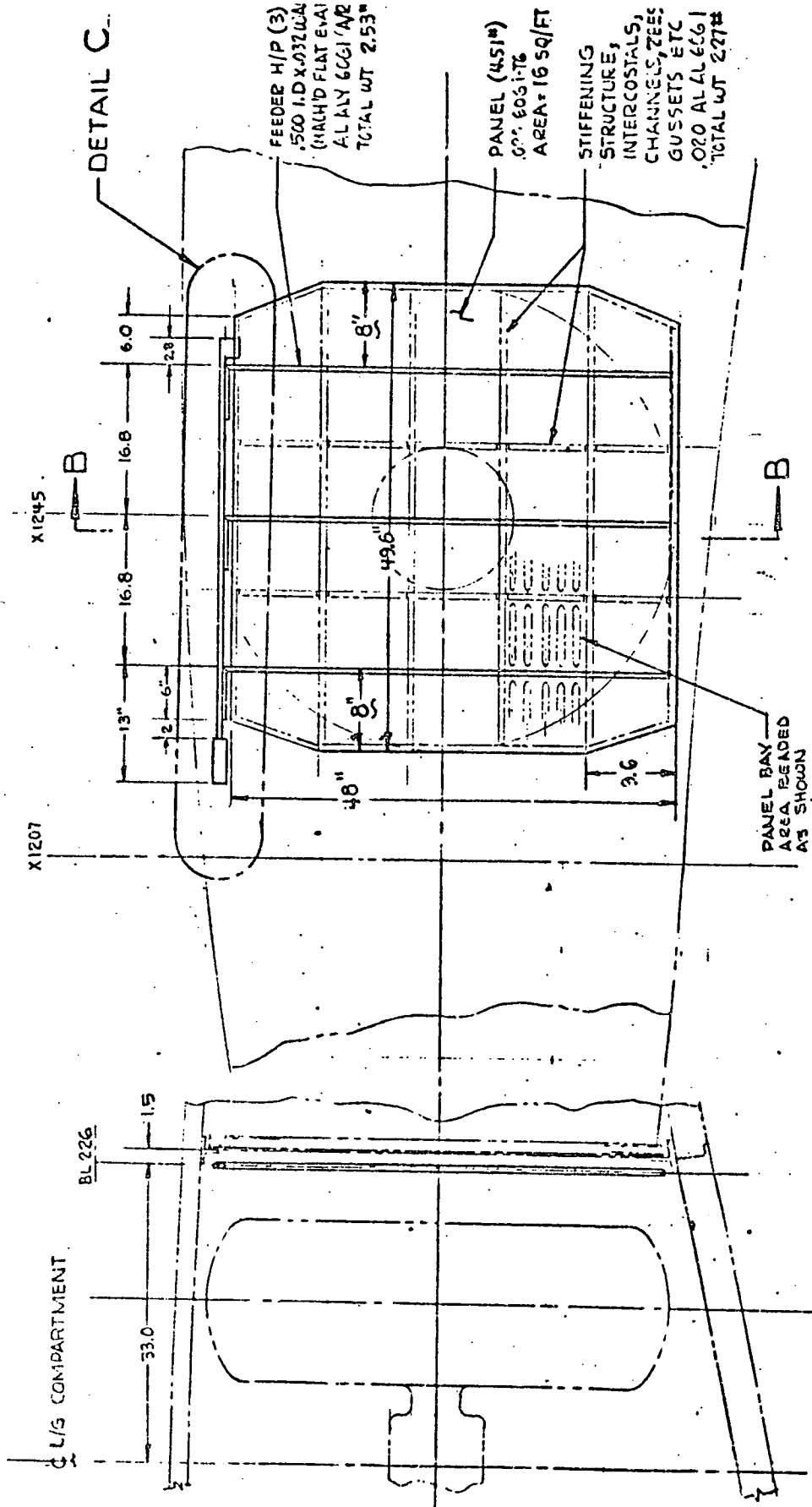


PLAN VIEW
 MAIN L/G (L.H.) GENERAL ARRANGEMENT
 (ORBITER VEHICLE)
 SCALE - 1/20

FIG. 9-5.15
 HEAT PIPE RADIATOR A
 MAIN L/G COMPARTMENT
 ORBITER
 SPL-103
 SHEET 1 OF 4



FIGURE 9-5.15



FEEDER H/P (3)
 .500 I.D X .032 WALL
 (NACH'D FLAT EVA)
 AL ALY 6061 /AR
 TOTAL WT 2.53#

PANEL (451#)
 .020 AL AL 6061
 AREA = 16 SQ/FT

STIFFENING
 STRUCTURE,
 INTERCOSTALS,
 CHANNELS, TEES,
 GUSSETS, ETC
 .020 AL AL 6061
 TOTAL WT 2.27#

SECTION A-A
 (SCALE 1/10)

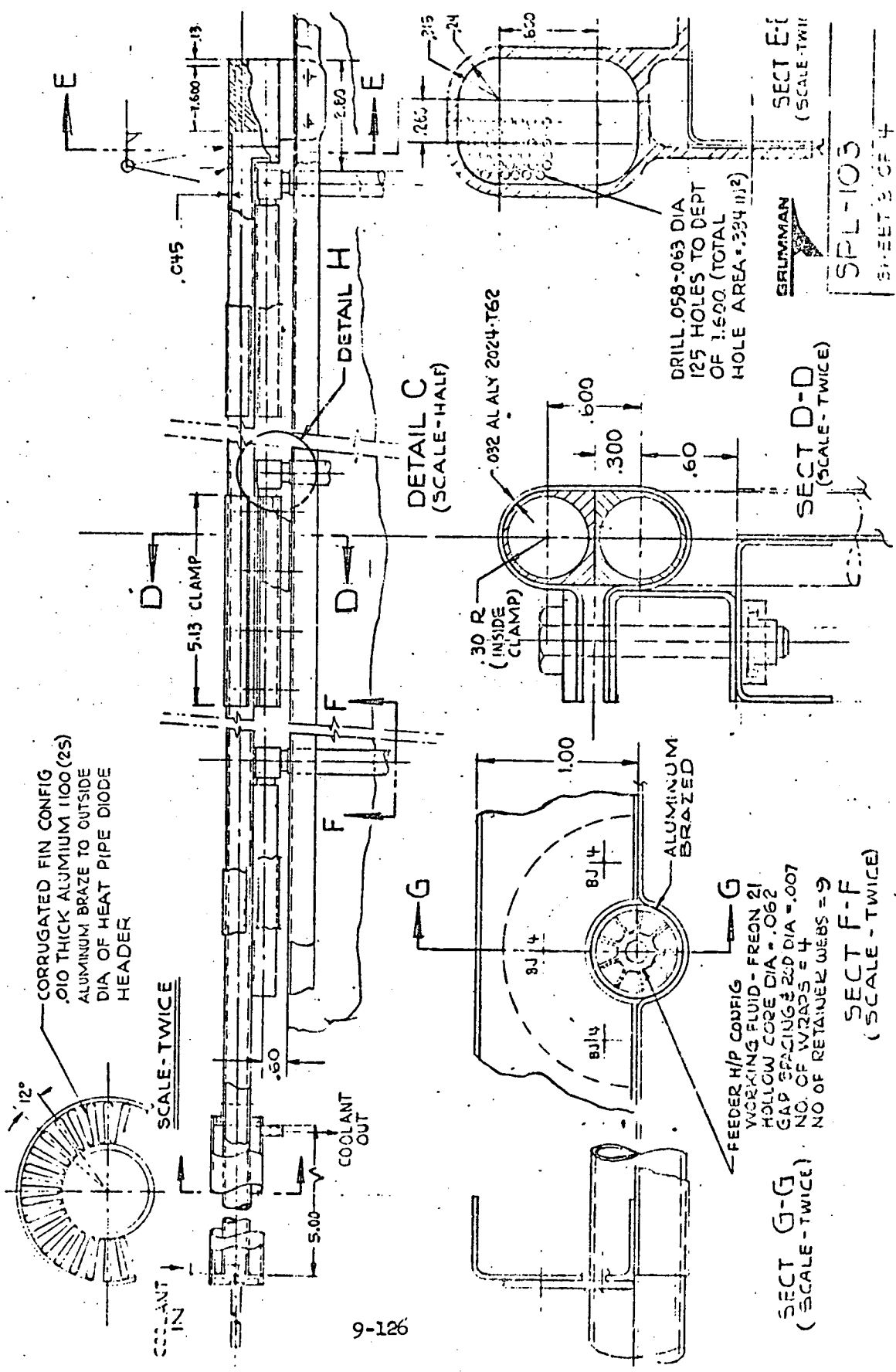
SECTION B-B



SPL-103
 SHEET 2 OF 4

BL 193

9-125



CORRUGATED FIN CONFIG
 .010 THICK ALUMIUM 1100 (2S)
 ALUMINUM BRAZE TO OUTSIDE
 DIA OF HEAT PIPE DIODE
 HEADER

SCALE - TWICE

COOLANT IN

COOLANT OUT

DETAIL C
 (SCALE - HALF)

.30 R
 (INSIDE CLAMP)

.032 AL AL 2024-T62

SECT G-G
 (SCALE - TWICE)

FEEDER H/P CONFIG
 WORKING FLUID - FREON 21
 HOLLOW CORE DIA = .062
 GAP SPACING 2RD DIA = .007
 NO. OF WRAPS = 4
 NO. OF RETAINER WEBS = 9

SECT F-F
 (SCALE - TWICE)

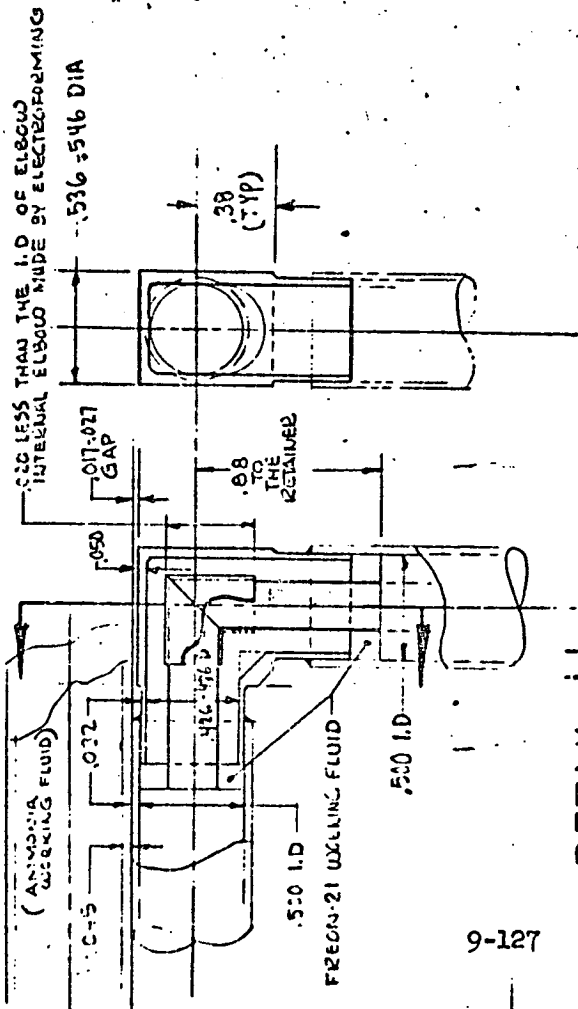
SECT D-D
 (SCALE - TWICE)

DRILL .058-.063 DIA
 125 HOLES TO DEPT
 OF 1.600 (TOTAL
 HOLE AREA = .234 IN²)

BRUMMAN

SECT E-E
 (SCALE - TWICE)

SPL-103
 SHEET 3 OF 4



DETAIL H
(SCALE - TWICE)

ESTIMATED WEIGHT BREAK-DOWN
FOR SINGLE PANEL CONFIGURATION

1- 16 FT ² SURFACE FIN PANEL AL ALY 6061T6 .020T x 48.0H x 49.6W	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 ATTACHMT 3 CLAMPS .032 AL ALY 2024-T62 ± .22 (12) RADIUS BLOCK .125 x .70 x .75 .076 (12) #10-32 BOLTS (1 3/16 LG) STL .132 (3) SPACER CHANNELS (5.13 LG) .032 AL ALY 2024-T62 .110	.54 #
5- DIODE HEADER (AMMONIA)	.70 #
6- HEAT EXCHANGER	.17 #
	<u>10.72 #</u>
10% ALLOWANCE, SUPPORT HDWARE, BACKUP/STRAKT	1.07 #
TOTAL WT FOR 16 SQ/FT PANEL	11.79 #
OR 16 SQ/FT AREA = .74# PER SQ/FT	

TOTAL FOR ORBITER VEHICLE
FOR FOUR PANELS = 47.2 #



SPL-103
SHEET 4 OF 4

SECTION 10

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°F to 150°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 safety 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.

Heat pipe systems intended for crew inhabited areas must use water as the primary working fluid because of stringent toxicity requirements. A slightly less toxic fluid, such as Freon-21, might be acceptable as a back-up if it exists in very small concentrations. Operating temperature and transport capacity requirements and material compatibility problems govern the selection of a working fluid outside the crew areas. Ammonia, Freon, methanol and acetone are all among the candidates. Only the unique requirements of a specific application and the working experience and practical knowledge of the contractor will determine the choice.

From a systems viewpoint, the external evaporator and condenser interfaces present the biggest problems in creating viable heat pipe thermal control systems. Interface conductances must be high to minimize thermal gradients as large gradients can easily compromise the benefits of a heat pipe system by requiring larger sink or source capacities to accommodate the required load.

Heat pipes which couple directly to a constant temperature, large capacity sink or source provide the most thermally efficient system since they result in the least overall temperature gradient from source to sink. In this case, the presence of the heat pipes does not affect its own operating temperature. With a non-uniform source such as a hot flowing fluid, the source temperature applied to the heat pipe evaporator varies from the fluid inlet to the fluid outlet; the lower outlet temperature being a function of the heat load transferred to the evaporator. Since the heat pipe evaporator section may be viewed as a heat exchanger between the fluid and the isothermal heat pipe vapor, the heat pipe vapor must operate at the less efficient fluid outlet temperature. With a non-uniform sink in contact with a heat pipe condenser, a HP system is similarly penalized by being forced to operate at the higher outlet temperature in order to reject its heat. This penalty is passed back through the system and results in a higher source temperature. When the system heat load requirements are not large, the resulting temperature penalties imposed by a non-uniform source or sink are small, and do not detract from the overall benefits of using heat pipes. But when carrying large loads, a heat pipe system coupled to a non-uniform source or sink can result in prohibitive weight and performance disadvantages.

From a heat pipe point of view, the most important parameters for successfully integrating heat pipes with a thermal control system are high evaporator and condenser film heat transfer coefficients. They are a must to minimize the overall temperature drop across the heat pipe. Circumferentially grooved heat pipe envelopes (up to 200 grooves/inch) have proven the most successful, to date, at satisfying this important need. Depending on the working fluid, evaporator film coefficients from 1500 - 2700 Btu/Hr-Ft² and condenser coefficients from 2000 - 3500 Btu/Hr-Ft² have been verified by test (Reference 18).

C. 5

SECTION 11
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 12

REFERENCES

1. "Alternate Space Shuttle Concepts Study," Grumman Aerospace Corporation Final Report (MSC 03810), July 1971.
2. "Space Shuttle System Space Thermal Control Study," North American Rockwell Study Report (SD 70-600-34), January 1970.
3. "Space Shuttle Phase B Final Report, Technical Summary," McDonnell-Douglas (MDC E308), June 1971.
4. "Development of Thermal Control System Design Concepts for Manned Space Station Applications," Grumman Final Report SPR-145-12, March 1971.
5. "Tentative Evaluation Procedure of Thermal Control System Candidates for an Earth Orbiting Space Station", SMO-078-05, April 14, 1970.
6. Evaluation of New Approaches to Packaging of Power Assemblies for Space Use, Final Report (Phase I), Contract NAS 9-10411.
7. Thermal Model Handbook, Apollo Lunar Module (LM) - Vol. III, Propulsion Systems, Section 5.1.
8. Grafton, John C., Transient Heat Transfer Analysis of Systems Involving Fluid Flow, NAA Report ASD-TDR-63-260.
9. "Electrical Load Analysis for GAC Shuttle," B35-160 MD-79, October 14, 1971.
10. A Joint Heat Transfer Data Critical Study and Design Guidelines, G.E. Report No. 71 SD4241.
11. Phase Change Materials Handbook, Lockheed Missiles and Space Co., Huntsville, Alabama, Sept. 1971, prepared for NASA-MSFC, Contractor Report NASA CR-61363.
12. Research and Development Study on Thermal Control by Use of Fusible Materials, Northrop Space Laboratories, Hawthorne, California, April 1966, prepared for NASA-MSFC, Contractor Report NAS 8-11163.
13. "Description of Heat Pipe Design Program," HP-38, January 1971.
14. Grumman Aerospace Corp. Shuttle Program Memo No. B31-155 M09, "Preliminary Design of EOS Environmental Control System Heat Transport Section," June 1971.

15. W.M. Kays and A.L. London, Compact Heat Exchangers, Second Edition, McGraw-Hill, Inc., 1964.
16. "Reusable Surface Insulation TPS Weights," B35-197 MO-31, December 18, 1971.
17. PDM-134V-2, "Selection of Baseline Design Coldplate for JIFDATS K-Band Transmitter on the OV-1 Relay Aircraft", dated 5 January 1972.
18. F. Edelstein, R. Kosson, R. Hembach, and M. Tawil, "A Tunnel Wick 100,000 Watt-inch Heat Pipe," AIAA paper 72-273, presented at the AIAA 7th Thermophysics Conference, April 1972.
19. Kreith, Frank, Principles of Heat Transfer, 2nd Edition, International Textbook Company, 1967.
20. "Test Results of a Compact Heat Exchanger for Use as a Heat Pipe Radiator Evaporator-Header Connection," Grumman A.D. Space Radiator Project Tech. Memo. 14, 29 August 1969.

APPENDIX A-1

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.1

APPLICATION: TPS Leading Edge to lower & equalize temperatures

SUBSYSTEM: Structure

LOCATION: Wing, Vertical Fin

BASELINE T/C SYSTEM: Carbon/Carbon

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Reentry	Fin: 2600	11.8 (10 ⁴) BTU/HR FT ²	<14' (LE to side panels)
	Wing: 2780-3180	15.2-23.2 (10 ⁴) BTU/HR FT ²	<65' (entire L.E.)

DESCRIPTION: (a) One application is an isothermolizer HP buried in the carbon/carbon and running along the leading edge. The working fluid could be contained in integral channels or in separate pipes placed in these channels.

(b) Another application is to transport the heat from the LE to the upper wing panels (for the wing) or side panels (for the fin).

ADVANTAGES

1. Lower temperatures, especially below 2900°F, mean a much longer life for carbon/carbon
2. Application of HP's minimize impact of uncertainties in predicted surface temperatures.

DISADVANTAGES

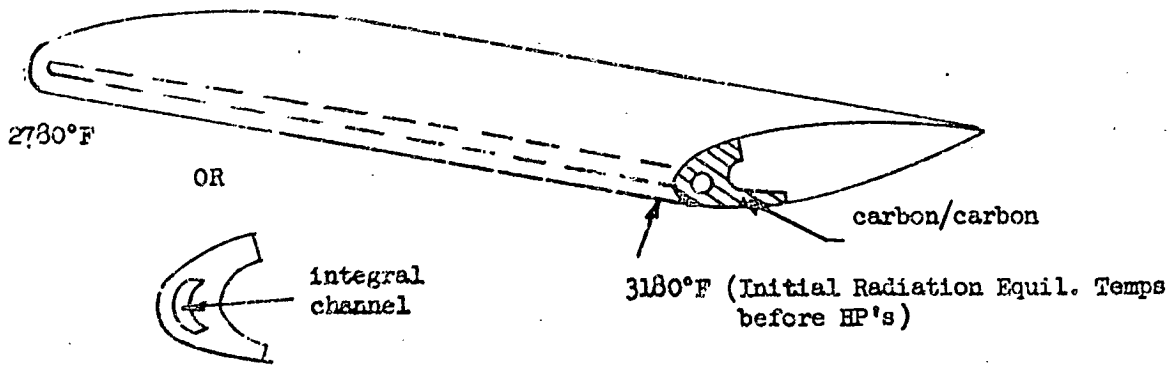
1. Materials compatibility problems at high temps (>1700°F) for liquid metal HP's.
2. Possible weight penalty.

COMMENTS:

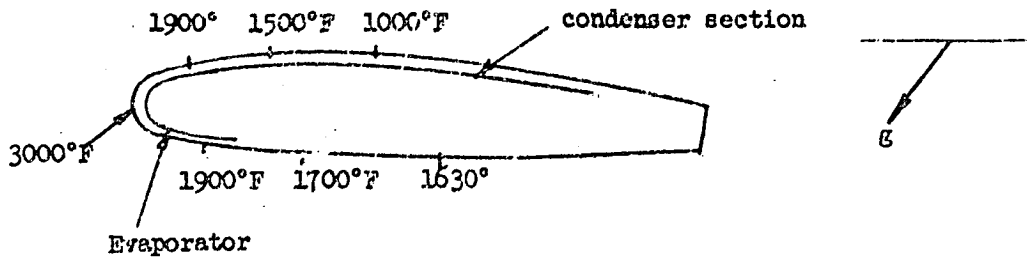
Requires liquid metal heat pipes. Pipes can be installed so that inertia helps pump fluid from cond. to evaporator.

HEAT SOURCE/SDIK IDENTIFICATION - CONTINUATION SHEET NO. 1

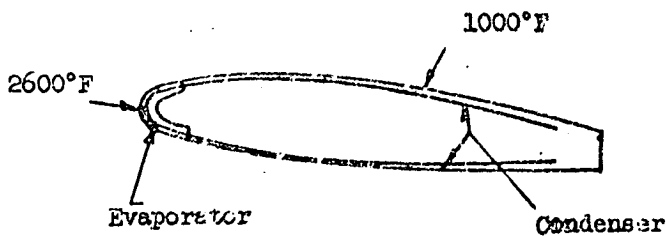
a. Wing LE Isothermalizer



b. Wing LE to Wing Panels



c. Fin LE to Side Panels



SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.2

APPLICATION: Landing Gear

SUBSYSTEM: Structure

LOCATION: Wheel Wells, Main and Fwd

BASELINE T/C SYSTEM: Fluid Loop Radiators or Electric Heaters

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On-Orbit	-65 to 270	1025 per compartment	Rad. HP: < 3
Reentry			Headers: < 10

DESCRIPTION: HP radiating surfaces line wheel well and transfer heat to tire compartment and hydraulics during on-orbit phase. Heat source is internal structure or ECS heattransport section. A diode heat pipe, permitting flow only toward wheel well, prevents leakage back into the heat transport section during reentry.

ADVANTAGES

1. No electrical power req'd.
2. No moving parts or circulating fluids
3. No fluid connections.
4. Self-controlling.
5. Improved reliability over fluid radiator since it can survive accidental impact of foreign object during landing.

DISADVANTAGES

1. May require flexible HP joint.

COMMENTS:

Protection from reentry can be obtained by increasing insulation thicknesses surrounding the compartments.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.3

APPLICATION: Avionics HP Circuit

SUBSYSTEM: Avionics

LOCATION: Cabin

BASELINE T/C SYSTEM: Fluid Loop (H₂O for heat transport section, F21 for heat rejection section).

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
All	40 to 90	6800	<12 (each pipe)

DESCRIPTION:

Series- parallel connection of HP's to transfer heat from electronics to fluid loop or to space radiator.

ADVANTAGES

1. No fluid couplings.
2. No moving parts or rotating machinery (no noise).
3. No electrical power req'd.
4. Offers high design flexibility during early design phases. No delicate flow splits or pressure drop analysis required for each loop change.

DISADVANTAGES

1. Possible high ΔT 's across HP interface couplings, resulting in lower radiator temp. or higher eqpt. temp.

COMMENTS:

Alternate sink necessary for other than on-orbit condition.

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)

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
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 Sensor Communications equip. (L Band Tacan, C band altimeter, L band transponder, VHF ATC transceivers)

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.5

APPLICATION: Air Transport Rack (ATR) Commercial and Military Aircraft Equipment

SUBSYSTEM: Avionics

LOCATION: Pressurized Cabin

BASELINE T/C SYSTEM: Equipment is off-the-shelf air cooled design

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
All	40 to 120	5000 to 6150 (Total Equipments)	

DESCRIPTION:

Modular HP to air heat exchanger elements within an air circulating enclosure remove heat from forced air passing over and through the electronic boxes. Eventual sink could be space radiator; (on-orbit) or boilers.

ADVANTAGES

1. Eliminates flow balancing problems associated with fluid Hx.
2. Easy assembly and maintenance.
3. Equipment growth flexibility.
4. Standardized approach for adaptation of ATR equipment cooling.
5. Low cost commercial equipment can be used in space with minimum design impact.
6. No fluid connections.

DISADVANTAGES

1. Leak resulting in loss of air would cause cooling failure, must provide backup for critical eqp't.

COMMENTS:

Baseline T/C system for air cooled equipment would also need backup in case of depressurization.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.6

APPLICATION: Flight & Voice Recorder

SUBSYSTEM: Avionics

LOCATION: Vertical Fin

BASELINE T/C SYSTEM: ECS Fluid Loop

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
All	0 to 120	140	

DESCRIPTION:

HP attached to fluid boiler or phase change heat sink provides high temp. heat rejection. Variable conductance heat pipe attached to internal structure controls low temp. condition.

ADVANTAGES

1. Eliminates long run of ECS line to this remote location.
2. Self controlling.

DISADVANTAGES

COMMENTS:

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.7

APPLICATION: HP Radiator W/HP Header

SUBSYSTEM: ECS

LOCATION: Inside Cargo Bay Doors

BASELINE T/C SYSTEM: F-21 Radiator and Header; Valve stagnation at Radiator and Regeneration and bypass control in radiator loop.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On-Orbit Only	0 to 80 (radiator)	23,900 (Min) to 48,800 (Max) HiTemp Elect: 3700-4100 Heat Transport: 10500-22300 Load Fuel Cell Load: 9700-14400	Rad HP: 6" evap, 6' cond. Header: 8' evap 11' cond 3' adiabatic

DESCRIPTION:

Each radiator panel will consist of about 22 fixed conductance HP's on 6 inch centers. Max. panel capacity is 3850 BTU/HR (55 BTU/HR FT²) at 70°F. Each HP has a 6" evap. and a 6 ft. condenser section at right angles. The radiator HP header is a cold reservoir variable conductance HP capable of 3850 BTU/HR. Radiator HP fluid is F-21. Header fluid is either F-21 or ammonia. HP Header can be connected directly to a HP circuit or thru a Hx to a fluid loop.

ADVANTAGES

1. Self controlling. Does not require sensor feedback & electro-mechanical regulators.
2. Eliminates moving parts & rotating machinery in heat rejection loop.
3. Reduce degradation caused by meteoroid & impact penetrations.
4. No electrical power required.
5. No fluid connections.

DISADVANTAGES

1. HP header needs flexible HP joint that must bend thru 135° of rotation.
2. Large Δ T's from header condenser to radiator surface. They can vary from 10°F to 37°F depending on contact conductance and working fluids.

NOTE: When interfacing with a fluid loop, the flexible HP joint can be eliminated by using a flexible coupling in the fluid lines.

COMMENTS: Q Absorbed = 60 BTU/HR FT², α_g = .20, ε = .90, η_{Fin} = .90

700 ft² of radiator area divided into 10 equal area panels. No inertia vector problem. Each panel designed for 55 BTU/HR FT² heat rejection at T_{Rad} = 70°F. Panels are made of .020" thick aluminum (2219).

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.8

APPLICATION: HP Radiator with Integral HP/Fluid Header

SUBSYSTEM: ECS

LOCATION: Cargo Bay Door

BASELINE T/C SYSTEM: F-21 Radiator and Header

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
--------------------------	----------------------------	-------------------------------	----------------------------------

See A-1.7 for Details

DESCRIPTION:

Same as A-1.7 except that it uses a flexible coupling in the fluid lines and an integral fluid to HP header Hx on the radiator panel.

ADVANTAGES

1. Panel self-controlling.
2. No circulating fluid in radiator - reduces fluid connections.
3. Reduce degradation caused by meteoroid & impact penetration. No need for meteoroid bumpers.

DISADVANTAGES

COMMENTS:

This application can also be adapted to radiator configuration of A-1.10.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.9

APPLICATION: ECS Cold Rail

SUBSYSTEM: ECS

LOCATION: Cabin, Electronic Bays

BASELINE T/C SYSTEM: None defined for high power density equipment.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
All	40 to 90	40 watts/inch	1.5

DESCRIPTION:

HP augmented cold rails diffuse high power densities over longer length before passing heat to fluid tubes. It reduces higher densities to an effective 3 watts/inch at fluid loop interface.

ADVANTAGES

1. Use existing flange designs and fluid flow rates.
2. HP augmented cold rails can easily accommodate unforeseen power increases and minimize impact on fluid temperatures.
3. Required for state-of-the-art eqpt. with power densities of 40 w/inch.

