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ICE PACK HEAT SINK SUBSYSTEM - PHASE I

VOLUME I

BY

GEORGE J. ROEBELEN, JR.

JUNE 1973

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PREPARED UNDER CONTRACT NO. NAS 2-7011

BY

HAMILTON STANDARD

DIVISION OF UNITED AIRCRAFT CORPORATION

WINDSOR LOCKS, CONNECTICUT

FOR

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

AMES RESEARCH CENTER

MOFFET FIELD, CALIFORNIA 94035



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ABSTRACT

ICE PACK HEAT SINK SUBSYSTEM - PHASE I

by

George J. Roebelen, Jr.

Contract No. NAS 2-7011

This report describes the design, development, fabrication, and test at one-g of a functional laboratory model (non-flight) Ice Pack Heat Sink Subsystem to be used eventually for astronaut cooling during manned space missions. In normal use, excess heat in the liquid cooling garment (LCG) coolant is transferred to a reusable/regenerable ice pack heat sink. For emergency operation, or for extension of extravehicular activity mission time after all the ice has melted, water from the ice pack is boiled to vacuum, thereby continuing to remove heat from the LCG coolant. This subsystem incorporates a quick connect/disconnect thermal interface between the ice pack heat sink and the subsystem heat exchanger.

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FOREWORD

This report has been prepared by the Hamilton Standard Division of the United Aircraft Corporation for the National Aeronautics and Space Administration's Ames Research Center in accordance with the requirements of Contract NAS 2-7011, Ice Pack Heat Sink Subsystem - Phase I.

Volume I contains the entire Ice Pack Heat Sink Subsystem - Phase I final report except for Appendix J, Ice Pack Heat Sink Subsystem Operating Instructions and Component Specifications.

Volume II contains Appendix J, Ice Pack Heat Sink Subsystem Operating Instructions and Component Specifications.

Appreciation is expressed to the NASA Technical Monitor, Mr. James R. Blackaby of the Ames Research Center, for his guidance and advice.

Hamilton Standard personnel responsible for the conduct of this program were Mr. F. H. Greenwood, Program Manager, and Mr. G. J. Roebelen, Program Engineer. Appreciation is expressed to Mr. J. S. Lovell, Chief, Advanced Engineering, Mr. P. F. Heimlich, Design Engineer, and Mr. E. H. Tepper, Analytical Engineer, whose efforts made the successful completion of this program possible.

TABLE OF CONTENTS

	<u>Page No.</u>
<u>SUMMARY</u>	1
<u>INTRODUCTION</u>	3
<u>CONCLUSIONS</u>	5
<u>RECOMMENDATIONS</u>	7
<u>NOMENCLATURE</u>	9
<u>SUBSYSTEM DESIGN AND ANALYSIS</u>	11
SUBSYSTEM CONCEPT	11
SUBSYSTEM FUNCTIONAL DESCRIPTION	16
SUBSYSTEM PHYSICAL DESCRIPTION	20
SUBSYSTEM ANALYSIS	24
<u>Conductance from LCG Water</u>	25
<u>Interface Contact Conductance</u>	27
SUBSYSTEM PHYSICAL DESCRIPTION	30
<u>Fin-To-Ice Conductance</u>	34
<u>Wicking Device</u>	37
<u>Ice Expansion Compensation Device</u>	40
<u>Ice Chest Sizing</u>	44
<u>Overall Subsystem Performance for Normal Operation</u>	45
<u>Emergency Operation</u>	45
<u>Overall Subsystem Performance for Emergency Operation</u>	55
<u>DESIGN FEASIBILITY TESTING</u>	63
<u>DEVELOPMENT TESTING</u>	87

TABLE OF CONTENTS (Concluded)

	Page No.
PRELIMINARY DEVELOPMENT TESTING	87
<u>Description of Preliminary Development Test Runs</u>	91
LAB MODEL CONFIGURATION DEVELOPMENT TESTING	94
<u>Normal Mode Testing</u>	95
<u>Emergency Mode Testing</u>	96
<u>Conclusions</u>	96
<u>Acceptance Testing</u>	97
APPENDIX A <u>VAPOR PASSAGE PRESSURE DROP SAMPLE CALCULATIONS</u>	A-i
APPENDIX B <u>ICE CHEST EXHAUST LINE PRESSURE DROP AND ORIFICE SIZES - SAMPLE CALCULATIONS</u>	B-i
APPENDIX C <u>DESIGN FEASIBILITY TEST PLAN</u>	C-i
APPENDIX D <u>DEVELOPMENT TEST PLAN</u>	D-i
APPENDIX E <u>DEVELOPMENT TEST LOG SHEETS</u>	E-i
APPENDIX F <u>ACCEPTANCE TEST PLAN</u>	F-i
APPENDIX G <u>ACCEPTANCE TEST LOG SHEETS</u>	G-i
APPENDIX H <u>SPECIFICATION FOR ICE PACK HEAT SUBSYSTEM</u>	H-i
APPENDIX I <u>REFERENCES</u>	I-i

LIST OF FIGURES

<u>Figure No.</u>	<u>Title</u>	<u>Page No.</u>
1	Ice Chest Heat Sink Subsystem Console Front View	12
2	Ice Chest Heat Sink Subsystem Left Side View	13
3	Ice Chest Heat Sink Subsystem Rear View	14
4	Ice Chest	15
5	Ice Pack Heat Sink Subsystem Schematic	17
6	Ice Pack Heat Sink Subsystem	18
7	Ice Chest/Heat Exchanger - Normal Operating Mode	22
8	Ice Chest/Heat Exchanger - Emergency Operating Mode	23
9	Normal Operation Predicted LCG Heat Exchanger Effectiveness Versus LCG Water Flow	26
10	Typical Temperature Profile	28
11	Contact Conductance Data	30
12	Thermal Contact Conductance of Selected Interstitial Materials	32
13	Comparison of Dimensionless Conductance for Selected Metal Interstitial Materials	33
14	Analytical Model for Fin-To-Ice Heat Transfer	35
15	Test Demonstration of Wicking Mechanics	37
16	#S4 Dacron Wicking Tests	41
17	Ice Expansion Compensation Device (PSIA)	42

LIST OF FIGURES (Continued)

<u>Figure No.</u>	<u>Title</u>	<u>Page No.</u>
18	Single Cell Structure	43
19	Heat Sink Performance with no Ice Melted	46
20	Heat Sink Performance with Ice Half Melted	47
21	Heat Sink Performance with Ice All Melted	48
22	Estimated Pressure Drop LCG Heat Exchanger Ice Pack Heat Sink Subsystem	49
23	Ice Chest Vapor Passage Configuration	52
24	Pressure Drop Versus Weight Flow from Porous Plate to Header Outlet ~ Emergency Mode	53
25	Emergency Operation Predicted LCG Heat Exchanger Effectiveness Versus LCG Water Flow	56
26	Emergency Operation Ice Pack Heat Sink Subsystem	58
27	Emergency Operation Ice Pack Heat Sink Subsystem	59
28	Emergency Operation Ice Pack Heat Sink Subsystem	60
29	Emergency Operation Ice Pack Heat Sink Subsystem	61
30	Emergency Operation Ice Pack Heat Sink Subsystem	62
31	Ice Chest and Heat Exchanger Thermocouple Locations	64
32	Feasibility Test Set-Up in Ambient	65
33	Feasibility Test Set-Up in Vacuum	66
34	Portable Flow Console	67
35	Vacuum Chamber	68
36	Performance Time Dependency for $Q_{\text{Sink}} =$ 879 J/S (3000 BTU/hr)	71

LIST OF FIGURES (Concluded)

<u>Figure No.</u>	<u>Title</u>	<u>Page No.</u>
37	Performance Time Dependency for $Q_{\text{Sink}} = 440 \text{ J/S (1500 BTU/hr)}$	72
38	Performance Time Dependency for $Q_{\text{Sink}} = 146.5 \text{ J/S (500 BTU/hr)}$	73
39	Heat Sink Performance with No Ice Melted	74
40	Heat Sink Performance with Ice Half Melted	75
41	Heat Sink Performance with All Ice Just Melted	76
42	Effectiveness vs. Percent Ice Melted, $879 \text{ J/S (3000 BTU/hr)}$	77
43	Effectiveness vs. Percent Ice Melted, $146.5 \text{ J/S (500 BTU/hr)}$	78
44	Effectiveness vs. Percent Ice Melted, $440 \text{ J/S (1500 BTU/hr)}$	79
45	Emergency Performance	81
46	Effectiveness vs. Time ~ Evaporative Mode	82
47	Emergency Performance	83
48	Emergency Mode Heat Sink Performance	84
49	Heat Exchanger Pressure Drop vs Actual Flow	85
50	Development Test Setup	88
51	Vapor Passage Shutoff Valve & Controller Setup	89
52	"Improved" Wick/Expansion Compensation Configuration	90
53	Lab Model Wick/Expansion Compensation Configuration	93

LIST OF TABLES

<u>Table No.</u>	<u>Title</u>	<u>Page No.</u>
I	Candidate Wicking Materials	39
II	Comparison of Ice Expansion Compensation Device Materials	40
III	Heat Exchanger Temperature Drop Across Contact Surface for "Ice All Melted" and "Evaporative" Cases	70
IV	Ice Chest/LCG Heat Exchanger Interface Characteristics	92
V	Predicted Ice Melting Time	95
VI	Melting Ice Test Results	96
VII	Acceptance Test Results	98

SUMMARY

A functional laboratory model (non-flight) Ice Pack Heat Sink Subsystem was designed, developed, and tested at one-g during this 12 month program. Two complete functional laboratory model subsystems, each having two spare ice pack modules, were fabricated and delivered to the NASA ARC. Complete detail and assembly drawings, operating instructions, and specifications of all mechanical components also were submitted.

In normal use the subsystem utilizes a six pass heat exchanger in contact with a reusable/regenerable ice pack heat sink module to remove heat from, and control temperature to, a liquid cooling garment (LCG) loop. For emergency operation, or for extension of extravehicular activity (EVA) mission time, when all the ice has melted, water from the ice pack module is automatically vented to vacuum, causing boiling, and thereby continuing the removal of heat from the LCG loop.

The ice pack module holds a minimum of 7.0 kg (15.4 lbm) of water. Cooling loads of 139.5, 440, and 879 J/s (475, 1500, and 3000 Btu/hr) can be satisfied for both the normal melting ice and emergency water boiling modes of operation, resulting in normal operation unit outlet temperatures of 302.6°K (85°F), 294.2°K (70°F), and 282.7 - 284.8°K (49-53°F) and in emergency operation unit outlet temperatures of 284.3 - 284.8°K (52-53°F), 284.3 - 287.8°K (52-56.5°F) and 293.7 - 295.3°K (69-72°F), respectively.

Development of a suitable process for applying a 0.1 mm (0.004 inch) lead coating to the heat transfer surface required considerable effort. The application process developed is an electro-plated layer, which was mechanically finished to obtain a suitably flat surface.

An additional problem area encountered was in the development of the ice expansion compensation sandwiches. Exposure of the sandwiches to vacuum caused them to grow and thereby push water out of the wicking material. The successful design now consists of a teflon laced sandwich, which is preloaded approximately 25 percent prior to the teflon lacing process. Preloading insures that all the slack was removed from the lacing.

It is significant to note that the final version of the Ice Pack Heat Sink Subsystem is essentially identical to the configuration generated analytically, and that all requirements of Specification Number 2-17753 were satisfied.

Based on the results of this program, the Ice Pack Heat Sink Subsystem concept has shown itself to be a viable system for astronaut cooling during EVA.

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INTRODUCTION

Future manned space exploration missions are expected to include requirements for astronaut life support equipment capable of repeated use and regeneration for many extravehicular activity sorties. In anticipation of these requirements, NASA ARC funded two contracts (NAS 2-6021 and NAS 2-6022) for the study of Advanced Extravehicular Protective Systems. The purpose of these studies was to determine the most practical and promising concepts for manned space flight operations projected for the late 1970's and 1980's, and to identify areas where concentrated research would be most effective in the development of these concepts.

One regenerative concept for astronaut cooling utilizes an ice pack as the primary heat sink for a liquid cooling garment (LCG) cooling system. In an emergency, or for extended operations, water from the melted ice pack could be evaporated (boiled) directly to space vacuum.

This report describes the design, development, fabrication, and test at one-g of a functional laboratory model of such an Ice Pack Heat Sink Subsystem.

Calculations and data pertaining to the execution of this program were made in U. S. customary units and then converted to SI units.

CONCLUSIONS

The results of this program effort to design, develop, fabricate, and test at one-g a functional laboratory model Ice Pack Heat Sink Subsystem lead to the following conclusions.

- The unit hold 7.0 kg (15.4 lbm) of water.
- Cooling loads of 139.5, 440, and 879 J/s (475, 1500, 3000 Btu/hr) can be satisfied for both the normal melting ice and emergency water boiling modes of operation, resulting in normal operation unit outlet temperatures of 302.6°K (85°F), 294.2°K (70°F), and 282.7-284.9°K (49-53°F) and in emergency operation unit outlet temperatures of 284.3-284.8°K (52-53°F), 284.3-287.8°K (52-56.5°F) and 293.7-295.3°K (69-72°F), respectively.
- The interface between the ice chest module and the LCG heat exchanger is sufficiently durable to withstand repeated removal/installation cycles as long as care is taken to wipe the mating surfaces prior to installation and to protect the mating surfaces from scratches or grooves.
- Manufacturing tolerances associated with the mating surfaces present no excessive problem with the concept utilizing a semi-flexible heat exchanger and a fluid interface preloading bladder.
- All aspects of NASA Ames Research Center specification number 2-17753, as amended by contract amendment/modifications No. 1, 2, 3, and 4, were met.

RECOMMENDATIONS

The studies and test results of this program evolved the following recommendations.

- Continuation of the basic hardware configuration generated by this program phase is recommended.
- Optimum data utilization could be obtained during any future testing if the following steps were taken.

Normal mode operation should be run at 139.5-440 J/s (475-1500 Btu/hr) over a wide LCG heat exchanger water flow spectrum, ignoring the unit mixed outlet temperature.

Accurate contact conductance measuring points should be utilized to obtain precise temperature differentials for a 879 J/s (3000 Btu/hr) load application in both normal and emergency operation.

The emergency water boiling mode should be run over the whole spectrum of LCG heat exchanger flow rates.

This test sequence would provide a complete series of unit effectiveness maps for both normal and emergency modes of operation.

- Additional effort is recommended to study alternate ice chest/ LCG heat exchanger interface configurations and interface pre-loading configurations to obtain a self-cleaning interface and a higher unit contact pressure.

NOMENCLATURE

A	effective area, cross sectional area
A _f	heat exchanger core face area
A _p	primary area
A _s	secondary area
Btu	British thermal unit
Btu/hr	British thermal unit per hour
Btu/hr-°F	British thermal unit per hour-degree Fahrenheit
Btu/hr-ft ² -°F	British thermal unit per hour-square foot-degree Fahrenheit
c	water specific heat
C ₁	LCG cooling water to heat exchanger end plate thermal conductance
C ₂	heat exchanger end plate to ice chest end plate conductance
C ₃	ice chest end plate to ice conductance
cm	centimeter
E	heat exchanger effectiveness
EVA	extravehicular activity
°F	degree Fahrenheit
g	gram
g/s	gram per second
gpm	gallons per minute
h	convection coefficient
hA	thermal conductance
H/X	heat exchanger
hr	hour
in	inch
J	joule
J/s	joule per second
J/s-°K	joule per second-degree kelvin
J/s-m ² -°K	joule per second-square meter-degree kelvin
°K	degree kelvin
k	thermal conductivity
K _{EFF}	thermal conductivity of parallel aluminium/water arrangement
kg	kilogram
K _L	thermal conductivity of water
K _M	thermal conductivity of aluminium
kN	kilonewton
kN/m ²	kilonewton per square meter

Nomenclature (Continued)

$\text{kN/m}^2\Delta$	kilonewton per square meter delta
ks	kilosecond
LCG	liquid cooling garment
lbm, lb	pound mass (avoirdupois)
lbm/hr, lb/hr	pound mass per hour
m	meter
mm	millimeter
N	newton
Np	number of flow passes in heat exchanger
P	plate open area divided by plate total area
psi	pounds force per square inch
psia	pounds force per square inch absolute
psi	pounds force per square inch delta
Q,q	heat transfer rate, heat load
s	second
T_{EVAP}	evaporant temperature of metal boiling surface
T_f	freezing point temperature of water
T_i	water inlet temperature
T_o	water outlet temperature
torr	pressure measured in millimeters Hg
UA	overall subsystem thermal conductance
V	heat exchanger core volume
W	water mass flow rate
X	fin height, length of heat transfer path
ΔT	temperature drop
η	fin efficiency, dimensionless conductance
η_f	fin efficiency
μ	pressure measured in micron Hg

SUBSYSTEM DESIGN AND ANALYSIS

The Laboratory Model Ice Pack Heat Sink Subsystem delivered as an end item of this contract consists of an ice chest/liquid cooling garment (LCG) heat exchanger combination whose operation was demonstrated during the feasibility testing portion of this program and whose performance capabilities were proven during the development portion, and an ice pack console. This console is designed to maintain contact between the ice chest and liquid cooling garment heat exchanger, and provides the liquid loop components which circulate the liquid, chilled in the heat exchanger by the ice chest, to the external liquid cooling garment. Figure 1 is a front photograph of the Ice Pack Heat Sink Subsystem Console. Figures 2 and 3 are left side and rear photographs of the Ice Pack Heat Sink Subsystem Console. Figure 4 is a photograph of the Ice Chest.

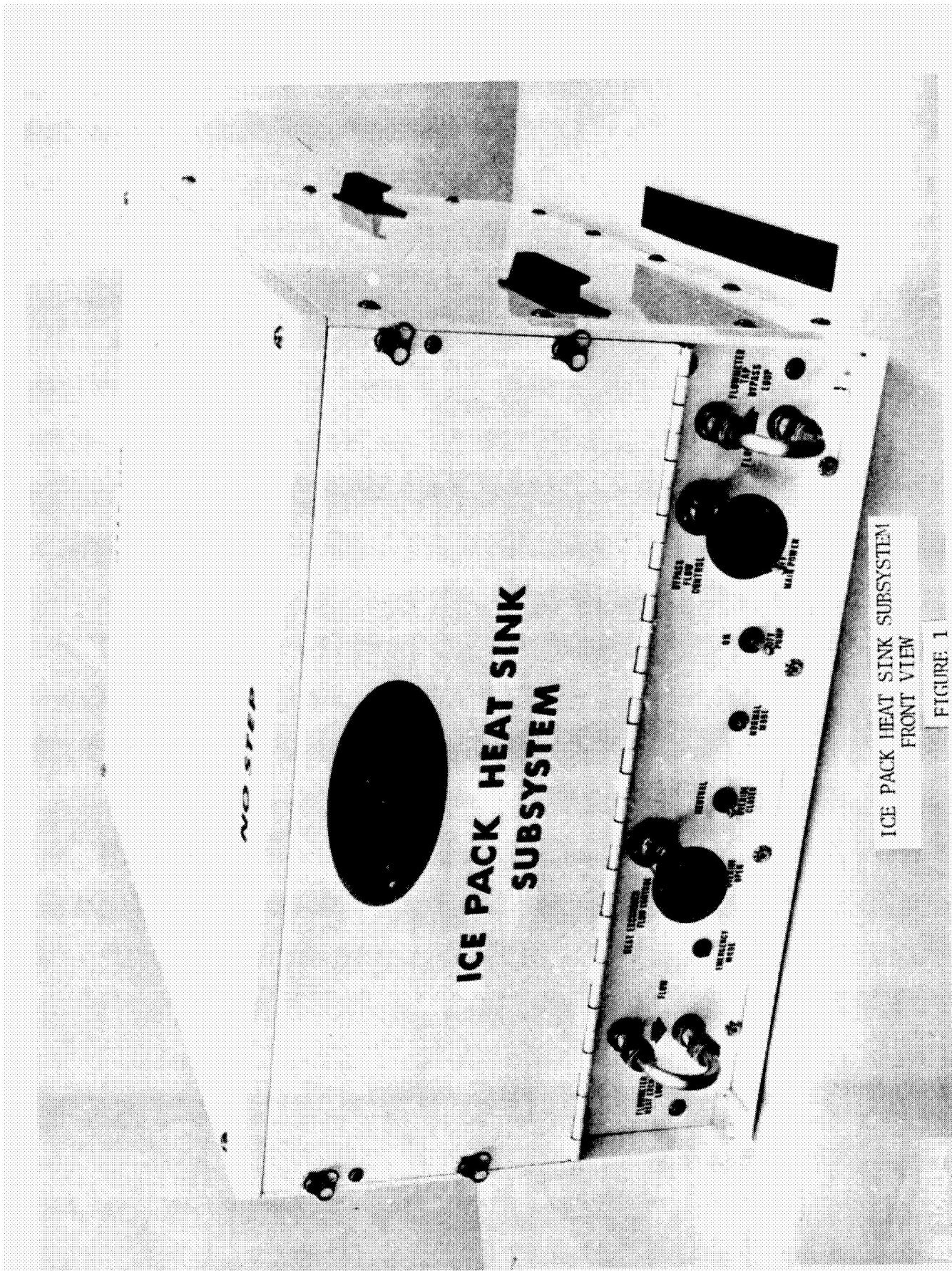
SUBSYSTEM CONCEPT

As designed, the Ice Pack Heat Sink Subsystem contains an independent ice chest which is mounted to a plate-fin LCG heat exchanger in which the LCG coolant is circulated. The independent ice chest is filled with water and frozen. The ice is contained in wicking material, and fins are incorporated to provide adequate heat transfer from the ice chest mounting surface to the ice. Closed cell foam is incorporated in the ice chest to serve as an ice expansion compensation device.

As the LCG coolant is circulated through the plate-fin LCG heat exchanger, the melting ice provides a heat sink which cools the LCG coolant over the complete range of heat loads. Temperature control is provided by the manual temperature control valve which determines the flow split between the heat exchanger and the bypass loop.

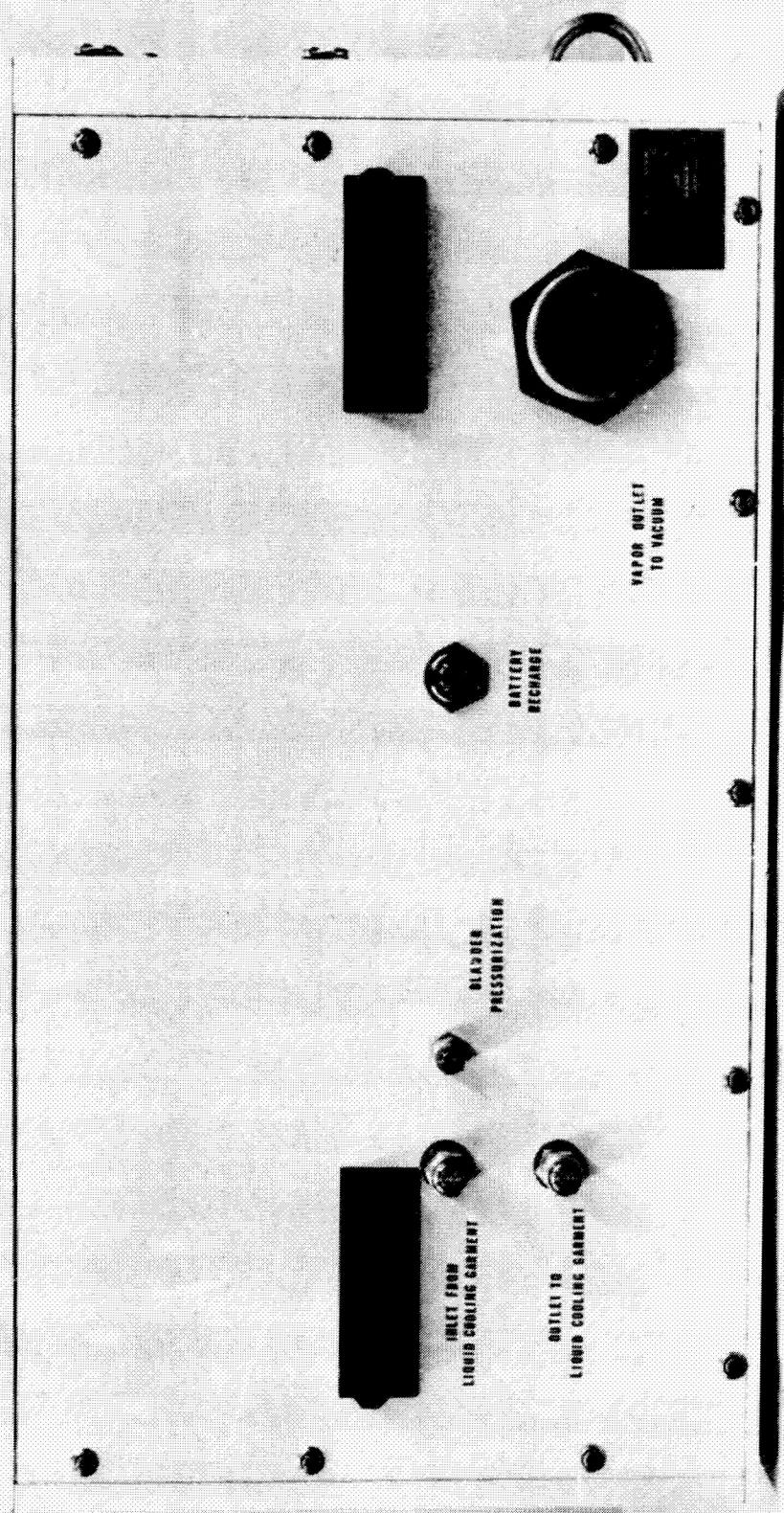
When the temperature in the heat exchanger outlet exceeds 288.7°K (60°F), a sign that all the ice has melted and that the unit can no longer satisfy the cooling requirements under normal operation, the ice chest may be set to automatically convert into a water boiler by venting the vapor passages to a vacuum ambient. Water within the ice chest is then internally **transported** by the wicking material to a metal boiling surface, where the subsequent evaporation of water is sufficient to cool the LCG coolant.

The initial charge and subsequent emergency recharges in the space vehicle are accomplished by filling the ice chest with water through a fill port. When the ice chest is filled with water, the fill port is capped and the ice chest is refrigerated. The ice chest is available for use as soon as all the water in it has frozen.



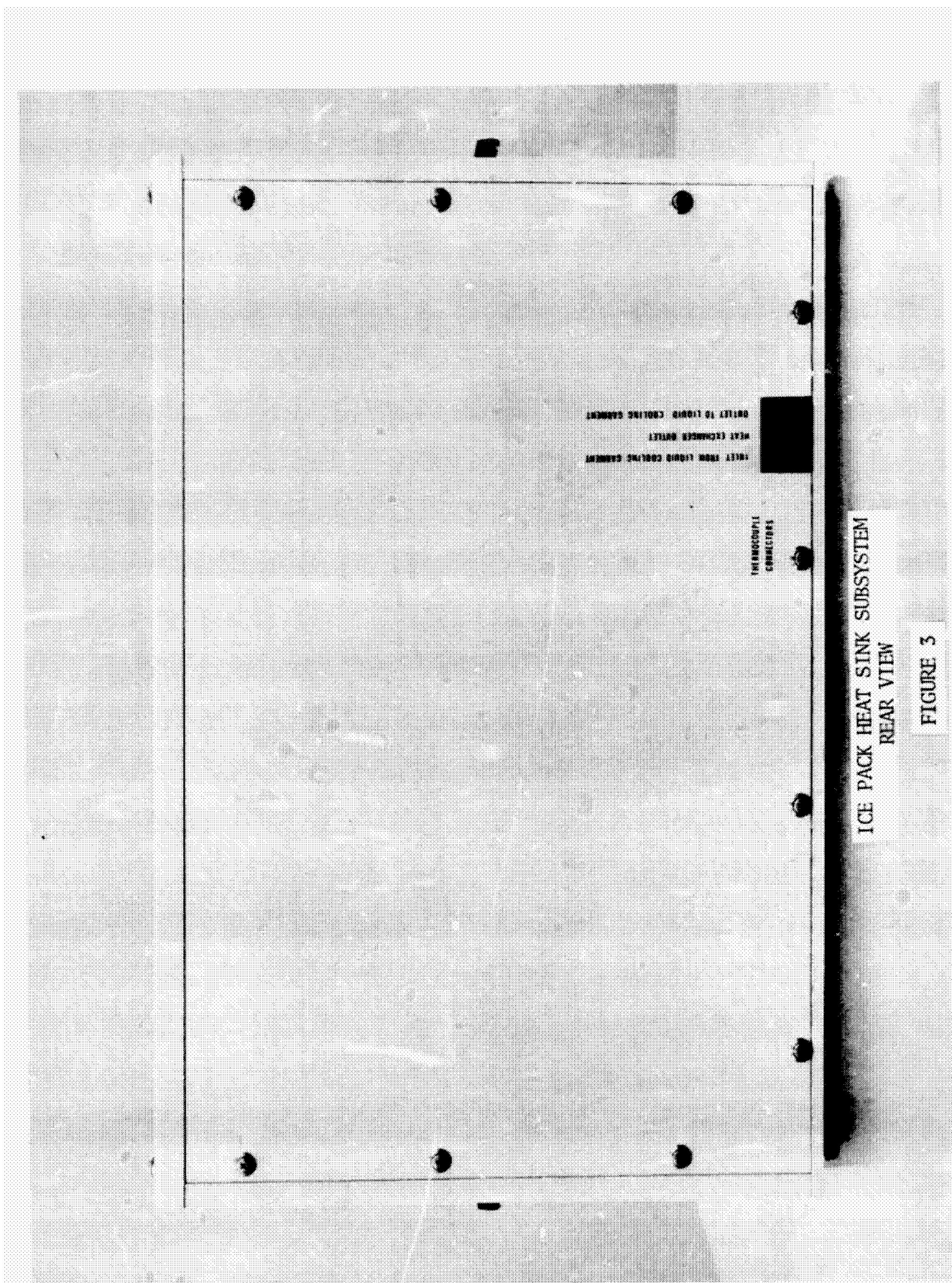
ICE PACK HEAT SINK SUBSYSTEM
FRONT VIEW

FIGURE 1



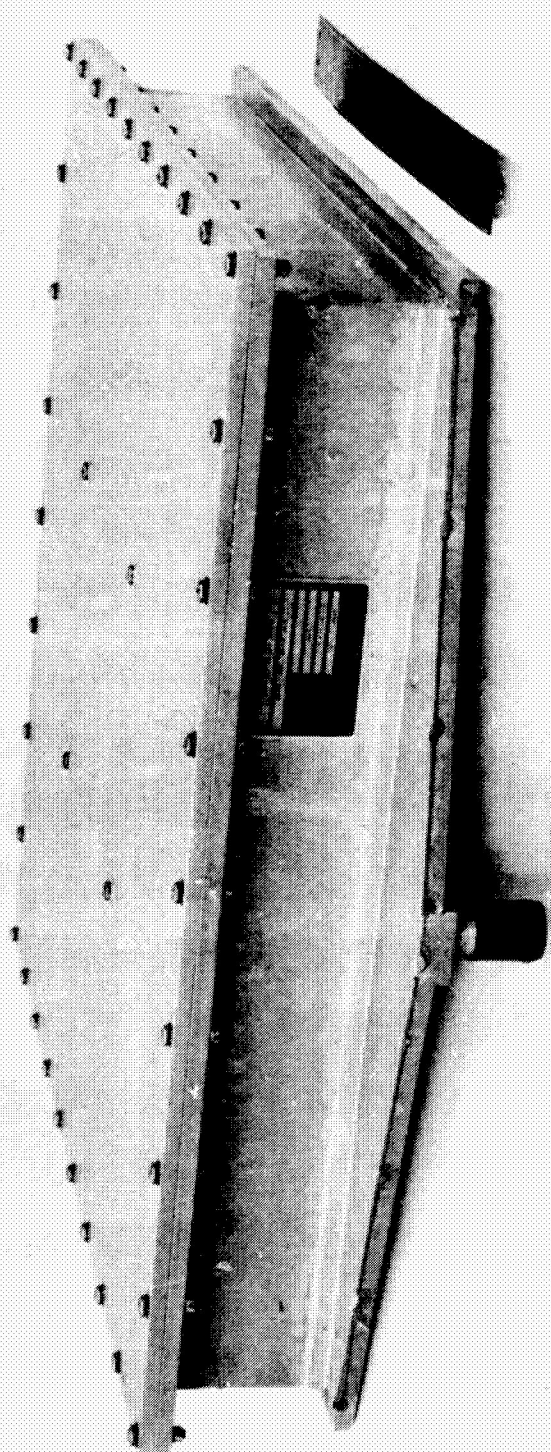
ICE PACK HEAT SINK SUBSYSTEM
LEFT SIDE VIEW

FIGURE 2



ICE PACK HEAT SINK SUBSYSTEM
REAR VIEW

FIGURE 3



ICE CHEST FRONT VIEW
FIGURE 4

Using this concept, recharging during EVA is conveniently accomplished by mechanical removal and replacement of the ice chest portion of the subsystem.

Detailed evaluation of the many facets of this design including conductances, wicking devices, ice expansion compensation device, ice chest sizing, evaporation mechanism and overall subsystem performance for both normal and emergency operation are included in the following sections.

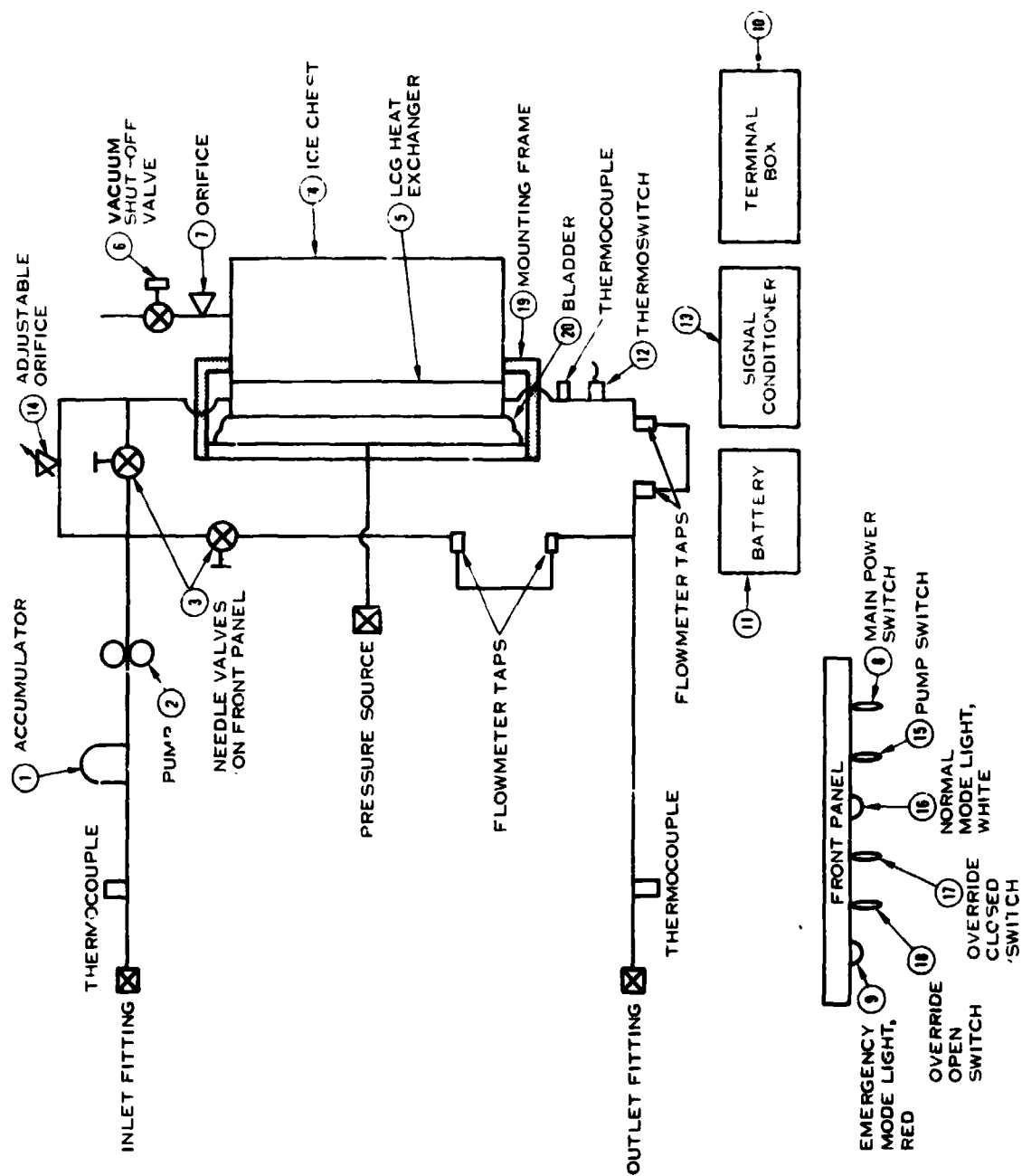
SUBSYSTEM FUNCTIONAL DESCRIPTION

The Ice Pack Heat Sink Subsystem is a regenerable thermal control subsystem containing 7kg (15.43 lbm) of ice and capable of rejecting heat in a range from 139.5 J/s to 879 J/s (475 Btu/hr to 3000 Btu/hr). A functional schematic of the subsystem is presented in figure 5.

The Ice Pack Heat Sink Subsystem consists schematically of a liquid heat transport loop containing inlet and outlet fittings, an accumulator, a pump, an ice pack heat sink, valves, a temperature sensor, and thermocouples. The inlet and outlet fittings are attached to a thermal heat source to simulate a space suit liquid cooling garment (LCG). By measuring total flowrate in the LCG and the temperature difference at the two thermocouples situated at the inlet and outlet, the heat source may be measured and controlled.

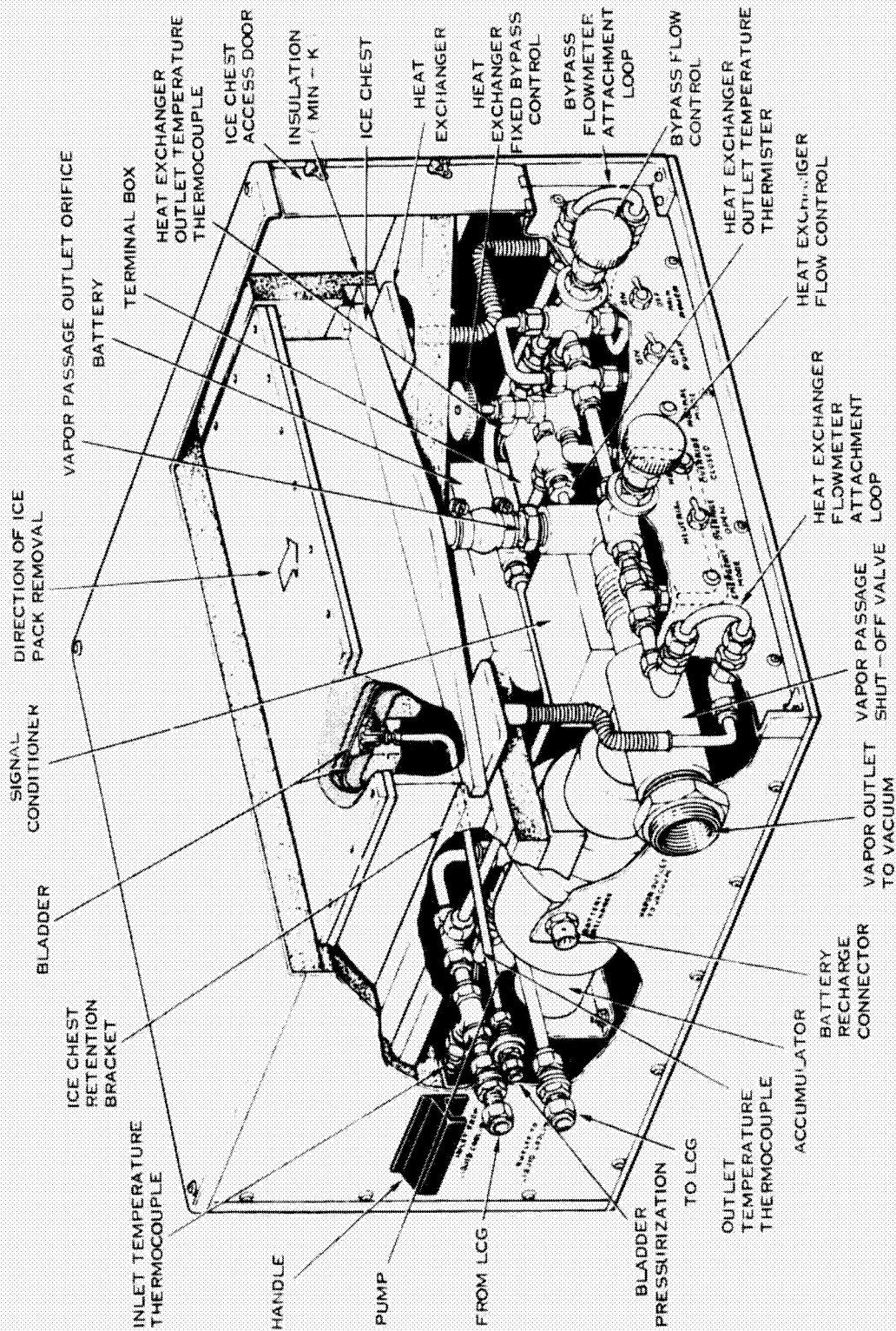
The accumulator (Item 1), which pressurizes the LCG coolant upstream of the pump to 143 kilonewtons/m² absolute, minimum (20.7 psia, minimum), is a spring loaded diaphragm device. Locating it upstream of the pump (Item 2) prevents pump cavitation. Since the accumulator is a gage device, the pump inlet pressure will be 41 kilonewtons/m² absolute, minimum (6 psia, minimum), if the total system is placed in a vacuum.

Two needle valves (Item 3), one in the LCG heat exchanger loop and one in the heat exchanger bypass loop, allow adjustment of the coolant flow through each loop thereby allowing the total flow and the flow split to be precisely set. During low heat load conditions most of the coolant flow is bypassed around the LCG heat exchanger, and during maximum heat load conditions all of the flow is circulated through it. A preset adjustable orifice (Item 14) is included to bypass the needle valve in the LCG heat exchanger loop. This adjustable orifice is set to maintain a minimum flow through the heat exchanger in the event the heat exchanger loop needle



ICE PACK HEAT SINK SUBSYSTEM SCHEMATIC

FIGURE 5



ICE PACK HEAT SINK SUBSYSTEM

FIGURE 6

valve is inadvertently shut-off with the ice chest installed. The orifice thereby prevents the heat exchanger coolant flow from being stagnated, which would allow it to freeze and plug the heat exchanger.

Flowmeter taps are provided in both the LCG heat exchanger loop and the bypass loop. These taps are located on the front panel and are utilized by removing jumper tubes and installing the appropriate flowmeters.

The heat sink portion consists of the ice chest, (Item 4), the LCG heat exchanger (Item 5), a bladder (Item 20) and mounting frame (Item 19), a pressure source port, a thermoswitch (Item 12), a vacuum shut-off valve (Item 6), and two interchangeable orifices (Item 7). The two most important parts of the heat sink portion are the heat exchanger and the ice chest.

The heat exchanger is a plate-fin, aluminum, six-pass, single passage device through which the coolant water flows. The heat load is transferred from the heat exchanger to the ice chest by conductors and is finally stored in the ice chest by the melting of the ice during normal operation. Emergency operation is initiated by the thermoswitch (Item 12) when a high temperature, 288.7°K (60°F), at the outlet of the heat exchanger is reached. The thermoswitch actuates the vacuum shut-off valve (Item 6) via the signal conditioner (Item 13). When this occurs - assuming all the water in the ice chest has melted - LCG thermal conditioning is maintained by the transformation of the ice chest into a water boiler. Two orifices are provided in the test hardware to control flow. An orifice diameter of 0.79 cm (.311 inch) provides a freezing condition at 117.2 J/s (400 Btu/hr) and effective cooling up to 411 J/s (1400 Btu/hr). The second orifice diameter of 1.53 cm (.590 inch) will freeze-up under a heat load of 366 J/s (1250 Btu/hr) or less and provides effective cooling up to the maximum heat load of 879 J/s (3000 Btu/hr). Attachment of the ice chest to the heat exchanger is accomplished by sliding the ice chest between the mounting frame assembly and the heat exchanger. The back side of the heat exchanger is pressurized from suit pressure 54 kilonewtons/m², (8 psi), thereby clamping the ice chest between the mounting frame and the heat exchanger to provide predictable heat transfer conductance.

A terminal box (Item 10) contains all electrical wire lead junctions. A battery (Item 11) provides system power.

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On the front panel are mounted the main power switch (Item 8), the pump switch (Item 15), the normal mode indication white light (Item 16), the emergency mode indication red light (Item 9), and the switches for providing override of the vacuum shut-off valve - the override closed switch (Item 17) and override open switch (Item 18).

In an actual flight system application, the ice chest will be replaced periodically during the mission. It is assumed that a cart or vehicle will be readily accessible to the crewman, where the replacement ice chest will be stowed in an insulated compartment. The replacement ice chest is visually inspected and readied for insertion into the mounting frame. First the spent ice chest must be removed from the crewman's life support system. The bladder pressurization line is vented to vacuum by turning a three-way valve. This closes-off the suit line and vents the back side of the bladder thus relieving the pressure on the ice chest. The ice chest is then slid out from the mounting frame. The replacement ice chest is inserted and the bladder pressurization line is repressurized by turning the valve. The spent ice chest is stowed in the "used" compartment. The actual method employed for replacement is dependent on the suit/life support system packaging design. For a front mounted life support system or integrated suit, replacement is relatively easy because it may be accomplished within view of the crewman's line of sight. For a back mounted life support system, a donning station and procedure may be required.

SUBSYSTEM PHYSICAL DESCRIPTION

A cut-away view of the Ice Pack Heat Sink Laboratory Model is shown in figure 6. The ice chest and heat exchanger assembly are surrounded by 2.54 cm (1 inch) of Min-K insulation and mounted on top of a structural frame. For ice chest insertion in the direction shown, the front panel of the structure is opened.

The pump, accumulator, battery, vacuum shut-off valve, signal conditioner, and terminal box are attached directly to the structure base plate. The adjustable orifice is located on the back side of the front control panel and is accessible only when the front panel is opened.

The front panel contains the two flow regulating needle valves, the normal mode and emergency mode pilot lights, the main power switch, the pump switch, the override open switch, the override closed switch, and the flowmeter taps for the heat exchange and the bypass lines.

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The inlet and outlet fittings for connection to the liquid cooling garment or dummy load are on the left side as is the fitting for the bladder pressurization line.

Isometric pictures of the Ice Chest and Heat Exchanger are shown in figures 7 and 8 for the normal and emergency modes, respectively. This section describes the ice chest and the heat exchanger in detail.

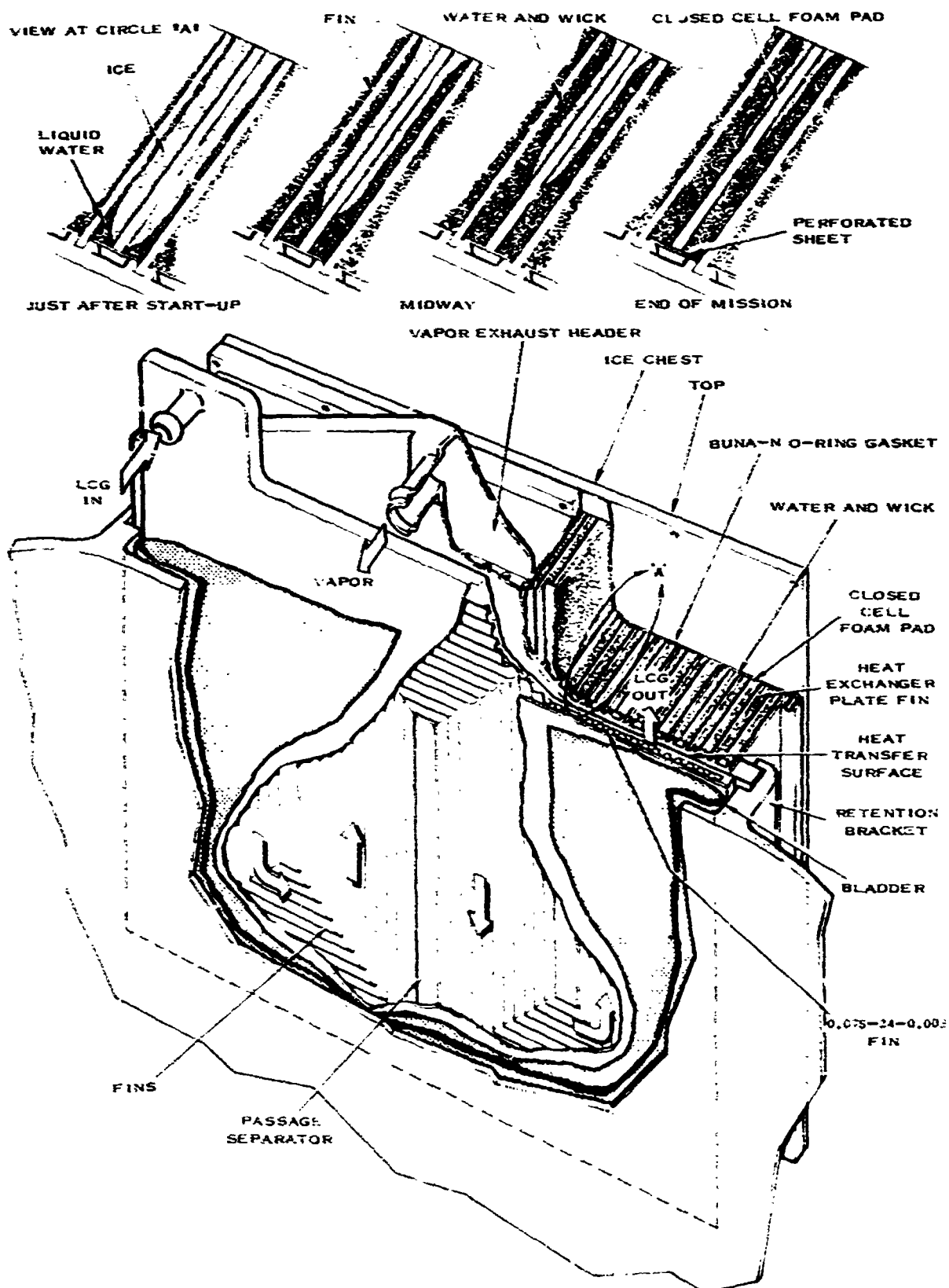
The heat exchanger is made from aluminum and its construction consists of a single passage, six-pass plate-fin heat exchanger. The selected fin for this application is a 1.9 mm (0.075 inch) high fin with 9.5 fins per cm (24 fins per inch) and a 0.076 mm (0.003) inch wall thickness. Headering between passes is accomplished by an internal headering arrangement consisting of fins cut and fitted at right angles and internal pass separators. The end sheet facing the ice chest - the heat transfer surface - is made extra thick to permit finish machining after brazing. The passage separators and the heat exchanger closure bars around the periphery of the heat exchanger are machined into this end sheet. The other end sheet - side opposite the ice chest - is made out of standard aluminum sheet stock. The inlet and outlet headers are brazed in place and are of standard heat exchanger header construction. The whole assembly is fluxless brazed.

The ice chest consists of a housing, top, buna-N O-ring gasket, wick, and closed cell foam pads. The housing is constructed predominantly of sheet metal wherein the sides and fins are spaced on a pitch of 1.5 cm (0.6 inch). The fins run the full length of the ice chest and from top to bottom. The bottom of the ice chest is constructed of aluminum plate stock machined with grooves in which the fins are placed and additional grooves which act as the vacuum passage during emergency operation. These grooves run the full depth of the ice chest between the fins and run into the vapor exhaust header. On top of the vacuum grooves, and in between the fins, is a perforated sheet. This perforated sheet acts as the boiling heat transfer surface during emergency operation. The entire sheet metal assembly described above is fluxless brazed to form a structural unit. After brazing, the underside of the bottom of the ice chest is finish machined to ensure flatness.

The spacing between each fin is 1.27 cm (0.50 inch). Inserted in this cavity are two layers of 5 mm (0.20 inch) thick wicking material separated by a layer of 2.5 mm (0.10 inch) thick closed cell foam and foam retainer. The closed cell foam is utilized to account for the expansion and contraction of ice. The theory is that upon being cooled, ice will form first at the fins and progressively freeze toward the closed cell foam. The lid is constructed of aluminum plate stock and the gasket is a buna-N O-ring.

