FOREWORD

This report was prepared by the Missile and Space Division of the General Electric Company, Philadelphia, Pennsylvania, under Contract No. AF 33(616)-6902 with the Life Support Systems Laboratory of the 6570th Aerospace Medical Research Laboratories. The work was performed in support of Project No. 6373, "Equipment for Life Support in Aerospace," Task No. 637303, "Nutritional Support Equipment." The study, design, and fabrication on this project were initiated 10 March 1961 and completed 10 April 1962. Contract Monitor was Mr. Courtney A. Metzger, Chief, Accommodations Section, Sustenance Branch, Life Support Systems Laboratory.

Appendix I in this report was prepared as a feasibility study for A Food Refrigeration System for Space Vehicles and constituted the basis for the design study phases of the program.

ABSTRACT

The purpose of this development project was to design, fabricate, and evaluate a food refrigeration and habitable atmosphere control system which will support a three-man crew for an extreme altitude mission of 14 days and have additional capabilities for the storage, heating and chilling of recovered water. The feasibility study and design study phases of the program indicated that a flight optimized system (i.e., a system with minimum power, weight, and volume characteristics) would be a system which utilizes a direct radiation to space concept to remove excess heat from the confines of a space vehicle. The equipment and systems were fabricated to assure their operability under the following extremes of environment: (1) cabin pressure will vary between 0.5 to 1.0 atmosphere, (2) equipment must operate in the presence of normal gravitational conditions as well as under a weightless condition and acceleration forces of up to 8 G's must be withstood.

PUBLICATION REVIEW

This technical documentary report has been reviewed and approved.

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (cont)

LIST OF TABLES

SECTION I

INTRODUCTION

Background and Objectives

The Food Refrigeration and Habitable Atmosphere Control System for Space Vehicles, or "Project COOL" as it is called throughout this report, is the result of a feasibility study, design, fabrication and test effort which originated as Project FROST, a program to develop an optimum method for the refrigeration of food aboard space vehicles.

The FROST program has been completed through its feasibility phase when the program objectives were extended to include the development of a system which had both Food Refrigeration and Habitable Atmosphere Control capabilities (Project COOL). Project FROST, A Food Refrigeration System for Space Vehicles, engineering feasibility study is included as Appendix 1 in this report.

The Project COOL phases of the program have been pursued along a course which results in the formulation of a flight optimized, integrated system for the preservation of food aboard a manned space vehicle, as well as providing a controlled habitable environment for the crew of such a vehicle. The food preservation means which are presented utilizes the liquid transport-direct radiation to space concept as recommended and studied in the Project FROST feasibility study (Appendix 1).

The habitable atmosphere maintenance system also utilizes the liquid transport-direct radiation to space concept since it also must perform, as its basic function, the removal of excess heat energy from the interior of a space vehicle; a job for which the liquid transport-direct radiation concept is optimum.

Ultimately heat rejection from a space vehicle must be accomplished purely by the mechanism of radiant heat transfer (excluding expendable refrigerants). Therefore, all methods of heat transport from the heat sources within the vehicle must convey waste heat to an effective space radiator before it can be rejected to space. Primary objectives of this program, as presented here, were to establish by comparison studies of various heat transport systems, the heat transport means which would most satisfactorily convey waste heat to the radiator for rejection to space. It was required that the space radiator be flight optimized by design study comparisons and trade-offs.

Design Description

The design of the Project COOL equipment using the liquid transport-direct radiation to space concept results in a system which is operable under zero gravity conditions and yet is optimum on the basis of a volume, weight, and power requirement. Equipment has been designed to preserve the dietary requirements and maintain an environment which is within the human comfort zone for a 3-man crew on a 14-day space mission.

The food preservation and habitable environment control (air conditioner) systems have been designed as an integrated package which is contained within a single modular envelope. The physical size and shape of the module permits its installation within the existing framework of the compact feeding console aboard the AMRL 3-man space cabin evaluator. It is designed to fit in the space formerly occupied by the thermoelectric food preservation freezer compartment, and, therefore, requires only minor structural modifications to the feeding console at installation. Furthermore, the air ducting subsystem is designed so that the air discharge from the conditioner heat exchanger plenum may be coupled at various locations along the length of the main duct trunk. This permits the feeding console to be located at various stations within the evaluator without hampering the efficiency of the air conditioning system or requiring any modifications to the COOL system, other than the extension of hydraulic and electrical interface connections.

SECTION II

SYSTEM DESIGN

1. System Performance Specifications - The following performance specifications are compiled from the contract technical exhibit and the findings of this project.

1.1 Operating Requirements

1.1.1 Food Preservation - Food quantities and containers as listed below are in accordance with the Exhibit and are sufficient to meet the requirements of a three-man crew for two weeks.

1.1.1.1 Freezer Compartment

Food container type: Metallic
Food container size: a. maximum outside height - 3.810"
b. maximum outside diameter - 2.250"
Food container shape: cylindrical
Food container number: 9 containers (cans)/man
Bread number: 53 bread items/man
Bread size: 3" x 3" x 3/4"

1.1.1.2 Chill Compartments

Food container type: Same as for freezer
Food container size: Same as for freezer
Food container shape: Same as for freezer
Food container number: 6 containers (total)
Bread number: 10 bread items (total)
Bread size: Same as for freezer
Tube type: Plastic
Tube size: 7 1/2" x 2 1/4" x 1 5/16"
Tube shape: Semi-flat cylinder
Tube number: 16 tube (total)

1.1.1.3 Food Temperature

Frozen food - Maintained at a temperature of $0^{o}F$ to $\pm 5^{o}F$
Chilled food - Maintained at a temperature of $32^{o}F$ to $40^{o}F$

1.1.1.4 Food Arrangement

Segregation - Food groups are separated and readily identifiable in the cabinet.

Protection against environment - The food is positively retained so that dynamic environments during launch and powered flight will not damage the food, containers or food storage compartment.

Ease of Removal - Containers are restrained within the cabinet and yet easily removed, one package or container at a time.

1.1.2 Habitable Environment Control

1.1.2.1 Cabin Environment

Cabin atmosphere temperature is to be maintained between 60°F and 75°F.

The humidity control devices are capable of maintaining the Cabin Relative Humidity between 30% and 50%. The habitable environment control system is capable of maintaining the temperature and humidity levels within the above limits for various cabin total pressure levels between the extremes of 7.35 psia and 14.7 psia.

1.1.2.2 Heat Sources (Maximum Values)

Sensible Load	
Cabin Equipment	13000 BTU/hr
Circulating Fan	1200 BTU/hr
Wall Load (100°F outer wall temp.)	2600 BTU/hr
Liquid Transport Pump	950 BTU/hr
3 men	900 BTU/hr
Total Sensible	18650 BTU/hr
Latent Load	
3 men	320 BTU/hr
Other Sources (cooking, etc.)	280 BTU/hr
Total Latent	600 BTU/hr

1.2 Launch, Landing and Flight Loads - (Equipment is designed to withstand the following):

Acceleration - 8 "G" peak axial and 4.5 "G" peak lateral for a total duration of ten minutes.
Deceleration - 2 "G" peak.
Shock - 25 "G" axial and 10 "G" lateral.
Vibration - 0 to 500 cps with inputs up to 5 "G" at the higher frequencies. Periods of continuous vibration for a maximum of 15 minutes.

1.3 Equipment Specifications

1.3.1 Food Storage Compartment

Insulation - The insulation choice for the compartment is polyurethane freon-filled foam.

The Food Storage Compartment is designed to fit as an integral part of the cabin structure, and encloses all compartments and controls for the radiator and interconnecting plumbing.

Door Opening - A front opening door is used with consideration given to such factors as minimizing the aisle clearance required for door swing, assuring good accessibility and ease of opening and closing.

1.3.2 Air Conditioner

The air conditioner heat exchanger is a compact airborne configuration with its weight, size and air side and fluid side pressure drops "traded off" to give an optimum design within the space available in so far as overall system weight and power requirements are concerned.

1.3.3 Humidity Control

Humidity control and water collection techniques are such that they will perform within the operating performance limits specified in a zero "g" environment as well as in the ground test facility.

1.3.4 Space Radiator

Design Considerations

Form - A fin type (two radiation surfaces) is utilized, simple in design and construction, and attached to the cabinet by means of interconnecting piping only.

Sink Available - The food storage compartment radiator is designed for equivalent conditions of an earth orbiting vehicle which is sun oriented. The sink used to simulate space is an evacuated chamber with a highly absorbent surface maintained at a temperature of -100°F (360°R). The radiator is compatible with such a test chamber. The air conditioner heat sink is a commercial condensing refrigeration unit of sufficient capacity to handle the maximum cabin heat load.

1.3.5 Heat Transport Fluid System

Fluid - Selected on basis of heat transport properties, ability to remain liquid through the operating temperature range (including a factor of safety against freezing at low temperatures), noncorrosive nature, and relative safety (non-toxic nor readily inflammable).

Substitution of Radiator - The refrigeration system is designed such that the radiator can be readily removed from the system and simulated operation produced by the attachment of alternate piping from which the heat can be removed by conventional means.

1.3.6 Power - The power source is dictated by minimum power, minimum weight, and minimum volume requirements.

1.4 General Requirements - All components.

Test - To be ground tested in a gravity field, however, convective heat transfer will be eliminated in the case of the radiator by operation in a vacuum.

Flight Capability - All components are designed and selected on the basis of minimum weight, volume and power required as well as the ability to withstand the environment of space flight. That is to say that the system should represent one suitable for actual space flight except where conflicts with the ground testing requirements take precedence. In these cases, however, the deviations from flight criteria are discussed and the modifications necessary to produce a flight capability are described.

2. Description of System and Concepts as Selected

The system concept choice used for maintaining thermal control within the vehicle is the Liquid Transport concept whereby a suitable fluid is used as the means of conveying the heat from its source to the radiator where it is to be rejected into space. A pump circulates the fluid within the refrigerator or air conditioner and the fluid is directed through connecting tubes to the radiator or the outer wall of the vehicle where heat is removed from it and radiated into space. Justifications for this concept are presented in the Concepts Comparison Section of this report and Appendix 1.

Figures 1 and 2, functional schematics of the system except for the water storage circuits, illustrate the integrated refrigeration - air conditioning system. The system has two separated liquid transport circuits, a low temperature circuit and a high temperature circuit, each of which will be discussed separately. The low temperature side of the circuit handles the food refrigeration heat load of 130 BTU/hr plus the cabin air conditioning moisture removal heat load of 600 BTU/hr. The high temperature circuit absorbs the gross sensible air conditioning load of 18650 BTU/hr.

Two separate liquid transport circuits are employed because by specialized design it is possible to decrease the total space radiator heat rejection area requirement by nearly a factor of two. This area reduction results in a radiator weight saving of 171 lbs. The refrigerator requires a transport fluid temperature of -10°F, but the heat load to be rejected from this system is small (1000 BTU/hr, with a safety factor) and so requires only 29 square feet of radiating surface. (Reference Appendix 3). If this same low temperature transport fluid system were used to convey the heat load of the air conditioner, (18650 BTU/hr) a total radiator surface of 570 ft^2 would be required since radiator area is directly proportional to the amount of heat to be rejected (if all other factors are held constant). The use of two separate radiators, a 29 ft^2 low temperature (-10°F) radiator and a 265 ft^2 high temperature (+40°F) radiator, results in a combined radiator surface of 294 ft^2 as opposed to the 570 ft^2 required for a single low temperature radiator. This is due to the fact that radiant heat transfer between the radiator and space is directly proportional to the difference between the absolute radiator temperature raised to the fourth power and the absolute space sink temp. raised to the fourth power. (Ref. Appendix 3). Hence, the higher the radiator temperature, the more heat it can reject per unit of area. Calculations indicate that a typical radiator will weight approximately 0.62 lbs/ft^2 of area (including fluid, tube and fin), therefore the reduction of radiator area requirement (276 ft^2) results in a 171 lb. weight saving, and gives a radiator weight of 182 lb.

Low Temperature Circuit Description

The low temperature liquid transport circuit utilizes a pump to convey the heat transport fluid through the heat transfer shelves within the box where the fluid absorbs heat from the compartment and its contents. There are two parallel fluid flow paths within the box and each is independently controlled by means of a thermostatic device. The controllers maintain food temperatures within the chill and freeze compartments at 36 ± 4°F and 0° ± 5°F respectively. Positive independent control of each of the compartments of the refrigerator is necessary in order to maintain food temperatures within the narrow limits specified. The chill circuit has sufficient heat removal capability to "pull-down" its entire food load from 75° F to 36° F over a period of 12 hours. Referring again to Figure 1, downstream of the freezer coils in the freezer flow circuit are the moisture freeze-out coils (zero-g water collector) of the habitable environment control portion of

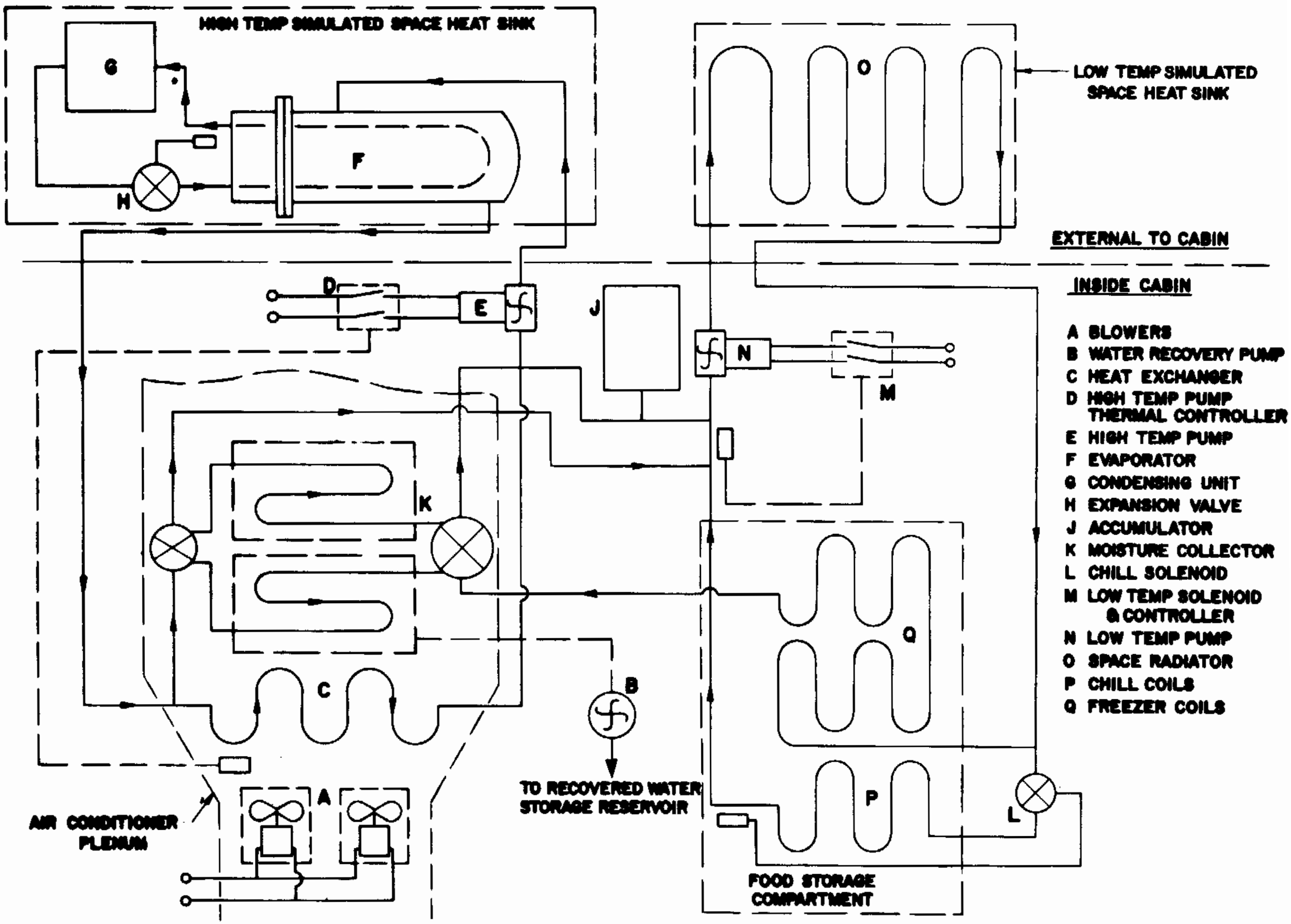


FIGURE 1 FUNCTIONAL SCHEMATIC

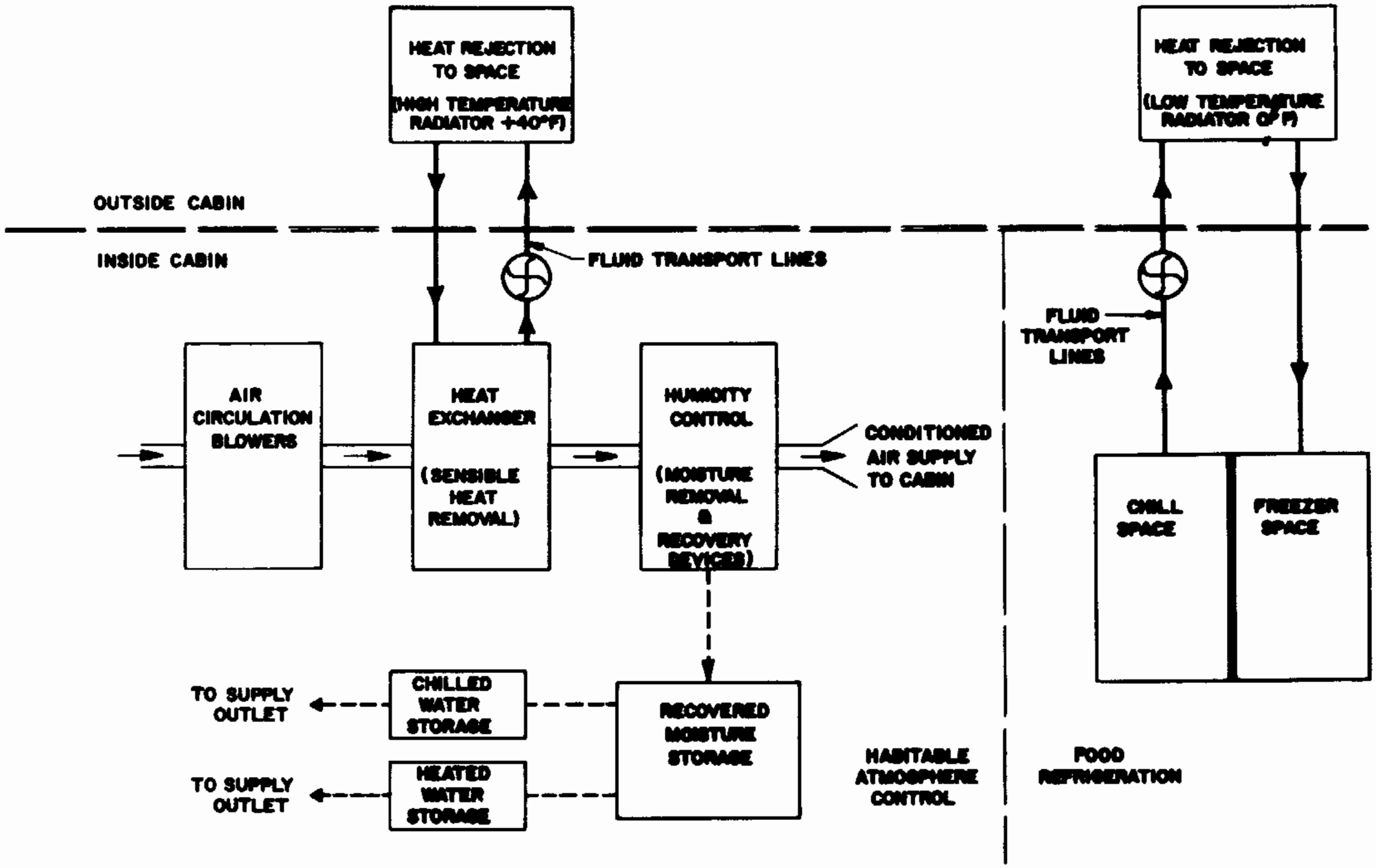


FIGURE 2 SYSTEM BLOCK DIAGRAM

the system which impose an additional 600 BTU/hr heat load on the low temperature side of the system. The low temperature fluid then passes through the pump which forces it through the space radiator where the heat energy gained from the refrigerator and moisture removal devices is rejected to space (a space heat sink simulator in this case). The only additional functional components required on the low temperature side of the system are the accumulators which compensate for fluid volume changes due to temperature variation. One accumulator connection to both the low and high temperature sides is used which results in a weight saving and reduces the complexity of the system.

High Temperature Circuit

As shown by Figure 1 the high temperature circuit contains an electrically driven pump to convey the heat transport fluid between the heat source, in this case the air conditioner heat exchanger, and the space heat sink. The space heat sink simulator for the high temperature circuit for Project COOL is a commercial condensing refrigerant unit of sufficient capacity to handle the anticipated maximum sensible heat load of 18,650 BTU/hr. The commercial condensing unit is an economical means of effectively providing a high temperature circuit heat sink. It is used to eliminate the high costs involved in fabricating a space type radiator and a high vacuum chamber with cryogenic walls as a simulated space heat sink of sufficient size to handle the high temperature circuit heat load. These costs were not warranted since the direct radiation to space liquid transport concept is effectively proven by the low temperature circuit of the system with its space type radiator and cryogenic vacuum chamber.

Thermal control of the high temperature side of the circuit (i.e., cabin dry bulb temperature) is accomplished by a transistorized temperature controller which monitors cabin air temperature and operates the liquid pump motor as required to maintain cabin air temperature at the set value.

The high temperature circuit utilizes an electronic thermal controller which has the fine control sensitivity necessary to enable the air conditioner coils to operate just above the dew point temperature of the incoming air mixture. The resulting temperature difference between the coils and the air mixture is the maximum (with a corresponding maximum heat transfer rate for a given size and configuration of heat exchanger) that can be obtained without having moisture condense on the coils. Moisture condensation on the heat exchanger coils in this case is not desirable since the system is designed to control humidity and collect moisture from the cabin atmosphere by freeze-out on the low temperature moisture collection surface. Freeze-out of moisture, rather than condensing, is a technique which can give positive collection under both laboratory tests and zero-g flight conditions. Also included in the high temperature fluid transport circuit are the liquid accumulators which as mentioned previously are shared with the low temperature circuit and serve to compensate for thermal expansions and contraction of the circulating fluid. The liquid transport fluid, ethylene glycol, selected because it has good thermal properties, presents no fire or explosion hazard and is non-toxic.

A ducting network and two parallel blowers, operating continuously and installed to deliver cabin air to the air conditioner complete the major functional parts required for the habitable environment control system.

The recovered water storage portion of the system is comprised of a 2 1/2-gallon main storage reservoir, a 3/4-gallon chilled water reservoir and a 3/4-gallon heated water storage reservoir. The main water-storage reservoir receives water from various recovery sources (humidity control devices, urine reclamation systems, etc.) and then supplies the chilled and heated reservoirs as well as the cabin temperature water outlet. It has provisions as described later for expelling any air entrapped within the water that is received from the water recovery systems.

The chilled water reservoir cooling coils are connected to the low temperature ($0^{o}F$) liquid transport circuit. The flow of the coolant fluid through the coils is controlled by a thermostatically actuated solenoid valve which maintains the chilled reservoir water at a temperature of + $40^{o}F$.

The heated water reservoir has its contents maintained at $170^{o}F$ by means of a 150-watt electrical cartridge heater which is thermostatically controlled. By careful overall system integration an actual space vehicle design could make use of a high temperature heat source to supply the energy required to heat the water.

3. System Packaging and Integration Philosophies

3.1 General Discussion

The design for the COOL system has been pursued along a course which results in a system that is flight optimized to the extent allowed by the fact that it has been integrated with an already existing feeding console structure aboard the space cabin evaluator. Other factors such as cost limitation, development lead times, and the desirability to have the capability of varying internal environmental parameters also hamper the utmost flight optimization of this system. The purpose of this section of the study is to explain and justify the system packaging philosophies.

3.2 Configuration

The volume available for the installation of the Project COOL equipment aboard the evaluator is that space which was formerly occupied by the frozen food storage compartment within the feeding console. The entire COOL system has been fitted into this space (33.75" high x 27.5" wide x 15.25" deep) with the exception of the blowers which are contained within the inlet duct section to the air conditioner heat exchanger. Figure 3 illustrates the overall system as packaged for installation and indicates how it is fitted within the existing feeding console complex.

The COOL module is mounted within the existing console frame in the same manner as was used to mount the thermoelectric freezer. The air conditioner inlet duct system is mounted in the existing available space beneath the module. The air discharge duct which connects to the main overhead conditioned air supply trunk is routed through the space formerly occupied by one of the overhead food storage bins of the feeding console which was in excess to the required food storage volume requirements.

The equipment module is designed as an integral self-supporting structure which can be installed within the feeding console structure with an absolute minimum of rework. It is mounted to the console frame by means of a frame of aluminum angles on its outer sides, top and bottom which are identical to those on the freezer that it replaces. The only rework required on the feeding console frame was the removal of two cross braces, one above the module and the other below it, to allow clearance for connection of the air

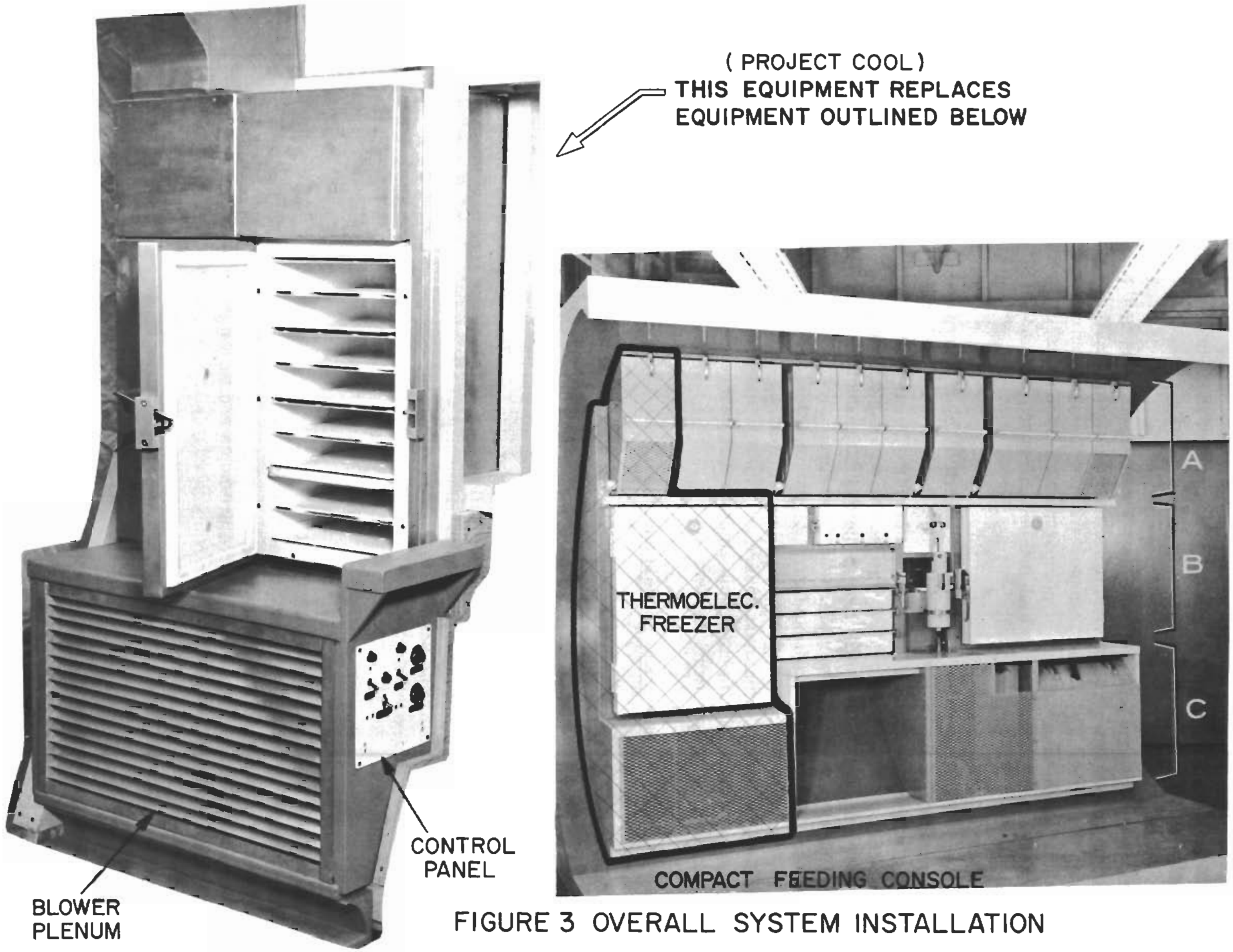


FIGURE 3 OVERALL SYSTEM INSTALLATION

conditioner inlet and discharge ducts. The removal of these braces does not materially affect the structural integrity of the console framework.

The various subsystems within the module, Figure 4, are arranged so that the air conditioner heat exchanger and water collector unit are on the left side and the food storage compartment is on the right. The liquid transport pumps, thermal control devices and accumulator are mounted above the refrigerator compartment. As can be seen in Figure 4, the system is broken down into submodules which can be independently removed from the main equipment module.

The ductwork installations did not require rework to the feeding console, other than the drilling of mounting holes or the installation of anchor nuts, to implement their installation. The main overhead duct trunk runs the entire length of the cabin and is designed so that branch outlets or discharge louvers may be fitted interchangeably at various locations along its length. Also, its connection to the discharge duct from the air conditioner plenum is designed so that their junction can be made anywhere along the length of the overhead trunk. This ductwork configuration permits the feeding console to be placed at any location along either wall of the cabin evaluator and also allows the varied arrangement of other internal heat source equipment without the necessity of reworking the entire duct network each time the cabin equipment is rearranged. Figure 5 illustrates the detail design of the ductwork system.

3.3 Pre-flight Operations and Food-Loading

Operational Checkout

The system has been designed so that pretest operational checkout procedures are minimized and simple to perform. The blower operational checkout will be merely to ascertain that electrical current draw and static pressure in the air conditioner plenum are within specified limits. Also required will be flow and pressure checks of the two liquid transport circuits to assure that both pumps are within operational limits. Thermal control devices are next to be checked for proper cut-in and cut-out temperatures. Liquid transport fluid level within the system is checked and this completes the pre-flight operational checkout.

Food Loading

Figure 6 illustrates the food storage submodule installed within the equipment module. It has been designed so that this complete assembly can be removed from the COOL module as a unit, loaded with the mission food requirements and then merely "plugged-into" the main module once more. Food removal procedures are explained in detail within the Equipment Design section.

Maintenance

The system has a minimum number of functional components which increases reliability. Maintenance procedures required are, as a result, also minimized. Should a component failure occur during a test or "flight" it can be easily replaced by access through the front of the equipment module. Special tools for servicing and maintenance are not required.

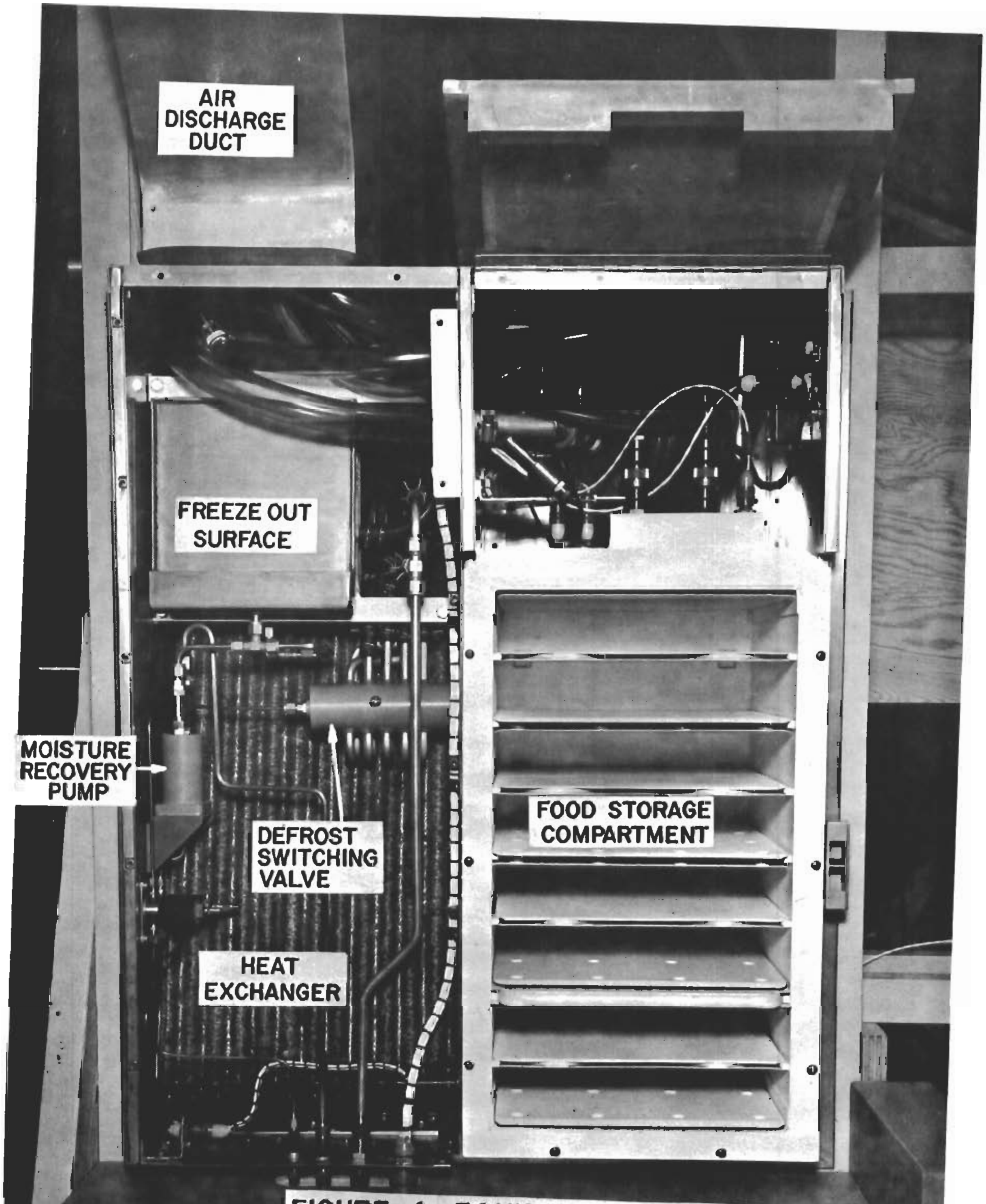


FIGURE 4: EQUIPMENT MODULE

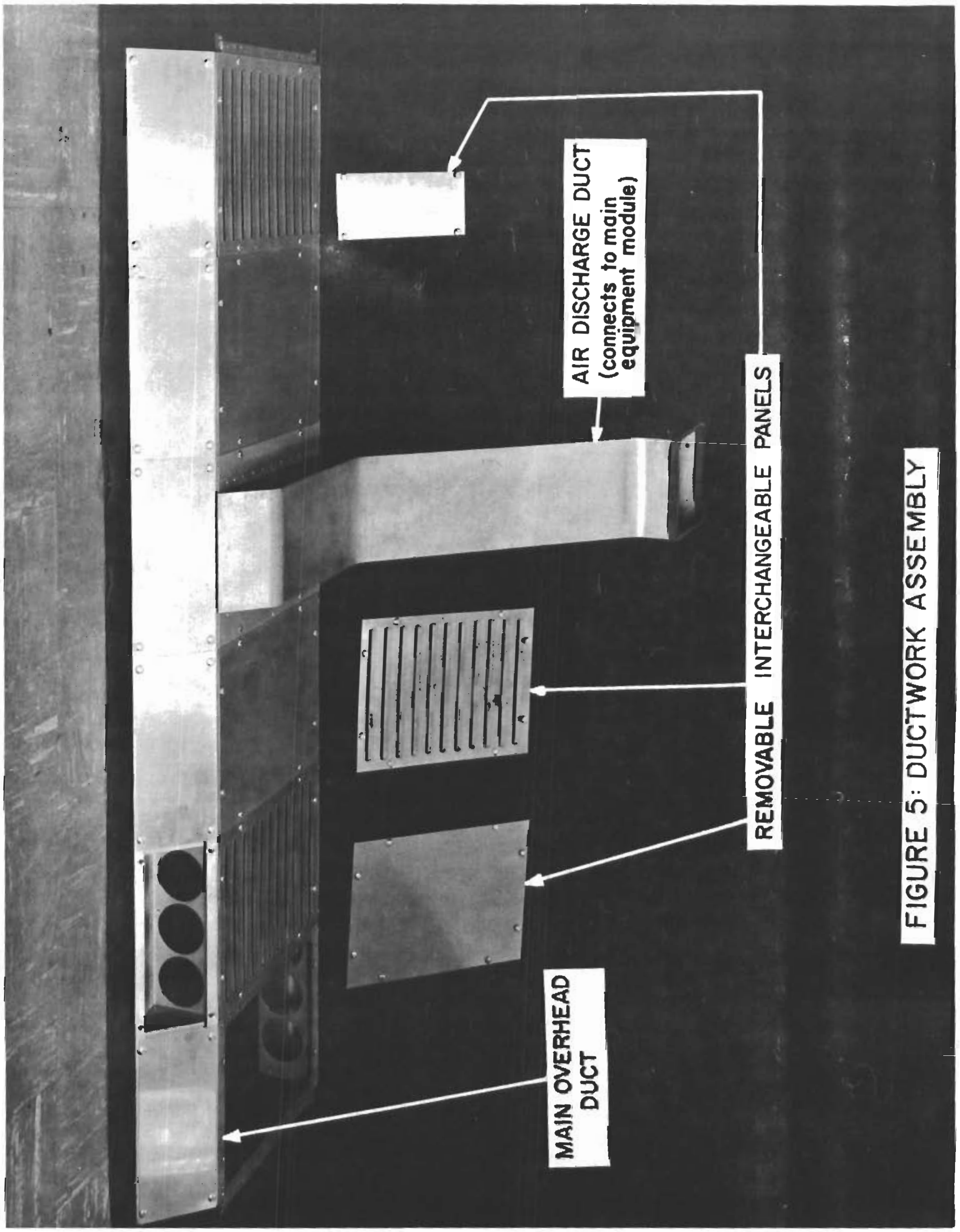


FIGURE 5: DUCTWORK ASSEMBLY

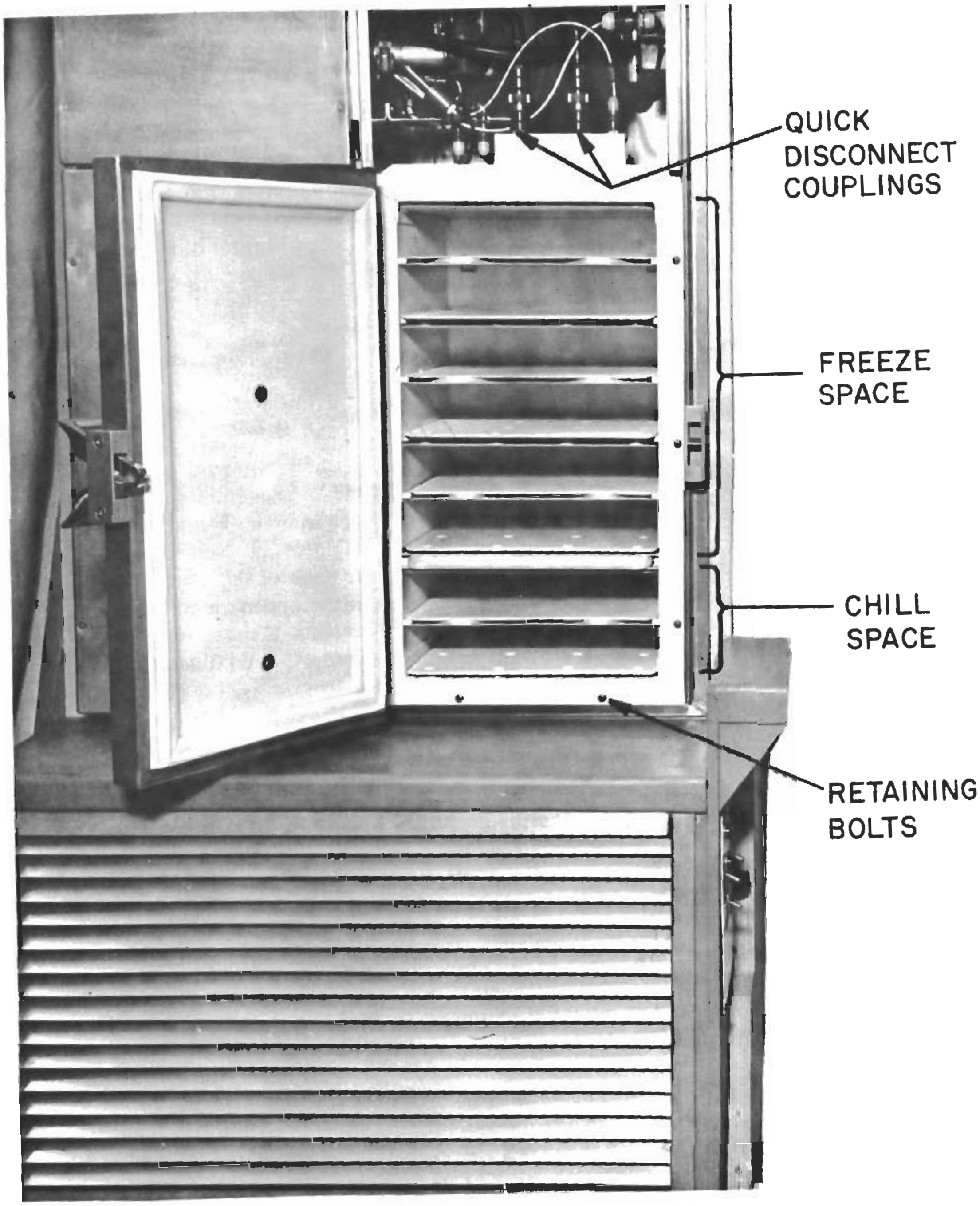


FIGURE 6: FOOD STORAGE COMPARTMENT

3.4 System Weight and Power Requirements

The Project COOL equipment has a total weight of 192 lbs. not including the food load and the weight of the external space radiators.

The total combined radiator weight as required for the two liquid transport circuits is estimated to be 182 lbs. including the liquid transport fluid. This configuration would be one which has both sides of its surface radiating to space and is the configuration that has been selected as the space radiator to be built for the low temperature circuit of the Project COOL system.

The total power requirement for the COOL system is 775 watts as itemized in Table 1.

4. Heat Removal Concepts Comparison

A comparison of total system estimated weights for various heat removal concepts which might be used aboard a space vehicle has been compiled and is presented in Table 2. The system weight is defined as the sum of internal cabin equipment weight plus external space radiating surface weight. It does not include the weight of the food to be stored. Also presented in the table are the power requirements for each of the concepts. From Table 2 it can be seen that the Liquid Transport concept is the optimum on a power and weight basis. Appendix 1 gives additional detailed comparisons of these concepts. Advantages and disadvantages of the different methods considered are discussed below.

4.1 Liquid Transport Concept

Advantages

a. Various components may be located at different places within a vehicle without penalizing the efficiency of the system other than the additional weight of the interconnecting tubing between components. By use of a single pump, it is possible to remove heat from several sources and convey it to a common radiator for rejection to space.

b. Since the liquid transfer fluid is not required to undergo any change of state, as is the case in a vapor compression cycle, it is not gravity dependent so far as its operation is concerned. With vapor compression the main design problem in space, aside from the greater power required, would be separation of liquid and vapor phases.

c. Also, the temperature levels of separate compartments (freeze and chill in the refrigerator), may be independently controlled at different levels within the same liquid transport system. This is accomplished with a minimum of compromise insofar as the operational efficiency of the system is concerned.

Disadvantage

The only disadvantage of the liquid transport system is that it can reject heat only when the external sink temperature is lower than the temperature of the cabin or refrigerator compartment temperature. A low temperature sink, however, is available in outer space and by proper orientation of the radiator during a mission it is possible to take advantage of this low temperature sink.

TABLE 1

WEIGHT TABULATION BY SUB-SYSTEM

SUB-SYSTEM	ITEM	ITEM WEIGHT (pounds)	ITEM POWER REQUIREMENT (watts)	SOLAR PHOTOVOLTAIC POWER SOURCE Weight 125 lb/KW (pounds)
Low Temperature	Fluid Pump (with motor)	4	90	11.25
	Radiator	12	--	-----
	Refrigerator (without food)	25	--	-----
	Thermostatic Control Values	1	20	1.25
	Humidity Controller	2	--	-----
	Accumulator - common to both systems	10	--	-----
	Fluid	10	--	-----
	Connecting Tubing's Hardware	4.5	--	-----
	TOTAL	68.5	110	12.50
High Temperature	Fluid Pump (with motor)	10	440	55
	Radiator	110	--	-----
	Heat Exchanger	20	--	-----
	Fluid Temperature Control	1	10	0.6
	Fluid	50	--	-----
	Control Solenoids	1	15	0.9
	Two Air Blowers	21	200	25
	Connecting Tubing Hardware	5	--	-----
	TOTAL	218	665	81.5
Structure	Unit Module	35	--	-----
	Air Ducting	50	--	-----
	Nut and Bolt Hardware	5	--	-----
	TOTAL	90	--	-----
System Total		376.5	775	94

TABLE 2

SYSTEM CONCEPT COMPARISON

SYSTEM CONCEPT	EQUIPMENT WEIGHT * (pounds)				POWER CONSUMPTION WATTS	SOLAR PHOTO-VOLTAIC POWER SOURCE WEIGHT (pounds) (estimated)	TOTAL EQUIPMENT FLIGHT WEIGHT INCLUDING POWER SOURCE (pounds)	SPACE RADIATOR SURFACE AREA (sq. ft.)
	REFRIGERATOR	AIR CONDITIONER	RADIATOR (with all fluid)	TOTAL				
Liquid Transport	66	77	237	380	640	80	460	294
Thermo-Electric	80	480	227	787	4950	618	1305	283
Vapor Compression	120	145	119	384	2800	350	734	148

* Not including weight of food and air ducting

4.2 Thermoelectric Cooling

Advantages

a. Silent operation
b. No moving parts
c. Unaffected by the absence of gravity
d. No refrigerant required
e. Relatively efficient for small cooling loads and capable of producing cooling effect over small area (good for spot cooling).

Disadvantages

a. Low coefficient of performance compared to other heat pumps for all cooling loads other than very small values (based on present state-of-the-art).

b. High temperature differences between source and sink reduce efficiency further.

c. Physical arrangement requires that heat sink be in close proximity to heat source otherwise and intermediate transport medium is necessary.

d. For large cooling loads, an extremely large volume of thermoelectric material is required resulting in large heat exchangers, large volumes, and sizable weights for the elements and heat exchangers alone.

e. For space vehicle use, where electrical energy at the present time is a scarcity, the power required to operate such a system is prohibitive. Where an inefficient heat transport means is used, the heat to be discarded increases greatly because the power used by the heat transport means and converted to heat is added to the original cooling load. This results in an increase in the exterior radiator required to ultimately reject the heat to space. Therefore, on a weight basis, the generating equipment required for the power, the additional radiator area necessary, and the weight of the elements themselves create an exorbitant weight penalty. (See Table 2)

4.3 Vapor Compression System

Advantages and Disadvantages

a. A system sized to handle the Project COOL heat load could be expected to have a COP* somewhere in the vicinity of 2 which means that for the 18,000 BTU/hr air conditioner heat load which must be rejected, a heat equivalent for the compressor work of approximately 9000 BTU/hr (2640 watts) must be supplied from vehicle power sources.

For use in a laboratory test bed application, if the only desired intent is to provide a reliable proven method of heat rejection, this vapor-compressions system would appear to be very desirable.

b. The weight of the two vapor-compression systems (refrigerator and air conditioner) required would be approximately 150 pounds not including radiator or evaporator weight. This is essentially 150 pounds more weight than the liquid transport system so it is obvious that the two do not compete in respect to weight. It would not be advantageous

*coefficient of performance

to use a single vapor compression system for both air conditioner and refrigerator functions since the low evaporator temperature required by the freezer would then dictate the hardware choice for the entire system. Since a major portion of the cabin load is actually at the higher temperature of the air conditioner evaporator, the COP of a single system based on Freezer requirements would be penalized. The consequence would be an increase in power consumption and hence, additional power source and radiator size.

c. Perhaps the decisive factor at this time in discarding a vapor compression system for space use is the fact that no proven hardware design configuration presently exists which could separate the two refrigerant phases (i.e. gaseous and liquid) under zero-g conditions. Without the development of practical techniques or devices to accomplish a positive separation of phases, it is not possible to efficiently operate a two-phase system.

d. The eventual development of a vapor compression system which will work in space would permit its use in applications where the orbit conditions were such that radiator orientation could not be controlled, or a minimum of area were available for radiator surface. In this case the vapor compression system, because of the higher temperatures at which it rejects heat, might be advantageous.

SECTION III

EQUIPMENT DESIGN

1. Equipment Module Shell Design

The outer shell structure for the equipment module, which houses the majority of the system, is designed as a riveted assembly and utilizes a sandwich construction for its wall sections. The wall is composed of inner and outer aluminum skins which enclose one-half inch thick polyurethane foam panels. The aluminum skins were bonded to the foam with an epoxy resin adhesive, each panel was cut to size and then the module shell formed by riveting the panels together in conjunction with an inner and outer corner framework of structural aluminum angle. The shell thus formed was mounted within the feeding console aboard the evaluator. The various subsystems were installed within the shell from the front. The removable front panels of the module shell provide easy access to any of the system components for servicing or replacement.

The foam sandwich wall construction gives a lightweight structurally stiff wall which is also a good thermal and acoustic insulator. The thermal insulating properties are required to prevent "sweating" (the condensation of moisture from the cabin atmosphere) on the outer wall of the shell adjacent to the food storage compartment and to reduce leakage of heat in the refrigerated space.

2. Refrigerator Sub-Module Design

The sub-module was constructed of an inner aluminum shell which gives structural support to the food restraint mechanism and heat transfer shelves and forms the vapor barrier for the compartment. One-half inch of polyurethane foam was "foamed-in place" to the outside of the shell and contributes approximately 40% of the thermal insulation to the refrigerator. The remainder of thermal insulation is provided by the foam sandwich wall of the system equipment module. This composite wall results in a steady state heat leak of approximately 80 BTU/hr (Appendix 2). Figure 7 is a section through the wall of the equipment module with the refrigerator sub-module installed in place. Also shown is the refrigerator door and door joint configuration.

The selection of the insulation material was predicated by factors other than those dealing with design optimization of weight and volume. The primary objective of this design study was to finalize a configuration for a food preservation technique which physically proves the feasibility of a Liquid Transport system rejecting heat by direct radiation to space. Therefore, any design decisions were heavily weighted in favor or proving this radiation concept.

"P-zero," the insulation which was recommended in Appendix 1 was not selected for this prototype because the time available for the fabrication was not sufficient to obtain the raw material (specially oriented glass fibres), or develop the special tooling required to fabricate a box from "P-zero."

In place of "P-zero," a polyurethane freon-filled foam was selected as the insulation material, because it is available "off-the-shelf" and readily lends itself to fabrication in model shop facilities. By referring to the FROST feasibility study report (Appendix 1) it can be seen that thermal conductivities of freon blown polyurethane foams are the

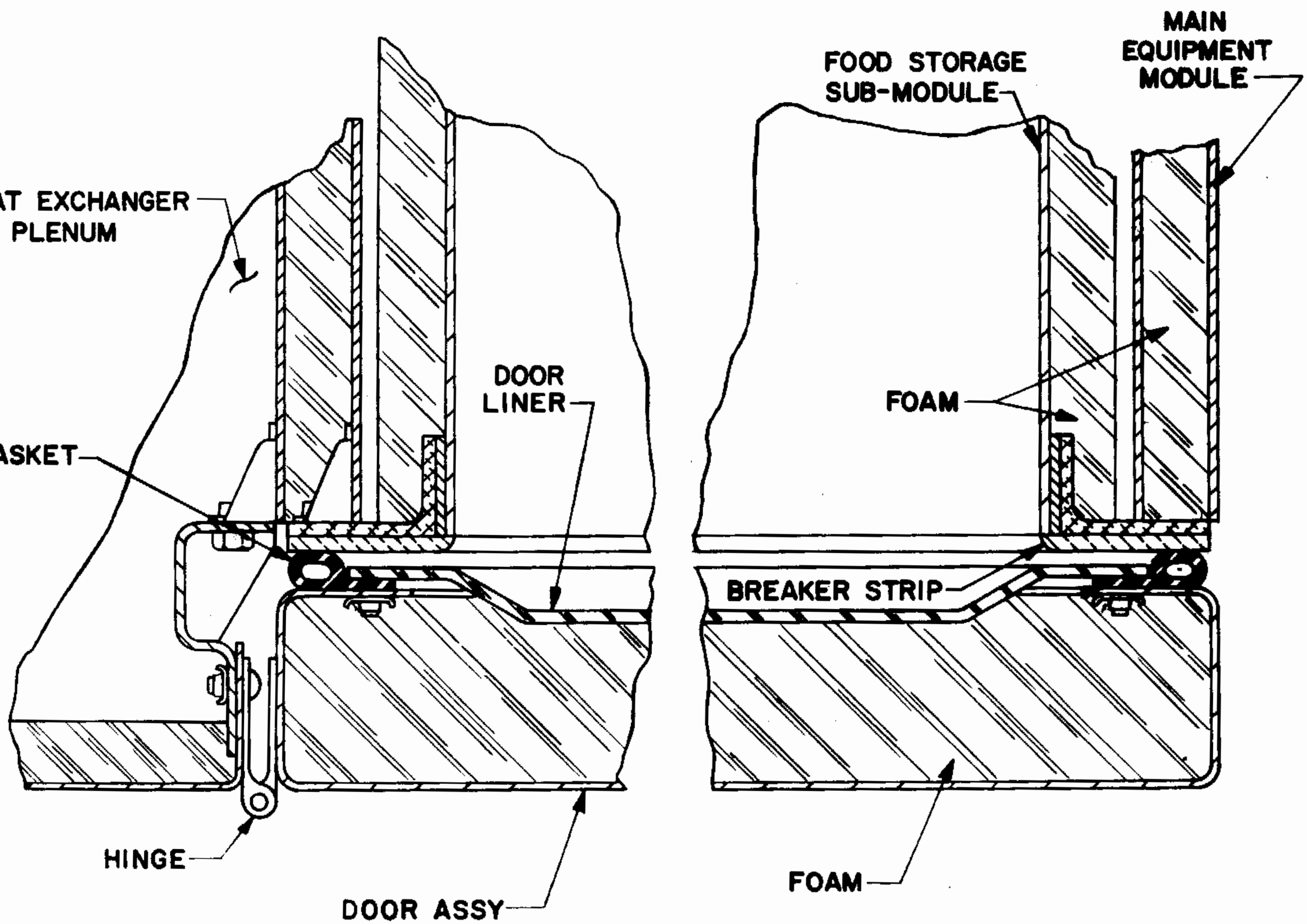


FIGURE 7

SECTION THROUGH FOOD STORAGE COMPARTMENT second choice with respect to "P-zero" when compared on the basis of weight and volume.

Refrigerator Door Design

The door design configuration also is depicted in Figure 7, a typical cross sectional view taken through the door joint. The door is hinged from the main equipment module rather than the refrigerator sub-module. This makes possible a longer heat conduction path around the perimeter of the door joint (resulting in less heat gain to the food compartment) and also lessens the weight of the refrigerator sub-module by some 6.50 lbs., making the module a more easily handled package for installation during food loading procedures.

The door was designed and constructed similar to the box envelope with the exception that the insulation was foamed in place within the preformed door structure. This type of construction makes possible the incorporation of a formed integral edge for the door. A swinging door was chosen over a sliding configuration to assure a good door seal joint, and also to minimize the complexity of the overall door design. Furthermore, sliding doors are more prone to "sticking" or jamming in their guide rails due to the interference of foreign objects, distortion of the rails or door warpage.

A single door common to both the chill and freeze compartments was determined to be the most efficient configuration, since it gives a minimum length of door gasket and reduces by half the number of door latches required. The only disadvantage of a single door is that contents of both compartments (chill and freeze) are exposed to the ambient atmosphere each time the door is opened, with the resulting exchange of cold dense compartment air for that of the relatively warm room ambient air. During weightless conditions of space flight, "air exchange" due to free convection will not occur, however, for laboratory test conditions free convection does exist and its effects must be considered.

Appendix 2 gives the calculations which were performed to determine how much heat energy must be rejected by the system each time the box door is opened. The value of 9 BTU was determined by assuming that a complete air change within the box occurred each time the door was opened, certainly a conservative assumption. If this additional heat load were imposed on the system as often as 3 times an hour (three door openings per hour), it would not over-tax the capacity of the radiator.

Gasket and Joint Configuration

In the design of an aperture in a wall which acts as a thermal insulator in earth environments, the edge leakage is a function of both conductive and convective heat transfer. Radiation gains or losses, as the case may be, are minimized due to the small cross sectional area of the gap between wall and door. Additionally, refrigerator door joints must have insulating properties which are adequate to give an outside (room side) surface temperature which is higher than the dew point of the most humid atmospheric conditions to be encountered. If the door joint is not designed to satisfy the latter requirements, moisture condensation (sweating) will occur around its perimeter.

The door sealing flange was made as wide as possible within the space available to give a maximum length to the conductive air path between the door and the seal surface of the box. This air space is the major thermal barrier to the transfer of heat in the door joint.

A "dead air" space is a good thermal insulator if free convection can be minimized, and to accomplish this, good practice in the refrigerator industry requires that the clearance between the door and the sealing flange of the box, when the door is secured, must be approximately one-quarter of an inch. The gasket seal around the perimeter of the joint affords a good vapor barrier and further reduces convective heat transfer. Polyethylene breaker strips were placed to minimize conductive heat transfer along the edges of the door joint. If the aluminum inner and outer box walls were thermally linked with metallic breaker strips, such as aluminum, the heat conducted through the door joint would be significantly increased. For example, the thermal conductivity of aluminum is approximately 1200 times that of polyethylene. Refer to Appendix 1 for detailed calculations of heat leakage through the door joint.

Polyethylene, a thermoplastic, was selected for the breaker strips because of its comparatively low thermal conductivity, good dimensional stability, and high impact strength, as well as the fact that it lends itself well to heat forming at comparatively moderate temperatures. No contamination should be imparted to the atmosphere or food from this material as it is presently widely accepted for use in the manufacture of food containers of all types.

Door Latch

The latch utilized for the refrigerator door (Figure 8) was especially developed to give ease of operation under the weightless conditions of space flight. In order to open the door the latch hand grips are squeezed together between the thumb and fingers. The force balance within the hand is such that the compressive forces applied by the fingers and thumb are equal and opposite and, therefore, cancel each other. The forces within the hand itself are, therefore, in balance and no external forces must be applied to keep it in equilibrium. In other words, under zero-g conditions it is possible for the crewman to unlatch the door and not pull himself toward or away from the door as he would if a conventional push or pull type latch release mechanism were utilized.

Internal Configuration

Because of weightless conditions encountered during space flights, the food storage racks within the box were designed to individually restrain the food containers. This has been done so that removal of a number of containers does not permit the remainder to "float" around within the box. Even worse, when the door is opened, the entire contents of the refrigerator could drift, or be easily displaced from the food compartment, into the cabin were they not restrained. The food container restraint racks also serve the additional function of carrying the cooling fluid through the refrigerator. This enables the pull down rate of the chill compartment (entire contents from 70°F to 36°F in twelve hours) to be more easily accomplished because each container to be chilled has a thermally optimum conductive path to the heat transport fluid.