DISADVANTAGES

COMMENTS:

Extensive application possibilities to high density cold rail mounted equipment such as is expected in the power conditioning and distribution avionics.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.10

APPLICATION: HP Radiator for fuel cell cooling

SUBSYSTEM: Power

LOCATION: Nose Module

BASELINE T/C SYSTEM: Fuel cells in heat rejection system, before radiators. They are coupled to the F-21 loop via a liquid to liquid Hx.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On-Orbit Only	0 to 80 (Rad)	9700 - 14400	Rad HP: 6" evap, 6" cond Header: 8' evap 11' cond 3' adiabatic Feeder HP: 39' total length

DESCRIPTION:

Self-contained fuel cell coolant is coupled to its own HP radiator panels via a HP header to fuel cell coolant Hx. The fuel cells are used during all mission phases and would require an alternate sink when the radiators are not deployed. This requirements also exists for the baseline system. The alternate sink could be a cryogenic Hx or water boiler.

ADVANTAGES

1. Self controlling. Doesn't require sensor feedback & controls.
2. Eliminates moving parts & rotating machinery in heat rejection system.
3. No electrical power required.
4. No fluid connections at radiator.

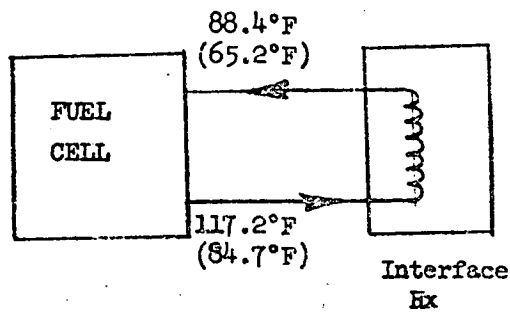
DISADVANTAGES

1. HP header needs flexible HP joint that must bend thru 135° of rotation.
2. Excessive ΔT 's from header condenser to radiator surface. They can vary from 10°F to 37°F.
3. Alternate sink requirement would mean development of a HP/LH₂ or HP/CH₂ Hx or a HP/water boiler Hx.

COMMENTS:

Requires HP to fluid Hx since fuel cell has its own self-contained cooling fluid. Requires alternate sink for fuel cell during boost to orbit and descent to landing phases of mission. These are also required by baseline system.

HEAT SOURCE/SINK IDENTIFICATION - CONTINUATION SHEET NO. 1



T_1 = temperature at 14,400 BTU/HR (high load)
(T_2) = temperature at 9,700 BTU/HR (low load)

FD-302

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.11

APPLICATION: Air Breathing Engine Compartment

SUBSYSTEM: Propulsion

LOCATION: Mid Module

BASELINE T/C SYSTEM: Electric Heaters

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	-30 to 150°	1020/compartment	

DESCRIPTION:

HP radiators lining compartment walls. Application is similar to landing gear wells.

ADVANTAGES

1. No electrical power req'd.
2. No moving parts or circulating fluids.
3. No fluid connections to leak.
4. Self controlling.
5. Improved reliability over fluid radiator panels.

DISADVANTAGES

1. May require flexible HP joint.

COMMENTS: Much of the air breathing engine components are designed around present MIL specs -65 to +160°F non-operational; providing this environment would permit max utilization of existing system component hardware.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.12

APPLICATION: OMS LH₂ Boiloff

SYSTEM: Structure

LOCATION: Nose Module

HEAT T/C SYSTEM: LH₂/fluid heat exchanger

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°R)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Descent	45°R - 500°R	90000	4
Launch		61000	
Ferry		50000	

DESCRIPTION:

Use Boiloff as heat sink in a HP/cryogenic Hx for 1) electronic equipment
2) ground & boost cooling of EC/LSS

ADVANTAGES

1. Convenient sink

DISADVANTAGES

1. Req'd boiloff greater than what's available it requires circulating LH₂ at about 50-60 lb/Hr. It would effectively amount to carrying extra propellant, 50 to 100#. (which is also required in baseline design).
2. Requires development of cryogenic/HP Hx.

REMARKS:

1840 ft³ of LH₂ in tank
Boiloff ≈ 1.5 lb/hr.

Cryogenic/HP Hx requires heat pipes to be used in the heat rejection system - this may not be the case.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.13

APPLICATION: High Intensity Lights - cool base.

SUBSYSTEM: Avionics

LOCATION: Cargo Bay & Exterior

BASELINE T/C SYSTEM: None established

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	400	Up to 1700	1-2

DESCRIPTION:

HP's applied to light socket provides cooling for lamp base in a vacuum environment. Structure serves as sink.

ADVANTAGES

1. Increase lamp life.
2. Increase lamp reliability.

DISADVANTAGES

1. May be slight weight penalty.

REMARKS:

Payload manipulator lights (500 w ea.) Target Illuminator (300 W) require 100 to 200 watts of cooling.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.14

APPLICATION: Battery

SUBSYSTEM: Avionics

LOCATION: Nose Module

BASELINE T/C SYSTEM: F-21 ECS loop

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Emergency Power	90°	3 KW-Hr. or 72000 BTU For 5 Hr. \approx 15000 BTU/HR	12

DESCRIPTION:

- a) HP's to isothermize heat rejection surfaces on cells. b) HP's connected to expendable sink, thereby eliminating fluid loop connection.

ADVANTAGES

1. No fluid connections.
2. Lower operational temperature gradients.

DISADVANTAGES

1. Weight of HP's & expendables.

COMMENTS:

Emergency use only.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.15

APPLICATION: Tracking Radar

SUBSYSTEM: Avionics

LOCATION: Cargo Bay

BASELINE T/C SYSTEM: Conduction coupling between electronic & antenna surfaces which in turn radiate to space.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	45 to 160	360	

DESCRIPTION:

Diode HP's coupling electronics to antenna surfaces provide good path to space during high temp. conditions, poor coupling during low temp. conditions.

ADVANTAGES

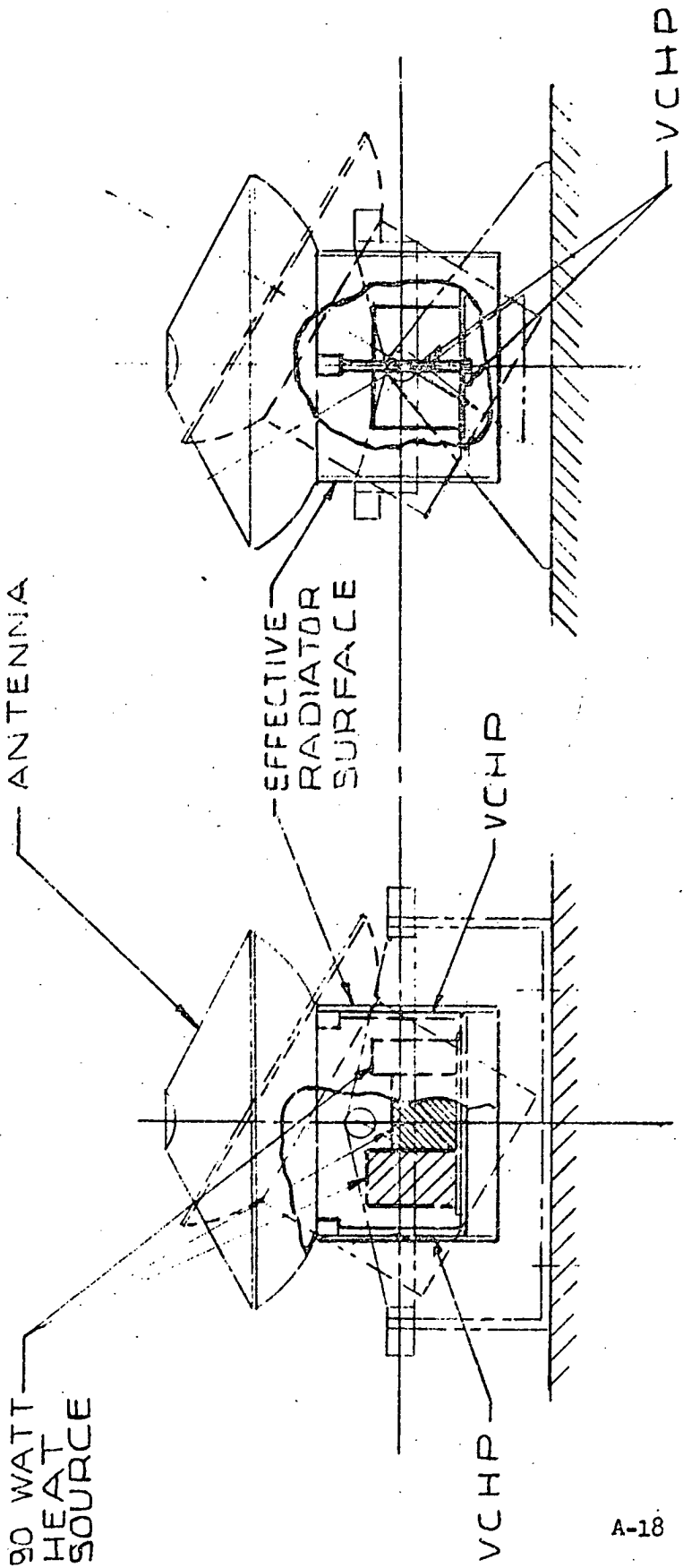
1. More efficient (less weight) than pure conduction coupling.
2. Diode feature - High pipe conductivity during operating temperature levels and low temperature cut-off minimizes the need for delicate thermal coatings and heater control loops.
3. Improved temperature regulations results in improved reliability.
4. Less heater power required.

DISADVANTAGES

1. Internal to manufacturer's hardware.

COMMENTS:

Thermal Control tied closely with manufacturer.



ELEVATION VIEW

TRACKING RADAR

28

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.15

APPLICATION: Fluid Evaporator

SUBSYSTEM: ECS

LOCATION: Fwd Module

BASELINE T/C SYSTEM: F-21 Heat Exchange with water

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
All above 100,000 ft.	40 to 120	40000 (max)	Up to 30

DESCRIPTION:

Multi-pass HP's transfer heat to expendable water.

ADVANTAGES

1. Eliminates running long ECS lines to out of the way sources.

DISADVANTAGES

1. Needs development of fluid to HP Hx

COMMENTS:

Required for ground support cooling. Need depends on existence of HP circuit which might not be the case.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.17

APPLICATION: Fuelage TPS - External Tank Inteference Heating Area

SUBSYSTEM: Structure

LOCATION: Upper Fwd. Mid Module

BASELINE T/C SYSTEM: Haynes 188

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Boost	1850	3.9 (10 ⁴) BTU/HR FT ²	20.
Reentry	1000	63 (10 ²) BTU/HR FT ²	

DESCRIPTION:

Isothermalizer HP's across inteference heating area to minimize & equalize temperatures.

ADVANTAGES

1. Minimize impact of uncertainties in predicted surface temps.
2. Lower peak surface temperatures.

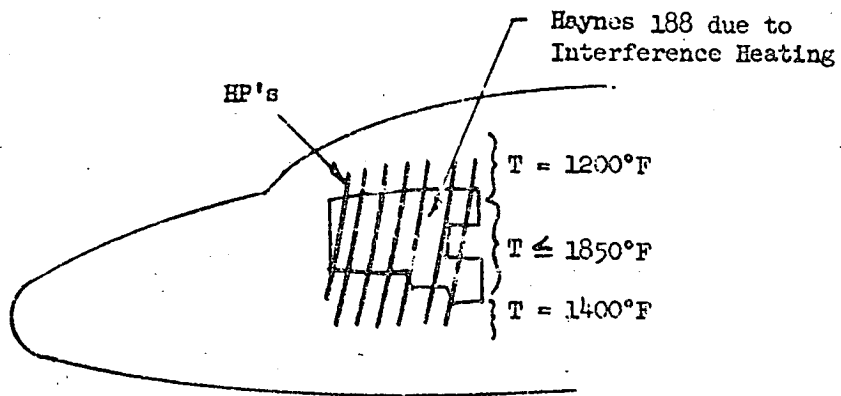
DISADVANTAGES

1. Applying fail safe criteria means HP's would be a weight penalty since passive TPS must function without them anyway and the passive system does not degrade.

COMMENTS:

No real payoff.

HEAT SOURCE/SINK IDENTIFICATION - CONTINUATION SHEET NO. 1



$$A_{1400^\circ} = (.5)(.6) \frac{(223)^2}{144} = 104 \text{ ft}^2$$

$$A_{1850^\circ} = (.4)(.4) \frac{(223)^2}{144} = 55 \text{ ft}^2$$

$$A_{1200^\circ} = (.4)(.3) \frac{(223)^2}{144} = 41.5 \text{ ft}^2$$

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.18

APPLICATION: TPS Panels

SUBSYSTEM: Structure

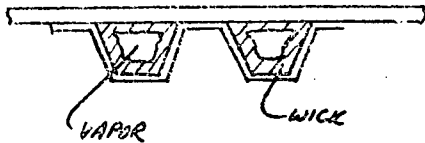
LOCATION:

BASELINE T/C SYSTEM: Superalloys & Insulation

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Reentry	2600 (Before)	$12 (10^4) \frac{\text{BTU}}{\text{HR FT}^2}$	20 to 50
Boost	1600 (After HP's)		

DESCRIPTION:

Integral heat pipe-structural panel for external shuttle surfaces. This will decrease and isothermize temp. levels.



ADVANTAGES

1. Decrease surface temps and allow use of less exotic materials.
2. Reduce temp. gradients across panel.
3. Possible wt. advantage.

DISADVANTAGES

1. Uncertainty of materials compatibility, 1 g effects, start-up time and long liquid metal HP's.
2. Possible large ΔT 's across HP interface.
3. There's a weight penalty if panels must be fail safe and function without the heat pipes.

COMMENTS:

No NASA interest.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.19

APPLICATION: Control Surface Pivots

SUBSYSTEM: Structure

LOCATION: Vertical Fin
Wing Elevon

BASISLINE T/C SYSTEM:

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Reentry	1300-1600°F (Fin) 1300-1600°F (Wing)	(Fin) 2880 ea.	(Fin) 38' max to sink (Wing) 34' along hinge

DESCRIPTION:

Heat pipes attached to pivot lugs and connected to a boiler type heat sink.

ADVANTAGES

1. Lower operating temps. at pivots will increase life and reliability.
2. Insure against uncertainties in predicted temperature levels.
3. Tolerate greater trajectory variations without necessitating design change.

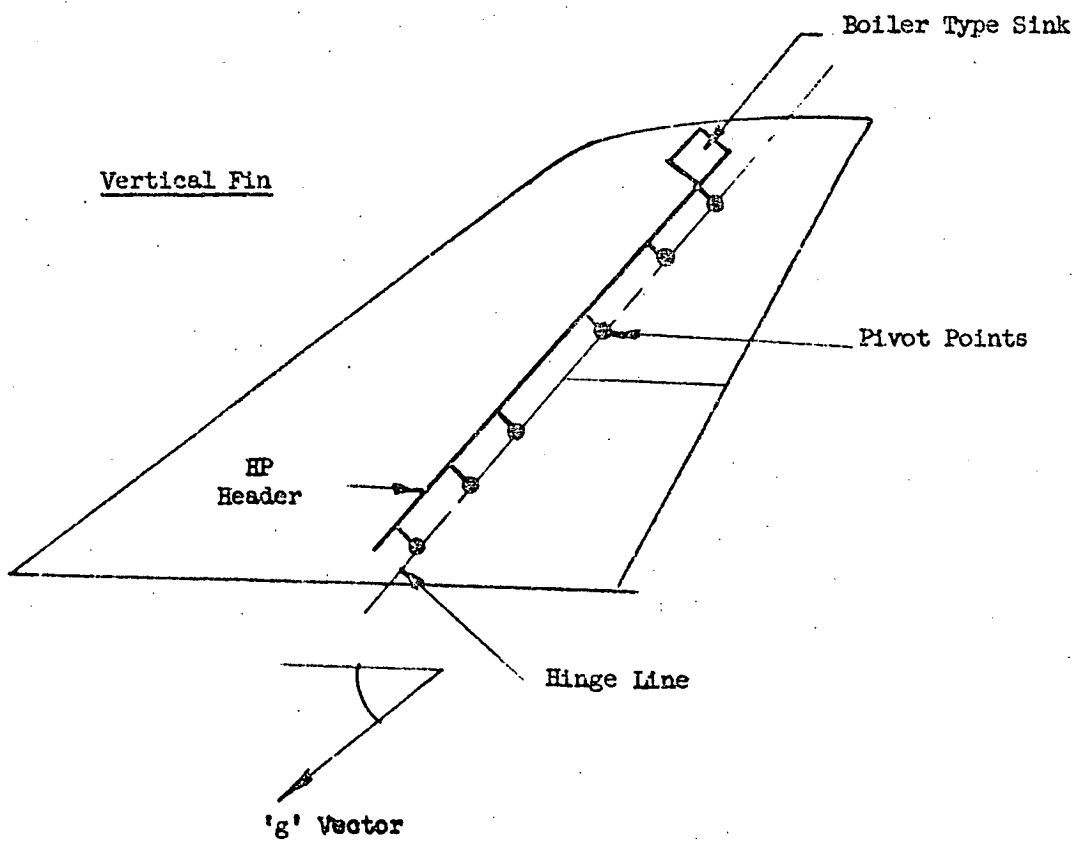
DISADVANTAGES

1. Large ΔT 's due to poor conductance are a distinct possibility.
2. Fail safe design required.

COMMENTS:

High temperatures dictate liquid metals as heat pipe working fluids. Since fail safe design is required there is no real payoff.

HEAT SOURCE/SINK IDENTIFICATION - CONTINUATION SHEET NO. 1



SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.20

APPLICATION: OMS LO₂ Boiloff (2 Tanks)

SUBSYSTEM: Structure

LOCATION: Aft Module

BASELINE T/C SYSTEM:

MISSION
PHASE

TEMP
LEVEL (°F)

HEAT
LOAD (BTU/HR)

TRANSPORT
LENGTH (FT)

Boiloff Negligible

DESCRIPTION:

ADVANTAGES

DISADVANTAGES

1. Boil off negligible

COMMENTS:

1 lb/hr boiloff

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.21

APPLICATION: Main LO₂ Tank Boiloff (Boost Tanks)

SUBSYSTEM: Structure

LOCATION: Mid Module (Fwd)

BASELINE T/C SYSTEM:

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Boost	165°R	61000	4

DESCRIPTION:

Use boiloff as heat sink in a HP/cryogenic Hx. for ground support and boost.

ADVANTAGES

1. Use of otherwise wasted resource
2. Convenient sink.

DISADVANTAGES

1. Required boiloff much greater than what's available. It would require circulating LO₂ at some fraction of its design flow rate of 168 lb/min, depending on the time required for the sink to be active and the Q is involved.
2. Low heat capacity as compared to Freon, water or hydrogen.

COMMENTS:

Unused propellant (LO₂) ≈ 3400 lbs. But tanks are emptied ≈ ½ hr from launch.

$\Delta h_{fg} = 100 \text{ BTU/LB.}$ Boiloff ≈ 1 $\frac{\text{lb}}{\text{hr.}}$

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.22

APPLICATION: C Band Directional Antenna

SUBSYSTEM: Avionics

LOCATION: Cargo Bay

BASELINE T/C SYSTEM: Conduction couplings.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>		<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
	<u>Gimbal</u>	<u>Elec.</u>		
On Orbit	-30 to 200	0 to 160	64.	
	-65 to 300	-65 to 200	0	

DESCRIPTION:

HP between power dissipating electronics and antenna which in turn radiates to space.

ADVANTAGES

1. High conductivity of pipes and achievable low temperature cut-off shall reduce the need for exotic thermal coatings and high power control loops.
2. Can eliminate antenna positional constraints imposed by long hot or cold hold conditions.
3. Better temperature regulation yields higher reliability.

DISADVANTAGES

1. Modifications are internal to manufacturer's equipment

COMMENTS:

Shuttle missions contain long duration full sun and full shadow hold conditions for the external antenna.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.23

APPLICATION: Electrical Wiring

SUBSYSTEM: Avionics

LOCATION:

BASELINE T/C SYSTEM: Wrap around ECS fluid lines.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	75 to 120		50

DESCRIPTION:

Isothermalizer HP within soft, insulated wire conduit linked via HP to a suitable sink.

ADVANTAGES

1. Avoids contact with grounded system.
2. Temperature level and gradients reduced to desirable levels improves performance of multiplexed wire systems.

DISADVANTAGES

1. Need for thermal control requirement not firmly established.

COMMENTS: Avionics people object to wrapping wire around the electrically grounded ECS due to increased chance of an electrical short. HP isothermalizer looks like a good alternate to them. Very little definition available.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.24

APPLICATION: Hydraulic Actuators

SUBSYSTEM: Power

LOCATION: Wing/Elevon, Fin/Rudder
Engine Gimbal, Landing Gear

BASELINE T/C SYSTEM: Electric Heaters or Recirculate
Hydraulic fluid continuously during on orbit.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u> (Of coolant)	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	-65 (limit) -20 (Desired)		25 - 30
Reentry	630 (limit) 400 (Desired)	4800 Ea.	

DESCRIPTION:

HP attached to hydraulic cylinder (or enclosing it) is connected to a suitable heat sink during high thermal loads, and to a heat source during a low temperature condition.

ADVANTAGES

1. Insure against uncertainties in predicted temps. both on orbit and during reentry.
2. No electrical power req'd.

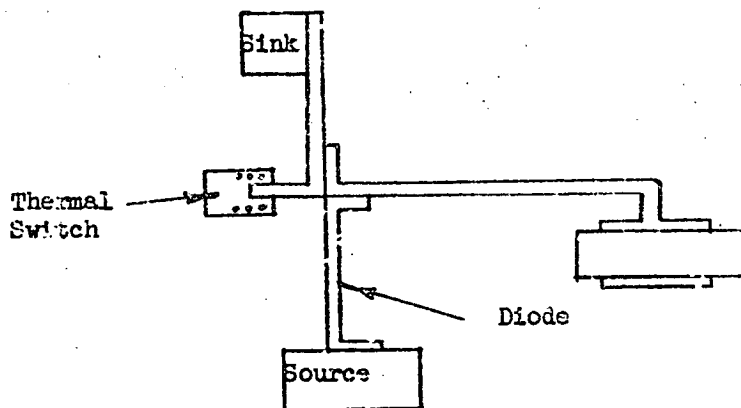
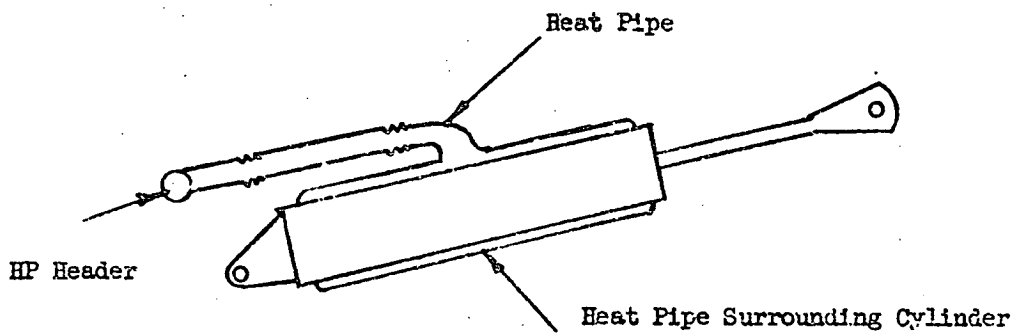
DISADVANTAGES

1. Relies on flexible HP joint.
2. Requires extensive development effort.

COMMENTS:

Structural environment is 600°F Max.

HEAT SOURCE/SINK IDENTIFICATION - CONTINUATION SHEET NO. 1



SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.25

APPLICATION: APU Fluid cooling

SUBSYSTEM: Power

LOCATION: Aft Mid Module

BASELINE T/C SYSTEM: Uses own fuel to cool lubricating & hydraulic fluids. Self-contained LH₂ Hx

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Launch-Docking	-65 to 275°	Case: 89145	
Reentry		Additional: 101,060	

DESCRIPTION:

HP heat exchanger removes heat from APU fluids and transports it to LH₂ sink.

ADVANTAGES

DISADVANTAGES

1. Use a LH₂ Hx directly. No real need for intermediate HP Hx.

COMMENTS: HP application would mean altering APU manufacturer's equipment. That's not Besides the APU's conveniently cool themselves using their own fuel (LH₂) as a sink in self contained heat exchangers - that's hard to beat.

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.26

APPLICATION: LO₂ Natural Recirculating System

SUBSYSTEM: Propulsion

LOCATION: Mid Module

BASELINE T/C SYSTEM: Electrical Heaters

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
Boost	181°R LO ₂ sink	600 <u>BTU</u> HR FT ²	
Reentry			

DESCRIPTION: Heat applied to riser pipe creates density gradient which promotes natural recirculation.

ADVANTAGES

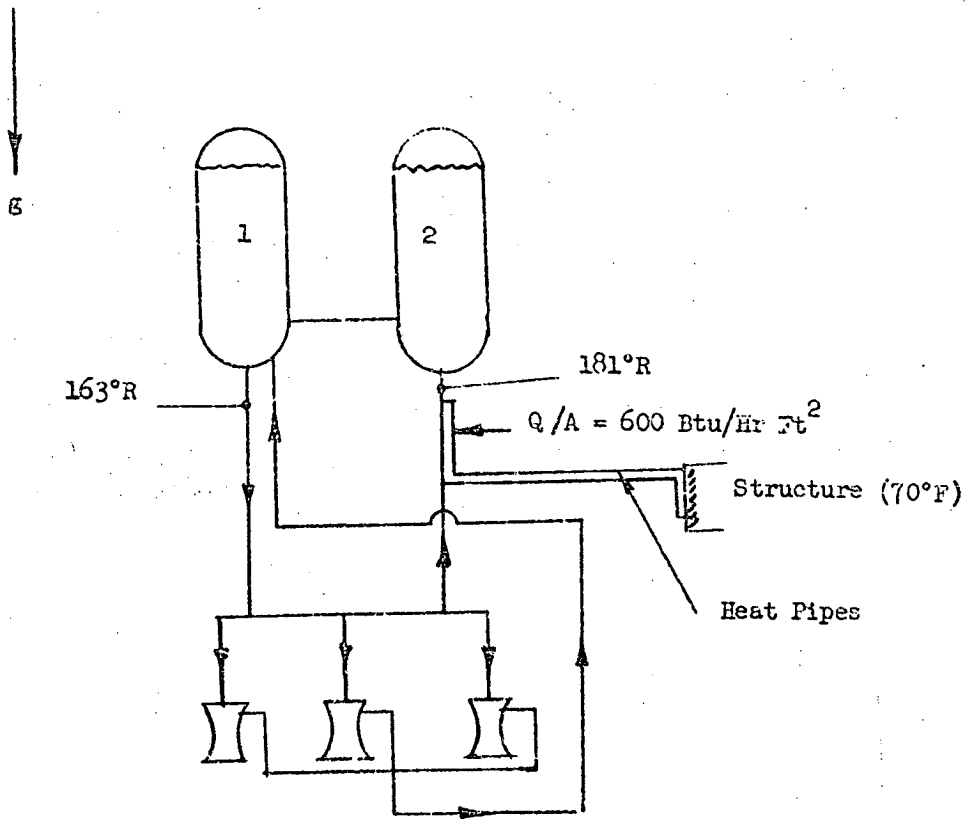
1. Eliminates electrical power req'mt with inherent danger of Apollo type failure.
2. Self actuating.

DISADVANTAGES

1. Reliable heat source questionable.

COMMENTS: LO₂ lines are uninsulated single wall pipes 16" dia. $\dot{m}_{LO_2} = 2.8 \text{ lb/sec}$

HEAT SOURCE/SINK IDENTIFICATION - CONTINUATION SHEET NO. 1



SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

A-1.27

APPLICATION: Water Chiller

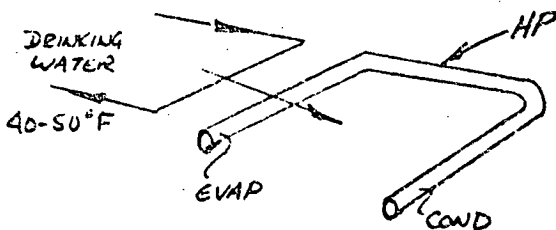
SUBSYSTEM: ECS

LOCATION: Cabin

BASELINE T/C SYSTEM: Water from heat transport loop (at 40°F) chills mixing water.

<u>MISSION PHASE</u>	<u>TEMP LEVEL (°F)</u>	<u>HEAT LOAD (BTU/HR)</u>	<u>TRANSPORT LENGTH (FT)</u>
On Orbit	40 - 50	10	10

DESCRIPTION: HP cools mixing water



ADVANTAGES

1. Provides high design flexibility ECS loop can be located in central locations.
2. Avoids possibility of contaminating water with ECS fluid.

DISADVANTAGES

1. Lowest water temp. available is about 15°F above sink.

COMMENTS:

No real improvement over baseline. It just complicates things.

APPENDIX A-2

LEADING EDGE HEAT PIPE APPLICATIONS

The feasibility of applying heat pipes between the wing leading edge and the upper wing surfaces to lower leading edge temperatures was studied. Significant reductions in temperature, from 2700°F to 1900°F, would permit the use of superalloys in place of the baselined REI and ablator TPS systems. Figure A-2.1 illustrates the concept and defines the important energy terms.