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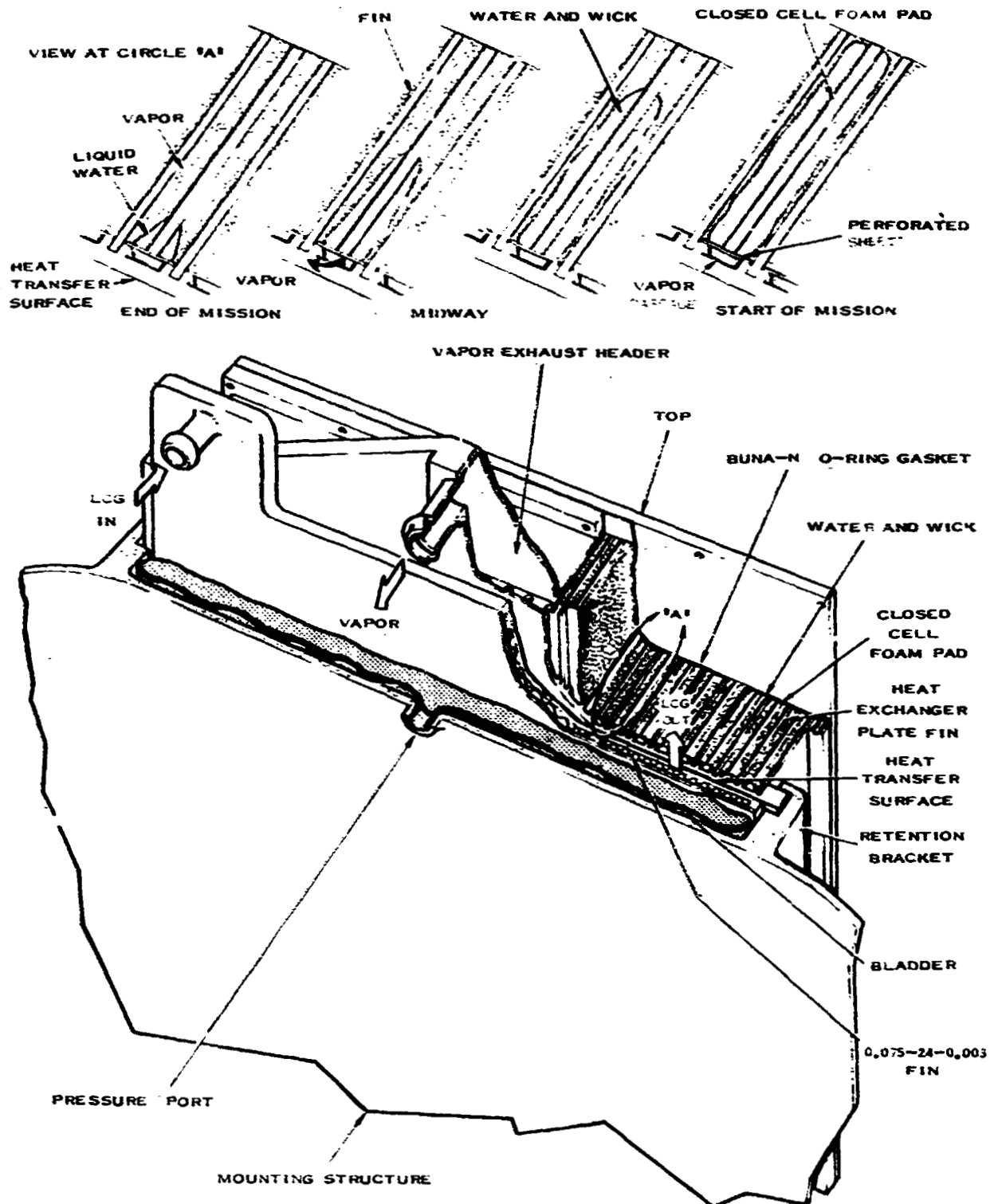


ICE CHEST/HEAT EXCHANGER - NORMAL OPERATING MODE

FIGURE 7

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ICE CHEST/HEAT EXCHANGER - EMERGENCY OPERATING MODE

FIGURE 8

The wick is the means by which the water is transferred to the perforated sheet metal boiling surface during emergency operation; i.e., a wick-fed water boiler. Therefore, during normal operation the water is contained within the wick and is frozen within it. The progression of melt/freeze lines is depicted at the top of figure 7 from beginning to the end of the mission.

For emergency operation, the electrically actuated shut-off valve is activated, exposing the vapor exhaust passage to vacuum. When the vapor pressure at the perforated sheetmetal boiling surface falls to the saturation pressure consistent with the heat transfer input rate, boiling occurs. Vaporization also occurs at the fins to maintain pressure equalization and to fill the void formed by the water being wicked to the boiling heat transfer surface. A progression of vapor/liquid lines as shown in figure 8 depicts the path the water will take within the wick from start to finish of the emergency mission mode.

A complete description of each piece of hardware is contained in Appendix J, Ice Pack Heat Sink Subsystem Operating Instructions and Component Specifications.

SUBSYSTEM ANALYSIS

Overall conductance from the ICG cooling water to the ice (or boiling water) was broken down into three conductances:

- ICG cooling water to heat exchanger end plate (convection).
- Heat exchanger end plate to ice chest end plate (contact conductances).
- Ice chest end plate (fin base) to ice (conduction).

These conductances are evaluated in detail in the following three sections. These individual conductances were then combined to obtain an overall conductance using the relationship.

$$\frac{1}{UA} = \frac{1}{C_1} + \frac{1}{C_2} + \frac{1}{C_3}$$

Heat exchanger effectiveness was determined utilizing counter flow heat exchanger data (a good approximation for the six-pass, cross flow configuration).

The effectiveness then was used to calculate outlet water temperature for various inlet water temperatures:

$$T_o = T_i - E(T_i - T_f),$$

where T_o is water outlet temperature, T_i is water inlet temperature, E is effectiveness, and T_f is the freezing point of water. The heat transfer rate then was determined:

$$Q = W c (T_i - T_o),$$

where W is the water flow rate and c is the water specific heat. Finally, graphs were made to show the relationships among heat transfer rate, water flow rate, heat sink inlet temperature, and time (percent of ice melted). Figure 9 shows effectiveness versus water weight flow for various times during normal operation.

The following three sections evaluate the individual rate-controlling conductances from the LCG cooling water to the ice. For normal operation at time zero and for emergency operation, the fin base-to-ice (or to boiling water) conductance is not rate controlling, and it is therefore taken as infinite.

Conductance from LCG Water

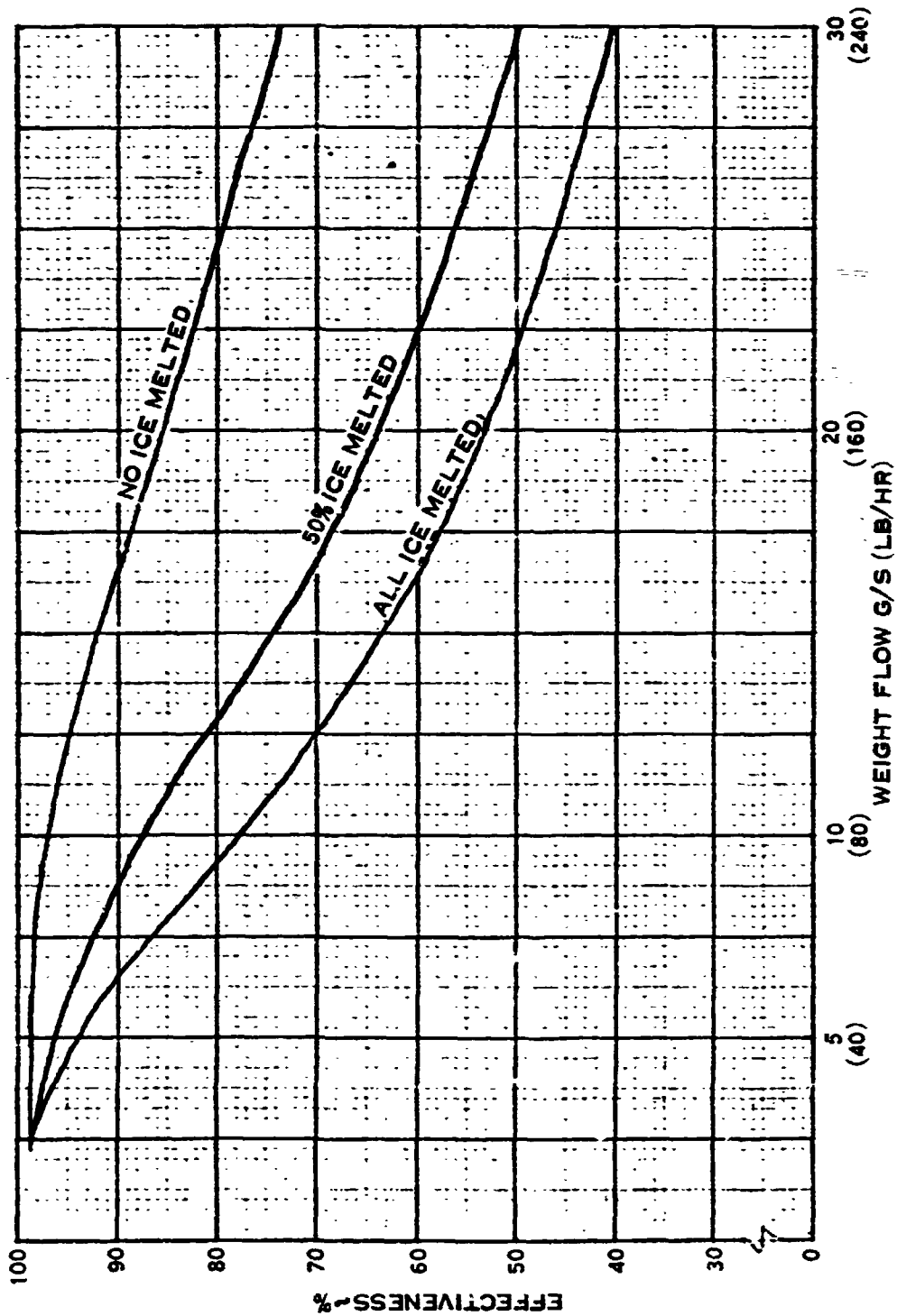
As the LCG cooling water flows through the finned, multipass heat exchanger, heat is transferred from the water, through the fins, to the heat exchanger end plate. The corresponding conductance is hA , where h is the convection coefficient at the interface and A is the effective area. This area

$$A = A_p + A_s \eta,$$

where A_p is the primary area, A_s is the secondary (fin) area, and η is the fin efficiency.

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NORMAL OPERATION PREDICTED LCG HEAT EXCHANGER
EFFECTIVENESS VERSUS LCG WATER FLOW

FIGURE 9

The value of the conductance depends on heat exchanger configuration; that is, on number of passes, face area, flow length, and fin height, spacing, thickness, and material. For a given heat exchanger configuration and fluid, conductance depends on fluid flow rate. Based on the proposed design, conductance was determined for several flow rates, as follows:

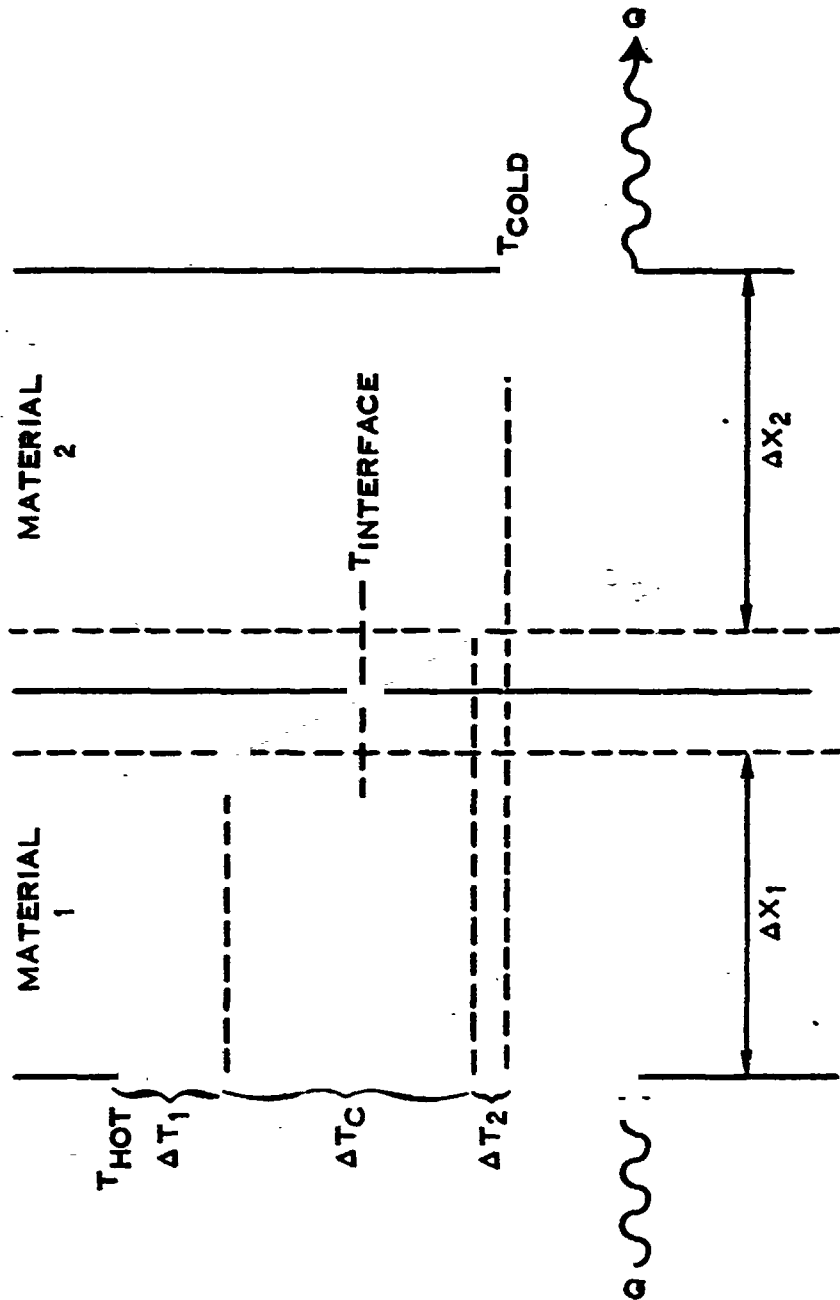
<u>Flowrate</u>		<u>Conductance</u>	
<u>g/s</u>	<u>(lb/hr)</u>	<u>J/s - °K</u>	<u>(Btu/hr - °F)</u>
2.5	(20)	970	(1838)
7.6	(60)	1233	(2333)
15.2	(120)	1950	(3685)
22.8	(180)	2616	(4945)
30.4	(240)	3200	(6065)

The values of conductance, hA , were determined from Hamilton Standard test data of this fin configuration which relates hA/V versus WN_p/A_f , where V is heat exchanger core volume, W is water flow rate, N_p is number of passes, and A_f is core face area.

Interface Contact Conductance

When two surfaces are brought together forming an interface across which heat must flow, a discontinuity in the system temperature profile (figure 10) will occur at the interface. The temperature profile within materials (1) and (2) will be a direct function of their thermal conductivities, k_1 and k_2 , but the definition of the contact temperature discontinuity is not so easily described. Considerable attention to this definition has been generated since the late 1950's due to stringent aerospace requirements. In the actual design, the contact resistance comprises a major portion of the total system temperature drop and was the subject of considerable attention during the feasibility phase. Although this problem could have been avoided through the utilization of a one piece ice chest/LCG heat exchanger, the inherent drawbacks of that configuration (logistics, performance and potential LCG freeze-up) would have produced problems with substantially higher development risk.

The actual area of the two materials in contact is a rather small fraction (possibly one to ten percent) of the projected area and is a direct function of the contact pressure or force holding the joint together and the hardness of the materials. Roughness and flatness of the two contacting surfaces result in peaks which join to form the effective heat flow area of the joint. The valleys between the peaks produce a gap which essentially insulates that portion of the heat flow path (especially in a vacuum environment). Conductance can be



$$Q = \frac{K_1 A \Delta T_1}{\Delta X_1} = h_c A \Delta T_C = \frac{K_2 A \Delta T_2}{\Delta X_2}$$

TYPICAL TEMPERATURE PROFILE

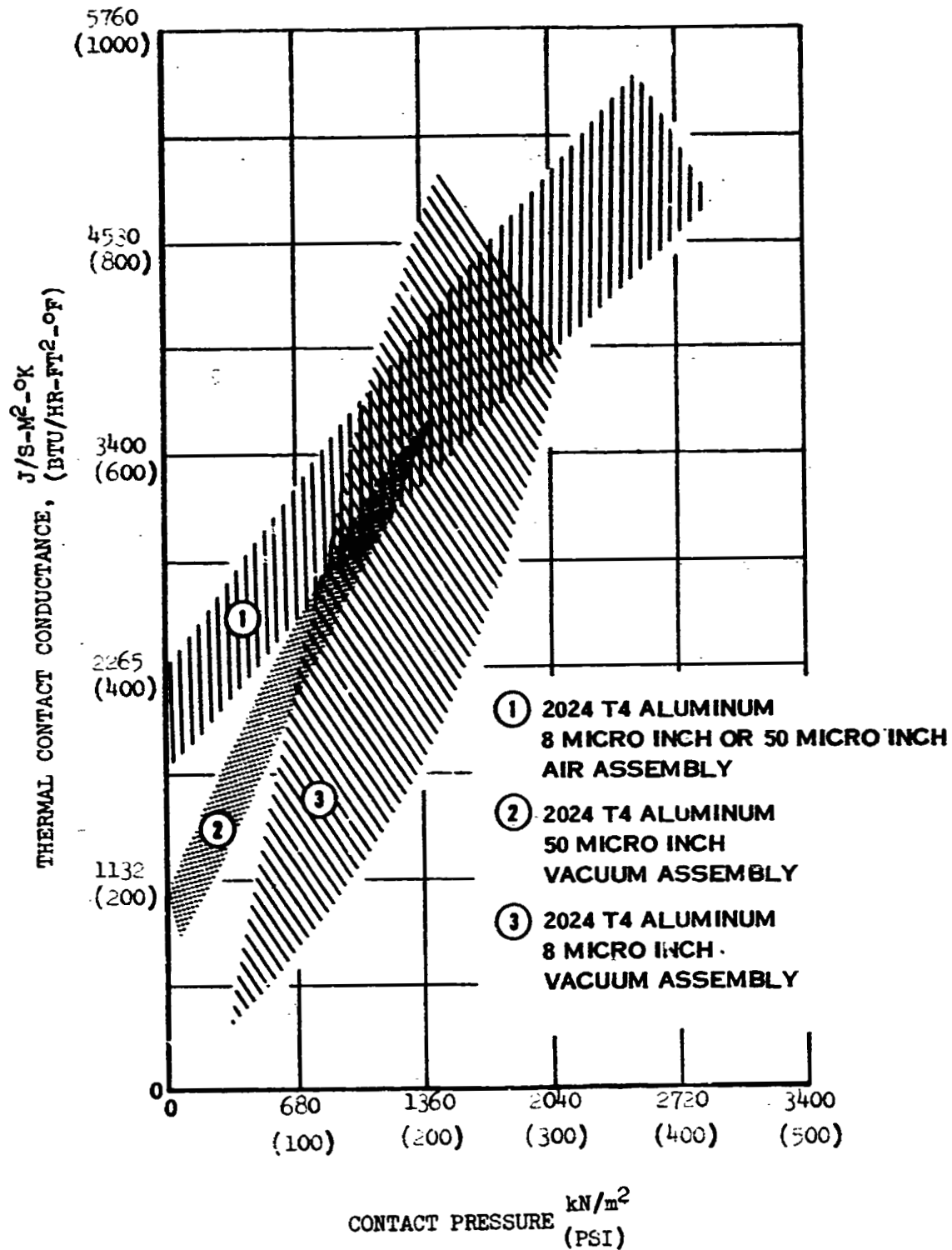
FIGURE 10

increased by increasing the contact pressure, which produces plastic deformation of the peaks thus increasing the effective heat flow area. Also, soft materials may be applied to the interface to deform under low load and fill the valleys providing parallel paths for heat flow. Obviously, the higher the plasticity and thermal conductivity of the interstitial material, the greater the beneficial effect.

For this design, the greatest hurdle was in the extrapolation of reported data down to a range of contact pressure achievable by a crewman during EVA. Much data is reported at loadings above 680 kN/m^2 (100 psi) while loadings for this effort will be 54 kN/m^2 (8 psi) or less. Data from Reference 1, Appendix I are presented in figure 11 and are the basis for the following conclusions.

- Contacts assembled in a pressurized environment will have substantial conductance at low applied loads as a result of gas trapped in the interface. This gas persisted in the joint throughout a one-week period during the Reference 1 testing.
- Materials having a very fine finish, which are assembled in vacuum, have essentially zero conductance at zero contact pressure.
- Rougher materials, assembled in a vacuum, exhibit some conductance at low contact pressures but verification of extrapolations to very low loads is required. The rate of conductance drop-off apparently is quite high in this region.

A contact conductance of $1132 \text{ J/s-m}^2\text{-}^\circ\text{K}$, ($200 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$) was assumed for the assembly. This value is considered conservative for contacts assembled in a pressurized environment such as the lunar base or spacecraft cabin, but is unrealistic for a vacuum assembly such as resupply during EVA.



CONTACT CONDUCTANCE DATA

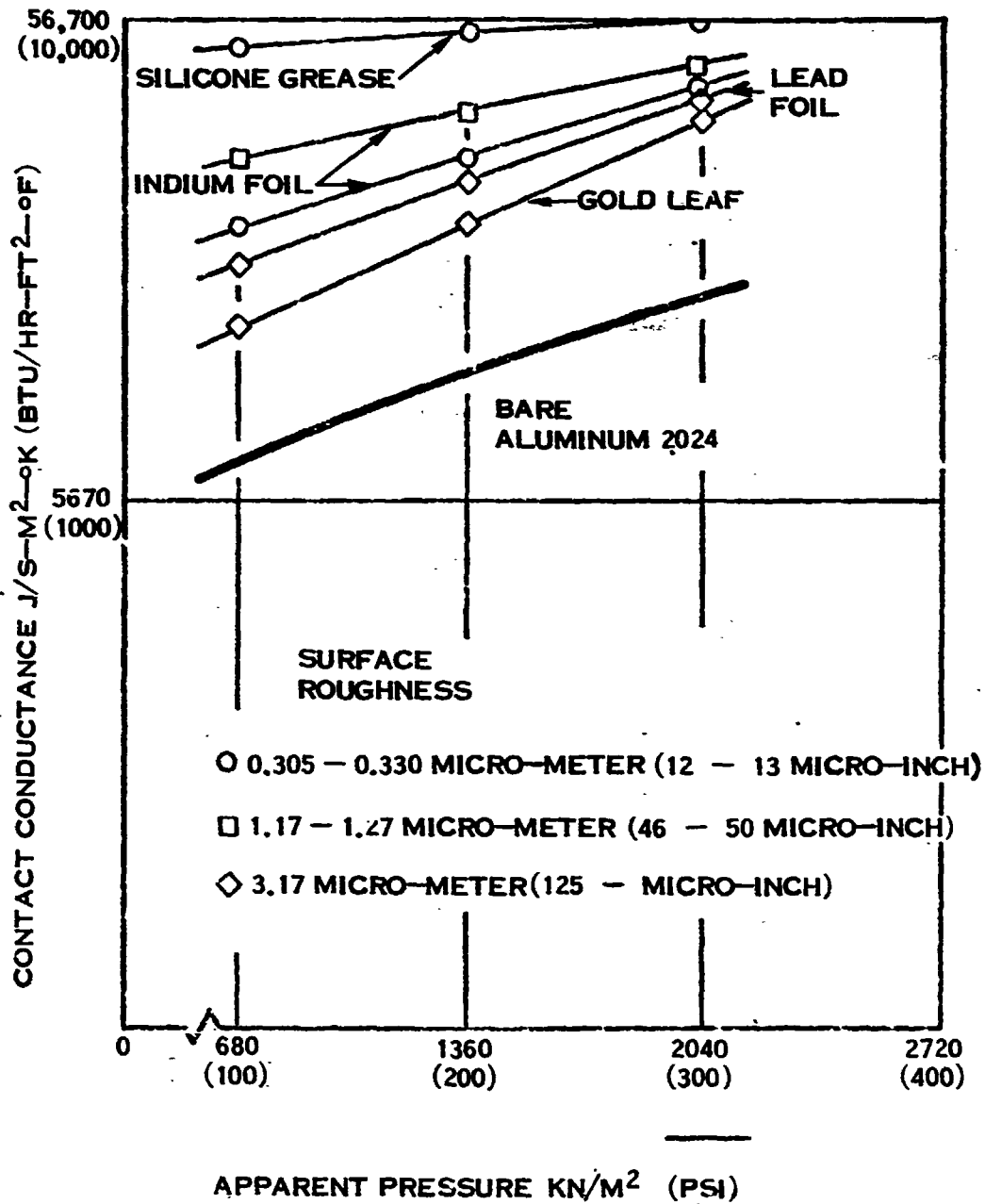
FIGURE 11

To attack this problem and improve conductance for all modes of assembly, the inclusion of an interstitial material is required. Data presented in Reference 2 and shown in figure 12 show an order of magnitude improvement in contact conductance through the application of silicone vacuum grease and substantial improvements with indium, lead or gold. Although the grease appears to hold the highest performance potential, it poses the practical problems of contamination during assembly and excessive force requirements for disassembly. Data presented in Reference 3 and shown in figure 13 substantiates the findings in Reference 2 with contact pressures in the range of 136 - 204 kN/m² (20 - 300 psi).

The designed unit has lead plating applied to the relatively rough, approximately 0.81 micro-meter (32 micro-inch) finish of the LCG heat exchanger. A rough finish will provide the peaks necessary to load and plastically deform the lead with the intended aim of filling the valleys and voids between the two contacting surfaces. For this reason it also would be desirable to provide an equally "rough" finish on the mating surface of the ice chest. Potential disassembly problems, however, have dictated a 0.1 micro-meter (4 micro inch) finish on this part to minimize friction.

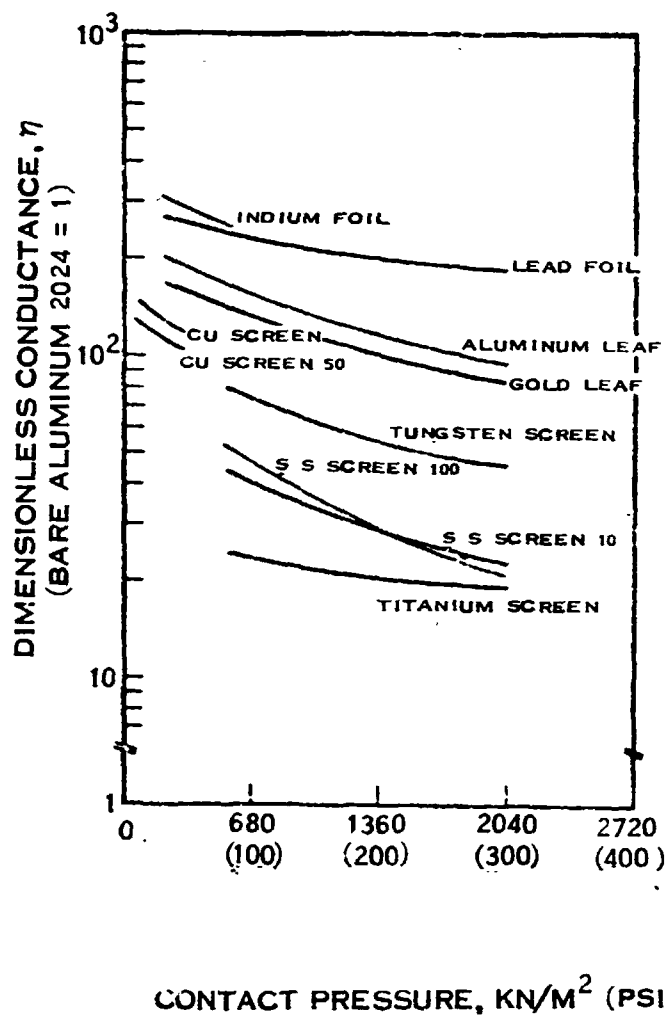
Several other materials and applications were considered because of their effect on contact conductance. Copper was eliminated by corrosion and storage requirements. Aluminum was disregarded because identical materials will cold weld under vacuum conditions. Although indium has a desirable effect on contact conductance, its cost is prohibitive when compared to lead. A plating procedure for the lead was deemed more applicable than leaves and foils, as leaves and foils will tear, necessitating replacement when the ice chests are removed and replaced. The lead plating is thus the most effective means of improving contact conductance by use of interstitial materials.

As presented in Reference 3 and figure 13, the use of lead plating will increase the contact conductance by approximately 200 times over that obtained with bare aluminum 2024. Thus, to get an overall contact conductance of 1132 J/s-m²-°K (200 Btu/hr-ft²-°F) requires a base metal junction contact conductance of only 5.67 J/s-m²-°K (1 Btu/hr-ft²-°F). Therefore, this portion of the design more than meets the requirements, holding the discontinuity in the system temperature profile (figure 10) to a minimum.



THERMAL CONTACT CONDUCTANCE OF SELECTED
INTERSTITIAL MATERIALS

FIGURE 12



COMPARISON OF DIMENSIONLESS CONDUCTANCE
FOR SELECTED METAL INTERSTITIAL MATERIALS

FIGURE 13

Fin-To-Ice Conductance

The final heat rejection stage is transfer of heat from the fin base to the melting ice. Most of the heat passes up the fin, through the water (melted ice), and into the ice. The rest of the heat flows directly from the fin base, through the water and into the ice. Early in the mission, when ice is in contact with the fins, the heat goes directly from the fins to the ice.

To determine the fin-to-ice conductance vertical and lateral recession analytical models were used, and the results were averaged. The resulting values were as follows:

<u>Percent of Ice Melted</u>	<u>Conductance</u>	
	<u>J/s-°K</u>	<u>(Btu/hr-°F)</u>
0	∞	∞(relative)
50	180	(341)
100	105.5	(200)

With no ice melted the conductance is of course finite, but it may be considered infinite relative to the conductances discussed in previous sections. The analytical models discussed here are represented in figure 14.

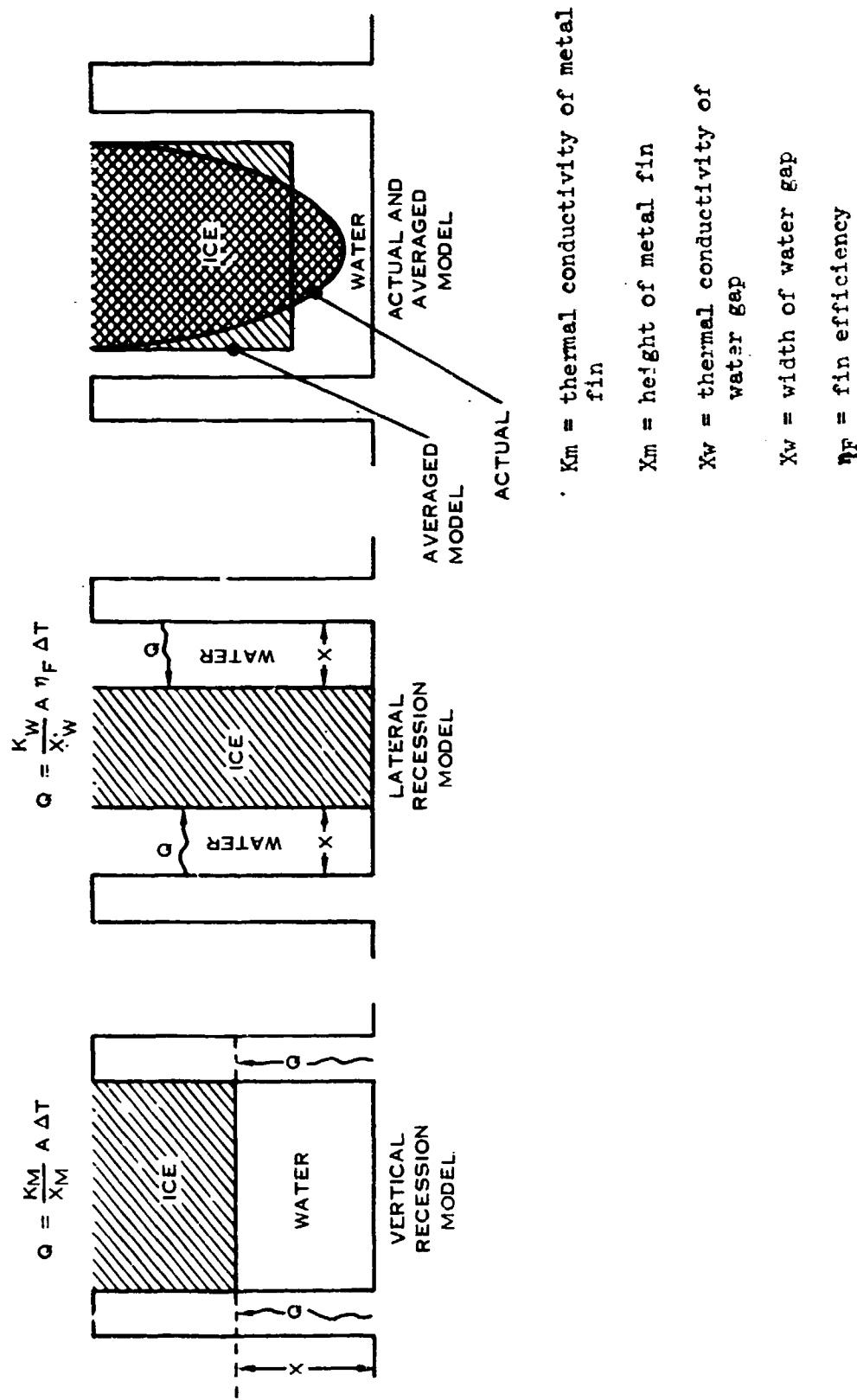
In the vertical model, the ice is always in contact with part of the fin, as it recedes vertically from the base to the top of the fin. Heat transfer through the water from the fin and fin base is negligible. Essentially all of the heat passes upward through the fin to the fin-ice interface at 273.1°K (32°F). The rate of heat transfer is therefore described by

$$q = kA\Delta T/x$$

where k is the thermal conductivity of the fin, A is the cross sectional area of the fin, ΔT is the temperature drop from the fin base to the fin-ice interface, and x is the height of the fin. For any heat transfer rate q, the temperature drop may then be calculated as

$$\Delta T = q x/(kA),$$

as x varies from zero at the start of the mission to the total fin height at the end of the mission.



ANALYTICAL MODEL FOR FIN-TO-ICE HEAT TRANSFER

FIGURE 14

In the lateral model, the ice is never in contact with the fin (except at time zero), and its surface is a uniform distance from the fin. Heat is transferred from the fin base upward along the fin and laterally through the water to the melting ice. Heat also flows from the fin base directly to the ice, but this was neglected to generate a conservative design. The heat flux from any point of the fin to the ice is described by

$$dq/dA = k \Delta T/x$$

where k is the thermal conductivity of water, ΔT is the temperature difference between the fin and the ice, and x is the distance from the fin to the ice. For the entire fin, the heat transfer rate is therefore described by

$$q = kA\eta \Delta T/x$$

where A is the fin lateral area, η is the fin efficiency, and ΔT is the temperature drop from the ice to the fin base. In this expression,

$$\eta = \tanh(2k_w L/k_f t x) / (2k_w L/k_f t x)$$

where k_w is thermal conductivity of the water, L is fin height, k_f is thermal conductivity of the fin, t is fin thickness, and x is the distance from the fin to the ice. For any heat transfer rate q , the temperature drop may then be calculated as

$$\Delta T = qx/kA\eta$$

as x varies from zero at the start of the mission to half the fin spacing at the end of the mission.

Fin temperature drops were then calculated for each model to account for the condition when the ice has all melted under a heat load of 870 J/s. (3000 Btu/hr). The temperature drop for the horizontal model is 7.45°K (13.4°F) and for the vertical model, 9.23°K (16.6°F). The average of these two values is 9.34°K (15.0°F).

Results for the two models are in fairly good agreement so that results are probably accurate within five percent. The right hand side of Figure 14 shows pictorially how averaging the two analytical models approximates the time situation.

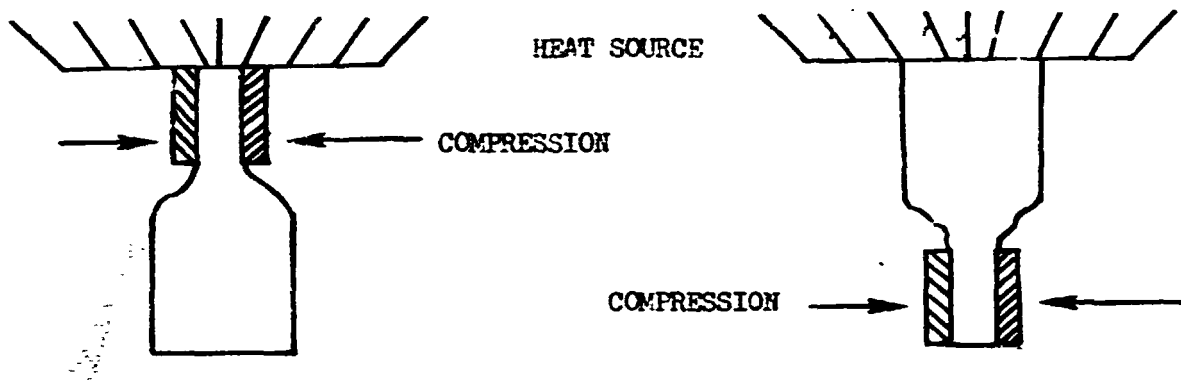
Wicking Device

A wicking device is used in the ice chest as a water transport medium to the boiling plate surface when all the ice has melted. When the solenoid actuated shut-off valve is activated exposing the vapor exhaust header to vacuum, water evaporates off the boiling plate surface. The wick is utilized to keep water always in contact with the surface.

Water flow in a wick not fed by a reservoir is caused by a wick density driving force; i.e., the small capillaries pull water from the larger capillaries. A wick with uniform capillary diameters will not induce flow. These facts were corroborated by test during the design of the ice chest. These tests are represented pictorially in figure 15.

Fig. 15 a) COMPRESSED AT HOT END

Fig. 15 b) COMPRESSED AT COLD END



TEST DEMONSTRATION OF WICKING MECHANICS

FIGURE 15

As represented by figure 15a., a wet wick was held against a hot surface and compressed to produce small capillaries at the hot end. In this case, water vapor could be observed forming at the hot end. Figure 15 b depicts the case where a wet wick was held against a hot surface and compressed at the end of the wick away from the hot surface. In this case, no water vapor was observed, and the wick began to burn at the hot end. These rather simple tests dramatically demonstrated both the wick density driving force and the need for the wick to be compressed in a controlled manner at the ice chest metal boiling surface, to ensure water flow to the boiling surface.

Along with the need for the wick to be compressible to ensure water transport, several other facets were also considered. One of these was capillary wicking head, which is the height of water that a wick can draw when placed vertically with one end in a liquid supply. A wick with sufficient capillary wicking head will provide the proper water retention in the ice cell when in a one-g testing environment. This precludes water dripping into the vapor passages where the possibility of freezing exists when the passages are vented to vacuum.

Another design criterion was that the wick should have high porosity. Porosity is defined as the wick void volume divided by the wick total volume. A wick of high porosity will provide for a smaller ice chest size than one with low porosity.

Table I shows candidate wicking materials and comments on each. From this information Dacron #67DA 18/.068 was chosen based on compressibility, capillary wicking head, and porosity considerations. A comparable wicking material was then tested to determine water flow characteristics when a heat load is applied.

A model ice chest cell was made of plexiglas, filled with wicking material, expansion foam and water, and hot air was blown over it. The purpose was to ascertain whether, for the range of conditions in which the Heat Sink Subsystem is to be used, the wick can transport enough water to satisfy the heat load requirements.

The heat sink available in any evaporating mechanism is dependent on the amount of mass evaporated and the heat of vaporization at the evaporant temperature. Any wicking device is flow limited at some maximum flow due to surface tension within the capillaries. Furthermore, there will always be some water retained in the wick which can not be drawn out, again due to surface tension effects. The test was run to determine the water transport and retention characteristics of the Dacron S4 materials, which exhibit characteristics similar to Dacron 67DA material.

TABLE I
CANDIDATE WICKING MATERIALS

Material	Description	Manufacturer	Porosity ($\frac{\text{Void Vol.}}{\text{Total Vol.}}$)	Wicking Capillary Head		Comments
				cm H ₂ O	in. H ₂ O	
Rayon	#432 Woven Synthetic	Troy Felt	0.580	2.22	0.875	Insufficient Capillary Head
	#182	Troy Felt	0.748	2.22	0.875	Insufficient Capillary Head
Dacron	#34-.070	Troy Felt	0.461	10.8	4.25	Insufficient Porosity
	#S4-.125	Troy Felt	0.525	11.8	4.63	Insufficient Porosity
	#62DA 11/125	American Felt	0.937	3.8	1.50	Insufficient Capillary Head
	#67DA 18/.068	American Felt	0.835	12.7	5	Good
Orlon	#2B-9	Troy Felt	0.920	1.27	0.5	Insufficient Capillary Head
Polypropylene	#342-1000	Troy Felt	0.774	7.6	3.0	Insufficient Porosity
	#342	Troy Felt	0.742	5.7	2.25	Insufficient Porosity and Capillary Wicking Head
Nylon Felt	452-24-344	Various	0.513	2.8	1.1	Insufficient Porosity and Capillary Wicking Head
Rafrasil Cloth	C100-96	Hiltco	0.580	25	10	Insufficient Porosity

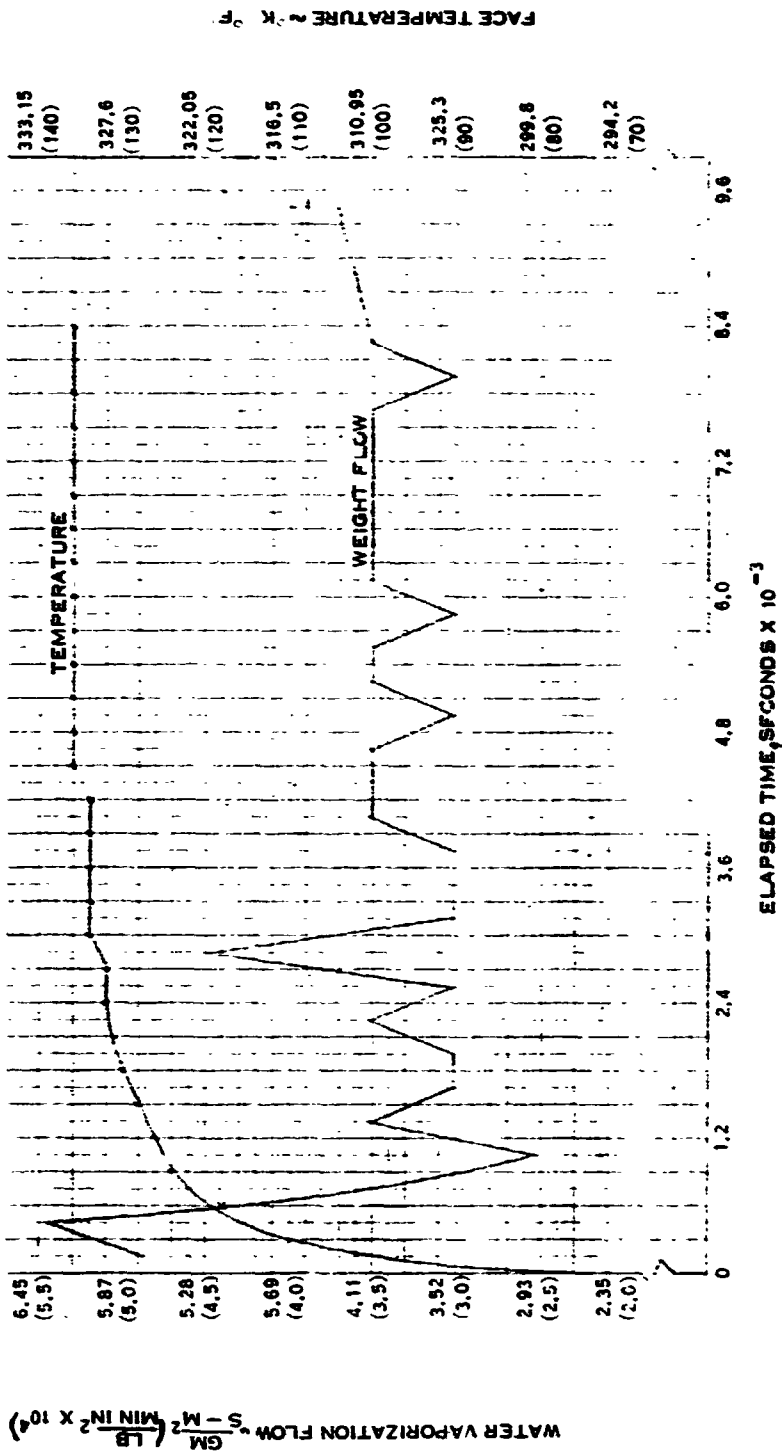
Test results are plotted in figure 16. The model cell was run on a weight scale with readings taken over specified time intervals to determine the amount of water evaporating. Temperature measurements were taken by a thermocouple at the cell model evaporating surface. The results indicate that for this application the wick can pass sufficient water flow to satisfy all heat loads required during emergency operation in the Ice Chest Heat Sink Subsystem. The results also indicate that 60% to 70% of the water in the ice chest can be wicked to the boiling surface before the wick becomes flow limited by the emptying of the large capillaries.

Ice Expansion Compensation Device

Each water cell in the ice chest heat sink has a layer of closed cell foam utilized as an ice expansion compensation device (reference figures 7 and 8). Prior to ice chest use as a heat sink, the unit is filled with water and frozen. Without an ice expansion compensation device, the volumetric expansion of the freezing water would either cause the unit to rupture or force water and ice into the vapor header passages, which could cause difficulties during emergency operation. Therefore, a compressible expansion device was needed to avoid these inherent difficulties. Table II shows a comparison of several candidate materials for the ice expansion compensation device.

TABLE II
COMPARISON OF ICE EXPANSION COMPENSATION DEVICE MATERIALS

Material	Construction	Comments
Polyethyl Impac (II)	Open Cell	Water can be forced through, stiffens below -55.35°F (-50°F).
Polyethyl Impac (III)	Open Cell	Water can be forced through, stiffens below -55.35°F (-50°F).
Neoprene SCE-42	Closed Cell	Strong skin, no contaminants.
Lockfoam E-304		Freon blown, may introduce contaminants.
Lockfoam A-200		Very rigid.
Lockfoam A-210		Heavy, rigid, possible water absorbent.

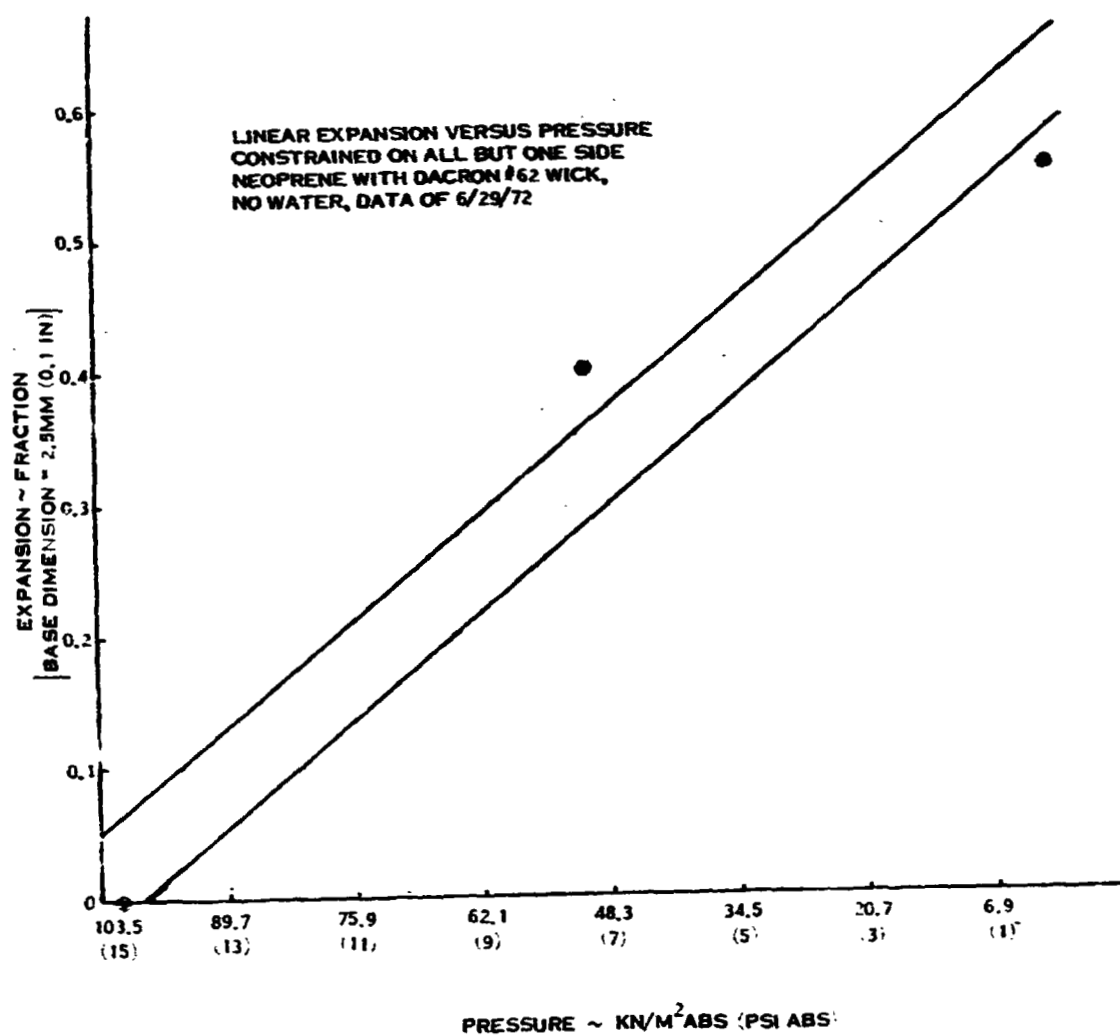


#54 DACRON WICKING TESTS

FIGURE 1

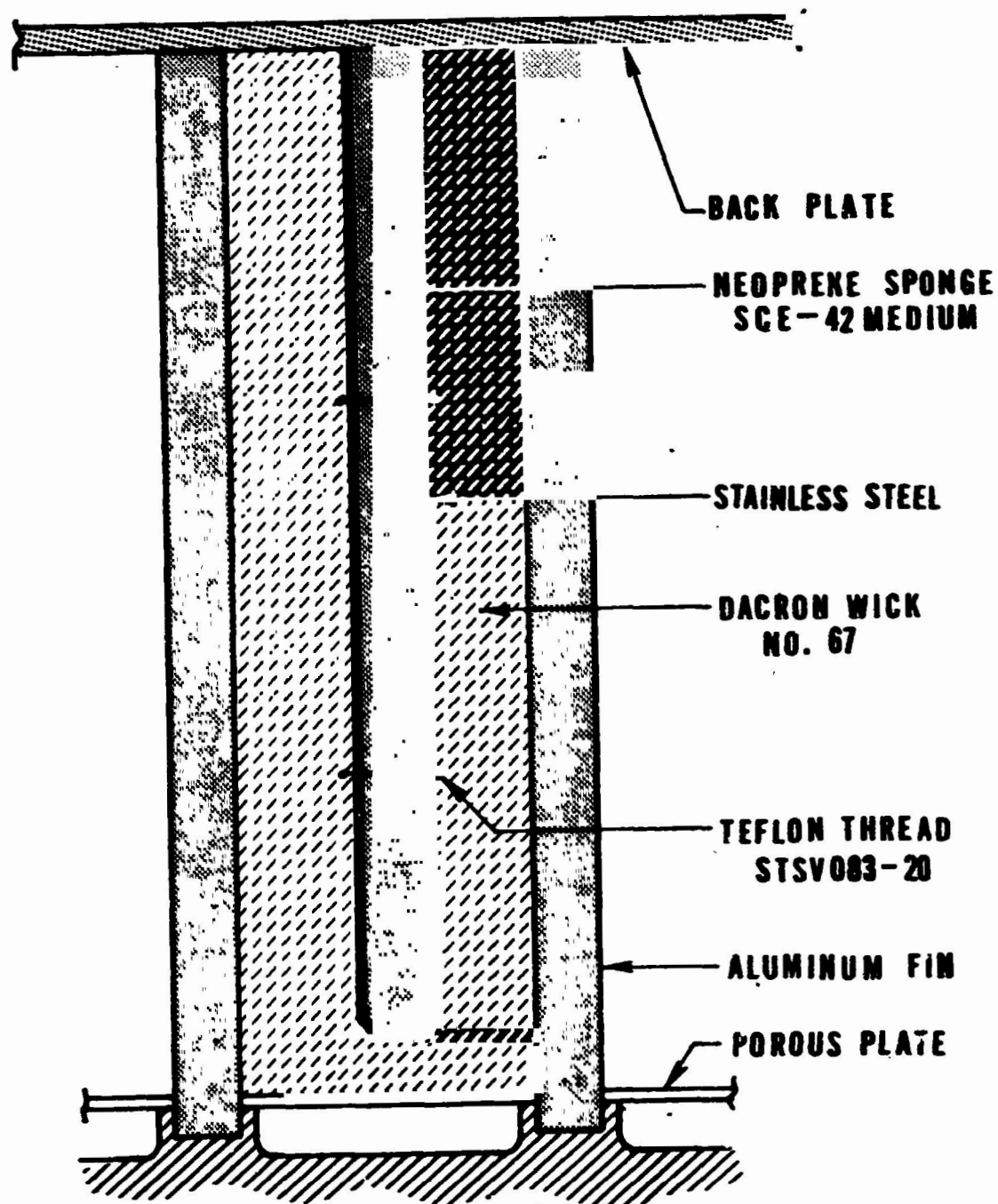
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ICE EXPANSION COMPENSATION DEVICE (PSIA)

FIGURE 17



SINGLE CELL STRUCTURE

FIGURE 18

Of the candidate materials, neoprene was chosen for its compressibility, closed cell, non-absorbent properties, and lack of potential contaminants. The neoprene was then checked for behavior in a vacuum environment.

