NASA CR-165238
2-53020/OR-52578

STUDY OF THERMAL MANAGEMENT FOR SPACE PLATFORM APPLICATIONS

by J. A. Oren

VOUGHT CORPORATION

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS3-22270

DISPLAY 05/6/1
81N21106*#   ISSUE 12   PAGE 1587   CATEGORY 15   RPT#: NASA-CR-165238
REPT-2-53020/OR-52578   CNT#: NAS3-22270   80/12/00   184 PAGES
UNCLASSIFIED DOCUMENT

UTTL: Study of thermal management for space platform applications

AUTH: A/OREN, J. A.

CORP: Vought Corp., Dallas, TX.; Hughes Aircraft Co., Los Angeles, CA.; TRW Systems, Redondo Beach, CA.; Hamilton Standard, Hartford, CT.; General Dynamics/Astronautics, San Diego, CA.; Lockheed Missiles and Space Co., Sunnyvale, CA.   AVAIL.NTIS

SAP: HC A09/MF A01

CIO: UNITED STATES Prepared in cooperation with Hughes Aircraft Co., Los Angeles, TRW Systems, Redondo Beach, Calif., Hamilton Standard, Hartford, Conn., General Dynamics/Astronautics, San Diego, Calif., and Lockheed Missiles and Space Co., Sunnyvale, Calif.

MAJS: /*AEROSPACE TECHNOLOGY TRANSFER/*HEAT PIPES/*HEAT RADIATORS/*SPACE PLATFORMS/*THERMAL ENERGY

MINS: / FEASIBILITY ANALYSIS/ FUEL PUMPS/ HEAT TRANSFER/ THERMODYNAMIC PROPERTIES

ABA: E.A.K.

mk JF

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA CR165238		

4. Title and Subtitle	5. Report Date
Study of Thermal Management for Space Platform Applications	December 1980
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
John A. Oren	2-53020/OR-52578
	10. Work Unit No.
9. Performing Organization Name and Address Vought Corporation P.O. Box 225907 Dallas, Texas 75265	11. Contract or Grant No. NAS3-22270
	13. Type of Report and Period Covered Contractor Report
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Langley Research Center 21000 Brookpark Road, Cleveland, Ohio 44135	14. Sponsoring Agency Code

15. Supplementary Notes

Project Manager: Sol Gorland, Space Propulsion Division, NASA Lewis Research Center, Cleveland, Ohio

16. Abstract

An evaluation was made of the candidate techniques for management of the thermal energy of large, multi-hundred kilowatt, ten year life space platforms. A number of promising concepts for heat rejection, heat transport within the vehicle and interfacing were analyzed and compared. The heat rejection systems were parametrically weight optimized over a wide range of conditions for heat pipe and pumped fluid approaches. Two approaches for achieving reliability were compared. An approach was recommended for each function of the thermal management system after comparing the candidates for a number of criteria including performance, weight, volume, projected area, reliability characteristics, cost and operational characteristics. An assessment of the technology needs was made and technology advancement recommendations were made.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Space Platforms Thermal Control Heat Pipes Heat Rejection	Unclassified

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified		

* For sale by the National Technical Information Service, Springfield, Virginia 22161

NASA-C-168 (Rev. 10-75)

N81-21106 #

FOREWORD

The study presented herein was supported by one subcontractor and four team members, as follows:

Hughes Electron Dynamics
Subcontractor for Heat Pipe Concepts Development

TRW Systems
Team Member, Space Processing Requirements

Hamilton Standard
Team Member, Life Support Systems Requirements

General Dynamics Astronautics
Team Member, Power Systems Requirements

Lockheed Missiles and Space Co.
Team Member, Power Systems Requirements

In addition, the USAF Aeropropulsion Laboratory and Martin Marietta Aerospace provided considerable useful information on battery technology and requirements.

The author is grateful for the support of the personnel from the above named organizations, the many Vought personnel who supported the study, and for the helpful direction of Mr. Sol Gorland, NASA Project Manager.

TABLE OF CONTENTS

TABLE OF CONTENTS (CONTINUED)

LIST OF FIGURES

LIST OF FIGURES (CONT'D)

LIST OF TABLES

LIST OF TABLES (CONT'D)

LIST OF TABLES (CONT'D)

1.0 SUMMARY

This report documents a study conducted by the Vought Corporation under Contract NAS3-22270 for the NASA Lewis Research Center during the period of 16 November 1979 through 26 August 1980. Objectives of the study were: (1) identification of promising thermal management concepts for a 250 kW Space Platform, (2) selection of a baseline concept along with alternate approaches that promise significant benefit, and (3) identification of the technology effort required to achieve a 1990 readiness for the baseline design.