Energy balances at the leading edge and the radiating upper surface result in the following expressions:

Leading Edge

$$(1) \quad q_{c_{Le}} - q_i = q_{HP} + \epsilon \sigma T_e^4$$

Radiating Surface

$$(2) \quad q_{c_R} + q_{HP} = \epsilon \sigma T_R^4$$

Considering areas, A_e and A_R :

Leading Edge

$$(3) \quad Q_{HP} = A_e [q_{c_{Le}} - q_i - \epsilon \sigma T_e^4]$$

Radiating Surface

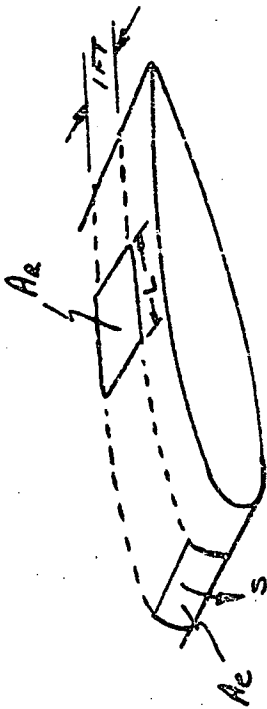
$$(4) \quad Q_{HP} = A_R [\epsilon \sigma T_R^4 - q_{c_R}]$$

Expressions 3 and 4 can be equated and then solved for the ratio A_e/A_R .

$$(5) \quad \frac{A_e}{A_R} = \frac{\epsilon \sigma T_R^4 - q_{c_R}}{q_{c_{Le}} - q_i - \epsilon \sigma T_e^4}$$

where T_e and T_R are the desired operating temperatures. Substituting values for q_{c_R} (≈ 3220 BTU/HR FT²) and $(q_{c_{Le}} - q_i)$ ($\approx 13.8 (10^4) \frac{\text{BTU}}{\text{HR FT}^2}$) gives

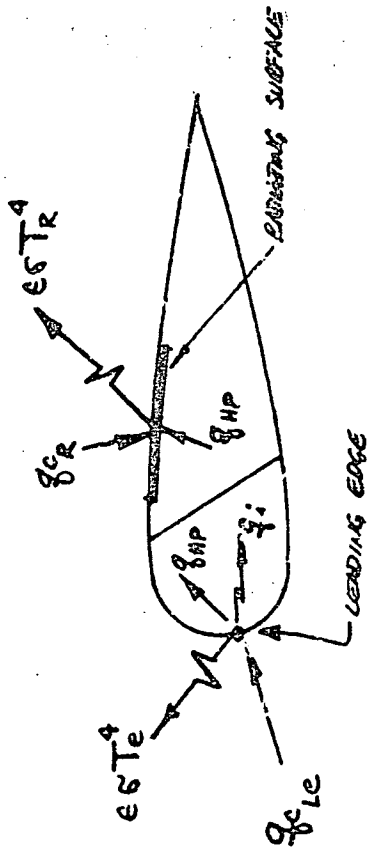
$$(6) \quad \frac{A_e}{A_R} = \frac{\epsilon \sigma T_R^4 - 3220}{13.8 (10^4) - \epsilon \sigma T_e^4}$$



$$A_e = \frac{1}{2} \pi D^2 = \pi r^2 = S$$

$$A_R = L D = L$$

$$\frac{A_e}{A_R} = \frac{S}{L}$$



q_c = CONVECTION HEATING

q_{HP} = HEAT TRANSFERRED BY HEAT PIPE

q_i = RADIATION TO INTERNAL STRUCTURE AND INSULATION

T_e, T_r = RADIATION EQUILIBRIUM TEMPERATURE

e = SURFACE EMITTANCE (ASSUME = 1.)

FIG A-2.1 - LEADING EDGE HEAT PIPE APPLICATION

The value for $(q_{c_{Le}} - q_i)$ is the decrease in energy due to internal radiation at the leading edge and corresponds to a decrease in radiation equilibrium temperature of 250°F from an initial value (without internal radiation) of 2950°F .

From the geometry of a one foot span the ratio can be expressed as

$$(7) \frac{A_e}{A_R} = \frac{S}{L}$$

Assuming a usable chord length of 4.5 feet and a leading edge radius of 2.5 inches results in a ratio of $\frac{A_e}{A_R} = .145$ for each foot of span, which means the wing configuration can accommodate temperatures that require the radiating area to be up to 6.9 times $(1/.145)$ as large as the edge surface area in satisfying equation 6. Fixing the maximum leading edge temperature at 1900°F , this means a required radiating surface temperature of 1310°F - an unacceptably high increase above its initial 775°F which would require a change to another, higher temperature, material (Rene 41) instead of titanium (Ti - 6AL - 2 Sn - 4 Zr - 2 Mo).

If this material change were acceptable, the heat pipes would be required to transfer about 18,000 watts at about 1400°F for each foot of span.

PRECEDING PAGE BLANK NOT FILMED

APPENDIX A-3

SLA BACKFACE TEMPERATURE CONTROL USING HEAT PIPES

The feasibility of using heat pipes to lower the backface temperatures of the shuttle's low density superlight ablator (SLA) was examined when emphasis was placed on using aluminum instead of titanium for primary structure. The goal was to decrease the backface temperature during entry from 600°F (Titanium structural limit) to 300°F (aluminum limit) without increasing the ablator thickness.

The possible benefits would be the lower material and manufacturing costs associated with using aluminum instead of titanium, and a reduction of about 1.3 lb/ft² of surface area in ablator weight.

The heat pipes would be required to transport the necessary heat from the SLA backface to a convenient and useable interval structural heat sink, e.g., bulkheads.

Two entry cases were examined. Case I specified a 6000 nautical mile downrange and 27° entry angle of attack; Case II specified a 3500 nautical mile downrange and 50° entry angle of attack. Heating rates for the Shuttle underbody were used as representative values, resulting in a total integrated heating rate* of 15,000 BTU/FT² over an entry time of 2000 seconds for Case I, and a total integrated heating rate of 3,800 BTU/FT² over 1100 seconds for Case II. The initial structural sink temperature was assumed to be 100°F for both cases.

For the operating temperature required (≈ 300°F) water is thermodynamically the most efficient working fluid; its liquid transport factor $\frac{\sigma \rho \lambda g_0}{\mu}$ is about 10.5 (10¹¹) BTU/HR FT².

The capacity of a one inch ID water heat pipe as a function of effective length and operating temperature is given in Figure A-3.1, which was the design curve used in this study.

* $\int_0^{\tau} q_0 d\tau$ = Total integrated heating rate

From Figure A-3.2 and using the total entry times previously given, the heat pipes would be required to carry 22,800 BTU/HR FT² of ablator area for Case I, and 9,800 BTU/HR FT² for Case II. Assuming an overall heat pipe conductance of 1,000 BTU/HR FT² and a 300°F operating temperature, the available temperature rise in the aluminum structure (initially at 100°F) would be 127°F for Case I and 191°F for Case II. One hundred pounds of aluminum (Cp = .25 BTU/lb°F) would provide a 3180 BTU sink for Case I and a 4780 BTU sink for Case II, resulting in a requirement for 393 lbs of aluminum per ft² of surface area for Case I and 63 lbs/ft² for Case II.

Using the results in Figure A-3.1, the structural sink must be within 2.4 ft of the source in order for the heat pipes to transfer the Case I load and within 9 ft of the source to transfer the Case II load. These requirements are impossible to meet since the average weight area density is about 5 lb/ft² of surface area for the body group - to use one example.

CONCLUSIONS AND COMMENTS

1. The proposed application of heat pipes is technically feasible with certain restrictions. They cannot be applied in stagnation regions due to excessive heating rates, and their use on other parts of the Shuttle depends upon the entry angle of attack and downrange requirements. For example, their use on the underbody surfaces is more practical with higher entry angles of attack and shorter downrange distances. Otherwise, the heating loads approach the theoretical capacities of the heat pipes in the system.

2. Since aluminum heat pipe envelopes are not compatible with water due to gas generation problems, more innocuous materials such as copper or monel would have to be used. This means that the heat pipe envelopes would have to be bonded to the aluminum backface and could not be manufactured as integral structure. This translates directly into an obvious weight penalty for the heat pipe system, a weight penalty that would have to be less than the ablator weight savings of 1.3 lb/ft². It is estimated that the heat pipes would weigh about .68 lb per ft² of evaporator area per foot of heat pipe length.

The material and manufacturing costs for the above heat pipe system must also be less than the costs of a titanium structure, and it is not clear that this is the case. The additional steps required to fabricate a heat pipe could make such a system more expensive, but enough information is not presently available to come to a definite cost conclusion.

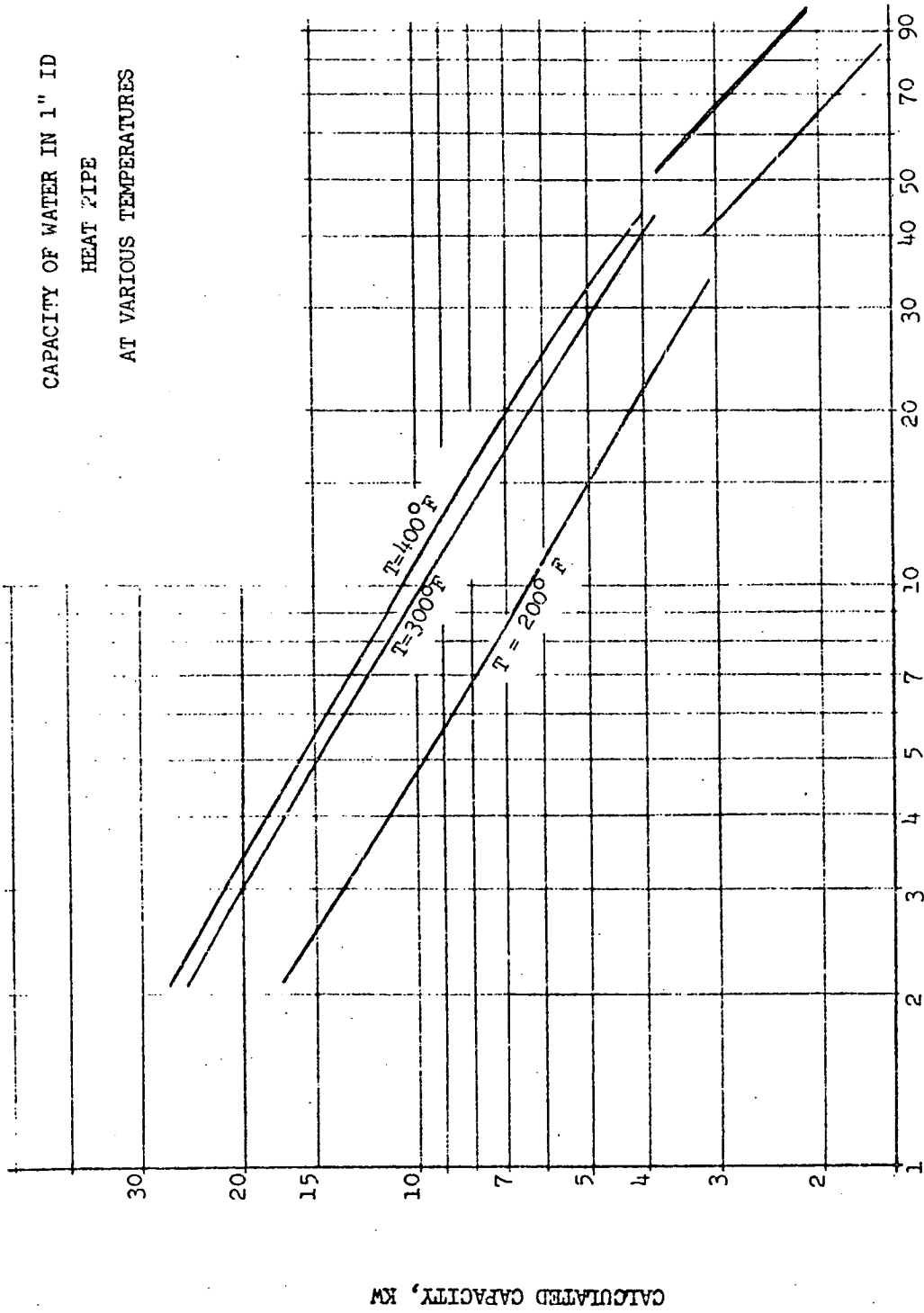
3. The heat pipe application, although technically feasible, is not practical since there is not enough available structural heat sink to serve the entire surface area requirement. The Shuttle Body Group weighs about 5 lbs per ft² of surface area; the wing group weighs about 7 lbs per ft² of exposed wing area (Reference 1). The volume of structural heat sink which would be required for the extensive application of heat pipes is an order of magnitude greater than that actually available. The heat pipe system might be able to satisfy some local conditions if the heating rates are not high and there is enough heat sink available, but the same results can be achieved by using more ablator.

4. The problems involved with routing the heat pipes around, along and through vehicle structure (stringers, ribs, longerons, spars) to get to adequate structural sinks cannot be evaluated until a closer examination of the structure is made, but solving such problems would be major design considerations.

5. Starting up a frozen water heat pipe, which would occur during entry from earth orbit, requires investigation. However, it is felt that such pipes would probably thaw and function satisfactorily due to the thermal lag inherent in the structure.

6. The heat pipes must be dicde types to preclude draining heat from the internal structure to the outer surfaces and into space during on-orbit operations.

CAPACITY OF WATER IN 1" ID
HEAT PIPE
AT VARIOUS TEMPERATURES

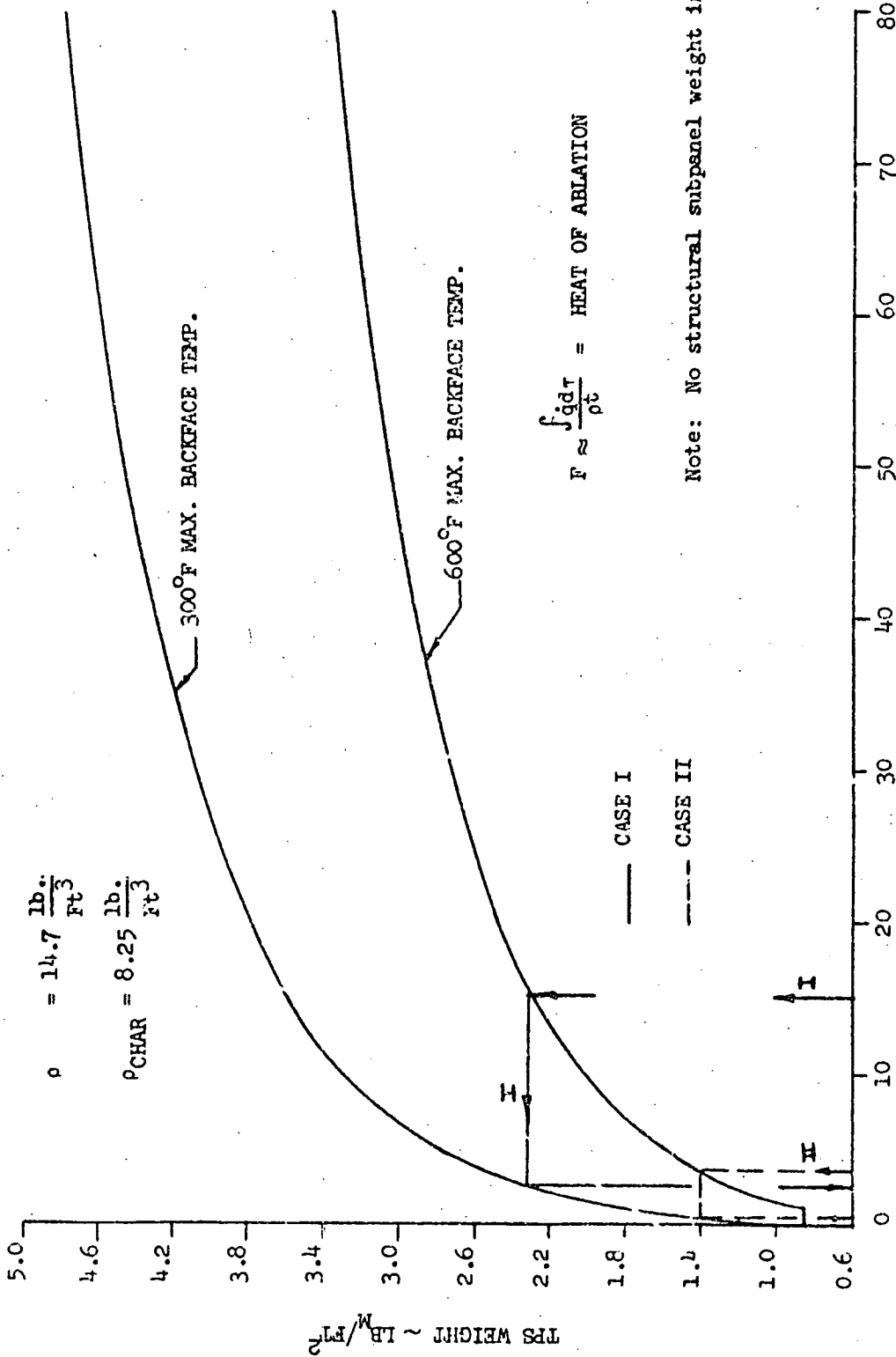


EFFECTIVE LENGTH, FEET

FIGURE A-3.1



MARTIN SLA 561 UNIT WEIGHTS



$\rho = 14.7 \frac{\text{lb}}{\text{ft}^3}$
 $\rho_{\text{CHAR}} = 8.25 \frac{\text{lb}}{\text{ft}^3}$

$$F \approx \int \frac{\dot{q} dt}{\rho t} = \text{HEAT OF ABLATION}$$

Note: No structural subpanel weight included.

TOTAL INTEGRATED HEATING (at surface of ablator) $\sim \frac{\text{BTU}}{\text{ft}^2} \times 10^{-3}$

FIGURE A-3.2



PRECEDING PAGE BLANK NOT FILMED

APPENDIX A-4

HEAT PIPE APPLICATIONS FOR HYPERGOLIC FUELED APU

Hydrogen/oxygen fuel cells are the primary source of electrical power during all Space Shuttle mission phases. However, the high power hydraulic system demands uniquely imposed during powered flight phases are met by four APUs.

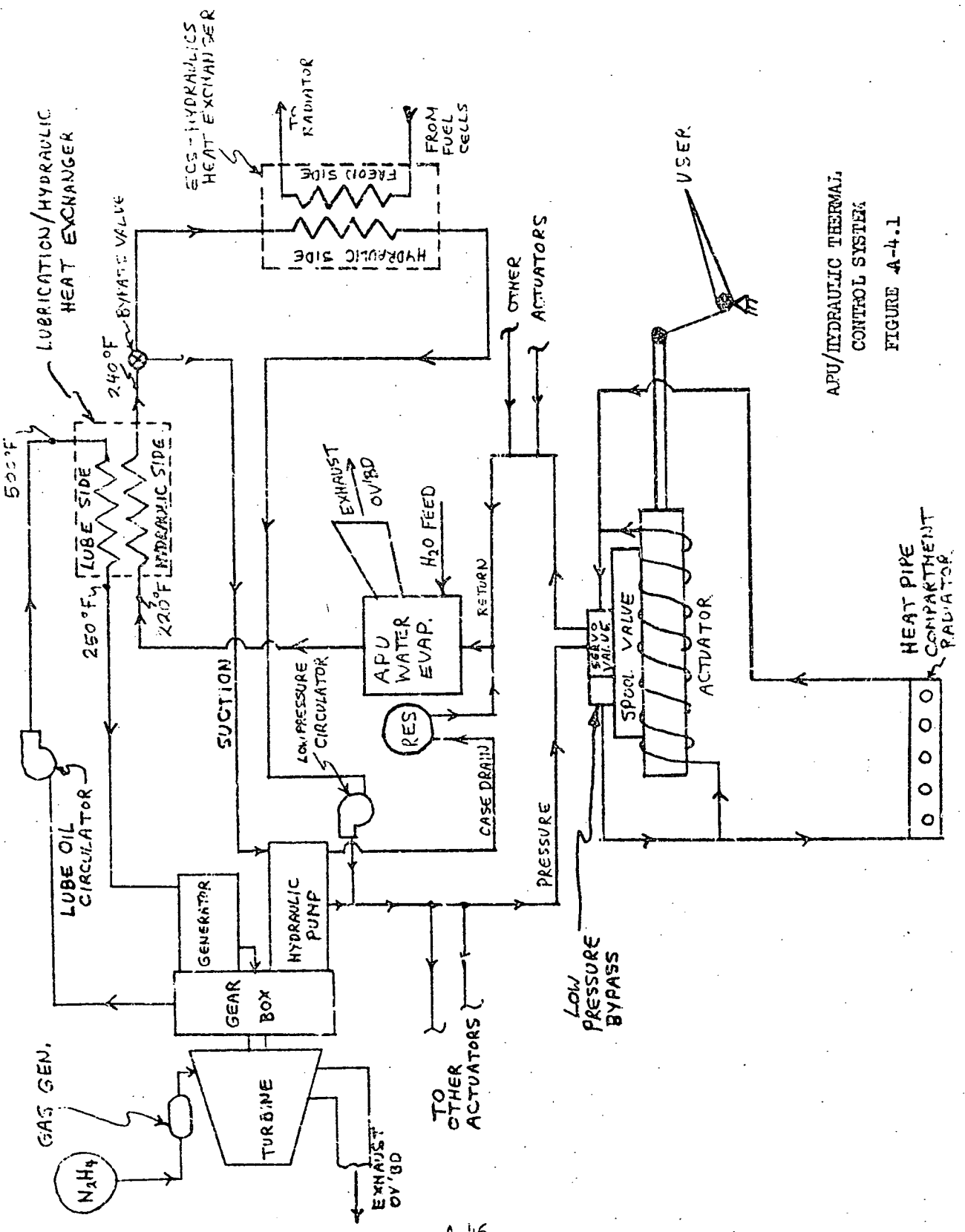
Each APU consists of a gas turbine driving a gear box which in turn drives a 5 KVA generator and a hydraulic system pump. The hot turbine exhaust gas is vented directly overboard, while the thermal conditioning of the gearbox and generator is accomplished by a circulating lubrication oil loop. This lube oil must be cooled (approximately 20,000 Btu per hour per APU) to maintain acceptable oil and bearing temperature limits (500°F) thereby avoiding breakdown of the oil and contamination of the gearbox bearings.

One of the twenty-seven initial heat pipe application concepts considered used heat pipes to transport the heat load from the APU fluids to a heat sink - at the time, LH₂. The baseline (no heat pipe) approach to removing this heat was to use a direct oil/LH₂ heat exchanger - thus using the APU's waste heat to pre-heat its own incoming fuel. No real purpose would have been served in introducing an intermediate LH₂ heat exchanger and this heat pipe concept received a rating of zero (outright rejection).

Since that time, a fundamental Space Shuttle program change was implemented; the substitution of hypergolic fuels for LH₂ and LOX in the Orbital Maneuvering System (OMS). One result of this change was to turn-around the APU design - from LH₂ fueled to hydrazine fueled. With no LH₂ in the system, a new baseline heat rejection approach had to be developed.

These changes prompted the thought that the new APU system may present design problems more readily solved by the application of heat pipe technology. Therefore, a study of the new baseline APU thermal control design was undertaken.

Figure A-4.1 is a schematic representing the present baseline APU thermal control system. During powered operation, the turbine exhaust gases ($\approx 1200^{\circ}\text{F}$)



APU/HYDRAULIC THERMAL CONTROL SYSTEM
FIGURE A-4.1

are vented overboard. The lube oil exits the gearbox ($\approx 500^{\circ}\text{F}$ max) and rejects its heat to the lubrication/hydraulics heat exchanger. The hydraulic loop water evaporator boils water to reject the heat generated in the hydraulic system and absorbed from the lube oil. (The ECS/hydraulic heat exchanger is bypassed).

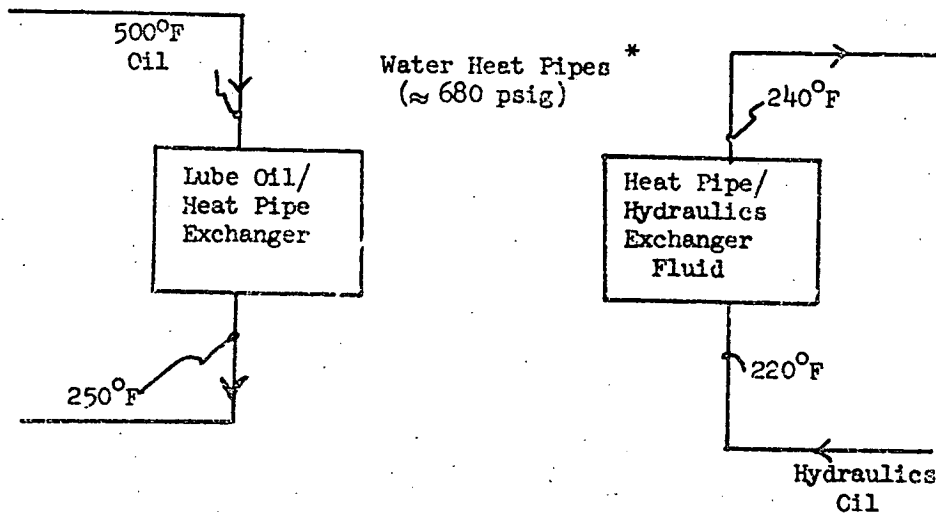
On orbit, APU thermal control is accomplished by actively circulating low pressure hydraulic oil to maintain minimum acceptable temperatures (40°F). Waste heat from the fuel cells is absorbed in the hydraulics fluid through the hydraulics/ECS heat exchanger and lube oil is circulated periodically to distribute heat to the APU components (replacing heat lost to space through the exhaust gas dump). In an alternate concept, for low heat leaks to space, no circulation of the lube oil would be necessary. Instead, the system would use low pressure hydraulics oil directly, in conjunction with a heat-pipe compartment radiator panel, thus radiatively supplying heat to the APU.

Studies were conducted to determine the optimum method of addressing the primary APU thermal problem, i.e., how to reject the 20,000 BTU/hr heat load during powered phases. The method selected (lube oil to hydraulics heat rejection) resulted in the minimum weight system. A potential heat pipe application would be to separate the lube/hydraulic interface with heat pipes (Figure A-4.2). The advantage of this scheme would be to prevent the leakage of high pressure hydraulic system oil into the APU lube oil with the resultant permanent contamination of the APU's bearings and other internal components.

Table A-4.1 is a "grading" sheet for this heat pipe application. As shown, the scheme would receive a rating of +1 (possible contender). This rating would have excluded it from the list of eleven prime contenders (+2 rating) selected at this point, but nevertheless, it does possess some merit. Had the concept been carried over into the next evaluation phase it would not have survived the subsequent cuts which reduced the field of prime candidates to a total of six. This is shown in the evaluation sheet in Table A-4.2. The safety advantage (no permanent APU damage due to hydraulics leak) is offset by a weight penalty - resulting in a net rating of C.

FIGURE A-4.2

APU LUBRICATION OIL/HEAT PIPE/HYDRAULIC FLUID HEAT EXCHANGER CONCEPT



* Approx.

12, $\frac{1}{2}$ inch ID (square), 24" long water heat pipes built to contain 680 psia (500°F), would be required to maintain an evaporator heat flux of 25 watts/in² with a total ΔT evaporator-condenser of $\approx 11^\circ\text{F}$. These pipes, having an effective length of 12 inches, would have no difficulty transporting the requisite heat load.

TABLE A-4.1

SHUTTLE HEAT PIPE APPLICATION - IDENTIFICATION SHEET

APPLICATION: APU Lubrication Oil/Hydraulic Fluid Heat Exchanger

SYSTEM: Power

LOCATION: Aft Mid Module

BASELINE T/C SYSTEM: Uses direct lube oil to hydraulic fluid heat exchanger to transfer lube oil heat load

<u>Mission Phase</u>	<u>Temp Level (°F)</u>	<u>Heat Load (Btu/hr)</u>	<u>Transport Length (ft)</u>
Ascent	$\left. \begin{array}{l} 500^{\circ}\text{F} \text{ (Lube oil inlet)} \\ 250^{\circ}\text{F} \text{ (Lube oil outlet)} \\ 220^{\circ}\text{F} \text{ (Hydraulic Inlet)} \\ 240^{\circ}\text{F} \text{ (Hydraulic Outlet)} \end{array} \right\}$	20K per APU	
Deorbit		19K	
Entry		20K	
APP/Land		20K	
Go Around		20K	

DESCRIPTION: Separate lube oil and hydraulic fluid flow paths with intermediary heat pipes

ADVANTAGES

1. Possibility of hydraulic fluid leaking into APU lube system and causing irreparable damage is totally eliminated.

DISADVANTAGES

1. HP ΔT decreases effective LMTD available to accomplish heat transfer between lube oil and hydraulic oil - resulting in higher heat exchanger weight.
2. Heat pipe weight is direct penalty.