The choice of this purely conductive heat transfer path between the food and the liquid cooling coils was made over a heat transfer path which uses an intermediate convective heat transfer means between the food and cooling coils located in the walls. Cold plates have several advantages in this application. First, since the food containers must be individually restrained it is efficient to have these restraint racks also carry the heat transfer fluid. Furthermore, since as previously stated free convection will not exist in outer space, it would be necessary to provide a means of forced convection in each

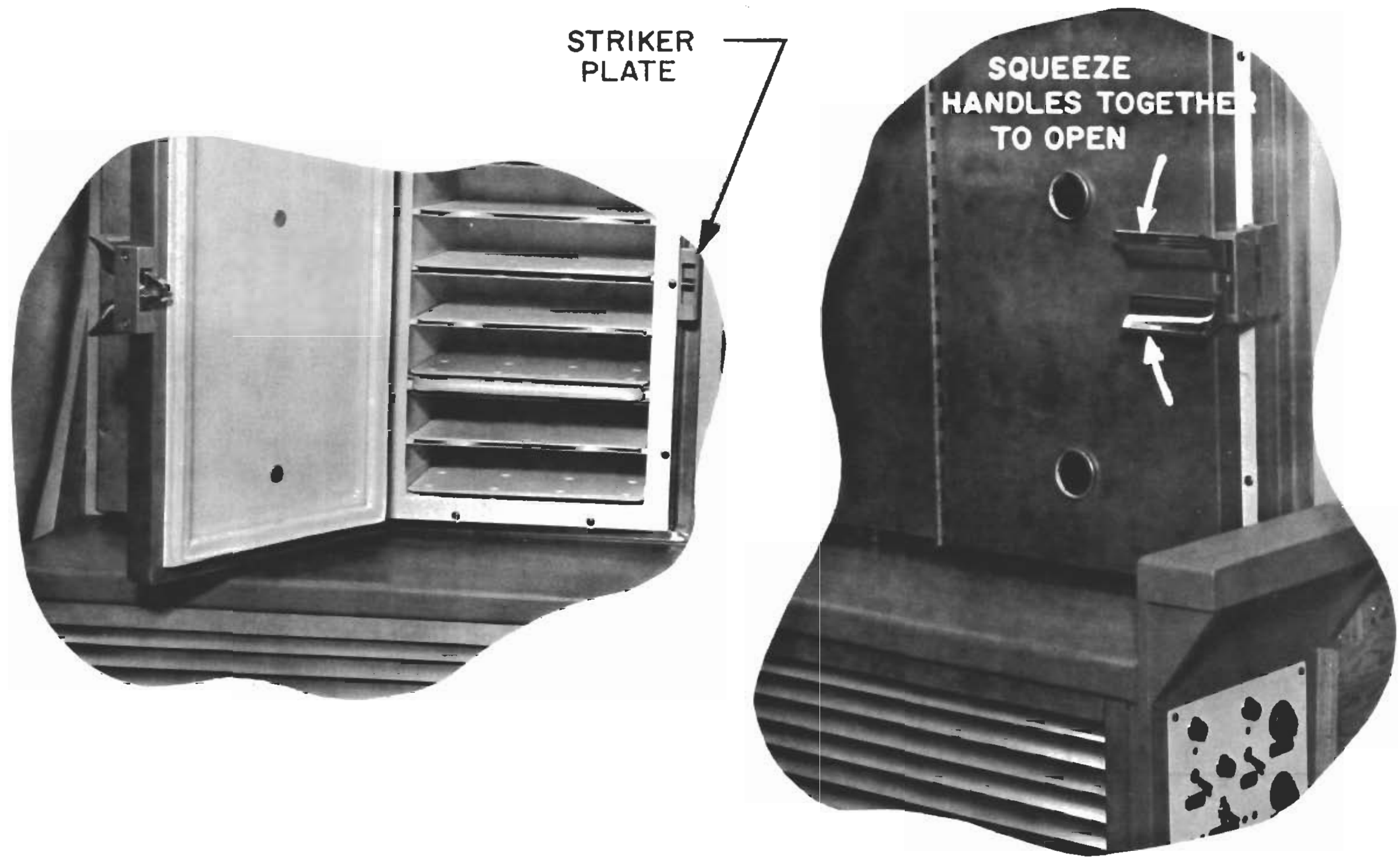


FIGURE 8
REFRIGERATOR DOOR LATCH

compartment by use of a small blower which would consume some 8 to 15 watts of electrical power each, depending on the pressure level in the box. The additional 16 to 30 watts of power would have to be supplied to the fan motors and then rejected as heat from the box to the radiator. This higher heat load would result in an increase in radiator size, and furthermore, if the fluid coils were on the inside of the box walls, the temperature differential (ΔT) across the box wall would be greater. This greater Δ T would generate a greater rate of heat transfer across the walls, imposing a larger heat load on the radiator and further increasing the surface area required to do the job. In other words, the placement of the cooling coils within the food support shelves permit the air space inside the box and the food itself to act as additional thermal insulation. The additional insulation decreases the overall coefficient of heat transmission between the cooling fluid and the cabin atmosphere which decreases the heat transferred to the fluid. From the above discussion it is concluded that the exclusively conductive cooling means, which has been selected, is more efficient as regards power, weight, space and temperature gradients.

Food Container Restraint

Referring to Figure 9, it can be seen that two types of shelves (coolant transport shelves and restraint shelves) are alternated in their arrangement within the box. This means that each and every container has one side in direct bearing with a cold fluid carrying shelf and its opposite side in contact with a spring loaded restraint shelf. The removal of food containers is accomplished by pulling out a "pallet" of food containers far enough to remove the number of containers desired. The pallet, along with the remaining containers, is pushed back all the way into the box. The force required to extract or insert the pallets is on the order of 5 or 6 pounds. This arrangement has a minimum of mechanism and moving parts. For this reason, it should be the least affected by frost accumulations within the box.

The contact with the fluid carrying shelf assures a good heat transfer path with resulting efficient "pull down" capability in the chill compartment and an even temperature profile within the freeze compartment. The heat transfer paths are further optimized by the positive force of the spring loaded shelves which forces each container against the heat exchange surface with a pressure sufficient to maintain the ΔT between the container and shelf at less than 1°F.

3. Low Temperature Fluid Pumping Requirements

The heat rejection capacity of the low temperature liquid transport circuit must be sufficient to convey the waste heat from the food storage compartment of approximately 130 BTU/hr, plus the heat load of the moisture freeze-out coils which are designed to remove 600 BTU/hr. This gives a total calculated heat load of 730 BTU/hr. However, in order to have a reasonable margin of safety, the space radiator design was sized to give a heat rejection capability of 1000 BTU/hr. The fluid pumping rate was determined by defining the maximum allowable fluid temperature rise through the freezer coils as 2°F and assuming that 1000/730 (margin of safety) x 130 BTU/hr (total box heat load) = 178 BTU/hr are to be required to give this 2° temperature rise to the fluid. By solving the following equation for w, fluid mass flow, the pumping rate can be determined:

$$Q = w\, C_p\, \Delta T \tag{1}$$

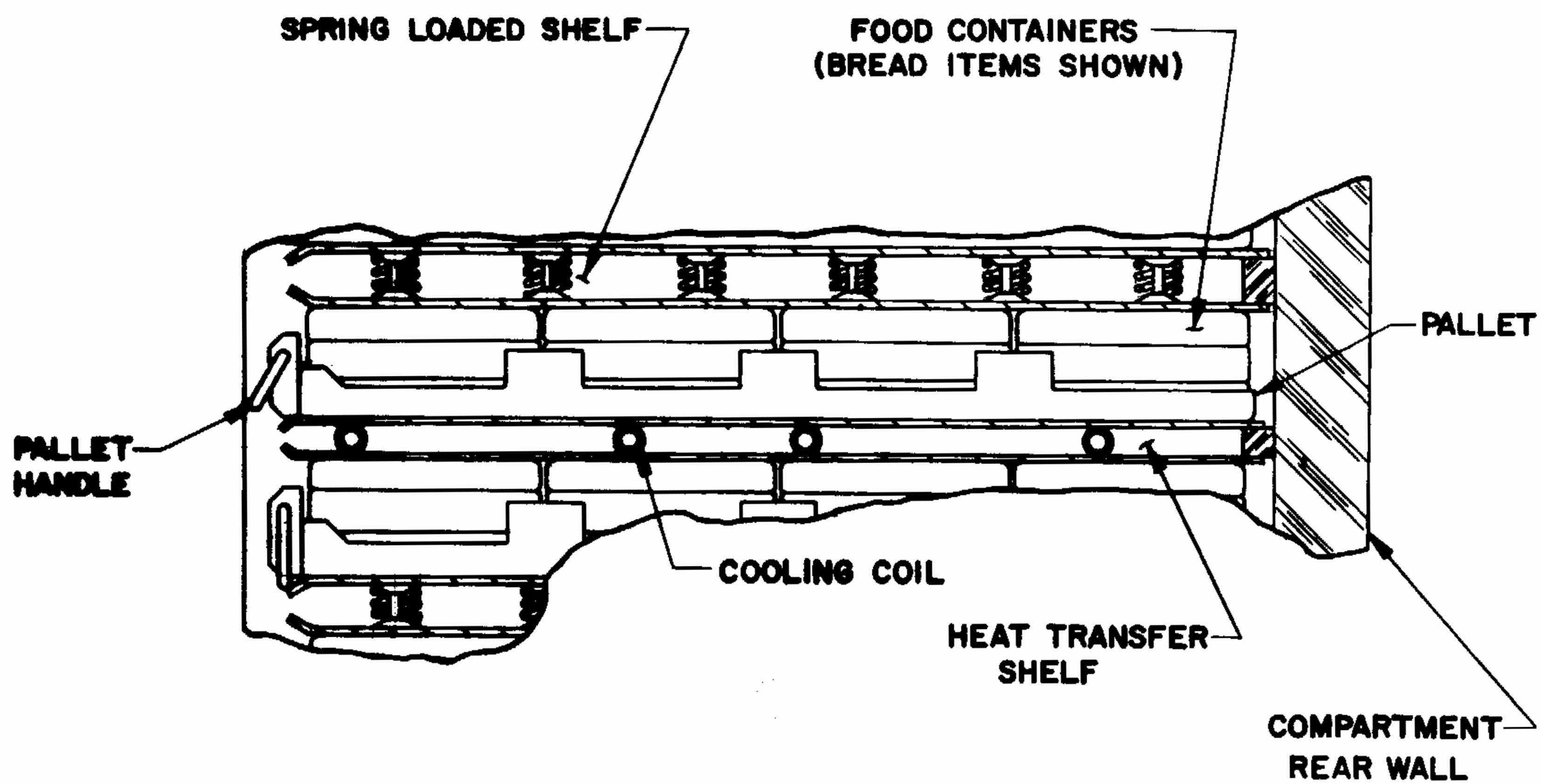


FIGURE 9
FOOD RESTRAINT MECHANISM

where

C_p = specific heat, 0.78 BTU/^{o}R# for 50%-50% glycol water @ 0^{o}F

w = fluid mass flow, #/hr (unknown)

Q = heat load, 178 BTU/hr

ΔT = temperature rise, ^{o}F

Using 50%-50% glycol-water as the heat transfer fluid, a pumping rate of w = 178/.78 = 114 lbs/hr or .214 gal/min of glycol-water is required.

The pump selected has the capability of delivering 0.6 gal/min at a head of 30 ft. of fluid. It is a centrifugal pump which is packaged as an integral pump-motor assembly, embodies aircraft type construction and meets the environmental requirements of the system specifications.

4. Accumulators

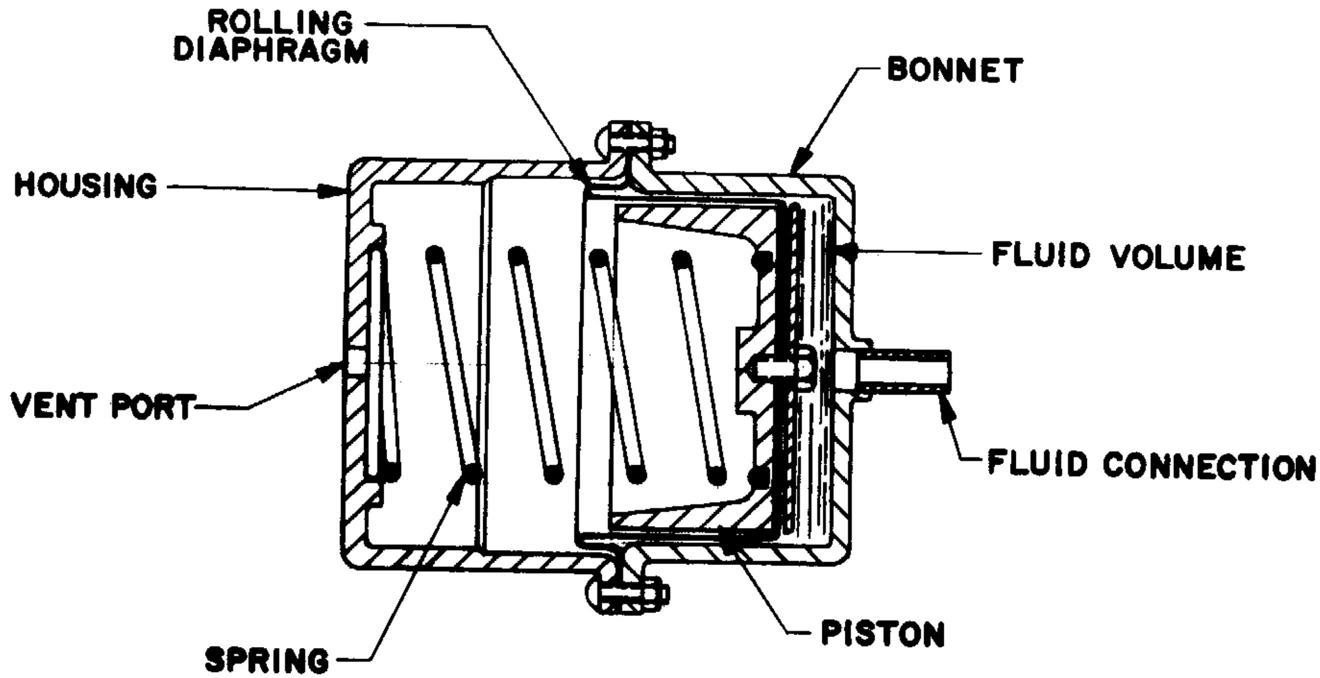
As previously mentioned, the hydraulic circuit must be totally filled with fluid to prevent the pump from becoming airbound which would interrupt the flow of liquid transport fluid. Since the specific volume of any fluid varies with changes in temperature, an expansion chamber (accumulator) must be provided in the system to give the necessary "variable volume," thus preventing excessive fluid pressure build-up and possible rupture in some section of the circuit. As discussed previously, a single accumulator circuit common to both fluid circuits is used for the COOL system.

The accumulator design configuration is comprised of a cylindrical vessel containing a spring loaded piston which has a rolling diaphragm seal. The vented side of this piston is referenced to cabin pressure. The accumulator will have sufficient capacity to compensate for change in fluid volume which result from temperature variation between -50^{o}F and + 165^{o}F. Fluid pressure within the system as a result of piston spring force will vary between 1 and 3 psi above the pressure level on the vented (reference) side of the piston.

The rolling diaphragm piston seal is a commercially available device which has many advantages over other piston sealing configurations. The rolling action of this diaphragm can be visualized by referring to Figure 10. As the diaphragm and piston are moved in an axial direction, due to an applied pressure, the diaphragm rolls off the piston sidewall, and onto the cylinder sidewall, with a smooth and continuous motion. The diaphragm is approximately .025 inches thick and is made up of a fabric overlay which is impregnated with an elastomeric sealant. The fabric-elastomer combination to be used will give satisfactory service over the entire operating temperature range of the system. The rolling diaphragm gives the following specific advantages for this application.

a. A positive static seal is provided by the membrane with no possibility of "blow-by" leakage past the piston.

b. The long stroke capability makes possible the optimization of the bore to stroke ratio for the accumulator as regards its packaging in the system.



(Section Thru Centerline)

FIGURE 10
ACCUMULATOR

c. The device provides an automatic de-icing action due to its rolling action and thereby prevents a "freeze-up" on the vented side of the piston.

d. Close tolerances are not required between cylinders and walls to accomplish the seal and, therefore, minor dents, scratches, or temperature variations will not cause binding or seizing.

e. Fatigue life of the device has exceeded 100 million full stroke cycles, in some cases, since the material is not stretched at any time throughout its stroke.

By referring again to Figure 10, it can be seen that the accumulator body to bonnet seal is accomplished by the retaining lip around the perimeter of the diaphragm, thus eliminating the need for a separate gasket to seal the joint. The body and bonnet are machined from 6061-T6 aluminum alloy and protected with a chromic acid anodic coating.

5. Habitable Atmosphere Control Equipment

5.1 Air Conditioner Heat Exchanger

5.1.1 Design Criteria

The heat exchanger configuration for the air conditioner is the optimum so far as the overall system weight, power source weights and volume available are concerned. The criteria for the establishment of the configuration were:

a. That the overall exchanger size was not to exceed the space available within the allocated volume for the system.

b. Air side and liquid side pressure drops were to be minimized which correspondingly results in minimum pump and blower power requirements.

c. The exchanger coil temperature was to be maintained at or above the highest dew point temperature to be encountered within the cabin. Maximum allowable cabin temperature and relative humidity are 75°F and 50%, respectively, which result in a dew point temperature (the temperature at which this vapor partial pressure is at the saturation point, 100% R.H.) of 55°F. This prevents the condensation of moisture on the exchanger coils since, as discussed later, it is desirable to control humidity and collect moisture by freeze-out techniques.

d. The heat exchanger as designed will remove a 18650 BTU/hr gross sensible heat load and maintain the cabin temperature at 75°F. The cabin temperature may be set and maintained if desired at a temperature as low as 60°F and 50% relative humidity. This is possible since the dew point temperature corresponding to the latter cabin air mixture conditions is approximately 40°F. The coil temperature under these conditions may be dropped considerably below the 55°F dew point temperature corresponding to a 75°F 50% R.H. air mixture and still not result in moisture condensation on the coils. It is emphasized here that the humidity control and moisture removal from the cabin air stream are accomplished separately on the low temperature moisture freeze-out surface. This freeze-out element has sufficient capacity to maintain the vapor partial pressure below 0.07 psia which is the partial pressure that results in a 30% Relative Humidity at 60°F. Therefore, it is possible with the proper manual control device settings to maintain the cabin temperature at various settings between 60°F and 75°F and yet prevent condensation in the exchanger coils.

5.1.2 Method of Determining Size Requirements and Configuration

The heat transfer capacity, Q, of a heat exchanger, may be expressed by the following relationships:

$$Q = U A \Delta t_m \tag{2}$$

where

Q = heat transfer rate, BTU/hr

U = overall coefficient of heat transmission, BTU/hr - °F-Ft.^{2}, which is the series sum of several thermal conductances

A = area of heat transmitting surface

t_m = log mean temperature difference

and

$$t_m = \frac{\Delta t_A - \Delta t_B}{\log_e \Delta t_A / \Delta t_B} \tag{3}$$

where Δt_a and Δt_b are the temperature differences at inlet and outlet ends respectively.

If the temperatures of the fluids entering and leaving are known, Δt_m is determined from Eq. 3. This value of Δt_m and a suitable value of U make it possible to find the area A required to transmit Q BTU/hr. The determination of a suitable value for U is usually a reiterative process since it is composed of the sum of the following thermal conductance coefficients in series:

a. Cold Side surface film coefficient of conductance.

b. A wall conduction component. Very often the wall resistance value is negligible and may be omitted.

c. Hot Side surface film coefficient of conductance including the temperature ineffectiveness on the extended area on this side (the air side in this case).

The above outlined method of area requirement calculation is known as the log mean temperature difference method and has been used extensively in the past. Recently, a second method, the "Heat Exchanger Effectiveness - N T U" method has become the recommended and accepted method for heat exchanger calculation. This method has been used to design the COOL air conditioner heat exchanger. Reference 2 gives a detailed explanation of the method and its background.

The effectiveness - NTU method for the sizing of heat exchangers has certain inherent advantages over the LMTD method: (Reference 2).

a. The effectiveness is a thermodynamically significant parameter (much like an efficiency factor).

b. The effectiveness - NTU method clearly shows the application of both the heat transfer rate equation and the energy balance principles to heat exchanger design.

c. This approach simplifies the calculations involved in predicting the performance of complex flow arrangements.

5.1.3 Description of Configuration Selected

The heat exchanger assembly is depicted in Figures 11 and 12 and the calculations to support the design of the configuration are included in Appendix 4. The heat exchange surface selected is a spine finned tube outer surface with a tube inner surface which is in the form of a set of helical lands and grooves. It is fabricated from aluminum and is presently used extensively as a high performance evaporator surface in General Electric Portable Home Air Conditioners and it was applied in the Discovered Life Cell Air Conditioner. Reference 3 gives reliable performance data for this surface and makes possible the accurate calculation of the surface area requirements. This combination of accurately defined performance characteristics, plus a relatively high overall U (overall coefficient of heat transmittance) and its light-weight construction predicated the choice of this material for a heat exchange surface.

Referring once again to Figure 11 it can be seen that a total of 48 lengths of tube surface are arranged between tube sheets with 180° return bends connecting the liquid side passes. The entire assembly is fabricated to form a single integral assembly which may be easily removed and replaced should it become necessary. The liquid flow configuration through the tubes is composed of twelve parallel paths of four passes each. The air side flow path is a modified cross flow pattern over the tube surface. Figure 13, a side elevation cross sectional view of the heat exchanger plenum, illustrates the attitude of the heat exchanger with arrows to indicate the air flow paths. This arrangement results in a design which allows the maximum air side free flow area to be obtained within the volume available. A maximum free flow area results in minimum air stream velocities across the exchanger surface and, hence, minimum air side friction losses. Friction losses for a liquid-to-liquid exchanger are small because of the comparatively low power requirements involved in pumping high density fluids. However, in the case of the COOL exchanger, a gas-to-liquid exchanger, the air side friction losses involved in moving the low density medium assume importance equal to or greater than the heat transfer characteristics and must be investigated.

The air side heat transfer coefficient h_a, for diatomic gases and forced convection across a tube surface may be approximated as:

$$h_a = 0.24 \, (D. \, \rho \, V_s)^{0.6} \; x \frac{K_f}{D_o} \tag{4}$$

where D_o = outside tube diameter

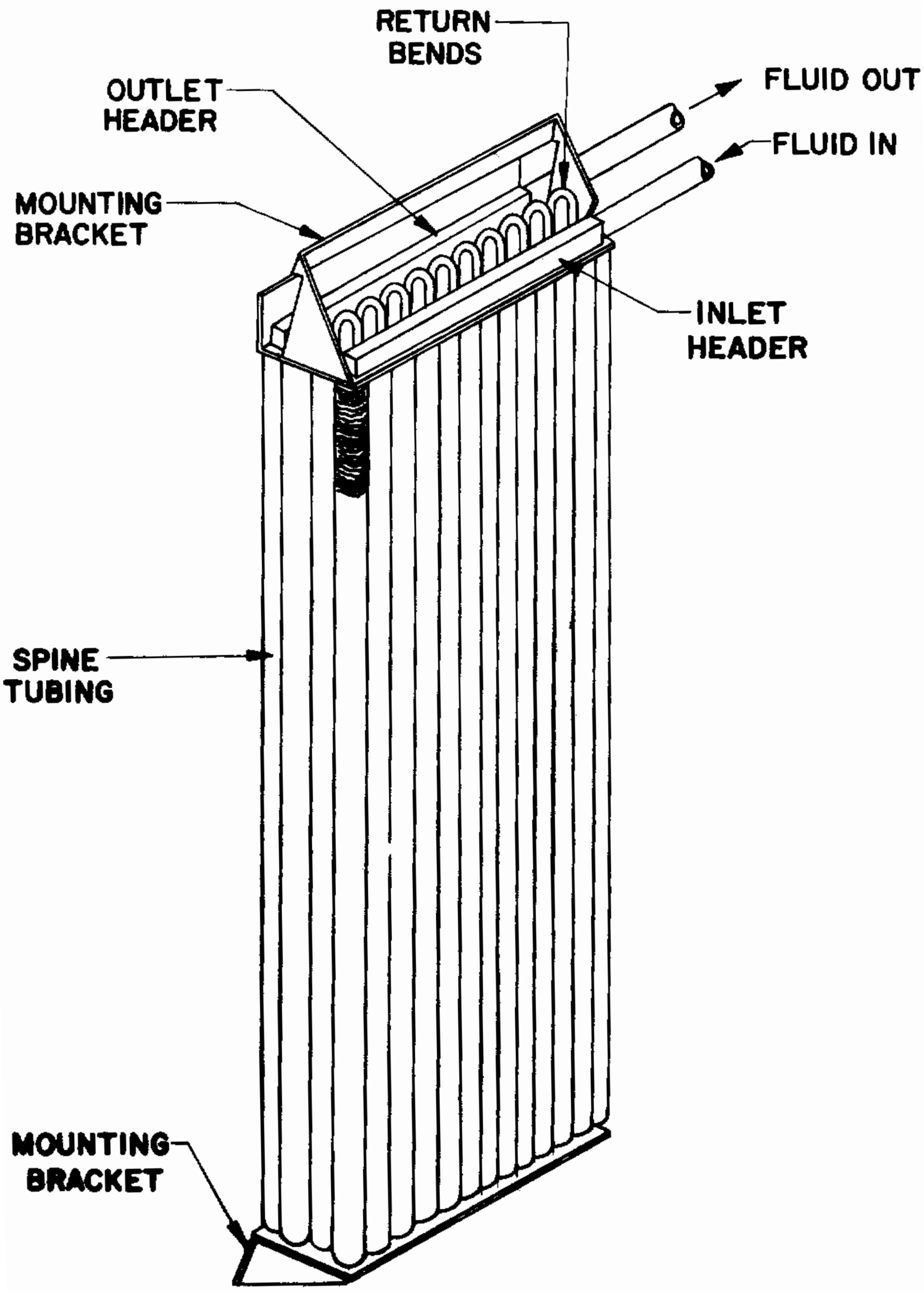


FIGURE II
HEAT EXCHANGER

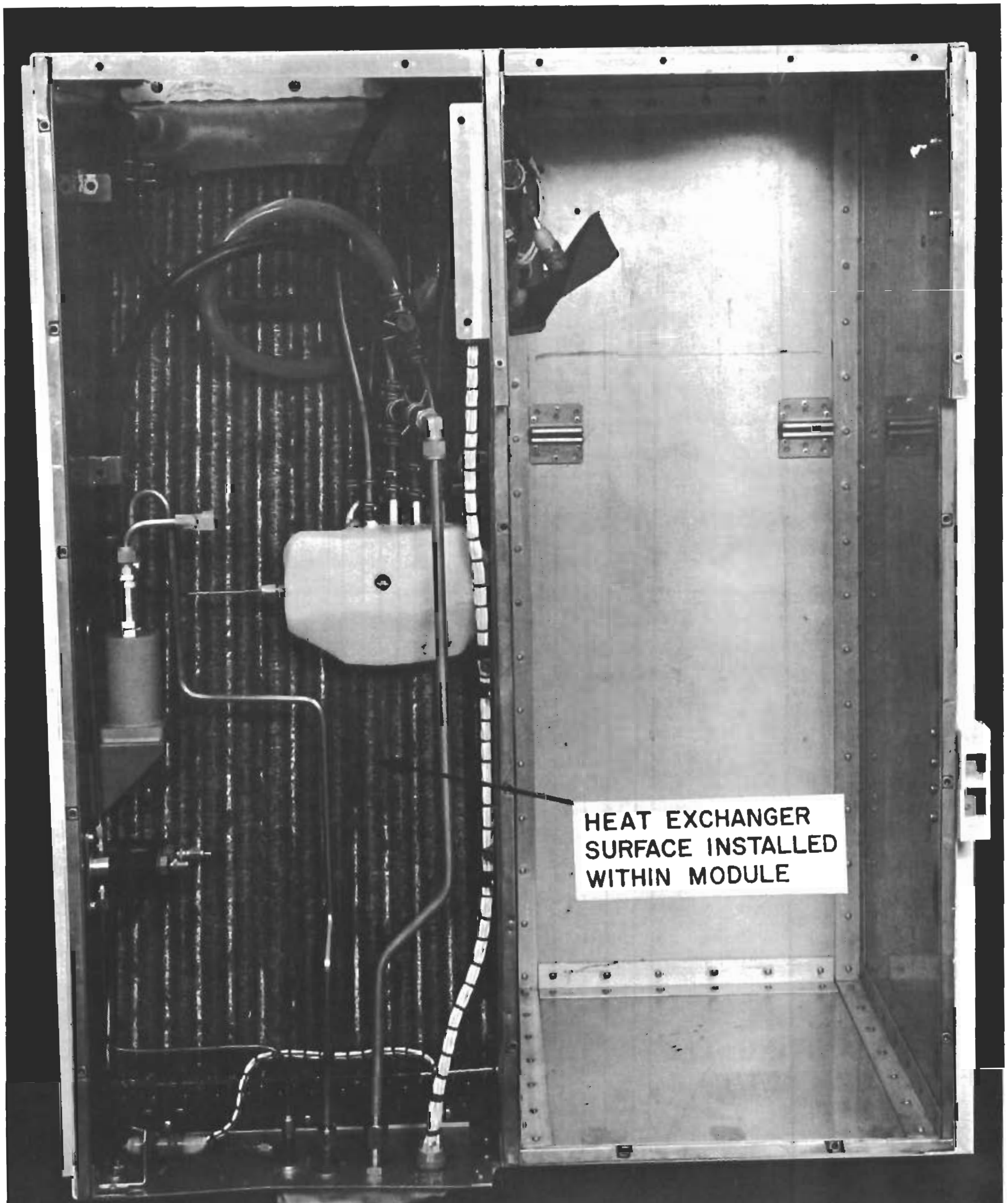


FIGURE 12: HEAT EXCHANGER INSTALLATION

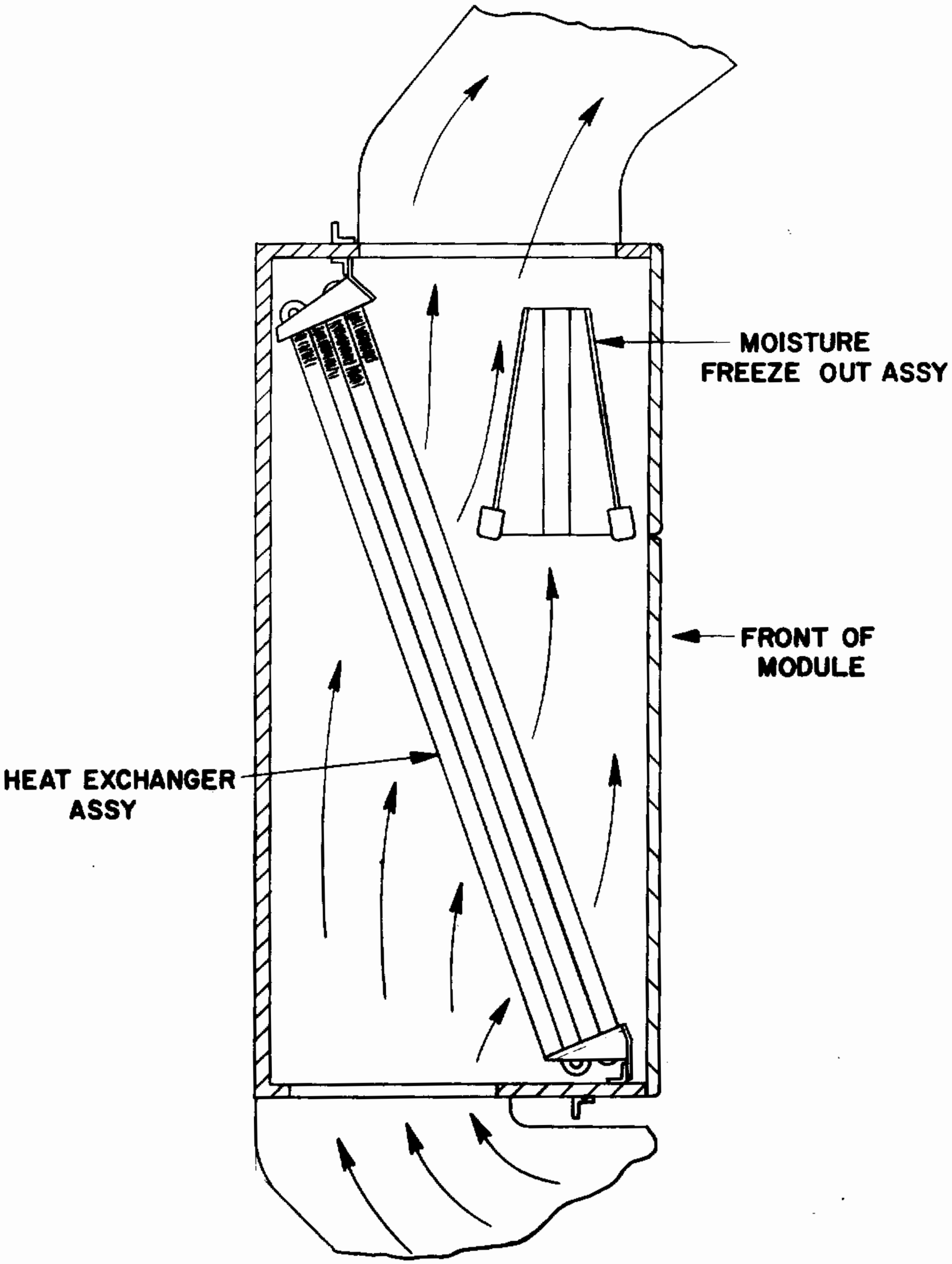


FIGURE 13
HEAT EXCHANGER & FREEZE OUT
INSTALLATION ATTITUDES

e = gas density

K_f = thermal conductivity of gas film

m_f = absolute viscosity of gas film

If gas temperature, pressure and exchanger surface configuration remain fixed the above reduces to $h_a = K (V_s)^{0.6}$

$$\text{where } K = 0.24 \frac{K_f}{D_o} (D_o \rho)^{0.6} \ \& \ V_s = \text{face velocity}$$

The compactness of a heat exchanger is increased by raising its overall coefficient of heat transmission, U, which is equal to:

$$\frac{1}{UA} = \frac{1}{h_a A_a} + \frac{1}{h_f A_f} \qquad (5)$$

when h_a and h_f are air side and liquid side heat transfer coefficients respectively. The thermal resistance of the exchanger surface itself is neglected here. From the above equation it can be seen that the air side coefficient only partially determines the thermal transmissibility of the exchanger and, therefore, the transmissibility coefficient U will vary as some power less than h_a to the 0.6 power. The air pressure drop across the exchanger surface for a given configuration varies approximately as the square of the velocity. Air horsepower, or fan power output is determined from the product of air volume (CFM) and pressure rise. Therefore, it can be stated that for a constant flow rate and specific exchanger configuration, the air horsepower required varies directly as the square of the face velocity.

To summarize in a single statement, it can be stated that if air velocity across a given exchanger configuration is increased, the thermal transmissibility, U, will increase directly as V_s 0.6 (at best) and the air horsepower requirement for a given mass flow will increase approximately as V^2. Therefore, to flight optimize an exchanger the air side surface face velocity must be minimized by designing for the maximum free flow face area that can be packaged within the volume allocated.

This theory may be validated by comparing an alternately sized exchanger with the COOL design. As mentioned previously, the COOL exchanger has a minimum air side heat transfer coefficient (maximum free flow area) which results in an exchanger weight of twenty lbs. The two blowers weigh 21 lbs. combined and consume a maximum of 350 watts. Three hundred and fifty watts require 44 lbs. of power source (solar voltaic cells). This gives a total combined weight of 85 lbs. Now if the air side free flow area were halved the face velocity would be doubled. This increase in velocity would increase the air side heat transfer coefficient approximately 57% and the heat exchanger weight would be decreased to 13 lbs. However, the air side pressure drop would increase by a factor of four. This means that the blower power requirement would be quadrupled assuming that motor and form efficiencies remain constant. The total combined weight is now 13 lbs. (exchanger) + 30 lbs. (blower) + 176 lbs. of power source weight for a total of 219 lbs. It is evident that the minimization of air side friction losses is a prime requisite in the design of space cabin heat exchangers.

The four-pass liquid flow path permits a close approximation to counter-flow performance (the most effective flow configuration, Reference 2) without becoming encumbered by heater and ducting difficulties which accompany counterflow arrangements. The twelve parallel flow paths result in liquid pumping velocities which give Reynolds numbers in the transition range between laminar and turbulent flow. If pump losses were to be lowered by decreasing fluid velocity (24 parallel flow paths for example) the overall coefficient of heat transmission, U, is decreased to a value which requires a larger heat transfer area to handle the 18650 BTU/hr load. The only method of increasing area within the space available is to place more tubes in series across the air flow path of the air conditioner plenum. This would increase the air side pressure drop and result in a fan power increase that would greatly overshadow the liquid pumping power savings.

A very high area compactness ratio (heat transfer area per unit of volume) is obtainable by use of plate-fin extended surface configurations when both fluids are gases and an extended surface can be effectively utilized on both sides of the exchanger. Various plate-fin configurations were investigated and discarded because the COOL heat exchanger is a liquid-to-gas type of exchanger which more efficiently utilizes the finned tube surface. This type has an external surface (gas side) area many times the internal (liquid side) surface area and is desirable because gases characteristically tend to operate with lower surface conductances than liquids and more area is required in the gas side of the surface for a balanced design.

The physical and operating characteristics for the exchanger are summarized here as extracted from the contents of Appendix 4.

Est. Wt. --- 16#
Size --- 4" x 12" x 33"
Air Side Pressure drop = 0.5" H_2O
Liquid Side Pressure Drop = 6.0 ft. fluid
Fluid Flow Rate - 6.0 GPM, 50%-50% glycol water
Air Flow Rate - 900 STD. CFM

5.2 Moisture Removal and Humidity Control

A totally closed environmental system such as the AMRL evaluator or a manned space cabin must have some means of removing the moisture which accumulates in vapor form within the cabin atmosphere. Without such a moisture removal device the cabin water vapor partial pressure would soon reach saturation with resulting moisture condensation on cabin surfaces and uncomfortable living conditions.

There are two main sources of moisture within a closed system:

a. Approximately two lbs. per man per day as exhaled vapor and evaporated perspiration.

b. Water vaporized during food preparation.

The project COOL specification states that cabin Relative Humidity is to be maintained between 30 and 50%. Numerous methods have been proposed to perform this control and removal function under zero-g conditions of space flight. Table 3 presents a condensed compilation of several possible methods. Major advantages and disadvantages of each approach are also cited.

5.2.1 Discussion of Moisture Freeze Out Technique

The moisture freeze-out technique has been selected for development and integration within the COOL habitable environment control system. It is the only method which will permit the recovery of the moisture for re-use without expending any electrical power other than that required to pump the liquid transport fluid through the collector coils. A low temperature fluid already exists in the refrigerator liquid transport circuit. The additional latent and sensible heat load imposed by the freeze-out device does necessitate that the low temperature radiator surface be increased by some 18 square feet. This additional 18 square feet of radiator results in a weight increase of approximately 14 lbs. plus a heat transport fluid weight of 3.5 lbs. The freeze-out device itself is estimated to weigh approximately 3lbs., bringing the total combined weight addition to accomplish this function of 20.5 lbs.

TABLE 3

METHODS OF MOISTURE REMOVAL FROM THE AIR OF AN ARTIFICIAL ENVIRONMENT

Method	Advantages	Disadvantages
1. Absorption of moisture by passing air through a bed of Lithium Chloride.	Capable of absorbing up to 0.6 pounds of water per pound of dry LiCl.	Not recoverable and is lost, for future consumption.
2. Absorption of moisture by passing air through a silica gel.	Water is recoverable by thermal regeneration or silical gel is recoverable by vacuum regeneration.	Absorbs only 15% of dry weight thermal regeneration requires 0.052 KW per man. Vacuum regeneration all water is lost to space.
3. Absorption of moisture by passing air through a molecular sieve.	Water is recoverable by thermal regeneration or the sieve is recoverable by vacuum regeneration.	Absorbs only 10% of dry weight thermal regeneration requires 0.066 KW per man. Vacuum regeneration all water is lost to space.
4. Condensation and Freeze-out of moisture on a cold surface.	Water is recoverable and a low temperature fluid is available to condense and freeze the moisture.	A method of water collection must be developed.

The moisture freeze-out collection surfaces are illustrated in the air conditioner plenum on the downstream side of the heat exchanger coils. The cold surface of the freeze-out plate is maintained at a temperature of approximately $0^{O}F$ and each has approximately 80 square inches of active area. A fluid carrying coil is dip brazed to the back side of the freeze-out surface and is insulated with one inch of polyurethane foam to prevent frost collection on the "back" side.

The freeze-out surface is shrouded from the main air stream. An opening around the perimeter of the shroud permits the migration of vapor molecules to the cold surface.

The lowering of the water vapor partial pressure in the area immediately adjacent to the cold surface, by condensation and freeze-out, causes vapor molecules to flow to this area of lowered vapor pressure in an attempt to reach an equilibrium pressure. Thus, continuous freeze-out and control is assured without the necessity of imposing a restriction on the entire conditioned air stream. This means that moisture can be removed even when there is no requirement for cabin cooling.

5.2.2 Automatic Moisture Removal and Recovery Apparatus

The technique which utilizes moisture freeze-out to maintain humidity control necessitates the use of some means that periodically and automatically defrost the freeze-out surfaces. If frost accumulations were not periodically removed from the freeze-out plate its efficiency would be lowered and in time the frost accumulations would build up sufficiently to restrict the air flow paths surrounding it. Furthermore, an additional source of potable water is available here for recovery. The automatic moisture removal and recovery apparatus is comprised of two finned plate surfaces (one active and the other defrosting) exposed to the conditioned airstream as it exhausts from the air conditioner heat exchanger. The temperature of each surface is independently maintained within narrow limits at either of two temperature levels. The active "freeze-out" plate is maintained at approximately 0°F by allowing fluid from the low temperature liquid transport circuit to flow through its tube paths. At the same time, the inactive (defrosting) plate has its tubes filled with fluid at approximately 50°F flowing from the high temperature liquid transport circuit. Each plate has sufficient area to efficiently freeze-out excess moisture from the cabin atmosphere and maintain humidity between 30 and 50% for a period of approximately one hour. At the end of one hour, the "active" plate is switched to its defrost phase of the cycle and the defrost plate assumes the function of freezing out moisture. The switch is accomplished by means of an automatically timed valve which alternately reverses the fluid flow paths of each of the plates from low temperature to high temperature circuit or vice versa. As the frozen moisture on the defrosting plate is thawed, it flows as a thin film between the accumulated frost layer and the surface of the plate. The flow direction of the liquid is determined by the suction side of a small pump which is manifolded to the surface of the plate. The pump suction is switched to the appropriate collection surface by means of the same timing devices which programs the liquid transport fluid switching valve. A recovered water storage reservoir to receive the discharge from the recovery pump completes the equipment necessary to provide a fully automatic moisture removal and recovery subsystem.

The automatic defrost technique outlined above is accomplished with a minimum of additional power consumption since the high temperature liquid transport circuit provides a sufficiently high temperature source of heat energy to melt the frost accumulations without expending any electrical power. The power to actuate the flow switching valve, programmer and recovery pump totals some 15 watts. No additional liquid fluid pumping electrical power is necessary.

The removal of excess moisture from the atmosphere of a space vehicle cabin by means of freeze-out techniques will function equally well under both zero g and laboratory ground test conditions. The frozen moisture removed from the atmosphere as a solid poses none of the problems associated with confinement of liquid matter. However, automatic collection and recovery of this moisture in the absence of gravitational field requires the solution of several problems, the solution of which can only be positively validated by actual zero g tests. The induction of air into recovered water storage vessels during collection is undesirable since it will result in only partial filling of drinking vessels and the feeding containers used for the reconstitution of dehydrated foods. Unfortunately, the major problem associated with liquid collection under zero g is that of separating the liquid and gaseous phases of matter and the successful solution of this problem is the key to obtaining an efficient zero g operable moisture collection system. Following is a description of an approach which will provide the solution to these problems. As mentioned previously, the frost layer accumulates on the exposed side of a plate until the freeze-out efficiency of the plate is lowered. At this time, the defrost phase of the cycle is initiated by switching the liquid transport fluid flow through the plate coils from the low temperature to the high temperature liquid transport circuit. Melting of the frost layer now begins mainly at its interface with the freeze-out plate because heat transfer coefficients on the air side of the frost layer will be considerably less than those at the freeze-out plate during defrost due to molecular adhesion (surface tension) forces between the liquid film, the plate and frost layer. The moisture recovery pump draws from this liquid film formed between the frost layer and the freeze-out plate.

During its induction stroke the recovery pump draws a mixture of the condensed moisture from the plate and cabin air as well. This induced air must be separated from the recovered liquid. An effective means to separate the entrapped air has been incorporated in the main water storage reservoir of the water supply system.

The 2-1/2-gallon storage reservoir contains a spring loaded piston with a sliding diaphragm seal. Pressure is maintained within the reservoir at between 1 and 5 lb/in^2 gage. The low sliding friction forces and minimum hysteresis characteristics of the diaphragm permit the piston to compensate for changes in fluid storage volume without sticking even though it is operating with such a low pressure differential across it. The nonwettable permeable barrier in the head end of the reservoir acts as a "phase separator" which permits entrapped air which has been recovered along with the water from the moisture collector to escape from the reservoir. This barrier will not permit the passage of water if the pressure differential across it is kept below approximately 20 lb/in^2. Above this pressure differential, the barrier will become "wetted." That is to say, if the pressure forces across the barrier are greater than surface tension forces between molecules, the liquid phase will also flow through the barrier. Thus, the use of the phase separating barrier permits a supply of water to the dispenser outlets which is essentially free of entrapped air.

The inlet fitting arrangement to the reservoir is such that the incoming air-water mixture is sprayed across the surface of the semi-permeable membrane at the end of the reservoir. This assures that the entrapped air is exposed directly to the membrane surface and thereby hastens the escape of the entrapped air from the reservoir. The water discharge fitting is located so that water leaves the reservoir at a point approximately 2" away from the membrane end of the reservoir. The anodized aluminum housing is immune to corrosion and will not contaminate the water supply. Spare inlet fittings are provided so that other water recovery sources can be fed into the storage reservoir with a minimum of rework.

5.2.3 Conditioned Air Circulation and Distribution

Two blowers (Figure 14) installed to give parallel flow delivery of the conditioned air are supplied within the system. The blowers give sufficient mass flow at cabin pressure levels between 0.5 and 1.0 atmosphere to convey the maximum anticipated cabin sensible heat load (18000 BTU/hr) from the cabin heat sources to the cabin heat exchanger coils.

The blowers are of the vanaxial type which are packaged with a direct drive motor mounted within the fan shroud. These integral assemblies are of aircraft construction, fabricated mainly from aluminum and weigh 10.5 lbs. each. Design characteristics when the motors are powered from a 28 VDC source are as follows:

H.P. = 0.2
Speed = 5000 RPM
Power Requirement = 240 watts
Design Point Operating Characteristic = 840 CFM delivered at a static pressure of 1.0" water gage.

Originally, the blowers were sized so that either of them was capable of delivering the required mass flow through the cabin air conditioner coils. The second blower was installed merely as a standby in case of failure of the first unit, or if under certain heat load - cabin pressure combinations, additional mass flow was required through the heat exchanger coils.

However, during the performance of system check out tests it was determined that the noise level resulting from a signle blower operating at 28 VDC was high. As a result both blowers were energized by wiring them in series across the 28 VDC supply. The air mass flow delivered by this parallel air flow delivery series electrical arrangement was adequate (approximately 1000 cfm) and the noise level was lowered considerably. A longer motor brush and bearing life can be expected due to the lower rotational speed resulting from the 14 VDC potential across each of the motors. Also the total blower power consumption under these conditions is 200 watts as opposed to a power drain of 240 watts when one blower is operated at 28 VDC.

6. Space Radiators and Heat Sinks

As mentioned previously the fabrication of the COOL system as a package for integration aboard the AMRL Evaluator was implemented to physically prove the flight suitability of the direct radiation to space concept. The system equipment was delivered for installation as flight functional hardware with the exception of the high temperature circuit space radiator.

6.1 Low Temperature Radiator Description

The food storage compartment and the humidity control apparatus require an inlet fluid transport temperature of -10°F and a fluid pumping rate of 0.2 gallons per minute as determined previously. The design heat rejection capability of this radiator is 1000 BTU/hr (a 37% safety factor) which can be dissipated to a near earth orbit spatial heat sink of -100°F by a radiator configuration with an area of 29 ft^2. The effective space heat sink temperature was established as typical value for a near earth orbiting vehicle (See Appendix 1). The radiator surface (Figures 15 and 16) is a 24-inch diameter x 30-inch long hollow cylinder

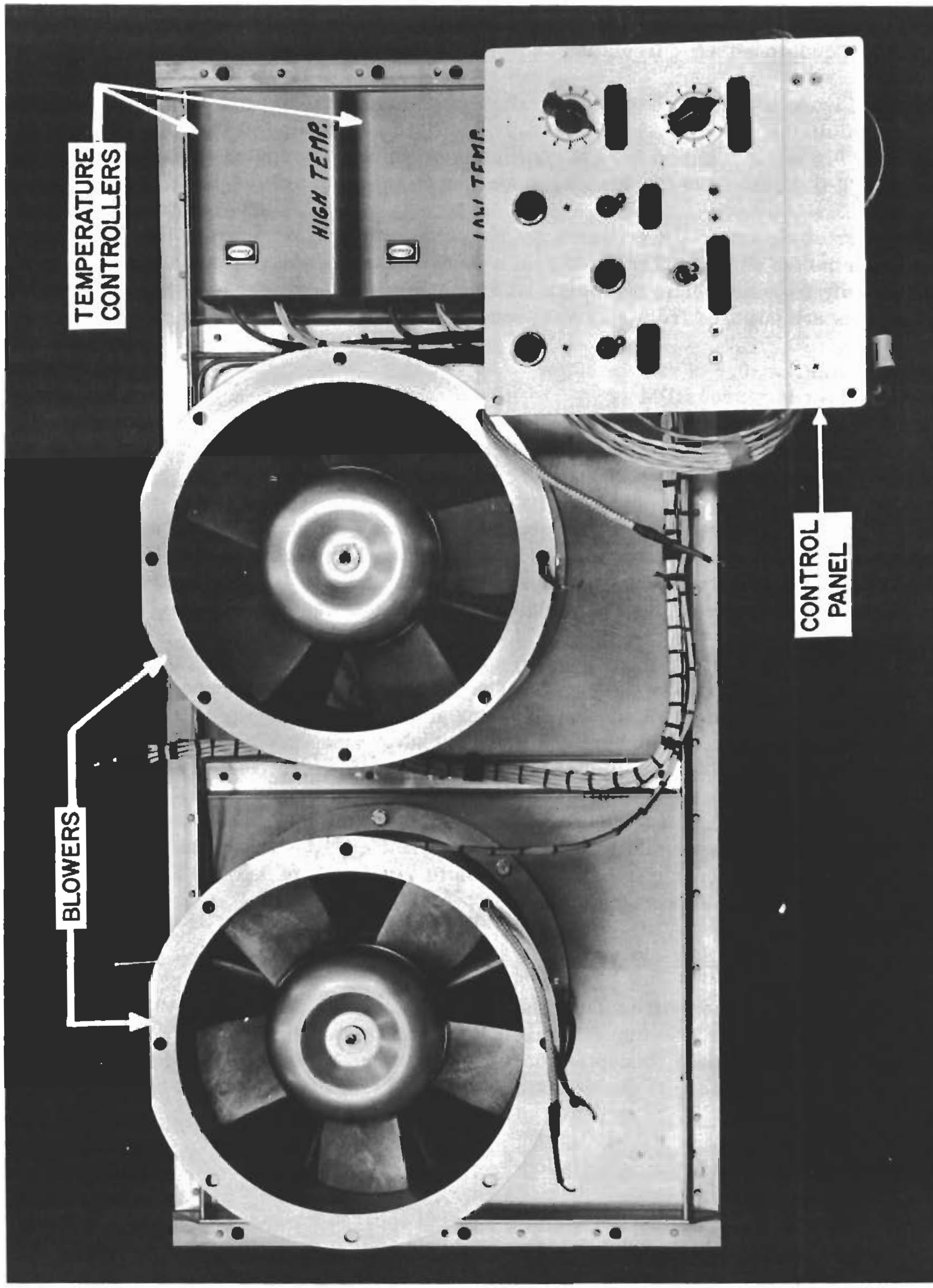


FIGURE 14: AIR CIRCULATING BLOWERS

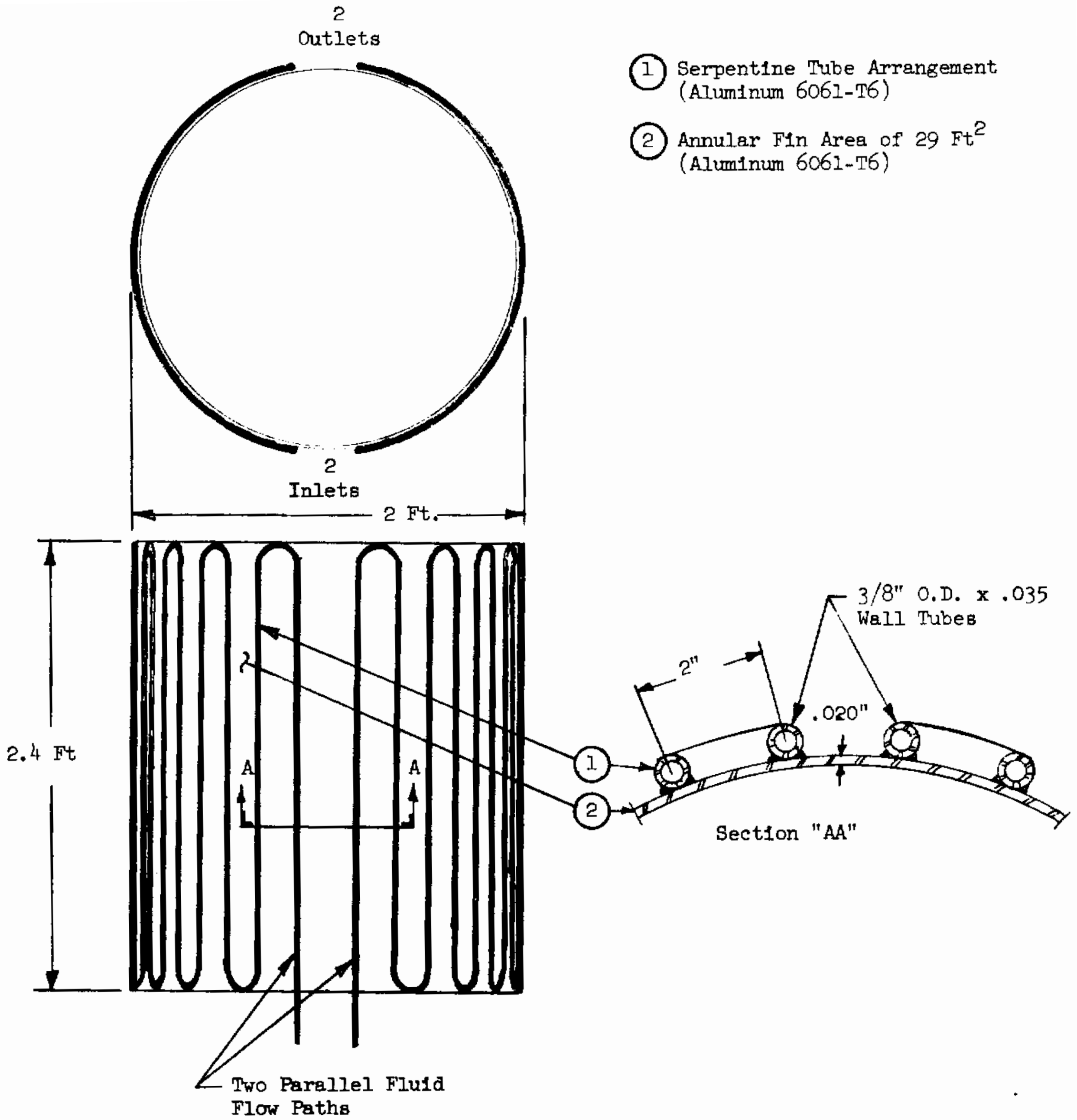


FIGURE 15 LOW TEMPERATURE RADIATOR

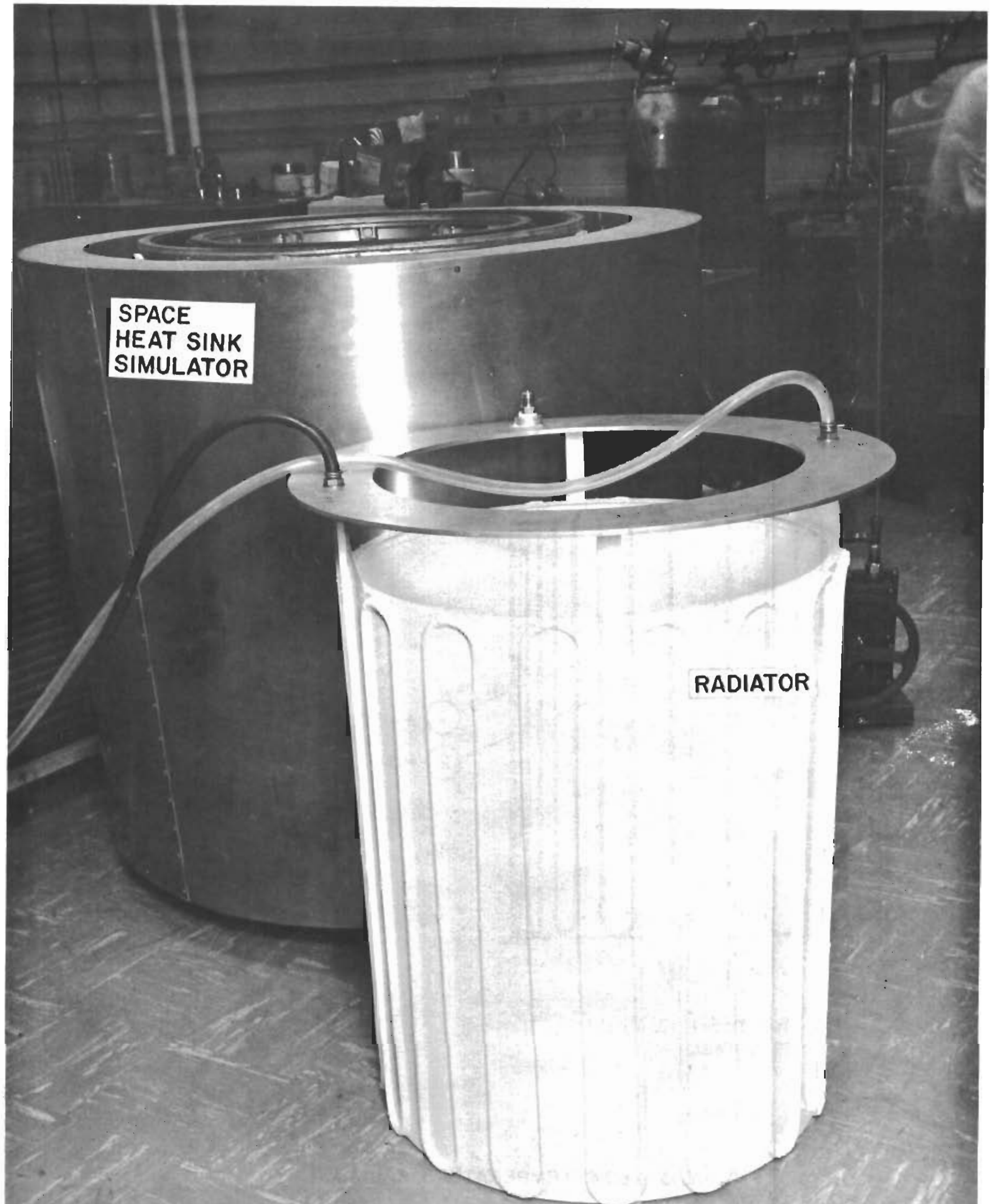


FIGURE 16: LOW TEMPERATURE RADIATOR INSTALLATION

of .020 inch 6061-T6 aluminum, a configuration which readily lends itself to installation within a space heat sink simulator. The radiator tubes are 3/8" OD x 0.035" wall arranged in a serpentine pattern on the surface of the cylinder on 2-inch centers. Approximately 100 feet of tubing were required to fabricate the two parallel fluid paths with each path bonded to the outside of 180° segment of the cylinder. The two parallel flow paths are necessary to maintain laminar flow conditions within the tubes.

The radiator was fabricated by roll forming its fin surface from a single sheet and then bonding on the fluid carrying tubes with an aluminum filled epoxy. The resulting structure was sufficiently rugged to withstand normal handling during use and yet have heat rejection characteristics which would be comparable to actual hardware as discussed later under the radiator design considerations section of this report. Its total weight including heat transfer fluid is 12 lbs.

6.1.1 Radiator Design Considerations

The radiator configuration was established by the trade off of several criteria: (1) area and volume, (2) weight, (3) fabrication and (4) structure. Actually, these are not separate and distinct items, but are closely interwoven. The design limitations they require are discussed briefly here. Formulas and calculations are shown in Appendix 3. In some designs, space limitations could dictate the maximum area, shape and volume in which the radiator must fit. In this study, no such limitations were made. It had been agreed, however, to have a tube fin construction with radiation from two surfaces. The total weight of the radiator is the sum of the weights of fins, tubes and fluid. Since the radiator was designed for space use, minimum weight was a major criteria. In the final selection of radiator size, however, minimum weight was compromised to provide for handling qualities to give a suitable test configuration.

Fabrication Methods Considered

Several configurations were investigated as shown in Figure 17. Figure 17a shows tubes welded to fins at the junction of tube and fin. For sheet thickness of 0.020" or less, this technique would be extremely difficult if not impossible. Figure 17b shows tubes and fins extruded as one piece, then seam welded to each other at the fin which would probably permit much easier fabrication, but end connections of tubes would be difficult.

Figures 17 g and 17h show more seam welding techniques. In the former, the tube pattern is essentially embossed with the top and bottom sheets welded or bonded together. Figures 17c through 17f are designs which can be joined together by epoxy bonding (the method utilized for COOL) or dip brazing.

Radiator Material

One of the criteria for a radiator material is its ratio of thermal conductivity to weight. The higher this ratio, the lighter the construction can be. Table 4 shows the ratio for several common materials.