The study was conducted in four major tasks. A schedule of the study is shown in Figure 1. Task I was to determine the thermal management requirements for the 250 kW Space Platform. The baseline vehicle description and a preliminary set of requirements were supplied for the study by NASA-LeRC. During the requirements study, the various team members were visited to solicit requirement inputs. These team members included TRW Systems (Space Processing), Hamilton Standard (Life Support System), General Dynamics Astronautics (Power Systems), and Lockheed Missiles and Space Company (Power System). Task II identified the best thermal management concepts for the space platform. This task involved generating and evaluating concepts for the various functions of heat transport and heat rejection and evaluating the various concepts through trade studies. As a result of the conceptual design studies, a baseline approach was selected for both heat transport and heat rejection within the guidelines of the study. Alternative approaches were selected that promise significant benefit. In Task III the technology development required to provide technology readiness for the baseline system and alternate approaches for the 1990 time period was identified. Task IV consisted of the study reporting including monthly reports and the Final Report.

The baseline 250 kW Space Platform vehicle configuration for the thermal management study is shown in Figure 2. This platform consist of a 250 kW Power Module with planar solar arrays, a centralized heat rejection system, and a berthing module into which the various payloads are docked. The docked

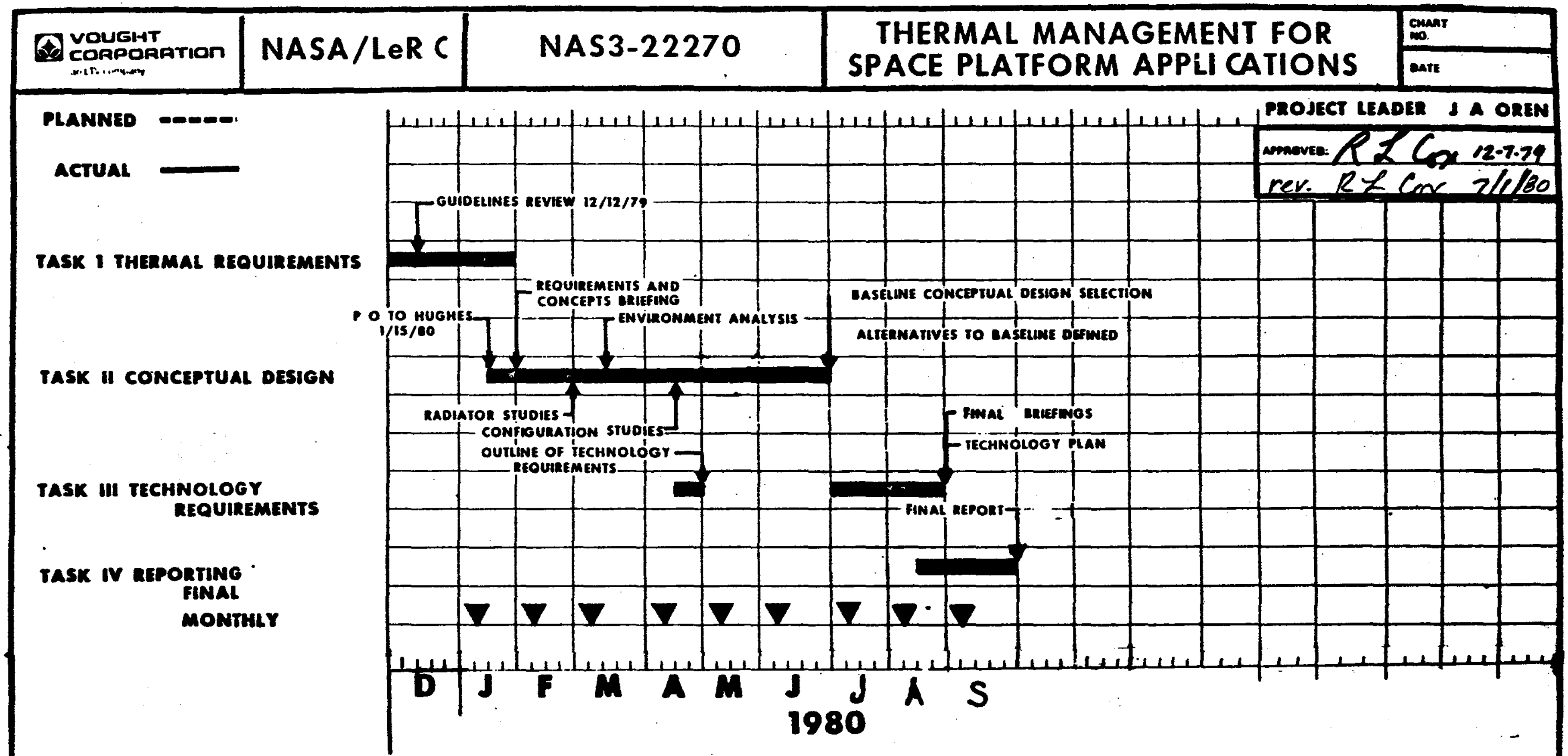

VOUGHT CORPORATION
NASA/LeR C
NAS3-22270
THERMAL MANAGEMENT FOR SPACE PLATFORM APPLICATIONS
CHART NO.
DATE
PROJECT LEADER J A OREN
APPROVED: R L Cox 12-7-79
rev. R L Cox 7/1/80
PLANNED
ACTUAL
GUIDELINES REVIEW 12/12/79
TASK I THERMAL REQUIREMENTS
REQUIREMENTS AND CONCEPTS BRIEFING
P O TO HUGHES 1/15/80
ENVIRONMENT ANALYSIS
BASELINE CONCEPTUAL DESIGN SELECTION
ALTERNATIVES TO BASELINE DEFINED
TASK II CONCEPTUAL DESIGN
RADIATOR STUDIES
CONFIGURATION STUDIES
OUTLINE OF TECHNOLOGY REQUIREMENTS
FINAL BRIEFINGS
TECHNOLOGY PLAN
TASK III TECHNOLOGY REQUIREMENTS
FINAL REPORT
TASK IV REPORTING
FINAL
MONTHLY
D J F M A M J J A S
1980