COMMENTS: None

RATING: 1

TABLE A-4.2

EVALUATION MATRIX - HEAT PIPE APPLICATIONS

APPLICATION: APU Lubrication Oil/Hydraulic Oil Heat Exchanger

<u>Criteria</u>	<u>Rating Compared to Baseline</u>		<u>Comments</u>
	<u>Worse Than</u>	<u>Better Than</u>	
1. Temperature Gradient	-	-	
2. Capacity Margin	-	-	
3. Power Requirements	-	-	
4. Control Requirements	-	-	
5. Weight	X		HP weight + increased exchanger weight
6. Safety		X	Possibility of hydraulic fluid permanently contaminating APU components is eliminated (safety to equipment)

REMARKS: Net Rating - 0 (1 worse, 1 better, 4 same)

APPENDIX A-5

APPLICATION DESCRIPTIONS

PRIME CONTENDERS (RATING = 2)

1. TPS Leading Edge

Subsystem: Structure

High temperature liquid metal heat pipes can be applied to the wing leading edges in areas of high aero-heating to reduce circumferential temperature gradients in the oxidation resistant carbon-carbon reinforced pyrolyzed plastic (RPF) composite structure. By reducing local maximum temperatures and temperature gradients, the number of missions the leading edge surfaces can fly without refurbishment is increased substantially.

2. Landing Gear

Subsystem: Structure

During certain orbital conditions, heat must be applied to the landing gear to prevent the tires and hydraulic fluid from becoming colder than the allowable limits. By using heat pipe radiator panels drawing heat (by means of a diode heat pipe) from the waste heat rejection loop, improvements in weight and control can be achieved over the baseline systems which use electrical heaters or fluid loop radiators. The diode heat pipe prevents reverse heat transfer from the compartment back into the waste heat rejection system during entry.

3. Avionics HP Circuit

Subsystem: Avionics

Heat pipes can be used to replace part or all of the coolant fluid loop. Equipment to be cooled can be mounted on heat pipe cold rails, which would transfer the waste heat to other heat pipes and finally to either a space radiator or the fluid loop. This approach requires no electrical power and no rotating machinery, unlike the pumped fluid loop baseline.

4. Modular Sinks

Subsystem: Avionics

It is simpler to cool a remotely located, low average power output component by some local means rather than extending a pumped fluid coolant loop to these remote locations. One system which does this is the heat pipe/phase change material modular sink, consisting of a heat pipe cold plate (on which equipment is mounted), and a transport heat pipe which connects the cold plate to a diode heat pipe; the diode is coupled to vehicle structure. A canister of phase change material is placed around the transport heat pipe and absorbs heat at times when the structure is too hot to act as a sink. The diode shuts off at these times, preventing structural heat from leaking back into the phase change material.

5. Air Transport Rack Commercial and Military Aircraft Equipment

Subsystem: Avionics

As a cost-saving measure, the use of standard commercial and military avionics equipment is planned on the shuttle. This equipment has been designed for standard racks and requires a convective atmosphere for proper temperature control, which would be provided in the shuttle by an enclosed rack with a circulating forced air system. The circulating air could be cooled by an air-to-heat-pipe heat exchanger, and the heat pipes could transfer the waste heat either to a radiator or to a pumped coolant loop by means of another heat exchanger. Use of such heat pipes makes fluid line penetrations of the enclosure unnecessary, minimizing the risk of equipment damage due to leakage.

6. Flight and Voice Recorders

Subsystem: Avionics

The flight data and voice recorders for the shuttle are FAA Type III, and are currently located in the lower aft portion of the vertical fin. The recorders are required to function continuously in orbit, and must be maintained between -68°F and 160°F . Analysis shows that surrounding structure can be used as a sink only during orbit.

In order to avoid long, out-of-the-way runs of coolant lines, heat pipes internal to a cold plate type mounting surface can transfer waste heat to either an intermediate sink (phase change material or boiler) or to structure

as the situation demands. A diode heat pipe would be used to prevent structural heat from leaking back into the intermediate sink.

7. HP Radiator with HP Header

Subsystem: ECS

This application is a radiator system for waste heat rejection consisting of ten heat pipe radiator panels. Waste heat from the shuttle electronics is taken from the coolant loop (by means of a heat exchanger) or from the heat pipe circuit and carried to the radiator panel by a VCHP header with a flexible section at the cargo bay door hinge line. Heat is distributed throughout the panel by a number of small feeder heat pipes. Use of a VCHP header eliminates the need for electromechanical control systems, increasing reliability, and a heat pipe radiator panel requires less micrometeoroid shielding than a conventional panel.

8. HP Radiator with Integral HP/Fluid Header

Subsystem: ECS

This waste heat rejection heat pipe radiator concept eliminates the requirement in (7) above for a flexible heat pipe leader. Flexible fluid lines run past the cargo bay door hinge line to a fluid header on the radiator panel. The heat pipe header evaporator is immersed in a fluid header, and the waste heat is carried to the rest of the panel by feeder heat pipes connected to the heat pipe header's condenser. Control is provided by making the heat pipe header a VCHP.

9. ECS Cold Rail

Subsystem: ECS

Some of the flange-mounted electronics modules proposed for the shuttle have dissipation levels of 40 watts per linear inch per side. This value is approximately 16 times greater than the design value in the Apollo vehicle and exceeds the capacity of simple fluid cold rails.

By placing a heat pipe in the center of the rail, the high watt density heat load may be spread over the length of the fluid passages rather than just a narrow section of the rail. By increasing the utilized fluid area, the temperature drop into the fluid passage should decrease enough to allow the

use of the proposed high watt density components.

10. HP Radiator For Fuel Cell

Subsystem: Power

If the decision is made to employ heat pipe radiators on the shuttle, the most thermally efficient concept is to segregate the loads into high and low temperature heat loads. Most of the high temperature load comes from the fuel cells, so the efficiency of the system could be enhanced by giving the fuel cells their own radiators. The radiator configuration could be physically the same as that in (7) or (8).

11. Air-Breathing Engine Compartment

Subsystem: Propulsion

If air-breathing engines are used on the shuttle in the space mission configuration, on-orbit heating of the engine components will be required to maintain the engine within present MIL specifications: - 65°F to 160°F non-operational. In this application, diode heat pipes are used to take waste heat from the pumped coolant loop system to radiator panels on the walls of the air-breathing engine compartment. Heat would be distributed throughout the panel by means of a number of small feeder heat pipes. A diode heat pipe is used to prevent heat from leaking back into the coolant loop during hot conditions such as entry.

POSSIBLE CONTENDERS (RATING = 1)

12. OMS LH₂ Boiloff

Subsystem: Structure

The boiloff from the nose-mounted OMS LH₂ tank could be used as a heat sink by venting it through a heat pipe heat exchanger to cool local electronics mounted on a heat pipe cold plate.

13. High Intensity Lights

Subsystem: Avionics

By using heat pipes to couple the cargo bay and exterior high intensity light bases to cooler surrounding structure, life of these expensive lights can be greatly extended because of the lower base temperatures provided.

14. Battery

Subsystem: Avionics

There are two heat pipe applications to the emergency power batteries in the nose module: as isothermalizers within a battery, and as heat transport devices from the batteries to an expendable sink. By causing all the cells within a battery to operate at the same temperature, battery life and performance can be increased. If these isothermalizing heat pipes are in turn connected to a local sink (e.g., a water boiler) by a transport heat pipe, fluid loop connections to the batteries (unneeded except in emergencies) could be eliminated.

15. Tracking Radar

Subsystem: Avionics

In the baseline shuttle tracking radar, electronics gear is cooled by conductively coupling it with the antenna surface which then radiates waste heat into space. By coupling equipment to the antenna with a heat pipe, a better coupling can be attained at less weight.

16. Fluid Evaporator

Subsystem: ECS

During mission phases other than orbit, it is impractical to use the waste heat rejection radiator system as a heat sink. Heat pipes could carry such waste heat to fluid evaporators, which would be placed at locations on the shuttle near the heat sources they serve.

REJECTED (RATING = 0)

17. Fuselage TPS - External Tank
Interference Heating Area

Subsystem: Structure

Liquid metal heat pipes can be used in skin structure in the interference heating area to isothermalize that area, minimizing peak temperatures and reducing the impact of uncertainties in predicted surface temperatures.

18. TPS Panels

Subsystem: Structure

Liquid metal heat pipes integral to the surface structural panels can be

used to isothermialize large surface areas, decreasing surface temperatures and allowing use of less exotic surface materials.

19. Control Surface Pivots

Subsystem: Structure

If the operating temperatures of control surface pivots can be lowered, life and reliability will be increased. One means of doing this is to attach small feeder heat pipes to pivot lugs and connect these feeder pipes to a heat pipe header. Liquid metal heat pipes would have to be used because of the temperature levels involved; boiler type heat sinks could be employed.

20. OMS LO₂ Boiloff

Subsystem: Structure

A possible sink for use with a heat pipe thermal control system is the LO₂ boiloff from the OMS storage tanks. The cold vapor could be vented through a heat pipe heat exchanger, thereby acting as a sink and allowing the heat pipe to reject waste heat.

21. Main LO₂ Tank Boiloff (Boost Tanks)

Subsystem: Structure

A heat pipe/cryogenic heat exchanger can be developed for use as a sink for heat pipe cooled equipment using the boiloff from the internal LO₂ tank in the Grumman H-33 configuration. This sink would be available during the ground support and boost phases of the mission, time periods when the space radiators are unavailable.

22. C Band Directional Antenna

Subsystem: Avionics

Power dissipating electronics are cooled by conductively coupling them to the antenna, which radiates the waste heat into space. By using heat pipes to couple the electronics to the antenna, the weight involved in providing a good conductive coupling can be eliminated, as is the need for exotic coatings on the antenna.

23. Electrical Wiring

Subsystem: Avionics

In order to avoid overheating the electrical wiring, one of the suggestions

for the baseline is to wrap the wiring around ECS lines. ECS lines are electrically grounded, and there is a chance of a short circuit. To eliminate this possibility, a heat pipe coupled to a sink could be placed within the wire conduit to cool the wire bundle.

24. Hydraulic Actuators

Subsystem: Power

The desired minimum temperature for hydraulic cylinders is -20°F . To attain this level on orbit, heat must be supplied. A heat pipe attached to the hydraulic cylinder (or enclosing it) is connected to a suitable heat source during low temperature conditions. Such a heat pipe application requires a flexible joint, as the actuator moves during the entry and landing mission phases.

25. APU Fluid Cooling

Subsystem: Power

The high heat loads generated by the APU create a cooling requirement for the APU's lubricating and hydraulic fluids. A fluid-to-heat pipe-to-fluid heat exchanger system can be developed using the LH_2 of the cryogenically fueled APU to cool the two hot fluids.

26. LO_2 Natural Recirculating System

Subsystem: Propulsion

During boost, oxygen in the internal LO_2 tanks is circulated within the tanks by providing small amounts of heat to provide natural density gradients and thereby establish convection currents. This heat could be provided by heat pipes coupling some local heat source during boost to the riser pipe "beneath" the tanks.

27. Water Chiller

Subsystem: ECS

The drinking water can be chilled by coupling the water to the ECS coolant loop by means of heat pipes. This approach avoids the possibility of contaminating the drinking water with ECS fluid.

APPENDIX B-1

HEAT PIPE JOINING TECHNIQUES

This appendix contains descriptions and drawings of several possible heat pipe joining techniques. Both the conductive interface type and the internal mating type are presented. The former are efficient means of thermally connecting two complete heat pipes; the latter method enables one large heat pipe to be made of several smaller subsections.

Figure B-1.1 - Conductive Interface - Saddle Bolted

This can be used to join conventional heat pipes together in series. In configuration (a) the heat pipes are completely encased between two halves of machined saddles; contact pressure on the HP/saddle interface is provided by the bolt torque. In configuration (b) the heat pipes are held in place by welding the tubes to the saddle recesses; the saddles are then bolted back to back. Contact pressure depends on preload at welding. Both configurations employ a high thermal conductance grease between mating surfaces.

Figure B-1.2 - Conductive Interface - Flange Bolted

These are extended flanged heat pipe tubes which are butt spliced to each end of the heat pipes to be joined. The unspliced end of the extended section is sealed. The joint between the pipes is made by screwing or bolting the flanged sections back to back. A high thermal conductance grease is used between the flanged mating surfaces.

Figure B-1.3 - Conductive Interface - Clamped Saddle

This can also be used to join conventional heat pipes in series. The heat pipes are held between the halves of machined saddles. Control pressure between the pipes and saddles is provided by strap type clamps every two to three inches as required.

Figure B-1.4 - Conductive Interface - Socket Type

The male evaporator end of one heat pipe is inserted into the female con-

denser end of the second heat pipe. The stepped wick transports condensed fluid to the artery gradually over a large surface area.

Figure B-1.5 - Conductive Interface - Heat Pipe Augmented Saddle

A thick wall tube containing integral heat pipes overlaps both ends of the pipes to be joined. Heat is transferred from the condenser end of one heat pipe, through the splicing tube, to the evaporator end of the second heat pipe.

Figure B-1.6 - Artery "Y" Splice

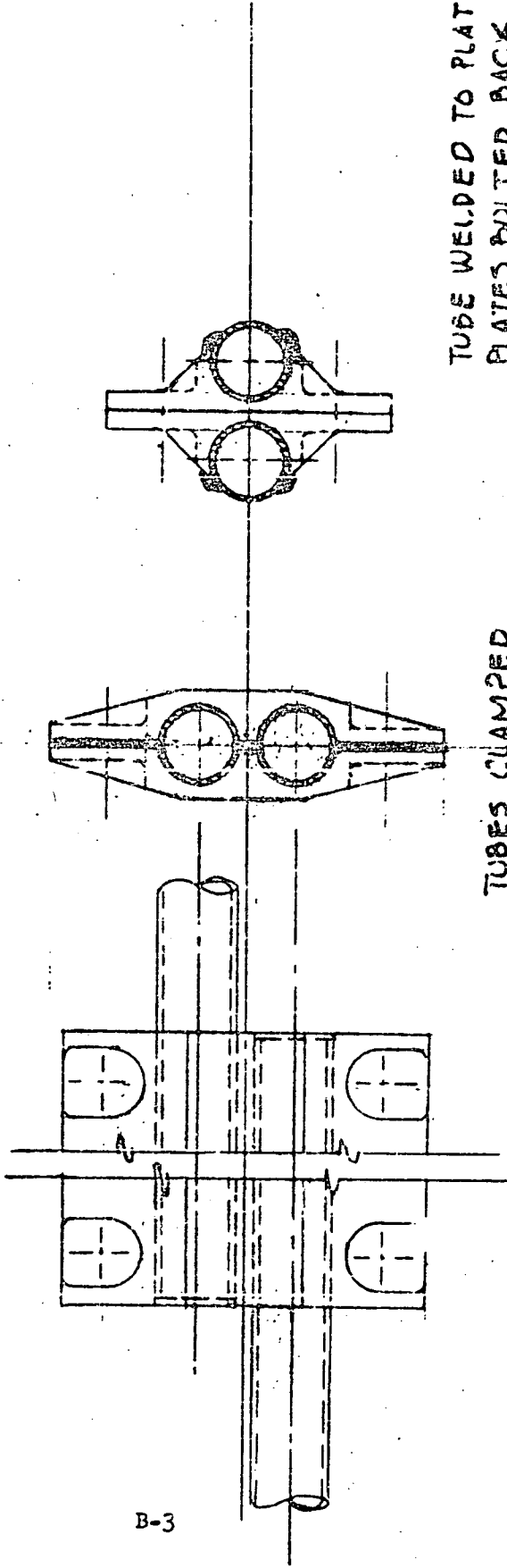
This method can be used to split heat pipe flow paths.

Figure B-1.7 - Butt Splice

This type of splice can be made in place on the vehicle by using portable induction heating elements which braze the two stainless steel ends of the heat pipe sections together. It's necessary to use a stainless steel section because the gold/nickel brazing alloy is not compatible with aluminum. The required aluminum/stainless transition tubes are commercially available. The internal arteries are butt spliced together using a sleeve made of screen mesh or thin wall tubing.

Figure B-1.8 - Butt Splice using Standard Fitting

Heat pipe butt splices can be made in place on the vehicle using standard Gamah fittings for joining the pipe envelopes; sleeve splices can be used for the arteries. The sketch shows how a JTL4 fitting can be used to join two 1 inch O.D. heat pipes.



(a)

(b)

TUBES CLAMPED
IN TWO SADDLES.
SECURED BY BOLTS -
SADDLES SHIMMED
IN BOLT AREA
(1.24[#]/FT W/O PIPES)

TUBE WELDED TO PLAT
PLATES BOLTED BACK
TO BACK
(.91[#]/FT W/O PIPES)

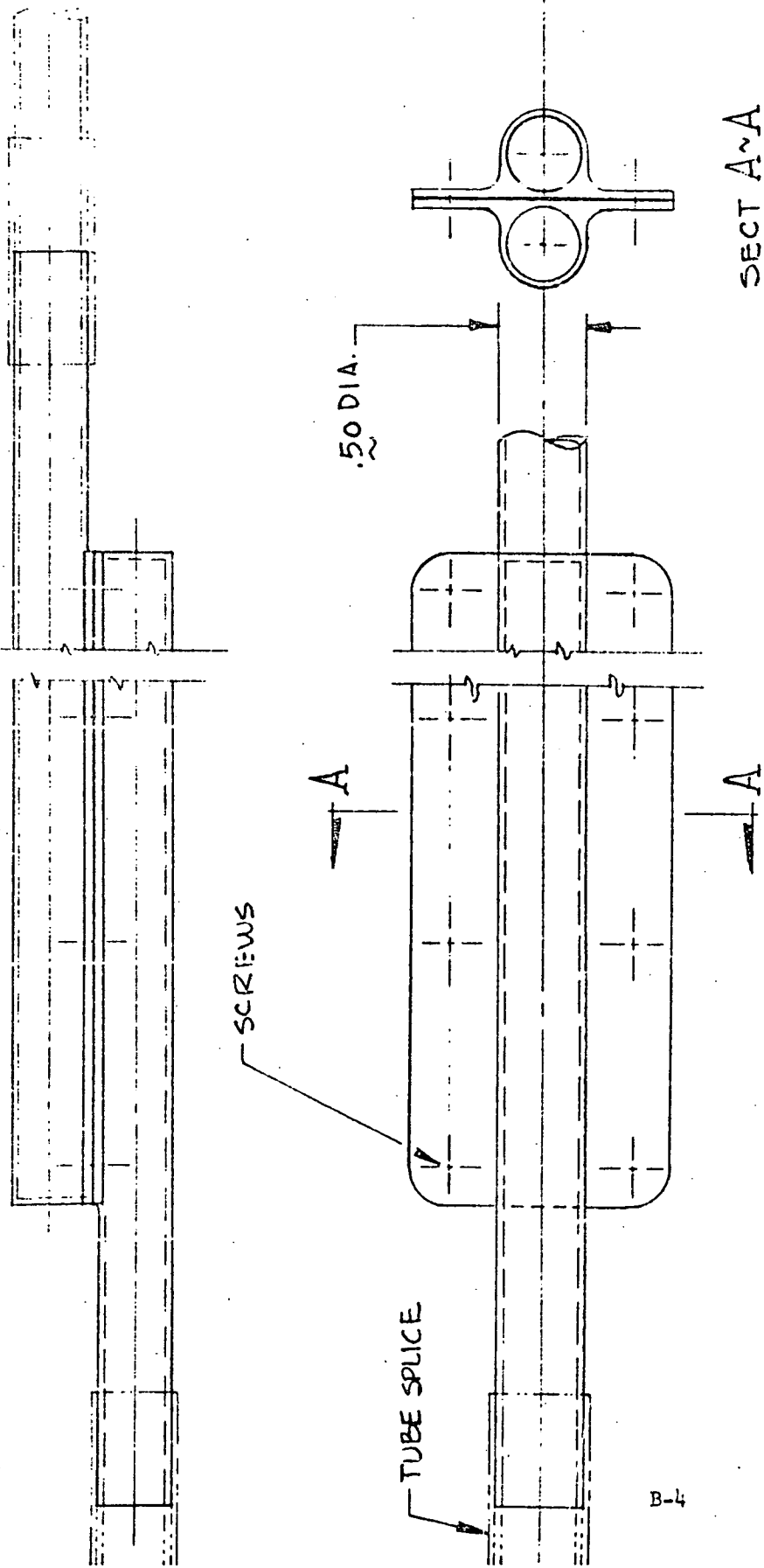
(AVG. 1.12[#]/FT)

FIGURE B-1.1
CONDUCTIVE INTERFACES - SADDLES BOLTED

H.P. EXTENSION
CLAMPING



7/14



TWO EXTRUDED SECTIONS
 BACK TO BACK.- AL ALLOY 2219.
 ONE END SEALED (WELDED).
 OTHER END MACHINED TO
 REMOVE FLANGE AND SIZE
 FOR ADJACENT TUBE AND
 SLEEVE. SECT. A-A WT. = .387" / FT
 LESS HARDWARE.

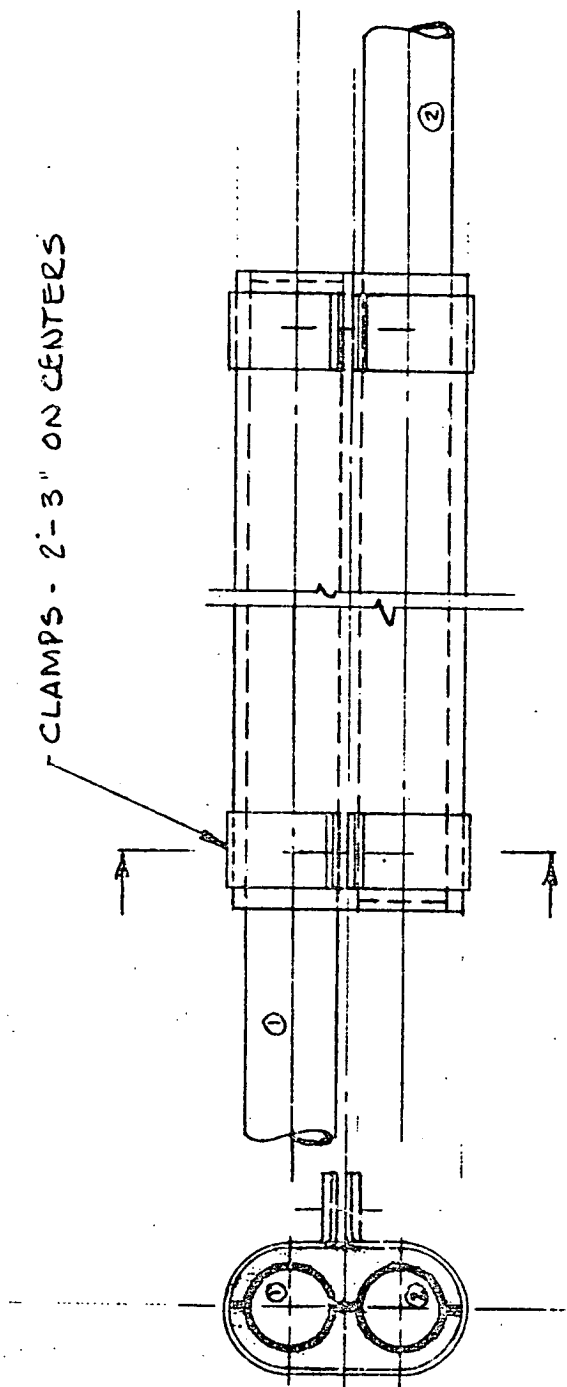
CONDUCTIVE INTERFACES--FLANGE BOLTED

FIGURE B-1.2



Hand-drawn notes and markings along the right edge of the page, including a vertical line and some illegible scribbles.

CLAMPS - 2-3" ON CENTERS



B-5
1/2 O.D. PIPES
.46"/FT W/O PIPES
.04"/CLAMP
(.62# TOT./FT APPROX)

CONDUCTIVE INTERFACES - SADDLES CLAMPED

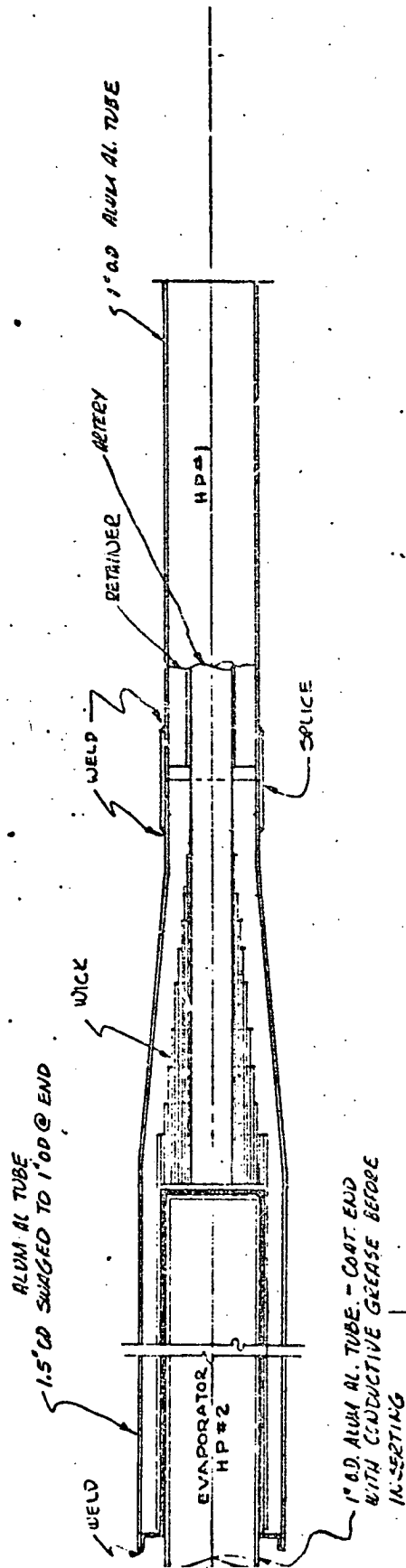
FIGURE B-1.3



1/2" O.D. PIPES

10/1/77

DIRECTION OF HEAT FLOW



HEAT PIPE CONDUCTIVE INTERFACE - SOCKET TYPE

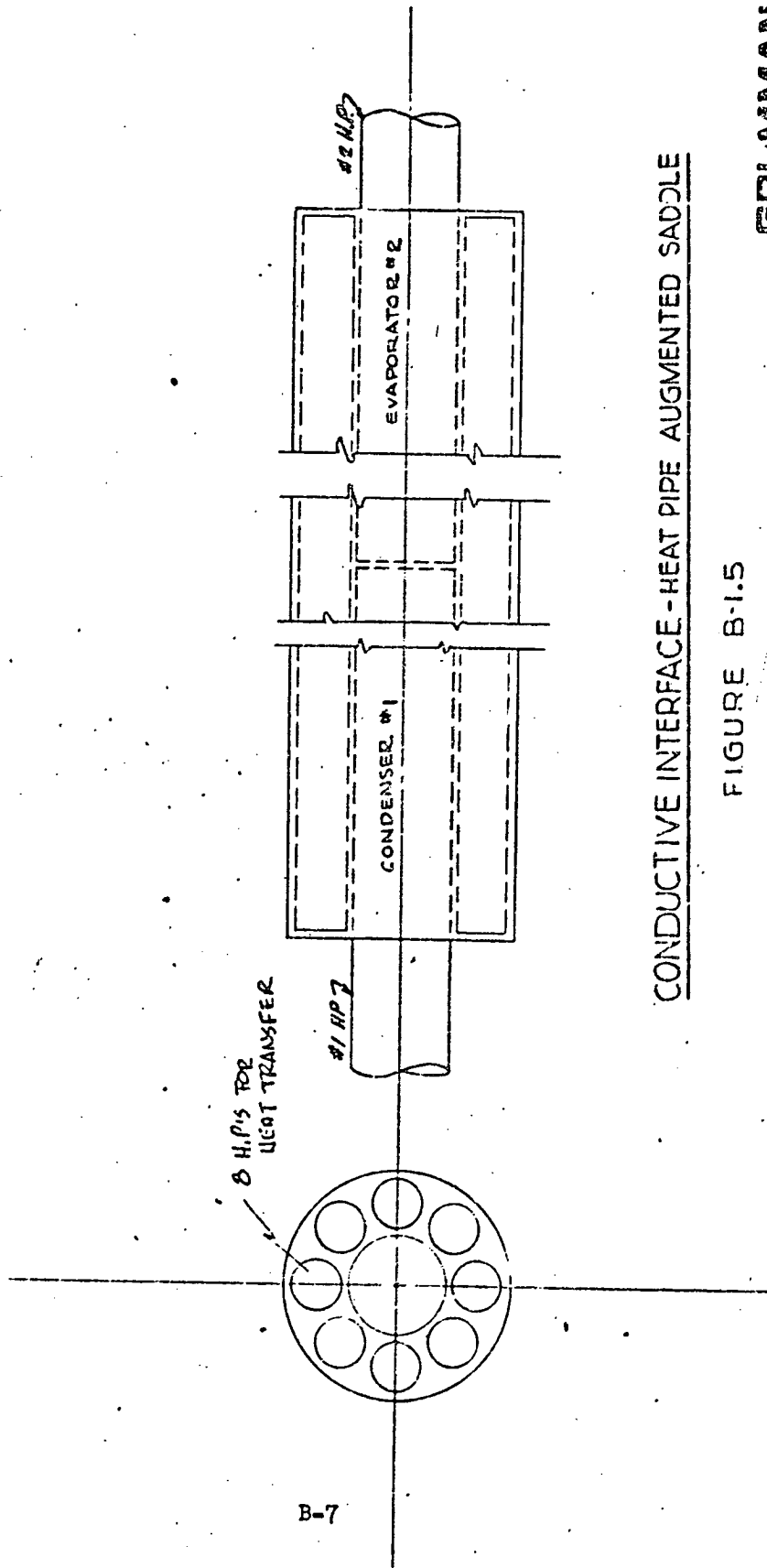
END OF 2ND H.P. NESTS INTO END OF 1ST H.P.