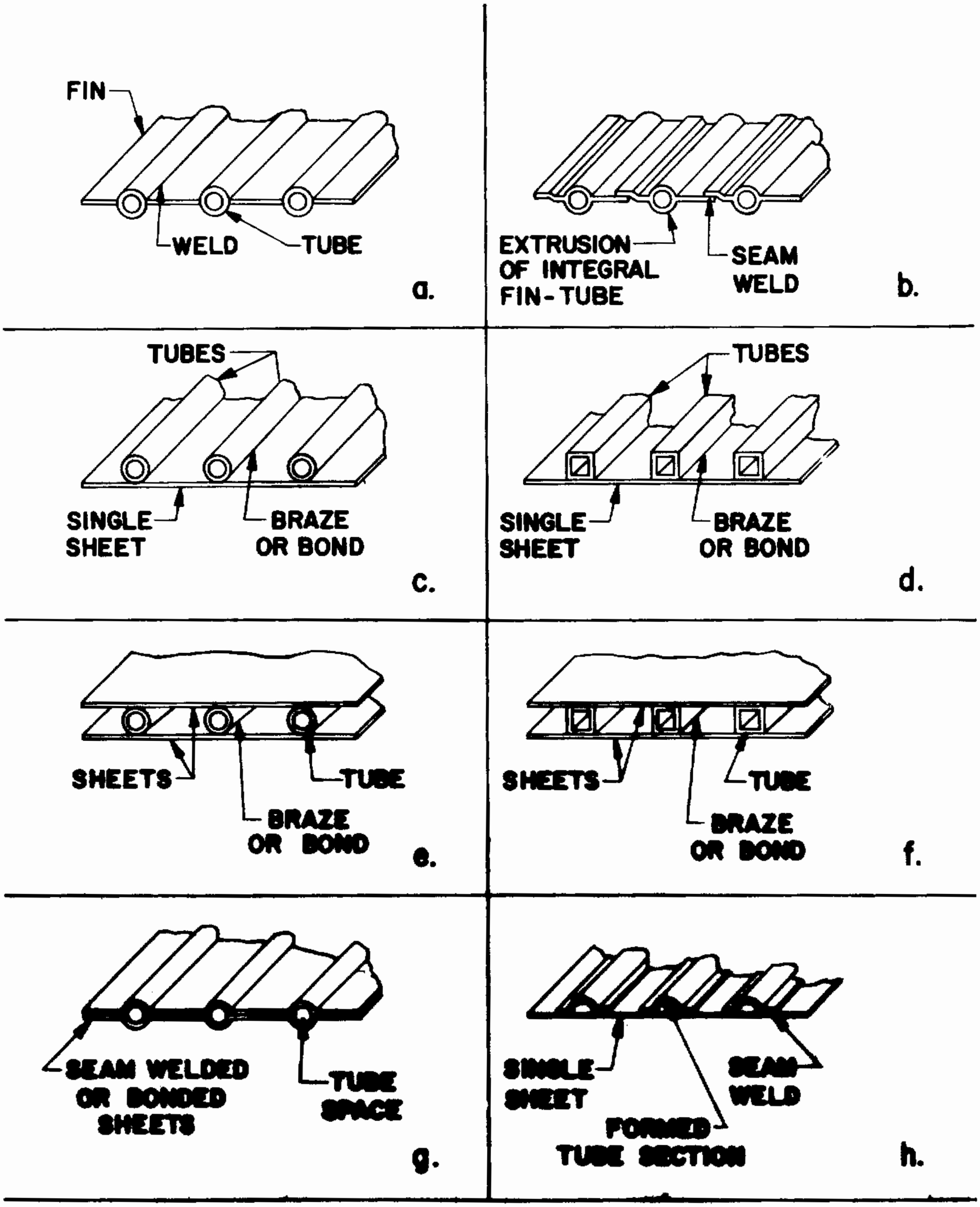


FIGURE 17

RADIATOR CONSTRUCTION METHODS

TABLE 4

RATIO OF THERMAL CONDUCTIVITY TO DENSITY OF SEVERAL METALS

Material	Conductivity BTU/ hr. ft. °R	Density ρ lb./ft^3	k/ρ
Aluminum 1100-0	128.	169	0.758
Al Alloy 3003 H -4	91.5	171	0.536
Al Alloy 2024 T -3	70.0	173	0.405
Al Alloy 6061 T -6	89.0	169	0.526
Magnesium (pure)	90.7	110	0.825
Mg Alloy AZ 31B	55.6	110	0.506
Copper	223.	556	0.401
303 Stainless Steel	9.4	500	0.019

Another criterion is the ratio of modulus of elasticity E to density ρ. This ratio indicates the resistance to penetration by meteoroids. The higher the ratio the lighter the (fluid bearing) tubes. Several metals are compared in Table 5.

TABLE 5

RATIO OF MODULUS OF ELASTICITY TO DENSITY OF SEVERAL METALS

Material	Density ρ lb/ft^3	Mod. Of Elast. E lb/in^2	E/ρ
Al Alloy 6061 - T 6	169	10×10^6	5.91×10^4
Mg Alloy AZ 31B	110	6.5×10^6	5.90×10^4
Copper	556	17×10^6	3.05×10^4
303 Stainless	500	29×10^6	5.80×10^4
Beryllium	115	42.7×10^6	37.1×10^4

From Table 4 the lightest fin would be made from pure magnesium which has the highest ratio. When alloys are considered, aluminum 3003 and 6061 are best.

From Table 5 the lightest tubes would result if beryllium were used; however, beryllium is very difficult to fabricate, is poisonous and expensive. Aluminum and magnesium alloys follow beryllium and are about equal in E/ρ ratio with steel very close behind. Thus, theoretically, pure magnesium tubes and fins would be desired for the lightest radiator; however, pure magnesium is not readily workable. One of its alloys, such as AZ 31B could be used. But a radiator of magnesium alloy will be heavier than one of aluminum alloy. Since 6061 aluminum is workable and can be welded or brazed, it was selected as the radiator material.

Radiator Finish

In addition to requirements of stability, durability, and ease of application, the surface treatment must include the requirements of space use and emissivity. The emissivity of the radiator surface will greatly affect the required area, as explained in Appendix 3. A satisfactory finish was easily obtained with a particular paint composed as follows:

Pigment: Rutile (T_i O_2) 45% by volume

Vehicle:	Dow Corning 808 Silicone Resin
Thickness:	3 to 5 mils

This coating applied to aluminum has undergone limited thermal cycling from 200° °F to -200°F without degradation of performance.

Meteoroid Protection

The subject of meteoroid protection is one of controversy because it is based on probability. The prediction of a meteoroid hitting a vulnerable section of the radiator is an extrapolation from measured data of the frequency of meteoroids hitting the earth. The vulnerable portion of the radiator is the tubing, since a tube puncture would cause loss of heat transport fluid. A fin puncture would have negligible effect on the radiative capabilities.

The depth of meteoroid puncture is a function of the meteoroid size and velocity, and the modulus of elasticity of the protective material. The large meteroids (mass of 1×10^{-7} grams or more) occur so infrequently in space, that their probability of hitting a small area, such as a radiator tube, is negligible. Protection against the small high-velocity particles can be most simply provided by using a tube of sufficient wall thickness which can be determined from the following equation:

$$d = 0.0436 \left[\frac{A_v \ \tau}{1 - P}\right] 0.3 \tag{6}$$

where d = wall thickness of aluminum tube, inches
A_v = projected area of tube, ft^2
τ = time of mission, years
P = probability of return with no penetration of tube wall in time τ, expressed as a decimal.

The constant 0.0436 takes into consideration the tube material, number of meteoroids per square foot of Earth's surface, and the estimated mass of the meteoroid. There is doubt as to the accuracy of the number of meteroids and their specific gravity in space; however, Equation 6 will give conservative results.

In the feasibility study for FROST (Appendix 1) a multi-walled radiator structure was discussed. In this construction, a protective wall is held at a short distance from the vulnerable area. The theory is that if a meteoroid hits and pierces the protective wall, the meteoroid will shatter. The resulting smaller particles would have less kinetic energy; thus, the tube wall would not have to be as thick as shown by Equation 5.

By adding a second tube circuit, equally capable of handling the fluid and heat transfer, increased reliability can be obtained. For example, if a single tube circuit is designed so that the probability of puncture (1-P) is 0.05, the probability of puncture of two such tube circuits is the product (0.05) (0.05) = 0.0025 or 99.75% safe. Vulnerable area will double for two circuits and call for an increase in d (and weight) of $(2)^{0.3}$ or 1.23. But to go from (1-P) of 0.05 to (1-P) of 0.0025 for one tube circuit would increase d by $(1-P)^{.3}$/(0.05) or 2.46 times. Thus, if very high reliability is desired, it can be obtained with redundant circuits at less tube weight. The total weight may not be reduced, however, since twice the fluid would be needed, and valves or controls would have to be added.

For COOL, a redundant low temperature radiator circuit was not included. Even if the radiator were to be used in space and punctured, the thermal mass of the food would enable a safe return of the astronauts. Also, the use of dehydrated foods will give a partial food supply. For a long mission, redundancy or repairability would be desired.

6.2 Low Temperature Heat Sink

The dissipation of the low temperature system heat load from the radiator to a simulated space heat sink is desirable in order to prove the validity of the design concepts and configuration. In order to simulate a spatial environment, two characteristics are required from the heat transfer point of view.

a. A low pressure of 10^{-2} mm Hg will eliminate the conduction and free convection to the heat sink wall of the simulator.

b. A heat sink temperature of -100°F, simulation of a 200-600 mile earth orbit can be obtained economically by a method using the sublimation of dry ice in a methyl alcohol solution which surrounds the walls of the space heat sink simulator.

Because of the size of the radiator and the high vacuum environment that must enclose the radiator, a cylindrical chamber was designed to withstand the atmospheric pressure load. The chamber (Figures 18 and 19) is built up of a series of concentric annular cylindrical volumes. The inner cylinder is filled with a dry ice and methyl alcohol solution. The middle chamber volume contains the radiator, whose pressure is reduced to 10^{-2} mm Hg for test purposes. The outer annular volume is designed to contain a volume of dry ice and methyl alcohol solution that is equal to the volume within the inner volume.

Dry ice sublimes at -109°F. This temperature is maintained as long as some--dry ice remains as a solid within the solution. Sublimation rates as expected should not require addition of dry ice to the chamber more often than once a day.

6.3 High Temperature Radiator

As mentioned before, a high temperature radiator is not to be applied as part of the COOL hardware; however, the design of a typical radiator configuration is presented here.

The AMRL evaluator dimensions are 8 ft. outside diameter by 20 ft. long. If the radiator were so constructed that the fin of the radiator and the outer skin of the cabin were the same, the maximum available surface for radiator is 503 square feet. For the Liquid Transport System, the required radiator area is 294 square feet total (the low temperature system requires 29 square feet and the high temperature system requires 265 square feet) which is 58.5% of the available 503 square feet if the radiator were made an integral part of the vehicle's skin as in Figure 20.

Figure 21, a schematic layout, shows three radiator sections in parallel. This would reduce the radiator vulnerability. For example, if one section were hit by meteoroids, valves would close, sealing off the section. The remaining units would continue to operate at a higher load per section until the break could be repaired or the mission completed. Varying the number of sections would vary the complexity of the system,

1. POLYURETHANE FOAM INSULATION - 6 INCHES THICK OVER ENTIRE SIMULATOR.
2. OUTER ALUMINUM CONTAINER FOR METHYL ALCOHOL AND DRY ICE GIVING A SINK TEMPERATURE OF BELOW -100°F.
3. VACUUM EVACUATED CHAMBER TO 10 $^{-2}$ MM HG.
4. RADIATOR SEE FIGURE.
5. INNER ALUMINUM CONTAINER FOR METHYL ALCOHOL AND DRY ICE.

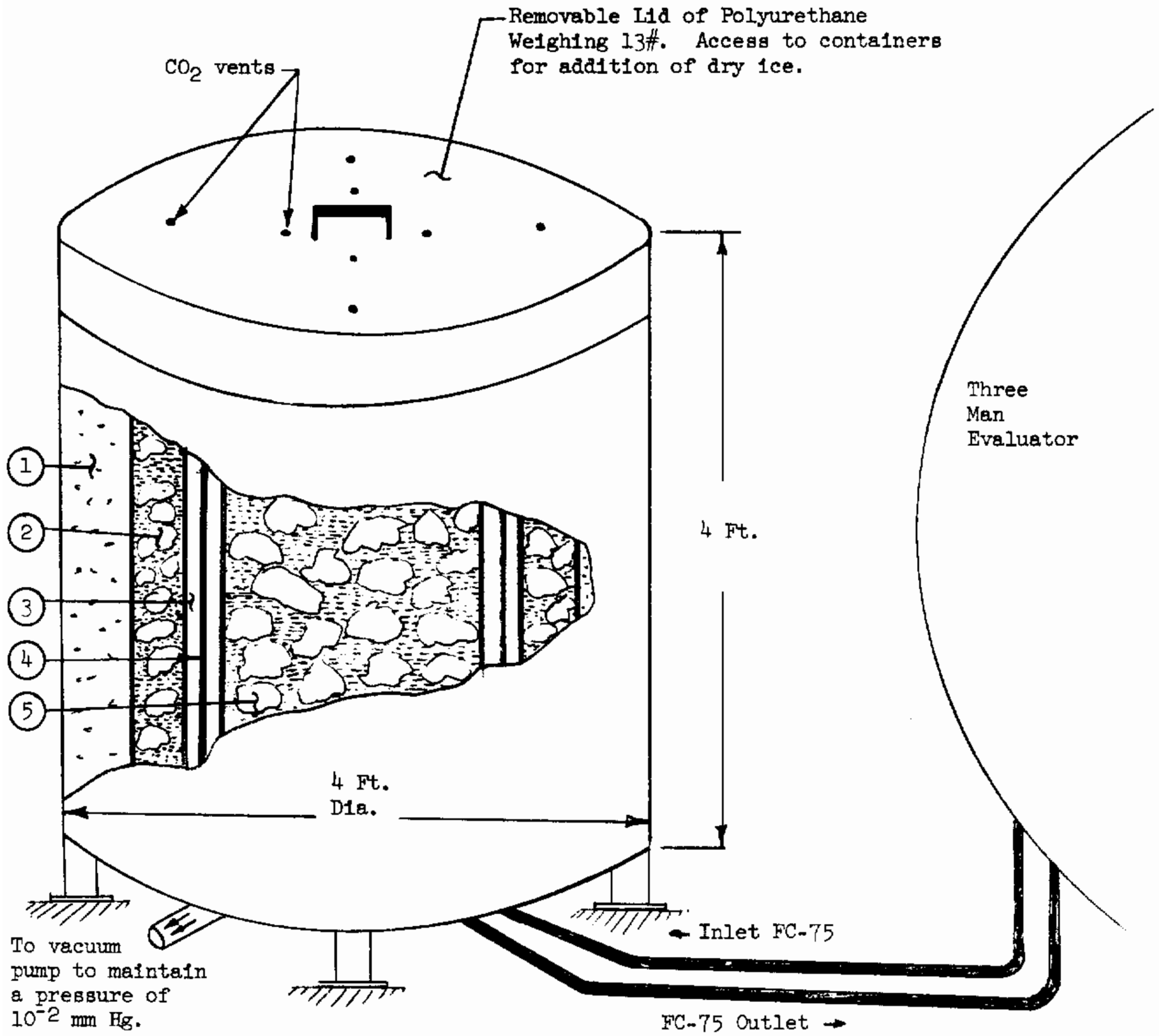


FIGURE 18 LOW TEMPERATURE HEAT SINK

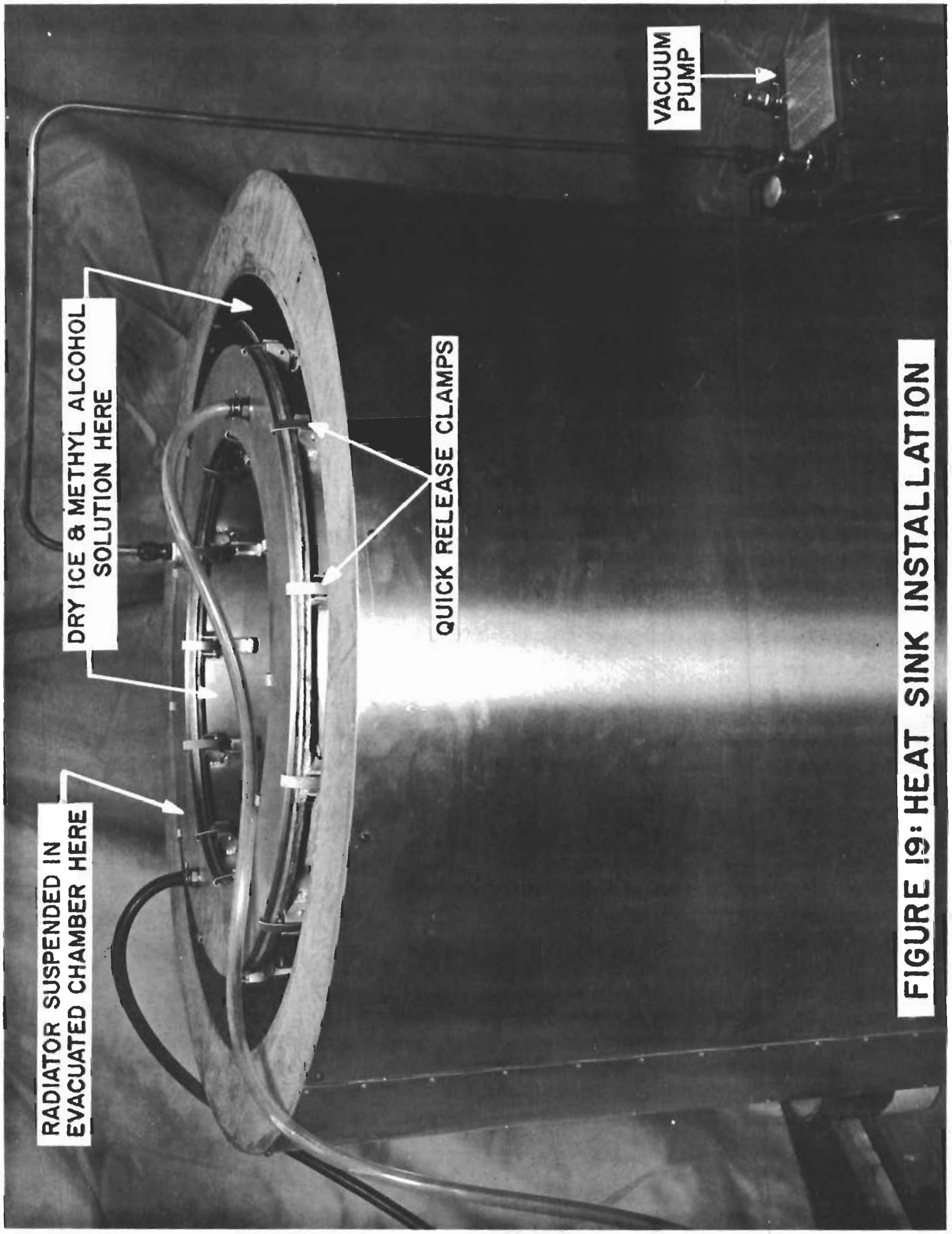


FIGURE 19: HEAT SINK INSTALLATION

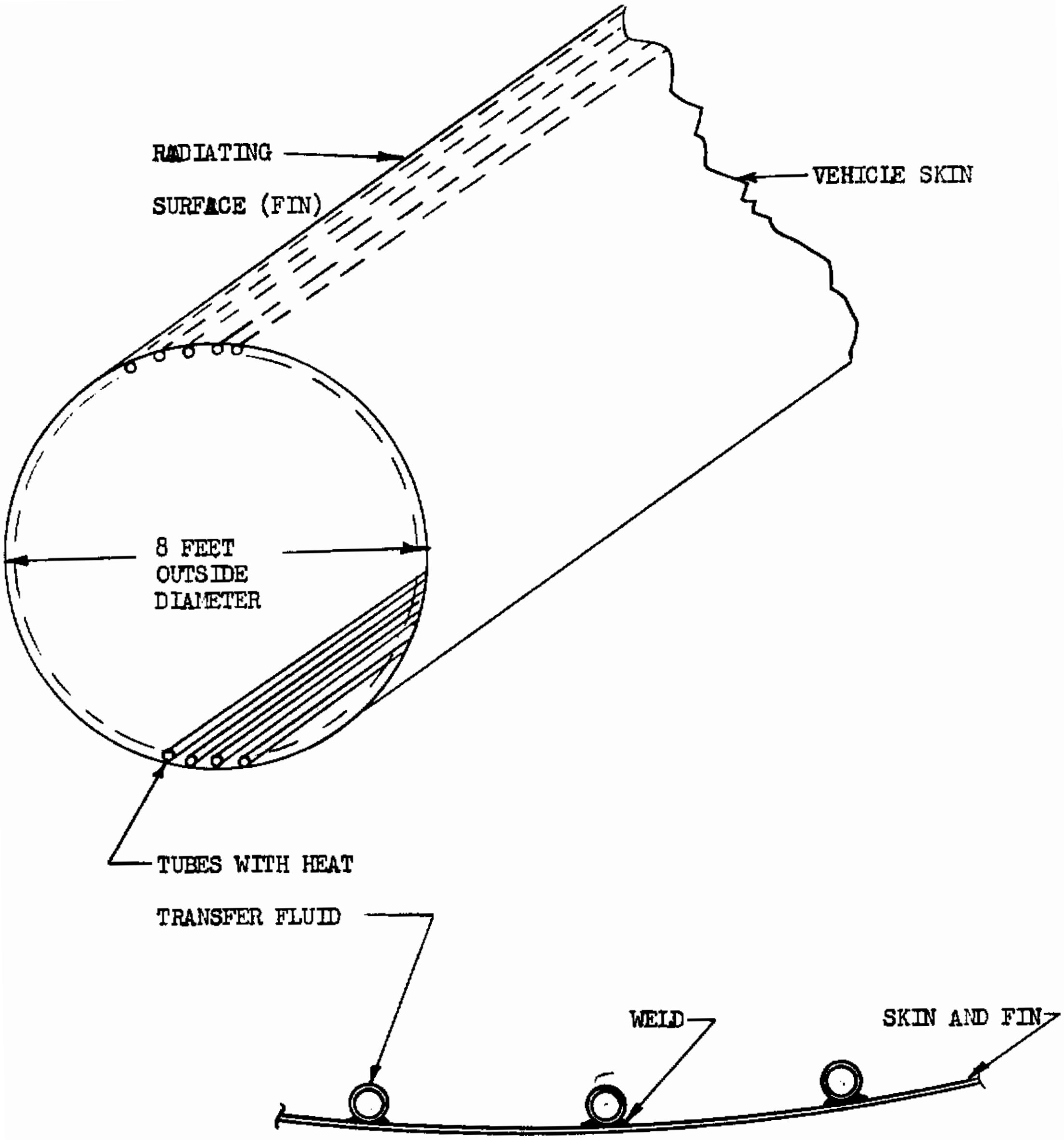


FIGURE 20 HIGH TEMPERATURE RADIATOR CONFIGURATIONS

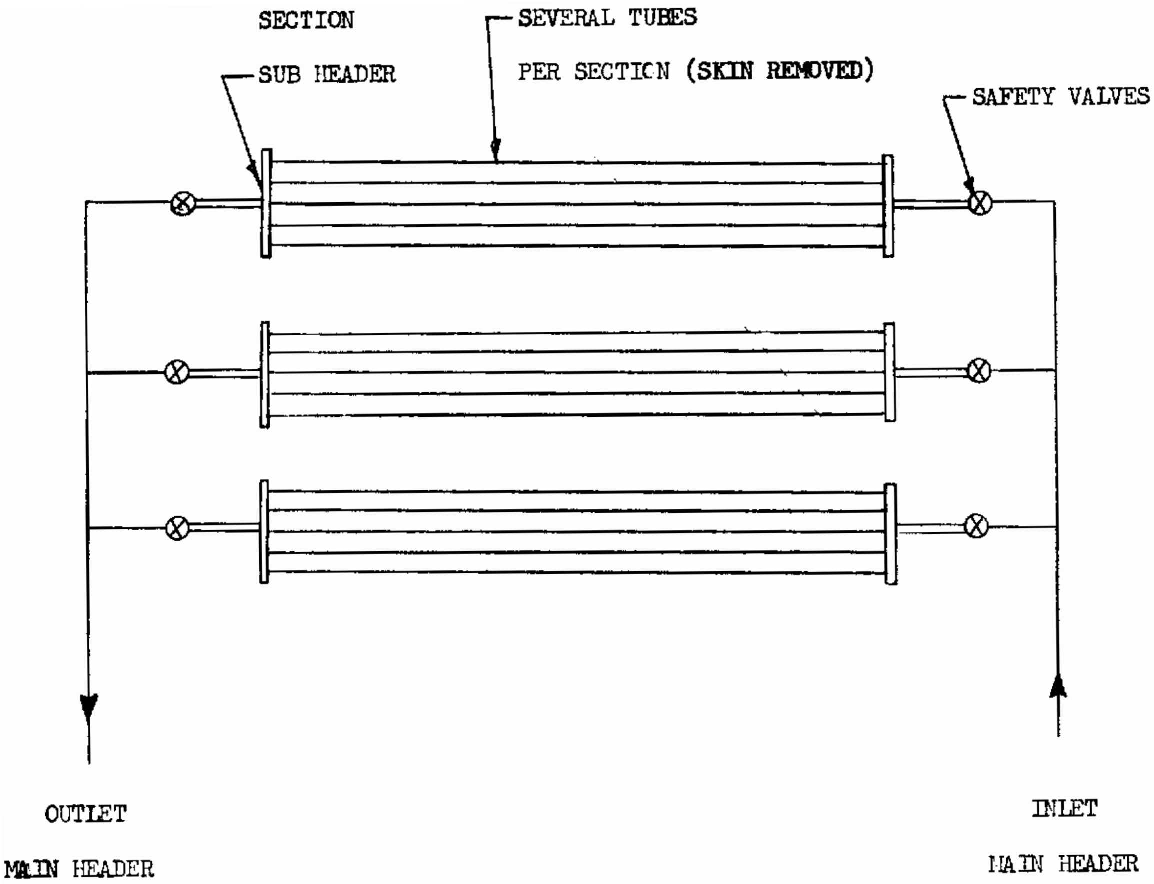


Figure 21. Schematic of Radiator Sections

so that a trade-off of safety versus complexity would be required to satisfy particular mission requirements. Also only about one half the radiator surface area can be used to dissipate the heat load to a spatial heat sink; the other half would be facing the sun and would require the shut-off valves to separate this area from the remainder of the system. An alternate system would be the use of a thermal shield mounted between the sun and the radiator which would shadow the radiator area and also give the main vehicle protection from meteoroids.

Specific calculations to determine a radiator size adequate to handle the 18,650 BTU/hr gross high temperature load at temperature levels and fluid flow rates which are compatible with those required by the heat exchanger are given in Appendix 5.

6.4 Method Used to Simulate the Space Radiator and Its Heat Sink

The evaporator used to duplicate the high temperature system space radiator is of the dry expansion tube-and-shell type with the liquid transport medium circulated through the tubes inside the shell. The refrigerant in the shell absorbs heat from the medium. The tube area, inside the evaporator, required for this system is 45.5 ft^2 and is contained in a shell 6 inches in diameter and 100 inches long (Appendix 6). The commercial hermetic condensing unit used to duplicate the space heat sink has a nominal 35,000 BTU/hr capacity with a 40^{o}F evaporator suction temperature. A thermostatic expansion valve is used to control evaporator temperature. The compressor drive motor is rated at three horsepower and operates on 220 VAC single phase power. Figure 22 shows the condensing unit and evaporator as it was set up for laboratory evaluation tests.

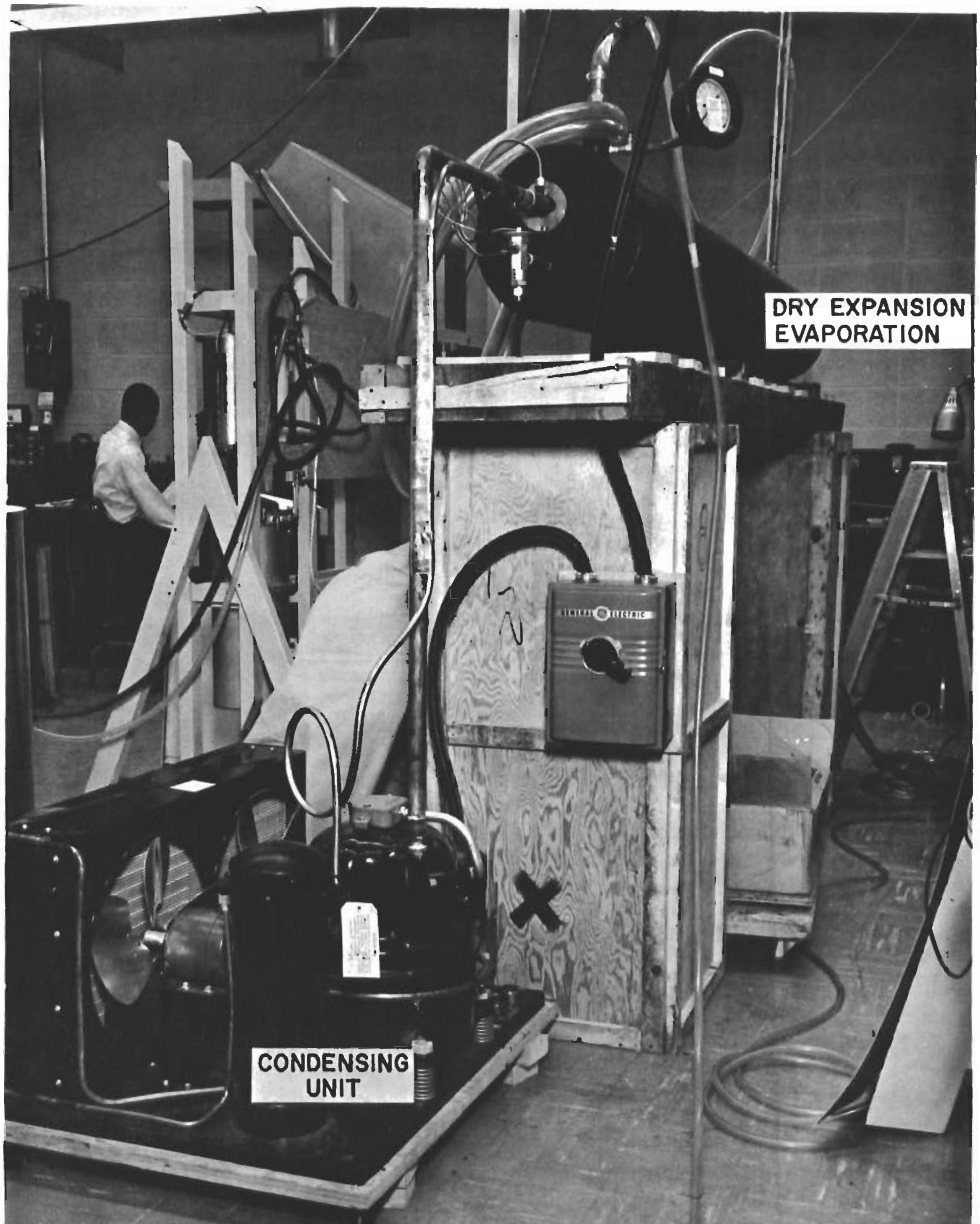


FIGURE 22: HIGH TEMP HEAT SINK INSTALLATION

SECTION IV

EVALUATION TEST PROGRAM

Following the completion of the program hardware fabrication phase the equipment underwent performance evaluation tests prior to its being installed in the AMRL space cabin evaluator at Dayton. All evaluation tests were run under static load conditions only, since operational check out testing under dynamic loading conditions at the shock, vibration and acceleration levels specified in the system performance specification would have required a test program of considerable expense due to the size and complexity of the COOL equipment. These expenditures were not considered to be warranted because the system is a prototype and it will never be subjected to flight loading conditions.

1. Test Objectives

The primary overall objective of the program was to prove the feasibility of the liquid transport-direct radiation to space concept as a workable means for conveying the waste heat from inside a space vehicle and rejecting it to space. Specifically the test objectives were:

a. Habitable Atmosphere Control - To ascertain the ability of the equipment to handle the heat load of the AMRL Evaluator and its capabilities regarding the effective control of relative humidity within specified limits.

b. Food Refrigeration - To ascertain that the freeze and chill food storage compartments are able to maintain food at $0^{o}F \pm 5^{o}F$ and $+ 32^{o}$ to $+ 40^{o}F$ respectively. Also, to determine the "pull-down" capability of the chill compartment.

2. Summary of Initial Check Out Test Results

The check out tests were performed in the laboratory with the entire system housed within a mock-up supporting structure so that all the various components were located in proper relation to one another (Figure 23). Several runs were made on succeeding days in order to establish the stabilization times for the various subsystems, their maximum heat removal capabilities and operational stabilities.

2.1 Food Refrigeration System

Pull down tests were run to determine the times required to reach specified food storage temperatures. Tests were made both with and without food stored in the box. Temperature differences between the food and cooling shelves were determined as well as thermal gradients throughout each of the compartments. Test results included herein in tabular and graphical form show that both compartments are capable of maintaining food temperatures within specified limits and that the chill compartment has adequate heat removal capacity to meet the "pull-down" requirements. No sweating on any of the box exterior surfaces occurred during the tests under ambient conditions of $80^{o}F$ and 50% R.H. Frost Accumulations within the compartment did not hamper the removal of food items but a continuous 14-day test is necessary to determine if long term effects of frost build-up will impair the functioning of food removal mechanisms.

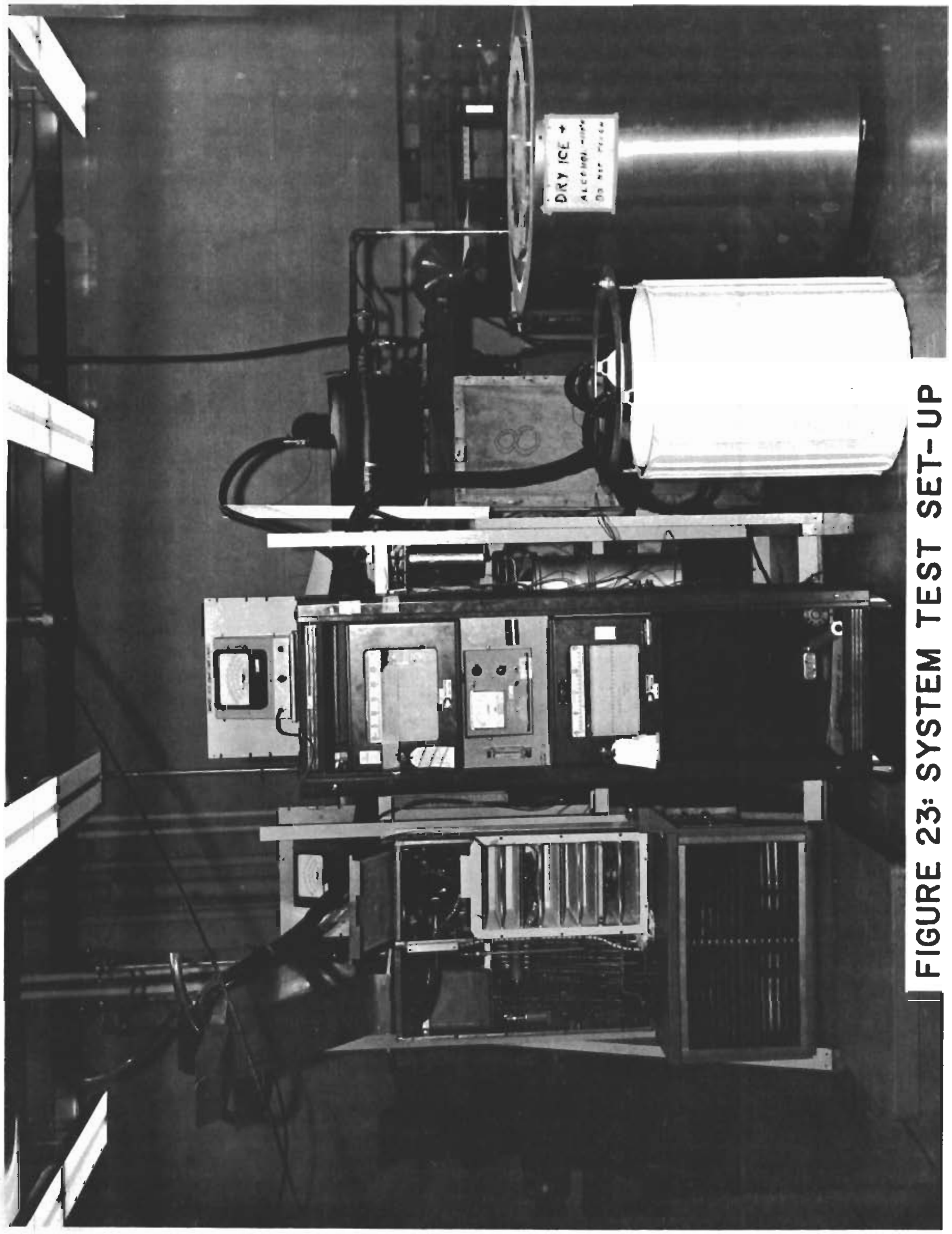


FIGURE 23: SYSTEM TEST SET-UP

Tabulation of Food Refrigeration Test Results

a. Freezer Compartment

Stabilized Shelf Temp: -2°F to +5°F
Food Temp: None
Pull Down Time: Approximately 2 hours

b. Chill Compartment

Shelf Temp: +35°F to +38°F
Food Temp: None (Compartment Air Temp, +32°F to +38°F)
Pull Down Time: Approximately 30 minutes

c. Low Temperature Heat Transport Circuit Power Consumption

Low Temperature Circulating Pump - 90 watts, 115-VAC, single phase, 60 cps., pump operated approximately 80% of the time.

d. Thermal Control Equipment

Temperature Controller - 10 watts, 115-VAC, single phase, 60 cps, continuous.

e. Chill Compartment Solenoid

10 watts, 28-VDC, 60% of the time.

Notes: Refrigerant circulating pump operated approximately 80% of the time. Entire low temperature heat load was imposed on radiator (e.g., humidity control apparatus, chilled water reservoir and food storage compartment). Tests were conducted under 80°F and 50% R.H. ambient conditions.

2.2 Habitable Atmosphere Control System

The habitable atmosphere control test results indicate that the system has more than adequate heat removal capabilities. All components performed as anticipated. Their performance is indicated in the summarized tabulation below. No quantative results were obtained on the moisture freeze-out apparatus (humidity control) during these tests. Quantitative tests will be performed subsequent to installation within the test cabin evaluator. Post-installation tests to be performed at Dayton will give more significant results because operation within a closed system environment will enable the humidity control device to function independent of the open laboratory environment which surrounded the system during check-put tests.

Tabulation of Habitable Atmosphere Control Results

a. Maximum Heat Removal Capacity Data

Laboratory Ambient Temp. - 74°F
Conditioned Air Flow Delivery Rate - 940 cfm @ 1.0 ATM & 72°F
Maximum Cooling Effect - 26,400 BTU/hr

Heat Exchanger Liquid Side Temp Rise - 8°F
Heat Exchanger Air Side Temp. Drop - 26°F
Heat Transfer Fluid Pumping Rate - 8.0 gpm @ 40°F fluid temp.
Conditioned Air Discharge Temp. - 48°F

Remarks: In order to obtain these maximum cooling effect results, the heat exchanger coil temperature was allowed to stabilize at its lowest attainable extreme. Air circulating blowers were connected electrically in series across the 28 VDC power source and installed to give parallel air flow delivery.

b. High Temperature Circuit Power Consumption

High Temp. Liquid Circulating Pump - 440 watts, 115 VAC, single phase 60 cps, continuous during maximum load conditions only.
Air Circulating Blowers - 200 watts, 28 VDC, continuous
Temperature Controller - 10 watts, 115 VAC, single phase 60 cps, continuous.

2.3 Water System

Various performance tests were conducted on the water recovery and storage systems to evaluate their effectiveness, establish capacities and determine leak resistance. Following are the tabulated test results:

a. Moisture Recovery Pump

Delivery Rate - 0.015 gallon/stroke, programmed to function at a rate of twelve evenly spaced delivery strokes per hour.

Function - Satisfactory. No measurable leakage, check valve functioned satisfactorily.

b. Water Storage Reservoirs

Main Storage Reservoir

Capacity - 2.5 gal.
Leakage - None measurable
Working Pressure - 0 to 5 psig (maintained by a variable spring force acting upon piston within reservoir)
Pressure Check - checked at 15 psig, semi-permeable membrane did not allow passage of water. No other measurable leakage.
Relief Valve - opened at 8 psig, fully re-seated at 4.5 psig.

Chilled Water Reservoir

Capacity - 0.374 gallon
Leakage - None measurable
Working Pressure - 0 to 5 psig, maintained at same pressure as main reservoir.
Water Temperature - Average 42°F, maintained by thermostatically controlled solenoid valve which meters low temperature liquid transport fluid through coils on outer wall of reservoir.
Power Requirements - 10 watts, 28 VDC for solenoid. Full time operation will supply a maximum of 48 oz. of chilled water per hour.

Heated Water Storage Reservoir

Capacity -0.375 gallons
Leakage - None measurable
Water Temperature - 168°F
Working Pressure - 0 to 5 psig
Power Requirements - 150-watt cartridge heater. Duration of operating cycling depends on use frequency. Heater has capability of providing 48 oz. of 170°F water per hour.

SECTION V

CONCLUSIONS

1.0 All the phases of this program as reported herein have physically proven the objectives set forth at the inception of the program and the following conclusions are derived as a result of this effort:

1.1 All subsystems functioned as intended and their performance was within the specification limits.

1.2 The habitable atmosphere control subsystem gives a suitable means for accomplishing thermal control within the confines of a space vehicle. The water recovered from the humidity control devices is acceptable as a potable water source.

1.3 The refrigerator subsystem satisfactorily preserves the refrigerated food requirements of a three-man crew for a 14-day mission.

1.4 The recovered water storage system will adequately supply the chilled and heated water requirements for a three-man crew.

2.0 The following conclusions can be drawn as the result of acceptance tests which were performed with the entire system installed within the AMRL Evaluator at Dayton.

2.1 The habitable atmosphere control system has sufficient heat removal capacity to maintain a comfortable environment within the evaluator. Water recovery from the humidity control devices can be accomplished.

2.2 The refrigeration sub-system functioned satisfactorily as installed within the evaluator. However, the physical details of the plumbing connections that were routed between the external radiator-heat sink and the equipment inside the evaluator were such that the resulting additional heat load imposed by these connections was sufficient to prevent the freezer from pulling down below 20°F.

The heat sink was located approximately thirty feet from the evaluator. These long lines plus penetration fittings through the evaluator wall which give appreciable conductive heat transfer from the wall to the cooling fluid resulted in higher than anticipated external heat loads (line losses). The radiator-heat sink did not have sufficient capacity to remove this additional heat.

However, the direct radiation to space concept has been adequately proven by the evaluation tests at GE, Philadelphia where line losses were within the limits that were used in sizing the radiator.

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APPENDIX I
FOOD REFRIGERATION FOR SPACE VEHICLES FEASIBILITY STUDY
SECTION 1

INTRODUCTION

Background

Man in space will, of course, require food as he does here on earth. Initial flights of approximately several days will require nothing more substantial than solid food bars and water or perhaps a mixture of the two. For extended flight periods, man will be subjected to many unfamiliar stresses. Food supplied and prepared in a familiar manner can reduce these stresses. Thus, it is essential that palatable and highly acceptable food be provided. While some of this type of food can be provided dry or dehydrated, the majority will require refrigeration for preservation. Because of the special environmental considerations of space (such as the absence of gravity), as well as the necessity to minimize weight, volume and power required, conventional refrigeration systems cannot be considered adequate. Project FROST was undertaken in order to study means whereby the required refrigeration and related food storage could be provided.

Requirements

Specific requirements which have been established for Project FROST are described. Two temperature zones are to be provided in the refrigerator. A chill space maintained at 32 to 40^{o}F., and a frozen food compartment maintained at -20 to 0^{o}F. The food to be stored in the refrigerator will be of two basic forms. The first of these will be the primary food, packaged in special containers in order to permit consumption in the absence of gravity. The second type of food to be considered for refrigerated storage will be buttered bread, provided in the form of sandwiches (two slices face to face) wrapped in plastic wrappers. In addition to food, consideration has also recently been given to storing fecal matter in the refrigerator. This would be packaged material which has been collected and processed by waste handling equipment and is to be stored at a low temperature in order to provide a means for preventing deterioration. The range of missions to be considered for this application vary from a period of one week to three years with a crew complement varying from one man to 20 men. A very short mission duration of one man for one week is envisioned as an earth orbiting type of mission. The intermediate range of missions could be earth-orbiting space laboratory types, lunar missions or near earth space explorations. The very long mission durations of 20 men for three years are imagined to be deep space penetrations. It is desired that the feasibility of a food refrigeration system for a space vehicle be established which will function in the environment of space and at the same time minimize weight, volume and power required. The type of food and method of food packaging are outside the scope of Project FROST.

Approach

As the first step in the search for an adequate refrigeration system for a space vehicle, a literature survey was conducted. The search was centered about new or novel refrigeration techniques which might be applicable to use aboard a space vehicle. The "state of the art" of existing refrigeration techniques was also examined in order to determine applicability. Certain details associated with the specific requirements of this space application were also studied, in order to determine the exact refrigeration requirements applicable over the wide range of mission durations to be considered. Integration of the refrigeration system with respect to the overall space vehicle was also considered. Those systems which appear to be best suited for our application were then examined in greater detail. System comparisons were made, and final recommendations and conclusions were drawn.

SECTION II

REVIEW OF AVAILABLE REFRIGERATION TECHNIQUES

A. Basic Concepts

On board a space vehicle, there are basically only two ways of discarding heat. The first of these would be internal and would consist of the rejection of heat to a cold source already available on the vehicle, which required the addition of heat. Such a system could utilize liquid oxygen, or other materials such as fuel for in-flight correction, stored at low temperatures aboard the vehicle. These materials would require the addition of heat before they are ready for use by the vehicle. They could, therefore, be utilized as a heat sink for a refrigerator at the appropriate temperature level, but would require a carefully balanced system. In addition to the heat balance problem, additional mechanical problems exist, since the fuel storage and engine spaces would probably not be in the immediate proximity of the refrigerator. This would mean piping low temperature material through the vehicle and the refrigerator and back again to the engine, not a very practical arrangement. The second concept which is available is that of complete removal of the heat to be rejected from the vehicle. This can be accomplished by two means; either radiation to space, or by material removal, e.g., expendable refrigerant systems. Depending on mission requirements, a system involving radiation to space would appear to be generally most desirable because the concept of material removal involves a launch weight penalty for all the material which is to be eventually discarded. The engineering feasibility study for project FROST has been directed toward the latter of these two concepts. The restrictive nature of the former was not considered compatible with the objectives of the FROST program.

B. Techniques Available

The following summarizes the many systems investigated and indicates those considered in greater detail as most probably applicable to Project FROST.

1. Direct Radiation to Space

A direct radiation system in its simplest form would consist of a radiator with surface properties such that heat inputs from external sources such as the earth, sun, reflected sunlight, etc., are exactly equal to the amount of internally produced heat to be discarded. With fixed inputs to the system and a fixed output, the system temperature can be maintained at a desired constant level. Such a system with no active elements is often called a passive radiation system. It is also possible to control the surface properties of the radiator sufficiently to enable compensation for loading variations over a limited range. If, however, the range of possible variations is great, such a technique could not be employed and an actively controlled radiation system is required. This technique will be considered in greater detail in a following section.

2. Heat Pumping

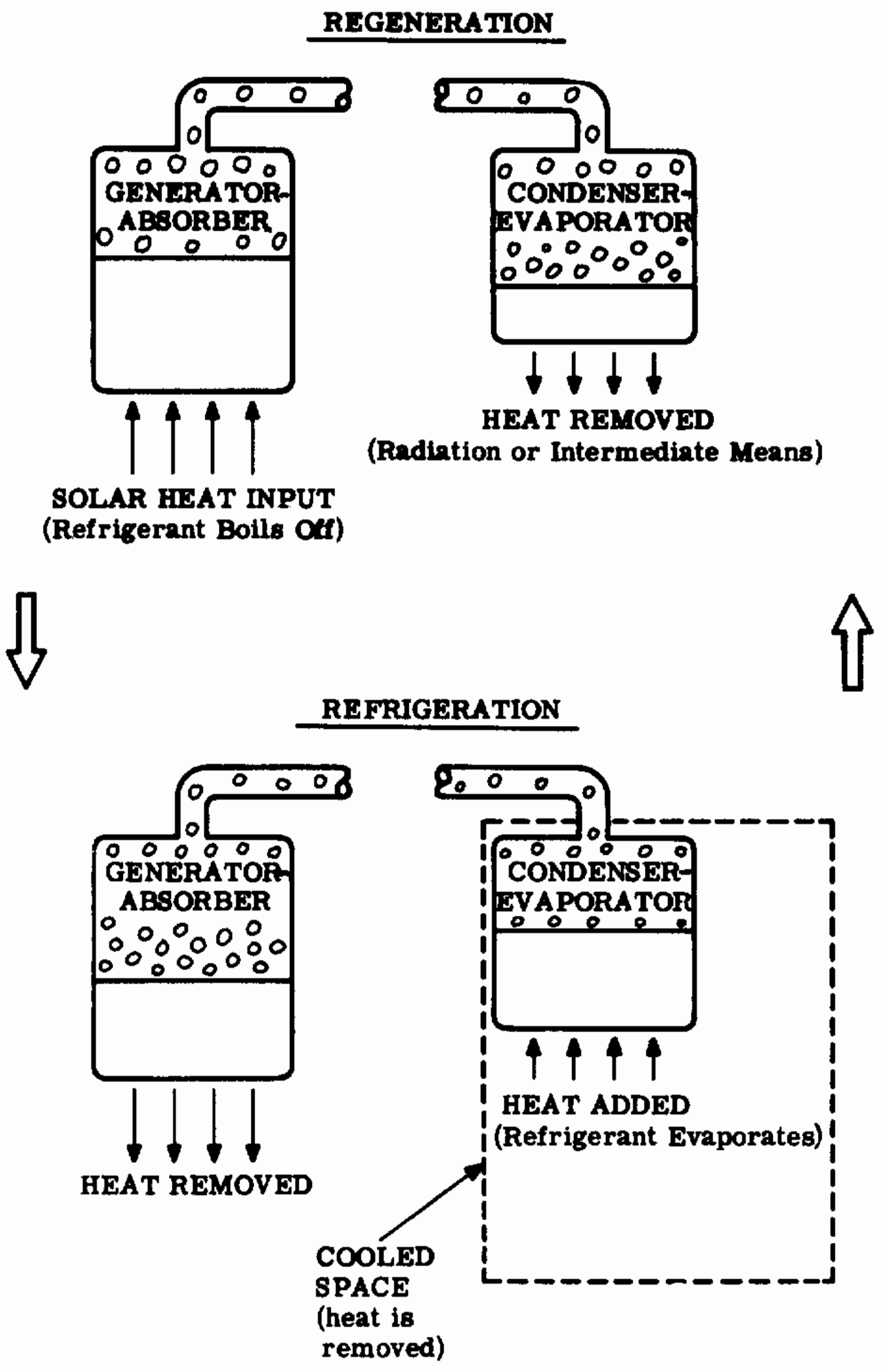
A heat pumping technique is normally utilized when the heat to be discarded is at a level lower than that of an available heat sink. An artificial heat sink is thereby created at the appropriate low level, and the heat is pumped up to a level above a readily available heat sink and rejected at this temperature. Various types of common systems are completely described in any elementary thermodynamics text. Examples of the common types are the vapor compression absorption, steam ejector, and air cycle systems. Variations of these heat pumping techniques have been used to produce a refrigeration effect on earth-bound vehicles and stationary installations for a considerable period of time. Special purpose types of heat pumps are also available, such as the vortex tube and various cryogenic techniques such as the demagnetization of a solid.

A technique which has recently undergone considerable development and which is attracting a great deal of attention currently, is the thermoelectric system. However, by using space as a heat sink, a temperature below that of the heat to be discarded is readily available and it appears inadvisable to attempt to pump heat in order to reject it. This subject will be explored in greater detail in a following section. In addition, the systems mentioned are probably not applicable for space vehicular use in that weight, volume, and power requirements do not appear compatible. Furthermore, operation in the absence of gravity would require severe modification to existing types of equipment and as a result, it is possible that efficient performance will be degraded. Only the thermoelectric system is suitable for space vehicular use in that the absence of gravity would have no effect on the operation of such a system. This heat pump system has, therefore, been selected for further study and will be described in greater detail in a following section. The only other information of interest which was uncovered by the literature survey was the apparent potentiality of driving an absorption system by using solar energy. A solar power absorption system is illustrated in Figure 24. This type of system has been proposed for earth use in areas where solar energy is readily available. Problems involved in this type of system are related to the fact that solar energy must be added to the same container from which heat is later extracted by the process of condensation. This requires that a mechanical arrangement be devised for moving this unit from one position external to the refrigerator to a position internal to the refrigerator, or a rather complicated pumping system must be used.

If a liquid absorber were used, problems of liquid-vapor separation under weightlessness conditions would exist. If a solid absorbent were used, liquid-vapor separation would no longer be a problem, but it would be impossible to pump the adsorbent from one location to another and the mechanical movement of the adsorbent would be required. Still another factor which should be considered is the necessity to reject more heat than was originally required for refrigeration, because of the addition of external heat (in this case solar energy), a situation common to all heat pumps. Thus, even this system which appeared initially of interest, because no external energy other than readily available solar energy would be required, does not seem suitable because of the complications described.

3. <u>Expendable Refrigerants</u>

This technique is essentially an open cycle system wherein the medium used to absorb the heat is discarded along with the heat to be rejected. The



INTERMITTENT ABSORPTION COOLER WITH SOLAR REGENERATION
(ALTERNATING CYCLES)

Figure 24

common forms of this system utilize a change of state of the refrigerant to absorb the heat and produce the refrigeration effect. In this way, high heats of vaporization or sublimation are available and the resulting vapor is discarded to space. Re-entry nose cones make use of a form of this principle to discard heat upon their re-entry into the earth's atmosphere by the ablation of a surface coating. A transpiration technique has been considered for the cooling of high speed aircraft, wherein a fluid is evaporated from the pores of the surface to be cooled. An obvious limitation to this concept is the requirement to provide at launch all of the refrigerant which will be required. However, for a short mission duration, a relatively simple expendable refrigerant system appears feasible. Therefore, further effort has been devoted to this particular concept in order to better establish the limits of possible use.

SECTION III

DETAIL CONSIDERATIONS

A. Establishment of Refrigerator Physical Size

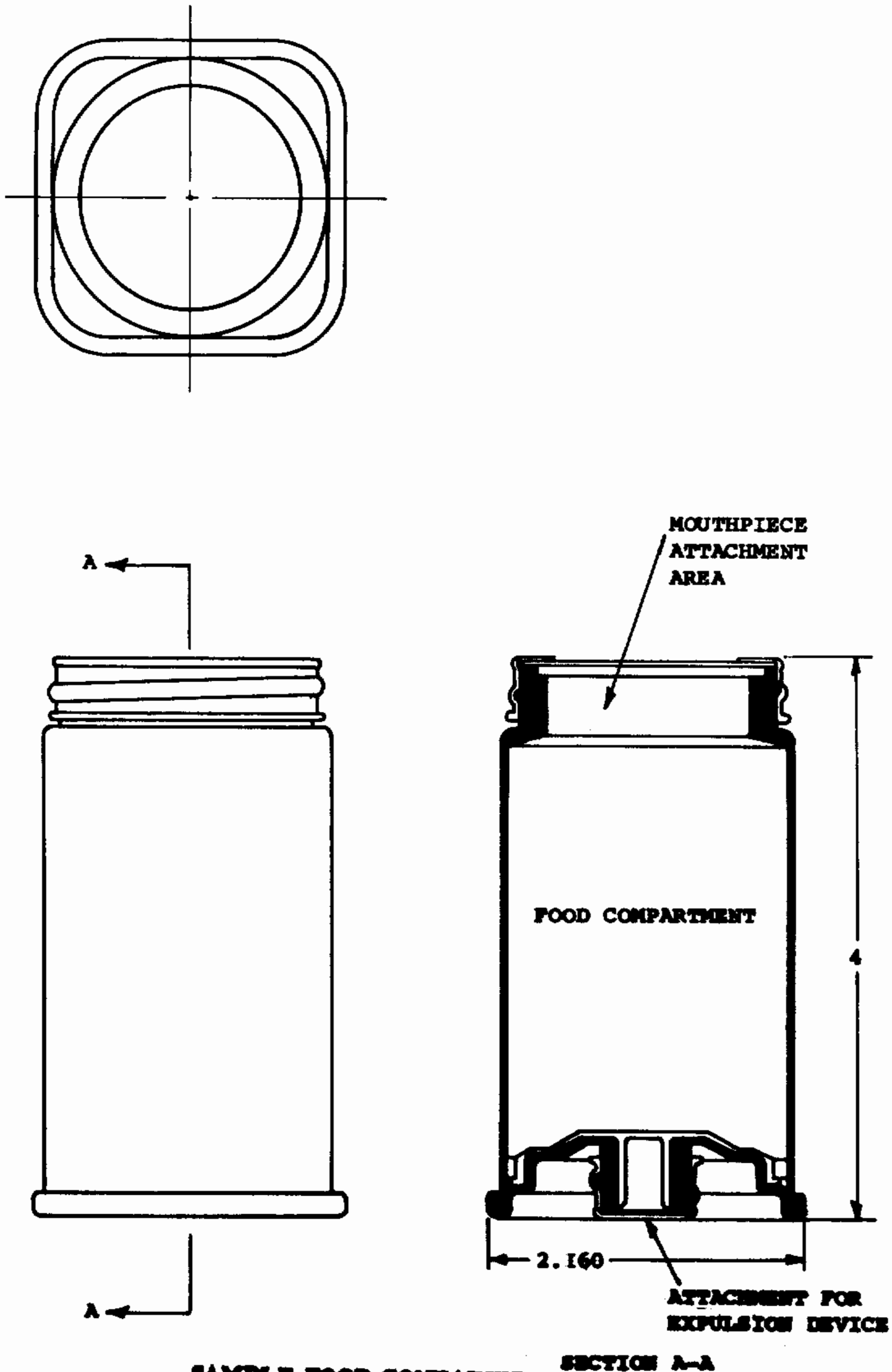
In order to determine the weight and volume requirements for the food to be stored on a mission basis, the following food requirements specified by WADD were considered.

Three six-ounce containers of food will be provided per man for each of three meals per day. These food containers are to be stored in the freezer compartment.

One bread sandwich consisting of two slices of bread with butter spread between will be provided per man for each of three meals per day. This bread sandwich is also to be stored in the freezer.

Two six-ounce juice containers are to be provided per man for each day. These juice containers are to be stored in the chill compartment.

Two types of containers for these foods are presently being considered. The first of these is an aluminum container illustrated by Figure 25. A second type of container being considered would be all plastic. Bread sandwiches will probably be packaged in a plastic wrapper in any case. At this point it is important to stress the need for proper integration of the food containers and the refrigerator. Thermal properties, physical size, and the shape of the food containers will be determining factors in the internal design of the refrigerator. In a minimum volume, minimum weight system, it is necessary to specifically insure that the container will exactly fit the shelving provided and will have any necessary appendages required to insure storability within the refrigerator. For example, a simple notch on the face of the container could be utilized to provide a stop for the individual containers as they are removed from the refrigerator. However, at this preliminary stage in the program, information on containers is not completely available. One container design has been made available and detailed information related to this design was used in the preliminary estimate of space and volume requirements. Since the food will be liquid, semi-solid or diced solid material, the density of water has been used as the approximate average density. On the basis of the quantities previously stated, the weight consumption per man day is 4.83 pounds. This value consists of using a weight of 0.375 pounds per six ounce container and 0.20 pounds estimated for a bread sandwich. The added weight estimated for the container is approximately 0.25 pounds based on the aluminum design referenced, and a small weight of 0.02 pounds has been added for the plastic wrapper for the bread sandwich. Thus, the total weight to be carried on a per man day basis would be 7.54 pounds. The volume for the containers and bread sandwiches has also been estimated again based on the container information available. A container volume of 16.65 cubic inches has been established. Since it would be necessary to allow some clearance surrounding the containers in order to provide restraint (shelves or other means) as well as an avenue for heat transfer, a volume of 20 cubic inches per can has been used. Thus, on a per man day basis, 180 cubic inches for frozen foods and containers must be provided. In addition, a volume equivalent to that of three bread sandwiches must be provided for in the freezer compartment. Again allowing for some small clearance, 20 cubic inches per bread sandwich has been estimated resulting in a total of 60 cubic inches per man day for bread sandwiches. The total freezer requirement per man day is then 240 cubic inches. A chill space requirement results from the need to



SAMPLE FOOD CONTAINER

Figure 25

provide for two six-ounce containers of juice per day of 40 cubic inches per man day. The total volume, therefore, required for all food in the refrigerator is 240 + 40 or 280 cubic inches per man day. This is equivalent to a value of 0.162 cubic feet per man day and is a relatively high value as compared to a figure of about 0.1 cubic feet per man day which is estimated for food storage space aboard a submarine. However, this is the volume penalty which must be paid when the food is individually packaged as must be the case for consumption in the absence of gravity. Aboard a submarine, bulk packaging is utilized wherever possible. Table 6 is provided as a summary table of food weight and internal refrigerator volume required for representative mission durations. From this table it can be readily seen that the weight and volume requirements for the many man, long duration missions, are considerable. The refrigerator volume required as a continuous function of mission duration has been plotted in Figure 26. A 10% excess food capacity has been allowed in determining both weight and volume requirements. In addition, the possibility of denser packaging for the very large volumes has been considered.

A 30-day, three-man mission can be examined to illustrate the specific internal requirements. The total volume of 16 cubic feet would provide for 900 cans and 300 bread sandwiches stored in the freezer compartment, and 200 cans stored in the chill compartment. In addition, it has recently been requested that consideration be given to the possible storage of fecal matter in this refrigerator. This would mean about 1/4 to 1/2 pound of material would be added to the refrigerator for each man per day. Since this material is 70% water, a density value equivalent to that of water can be used. On the basis of the maximum value of 1/2 pound per man day and using the desnity of water, a volume of 14 cubic inches per man day would be required. Since this is a very small volume compared to the space vacated by consumed food on a daily basis (280 cubic inches), a procedure for storage of this waste material in the vacated volume should be utilized. In fact the volume of fecal matter to be stored for one man day is less than the volume of a single container (16.65 cubic inches). Thus, a technique of storing the waste material directly in the empty food containers should be considered, or if aluminum food containers are used, it is conceivable that these containers could be collapsed and the volume formerly occupied by the containers could be used for the storage of the fecal matter. On the other hand, if the food is stored in flexible plastic containers, the empty container would certainly allow sufficient volume for the addition of the fecal matter. This subject will again be considered when the details of the heat load applied to the refrigerator are studied.