FIGURE 1

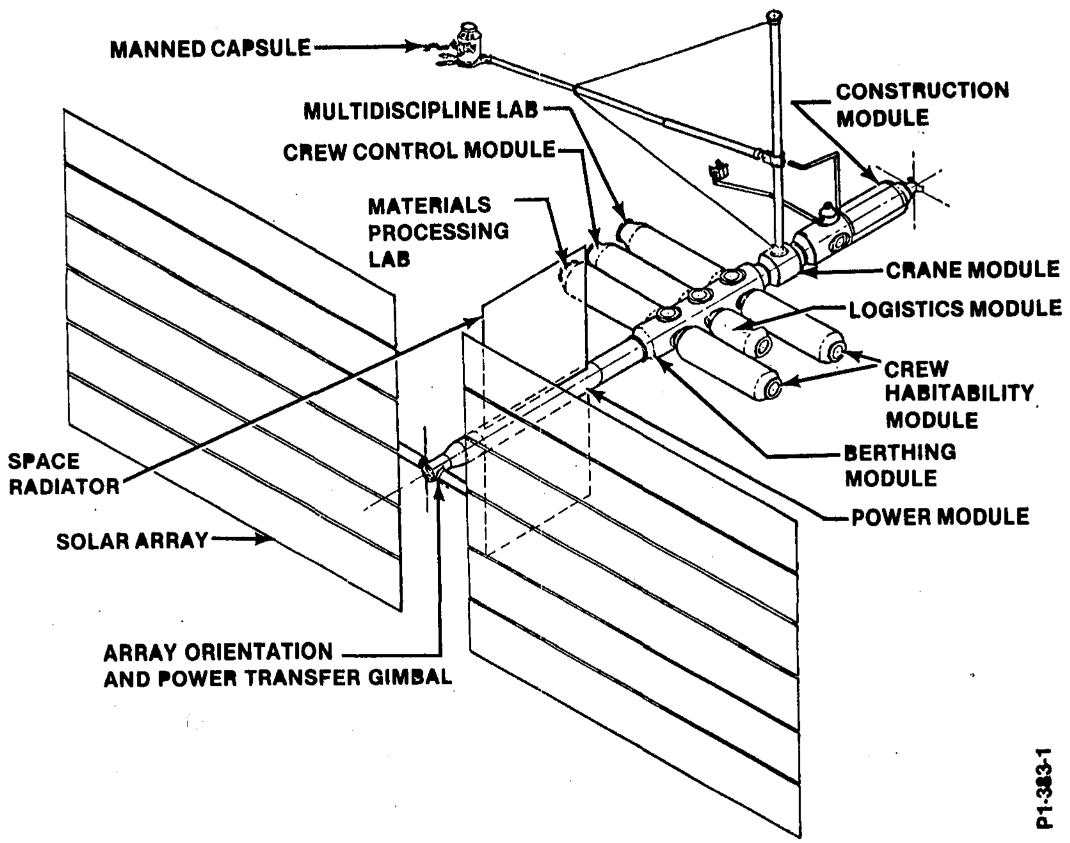


FIGURE 2 NASA BASELINE SPACE PLATFORM

modules include two habitability modules, a logistics module, a crew control module, a multidiscipline lab, a materials and processing lab, a construction module, and a crane module. Not shown in Figure 2 is an unmanned pallet containing scientific instruments. The requirements for the unmanned module were derived from studies on the Science and Applications Space Platform (SASP) (Ref. 1). The Power Module delivers 250 kW to the users on the berthing module and thus the heat rejection is larger than 250 kW by the amount of the power processing heat in the Power Module. Heat loads and temperatures for each module were established in the requirements study based on previous studies and discussions with team members. The life requirement of the platform was baselined to be 10 years and an indefinite life with on-orbit maintenance and replacement. The platform was to be capable of an orbital altitude of 370 to 650 km and orbital inclinations of 0° to 90°. In addition to the baseline thermal requirements, three special experiments and equipment were included on the platform. These were a microwave power transmission experiment which had a series of pulses at 75 kWe into the transmitter; a particle beam injection experiment which had 500 kW peak power pulses for just a few seconds duration with a 10% duty cycle, and propellant reliquification facility which processes the daily heat loads into the liquified hydrogen and oxygen stored on-orbit.

Concept studies were conducted to identify the most promising heat transport systems for the 250 kW Thermal Management System to meet the requirements. Eight promising concepts were defined and evaluated for comparison in the trade studies. These eight concepts included three variations of pumped liquid concepts and five variations of two-phase condensing and evaporating flow concepts. All the heat transport concepts involved a centralized thermal control system. The following major conclusions were reached as a result of the heat transport system trade studies.

(1) Two-phase thermal buss approaches offer the advantage of isothermal temperature sources for either cooling or heating and provide the potential for higher heat transfer due to evaporation and condensation. This approach may be especially attractive for unmanned payloads.

(2) A single phase pumped liquid water loop is the best choice for space platform heat transport when manned cabins are

involved and isothermal sink and sources are not required.

(3) Osmotic heat pipe approaches which offer the appeal of a completely passive all heat pipe system are still in the laboratory stage. Meaningful weight and cost projections for this approach cannot be made at this time.

(4) Use of vapor compression heat pumps for local cooling offers the promise of lowering radiator area and possibly system weight depending upon power system weight penalties.

(5) Multiple discrete temperature thermal busses offer promise of significant weight and radiator area reduction for both single phase and two phase approaches.

(6) Technology development is needed for efficient connectable thermal interfaces between the heat transport system and the individual modules.