FIGURE B-1.4

B-6

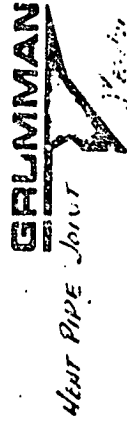


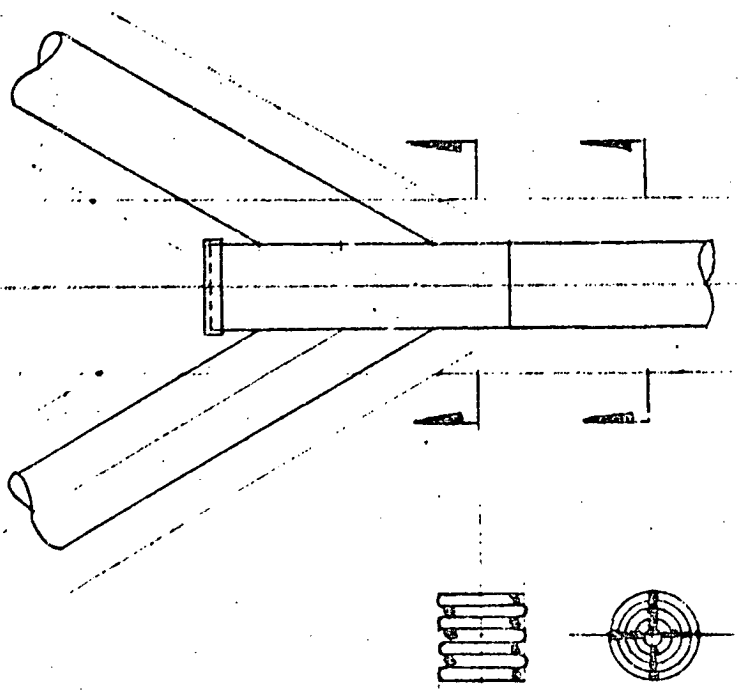
Smith



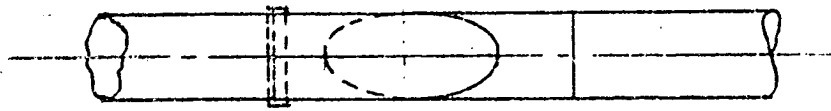
CONDUCTIVE INTERFACE - HEAT PIPE AUGMENTED SADDLE

FIGURE B-1.5

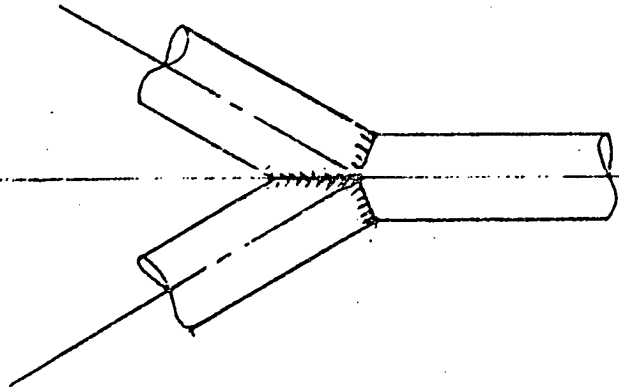




ARTERY "Y" SPLICES
FIGURE B-1.6

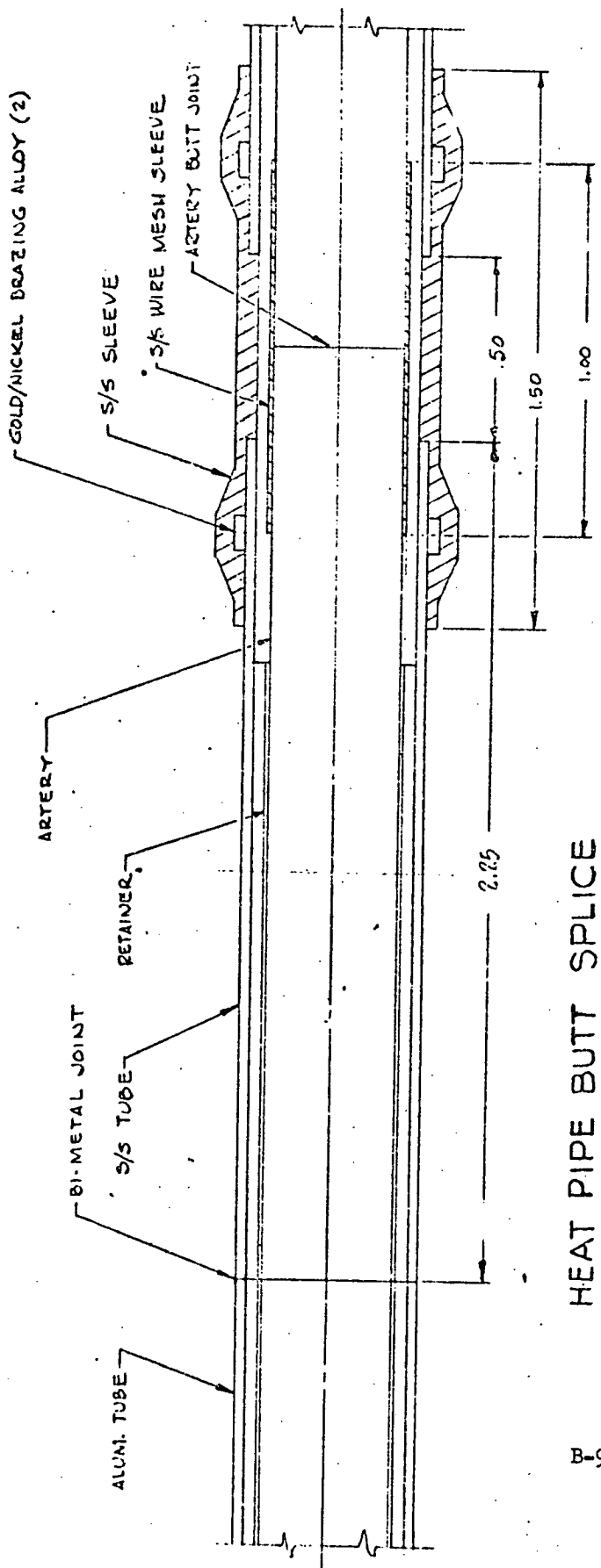


GERLMAN



ALTERNATE

WICK "Y" JOINT

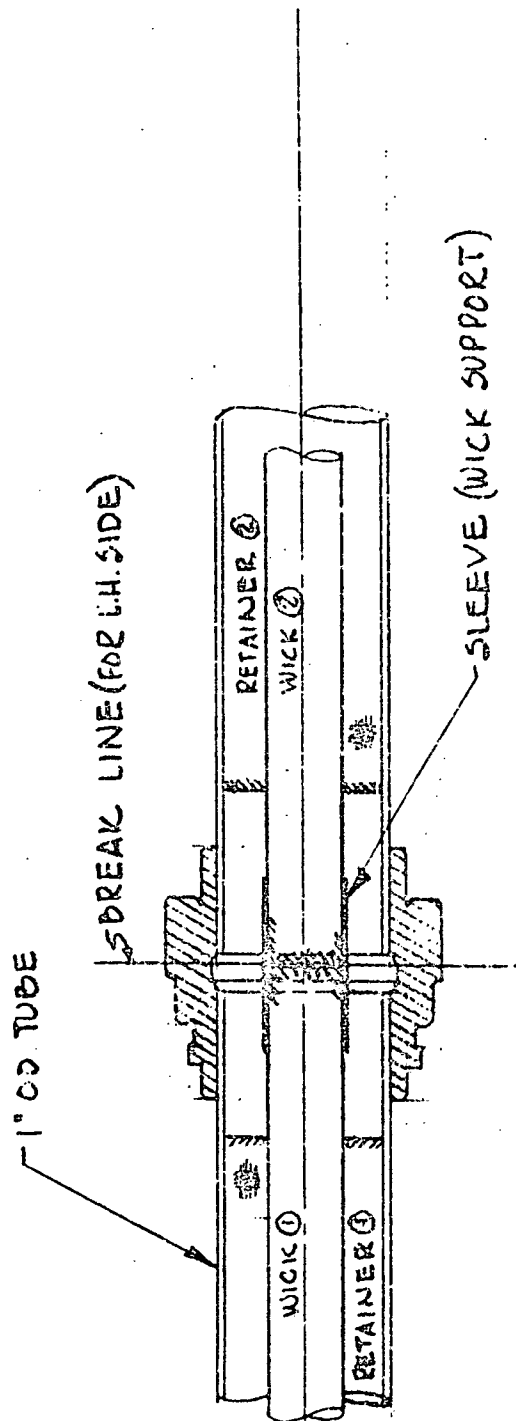


HEAT PIPE BUTT SPLICE

FIGURE B-1.7

H.P. JOINT
(4x SIZE)





HEAT PIPE BUTT SPLICE
 USING JT14 GAMAH FTG'S

FIGURE B-1.8

GRUMMAN

8/8/71

APPENDIX B-2

DESIGN CURVES FOR SELF-PRIMING HIGH-CAPACITY HEAT PIPES

In most potential aerospace applications of heat pipes, minimum weight is a requirement, forcing the designer to use the heat pipe of smallest diameter which is capable of transporting the heat load. Although many parametric computer runs were made to determine the effects of wick parameters on performance, this information was not sufficient to enable the designer to choose the pipe diameter and working fluid to be used in a particular heat pipe application. For this reason, it was decided to see if a relationship could be found between capacity and general heat pipe characteristics which would serve as a preliminary sizing tool for design.

The first step taken was to optimize wick parameters for heat pipes of several diameters and working fluids in order to maximize capacity (based on groove dryout). It was decided to limit this study to pipes with inside diameters of 0.5, 0.75, and 1.0 inches, and, initially, to the working fluids ammonia, water, and Freon-21. The fluid properties used were those of ammonia and Freon-21 at 90°F, and those of water at 100°F, as these are normal operating temperatures of electronics equipment, a typical shuttle heat pipe application. The evaporator and condenser sections of the heat pipes for this study are threaded to minimize fluid layer thickness and temperature drops through the fluid. Threads as fine as is practical are used; for this study, a pitch of 150 threads per inch, a width of 0.002 inches, a root of 0.001 inches, and a depth of 0.005 inches were used, these figures being easily attainable. The wick parameters varied to produce optimized wick designs were the tunnel core diameter, the gap spacing, the number of retainer webs, and the number of spiral wraps (Figure B-2.1). A limitation on the diameter of the tunnel core of 0.2 inches was set. Although larger diameters are possible and theoretically provide a greater maximum capacity, they represent an extension of test results and are therefore less conservative. Core diameters as small as 0.1 inches were used. These parameters were varied to produce the maximum calculated capacity in a heat pipe with a 48 inch evaporator section and a 48 inch condenser section. These dimensions were chosen because some parametric information on heat pipes of these dimensions was already available.

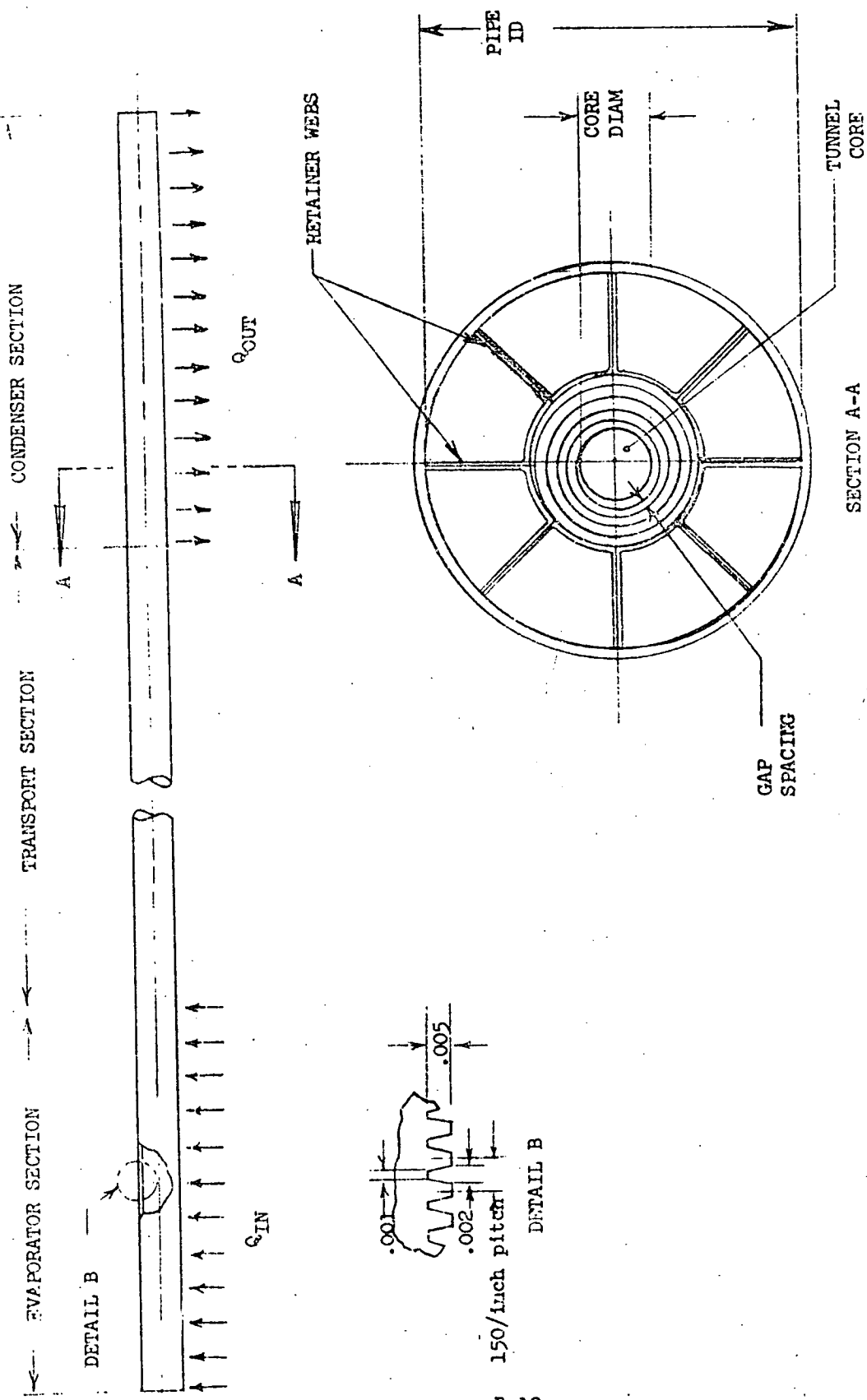


Figure B-2.1



Once optimum wick designs had been chosen, the capacities of heat pipes with 48 inch evaporator sections, 48 inch condenser sections, and adiabatic transport sections of differing lengths were determined. Using the 1.0 inch ID ammonia-filled heat pipe as an example, these capacities were plotted against the effective length of the heat pipe (Figure B-2.2). As the resulting curves appeared hyperbolic, it was decided to plot the same data against a logarithmic scale (Figure B-2.3). A pair of straight lines resulted, one showing the relationship between capacity and effective length when the vapor flow was turbulent; the other (for longer lengths) showed this relationship for laminar vapor flow.

Similar data points were generated for 0.75 inch and 0.50 inch diameter pipes and for water and Freon-21, in all pipe diameters. These data are plotted, by fluid type, in Figures B-2.4, B-2.5, and B-2.6.

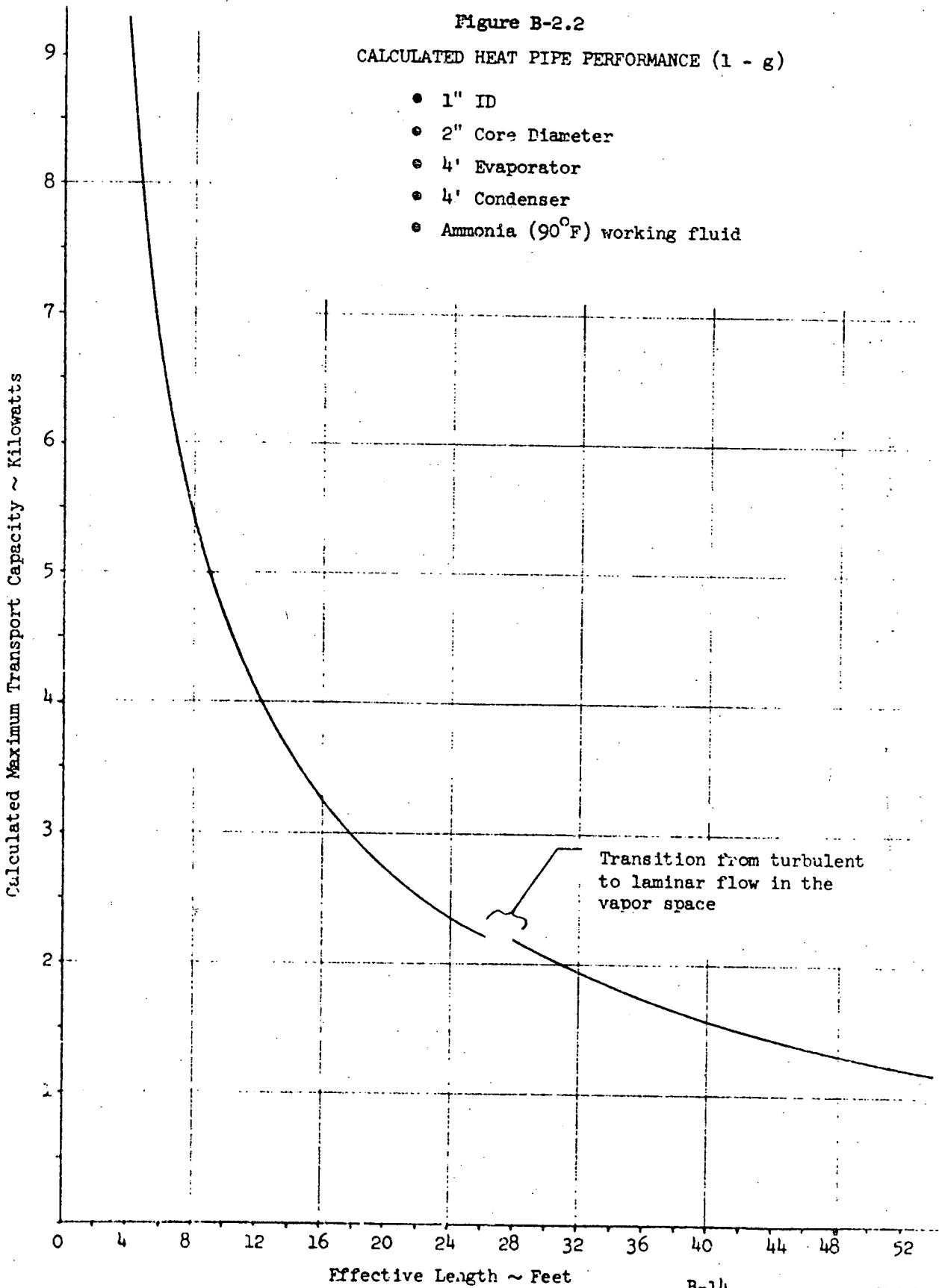
To expand the applicable scope of this information similar curves were generated with 24 inch evaporator and condenser sections and plotted on the same graphs to show the effects of varying the evaporator and condenser lengths. No significant difference in performance was noted except in heat pipes of very short effective length. A heat pipe with a 48 inch evaporator and a 48 inch condenser has the same effective length (4 feet) as a heat pipe with a 24 inch evaporator, 24 inch adiabatic transport section, and a 24 inch condenser section, but the former will have a higher capacity. The slight difference in capacity is due to the increased pressure drop in the grooves and webs at the higher heat flux levels associated with the shorter evaporator and condenser lengths.

The curves in Figures B-2.4, B-2.5, and B-2.6 were then used as design tools in each specific shuttle application. The required load to be transported and the distance from heat source to sink being known, the corresponding design point could be found on each of the three charts. If it appeared below the optimum design line, the pipe corresponding to that line was used; if it appeared above the line, either a different heat pipe or a number of heat pipes connected in series or parallel would be used.

Figure B-2.2

CALCULATED HEAT PIPE PERFORMANCE (1 - g)

- 1" ID
- 2" Core Diameter
- 4' Evaporator
- 4' Condenser
- Ammonia (90°F) working fluid



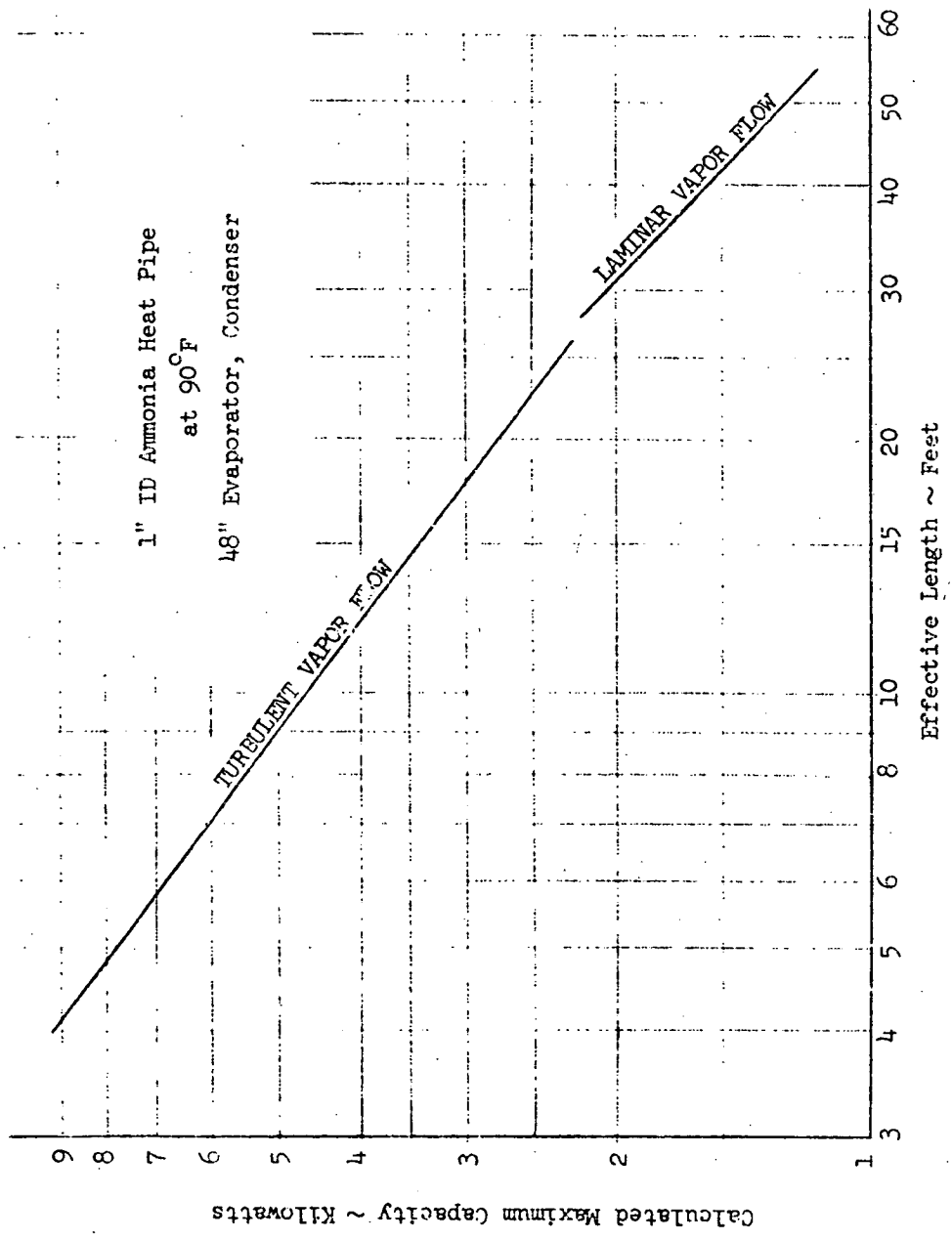


Figure B-2.3

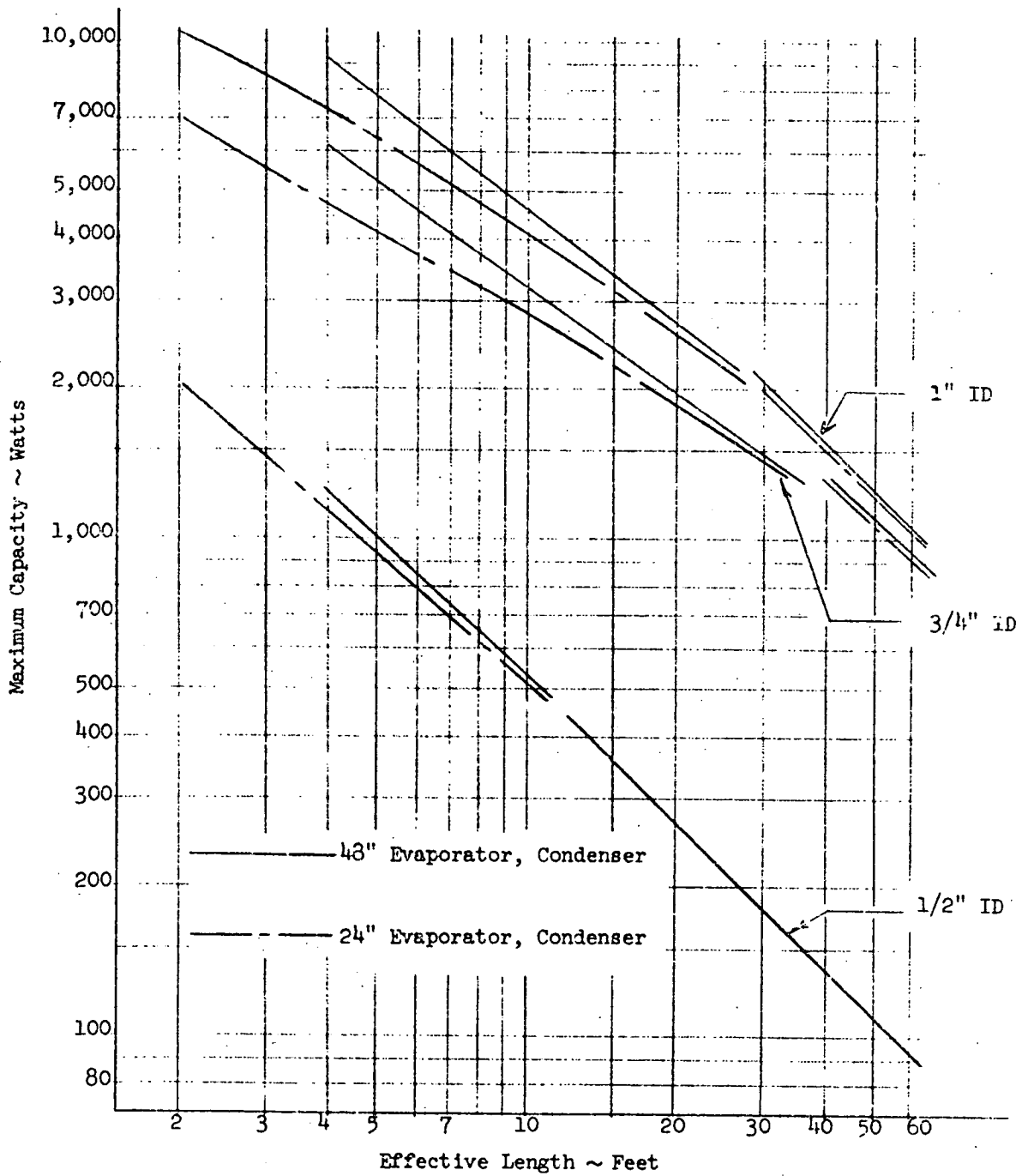


Figure B-2.4: HEAT PIPE DESIGN SIZING CURVES FOR AMMONIA (90°F)