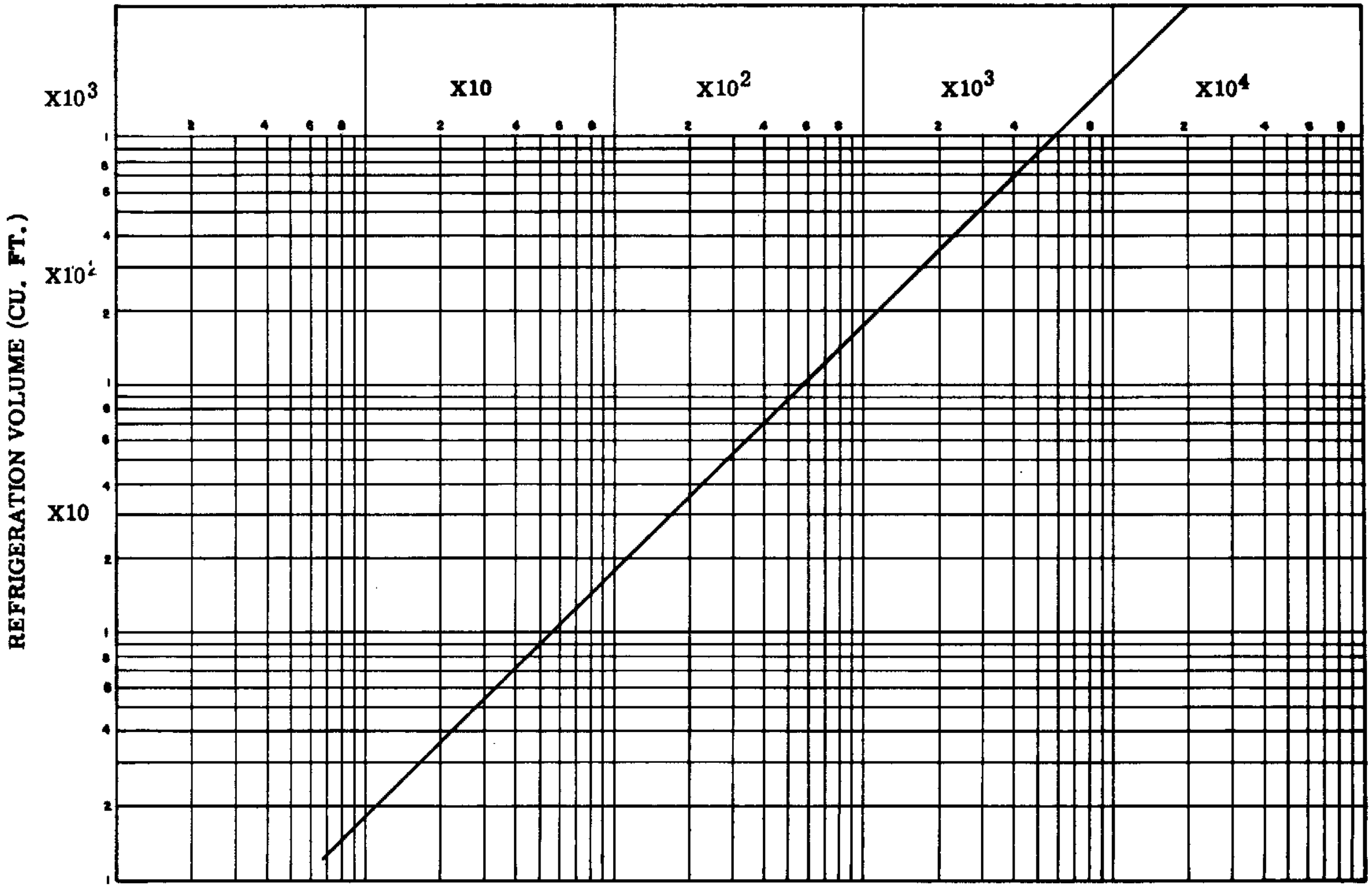
B. Insulation Evaluation

Now that the basic sizes of the refrigeration space required have been established as a function of mission duration, the next point to be investigated will be the kind of internal insulation suitable for use in the refrigerator. This subject will form the basis for a further discussion of the over-all refrigerator heat loads. The insulations available for use can be divided into two major classes. The first of these is insulation containing entrapped air or other intentionally entrapped gases at essentially atmospheric pressure. The second category would be vacuum insulations wherein atmospheric air has been removed. Figure 27 illustrates the kind of thermal conductivity factor available with present day insulating materials as a function of temperature. From this data it appears that three specific types of insulation bear further investigation. The first of these is fibre-glass. Fibre-glass batts can be commercially purchased in densities of three pounds per cubic foot, and with a coefficient of thermal conductivity, k, of 0.24 BTU/Hr-Sq. Ft.-^{o}F/In. in the temperature range of our intended use. This material

Table 6. Food Weight and Volume Required for Representative Missions

FOOD STORAGE REQUIREMENTS

Mission (Man Days)	Mission Type	Ref. Size FT^3	Weight of food (lbs)	Total Wt. Food and Cont. (lbs)
7	1 man - 1 wk.	1.25	37.2	58.1
90	3 men - 1 mo.	16	480	746
3650	10 men - 1 yr.	600	19,300	30,300
21,900	20 men - 3 yr.	3400	116,000	182,000



MISSION DURATION (MAN DAYS)

Figure 26

MODERATE TEMPERATURE THERMAL INSULATIONS

	INSULATION	DENSITY	REMARKS
(1)	P-ZERO (10^{-3} mm Hg)	20. LB/FT^3	0.00014" DIA FIBERS ORIENTED #
(2)	POLYURETHANE FOAM - FREON FILLED @ 1 atm	2.0 LB/FT^3	CLOSED CELLS
(3)	FIBERGLASS-FREON		CAUTION: DIFFUSION THROUGH CONTAINER
(4)	SILICA AEROGEL - AIR @ 1 atm	5.3 LB/FT^3	CAUTION: MOISTURE ABSORPTION
(5)	CORKBOARD - AIR @ 1 atm	6.9 LB/FT^3	
(6)	STYROFOAM - AIR @ 1 atm	1.5 LB/FT^3	
(7)	CELLULAR UREA FORMALDEHYDE AIR@ 1 atm	0.3 LB/FT^3	
(8)	FIBERGLASS-AIR @ 1 atm	3.0 LB/FT^3	NO BINDER
(9)	FIBROUS POTASIUM TITANATE AIR @ 1 atm	15.0 LB/FT^3	
(10)	MIN. K501	10.0 LB/FT^3	

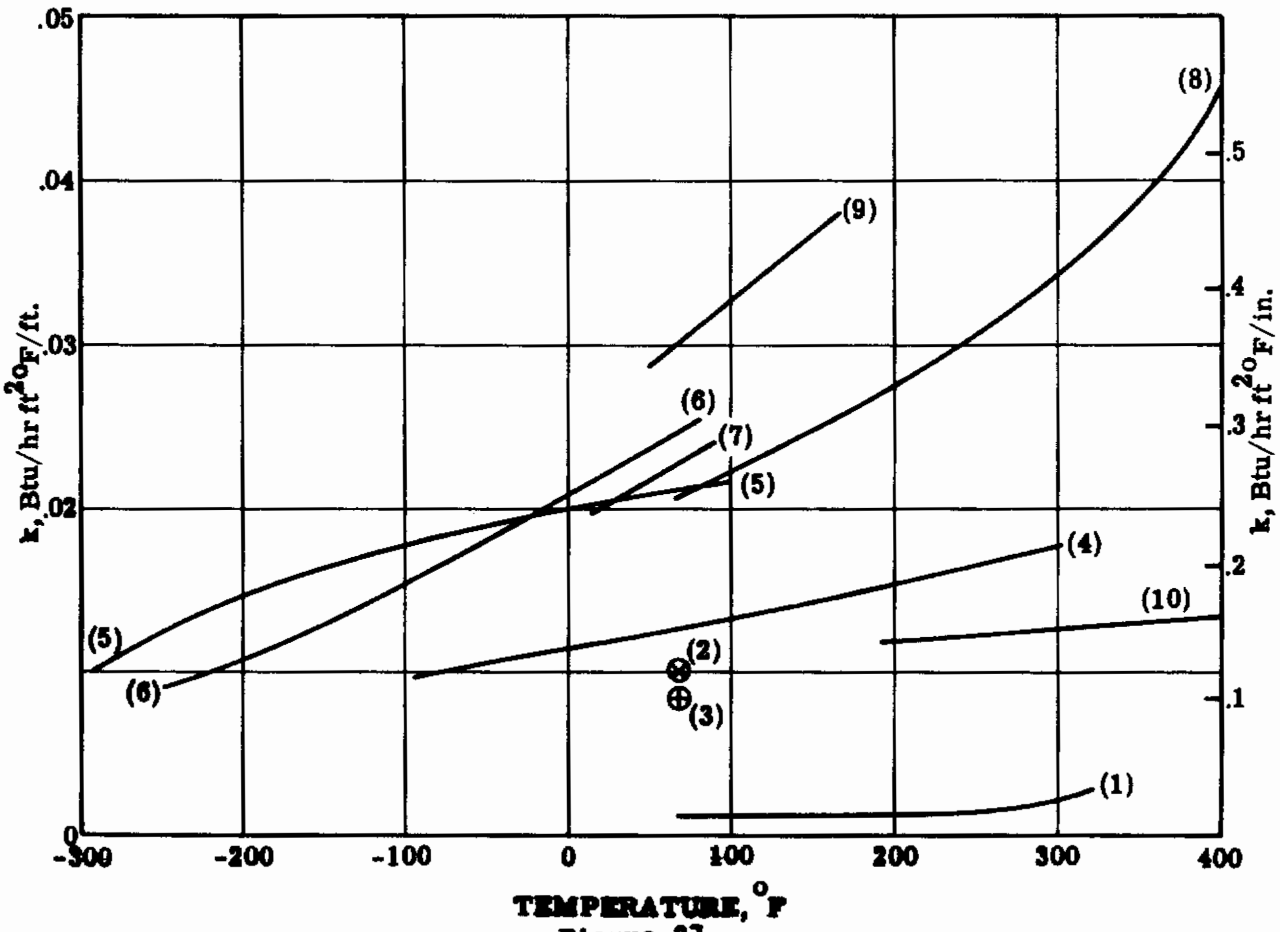


Figure 27

constitutes no special problems other than a disagreeable handling condition. However, once installed with a suitable envelope, this material will provide no added difficulty, and is representative of about the best insulation available in a solid material with no further requirements. In recent years, emphasis has been placed on foam materials for insulation applications, particularly for use as refrigerator insulation. Polyurethene foam, for example, filled with freon has a coefficient of thermal conductivity of 0.12 BTU/Hr.-Sq. Ft.-^{o}F/In. at a density of two pounds per cubic foot. This low thermal conductivity is the order of magnitude of the minimum obtainable without the use of vacuum techniques. However, this type of insulation does suffer to a degree from degradation due to the diffusion of nitrogen from the atmosphere into the insulation. Therefore, for comparative purposes, a minimum value will not be used, but a value of thermal conductivity which accounts for a degree of diffusion will be used. As a representative of the third (vacuum) type of insulation to be considered, G. E. P-O vacuum insulation has been selected. This is an insulation which employs vacuum techniques for use at room temperatures and was especially developed for household refrigerator use. In addition, this material provides the structural capability of supporting the pressure differential imposed by internal vacuum and external atmospheric pressure with minimum structural support. The density of this material is 20 pounds per cubic foot, and a value of thermal conductivity of 0.0144 BTU/Hr.-Sq. Ft.-^{o}F/In. is a tested and proven value for this material. As a final comment, vacuum type insulation appears especially suitable for this application because of the readily available vacuum condition existing in space. Thus, the problems sometimes encountered in maintaining vacuum conditions within the insulation can be alleviated by venting to space subsequent to launch. In order to evaluate the insulation situation, the factors to be considered are weight, volume and insulation effect. Obviously, the lowest k value material will result in a lowest heat load into the refrigerator for the same thickness of material. Or on the other hand, a reduced thickness of lower k value material could be used to result in the same heat load as that of a higher k value material. The weight penalty of providing different levels of heat load into the refrigerator must also be considered. Therefore, a relationship has been established combining the effects of insulation weight, and ultimate radiator weight required to discard the heat allowed to leak into the refrigerator. Using some numbers which will be developed in subsequent sections of this report, a figure for radiator weight which can be used is one pound per square foot of surface area required. The surface area required can be described by the heat flow q divided by the radiative capability per unit surface area q''. A good value for q'' for this application, which is also derived in a later section on radiation capabilities, is 25 BTU/Hr.-Sq. Ft. The heat flow, q, through the insulation is described by the following expression:

$$q = \frac{kA\,\Delta T}{t}$$

where k = coefficient of thermal conductivity
A = area (sq. ft.)
ΔT = temperature differential (^{o}F)
t = thickness (ft.)
then q = BTU/Hr.

For a particular application the area of the external surface of the refrigerator would be fixed as well as the temperature difference existing. Thus, the effect of variation of thickness of the insulation, and the thermal coefficient can be evaluated. The 16 cubic ft. refrigerator which results in a surface area of 38.1 sq. ft. will be examined in detail. On the subject of the temperature difference to be considered, an extreme low temperature for the refrigerator of $-20^{\circ}F$ has been assumed. It is understood that this temperature is not especially desirable for all of the food, since this low temperature would require additional heat input to the food in order to warm it prior to consumption. However, in order to examine the refrigeration capability, the lowest refrigerator temperature required is best considered. To maintain a temperature of -20° for the food, an average refrigerator inner surface temperature of about -25° has been estimated. This means a temperature difference of $95^{\circ}F$. between refrigerator inner surface and the cabin, assuming a cabin temperature of $70^{\circ}F$. If we combine the above values, the radiator weight can be expressed as follows:

Radiator Weight = Radiator Area x Specific Weight where

$$A_{rad} \text{ (Radiator area) } = \frac{q}{q''}$$

or substituting for q

$$A_{rad} = \frac{kA\Delta T'}{t} \times \frac{1}{q''}$$

for the particular case under consideration

$$\text{Rad. Weight} = \text{Rad. Area} \times 1$$

$$= \frac{k\,(38.1)\,(95)}{t \times 25}$$

$$= \frac{144.8k}{t}$$

The insulation weight is defined by the surface area, A, of the refrigerator multiplied by the thickness, t, and the density, ρ, of the insulation. Thus, for the 16 cubic ft. refrigerator, we have,

$$\text{Insul. Weight} = A \times \frac{t}{12} \times \rho$$

$$= 38.1 \times \frac{t}{12} \times \rho$$

$$= 3.17 \times \rho \times t$$

Thus, the total weight, W_t, of the radiator and insulation is equal to the sum or,

$$W_t = \frac{144.8k}{t} + 3.17\rho t$$

This weight neglects the weight of accessory structure and controls which will be essentially the same for all the systems. These weights will be considered in the design study. For a particular insulation where k and ρ have been established, the value of t which will produce a minimum W_t can be found. Taking the derivative of W_t with respect to t

$$\frac{D\,(W_t)}{dt} = -\frac{144.8k}{t^2} + 3.17\rho \text{ and equating to zero}$$

$$t^2 = \frac{144.8k}{3.17\rho} \text{ or the min. } W_t,\ t = \left[\frac{144.8k}{3.17\rho}\right]^{1/2}$$

For the three materials being considered, this relationship was solved and is presented in Table 7. Figure 28 illustrates the continuous variation of system weight (W_t) vs. insulation thickness for these three materials and visually demonstrates the thickness to weight tradeoff available. A further parameter which must be considered is the volume occupied by these various materials. For example, it can be readily seen that the polyurethane foam results in the lowest system weight (21 pounds); however, the required insulation thickness is 1.656". The P-zero, on the other hand, has a minimum system weight of 23 pounds for a corresponding 0.1814 thickness. Thus, at the cost of a slight increase in weight, a considerable amount of volume is saved by the use of this kind of insulation. An additional characteristic which is found in this curve is the rate of change of system weight with increase in thickness. For example, if a thickness slightly greater than the minimum value for the P-zero material were used, say a value of 1/4", the system weight would increase only a small amount to about 24 pounds. This would have the effect of decreasing the heat flow into the refrigerator. This effect is illustrated by Figure 29, a plot of heat flow into the refrigerator vs. the insulation thickness. Examining this figure, it can be seen that the heat flow is reduced from 287 BTU's per hour to 209 BTU's per hour by increasing the insulation thickness of the P-zero from the minimum value to a value of 1/4". This would result in a significant saving in the required radiator size at the cost of only a small amount of additional weight.

In summary, we find that the polyurethane foam filled with freon results in the lowest system weight. However, a substantial volume is required and some provision for pressure differential would have to be made since the ground environment of the space capsule would probably be one atmosphere, whereas the space environment of this capsule may be in the neighborhood of 1/2 atmosphere. The P-O vacuum material looks especially promising since the minimum system weight is very nearly as low as the gas filled foam and the required volume is considerably less. A problem to be expected with this type of insulation would be the heat flow due to edge leakage since the thickness of the insulation being considered is small. Special design features may be required in order to minimize this situation. The P-O was selected as representative of vacuum insulations, because the weight penalty required to provide for the pressure difference applied across the walls of the refrigerator has been accounted for in the high density of the material. In addition, the k value provided for this particular material has been established for operation at room temperature. This information is not readily available for most vacuum insulations which are normally operated at cryogenic temperatures. At the relatively higher temperature which is under consideration (room temperature), radiation is a significant effect contributing greatly to the thermal conductivity of the material.

The fibreglass insulation would not be a structural problem, but has a relatively low insulation factor to weight ratio. In addition, the minimum system weight is fairly high, there is a high heat input into the refrigerator, and the volume occupied is great. Thus, it appears that the most suitable insulation would be something like the P-O vacuum type insulation. The exact insulation to be used will not be selected at this point, but has merely been evaluated in order to establish the level of heat load into the refrigerator. During the design study phase, a more detailed evaluation of the insulation will be made in order to decide upon the specific form of the insulation to be applied. At this point, it appears that reasonable value to use for heat leakage through the walls of this particular refrigerator (16 cubic ft.) would be a value of about 200 BTU's per hour.

Table 7. Summary of Performance Data for Primary Insulations Considered

INSULATION EVALUATION SUMMARY TABLE

Material	k BTU/hr.-sq. ft.-^{o}F/in	lbs/ft.^{3}	Min. Wt. lbs.	Min. t inches
Fiberglass batts	0.24	3	36.4	1.912
Freon filled Polyurethane Foam	.0.12	2	21.0	1.656
P-O	0.0144	20	23.0	0.181

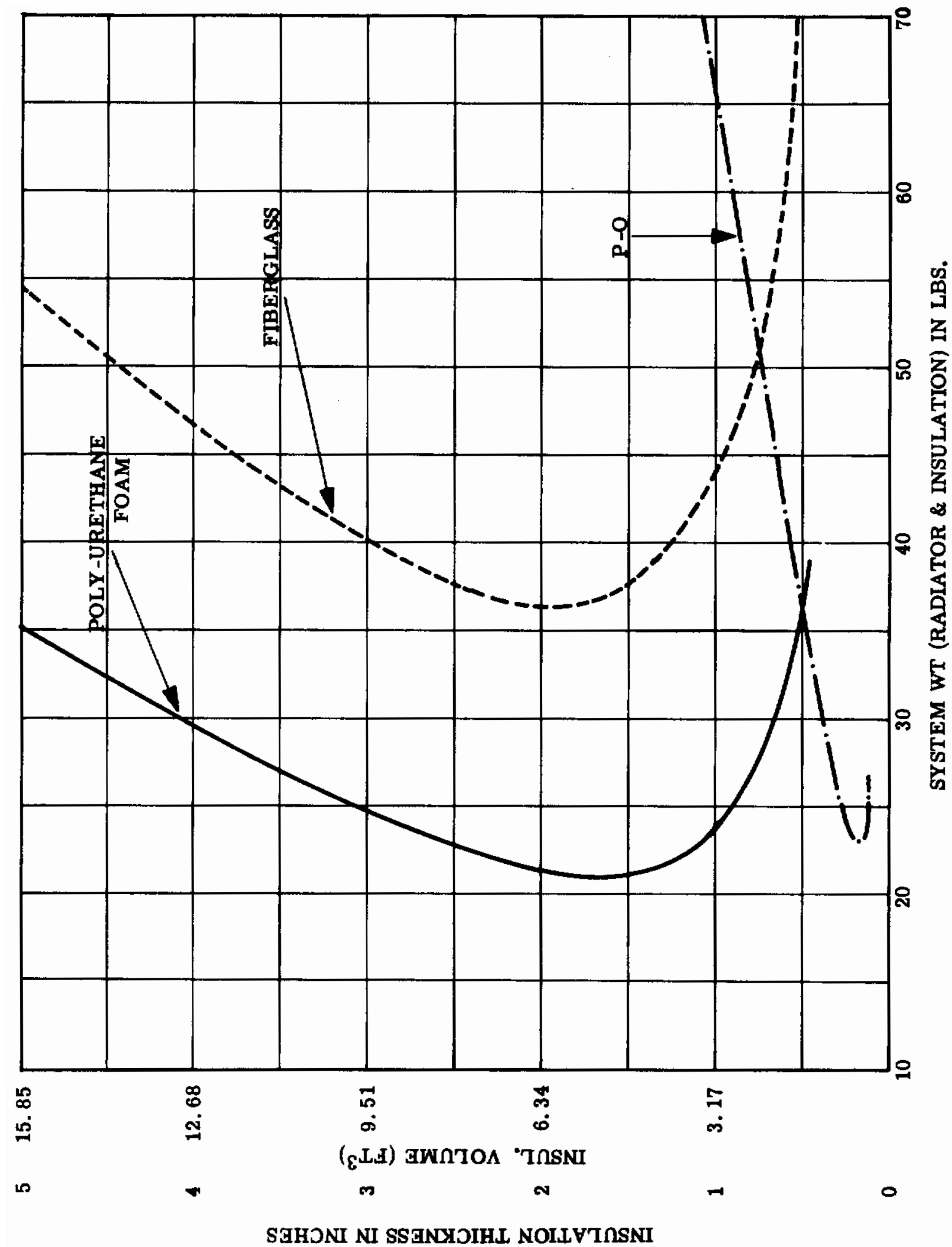

POLY-URETHANE FOAM
FIBERGLASS
P-O
SYSTEM WT (RADIATOR & INSULATION) IN LBS.
10
20
30
40
50
60
70
INSUL. VOLUME (FT³)
15.85
12.68
9.51
6.34
3.17
INSULATION THICKNESS IN INCHES
5
4
3
2
1
0

Figure 28

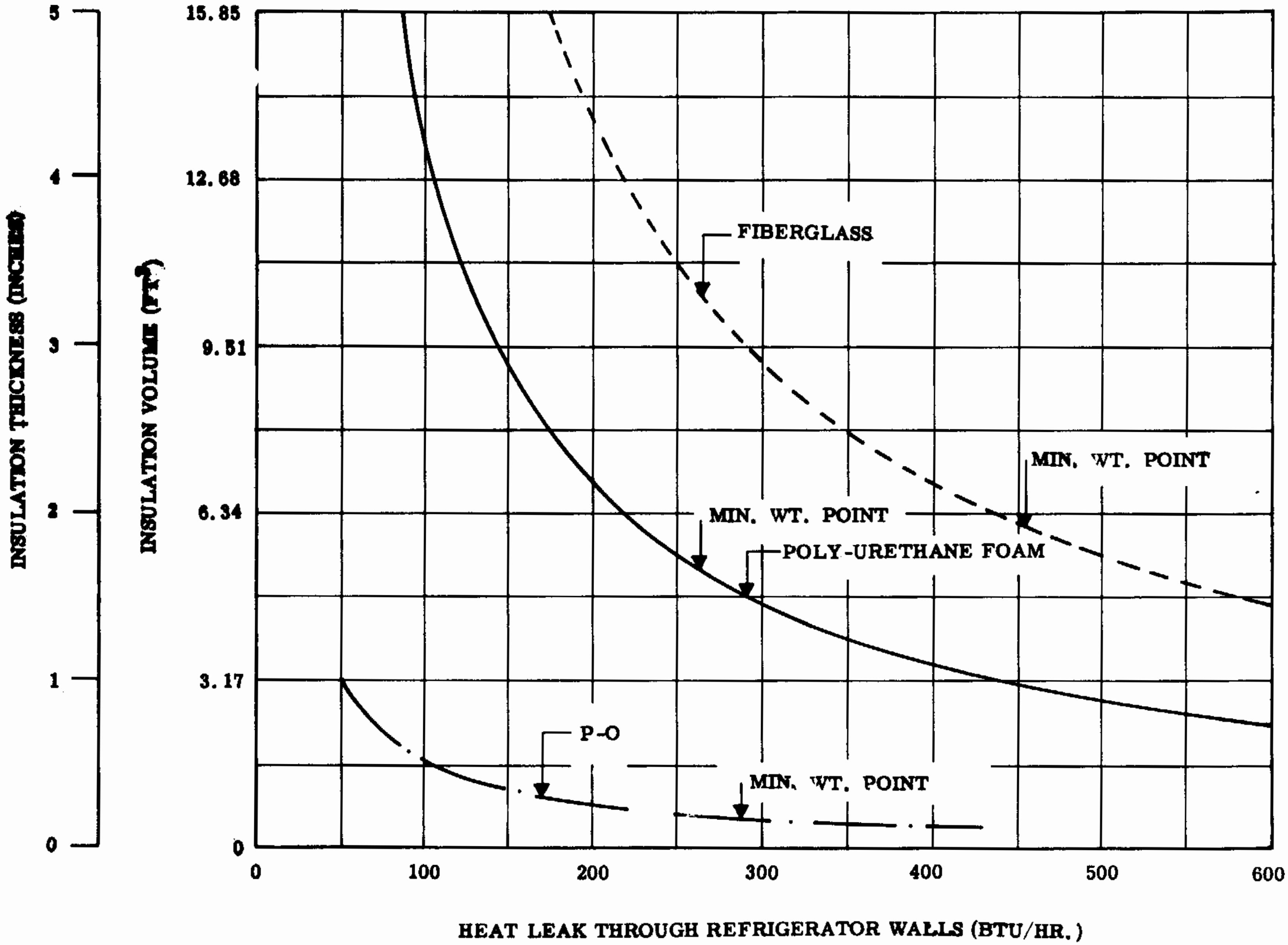

INSULATION THICKNESS (INCHES)
5
4
3
2
1
0
INSULATION VOLUME (FT^3)
15.85
12.68
9.51
6.34
3.17
0
FIBERGLASS
MIN. WT. POINT
MIN. WT. POINT
POLY-URETHANE FOAM
P-O
MIN. WT. POINT
0
100
200
300
400
500
600
HEAT LEAK THROUGH REFRIGERATOR WALLS (BTU/HR.)

Figure 29

C Overall Refrigerator Heat Loads

In addition to the heat transfer through the walls which has been evaluated in the previous section, heat will have to be removed from the refrigerator because of the following items: opening of the doors of the refrigerator allowing room temperature air to enter, conduction through the attaching structure of the refrigerator, cooling of the used containers which will be returned to the refrigerator, and also the capacity is to be provided to chill some dehydrated juices after they have been reconstituted. Finally, an added heat load which is to be considered, is the possibility of storing fecal matter in the refrigerator. Examining in detail the 16 cubic ft. refrigerator, (3-man, 30-day mission):

1. Heat Transferred Through Walls

The value of 200 BTU's per hour previously determined will be used.

2. Heat Transferred as a Result of Opening Doors

A conservative assumption that 10% of the refrigerator air will be replaced by air at cabin temperature each hour will be made. This is based on a rough estimate of one opening per hour. Since the three men may be required to consume their meals at separate times, at least nine meals or nine openings must be considered. Therefore, a conservative allowance for 24 refrigerator openings, or one per hour has been made. The 20% air change per opening added to the refrigerator is also conservative because the basic design will insure that this effect is minimized. On the basis of this conservative estimate, the following heat load can be calculated.

Assuming the cabin environment to be air at one atmosphere, the weight, W_a, to be cooled per hour corresponding to 10% of 16 cubic ft., or 1.6 cubic ft., is

$$W_a = \frac{pV}{RT} = \frac{14.7 \times 144 \times 1.6}{53.3 \times 530} = 0.120 \text{ lb./hr.}$$

where p = absolute press. (lb./sq. ft.)
V = Volume (ft^3)
R = gas constant (air = 53.3)
T = absolute temperature ^{o}R

The relationship which determines the amount of heat to be removed in order to cool this air and freeze out most of the water contained in it is

$$q = W_a \left[(h_o - h_s) + (w_o - w_s)\ h_{sub} \right]$$

where h_o = enthalpy of air supplied

h_s = enthalpy of air in the refrigerator

w_o = lb. of H_2O vapor/#dry air supplied

w_s = lb. of H_2O vapor/#dry air in the refrigerator

h_{sub} = enthalpy of sublimation (1220 BTU/#)

Assuming the cabin air to be at a temperature of 70^{o}F and the relative humidity to be a maximum of 60%, then

h_0 = 28 BTU/lb. and w_0 = 0.008 lb./lb. dry air.

The conditions inside the refrigerator are -20^{o}F and 100% R.H. then

h_s = -5 BTU/lb. and w_s = 0.0004 lb./lb. dry air

Thus, the heat to be removed in order to cool the air is

$h_o - h_s = 28 - (-5) = 33$ BTU/lb. of air

and the heat to be removed in order to freeze out the water vapor is

$h_{sub} (w_o - w_s) = 1220\ (0.008 - 0.0004) = 1220 \times 0.0076$

$= 9.3$ BTU/lb. of air

Therefore, the total heat to be removed for this effect is

$q = 0.120\ (33 + 9.3) = 5.1$ BTU/hr.

3. Heat Transferred by Conduction Through the Structural Members

This value will not be calculated in detail, since the calculation is complex and is dependent on the refrigerator design. A relatively small value will be assumed considering the use of thermal insulators, either as structural members themselves or separating the structural members from the refrigerator. An approximate value of 10 BTU/hr. has been assigned to this heat path.

4. Removal of Heat from Containers Replaced in the Refrigerators After Use

$Q = WC\Delta T$ - where W/can = 0.25 and C for alum. = 0.22

considering the 33 cans per day used for this mission

$Q = (33 \times 0.25)\ 0.22 \times 90 = 163.4$ BTU

distributing this over a 24 hour period the average heat flow rate is

$q = \frac{163.4}{24} = 6.8$ BTU/hr.

5. Provision of the Capacity to Chill Juice Containers

Estimating that two additional six-ounce juice containers per day consisting of reconstituted juices will be chilled for each man, it will be necessary to provide the capability to chill a total of six such containers each day. However, these containers need only be reduced from room temperature to a chill temperature of $35^{\circ}F$.

Again $Q = WC\Delta T$

for the contents of the containers

$W = 6 \times 0.375 = 2.25\#$, $\Delta T = 70^{\circ} - 35^{\circ}F$, and $C = 1$
$Q = 2.25 \times 1 \times 35 = 78.7$ BTU

for the containers themselves (assumed alum. similar to others)

$W = 0.25 \times 6 = 1.50\#$, $\Delta T = 35^{\circ}F$ and $C = 0.22$
$Q = 1.50 \times 0.22 \times 35 = 11.5$ BTU

Thus, the total heat load per day required for the chilling of these containers is 90 BTU's. On a very conservative basis, that is, requiring that all six containers be reduced in temperature in a period of three hours, the average heat load is approximately 30 BTU's per hour. This conservative value will be used, although it would not be a constant requirement for the entire 24 hour period. Obviously, a considerable lower value would result if the 90 BTU's were equally divided over a 24-hour period.

6. Storage of Fecal Matter

The final item to be considered is the possible necessity of storing fecal matter in the refrigerator. Using the maximum rate of supply of 0.5 lbs. per man per day, and using the specific heat of water, the principle constituent, the following heat load results.

Assuming an original temperature slightly higher than room temperature, say $80^{\circ}F$, and cooling this to $-20^{\circ}F$ then $T = 100^{\circ}F$ $W = 3 \times 0.5 = 1.5$ and $C = 1$.

Again $Q = WC\Delta T$ $\qquad$ $Q = 1.5 \times 1 \times 100 = 150$ BTU

This can be distributed over the entire 24-hour period and the average heat flow rate is:

$$q = \frac{150}{24} = 6.25 \text{ BTU/hr. for a max. value}$$

Thus, in summary, all of the heat load elements considered have made a relatively small contribution to the refrigeration requirement except for the heat leakage through the walls. The only other substantial value has been the provision for the capability of chilling items added to the refrigerator.

The capability to pull down the entire refrigerator to the required low temperature will be briefly considered. The total weight of food and containers to be carried for a 30-day, 3-man mission is 746 pounds, excluding internal elements of the refrigerator itself. With a substantial weight such as this, involved, the refrigerator capacity for lowering this weight a significant number of degrees would be exceedingly large, much out of proportion to the other requirements described. However, since the refrigerator will be loaded with cold food (the food could be even colder than usual for initial loading, or the box itself could be pre-chilled), the thermal inertia of such a massive quantity would be high. This will be shown in detail in a subsequent section. Thus, initial pull down would not be required, nor would ground cooling prior to launch be necessary. Should failure occur in flight, a considerable amount of time would be available for effecting minor repairs without a serious increase in the refrigerator temperature. If a serious failure or damage should occur in flight, it would probably be necessary to abort the mission in any case, thus, pull down capability will not be considered. The total value for refrigerator heat load independent of the requirements for storage of fecal matter is 251.9 BTU's per hour as summarized in Table 8. This will be rounded out to approximately 250 BTU's per hour for a 16 cubic ft. refrigerator, accommodating three men for 30 days. The addition of the requirement for the storage of fecal matter would not make a significant difference, since the BTU per hour value is small, and in light of the fact that a significant additional capacity was provided in assigning a value of 30 BTU's per hour for the capability of chilling the juice containers. Thus, this value of 250 BTU's per hour can be used as a basis for establishing the relationship between mission duration and refrigeration requirement. (See Table 9.). This relationship as a continuous function is shown as Figure 30.

D. Thermal Inertia Considerations

Since a considerable weight of food is to be stored in the refrigerator, the thermal inertia aspects of the food storage problem should be examined. For a three-man, 30-day mission approximately 746 pounds, including containers, will be initially stored in the refrigerator. Of this total weight, 480 pounds is food which will be consumed. However, the used containers will be returned to the refrigerator thereby insuring a weight of 266 pounds remaining at the end of the mission. Of course, if the weight of the containers is significantly reduced, both total and final values of weight will be similarly reduced. On the other hand, the addition of fecal matter would add weight. However, using the weights presently available, and neglecting the possible addition of fecal matter, the following analysis was performed. Failure of the refrigerator was considered at different times throughout the mission and the corresponding weight of the refrigerator contents was used. The same magnitude of heat loads as originally required was used, that is, the 250 BTU's per hour as previously established, except for a correction due to the increasing temperature within the refrigerator. Both the specific heats of the food and the containers were considered. The specific heat of the food was estimated to be equivalent to that of water and the specific heat of the containers was assumed to be that of aluminum. By using an approximation process consisting of first estimating the temperature drop which could be expected, and correspondingly reducing the heat leak into the refrigerator, the actual temperature drop could be determined by a reiterative process.

The results of this analysis are summarized on Table 10. This table illustrates the effect of the thermal inertia on the temperature of the refrigerator corresponding to failure or shut down of the refrigeration system during the first day, and on five day

Table 8. Summary of Heat Loads Estimated for the 16 Ft.^{3} Refrigerator (three-Man Thirty-Day Mission)

HEAT LOAD SUMMARY

Cause	Heat Load BTU/hrs.
1. Transferred Through Walls	200
2. Result of Opening Doors	5.1
3. Conduction Through Structure	10
4. Cooling Used Containers	6.8
5. Capacity for Chilling Added Juice Containers	30
TOTAL	251.9

REFRIGERATION REQUIREMENT

Utilizing the information developed for the 16 cu. ft. refrigerator as a basis, the refrigeration requirement as a function of mission duration and corresponding refrigerator size will be established.

$$250/16 = 15.6 \text{ BTU/hr/ft}^3$$

For the very large sizes (over 1000 ft^3) a requirement of about half of this, 8 $BTU/hr/ft^3$, will be used. This results from the fact that volume increases at a more rapid rate than does surface area.

A value of 10 $BTU/hr/ft^3$ will be used for the in-between size (600 ft^3), and a relatively high value of 25 $BTU/hr/ft^3$ will be used for the very small size (1.25 ft^3).

Table 9. Relationship Established Between Mission Duration and Refrigeration Required

TABULATED REQUIREMENTS

Mission Man-Mos.	Size (ft^3)	Sizing Rate $BTU/hr/ft^3$	Refrig. Req. BTU/hr.	Mission (Man-Days)
0.25	1.25	25	30	7
3	16	15.6	250	90
120	600	10	6000	3650
720	3400	8	27,200	21,900

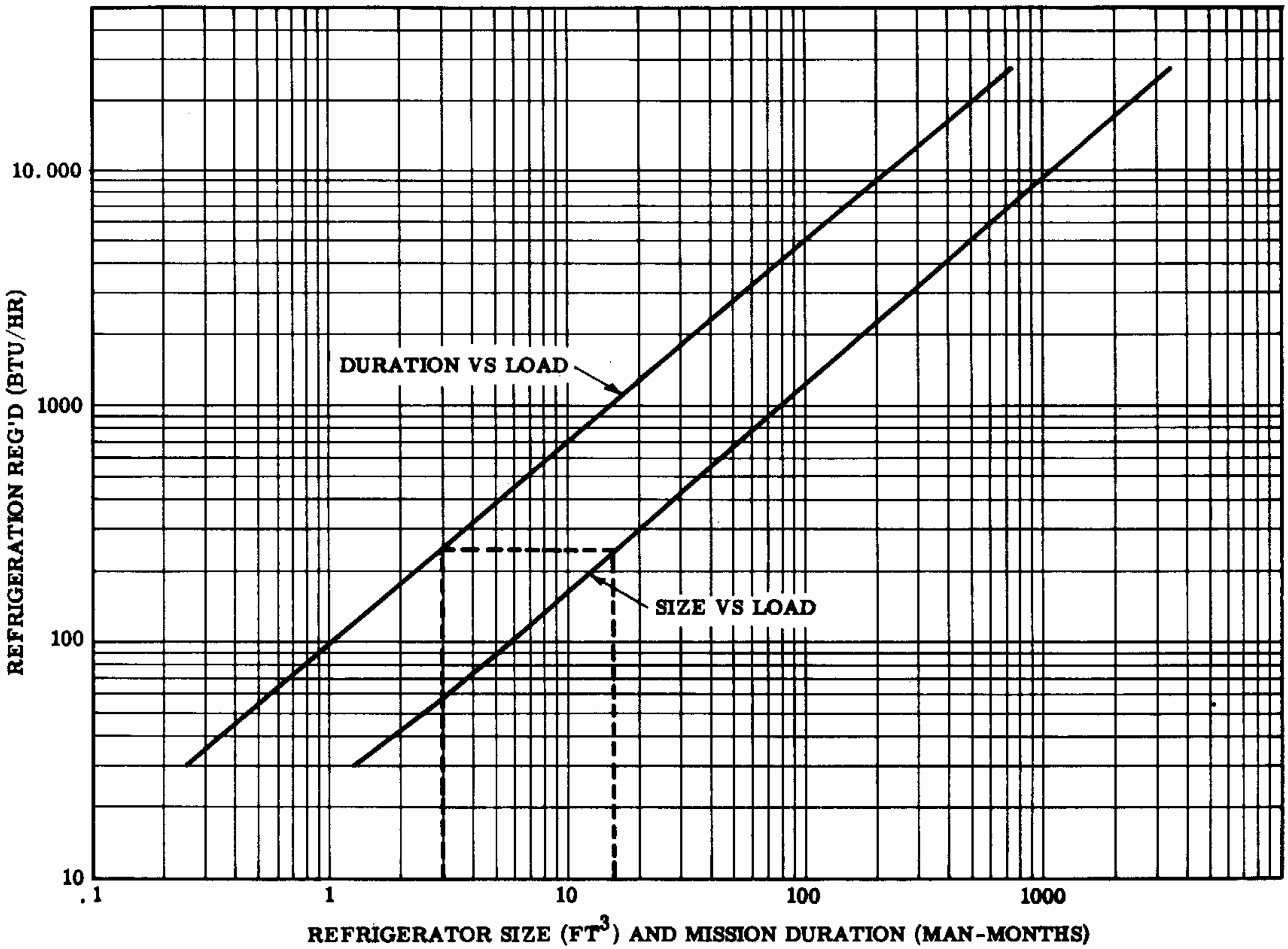

REFRIGERATION REQ'D (BTU/HR)
10.000
1000
100
10
DURATION VS LOAD
SIZE VS LOAD
.1
1
10
100
1000
REFRIGERATOR SIZE (FT³) AND MISSION DURATION (MAN-MONTHS)

Figure 30

Table 10. Thermal Inertia Situation for a Three-Man-Thirty-Day Mission

THERMAL INERTIA OF SYSTEM

Assuming Ref. Coils at -25°F Initially

Day of Failure or Shut Down	First Day T	Temp at End of 1st Day	Days to Reach 0°F	Days to Reach 30°F
0	11.3	-13.7°F	2.3	6.1
5	12.8	-12.2°F	1.9	4.6
10	16.0	- 9.0°F	1.6	3.6
15	19.8	- 5.2°F	1.3	2.9
20	25.5	+ 0.5°F	1.0	2.3
25	36.1	+11.1°F	0.7 (17 hrs.)	1.7

intervals subsequent thereto. The temperature drop resulting from the first 24 hours following shut-down is indicated as well as the final temperature at this time. The number of days of non-operation of the refrigerator required for the temperature to reach zero degrees F, and also 30°F are included in this table. It can be seen that a day or more of thermal inertia is available. The minimum value is, in fact, 17 hours before the temperature of the walls will rise from -25°F to 0°F. In addition, two or more days are available prior to the temperature of the walls reaching 30°F. The food at a slightly lower temperatre than the walls is considered to be at the threshold of defrosting at this point. Of course, these values would have to be modified if it were desired that the refrigerator temperature be maintained closed to 0°F. However, in order to take advantage of the maximum thermal inertia which could be made available, it appears advisable to maintain the average temperature in the refrigerator at 10°F or possibly lower. During the design study phase, consideration will be given to an arrangement whereby the temperature of those containers which are about to be used could be at a slightly higher temperature (nearer the desired value of 0°F) while the overall refrigerator temperature is maintained at a lower value.

The thermal inertia characteristic has several other interesting aspects. It can be seen, for example, that the refrigerator, fully loaded, could remain on the ground with the vehicle in the launch position for 2.3 days before the temperature in the refrigerator would reach 0°F. Thus, there would be no requirement for ground cooling. If necessary, the original temperature of the box could be reduced to a value even lower than -20°F in order to insure a launch temperatore of -10°F. Another situation during which the thermal inertia of the refrigerator would of use would be at a time when the space vehicle is in close proximity is to a large heavy body such as might be the case of a lunar observing space vehicle. If it is required to approach the moon at a very low altitude, the thermal radiation effects of the lunar body as well as reflected solar radiation could create a serious problem for a refrigeration technique employing radiation to space. However, due to the availability of thermal inertia, the refrigeration system could be shut down for long periods of time without serious mishap. Finally, the thermal inertia available could be used to determine the requirement for dry emergency rations. If the three-man, 30-day mission were a circum-lunar type, it can be seen that after the first 15 days, less than three days of thermal inertia are available to prevent food spoilage. If it were desired to provide five days food at all times, even under emergency conditions, then approximately three days of dry emergency rations could be provided (certain dry foods will already be stored aboard the vehicle and this may be sufficient).

E. Heat Pumping Considerations

Since the ability to radiate heat is a function of the fourth power of the temperature of the radiator, it would appear desirable to increase the temperature of the heat to be discarded. However, not so obvious is the fact that the process of heat pumping itself adds heat to the original refrigeration requirement. The total heat that must be discarded is then dependent upon the efficiency of the heat pump. In addition, the weight penalty of the power required must be accounted for, as well as the weight penalty of the heat pump itself. Thus, in order to evaluate the desirability of pumping heat, a break-even situation will be examined. That is the point at which the weight of a direct radiation system (without the heat pump) equals a heat pump system radiating to space at a higher temperature. Initially, black body conditions only will be considered. That is, the effect of the earth, the sun, and solar reflected radiation will be neglected, and we will consider the so-called perfect radiator. This is a situation which can be approached in space and will be better described in a following section of this report.

The only element to be considered in the direct radiation system is the radiator itself. The weight of this radiator is determined on the basis of the surface area required. The surface area required can be expressed in the form

$$A_o = \frac{q_r}{\sigma \epsilon t_o^4}$$

where A_o = surface area (sq. ft.)
q_r = refrigeration effect (BTU/hr.)
σ = Stefan - Boltzmann constant
ϵ = emissivity (dimensionless)
T_o = radiator temp. ^oR

Since we are considering a perfect radiator, $\epsilon = 1$ and $A_o = q_r/\sigma\ T_o^4$. To convert this to weight, a value for the specific weight of the radiator, w_r (lbs/sq. ft.), must be applied. Since a reasonable value for w_r has been found to be one pound per square foot (see section IV-B-3), then the corresponding weight of this radiator, R_{wo}, is $w_r A_o$ or

$$R_{wo} = \frac{q_r}{\sigma T_o^4}$$

For the corresponding heat pumping situation, several weight elements are to be considered. The first of these is the weight of the new radiator. This new radiator must discard both the heat due to the refrigeration effect, q_r, and also the additional heat due to the energy input to the heat pump, W. The ratio of these two values is defined as the coefficient of performance (COP). Thus, the total heat to be discarded is $q_r + W$, and since

$$COP = q_r/W$$

then the total heat to be rejected can be expressed as

$$q_t = q_r + q_r/COP \qquad \text{or} \qquad q_t = q_r\left(1 + \frac{1}{COP}\right)$$

Due to the heat pumping effect, this heat will be rejected at a new temperature T_n (^oR). The radiator area required is then described by

$$A_n = q_r \left(\frac{1 + \frac{1}{COP}}{\sigma T_n^4}\right)$$

and since a radiator specific weight of one pound per square foot is to be used, this expression also describes the required radiator weight. That is

$$R_{wn} = q_r \left(\frac{1 + \frac{1}{COP}}{T_n^4}\right)$$

The weight penalty for the provision of the input energy, W, necessary to drive the heat pump must also be determined. Once again expressing W in terms of q_r and COP and dividing by a value of power supply specific weight, w_p (in BTU/lb.). The weight of furnishing the power, P_{wn}, can be expressed as

$$P_{wn} = q_r/COP \times w_p$$

The final weight which must be considered is the weight of the heat pump itself, H_{wn}. This weight can be described as being a function of the refrigeration effect, q_r, and will be q_r/w_h, where w_h is the specific weight of the heat pump in BTU/hr/lb. Therefore, the weight of the entire heat pumping, HP_w, system can be described by the sum of these three terms.

$$HP_w = R_{wn} + P_{wn} + H_{wn}$$

or

$$HP_w = \frac{q_r \left(1 + \frac{1}{COP}\right)}{\sigma\ T_n^4} + \frac{q_r}{COP \times w_p} + \frac{q_r}{w_h}$$

For break-even, the relationship for R_{wo}, the weight of the direct radiation system can be set equal to HP_w, the weight of the heat pumping system. This equation will be solved for COP in terms of T_n (the temperature at which heat is to be discarded using the heat pump system).

$$\frac{q_r}{\sigma T_o^4} = \frac{q_r \left(1 + \frac{1}{COP}\right)}{\sigma\ T_n^4} + \frac{q_r}{COP\ (w_p)} + \frac{q_r}{w_h}$$

It can be seen that q_r appears in all the terms and thus will be divided out of the equation; that is, the relationship is independent of the refrigeration effect term or

$$\frac{1}{\sigma\ T_o^4} = \frac{\left(1 + \frac{1}{COP}\right)}{\sigma\ T_n^4} + \frac{1}{COP \times w_p} + \frac{1}{w_h}$$

simplifying and solving for COP

$$\frac{1}{\sigma\ T_o^4} = \frac{COP + 1}{COP \times \sigma T_n^4} + \frac{1}{COP \times w_p} + \frac{1}{w_h}$$

$$\frac{COP}{\sigma T_o^4} = \frac{COP + 1}{T_n^4} + \frac{1}{w_p} + \frac{COP}{w_n}$$

at this point some new constants will be defined and introduced

$$R_n = \sigma T_n^4 \qquad \text{(the independent variable)}$$

$$a = \frac{1}{\sigma T_o^4} \qquad \text{(a constant)}$$

$$b = \frac{1}{w_p} \qquad \text{(a constant)}$$

$$c = \frac{1}{w_n} \qquad \text{(a constant)}$$

thus the equation can be represented by

$$a\,(COP) = \frac{COP + 1}{R_n} + b + c\,(COP)$$

continuing to simplify and solve for COP

$$aR_n\,(COP) = COP + 1 + bR_n + cR_n\,(COP)$$

$$COP\left[R_n\,(a-c) - 1\right] = bR_n + 1$$

Finally, we have the relationship for COP in terms of the temperature of the heat to be discarded for the heat pumping system.

$$COP = \frac{bR_n + 1}{R_n\,(a-c) - 1}$$

This is the break-even COP. If a heat pump COP greater than its value is attainable at the new temperature T_n, representing a heat pump ΔT of $T_n - T_o$, then the heat pumping system weight would be lower than that of the direct radiation system. Figure 31 is a plot of the break-even COP as a function of ΔT. It can be seen from the figure that extremely high COP values are required for break-even at moderate ΔT conditions. As the ΔT increases, the COP required for break-even decreases, but is still much higher than present technology permits, or even that which would appear possible in the

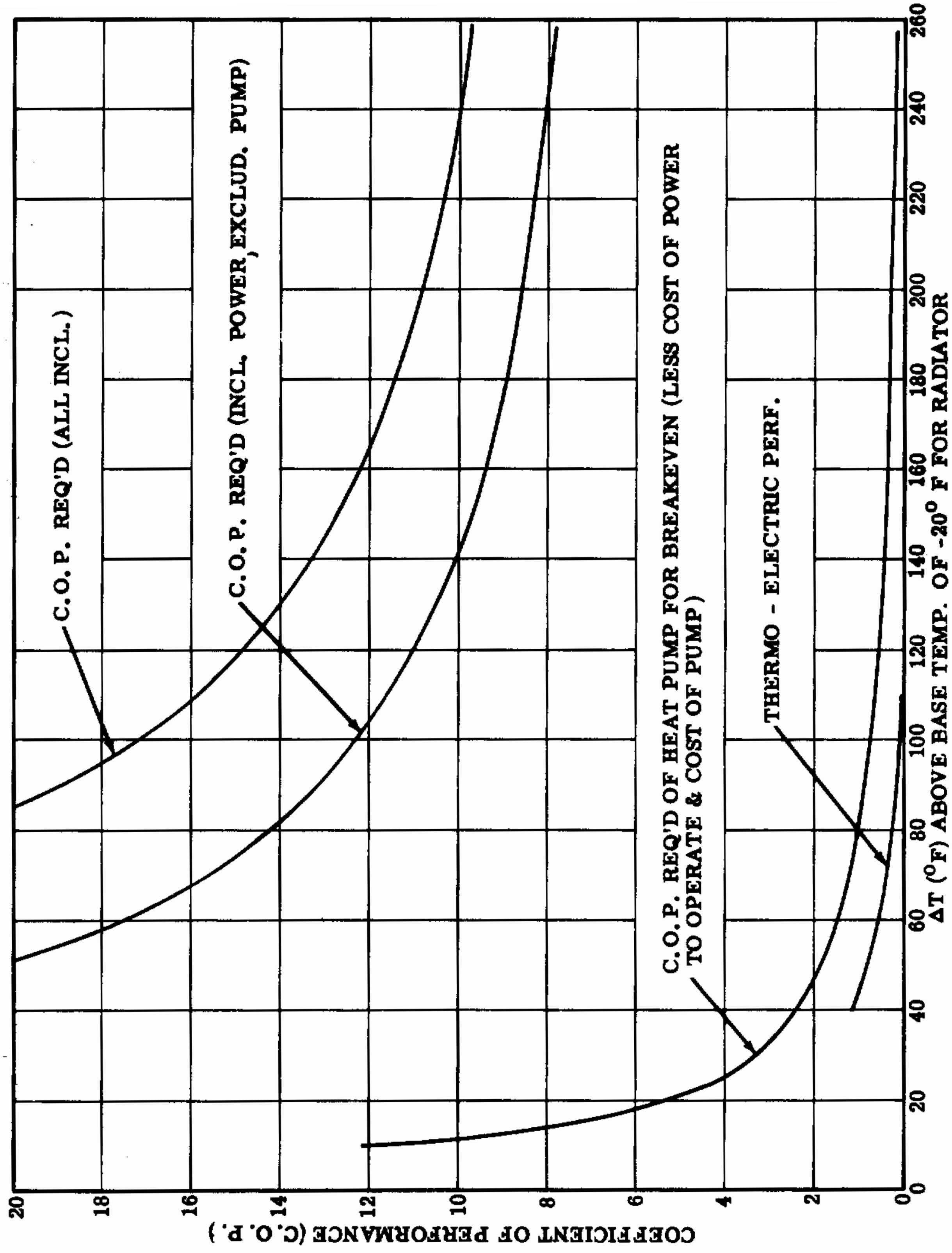


Figure 31 **BREAKEVEN C.O.P. REQUIRED USING HEAT PUMP**

foreseeable future. In order to demonstrate the cumultative effect of the added factors requiring successively higher COP's for break-even, three curves have been plotted on Figure 31, and are compared to the capability of a thermoelectric heat pump. The first of these curves neglects the weight addition due to the power required, and also the weight of the heat pump itself (b = c = 0). The expression for COP for this case is

$$COP = \frac{1}{\left(\frac{T_n}{T_o}\right)^4 - 1}$$

The second curve adds in the effect of accounting for the weight of the power required, but excludes the weight of the pump. In this case c = o, and could represent a very high specific weight for the heat pump, then

$$COP = \frac{bR_n + 1}{aR_n - 1}$$

The third curve accounts for all of the factors and is the entire equation. At this point, it is interesting to note that if c is greater than a, the COP is negative, an impossible situation indicating that there is no real value of COP which will make the weight of the two systems equal. This is actually the case in the situation being considered, if w_h is sufficiently low. This will be demonstrated by evaluating the constants a, b, and c.

$$a = \frac{1}{\sigma \, T_o^4}$$

since $\sigma = 0.1713 \times 10^{-8}$ and using a radiator temperature of -20^oF for the direct radiation system

or

$$a = \frac{1}{0.1713 \left(\frac{440}{100}\right)^4} = \frac{1}{64.2}$$

$$= 0.01558$$

Solving for b

For, W_p, a reasonable value which can be found in the literature is three watts per pound, or approximately ten BTU's per pound. This is slightly optimistic, but is one which should be attainable in the relatively near future. Therefore, b = 0.1.

Solving for c

Establishing a value for w_h is a little more difficult. This value can vary anywhere between 20 BTU's per pound (commercially available Westinghouse WX814 thermoelectric units for a 1-watt couple and a ΔT of $90^\circ F$), and perhaps 400 BTU's per pound (large vapor compression unit). Thus, c can vary between 1/20 and 1/400 or between a range of 0.05 to 0.0025.

Since a value of c = 0.05 is greater than a = 0.01558 from above, a break-even COP does not exist. The curves provided have been plotted in terms of the coefficient of performance because this is the usual manner of describing the performance of refrigeration devices, and the current state of the art of existing refrigeration techniques can be readily found in terms of this coefficient.

One final situation will be considered, and that is the special case where the radiator capability is reduced because of earth effects or other external heat inputs. Let us examine the case where the radiator capability is reduced to 25 BTU's per sq. ft., for a radiator temperature of approximately $-25^\circ F$. For this case a more realistic value of $\epsilon = 0.9$ will be used rather than unity as for a perfect radiator. The following conditions will also be established: (1) A refrigerator temperature of $440^\circ R$ which will require a sink temperature of at least $430^\circ R$. For the sake of comparison this will be considered the temperature of the radiator for the direct radiation system, and also the cold temperature for the heat pump. (2) Considering a ΔT of $100^\circ F$. for the heat pump, the hot side temperature would be $530^\circ R$ and a corresponding radiator temperature of $520^\circ R$ can be used. Thus, $T_o = 430^\circ R$ and $\sigma \epsilon T_o^4 = 52.7$ BTU/hr/sq. ft. (3) Since we are considering the net radiative heat flow per square foot of area to be 25 BTU/hr., then 27.7 BTU/hr. must result from the external inputs,

and $R_n = \sigma \epsilon T_n^4$ - input flux where $T_n = 520^\circ R$

$= 112.7 - 27.7 = 85.0$ BTU/hr/sq. ft.

also $a = \dfrac{1}{25} = 0.04$ A value of b = 0.1 will be used as before.

For W_h, an optimistic value of 100 BTU/lb. will be used or c = 0.01. Thus, to solve for the break-even COP

$$COP = \frac{bR_n - 1}{R_n (a-c) - 1} = \frac{(0.1 \times 85) - 1}{85\,(0.04 - 0.01) - 1}$$

$$= \frac{9.5}{1.55} = 6.3$$

Thus, it can be seen that the required coefficient of performance would have to be 6.3 for break-even, which is an extremely high value for the conditions established. It is possible for a large unit, that is, one capable of a very large refrigeration effect, that W_h might be higher than the value used, but the combination of a large ΔT, high W_h, and high COP required is not yet possible. This would be an area for possible future development. However, a further consideration must be that of operation under zero G conditions, that is to say these high coefficient of performance requirements must be maintained in the absence of gravity. If a heat pump is ever to be developed for use in the type of situation being analyzed, the following requirements would have to be satisfied;

adequate functioning in the absence of gravity, a high ΔT capability must exist combined with high W_h and a high COP, and the power to operate such a unit must be provided with a high W_p.

Thus, in summary, it can be seen that existing equipment or equipment which can be imagined will exist in the foreseeable future, cannot be expected to have the efficiency of operation required to justify pumping and the heat to be discarded to a higher temperature.

SECTION IV

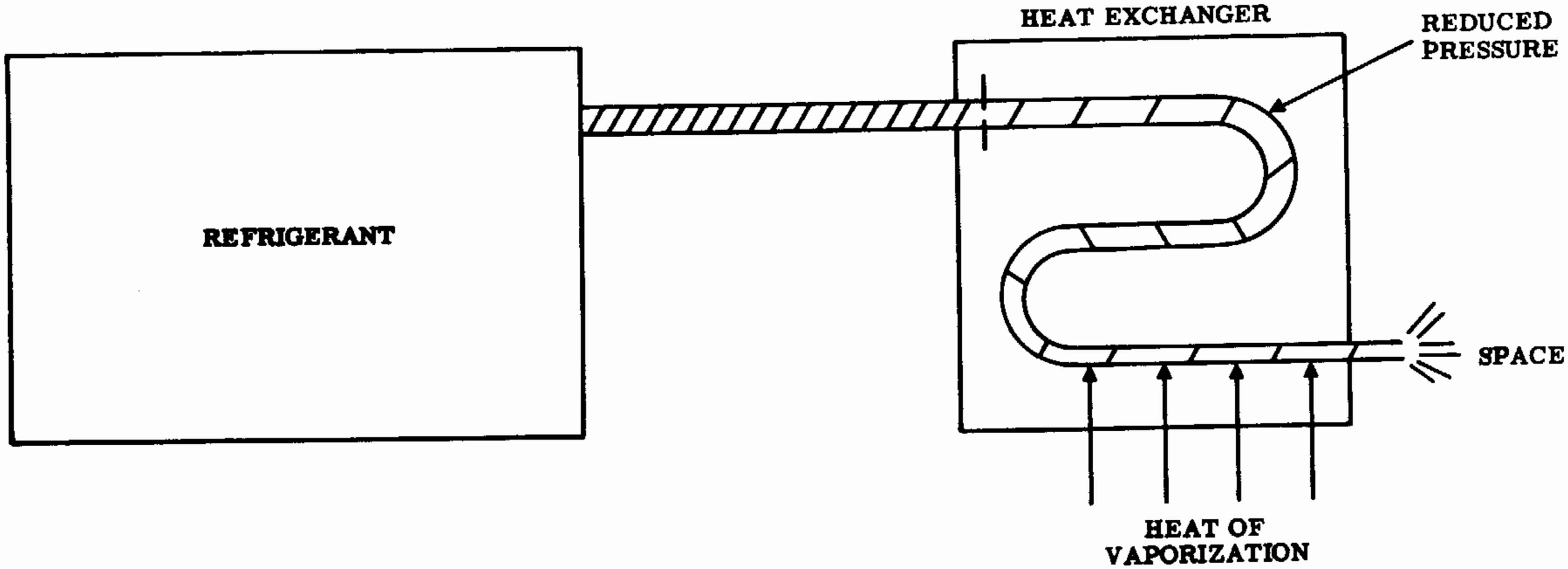
TECHNIQUES CONSIDERED BEST SUITED FOR SPACE VEHICLE APPLICATION

As a result of the literature survey and preliminary analysis of the systems available, the following techniques appeared to offer the best possibilities and have, therefore, been examined in detail.