Heat rejection system studies were conducted to determine the most promising concept meeting the requirements of the 250 kW power platform. The objectives of the concept studies were (1) determining the best radiator type (pumped fluid, heat pipe, constructable), (2) determining methods for achieving reliability goals and (3) obtaining a design description of the best concepts. Loop studies were conducted to determine reliabilities of the various radiator loop configurations. Reliabilities were then combined with micrometeoroid penetration probabilities to determine the optimum subsystem size (and number of independent subsystems). Weight trades were conducted for the heat rejection system to compare the different panel designs (pumped fluid panels, two designs for hybrid heat pipe panels and the constructable radiator concept). Studies were conducted to determine the effect of rejecting heat from the individual modules as opposed to the centralized system which was baselined for the majority of the study. Cost trades were also conducted to compare the costs of the different radiator designs. The RCA PRICE routine was used for these cost analyses. The following conclusions were reached as a result of the heat rejection system concept trade studies.

1. A deployed integral manifold single subsystem hybrid heat rejection system was selected as the baseline heat rejection system approach.

2. The space constructed radiator with no deployment mechanism and the fluid interface completely enclosed in the Power Module structure offers significant advantages and is the best overall approach. However, this concept has the disadvantage of requiring significant advances in heat pipe technology. This was selected as the alternate high technology approach with significant promise.
3. The use of the outer surface of the module to augment heat rejection offers significant savings in deployed area with little impact in cost or weight. However, increased system complexity and sensitivity to radiator coating degradation results.
4. Concepts with automatic deployment of the pumped fluid system cost about 10% less than the heat pipe but weighs about 10% more.

A technology assessment was conducted for the baseline approaches selected from the concepts studies and the alternate approaches. Current state-of-the-art for meeting the various functions of the thermal management system were assessed and compared to those required to meet the various functions for a 1987 technology readiness for a 1990 Space Platform launch. From this assessment, a set of technology recommendations were derived for meeting the 1987 technology readiness. Technology items requiring development for the heat transport systems and its interfaces are as follows:

1. The proven pump life for a pumped liquid heat transport system should be increased by a factor of 4 from the current 2-1/2 year life to 10 years proven life.
2. The capacity of developed pumps should be increased by an order of magnitude.
3. The technology to support the development of an advanced, high capacity thermal buss should be initiated. This includes increasing the heat transport capacity of heat pipes and pseudo-heat pipe type systems by 3 orders of magnitude.
4. Contact heat exchangers should be developed for integration into docking ports for thermal interfacing upon docking the payload modules into the platform. The interface heat

exchanger should be a fluid-to-fluid heat exchanger. A potential need also exists for a fluid-to-heat pipe and heat pipe-to-heat pipe contact heat exchanger.