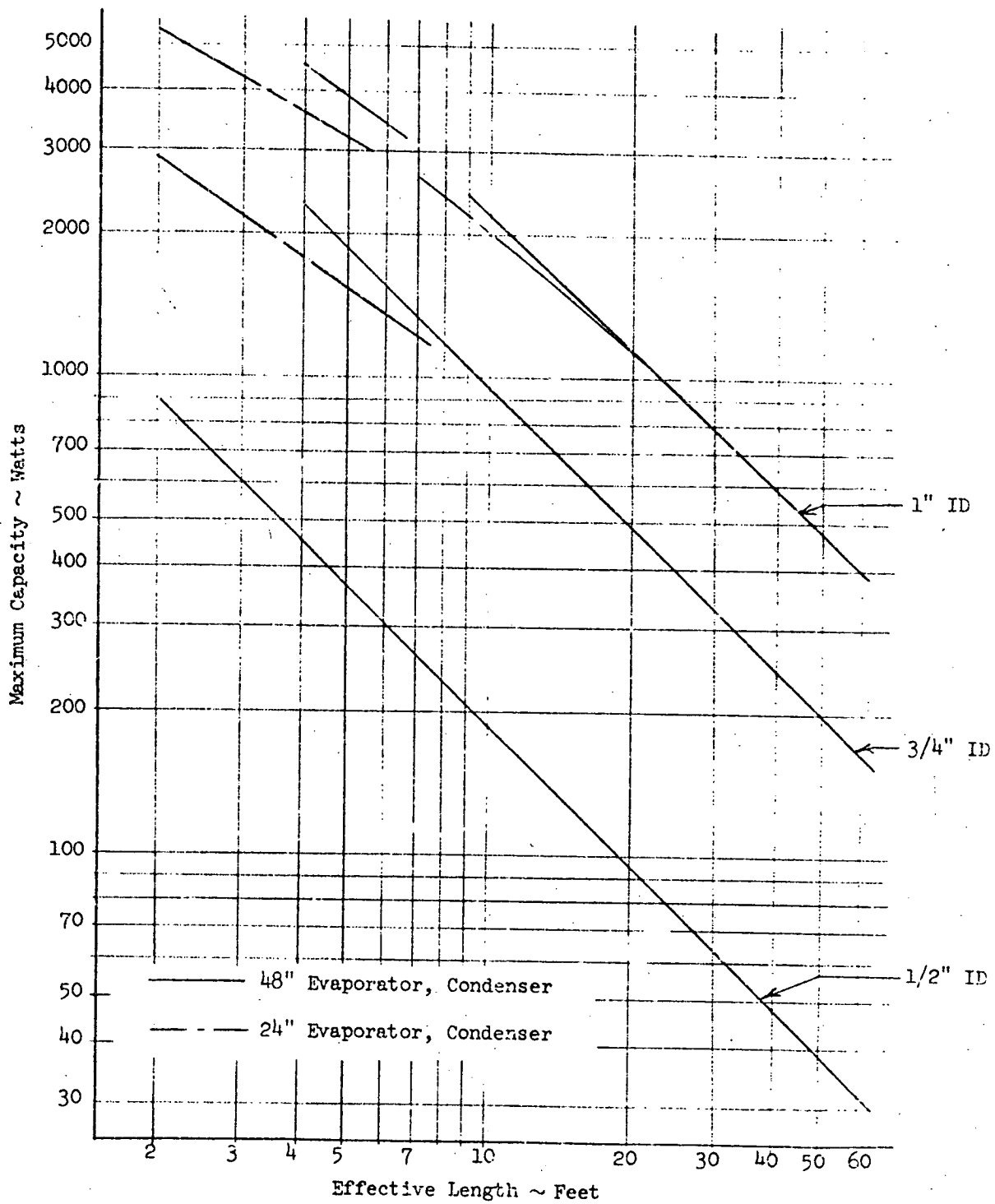


Figure B-2.5: HEAT PIPE DESIGN SIZING CURVES FOR WATER (100°F)



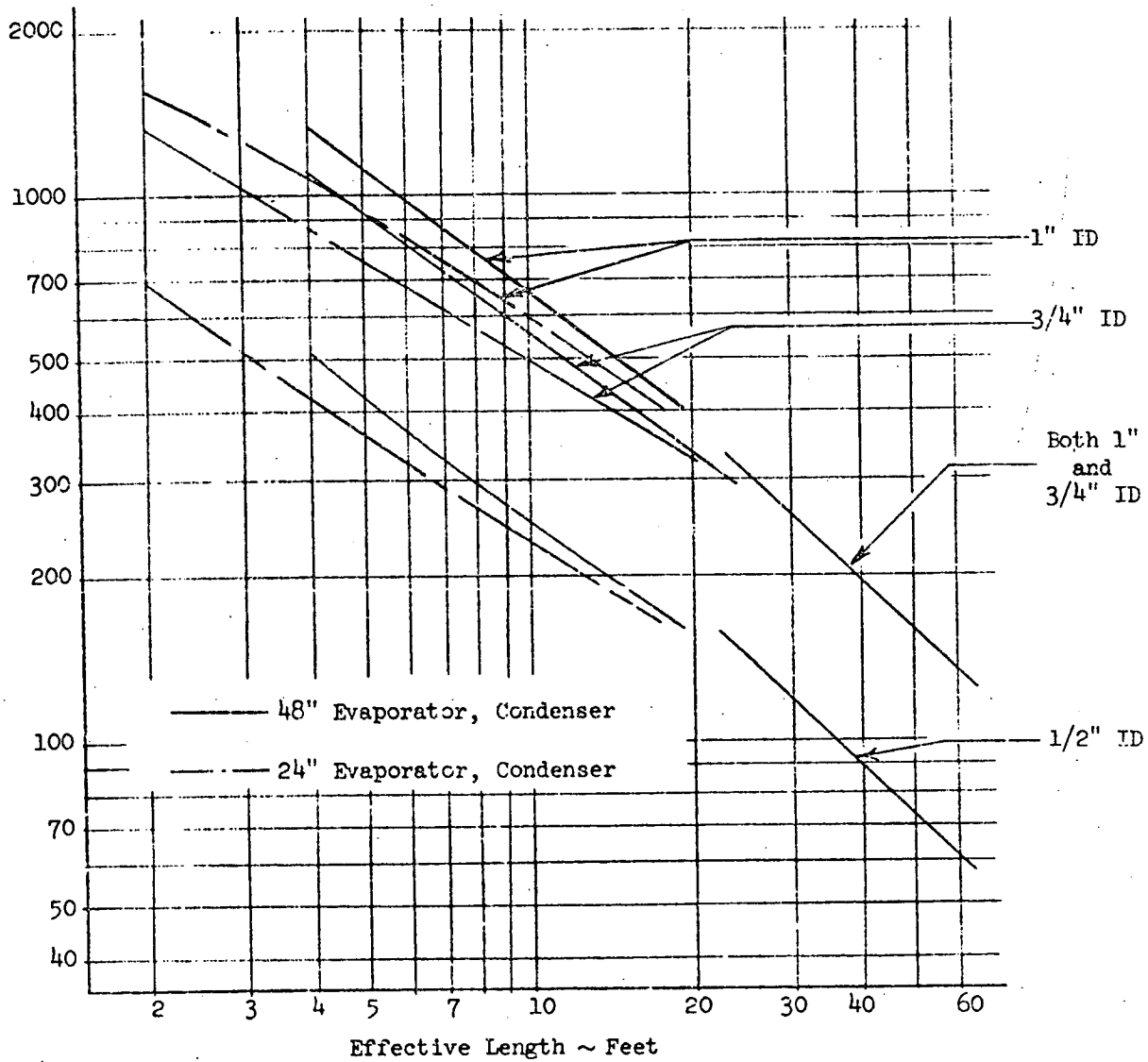


Figure B-2.6: HEAT PIPE DESIGN SIZING CURVES FOR FREON-21 (90°F)

As a confirmation of the pipe selections the capacities of all the heat pipes used in the preliminary application designs were calculated. In no case was the design point based on the curves significantly different from the specific calculations.

Temperature Drop - It is not sufficient to know the capacity of a heat pipe; the penalty that must be paid to transport this load, in the form of a temperature drop, must also be known for proper system definition. The temperature drop through the pipe is essentially a function of the fluid film coefficients at the evaporator and condenser. Overall heat transfer coefficients (experimentally determined for ammonia and water, and estimated for Freon-21) are given in Table B-2.1. They specify the rate of heat transfer from the outside wall of the heat pipe to the vapor space. The area through which the heat flux takes place was taken to be that of the inside wall of the pipe.

TABLE B-2.1

OVERALL HEAT TRANSFER COEFFICIENTS (Btu/hr-ft²-°F)

Fluid	Evaporator	Condenser
Ammonia	900	2900
Water	500	2000
Freon-21	450	1740

Note: Subsequent development and testing at Grumman has resulted in increases in heat transfer coefficients by a factor of 3.5 for the evaporator and 1.5 for the condenser.

A chart was sought that would make detailed, repeated calculations using this data unnecessary. Since:

$$Q = UA \Delta T = UL\pi D \Delta T, \text{ then}$$

$$\Delta T = \frac{1}{U\pi D} \frac{Q}{L},$$

giving a relationship between the temperature drop and the loading of the heat pipe per unit length, for a given diameter heat pipe. Using the data in Table

B-2.1, curves were plotted to produce Figures B-2.7, B-2.8, and B-2.9 for the three fluids, and pipe diameters considered in this memorandum.

These curves are used by calculating the load per unit length of the evaporator and condenser in watts/inch, reading up to the appropriate curves, and then reading across to find the temperature drops in the evaporator and condenser sections in $^{\circ}$ F. These are added to determine the temperature drop through the entire heat pipe. Temperature drops through any conductive interfaces must be calculated separately.

If the heat pipe is attached to a surface by means of a saddle, only a fraction of the pipe area at the inside diameter may be considered as effective heat transfer area, and the temperature drop obtained from Figures B-2.7, B-2.8, and B-2.9 must be multiplied by the inverse of this fraction to obtain the true temperature drop.

The coefficients in Table B-2.1 come from heat pipes with 80 to 90 threads per inch, and so must be used with caution, as the pipes considered in this memorandum have 150 threads per inch (see note in table). This should make the curves slightly conservative, as the finer threads produce a thinner, more evenly spread fluid layer. Development work in this area is continuing and the data will be presented as it becomes available (see, for example, reference 18).

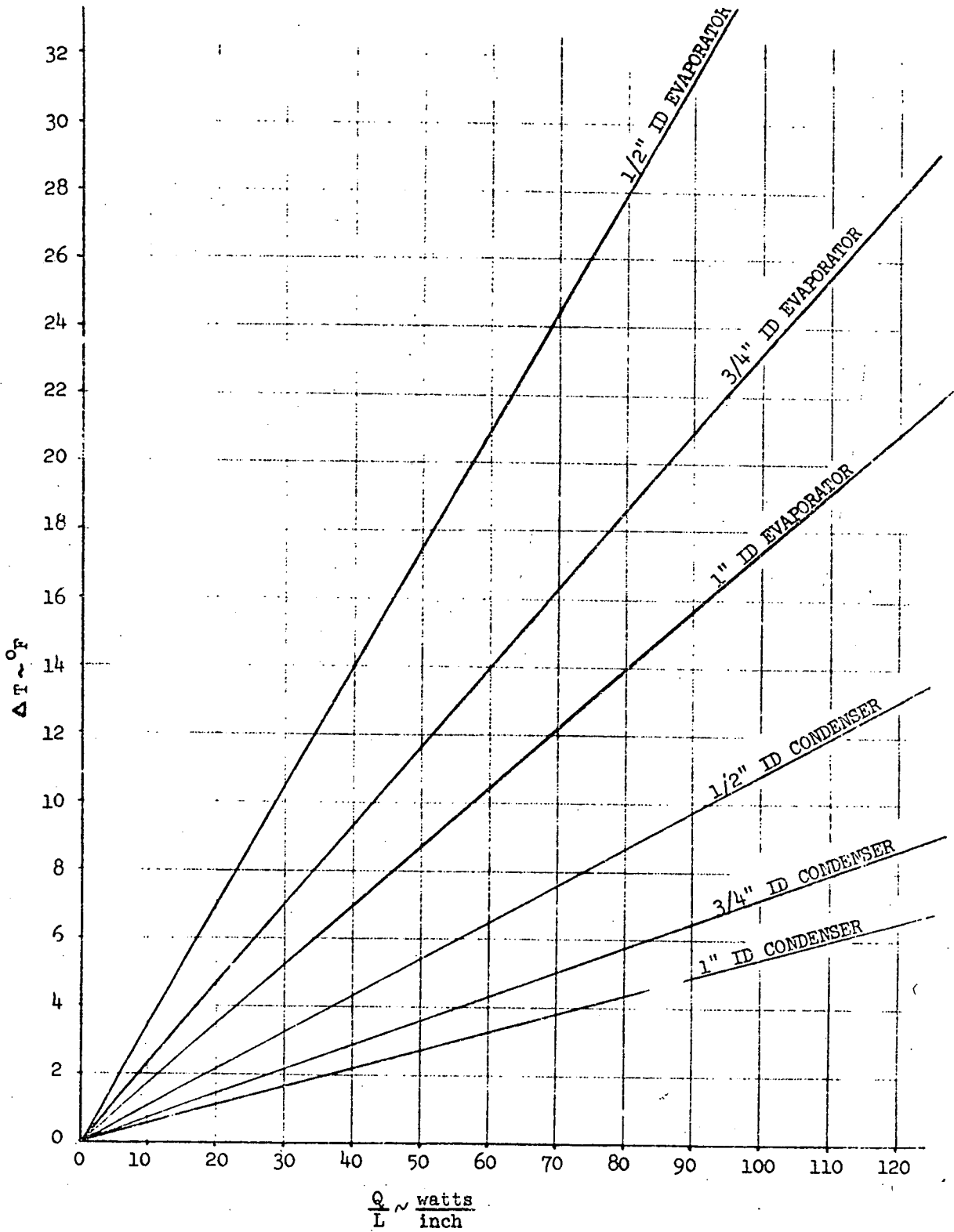


Figure B-2.7: ΔT ESTIMATION CHART FOR AMMONIA HEAT PIPES

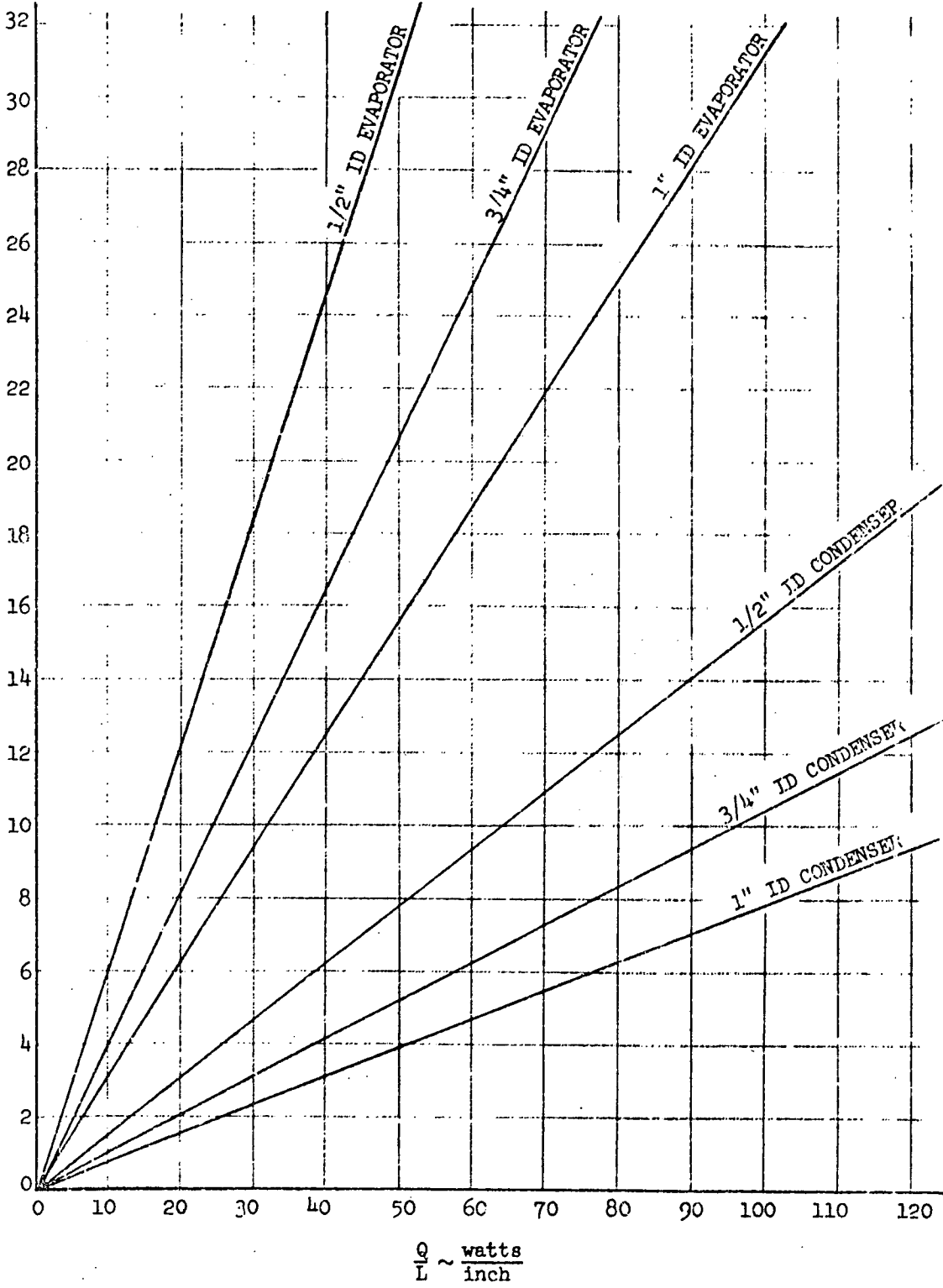


Figure B-2.8: ΔT ESTIMATION CHART WATER HEAT PIPE

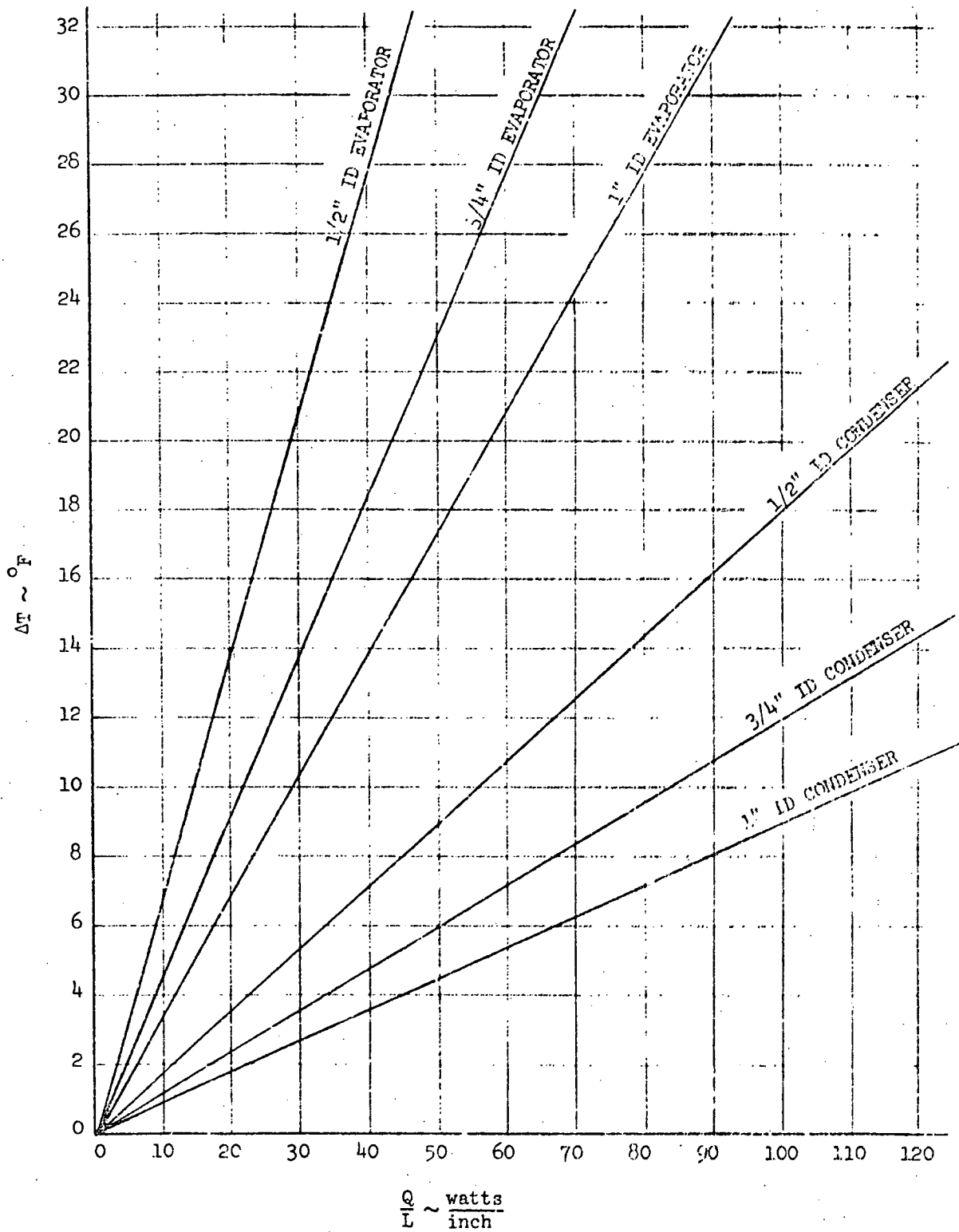


Figure B-2.9: ΔT ESTIMATION CHART FREON-21 HEAT PIPE

APPENDIX C-1

Evaluation Matrix - Heat Pipe Applications

C-1.1

APPLICATION: Leading Edge TPS - Heat pipes isothermalyze circumferential and longitudinal temperature gradients in local high heating rate areas.

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient		X	Isothermalizes Leading Edge
2. Capacity Margin		X	Heat pipes can transfer twice the load required of them.
3. Power Requirements	-	-	None Required
4. Control Requirements	-	-	None Required
5. Weight	X		Weight of heat pipe structure
6. Safety	-	-	Must be failsafe in any case.

REMARKS: Requires development effort in bonding heat pipes to carbon/carbon backface or coming up with an integral carbon/carbon heat pipe envelope.

Evaluation Matrix - Heat Pipe Applications

C-1.2

APPLICATION: Landing Gear - on-orbit temperature control of wheel well compartments using F-21 HP radiator.

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	Brazed Joints
2. Capacity Margin		X	Excess capacity inherent in HP design. It's not flow rate limited.
3. Power Requirements		X	Pumped loop incurs pressure drop through radiator.
4. Control Requirements		X	Built in diode control prevents hot case compartment load feedback to ECS.
5. Weight		X	.95 lb/ft ² vs 1.30 lb/ft ²
6. Safety	-	-	Both use F-21 which is non-toxic (Group 4-5)

REMARKS: Baseline system uses F-21 fluid radiator; HP system uses F-21 working fluid.

Evaluation Matrix - Heat Pipe Applications

C-1.3

APPLICATION: Heat Transport System - All HP system from source to sink.

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	X		Excessive ΔT 's. Transport lengths too long; too many joints.
2. Capacity Margin		X	Excess capacity in heat pipes
3. Power Requirements		X	No pumping power required for HP circuit.
4. Control Requirements		X	None Required.
5. Weight	X		Additional radiator weight needed for lower rejection temperatures.
6. Safety	X		Ammonia heat pipes needed for external transport system.

REMARKS: Temperature gradients are so large that system becomes marginal, radiator area is insufficient.

The avionics HP circuit is treated in a separate evaluation.

Evaluation Matrix - Heat Pipe Applications

C-1.4

APPLICATION: Avionics HP Circuit - HP's couple avionics to ECS fluid header.

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	Equipment can operate at higher temperatures and still reject heat. Some ΔT at fluid header interface.
2. Capacity Margin		X	Excess capacity in heat pipes
3. Power Requirements		X	No flow pressure losses with HP's.
4. Control Requirements		X	Eliminates flow splits and balance problems.
5. Weight	-	-	About same
6. Safety	-	X	No fluid connections, therefore, less leaks and contamination.

REMARKS:

C-4

Evaluation Matrix - Heat Pipe Applications

C-1.5

APPLICATION: HP/Phase Change Remote Sink.

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	
2. Capacity Margin	-	-	
3. Power Requirements		X	No flow pressure losses
4. Control Requirements		X	None Required for HP's
5. Weight	-	-	
6. Safety	-	-	

REMARKS: Although feasible, there are no definite applications on the Shuttle at this time.

Evaluation Matrix - Heat Pipe Applications

C-1.6

APPLICATION: Air Cooled Commercial and Military Equipment (ATR Equipment)

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	
2. Capacity Margin	-	-	Depends on Air Hx design
3. Power Requirements		X	Lower pressure drop for ECS fluid .
4. Control Requirements	-	-	
5. Weight	-	-	
6. Safety		X	No fluid connections near equipment; no leaks and no contamination.

REMARKS: Baseline system would use ECS fluid/air heat exchanger in the enclosed equipment rack.

Evaluation Matrix - Heat Pipe Applications

C-1.7

APPLICATION: Flight and Voice Recorder - Remote Heat Sink

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	Both systems designed for same requirement.
2. Capacity Margin	-	-	Limited by expendable sink
3. Power Requirements		X	None required - no flow losses
4. Control Requirements		X	HP's are self-controlling
5. Weight		X	Eliminates long fluid line runs and controls.
6. Safety	-	-	Less leakage hazard with heat pipes.

REMARKS:

Evaluation Matrix - Heat Pipe Applications

C-1.8

APPLICATION: HP Radiator

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	Drazed HP attachments
2. Capacity Margin	-	-	Both systems would be designed for same capacity
3. Power Requirements		X	Less power required due to reduced pressure losses in radiator panel.
4. Control Requirements		X	Eliminates flow splits and balance problems.
5. Weight		X	.95 lb/ft ² <u>vs</u> 1.30 lb/ft ²
6. Safety		X	<u>Baseline</u> : Fitting leaks, tube puncture probability. <u>HP</u> : Limited amount of ammonia in HP header (≈ 200 gms)

REMARKS: Supply Header Possibilities

1. VCHP header coupled to all heat pipe transport loop.
2. Hybrid VCHP/fluid header serving as a heat exchanger interface between the heat pipe radiator panel and a fluid heat transport loop.
3. Individual feeder heat pipes on the panel coupled directly to a fluid supply header.

Evaluation Matrix - Heat Pipe Applications

C-1.9

APPLICATION: Heat Pipe Augmented Cold Rail

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient		X	Baseline can't handle power densities
2. Capacity Margin		X	Baseline can't handle power densities
3. Power Requirements		X	Eliminates requirement for less dense packing and longer lengths of rail.
4. Control Requirements	-	-	Not applicable.
5. Weight		X	Eliminates requirement for less dense packing and longer lengths of rail.
6. Safety	-	-	

REMARKS: Baseline System (LM-type ECS cold rail) cannot handle high density packaging; this unit can. Without the heat pipe augmented cold rail, previous generation electronics would have to be substituted for the newer and more compact equipment. This would result in heavier pieces of equipment and fewer "boxes" mounted per rail. The latter requires longer lengths of cold rail which would result in increased flow pressure losses and rail weight.

Evaluation Matrix - Heat Pipe Applications

C-1.10

APPLICATION: HP Radiator System for Fuel Cell

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	
2. Capacity Margin	-	-	
3. Power Requirements		X	See C-1.8
4. Control Requirements		X	
5. Weight		X	
6. Safety		X	

REMARKS: Same as C-1.8 except at higher temperature. Will not be developed as a separate application, but instead will be lumped with C-1.8.

Evaluation Matrix - Heat Pipe Applications

C-1.11

APPLICATION: Air Breathing Engine Compartment

CRITERIA	RATING COMPARED TO BASELINE		COMMENTS
	WORSE THAN	BETTER THAN	
1. Temperature Gradient	-	-	
2. Capacity Margin		X	See C-1.2
3. Power Requirements		X	
4. Control Requirements		X	
5. Weight		X	
6. Safety	-	-	

REMARKS: Similar in concept to C-1.2. It will not be treated as a separate application, but will be lumped with C-1.2.

PRECEDING PAGE BLANK NOT FILMED

APPENDIX C-2

HEAT PIPE RADIATOR FOR WASTE HEAT REJECTION

Before Grumman was requested to halt investigation of heat pipe waste heat rejection radiators under this study contract, some preliminary analysis had been performed. This appendix documents that work and the computer program written to calculate the heat rejection capacity of a heat pipe radiator.

The baseline shuttle radiator (reference 1) consists of 700 ft² of surface area broken into ten radiator panels mounted on the inside of the cargo bay doors. Under maximum heat load conditions, fluid inlet temperature to the system is 117.2°F, and fluid outlet temperature is 35.5°F. The working fluid is Freon-21, flowing at 2,000 lb/hr in parallel through the panels. The heat load to be rejected is 40,800 Btu/hr; the environmental heat flux absorbed is 45 Btu/hr ft². Overall radiator efficiency is 0.9.