A. Expendable Refrigerant

The concept of an expendable refrigerant appeared to be most attractive for short mission durations. Utilizing such a system, a radiator and a heat transport fluid would not be required. The only unique elements necessary would be adequate storage space for the quantity of refrigerant required and an evaporation chamber of some sort. Since the near zero pressure of space is readily available, a system using a low pressure can be considered. The schematic diagram for such a system is illustrated as Figure 32. Optimum design requires that the average temperature of the heat exchanger be very nearly that of the temperature of the refrigerator and it is desirable that only a small temperature gradient exist across the heat exchanger. A refrigerant which would change state at the required temperature was considered desirable. A number of possible refrigerants were examined and a careful study of the literature was made. Metallic solids at extremely low pressures could possibly be utilized since they have extremely high heats of vaporization or sublimation. However, their rate of sublimation would be extremely slow, and it appeared that the problem of providing sufficient surface area of the refrigerant in order to remove the desired heat at the proper rate would be severe. Among the liquid refrigerants, water, as usual, had the best heat of vaporization. However, water is solid at $-20^{\circ}F$ at ordinary pressures. This is no problem and, in fact, is of some benefit, since at an appropriately low pressure the ice will sublimate and convert directly to water vapor with an energy input equal to the heat of sublimation. This could then be used to produce the required refrigeration effect. The phase diagram for water at low pressures is illustrated as Figure 33. From this figure, it can be seen that a temperature of $-30^{\circ}F$ can be produced with a pressure of 0.0035 pounds per square inch or a temperature of $-10^{\circ}F$ can be produced with a pressure of 0.011 pounds per square inch. Temperatures slightly lower than that required for the food have been considered in order to permit some thermal gradient within the refrigerator. Thus, if a chamber containing ice were maintained at the pressure levels prescribed, heat could be removed and the corresponding temperatures would be maintained. The only added element necessary for such a system would be a pressure relief valve which maintains the pressure within these limits which are somewhat above the essentially zero pressure available in space. A rudimentary laboratory model of such a system was assembled and tested. A descriptive diagram of the ice sublimation system tested is illustrated in Figure 34. Ice cubes were used during this testing rather than a solid block of ice, because it was found that more uniform sublimation occurred and the thermal gradient across the ice was reduced. That is the heat input could be distributed to a larger surface area of the ice insuring uniform sublimation, rather than concentrated action at the edges of the ice adjacent to the heat input plate.

The elements of the system will be described. The ice cubes were held against the sublimation plate by spring pressure. A heat input simulating the refrigerator loading was added to the sublimation plate by the use of a heater attached to it. The heating element was surrounded by insulation in order to assure that essentially all of the heat



EXPENDABLE REFRIGERANT

Figure 32

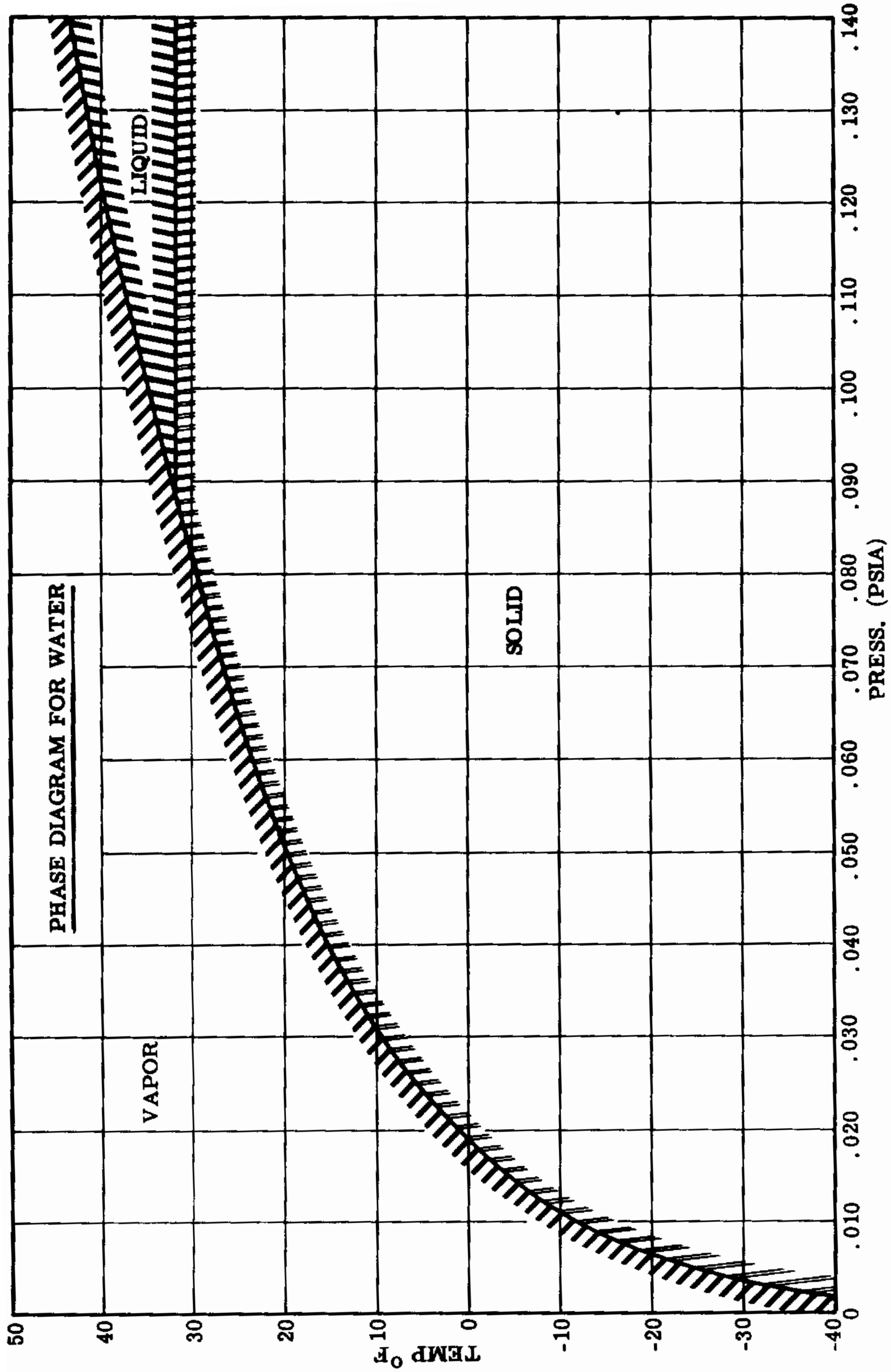
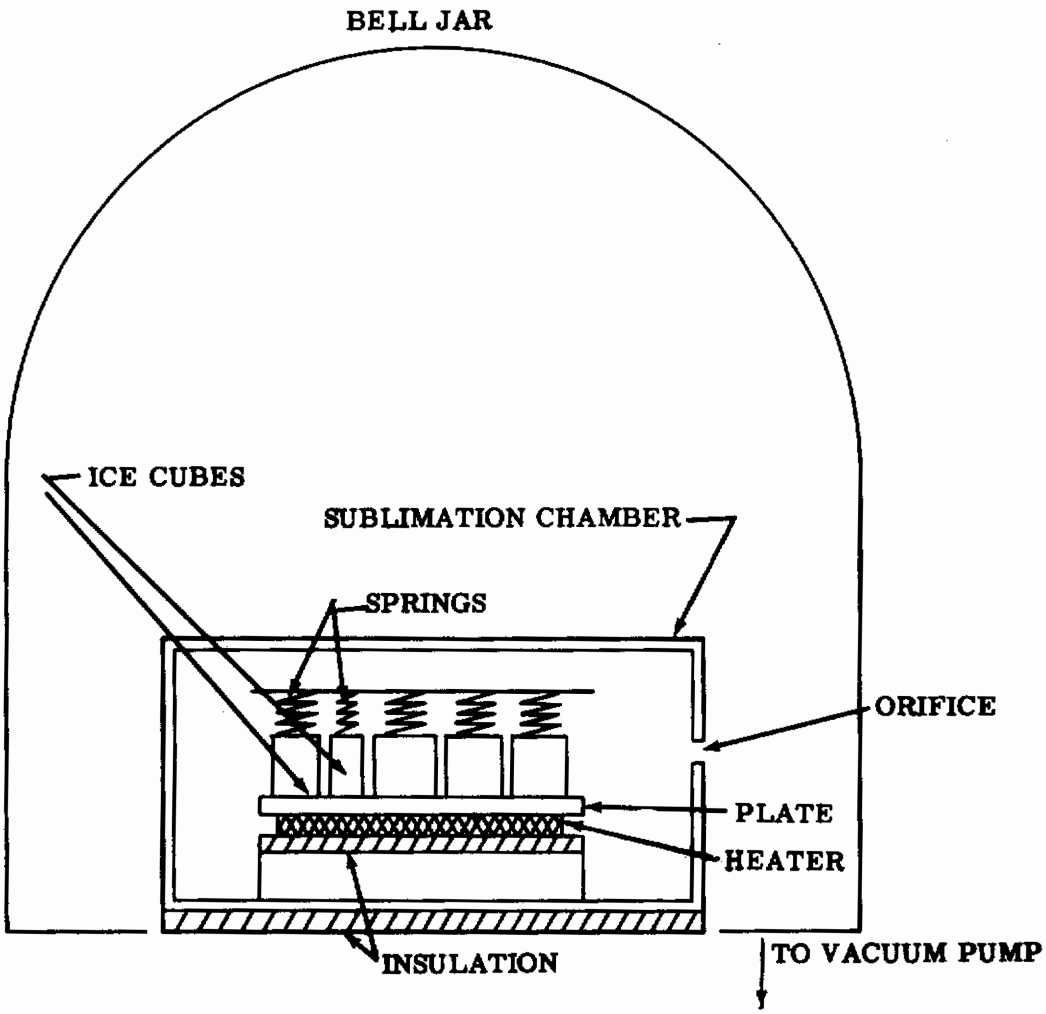

PHASE DIAGRAM FOR WATER
LIQUID
VAPOR
SOLID
TEMP °F
50
40
30
20
10
0
-10
-20
-30
-40
PRESS. (PSIA)
0
.010
.020
.030
.040
.050
.060
.070
.080
.090
.100
.110
.120
.130
.140

Figure 33



TEST SETUP FOR ICE SUBLIMATION SYSTEM

Figure 34

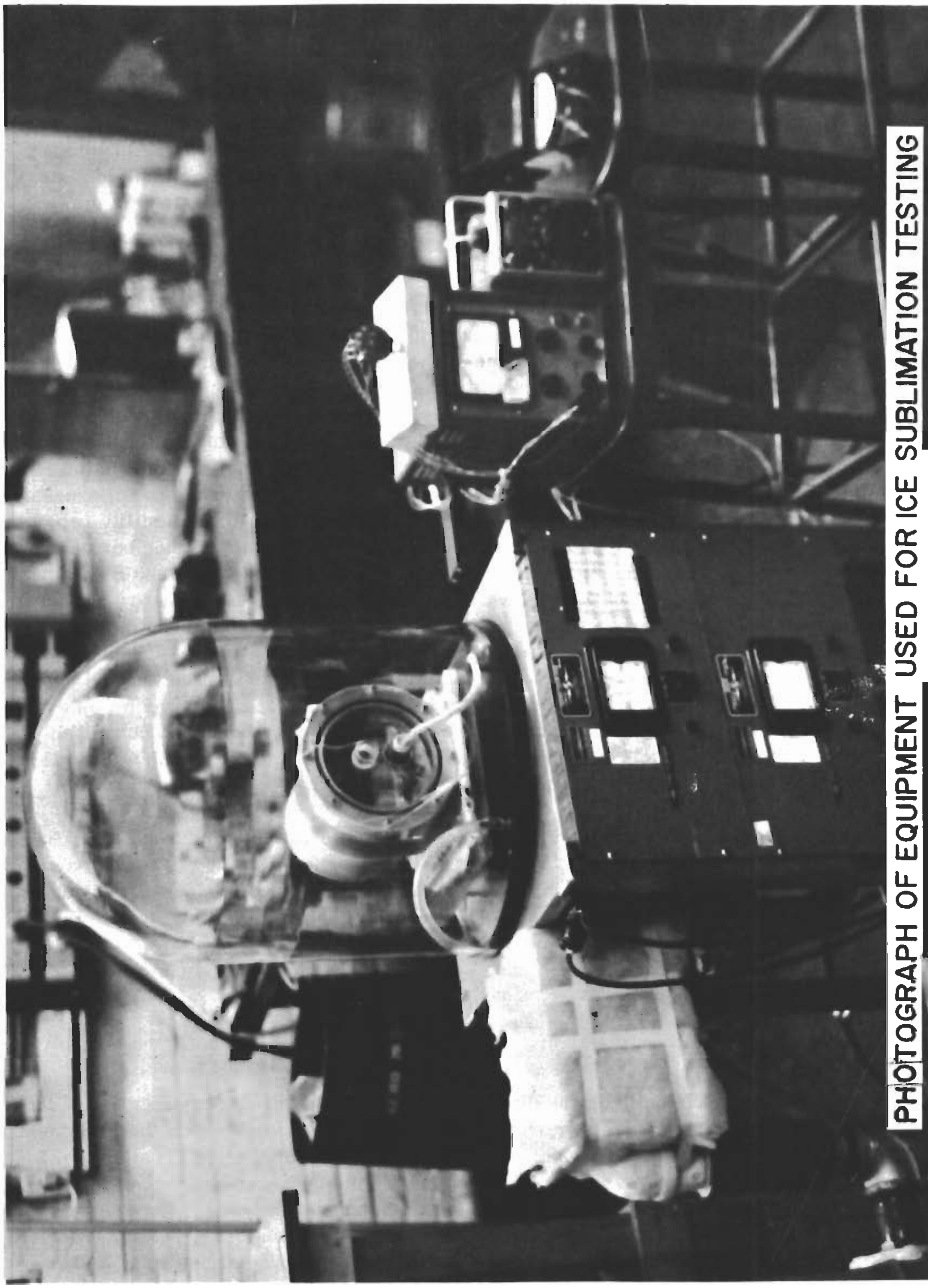
produced by the heater would be transmitted to the sublimation plate. The simulated sublimation chamber contained an orifice which was intended to produce a pressure difference between the sublimation chamber and the surrounding bell jar. In this way, a low vacuum pressure could be produced in this bell jar, simulating space conditions, and slightly higher pressure would then exist inside the sublimation chamber. By controlling the pressure in the bell jar, the pressure within the sublimation chamber was maintained near the desired value, thereby insuring control of the temperature of the sublimation plate. A slight difficulty was encountered in maintaining the exact pressure desired in the bell jar, therefore, slightly lower sublimation chamber pressures were achieved than that intended. Therefore, correspondingly lower plate temperatures were realized. A photograph of the test set-up is included as Figure 35. Summarizing data of the test conducted is shown in Table 11. This fundamental arrangement demonstrated the ability to add heat to the system which was absorbed by the conversion of the ice to water vapor, and which in turn was discarded to the simulated space of the bell jar and vacuum pump. The arrangement shown was capable of holding the plate temperature below $-20^{\circ}F$ with a heat input of five watts (17 BTU/hr.) for several hours with only a relatively small reduction in the size of the ice cubes. A close-up photograph of the ice cubes following this test is provided as Figure 36. Thus, it appears highly feasible that an ice sublimation system is suitable provided that the quantity of ice which must be initially furnished is not excessive. It is interesting to note that this return to an ice box type of food storage facility represents a considerable modernization over that available in the old home ice box of a generation ago. For this space age system a refrigeration capability of 1220 BTU's per pound of ice (the heat of sublimation) is available rather than the relatively low heat of fusion (144 BTU/lb.) originally available in the old ice box. An obvious limitation of such a system is one in which large quantities of ice would have to be provided at launch. It is, however, conceivable that a system such as this could be used on a continuous supply basis. That is to say, water could be added to the sublimation chamber which would freeze, thereby providing additional ice, or the sublimation chamber could conceivably be recharged with ice that was frozen in the refrigerator, assuming that the water balance for the vehicle is such that surplus water is available.

Finally, the system is simple and appears to be quite reliable and would be satisfactory for relatively short mission durations. For mission durations of 20 man days or less, an ice sublimation refrigeration system should certainly be considered. It would also appear desirable to investigate the potentiality of a continuous supply system for extending the range of suitability of this concept.

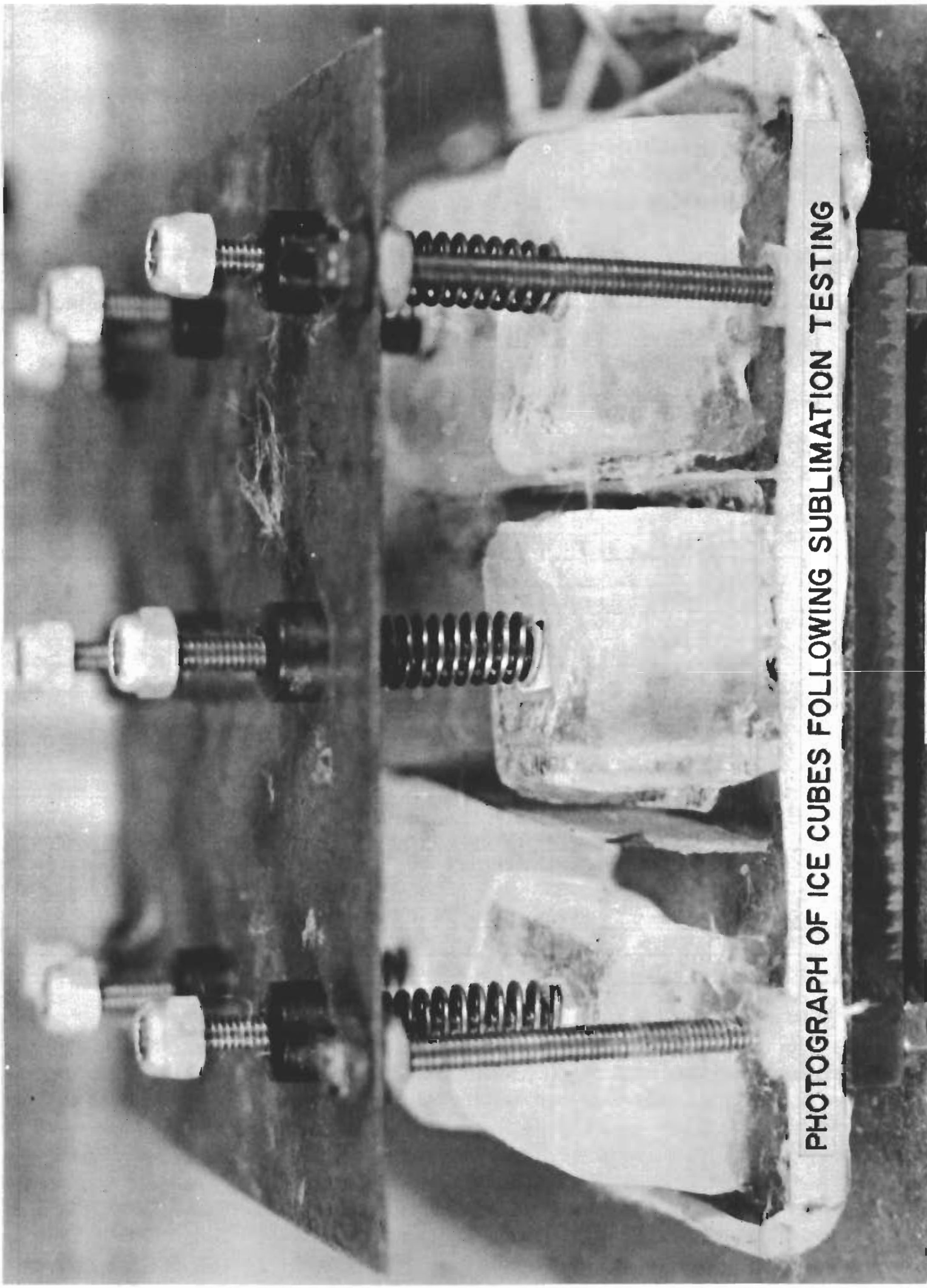
B. Direct Radiation

1. Space as a Heat Sink

With space available as the ultimate heat sink, a thorough investigation of the ability to discard heat was required. It was necessary to determine the range of temperatures which could be maintained on a radiator surface for all situations. For example, a radiator surface in deep space shielded from the sun by the vehicle would effectively have no external heat input whatsoever and would very closely approximate a perfect radiator. On the other hand, a radiator surface on a vehicle orbiting the earth would be required to account for earth radiation, solar and reflected solar radiation, vehicle effects, etc., all of which would reduce the radiative capability. Furthermore, varying orientations of the radiator would produce varying effects. Thus, a very generalized parametric study of the radiator in space



PHOTOGRAPH OF EQUIPMENT USED FOR ICE SUBLIMATION TESTING
FIGURE 35



PHOTOGRAPH OF ICE CUBES FOLLOWING SUBLIMATION TESTING

FIGURE 36

Table 11. Test Data for Ice Sublimation System

Test: Ice Sublimation System

Date 7-19-60 Performed by: R. G. Roos

Approved by: S. Halpert

Laboratory Conditions - Temp 73°F, Pressure 760.6mm Hg.

Data

Time	Temperature °C		Pressure-Microns		Heater Power	Remarks
	Ice	Sub. Plate	Bell Jar	Sub. Chamber	Applied-Volts	
1000	- 3.5	- 3.5	Atm.	Atm.	0	
05	-31	-29	230	400	0	
10	-36	-35.5	250	155	0	
15	-38	-37.5	118	181	0	
25	-38.5	-37	105	181	3.75	Heat On
30	-39	-37	95	165		
35	-39	-37	96	168		
45	-39	-37	95	165		
1100	-39	-37	98	158		
115	-39	-37	98	160		
30	-38	-36.2	100	162		
45	-38	-36	101	165		
1215	-38	-36	86	132	3.75	
		Vacuum Pump Overheating, Shut-down required				
1330	-27	-20	115	210	3.75	*See explanation
40	-36.5	-28	98	162		
50	-38	-28.5	93	152		
1400	-38	-28.5	90	150		
30	-38	-28	89	145		
1510	-38	-26	90	150		
1535	-37.5	-23.5	120	162		
1600	-34.5	-22	172	210	3.75	

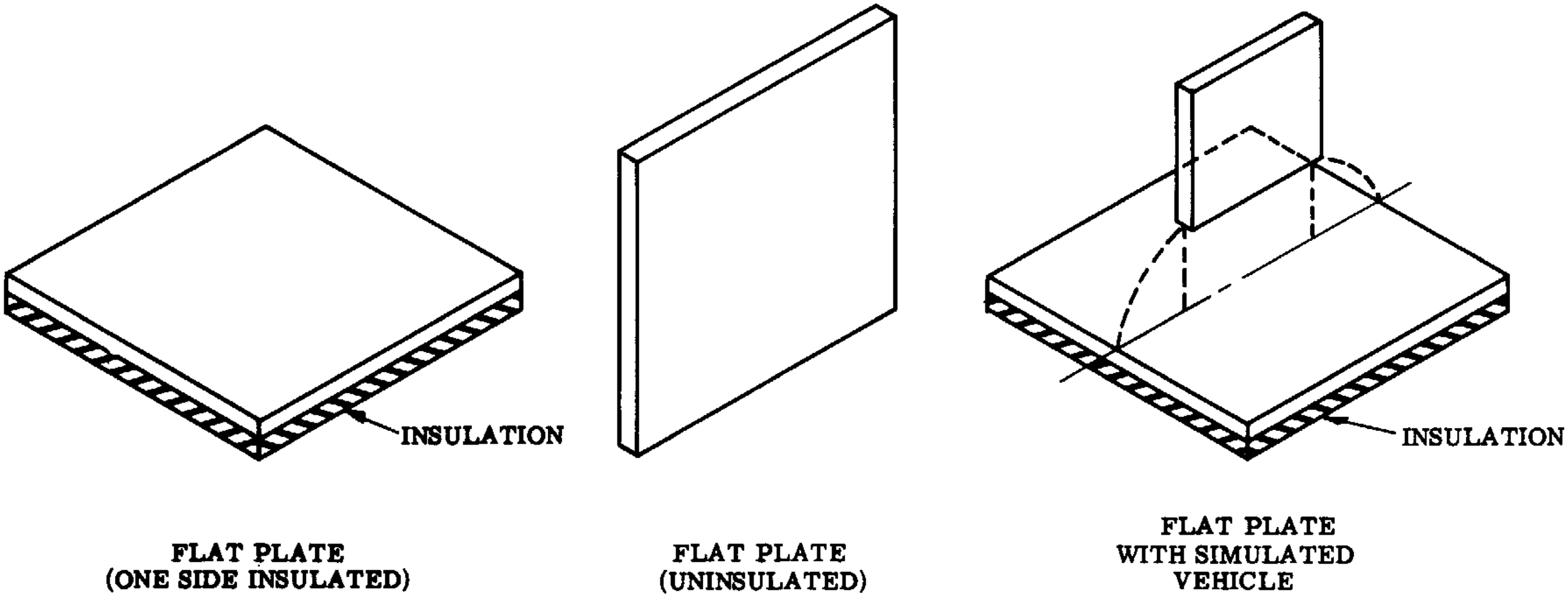
*Note: The ice cubes apparently separated from the plate as a result of pump shut-down and imperfect thermal contact resulted when the test was continued, thus the greater temperature difference between the ice and plate for the second half of the test.

was undertaken. Three basic radiation configurations were considered. The first of these was a flat plate radiator having one face adiabatically shielded. This configuration is representative of a portion of the surface area of the vehicle itself used as a radiator. The second radiator configuration considered was that of a flat plate with both surfaces free to accept and reject heat. This arrangement was representative of a fin-type radiator, but omitted the effect of the vehicle. The third configuration considered was again a fin-type radiator in the presence of a simulated vehicle. These configurations are illustrated as Figure 37. In order to establish the extremes of possible temperature variation of these radiator configurations, a simplified scheme of positionally fixed earth and sun, and a rotating radiator was analyzed for steady state conditions. For each of the radiator configurations, various conditions of exposure were studied, such as exposure to earth alone, to sun alone, and to various combinations of sun and earth. The most significant range of altitudes representative of earth orbits was assumed to be 200 to 600 miles. Since altitudes above 600 miles would be within the Van Allen radiation belt, it is assumed that orbiting manned vehicles would not be desired above this altitude. Extreme altitude orbits outside of the Van Allen belt would not be of interest since these would not represent as severe a condition as a vehicle in close proximity to the earth. That is, the earth's effect on the radiator would be greatly diminished. One other variable was applied as part of this study and that is the ratio of fin size to simulated vehicle size. These variations are illustrated as Figure 38.

As a result of this study, numerous curves were established describing the radiator temperature as a function of position. The temperature which was determined was a steady state value for each positional increment. While this is not entirely accurate in that transient and thermal inertia effects would tend to translate the curve with respect to angular position and diminish the extremes indicated. These effects are not assumed to be very large since at the lowest altitude considered (200 miles), orbit time is approximately one and a half hours and also the mass of the radiator would not be very great. Later in this section, it will be demonstrated that a value of approximately one pound per square foot of surface area can be assumed. A representative curve resulting from this study is shown as Figure 39. As would be expected, the temperature of the radiator is greatest for the particular case when the flat plate radiator surface is perpendicular to the sun's rays. As the radiator continues to rotate, the temperature falls off until the radiator surface is parallel to the sun's rays. At this point, the temperature would remain constant throughout the next 180° of rotation, since the earth's effect is constant. When the radiator again passes the point where it is parallel to the sun's rays the temperature would rise until the surface is once again perpendicular to the solar flux. Appendix VII contains the details of this study.

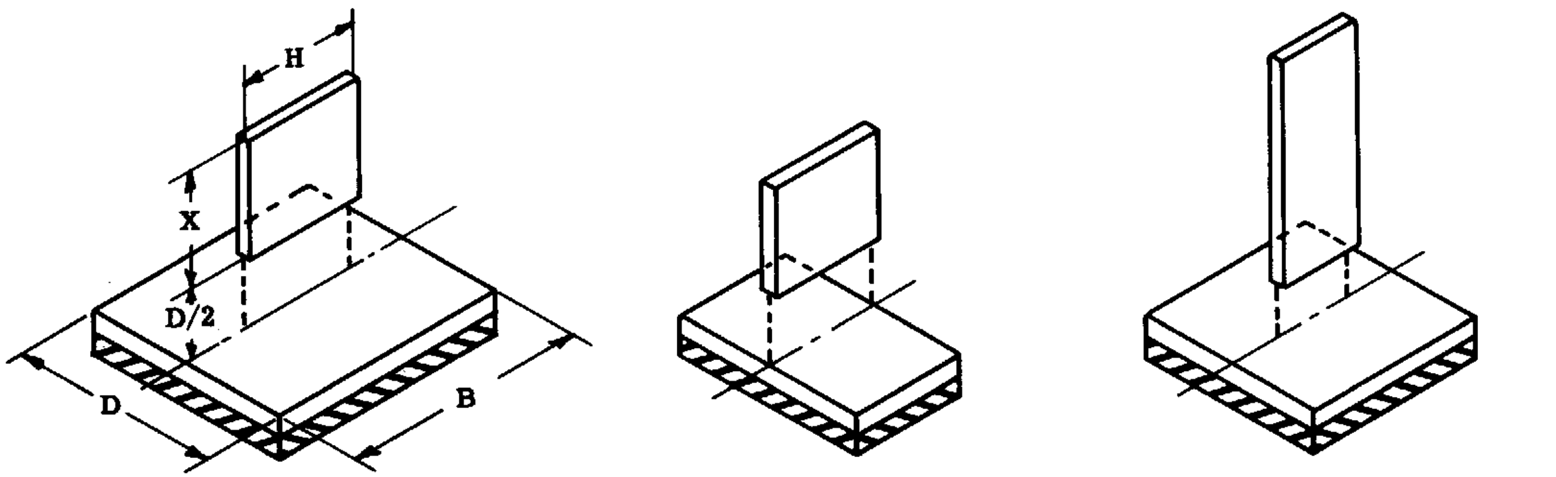
At this point, it becomes necessary to clarify the value of α/ϵ which has been used for this study. The α/ϵ ratio, or the ratio of absorbtivity of high frequency solar energy to emissivity of low frequency earth level radiation, contributes greatly to the radiator temperature and, therefore, correspondingly to the refrigerative capability of such a radiator.

A low α/ϵ value indicates the ability to radiate much more low level energy than is absorbed from a high level energy source such as the sun. This can be better understood by examining the relationship for the ability of a perfect radiator to discard heat, which has been used previously in this report.



RADIATOR CONFIGURATIONS

Figure 37



D = 3H = B
X = H

H = B = X
D = 3H

D = 3H = B'
X = 3H

FIN SIZE TO SIMULATED VEHICLE RATIOS CONSIDERED

Figure 38

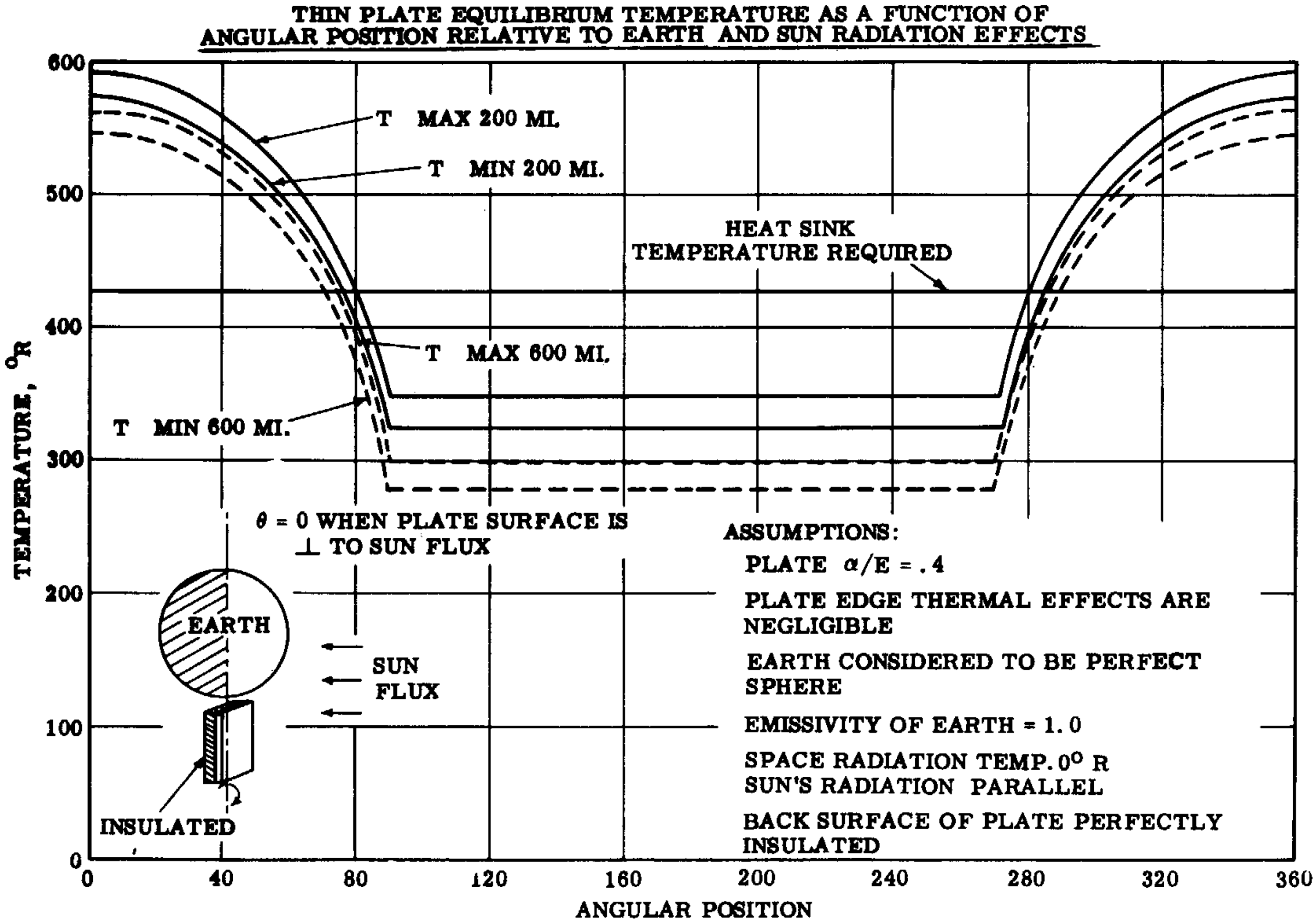
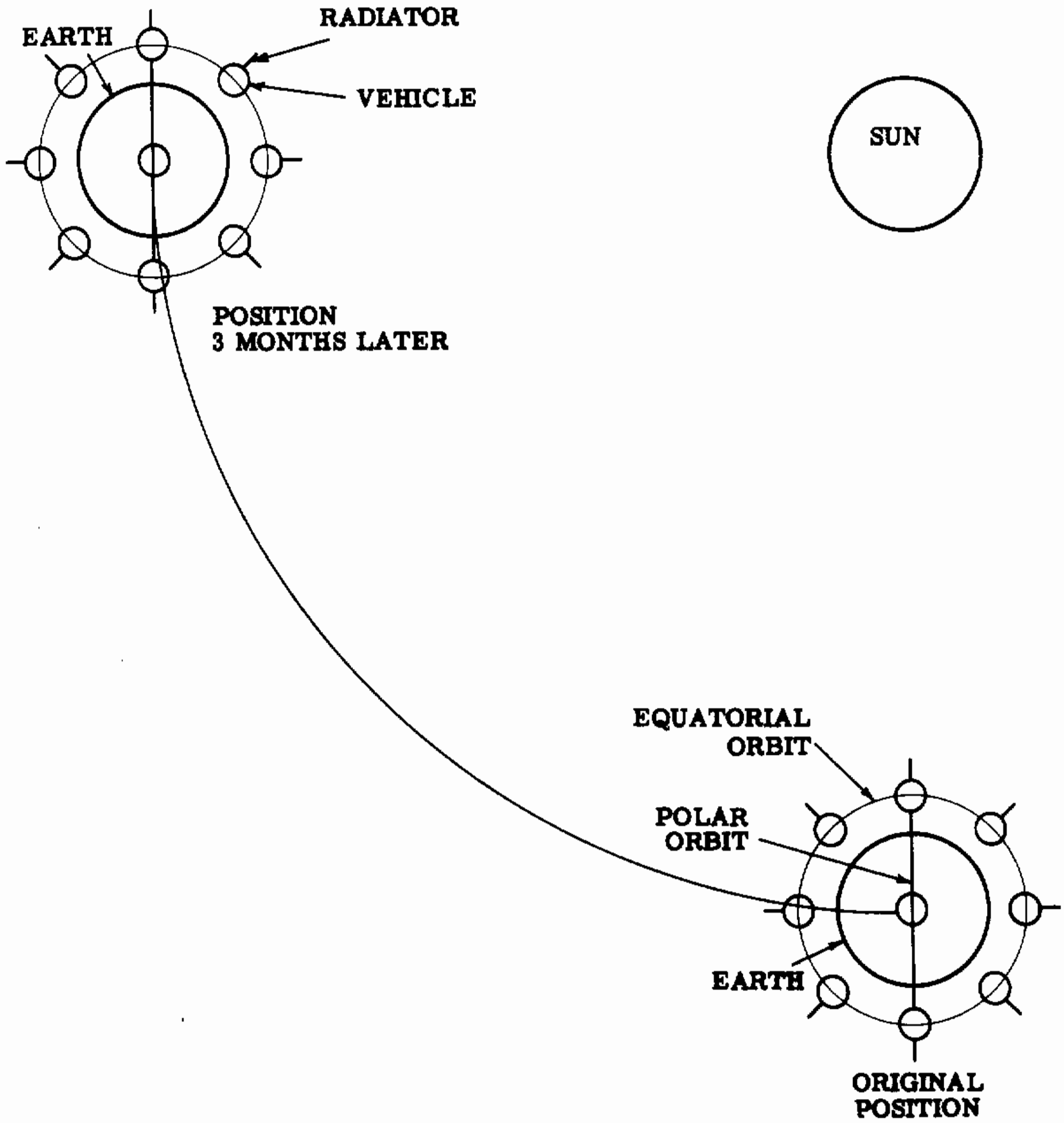

THIN PLATE EQUILIBRIUM TEMPERATURE AS A FUNCTION OF
ANGULAR POSITION RELATIVE TO EARTH AND SUN RADIATION EFFECTS
T MAX 200 MI.
T MIN 200 MI.
HEAT SINK
TEMPERATURE REQUIRED
T MAX 600 MI.
T MIN 600 MI.
θ = 0 WHEN PLATE SURFACE IS
⊥ TO SUN FLUX
EARTH
SUN
FLUX
INSULATED
ASSUMPTIONS:
PLATE α/E = .4
PLATE EDGE THERMAL EFFECTS ARE NEGLIGIBLE
EARTH CONSIDERED TO BE PERFECT SPHERE
EMISSIVITY OF EARTH = 1.0
SPACE RADIATION TEMP. 0° R
SUN'S RADIATION PARALLEL
BACK SURFACE OF PLATE PERFECTLY INSULATED
TEMPERATURE, °R
600
500
400
300
200
100
0
0
40
80
120
160
200
240
280
320
360
ANGULAR POSITION

Figure 39

$$q = \sigma \epsilon A T^4$$

For a perfect radiator the value of ϵ would be unity. For materials which would be available for our use in this kind of application a value for $\epsilon = 0.9$ could be expected. Thus, for a radiator at a particular temperature this relationship describes the total heat which could be discarded as a function of the surface area available. If, however, external inputs exist such as solar radiation, earth's radiation, or solar radiation reflected from the surface of the earth, these fluxes subtract from the radiator's ability to discard heat. A complete heat balance equation must be established for this condition. The solar flux input would be the major element to be accounted for, and the amount of this flux which would be accepted by the radiator is a function of the absorptivity α. Thus, with α as small a number as possible and as large a number as possible the net effect of solar flux is diminished. Values of a very low α/ϵ have been found to exist under laboratory conditions. White paint for example has an α/ϵ of approximately 0.15, however, the question arises as to the repeatability and life expectancy of such a low value. That is, can such a low value be consistantly expected, and more important, for how long a period of time could such a low value be maintained as a result of exposure to the space environment. It is known that micrometeorite bombardment, nuclear radiation, and high temperatures resulting from launch conditions or initial passage through the earth's atmosphere, could have deteriorating effects upon a low α/ϵ ratio. It, therefore, appeared that the best all around value which could be expected to be consistently obtained and readily maintained was a value of 0.4, and the effect of a variation of this ratio was included as a part of the study.

The result of this study was a better understanding of the heat sink available, both in the vicinity of earth, and in outer space as well. In addition, any orbital situation can be constructed using these curves. Two orbital extremes are illustrated by Figure 40, as well as the effect of the earth's rotation about the sun. The temperature variation of a fin-type radiator attached to such a vehicle is illustrated by Figure 41. It can be seen that a particular polar orbit, shown as the original position, will initially provide temperatures significantly below the required heat sink for the refrigerator of 430°R. However, after three months time, this particular orbit will result in a position wherein one radiator surface is held perpendicular to the sun's rays at all times. For the equatorial orbit, the radiator will pass through various positions with respect to the sun's rays for each orbital pass. This motion will result in relatively high temperatures when the radiator is facing the sun, and low temperatures when the radiator and vehicle are shielded from the sun by the earth. This type of orbit is relatively unaffected by the earth's rotation about the sun. It can be seen by these characteristics that it would be undesirable to allow the radiator surface to face the sun, that is, the best heat sink characteristics are available when only the edge of the radiator is presented to the sun. Even if a lower value of α/ϵ were used, such as a value of 0.2, a considerable amount of radiative capability would be lost as a result of permitting the radiator to face the sun during part of the orbit. A relatively simple solution to the problem would be to maintain a constant condition wherein the edge of the radiator faced both sun and earth at all times. Similarly it would be undesirable to place the radiator parallel to the surface of the earth, since the resulting temperature would be excessive.



EXTREME ORBITS FIXED RADIATOR

Figure 40

STEADY STATE TEMPERATURE VARIATION

EQUATORIAL AND POLAR ORBIT

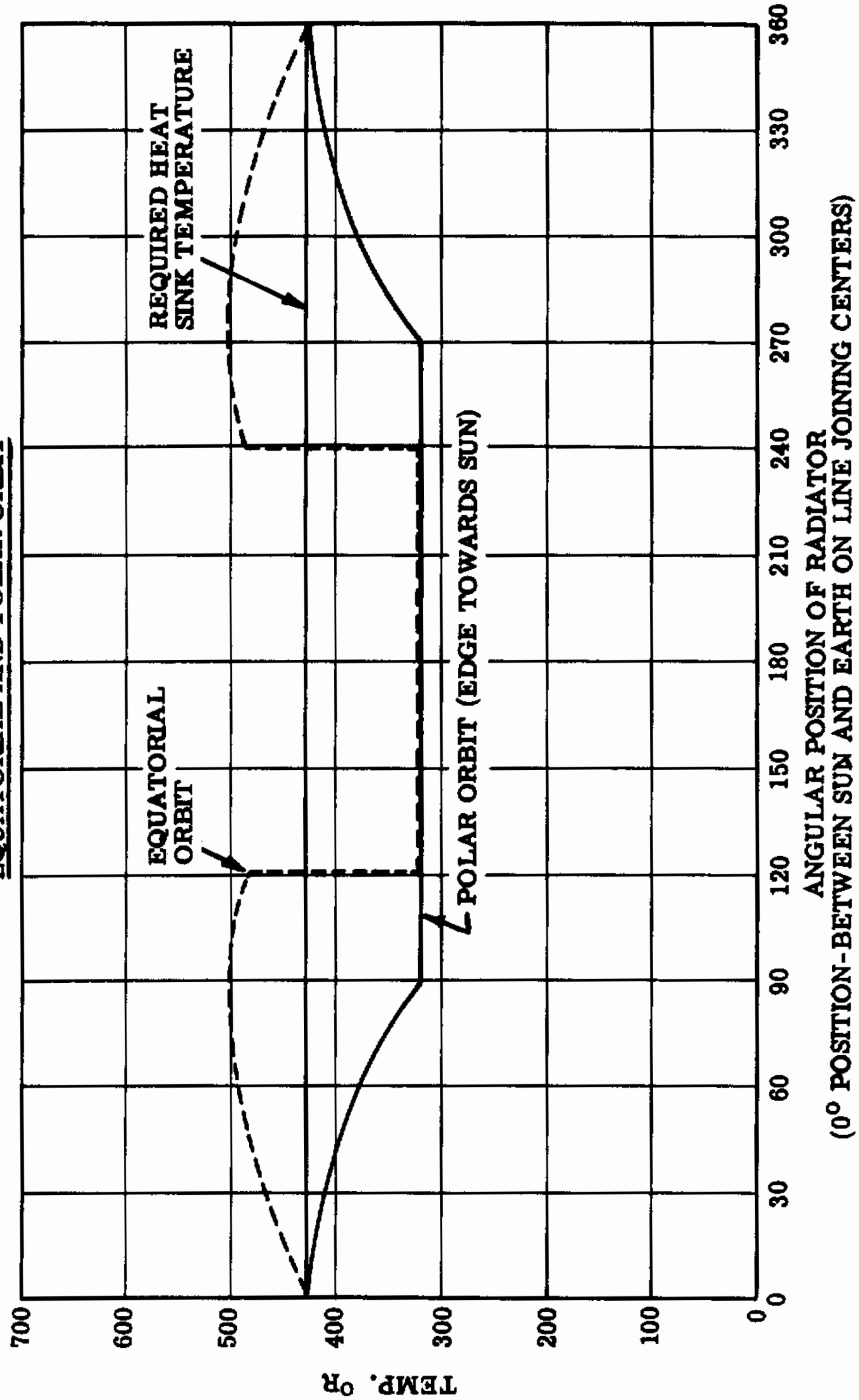


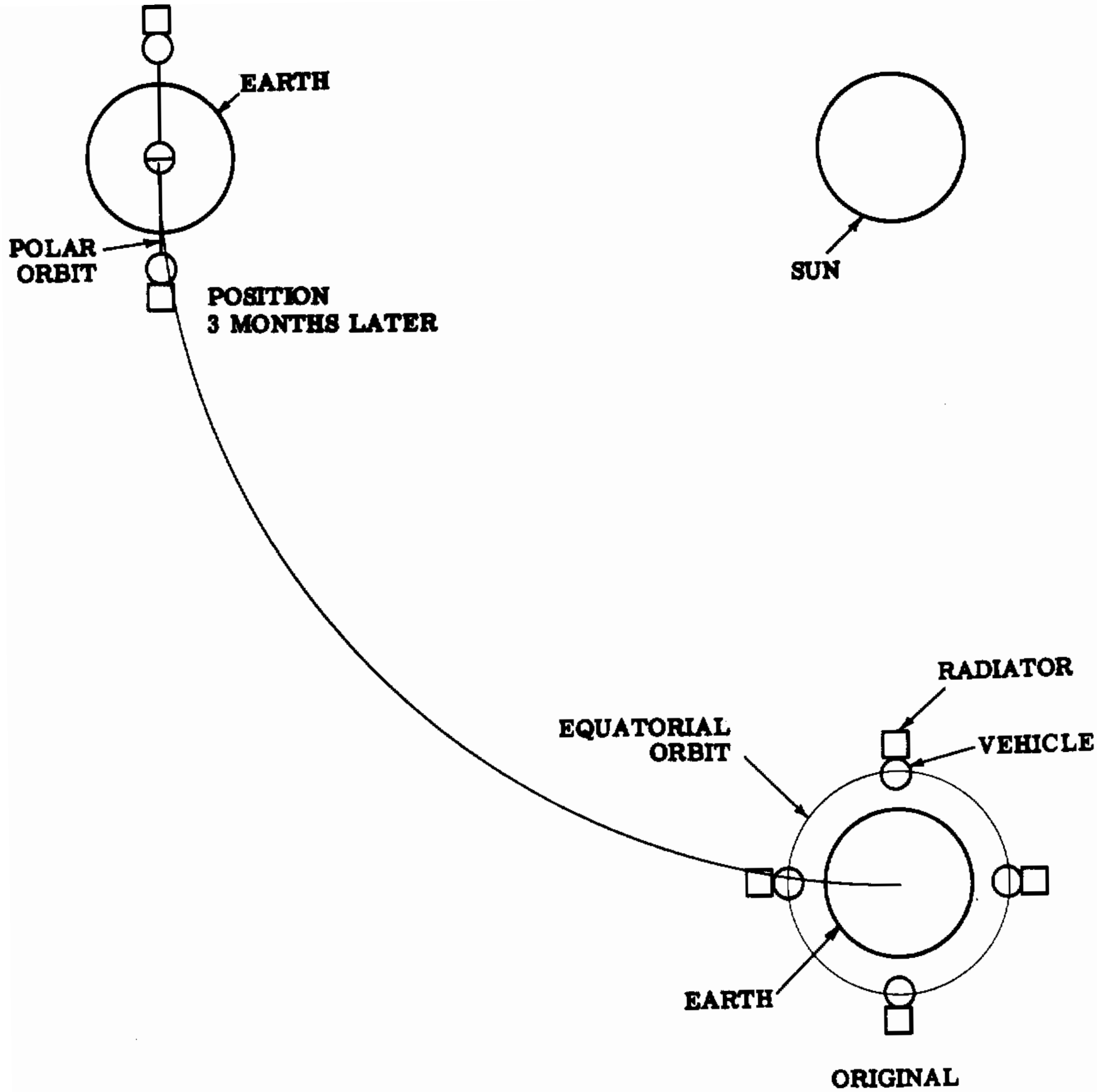
Figure 41

Of course, the need to have the edge of the radiator presented to the earth would only be meaningful for an earth orbiting vehicle. The vehicle out in space would be concerned with only maintaining an edge facing the sun. Figure 42 illustrates the effect of control of either the vehicle or the radiator in relation to the orbits previously discussed.

The question of whether the vehicle or the radiator should be oriented merits some discussion at this point. The moving radiator presents several severe difficulties. Maintaining a good thermal path between two parts moving relative to one another can be a rather nasty problem. That is, the problem of transferring the heat to be rejected out to the radiator becomes complicated by the requirement that the radiator be free to rotate. In addition, moving parts tend to fuze at very low pressures and difficulties are being encountered by other space programs requiring the provision of rotating members. An alternate arrangement of several radiators distributed around the vehicle could be considered. A heat transport fluid pumped to the radiator at the proper temperature would be necessary together with appropriate sensing and valving equipment. This concept appears to add unnecessary complication, and for this system as well as that of the moving radiator, the added weight of the auxiliary equipment must be accounted for on a system comparison basis.

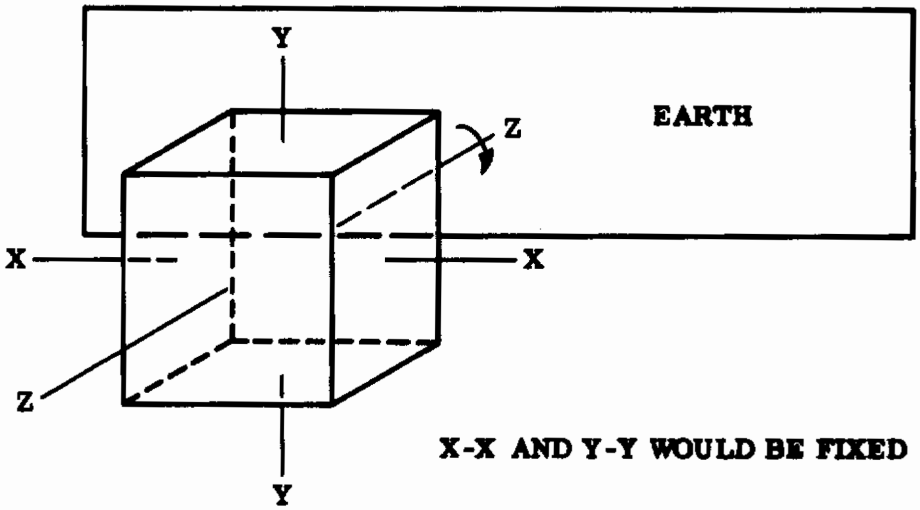
On the other hand, a fixed radiator and an established vehicle orientation is not difficult to conceive of, since solar energy is the primary source of power presently being considered. Sun orientation of some element of the vehicle, such as solar cells or a solar collector, will have already been established. Even for a nuclear power plant, it would be probable that the radiator used to discard the waste heat would be sun oriented. As an example of how our radiator could be located aboard a sun oriented vehicle, we can examine a simple earth orbiting vehicle which is also controlled with respect to the earth. This is illustrated in Figure 43A. For this figure, two axis control, the XX and YY axes, is necessary in order to maintain a vehicle face towards the earth at all times. If this vehicle were to have solar paddles attached, as illustrated in Figure 43B, it would be necessary to establish control of the third axis, the ZZ axis, in order to insure that the plane of the solar paddles was perpendicular to that of the sun's rays at all times. For this particular case it is also necessary to have control of the rotational position of the paddle itself since establishment of a fixed position of the XX plane is insufficient to insure maximum surface area facing the sun's rays at all times. This additional control, however, is not necessary for the radiator as can be seen in the illustration. Merely placing the radiator in the Z-Y plane is sufficient to insure that an edge will be facing the sun regardless of the sun's position. Thus, the vehicle can freely rotate about the earth with the Z axis maintained as an extension of an earth radii, and controlled in order to maintain the Z-Y plane parallel to the sun's rays. For this concept, only one radiator is necessary, and no auxiliary control equipment penalty can be charged against the refrigeration system.

Next the capability of such a radiator to discard heat can be established under several different conditions. The first condition to be investigated is the radiative capability in the vicinity of the earth. The extreme temperature distribution characteristic of the polar orbit condition of Figure 41 will be used assuming controlled orientation. From this figure, the average radiator temperature resulting from external thermal inputs only can be determined to be $355^{\circ}R$. This temperature could be raised to a value between $400^{\circ}R$ and $430^{\circ}R$, in order to permit dissipation



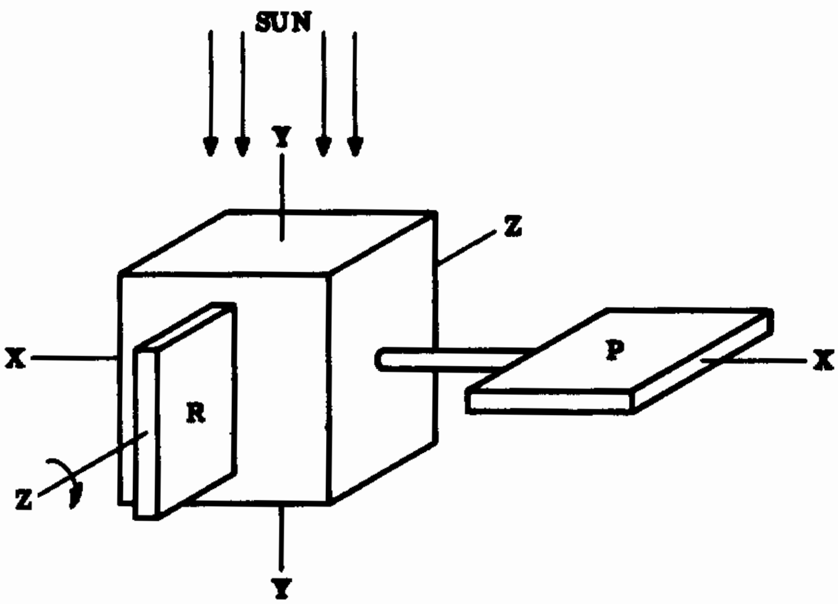
EFFECT OF RADIATOR ROTATION ON ORBITAL POSITION

Figure 42



Two Axis Controlled Vehicle

Figure 43 a



Third Axis Control Required for Solar Orientation

Figure 43 b

FIXED RADIATOR AND SOLAR ORIENTED VEHICLE

Figure 43

of an internal thermal load (the refrigeration effect). By using the relationships developed in Appendix VII for the fin and vehicle arrangement under the conditions established, and permitting a radiator temperature of 400°R, it can be determined that 17.4 BTU/hr. of internally produced heat can be discarded per square foot of radiator surface area. If the thermal interface problems between the radiator and refrigerator are found to be not quite as severe, and a radiator temperature closer to that of the required refrigeration temperature can be utilized, the radiator capability would be increased correspondingly. For example, a radiator temperature of 430°R would provide the capability of radiating 32.75 BTU/sq. ft. of surface area.

If a different vehicle situation is considered, that of a vehicle distant from the effects of the earth, and again either shielded from or parallel to the sun's rays, the only factor which would affect the radiative capability would be the vehicle effects. Under such a situation, and with a radiator temperature at 400°R, it can be found that the radiative capability is 37.3 BTU/hr. per square foot of surface area. If the vehicle effect is neglected, a value of 39.5 BTU/hr. per square foot results.

A final value which can be presented for the sake of comparison, is the maximum radiative capability of a radiator in space at the corresponding temperature (a perfect radiator). The value for such a radiator at 400°R is 43.8 BTU/hr. per square foot. Figure 44 is a plot of radiative capability as a function of radiator temperature for a perfect radiation. A table summarizing the radiator capability under the various situations described is presented as Table 12, for radiator temperatures of 400°R and 430°R. Since the radiative capability varies from 17.4 BTU/hr. at 400°R in the vicinity of the earth, to a value of higher than 50 BTU/hr. in space at a temperature of 430°R, a value on the conservative side of approximately 25 BTU/hr. will be selected for general use as radiative capability. This value is representative of the worst situation, that is, earth orbit, at a temperature relatively close to that of the refrigerator. It is doubtful that a low radiator temperature of 400°R would be required. At this point, it can be seen that a universal refrigeration system, or rather a universal radiator design, would be undesirable. If the radiator were designed for a lower value such as occurs in a situation of earth orbit, this would be an inefficient arrangement for use in a vehicle primarily intended for interplanetary or distant space use.

One other situation will be briefly discussed, and that is the radiative capability of a vehicle orbiting the moon. The moon's temperature extremes have been estimated at 213°F on the hot side and -224°F, on the cold side, resulting in an average temperature of -15°F, a value lower than that of the earth. In addition, the albedo factor is estimated at 0.125, again a value considerably lower than that of the earth. Since the moon is a body smaller than that of the earth, the only condition under which the moon's effect would be more severe than that of the earth's, would be a situation where the vehicle were to operate at a low altitude, or were to hover over the hot side for a considerable period of time. Even for these situations, the system could be shut down, and the thermal inertia of the refrigeration system, as described in a previous section, would be sufficient to insure that the temperature of the refrigerator did not increase severely for reasonable exposure periods.

So it can be seen, that for a mission such as a circum-lunar mission, various conditions will be encountered, i.e., the effect of the earth while in close proximity

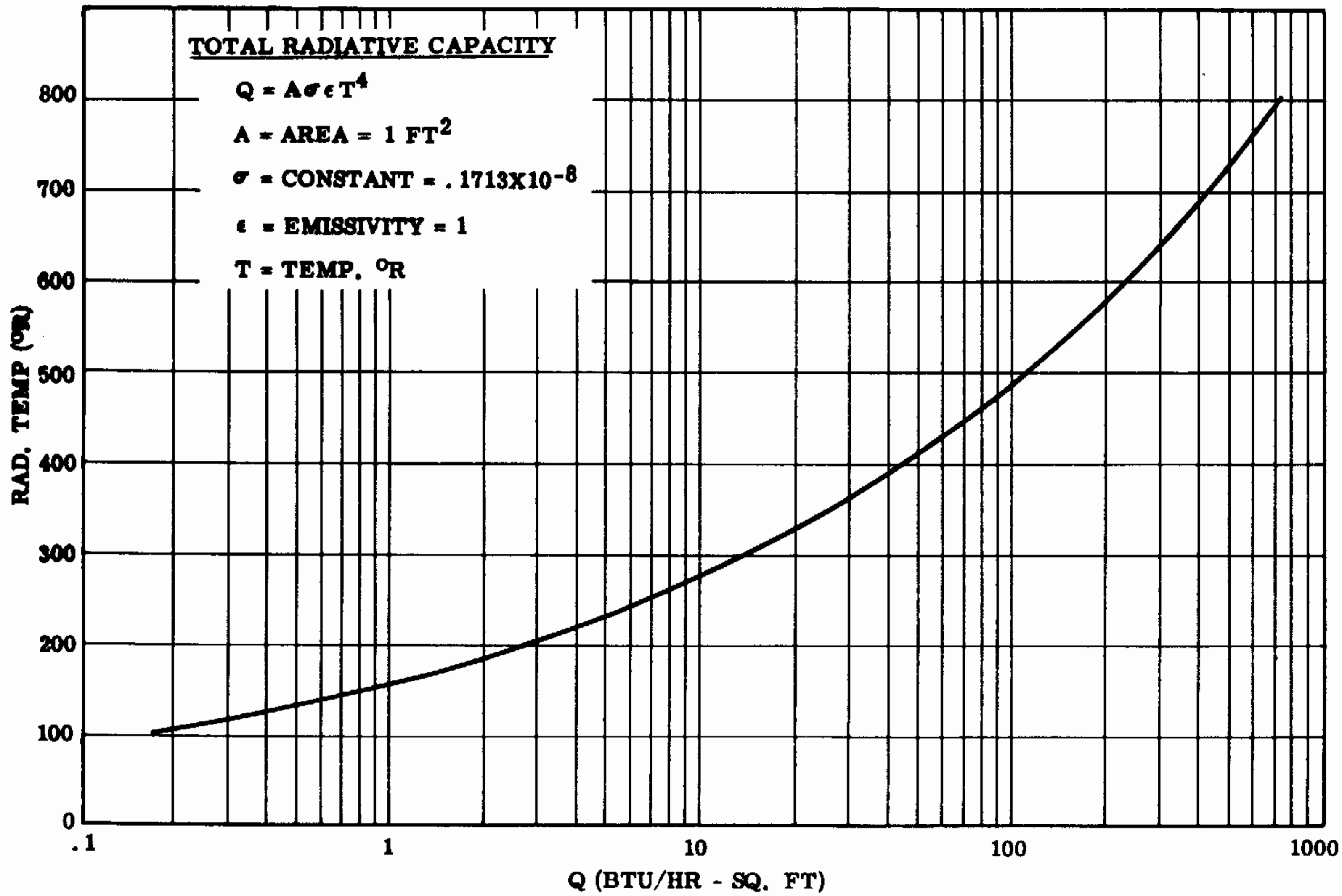

TOTAL RADIATIVE CAPACITY
$Q = A \sigma \epsilon T^4$
A = AREA = 1 FT^2
σ = CONSTANT = $.1713 \times 10^{-8}$
ϵ = EMISSIVITY = 1
T = TEMP. °R
RAD. TEMP (°R)
0
100
200
300
400
500
600
700
800
.1
1
10
100
1000
Q (BTU/HR - SQ. FT)

Figure 44

Table 12. Space Radiator, Heat Rejection Capability under Various Conditions

DIRECT RADIATION HEAT REJECTION CAPABILITY

on the basis of a square foot of radiator surface				
Rad. Temp.	Rad. Situation		Max. Rad. in Space ($\epsilon = 0.9$)	Per. Rad. $\epsilon = 1.0$
	Earth Orbit	Space		
400°R	17.4	37.3*	39.5	43.8
430°R	32.75	51.0*	52.8	58.6

*includes vehicle effect

to the earth, space conditions while between the earth and the moon, and conditions in the vicinity of the moon during that period of the trip. It has been demonstrated, however, that a direct radiation system would be entirely suitable for such a mission, and the only detailed problems involved would be a closer examination of the exact radiative capability to be used for design purposes.