The perimeter of a test piece of neoprene was constrained on all but one side, and the test piece was placed in a bell jar. Pressure was reduced from 101.3 kN/m² abs (14.7 psia) to 3.45 kN/m² abs (0.5 psia) and the expansion of the unconstrained end of neoprene was measured. Results are plotted in figure 17. From the figure, it can be seen that linear dimensions of the neoprene closed cell foam will increase 65% without rupture. If left unconstrained this closed cell foam will force liquid water through the perforated metal boiling surface when the vapor exhaust header is vented to vacuum conditions. This situation could then cause icing conditions in the vapor passages or solenoid shut-off valve. Therefore, the closed cell foam ice expansion compensation device is constrained against expansion.

Figure 18 gives a detailed schematic of the designed ice chest water cell, showing the constraint of the closed cell foam and the means of maintaining the wicking material against the metal boiling surface.

Ice Chest Sizing

The ice chest is designed to retain 7 kilograms (15.43 pounds) of water. The face area in contact with the LCG heat exchanger is 0.186 m² (2 ft²) to provide an overall thermal contact conductance of 211 J/s-°K (400 Btu/hr-°F). Basic water cell dimensions (reference figure 18) are 2.5 mm (0.10 inch) width for the ice expansion compensation device, 5.1 mm (0.20 inch) width on either side of the ice expansion compensation device for water filled wicking material, and 2.5 mm (.10 inch) wide aluminum fins on the outside of the wicking material.

An attempt was made to keep the ice chest - LCG heat exchanger interface area as square as possible to minimize deflection of the interface. Based on these requirements, the ice chest is composed of 28 cells (29 fins) on an overall dimension of 43 cm (16.9 inches). This requires a 43.3 cm (17.03 inch) height to provide the required 0.186 m² (2 ft²) surface area. Allowing 20% blockage for the wick, a cell height of 7.11 cm (2.8 inches) is needed to hold the required 7 kilograms (15.43 lbs) of water.

Overall Subsystem Performance for Normal Operation

Thermal performance of the ice pack during normal operation is represented by figures 19 through 21. These plot heat transfer rate from the LCG cooling water to the ice pack heat sink versus water flow rate through the heat sink heat exchanger, with inlet water temperature as a parameter. Each graph is for a different point in time when no ice is melted, half the ice is melted, and the end of the mission (all ice just melted). Only three temperatures are shown, but interpolation is linear. Application of these graphs may be illustrated by considering Figure 19. At the start of the mission with a 297.0°K (75°F) heat sink inlet temperature, an LCG cooling water flow rate of 5.66 g/s (45 lb/hr) through the heat sink is required for a heat load of 586 J/s (2000 Btu/hr). The rest of the 30.2 g/s (240 lb/hr) LCG cooling water is bypassed around the heat sink. Water temperature at the inlet of the LCG is:

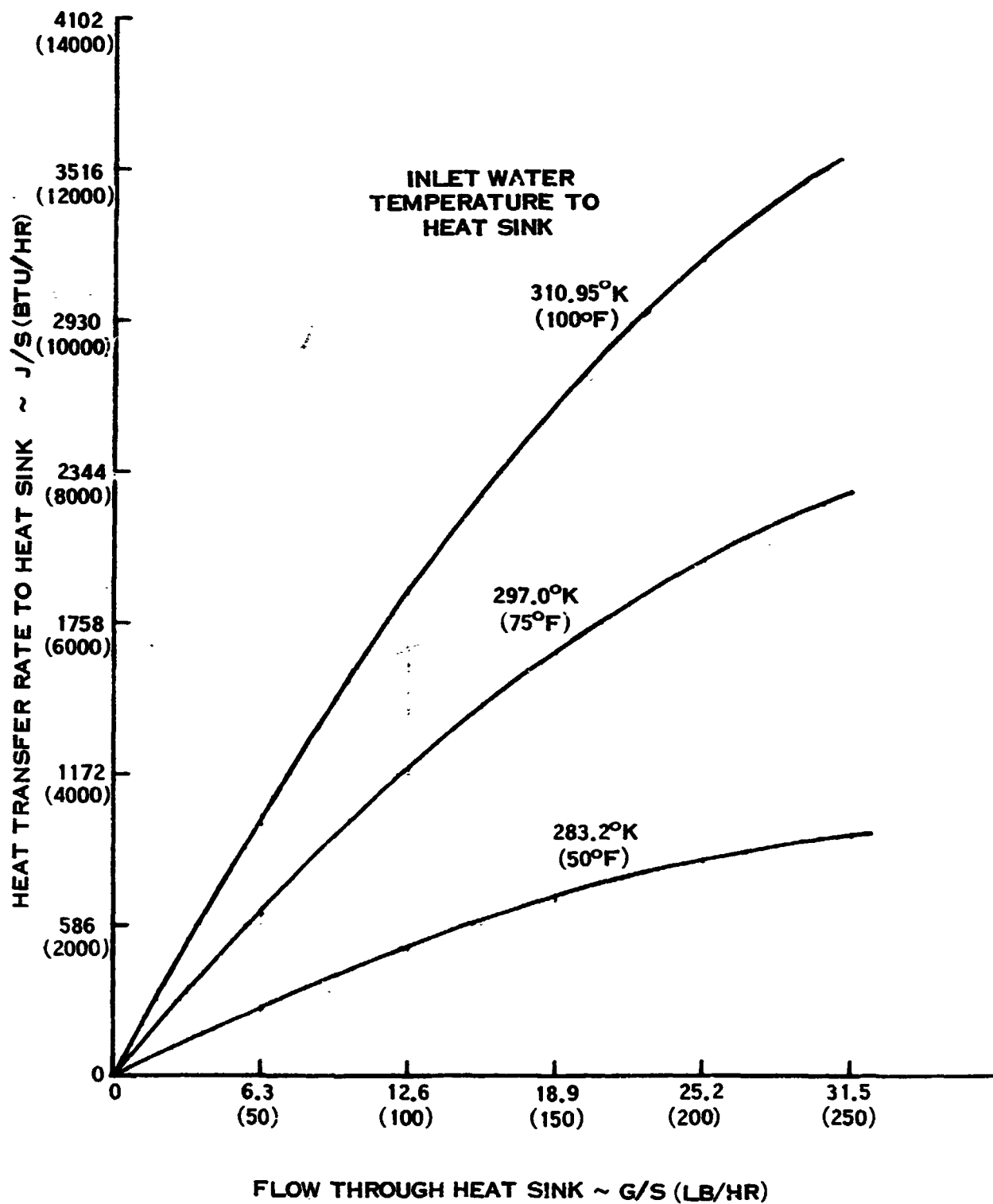
$$T - 0.238Q/30.2 = 297.0 - 0.238 \times 586/30.2 = 292.5^\circ\text{K}$$

$$(T - Q/240 = 75 - 2000/240 = 66.7^\circ\text{F})$$

LCG heat exchanger core pressure drops were also calculated for LCG water inlet temperatures of 283.2°K - 305.3°K (50°F - 90°F) over a flow range of 0 - 30.2 g/s (0 - 240 lb/hr). The LCG heat exchanger is aluminum, 43 cm (16.9 inches) by 43.3 cm (17.03 inches) single passage, 6 pass, with ruffled aluminum fins. The fins are 1.9 mm (0.075 inches) high, 0.076 mm (0.003 inches) thick, with 9.5 fins per cm (24 fins per inch). Core pressure drops were calculated based on Hamilton Standard test data for this fin and are presented in figure 22.

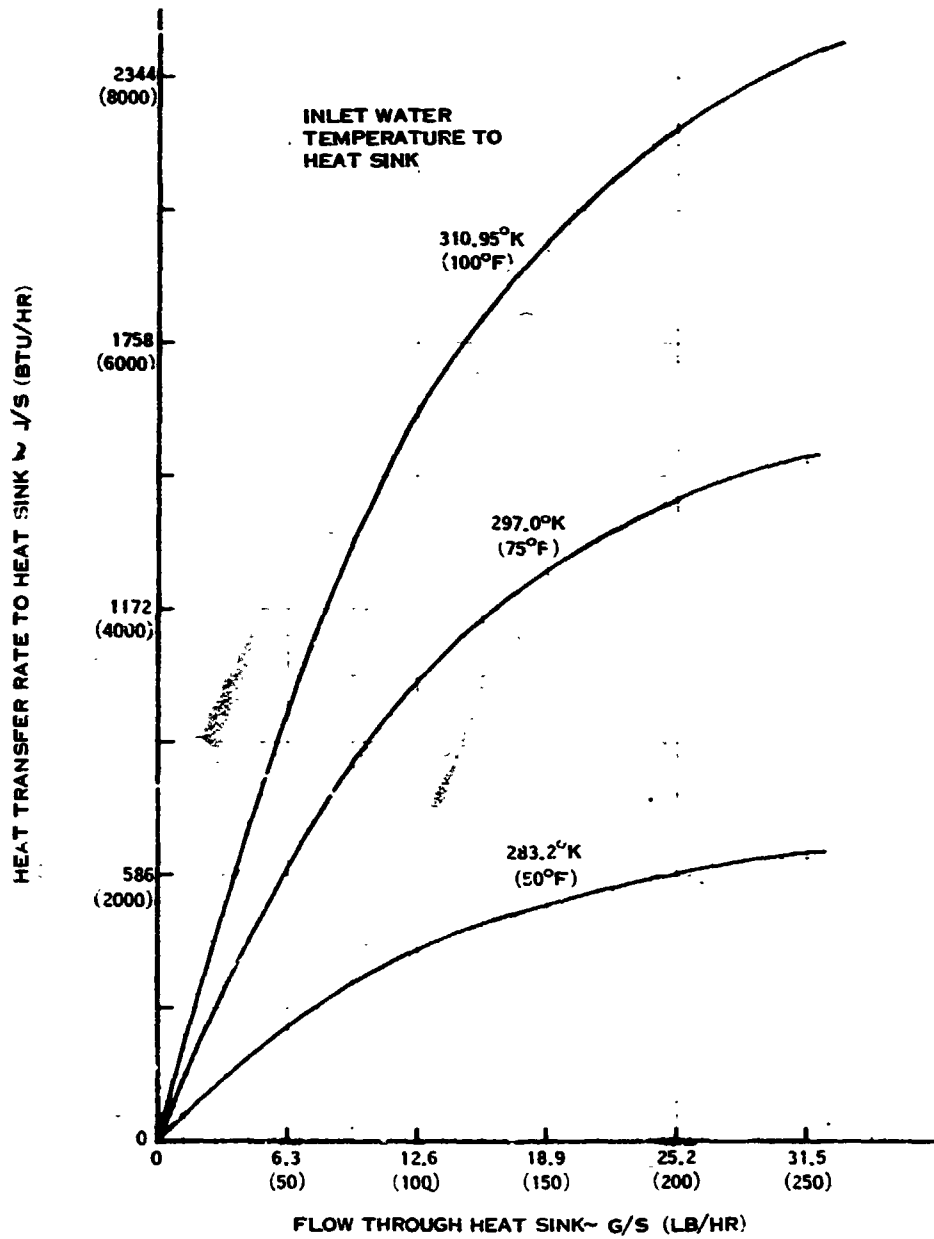
Emergency Operation

An emergency operation is considered to exist during EVA when all the ice in the ice chest has melted. A thermostwitch at the LCG outlet of the heat exchanger activates a vacuum shut-off valve which vents the ice chest vapor passages to ambient vacuum. The reduction in pressure causes water to boil at the metal boiling surface. LCG thermal conditioning is thus maintained by transforming the ice chest into a water boiler.



HEAT SINK PERFORMANCE WITH NO ICE MELTED

FIGURE 19

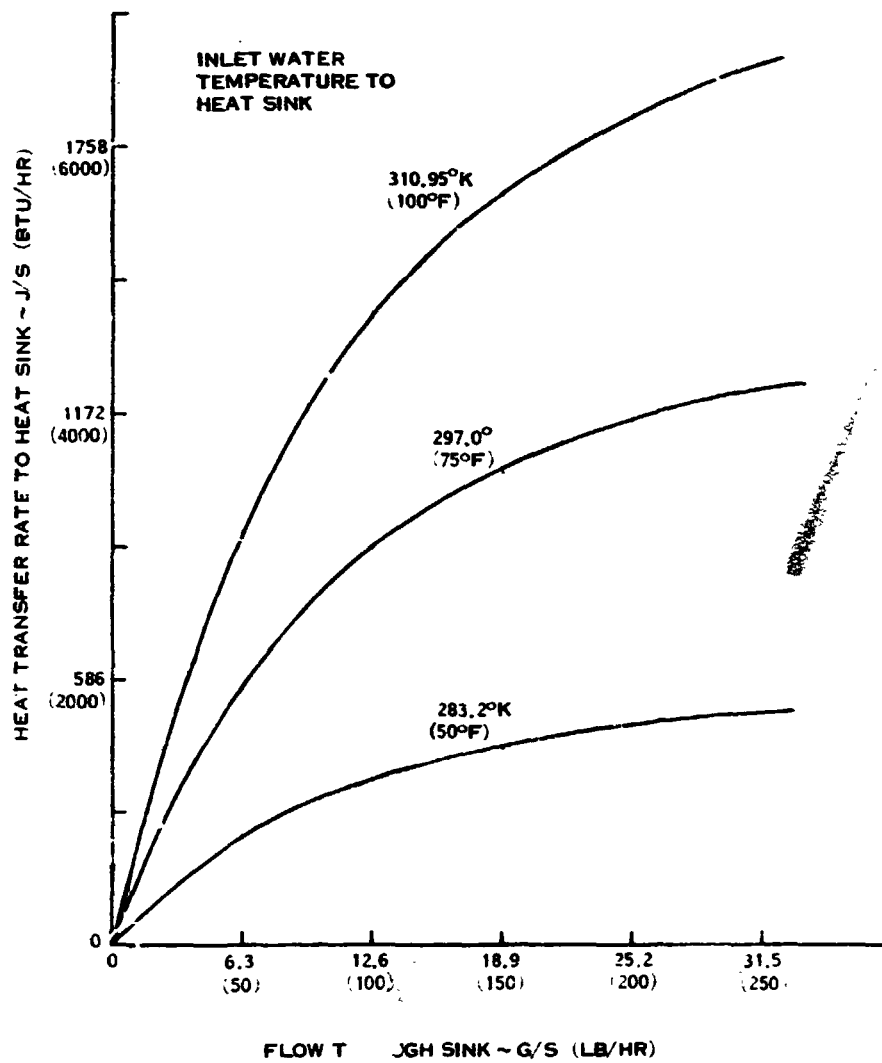


HEAT SINK PERFORMANCE WITH ICE HALF MELTED

FIGURE 20

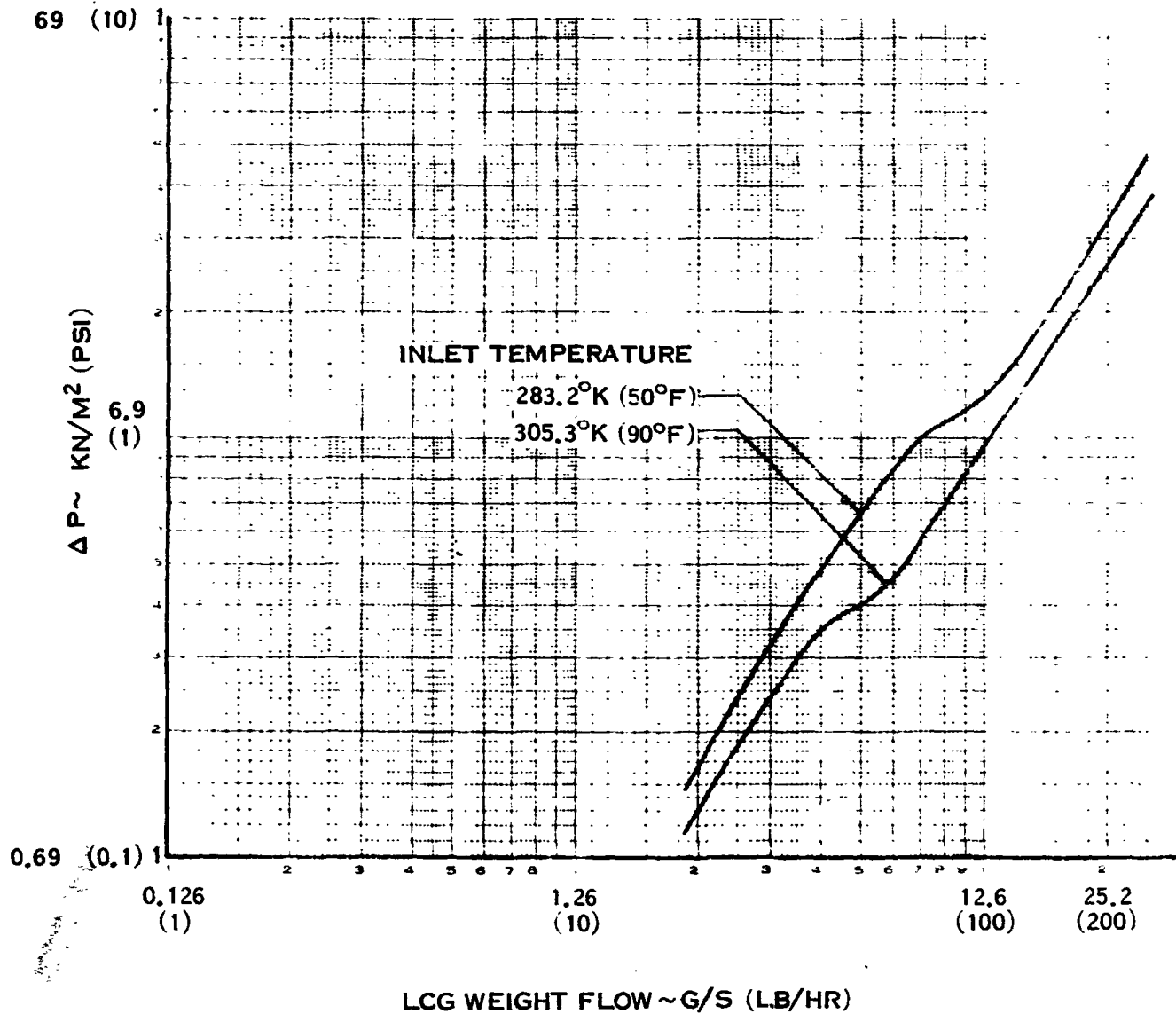
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HEAT SINK PERFORMANCE WITH ICE ALL MELTED

FIGURE 21



ESTIMATED PRESSURE DROP LCG HEAT EXCHANGER
ICE PACK HEAT SINK SUBSYSTEM

FIGURE 22

The available heat sink for emergency operation depends on the water wicked to the metal boiling surface and the heat of vaporization at the evaporant temperature of the metal boiling surface. The evaporant temperature is governed by the pressure at the metal boiling surface which is controlled by the pressure drop through the vapor header passages and across the flow limiting orifice. The overall effectiveness of the emergency mode unit can then be determined by an analysis of the conductive heat transfer path from the metal boiling surface to the LCG water.

The metal boiling surface is designed to provide a proper medium for water boiling during emergency operation. High wicking capillary head and constraint of the Neoprene ice expansion compensation device eliminated metal boiling surface water retention capabilities as a design criteria. The material chosen was aluminum, which provides for similar material throughout the ice chest, thus facilitating the brazing assembly procedure. The metal boiling surface is 1.57 mm (0.062 inch) thick with approximately 11% open area, which can be readily purchased from any of several vendors.

The range of open areas (10 - 13%) was picked to maintain high thermal conductivity through the metal boiling surface. Since the surface is alodine 1200 treated aluminum, which is a wetting surface, the metal boiling surface is assumed to be hydrophilic, i.e., the water is retained in the metal boiling surface at the downstream end of the open area. This assumption eliminates pressure drop calculations across the metal boiling surface. Furthermore, the boiling surface conductivity can then be calculated by treating the conductive heat transfer path as a parallel arrangement of aluminum and water. Therefore,

$$K_{EFF} = (1-P) K_M + PK_L$$

where K_{EFF} is the thermal conductivity of the parallel aluminum/water arrangement, K_M is the thermal conductivity of the aluminum, K_L is the thermal conductivity of water, and P is the fraction of plate open area divided by the plate total area. By holding metal boiling surface area fraction P to a minimum, the effective thermal conductivity K_{EFF} is held high, since $K_M > K_L$. For any given heat load Q , the temperature drop ΔT can be held to a minimum since

$$Q = \frac{KA}{x} \Delta T$$

where A is the cross sectional area of the heat transfer path and x is the length of the path.

For this unit, the effective thermal conductivity is $645 \text{ J/s} \cdot \text{m}^2 \cdot ^\circ\text{K}$ ($113.9 \text{ Btu/hr} \cdot \text{ft} \cdot ^\circ\text{F}$) with 11 percent open area.

To accurately predict the evaporant temperature at the metal boiling surface, it is necessary to have pressure drop characteristics as a function of weight flow, since evaporant temperature is dependent on the pressure at the metal boiling surface. Pressure drops during emergency operation consist of vapor passage pressure drops and orifice pressure drops.

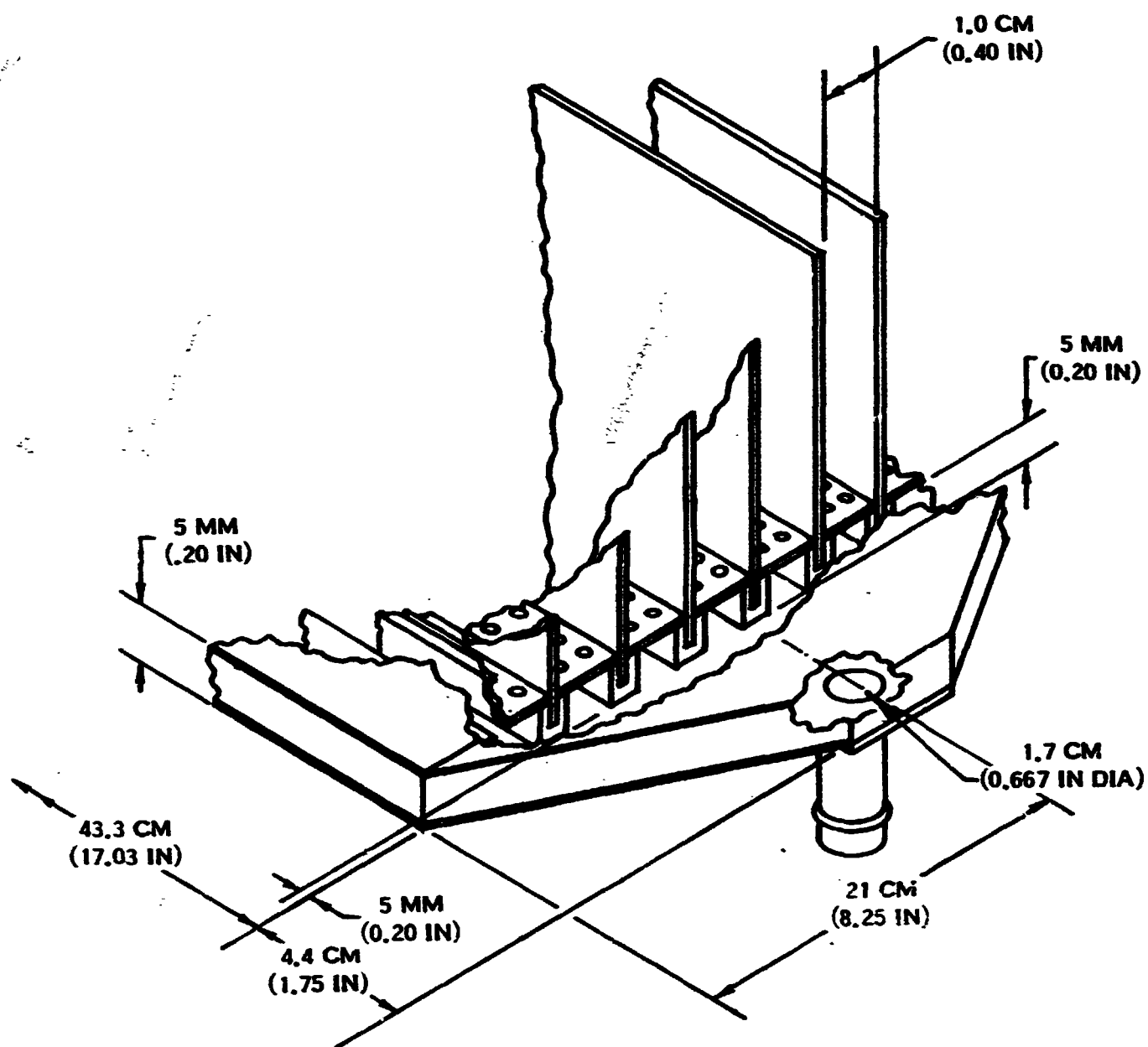
Figure 23 presents a schematic of the vapor passages. The total pressure drop is subdivided into four parts.

- Frictional pressure drop in the cell.
- Pressure drop for a square mitered elbow bend of 79.2° based on the flow in the outside cell.
- Frictional pressure drop in the header.
- Pressure drop due to contraction in the exhaust port.

Figure 24 presents the vapor passage pressure drop curve as a function of evaporant water flow. Appendix A presents sample calculations for this procedure.

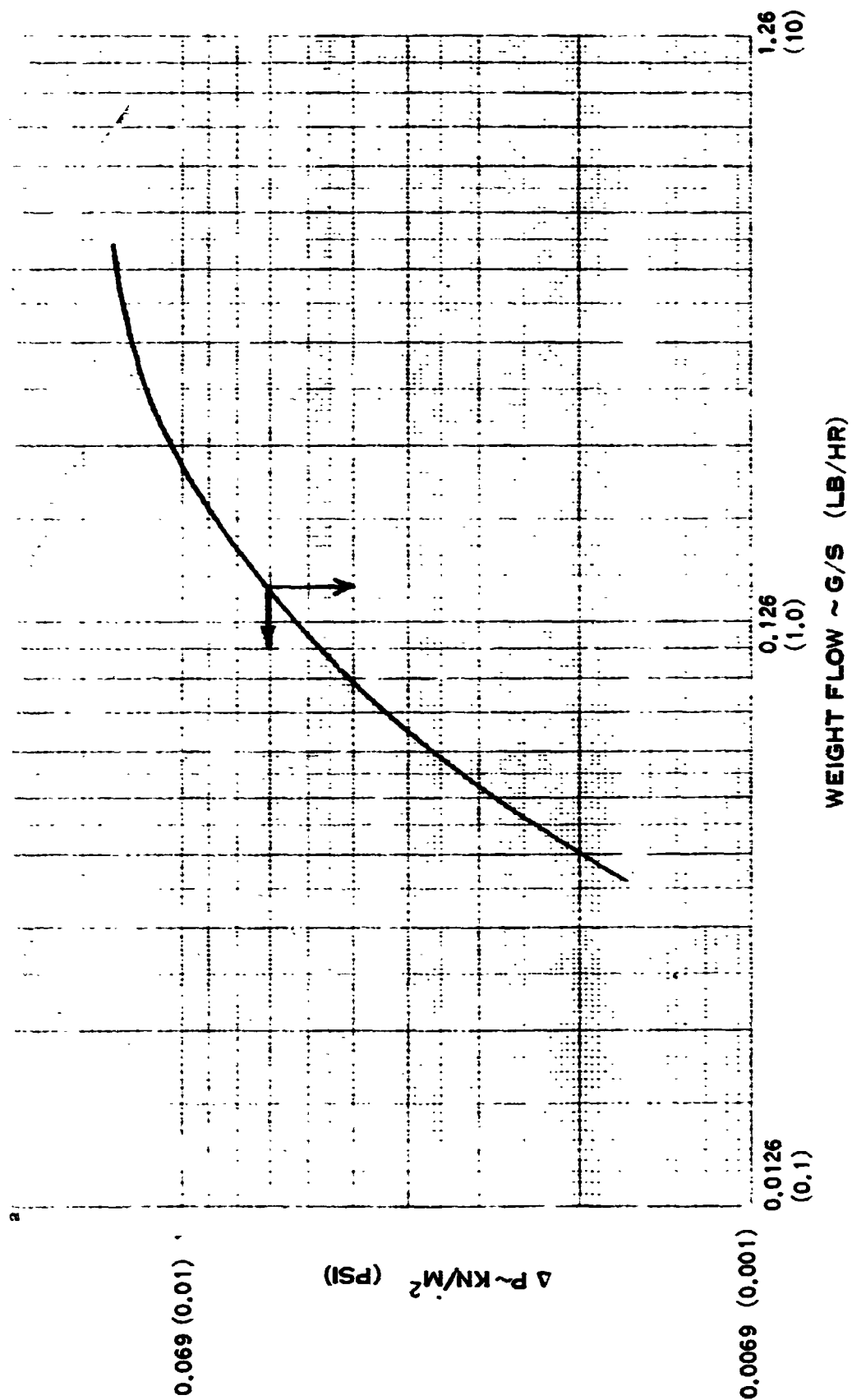
An orifice is placed downstream of the vapor passage exhaust port to control the overboard flow of water vapor. For any orifice size, there is a point of freeze-up. This is caused by the vapor pressure falling below 0.61 kN/m^2 absolute (0.0885 psia), which is equivalent to a saturation temperature of 273.1°K (32°F). At this point, ice is formed in the orifice and the minimum heat load has been reached. At the other end of the spectrum, there is also a maximum cooling load that the orifice can accommodate. This is a result of higher pressure drops caused by an increased flow requirement at higher heat loads.

As the pressure at the metal boiling surface increases due to the increased pressure drop, the evaporant temperature at the metal boiling surface also increases. Eventually, the point is reached where the higher evaporant temperatures can no longer satisfy the LCC flow/temperature requirements.



ICE CHEST VAPOR PASSAGE CONFIGURATION

FIGURE 23



PRESSURE DROP VERSUS WEIGHT FLOW FROM POROUS PLATE
TO HEADER OUTLET ~ EMERGENCY MODE

FIGURE 24

For this application, two orifices are employed. The first orifice diameter of 7.9 mm (0.311 inch) will freeze-up for a heat load of 117.2 J/s (400 Btu/hr) and provide effective operation up to 411 J/s (1400 Btu/hr). Allowing some overlap, the second orifice diameter of 15.0 mm (0.590 inch) is designed to freeze at 366 J/s (1250 Btu/hr) and to provide required cooling for the unit under a 879 J/s (3000 Btu/hr) heat load.

A sample of orifice calculations is included in Appendix B.

The following comments on the exhaust tube design resulted from an analysis:

- The line between the ice chest outlet and the orifice, because of its size and its sharp edged inlet, does not provide a flow pattern which enables accurate prediction of orifice size required for a given flow rate and upstream pressure. A rounded inlet, larger diameter tube, and possibly greater length are required to establish the fully developed flow upstream of the orifice required for accurate prediction.
- The high Mach numbers in the line upstream of the orifice, particularly in the vena contracta at the inlet, increase the probability of vapor condensation shocks occurring in the exit line. Relatively high Mach numbers may also be present downstream of the orifice at high heat loads. Vapor condensation shocks will result in depositing liquid in the exhaust line which will subsequently freeze. An external liquid trap is required to drain off condensate.
- The calculated pressure drop in the exhaust line was sufficiently marginal with respect to requirements that manufacturing tolerances or distortion of the line due to handling could cause an increased pressure drop, which would back-pressure the boiler causing an increase in evaporant temperature. Careful establishment of tolerances and relatively non-flexible lines have been incorporated.

Heat exchanger performance was calculated by an analysis of the conductive heat transfer path from the LCG cooling water to the boiling vapor. This path included the aforementioned contact conductance and the pure metal mass of the ice chest end plate. The conductivity of the metal boiling surface is:

$$K_{EFF} = (1-P) K_M + PK_L$$

where P is the fraction of plate open area to total area, K_M is the thermal conductivity of the metal, K_L is the thermal conductivity of the water, and K_{EFF} is the overall thermal conductivity of the metal boiling surface. The rate of heat transfer down the metal boiling surface is therefore described by

$$q = K_{EFF} \cdot A \cdot \Delta T/x$$

where A is the cross sectional area of the metal boiling surface, x is the distance from the edge of the metal boiling surface to the ice expansion compensation device, and ΔT is the temperature drop along the plate.

For any heat transfer rate q, the temperature drop attributable to each of these conductances can be calculated. The total temperature drop can then be used to calculate an overall conductance UA by use of the relationship

$$\frac{1}{UA} = \frac{\Delta T}{q}$$

where ΔT is the total temperature drop from the LCG cooling water to the evaporant surface. This overall conductance is assumed to be a constant over the range of heat loads encountered during emergency operation. Overall effectiveness during emergency operation was determined based on counter flow theory and is presented in Figure 25.

Overall Subsystem Performance for Emergency Operation

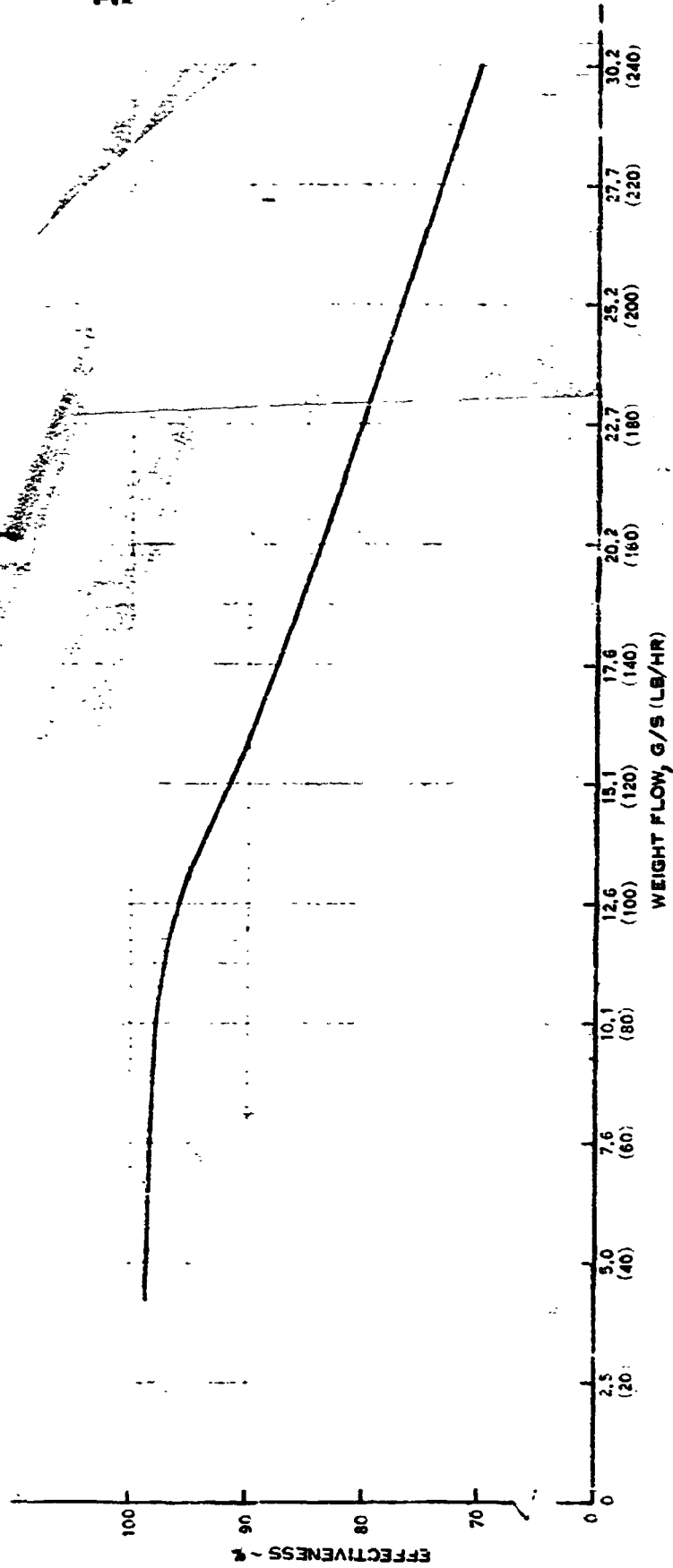
The overall heat transfer rate during emergency operation is defined by

$$Q = W c (T_i - T_o)$$

where T_o is water outlet temperature, T_i is water inlet temperature, W is the water flow rate, and c is the water specific heat. The effectiveness is used to determine water outlet temperature for various water inlet temperatures

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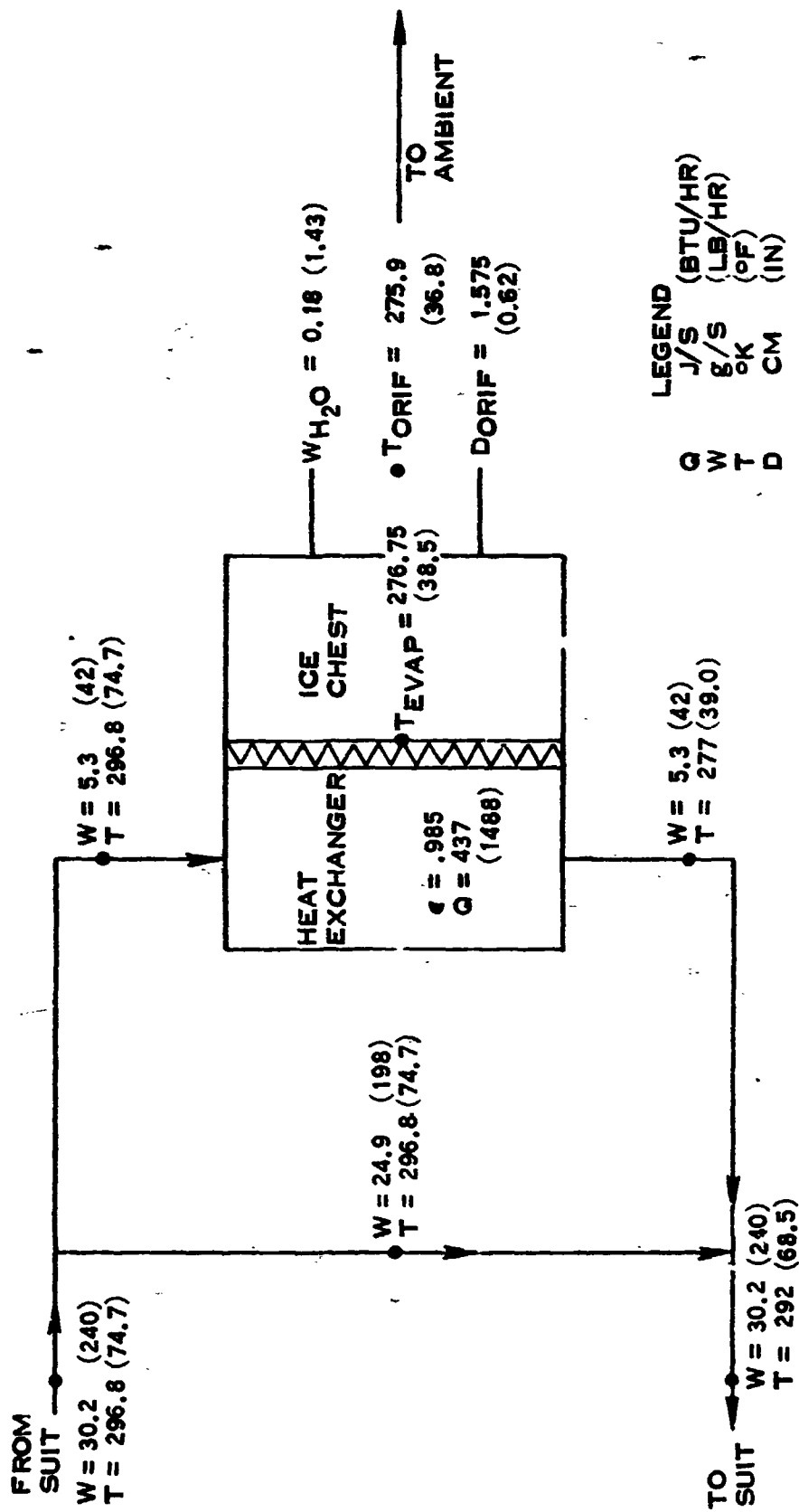
EMERGENCY OPERATION PREDICTED LCG HEAT EXCHANGER EFFECTIVENESS
VERSUS LCG WATER FLOW

FIGURE 25

$$T_o = T_i - E (T_i - T_{EVAP})$$

where E is effectiveness and T_{EVAP} is the evaporant temperature of the metal boiling surface.

Ice Pack Heat Sink Subsystem performance during emergency operation with each of the orifices is presented by figures 26 through 30. Appendix B contains sample calculations for these conditions.



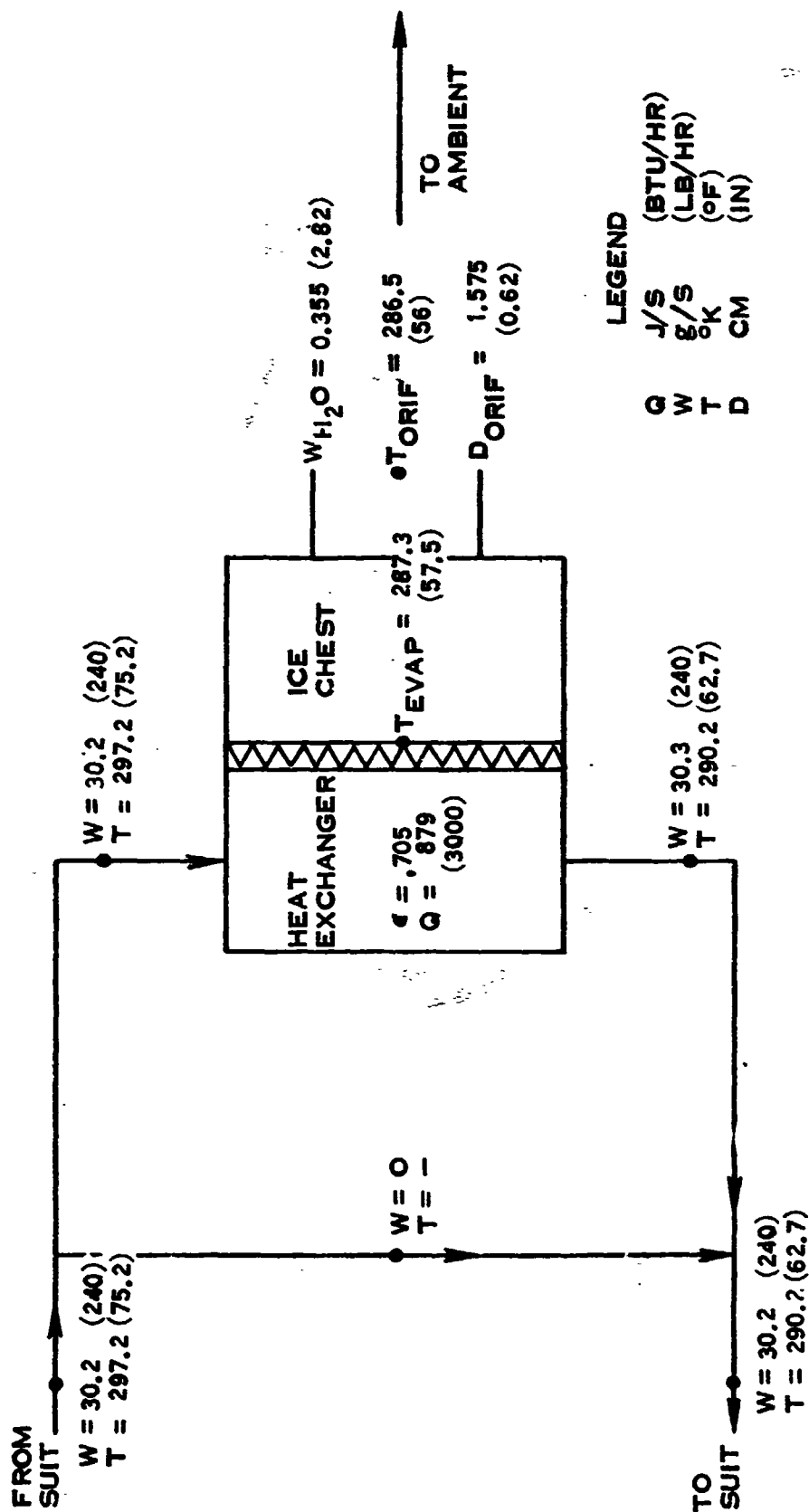
LEGEND

Q (BTU/HR)
J/S (LB/HR)
W (°F)
T (°K)
D (IN)

CONDITION Q=437 (1488)

EMERGENCY OPERATION ICE PACK HEAT SINK SUBSYSTEM

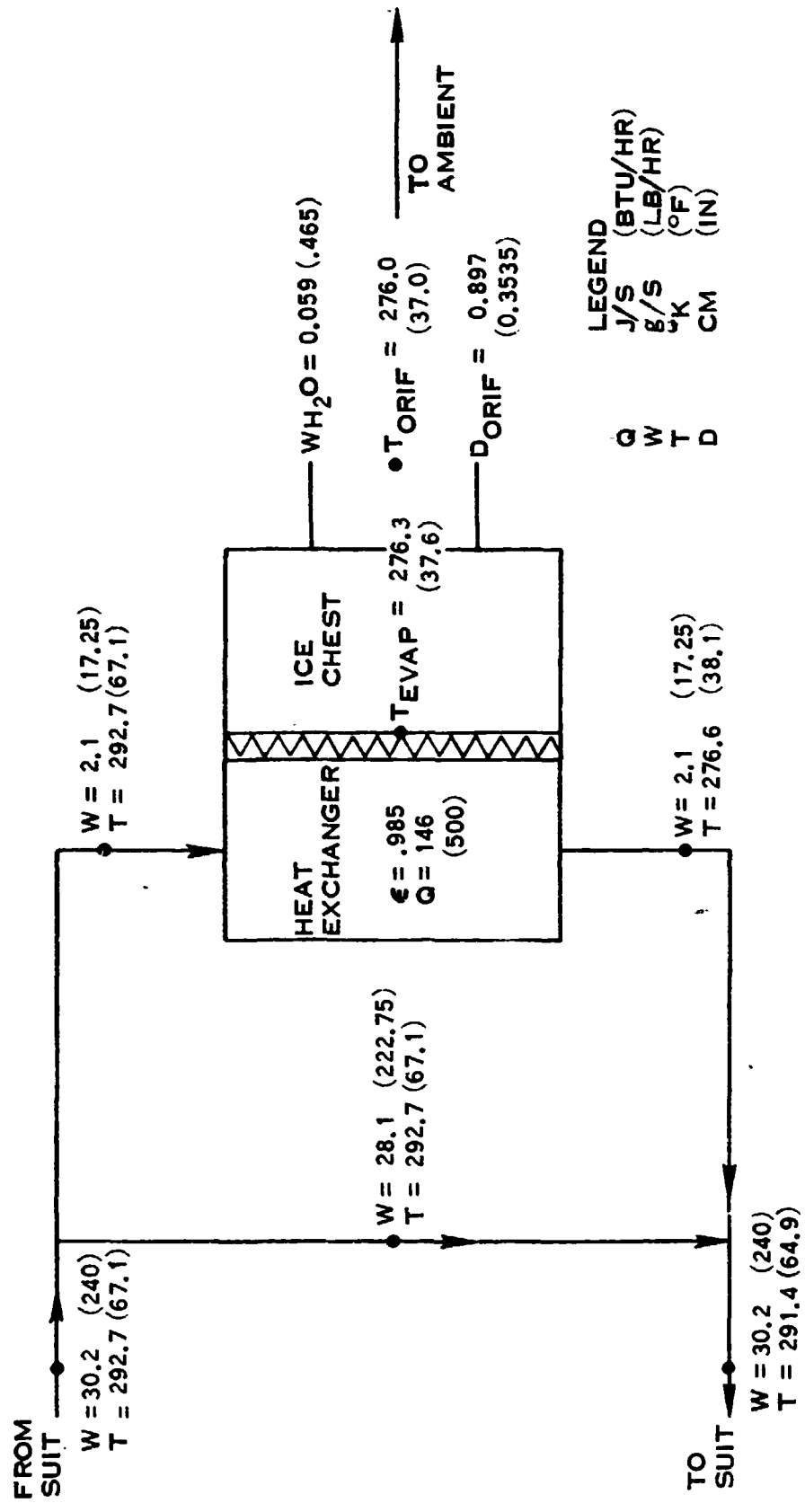
FIGURE 26



CONDITION $Q = 879 \text{ (3000)}$

EMERGENCY OPERATION ICE PACK HEAT SINK SUBSYSTEM

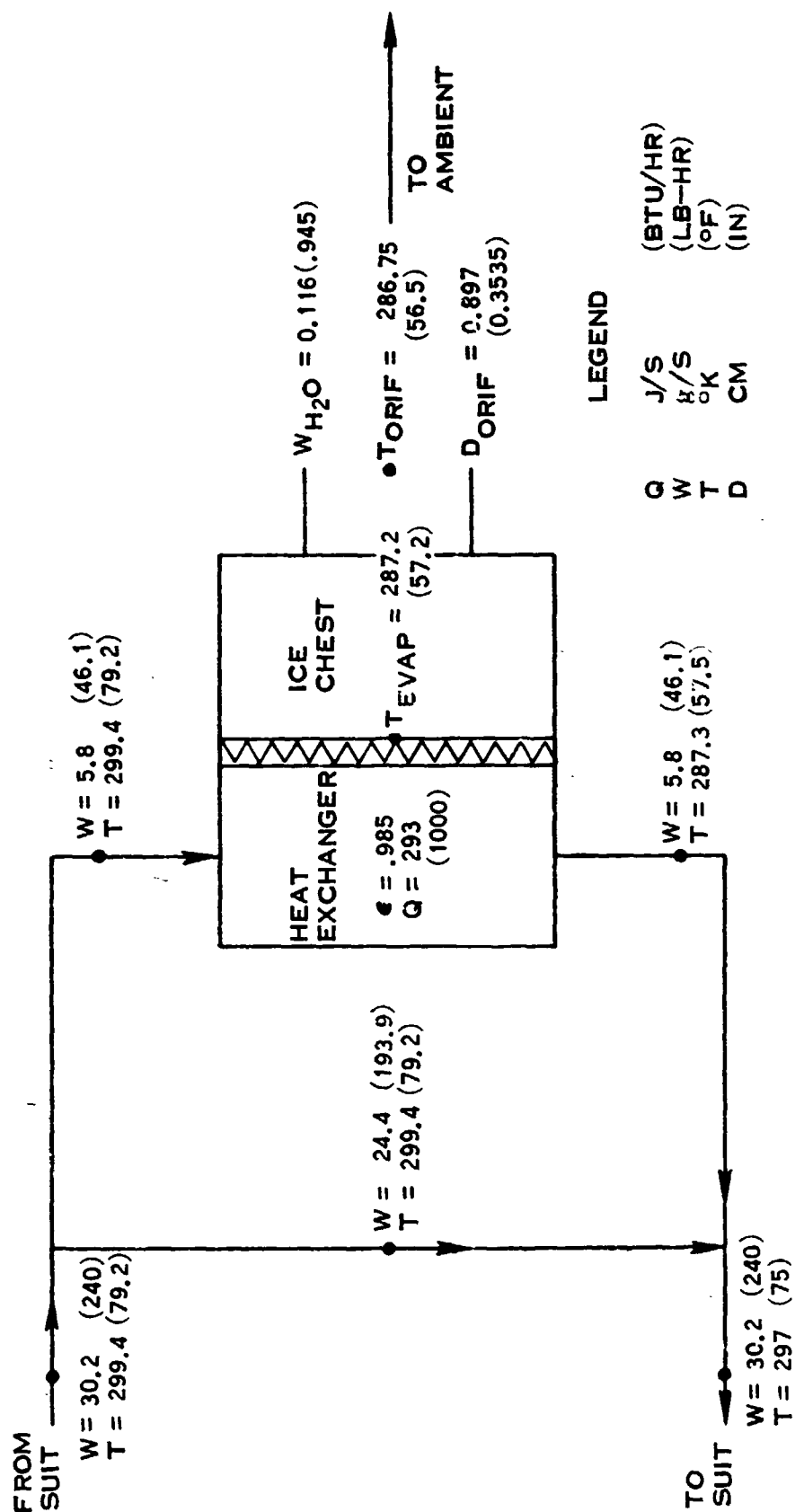
FIGURE 27



CONDITION $Q = 146$ (500)

EMERGENCY OPERATION ICE PACK HEAT SINK SUBSYSTEM

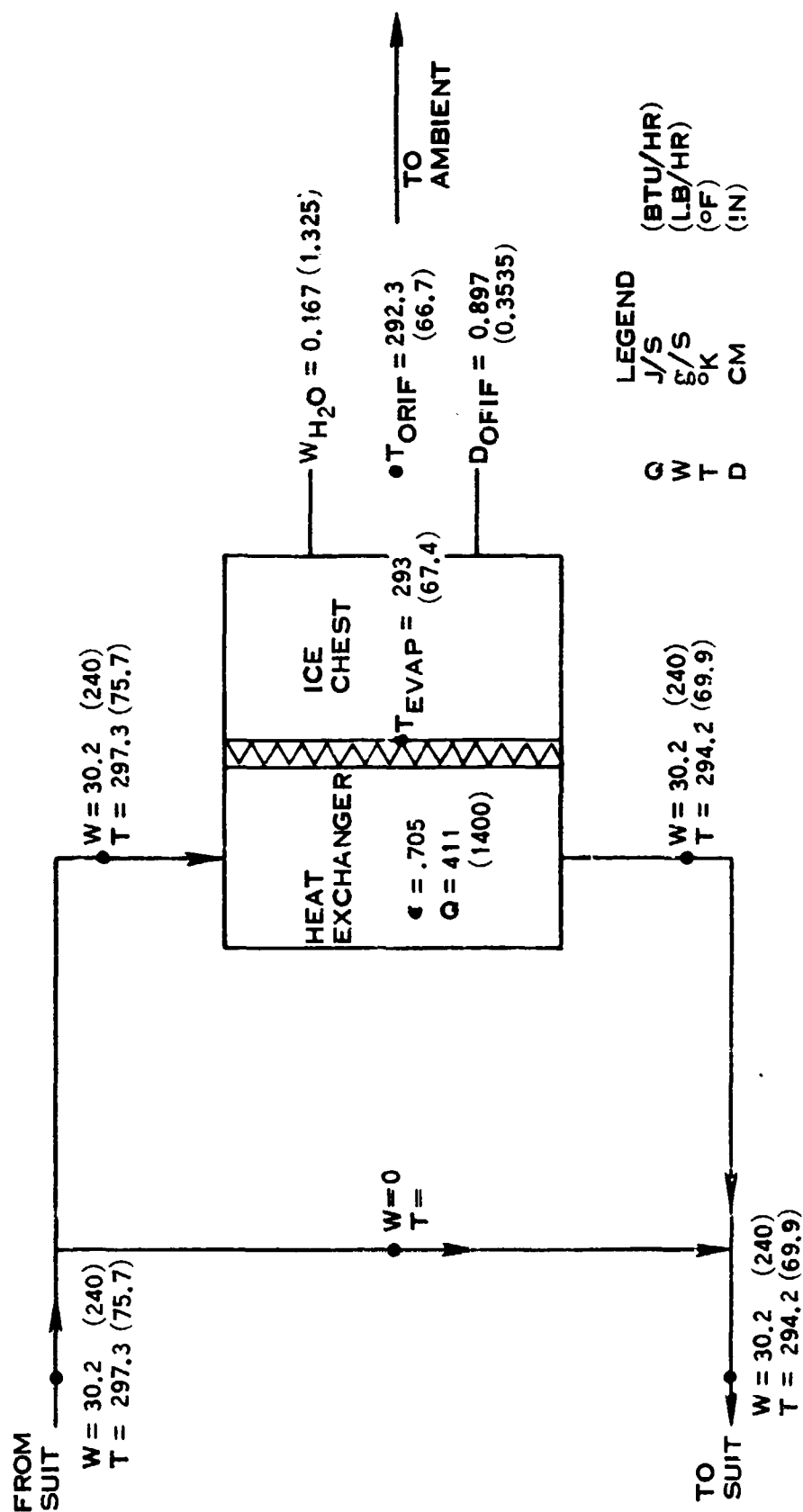
FIGURE 28



CONDITION $Q=293$ (1000)

EMERGENCY OPERATION ICE PACK HEAT SINK SUBSYSTEM

FIGURE 29



CONDITION $Q = 411$ (1400)

EMERGENCY OPERATION ICE PACK HEAT SINK SUBSYSTEM

FIGURE 30

DESIGN FEASIBILITY TESTING

The purpose of conducting feasibility testing was to obtain data of actual performance, to be correlated with analytically predicted performance. This performance correlation verified design feasibility and allowed more precise performance calculations to be made for the laboratory demonstration model hardware. A design feasibility hardware test plan was written to direct the feasibility hardware testing and is included in Appendix C.