A computer program to analyze the capability of any of the header configurations depicted in Section 5 of this report was written. To achieve maximum flexibility, the program was made as general as possible. It assumed a flat radiator with an unrestricted view to space. A segment of the model is shown in Figure C-2.1. Fluid inlet and outlet temperatures, and a mean fluid

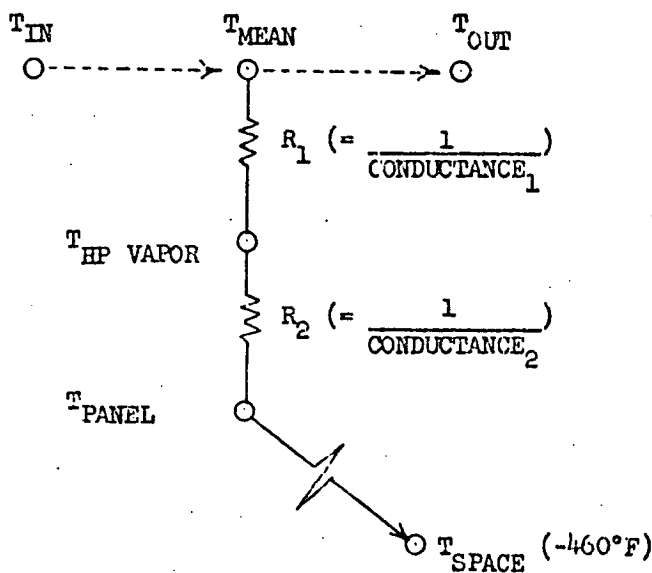


FIGURE C-2.1
C-13



temperature, are indicated. Heat is transferred from the fluid to the heat pipe vapor temperature, which could correspond either to the vapor temperature of a VCHP header or a feeder heat pipe. From this point heat is transferred into the panel, and then radiated to space. The following equations describe the performance of this model:

$$Q_{REJ} = \eta \left[\epsilon A \sigma T_{PANEL}^4 - Q_{ABSORBED} \right] \quad (1)$$

$$T_{OUT} = T_{IN} - \frac{Q_{REJ}}{\dot{m} C_P} \quad (2)$$

$$T_{MEAN} = T_{VAPOR} + \frac{T_{IN} - T_{OUT}}{\ln \left(\frac{T_{IN} - T_{VAPOR}}{T_{OUT} - T_{VAPOR}} \right)} \quad (3)$$

This equation uses the logarithmic mean temperature difference from heat exchanger theory as a ΔT .

$$T_{VAPOR} = T_{MEAN} - \frac{Q_{REJ}}{COND_1} \quad (4)$$

$$T_{PANEL} = T_{VAPOR} - \frac{Q_{REJ}}{COND_2} \quad (5)$$

These represent five equations in five unknowns. Defining $R' = \epsilon A \sigma T_{PANEL}^4 - Q_{ABSORBED}$, and combining the above five equations:

$$T_{IN} - T_{PANEL} + \left(\frac{\eta}{\dot{m} C_P} \frac{e^{\frac{COND_1}{\dot{m} C_P}}}{1 - e^{\frac{COND_1}{\dot{m} C_P}}} - \frac{\eta}{COND_2} \right) R' = 0 \quad (6)$$

Since $R' = R'(T_{\text{PANEL}})$, this equation can be solved numerically for T_{PANEL} . Once T_{PANEL} is known, Q_{REJECTED} and then T_{OUT} may be found. To solve a radiator system containing a number of such sections (or panels) in series, let T_{IN} equal the calculated value of T_{OUT} and resolve the equation. A computer program which solves equation (6) by Newton's method and, if required, solves it again using its own calculated values as input for a series of panels is presented in this appendix.

By properly defining R_1 and R_2 , panels with fluid headers and feeder heat pipes and panels with hybrid fluid/VCHP headers can be modelled. For the hybrid fluid/VCHP header:

$$R_1 = \left(\frac{1}{hA}\right)_{\text{fluid to wall}} + \left(\frac{1}{hA}\right)_{\text{VCHP evaporator}}$$

$$R_2 = \left(\frac{1}{hA}\right)_{\text{VCHP condenser}} + \left(\frac{1}{hA}\right)_{\text{VCHP/feeder HP interface area}}$$

$$+ \left(\frac{1}{hA}\right)_{\text{feeder evaporator}} + \left(\frac{1}{hA}\right)_{\text{feeder condenser}}$$

Using these values, $T_{\text{HP VAPOR}}$ corresponds to the vapor temperature in the VCHP, and T_{PANEL} corresponds to the outside wall temperature of the feeder heat pipe condensers, the "root of the fin" temperature of the panel. For the case with a fluid header and feeder heat pipes in the panel:

$$R_1 = \left(\frac{1}{hA}\right)_{\text{fluid to wall}} + \left(\frac{1}{hA}\right)_{\text{feeder evaporator}}$$

$$R_2 = \left(\frac{1}{hA}\right)_{\text{feeder condenser}}$$

Using these values, $T_{\text{HP VAPOR}}$ corresponds to the vapor temperature of the feeder heat pipe, and T_{PANEL} to the outside wall of the feeder heat pipe, again the "root of the fin" temperature.

The first concept to be investigated was that of the heat pipe radiator system closest to the baseline: ten panels in parallel, with hybrid fluid/VCHP headers. The capacity of such a system is insufficient, as can be quickly demonstrated. The VCHP must operate at a temperature below the fluid outlet temperature, as the heat pipe can be looked at as cooling the fluid to that level. Further, the feeder heat pipes must in turn operate at a lower temperature than the VCHP header in order to pick up heat from the header. The effective panel temperature will be even lower.

The required heat rejection rate for one panel is readily calculated; including the environmental heat input, it is 103.6 Btu/hr ft². In order to reject heat at this rate, a black body must be at 36.4°F, higher than the fluid outlet temperature. Such a heat pipe radiator arrangement is therefore impractical.

The radiator panel configuration is shown in Figure SPL-113, Sheet 2, in Section 5 of this report. The 1" ID VCHP ammonia-filled header has a 6.5' evaporator immersed in an internally finned fluid header and a 6.5' condenser assumed bonded (by brazing or soldering) to the feeder heat pipe evaporator sections. The ammonia feeder heat pipes (13 per panel) have 5.75" evaporators and 11' condensers. R_1 and R_2 were calculated, and the computer program run assuming two parallel groups (one on each door) of five panels in series. Methods used to calculate the resistances are the same as those used in Section 9.2 of this report, the heat pipe circuit.

Results of this computer run, with $R_1 = 2.28 \times 10^{-3} \frac{^\circ\text{F}}{\text{Btu/hr}}$ and $R_2 = 2.102 \times 10^{-3} \frac{^\circ\text{F}}{\text{Btu/hr}}$, are listed in Table C-2.1. Using the computer results, operating temperatures of all the heat pipes employed were calculated and are included in this table.

As shown, the heat rejected by the uncontrolled system exceeds the requirements slightly (in practice, part of one of the panels would be shut off by the VCHP, providing the desired outlet temperature).

Such a system could easily be built with redundant headers, and micro-meteoroid protection of feeder heat pipes is unnecessary, so in terms of safety, the self-controlled heat pipe radiator system is the equal of the baseline. Since the heat pipe system contains no valves or moving parts, it is also more reliable than the baseline.

HEAT PIPE RADIATOR ROUTINE

```

C
C INPUT AND INITIALIZATION
  READ (2,100) TP, LM
  READ (2,101) TIN, ETA, CPM, COND1, COND2, QABS, EPS, AREA
  EX=EXP(COND1/CPM)
  Z=(ETA/CPM)*(EX/(1.-EX))-ETA/COND2
  I=0
  TPR=TP+460.
  RTP=EPS*AREA*.1713E-8*TPR**4-QABS
C BEGIN CALCULATIONS USING NEWTON'S METHOD
  F=TIN-TP+Z*RTP
20 I=I+1
30 FP=-1.+Z*4.*EPS*AREA*.1713E-8*TPR**3
  TPN=TP-F/FP
  TPNR=TPN+460.
  RTPN=EPS*AREA*.1713E-8*TPNR**4-QABS
  F=TIN-TPN+Z*RTPN
C CHECK SOLUTION FOR CONVERGENCE
  IF (ABS(F)-.001) 50,50,40
40 TP=TPN
  TPR=TP+460.
  GO TO 30
C SOLUTION FOUND--FIND RELATED VARIABLES
50 QREJ=RTPN*ETA
  TOUT=TIN-(ETA/CPM)*RTPN
  WRITE (3,102) I, TIN, TOUT, TPN, QREJ
  TIN=TOUT
  IF (I-LM) 20, 60, 60
60 CONTINUE
100 FORMAT (F10.5, I5)
101 FORMAT (R10.5)
102 FORMAT (I10, 5X, I3, 5X, F6.2, 5X, F6.2, 5X, F6.2, 5X, E19.9)
  CALL EXIT
  END
// XEQ

```

where

AREA = area of one radiator panel or section, in ft^2
COND 1 = conductance 1, $\text{Btu/hr ft}^2 \text{ } ^\circ\text{F}$
COND 2 = conductance 2, $\text{Btu/hr ft}^2 \text{ } ^\circ\text{F}$
CPM = $\dot{m} C_p$, $\text{Btu/hr } ^\circ\text{F}$
EPS = ϵ , emissivity of radiator panel
ETA = η , radiator fin effectiveness
EX = fraction involving exponential terms in equation (6)
F = function on left side of equal sign in equation (6)
FP = derivative of F w.r.t. T_{PANEL}
LM = number of panels or sections in series
QABS = heat absorbed by AREA, Btu/hr
QREJ = heat rejected by AREA, Btu/hr
RTP = $R'(TP)$
RTPN = $R'(TFN)$
TIN = fluid inlet temp., $^\circ\text{F}$
TOUT = fluid outlet temp., $^\circ\text{F}$
TP = T_{PANEL} , $^\circ\text{F}$, "root of fin" radiator temperature
TPN = T_{PANEL} , $^\circ\text{F}$, after iteration
TPNR = T_{PANEL} , $^\circ\text{R}$, after iteration
TPR = T_{PANEL} , $^\circ\text{R}$
Z = part of constant not including EX on left side of equal sign in equation (6)

TABLE C-2.1

Panel No.	T _{IN} (°F)	T _{OUT} (°F)	T _{HEADER} HP(°F)	T _{FEEDER} HP's(°F)	T _{PANEL(Fin-root)} (°F)	Q _{REJECTED} ($\frac{\text{Btu}}{\text{hr}}$)
1	117.2	94.2	89.3	77.5	77.2	5,760
2	94.2	75.2	71.2	61.4	61.2	4,750
3	75.2	59.3	56.0	47.8	47.7	3,960
4	59.3	45.9	43.7	36.3	36.1	3,340
5	45.9	34.6	32.2	26.3	26.2	2,840
TOTAL						= 20,650
TOTAL FOR 10 PANELS						= 41,300 $\frac{\text{Btu}}{\text{hr}}$

APPENDIX D-1: FLANGE CONDUCTANCE STUDY FOR HP AUGMENTED COLD RAIL

The initial calculations of the temperature drop from the module box flange root to the cold rail box flange root were performed using the simple model shown in Figure D-1.1. With dimensions and materials as specified in reference 6 ($d_1 = .25"$, $d_2 = .15"$, module box of Magnesium with $K = 70 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}$), and the 1.8 inch wide flange specified for the thermal

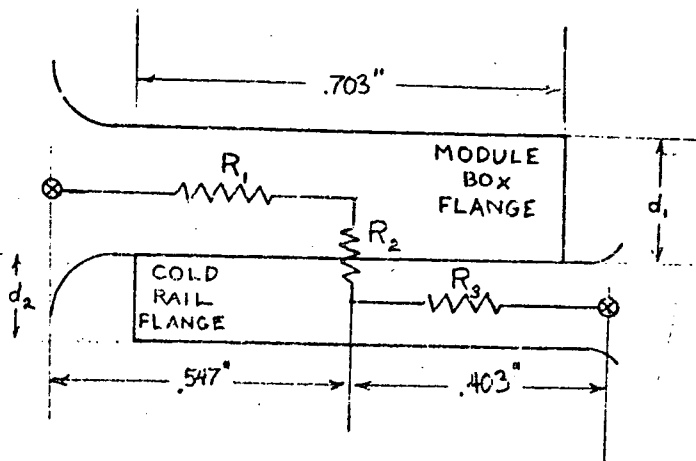


FIGURE D-1.1

model, the total $R = .547 \frac{^\circ\text{F}}{\text{Btu/hr}}$. At the mounting site for the high power density equipment module, with a 70 watt (239 Btu/hr) thermal input, this implies a temperature drop of 131°F . Since the maximum allowable box flange root temperature is 140°F , the maximum permissible cold rail flange root temperature in this configuration would be 9°F .

With a 70°F fluid inlet temperature, holding the box temperature to 140°F cannot be done.

A higher conductance was needed. A step at a time, the material of both the box flange and the cold rail flange was changed to 6101 Aluminum alloy ($k = 125 \frac{\text{Btu}}{\text{hr ft } ^\circ\text{F}}$), the thickness of both flanges increased ($d_1 = 0.5$ inches, $d_2 = 0.25$ inches), and, based on reference 10, the interface conductance raised to $1,500 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}$. Again using the simple model, the total $R = .221 \frac{^\circ\text{F}}{\frac{\text{Btu}}{\text{hr}}}$, implying a temperature drop at the high watt density

mounting site referred to above of 53°F . While this requires maintaining the rail center at 87°F , this seems feasible with a 70°F fluid inlet temperature.

With flanges as thick as these, the simple model originally used is

not adequate. A more detailed nodal network representing the two flanges and the interface was constructed and run using a simple steady state thermal analysis program; the resulting temperature distribution is shown in Figure D-1.2. Using this model, and taking a mean box flange root temperature of 140°F, a more accurate thermal resistance can be calculated:

$$R = \frac{\Delta T}{Q} = \frac{(140.0 - 92.3)}{240} = \frac{47.7}{240} = .199 \frac{^{\circ}\text{F}}{\text{Btu/hr}}$$

With a load of 239 Btu/hr, a cold rail flange root temperature of 92.5°F results. This thermal resistance is the one finally used in the detailed thermal model.

$60 \frac{\text{Btu}}{\text{hr}}$	140.3	137.4	134.8	132.4	130.4	127.9	127.8	127.2	
$60 \frac{\text{Btu}}{\text{hr}}$	140.1	137.2	134.4	132.0	130.0	127.4	127.3	126.7	
$60 \frac{\text{Btu}}{\text{hr}}$	139.9	136.8	133.7	131.1	129.0	127.3	126.1	125.5	
$60 \frac{\text{Btu}}{\text{hr}}$	139.6	136.3	132.5	129.7	127.5	125.7	124.4	123.5	
			112.9	111.8	109.9	107.1	103.3	98.4	92.3
			111.8	110.8	108.9	106.1	102.3	97.5	92.3

FIGURE D-1.2 DETAILED NODAL MODEL RESULTS
(All Temperatures in °F)



APPENDIX D-2: Computer Subroutine, Heat Pipe Augmented Cold Rail

FILE: CLC1

FORTRAN P1

CALLDATA PIPE - SHARING

```

SUBROUTINE CLC1
C THIS SUBROUTINE IS CALLED AT THE START OF EVERY ITERATION
COMMON STG(5000),NPOS(50),L1(50),L2(50),HLAST(50),PCT(50)
COMMON KILI,NTABS
COMMON RUFF(20),T(1000),CAP(1000),CORD(2000),RAD(2000),D(100)
1  NCOND(2,2000),NRAD(2,2000),NTN(1000),NBN(250),KEY(5),NQ(100)
2  KAY(4),FLUX(1000)
COMMON TIME,DTIME,PTIME,DUM,KNIN,KNDN,N4,N5,KODE,KNO,
1  KOUT,KOUNT
C THIS SECTION OF THE SUBROUTINE CALCULATES THE HEAT PIPE
C VAPOR TEMPERATURE
DIMENSION AREA(48)
DO 5 I=1,48
5 AREA(I)=1.0
J1=24
HCNDR=16.0
HCNDF=11.72
HEVPR=7.92
HEVPF=9.77
J2=J1+1
JN=2*J1
DO 20 I=1,J1
IF (T(I)-T(JN+1)) 10,10,15
C NODE I IS ACTING AS A CONDENSER
10 COND(I)=AREA(I)*HCNDF
GO TO 20
C NODE I IS ACTING AS AN EVAPORATOR
15 COND(I)=AREA(I)*HEVPF
20 CONTINUE
DO 24 I=J2,JN
IF (T(I)-T(JN+1)) 21,21,22
C NODE I IS ACTING AS A CONDENSER
21 COND(I)=AREA(I)*HCNDF
GO TO 24
C NODE I IS ACTING AS AN EVAPORATOR
22 COND(I)=AREA(I)*HEVPR
24 CONTINUE
C SOLVE FOR STEADY-STATE VAPOR TEMPERATURE
25 SUMC=0.0
SUMTC=0.0
DO 30 I=1,JN
SUMC=COND(I)+SUMC
SUMTC=COND(I)*T(I)+SUMTC
30 CONTINUE
TVAP=SUMTC/SUMC
C DETERMINE WHETHER CONDUCTANCES MUST BE CHANGED
KLANG=0
DO 50 I=1,JN
IF ((TVAP-T(I))/(T(JN+1)-T(I))) 35,50,50
35 KLANG=1
IF (T(I)-TVAP) 40,40,45
C T(I) WILL NOW BE CONSIDERED A CONDENSER
40 IF (I-J1) 36,36,37
36 COND(I)=HCNDF*AREA(I)
GO TO 50

```

COL00010
COL00020
COL00030
COL00040
COL00050
COL00060
COL00070
COL00080
COL00090
COL00100
COL00110
COL00120
COL00130
COL00140
COL00150
COL00160
COL00170
COL00180
COL00190
COL00200
COL00210
COL00220
COL00230
COL00240
COL00250
COL00260
COL00270
COL00280
COL00290
COL00300
COL00310
COL00320
COL00330
COL00340
COL00350
COL00360
COL00370
COL00380
COL00390
COL00400
COL00410
COL00420
COL00430
COL00440
COL00450
COL00460
COL00470
COL00480
COL00490
COL00500
COL00510
COL00520
COL00530
COL00540
COL00550

FILE: CICI FORTRAN P1

CALLDATA TIME - SHARING

37 COND(I)=HCMDR*AREA(I)	COL0056
GO TO 50	COL0057
T(I) WILL NOW BE CONSIDERED AN EVAPORATOR	COL0058
45 IF (I-J1) 46,46,47	COL0059
46 COND(I)=HEVPP*AREA(I)	COL0060
GO TO 50	COL0061
47 COND(I)=HEVPR*AREA(I)	COL0062
50 CONTINUE	COL0063
T(JN+1)=TVAP	COL0064
IF CONDUCTANCES HAVE CHANGED, RECALCULATE VAPOR TEMPERATURE	COL0065
IF (KLANG) 55,55,25	COL0066
55 CONTINUE	COL0067
THIS SECTION OF THE SUBROUTINE CALCULATES THE FLUID	COL0068
TEMPERATURES, USING STEADY-STATE METHODS FROM GRAFTON	COL0069
DO 140 I=100,111	COL0070
FIRST CALCULATE FLUID 'AVERAGE' TEMPERATURES FOR BOTH FLUID	COL0071
PASSAGES.	COL0072
$T(I) = (T(I-50) * COND(I-50) + T(I-99) * COND(I+68)) / (COND(I-50) + COND(I+68))$	COL0073
$T(I+12) = (T(I-38) * COND(I-38) + T(I-87) * COND(I+80)) / (COND(I-38) + COND(I+80))$	COL0074
IF (I-111) 120,140,140	COL0075
CALCULATE FLUID INLET TEMPERATURES FOR EACH SECTION.	COL0076
120 T(I-49)=2.*T(I)-T(I-50)	COL0077
T(I-37)=2.*T(I+12)-T(I-38)	COL0078
140 CONTINUE	COL0079
CALCULATE FLUID OUTLET TEMPERATURES	COL0080
T(98)=2.*T(111)-T(61)	COL0081
T(99)=2.*T(123)-T(73)	COL0082
RETURN	COL0083
END	COL0084
	COL0085
	COL0086

PRECEDING PAGE BLANK NOT FILMED

APPENDIX D-3

HEAT TRANSFER FILM COEFFICIENTS FOR 3/8" ID INTERNALLY-FINISHED TUBE

In order to formulate a thermal model for the heat pipe augmented cold rail, an accurate estimate of the heat transfer coefficient from the coolant fluid to the inside wall of an internally-finned tube must be available. A curve (Figure D-3) giving the experimentally-determined heat transfer coefficient for such a tube in terms of flow rate in gallons per minute was available, but for the Apollo Project ECS Fluid, which was a mixture of ethylene glycol (60%) and water (40%). Some work had to be found to account for differences in the fluid properties.

The Sieder-Tate equation (reference 19, page 392) provides a means of relating the Nusselt number with the Graetz number:

$$\overline{Nu} = 1.86 \left(\frac{4}{G_z} \right)^{.33} \left(\frac{\mu_w}{\mu} \right)^{.14}$$

Since $G_z = \frac{\dot{m} c_p}{\pi k L}$, and neglecting the viscosity correction term, this reduces to

$$\frac{hD}{k} = 1.86 \left(\frac{4 \dot{m} c_p}{\pi k L} \right)^{1/3}$$

or

$h = (\text{Const.}) k^{2/3} (\dot{m} c_p)^{1/3}$ where the constant is solely dependent on geometry.

The film coefficient for distilled water at the same flow rate and in the same tube is obtained from that for 60-40 glycol by using a ratio based on the above equation, dropping geometry and flow rate terms:

$$\frac{h_{\text{water}}}{h_{\text{glycol}}} = \frac{(k^{2/3} c_p^{1/3})_{\text{water}}}{(k^{2/3} c_p^{1/3})_{\text{glycol}}}$$

As fluid property data for both water and a 60-40 glycol/water mixture are readily available, this correction is easy to apply.

LYTRON TUBE (3/8 ID)

318 x 3116 Inner Fin

14 Point

$A_{3/8 \text{ ID}} = .098 \text{ Ft}^2/\text{Ft of Length (Full Circumference Area)}$

REF. (Lytron Test Data)

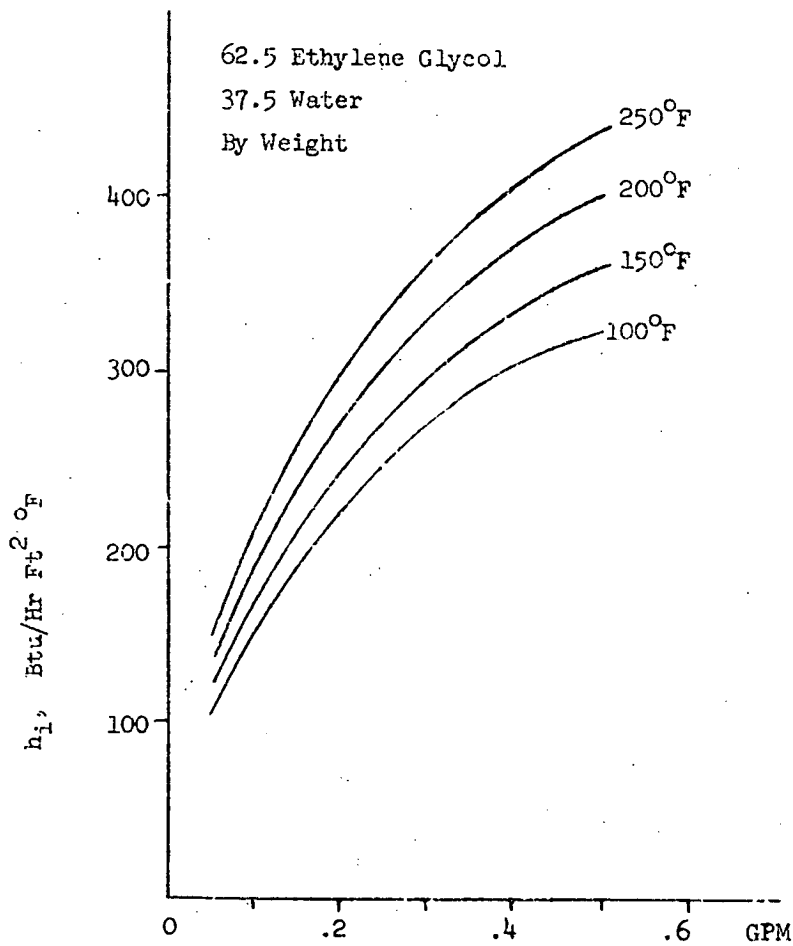


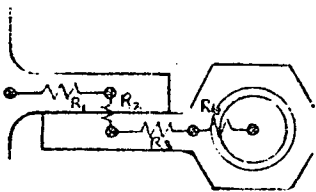
FIGURE D-3



APPENDIX E - AVIONICS HEAT PIPE CIRCUIT ANALYSIS DETAIL

(1) Conductances all conductances are of the $\frac{L}{kA}$ or $\frac{1}{hA}$ type.

- box flange-to-feeder heat pipe: (same module box and flange as in Appendix D-1)



$$R_1: \frac{.547 (x12)}{125 \times .2 \times 44} = .00596^\circ\text{F/Btu/hr}$$

$$R_2: \frac{(144)}{.703 \times 44 \times 1000} = .00466$$

$$R_3: \frac{.403 (x12)}{125 \times .125 \times 44} = .00704$$

$$R_4: \frac{.25 (x12)}{125 \times .24 \times 44} + \frac{(144)}{500 \times 44 \times .4 \times .5} + \frac{(144)}{2000 \times 44 \times .4 \times .5} =$$

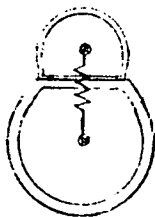
$$.00227 + .01040 + .00260$$

Total resistance on 44-inch flange = $.112^\circ\text{F/watt}$

Resistance on rail (two flanges) = $.056^\circ\text{F/watt}$

As indicated, the flange contact conductance is $1000 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}$, and a conductance of $500 \frac{\text{Btu}}{\text{hr ft}^2 \text{ } ^\circ\text{F}}$ is used between the aluminum rail extrusion and the copper sleeves insert to simulate poor thermal contact.

- Feeder heat pipe-to-heat header:



$$R_{\text{feeder condenser}} = \frac{(144)}{2500 \times .7 \times 7.0} = .0117^\circ\text{F/Btu/hr}$$

$$R_{\text{interface}} = 0 \text{ (soldered or brazed joint)}$$

$$R_{\text{header evaporator}} = \frac{(144)}{2000 \times 7. \times 1.5} = .0069$$

$$R_{\text{TOTAL}} = .0634^\circ\text{F/watt}$$

(2) Feeder heat pipe vapor temperature

With a maximum load of 200 watts evenly distributed over the cold rail, the flange root temperature is to be maintained at or below 140°F. The temperature drop from the 140°F flange to the heat pipe vapor, calculated using the conductance just obtained, is

$$.056 \text{ F/watt} \times 200 \text{ watts} = 11.2^\circ\text{F},$$

and the feeder heat pipe vapor temperature is 128.8°F.

(3) Header heat pipe vapor temperature

The temperature drop between the 128.8°F feeder heat pipe and the header heat pipe vapor is $200 \times .0634 = 12.7^\circ\text{F}$, so that header heat pipe vapor temperature is 116.1°F.

(4) Heat exchanger over-all conductance (Uo Ao)

Heat flux between the heat pipe header and the coolant equals the overall heat transfer coefficient of the exchanger times the logarithmic mean temperature difference between the heat pipe and the fluid. Hence

$$U_o A_o = \frac{Q_{\text{total}}}{T_{\text{out}} - T_{\text{in}}} = \frac{700}{96.3 - 68.1} = \frac{700}{31.7} = 22.1 \frac{\text{watts}}{^\circ\text{F}}$$
$$\ln\left(\frac{T_{\text{HP}} - T_{\text{IN}}}{T_{\text{HP}} - T_{\text{out}}}\right) = \ln\left(\frac{116.1 - 68.1}{116.1 - 96.3}\right)$$

(5) Heat exchanger design

A heat exchanger design with a calculated overall heat transfer coefficient of $22.1 \frac{\text{watts}}{^\circ\text{F}}$ is required, with as small a pressure drop as possible. In order to use the area of the heat pipe most efficiently, the pipe should be immersed in the fluid coolant. This suggests using bare concentric cylinders, but when calculations were performed, the required

length of such an exchanger is excessive

A conclusion of reference 20 is that the curves of Colburn j-factor in reference 15 for flat plate fin heat exchanger elements can be used for equivalent radial fin surfaces without modification. This allows the calculation of overall heat transfer coefficients from flow rate and exchanger geometry.

The standard formula for heat exchangers fitting the suggested concentric geometry is

$$\frac{1}{U_o A_o} = \frac{1}{h A_{\text{heat pipe side}}} + \frac{1}{k A_{\text{wall}}} + \frac{1}{h A_{\text{fluid side}}}$$

The term on the left hand side has been previously calculated, and the heat pipe and the wall terms are both included in the measured condenser film coefficient, leaving only the fluid side term to be determined on the basis of exchanger configuration.

To determine the fin geometry, reference 15 was used as a catalog, with the most desirable fin being a plate fin element with few fins per inch (to minimize pressure drop) and with short fins (to avoid large distortions of fin geometry and increase fin efficiency). The fin design chosen is that of reference 15, figure 10-26, page 195, having 11.1 fins per inch 0.25 inches high.

With water flowing through this fin geometry wrapped around the 0.875 inch O D. heat pipe at 85 lb/hr, the Reynolds Number is 75.2, and the Colburn j-factor is .0349. The exchanger film coefficient and efficiency are $38.3 \frac{\text{Btu}}{\text{hr ft}^2 \text{ }^\circ\text{F}}$ and .901, respectively. Total heat transfer area is 1.812 times L, where L is the exchanger length in feet.