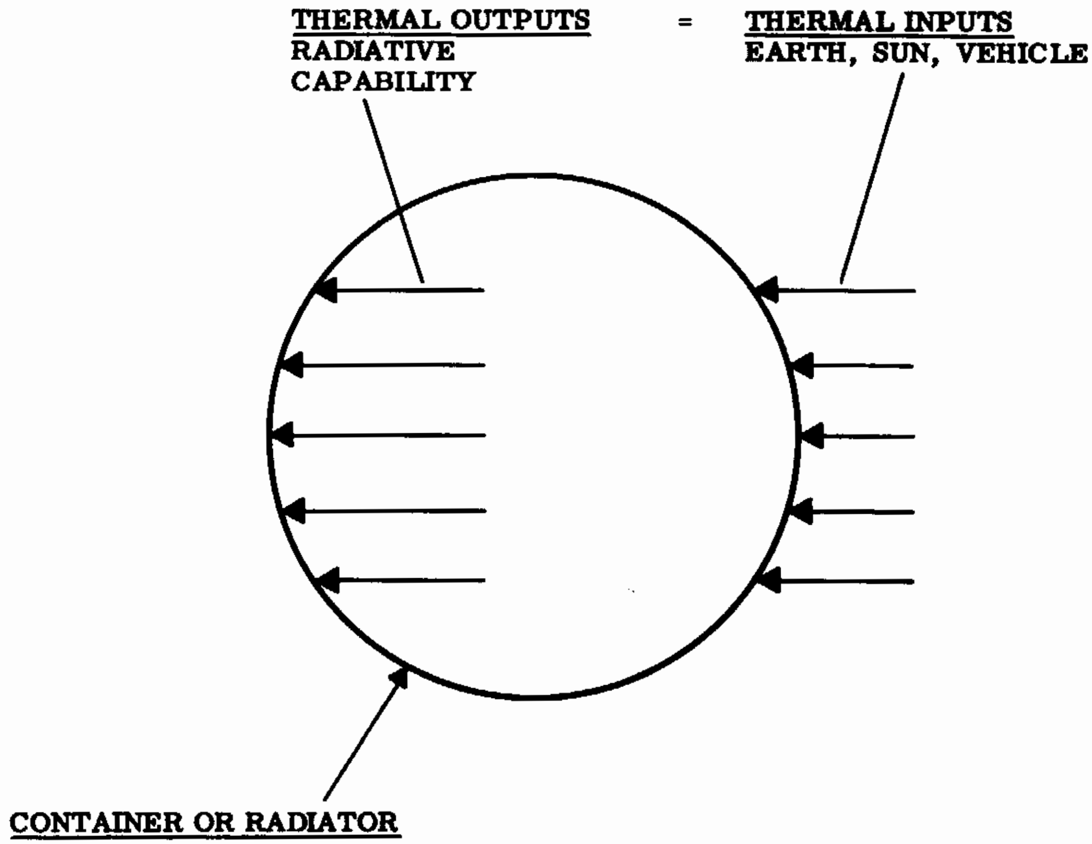
2. Refrigerator Control

As a result of previous discussions, it can be seen that varying factors will exist for the space radiator. Surface characteristics of the radiator may change. The heat input of the refrigerator will probably vary. External effects upon the radiator will also vary. It, therefore, does not appear feasible for a strictly passive radiation system, Figure 45, to be utilized by vehicles in the near future, and it is similarly apparent that some sort of a control system would be required. The simplest form of such a control system is illustrated by Figure 46, and is merely a means of providing a thermal link between the refrigerator and the radiator. With the pump inactive, the heat transport fluid does not circulate between the refrigerator and radiator and only a very small quantity of heat is transferred by conduction through the heat transport fluid to the radiator. The refrigerator temperature will increase until a predetermined value is reached and the thermostat will then actuate the pump and circulate the heat transport fluid. This then actively transports the heat from the refrigerator to the radiator where it can be rejected to space. A heat transport fluid which remains liquid at a rather low temperature is desired. Freon 21, which is liquid between the temperatures of 48°F and -197°F at atmospheric pressure, appears best suited. By raising the pressure slightly, an increase of boiling temperature can be readily achieved. The presence of the vehicle, as an external effect, and some small heat flow through the insulation used to separate the radiator from the vehicle will assure that the radiator temperature never drops below the freezing point of the heat transport fluid.

Thus, such a system appears to be highly desirable and can be accomplished with very little additional weight penalty to the system. For example, in the case of the 16 cubic foot refrigerator previously considered, with a heat load of 250 BTU per hour to be removed from the refrigerator, only nine gallons per hour would have to be circulated through the heat transport system, if a fluid such as Freon 21 were used. This would require only a very small pump and very low power.

3. Radiator Design and Specific Weight Determination

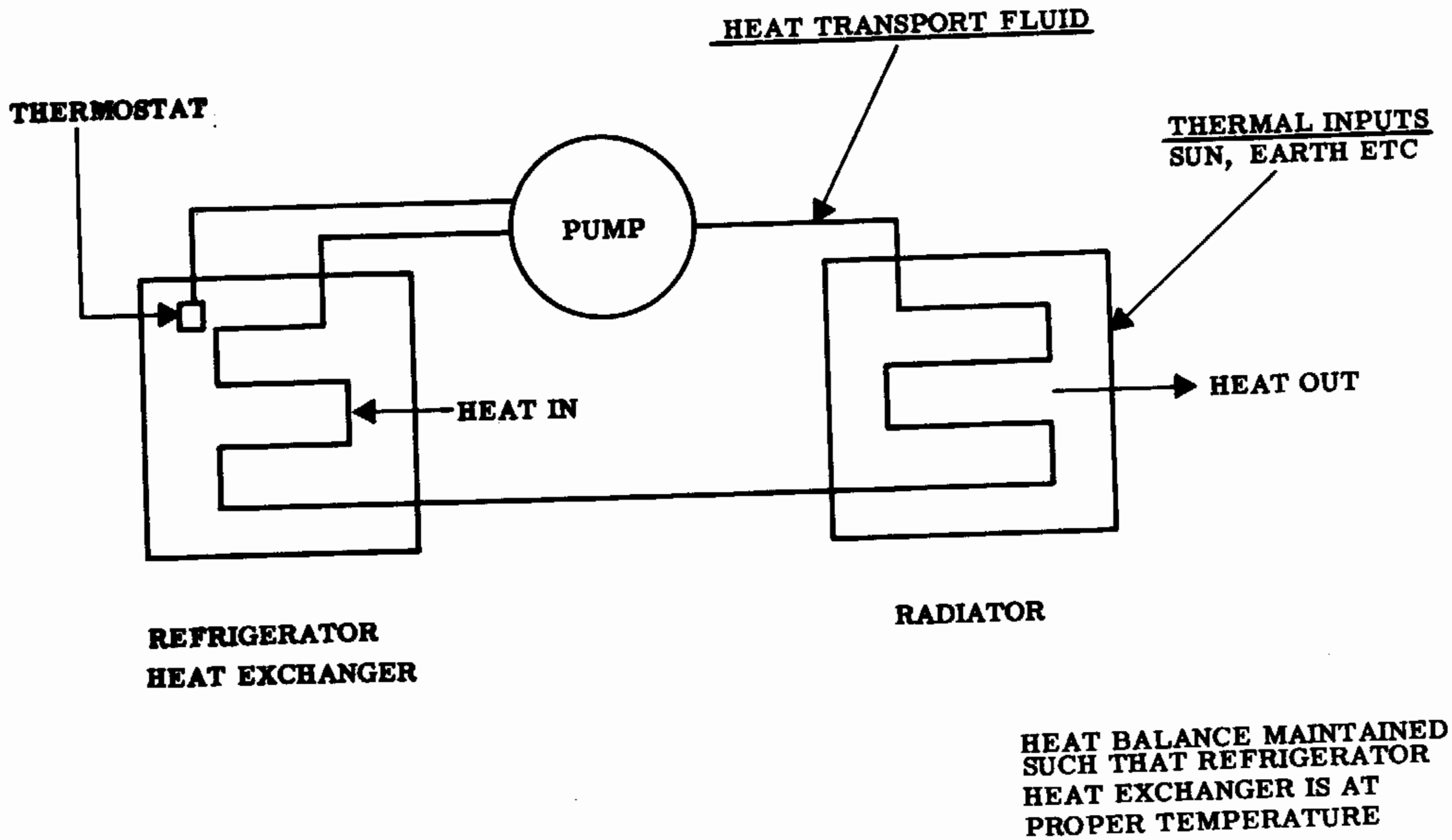
In order to formulate a concept for the specific design of the radiator, numerous references were consulted. A large body of material exists concerning optimum radiator design, and such problems as meteorite bombardment and minimization of weight are considered. Two representative designs of fin type construction are illustrated as Figures 47a and 47b. In the case of Figure 47a, a sufficiently thick skin has been considered in order to insure adequate protection against meteorite penetration. The design concept for Figure 47b, is the utilization of meteorite barriers which provide a minimum weight design. The details of the weight estimate made for both designs follow:



HEAT BALANCE MAINTAINED BY CONSTANT INPUTS AND SURFACE PROPERTIES

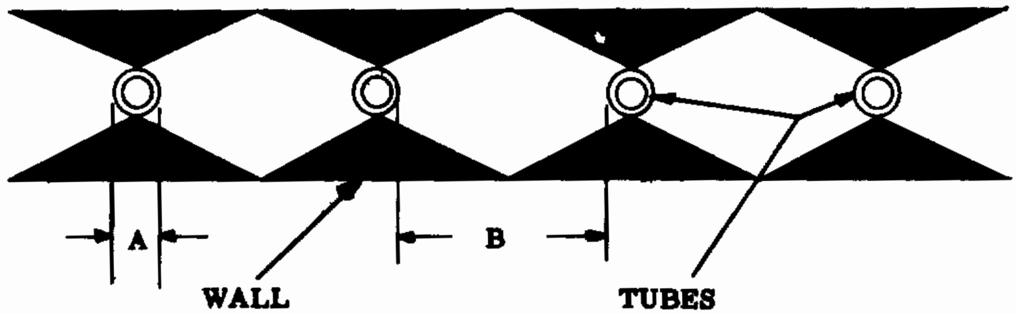
A PASSIVE DIRECT RADIATION SYSTEM

Figure 45



DIRECT RADIATION ACTIVELY CONTROLLED

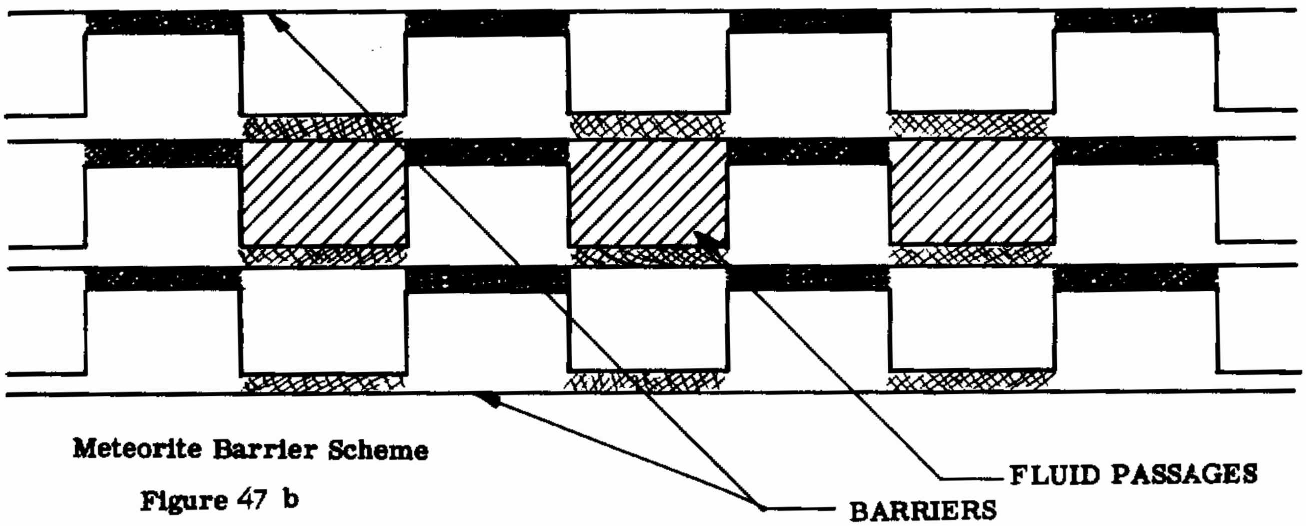
Figure 46



DIMENSION A = 1/2B

Solid Walled

Figure 47 a



Meteorite Barrier Scheme

Figure 47 b

SPACE RADIATOR DESIGN CONCEPTS

Figure 47

Thick Skin Design (See Figure 47a)

As a result of reviewing various papers, it appears that a radiator skin thickness, (considering aluminum), of 0.125 inch is required to insure an adequately low probability of meteorite penetration for a five sq. ft. radiator surface. This is the size which would be required for the three-man, 30-day mission using the values previously established of;

$\frac{250 \text{ BTU/hr}}{25 \text{ BTU/hr/sq.ft.}}$ = 10 sq. ft., or for a fin type radiator,

the surface area of each side would be five sq. ft. (a thicker wall would be required for larger radiators and longer mission durations).

Using a wall thickness of 0.125 inch as the maximum necessary in the vicinity of the tubes, an average wall thickness of 2/3 x 0.125 = 0.0835" can be established. Using a density for aluminum of 0.1 lb/cu. in. and accounting for both walls;

wall weight (1 sq. ft. of rad.) = 0.0835 x 2 x 144 x 0.1 = 2.40 lb/sq. ft. The weight of aluminum tubes is;

π x thickness x dia. x length x no. of tubes/in. x 12 x density. Assuming a tube diameter of 0.06 inch and a wall thickness of 0.01 inch;

wt. of tubes = 3.14 x 0.01 x 0.06 x 12 x $\frac{16}{3}$ x 12 x 0.1 = 0.14 lb/sq. ft. The weight of the liquid in the tubes (assuming specific gravity of water), is ;

Cross section area of tube x length x no. of tubes/in. x density.

Wt. of liquid = 0.003 x 12 x $\frac{16}{3}$ x 12 x 0.0361 = 0.088. Thus the total weight is 2.40
0.14
0.08
2.62 lb/sq. ft. for the entire radiator of considering both sides a value of 1.31 lb/sq. ft. of radiator surface area can be used.

Radiation Barrier Design (See Figure 47b)

For this concept, seven sheets of 0.003 stainless steel would be required, thus the volume of these sheets (considering one sq. ft.) would be;

4 x 0.003 x 12 x 12 = 1.73
3 x 0.003 x 24 x 12 = 2.59
4.32 cu. in.

Using a value of 0.293 lb/cu. in. for density, the weight = 4.32 x 0.293 = 1.27 lb/sq. ft. If a nominal amount is added for liquid, say 0.25 lb/sq. ft. the total weight is about 1.50 lb/sq. ft. for the entire radiator or a value of 0.75 lb/sq. ft. of surface area can be used.

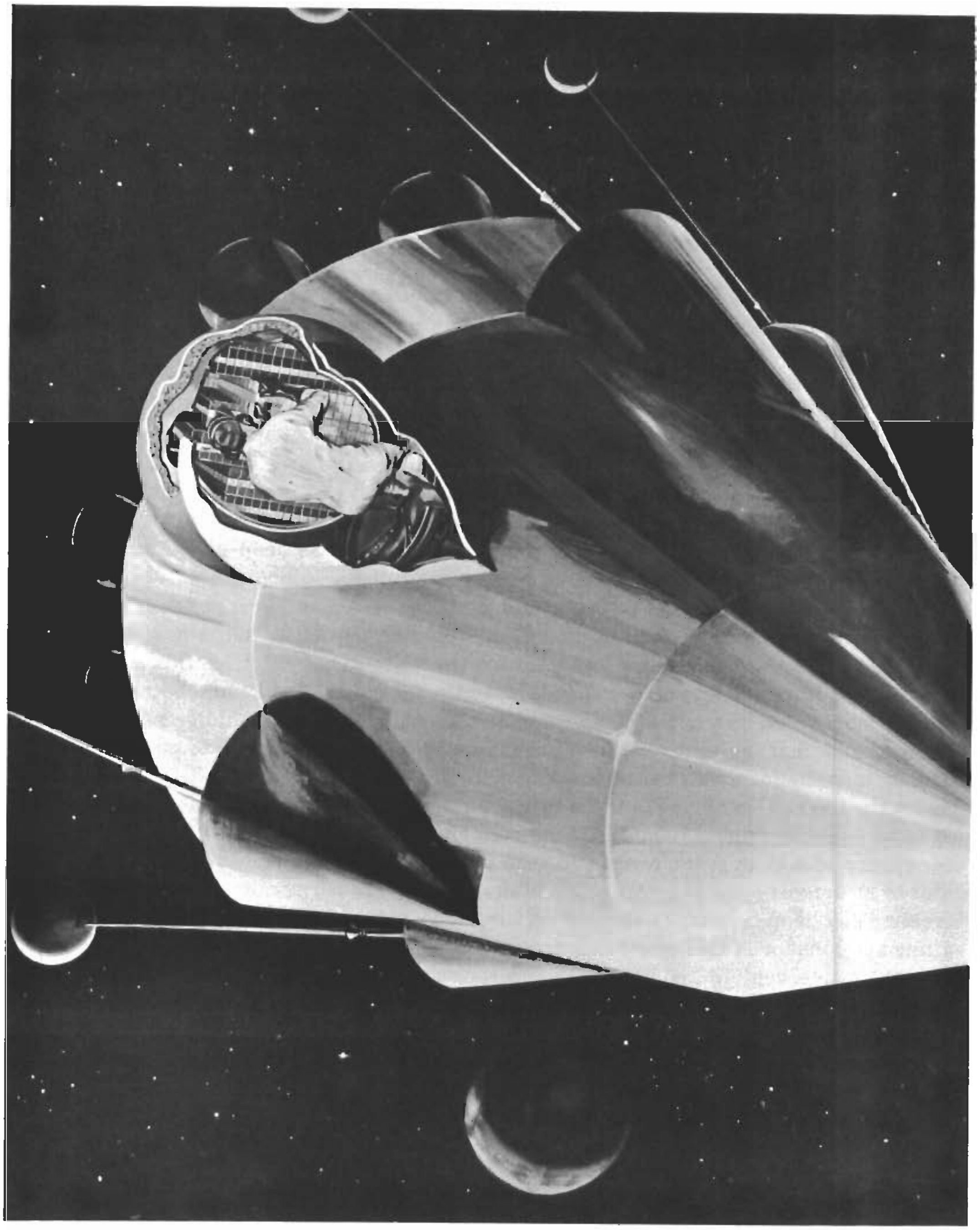
Thus, an intermediate value of one pound per square foot is a representative value to use as a specific weight of a radiator surface. This is also a value commonly used by numerous authors. In addition, this value appears to be a reasonable one to use for a radiator which would consist of the external wall of the vehicle, since it will be necessary to add to this wall, tubing to carry the heat transport fluid. It might be possible to achieve a radiator weight saving by such an arrangement, but obviously twice as much surface area would be required, since only one side of this radiator is exposed to space.

4. Applications

As a result of the general discussions previously presented, it can be seen that a system of direct radiation, modified to the extent that some control is maintained, appears suitable and, in fact, desirable for use aboard space vehicles. This system appears to be the best all around system available, and would be limited only by the maximum size of the radiator which could be utilized. This system would require a smaller radiator than that of any other system, since all systems with the exception of the expendable refrigerant will require a radiator. The radiator limitations imposed upon this system are imposed upon all systems. For futuristic applications, several other concepts are available which could be incorporated into a direct radiation system. For very long missions such as deep-space probes, it is conceivable that it would not be necessary to limit the temperature requirements of the food as stringently as those provided. That is to say, it is possible that the only requirement which would be imposed, would be that of maintaining a temperature below 0°F, and the lower temperature limit would not exist. In this case, active control is not as critical as previously described, and the food could be placed in a container which was oriented perhaps on the shady side of a vehicle and insulated from it such that the temperature of the compartment was reduced to perhaps -200°F. This compartment would represent a long-term storage compartment, and periodically food would be removed from this area and stored in a local galley refrigerator at higher temperatures, such as those considered for this report. It would not be necessary to add as much heat to food in the vicinity of 0°F as it would to heat food stored at the very low temperature. Thus, a three-zoned arrangement (chill, freeze, and deep freeze) is envisioned for multi-manned long range missions. An alternate concept is that of storing the food in a sphere which would be trailed behind the space vehicle and exposed to an essentially steady state condition of solar flux. Relatively constant surface characteristics of the material of this sphere and a constant solar input would assure a relatively constant temperature of the food compartment. An artist's representation of this arrangement is illustrated as Figure 48. This concept would require transfer to the space vehicle proper on a periodic basis. The men's immediate food needs would again be handled by an internal refrigerator.

C. Thermoelectric Refrigeration

A well-known thermoelectric phenomena known as the Seebeck effect has been used for a considerable period of time as a means of measuring temperature. This is the well-known thermocouple, where a circuit consisting of junctions of two dissimilar metals are placed at two different temperature levels causing current to flow in a circuit. The reverse of this arrangement is called the Peltier effect. If a voltage is applied to such a circuit, and current caused to flow, a temperature differential appears between the two

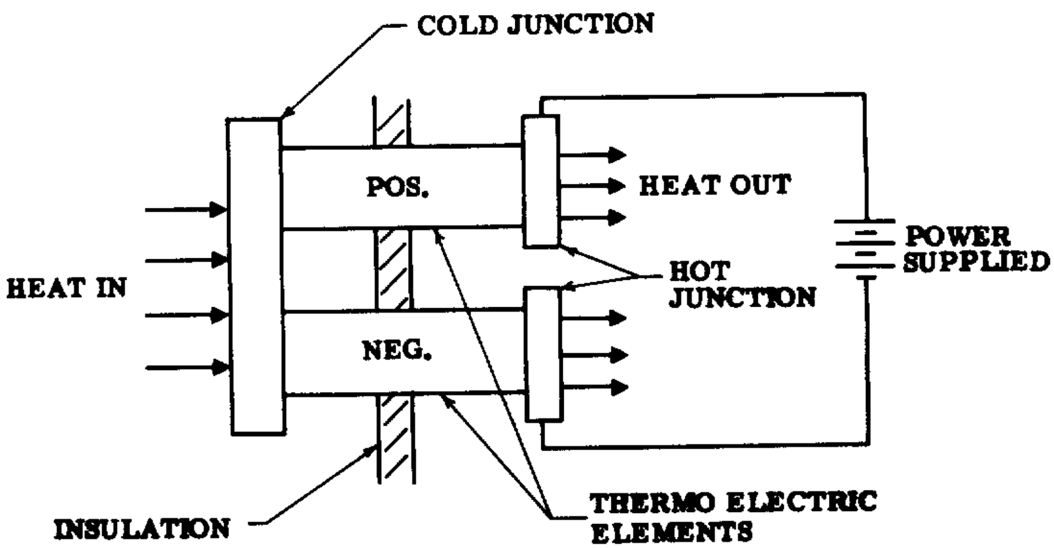


PROPOSED FUTURISTIC APPLICATION FOR PASSIVE RADIATION SYSTEM USING TRAILING SPHERES

Figure 48

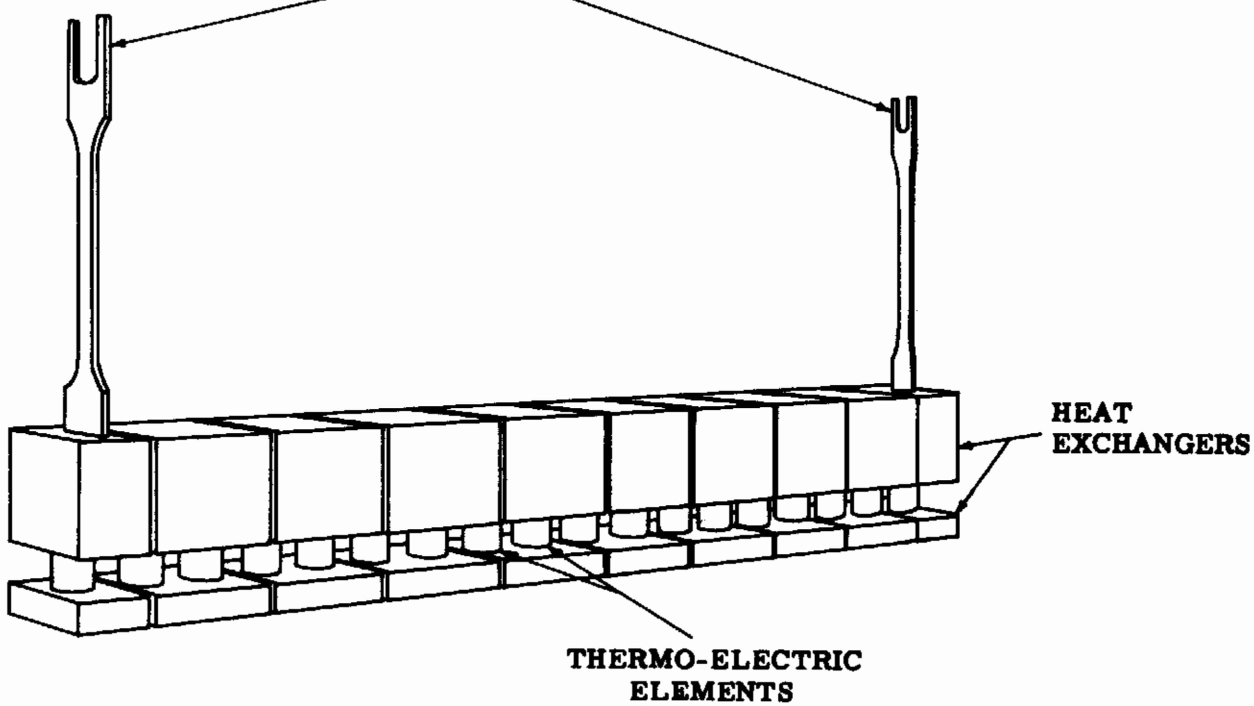
junctions; thus, a heat pumping effect can be produced. The advent of semi-conductor materials provided the opportunity to convert a laboratory curiosity to a practical engineering development. A circuit consisting of a positive semi-conductor element and a negative semi-conductor element as illustrated by Figure 49 is responsible for the current interest in the thermoelectric phenomena. These materials have provided the required combination of physical characteristics to a greater extent than had been previously available. Nevertheless, the over-all efficiency of such an arrangement is still considerably less than the efficiency available from standard systems such as a vapor compression refrigerator. Thermoelectric refrigeration has aroused interest because of certain advantages not obtainable with other systems, such as the following: no moving machinery, no problems from possible contamination by the refrigerant, silent operation, apparently good reliability, and a reasonable level of efficiency for small refrigeration effect. A great quantity of current literature exists on the theoretical aspects of thermoelectric refrigeration, and indeed some commercial applications of this technique are at present under-way. Thermoelectric materials and complete thermoelectric cooling modules are presently available on the commercial market. Since relatively high currents and low voltages are required for the operation of the single couple, common usage requires a number of couples in series as the best design arrangement. This concept is illustrated by Figure 50.

The overall question of the desirability of heat pumping has been discussed in a prior section, and the kind of coefficient of performance that would be required in order for such a system to be advantageous has been demonstrated. The type of performance which could be expected from a thermoelectric heat pump, utilizing present technology, can be explained. Figure 51 illustrates the performance for a thermoelectric cooling module which is being considered for a submarine refrigeration application. From this figure, it can be seen that the maximum coefficient of performance is 0.3 for a corresponding ΔT of $70^{\circ}F$. The heat pumping capacity of such a module is seen to be approximately 270 BTU per hour. Since the module consists of 432 couples, this means that each module is capable of pumping about 0.6 BTU per hour. For our application, it has previously been established that a ΔT of approximately $90^{\circ}F$ is required. For this value, a COP of 0.1 exists, and a heat pumping capacity of 500 BTU's per hour or slightly greater than one BTU per hour for each couple. The best combination of heat pumping capacity and coefficient of performance is obtained at a ΔT of approximately $85^{\circ}F$, and even at this value the COP is still quite low as well as the heat pumping capacity. In order to establish the kind of maximum coefficient of performance which could be expected for the exact conditions of operation of our space vehicular refrigeration system, Figure 52 has been prepared. This figure is characteristic of maximum COP design, representing current thermoelectric capability as a function of the hot junction temperature with cold junction temperature fixed at $-10^{\circ}F$. A figure of merit of 2.5 for the material was used. This is a value typical of material presently available. The literature indicates that future developments should appreciably improve this figure of merit, and it is, therefore, expected that the coefficient of performance illustrated would similarly be improved. It can be seen from this figure that a hot junction temperature of $530^{\circ}R$ ($70^{\circ}F$) would result in a maximum coefficient of performance of 0.32. Therefore, a value of approximately 1/3 for the expected coefficient of performance for this kind of a temperature difference has been used for the purposes of system comparison. That is to say that for every 100 BTU's of refrigeration effect produced, 300 BTU of power must be furnished, and the corresponding weight penalty for the provision of this power must be accounted for.



THERMO-ELECTRIC REFRIGERATION SYSTEM

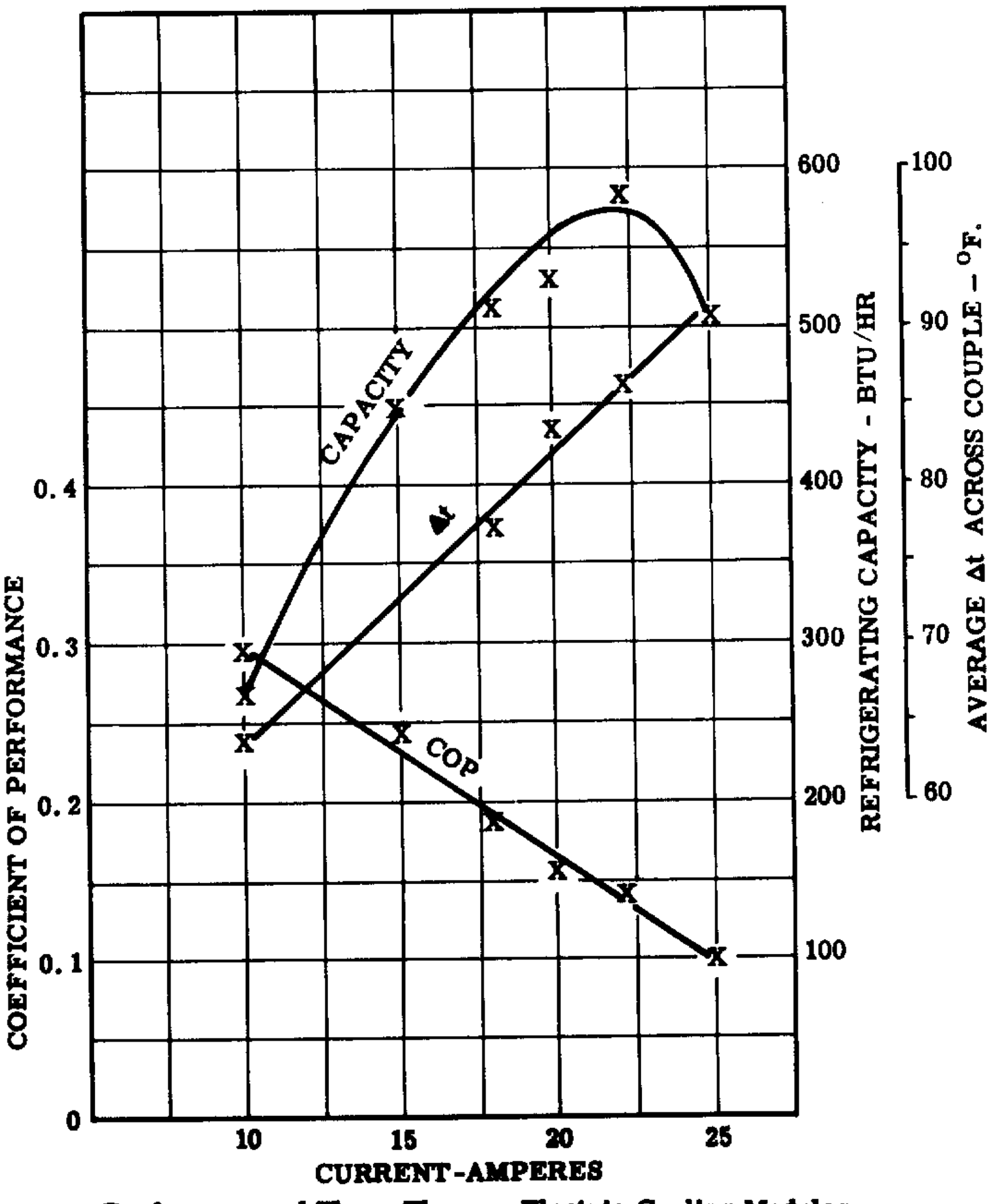
Figure 49



<u>THERMO-ELECTRIC ELEMENT SERIES ARRANGEMENT</u>

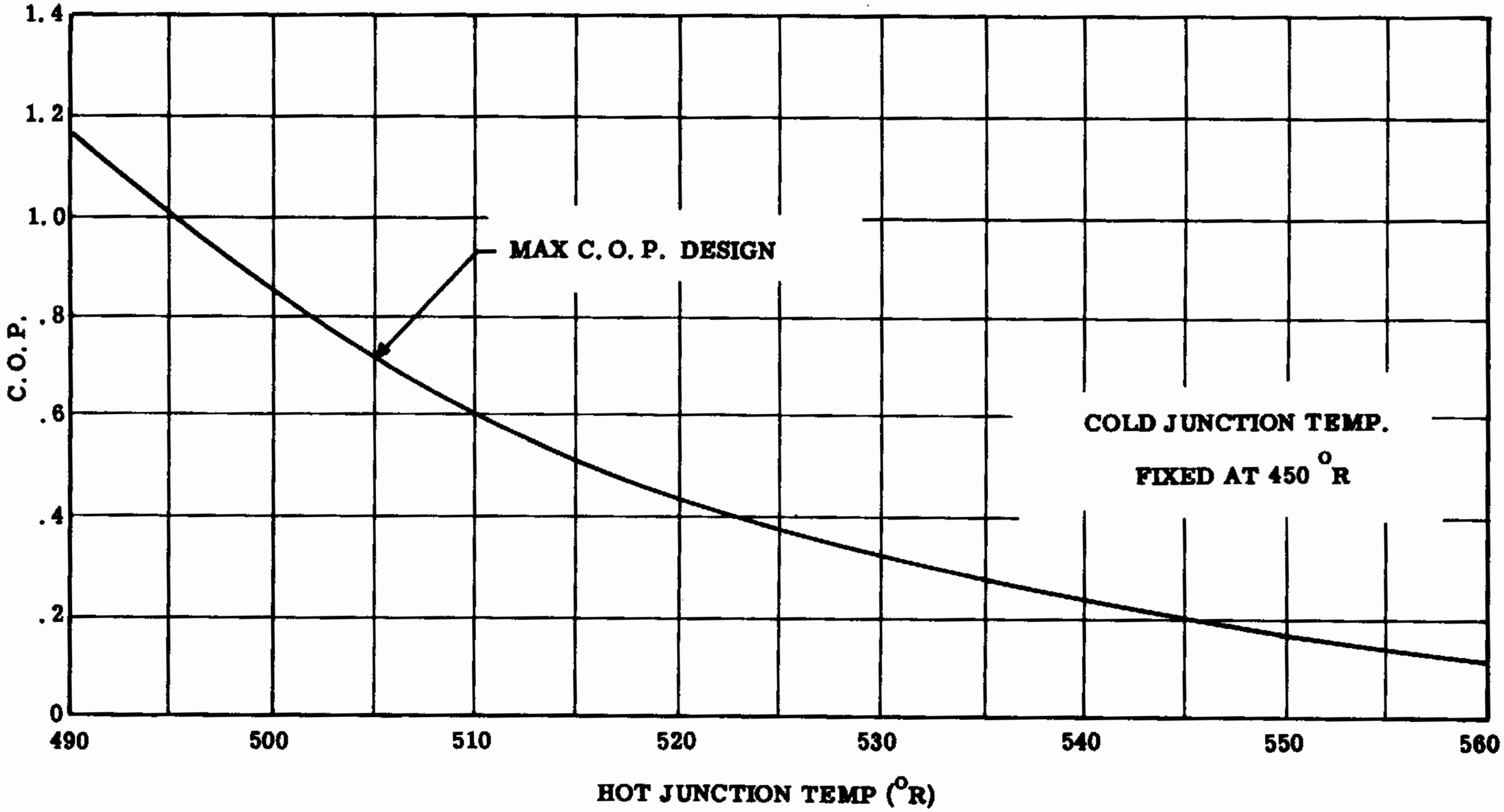
Figure 50

TYPICAL THERMO-ELECTRIC PERFORMANCE



Performance of Three Thermo-Electric Cooling Modules, 432 Couples, 0° F Air Temperature, 55° F Cooling Water

Figure 51

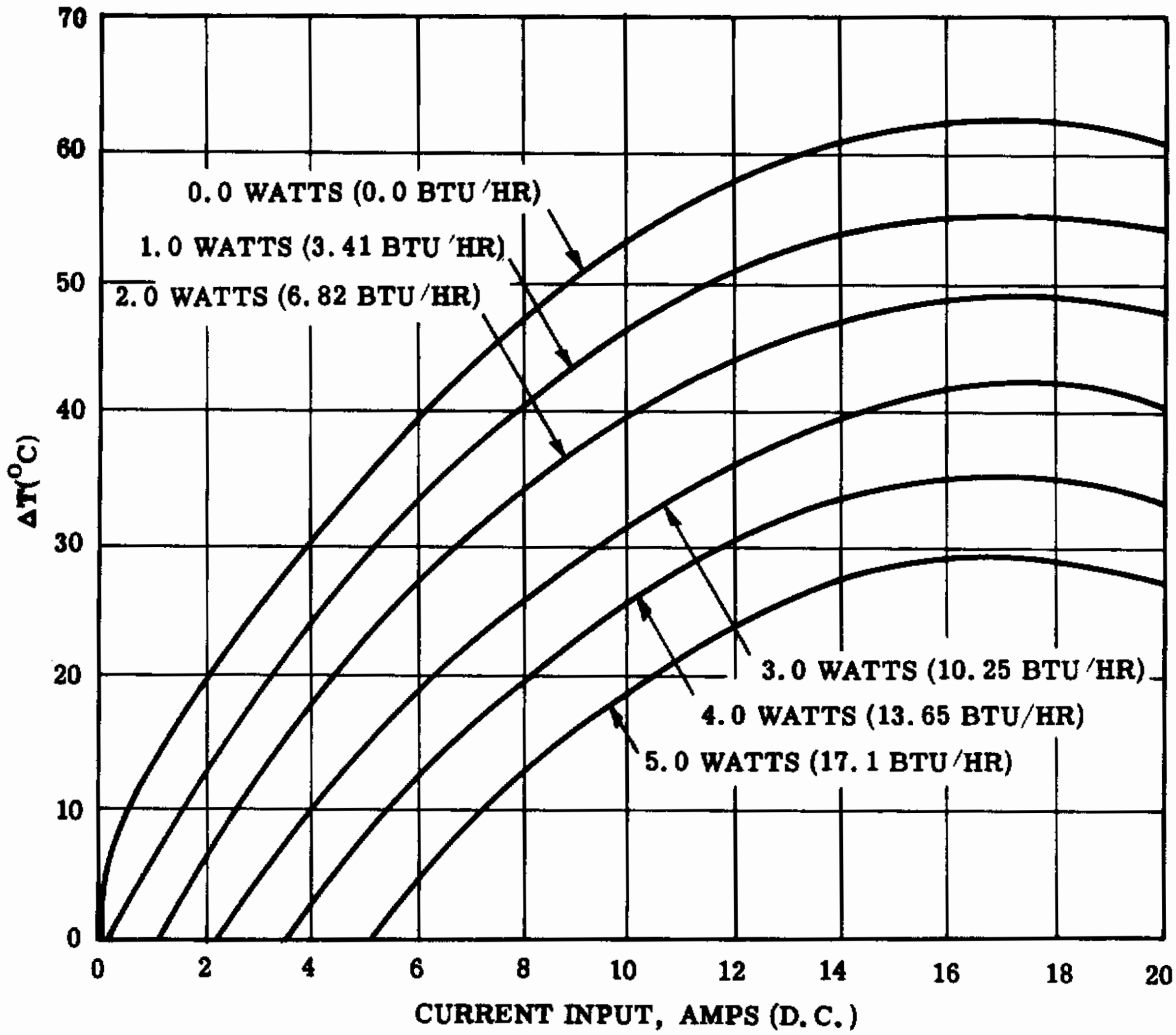


THERMO-ELECTRIC PERFORMANCE (ESTABLISHED CONDITIONS)

Figure 52

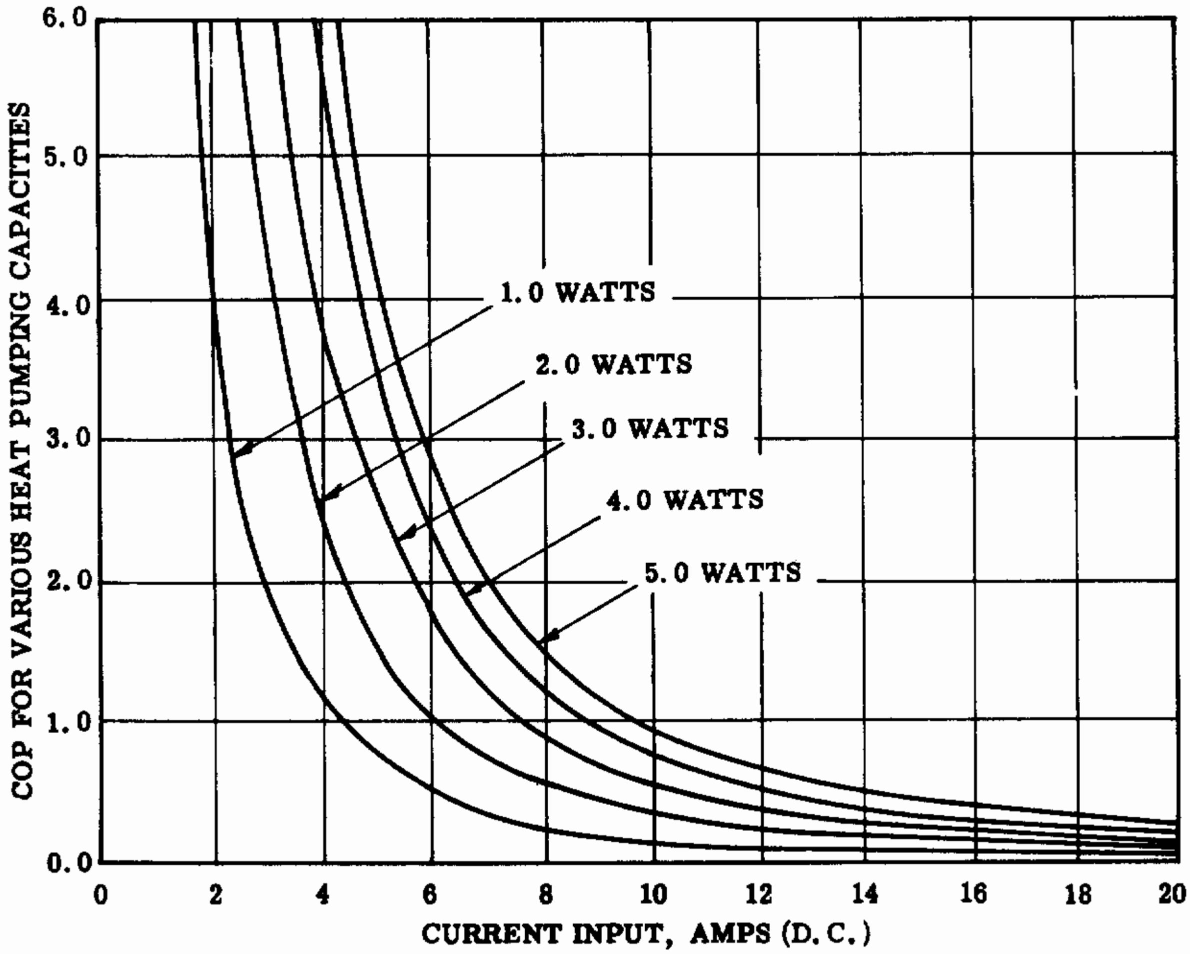
As a further indication of the "state of the art" of thermoelectric materials, Figures 53 and 54 have been included. These curves describe catalog advertised performance for thermoelectric materials available from a particular manufacturer. The coefficient of performance which can be expected for a corresponding temperature difference can be seen, as well as the heat pumping capacity of the couples under varying conditions. An individual couple of this type weighs approximately 2 1/2 ounces, and this is the weight penalty which has been assigned for the thermoelectric elements in the comparison of systems appearing in the next section of this report.

In summary, a thermoelectric refrigeration system has advantages over many other heat pumping systems which have been considered. However, the relatively low efficiency of this concept and the accompanying power and weight penalty produces serious disadvantages. Not only is the power required to operate such a system of concern, but also the fact that this power is converted to heat and must be rejected along with refrigeration effect appreciably increases the problem of heat removal. Increasingly efficient units will tend to counteract this disadvantage and perhaps future thermoelectric systems will be available with a less severe power penalty required. As a final possibility, a thermoelectric refrigeration system could be considered for use in discarding heat to the manned compartment, provided that the heat load from this effect is a relatively small portion of the compartment heat load. In this way, any excess capacity available from the conditioning system of the manned compartment of the vehicle could be utilized. For example, with entire vehicle skin acting as a radiator, it is conceivable that an overall heat balance could be achieved resulting in the required internal compartment temperature. The refrigerator load would then be automatically accounted for and a special refrigerator radiator would not be required. It remains, however, for these possibilities to be put into actual practice, and manned space vehicles will have to be built and tested under operating conditions, before the entire thermal condition of such a vehicle is established. In the interval, it appears reasonable to assume that the overall heat balance of a space vehicle will be such that heat will ultimately need to be rejected to space by the use of a radiator, other than the skin. Our refrigeration radiator could either be a part of this overall radiator or an independent unit. In any case, it seems reasonable to assign a radiator weight penalty to any system which requires the rejection of heat.



TYPICAL THERMAL AND ELECTRICAL CHARACTERISTICS OF COMMERCIALLY AVAILABLE THERMO-ELECTRIC COUPLES ΔT VS CURRENT

Figure 53



TYPICAL THERMAL AND ELECTRICAL CHARACTERISTICS OF COMMERCIALLY AVAILABLE THERMO-ELECTRIC COUPLES COP VS CURRENT

Figure 54

SECTION V

COMPARISON OF SYSTEMS

The three major systems which have been considered have been compared on the basis of system weight. This system weight is defined as a value independent of the weight of the box itself since the basic box will essentially be the same for all three systems. The system weight consists of those auxiliaries and appendages required in order to make the individual systems operate satisfactorily. The actual total weight of the system will, of course, require the addition of the weight of the box. Table 13 is illustrative of the values which have been considered in compiling this system tradeoff characteristic. These system comparisons have been made for the short mission durations only because of the rapid divergence of the curves involved and the obvious desirability of using a direct radiation system for long mission durations.

For the direct radiation system two variations have been evaluated. The first of these is a system consisting of one fixed radiator, which is probably the most likely situation. The elements which have been considered for this system are the weight of the radiator itself, the weight penalty of the heat transport pump and associated piping, and also the weight penalty for the power to operate the pump. A modification of the direct radiation system providing for more than one radiator or the addition of radiator orienting equipment has also been considered. An appropriate weight penalty has been assigned to these additional elements. The auxiliary elements of the ice sublimation system are the weight of the ice itself and a value added for the sublimation chamber which would not be required for the other two systems. In the case of the thermoelectric system, the elements which have been considered have been the weight of the couples themselves, the weight of the radiator area required for the heat to be rejected, the weight penalty for the power required to operate such a system, and a small weight penalty to account for the necessity of transporting the heat to be rejected to the radiator.

The resulting characteristics are illustrated by Figure 55. The ice sublimation system can be seen to provide the minimum system weight for the shortest mission duration considered. However, this system is doubly time dependent. The refrigeration effect (BTU's/hour) increases with an increase in mission duration because the quantity of food to be provided increases, and also the total amount of refrigeration work (total BTU's) increases with greater mission durations. As a result, the weight of ice required increases with greater mission durations. Thus, the system is no longer competitive with direct radiation beyond a mission duration of 20 man-days. The thermoelectric system is the least efficient system of all for short mission durations until a level of approximately 60 man-days duration is reached, at which time the ice sublimation system weight exceeds that of the thermoelectric system. Both the direct radiation system and the modified direct radiation system indicate the lowest overall system weight for the majority of the mission durations. It can be seen that the need for orienting the radiator or providing multiple radiators increases the overall system weight of a direct radiation system by about a factor of two for the major portion of the range of mission durations considered. Even so, the modified direct radiation system far exceeds either the ice sublimation of thermoelectric system for mission durations in excess of 30 man-days.

The suitability of the systems can be generally summarized. The technique of direct radiation, including the modified system, appears best suited for very nearly every application, especially the extremely long range missions not represented by the figure.

Table 13. Weight Factors Considered for Comparison of Systems, Short Mission Durations

Direct Radiation System					
Mission Duration (Man-Days)	Heat Load (BTU/hr)	Rad. Surface Req'd. $ft.^2$	Weight (lbs.)		
			Rad. Only	Heat Trans. Sys.	Total
7	30	1.2	1.2	7.0	8.2
28	100	4.0	4.0	8.8	12.8
42	140	5.6	5.6	9.4	15.0
90	250	10.0	10.0	11.0	21.0

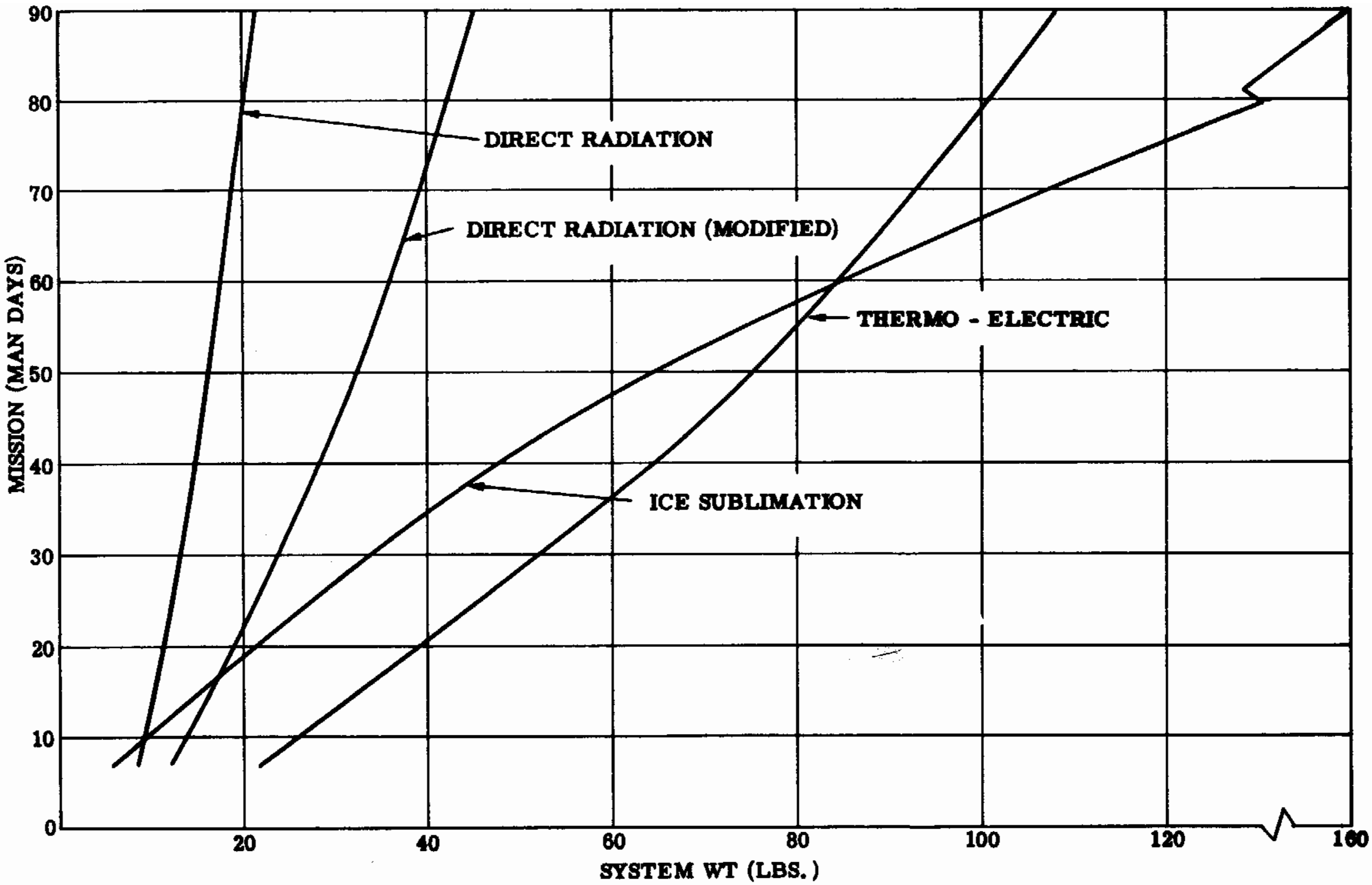
Direct Radiation (modified)			
Mission Duration (Man-Days)	Total Weight of Rad. System from above (lbs)	Added Factor for Mult. Rad. or Orient. Equip. (lbs.)	Total Weight (lbs.)
7	8.2	3.8	12.0
28	12.8	10.2	23.0
42	15.0	14.0	29.0
90	21.0	24.0	45.0

Ice Sublimation System							
Mission Duration			Heat Load		Weight (lbs.)		
man-days	days	hrs.	BTU/hr.	Total BTU	Ice only*	Sub. Ch.	Total
7	7	168	30	5,040	4.6	1.0	5.6
28	14	336	100	33,600	30.5	3	33.5
42	14	336	140	47,100	42.8	4	46.8
90	30	720	250	180,000	149	10	159

*Note: Using only 1100 out of 1220 BTU/lb. available - allows 10% residue

Thermo-Electric (COP = 0.3)									
Mission Dur. Man-Days	Heat Load BTU/hr.	No. of Elements Required	Total Heat to be Discarded BTU/hr	Power Added BTU/hr	Weight (lbs.)				
					T.E. Elem.	Rad.*	Power	Heat Trans.	Total
7	30	9	120	90	1.4	1.5	8.8	10.0	21.7
28	100	30	400	300	4.7	4.7	29.3	10.6	49.3
42	140	41	560	420	6.4	6.6	41.0	10.8	66.8
90	250	74	1000	750	11.5	10.6	73.3	11.5	108.1

*Note: Rad. Wt. estimated on basis of 85 BTU/sq. ft. for rad. temp. of 520°R.



SYSTEM WEIGHT COMPARISON
SHORT MISSIONS
Figure 55

Even in the case of the very short mission duration, while an ice sublimation system appears to provide the absolute minimum system weight, a direct radiation system is somewhat competitive. The ice sublimation system appears suitable for use for very short mission durations only, and would appear to be out of consideration beyond a value of 20 man-days unless a system of ice resupply could be made available. Of course, numerous estimates have been made in the preparation of each of these curves such that they should properly be represented by bands rather than single lines. Thus, it is improper to establish an exact cross-over point beyond which one system far exceeds another. Therefore, the cross-over point itself has not been used to establish the useful limit of the system, but an added margin has been used instead. If a real need for a 20 man-day mission duration or less were established, the ice sublimation system should be seriously considered since in addition to weight factor, several other advantages exist. The system is extremely simple and does not require a radiator with the corresponding elimination of concern regarding orientation. A thermoelectric system has no apparent advantage regarding system weight, as was to be expected from the analysis of the heat-pumping situation made earlier in the report. However, this is a system which is in operation here on earth and has proven capability. It is also simple and quite reliable such that operation in space could not be expected to result in a deterioration of performance. The system weight, however, diverges so rapidly from that of direct radiation that it does not appear feasible to consider this system for any mission where minimum weight would be a prime requirement. The major element contributing to this large system weight is, of course, the power required, again a factor of serious concern for space vehicular application. It appears that a considerable improvement in the efficiency of such a system would have to be accomplished before this type of system would be feasible for use for space vehicular use.

SECTION VI

DISCUSSIONS OF RESULTS

In many areas throughout the course of this study it was necessary to examine the borderline of knowledge in order to predict the direction in which future development would take place. However, in order to make reasonable assumptions and arrive at a comparison of systems, it was necessary to select values which represent present "state of the art." Values were selected which appeared realistic and deemed capable of being produced if a model of the particular system under consideration were to be built. In a few cases, the "state of the art" has already advanced beyond the values selected at that point in the study. However, it has been attempted to demonstrate the net effect that such a change would produce.

Since a system of direct radiation appears to be best suited for practically all missions which can be conceived of, it is this concept which is primarily recommended for future effort. A model of such a system could be readily constructed and tested in order to further evaluate system performance. None of the other systems considered appear worthy of further consideration, with the possible exception of the ice sublimation system, which, however, is severely mission limited. A model of such a system could be built and tested if the requirement for a short mission duration is established.

A further development resulting from this study is the apparent need for complete information on the α/ϵ, or absorbtivity to emissivity ratio of radiator coating materials and the capability to sustain this value after exposure in space. It can be seen that if a sufficiently low value can be established and maintained, it would no longer be necessary to be concerned about orientation with respect to the sun. That is to say, by proper adjustment of this ratio, solar energy can effectively be shut out of the radiator and only low level radiation in the temperature range of the refrigeration effect could be emitted by this radiator. Earth orientation, however, would still remain a problem in that it would not be desirable to maintain the radiator parallel to the surface of the earth at a relatively low altitude. Since the temperature level of the earth's radiation is of the order of magnitude of the radiation being emitted from the radiator, the radiator would be free to accept this large quantity of earth radiation. Therefore, for a low altitude earth-orbiting vehicle, complete control of surface characteristics would not entirely provide a fool-proof system without requiring further control of the situation.

Another situation which should be carefully examined is the complete integration of the containers, the food warming system, and the refrigerator. It is obviously necessary to provide a completely integrated food preservation and servicing system intended for a specific vehicle. When a specific vehicle is to be considered, the availability of an existing heat sink in the form of an appropriate low temperature medium should also be very carefully evaluated.

The problem of providing food in containers permissive of temperature storage and also the addition of heat in order to warm the food prior to consumption is a problem closely related to the design of the refrigerator. Thus, the philosophy of the interval design of the refrigerator itself must be considered. Here we must examine two conflicting concepts. From the standpoint of the necessity to ultimately heat the food and its container, it would appear desirable to refrigerate the food at the highest possible temperature compatible with the ability to prevent deterioration. On the other hand, the establishment of tight tolerance limits on the refrigerator increases the difficulty of the temperature control problem. That is to say, the refrigeration system control would greatly be simplified if a maximum temperature were all that were established. Thus, if the food were required to be maintained at a temperature below $0^{\circ}F$, and one were not concerned with how low the temperature became, control would not be quite as serious. There is, of course, the possibility that extemely low temperatures would be harmful to the food, or in some way reduce its acceptability. This should be determined. If it appears suitable for the food to be stored for long periods of time at extremely low temperatures a three-zoned refrigeration system as has been mentioned previously appears to be appropriate. This would mean that most of the food would be stored at a relatively low temperature with a corresponding relatively wide tolerance range established on this particular temperature. Prior to consumption, the food would be placed into a temperature zone approaching the upper limit necessary for preservation. This would then require the addition of a minimum amount of heat in order to warm the food prior to consumption. The three-zone provided would be a chill area. A form of this concept appears possible for use even in a relatively small refrigerator. It is possible, for example, to provide a built-in thermal granient within the refrigerator, let us say from the back end to the front end, where access to the refrigerator is provided. The food would then advance from a relatively colder portion of the refrigerator to the warmest area immediately prior to removal for use.

The question of providing a chill space in the refrigerator also requires some discussion. A separate control system and separate radiator surface maintained at the specified chill temperature appears to be an unnecessary additional complication. An adequate solution to the problem of providing a chill space appears to be that of proper control of the insulation arrangement. That is to say, a buffer zone of the required higher chill temperature can be provided between the outside of the refrigerator and the low temperature freezer compartment. If a thermoelectric or other heat pumping system were used, it then might be relatively easy to provide a specially controlled chill compartment. However, utilizing a direct radiation technique, this arrangement is considered inadvisable.