The feasibility hardware assembly task consisted of loading the ice chest with the expansion compensation sponge rubber and stainless steel containment plates, and packing in the water containment wick material.

Instrumentation was added to the ice chest to obtain temperature readings for various areas within the ice chest and pressure readings for various positions within the ice chest vapor passage. Ice pack thermocouple locations are identified in figure 31. Pressure probes were located at the right rear, right front, center, and outlet portions of the vapor passages.

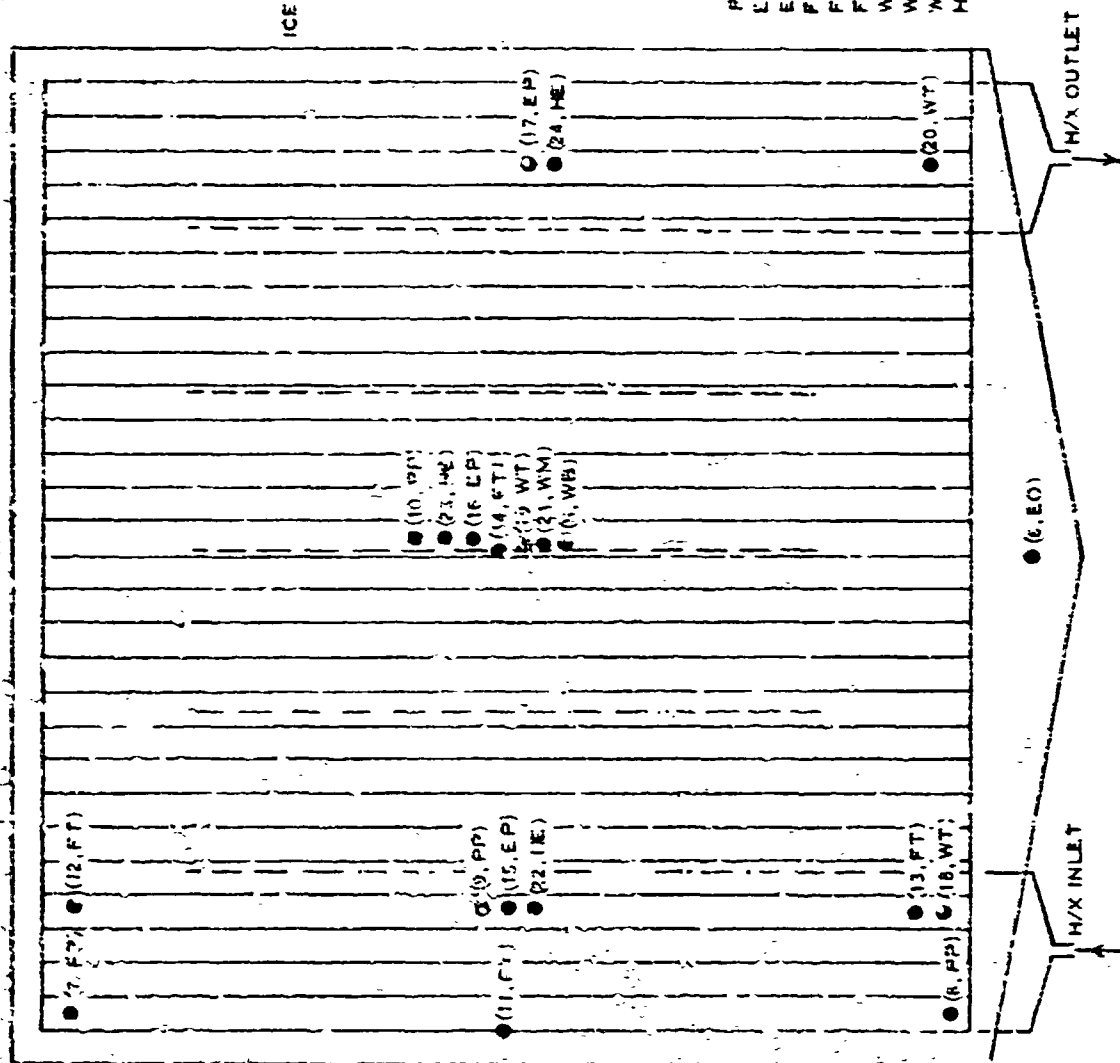
The cover and cover seal were bolted in place and the assembly was pressurized to ensure pressure integrity. Slight pressure loss was encountered along the thermocouple wires but this leakage was judged to be acceptable for the feasibility testing. When hard vacuum testing was required to prove out the interface surface heat conductance the twenty unwelded thermocouples were replaced with three welded (sealed) thermocouples. The assembled ice chest, including the optimum wicking material, was vacuum loaded with water and the water containment capacity was verified at 7000 ml minimum.

The design and assembly of the design feasibility hardware test rig was completed in accordance with the Design Feasibility Test Plan. The Portable Flow Console was assembled in the Space System Department (SSD) Advanced Engineering Laboratory, per figure 32, and checked out in ambient utilizing the Liquid Cooling Garment (LCG) heat exchanger and the ice chest for a series of ice melting runs. After the operation of the Portable Flow Console had been proven out in ambient, the console was moved to the Space Laboratory Rig 25 vacuum chamber, reassembled to the LCG heat exchanger and checked out. Figure 33 schematically illustrates the Feasibility Hardware Vacuum Test Rig. Figure 34 is a photograph of the Portable Flow Console with key components identified and figure 35 is a photograph of Rig 25 vacuum chamber with the ice chest and LCG heat exchanger installed.

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ICE CHEST PLAN VIEW



LOCATION
PP PERFORATED PLATE
EP EXHAUST PASSAGE
EO EXHAUST OUTLET
FT FIN TOP
FM FIN MIDDLE
FB FIN BOTTOM
WT WICK TOP
WM WICK MIDDLE
WB WICK BOTTOM
HE HX EXTERIOR

ICE CHEST AND HEAT EXCHANGER THERMOCOUPLES
LOCATIONS

FIGURE 11

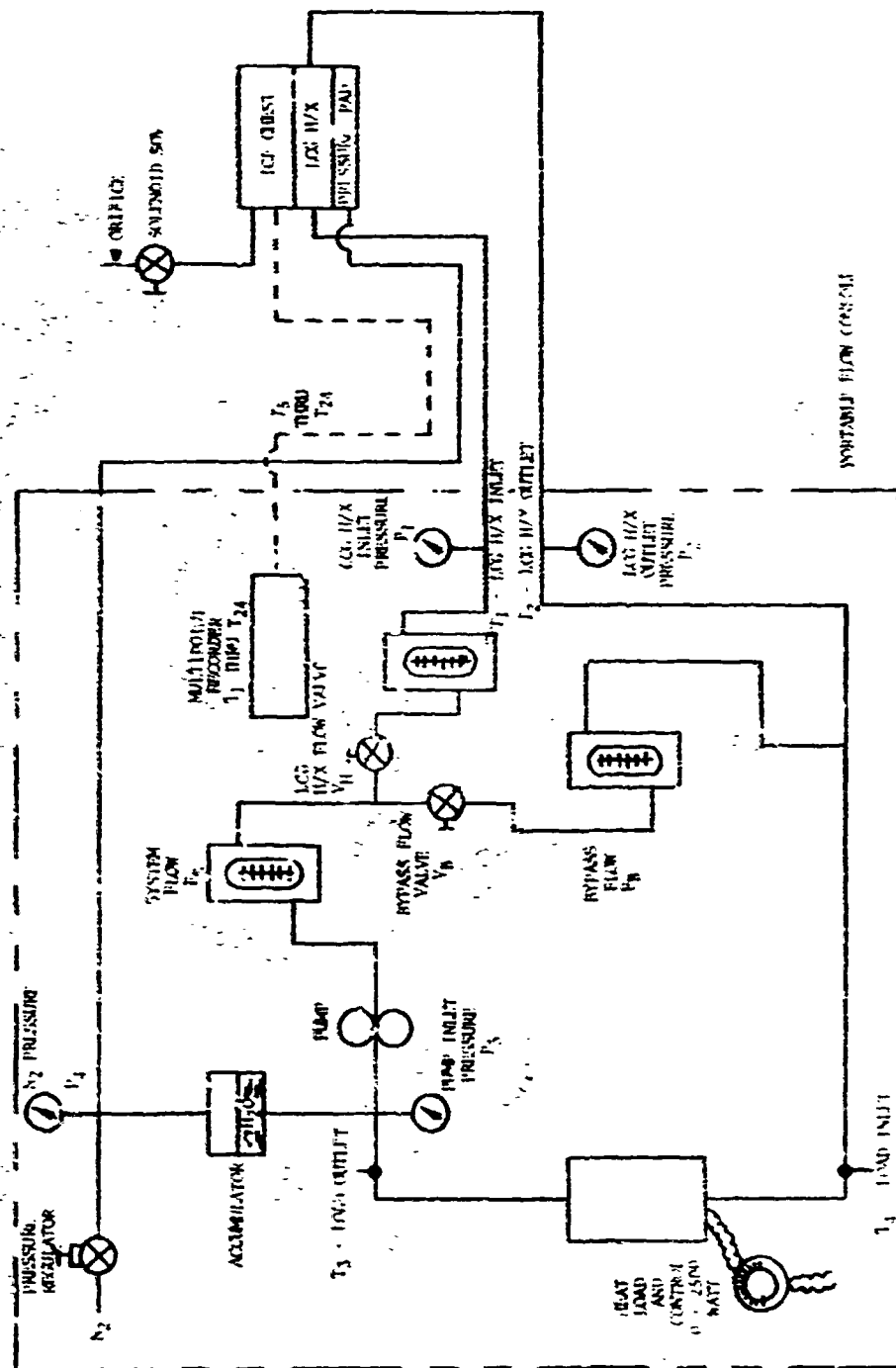
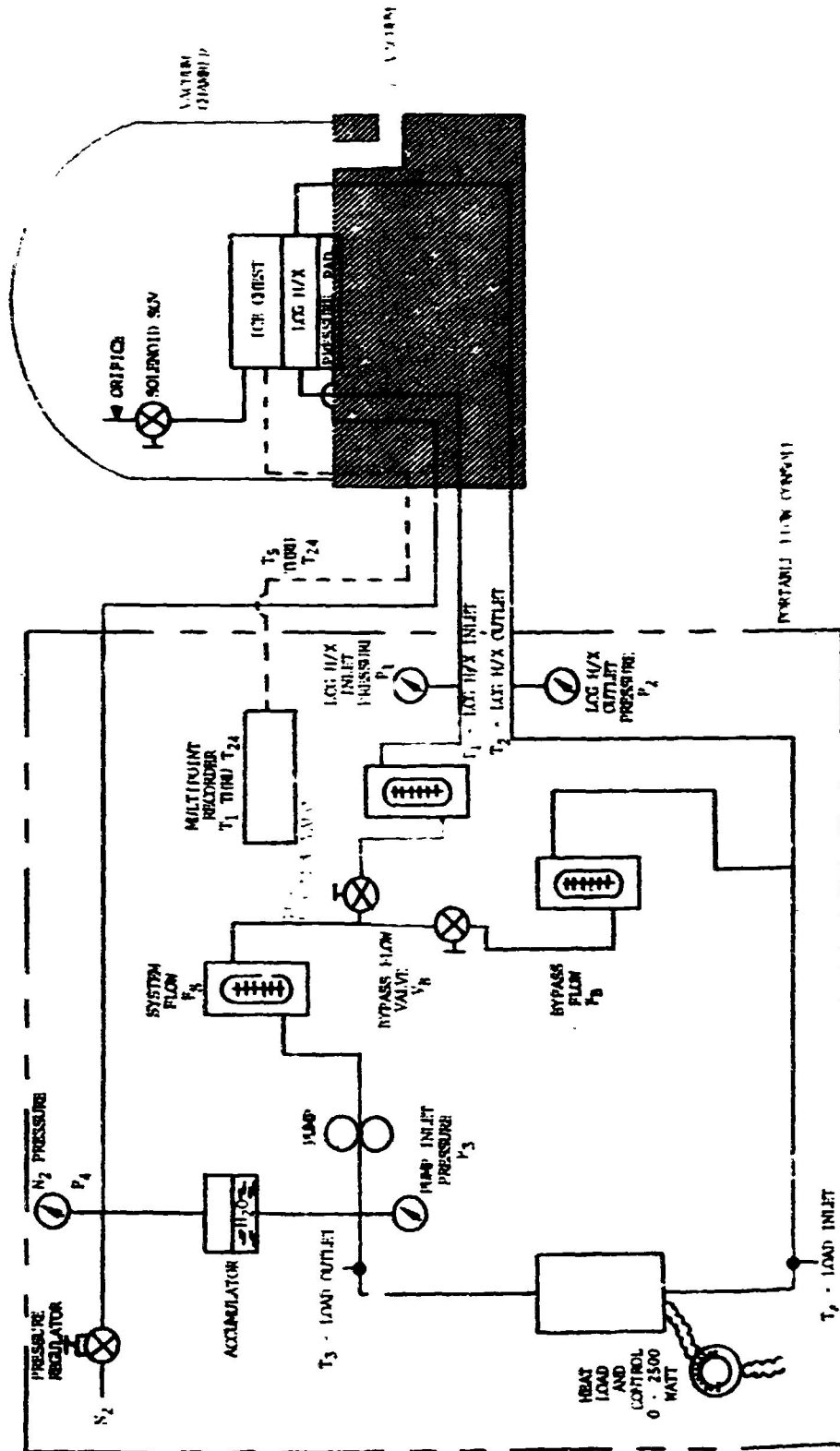


FIGURE 1. TEST OF FLOW CONTROL

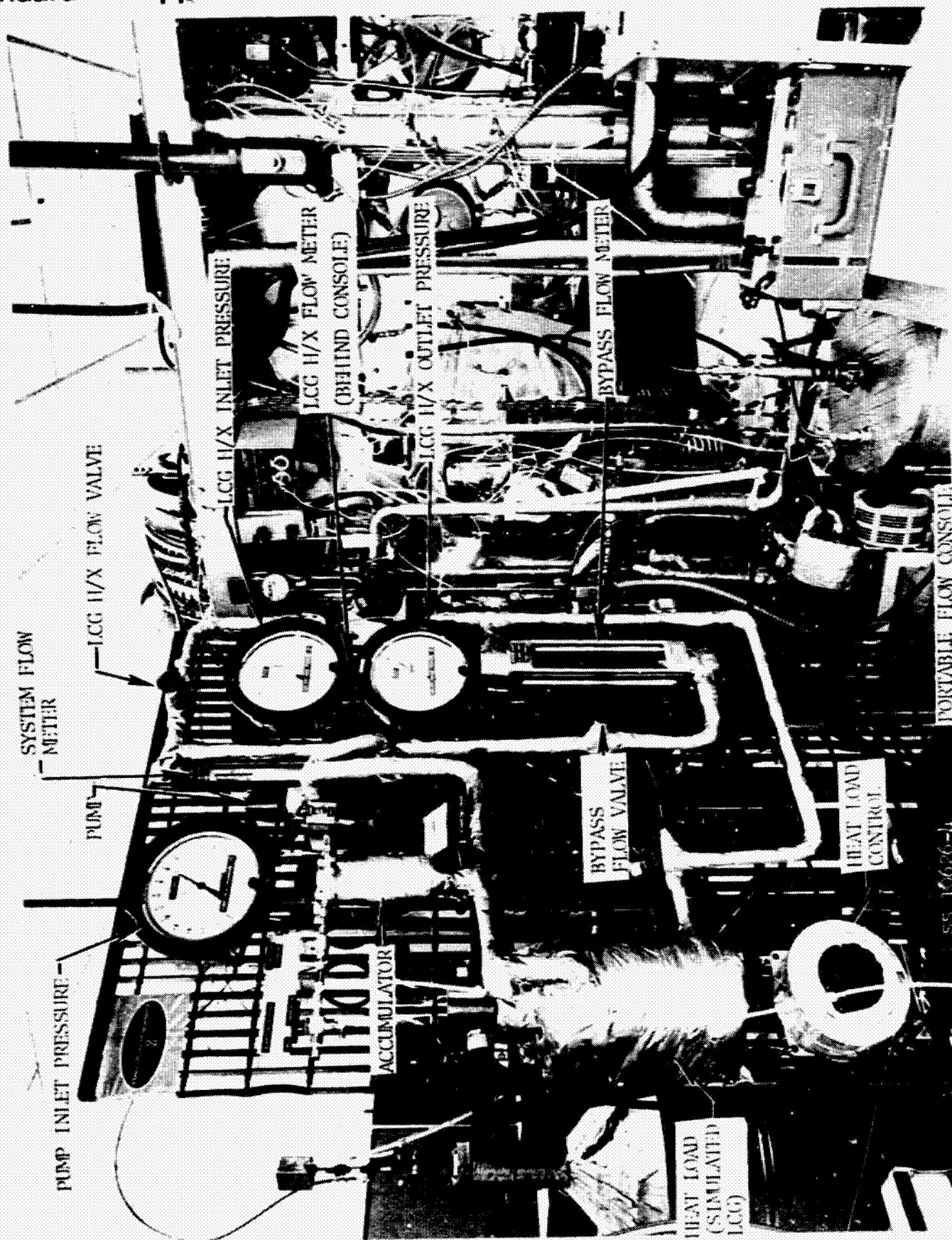


CRYOSTAT TEST SET UP - VARIATION

FIG. 10

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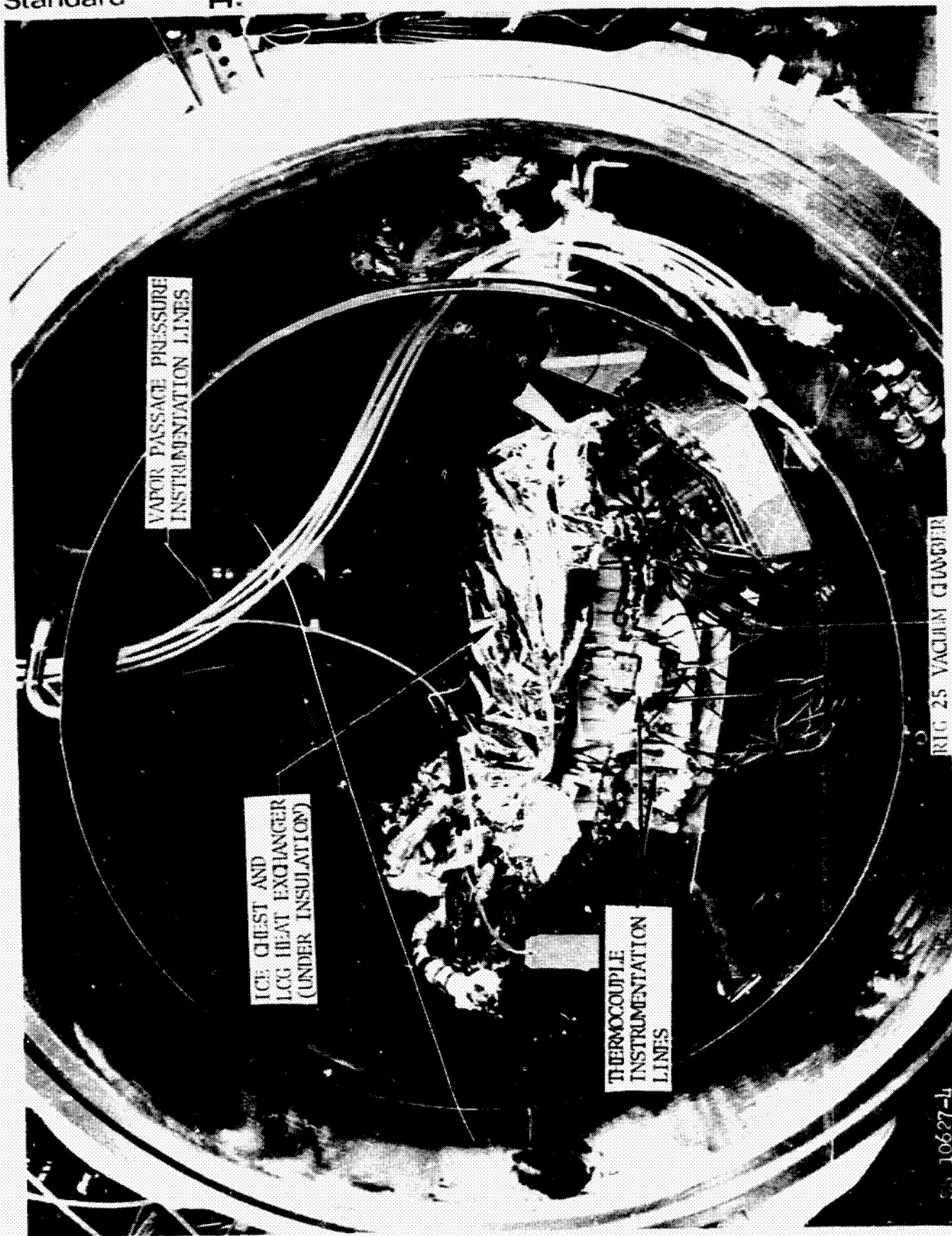
PORTABLE FLOW CONSOLE

SS 15626-4

FIGURE 54

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Full scale design feasibility testing of the ice chest and air-cooled heat exchanger was initially performed at room ambient conditions and provided data necessary to evaluate the ability of the ice chest to withstand repeated freeze-thaw cycling. Further room ambient testing was performed on the ice chest/LCG heat exchanger combination to debug the Portable Flow Console and develop proficiency in rig operation and hardware handling techniques.

Ice chest operation and water boiler operation then were verified in vacuum environment utilizing the procedures specified in sections 4.3 and 5.4 of the Design Feasibility Test Plan, except the hard vacuum evaluation of the heat transfer interface surface could not be accomplished. The vacuum chamber could not pull down to 10^{-4} mmHg vacuum with the instrumented ice chest in the chamber because of the previously mentioned leakage along the thermocouple wires. Therefore, the hard vacuum evaluation of the heat transfer interface surface was rescheduled to be accomplished during the Development Test, in accordance with the original program plan.

The initial testing conducted at room ambient provided considerable experience with the test setup. The usefulness of these tests for performance evaluation, however, was limited by the fact that the chest contained an interim wick material which had a relatively inferior water absorption capability. As a result, large air pockets were present in the chest, degrading the unit's potential performance. This data therefore was not extensively analyzed.

In the subsequent thermo-vacuum tests the ice chest was reassembled with the optimum wicking material and an improved loading procedure devised to obtain a satisfactory charge. Four combined tests were conducted in the ice operating mode and two in the boiling (evaporative) mode. In the ice mode the tests evaluated operation at heat loads of 205, 440, and 879 J/s (700, 1500, and 3000 Btu/hr). In the emergency mode heat loads of 293, 440, 586, and 733 J/s (1000, 1500, 2000, and 2500 Btu/hr) were imposed. Ice pack operational performance in these tests was extensively analyzed.

The performance was compared to predicted performance by superimposing the test data on the predicted performance plots, figures 36-49. Review of the data shows that the ice mode performance was as expected at 879 J/s (3000 Btu/hr) while at 440 and 205 J/s (1500 and 700 Btu/hr) performance was satisfactory, but lower than predicted.

The temperature drop across the contact surface previously had been predicted to be 4.16, 2.08, and 0.97°K (7.5, 3.75, and 1.75°F) for respective heat loads of 879, 440, and 205 J/s (3000, 1500 and 700 Btu/hr) for a vacuum contact. Lower temperature drops were anticipated for contacts with trapped air. Table III gives the heat exchanger contact temperature differential near the outlet and middle of the heat exchanger at several heat loads.

TABLE III

HEAT EXCHANGER TEMPERATURE DROP ACROSS CONTACT
SURFACE FOR "ICE ALL MELTED" AND "EVAPORATIVE" CASES

Mode	Contact Condition	Heat Load		Contact ΔT , °K (°F)	
		J/s	(Btu/Hr)	Outlet	Middle
Evaporative	"vac"	293	(1000)	263.2 (14)	252.1 (12)
Evaporative	"vac"	440	(1500)	266.5 (20)	266.0 (19)
Evaporative	"vac"	586	(2000)	260.4 (9.0)	259.7 (7.5)
Evaporative	"vac"	733	(2500)	260.7 (9.5)	261.3 (10.5)
Ice	"vac"	498	(1700)	266.3 (19.5)	261.0 (10)
Ice	"vac"	469	(1600)	263.2 (14)	257.8 (4.5)
Ice	"vac"	938	(3200)	259.3 (7)	258.8 (6)
Ice	AMB	403	(1375)	257.2 (3)	258.8 (6)

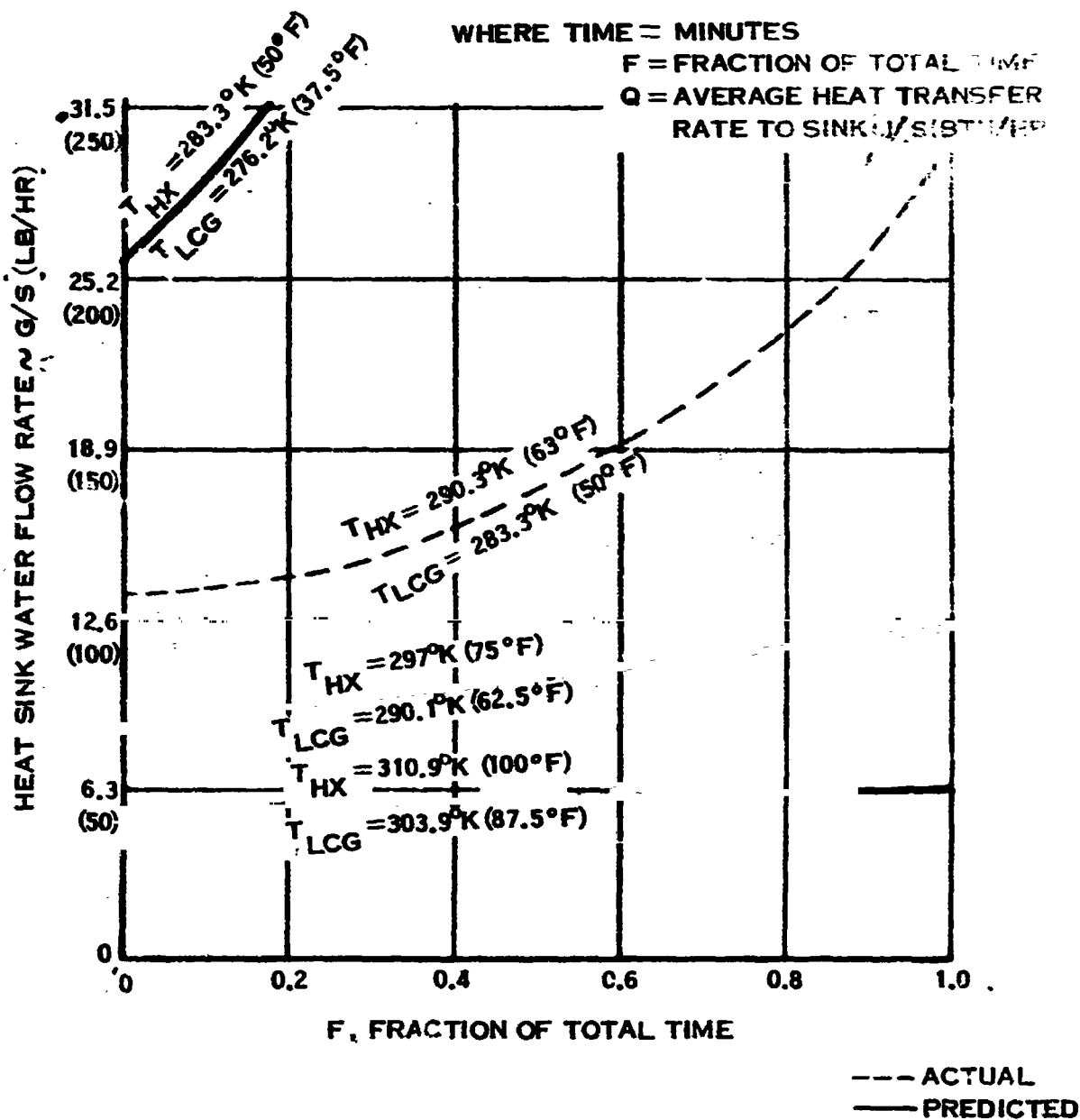
NOTE :

$$\text{ACTUAL TIME} = 3.88 \times 10^4 \frac{F}{Q} \quad (1.327 \times 10^5 \frac{F}{Q})$$

WHERE TIME = MINUTES

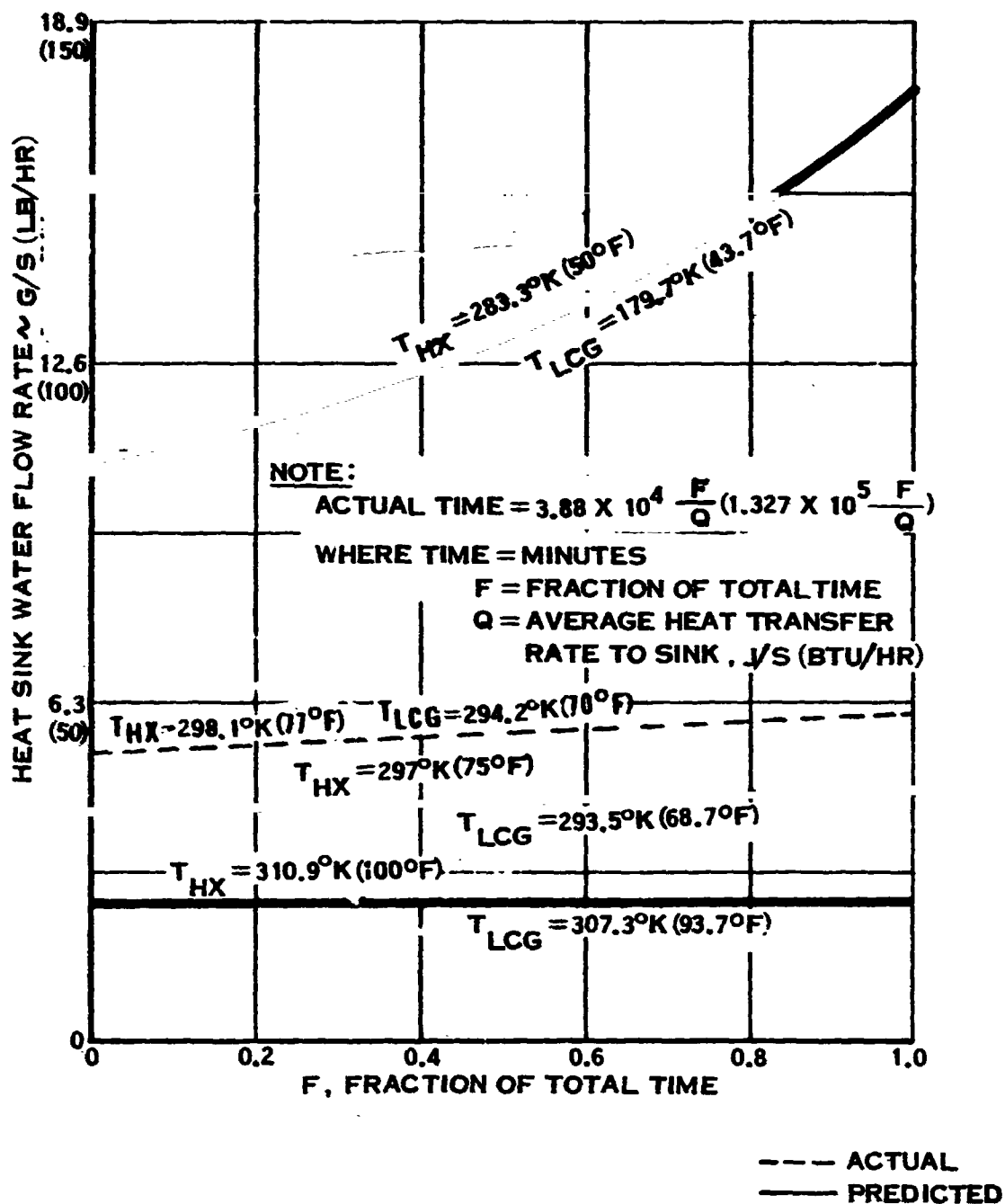
F = FRACTION OF TOTAL TIME

Q = AVERAGE HEAT TRANSFER
RATE TO SINK (J/S) (BTU/HR)



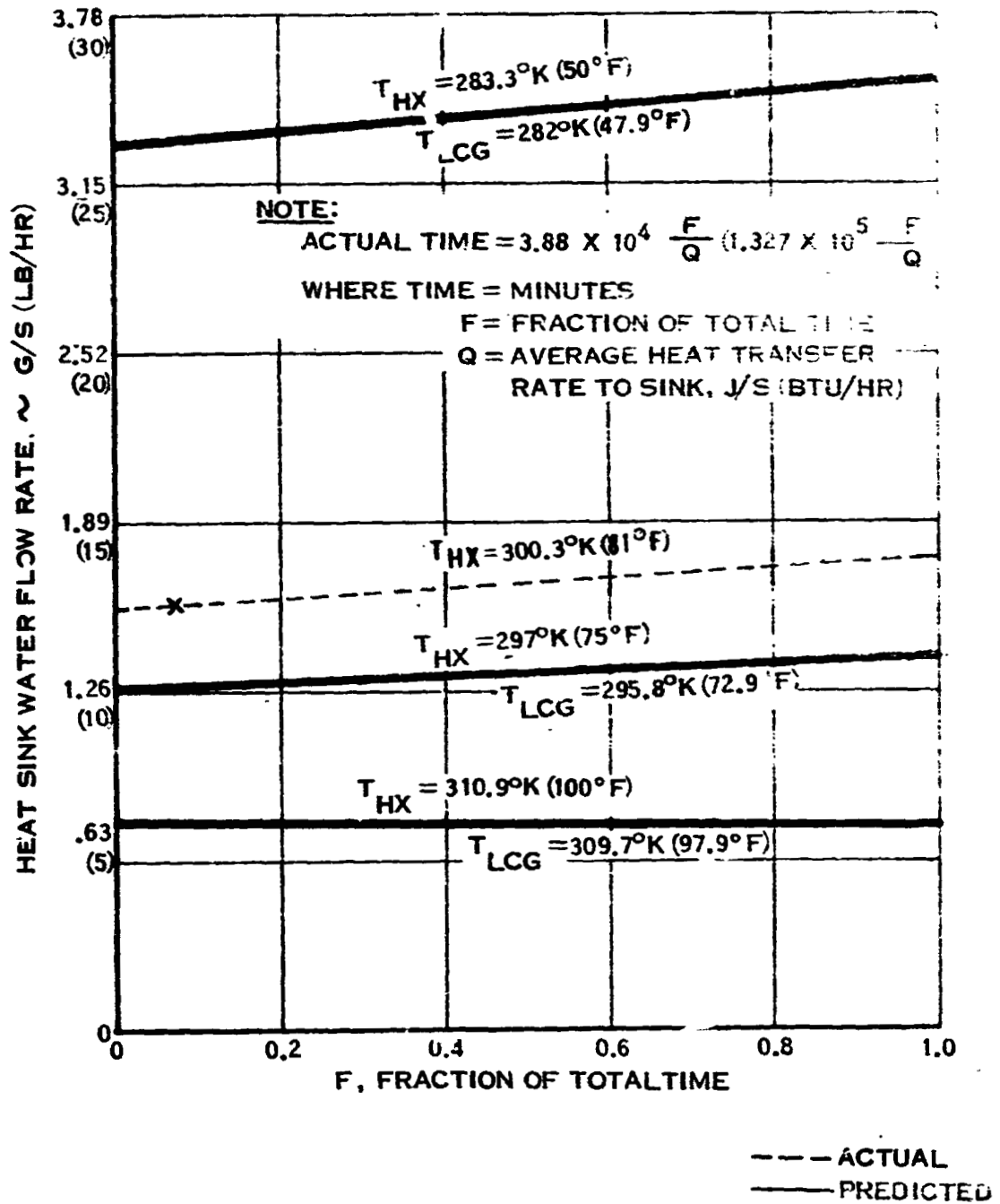
PERFORMANCE TIME DEPENDENCY FOR $Q_{SINK} = 879 \text{ J/S (3000 BTU/HR)}$

FIGURE 36 *



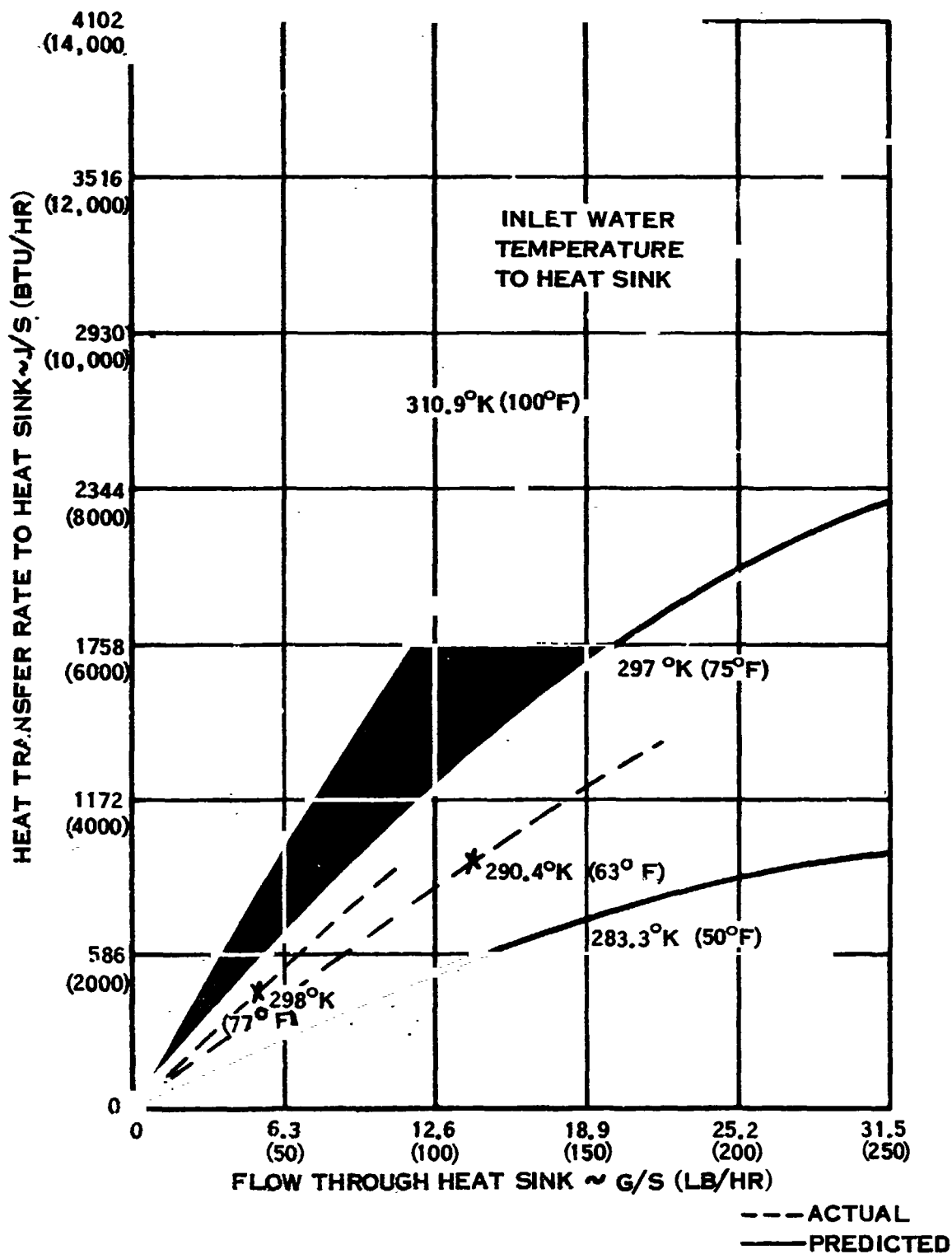
PERFORMANCE TIME DEPENDENCY FOR
 $Q_{SINK} = 440 \text{ J/S (1500 BTU/HR)}$

FIGURE 37



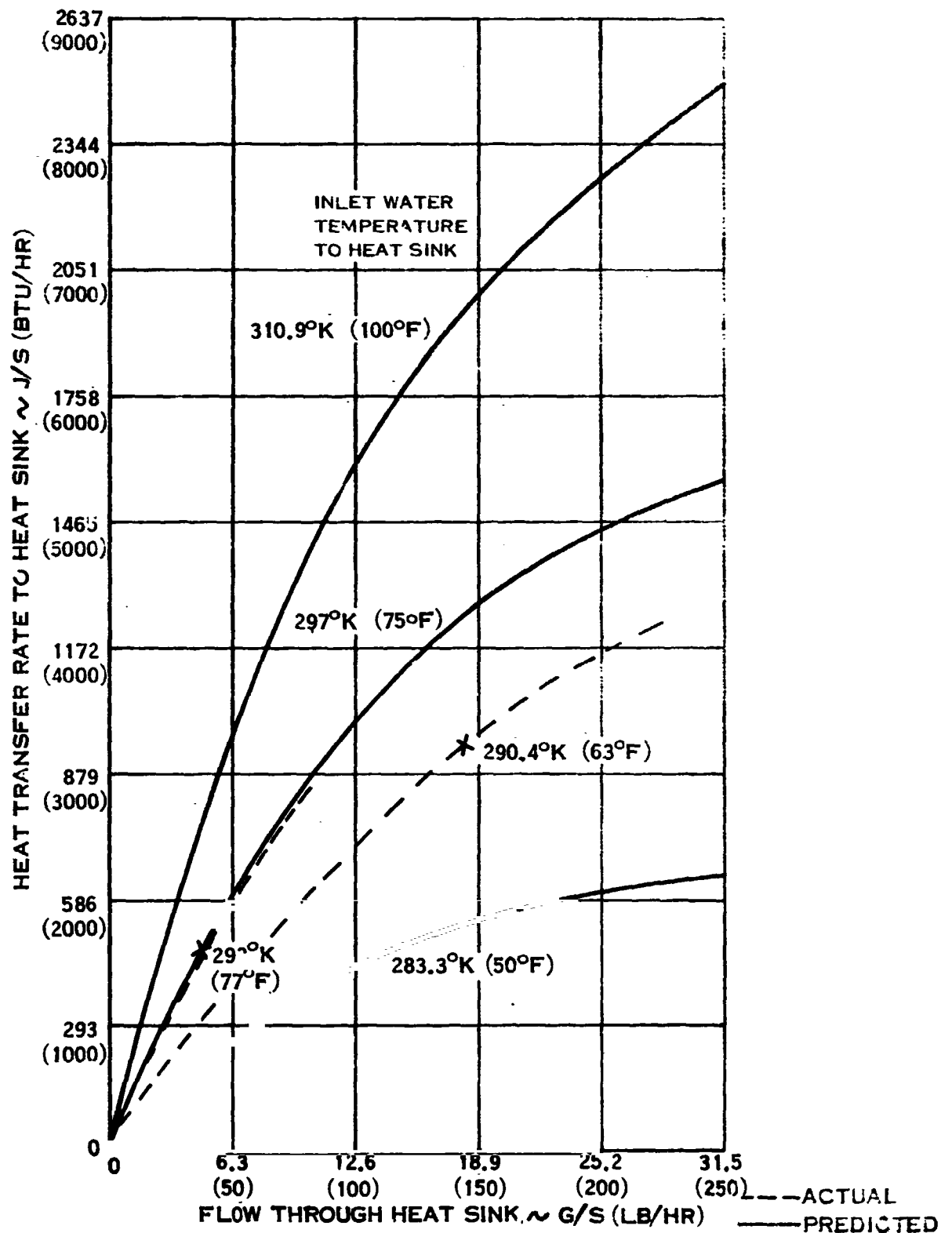
PERFORMANCE TIME DEPENDENCY FOR
 $Q_{SINK} = 146.5 \text{ J/S (500 BTU/HR)}$

FIGURE 38



HEAT SINK PERFORMANCE WITH NO ICE MELTED

FIGURE 39.



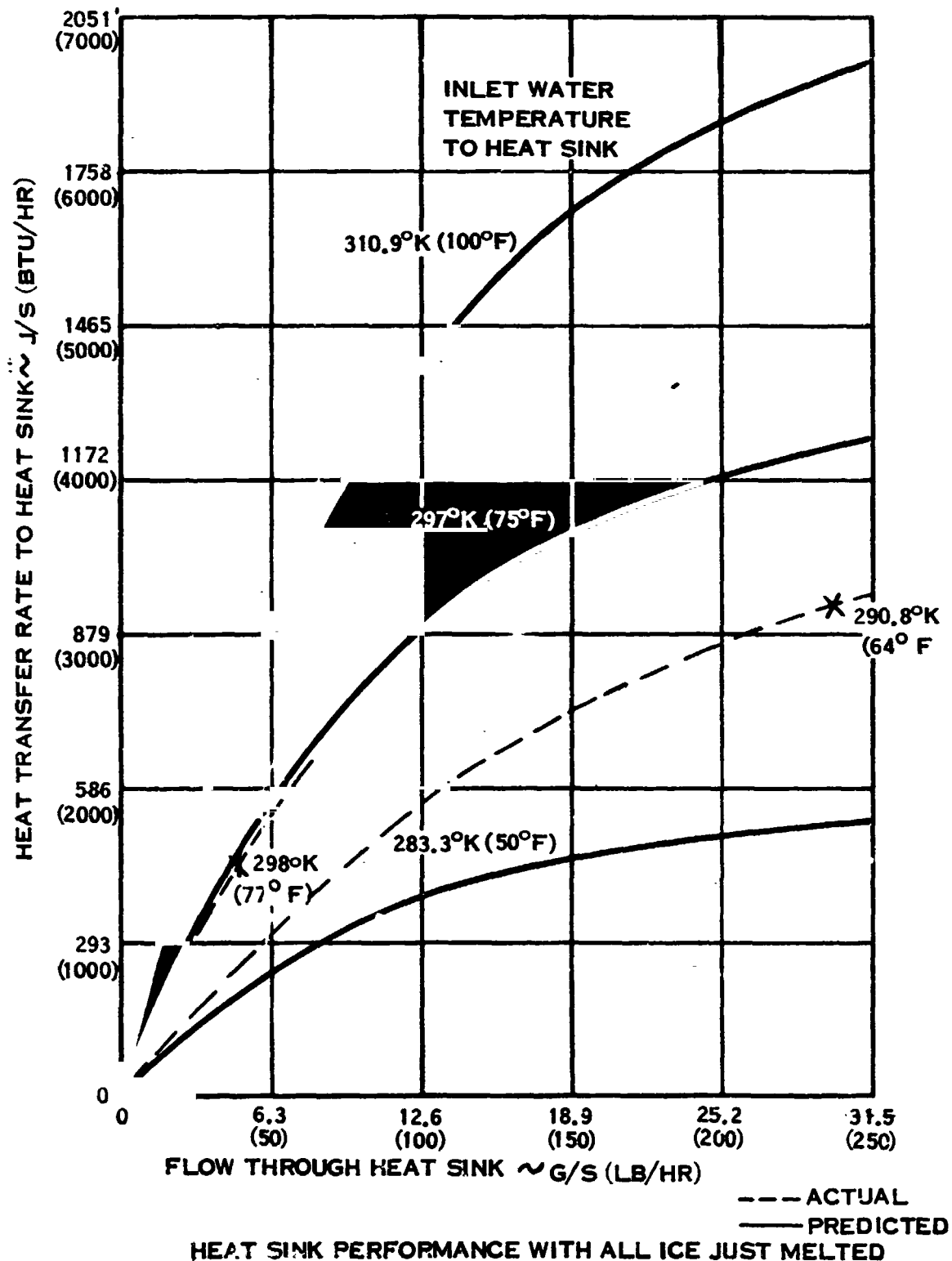


FIGURE 41.

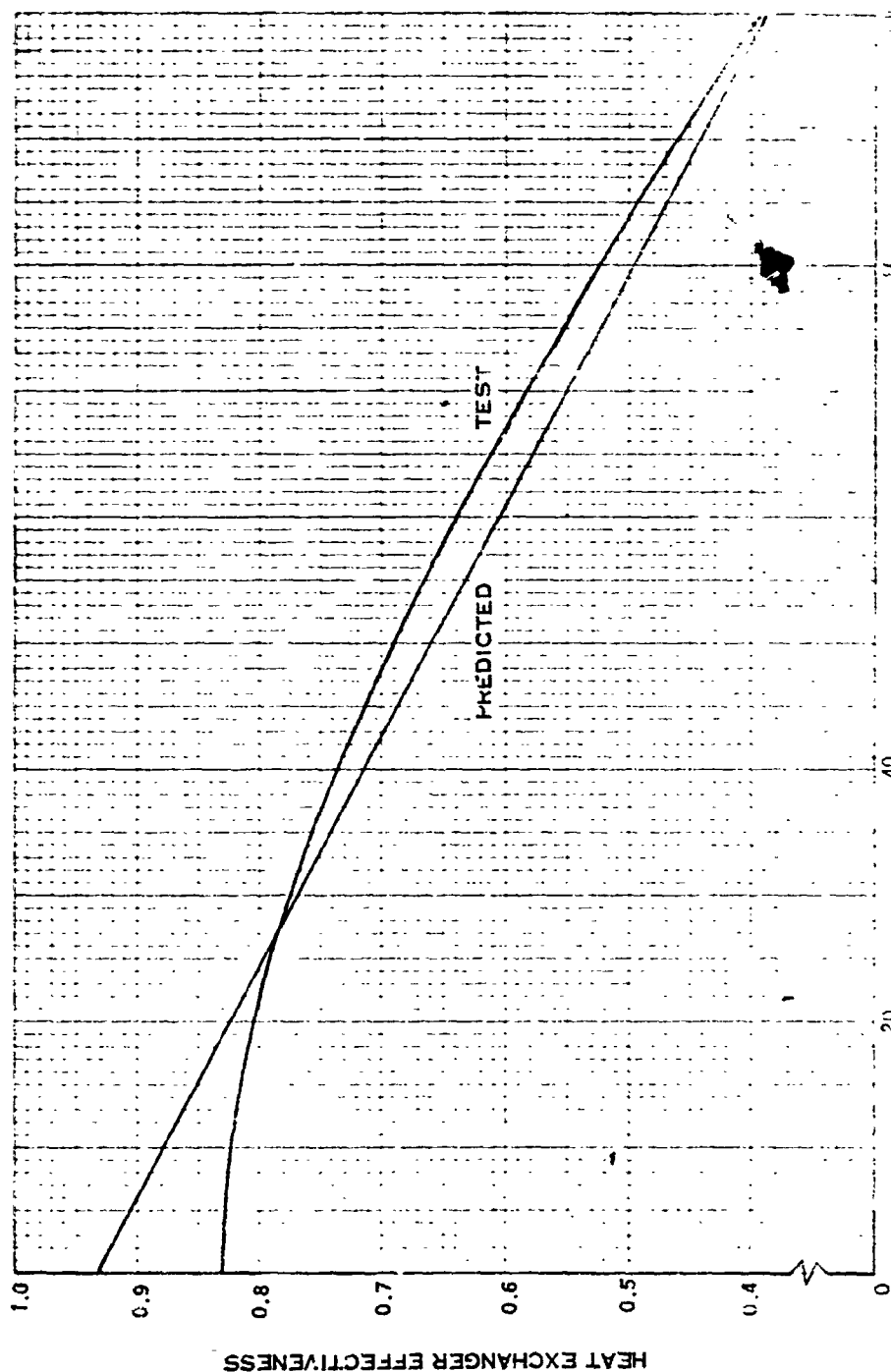


FIG. 1. MELTED %
EFFECTIVENESS VS. PERCENT ICE MELTED, 0% / 100% STU / 100
FIGURE 10.