5. A 360° rotation, no leak, long life fluid swivel should be developed. Also there is a potential need for a heat pipe swivel.
6. Technology should be developed to permit the analysis, design and fabrication of two phase flow systems in zero gravity. This technology will support the use of vapor compression systems in space.
7. A long life zero gravity compressor for use in vapor compression systems needs to be developed.
8. A need was identified for an in-line thermal storage system with approximately 5000 to 10000 watt-hours of energy storage capacity

Primary technology development required for the heat rejection systems is as follows:

1. Efficient lightweight fluid-to-heat pipe panel heat exchanger technology must be developed.
2. A radiator panel thermal coating with 10 year end of life thermal properties of α/ε of less than 0.2 should be developed to reduce the radiator area and the maintenance required for long life radiator systems.
3. Methods for deploying large radiator systems should be developed which are efficient in both weight and stowed volume.
4. A heat pipe contact heat exchanger should be developed specifically for use on the space constructable radiators.

2.0 INTRODUCTION

All energy utilized by any spacecraft must be rejected, either in the form of thermal energy or radiation in other electromagnetic wavelengths such as microwave or laser radiation. For the vast majority of projected future space missions, all of the energy generated by the power system must be rejected to the space environment via thermal radiation. Present long life spacecraft utilize only a few kilowatts of electrical power and their thermal management system has consisted of an "add-on" heat rejection subsystem. However, spacecraft being projected for the 1990's will require orders of magnitude increase in power capability to the hundreds of kilowatt range. These spacecraft will require comparably large heat rejection systems with radiator areas of a thousand square meters. Because of their large size and dependence on view factor constraints, the radiator can become a principal driver on the overall configuration of the spacecraft.

The large space platforms of the multihundred kilowatt power class will likely supply all the utilities to a diverse and continually changing mix of payloads. These utilities will include electrical power, thermal control and attitude control. Thus the thermal management system of the future must interface with ever changing thermal control requirements of the payloads. It must also provide the function of transporting the waste heat from the payloads to the heat rejection system. Because of the projected large quantities of heat, long transport distances, diverse interface requirements and long spacecraft life, new thermal management methods may be required for the future spacecraft. The potential of integrating the heating and cooling requirements for the total spacecraft may alleviate some of the problems due to the large sizes.

The primary purpose of the study was to determine the thermal management technology development required for the platforms of the 1990's. Promising candidate approaches were evaluated and the most promising selected for each thermal management function. The benefits of the most promising approaches were identified. Recommendations were made as to areas of technology development needed to provide technology readiness for the future space platforms.

3.0 REQUIREMENTS

A requirements study was conducted to identify the major requirements for the thermal management system for the 250 kW Space Platform. This included reviewing previous studies (such as the Multihundred Kilowatt studies (Ref. 2) being conducted by NASA), visiting our supporting team members , and assembling the available data to come up with the overall requirements for the thermal management system. The requirements resulting from these studies are discussed in the following sections.

3.1 System Definition

The Space Platform thermal management system includes all of the functions associated with:

- o Collection of waste heat from heat sources at rates which maintain the sources at acceptable temperature levels.
- o Transmission of the waste heat from the source to the heat rejection system.
- o Rejection of the waste heat to the space environment.

The thermal management system (TMS) provides an acceptable interface at each heat source but does not include internal mechanisms for transferring heat to the TMS interface within the components (such as electronic boxes or batteries).

3.2 General Description

Major elements of the Space Platform thermal system are identified in Figure 3 along with analogous elements for the power management system. The thermal management system includes the major elements of rejection, transport, and collection of the waste heat. The heat rejection function includes space radiators for rejection and possibly fluid slip rings to permit articulation of the radiator panels. The collection function includes interfacing with payloads and transferring heat from the payloads to the thermal buss. The transport function includes moving the heat from the collection points to the rejection points and also controlling the temperature levels at the heat removal points. Heat transport functions must also include storage of the thermal energy when necessary. The size of the thermal system for a 250 kW useful load is 250 kW plus power processing waste heat in the Power Module. The space processing power is about 25% of the total power load or 33% of the useful load. Temperature levels of the thermal buss were to be determined during the trade studies.