Regarding the capability to store fecal matter in the refrigerator, aside from the aesthetic undesirability of so doing, there appears to be no engineering reason why this cannot be accomplished. The heat load produced is not significant, and the volume occupied is relatively small, especially if provisions were made to incorporate the ability to store this material into used containers. It would appear that this would be the best manner in which to store this material, since the provision of a separate compartment would entail reservation of a comparatively large volume. The volume necessary would, of course, already be made available within the refrigerator proper as food is consumed. The interior of the refrigerator could be appropriately arranged such that the fecal matter would in effect be separated from the food yet to be consumed. One could summarize this situation by saying that if the astronauts don't mind this arrangement, it could certainly be designed into the system.

The foregoing has primarily dealt with those applications deemed most likely in the immediate future. The concept of multi-man missions for relatively longer periods in space requires a somewhat different approach. Food volume and food weight can be seen to rapidly come to significant numbers. In terms of the concepts proposed, it would be obvious that an immediate remedy for this situation would be a reprovisioning arrangement. With food regularly resupplied, very large quantities would not be required at any one time. This would not basically alter the refrigerator concepts which have been developed, but would instead modify the size of the refrigerator to be installed on board the vehicle. Thus, the multi-man space vehicular refrigerator is still envisioned as a direct radiation system utilizing a heat transport fluid.

Certain concepts, however, at this point become more attractive, such as the desirability of providing a passive system of radiation, if possible. The trailing spheres concept of maintaining steady state inputs and outputs, for example, should be considered. Other possible means for varying the absorbtivity to emissivity ratio of surfaces should similarly be investigated. Heat pump developments must be carefully watched. The problems existing in the use of heat pumps, as stated in this report, will probably be continually attacked with resulting improvements. Problems such as liquid vapor separation mentioned for the vapor compression system will be solved, as this concept is necessary in many systems other than that of refrigeration. Nuclear power plants utilizing Rankine Cycles will find it necessary to resolve this problem long before it is solved for the space refrigerator. Power requirements will be reduced and weight penalties resulting from the provision of power will likewise be reduced. It is entirely conceivable that futuristic vehicles will have available more energy than possibly could be used, in which case heat pumps, particularly simple and reliable ones such as the thermoelectric system should be seriously considered to produce the refrigeration effect required.

Another aspect of the problem must also be considered. If resupply were not possible such as in the case of a deep-space exploration, another look must be taken at the food provided. Since the weight and volume of the kind of foods which have been considered would be extremely large if used for deep-space penetrations, it would appear that some cutoff point for such foods would exist. There must certainly be a point beyond which we would not consider providing this highly acceptable food form individually packaged. We would, instead, in order to accomplish overall mission objectives, accept a denser packaging of foods at a lower weight per man-day stored. This would, of course, mean dehydrated food. With dehydrated food and a closed cycle water system, the volume and weight requirements could be greatly reduced. The food storage system under such an arrangement could be conceived of as a series of large hoppers packed with granules of concentrated food provided with a means of expulsion. Of course, with such a concept it would appear that no refrigeration would be required and these foods could be stored at room temperature.

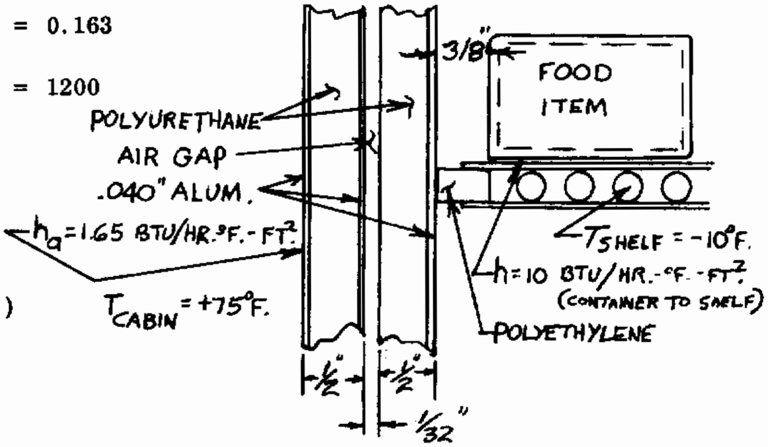
Even with such a system we would again reach a cutoff point. For a trip to a distant planetary system, the times involved might be so great that the space vehicle would have to be essentially one big food container. Thus, we reach a point where an open cycle system of providing food is entirely unacceptable and the concept of a closed cycle must be applied. That is, food would have to be regenerated from the waste products produced by man and his equipment on board the space vehicle. Such a system, of course, would be utopian, since the astronauts would then have the capability of sustaining their own existence for an unlimited period of time. There are perhaps some other regions between these which have been discussed, such as a partially closed cycle supplemented by stored components which we are unable to regenerate. However, there is no question that if we are to explore distant galaxies, we must ultimately be able to develop a completely closed system. If man is ever to set foot on all of that realm of the universe that he is presently aware of, he will have to learn to like algae or some other product resulting from a regenerative cycle.

APPENDIX 2

FOOD STORAGE COMPARTMENT HEAT LOAD CALCULATIONS

Heat Leak - Wall

Polyurethane	$K = 0.16$ BTU-in/hr.ft^{2}F^o
Polyethylene	$K = 1.0$
Gasket	$K = 1.35$
Air (still)	$K = 0.163$
Aluminum	$K = 1200$



$$Q = \frac{\Delta T \times A}{\epsilon \frac{L}{K} + \epsilon \frac{1}{h}}$$

Q = rate of heat transfer (BTU/hr.)

ΔT = temperature difference

Let A = 1 ft^2

L = length of heat path in in.

K = thermal conductivity BTU-in/hr.ft^{2}F^o

h = heat transfer coefficient BTU/hr.ft^{2}F^o

ϵ = Summation

$$Q = \frac{\left[75^oF - (-10^oF)\right] \times 1\ Ft^2}{\left(\underset{\text{out-side}}{\frac{1}{1.65}} + \underset{\text{Alum.}}{\frac{3(0.06)}{1200}} + \underset{\text{air gap}}{\frac{0.031}{0.163}} + \underset{\text{Foam}}{\frac{1}{0.16}} + \underset{\text{Poly strip}}{\frac{0.125}{1}} + \underset{\text{Food air gap}}{\frac{0.38}{0.163}} + \underset{\text{Food Cont.}}{\frac{2(0.06)}{1}} + \underset{\text{Food Cont. to Shelf}}{\frac{2-}{4.1}\ \frac{1}{10}}\right)\left(\frac{1}{(BTU/hr.ft^2F^o)}\right)}$$

$$Q = \frac{85^oF\ Ft^2}{(0.606 + 0.00015 + 0.19 + 6.26 + 0.125 + 2.33 + 0.12 + 0.488 + 0.1)\ (1/BTU/hr.ft.^2\ ^oF)}$$

$t\frac{QL/K}{A}$ $\Delta t = 5.04$ $\Delta t = 0$ $\Delta t = 1.58$ $\Delta t = 52$ $\Delta t = 1.04$ $\Delta t = 19.3$ $\Delta t = 1$ $\Delta t = 4.02$ $\Delta t = 4.02$ $\Delta t = 0.83$

$$Q = \frac{85^oF\ Ft^2}{10.21915\left(\frac{1}{(BTU/hr.ft^2F^o)}\right)} = 8.3\ \text{BTU/hr. for a 1 Ft}^2\ \text{Area}$$

	H	W	L
Total Loss Through Walls	21.75"	x 11.75"	x 12"
	1.8'	1'	1'

Front Side Top

$A = 2(1.8 \times 1) + 2(1.8 \times 1)\ 2(1 \times 1)$

$A = 3.6 + 3.6 + 2 = 9.2\ Ft^2$

Q Total = $8.3\ BTU/hr.Ft^2 \times 9.2\ Ft^2 = 76.4\ BTU/hr.$

__DOOR OPENING - HEAT LOSS__

Total Internal Volume (not counting food or shelves)

$V = 1.8 \times 1 \times 1 = 1.8\ Ft^3$

Say 1 air change for each opening

Air density @ STP e = $0.075\ lb/ft^3$ Specific heat $C_p = 24\ BTU/lb.F^o$

$\Delta t = 85^o$

Weight $W_{air} = 0.075\ lb/ft^3 \times 1.8\ Ft^3 = 0.133\ lb.$

$\Delta H = W_a C_p\ \Delta t = 0.133\ lb. \times 0.24\ BTU/lb.F^o \times 85\ F^o = 2.71$ BTU/opening

Heat Removal due to Moisture Influx

Air @75^oF and 50% RH has 64 gr H_2O/lb/dry air

7000 GR = 1 lb.

$\frac{64}{7000} \times 0.133$ lb. dry air = 0.00122 lb. moisture/air change

When cooled to 0^oF and 100% RH spec. humidity is $<$10 gr. moisture/lb. dry air

$64 - 10 = 54$

$\therefore \frac{54}{7000} \times 0.133\# = 0.001025$ lb. moisture will condense

Total Heat removed from moisture

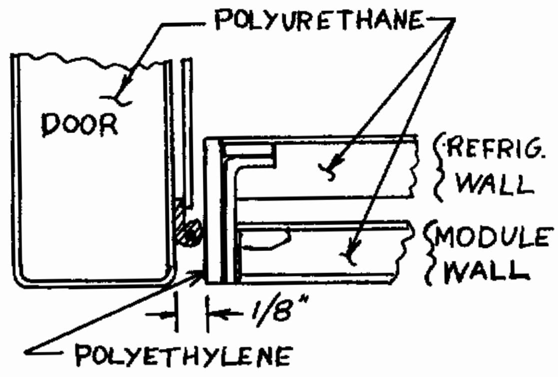
Sensible Heat Latent Heat
$0.00122\ lb. \times 1\ BTU/lb.\ F^o \times 85^o F + 144\ BTU/lb. \times 0.001025\ lb.$

0.1036 BTU + 0.148 BTU = 0.242 BTU heat removed from moisture

Assume 1 opening per hr.

Air change + ice forming
2.71 BTU + 0.242 = 2.952 Say 3 BTU/hr.

DOOR GASKET - HEAT LOSS



Conduction through breaker strip (one)

$K = 1.0 \quad A = \frac{1}{8} \times 2\ (12 - 21.75)$

$$A = \frac{8.44\ in^2}{144} = 0.0585\ ft^2$$

$$Q = KA\Delta t$$

$$Q = 1.0 \times 0.0585 \times 52 = 3.04\ \text{BTU/hr.}$$

Conduction through air gap and gasket

$Q = KA\Delta t$ K gasket = 1.35

$\Delta t = 52^o$ K air = 0.163

$A = 0.0585\ ft^2$

$$Q = \frac{0.0585\ ft^2 \times 52^o F}{\frac{1}{0.163} \times \frac{5}{1.35}} = \frac{3.04}{\frac{0.5}{0.22}} = \frac{3.04}{2.27} = 1.34\ \text{BTU/hr.}$$

Conduction through aluminum strip

Let $A = 1\ ft^2$

$$= \frac{\Delta t \times A}{\Sigma \frac{L}{K} + \Sigma \frac{1}{h}} = \frac{52 \times 1}{\underset{\text{POLY}}{\frac{0.125}{1}} + \underset{\text{AL}}{\frac{1}{1200}} + \underset{\text{OUTSIDE}}{\frac{1}{1.65}}}$$

$$= \frac{52}{0.125 \times 0.000834 + 0.606} = \frac{52}{0.731834} = 71\ \text{BTU/hr. per sq. ft.}$$

$$A = \frac{1}{8} \times 2\,(12 - 21.75)\,\frac{8.44\ in^2}{144\ in^2/ft^2} = 0.0586\ ft^2$$

$$Q = 71\ BTU/hr./ft^2 \times 0.0586\ ft^2 = 4.16\ BTU/hr.$$

Total Door Joint Heat Loss = 3.04 + 1.34 + 4.16 = 8.54 BTU/hr.

Total Heat Loss	Walls = 76.4
	Opening = 3.0
	Door = 8.54
	87.94 BTU/hr.

PULL DOWN - HEAT REMOVAL - CHILL COMPARTMENT

FOOD CONTAINERS

C_p = 0.23 (ALUMINUM)

C_p = 0.55 (POLYETHYLENE)

Wt = 0.1#

Assume pull-down as $75^{\circ}F$ to $32^{\circ}F$, (ΔT)

No. of containers	6 aluminum containers	15 in^3 each	approx.
	16 polyethylene tubes	34 in^3 each	

$\Delta H = W \times C_p\, \Delta t$

$\Delta H = 0.1 \times (16) \times 0.55\,(75^{\circ} - 32^{\circ})$ = 37.8 BTU must be removed from tubes

$\Delta H = 0.1 \times 6 \times 0.23\,(75^{\circ} - 32^{\circ})$ = 5.94 BTU must be removed from containers

FOOD C_p = 1.0 Food (Assume same as water)

$\Delta t = 43^{\circ}, (75^{\circ} - 32^{\circ})$

			ESTIMATE
FREEZER*	180 Bread Items	@	0.1# = 18#
	30 Container Items	@	0.4# = 12#
			30#
CHILL	16 Tube Items	@	0.4# = 6.4#
	6 Container Items	@	0.4# = 2.4#
	10 Bread Items	@	0.1# = 1.0#
			9.8#

*No pull down required.

H = 9.8 x 1 x (75-32) $\therefore$ 422 BTU (must be removed from food)
= 422 BTU

43.74 (BTU must be removed from tubes and containers)

TOTAL 465.74 BTU

Assume Pull Down over a 12 hr. period

$\frac{465.74 \text{ BTU}}{12 \text{ hrs.}}$ = 38.8 BTU/hr. for 12 hrs.

SUMMATION OF HEAT REMOVAL LOADS

76.4 BTU/hr. "Steady state" wall heat leakage

3.0 BTU/hr. Door openings (one per hour)

8.54 BTU/hr. Door Joint Leakage

38.8 BTU/hr. Pull down (for 12 hours)

126.75 BTU/hr. Total Load while chill compartment is being pulled down

Say 130 BTU/hr. for first 12 hours

Then 90 BTU/hr. for remainder of time.

APPENDIX 3

LOW TEMPERATURE RADIATOR DESIGN PROCEDURE

Stefan Boltzmann Law

The basic equation for radiation from any body is:

$$Q = \sigma \epsilon A T^4 \qquad (7)$$

where Q = heat rejected by radiation, BTU/hr

σ = Stefan Boltzmann constant

$= \dfrac{0.1713 \times 10^{-8} \text{ BTU}}{(\text{hr})(\text{ft}^2)(^{\circ}\text{R}^4)}$

ϵ = emissivity of the surface, dimensionless

A = total radiating area, ft^2

T^4 = temperature of radiating body, $^{\circ}R$

The terms in this equation are discussed separately below.

Emissivity and Absorptivity

From the above equation it can be seen that the higher the emissivity, ϵ, the greater the amount of heat that can be rejected from a surface. Furthermore, it should be stated here that an efficient space radiator should have a low solar absorptivity, δ, to minimize the amount of high temperature solar energy that is absorbed. The significance of the ratio of solar absorptivity to emissivity, δ/ϵ, regarding the determination of radiator heat rejection capabilities as well as the establishment of effective space heat sink temperature are discussed in detail in Appendix 1 and later in the appendix.

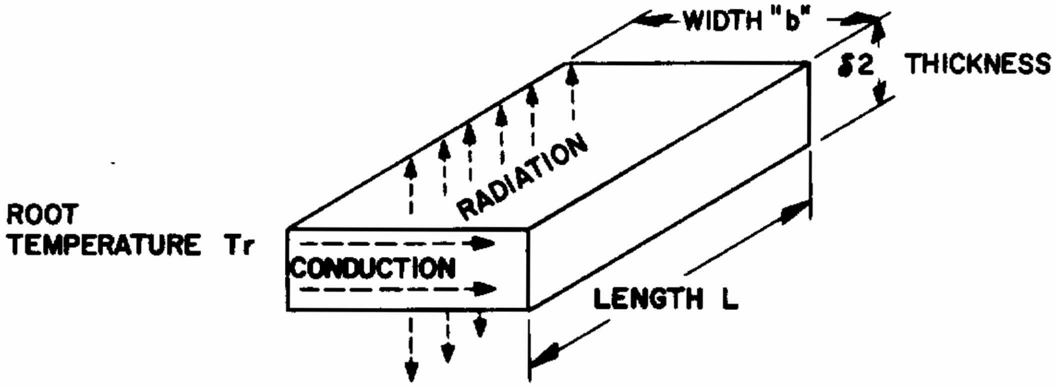
Temperature of Radiating Body

For a fin tube construction, there will be a temperature gradient in the fin. Figure 56 shows several fin profiles. As the fins become thinner the temperature at the tips becomes lower for a given heat input. Then the mean temperatures T_{m1}, T_{m2}, T_{m3}, change, and the area required to dissipate a given quantity of heat changes. To eliminate one of the variables, all temperatures are compared to the temperature at the root of the fin, T_r. From Figure 56, the following equation can be derived:

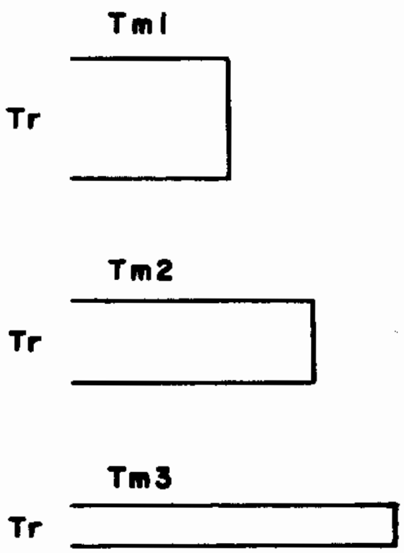
$Q_1 = Q_2 = Q_3$ = the heat in

$$\sigma \epsilon A_1 T_{m1}^4 = \sigma \epsilon A_2 T_{m2}^4 = \sigma \epsilon A_3 T_{m3}^4$$

$$A_1 T_{m1}^4 = A_2 T_{m2}^4 = A_3 T_{m3}^4$$



HEAT ENTERS AT ROOT TEMPERATURE Tr, AND RADIATES FROM BOTH SIDES AT MEAN TEMPERATURE Tm, FROM AREA A = 2 bL.



$$Q_1 = \sigma \epsilon A_1 T^4_{m1}$$

$$Q_2 = \sigma \epsilon A_2 T^4_{m2}$$

$$Q_3 = \sigma \epsilon A_3 T^4_{m3}$$

FIGURE 56

FIN TEMPERATURE AND RADIATION

Let $\eta_1 = \left(\frac{T_{m1}}{T_r}\right)^4 \quad \eta_2 = \left(\frac{T_{m2}}{T_r}\right)^4 \quad \eta_3 = \left(\frac{T_{m3}}{T_r}\right)^4$

Then $A_1 \eta_1 T_r^4 = A_2 \eta_2 T_r^4 = A_3 \eta_3 T_r^4$

$$A_1 \eta_1 = A_2 \eta_2 = A_3 \eta_3 = A \eta$$

In general

$$Q = \sigma \epsilon A \eta T_r^4 \tag{8}$$

Then, fin width, (hence, area) thickness, thermal conductivity and root temperature can be combined into a radiation modulus and plotted with fin efficiency η as in Figure 57.

The radiation modulus is given as:

$$Mr = \frac{\sigma \epsilon b^2 T_r^3}{k \delta} \tag{9}$$

where Mr = the radiation modulus, dimensionless

b = $\frac{1}{2}$ the tube spacing, feet

k = thermal conductivity of the fin material, BTU/(hr) (ft^2) ($^\circ$F)/ft

δ = $\frac{1}{2}$ the fin thickness, feet (for radiation from 2 sides)

Tr = temperature of fin at root, $^\circ$R

σ, ϵ defined above

Effect of Sink Temperature

Appendix I indicates that a flat plate in orbit with no heat input will experience cyclic temperature changes. For a given point in the cycle with a fixed position, orientation, solar absorptivity and emissivity, the plate will reach an equilibrium temperature, T_s. At this temperature, the incident heat (sun, earth and earth albedo) will be radiated to space at the rate of

$$Q = \sigma \epsilon A T_s^4 \tag{10}$$

This temperature will reduce the capabilities of the plate to radiate additional heat. Thus, equation 8 should be modified to read:

$$Q = \sigma \epsilon A (\eta T_r^4 - T_s^4) \tag{11}$$

where the new term T is the effective sink temperature. For COOL, T_s was established as 360°R. (Appendix I)

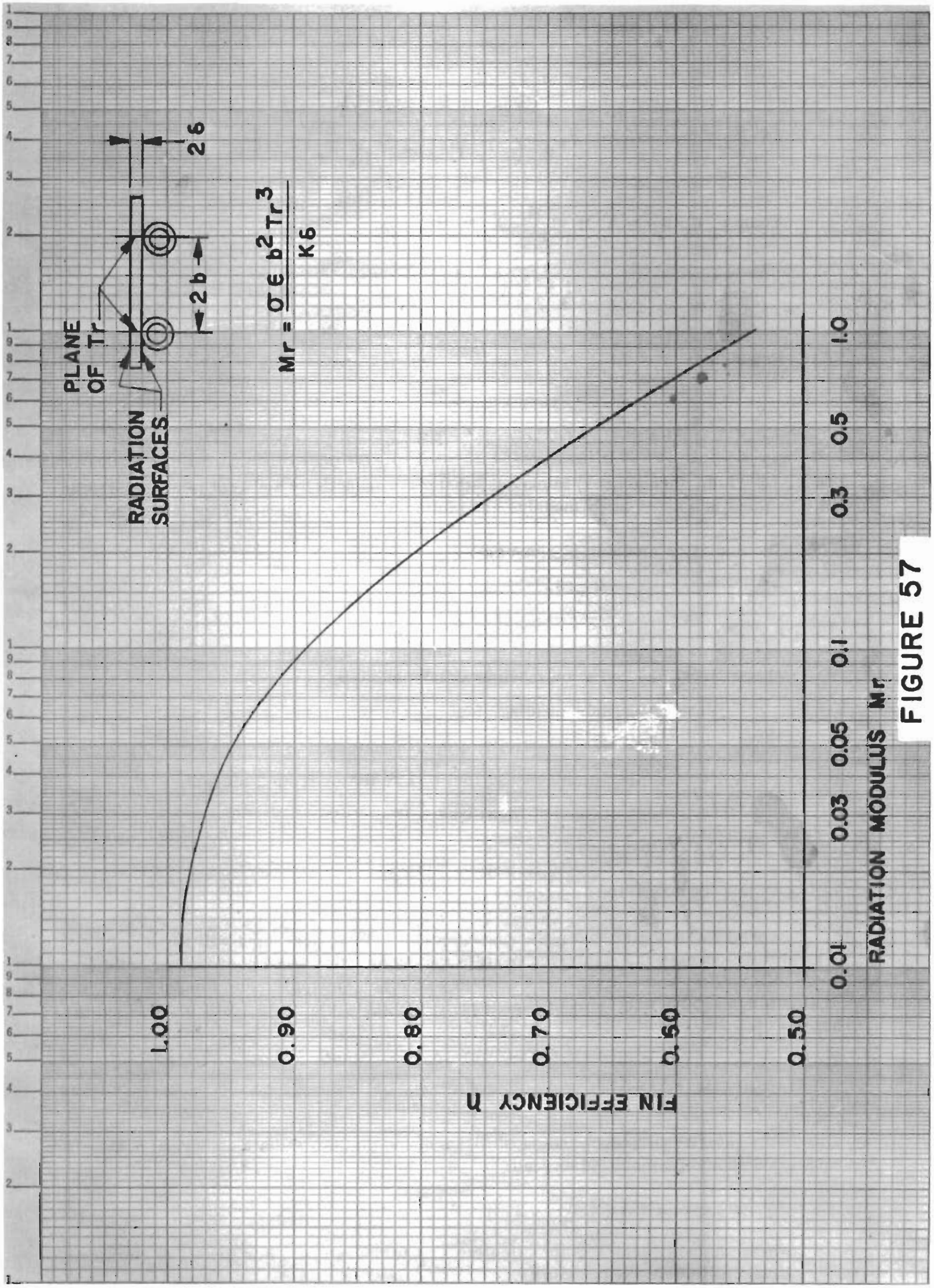


FIGURE 57

FIN EFFICIENCY VERSUS RADIATION MODULUS

The inclusion of T_s in equation 11 reduces the radiative capacity of a fin. Thus, the fin thickness given by equation 9 and Figure 57 will be greater than actually required to dissipate the heat of equation 11, resulting in a fin thickness design that is conservative.

Design Steps

For the synthesis of the radiator, more equations than 10 and 11 are needed. In fact, the number of equations that are available is less than the number of unknowns. (More equations introduce more unknowns.) This means that various assumptions were made. The equations and assumptions are given in the steps that follow.

Radiation Equation

$$Q = \sigma \epsilon A (T_r^4 - T_s^4) \quad \text{(11) repeated}$$

Q = 1000 BTU/hr (from refrigerator load with safety factor)

$\sigma = 0.1713 \times 10^{-8} \dfrac{\text{BTU}}{(\text{hr})(\text{ft}^2)(^\circ\text{R}^4)}$ (Stefan Boltzmann Constant)

ϵ = 0.9 (for TiO_2 paint)

$T_s = 360^\circ R$ (given)

Unknowns = A, η, T_r

Mean Fluid Temperature

$$Q = W c_p \Delta t_c \quad (12)$$

Q = heat to be dissipated, 1000 BTU/hr

c_p = specific heat of coolant, 0.780 BTU/lb $^\circ$R

Unknowns W = flow rate lb/hr

Δt_c = temperature drop in coolant, $^\circ$R

$$T_{cm} = T_i - \frac{\Delta t_c}{2} \quad (13)$$

where T_{cm} = Mean coolant temperature, $^\circ$R

T_i = Inlet coolant temperature, $^\circ$R

Fluid Convection

The heat transferred from the fluid to the tube wall by convection is given by the equation:

$$q = h_c \; S \Delta t_w \tag{14}$$

where $$q = Q/N \tag{15}$$

but $$S = \pi D L \tag{16}$$

and $$h_c = 1.86 \frac{k_c}{D} \left[\frac{DV\rho}{\mu} \; \frac{c_p \mu}{k_c} \; \frac{D}{L} \right]^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \tag{17}$$

thus $$t = \frac{q}{1.86 \; k \; L \; \frac{DV\rho}{\mu} \; \frac{c_p \mu}{k_c} \; \frac{D}{L}^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}} \tag{18}$$

where the new terms are:

L = length of tube, feet

k_c = thermal conductivity of fluid

$$\frac{\text{BTU}}{(\text{hr})\,(\text{ft}^2)\,(^\circ\text{F})/\text{ft}}$$

D = inside diameter of tube

V = fluid velocity, ft/hr

ρ = fluid density, lb/ft^3

μ = absolute viscosity of fluid at bulk Temperature lb/(ft) (hr)

μ_w = absolute viscosity of fluid at tube and temperature lb/(ft) (hr)

N = number of tubes

q = heat flow per tube, BTU/hr

h_c = convection coefficient,

$$\frac{\text{BTU}}{(\text{hr})\,(\text{ft}^2)\,(^\circ\text{R})}$$

S = total inside surface of tube, ft^2

Δt_w = temperature drop from bulk of fluid to tube wall, ^oR

The term $\frac{DV\rho}{\mu}$ is known as Reynold's number.

The velocity and density, however, are functions of the weight flow and tube diameter.

$$V\rho = \frac{4W}{\pi D^2} \tag{19}$$

then

$$\frac{DV\rho}{\mu} = \frac{4W}{\pi D \mu} \tag{20}$$

and

$$\frac{DV\rho}{\mu} \cdot \frac{c_p \mu}{k_c} \cdot \frac{D}{L} = \frac{4}{\pi} \quad \frac{Wc_p}{k_c L} \tag{21}$$

In order for equation (17) to apply,

$$\frac{4W}{\pi D \mu} < 2100 \text{ and } \frac{Wc_p}{k_c L} > 10 \tag{22}$$

The three unknowns are W, D, and L. Considering the refrigerator requirements and the inequalities (22), the flow rate W can be estimated. Then D and L have a limited range of values. As they are varied, and the number of tubes are varied, Δt_w varies. The diameter and length also affect pressure loss; hence, before proceeding, the pressure should be checked.

Fin Root Temperature

For a set of assumptions of N, W, D and L the temperature difference Δt_w can be determined. There will be a small temperature drop through the tube wall, but since other factors have been conservative, this temperature can be neglected.

By assuming W, Δt_c can be found from equation (12). The coolant inlet temperature T_i is set by refrigerator requirements (at 450^{o}R). Thus, T_{cm} can be found and T_r can be found by:

$$T_r = T_{cm} - \Delta t_w \tag{23}$$

With T_r determined from (23) only two unknowns remain in (11). These are fin area and fin efficiency. Assuming an efficiency, the total required area can be computed. Since L and N have been assumed, the fin width "b" can be found from:

$$b = \frac{A}{4NL} \tag{24}$$

where the factor 4 takes into account two fins per tube, with radiation from both sides of the fin.

At this point there is sufficient data to determine Mr from Figure 21 and fin thickness δ from equation 9.

Tube Wall Thickness

The thickness of tube wall is a function of meteorite size and probability of puncture There are several methods of computing thickness with various results. One equation giving somewhat conservative values for tube thickness is:

$$d = 0.0436 \left[\frac{A_v \tau}{1-P}\right]^{0.3} \quad \text{For Aluminum Tubes} \tag{25}$$

where d = tube wall thickness, inches

A_v = projected area of tube, ft^2

τ = time of mission, years

P = probability of return with no penetration of tube wall in time, expressed as a decimal.

Low Temperature Radiator Calculations

Following are calculations which established the specific radiator size required:

Radiation Equation

(I) $Q = \sigma \epsilon A \left[\eta T_R^4 - T_S^4\right]$

Unknowns

A = radiator area, ft^2

η = fin efficiency

T_R = temperature of fin at root, ^oR.

Given

Q = heat rejected by radiator, 1000 BTU/hr. (with a safety factor)

σ = Stefan-Boltzmann Constant 0.1713×10^{-8} BTU/hr. ft^2 $^oR^4$

ϵ = emissivity, 0.9 for T_1O_2 paint (II)

T_S = effective sink temperature, 360^oR (for a 200-600 mile earth orbit)

Mean Fluid Temperature

(2) $Q = WC_p \Delta t_c$

Unknown

Δt_c = temperature drop

Given

Q = heat rejected by radiator, 1000 BTU/hr.

C_p = specific heat of 0.78 BTU/lb. °R

W = flow rate @ 0.28 GPM, 114 #/Hr.

$$\Delta t_c = \frac{Q}{WC_p} = \frac{1000 \text{ BTU/hr.}}{114 \text{ lb/hr.} \times 0.78 \text{ BTU/lb. }^{\circ}\text{R}} = 11.3^{\circ}\text{R}$$

(3) $T_{cm} = T_i - \frac{\Delta t_c}{2}$

Unknown

T_{cm} = mean fluid temperature, °R

$T_{cm} = 460 - \frac{11.3}{2} = 454.4^{\circ}\text{R}$

Given

T_i = Inlet fluid temperature, 460°R

$\Delta t_c = 11.3^{\circ}\text{R}$

Fluid Flow

(4) $R_e = \frac{DVe}{\mu}$

Unknown

R_e = Reynolds number

Given

D = inside the tube diameter, 0.0254 feet

V = fluid velocity @ 0.214, 3440 ft/hr.

e = fluid density of glycol @ 0°F 66#/ft³

μ = absolute viscosity of glycol, 58 lb/ft hr.

$$R_e = \frac{0.0254 \text{ ft} \times 3440 \text{ ft/hr} \times 66 \text{ lb/ft}^3}{58 \text{ lb/ft hr.}} = 100 \text{ Laminar}$$

Laminar flow is desirable in order to reduce the frictional head loss in the tube and thus reduce the pump power required to circulate the transport fluid. Although laminar flow also reduces the heat transfer through the fluid film, pump power was used as the criterion for selecting fluid flow. Also the 3/8 inch O.D. tube is used in the above calculation because it is easily fabricated into a radiator configuration.

Therefore two radiator flow paths in parallel with the same tube size as above will be used in parallel thus decreasing V from 3440 ft/hr. to 1720 ft/hr.

$$\text{Now } R_e = \frac{DVe}{\mu} = 50$$

Fluid Convection for Laminar Flow

(5) $$\Delta t_w = \frac{q}{1.86\, \pi K_c L \frac{DVe}{\mu} \times \frac{C_p \mu}{K_c} \times \frac{D}{L}^{1/3} \left(\frac{\mu}{\mu_w}\right)^{-14}}$$

Given

q = heat rejected by each side radiator, 500 BTU/hr.

L = length of tube in each radiator side, 50 feet

K_c = thermal conductivity of fluid 0.24 BTU/hr. ft. ^{o}F

D = inside diameter of tube, 0.0254 feet

V = fluid velocity, 1720 ft./hr.

e = fluid density @ 66 lb./ft.^{3}

C_p = 78

$$\frac{\mu}{\mu_w}^{-14} = 1 \text{ for small } t_c$$

μ = absolute viscosity of fluid at bulk temperature of 0^{o}, 56 lb./ft.hr.

Unknown Δt_w = temperature drop from bulk of fluid to tube wall, ^{o}R. By substituting the above values in equation (5), a value of $\Delta t_w = 4.3^{o}R$ is obtained

(6) $T_R = T_{cm} - \Delta t_w$

Given

T_{cm} = mean fluid temperatore 454.4^{o}R (See Equation #3)

Δt_w = temperature drop from bulk of fluid to tube wall, 4.3^{o}R (See Equation #5)

Unknown

T_R = temperature of fin at root, ^{o}R.

T_R = 454.4 - 7.8 = 550^{o}R.

(7) $$MR = \frac{\sigma \epsilon b^2 T_R^{\ 3}}{K \delta}$$

Given

For maximum fin efficiency, let δ = 0.010 inch, = 0.000835 feet, b = 1.0 inch = 0.0835 feet (estimated from preliminary calculations).
$\sigma = 0.1713 \times 10^{-8}$ BTU/hr.ft.^{2} $^{o}R^4$ (See Equation #1)

ϵ = 0.9 TiO_2 paint

T_R = $450^{o}R$ (See Equation #5)

k = thermal conductivity 6061-16 aluminum, 90 BTU-ft/hr. ft^2 ^{o}R

Unknown

M_R = Radiator Modulus

b = 1/2 tube spacing, feet

δ = 1/2 fin thickness, feet (radiation both sides)

$$M_R = \frac{0.1713 \times 10^{-8}\ \text{BTU/hr ft}^2\ {}^{o}\text{R}^4 \times 0.9 \times (0.0835\ \text{ft})^2 \times (450^{o}\text{R})^3}{90\ \text{BTU Ft/Hr. Ft}^2\ {}^{o}\text{R} \times 0.000835\ \text{ft.}} = 0.0117$$

From Figure 57

η = 0.99 fin efficiency

Substituting the derived values into Equation #1

$$A = \frac{q}{\sigma \epsilon \left[\eta\, T_R^{\ 4} - T_S^{\ 4}\right]} = \frac{500\ \text{BTU/hr.}}{0.1713 \times 10^{-8}\ \text{BTU/hr. ft}^2\ {}^{o}\text{R}^4 \times 0.9 \left[0.99\,(450^{o}\text{R}) \cdot (360^{o}\text{R})^4\right]}$$

$A = 13.1\ ft^2$

x 2 two parallel radiators

26.2 ft^2 Total Radiator Area (or 26 sq. ft.) Required

APPENDIX 4

AIR CONDITIONER HEAT EXCHANGER DESIGN

USING NTU-EFFECTIVENESS METHOD (REF. 2)

I **SPINED FIN TUBE CHARACTERISTICS**

Total air side surface/lineal ft. of tube = 1.44 ft^2/ft

Total liquid side surface/lineal ft. of tube .211 ft^2/ft

Internal Hydraulic Radius

$$D_H = 4r_h = 4\,\frac{\text{AREA}}{\text{WETTED PERIMETER}}$$

$$D_H = \frac{4.4 \times 10^{-4} \times 4}{0.211} = 83.2 \times 10^{-4} \text{ ft.}$$

TUBE CROSS-SECTION

II **HEAT TRANSFER FLUID CHARACTERISTICS** 50% H_2O - 50% Ethylene Glycol

$$cp = 0.780 \frac{\text{BTU}}{\text{lb. }^{\circ}\text{R.}}$$

$$p = 66 \frac{\text{lb.}}{\text{ft.}^3}$$

$$M = 12.8 \frac{\text{lb.}}{\text{ft. hr.}}$$

$$K = 0.242 \frac{\text{BTU}}{\text{hr. }^{\circ}\text{F. ft.}}$$

Assume liquid side flow rate = 5.5 GPM

$$\text{WT. FLOW} = 66 \times 44.2 = 2920 \frac{\#}{\text{hr.}} \quad C_c = W_c \times cp = 2920 \times 0.78 = 2280 \frac{\text{BTU}}{\text{hr. }^{\circ}\text{R.}}$$

Cold Side Fluid Temperature Rise

$$Q = w c_p \Delta T_c$$

$$18650 = 3180 \text{ \#/hr.} \times 0.78\ \Delta T_c$$

$$\Delta T_c = 7.5^{\circ}\text{F}$$

III AIR FLOW RATE REQUIREMENTS

Gross sensible heat load = 18650 $\frac{BTU}{hr.}$

Design is to be such that coil temperature is to be above the dew point temperature for 75^{o}F and 50% R. H. air mixture (55^{o}F). Therefore, assumed that T_{hout} = 59^{o}F (discharge air temperature)

$$T_{hin} = 75^{o}F \text{ and } 50\% \text{ R. H.}$$

From Psychrometric Chart

$$h_{hin} = 28.3 \ \frac{BTU}{lb.}$$

For 100% sensible load (SHF = 1.0)

$$h_{hout} = 24.0 \ \frac{BTU}{lb.} \quad 59^{o}F \text{ and } 85\% \text{ R. H.}$$

$$W_{req'd} = \frac{18650}{28.3-24} = 4330 \ \frac{lb.}{hr.}$$

$$\text{CFM @ 1.0 atm} = \underline{962 \text{ CFM}}$$

$$\text{CFM @ 0.5 atm} = 1924 \text{ CFM}$$

$$C_H = w \times cp = 4330 \times 0.24 = 1040 \ \frac{BTU}{hr. \ ^{o}R.}$$

IV EXCHANGER ARRANGEMENT

(Liquid side) 12 parallel paths, 4 passes/path

Total high side area: Total A_h = 1.44 ft^2/ft x 48 lengths x 2.75 ft/length = 190 ft.

$$A_h = 1.44 \text{ ft}^2/\text{ft.} \quad \therefore \frac{A_h}{A_c} = 6.83$$

$$A_c = 0.211 \text{ ft}^2/\text{ft.} \quad \text{Total } A_c = 27.8 \text{ ft}^2$$

Air side free flow area = 1.58 ft^2 (Ref. 3)

$$\text{Air side velocity} = \frac{962}{1.58} = 600 \text{ FPM}$$

$$h_{air} \text{ side} = 29 \text{ BTU/hr. } ^{o}F \text{ ft}^2$$

Air Side Extended Surface Overall Effectiveness

Assume 90% fin efficiency, ηf

$$\eta\,oh = 1 - \frac{A_f}{A}\,(1 - \eta f)$$

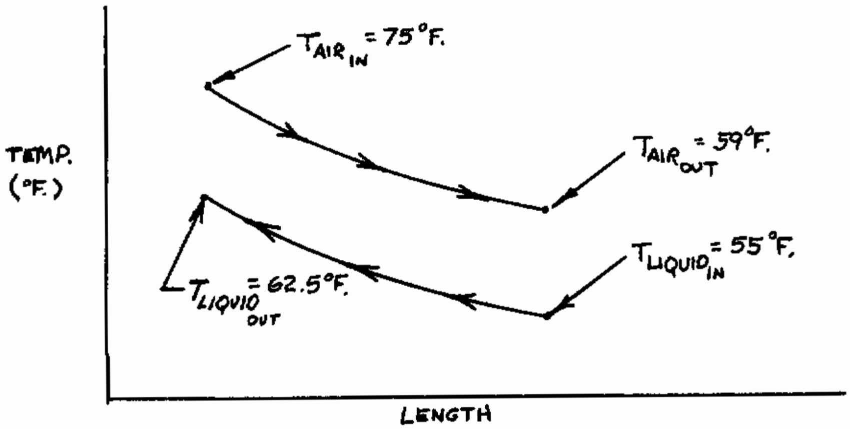
$$\eta oh = 1 - \frac{178}{190}\,(1 - 0.90)$$

$$\eta oh = 0.906$$

V DETERMINE UA REQUIRED CONFIGURATION CHOSEN (NTU-EFFECTIVENESS)

$$\Sigma = \frac{C_c\,(tc_o - t_{c_i})}{Cmin\,(t_{h_i} - t_{c_i})} = \frac{2280\ (7.5)}{1040\ (20)} \qquad \Delta t = 7.5$$

$$\Sigma = 0.824 \text{ (effectiveness)}$$



$$\frac{Cmin}{Cmax} = \frac{1040}{2280} = 0.456$$

Assume cross flow configuration

From Ref. 2 page H-135.

Using $\frac{Cmin}{Cmax}$ = 0.456 and = 0.823

NTU = 3.0

UA required = NTU x Cmin

UA required = 3.0 x 1040 = 3120 BTU/hr. - ^oR

VI DETERMINATION OF US "AVAILABLE" FOR CONFIGURATION AND SIZE ESTABLISHED

Liquid side Reynolds No. :

$$R_N = \frac{D_H V_s \rho}{\mu}$$

$$V_s = \frac{44.2 \text{ ft}^3/\text{hr.}}{12 \times 4.4 \times 10^{-4} \text{ ft}^2}$$

$$= 8350 \text{ ft/hr.}$$

$$M = 12.8 \text{ Lb./ft. hr.}$$

$$R_N = \frac{0.832 \times 10^{-2} \times 8350 \times 66}{12.80}$$

$$R_N = 358 \text{ (Laminar)}$$

$$\frac{h_c D_h}{K} = 2.02 \left(\frac{wc}{KL}\right)^{1/3} \quad \text{(Faires Form. 101, Ref. I)}$$

$$h_c = 2.02 \; \frac{(243 \times 0.78)}{(0.243 \times 11)}^{1/3} \times \frac{0.242}{0.832 \times 10^{-2}}$$

$$= 2.02 \; (71.5)^{1/3} \times 0.291 \times 10^2$$

$$= 2.02 \times 4.15 \times 29.1 = 245 \text{ BTU/hr } ^oF \text{ ft}^2$$

$$\frac{1}{UA} = \frac{1}{A_h \eta_{oh} h_h} + \frac{1}{A_c h_c}$$

$$\frac{1}{UA} = \frac{1}{190 \times 0.906 \times 29} + \frac{1}{27.8 \times 245}$$

$$\frac{1}{UA} = \frac{1}{5000} + \frac{1}{6820}$$

$$= 0.0002 + 0.00046 = 0.000346$$

UA = 2900 Available as opposed to 3120 BTU/hr ^{o}R required

CONCLUSION

This configuration will be adequate, since air flow path approaches counter-flow configuration rather than cross flow as assumed, a conservative assumption, since;

NTU_{max} for counter-flow = 2.4 (page H-135, Ref. II)

$$AU_{req'd} = NTU_{max} \times C_{min}$$

$$AU_{req'd} = 2.4 \times 1040$$

$$= 2500 \text{ BTU/hr. } ^{o}R$$

APPENDIX 5

HIGH TEMPERATURE RADIATOR SYSTEM

Radiation Equation

$$Q = \sigma \varepsilon A \left[\eta T_R^4 - T_S^4 \right]$$

Given

Q = heat rejected by radiator, 18650 BTU/hr.

σ = Stefan-Boltzman Constant
0.1713 x 10^{-8} BTU/hr. ft^2 $^oR^4$

ε = emissivity, 0.9 for TiO_2 point (II)

T_S = effective sink temperature 360^{o}R)
(for a 200-600 mile earth orbit)

η = fin efficiency same as low temperature system, 0.98

Mean Fluid Temperature

$$Q = WC_p \Delta tc$$

Δtc = temperature drop

$$\Delta tc = \frac{Q}{WC_p} = \frac{18650 \text{ BTU/hr.}}{9210 \text{ lb/hr.} \times 263 \text{ BTU/lb}^o\text{R}}$$

$\Delta t_c = 7.7°R.$

$$T_{cm} = T_i \frac{\Delta tc}{2}$$

Given

Q = heat rejected by radiator, 18650 BTU/hr.

C_p = specific heat of FC-75 @ 55^{o}F, 0.263 BTU/lb^{o}R

W = flow rate @ 103 GPM, 9210 lb/hr.

Unknown

T_{cm} = mean fluid temperature, ^{o}R

$$T_{cm} = 523.2\ ^oR \frac{-7.7^oR}{2}$$

$$T_{cm} = 519.3\ ^oR$$

Given

T_i = Inlet fluid temperature 523.2^{o}R

Δtc = 7.7^{o}R

Fluid Flow

$$R_e = \frac{DV_e}{\mu}$$

Unknown

R_e = Reynold's Number

Given

D = inside tube diameter 0.0254 feet

V = fluid velocity @ 10.3 gpm, 1.64 x 10^5 ft/hr.

e = fluid density of FC-75 @ 55^{o}F, 111 lb/ft.^{3}

μ = absolute viscosity of FC-75 @ 55^{o}F,
4.2 lb/ft. hr.

$$\Delta tw = \frac{340 \text{ BTU/hr.}}{1.86 \times 3.14 \times 0.083 \text{ BTU ft/hr. ft}^2\ {}^{o}R \times 35 \quad \frac{2000 \times}{1}\left[\frac{.265 \text{ BTU/lb. } {}^{o}R \times 4.2 \text{ lb/ft. hr}}{0.083 \text{ BTU ft/hr. ft}^2\ {}^{o}R} \times \frac{0.0254 \text{ ft}}{35 \text{ ft.}} \times 1\right]^{1/3}}$$

$$Re = \frac{0.0254 \text{ ft. } \times 1.64 \times 10^5 \text{ ft/hr. } \times 111 \text{ lb/ft}^3}{4.2 \text{ lb/ft. hr.}} = 110500 > 2100 \text{ Turbulent}$$

Make a minimum of 55 parallel passes

$Re = 2000 < 2100$ Laminar flow to reduce pump power

Fluid Convection for laminar flow

$$\Delta t_w = \frac{q}{1.86 \pi\ K_c L \left[\frac{DV_e}{\mu} \times \frac{C_p \mu}{K_c}\right] \times \frac{D}{L}\ 1/3 \left(\frac{\mu}{\mu_w}\right)^{0.14}}$$

Given

q = heat rejected by each radiation (55), 340 BTU/hr.

L = length of tube in each radiator, 35 feet (estimated from preliminary calculations)

K_c = thermal conductivity of fluid FC-75 @ 55°F, 0.083 BTU ft/hr. ft^2 °R

$$\frac{DVP}{\mu} = 2000$$

Cp = specific heat of FC-75 @ 55°F, 0.265 BTU/lb. °R

μ = absolute viscosity of FC-75 @ 55°F, 4.2 lb/ft. hr.

D = inside tube diameter, .0254 feet

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = 1 \text{ for small } \Delta tc$$

Unknown

Δtc = temperature drop from bulk of fluid to tube wall, °R

$\Delta t = 7.42^{\circ}R$

$T_R = T_{cm} - \Delta tw$

Given

T_{cm} = mean fluid temperature, $519.35^{\circ}R$ (Eq. #3)

Δtw = temperature drop from bulk of fluid to tube wall, $7.42^{\circ}R$ (Eq. #5)

Unknown

T_R = temperature of fin at root, $^{\circ}R$

$T_R = 5.19.35^{\circ}R - 7.42^{\circ}R = 511.93^{\circ}R$

Substituting the driven values into Equation #1

$$A = \frac{q}{e\,\epsilon\left[\eta T_R^4 - T_S^4\right]} \qquad \frac{340 \text{ BTU/hr.}}{0.1713 \times 10^{-8} \text{ BTU/hr. ft}^2\ {}^{\circ}R^4 \times 0.9 \left[0.98(511.93^{\circ}R)^4 - (360^{\circ}R)^4\right]}$$

$A = 4.83 \text{ ft}^2$

x 55 parallel radiators

265 ft^2 total radiator area

APPENDIX 6

EVAPORATOR

EVAPORATOR EQUATION

$Q = wC_p \Delta t$

UNKNOWN

Δt = change in fluid temperature, $^{\circ}R$

KNOWN

Q = heat transfer, 18650 $\frac{BTU}{hr.}$

W = flow weight, 9210 $\frac{lb.}{hr.}$

C_p = specific heat @ 55°F;.263 $\frac{BTU}{lb.\ ^{\circ}R}$

$$\Delta t = \frac{Q}{wC_p} = \frac{18650\ BTU/hr.}{9210\ lb./hr.\ \ 0.263\ BTU/lb^{\circ}R} = 7.7^{\circ}R$$

Fluid (FC-75) Flow Inside Shell

Section Area Shell =	28.2 in^2
Area Tubes	4.0 in^2
	24.2 in^2
	144 in^2/ft^2 = 0.168 ft^2

$$\text{Flow Rate} = \frac{9210\ lb/hr.}{111.3\ lb/ft^3} = 82.8\ ft^3/hr.$$

$$\text{Velocity} = \frac{\text{Rate}}{\text{Area}} = \frac{82.8\ ft^3/hr.}{0.168\ ft^2} = \frac{493\ ft/hr.}{3600\ sec/hr.} = 0.137\ ft./sec.$$

Circumference shell = 1.570 ft.

$$\text{Circumference tubes} = \frac{3.53\ ft.}{5.10\ ft.} = \text{wetted perimeter}$$

$$\text{Hydraulic Radius} = \frac{\text{Area}}{\text{Wetted perimeter}} = \frac{0.168\ ft^2}{5.1\ ft} = 0.033\ ft.$$

FLUID FLOW

$$Re = \frac{4R_h V \rho}{\mu}$$

Unknown

Re = Reynold's Number

Known

R_h = hydraulic radius, 0.033 ft.

V = fluid velocity, 0.493 ft/hr.

e = fluid density @ 55°F, 111.3 lb/ft^3

μ = absolute viscosity, 4.2 lb/ft. hr.

$$Re = \frac{4 \times 0.033 \text{ ft.} \times 493 \text{ ft/hr.} \times 111\ 3 \text{ lb/ft}^3}{4.2 \text{ lb/ft. hr.}} = 1815$$

Re = 1815 < 2100 Laminar Flow

FLUID CONVECTION for Laminar Flow

$$\frac{h_m 4R_h}{k} \frac{(\mu_s)^{0.14}}{(\mu)} = 1.86 \text{ Re} \frac{(Cp\,\mu)}{(k)} \frac{(4R_h)}{(L)}^{1/3}$$

Unknown

h_m = film coefficient for FC-75

Known

R_H = hydraulic radius = 0.033 ft.

k = thermal conductivity = $\frac{0.083 \text{ BTU ft.}}{\text{Hr. ft}^2 \text{ °R}}$

$\frac{(\mu_s)^{0.14}}{(\mu)}$ = 1 for small Δt

Re = Reynold's Number 1815

Cp = specific heat, $\frac{0.263 \text{ BTU}}{\text{lb. °R}}$

w = absolute viscosity; $\frac{4.2 \text{ lb.}}{\text{ft. hr.}}$

L = length of tube, 6 ft. (estimated from preliminary calculations)

$$\frac{hm \times 4 \times 0.033 \text{ ft}}{0.083 \text{ BTU ft/hr ft}^2 \, {}^oR} \times 1 = 1.86 \left[1815 \frac{(0.263 \text{ BTU/lb}^oR \times 4.2 \text{ lb/ft. hr.})}{(0.083 \text{ BTU ft/hr. ft}^2 \, {}^oR)} \frac{(4 \times 0.033 \text{ ft})}{(6 \text{ ft.})} \right]^{1/3}$$

$h_m = 27$ BTU/hr. ft^2 ^oR

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{h_{f}s} + \frac{x}{k} + \frac{R}{hms} + \frac{R}{hm}$$

<u>Known</u>

hf = film coefficient for refrigerant FREON 12, 770 BTU/ft^2 hr ^oR

hfs = scale coefficient for refrigerant, 2000 BTU/ft^2 hr. ^oR

hm = film coefficient for FC-75, 27 BTU/ft^2 hr ^oR

h_{ms} = scale coefficient for FC-75, 2000 BTU/ft^2 hr. ^oR

x = wall thickness, 0.003 feet

k = thermal conductivity of wall, 220 BTU/ft^2 hr. ^oR (copper)

R = Ratio of inside to outside surface area of tubes = $\frac{D_o}{D_1} = \frac{0.375}{0.305} = 1.23$

<u>Unknown</u>

U = overall heat transfer factor BTU/ft^2 hr. ^oR

$$\frac{1}{U} = \frac{1}{\text{BTU/ft}^2 \text{ hr. } {}^oR} \left[\frac{1}{770} + \frac{1}{2000} + \frac{0.003}{220} + \frac{1.23}{2000} + \frac{1.23}{27} \right]$$

U = 21.3 BTU/ft^2 hr. ^oR

$$Eff. = \frac{t_{mi} - t_{mo}}{t_{mi} - t_{fi}} \quad (Ref\ 2)$$

Unknown

$Eff.$ = effectiveness of heat exchanger

Known

t_{mi} = temperature FC-75 in, 63.2°F

t_{mo} = temperature FC-75 out, 63.2 - 7.7 = 55.5°F

t_{fi} = temperature refrigerant in, 40°F

$$Eff. = \frac{63.2 - 55.5}{63.2 - 40} = \frac{7.7}{23.2} = 0.332$$

$Eff. = 0.332 \therefore NTU_{max.} = 0.4$ for a cross-flow exchanger where $\frac{Cmin.}{Cmax.} = Q$

$NTU_{max.} = AU/C_{min.}$

Unknown

A = heat transfer area ft^2

Known

$NTU_{max} = 0.4$

$U = 21.3\ BTU/ft^2\ hr.\ ^oR$

$C_{min} = wCp = 2420\ BTU/hr.\ ^oR$

where w = flow weight, 9210 lb/hr.

Cp = Specific heat, 0.263 BTU/lb. ^oR

$$A = \frac{NTU_{max}\ C_{min}}{U} = \frac{0.4 \times 2420\ BTU/hr.\ ^oR}{21.3\ BTU/ft^2\ hr.\ ^oR}$$

$A = 45.5\ ft^2$

APPENDIX 7

PARAMETRIC STUDY OF RADIATOR THERMAL ENVIRONMENT

Purpose

To determine the extreme surface temperatures for the various locations and orientations which a radiative surface or fin would encounter on a satellite orbiting the earth at a 200 -600 mile altitude. A finned radiator in space in the vicinity of the earth's thermal effects is also explained.

Problem Approach

An analysis of five basic plate configurations (Figure 58) was conducted to determine stabilized plate temperatures for the various 200 and 600 mile radius orbit positions from the earth surface, and one position considering solar thermal effects only. This analysis related stabilized plate temperatures, surface absorptivity and emissivity, plate orientation, plate configuration, and the effects of distributed energy input (refrigerator load-in addition to earth and solar radiant effects). The resulting derived relations were evaluated for two extreme plate environmental conditions. These conditions consisted of:

Minimum Thermal Environment	Maximum Thermal Environment
Solar Flux (S_s) = 417 BTU/ft^2-hr	Solar Flux (S_s) = 474 BTU/ft^2-hr
Earth Flux (S_E) = 60.6 BTU/ft^2-hr	Earth Flux (S_E) = 81.1 BTU/ft^2-hr
Earth Albedo Factor = 0.20	Earth Albedo Factor = 0.52

The analysis was based on the following assumptions:

The earth was considered as a perfect sphere.
Plate edge thermal effects were neglected.
Emissivity of Earth = 1.0.
Space Radiation Temperature = 0^{o}R.
Solar radiant flux considered to be parallel rays.
Plate thermal capacity neglected. (steady state)
Thermal gradients through plate were neglected.
Insulation assumed perfect (where insulation is indicated).
Surface "A" conductively insulated from surface "K" for the perpendicular plate configurations.

Analysis and Results

I. The equilibrium temperature of a plate having one side perfectly insulated (Figure 58 Conf. 2) and subjected to earth radiant flux is given by:

$$T_A = \left(\frac{F_{A-E}\ S_E}{\sigma}\right)^{1/4} \qquad (1)$$

where: T_A = equilibrium temperature of plate

S_E = earth radiant flux

σ = Stefan-Boltzmann Constant

F_{A-E} = configuration factor between plate and earth (function of altitude and plate orientation).

This relation is shown in Figure 59 for the two plate environmental conditions defined earlier.

II. The equilibrium temperature of a plate having one side insulated (Figure 58 Conf. 2) and longitudinal axis is perpendicular to the sun's radiant flux is given by:

$$T_A = \left(\frac{\alpha_A S_s F_{A-S}}{\epsilon_A \sigma} \right)^{1/4} \tag{2}$$

where: α_A = solar absorptivity of plate surface

ϵ_A = emissivity of plate surface

S_s = solar radiant flux

F_{A-S} = configuration factor between plate and the sun

This relation is shown as Figure 59 for three values of α_A / ϵ_A and for the two plate environmental conditions defined earlier.

III. The equilibrium temperature of a plate having one side insulated (Figure 58 Conf. 2 with its longitudinal axis perpendicular to sun flux) and which is located between, and in line with, the centers of the earth and sun is given by:

$$T_A = \left[\frac{\left(\frac{\alpha_A}{\epsilon_A} \right) S_s \left(F_{A-S} + a\, F_{A-E} \right) + F_{A-E}\, S_E}{\sigma} \right]^{1/4} \tag{3}$$

where: a = earth albedo factor

This relation is shown in Figure 60 for the two plate environmental conditions defined earlier.

IV. The equilibrium temperature of an uninsulated plate (Figure 58 Conf. 1 - longitudinal axis perpendicular to sun flux) which is located between, and in line with, the centers of the earth and sun is given by:

$$T_A = \left[\frac{(\alpha_A / \epsilon_A)\, S_s (F_{A-S} + a\, F_{A-E}) + F_{A-E}\, S_E}{2\,\sigma} \right]^{1/4} \tag{4}$$

This relation is shown as Figure 61 for the two plate environmental conditions defined earlier.

V. The equilibrium temperature of a plate having one side insulated (Figure 58 Conf. 2 with its longitudinal axis perpendicular to sun flux), and which is located at the perpendicular intersection of a normal from the earth surface with the sun's radiant flux is given by:

$$T_A = \left[\frac{(\alpha_A / \epsilon_A)\, S_s F_{A-S} + S_E F_{A-E}}{\sigma} \right]^{1/4} \tag{5}$$

This relation is shown as Figure 62 for the two plate environmental conditions defined earlier.

VI. The equilibrium temperature of an uninsulated plate (Figure 58 Conf. 1-longitudinal axis perpendicular to sun flux), and which is located at the perpendicular intersection of a normal from the the earth's surface with the sun's radiant flux, is given by:

$$T_A = \left[\frac{\alpha_A / \epsilon_A)\, S_s F_{A-S} + S_E F_{A-E}}{2\sigma} \right]^{1/4} \tag{6}$$

Notes: This neglects the slight albedo effect at this location. Estimated error of < 3% for orbit radius ≤ 600 mi.

The above relation is shown as Figure 63 for the two plate environmental conditions defined earlier.

VII. The equilibrium temperature of a plate having one side insulated (Figure 58- Conf. 2-longitudinal axis perpendicular to sun flux and coincidental with an extension of the earth's radius), and which is located at the intersection of a normal to the earth's surface with the sun's radiant flux is given by the general expression (5).

This relation is shown as Figure 64 for the two plate environmental conditions defined earlier.

VIII. For the same plate orientation and location described in VII, except plate is not insulated (Figure 58 Conf. 1), the plate equilibrium temperature is given by the general expression (6).

This relation is shown as Figure 65 for the two plate environmental conditions defined earlier.

IX. For the same plate configuration and location described in VII, except that the plate receives an additional distributed energy input of 15 BTU/ft^2-hr, the plate equilibrium temperature is given by:

$$T_A = \left[\frac{(\alpha_A / \epsilon_A)\, F_{A-S}\, S_s + F_{A-E}\, S_E + Q}{\sigma} \right]^{1/4} \tag{7}$$

where: $Q = 15$ BTU/ft^2-hr

This relation is shown as Figure 66 for the two plate environmental conditions defined earlier.

X. The equilibrium temperature of a plate having one side insulated (Figure 58 - Conf. 2-longitudinal axis perpendicular to sun flux) which is located on the tangency of the sun flux to the earth surface (on the "dark side" of earth) is given by the general expression (5).

This relation is shown as Figure 67 for the two plate environmental conditions defined earlier.

XI. For the same plate configuration and location described in X, except that the plate receives an additional distributed energy input of 15 BTU/ft^2 -hr, the plate equilibrium temperature is given by expression (7). This relation is shown as Figure 68 for the two environmental conditions defined earlier.

XII. The equilibrium temperature of a plate fixed perpendicular to another plate (conductively insulated from each other) is dependent upon the relative dimensions of the plates, upon the distance between them and upon their thermal environment. The temperature relation for such a configuration (Figure 58 Conf. 3, 4, and 5) located in line with, and between, the centers of the earth and sun is shown as Figure 69. The thermal radiative effect of "A" upon "K" was considered negligible, but the effect of "K" on "A" was considered. The resulting expression being:

$$T_A = \left[\frac{S_s \,(\alpha_A/\epsilon_A)\,(F_{A-S} + a\,F_{A-E}) - S_E\,F_{A-E}}{2\,\sigma\,(1 + \epsilon_K\,F_{A-K})} + \epsilon_K\,F_{A-E}\,T_K^{\,4} \right]^{1/4} \qquad (8)$$

where: F_{A-K} = configuration factor between "A" and "K"

ϵ_K = coefficient of emissivity of plate "K" surface

T_K = Temperature of plate "K"

This relation is shown as Figure 69 for three different plate "K" & "A" configurations with the two plate environmental conditions defined earlier.