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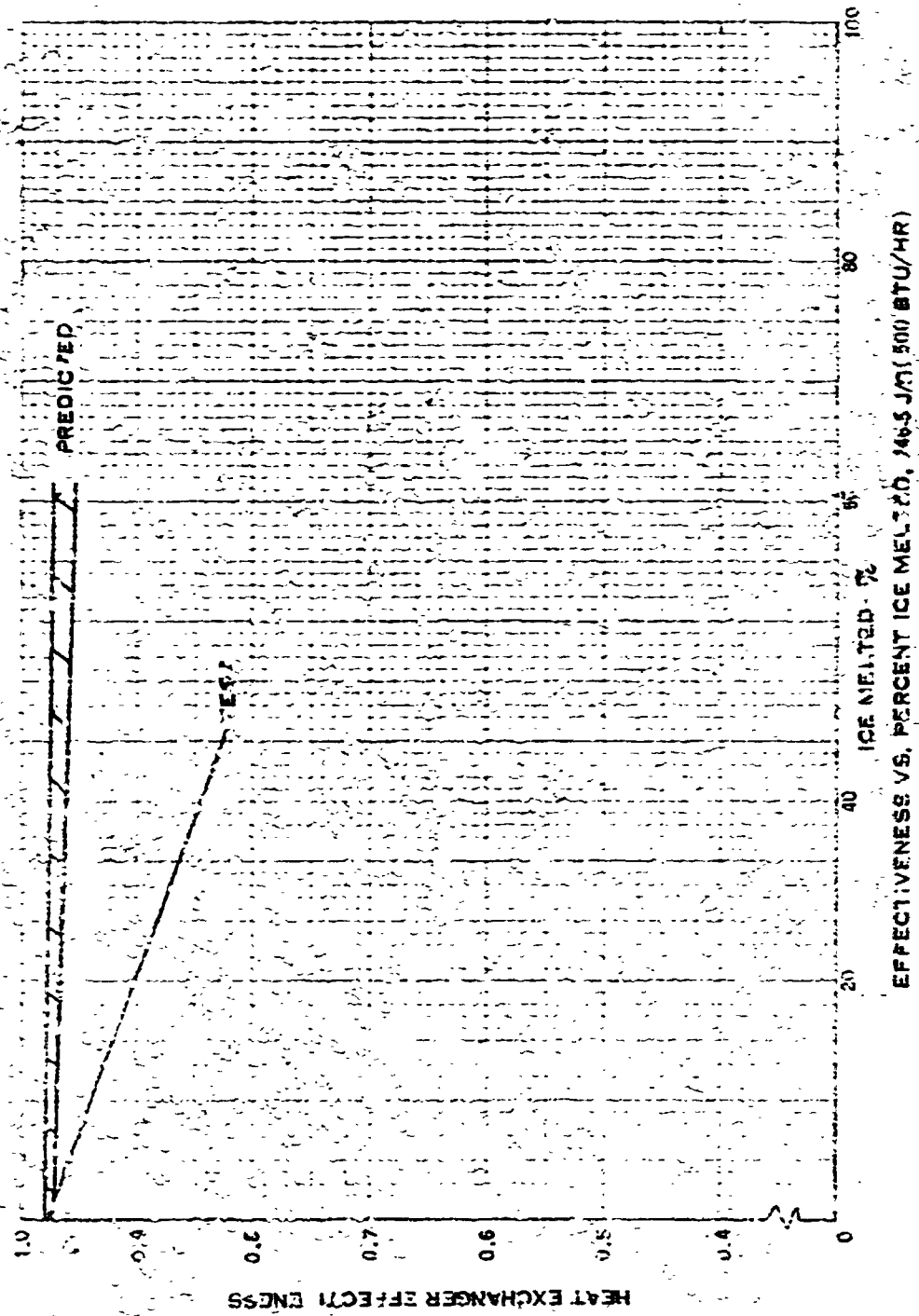
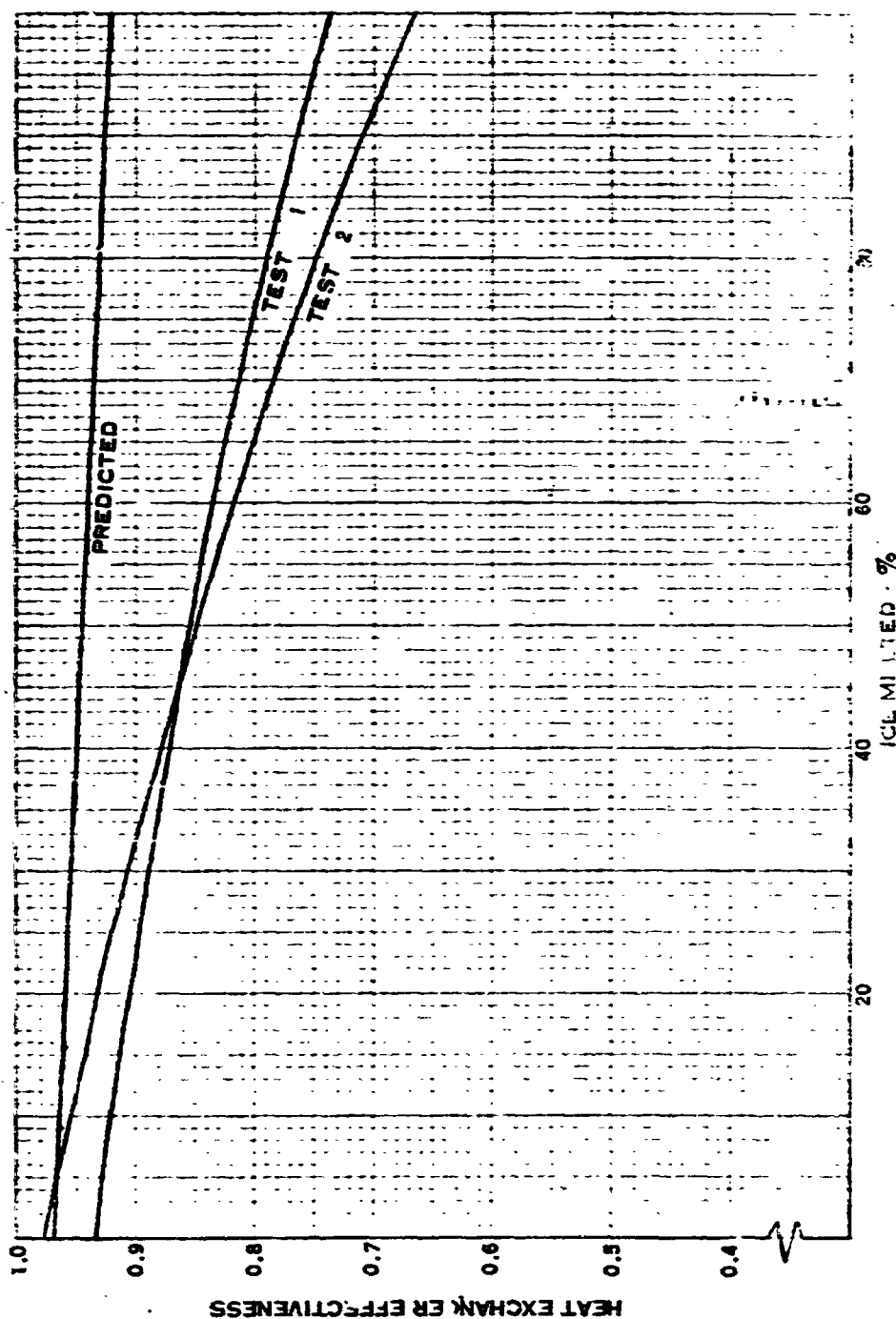


FIGURE 45



EFFECTIVENESS VS. PERCENT ICE MELTED, 400 (100 BTU / HR)

FIGURE 11

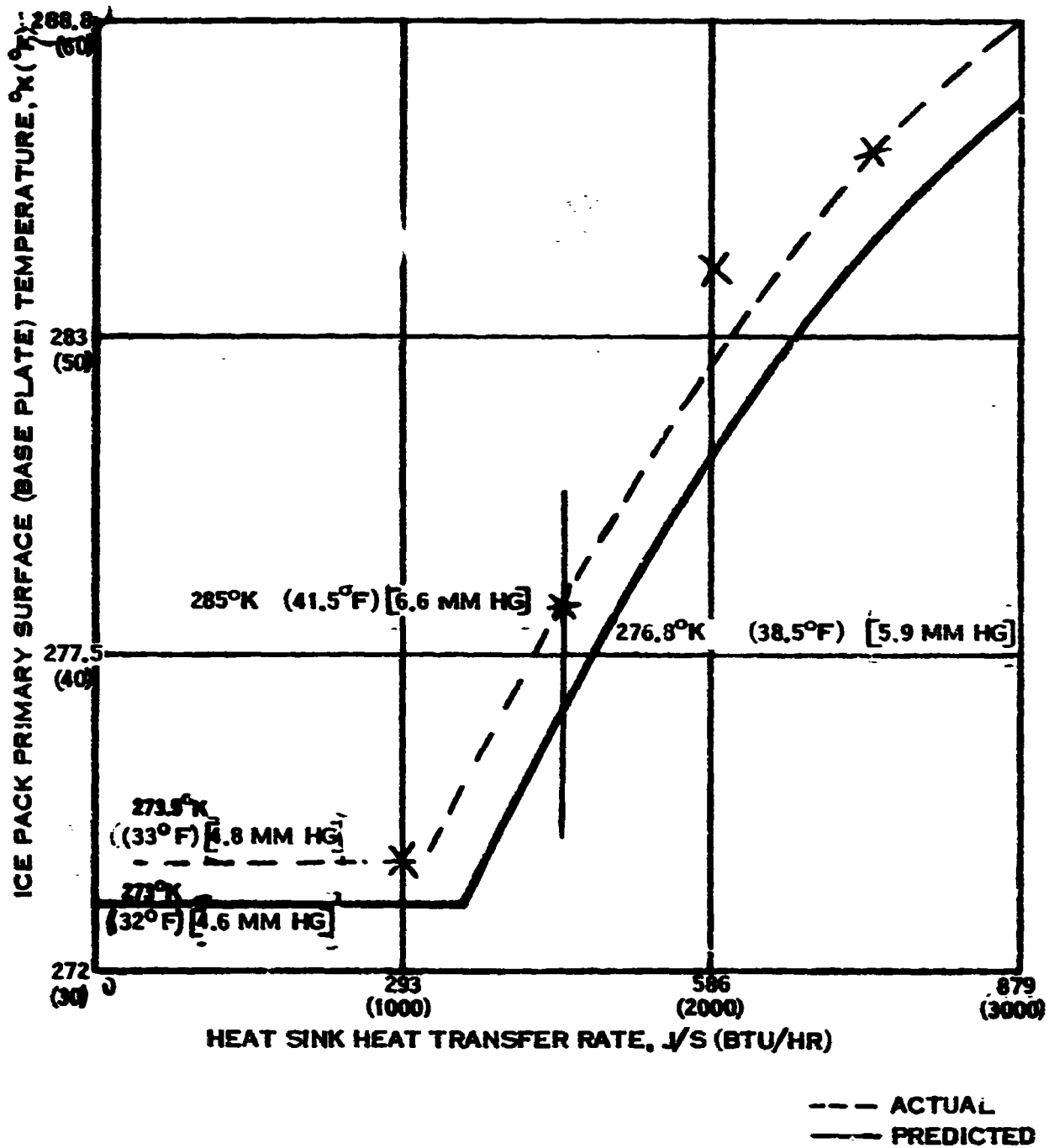
At 938 J/s (3200 Btu/hr) the contact temperature differential was as expected; however, for the lower heat loads it was much higher with the exception of the ambient test case. Furthermore, there is a lack of complete consistency in the data. A bluing test of the contact surface showed that the contact zones only covered approximately 20 percent of the total surface due to lead plating build-up at the edges. On this basis the contact ΔT might be expected to be five times the predicted values. This is in general agreement with the test data. The inconsistency in the data may have resulted from variations in the surface mating due to the lead build-up. The ambient case showed that the temperature drop was less than in vacuum, as expected.

Evaporation mode testing was done with a 1.59 cm (0.625 inch) diameter orifice. The performance of the unit is illustrated in Figures 45 through 49. Figure 45 shows that the evaporant surface temperature is approximately 1.67°K (3°F) above predicted. This indicates that this system exhaust pressure drop is nearly as expected.

Figure 46 shows effectiveness versus time for this test and is as predicted. Since the test started with a heat load of 293 J/s (1000 Btu/hr), freezing at the evaporant surface occurred as expected; however, it is pertinent that as the heat load was increased to 440 J/s (1500 Btu/hr) the frozen wick was unable to adequately feed water to the evaporant plate with the resultant degradation in effectiveness and performance noted in Figures 47 and 48. Nevertheless the unit continued to perform adequately. The test was not run to completion, so no assessment of effective utilization can be made. A second evaporative mode test at heat loads of 586 and 733 J/s (2000 and 2500 Btu/hr) showed that performance approximated predictions when the unit had not been previously frozen.

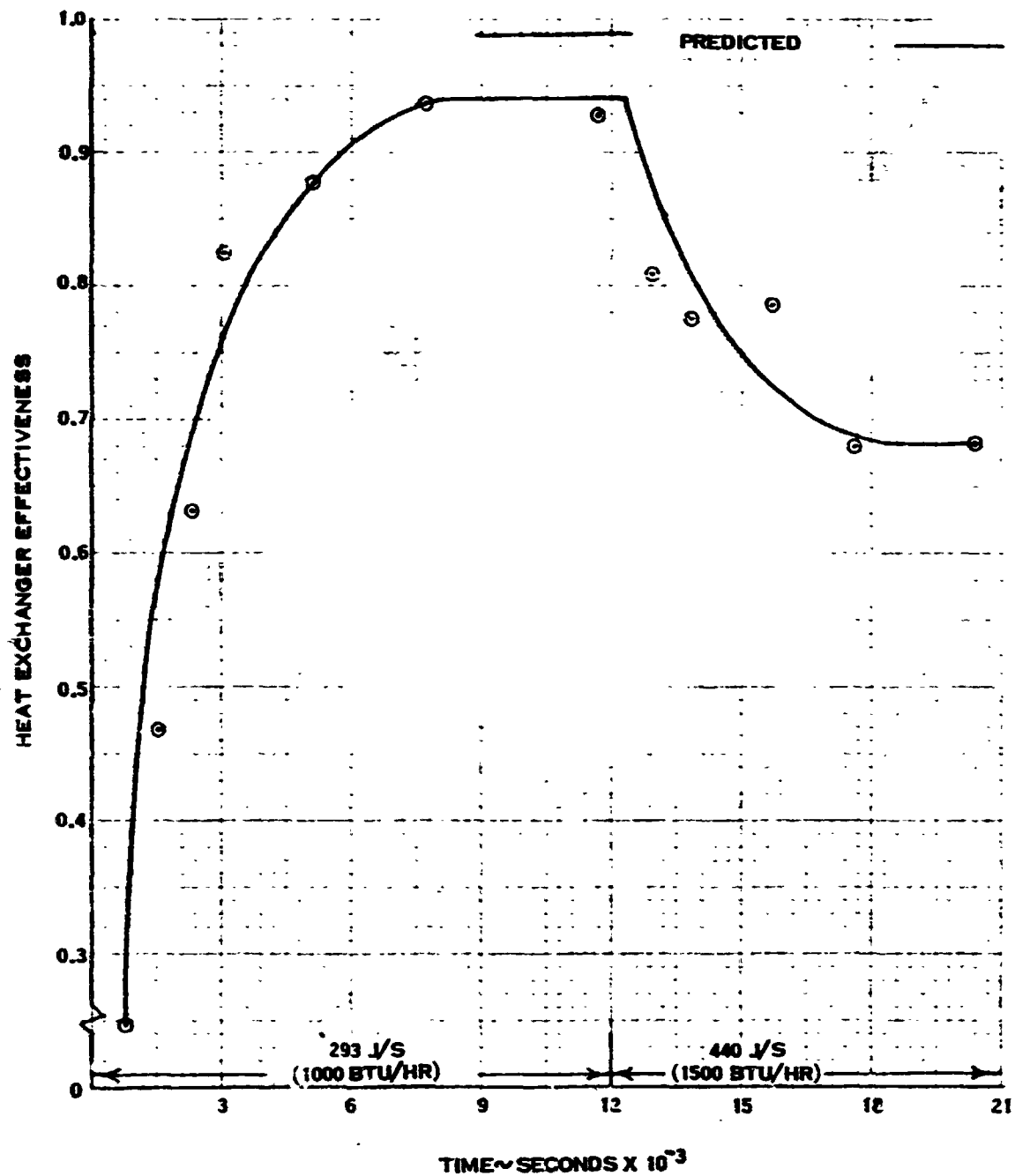
The predicted effectiveness for the respective heat loads was 0.96 and 0.69 versus 0.94 and 0.60 actually achieved.

Heat exchanger inlet and outlet absolute pressures were monitored during the tests. Figure 49 shows the pressure drop versus flow from all tests. Predicted pressure drop was approximately one-half that measured. The higher pressure drop is attributable in part to inaccuracy of measuring caused by measuring absolute pressures rather than a delta pressure and due to the fact that this data includes flowmeter and line pressure drops.



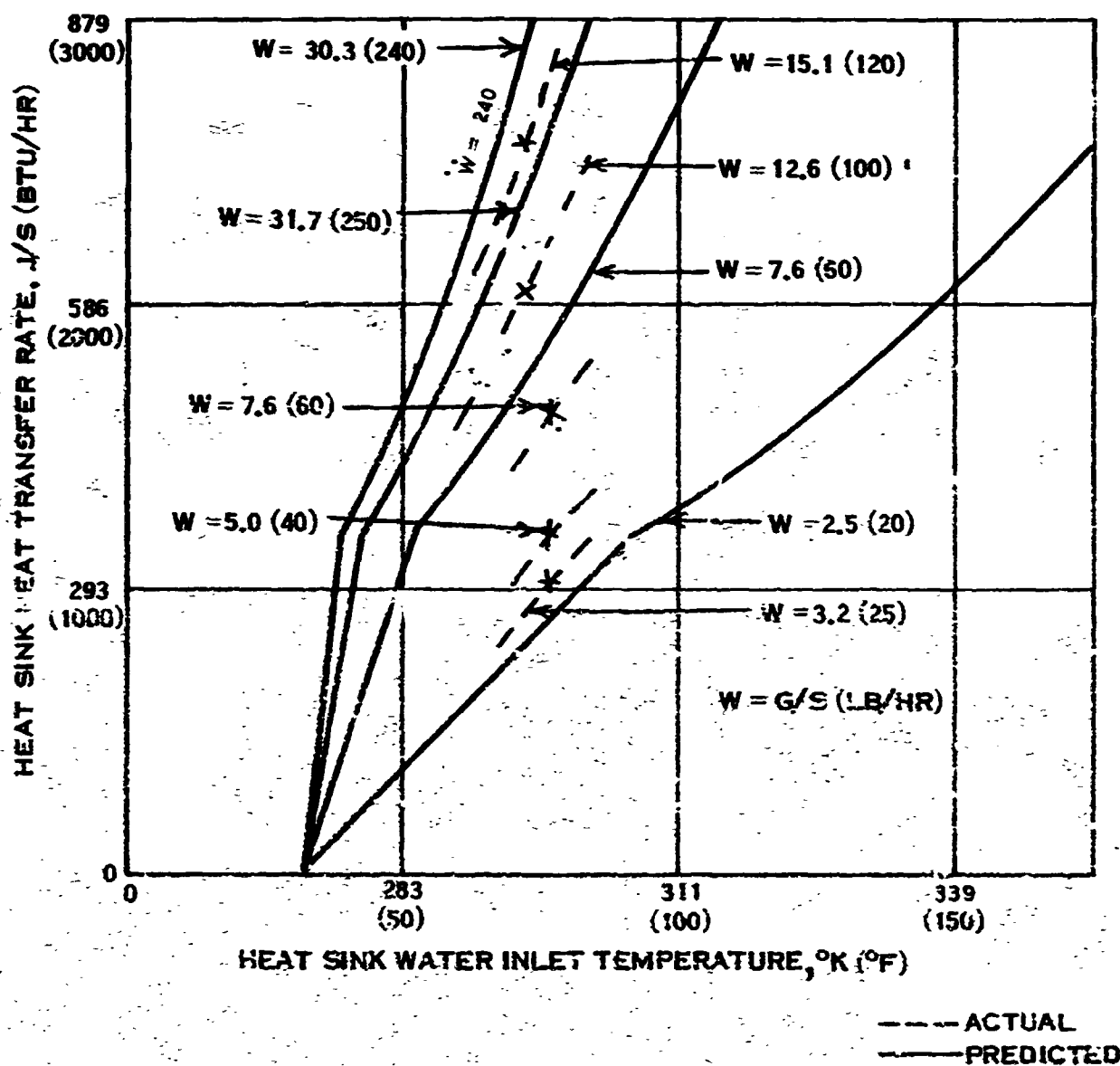
EMERGENCY PERFORMANCE

FIGURE 45.



EFFECTIVENESS VS. TIME ~ EVAPORATIVE MODE

FIGURE 46.



EMERGENCY PERFORMANCE

FIGURE 47.

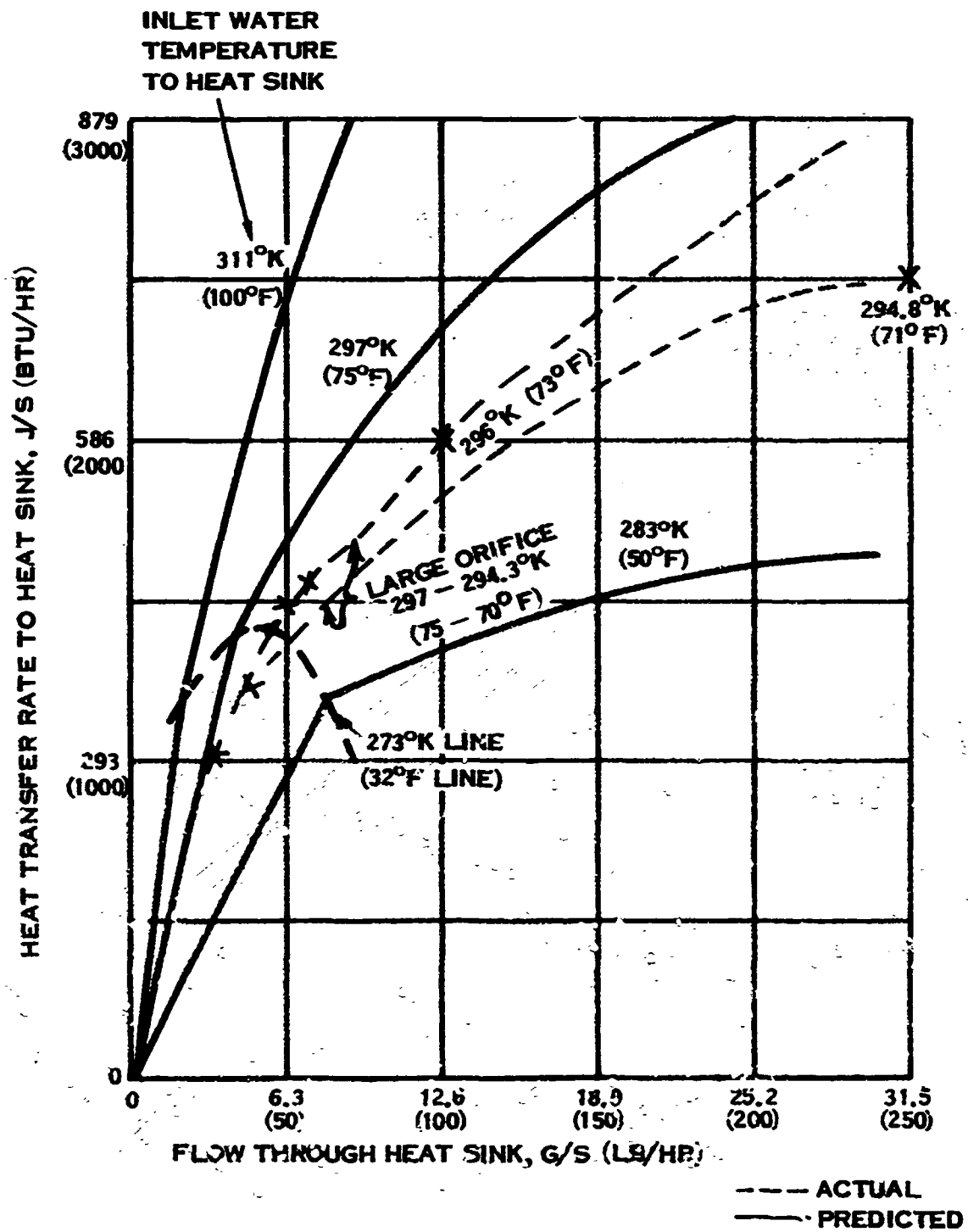
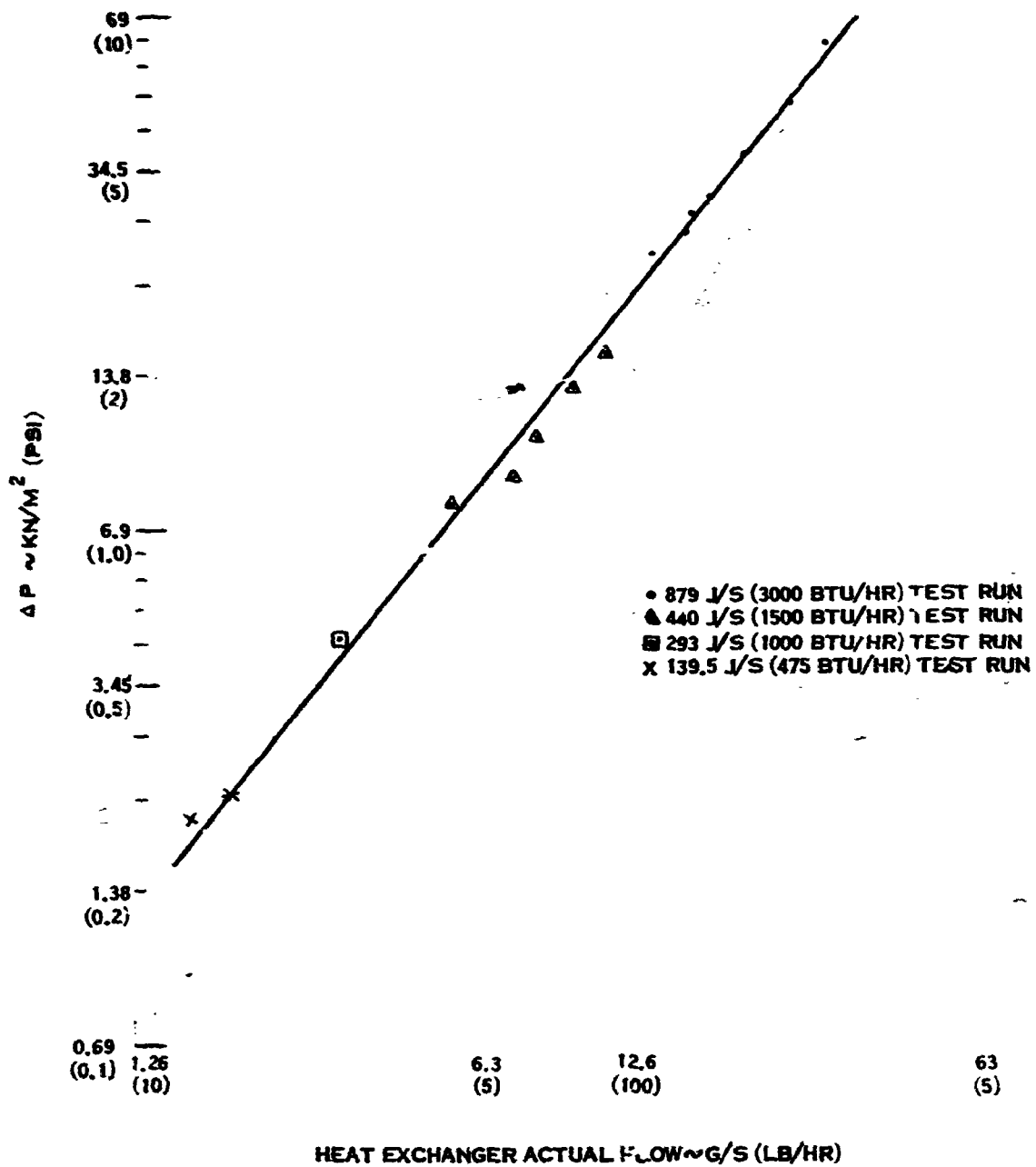


FIGURE 48.

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HEAT EXCHANGER PRESSURE DROP VS ACTUAL FLOW
FIGURE 49.

The conclusions obtained from the feasibility test program are listed below:

- Heat exchanger liquid pressure drop as recorded is twice predicted.
- Ice mode 879 J/s (3000 Btu/hr) performance agrees with prediction.
- Temperature drop across the heat transfer surface is higher than anticipated. Bluing test of contact zones correlates with test results.
- Further testing with an improved contact surface would be required to identify reasons for satisfactory, but lower than predicted, ice mode performance at 440 and 205 J/s (1500 and 700 Btu/hr).
- Emergency mode testing demonstrated basic feasibility of this operating mode. At 293 J/s (1000 Btu/hr) the large orifice resulted in freezing. Subsequent 440 J/s (1500 Btu/hr) heat load was met, but performance was degraded, probably due to wick dry-out. Further testing without having frozen the wick demonstrated that performance is approximately as expected.
- Evaporant surface temperature is as predicted - within 1.67°K (3°F).

Utilizing the results derived from the feasibility test program the sizing of the ice chest and LCG heat exchanger were judged to be satisfactory for incorporation into the Lab Model design. Further development effort was judged to be required to improve the flatness of the LCG heat exchanger lead coating and to fully evaluate the water retention capabilities of the existing wick expansion compensation device when used with deaerated water. Changes in this area were minor and in no way imposed a constraint on Lab Model system design and packaging.

The Lab Model system was designed utilizing system flow and heat transfer data developed primarily during the feasibility hardware analysis task.

DEVELOPMENT TESTING

The purpose of conducting development testing was to obtain data of actual performance, to verify satisfactory performance of the unit. Selected tests from the Development Test Plan were used to investigate various ice chest/LCG heat exchanger interface configurations and to check out an alternate wick/expansion compensation configuration. The Development Test Plan is included as Appendix D and the development test log sheets are included as Appendix E of this report.

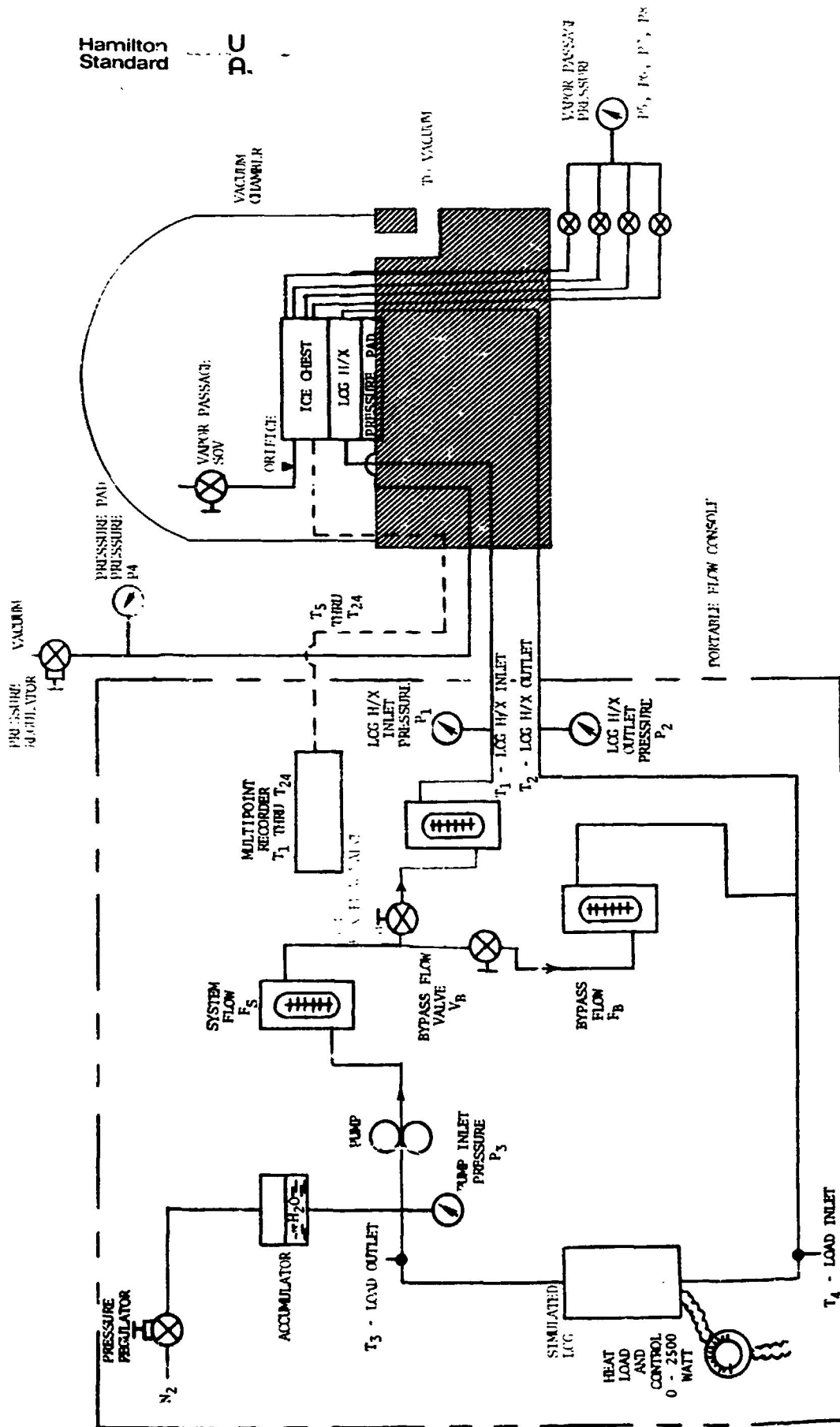
Development test hardware consisted of the previously manufactured LCG Heat Exchanger (S/N 1) and ice chest (S/N 1), and an additional LCG Heat Exchanger (S/N 2). Modifications were made to the LCG Heat Exchanger plating technique to allow investigation of different interface configurations. Internal ice chest hardware was manufactured to allow testing of an alternate wick/expansion compensation configuration.

The design and assembly of the development test rig was completed in accordance with the Development Test Plan. This test rig utilized the feasibility hardware test rig with modifications to provide for better deaeration of the transport fluid, and the addition of circuitry to allow checkout of the vapor passage shutoff valve and controller. Figure 50 is a schematic of the development test setup and figure 51 is a schematic of the vapor passage shutoff valve and controller setup.

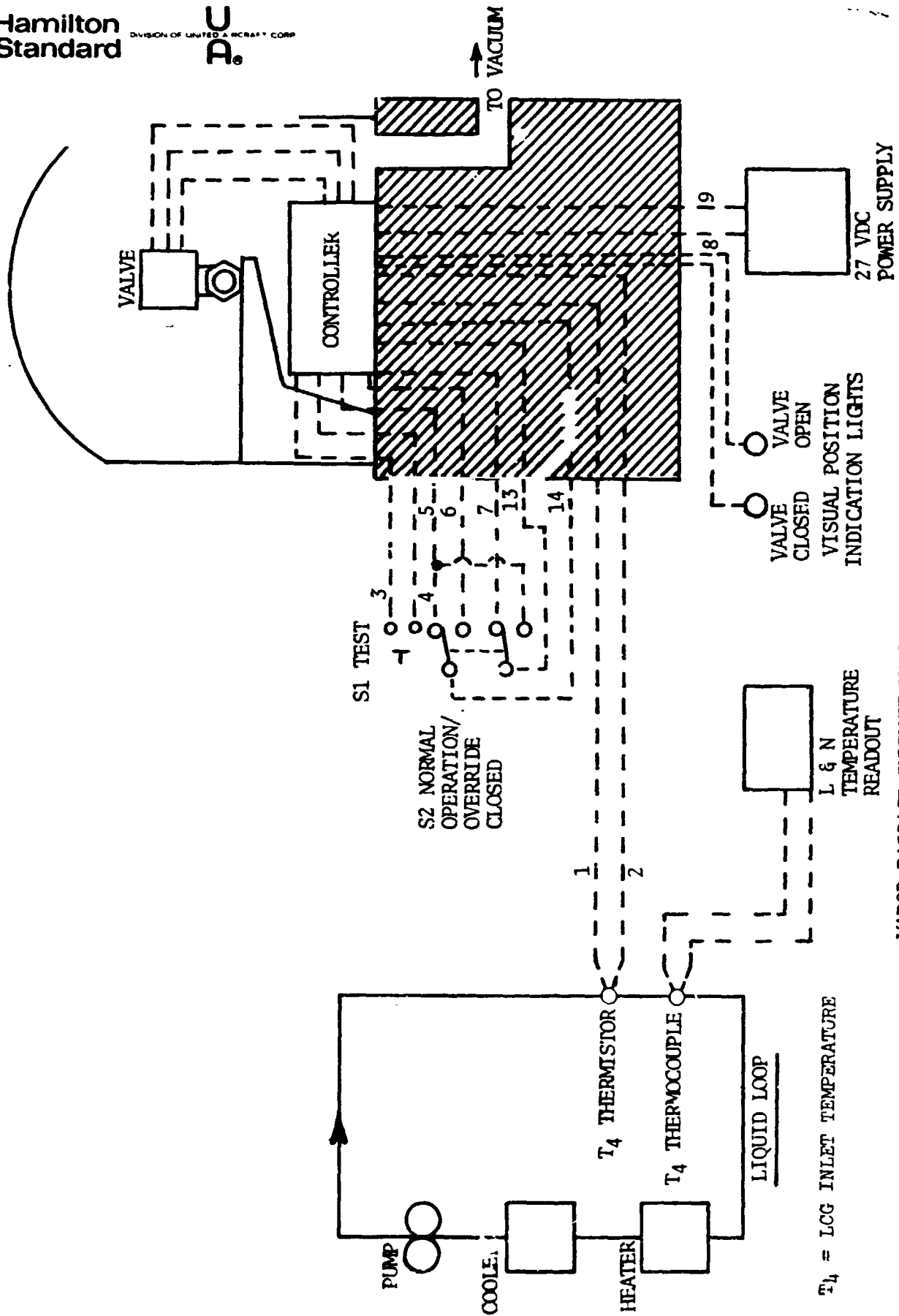
PRELIMINARY DEVELOPMENT TESTING

Preliminary development testing was performed on an "improved" wick/expansion compensation configuration. The main advantage of this configuration, shown in figure 52, is that the expansion compensation device - large open cell semi-rigid foam - is pressure balanced, thereby eliminating the forcing out of water when the ice chest vacuum passage is depressurized. The configuration performed very satisfactorily as a water boiler and dumped little or no water (< 60 ml) into the exhaust duct trap. Unfortunately, however, the configuration was poor in the ice melting mode; the semi-rigid foam was insufficient to preload the frozen wicks against the ice chest fins when the wicks started to thaw and contract. It is speculated that gaps opened up between the wicks and the fins during thawing. The closed cell sponge material initially used has sufficient preload to push the wicks into contact with the fins. In any event, the semi-rigid open cell foam version did not perform satisfactorily in the ice melting mode.

The original restrained closed cell expansion compensation device was readopted with a modification instituted to minimize the objectionable dumping of water during vacuum operation. Specifically, the change consisted of applying a preload to the metal containment plates to squeeze the foam rubber approximately 25 percent prior to installing the teflon restraint cording. This procedure ensured that all of the slack was taken out of the teflon restraint cording and thereby minimized further growth during vacuum exposure



DEVELOPMENT TEST SETUP
FIGURE 50

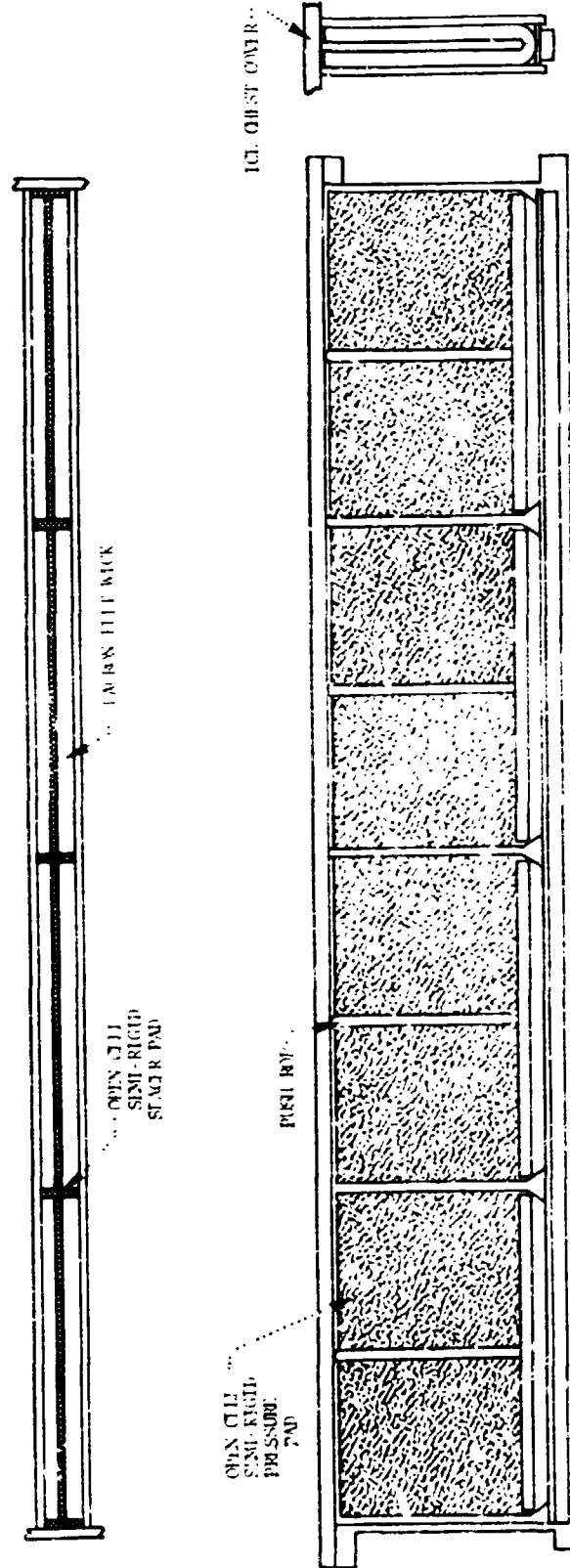


VAPOR PASSAGE SHUTOFF VALVE & CONTROLLER SETUP

FIGURE 51

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FIG. 1

FIG. 2

as illustrated in figure 53 . Test data indicated that the modification did, in fact, improve the water dumping problem, lowering it into the 600 ml range. This quantity, while significant, is small enough so that it does not clog the liquid trap and hence causes no operational problems.

Full scale interface investigation testing was initiated to assess the interface temperature drop for the interface conditions given in Table IV.

Testing was performed in the boiling mode utilizing the "improved" wick/expansion compensation configuration. The boiling mode was chosen because it allowed continuous testing without requiring refreezing between runs. Also, the greater endurance capability in the boiling mode allowed longer runs at deep vacuum (10^{-4} torr), thereby making more efficient use of the test facility. The temperature drop across the interface for each configuration was corrected to account for slightly different heat loads and compared to the feasibility hardware temperature drop. This comparison allowed performance prediction based on the performance obtained during feasibility testing.

Description of Preliminary Development Test Runs

Eight runs were conducted during the preliminary development testing as summarized in Table IV. A brief discussion of each run is presented in the following sections.

Run #1

As expected, the bare aluminum on bare aluminum surface condition produced very poor heat transfer, resulting in a relative temperature drop of approximately 2. Relative temperature drop is defined as the ratio obtained by dividing the interface temperature drop of the run under investigation by the interface temperature drop of the feasibility configuration run.

Run #2

A 0.10 mm (0.004 inch) thick lead foil "gasket" was inserted between the bare aluminum surfaces and produced a relative temperature drop of 1.06. It is believed that the two air gaps (aluminum to lead and lead to aluminum) increased the temperature drop.

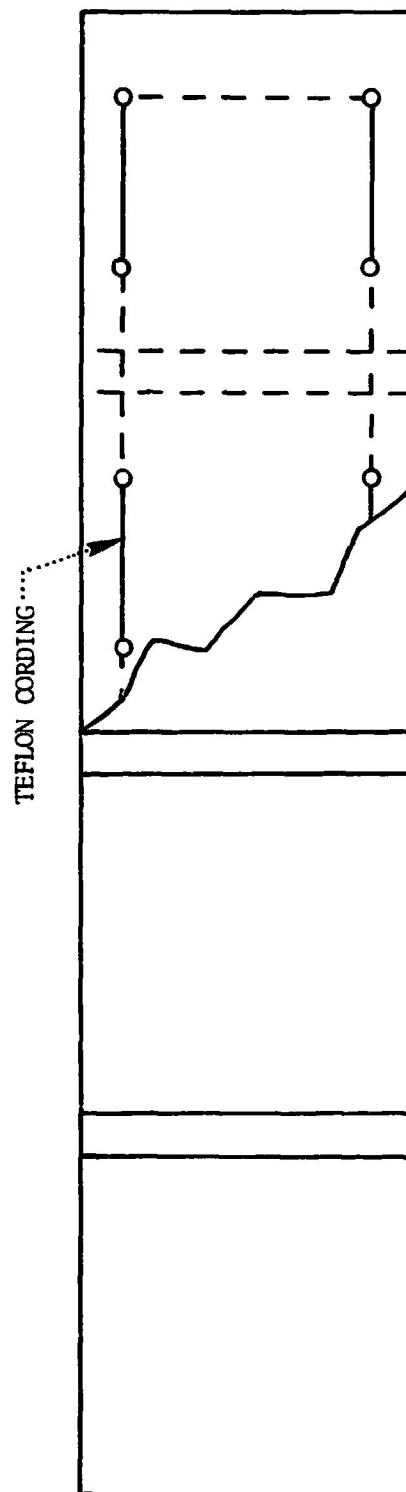
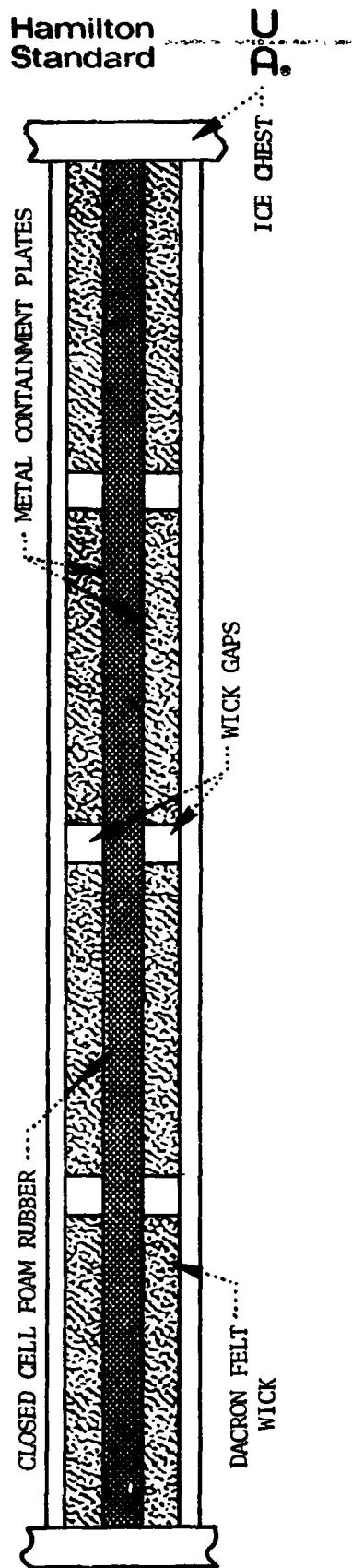
Run #3

The lead foil was epoxy bonded to the heat exchanger to eliminate one aluminum to lead air gap. Unfortunately the relative temperature drop increased to 1.25. This increase can be attributed to two factors: the drop through the relatively thick, approximately 0.075 mm (0.003 inch), epoxy and the uneven lay of the bonded foil. It is believed that improved bonding techniques could improve this interface configuration.

TABLE IV
ICE CHEST/LCG HEAT EXCHANGER INTERFACE CHARACTERISTICS

RUN NO.	SURFACE CONDITION	INTERFACE PRESSURE		CHAMBER PRESSURE AT INTERFACE MATE μ = microns Hg torr = millimeters Hg	RELATIVE TEMPERATURE DROP
		LOADING PRESSURE $\frac{\text{KN}}{\text{M}^2 \Delta}$	(psid)		
1	Bare aluminum on bare aluminum	55.2	(8)	200 μ	1.94
2	0.10 mm (0.004") thick lead foil between bare aluminum surfaces	55.2	(8)	200 μ	1.06
3	0.10 mm (0.004") thick lead foil bonded to heat exchanger surface	55.2	(8)	200 μ	1.25
4	0.10 mm (0.004") thick plated heat exchanger (feasibility conf.)	55.2	(8)	200 μ	1.0
5	0.10 mm (0.004") thick improved lead plated heat exchanger	55.2	(8)	200 μ	0.85
6	0.10 mm (0.004") thick improved lead plated heat exchanger	55.2	(8)	< 10 ⁻⁴ torr	0.94
7	0.10 mm (0.004") thick improved lead plated heat exchanger	34.4	(5)	< 10 ⁻⁴ torr	1.03
8	0.10 mm (0.004") thick improved lead plated heat exchanger	20.7	(3)	< 10 ⁻⁴ torr	1.05

Note: Relative Temperature Drop is defined as the ratio obtained by dividing the interface temperature drop of the run under investigation by the interface temperature drop of the feasibility configuration run.



LAB MODEL WICK/EXPANSION COMPENSATION CONFIGURATION

FIGURE 53

Run #4

This run utilized the feasibility configuration and was considered as the standard run.

Run #5

The lead plated heat exchanger used in run #4 and in the feasibility testing was reworked to remove the lead ridge around the outer border. The surface was scraped to the point where a bluing test showed no high points. This configuration was tested and showed a relative temperature drop of 0.85. It was decided to run the deep vacuum testing with this configuration.

Run #6

Deep vacuum testing (10^{-4} torr) of the run #5 configuration produced a relative temperature drop of 0.94. This was extremely encouraging because it showed that the interface temperature drop characteristics of this configuration at deep vacuum were better than those of the feasibility hardware at 200 μ . The feasibility hardware was good enough to more than meet program performance requirements.

Run #7

Decreasing the interface loading pressure from run #6 to 34.4 kN/m² Δ (5 psid) produced a relative temperature drop of 1.03.

Run #8

A further decrease of the interface loading pressure to 20.7 kN/m² Δ (3 psid) produced a relative temperature drop of 1.05.

LAB MODEL CONFIGURATION DEVELOPMENT TESTING

At this point in time sufficient data had been compiled to allow selection of a configuration for the lab model hardware to meet the program requirements.

A lead plated heat exchanger with the lead ridge around the outer border removed was chosen as the lab model configuration. This configuration provides satisfactory performance and appears to be reasonably durable, judging from the number of runs made during feasibility and development testing.

The configuration selected for the ice chest wick/expansion compensation assembly is the same basic one utilized in the feasibility hardware. Two improvements were made to this configuration. The wicks were cut into smaller pieces and separated, to prevent the wicks from losing water if the ice chest were set up on end, as illustrated in figure 53. The expansion compensation sandwich, consisting of two metal plates with an inside piece of closed cell foam, is held together by teflon cording. The sandwich is precompressed approximately 25 percent by a fixture during the cording process so that no

slack exists in the cords of the completed sandwich. This is important because any slack in the cords would allow the sandwich to grow during vacuum (boiler) operation thereby expelling water into the vapor passages. Runs made during lab model configuration development testing indicated that the water expelled during vacuum testing had been cut in half compared to the water expelled by the feasibility configuration.