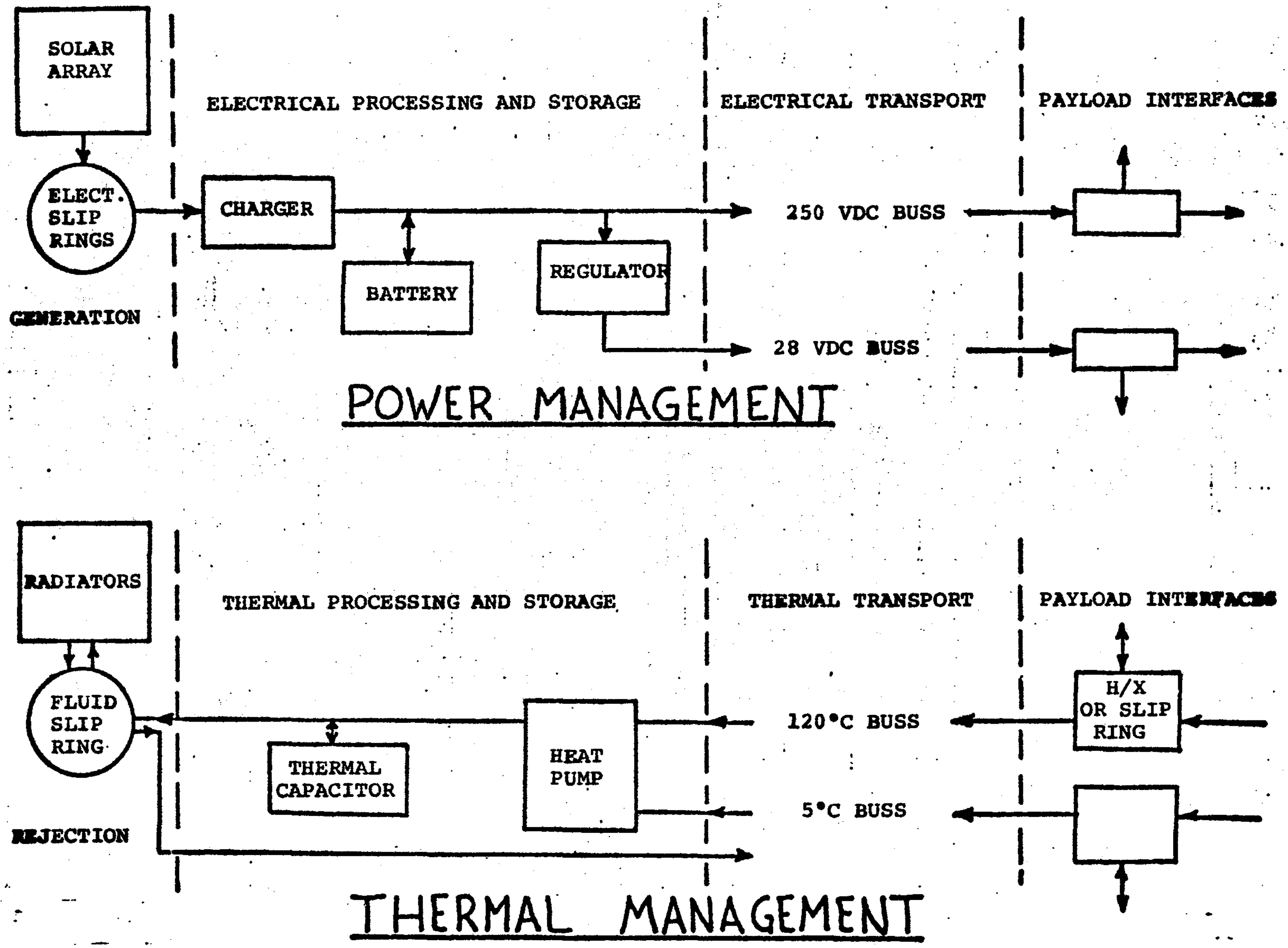


FIGURE 3 MAJOR FUNCTIONS OF THE THERMAL MANAGEMENT AND POWER MANAGEMENT SYSTEMS

3.3 Baseline Space Platform

The baseline configuration for the Space Platform is shown in Figure 2. This configuration, projected for the early 1990's era, was baselined by NASA for the study. It would evolve through a succession of build-up and operational phases of Shuttle compatible modular elements.

In the configuration shown, the Space Platform has the capability for continuous manned operation with a crew of 20. It would be supported by periodic logistics flights of the Space Shuttle for resupply of materials and consumables, crew rotation, and delivery of space manufactured products and waste materials to earth. A regenerative life support system is employed to reduce the amount of life support system consumables carried in the logistics flights.

The major elements that comprise the baseline Space Platform cluster are the power module, berthing module, modules for operational control, crew habitats, laboratories, construction, and cargo storage, as well as a space crane. The functional interfaces between the platform elements are identified in Figure 4. A brief description of each of the elements is provided below.