XIII. This same general relation (8) is shown as Figure 70 for the two plate configurations (Figure 58 conf. 3) located at the perpendicular intersection of the sun flux and a normal to the earth's surface. The figure shows the effects of two orientations of the plate configuration with the respect to the sun as well as the effects of the two earlier defined extremes of thermal environment.

XIV. The equilibrium temperature for this same plate configuration (described in XIII) and located at the perpendicular intersection of the sun flux with a normal to the earth's surface, but with rotation about the projected intersection of the two plates as shown on Figure 71. The axis of rotation for this case is perpendicular to a normal from the earth's surface.

XV. The equilibrium temperature for the configuration and location described in XIV, except with the axis of rotation being in line with a normal to the earth's surface, is shown on Figure 72.

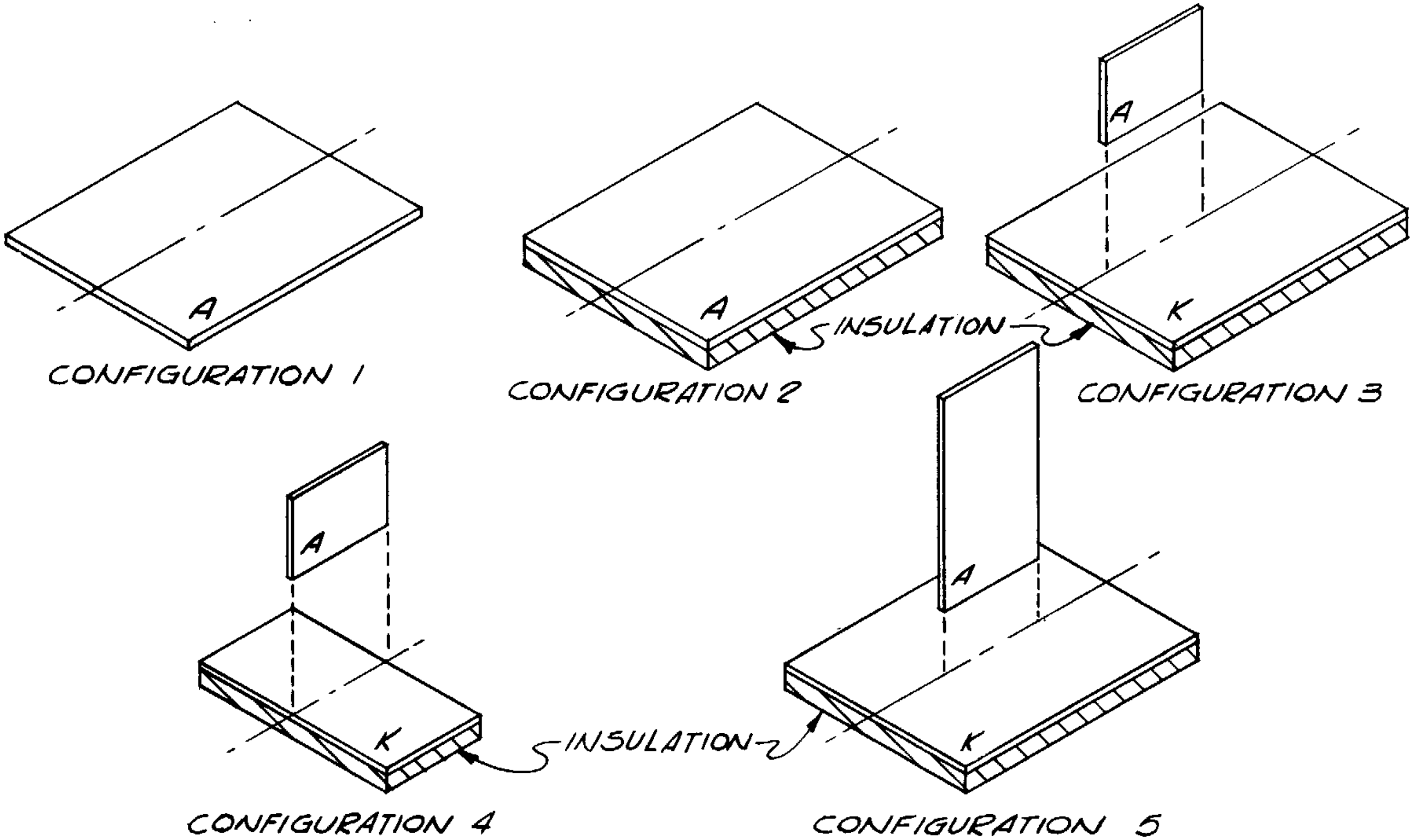


FIGURE 58 - PLATE CONFIGURATIONS

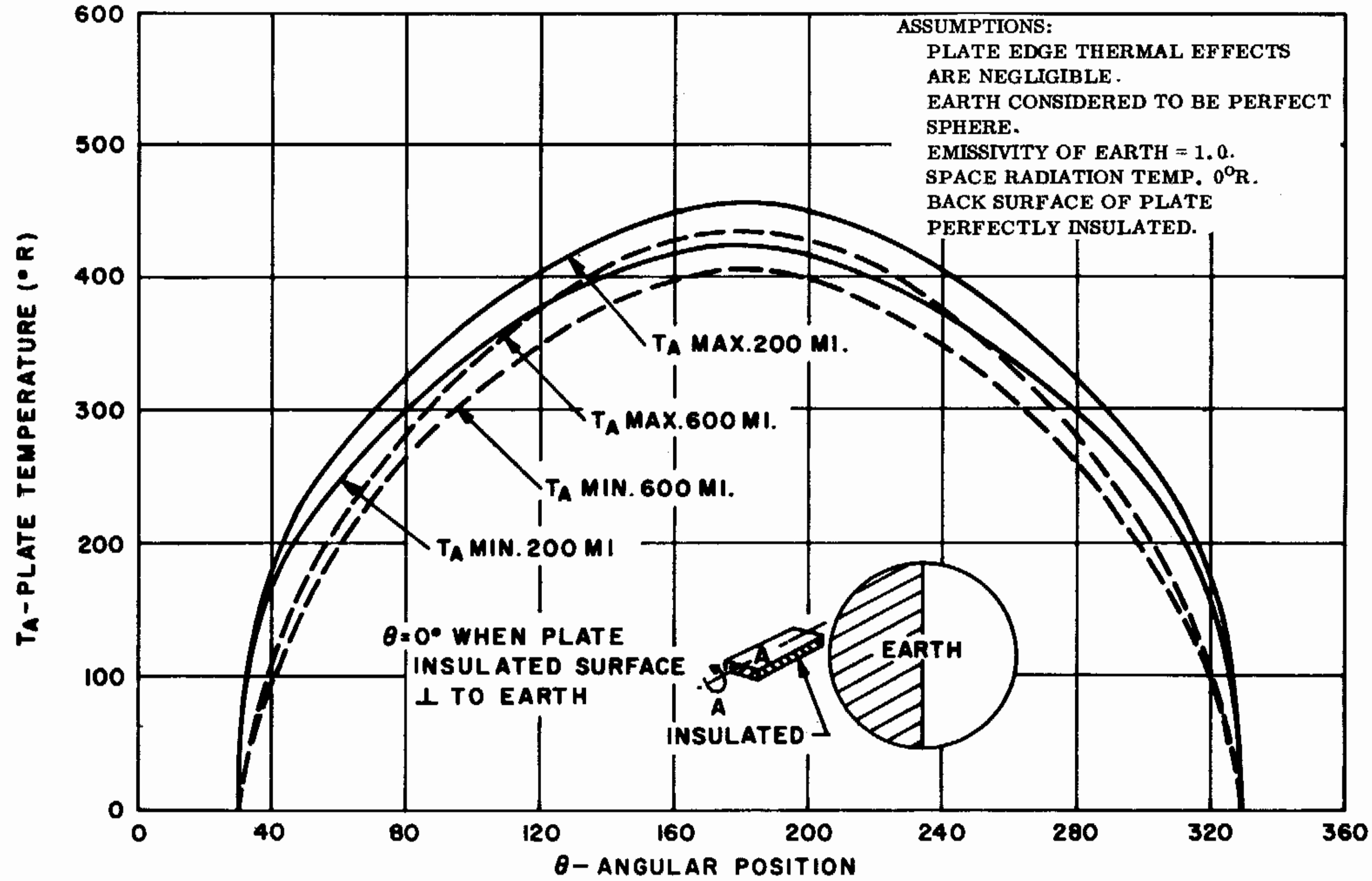


Figure 59. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth Radiation Effects

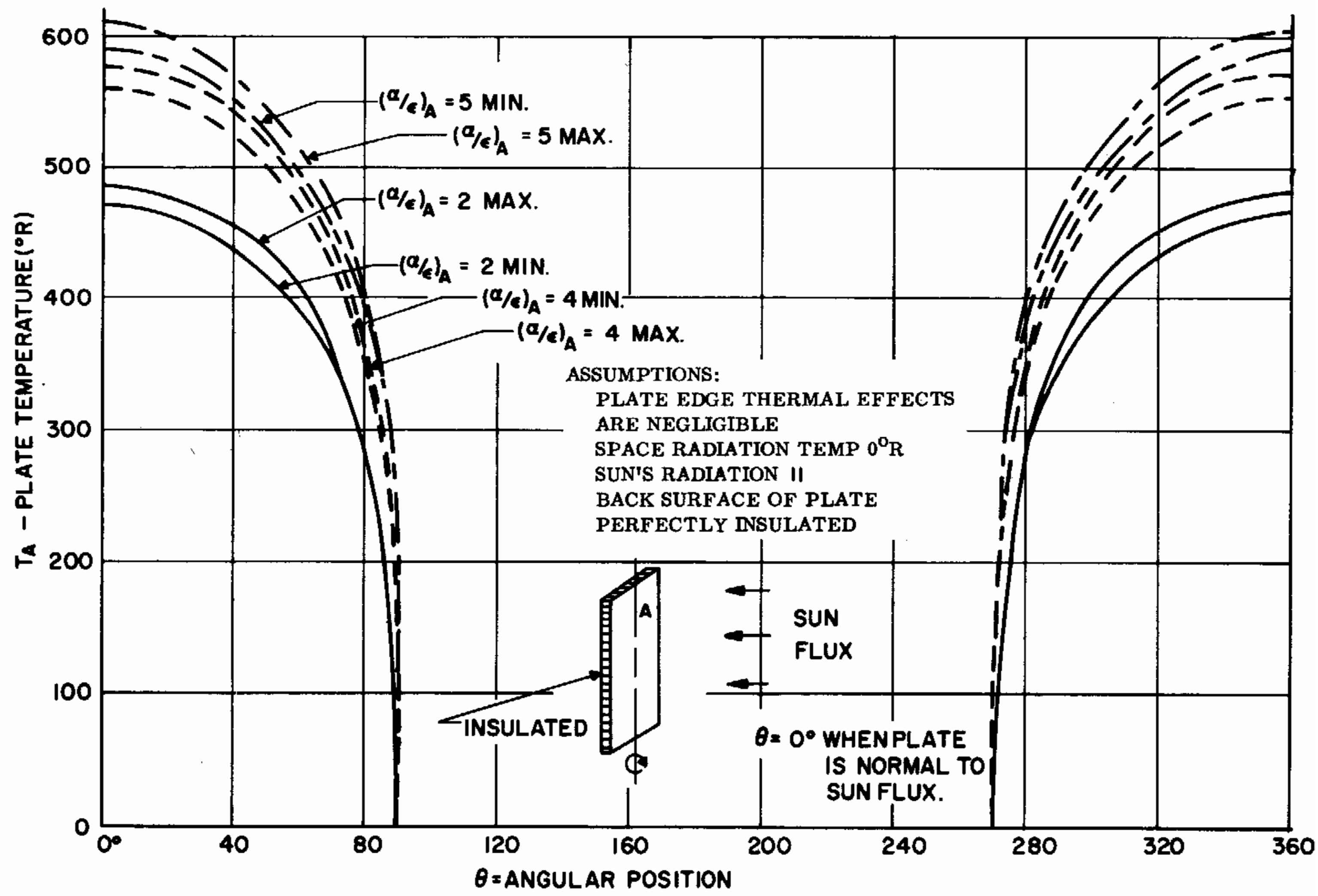


Figure 60. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Sun Radiation Effects

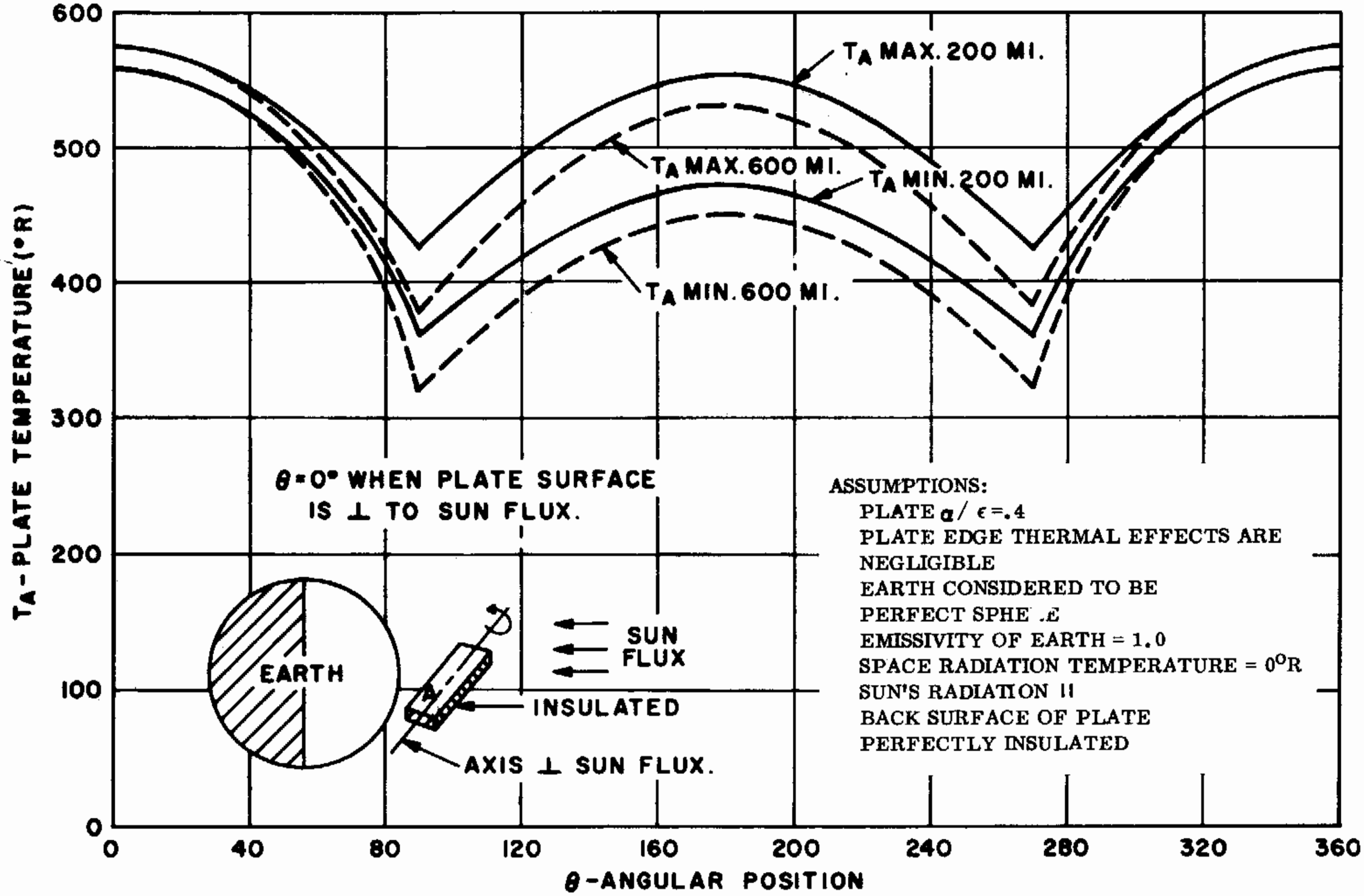


Figure 61. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

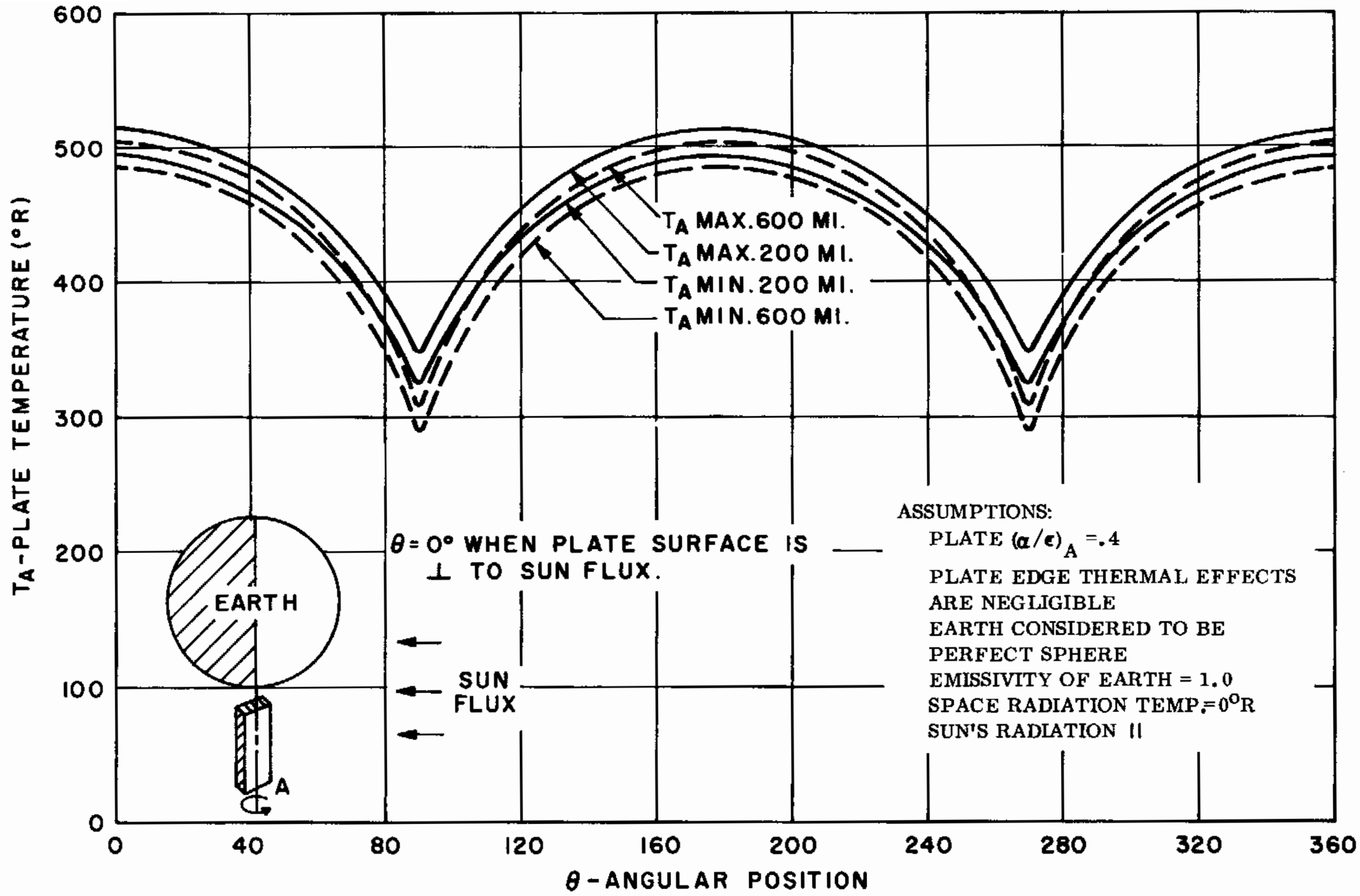


Figure 62. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

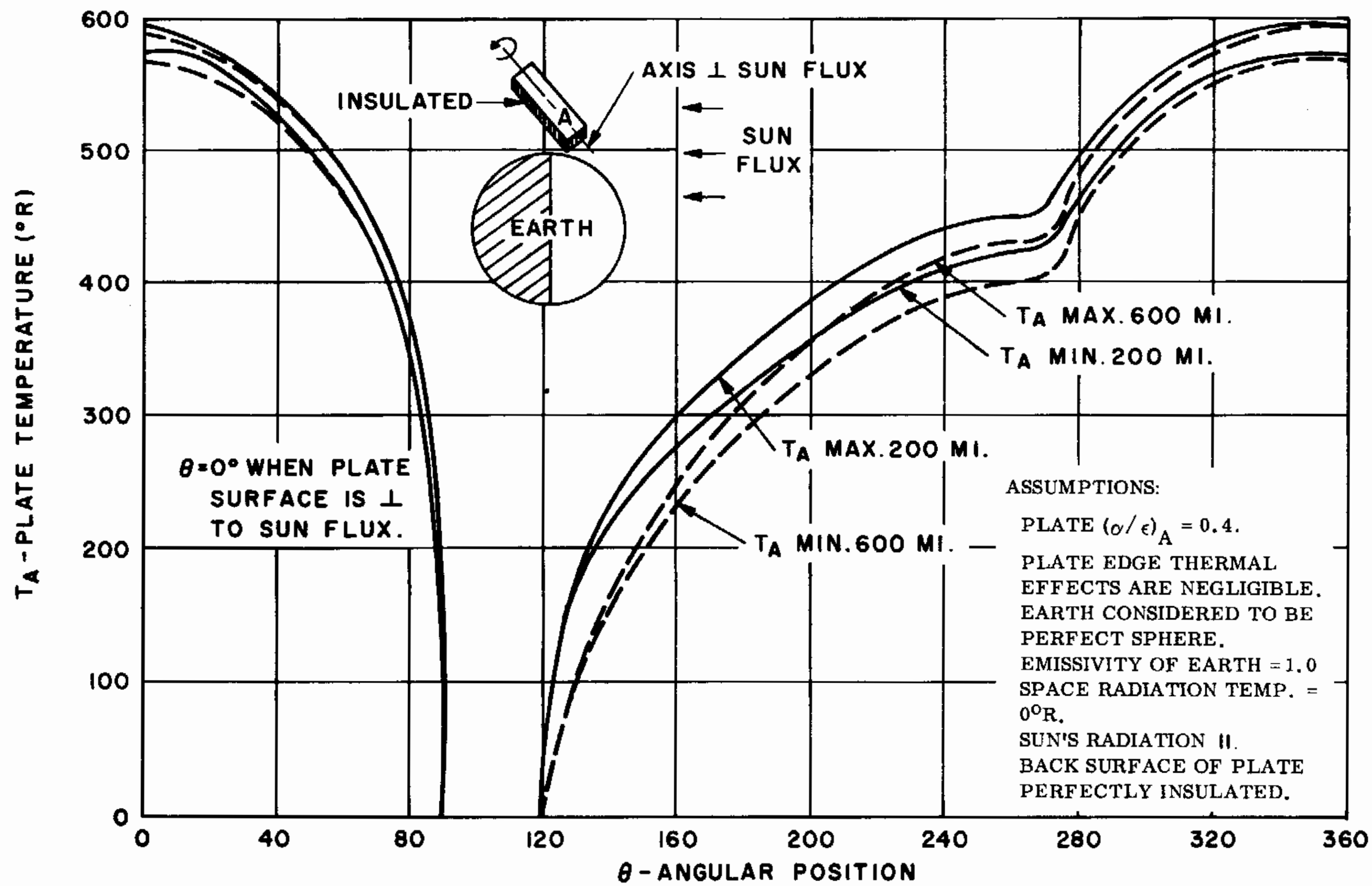


Figure 63. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

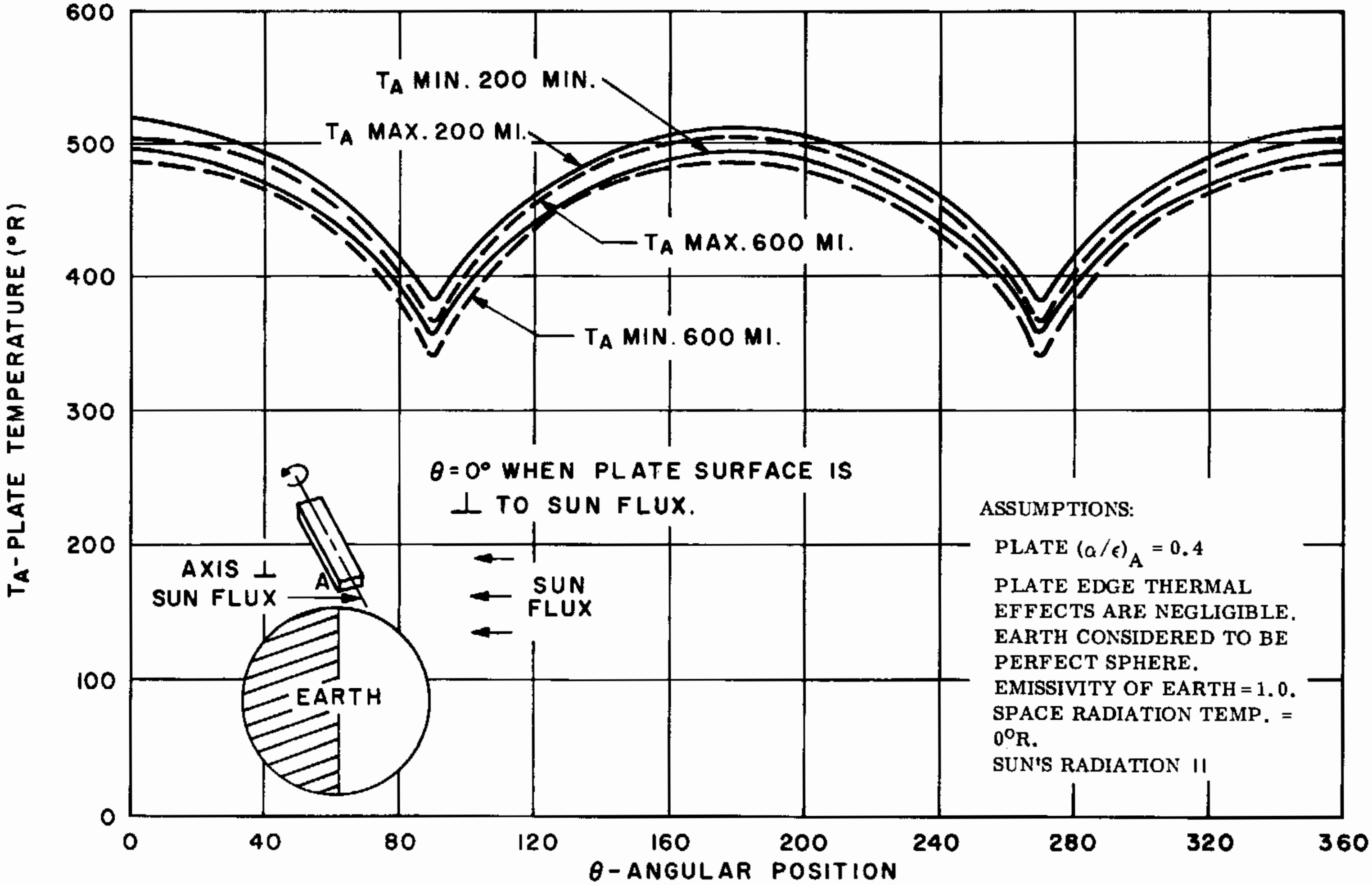


Figure 64. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

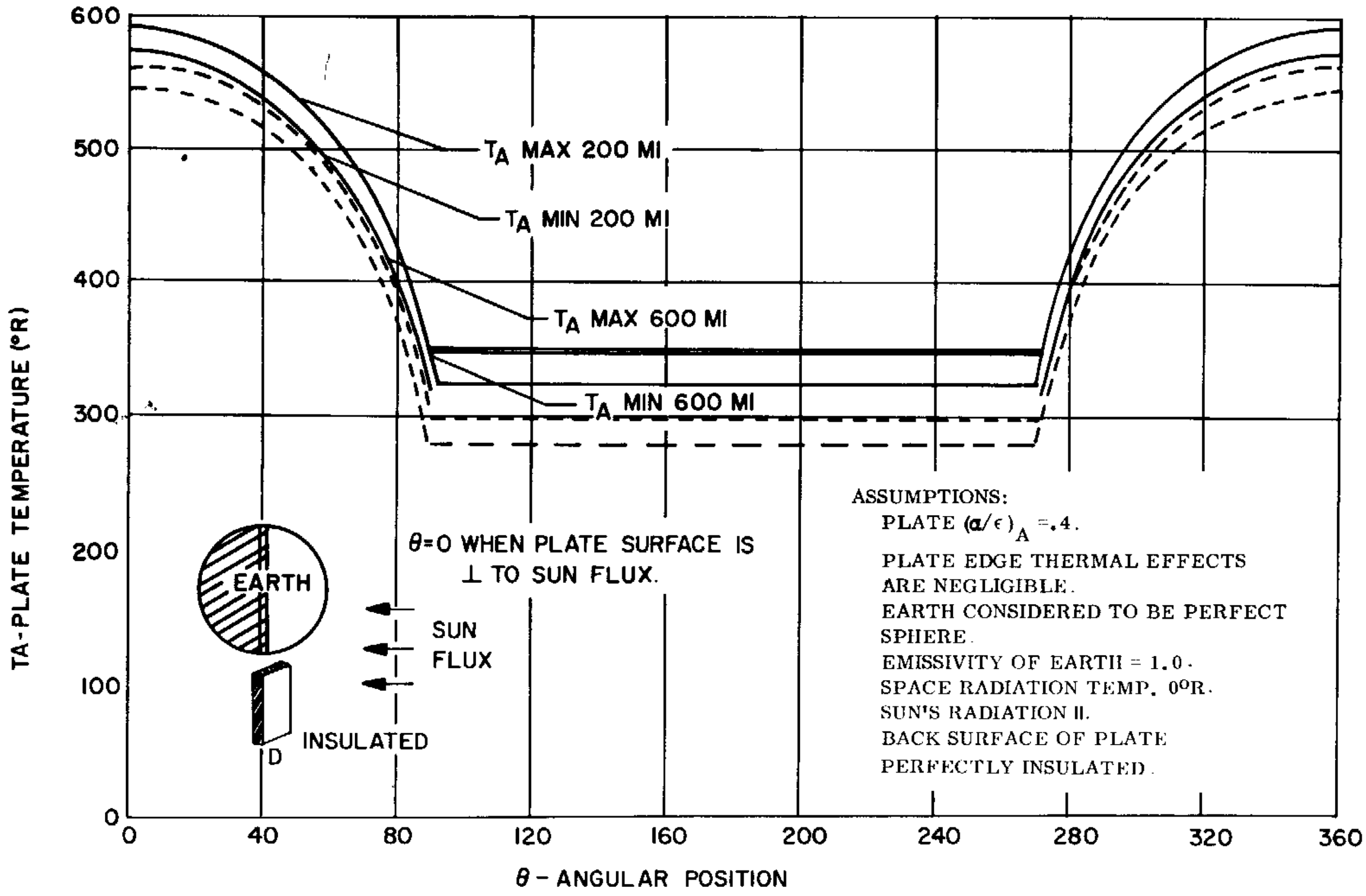


Figure 65. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

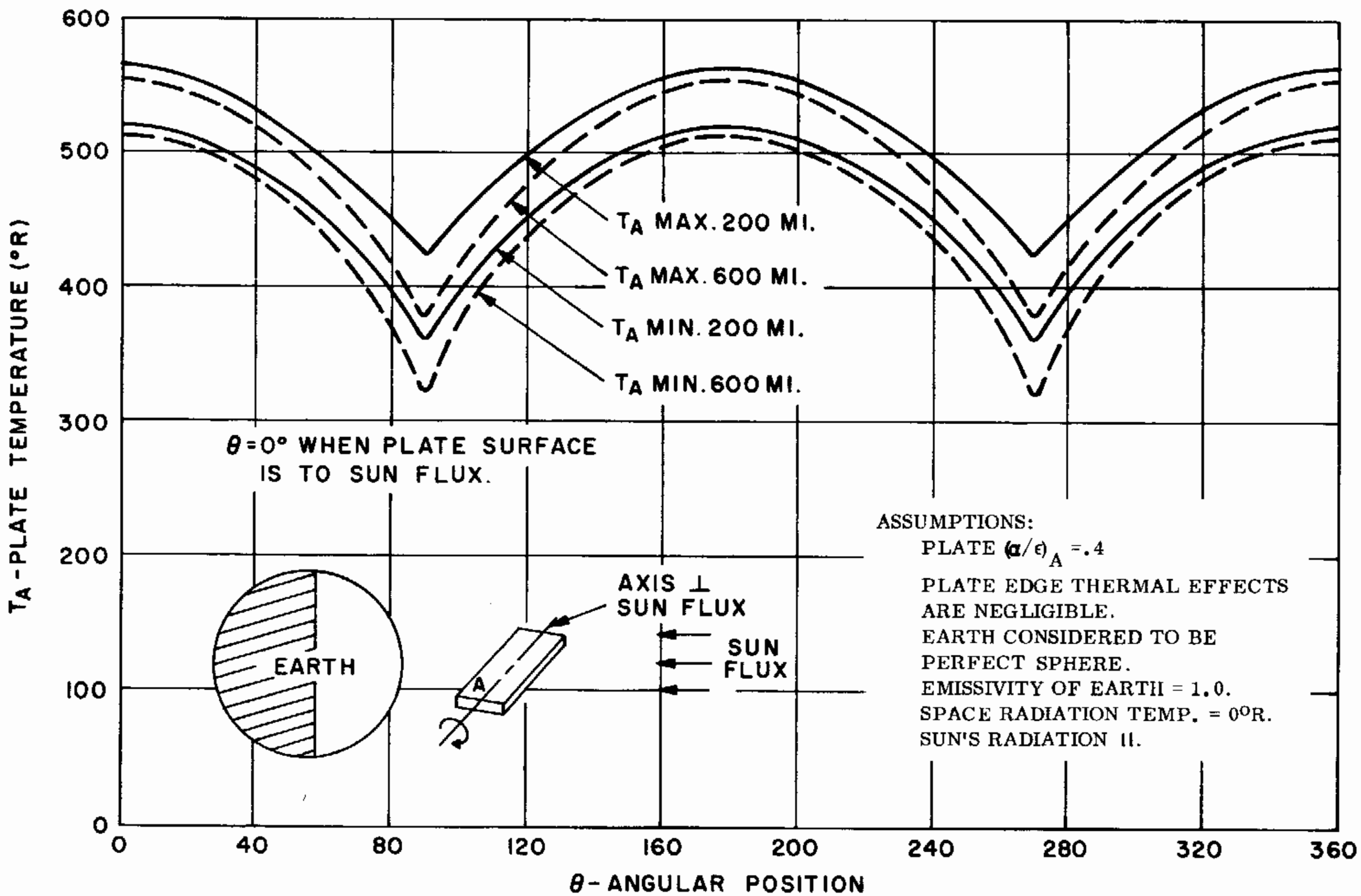


Figure 66. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

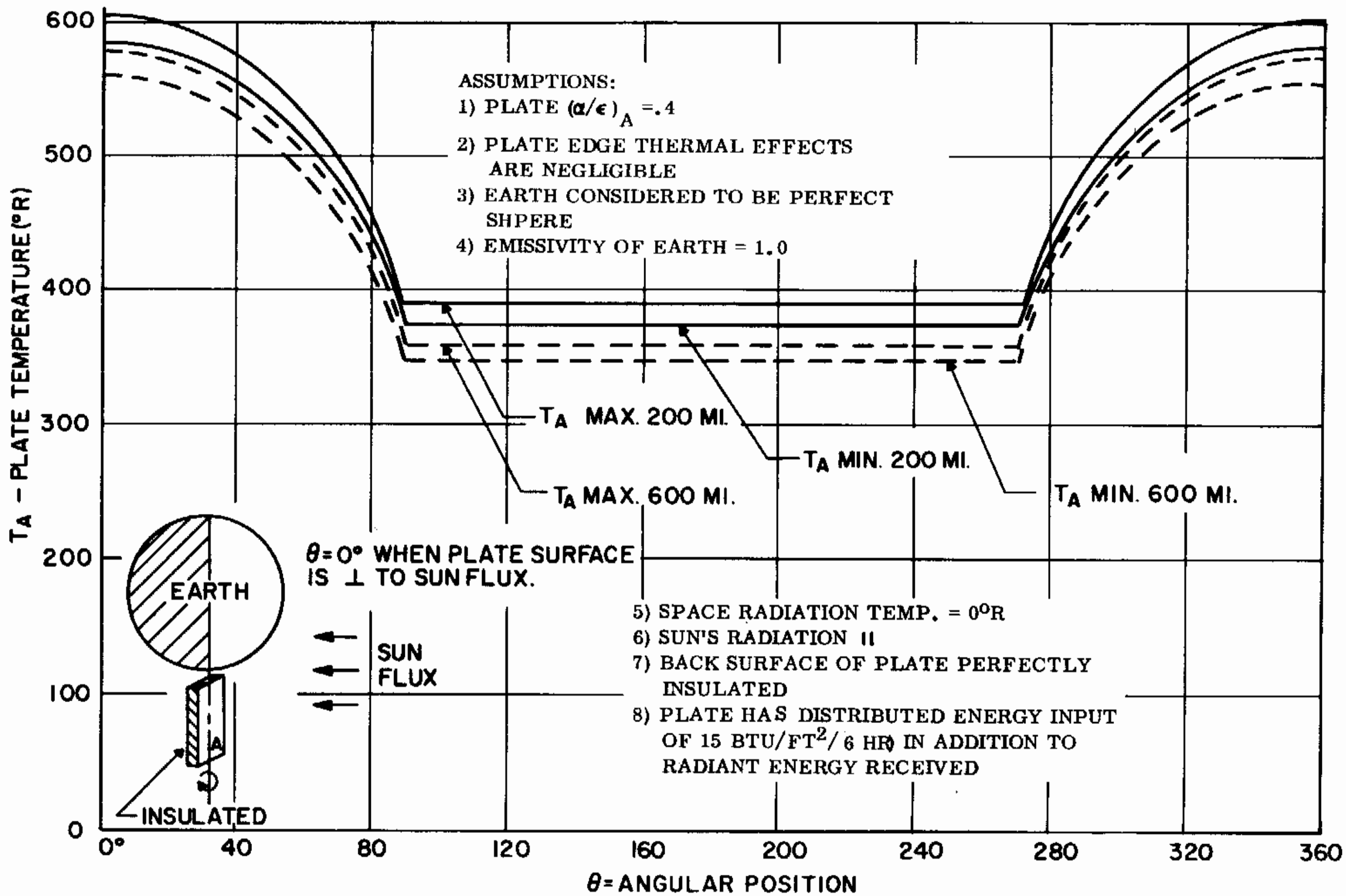


Figure 67. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

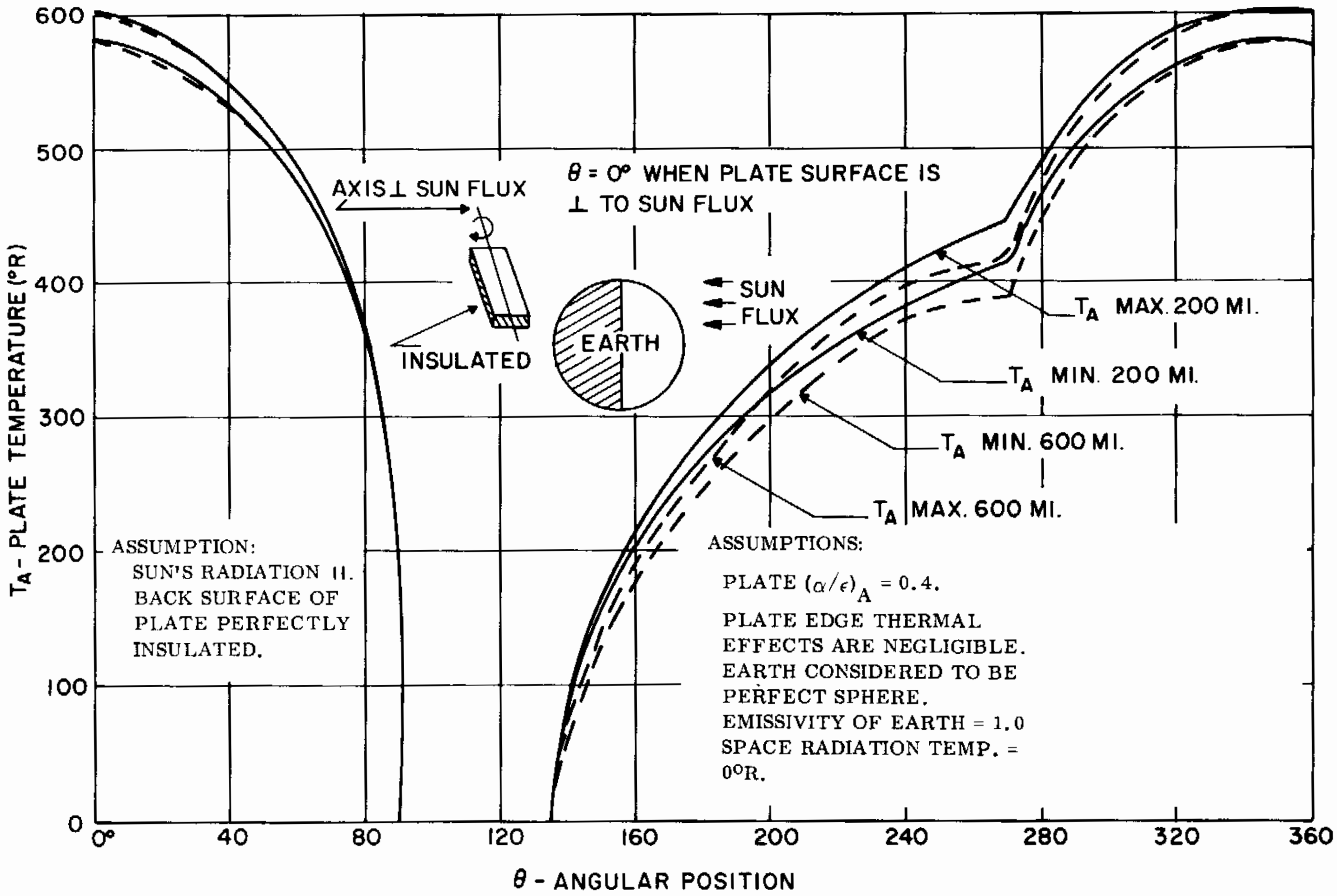


Figure 68. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

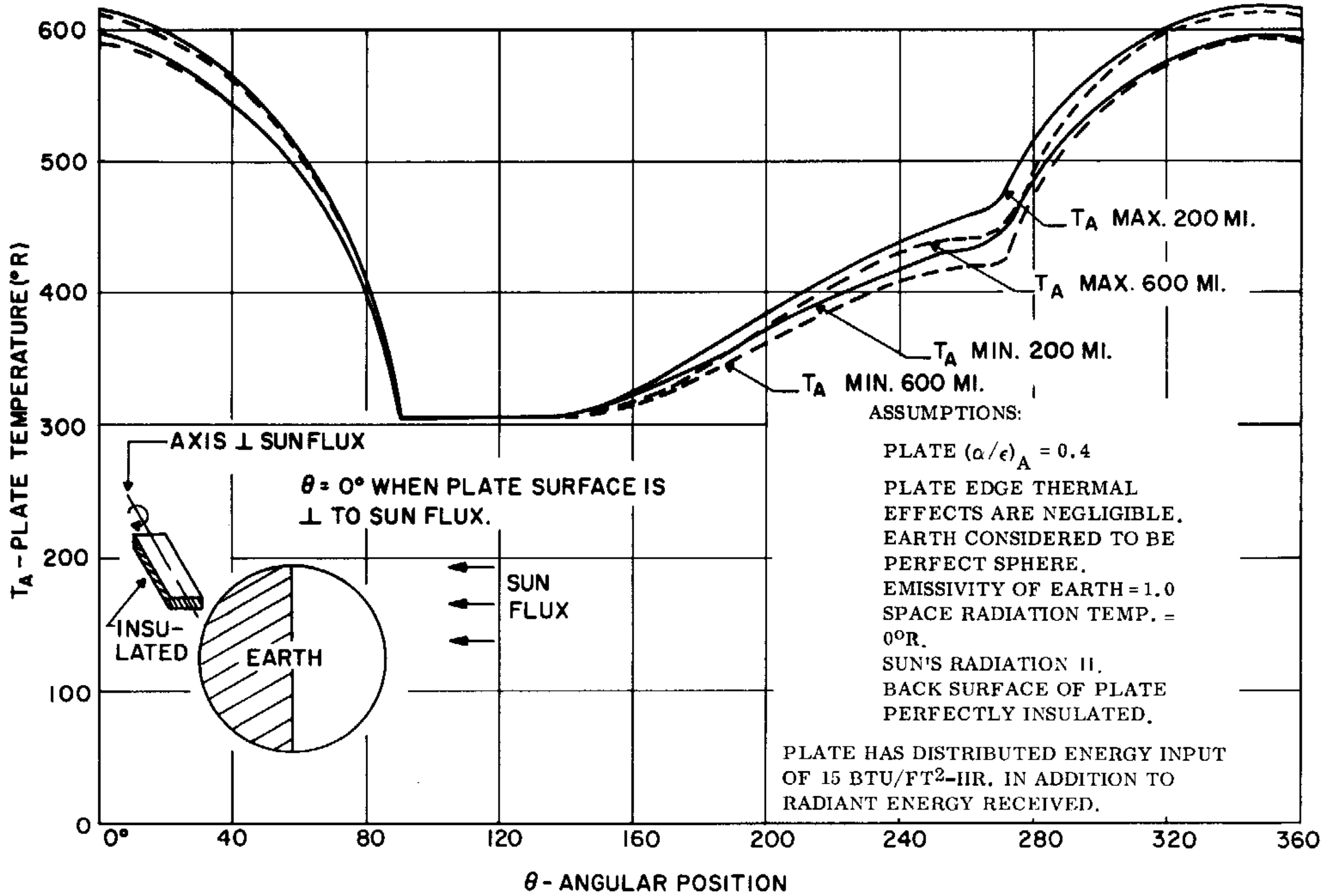


Figure 69. Thin Plate Equilibrium Temperature as a Function of Angular Position Relative to Earth and Sun Radiation Effects

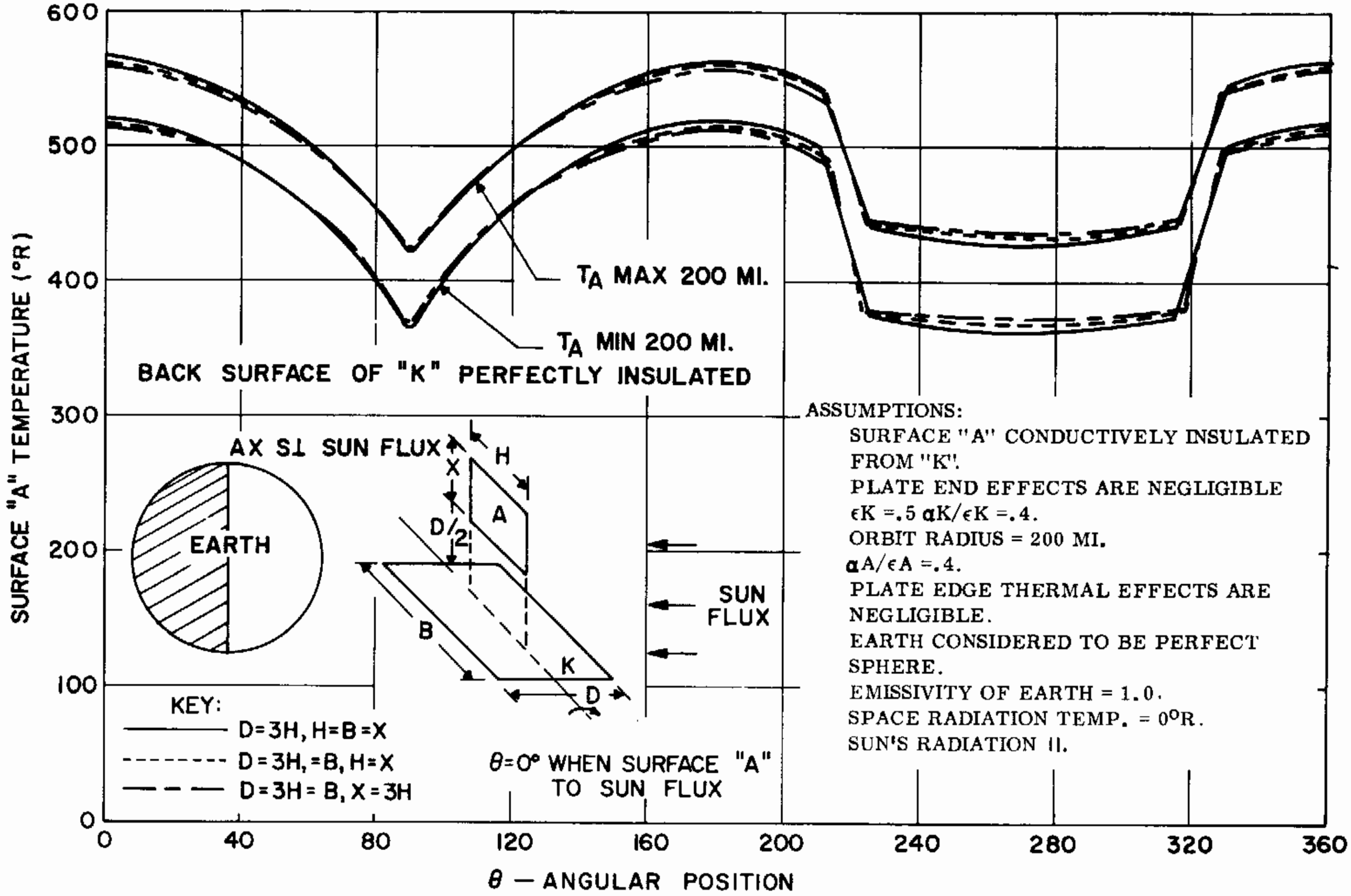


Figure 70. Equilibrium Temperatures of Two Thin Perpendicular Plates as a Function of Position Relative to Earth and Sun Radiation Effects

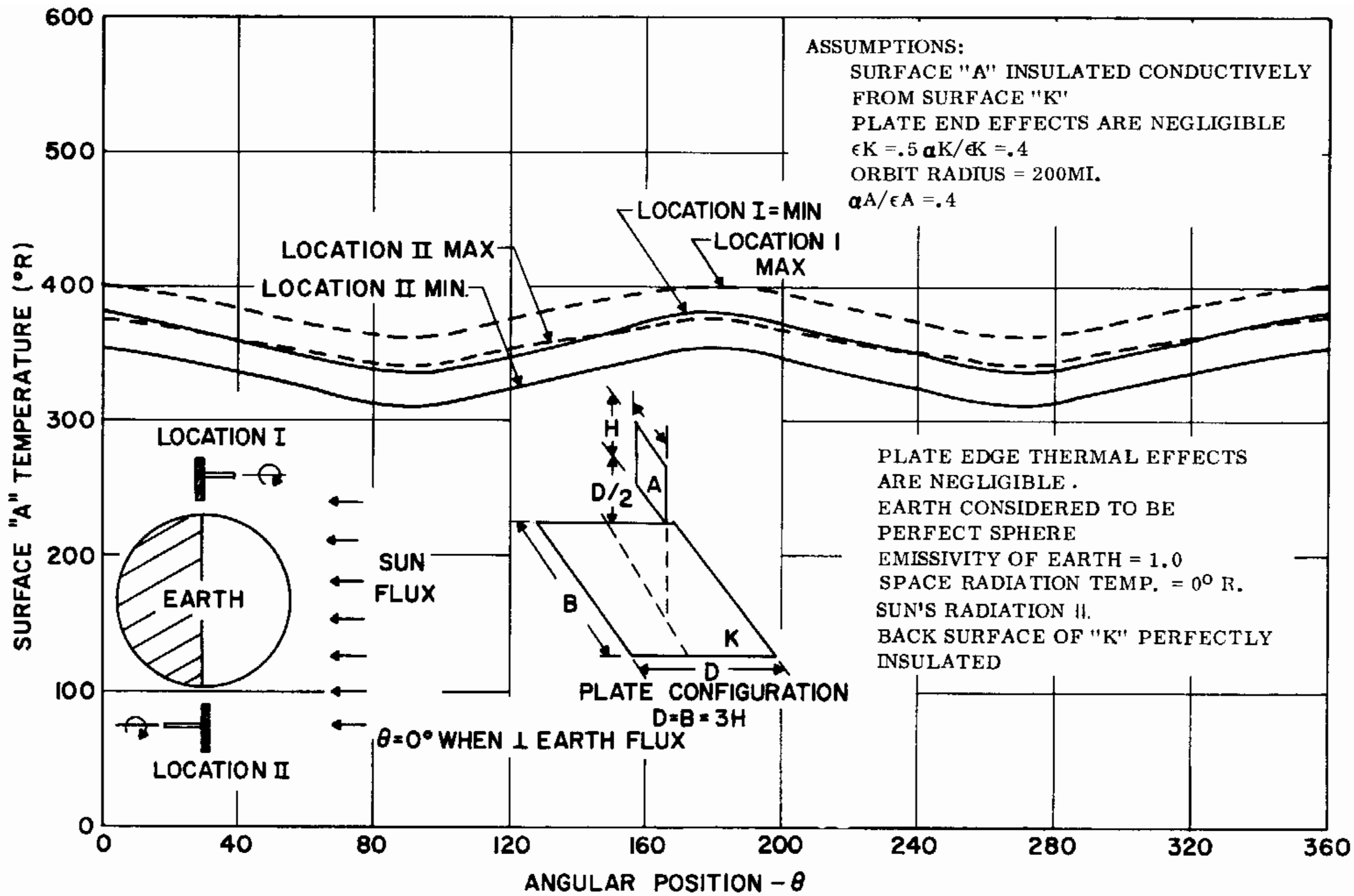


Figure 71. Equilibrium Temperatures of Two Thin Perpendicular Plates as a Function of Position Relative to Earth and Sun Radiation Effects

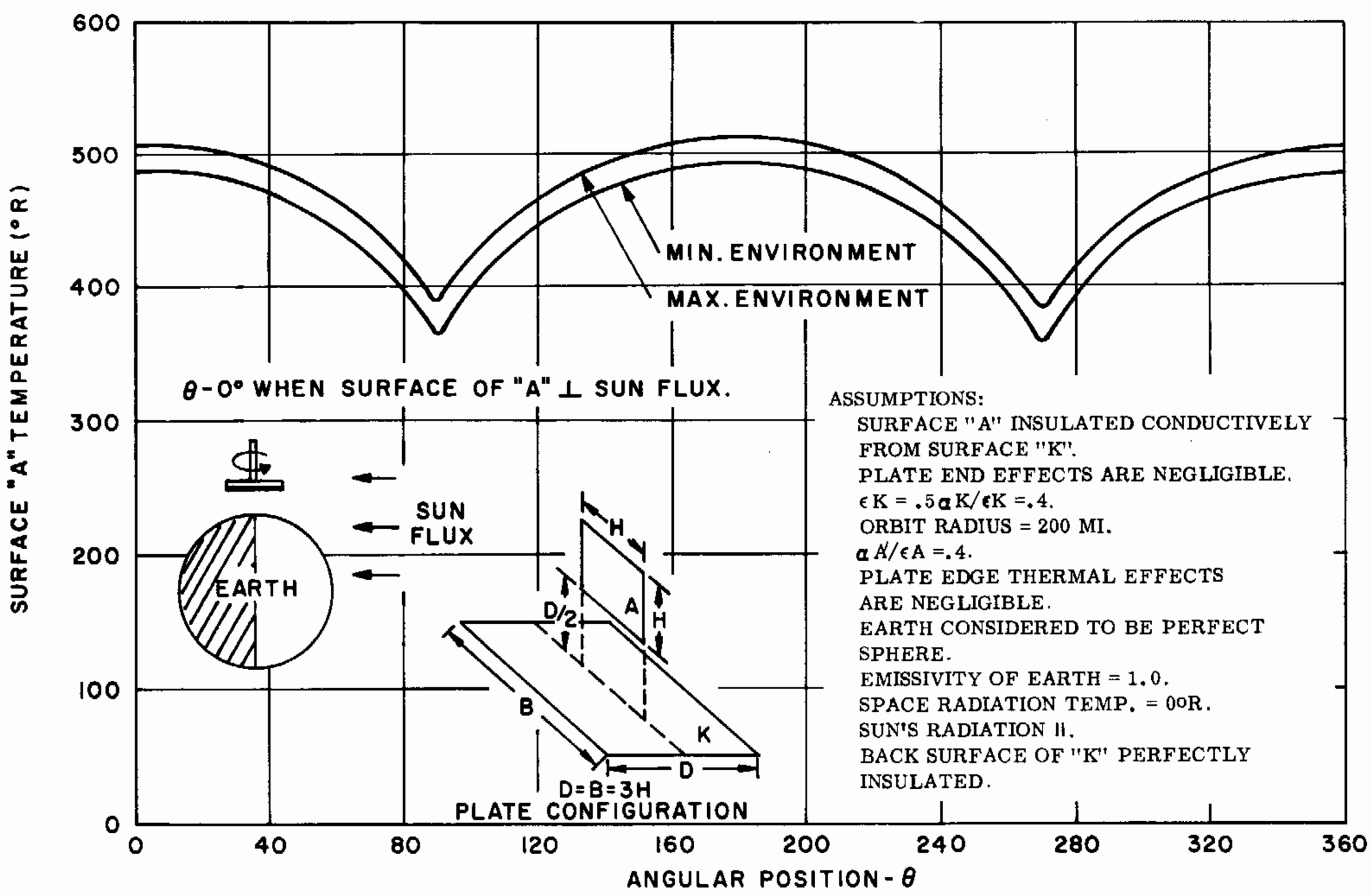


Figure 72. Equilibrium Temperatures of Two Thin Perpendicular Plates as a Function of Position Relative to Earth and Sun Radiation Effects

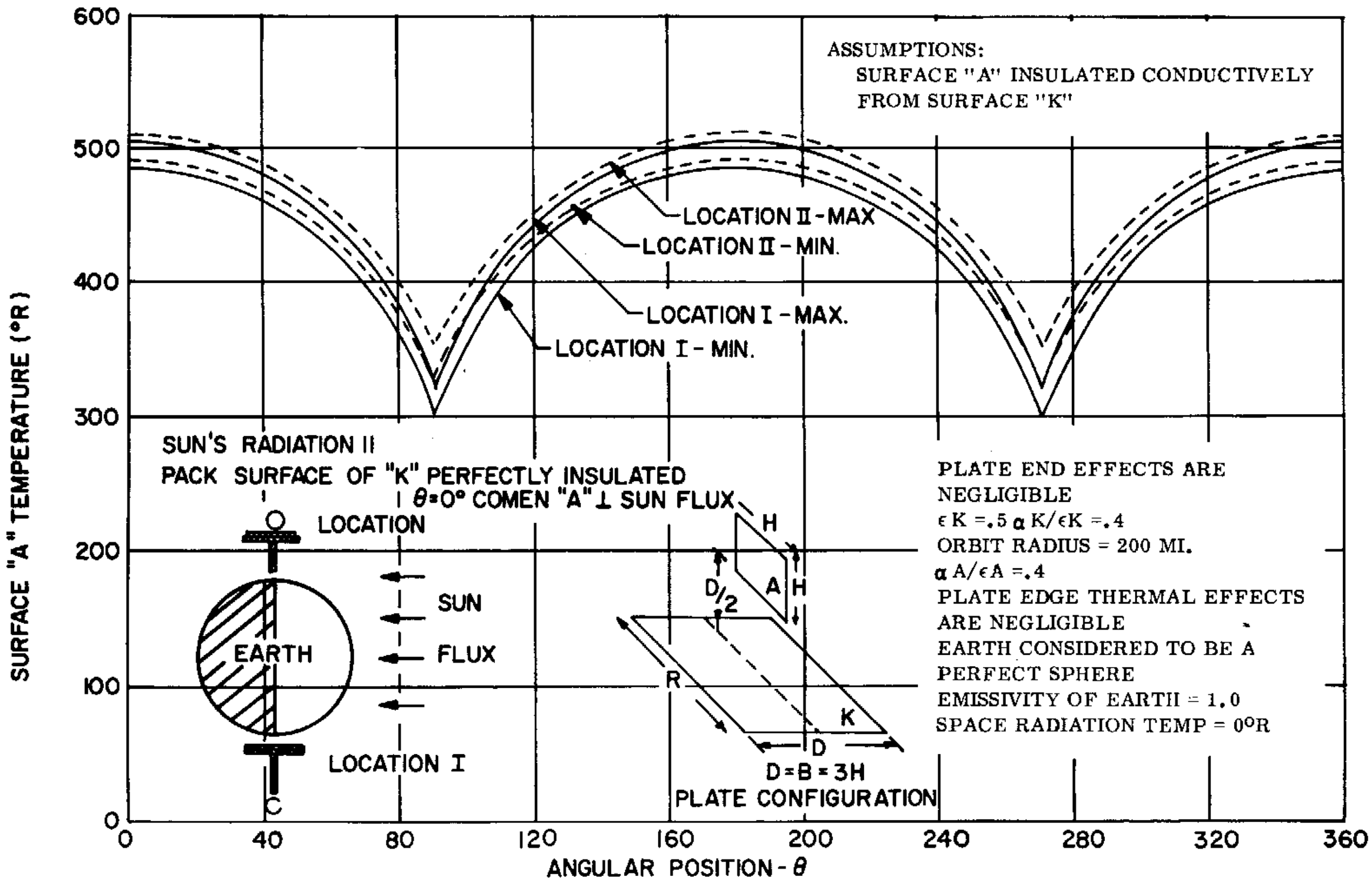


Figure 73. Equilibrium Temperatures of Two Thin Perpendicular Plates as a Function of Position Relative to Earth and Sun Radiation Effects