Full scale development testing was conducted on the final Lab Model hardware configuration. The analysis of this one-g operation data had two specific purposes. First, the analysis was intended to confirm that the Lab Model Ice Pack Heat Sink Subsystem meets all requirements of the NASA ARC "Specification for Ice Pack Heat Subsystem for Extravehicular Activity", Specification No. 2-17753, contained in Appendix H. Second, the analysis was used to generate optimum methods for collecting data necessary for predicting the performance of subsequent ice chest designs.

Normal Mode Testing

Testing of the melting ice mode of operation confirmed that the unit holds the design water mass of 7 kg (15.4 lbm) and that cooling loads of 139.5-879 J/s (475-3000 Btu/hr) could be maintained. For an ice mass of 7 kg (15.4 lbm) the predicted times for all of the ice to melt while providing the required range of cooling loads are presented in Table V.

TABLE V
PREDICTED ICE MELTING TIME

Heat Load		Time to Melt	
J/s	(Btu/hr)	ks	(hrs - min.)
139.5	(475)	16.56	(4 hrs. 36 min.)
440	(1500)	5.28	(1 hr. 28 min.)
879	(3000)	2.64	(44 min.)

During the melting cycles, various LCG heat exchanger/bypass water flow splits were used to maintain appropriate unit outlet temperatures. The results of these melting cycles are depicted in Table VI.

TABLE VI
MELTING ICE TEST RESULTS

Heat Load		Unit Outlet Temperature		Time to Melt Ice	
J/s	(Btu/hr)	°K	(°F)	ks	(hrs - min.)
139.5	(475)	302.6	(85)	15.12	(4 hrs. 12 min.)
440	(1500)	480.8	(70)	4.26	(1 hr. 11 min.)
879	(3000)	282.7-284.8	(49-53)	3.06	(50 min.)

Emergency Mode Testing

Testing of the water boiling mode of operation was conducted over the range of heat loads of 139.5-879 J/s (475-3000 Btu/hr) for a maximum LCG water flow rate of 31.5 g/s (250 lbm/hr).

For a unit cooling load of 879 J/s (3000 Btu/hr), LCG outlet temperatures between 293.7°K and 295.3°K (69-72°F) were maintained for over 7.2 ks (two hours). LCG outlet temperatures of 284.3-287.8°K (52-56.5°F) were sustained for the 440 J/s (1500 Btu/hr) load, while the LCG outlet temperature was maintained at 284.3-284.8°K (52-53°F) for the 139.5 J/s (475 Btu/hr) load operation.

For this emergency evaporative mode of operation, overall unit effectiveness ranged from 0.477 to 0.615 for an LCG water flow rate of 31.5 g/s (250 lbm/hr), as compared to a predicted effectiveness of 0.680. As the contact conductance between the LCG heat exchanger and the ice chest is the predominant factor affecting overall unit effectiveness, analysis of the test data indicated unit contact conductances of 543-862 J/s-m²-°K (96-152 Btu/hr-ft²-°F) in the development unit as opposed to a design value of 1134 J/s-m²-°K (200 Btu/hr-ft²-°F). For a nominal ice chest emergency mode effectiveness of 0.546, the contact conductance would be 685 J/s-m²-°K (121 Btu/hr-ft²-°F).

Conclusions

The following conclusions were derived from the development test program:

- The unit holds 7.0 kg (15.4 lbm) of water.
- Cooling loads of 139.5, 440, and 879 J/s (475, 1500, and 3000 Btu/hr) can be satisfied for both the normal melting ice and emergency water boiling modes of operation.
- Appropriate water temperatures can be maintained. For normal operation unit outlet temperatures at 302.6°K (85°F), 294.2°K (70°F), and 282.7-284.8°K (49-53°F) correspond to heat loads of 139.5 J/s (475 Btu/hr), 440 J/s (1500 Btu/hr), and 879 J/s (3000 Btu/hr) respectively.

For emergency operation unit outlet temperatures of 284.3-284.8°K (52-53°F), 284.3-287.8°K (52-56.5°F), and 293.7-295.3°K (69-72°F) correspond to heat levels of 139.5 J/s (475 Btu/hr), 440 J/s (1500 Btu/hr), and 879 J/s (3000 Btu/hr) respectively.

- While the development test results do not provide sufficient data to precisely extrapolate performance maps for normal and emergency modes of operation, contact conductance is 50-60 percent of the analytical value of 1134 J/s-m²-°K (200 Btu/hr-ft²-°F) resulting in a lesser overall unit effectiveness than that predicted analytically but still within the effectiveness required to meet performance requirements.

Reduction of the test data indicates that optimum data utilization may be obtained if the following steps were taken during any subsequent testing.

- Normal mode operation should be run at 139.5-440 J/s (475-1500 Btu/hr) over a wide LCG heat exchanger water flow spectrum, ignoring the unit mixed outlet temperature.
- Accurate contact conductance measuring points should be utilized to obtain precise temperature differentials for a 879 J/s (3000 Btu/hr) load application in both normal and emergency operation.
- The emergency water boiling mode should be run over the whole spectrum of LCG heat exchanger water flow rates.

This test sequence would provide a complete series of unit effectiveness maps for both normal and emergency modes of operation.

Acceptance Testing

Acceptance testing was performed on each of the two functional laboratory model Ice Pack Heat Sink Subsystem in accordance with the Acceptance Test Plan in Appendix F. Acceptance Test Log sheets are included in Appendix G.

Table VII summarizes the results of the Acceptance Test. All acceptance test requirements were met.

TABLE VII
ACCEPTANCE TEST RESULTS
(External Power Supply Except As Noted)

	Unit #1	Unit #2	Specification
Main Power Switch Operation	OK	OK	Main power on/off
Pump Switch Operation	OK	OK	Pump power on/off
Normal mode pilot light	OK	OK	On when main power switch is on, override open switch in normal position, H/X outlet temperature less than 287°K (57°F), and override closed switch moved first to override closed and then returned to normal position.
Pump Operation	OK	OK	Pump starts/stops in response to pump power switch.
139.5 J/s (475 Btu/hr) flow condition	1.315 g/s (0.021 gpm)	1.315 g/s (0.021 gpm)	Pump on. H/X flow control valve closed. Adjust bypass flow control valve to obtain system flow of 31.3 g/s (0.5 gpm). H/X flow to be 1.315 ± 0.313 g/s (0.021 ± 0.005 gpm).
440 J/s (1500 Btu/hr) flow condition	OK	OK	Pump on. Adjust H/X flow control valve and bypass flow control valve to obtain system flow of 31.3 g/s (0.5 gpm) and bypass flow of 26.3 g/s (0.42 gpm). Note compliance.

TABLE VII (Concluded)

	Unit #1	Unit #2	Specification
879 J/s (3000 Btu/hr) flow conditions	OK	OK	Pump on. Adjust H/X flow control valve and bypass flow control valve to obtain system flow of 31.3 g/s (0.5 gpm) and bypass flow of 8.75 g/s (0.14 gpm). Note compliance.
Override open switch	OK	OK	Valve closed. Actuate override open switch with H/X outlet temperature operation less than 287°K (57°F). Valve should open and remain open.
Override closed switch	OK	OK	Valve open. H/X outlet temperature less than 287°K (57°F). Actuate override closed switch to override position and return to normal position. Valve should close and remain closed.
Vapor passage shutoff valve and controller	288.8°K (60.2°F)	288.9°K (60.4°F)	Pump on. System flow 31.3 g/s (0.5 gpm) and bypass flow 8.75 g/s (0.14 gpm). Apply heat to LCG H/X and note temperature where valve opens as noted by change in pilot light from normal to emergency. Temperature to be 288.7 ± 1.7°K (60 ± 3°F)
Emergency mode pilot light	OK	OK	
Override closed switch	OK	OK	Valve opened by raising H/X outlet temperature. Actuate override closed switch to override and leave in override. Valve should close.
Battery operation checkout	OK	OK	Disconnect external power supply and connect battery. Pump on. Adjust H/X flow control valve and bypass control valve to obtain system flow of 31.3 g/s (0.50 gpm) and bypass flow of 8.75 g/s (0.14 gpm). Cycle valve with override switches. Pump should continue to operate and valve should cycle properly.

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APPENDIX A

VAPOR PASSAGE PRESSURE DROP

SAMPLE CALCULATIONS

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For the case of 1500 Btu/hr heat load, assume evaporative temperature of approximately 37°F, giving a vapor pressure of .108 psia (steam tables). For 37°F, absolute viscosity in a near vacuum is

$$8.8 \times 10^{-5} \text{ poise or}$$

$$2.13 \times 10^{-2} \text{ lb}_m/\text{hr-ft.}$$

At 37°F,

$$h_{fg} = 1073 \text{ Btu/lb}_m.$$

Therefore,

$$\omega = \frac{1500 \text{ Btu/hr}}{1073 \text{ Btu/lb}_m} = 1.40 \text{ lb}_m/\text{hr}$$

Analyzing each section separately:

1) Cell drop: Cell cross sectional area = A; equivalent diameter = Deq

$$A = .38 \times .2 = .076 \text{ inches}^2$$

$$\text{Deq} = \sqrt{.076/\pi/4} = .311 \text{ inches}$$

Since flow has an uneven distribution take conditions at the half-way point and apply them for half the total length. L = half length.

$$L = 17.03/2 = 8.515 \text{ inches}$$

$$L/\text{Deq} = 8.515/.311 = 27.4$$

Half flow per cell = ω_c

$$\omega_c = 1.40/(28)(-) = .025 \text{ lb}_m/\text{hr/cell}$$

Calculate Reynolds Number. Reynolds Number = R_e

$$R_e = \frac{\omega_c \times \text{Deq} \times 12}{\mu \times A}$$

$$R_e = \frac{.025 \times .311 \times 12}{.076 \times 2.13 \times 10^{-2}} = 57.7$$

Since this is laminar flow, smooth ducts, set

$$f = 64/R_e = 1.11$$

therefore,

$$\Delta P = f \left(\frac{L}{D} \right) q$$

where q = dynamic head in psia.

By definition,

$$q = \frac{1}{2} \frac{V^2}{vg} \times 144$$

$$\text{and } V = \frac{v\omega}{A} \times \frac{144}{3600}$$

where v is specific volume in ft^3/lb_m

ω is weight flow in lb_m/hr

A is area in inches^2

g is gravitational constant

V is velocity in ft/sec

$$V = \frac{(2732)(.025)}{.076} \times \frac{144}{3600} = 35.9 \text{ ft/sec}$$

$$q = \frac{1}{2} \frac{(35.9)^2}{(2732)(32.2)(144)} = .0000509 \text{ psi}$$

$$\Delta P = 1.14 (27.4)(.0000509) = .0015 \text{ psi}$$

2) Mitered bend:

$$\Delta P = \lambda q$$

For a 79.2°F bend, $\lambda = 1.25$

Using the outside cell

$$\omega = 1.40/28 = .05 \text{ lb}_m/\text{hr}/\text{cell}$$

$$v = 2732 \text{ ft}^3/\text{lb}_m$$

$$A = .076 \text{ inches}^2$$

$$V = \frac{(2732)(.05)}{.076} \times \frac{144}{3600} = 71.9 \text{ ft/sec}$$

$$q = \frac{1}{2} \frac{(71.9)^2}{(2732)(32.2)(144)} = .000204 \text{ psi}$$

$$\Delta P = (.000204)(1.25) = .00026 \text{ psi}$$

3) Frictional header loss:

for a rectangle,

$$D_e = \frac{2(a)(b)}{(a+b)}$$

Defining at the midpoint

$$\begin{aligned} a &= .2 \text{ inches} \\ b &= .975 \text{ inches} \\ D_e &= .332 \text{ inches} \\ \text{half length } L &= 4.125 \text{ inches} \\ \frac{L}{D} &= 12.4 \end{aligned}$$

In the header take the midpoint of each half

$$A = .2 \times .975 = .1950 \text{ inches}^2$$

and a quarter of the total flow

$$\omega = 1.40/4 = .35 \text{ lb}_m/\text{hr}$$

to approximate the flow distribution.

Calculate Reynolds number.

$$R_e = \frac{(.35)(.332)(12)}{(.1950)(2.13 \times 10^{-2})} = 335.7$$

Laminar, therefore

$$f = 64/R_e = .191$$

Following the same procedure as above

$$V = \frac{(2732)(.35)(144)}{.195(3600)} = 196.1 \text{ ft/sec}$$

$$q = \frac{1}{2} \frac{(196.1)^2}{(32.2)(2732)(144)} = .00152 \text{ psi}$$

$$\Delta P = f\left(\frac{L}{D}\right)(q) = (.191)(12.4)(.00152) = .0036 \text{ psi}$$

4) Abrupt contraction into outlet port:

Outlet diameter - .667 inches
 Outlet area = .349 inches² = A_{out}
 Inlet area = A_{in} = (.2)(.38)(28) = 2.128 inches²

$$\frac{A_{out}}{A_{in}} = \frac{.349}{2.128} = .164$$

For an abrupt contraction,

$$A_2/A_1 = .164$$

$$\Delta P = \lambda q \quad (q \text{ defined in contraction})$$

and

$$\lambda = K_c C_c = (.4)(1) = .4 \quad \text{From SAE Aeronautical Report No. 23.}$$

Defining velocity head at the outlet, total flow

$$V = \frac{(1.40)(2732)}{.349} \frac{(144)}{(3600)} = 438.4 \text{ ft/sec}$$

$$q = \frac{1}{2} \frac{(438.4)^2}{(2732)(32.2)(144)} = .0076 \text{ psi}$$

$$\Delta P = .4 \times .0076 = .0030 \text{ psi}$$

Total pressure drop:

cell	- .0015
bend	- .00026
header	- .0036
contraction	- .0030
	<u>.00836</u> psi

APPENDIX B

ICE CHEST EXHAUST LINE PRESSURE DROP AND ORIFICE SIZES

SAMPLE CALCULATIONS

A. Pressure Drop Calculations (see figure 1-B for exhaust line geometry).

Exhaust line pressure drop will be calculated assuming:

1. A flow corresponding to 1250 Btu/hr heat load on the ice chest -

$$\dot{m} = \frac{Q}{H_{fg} H_2O} = \frac{1250}{1074} = 1.16 \text{ lb.hr}$$

2. A total pressure entering section 1 corresponding to saturation pressure of H_2O at $32^\circ F$, i.e. $.08854 \text{ lb/in}^2$.
3. The isentropic critical pressure ratio across the orifice.

Pressure drop calculations proceeded from the exhaust line inlet, section 1, to the exit. The purpose of these calculations was to determine that the exhaust line geometry permits the assumed flow rate at the assumed inlet and exit pressures, i.e., that the calculated pressure at the exhaust line exit is greater than the nominal ambient pressure of 400μ . Consequently the calculations do not predict an actual exhaust line ΔP (which will of course always be the inlet minus the ambient pressure) or the pressure profile in the line.

In sections of the line where the calculated Mach number is less than .3, incompressible relationships were used:

$$\Delta P = 4f \frac{L}{D} \frac{G^2}{2g\rho}$$

$$f = f(Re)$$

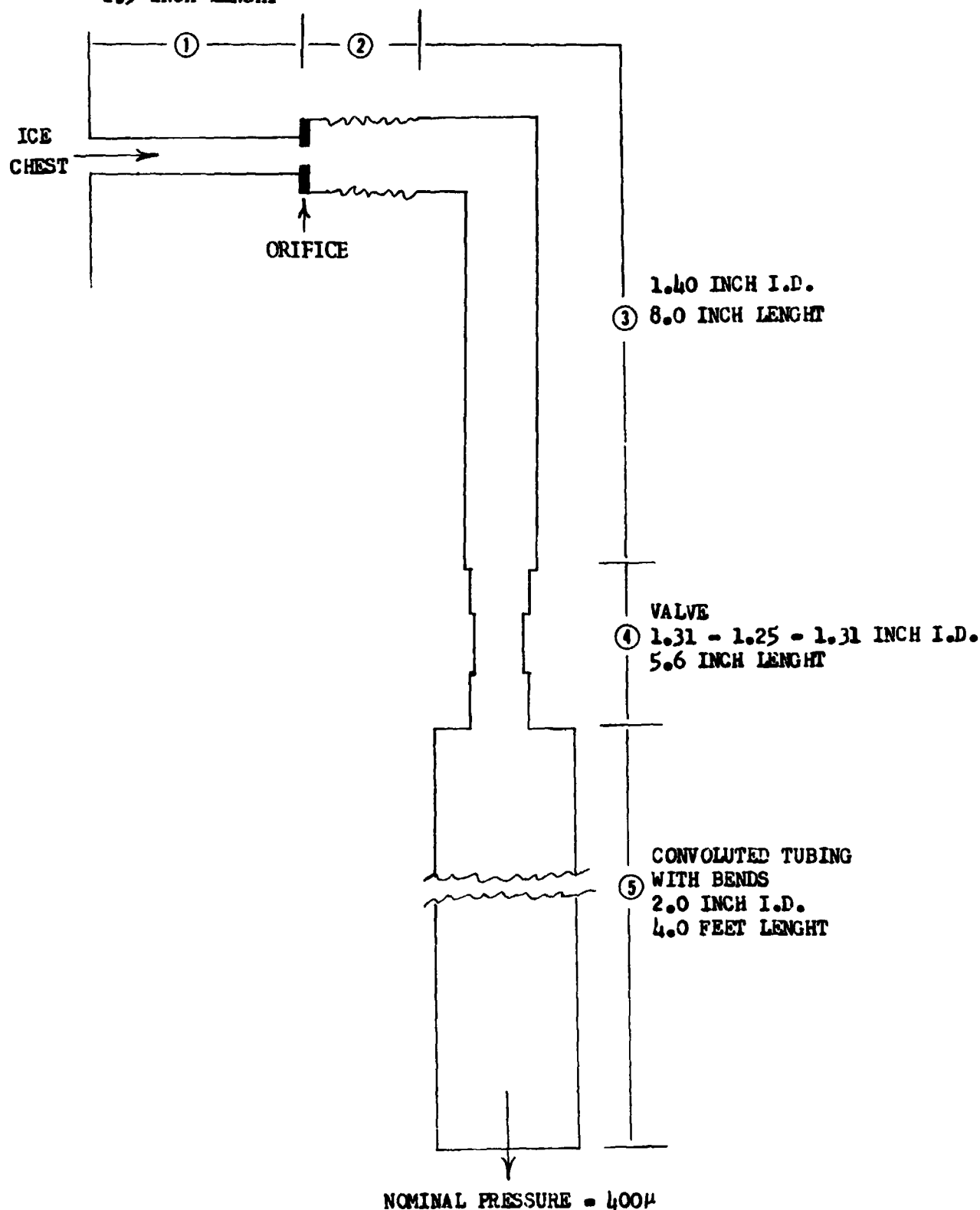
Where the Mach Number is greater than .3, tabulated Fanno line (compressible adiabatic flow with friction) relationships were used. In using these relationships, $4f \frac{L}{D}$ is evaluated and subtracted from the $4f \frac{L_{max}}{D}$ associated with the Mach Number at the beginning of the section. This gives the $4f \frac{L_{max}}{D}$ at the end of the section, and values for exit Mach Number and total pressure associated with this $4f \frac{L_{max}}{D}$ may be obtained. Figure 2-B is a plot of Fanno line relationships for $\gamma = 1.3$ (the approximate value for water vapor).

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FLEX HOSE
1.50 INCH I.D.
1.75 INCH LENGHT

.65 INCH I.D.
2.9 INCH LENGHT



ICE CHEST H₂O EXHAUST LINE

FIGURE 1-B

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FANNO LINE 7-13

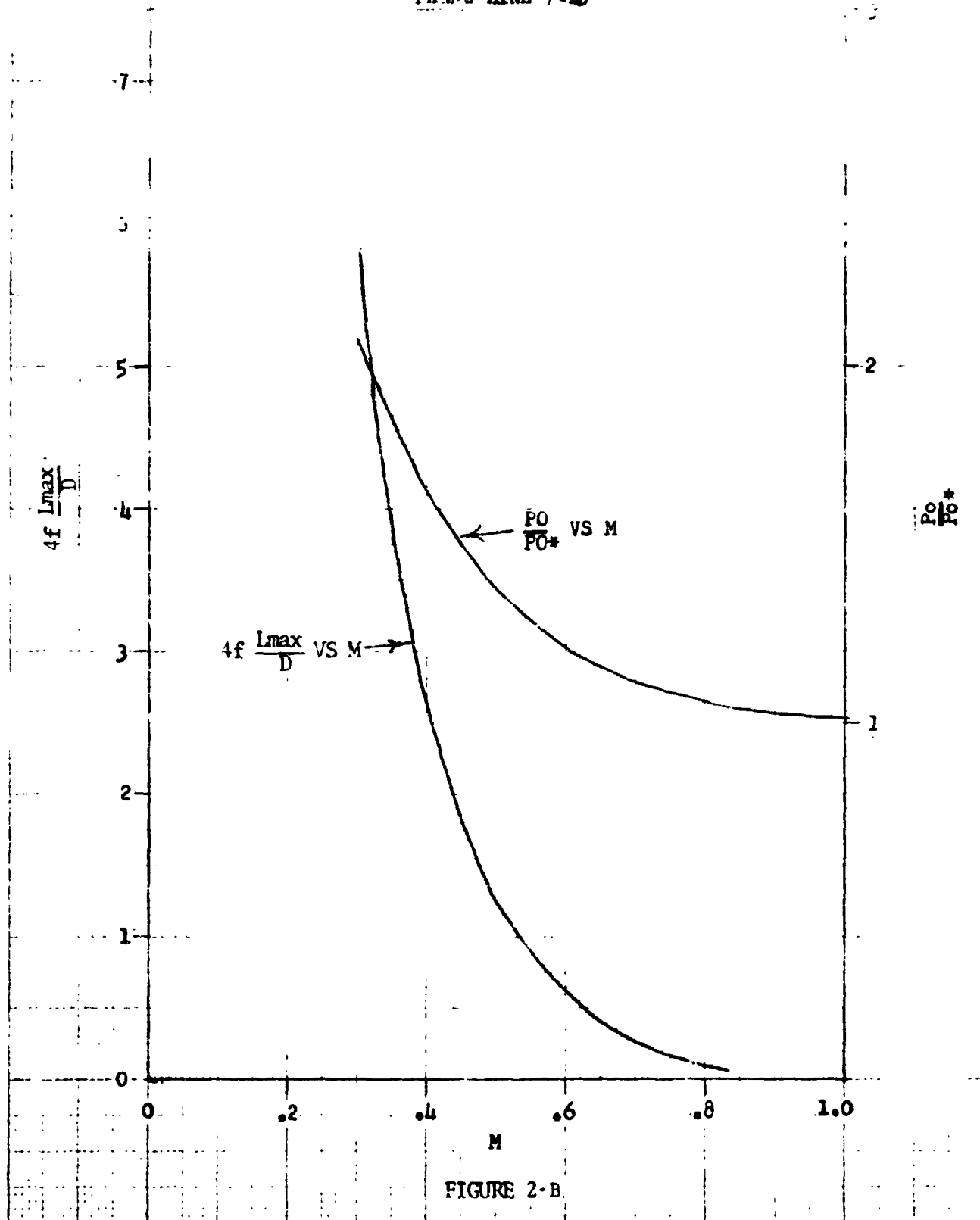


FIGURE 2-B.

Sample Calculations

I. Section 1

The total density at the inlet is:

$$\rho_o = \frac{P}{RT} = \frac{(.08854 \times 144)}{\left(\frac{1545}{18}\right) \times 492} = .000302 \text{ lb/ft}^3$$

The velocity and Mach Number which would be associated with this total density are:

$$V_o = \frac{\omega}{A \rho_o} = \frac{(1.16/3600)}{\left(\frac{\pi}{4} \times \left(\frac{.65}{12}\right)^2\right) \times .000302} = 461 \text{ ft/sec}$$

$$M_o = \frac{V_o}{C} = \frac{461}{1340} = .344$$

However, the velocity depends on the static density. Evaluating the static density at a Mach Number of .344:

$$\rho = \left(\frac{\rho}{\rho_o}\right)_{\text{isentropic}} = .942 \rho_o$$

$$\text{Therefore } M \approx \frac{.344}{.942} = .366$$

For more precise evaluations the static density and Mach Number should be further iterated upon, but the above value of Mach Number was used for the following calculations:

$$Re = \frac{d G}{\mu} = \frac{\frac{.65}{12} \times \frac{1.16}{\pi/4 \times \left(\frac{.65}{12}\right)^2}}{.021} = 1290$$

$$4f \text{ (laminar)} \approx \frac{64}{Re} = \frac{64}{1260} = .0496$$

$$\text{and } 4f \frac{L}{D} = .0496 \times \frac{2.9}{.65} = .222$$

from figure -2, the inlet $4f \frac{L_{max}}{D} \approx 3.30$

and the exit $4f \frac{L_{max}}{D} = 3.30 - .22 = 3.08$

The exit $\frac{P_o}{P_o^*} = 1.71$, and the exit total pressure is:

$$P_{o_{exit}} = \frac{\left(\frac{P_o}{P_o^*}\right)_{exit}}{\left(\frac{P_o}{P_o^*}\right)_{inlet}} \times P_{o_{inlet}} = \frac{1.71}{1.76} \times .08854 = .0858$$

II. Orifice

The isentropic critical pressure ratio for $\gamma = 1.3$ is .546. Therefore, the value for total pressure downstream of the orifice assumed was:

$$P_o = P_{o_{upstream}} \times .546 = .0858 \times .546 = .0468 \text{ lb/in}^2$$

III. Section 2

The total density at the inlet is:

$$P_o = \frac{P_o}{R T_o} = \frac{(.0468 \times 144)}{\left(\frac{1545}{18}\right) \times 492} = .00016 \text{ lb/ft}^3$$

The velocity and Mach Number at this density are:

$$V_o = \frac{w}{A \rho} = \frac{1.16/3600}{\left(\frac{\pi}{4} \times \left(\frac{1.5}{12}\right)^2\right) \times .00016} = 164 \text{ ft./sec}$$

$$M_o = \frac{164}{1340} = .122$$

At this Mach Number flow is for all practical purposes incompressible, so total density is used in the following calculations:

$$Re = \frac{\frac{1.5}{12} \times \frac{1.16}{\frac{\pi}{4} \times \left(\frac{1.5}{12}\right)^2}}{.021} = 558$$

The manufacturer of this flex hose recommends multiplying the friction factor for a smooth tube with an I D equal to the inner corrugation diameter of the flex hose by a factor of 4 to obtain an effective friction factor. Therefore:

$$4f_D = \frac{64}{558} \times \frac{1.75}{1.5} \times 4 = .535$$

$$\Delta P = 4f_D \frac{L}{2g} \frac{G^2}{K_f \times .00016} = .535 \times \frac{\left(\frac{1.16}{4 \times \left(\frac{1.5}{12} \right)^2} \right)}{K_f \times .00016} = .00025 \text{ lb/in}^2$$

$$\text{where } K_f = 2g \left(\frac{\text{sec}^2}{\text{hr}^2} \right) \left(\frac{\text{in}^2}{\text{ft}^2} \right) = 1.202 \times 10^{11}$$

The pressure at the exit to the section is therefore:

$$.0468 - .00025 \approx .0465 \text{ lb/in}^2$$

IV Sections 3, 4 and 5

At the flow rate considered, the remaining sections may be treated as incompressible, and the equations described for section 2 applied. Additional considerations in these sections are:

Section 3 - The 90° miter bend was assigned an equivalent $\frac{L}{D}$ of 59.

Section 4 - The entire valve flow length was considered to be the minimum ID.

Section 5 - Two 90° bends with r/d of 10 were assumed and assigned an equivalent $\frac{L}{D}$ of 29. A corrugation factor of 4 was applied to the friction factor in this section.

Calculated values for each section are presented in Table I-B.

B. Orifice Size Calculations

The orifice size required to pass a given amount of flow in the ice chest exhaust line cannot be calculated precisely because the flow in the tube upstream of the orifice is not in a fully developed state. Consequently two methods of calculation were used, and the smaller of the calculated values was recommended since it was desired to have an upstream pressure

LINE SECTION	REYNOLD'S NUMBER	MACH NUMBER (inlet)	EQUIV. $4f\frac{L}{D}$	INLET Po PSIA	OUTLET Po PSIA
1	1290	.366	.222	.0885	.0858
Orifice				.0858	.0468
2	558	.122	.535	.0468	.0465
3	600	.144	7.0	.0465	.0421
4	670	.195	.63	.0421	.0414
5	211	.106	60.7	.0414	.0225

CALCULATED THERMODYNAMIC CHARACTERISTICS OF

ICE CHEST H₂O EXHAUST LINE

TABLE I-B

higher rather than lower than the saturation pressure corresponding to a 32°F temperature.

The first method is from Shapiro "Dynamics & Thermodynamics of Compressible Fluid Flow". This method assumes a zero upstream velocity, and a sharp edged orifice discharge coefficient is calculated which is to be applied to the isentropic choked flow equation. For sonic velocity in the orifice vena contracta:

$$C = \frac{\pi}{\pi + \frac{2}{\left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{1}{\gamma - 1}}}} = .716$$

$$\text{isentropic } \frac{W}{A} = \frac{P_o}{\sqrt{T_o}} \sqrt{\frac{\gamma}{R} \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma + 1}{\gamma - 1}}}$$

$$= \frac{.0857}{\sqrt{492}} \sqrt{\frac{(1.3 \times 32.2)}{86} \left(\frac{2}{1.3 + 1}\right)^{\frac{1.3 + 1}{1.3 - 1}}} = .001645 \text{ lb/sec in}^2$$

Therefore, for a flow rate of 1.16 lb/hr (1250 Btu/hr heat load on the ice chest), the required orifice area:

$$A = \frac{1.16/3600}{.001645 \times .716} = .273 \text{ in}^2, \text{ and the diameter}$$

$$D = \sqrt{\frac{4}{\pi} \times .273} = .59 \text{ in}$$

The second method is from Crane, Technical Paper No. 410. The discharge coefficient is calculated based on the Reynold's number of fully developed flow upstream of the orifice, and an expansion factor, Y, is applied to the incompressible equation for flow through a sharp edged orifice. From plots in the Crane paper, pages A-20 and A-21, assuming an orifice to pipe diameter ratio of .8:

$$\text{At } Re = 1290, C \approx 1.23$$

and at the critical ΔP across the orifice,

$$Y \approx .78$$

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$$\frac{W}{A} = \frac{CY}{144} \sqrt{2g(144 \Delta P \rho)}$$

$$= \frac{1.23 \times .78}{144} \sqrt{2 \times 32.2 \times 144 \times .454 \times .000302} = .00221 \text{ lb/sec in}^2$$

The required orifice diameter is therefore:

$$D = \sqrt{\frac{4}{\pi} \frac{W}{A}} = \sqrt{\frac{4}{\pi} \frac{1.16/3600}{.00221}} = .64 \text{ in}$$

APPENDIX C

DESIGN FEASIBILITY TEST PLAN

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ICE PACK HEAT SINK SUBSYSTEM
DESIGN FEASIBILITY TEST PLAN

PREPARED UNDER CONTRACT NAS 2-7011

by

HAMILTON STANDARD
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT

for

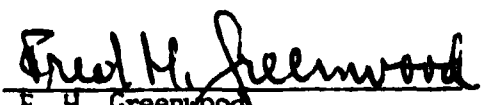
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
AMES RESEARCH CENTER
MOFFETT FIELD, CALIFORNIA

AUGUST 1972

Prepared by:


G. Roebelen
Program Engineer

Approved by:


F. H. Greenwood
Program Manager

1.0 Scope

This plan of test defines the Design Feasibility Tests to be performed by Hamilton Standard on the Ice Pack Heat Exchanger and Ice Chest per SVSK 86020 and SVSK 86016. This test program is intended to provide information necessary to verify operational performance of the Ice Pack Heat Sink Subsystem.

2.0 Test Sequence

This test program will consist of the following tests which will be performed in the sequence defined:

- 1) Evaluation of wicking device, screen device, and ice expansion compensation device.
- 2) Evaluation of ice chest operation - ambient environment.
- 3) Evaluation of ice chest operation - vacuum environment.
- 4) Evaluation of water boiler operation.

Deviation from the defined test procedure or test sequence will require prior approval of the cognizant project engineer.

3.0 Test Media

The test media for test sequence 1) and test sequence 2) will be ambient air. The test media for test sequence 3) and test sequence 4) will be vacuum.

4.0 Test Equipment

The tests for this program will be conducted as follows: Test sequence 1) and test sequence 2) conducted in HSD SSD Advanced Engineering Lab. Test sequence 3) and test sequence 4) conducted in HSD SSD Space Laboratory.

Except for the Rig 8 vacuum facility, portable equipment compatible with the test unit and the test requirements as defined by this plan of test will be utilized.

5.0 Definition of Tests

5.1 Evaluation of Wicking Device, Screen Device, and Ice Expansion Compensation Device

5.1.1 Instrumentation

<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
Scale	0-100 lbs.	± .1 lbs.
Flowmeter	0-1 GPM	± .05 GPM
Temperature Indication	0-100°F	± ¼°F

5.1.2 Test Setup

This test is performed on the bench. A freezer is required to freeze the water and a hot water source is required to melt the ice. Figure 1-C schematically illustrates the test setup.

5.1.3 Test Procedure

- With ice chest cover removed fill ice chest with distilled water, taking care to preclude any trapped air. Record volume of water added. Weigh ice chest before and after addition of water.
- Observe distribution of water in the ice cavities. Determine if there is any leakage of water into the vapor cavities.
- Refrigerate the ice chest until all the water is frozen. Observe the distribution of the ice and the adequacy of the ice expansion compensation device.
- Couple the ice chest to the LCG H/X and flow H₂O at 0.5 GPM and 90°F through the LCG H/X. Observe the melting process by probing the wicking material with a thin plastic rod. Make a sketch illustrating the plan view of the melting pattern in progressive stages.
- Repeat the freezing and thawing process 10 times and observe the results per c) and d) above.

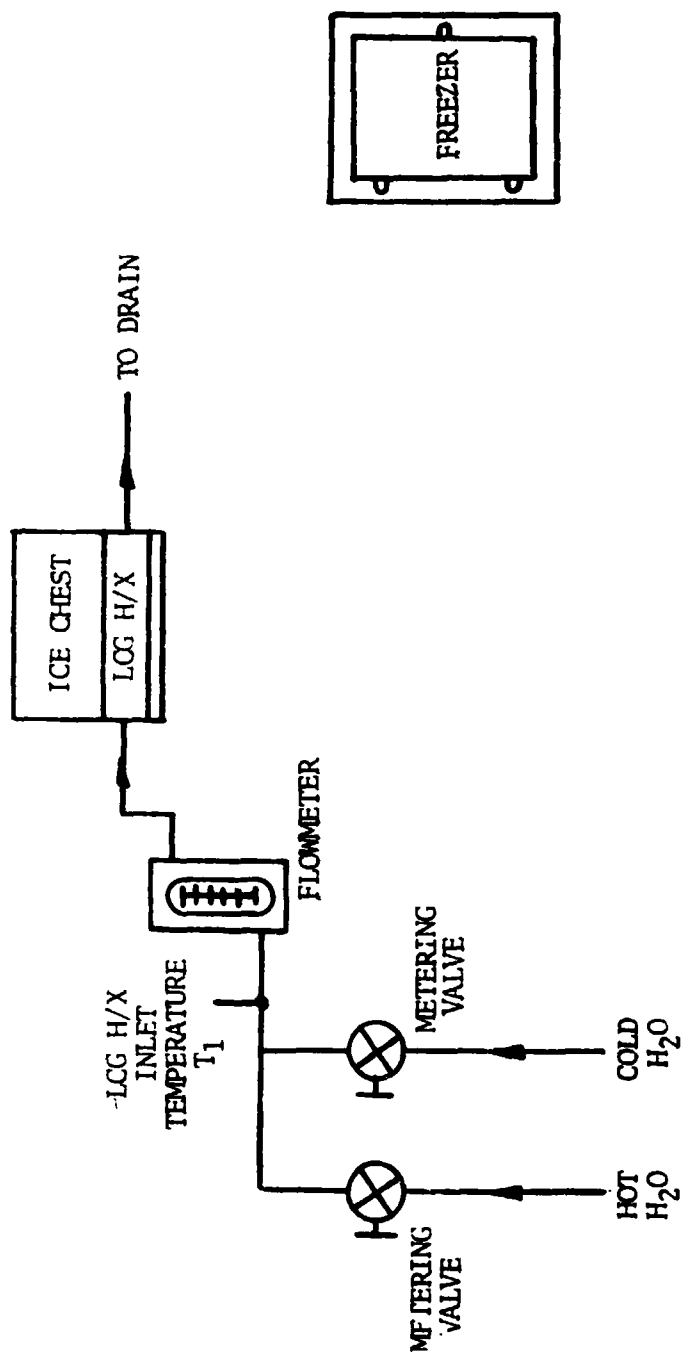
5.1.4 Test Requirements

The quantity of distilled water to be filled into the ice chest must be 15.4 lbs. minimum. Every attempt must be made to eliminate trapped air from the wick material.

5.2 Evaluation of Ice Chest Operation - Ambient Environment

5.2.1 Instrumentation

<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	24 Channel Temperature Recorder - copper/con.	0-90°F	± 0.1°F
2	Flowmeter	0-1 GPM	± .05 GPM
1	Wattmeter	0-2500 watt	± 10 watt
3	Pressure Gauge	0-30 PSIA	± .05 PSID



FEASIBILITY BENCH TEST SETUP

FIGURE 1-C

5.2.2 Test Setup

This test is performed in the SSD Advanced Engineering Lab. A freezer is required to freeze the water in the ice chest. Figure 2-C schematically illustrates the test setup.

5.2.3 Test Procedure

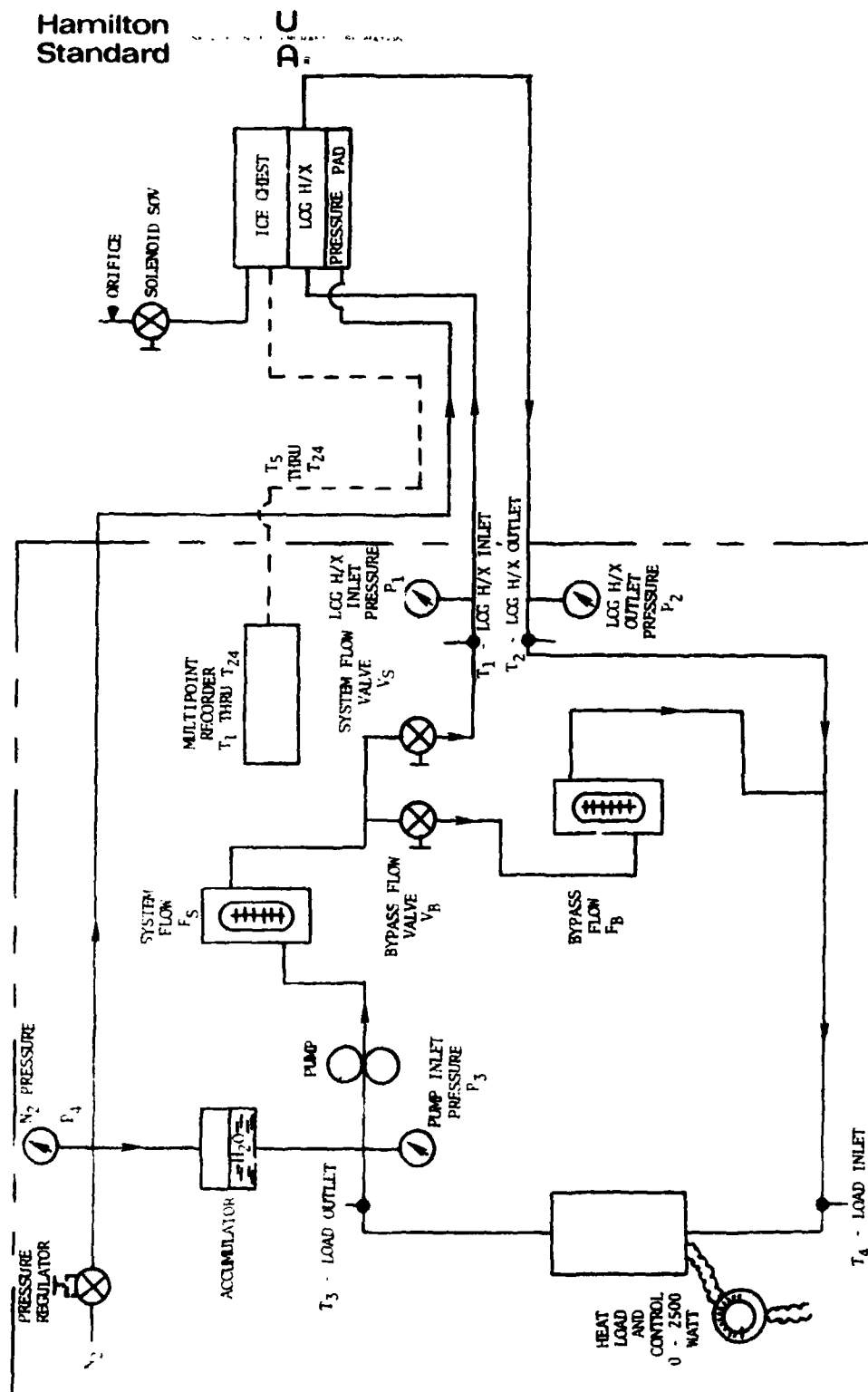
- a) Clamp an insulated frozen ice chest to the heat exchanger and install the feasibility test hardware in the laboratory bench test setup shown in figure 2.
- b) Apply a load of 3000 Btu/hr to the ice chest and by controlling heat exchanger flow, maintain the LCG inlet temperature at 50°F. Record all heater loads, temperatures, flows, and pressures per paragraph 5.2.4 of this document.
- c) Repeat steps a) and b) with a load of 1500 Btu/hr and an LCG inlet temperature of 70°F.
- d) Repeat steps a) and b) with a load of 475 Btu/hr and an LCG inlet temperature of 85°F.
- e) Repeat steps a) and b) with an LCG inlet temperature of 70°F and the following heat load profile:

Load (Btu/hr)	Time (minutes)
475	10
1000	10
2000	10
3000	10
2000	10
1000	10
2000	10
3000	Run until temperature at LCG H/X inlet cannot be stabilized.
1500	Attempt to restabilize LCG H/X inlet temperature.

5.2.4 Test Requirements

Each of the four tests must be run until the ice in the ice chest is completely exhausted. This condition can be recognized by the inability of the system to maintain relatively steady state conditions. For each run record the following steady state conditions:

1. Heater power.
2. Total system flow.
3. LCG H/X bypass flow.
4. Pump inlet pressure.
5. LCG H/X inlet pressure.
6. LCG H/X outlet pressure.



FEASIBILITY TEST SETUP IN AMBIENT

FIGURE 2-C

5.2.4 (Continued)

For each run record on log sheets at 3 min intervals, temperature vs time for each of the 24 thermocouples (Reference figure 3-C).

5.3 Evaluation of Ice Chest Operation - Vacuum Environment

5.3.1 Instrumentation

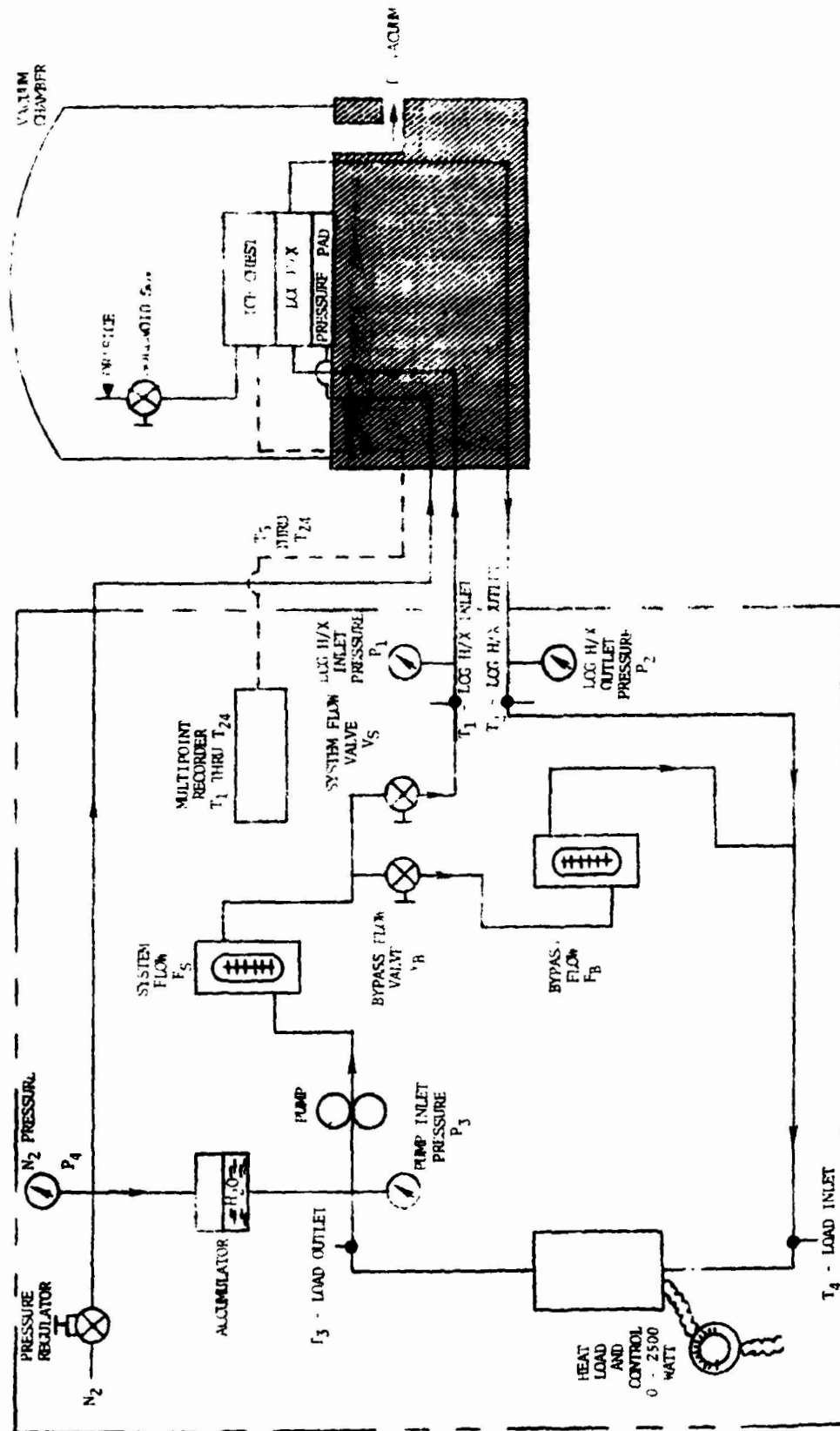
<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	24 Channel Temperature Recorder - copper/con.	0-90°F	± 0.1°F
2	Flowmeter	0-1 GPM	± .05 GPM
1	Wattmeter	0-2500 watt	± 10 watt
4	Pressure Gauge	0-30 PSIA	± .05 PSID
4	Pressure Gauge	0-15 PSIA	± .05 PSID

5.3.2 Test Setup

This test is performed on Rig 8 in the SSD Space Laboratory. Figure 4-C schematically illustrates the test setup.

5.3.3. Test Procedure

- Install the ice pack model minus the ice chest in the vacuum chamber of Rig 8 and plumb up the hardware per figure 4. Close the solenoid SOV and set the N₂ absolute pressure regulator to vacuum. Note: The vapor pressure passage pressures are not used during this portion of the test.
- Install an insulated frozen ice chest in the ice pack model. Evacuate the vacuum chamber to 500 microns or less and maintain a record of vacuum vs time for the entire run.
- Set the absolute pressure regulator to 8 PSIA and apply a load of 3000 Btu/hr to the ice chest and by controlling heat exchanger flow, maintain the LCG inlet temperature at 50°F. Record all heater loads, temperatures, flow, and pressures per paragraph 5.3.4 of this document.
- Repeat steps b) and c) with a load of 1500 Btu/hr and an LCG inlet temperature of 70°F.
- Repeat steps b) and c) with a load of 475 Btu/hr and LCG inlet temperature of 85°F.



FLASH-RIN TEST SETUP IN VACUUM
FIGURE 4 - C

5.3.3 (Continued)

- f) Repeat steps b) and c) while slowly increasing load above 3000 Btu/hr until it is no longer possible to maintain LCG inlet temperature at 50°F.
- g) Repeat step f) with absolute pressure regulator set at 5 PSIA.
- h) Repeat step f) with absolute pressure regulator set at 3 PSIA.

5.3.4 Test Requirements

Each of the tests must be run until the ice in the ice chest is completely exhausted or the system is overloaded. This condition can be recognized by the inability of the system to maintain relatively steady state conditions. For each run record the following steady state conditions:

- 1. Heater power.
- 2. Total system flow.
- 3. LCG H/X bypass flow.
- 4. Pump inlet pressure.
- 5. LCG H/X inlet pressure.
- 6. LCG H/X outlet pressure.
- 7. N₂ pressure.

For each run record on log sheets at 3 min intervals the following:

- 8. Temperature vs time for each of the 24 thermocouples.
- 9. Vacuum chamber pressure vs time.

5.4 Evaluation of Water Boiler Operation

5.4.1 Instrumentation

<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	24 Channel Temperature Recorder - copper/con.	0-90°F	± 0.1°F
2	Flowmeter	0-1 GPM	± .05 GPM
1	Wattmeter	0-2500 watt	± 10 watt
4	Pressure Guage	0-30 PSIA	± .05 PSIA
4	Pressure Guage	0- 15 PSIA	

5.4.2 Test Setup

This test is performed on Rig 8 in the SSD Space Laboratory. Figure 4-C schematically illustrates the test setup.

5.4.3 Test Procedure

- a) Install the ice pack model minus the ice chest in the vacuum chamber of Rig 8 and plumb up the hardware per figure 4-C.

5.4.3 (Continued)

Close the solenoid SOV and set the N₂ absolute pressure regulator to vacuum.

- b) Install an insulated unfrozen ice chest in the ice pack model. Evacuate the vacuum chamber and maintain a record of vacuum vs time for the entire run.
- c) Close the bypass needle valve and open the main flow needle valve.
- d) Set the absolute pressure regulator to 8 PSIA and apply a load of 3000 Btu/hr to the ice chest. When the LCG inlet temperature exceeds 85°F open for solenoid SOV. Adjust the bypass needle valve to maintain the LCG inlet temperature at 50°F. Run until LCG inlet temperature cannot be maintained. Record all heater loads, temperatures, flows, and pressures per paragraph 5.4.4 of this document.
- e) Repeat steps a), b), c) and d) with a load of 1500 Btu/hr and an LCG inlet temperature of 70°F.
- f) Repeat steps a), b), c) and d) with an initial load of 1500 Btu/hr and an LCG inlet temperature of 70°F and decrease the heat load 100 Btu/hr every 15 minutes until freeze-up occurs.
- g) Repeat d), e), and f) with an alternate orifice.

5.4.4 Test Requirements

Each of the tests must be run until the water in the wicking is depleted as much as possible. This condition can be recognized by the inability of the system maintain relatively steady state conditions. For each run record the following steady state conditions:

- | | |
|----------------------------|--------------------------------|
| 1. Heater power. | 6. LCG H/X outlet pressure. |
| 2. Total system flow. | 7. N ₂ pressure. |
| 3. LCG H/X bypass flow. | 8. Vapor passage pressure #1. |
| 4. Pump inlet pressure. | 9. Vapor passage pressure #2. |
| 5. LCG H/X inlet pressure. | 10. Vapor passage pressure #3. |
| | 11. Vapor passage pressure #4. |

For each run record on log sheets at 3 min intervals the following:

- 12. Temperature vs time for each of the 24 thermocouples.
- 13. Vacuum chamber pressure vs time.