The required length of the exchanger can now be calculated by using the above equation:

$$\frac{1}{22.1 \times 3.41} = \frac{12}{2500 \times .875 \times L} + \frac{1}{.901 \times 38.3 \times 1.812 \times L}$$

$$.01329 = \frac{.00175}{L} + \frac{.01600}{L}$$

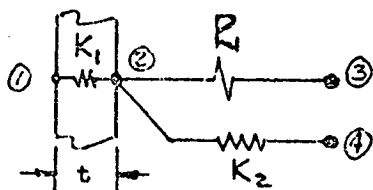
so $L_{\text{required}} = 1.335 \text{ feet} = 16 \text{ inches}$

APPENDIX F-1

System Heat Leaks

To minimize the heat transfer from the environment to the modular heat sink system all components will be covered with a $\frac{1}{4}$ inch thickness of TG-15000 insulation. The outside surface of the insulation will also be wrapped with a single layer of oxidized nickel foil to provide a radiation barrier.

1. Flat Plate Configuration (Cold Plate and box surfaces)



- (1) = inside insulation (107°F)
- (2) = outside insulation
- (3) = structure (207°F)
- (4) = atmosphere (207°F)

$$\frac{R_1}{A} = \sigma F_{23} \epsilon_2 \epsilon_3 ; \quad \begin{aligned} \epsilon_2 &= .05 \\ \epsilon_3 &= .20 \\ F_{23} &= 1 \end{aligned}$$

$$\frac{R_1}{A} = .12 (10^{-12}) \text{ BTU/Hr in}^2 \text{ R}^4$$

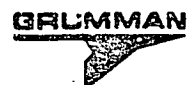
$$\frac{K_1}{A} = \frac{k}{t} = \frac{.00267}{t} \quad \frac{\text{BTU}}{\text{Hr in}^2 \text{ } ^\circ\text{F}}$$

$$\frac{K_2}{A} = h = .695 (10^{-3}) \quad \frac{\text{BTU}}{\text{Hr in}^2 \text{ } ^\circ\text{F}}$$

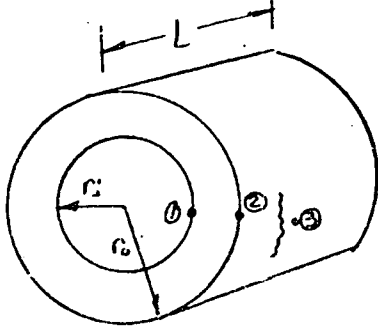
$$q_{2-1} < 3.2 \frac{\text{BTU}}{\text{Hr}} \quad \text{for } A = 28 \text{ in}^2$$

Neglecting radiation which is less than 10% of the heat gain anyway, the following general expression is obtained.

$$\frac{Q_{2-1}}{A(T_4 - T_1)} = \frac{.00267/t}{1 + \frac{.00267}{ht}} \quad (\text{Flat Plate})$$



2. Cylinder (PCM Container, Heat Pipes)



- (1) = inside insulation
- (2) = outside insulation
- (3) = atmosphere

$$\frac{C_{2-1}}{L(T_3 - T_1)} \approx \frac{2\pi r_o h k}{k + r_o h \ln\left(\frac{r_o}{r_i}\right)} \quad (\text{Cylinder})$$

For $h = .693 (10^{-3}) \frac{\text{BTU}}{\text{Hr in}^2 \text{ } ^\circ\text{F}}$

$k = .00267 \frac{\text{BTU}}{\text{Hr in } ^\circ\text{F}}$

$$\frac{Q}{L\Delta T} = \frac{.00165 r_o}{2.67 + .695 r_o \ln\left(\frac{r_o}{r_i}\right)}$$

PCM Container: $L = 10 \text{ in}$ $r_i = 1.56$ $r_o = r_i + .25$

$\Delta T \approx 100^\circ\text{F}$

$Q = 7.1 \text{ BTU/Hr}$

HP's: $L = 5 + 14 = 20 \text{ in}$ of exposed length

$r_i = .625$ $r_o = r_i + .25$
 $\Delta T \approx 100^\circ\text{F}$

$Q = 7.0 \text{ BTU/Hr}$

3. Diode conduction losses = 1 watt for a .028 wall

4. Bolt/washer conduction losses for a No. 10 steel bolt and .3 inch thick fiberglass washer = .048 BTU/Hr $^\circ\text{F}$ per connection.

Total ambient heat gain \approx 22 BTU/Hr

$$\% = \frac{22}{(35)(3.412)} = 18.5\% \text{ of dissipation}$$

use 20% for design

PRECEDING PAGE BLANK NOT FILMED

APPENDIX F-2

PCM Container Analysis

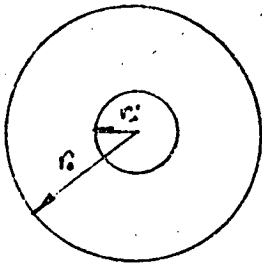
Assumptions:

- (1) Contact resistance between the filler material and cold plate or heat pipe is negligible.
- (2) Three dimensional heat transfer effects are neglected due to uniform application of heat flux and close proximity of filler material.

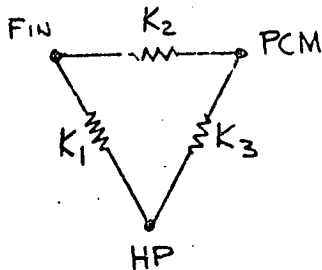
The required effective conductance of a PCM is defined as the ratio of heat flow into the PCM to the temperature difference between the PCM melting point and the maximum heat pipe wall or cold plate temperature.

$$K_{EFF} = \frac{Q}{T_{W_{Max}} - T_{MELT}}$$

1. Cylinder Container - Circular Fins



$$f = \frac{1}{t+s} = \text{FINS PER INCH}$$



$$K_1 = \frac{4\pi k_F t}{\ln(r_o/r_i)}$$

$$K_2 = k_{PCM} \frac{4\pi(r_o^2 - r_i^2)}{s}$$

$$K_3 = \frac{4\pi k_{PCM} s}{\ln(r_o/r_i)}$$

In a length, L, there are fL fins.

Effective Conductance

$$\frac{K_{HP-PCM}}{L} = f \left\{ \frac{\frac{4\pi k_F t}{\ln(r_o/r_i)} \frac{k_{PCM} (r_o^2 - r_i^2)}{s}}{\frac{k_F t}{\ln(r_o/r_i)} + \frac{k_{PCM} (r_o^2 - r_i^2)}{s}} + \frac{4\pi k_{PCM} s}{\ln(r_o/r_i)} \right\}$$

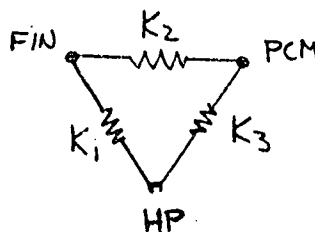
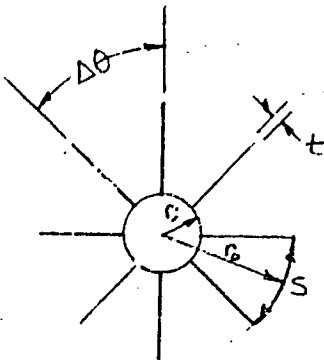
Fin Weight

$$\frac{W_F}{L} = f \rho_F t \pi (r_o^2 - r_i^2)$$

PCM Weight

$$\frac{W_{PCM}}{L} = \rho_{PCM} \pi (r_o^2 - r_i^2) (1 - f t)$$

2. Cylindrical Container - Longitudinal Fins



$$K_1 = \frac{2 t L k_F}{(r_o - r_i)} \quad K_2 = \frac{4 L r_o}{S} \ln \left(\frac{r_o}{r_i} \right) k_{PCM}$$

$$K_3 = \frac{\frac{S}{r_o} (r_o - r_i) k_{PCM}}{r_o - r_i \left(1 + \ln \left(\frac{r_o}{r_i} \right) \right)} \quad N = \frac{2 \pi r_i}{t + S} = \text{No. of fins}$$

For N fins in a length L:

Effective Conductance

$$\frac{K_{HP-PCM}}{L} = N \left[\frac{\frac{4 t k_F}{(r_o - r_i)} \frac{r_o}{S} \ln \left(\frac{r_o}{r_i} \right) k_{PCM}}{\frac{t k_F}{(r_o - r_i)} + \frac{2 r_o}{S} \ln \left(\frac{r_o}{r_i} \right) k_{PCM}} + \frac{\frac{S}{r_o} (r_o - r_i) k_{PCM}}{r_o - r_i \left(1 + \ln \left(\frac{r_o}{r_i} \right) \right)} \right]$$

Fin Weight

$$\frac{W_F}{L} = N \rho_F t (r_o - r_i)$$

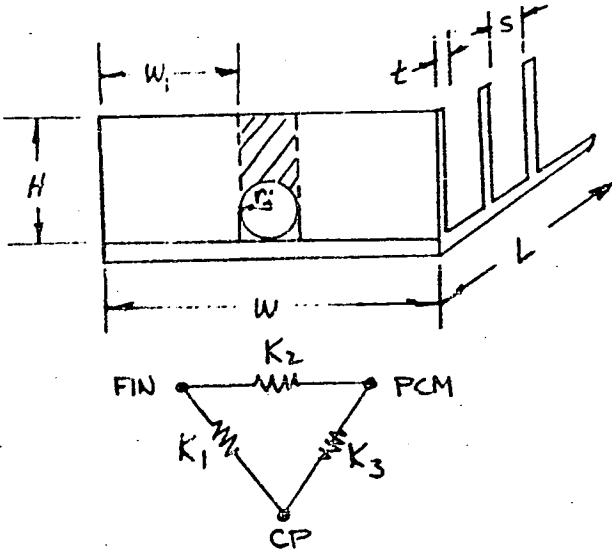
PCM Weight

$$\frac{W_{PCM}}{L} = \rho_{PCM} \pi (r_o^2 - r_i^2) \left(1 - \frac{2 r_i}{r_o + r_i} t \right)$$

Longitudinal Fin Results (t = .016, r_i = .31, r_o = 1.56)

N	$\frac{K}{L}$	$\frac{\text{BTU}}{\text{Hr } ^\circ\text{F-in}}$	$\frac{W_{PCM}}{L}$	$\frac{\text{lb}}{\text{in}}$	$\frac{W_{TOTAL}}{L}$
10		.302	.198		.218
20		.546	.194		.234
30		.345	.187		.247
40		.325	.182		.262

3. Rectangular Container - Straight Fins



$$f = \frac{1}{t + S} \quad \text{fins per inch}$$

$$K_1 = \frac{k_F W t}{\frac{H}{2}} \quad K_2 = \frac{k_{PCM} 2 HW}{S}$$

$$K_3 = k_{PCM} \frac{SW}{\frac{H}{2}}$$

Neglecting the contribution of the shaded area for conductance only:

Effective Conductance

$$\frac{K_{CP-PCM}}{H} = N \left\{ \frac{4 k_F k_{PCM} t w_1}{k_F t S + 2 k_{PCM} H^2} + \frac{2 k_{PCM} S w_1}{H^2} \right\}$$

Fin Weight

$$W_F = \rho_F t W H f L - \rho_F t f L \left(2 r_1^2 + \frac{\pi}{2} r_1^2 \right)$$

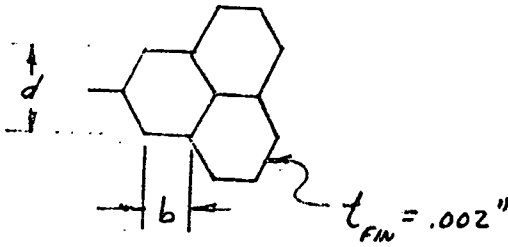
PCM Weight

$$W_{PCM} = \rho_{PCM} L H W (1 - ft) - \rho_{PCM} L (1 - ft) \left(2 r_1^2 + \frac{\pi}{2} r_1^2 \right)$$

2.7

Rectangular Container - Honeycomb

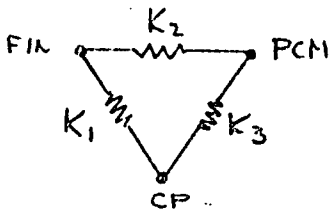
$$b = d \tan 30^\circ, \quad d = \text{core size}$$



$$K_1 = \frac{k_F A_F}{\frac{H}{2}}, \quad A_F = 6 t b$$

$$K_2 = 6 k_{PCM} H$$

$$K_3 = k_{PCM} \frac{3 (bd)}{H}$$



Effective Conductance

$$\frac{K_{CP-PCM}}{H} = \frac{12 k_F t b k_{PCM}}{2 k_F t b + k_{PCM} H^2} + \frac{3 k_{PCM} b d}{H}$$

Fin Weight

$$W_F = 6 t b H \rho_F$$

PCM Weight

$$W_{PCM} = 1.5 b d H \rho_{PCM}$$

APPENDIX G

DETAILED DESIGN CALCULATIONS FOR THE AVIONICS RACK HEAT EXCHANGER

A. Heat Pipe Optimization Calculations

The method used to determine the optimum number of one half inch square, 15 inch long pipes and the relative evaporator and condenser section lengths is discussed below.

The total heat pipe temperature gradient, ΔT_{HP} (evaporator section plus condenser section film losses... transport losses being negligible) may be expressed as a function of the number of pipes, N , and the evaporator length, L (the condenser length is equal to $14.75-L$ inches allowing 0.25 inches for a transport section). Additionally, the evaporator section unit heat flux, q_{evap} , must be determined as a function of N and L to verify that fluxes of less than 25 watts/in² are achieved.

$$\Delta T_{HP} = \frac{Q}{(hA)_{evap}} + \frac{Q}{(hA)_{cond}}$$

where

- Q = 1980 watts, design load
- h_{evap} = 2000 Btu/hr-ft²-°F, evaporator film coefficient
- h_{cond} = 2500 Btu/hr-ft²-°F, condenser film coefficient

An expression for the evaporating surface area, A_{evap} , and condensing surface area, A_{cond} , may be derived in terms of N and L .

$$A_{evap} = 2 \times [0.496 + 2 (0.248 \eta_1)] \times N \times L, \text{ and}$$

$$A_{cond} = 2 \times [0.496 + 2 (0.248 \eta_2)] \times N \times (14.75-L)$$

In the above expressions, η_1 and η_2 are the evaporator and condenser section conduction fin efficiencies of the two heat pipe sides not in direct contact with the fluid stream cores.

Using standard techniques, these efficiencies were calculated (based on applicable wall thickness, thermal conductivity, film coefficient and fin height) to be 0.66 for η_1 and 0.62 for η_2 .

Using these values, the area terms become

$$A_{\text{evap}} = 1.646 \times N \times L, \text{ in}^2$$

$$A_{\text{cond}} = 1.602 \times N \times (14.75 - L), \text{ in}^2$$

The expression for the total heat pipe temperature gradient therefore becomes:

$$\Delta T_{\text{HP}} = \frac{295.5}{NL} + \frac{242.5}{N(14.75-L)}, \text{ } ^\circ\text{F}$$

Similarly, the expression for the evaporator section heat flux may be derived as a function of N and L.

$$q_{\text{evap}} = \frac{Q}{A_{\text{evap}}} = \frac{1203}{NL}, \text{ watts/in}^2$$

B. Air Side Design Conditions

As calculated in section 9-4 of this report, the total available sum of thermal resistances on the air and water sides of the heat exchanger is:

$$\frac{1}{UA_{\text{Air}}} + \frac{1}{UA_{\text{Water}}} = 0.00362 \frac{^\circ\text{F-hr}}{\text{BTU}}$$

Since the air side film coefficients were expected to be quite lower than the water side, and the pressure drop requirements on the air side were quite stringent, it was decided to allocate the major portion of the available thermal resistances to the air side. The following split was made:

$$\begin{aligned} \frac{1}{UA_{\text{Air}}} &= 0.00278 \frac{^\circ\text{F-hr}}{\text{BTU}} ; & UA_{\text{Air}} &= 360 \frac{\text{BTU}}{\text{hr-}^\circ\text{F}} \\ \frac{1}{UA_{\text{Water}}} &= 0.00084 \frac{^\circ\text{F-hr}}{\text{BTU}} ; & UA_{\text{Water}} &= 1200 \frac{\text{BTU}}{\text{hr-}^\circ\text{F}} \end{aligned}$$

The allowable air system pressure drop corresponding to a fan power allowance of 180 watts can be calculated as follows:

$$Q_{\text{Fan}} = \frac{144 \times \Delta P \times \dot{w} \times \sigma}{27.7 (.0750) (778) (3.415) \times 10^2 \eta_{\text{Fan}}}$$

$$Q_{\text{Fan}} = \frac{.0261 \dot{w} \sigma \Delta P}{10^2 \eta_{\text{Fan}}}, \text{ watts}$$

where,

\dot{w} = air flow rate, lbs/hr

= 1000.

η_{Fan} = fan efficiency

= 0.35 (per vendor supplied information)

ΔP = pressure drop at operating temperature and pressure, in H₂O

σ = ratio of air density at operating conditions (10 psia, 125°F) to density at 14.7 psia, 70°F

= .0461/.075 = 0.614

Q_{Fan} = fan power allowance, watts

= 180.

Substituting values and solving for $\sigma \Delta P$, we obtain:

$$\sigma \Delta P = 0.914 \text{ inches of H}_2\text{O (total system loss)}$$

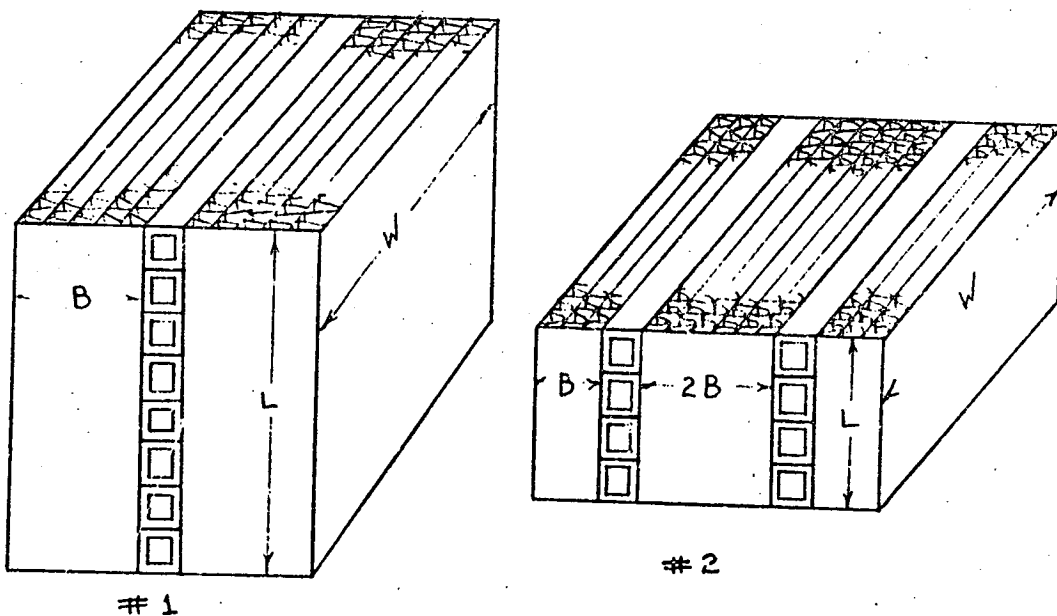
Allowing 10% of this figure for ducting losses, we obtain for our allowable core loss:

$$(\sigma \Delta P)_{\text{Core}} = 0.819 \text{ inches of H}_2\text{O}$$

The design of the air core was performed using a Grumman developed technique for sizing a laminar plate-fin heat exchanger (reference 17). The method uses a mathematical correlation of heat transfer data for the flow of air in rectangular ducts compiled in reference 15. In particular,

it is found that both the product of friction factor and Reynolds number, as well as Stanton number, Prandtl number (to the two thirds power) and Reynolds number, are constant in the laminar region. By specifying the allowable UA, pressure drop (ΔP), flow length, and one dimension of the frontal area, the method allows one to determine the other frontal dimension as well as the number of core layers and fins per inch required in the extended surface.

Two core/heat pipe configurations were considered in arriving at the final design. These two concepts are pictured below:



In concept #1, the air cores are positioned adjacent to a single row of 8 heat pipes. Although design calculations were performed, this concept was rejected. The B dimension required was excessively large (approximately 6 inches) consisting of approximately 40 layers of very small core material. In essence, therefore, the design requirements could not be met for this configuration.

Concept #2 positions 3 air cores about two rows of four pipes each. The center core is actually two of the outer cores back to back since we need an equal amount of core on either side of each row of heat pipes to achieve

uniform performance. This concept proved to be the one chosen for the final design.

Since $UA_{Air} = 360 \text{ Btu/hr } ^\circ\text{F}$, the required UA per core is $90 \text{ Btu/hr } ^\circ\text{F}$ (or $100 \text{ Btu/hr } ^\circ\text{F}$ including a 10% safety factor).

The design calculations using the method derived in reference 17, are shown below:

$$UA_{Air} = \sqrt{X} \left[\tanh \left[\frac{B}{WL} \left(\frac{D}{D-1} \right) \frac{1}{K} \sqrt{X} \right] \right]$$

where,

$$X = \frac{4 \epsilon_c}{.385} \left(\rho_{STD} \Delta P C_p \frac{N_{st}}{f} W^3 \text{ LB} \frac{K}{\dot{w}_{Air}} \frac{D-1}{D^3} \frac{1}{\lambda^2} \right)$$

where,

UA	=	requisite heat transfer coefficient, Btu/hr $^\circ\text{F}$
L	=	0.1835 feet, flow length
W	=	0.792 feet, heat pipe contact length
B	=	core depth
D	=	$\frac{\text{fin thickness} + \text{fin spacing}}{\text{fin spacing}} = 1.25$ for 20% solid core
K	=	125 Btu/hr-ft- $^\circ\text{F}$ (1100 aluminum)
ϵ_c	=	gravitational constant, ft/hr ²
ρ_{STD}	=	0.075 lb/ft ³
ΔP	=	0.819/12 inches of H ₂ O water
C_p	=	specific heat of air = 0.24 Btu/lb- $^\circ\text{F}$
N_{ST}	=	Stanton number
f	=	friction factor
\dot{w}_{Air}	=	250 lbs/hr per core
λ	=	$\frac{1.0 + \text{spacing plate thickness}}{1.0}$

From Ref. 15 for rectangular tubes having an aspect ratio, α , of 8,

$$N_{Re} N_{ST} N_{PR}^{2/3} = 6.7,$$

$$N_{Re} f \frac{T_W^{-1.35}}{T_A} = 21.0,$$

where,

T_W = wall temperature, and

T_A = air temperature, °F

$N_{PR} = 0.7$

or

$$\frac{N_{ST}}{f} = \frac{6.7 \left(\frac{T_W}{T_A} \right)^{-1.35}}{21.0 (0.7)^{2/3}} = 0.368$$

Substituting in our expression for X, we obtain,

$$X = 137,100 B$$

Assuming that the hyperbolic tangent term is equal to 1.0,

$$UA = \sqrt{X}, \quad \text{or}$$

$$B = \frac{UA^2}{952.5} = \frac{100^2}{137,100} = .0728 \text{ ft} = .874 \text{ inches}$$

To check, substituting these values into the complete expression for UA, we get,

$$UA = 100 \tanh(2.21) = 97.6 \text{ (good enough)}$$

The required hydraulic diameter, d_h is calculated from

$$\begin{aligned} d_h^2 &= \frac{.385}{\text{STD } g_c} \frac{\mu_{\text{Air}}}{\sigma \Delta P} f N_{Re} \frac{L \rho \lambda}{WB} \\ &= 15.5 \times 10^{-6} \text{ ft}^2 \\ d_h &= 3.935 \times 10^{-3} \text{ ft} = 47.2 \times 10^{-3} \text{ inches} \end{aligned}$$

the core opening, $b = d_h \frac{\alpha + 1}{2}$, where $\alpha = \text{aspect ratio} = b/a = 8$

Substituting, $b = 0.212$ inches.

The number of layers comprising the core, is

$$N = \frac{B}{b} = \frac{0.874}{0.212} = 4.12$$

Since we cannot fabricate a fractional layer, we will calculate the new geometry required for an integral number of layers.

From the two expressions above for d_n^2 and d_n , we can derive a relationship between B/λ and N . In particular,

$$(B/\lambda)^3 = 22.85 \times 10^{-6} N^2$$

For $N = 4$

$$\frac{B}{\lambda} = 7.16 \times 10^{-2} \text{ ft} = 0.860 \text{ inches}$$

$$d_h = \frac{2}{9} \frac{B'/A}{N} = 0.04775 \text{ inches}$$

$$b = B/\lambda N = 0.215 \text{ inches}$$

With a spacer plate thickness of 0.010 inches,

$$\lambda = 1.05$$

Therefore,

$$B_{\text{Total}} = .860 \times 1.05 = .903 \text{ inches}$$

$$A = \text{surface area} = \frac{4 WL}{d_h} \frac{B'}{\lambda} = 10.06 \text{ ft}^2$$

$$h = \text{film coefficient} = \frac{C_p M N_{ST} N_{Re}}{d_h} = 23.2 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\delta = \text{fin thickness} = \frac{b}{32} = 0.00672 \text{ inches}$$

$$\eta = \text{fin effectiveness} = \frac{\tanh(Z)}{Z}, \text{ where}$$

$$Z = \sqrt{\frac{2 h}{K \delta} \frac{\alpha + 1}{\lambda \alpha}} B$$

$$\gamma = \frac{\tanh(2.0)}{2.0} = 0.482$$

$$UA = h S \gamma = 112 \text{ BTU/hr-}^\circ\text{F}$$

Since this is somewhat higher than we require, we can recalculate using $N = 3$.

For $N = 3$

$$\frac{B}{\lambda} = 5.9 \times 10^{-2} \text{ feet} = 0.707 \text{ inches}$$

$$d_h = \frac{2}{9} \frac{B'}{N} = 0.0524 \text{ inches}$$

$$b = B/\lambda N = 0.236 \text{ inches}$$

$$\text{for } \lambda = 1.05 \text{ (0.010 inch spacer)}$$

$$B_{\text{Total}} = 0.707 (1.04) = 0.737 \text{ inches}$$

$$A = 7.54 \text{ ft}^2$$

$$h = 21.1 \text{ BTU/hr-ft}^2\text{-}^\circ\text{F}$$

$$\delta = 0.00737 \text{ inches}$$

$$\gamma = 0.603$$

$$UA = h S \gamma = 96.1 \text{ BTU/hr-}^\circ\text{F} \text{ (this is good enough)}$$

SUMMARY

The final air core design for the four cores required is as follows:

3 layers

$$b = 0.240 \text{ inches}$$

$$a = 0.030 \text{ inches}$$

$$\delta = 0.008 \text{ inches (32 gauge)}$$

26.3 fins/inch

0.010 inch plates between cores

C. Water Side Design Calculations

The water side plate-fin cores were designed using the same methods previously outlined for the air side, with the following restrictions:

$$UA_{\text{Allowable}} = 1200 \text{ Btu/hr-}^{\circ}\text{F} = 300 \text{ Btu/hr-}^{\circ}\text{F per core (4 cores in parallel)}$$

$$L = 0.1835 \text{ feet, flow length}$$

$$W = 0.4375 \text{ feet, heat pipe contact length}$$

$$B = \text{core depth}$$

$$\Delta P = \text{pressure drop, inches of water (4 psig allowable)}$$

$$\phi = 1.25 \text{ for 20\% solid core}$$

$$K = 125 \text{ Btu/hr-ft-}^{\circ}\text{F (1100 aluminum)}$$

$$\rho_{\text{STD}} = 62.1 \text{ lb/ft}^3$$

$$\sigma = 1.0$$

$$C_p = 1.0$$

$$\alpha = 8.0$$

$$N_{\text{PR}} = 5.49$$

$$N_{\text{Re}} \times N_{\text{ST}} \times N_{\text{PR}} = 6.47$$

$$fN_{\text{R}} = 20.5$$

Substituting these values into the expressions for X , UA , and d_h presented in section B, we may obtain,

$$\Delta P \times B = 1.412 \times 10^{-3}$$

$$d_h = 3.65 \times 10^{-3} \text{ feet}$$

$$= 43.7 \times 10^{-3} \text{ inches}$$

$$b = d_h \frac{\alpha + 1}{2} = 0.197 \text{ inch (.0164 (feet))}$$

By choosing $B = b$ (i.e., a single layer core),

$$\Delta P = \frac{1.412 \times 10^{-3}}{.0164} = 0.0862 \text{ inches of water (well below the allowable 4 psia)}$$

$$S = \frac{4 W L}{d_h \nu} \frac{B}{\lambda} = 1.154 \text{ ft}^2$$

$$h = \frac{C_p \mu^{N_{ST}} N_R}{d_h} = 636 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\delta = b/32 = 0.00615 \text{ inches}$$

$$Z = 2.45$$

$$\eta = \frac{\tanh(Z)}{Z} = 0.403$$

$$UA = h S \eta = 296 \text{ Btu/hr-}^\circ\text{F} \text{ (good enough)}$$