Power Module - The Power Module (PM) provides the photovoltaic power source, power conditioning equipment, energy storage, and some elements for power distribution, heat rejection, attitude control and stationkeeping. In the first phase of space platform operation, when orbital activities are relatively low and power demand is low, only a fraction of the solar array panels, storage batteries and power management system components are installed. Over a period of years additional equipment is added in several steps, culminating in the baseline power module which can deliver 250 kWe continuous average power when solar illumination is available for a minimum of 62% of the orbit period.

Berthing Module - The Berthing Module (BM) is an 18.3 m long by 4.6 m dia. pressurized structure which provides radial and axial docking ports for cluster build-up and subsystem interfacing. It also provides for inter-module access for crewmen to move from one module to another in a shirtsleeve environment. Hatches for ingress to the space crane and Shuttle

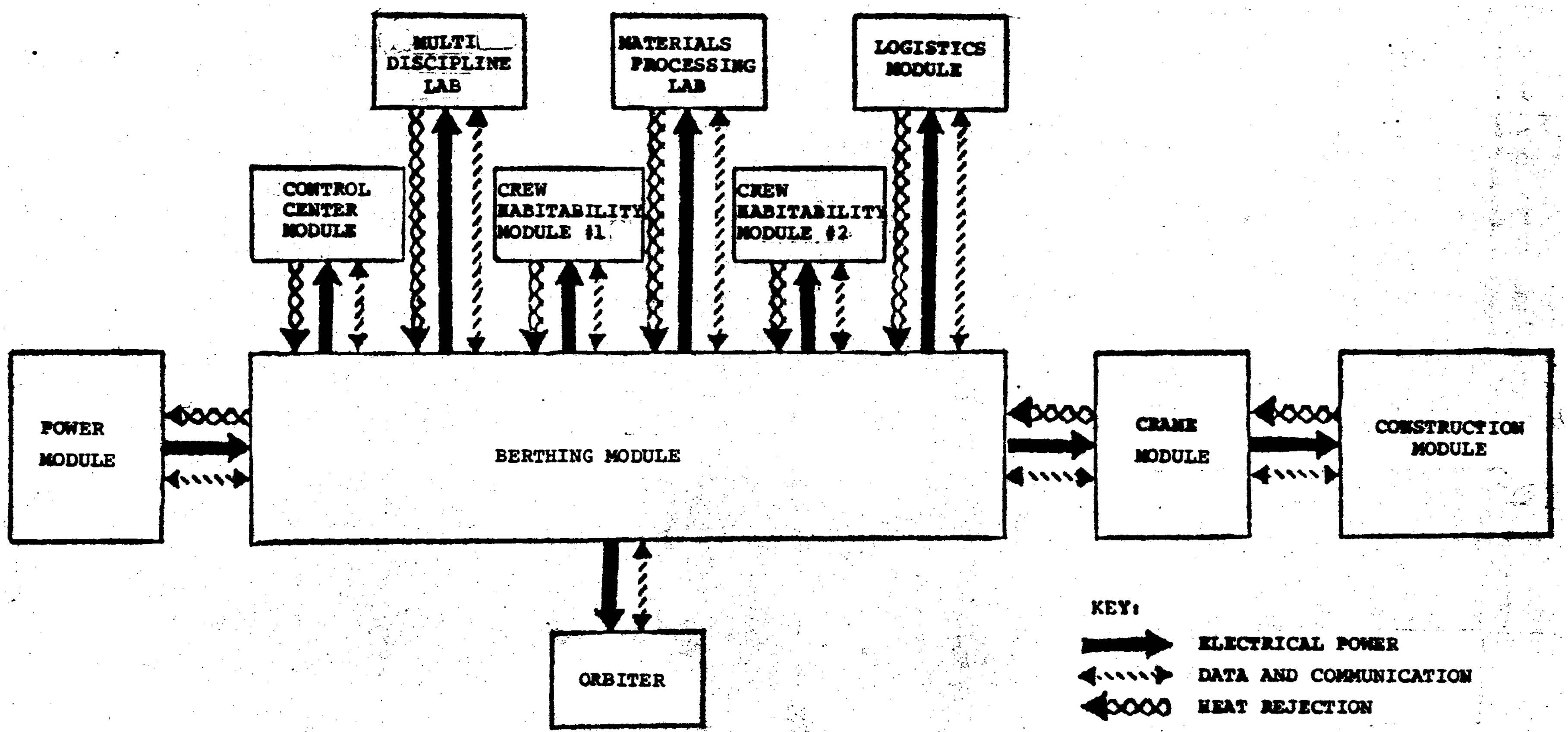


FIGURE 4 SPACE PLATFORM ELEMENTS AND INTERFACES

Orbiter and an EVA airlock are also provided. Standardized physical and utility interfaces are employed at each of the docking ports.