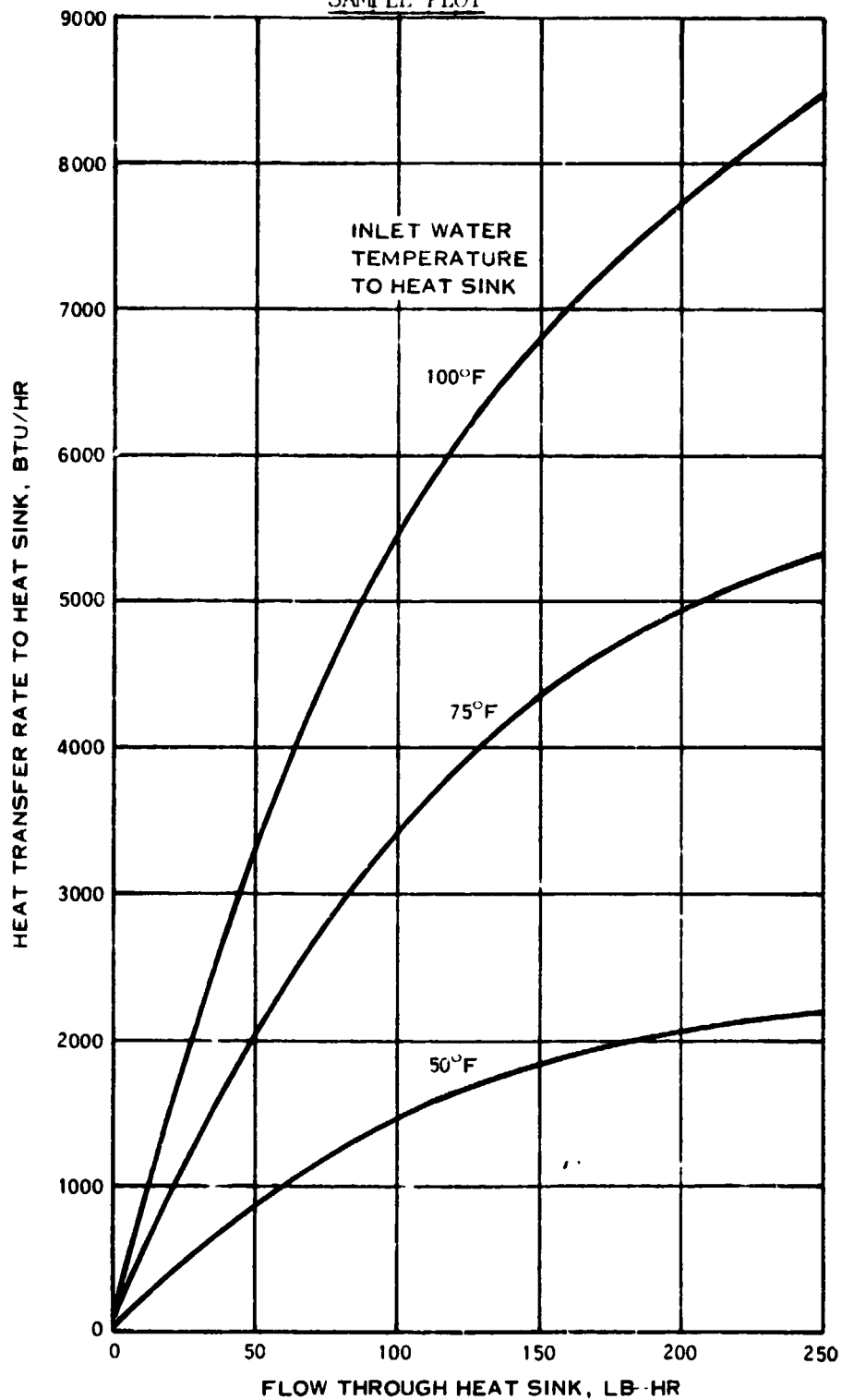
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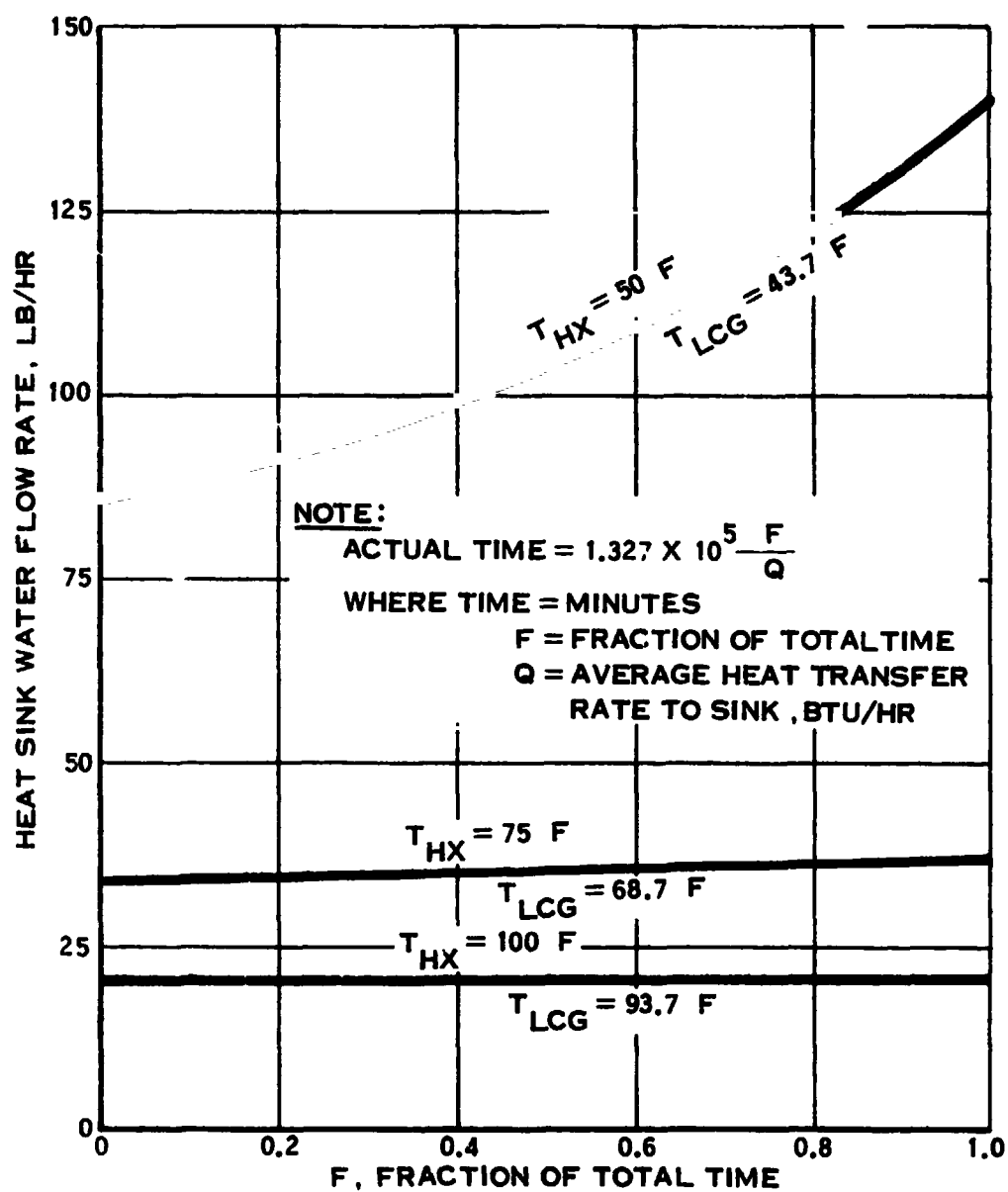
SAMPLE PLOT



HEAT SINK PERFORMANCE WITH ICE HALF MELTED

FIGURE 5-C

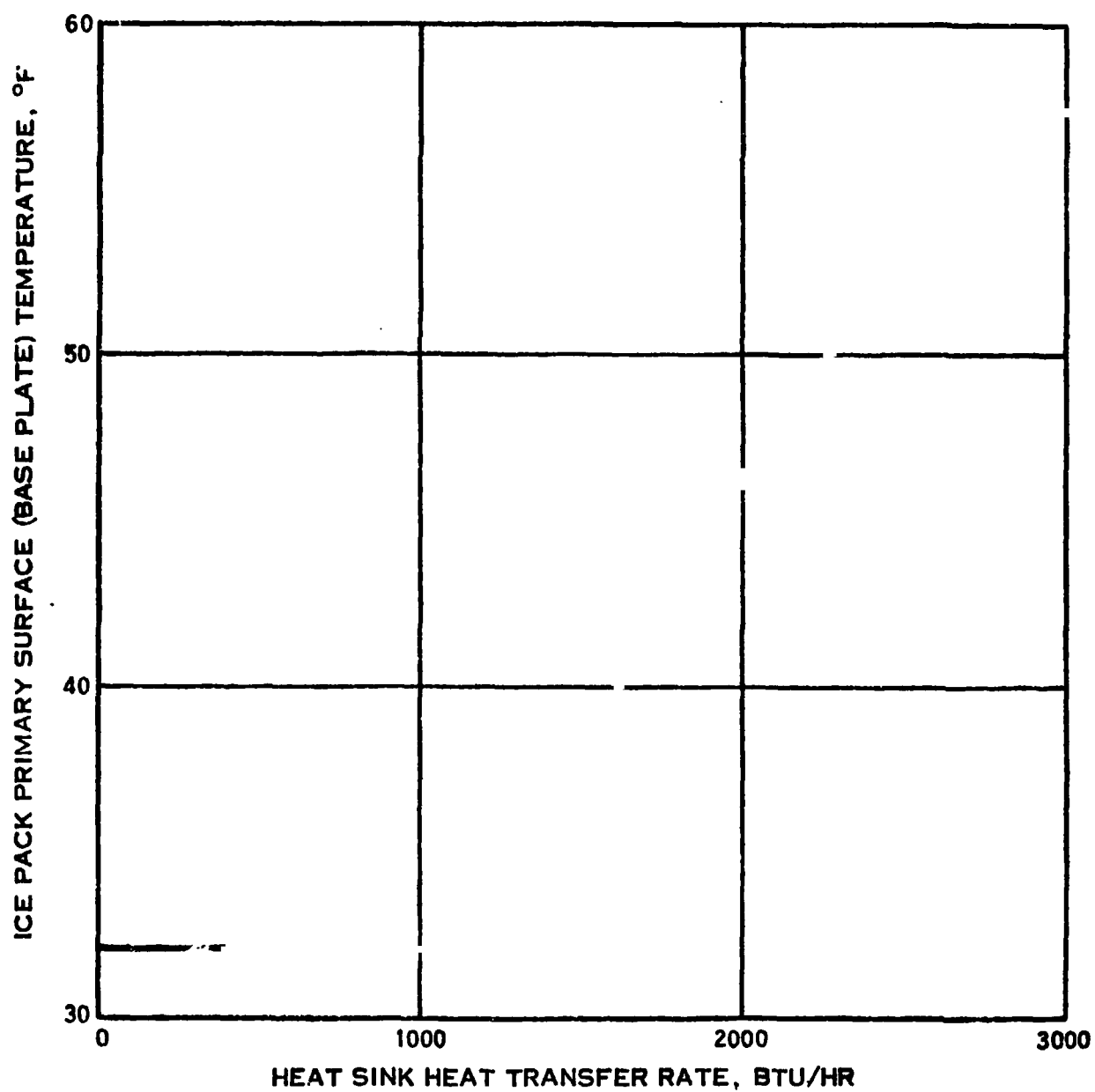
SAMPLE PLOT



PERFORMANCE TIME DEPENDENCY FOR Q SINK = 1500 BTU/HR

FIGURE 6-C

SAMPLE PLOT



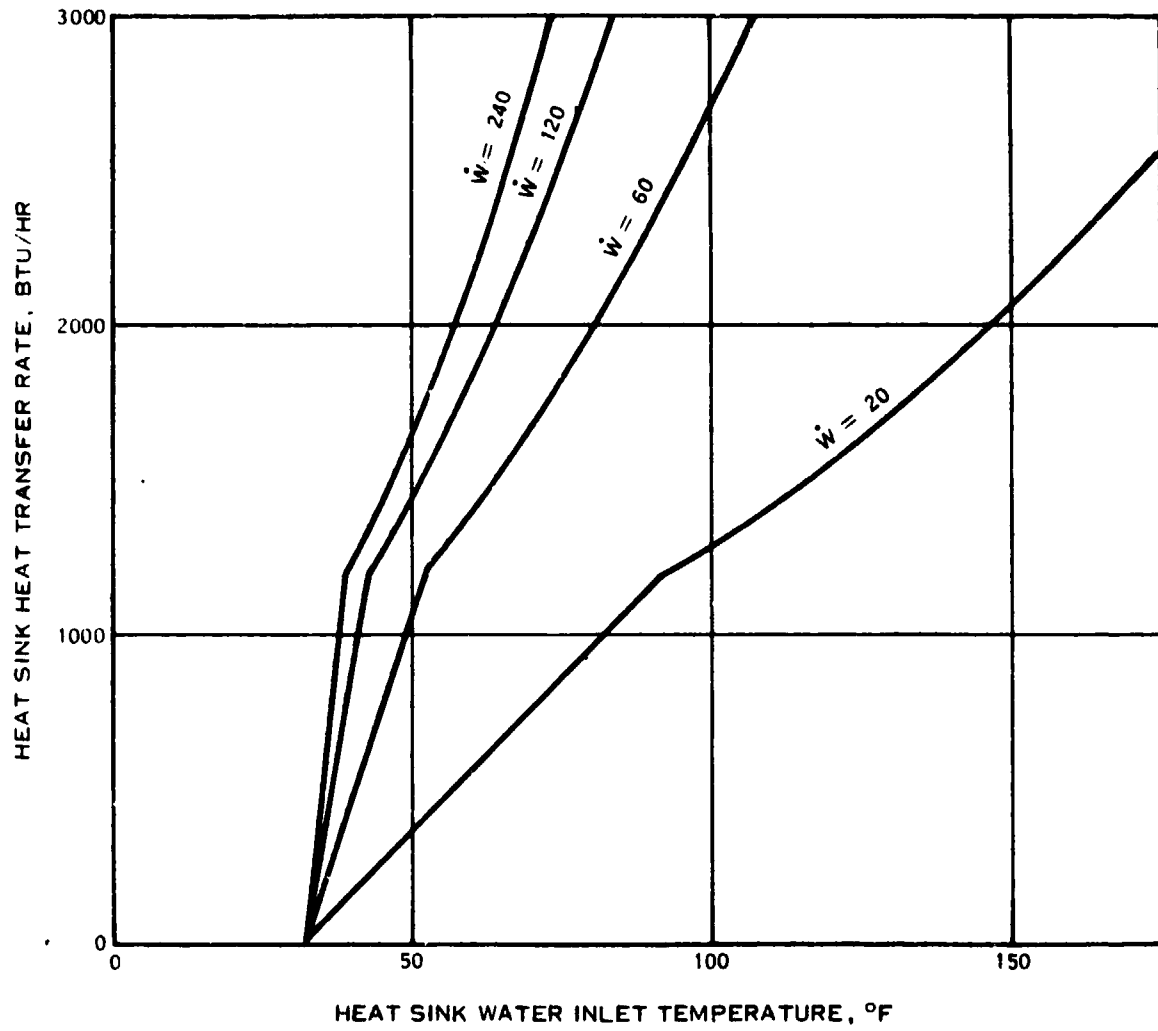
EMERGENCY PERFORMANCE

FIGURE 7-C

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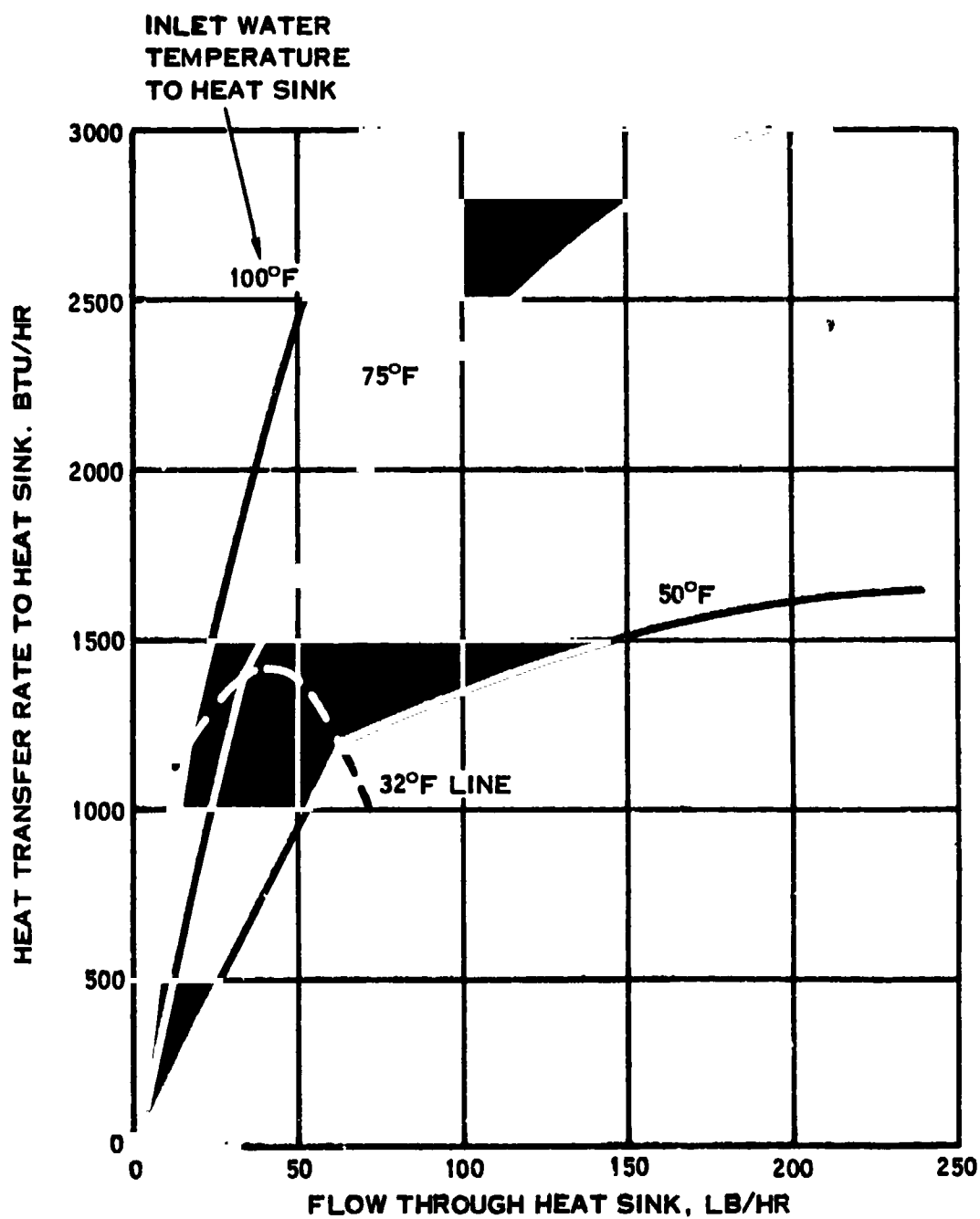
SAMPLE PLOT



EMERGENCY PERFORMANCE

FIGURE 8-C

SAMPLE PLOT



EMERGENCY MODE HEAT SINK PERFORMANCE

FIGURE 9-C

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APPENDIX D
DEVELOPMENT TEST PLAN

Hamilton
Standard



ICE PACK HEAT SINK SUBSYSTEM

DEVELOPMENT TEST PLAN

PREPARED UNDER CONTRACT NAS 2-7011

by

HAMILTON STANDARD

DIVISION OF UNITED AIRCRAFT CORPORATION

WINDSOR LOCKS, CONNECTICUT

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

AMES RESEARCH CENTER

MOFFETT FIELD, CALIFORNIA

DECEMBER 1972

Prepared by:

G. Roebelen
Program Engineer

Approved by:

F. H. Greenwood
Program Manager

1.0 Scope

This plan of test defines the Development Tests to be performed by Hamilton Standard on the Ice Pack Heat Exchanger and Ice Chest per SVSK 86020 and SVSK 86016 and the Vapor Passage Shut-off Valve and Controller per SVSK 86216. This test program is intended to provide information necessary to verify operational performance of the Ice Pack Heat Sink Subsystem.

2.0 Test Sequence

This test program will consist of the following tests which will be performed in the sequence defined:

- 1) Evaluation of ice chest/heat exchanger interface operation.
- 2) Evaluation of ice chest operation.
- 3) Evaluation of water boiler operation.
- 4) Verification of Lab Model operation.
- 5) Verification of vapor passage shut off valve and controller operation.

3.0 Test Media

The test media for all portions of this test will be vacuum.

4.0 Test Equipment

All portions of this test program will be performed in the Hamilton Standard Space Systems Department Space Laboratory. Except for the Rig 25 vacuum facility, portable equipment compatible with the test unit and the test requirements as defined by this plan of test will be utilized.

5.0 Definition of Tests

5.1 Evaluation of Ice Chest/Heat Exchanger Interface Operation

5.1.1 Instrumentation

QUANTITY	ITEM	RANGE	ACCURACY
1	24 Channel Temp Recorder	0-90°F	±.5°F
1	Copper/Con. Flowmeter	0-.1 GPM	±.005 GPM
2	Flowmeter	0-1 GPM	±.05 GPM
1	Wattmeter	0-1500 watt	±25 watt
4	Pressure Gauge	0-30 psia	±.05 psi
1	Pressure Gauge	0-20 mmHg ABS	±.05 mmHg

5.1.2 Test Setup

This test is performed on Rig 25 in the Space Systems Department Space Laboratory. Figure 1-D schematically illustrates the test setup.

5.1.3 Test Procedure

a) Install the ice pack liquid cooled garment (LCG) heat exchanger and holding fixture in the vacuum chamber of Fig 2- and install the hardware per figure 1-D. Close the vapor passage shutoff valve.

NOTE: The vapor pressure passage pressures are not used during this portion of the test.

b) Set the power supply to 27 VDC and start the Portable Flow Console pump. Close the bypass flow valve. Adjust the LCG heat exchanger flow valve to obtain a system flow of 0.5 GPM.

c) Install an insulated frozen ice chest in the holding fixture. Hook up the thermocouples. Close the vacuum chamber and evacuate to 500 microns.

d) Pressurize the pressure pad to 8 psia. When the outlet temperature of the LCG simulated load reaches 62°F, apply a continuous load of 3000 Btu/hr and adjust the bypass flow valve (V_b) and LCG heat exchanger outlet temperature at 50°F and to maintain the system flow at 0.5 GPM. (Ref.: Bypass flow is approximately 0.14 GPM.) Continue adjusting flow to maintain the LCG heat exchanger outlet temperature at 50°F and the system flow at 0.5 GPM. When the bypass flow is zero, continue running until the LCG heat exchanger temperature reaches 60°F. Shut off heat load and turn off Portable Flow Console pump. Repressurize chamber and remove ice chest.

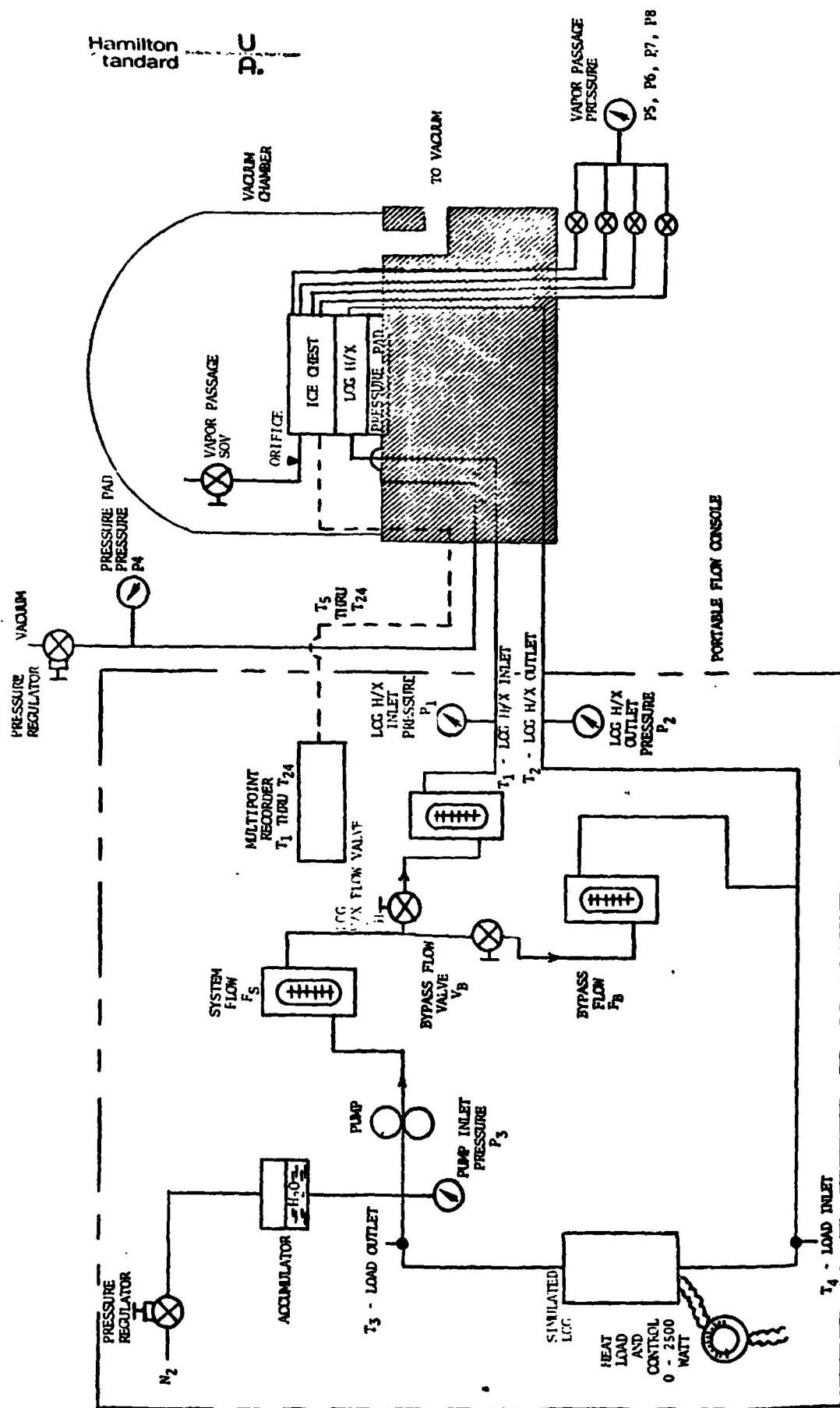
e) Repeat b), c), d) except evacuate vacuum chamber to 10^{-4} mmHg (0.1 micron) prior to pressurizing the pressure pad to 8 psia.

f) Repeat e) except pressurize the pressure pad to 5 psia.

g) Repeat e) except pressurize the pressure pad to 3 psia.

5.1.4 Test Requirements

Each of the tests must be run until the LCG heat exchanger outlet temperature reaches 60°F. For each run record the following



DEVELOPMENT TEST SETUP
FIGURE 1-D

relatively steady state conditions.

NOTE: Monitor these parameters and record any changes due to valve settings, etc.

- | | |
|----------------------------|---------------------------------------|
| 1. Heater power | 5. Pump inlet pressure |
| 2. Total system flow | 6. Pressure pad pressure |
| 3. Bypass flow | 7. LCG heat exchanger inlet pressure |
| 4. LCG heat exchanger flow | 8. LCG heat exchanger outlet pressure |

For each run record on log sheets at three (3) minute intervals the following:

9. Temperature versus time for each of the applicable thermocouples.
10. Vacuum chamber pressure versus time.

5.2 Evaluation of Ice Chest Operation

5.2.1 Instrumentation

QUANTITY	ITEM	RANGE	ACCURACY
1	24 Channel Temp Recorder	0-90°F	±.5°F
	Copper/Con.		
1	Flowmeter	0-.1 GPM	±.005 GPM
2	Flowmeter	0-1 GPM	±.05 GPM
1	Wattmeter	0-1500 watt	±25 watt
4	Pressure Gauge	0-30 psia	±.05 psi
1	Pressure Gauge	0-20 mmHg ABS	±.05 mmHg

5.2.2 Test Setup

This test is performed on Rig 25 in the Space Systems Department Space Laboratory. Figure 1-D schematically illustrates the test setup.

5.2.3 Test Procedure

- a) Install the ice pack LCG heat exchanger and holding fixture in the vacuum chamber of Rig 25 and plumb the hardware per figure 1. Close the vapor passage shutoff valve.

NOTE: The vapor pressure passage pressures are not used during this portion of the test.

- b) Set the power supply to 27 VDC and start the Portable Flow Console pump. Close the bypass flow valve. Adjust the LCG heat exchanger flow valve to obtain a system flow of 0.5 GPM.
- c) Install an insulated frozen ice chest in the holding fixture. Hook up the thermocouples. Close the vacuum chamber and evacuate to 0.1 micron.
- d) Pressurize the pressure pad to 8 psia. When the outlet temperature of the LCG simulator load reaches 62°F apply a continuous heat load of 3000 Btu/hr and adjust the bypass flow valve (V_B) and LCG heat exchanger flow valve ($V_{H/X}$) to maintain the LCG heat exchanger outlet temperature at 50°F and to maintain the system flow at 0.5 GPM. (Ref.: Bypass flow is approximately 0.14 GPM.) Continue adjusting flow to maintain the LCG heat exchanger outlet temperature at 50°F and the system flow at 0.5 GPM. When the bypass flow is zero, continue running until the LCG heat exchanger outlet temperature reaches 60°F. Shut off heat load and turn off Portable Flow Console pump. Repressurize chamber and remove ice chest.
- e) Repeat b), c), d) except heat up the system until the outlet temperature of the LCG simulator load reaches 76°F before pressurizing the pressure pad to 8 psia, apply a continuous heat load of 1500 Btu/hr, and adjust the flow control valves to maintain the LCG heat exchanger outlet temperature at 70°F. (Ref.: Bypass flow is approximately 0.42 GPM.)
- f) Repeat e) except heat up the system until the outlet temperature of the LCG simulator load reaches 87°F before pressurizing the pressure pad to 8 psia, apply a continuous heat load of 475 Btu/hr, and adjust the flow control valves to maintain the LCG heat exchanger outlet temperature at 85°F. (Ref.: LCG heat exchanger flow is approximately 0.021 GPM.)

5.2.4 Test Requirements

Each of the tests must be run until the LCG heat exchanger outlet temperature reaches 60°F. For each run record the following relatively steady state conditions.

NOTE: Monitor these parameters and record any changes due to valve settings, etc.

- | | |
|----------------------------|---------------------------------------|
| 1. Heater power | 5. Pump inlet pressure |
| 2. Total system flow | 6. Pressure pad pressure |
| 3. Bypass flow | 7. LCG heat exchanger inlet pressure |
| 4. LCG heat exchanger flow | 8. LCG heat exchanger outlet pressure |

For each run record on log sheets at three (3) minute intervals the following:

9. Temperature versus time for each of the applicable thermocouples.
10. Vacuum chamber pressure versus time.

5.3 Evaluation of Water Boiler Operation

5.3.1 Instrumentation

QUANTITY	ITEM	RANGE	ACCURACY
1	24 Channel Temp Recorder Copper/Con.	0-90°F	±.5°F
1	Flowmeter	0-.1 GPM	±.005 GPM
2	Flowmeter	0-1 GPM	±.05 GPM
1	Wattmeter	0-1500 watt	+25 watt
4	Pressure Gauge	0-30 psia	±.05 psi
1	Pressure Gauge	0-20 mmHg ABS	±.05 mmHg

5.3.2 Test Setup

This test is performed on Rig 25 in the Space Systems Department Space Laboratory. Figure 1-D schematically illustrates the test setup.

5.3.3 Test Procedure

- a) Install the ice pack LCG heat exchanger and holding fixture in the vacuum chamber of Rig 25 and plumb the hardware per figure 1. Open the vapor pressure shutoff valve.
- b) Set the power supply to 27 VDC and start the Portable Flow Console pump. Close the bypass flow valve. Adjust the LCG heat exchanger flow valve to obtain a system flow of 0.5 GPM.
- c) Install an insulated unfrozen ice chest containing the large exhaust orifice in the holding fixture. Hook up the thermocouples and vapor passage pressure lines. Close the vacuum chamber. Draw a vacuum of 500 microns on the vapor passage line and simultaneously evacuate the chamber to 10^{-4} mmHg (0.1 micron).
- d) Pressurize the pressure pad to 8 psia. When the outlet temperature of the LCG Simulated Load reaches 72°F apply a continuous heat load of 3000 Btu/hr. When the vacuum chamber pressure reaches 10^{-4} mmHg, depressurize the pressure pad for one minute and repressurize to 8 psia.

Continue to run until the temperature throughout the loop and ice chest stabilizes. Run at least thirty minutes more and shut down the system by turning off the heat load and Portable Flow Console pump. Repressurize the chamber and vapor passage and remove the ice chest.

e) Repeat b), c), and d) except apply a continuous load of 1500 Btu/hr.

f) Repeat e) except use small exhaust orifice, and start at 1500 Btu/hr with a heat load decreasing 200 Btu/hr every sixty minutes until ice chest freezing is observed.

5.3.4 Test Requirements

Each of the tests must be run until the LCG heat exchanger outlet temperature reaches 60°F. For each run record the following relatively steady state conditions.

NOTE: Monitor these parameters and record any changes due to valve settings, etc.

- | | |
|----------------------------|---------------------------------------|
| 1. Heater power | 5. Pump inlet pressure |
| 2. Total system flow | 6. Pressure pad pressure |
| 3. Bypass flow | 7. LCG heat exchanger inlet pressure |
| 4. LCG heat exchanger flow | 8. LCG heat exchanger outlet pressure |

For each run record on log sheets at three (3) minute intervals the following:

9. Temperature versus time for each of the applicable thermocouples.
10. Vacuum chamber pressure versus time.
11. Vapor passage pressures versus time.

5.4 Verification of Lab Model Operation

5.4.1 Instrumentation

QUANTITY	ITEM	RANGE	ACCURACY
1	24 Channel Temp Recorder Copper/Con.	0-90°F	±.5°F
1	Flowmeter	0-.1 GPM	±.005 GPM
2	Flowmeter	0-1 GPM	±.05 GPM
1	Wattmeter	0-1500 watt	±25 watt
4	Pressure Gauge	0-30 psia	±.05 psi
1	Pressure Gauge	0-20 mmHg ABS	±.05 mmHg

5.4.2 Test Setup

This test is performed on Rig 25 in the Space Systems Department Space Laboratory. Figure 1-D schematically illustrates the test setup.

5.4.3 Test Procedure

- a) Install the ice pack LCG heat exchanger and holding fixture in the vacuum chamber of Rig 25 and plumb the hardware per figure 1. Close the vapor passage shutoff valve.
- b) Set the power supply to 27 VDC and start the Portable Flow Console pump. Close the bypass flow valve. Adjust the LCG heat exchanger flow valve to obtain a system flow of 0.5 GPM.
- c) Install an insulated frozen ice chest containing the large exhaust orifice in the holding fixture. Hook up the thermocouples and vapor passage pressure lines. Close the vacuum chamber. Draw a vacuum of 500 microns on the vapor passage line and simultaneously evacuate the chamber to 10^{-4} mmHg (0.1 micron).
- d) When chamber has reached 10^{-4} mmHg, pressurize the pressure pad to 8 psia. When the outlet temperature of the LCG Simulated Load reaches 62°F apply a continuous heat load of 3000 Btu/hr and adjust the bypass flow valve (V_B) and the LCG heat exchanger flow valve ($V_{H/X}$) to maintain the LCG heat exchanger outlet temperature at 50°F and to maintain the system flow at 0.5 GPM. When the bypass flow is zero, continue running until the LCG heat exchanger outlet temperature reaches 60°F. At this point open the vapor passage shutoff valve and run until the temperature throughout the loop and ice chest stabilizes. Run at least thirty minutes more and shut down the system by turning off the heat load and Portable Flow Console pump. Repressurize the chamber and vapor passage and remove the ice chest.
- e) Repeat b), c), and d) except heat up the system until the outlet temperature of the LCG Simulated Load reaches 76°F before pressurizing the pressure pad to 8 psia, apply a continuous heat load of 1500 Btu/hr, and adjust the flow control valves to maintain the LCG heat exchanger outlet temperature at 70°F for as long as possible. (Ref: Bypass flow is approximately 0.42 GPM.)

5.4.4 Test Requirements

For each run record the following relatively steady state conditions.

NOTE: Monitor these parameters and record any changes due to valve settings, etc.

- | | |
|----------------------------|---------------------------------------|
| 1. Heater power | 5. Pump inlet pressure |
| 2. Total system flow | 6. Pressure pad pressure |
| 3. Bypass flow | 7. LCG heat exchanger inlet pressure |
| 4. LCG heat exchanger flow | 8. LCG heat exchanger outlet pressure |

For each run record on log sheets at three (3) minute intervals the following:

9. Temperature versus time for each of the applicable thermocouples.
10. Vacuum chamber pressure versus time.
11. Vapor passage pressures versus time.

5.5 Verification of Vapor Passage Shutoff Valve and Controller Operation

5.5.1 Instrumentation

QUANTITY	ITEM	RANGE	ACCURACY
1	L & N Thermocouple Readout	0-90°F	±.5°F

5.5.2 Test Setup

This test is performed on Rig 25 in the Space Systems Department Space Laboratory.

5.5.3 Test Procedure

- a) Hook up the vapor passage shutoff valve and controller per the schematic shown in figure 2-D.
- b) Evacuate the vacuum chamber to 10^{-4} mmHg.
- c) Open S3. Energize the 27 VDC power supply. Push S2. Observe the vacuum chamber windows. The valve should close and remain closed.
- d) Turn on the pump and chill the liquid loop to $T_4 = 50^\circ\text{F}$
- e) Close S3. The valve should remain closed.
- f) Push S1 and observe the valve. It should open and remain open. Push S2. the valve should close and remain closed.

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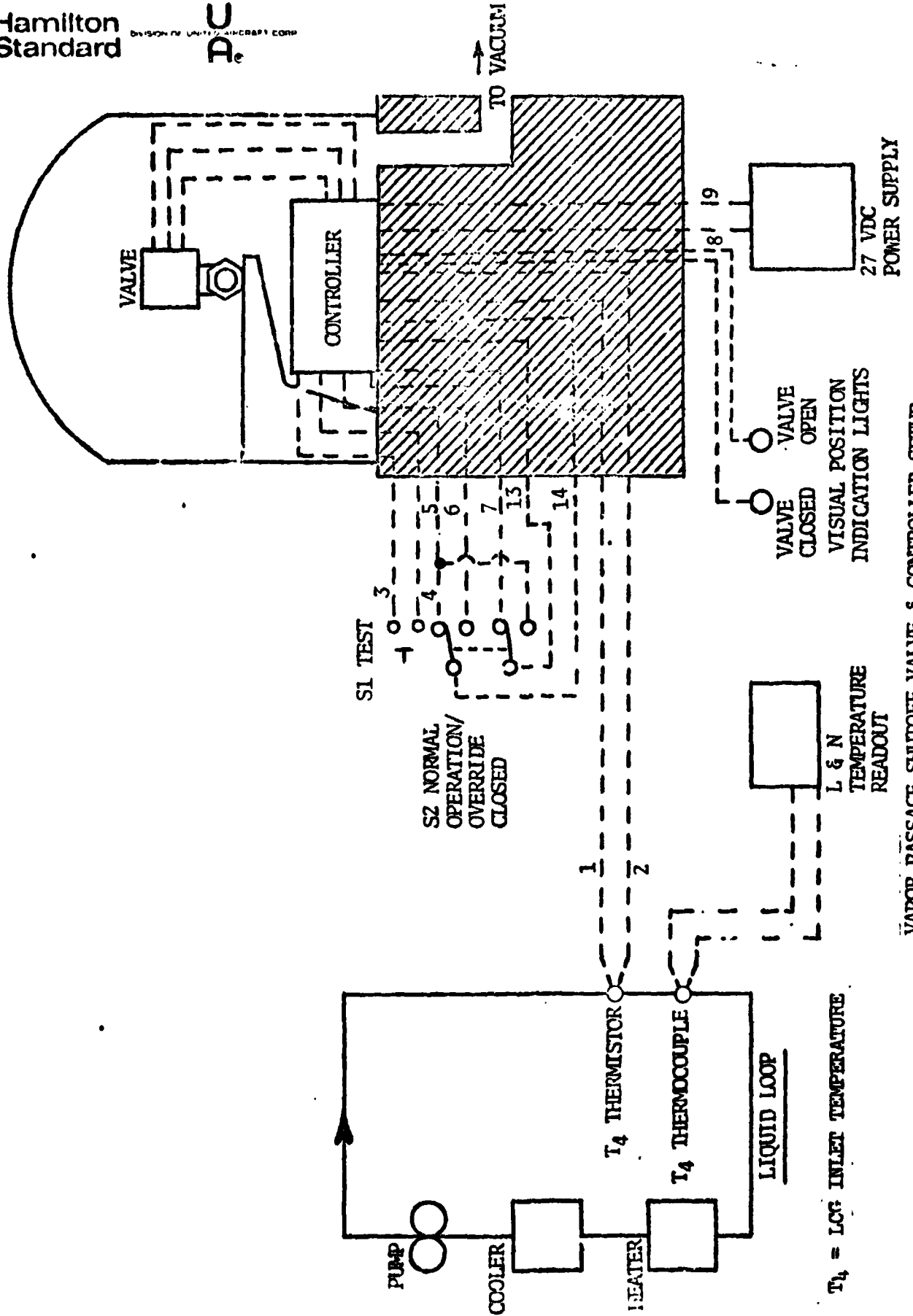


FIGURE 2-D
VAPOR PASSAGE SHUTOFF VALVE & CONTROLLER SETUP

g) Heat the liquid loop at the rate of approximately 1°F/minute. Record the temperature at which the valve opens. Cool the liquid loop to $T_4 = 50^\circ\text{F}$. The valve should not close. Push S2. The valve should close and remain closed.

h) Open S3. Heat the liquid loop to 80°F. The valve should not open.

i) Shut off the 27 VDC power supply and evacuate the vacuum chamber. End of test.

5.5.4 Test Requirements

For each operation record the operation performed, the initial valve position, the final valve position, and the value or values of T_4 at which any valve action takes place.

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CHART SAMPLES OF DATA REDUCTION

Hamilton Standard
WINN-DIXIE STORES, CONNECTICUT 06096

**SPACE & LIFE SYSTEMS LABORATORY**

LOG OF TEST

TYPE OF TEST Ice Pack Development Test	TP NO.	SHEET TEST PLAN NO	OF	DATE
TEST ENGINEER		MODEL NO		
NAME OF RIG		PART NO		
PROJECT & ENG ORDER NO		SERIAL NO		
		OPERATORS		

2017

[illegible]

NAME OF RIG

**ON VIDEO AND ORDER NO
PROJECT & ENG ORDER NO**

DATE	ON TIME	HTK POWER	TOTAL SVST. FLOW	BYPASS PUMP FLOW	LCG P/WX IP	LCG P/WX OP	NL P	VPP #1	VPP #2	VPP #3	VPP #4	AMB P	AMB T	VC P	LCG W/WX IP	LCG W/WX OP	LOT (T ₁)	LIT (T ₂)	ICE CHGTS TEMPERATURE (T ₁ - T ₂)
		WATTS (33.740 W/m ²)	CM ³ /HR (6.0 PM)	CM ³ /HR (6.0 PM)	PSIA (P ₁)	PSIA (P ₂)	PSIA (P ₃)	PSIA (P ₄)	PSIA (P ₅)	PSIA (P ₆)	PSIA (P ₇)	PSIA (P ₈)	°F (T ₁)	°F (T ₂)	°F (T ₃)	°F (T ₄)	°F (T ₅)	°F (T ₆)	°F (T ₇)

ACUMULATIVE RUNNING TIME PER TEST
(TAKE A SET OF READINGS EVERY 3 MIN.)
POWER AS SPECIFIED IN TEST PLAN
(WATTS = BTU/HR X 0.2931)

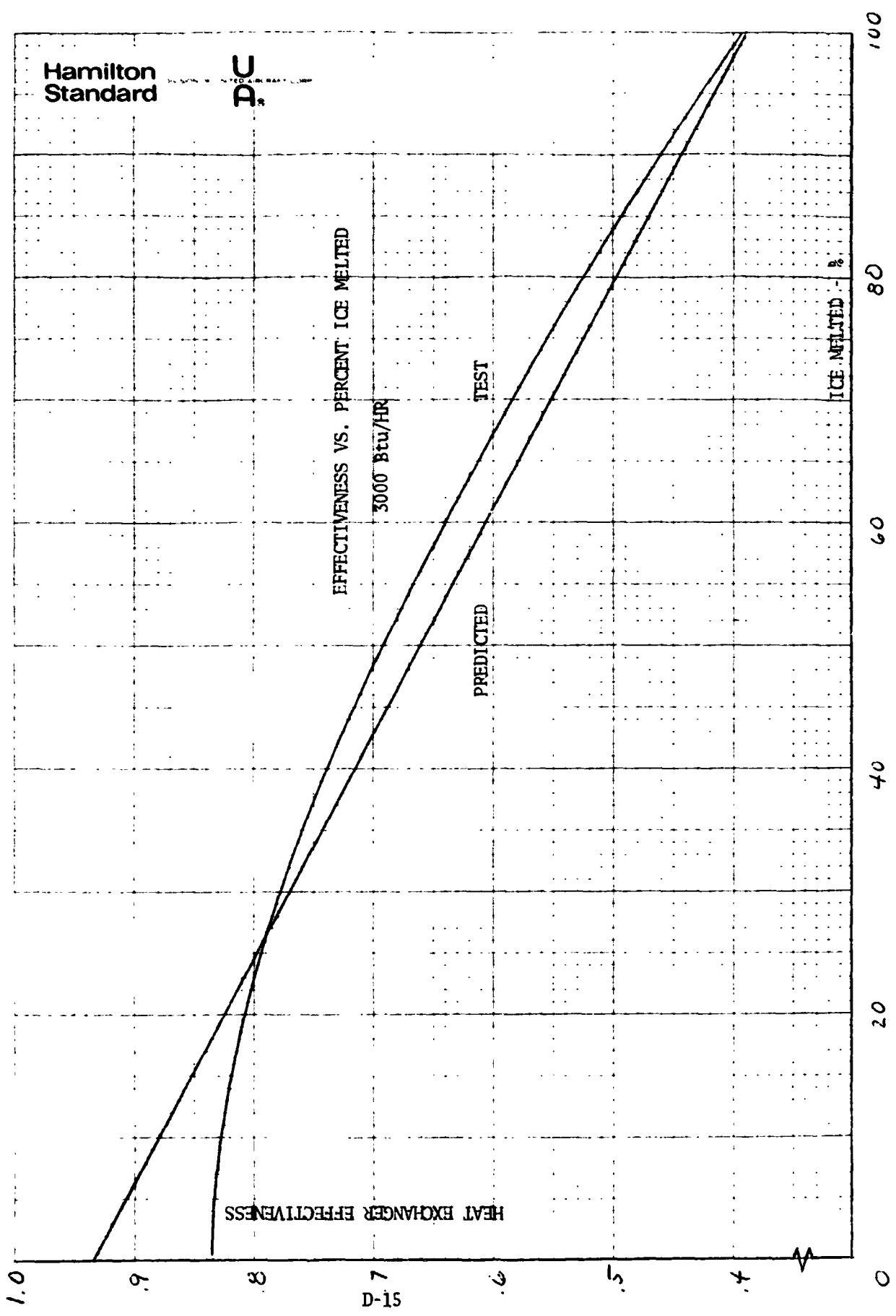
ICE CHEST TEMPERATURES - (RECORD)

12/11/2019

SAMPLE DATA SHEET

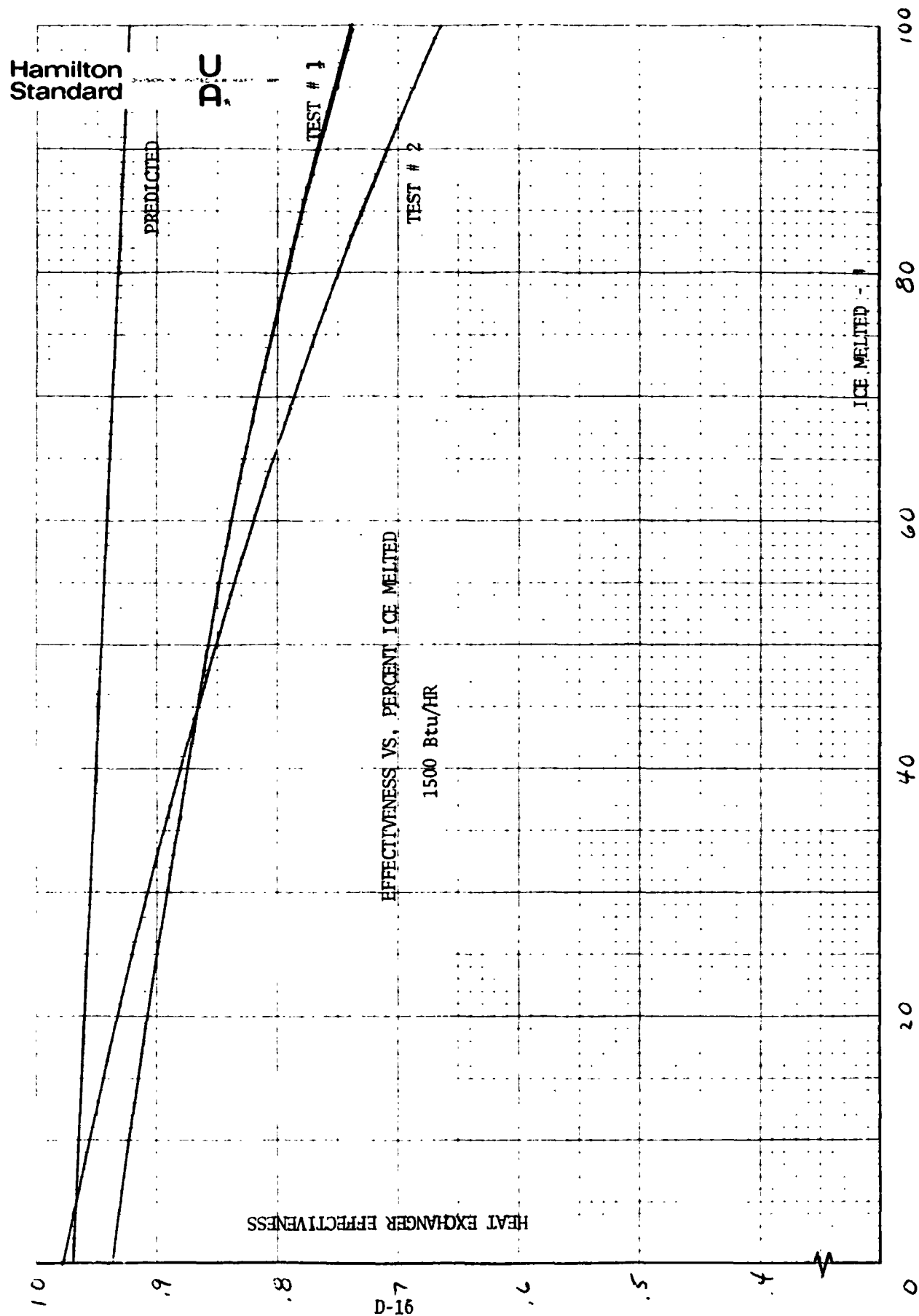
5163

NO. 340-10 DIETZEN GRAPH PAPER
10 X 10 PER INCH
EUGENE DIETZEN CO.
MADE IN U. S. A.



NO. 340 10 DIETZGEN GRAPH PAPER
10 X 10 PER INCH

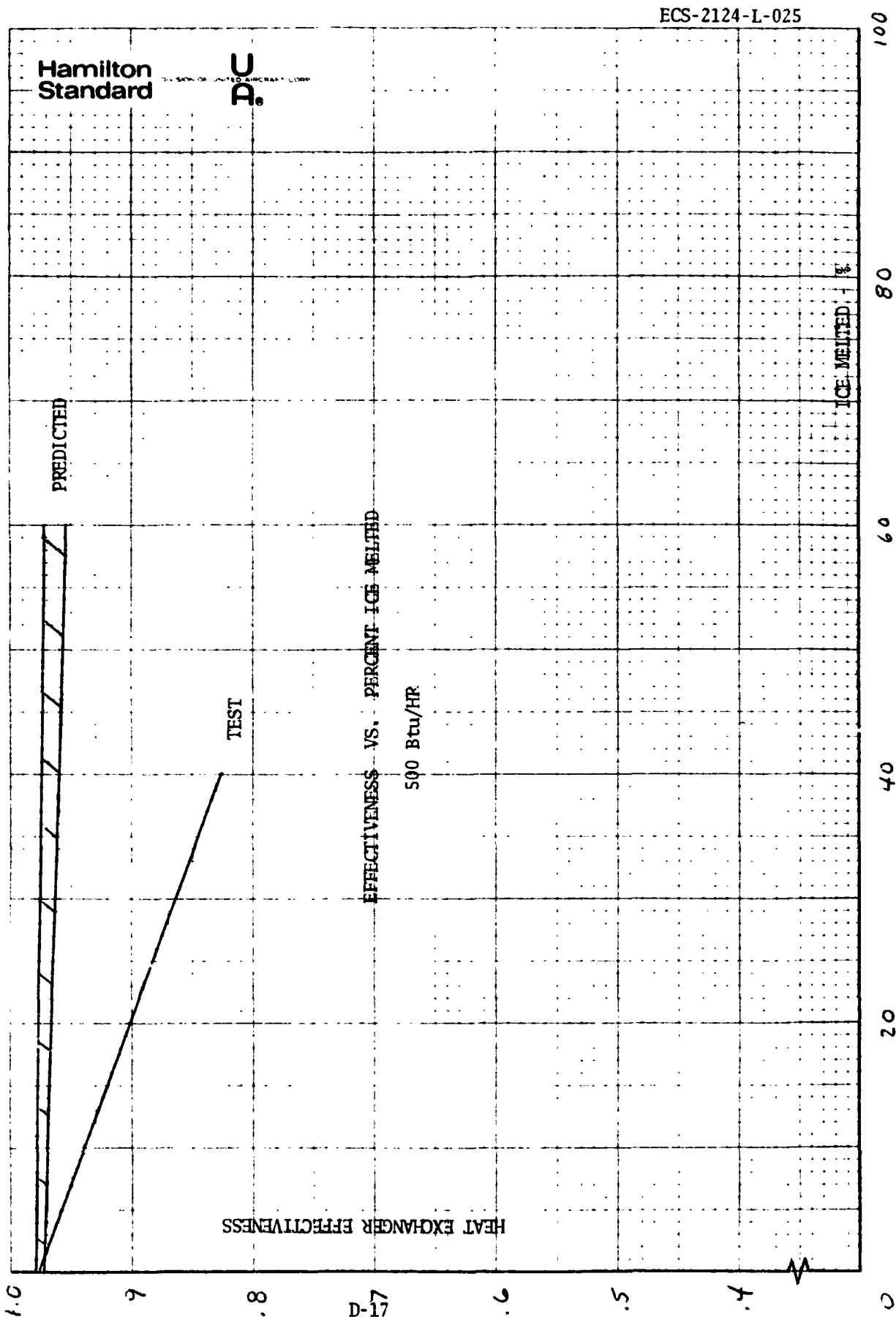
EUGENE DIETZGEN CO.
MADE IN U. S. A.



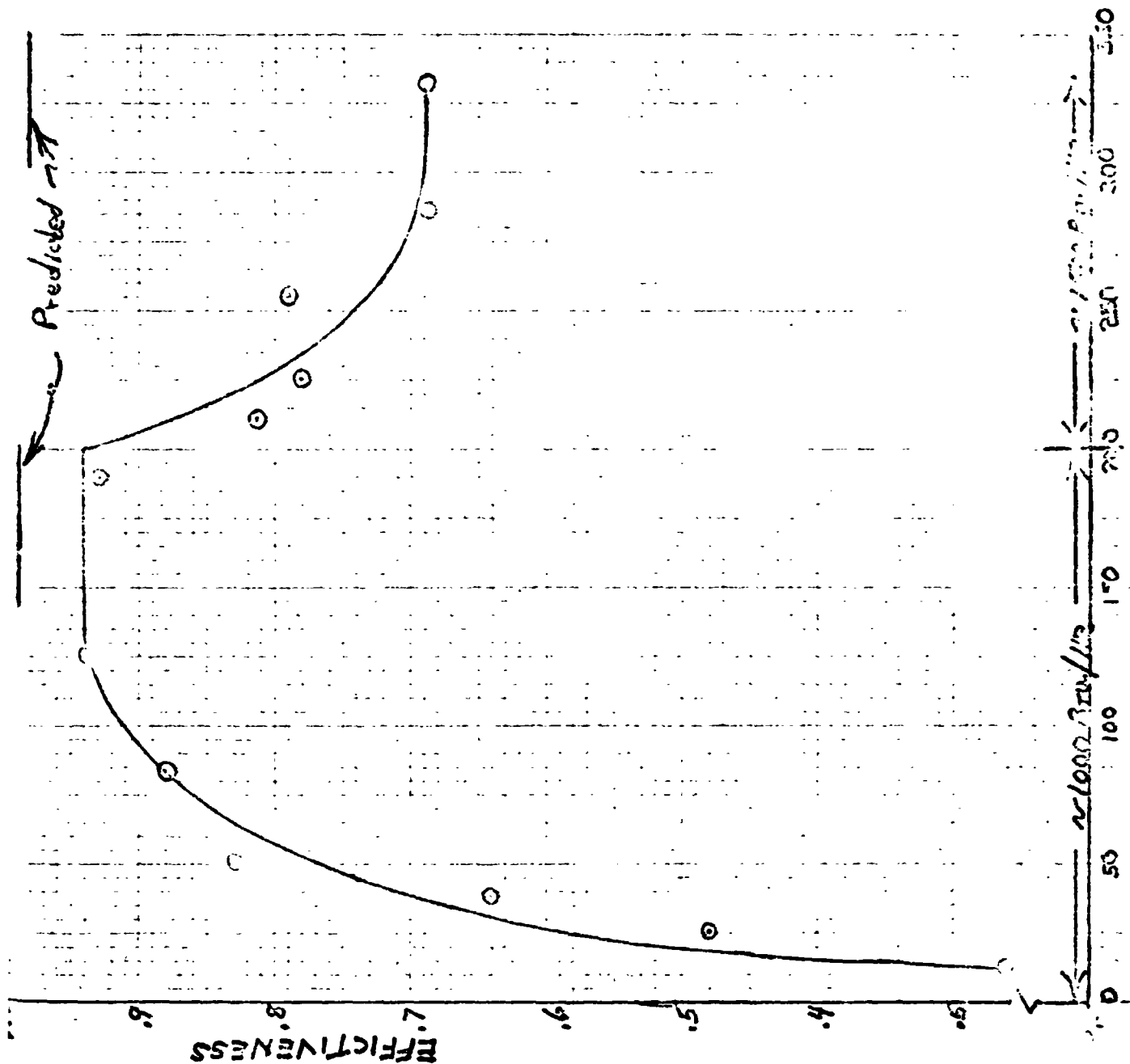
D-16

NO 340-10 DIETZGEN GRAPH PAPER
1 1/2 X 10 PER INCH

EUGENE DIETZGEN CO.
MADE IN U S A



**Hamilton
Standard**

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A



APPENDIX E

DEVELOPMENT TEST LOG SHEETS

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

REF 171141/40

Hamilton Standard
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

**U
A**

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST
ICE PACK HEAT SINK *P.S. 3.3*

TEST ENGINEER
E. ROE BELDEN

NAME OF RIG
3.5

PROJECT & ENG ORDER NO
B69-200-300A

SHEET **2** OF **2**

DATE **2-1-73**

TEST PLAN NO **ECS-3124-C-025**

MODEL NO **ICE PACK HEAT SINK**

PART NO

SERIAL NO

OPERATOR **LA 221244**

TIME	TEMP	HEATER VOLTS	HEATER AMPS	POWER WATTS	TOTAL FLOW	% GPM	FLUX	HA FLOW	HA PRESS	HA TEMP	HA PRESS	HA TEMP	HA PRESS	HA TEMP
		AC	AC	DC										
12:53	12:53	74	5.8	322	53	140	38	161	23.0	17.5	14.6	17	27	5.0
13:03	13:03	74	5.8		57	502	39.5	168	23.8	17.5	16.2	60.5	27	8.0
13:15	13:15	86	6.8	584.8	57	A	46	204	23.0	17.4	16.4	60	27	8.0
13:25	13:25	86	6.8	584.8	57		46.5	206	23	17.4	16.4	60	27	8.0
13:35	13:35	86	6.8		57		46.5	206	22	17.4	16.4	60	27	8.0
13:45	13:45	86	6.8		57		46.5	206	23	17.4	16.4	60	27	8.0
14:00	14:00	96	7.6	729.6	57		34.0	102	24.5	17.5	16.3	60.5	27	8.0
14:10	14:10	96	7.6		57		23.5	120	24.5	17.5	16.3	61	27	8.0
14:20	14:20	96	7.6		57		23.5	120	24.5	17.5	16.4	61.5	27	8.0
14:33	14:33	96	7.65	734.4	57		23	097	24.5	17.4	16.4	61.5	27	8.0
14:37	14:37	96	7.65		57		0	0	27.5	17.3	16.1	60.5	27	8.0
14:45	14:45	96	7.65		57		0	0	27.5	17.3	16.1	60.5	27	8.0

REMARKS

DEPLETED 5% & REIMPLEATED
140 ML. DRAINED FROM ICE PACK PRIOR TO START 12:53
60 ML. IN SET 1800 ML. DRAINED FROM COOLING M.D.

15003

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

WSP 178 1A 1/60

Hamilton Standard
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST
ICE PACK HEAT SINK

TEST ENGINEER
G. BOEHLER

NAME OF RIG
35

PROJECT & ENG ORDER NO
B84-300-300A

TEST PLAN NO
EC5 2124 -2-025

MODEL NO
ICE PACK HEAT SINK

PART NO.

SERIAL NO.

OPERATORS
LASZLO ZYK

SHEET 1 OF 2

DATE 2-2-73

TIME	TEST TIME	HEATER POWER		BTU	TOTAL FLOW		BTU/HR		HX FLOW	HX P101	HX P201	HX P301	HX P401	HX P501	HX P601	HX P701	HX P801	HX P901	HX P1001	HX P1101	HX P1201	HX P1301	HX P1401	HX P1501	HX P1601	HX P1701	HX P1801	HX P1901	HX P2001	HX P2101	HX P2201	HX P2301	HX P2401	HX P2501	HX P2601	HX P2701	HX P2801	HX P2901	HX P3001	HX P3101	HX P3201	HX P3301	HX P3401	HX P3501	HX P3601	HX P3701	HX P3801	HX P3901	HX P4001	HX P4101	HX P4201	HX P4301	HX P4401	HX P4501	HX P4601	HX P4701	HX P4801	HX P4901	HX P5001	HX P5101	HX P5201	HX P5301	HX P5401	HX P5501	HX P5601	HX P5701	HX P5801	HX P5901	HX P6001	HX P6101	HX P6201	HX P6301	HX P6401	HX P6501	HX P6601	HX P6701	HX P6801	HX P6901	HX P7001	HX P7101	HX P7201	HX P7301	HX P7401	HX P7501	HX P7601	HX P7701	HX P7801	HX P7901	HX P8001	HX P8101	HX P8201	HX P8301	HX P8401	HX P8501	HX P8601	HX P8701	HX P8801	HX P8901	HX P9001	HX P9101	HX P9201	HX P9301	HX P9401	HX P9501	HX P9601	HX P9701	HX P9801	HX P9901	HX P10001	HX P10101	HX P10201	HX P10301	HX P10401	HX P10501	HX P10601	HX P10701	HX P10801	HX P10901	HX P11001	HX P11101	HX P11201	HX P11301	HX P11401	HX P11501	HX P11601	HX P11701	HX P11801	HX P11901	HX P12001	HX P12101	HX P12201	HX P12301	HX P12401	HX P12501	HX P12601	HX P12701	HX P12801	HX P12901	HX P13001	HX P13101	HX P13201	HX P13301	HX P13401	HX P13501	HX P13601	HX P13701	HX P13801	HX P13901	HX P14001	HX P14101	HX P14201	HX P14301	HX P14401	HX P14501	HX P14601	HX P14701	HX P14801	HX P14901	HX P15001	HX P15101	HX P15201	HX P15301	HX P15401	HX P15501	HX P15601	HX P15701	HX P15801	HX P15901	HX P16001	HX P16101	HX P16201	HX P16301</
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6 MAY 1104	TESTED WITH MAX SANDED AT EDGES.
UNPAINED 300 ML	1130 PROB ICE PAIR AFTER TEST
315 MC	UNPAINED FROM DUT AFTER UNPAINTING ICE CRACK - 100 CMIST NOT DRIPPING

3884

DI

Hamilton Standard
WINDSOR LOCKS, CONNECTICUT 06096

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

REMARKS: ① 13:42 DECREASED GAS PRESS TO 3 PSIA
NO H₂O IN EXHAUST DUCT WATER TEST
DRAINED 4.50 MG H₂O FROM COLD TRAN

3819

Hamilton Standard
WINSTON LOCKS, CONNECTICUT 06096



DIVISION OF LABOR AND THE RATE OF CONSUMPTION

TYPE OF TEST	LEAD FILM H.X.
TEST ENGINEER	ICE PACIC HEAT SINK

SHEET 2 OF DATE 2-6-73
TEST PLAN NO FCS-2124-6-025
MODEL NO. ICE PAK MORT SINK
PART NO.
SERIAL NO
OPERATORS LASZCZYK

NAME OF RIG

25

PROJECT & ENG ORDER NO
1389-200-30

SPACE & LIFE SYSTEMS LABORATORY
LOG CF TEST

TEST TIME	H-POWER VOLT H/C	AMPS H/C	WATTS H/C	ESTD	TOTAL FLOW % GPM	PSI F/REV % GPM	PASS % GPM	HX FLOW GPM	HX P/N PSIA	HX POST PSIA	PUMP INLET PSIA	T3 OF	T4 OF	PUMP PRESS PSIA	BAG PRESS PSIA	CHECK PRESS PSIA	WATTS H/C
1245	96	7.7	739		87 502	0 0	0	502	28.0	18.6	17.7		85.5	27	5.0	8.8	2.9 X 10 ⁵
1255	96	7.7	731		87 502	0 0	0	502	27.9	18.6	17.65		85.5	27	5.0	8.8	2.7 X 10 ⁵
1305	96	7.7	731		87 502	0 0	0	502	28.3	18.3	18.4		94	27	3.0	7.5	2.6 X 10 ⁵
1315	96	7.7	731		87 502	0 0	0	502	28.4	18.65	18.7		97	27	3.0	1.4	2.6 X 10 ⁵
1325	96	7.7	731		87 502	0 0	0	502	28.4	18.6	18.6		98	27	3.0	7.45	2.6 X 10 ⁵
1335	96	7.7	734		87 502	0 0	0	502	28.2	18.4	18.4		96	27	3.0	8.25	2.5 X 10 ⁵
1345	96	7.7	731		87 502	0 0	0	502	28.2	18.2	18.2		98	27	3.0	8.45	2.5 X 10 ⁵
1400	86	6.7	583		87 502	0 0	0	502	26.3	16.7	15.9		74	27	8.0	8.2	2.5 X 10 ⁵
1410	86	6.7	593		87 502	0 0	0	502	26.4	16.7	15.9		73.5	27	8.0	8.1	2.4 X 10 ⁵
1420	86	6.7	593		87 502	0 0	0	502	26.6	16.4	15.9		72.5	27	8.0	8.05	2.5 X 10 ⁵
1430	86	6.7	593		87 502	0 0	0	502	26.5	16.7	15.9		72.5	27	8.0	8.0	2.3 X 10 ⁵
1440	86	6.7	593		87 502	0 0	0	502	26.6	17.0	16.0		72.5	27	8.0	7.7	2.0 X 10 ⁵
1450	86	6.7	593		87 502	0 0	0	502	26.6	17.1	16.1		72.5	27	8.0	7.7	2.3
1500	86	6.7	593		87 502	0 0	0	502	26.6	17.1	16.1		73	27	8.0	7.5	2.5

ITEM A-201.3

① 1957 ACQUED L.M. PUNTS 1-3 PM
② 1947 ACQUED L.M. PUNTS - 1:30 PM
ACQUED L.M. PUNTS DIRECT HILTON TEST
25:00 PM DRAMACH ENCL. OLD TRAP

381'7

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DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

**SPACE & LIFE SYSTEMS LABORATORY**

LOG OF TEST

TEST	H.A. POWER	TOTAL	BYPASS
Time	V.LTS	FLOW	FLOW
	N.C.	% GPM	% GPM
1040	105	87	67
1048	105	502	502
1055		67	54
1100		87	51
1107	105	86	49
		86	49
1122		FLOW STOPPED	
1130	105	85	49
1133	105	86	49

[illegible]

PUMP	T ₄	PUMP PRESS	BAB CMLT NAME
16.4	49	37 F.O.	PRESS PLS PR. SS
16.4	50	37 F.O.	PLS PR. SS
16.4	50	37 F.O.	PLS PR. SS
16.4	50	37 F.O.	PLS PR. SS
16.4	53	37 F.O.	PLS PR. SS
16.4	58	37 F.O.	PLS PR. SS
16.4	60	37 F.O.	PLS PR. SS

MARK

8152

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

REMARKS:

7761

SPACE & LIFE SYSTEMS LABORATORY[illegible]

10

7759

HAMILTON STANDARD
 DIVISION OF UNITED AIRCRAFT CORPORATION
 WINDSOR LOCKS, CONNECTICUT 06096
 MAP 175 1A1/86

DC

Hamilton Standard.
WINDSOR LOCKS, CONNECTICUT 06096

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TEST TIME	HEATER POWER		TOTAL FLOW	BYPASS FLOW	MAX FLOW	HX		HX PUMP	T ₁	T ₂	PUMP VDC	BAC PRESS	QUEST VMMG	
	AC VOLT	AC AMP				AC	IN							OUT
1120	75	5.85	439	0		26.1	15.8	14.8	51	27.1	27	8.0	6.7	45
1130	75	5.45	439	0		26.1	15.8	14.8	50.5	27	27	8.0	6.6	43
1145	75	5.45	439	0		26.1	15.8	14.8	50.5	27	27	8.0	6.5	35
1319 START				ORIFICE IN LINE								8.0		80
1330	43.5	3.2	137	0		26.0	15.7	14.7	53	27	27	8.0	8.75	47
1340	43.5	3.2	137	0		26.0	15.7	14.7	52.5	27	27	8.0	8.7	41
1350	43.5	3.2	137	0		26.1	15.7	14.7	52	27	27	8.0	8.65	32
1400	43.5	3.2	137	0		26.1	15.7	14.7	52	27	27	8.0	8.6	27
1410	43.5	3.2	137	0		26.1	15.7	14.7	51.5	27	27	8.0	8.5	22
1420	43.5	3.2	137	0		26.1	15.7	14.7	51.5	27	27	8.0	8.45	23
1430	43.5	3.2	137	0		26.1	15.7	14.7	51.5	27	27	8.0	8.40	22

SECRET

7755

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

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UNIVERSITY OF CONNECTICUT AERONAUTICAL CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

SPACE & LIFE SYSTEMS LABORATORY

TYPE OF TEST: **1500 BTU, 3000 BTU**
ICE PACK HEAT SINK
 TEST ENGINEER: **G. R. B. B. B.**
 NAME OF RIG: **25**
 PROJECT & ENG ORDER NO: **887-201-300A**

SHEET **1A** OF **DATE 2-27-73**
 TEST PLAN NO: **5-55-2124-1-025**
 MODEL NO: **1500 BTU HEAT SINK**
 PART NO:
 SERIAL NO:
 OPERATIONS: **1022025**

TIME	WATER FLOW		PRESS		TEMP		HUMIDITY		HEAT SINK		PUMP		MOTOR		REMARKS	
	AC	AC	AC	AC	AC	AC	AC	AC	AC	AC	AC	AC	AC			
1310	105	5.4	882		86	498	0		26.5	17.3	16.3	72	27	8.0	11.2	73
1330	105	5.4	882		86	498	0		26.5	17.3	16.3	70	27	8.0	11.05	52
1340	105	5.4	882		86	498	0		26.5	17.4	16.4	70	27	8.0	11.05	50
1350	105	5.4	882		86	498	0		26.5	17.4	16.4	70	27	8.0	11.05	46
1406	105	5.4	882		86	498	0		26.5	17.4	16.4	70	27	8.0	11.05	43
1410	DEFLECTED	1346	76.8	1 MINUTE												
1415	105	5.4	882		86	498	0		26.5	17.3	16.3	65.5	27	8.0	10.75	39
1425	105	5.4	882		86	498	0		26.5	17.4	16.4	69	27	8.0	10.9	38
1435	105	5.4	882	3000	86	498	0		26.5	17.4	16.4	69	27	8.0	11.0	37
1445	75	5.65	432	1500	86	498	0		26.4	17.0	16.0	65	27	8.0	10.0	36
1455	75	5.65	432		86	498	0		26.3	16.6	15.6	60.5	27	8.0	8.75	32
1510	75	5.65	432		86	498	0		26.3	16.4	15.4	56	27	8.0	7.85	30
1520	75	5.65	432		86	498	0		26.4	16.3	15.3	54.5	27	8.0	7.4	32
1530	75	5.65	432		86	498	0		26.4	16.3	15.3	53.5	27	8.0	7.2	32
1542	75	5.65	432		86	498	0		26.4	16.3	15.3	52.5	27	8.0	7.0	30
16030	75	5.65	432		86	498	0		25.9	16	15	56.5	27	8.0	8.1	110
1640	75	5.65	432		86	498	0		25.9	16	15	55	27	8.0	7.5	63
1650	75	5.65	432		86	498	0		26.0	15.7	14.9	54	27	8.0	7.3	55
1700	75	5.65	432		86	498	0		26.1	15.8	14.8	53	27	8.0	7.0	50
1710	75	5.65	432		86	498	0		26.1	15.8	14.8	52	27	8.0	6.85	48

REMARKS: **EVACUATED 34% OF H2O FLOW CHEST PRIOR TO TEST**
RECOMMENDED 2-28-73 - EVACUATED 83% OF H2O PRIOR TO TEST

LOG OF TEST

7757

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

WYOMARKS.

7752

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

REMARKS:

7753

HAMILTON STANDARD DIVISION OF UNITED AIRCRAFT CORPORATION WINDSOR LOCKS, CONNECTICUT 06096				LOG OF TEST				TYPE OF TEST ICE PACIC HEAT SINK				SHEET 28 OF 28 DATE 3-1-73			
PROJECT & LIFE SYSTEMS LABORATORY				PROJECT & ENG ORDER NO 889-200 300A				TEST ENGINEER G. ROBERTSON				TEST PLAN NO E-25-2124-2-0-5			
NAME OF MFG 25				NAME OF MFG 25				MODEL NO 15E PACIC HEAT SINK				PART NO.			
SERIAL NO.				SERIAL NO.				OPERATORS LASSCZAK				SERIAL NO.			
TIME	OUT	IN	TEMP	WATER	IN	OUT	HA	HA	HA	HA	HA	HA	HA	HA	HA
	1	2	3	7			25	24	27	22	29	30	31	32	33
1251	89	48	70	89			85	37	54	51	35	64	45	34	
1301	89.5	48.5	70.5	89.5			86	38	55	53	36	65	47	34	
1316	88	47	70.5	88			85	40	57	55	38	67.5	49.5	35.5	
1326	88	46.5	70.5	88			85	41	58	56	38.5	68	51	36.5	
1336	88.5	52	71	88.5			85	42	59	58	40	68.5	52.5	37.5	
1346	89	52	71	89			85.5	44	60.5	58.5	42	69.5	54	38.5	
1356	89	53	71.5	89			85.5	47	62.5	60.5	45.5	71	56	43	
1406	88.5	54.5	71.5	88.5			85.5	50.5	64	62	50	72	57.5	44	
1416	88.5	55	71.5	88.5			85.5	54	65.5	64	53.5	73	60	52.5	
1426	88.5	60.5	71.5	88.5			86	57	64	67	57	74.5	63	56.5	
1436	88.5	63	71.5	88.5			86	61	70.5	69	61	76	65	60	
1446	88.5	67	71.5	88.5			86	63.5	72.5	71.5	63	77	68	62.5	
1456	88.5	68.5	71.5	88.5			86	67	74.5	73.5	66.5	78.5	71	66	
1506	89	71	72	89			86.5	69	76.5	75.5	68	80	72.5	68	
1516	89	73	72	89			86.5	72	77.5	77	71	81	75	71	
1526	89	75	72	89			86.5	75	79.5	79.5	74.5	82	77	73.5	
1536	89	78	72.5	89			86.5	78	81.5	81.5	77.5	85	80	76.5	
1546	89	81	72.5	89			87	80.5	84	83.5	80.5	89	82	79.5	
1556	91	85	70	91			87	84	88.5	88.5	83.5	91.5	85	82.5	
1604	89	87	70	89			87	85.5	90.5	90.5	85.5	94	85	84.5	

7751

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TIME	TEST	HEATER POWER		TOTAL ELCTR %	BYPASS FLOW %	HX FLOW %	HX PUMP	T3 °F	T4 °F	PUMP PRESS	DAG PRESS	CHECK PRESS
		AC	AC									
1030		105	84	882								
1035		105	84	882								
1040		105	84	882								
1045		105	84	882								
1050		105	84	882								
1055		105	84	882								
1100		105	84	882								
1105		105	84	882								
1110		105	84	882								
1115		105	84	882								
1120		105	84	882								
1125		105	84	882								
1130		105	84	882								
1135		105	84	882								
1140		105	84	882								
1145		105	84	882								
1150		105	84	882								
1205		105	84	882								
1210		105	84	882								
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1305		105	84	882								
1310		105	84	882								
1315		105	84	882								
1320		105	84	882								
1325		105	84	882								
1330		105	84	882								
1335		105	84	882								
1340		105	84	882								
1345		105	84	882								
1350		105	84	882								
1355		105	84	882								
1400		105	84	882								

REMARKS:

(1) CHANGED LMR DGMAR
EVACUATED SC. MC H₂ - PRIOR TO TEST

7748

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

[illegible]

11 (MAR 66) LN2 11-4A1C

7743

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST

UNION PRESS. SHUTTLE VALUE AND CONTRACTION
OF PAPER
TEST ENGINEER

NAME OF RIDER

PROJECT & ENG. ORDER NO.

1359-200-361A

QUEST

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DATE

TEST PLAN NO. PCC-2124-L-025

MODEL NO.

PART NO.

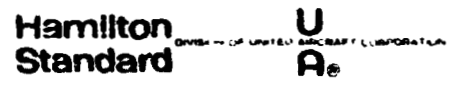
SERIAL NO.

OPTIMATIONS 61953024K

[illegible]

STEWART

8673



APPENDIX F
ACCEPTANCE TEST PLAN

ICE PACK HEAT SINK SUBSYSTEM

ACCEPTANCE TEST PLAN

Prepared Under Contract NAS 2-7011

by

HAMILTON STANDARD

DIVISION OF UNITED AIRCRAFT CORPORATION

WINDSOR LOCKS, CONNECTICUT


for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

AMES RESEARCH CENTER

MOFFETT FIELD, CALIFORNIA

Prepared by:


George Roebelen
Program Engineer

Approved by:


Fred H. Greenwood
Program Manager

1.0 Scope

This plan of test defines the acceptance tests to be performed by Hamilton Standard on the Ice Pack Heat Sink Subsystem, SVSK 86164. The acceptance test program verifies the functional operation of the Ice Pack Heat Sink Subsystem.

2.0 Test Sequence

The acceptance test program consists of the following tests, performed in the sequence defined:

- 1) Verification of functional operation of fluid circulation loop.
- 2) Verification of functional operation of vapor passage shut-off valve and controller.
- 3) Verification of battery operation.

Deviation from the test sequence or test procedure requires approval of the cognizant program engineer.

3.0 Test Environment

The test environment for all portions of this test will be room ambient temperature and pressure.

4.0 Test Equipment

All portions of this acceptance test program are performed in the Hamilton Standard Advanced Engineering Laboratory. Portable equipment compatible with the test unit and the test requirements as defined by this plan of test are utilized.

5.0 Description of Tests

5.1 Verification of Functional Operation of Fluid Circulation Loop

5.1.1 Instrumentation and Equipment

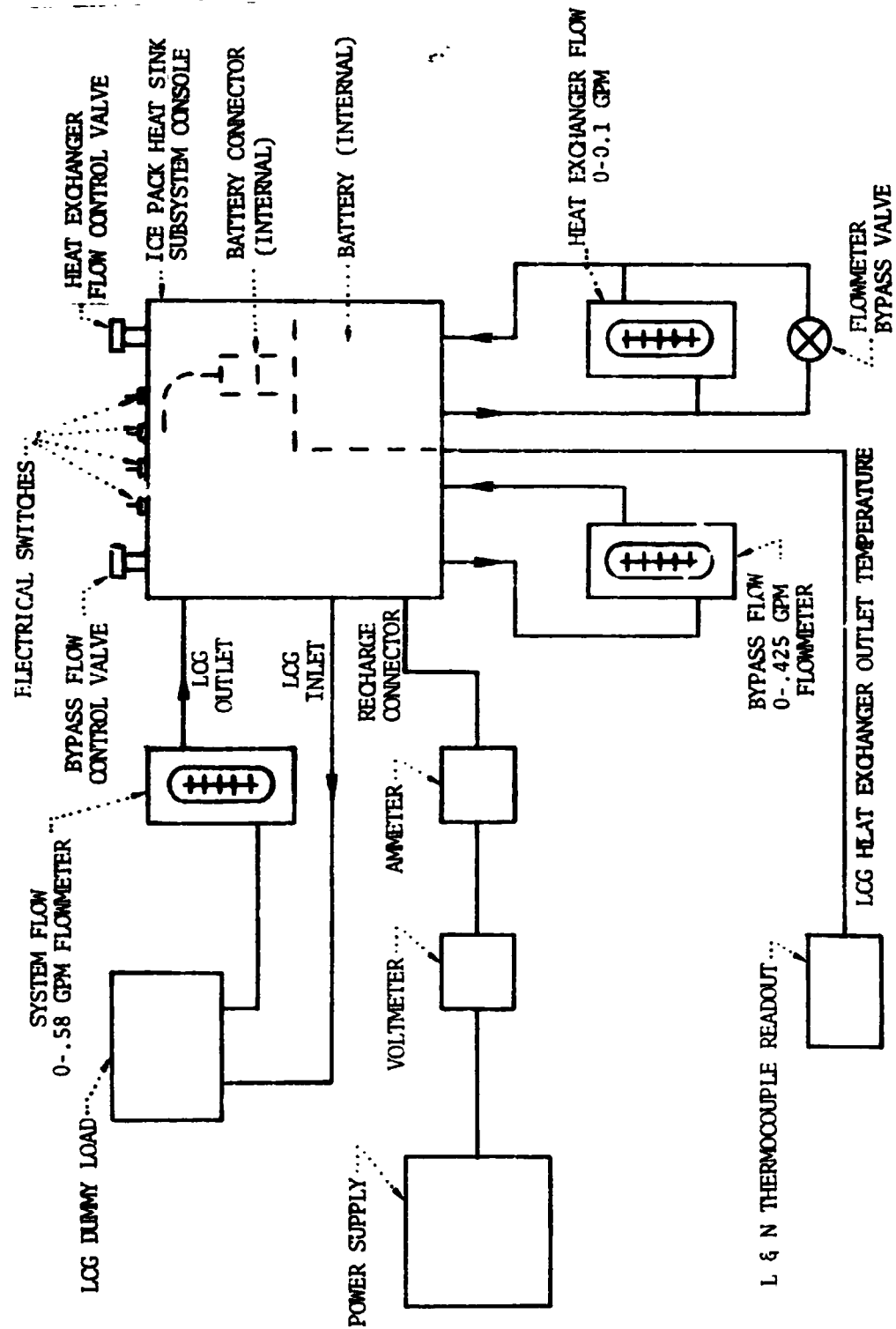
<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	DC Power Supply	0-30 VDC @ 3 ampere	± 0.5 volt
1	DC Voltmeter	0-30 VDC	± 0.05 volt
1	DC Ammeter	0-3 ampere	± 0.1 ampere
1	Flowmeter	0-0.1 gpm	± 0.005 gpm
1	Flowmeter	0-0.425 gpm	± 0.025 gpm
1	Flowmeter	0-0.58 gpm	± 0.025 gpm
1	LCG Dummy Load	3.75 psi Δ P @ 0.5 gpm	± 10%
1	L & N Thermocouple Readout	0-100°F	± 0.5°F

5.1.2 Test Setup

Figure 1-F schematically illustrates the test setup.

5.1.3 Test Procedure

- a) Disconnect the internal battery connector from the internal battery. Set the front panel electrical switches as follows:
 Main Power Switch.....OFF
 Pump Switch.....OFF
 Override Open Switch.....NEUTRAL
 Override Closed Switch.....~~OVERRIDE CLOSED~~
- b) Set up the test as shown in figure 1-F. Turn on the power supply and set the voltage at 27 VDC. Fully close the front panel Heat Exchanger Flow Control Valve. Fully open the Bypass Flow Control Valve. Close the Flowmeter Bypass Valve (external to console).
- c) Turn the main power switch to ON. Check that Normal Mode Light is ON. Position Override Closed Switch to NEUTRAL. No changes will occur provided the LCG H/X Outlet Temp. as indicated on the L & N Thermocouple Readout is less than 57°F. If the LCG H/X Outlet Temp. is greater than 57°F it is necessary to cool the LCG Heat Exchanger, using an ice bag applied directly to the heat exchanger, to bring the LCG H/X Outlet Temp. to approximately 50°F. When the LCG H/X Outlet Temp. is less than 57°F, position the Override Closed Switch to ~~OVERRIDE CLOSED~~ and return to NEUTRAL when the Normal Mode Light is ON. A
- d) Turn the Pump Switch to ON and close the Bypass Flow Control Valve until system flow is 0.5 gpm. The heat exchanger flow will read 0.021 gpm ± .005 gpm. This condition corresponds to an LCG heat load of 475 Btu/hr.
- e) Open the Flowmeter Bypass Valve (external to console). Adjust the Bypass Flow Control Valve and the Heat Exchanger Flow Control Valve to obtain a system flow of 0.50 gpm and a bypass flow of 0.42 gpm. This condition corresponds to an LCG heat load of 1500 Btu/hr.
- f) Adjust the Bypass Flow Control Valve and the Heat Exchanger Flow Control Valve to obtain a Systems Flow of 0.50 gpm and a Bypass flow of 0.14 gpm. This condition corresponds to an LCG heat load of 3000 Btu/hr.



ACCEPTANCE TEST SETUP

FIGURE 1-F

- g) Turn the Pump Switch to OFF. Turn the Main Power Switch to OFF. Shut off the power supply.

5.1.4 Test Requirements

The response of the Ice Pack Heat Sink Subsystem Console must be exactly as outlined in paragraph 5.1.3 for the unit to pass this portion of the acceptance test. Record all actions and responses.

5.2 Verification of Functional Operation of Vapor Passage Shutoff Valve and Controller

5.2.1 Instrumentation and Equipment

<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	DC Power Supply	0-30 VDC @ 3 ampere	± 0.5 volt
1	DC Voltmeter	0-30 VDC	± 0.05 volt
1	DC Ammeter	0-3 ampere	± 0.1 ampere
1	Flowmeter	0-0.1 gpm	± 0.005 gpm
1	Flowmeter	0-0.425 gpm	± 0.025 gpm
1	Flowmeter	0-0.58 gpm	± 0.025 gpm
1	LOG Dummy Load	3.75 psi ΔP @ 0.5 gpm	± 10%
1	L & N Thermocouple Readout	0-100°F	± 0.5°F

5.2.2 Test Setup

Figure #1 schematically illustrates the test setup.

5.2.3 Test Procedure

- a) Set all conditions that existed at the end of the previous test sequence.
- b) Turn on the power supply and set the voltage at 27 VDC. Turn the Main Power Switch to ON. Turn the Pump Switch to ON.
- c) Adjust the Bypass Flow Control Valve and the Heat Exchanger Flow Control Valve to obtain a Systems Flow of 0.50 gpm and a Bypass Flow of 0.14 gpm. This condition corresponds to an ICG heat load of 3000 Btu/hr.
- d) Actuate the Override Closed Switch to OVERRIDE CLOSED. When the valve actuation has been completed as indicated by the change of indicator lights, return the Override Closed Switch

A

to NEUTRAL. The valve will remain closed provided the LCG H/X Outlet Temp. is less than 57°F. If the LCG H/X Outlet Temp. is greater than 57°F it is necessary to cool the LCG Heat Exchanger, using an ice bag applied directly to the heat exchanger, to bring the LCG H/X Outlet Temp. to approximately 50°F. When the LCG H/X Outlet Temp. is less than 57°F, position the Override Closed Switch to OVERRIDE CLOSED and return to NEUTRAL when the Normal Mode Light is ON.

- e) Apply heat to the LCG Heat Exchanger using a heat gun and continue to run the unit until the valve automatically opens. The LCG H/X Outlet Temp. at the point where the valve actuates must be 60°F ± 3°F.
- f) With the LCG H/X Outlet Temp. higher than the actuation temperature, actuate the Override Closed Switch to OVERRIDE CLOSED. The valve will close and remain closed.
- g) Turn the Pump Switch to OFF. Turn the Main Power Switch to OFF. Shutoff the power supply.

5.2.4 Test Requirements

The response of the Ice Pack Heat Sink Subsystem Console must be exactly as outlined in paragraph 5.2.3 for the unit to pass this portion of the acceptance test. Record all actions and responses.

5.3 Verification of Battery Operation

5.3.1 Instrumentation & Equipment

<u>Quantity</u>	<u>Item</u>	<u>Range</u>	<u>Accuracy</u>
1	DC Power Supply	0-30 VDC @ 3 ampere	± 0.5 volt
1	DC Voltmeter	0-30 VDC	± 0.05 volt
1	DC Ammeter	0-3 ampere	± 0.1 ampere
1	Flowmeter	0-0.1 gpm	± 0.005 gpm
1	Flowmeter	0-0.425 gpm	± 0.025 gpm
1	Flowmeter	0-0.58 gpm	± 0.025 gpm
1	LCG Dummy Load	3.75 psi ΔP @ 0.5 gpm	± 10%
1	L & N Thermocouple Readout	0-100°F	± 0.5°F

5.3.2 Test Setup

Figure 1-F schematically illustrates the test setup.

5.3.3

Test Procedure

- a) Disconnect the external power supply. Set the front panel electrical switches as follows:

Main Power Switch.....OFF
Pump Switch.....OFF
Override Open Switch.....NEUTRAL
Override Closed Switch.....OVERRIDE CLOSED

Connect the internal battery connector to the internal battery. Set all other conditions that existed at the end of the previous sequence except the LCG H/X Outlet Temperature must be lowered below 57°F by applying an ice bag to the LCG Heat Exchanger.

- b) Turn the Main Power Switch to ON. Turn the Pump Switch to ON.
- c) Adjust the Bypass Flow Control Valve and the Heat Exchanger Flow Control Valve to obtain a Systems Flow of 0.50 gpm and a Bypass Flow of 0.14 gpm. This condition corresponds to an LCG heat load of 3000 Btu/hr.
- d) Actuate the Override Closed Switch to NEUTRAL. The valve will remain closed as indicated by the Normal Mode Light.
- e) Apply heat to the LCG Heat Exchanger using a heat gun and continue to run the unit until the valve automatically opens. The LCG H/X Outlet Temp. at the point where the valve actuates must be 60°F ± 3°F. A
- f) Turn the Pump Switch to OFF. Turn the Main Power Switch to OFF. Shutoff the power supply.

5.3.4

Test Requirements

The response of the Ice Pack Heat Sink Subsystem Console must be exactly as outlined in paragraph 5.3.3 for the unit to pass this portion of the acceptance test. Record all actions and responses.

Hamilton Standard DIVISION OF UNITED AIRCRAFT CORPORATION
COMMERCIAL LOCK, CONSTRUCTION EQUIPMENT

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SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST	ACCEPTANCE TEST	TEST ENGINEER
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NAME OF RIG

PROJECT & ENG. ORDER NO.

SHEET	OF	DATE
TEST PLAN NO.		
MODEL NO.		
PART NO.		
SERIAL NO.		
OPERATIONS		

[illegible]

3879

SAMPLE # DATA SHEET



APPENDIX G

ACCEPTANCE TEST LOG SHEETS

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST	ICE PACK ACCEPTANCE TEST
TEST ENGINEER	G. ROEBELEN
NAME OF RIG	ACCEPTANCE TEST RIG
PROJECT & ENG. ORDER NO.	

SHEET 1	OF 3	DATE 5/24/73
TEST PLAN NO ECS 2124-6-036A		
MODEL NO.		
PART NO.	SVSK 86164	
SERIAL NO.	001	
OPERATORS G. ROEBELEN		

① Unit set up per figure #1 of test plan:

② (75.1.3)

Internal battery disconnected.	Main Power Switch OFF.	Pump Switched OFF
Overvoltage 2 sec Switched NEUTRAL. Overvoltage Closed Switched OVERVOLTAGE SWITCHED		

Power normally out at 2-7 VDC.	H/X Flow Control Valve	fully closed.
Bypass Control Valve fully open.	Flowmeter Bypass Valve	closed.

Main Power Switch turned ON. Normal Mode Light in ON LCG H/X Outlet Temp at 65°F. Ice bag applied to LCG H/X and temperature lowered to approx 56°F. Override Choral Switch was set to OVERRIDE CLOSED and was now set to NEUTRAL. Valve remained in normal mode and Normal Mode Light remained ON.

Pump Parallel turned ON: Bypass of low Contd Valve closed until system flow is reached 0.5 gpm.
 $H \times flow = 0.021 \text{ gpm}$ (sec 0.021 \pm 0.005 gpm)

Flowmeter Bypass Valve closed. Bypass Flow Controller Valve and H/x Flow Controller Valve set to allow System Flow = 0.50 gpm and Bypass Flow = 0.42 gpm.

SECRET

Q. C. APPROVED
WMA 5/25/13
NAME DATE

9918

Hamilton Standard.
HAMILTON LOCKS, CONNECTICUT 06096

**SPACE & LIFE SYSTEMS LABORATORY**

LOG OF TEST

TYPE OF TEST	ICC PACK ACCEPTANCE TEST
1. Visual inspection of the pack for damage, discoloration, or other defects.	
2. Weighing the pack to ensure it meets the required weight specifications.	
3. Testing the pack for moisture content to ensure it is within the acceptable range.	
4. Conducting a leak test to ensure the pack is properly sealed and does not leak.	
5. Performing a stability test to ensure the pack remains intact and secure during handling and transport.	
6. Checking the pack for proper labeling and identification.	
7. Conducting a final inspection and approval by a qualified personnel.	

TEST ENGINEER

G. ROEBELIN

NAME OF RIG

ACCENTANCE TEST RIG

PROJECT & ENG ORDER NO

SECRET 2 of 3 DATE 5/24/73

TEST PLAN NO. ESC 2. 1-036A

ON 1100M

PART NO. SVSK 86164

SERIAL NO. 901

OPERATORS G. ROEBELIN

Bypass Flow Control Valve and H/K Flow Control Valve set to obtain System Flow = 0.50 gpm and Bypass Flow of 0.14 gpm

Purge Switch turned OFF, Main Purge Switch turned OFF, Purge Assembly OFF. (END # 5.1.3)

(# 5.2.3)

Purge assembly turned ON (27VDC). Main Purge Switch ON. Purge Flow Switch ON.

Bypass Flow Control Valve and H/K Flow Control Valve adjusted to obtain System Flow of 0.50 gpm and Bypass Flow of 0.14 gpm.

Override Closed Switch positioned to OVERRIDE CLOSED. Normal Mode Light is ON. LSG H/K Outlet Temp $> 60^{\circ}\text{F}$. The bag attached to H/K to bring H/K Outlet Temp to ambient. 56°F . Override closed Switch positioned to NEUTRAL. No change occurred.

Heat applied to system using heat gun. H/K Outlet Temp rising. The valve opened at H/K Outlet Temp = 60.2°F (SPEC $60 \pm 3^{\circ}\text{F}$). Normal Mode Light turns OFF. Emergency Mode Light turned ON. Override closed Switch positioned to OVERRIDE CLOSED. Valve closed and remained closed. Normal Mode Light ON. Emergency Mode Light OFF.

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NAME *2500 27013*

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Hamilton Standard
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

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SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST

ICE PACK ACCEPTANCE TEST

TEST ENGINEER

G. ROEBELEN

NAME OF RIG

ACCEPTANCE TEST RIG

PROJECT & ENG ORDER NO

SHEET 3

OF 3

DATE 5/24/73

TEST PLAN NO. EGS 2124-L-026A

MODEL NO.

PART NO. SVSKAG164

SERIAL NO. 001

OPERATOR G. ROEBLEN

Pump Switched ON. Main Power Switched OFF. Power Supply OFF.
(END OF P.S. 2.3)

5/25/73

(P.S. 2.3)

④ External Power Supply disconnected. Main Power Switched OFF. Power Supply OFF.
Override Open Switch in NEUTRAL. Override Check Switch in OVERRIDE CLOSED.
Internal Battery connected. The bag connected to Lower LCG HX Outlet
Temp to approx 56°F.

Main Power Switched ON. Power Switched ON (Power Supply at 29VDC)

Bypass Flow Control Valve and HX Flow Control Valve adjusted to obtain
System Flow = 0.50 gpm and Bypass Flow = 0.14 gpm.

Override Check Switch set to NEUTRAL. Valve remained closed.

Heat applied to system. HX Outlet Temp. rising. The valve opened at
60.2°F (SPEC 60°F ± 3°F)

Power Switched OFF. Main Power Switched OFF.
(END OF P.S. 2.3)

END OF TEST

REMARKS:

Q. C. APPROVED
NAME DATE
5/25/73

9020

Hamilton Standard

WINDSOR LOCKS, CONNECTICUT 06096

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF YEAST

ICE PACK ACCEPTANCE TEST

TEST ENGINEER

G ROEBLEN

NAME OF RIG

ACCEPTANCE TEST RIG

PROJECT & ENG ORDER NO

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303

DATE 5/24/73

TEST PLAN NO ECS 2124-6-036A

ON 72074

PART NO. SVSK 86164

SERIAL NO 002

OPERATORS G. ROUSELL

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D. C. APPROVED

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Hamilton Standard
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT 06096

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A

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST
ICE PACE ACCEPTANCE TEST

TEST ENGINEER
G. RUEBELEN

NAME OF RIG
ACCEPTANCE TEST RIG

PROJECT & ENG ORDER NO

SHEET 2 OF 3

DATE 5/24/73

TEST PLAN NO. EES 2124-L-036A

MODEL NO.

PART NO. SVSK 86164

SERIAL NO. 002

OPERATORS G. RUEBELEN

Bypass Flow Control Valve and H/X Flow Control Valve set to obtain Asystem Flow = 0.50 gpm and Bypass Flow = 0.14 gpm.

Pump Switched turned OFF. Main Power Switch turned OFF. Power On only OFF. (EAS 5.1.3)

② (EAS 5.2.3)

Power Supply turned ON (27 VDC). Main Power Switch ON. Pump Power Switch ON.

Bypass Flow Control Valve and H/X Flow Control Valve adjusted to obtain Asystem Flow of 0.50 gpm and Bypass Flow of 0.14 gpm.

Overdrive Circuit Switch positioned to ~~normal~~ OVERDRIVE. Normal Mode Light in ON. LSG H/X Outlet Temp > 60°F. The bag approached to H/X & bring H/X Outlet Temp to approx. 56°F. Overdrive Circuit Switch positioned to NEUTRAL. No change occurred.

Heat applied to system using heat gun. H/X outlet Temp rising. The valve opened & H/X Outlet Temp = 60.4°F (SPEC 60°F ± 3°F). Normal Mode Light turned OFF. Emergency Mode light turned ON. Overdrive Circuit Switch positioned to OVERDRIVE. Value closed and remained closed. Normal Mode Light ON. Emergency Mode Light OFF.

Q. C. APPROVED
DATE 5/25/73
NAME DATE

9022

Hamilton Standard
 DIVISION OF UNITED AIRCRAFT CORPORATION
 WINDSOR LOCKS, CONNECTICUT 06094

 U
 A

SPACE & LIFE SYSTEMS LABORATORY

LOG OF TEST

TYPE OF TEST

ICE PACK ACCEPTANCE TEST

TEST ENGINEER

G. RUEBELEN

NAME OF RIG

ACCEPTANCE TEST RIG

PROJECT & ENG ORDER NO.

SHEET

3 OF 3

DATE 5/24/73

TEST PLAN NO. EES 2124-L-036A

MODEL NO.

PART NO. SVSK B6164

SERIAL NO. 002

OPERATOR G. RUEBELEN

Pump Switched turned OFF. Main Pump Switched turned OFF. Power Supply OFF
 (END OF TEST 2.3)
 5/25/73 (9.5.3.3)
 (4) External Power Supply disconnected. Main Pump Switched OFF. Power Supply OFF
 Override Open Switch is NEUTRAL. Override Closed Switched in OVERCLOSURE CLOSED.
 Internal Battery connected. The bag applied to lower LCC HX Outlet
 Temp to approx 56°F.
 Main Pump Switched turned ON. Pump Switched turned ON (Power Supply at 27VDC)
 Bypass Flow Control Valve and HX Flow Control Valve adjusted to obtain
 System Flow = 0.50 gpm and Bypass Flow = 0.14 gpm.
 Override Closed Switch set to NEUTRAL. Valve remained closed.
 Heat applied to system. HX Outlet Temp rising. The valve opened at
 60.4°F (SPEC 60.1° ± 3°F)
 Pump Switched OFF. Main Pump Switched OFF. ~~Power Supply~~
 (END OF TEST 3.3)

END OF TEST

REMARKS

 Q. C. APPROVED
 Date 5/25/73
 NAME DATE

9023



APPENDIX H

SPECIFICATION FOR ICE PACK HEAT SUBSYSTEM

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

AMES RESEARCH CENTER

MOFFETT FIELD, CALIFORNIA

SPECIFICATION

FOR

ICE PACK HEAT SINK SUBSYSTEM FOR EXTRAVEHICULAR ACTIVITY

Specification No. 2-17753

December 30, 1971

1. Introduction

Future manned space exploration missions are expected to include requirements for astronaut life support equipment capable of repeated use and regeneration for many extravehicular activity (EVA) sorties. In anticipation of these requirements, NASA-Ames funded two contracts (NAS 2-6021 and NAS 2-6022) for the study of Advanced Extravehicular Protective Systems (AEPS). The purpose of these studies was to determine the most practical and promising concepts for manned space flight operations projected for the late 1970's and 1980's, and to identify areas where concentrated research would be most effective in the development of these concepts.

One regenerative concept recommended for development for astronaut cooling would utilize an ice pack as the primary heat sink for a liquid cooled garment (LCG) cooling system. In an emergency, or for extended operations, water from the melted ice pack could be evaporated directly to space or supplied to an evaporative-type LCG heat exchanger. A development and test program is, therefore, proposed for an Ice Pack Heat Sink subsystem for LCG cooling. The program is planned in two phases:

- a. Phase I will include the design of the Ice Pack Heat Sink Subsystems for operation at 0 g to 1 g in accordance with the requirements set forth below; fabrication of two functional laboratory models and performance verification in 1 g operation with revisions as required to attain design performance; limited component or concept checkout, if necessary and if approved by NASA-Ames, in an aircraft simulated 0 g environment flight; and, delivery of two (2) complete functional laboratory model subsystems with two (2) spare ice pack modules each, along with operating instructions, complete detail and assembly drawings, and specification of all mechanical components.
- b. Phase II will include the construction, assembly and checkout of a prototype subsystem, plus the test equipment and procedures required for 0 g flight tests; performance testing of the prototype Ice Pack Heat Sink Subsystems at 0 g; and, redesign, refabrication and retesting

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At 0 g and/or 1 g necessary to attain the required performance.

Phase I shall be completed before Phase II is initiated.

2. Reference Reports

- a. Advanced Extravehicular Protective System (Interim Report), Contract NAS 2-6021, NASA CR 114320.
- b. Advanced Extravehicular Protective Systems (AEPS) Study, (Interim Report), Contract NAS 2-6022, NASA CR 114321.

3. Requirements - Phase I

The Contractor shall design, develop, fabricate and test at 1 g, a functional laboratory model Ice Pack Heat Sink Subsystem. Consideration shall be given to the following:

- a. The subsystem shall be designed as a compact, portable assembly to be used as the temperature control and coolant pumping unit for an LCG for maintenance of astronaut thermal comfort during EVA either in orbit, or on the lunar surface. In normal use, excess heat in the LCG coolant will be transferred to a reusable/regenerable ice pack heat sink. For emergency operation, or for extension of EVA mission time, when all the ice has melted, water from the ice pack is to be vented directly to space or metered to an evaporator/heat exchanger which will continue to remove heat from the LCG coolant. Switchover from normal to emergency mode should be initiated automatically with an appropriate warning signal to the astronaut.
- b. The subsystem will be considered to consist, essentially, of three units:
 1. An equipment module containing the LCG coolant pump; replaceable/rechargeable batteries; LCG coolant diversion valve to control LCG inlet temperature by regulating the flow through the heat exchanger; and, sensors, valves, regulators, controls, indicators, etc., as may be required for automatic operation, manual override, and status indication.
 2. An evaporator/heat exchanger module including necessary valving, LCG coolant flow passages, and evaporator water supply passages as required, depending on the type of extended, or emergency, operation selected. For normal operation, ice packs are visualized as being held in intimate contact with a primary heat exchanger. For extended, or emergency operation, if ice pack water is metered to a sublimator-type cooler, an additional heat exchanger and appropriate flow passages and valving may be required; however, if the ice pack water is to be evaporated directly to space, the primary heat exchanger may be all that is required, other than

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the vent valves. (Note: where an "evaporator" is referred to in this Statement of Work, it is understood to include any suitable evaporator- or sublimator-type cooler).

3. Replaceable, refillable, reusable ice packs to be utilized as described in 2 above. They must be quickly and easily replaceable by the astronaut without assistance. Ice packs may be used in pairs if design and packaging considerations show a smaller pack size to be more effective than a single large pack. However, the singular term "ice pack" will continue to be used in this Statement of Work to denote the total amount of ice specified for the Ice Pack Heat Sink Subsystem.

c. Nominal specifications for the Subsystem are as follows:

1. LCG coolant flow rate, 1800-2000 gm/min (0.48 to 0.53 gal/min).
2. LCG inlet temperature range, 10 to 32 deg C (50 to 90 deg F) (for thermal comfort at maximum and minimum work rates).
3. Temperature rise through LCG, up to 7 deg C (12.6 deg F) at maximum work rate.
4. Heat rejection rates:

Average, 1600 kilojoules/hr (1500 Btu/hr)
Minimum, 500 kilojoules/hr (475 Btu/hr)
Maximum, 3200 kilojoules/hr (3000 Btu/hr)
5. LCG inlet pressure, $15.5(10^4)$ N/m² (22.5 psi), maximum.
6. Pressure drop through LCG, $2.6(10^4)$ N/m² (3.75 psi).
7. Ice pack water weight, 7 kg (15.4 lb).
8. Emergency flow to the evaporator/heat exchanger should be sufficient to provide the above heat rejection rates. A variable orifice and constant LCG coolant flow rate can be used, or, alternatively, a fixed orifice (constant evaporative rate) can be used in conjunction with variable LCG coolant flow rate through operation of the LCG coolant diversion valve.

- d. No operating time limit per ice pack is specified for the Subsystem. Time will be dependent on the actual heat rejection rate and on the practical limits of the initial and final temperatures of the ice pack (i.e., the degree of cooling below 0 deg C for the fresh ice pack, and the effective cooling range for the melted ice above 0 deg C). It is to be expected, however, that the heat sink capability of the melting ice pack remain essentially undiminished until it is completely melted.

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e. Within the constraints of the general requirements above, and other specific provisions of this contract, the Contractor is encouraged to be innovative and original in the design and fabrication of the Subsystem. No operational lifetime is specified; however, consideration should be given to maintainability, reliability, and parts replacement and repair to insure a reasonable lifetime for the delivered Subsystems, and to insure their operational integrity, if required, for support of Phase II of the development program.

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APPENDIX I

REFERENCES

REFERENCES

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