Control Center Module - The Control Center Module (CCM) is a 15.2 m long by 4.6 m dia. pressurized structure which houses the central control and display console, communications and data management equipment, wardroom, food storage and preparation facilities, and dining area. This module is the control nerve center of the space platform and contains all of the command, control and communications equipment to support the platform in its normal operating mode. In addition to crew sustenance it provides facilities for crew briefings, training, recreation, and medical services.

Crew Habitat Module - The Crew Habitat Module (CHM) is a 15.2 m long by 4.6 m dia. pressurized structure that provides private sleeping quarters for 10 crewmen and personal hygiene compartments with shower and waste management provisions. The baseline space platform configuration includes two crew habitat modules to accommodate the crew of 20.

Multi-discipline Lab - The Multi-discipline Lab (MDL) is a 15.2 m long by 4.6 m dia. pressurized structure that provides both general purpose and specialized laboratory facilities to support experiments and observations for a wide range of science and applications disciplines. As with the present Spacelab, the complement of experiment dedicated equipment will change over a period of time but general purpose support equipment such as an airlock, viewports, data collection and display consoles, work benches, freezers, ovens, and storage lockers will be available. Utilities in the form of electrical power of various types, compressed gas, vacuum, and water will also be provided.

Materials Processing Lab - The Materials Processing Lab (MPL) is a 15.2 m long by 4.6 m dia. pressurized structure that houses research and development equipment and pilot production plant facilities for processing materials in space. Typical types of equipment employed are furnaces, electrophoresis separation columns, refrigerators and freezers, analytical instruments and data collection and display equipment. Typical product development candidates are biological processing (Urokinase), high purity glass production, and silicon ribbon production.

Logistics Module - The Logistics Module (LM) is an 8.5 m long by 4.6 m dia. pressurized structure that is used in conjunction with the Space Shuttle to deliver materials and consumables to the Space Platform and store them on-orbit, and to return waste materials and space manufactured products to earth. The interior of the LM is arranged into a series of cells to support the transported equipment against flight loads, and to provide organized zero-g storage areas for supplies, waste containers, and other materials. External attachments and safety shields are provided for liquid storage tanks and high-pressure gas tanks.

Construction Module - The Construction Module (CM) is an 18.3 m long by 4.6 m dia. structure that incorporates a pressurized control station and an unpressurized work section. The module accommodates the beam fabrication machine, assembly tools, jigs and fixtures, EVA work stations and remote manipulators. In conjunction with the space crane, a variety of structural configurations can be assembled by this basic construction module. For building a large planar array, this CM would be exchanged for another CM configured for the specific geometry and assembly techniques required.

Space Crane - The Space Crane (SC) consists of a rotating, telescopic boom with a manned capsule at the outboard end, equipped with a manipulator system controlled by the crewmen. The crane has a reach of approximately 50 m radius by 70 m high. The crane is used to transport structural assemblies from the construction module, emplace them relative to other elements of the structure under construction, align and assemble structural elements and to install subsystem components and cabling. The crane can also be used to extract modules from the Orbiter cargo bay and transport then to docking ports on the berthing module.

Unmanned Pallet - The unmanned pallet is assumed to be an open truss design similar to those being evolved in the Science and Applications Space Platform (SASP) study (Reference 1). The open truss platform would accommodate a broad spectrum of unmanned scientific payloads, including Earth Viewers, Magnetic Field Viewers, Celestial and Solar Viewers and other experiments. The experiment duration would vary from one to ten years in length. The thermal heat loads that require rejection vary from 10 to 25 kW with a nominal of 16 kW identified. Cooling temperatures for the heat load were identified to be $16^{o}C$ ($60^{o}F$) coolant supply temperature and $43^{o}C$ ($110^{o}F$) return temperature.

All of these modules include a life support and environmental control system sized for the number of crewmen accommodated and the equipment heat loads dissipated in the module. One oxygen regeneration plant that services the entire cluster is located in the berthing module. To conserve oxygen and nitrogen, the airlock is pumped down to low pressure and the air is stored in pressure vessels prior to opening the external hatch.

Internal illumination is provided primarily by fluorescent lamps, with a few small incandescent high intensity lamps used in areas where detailed observation or dexterous manipulation is required. External illumination is both by fluorescent floodlamps and incandescent spotlamps.

Figure 5 is representative of the distances between the various modules.

3.4 Description of Major Subsystems

Two subsystems of the space platform have sufficient influence on the design of the thermal management system to warrant additional definition. These are the Electrical Power Subsystem and the Life Support Subsystem. An assumed definition of each of these for this study is given below.

3.4.1 Electrical Power Subsystem - The electrical power system provides a nominal 250 kW of electrical energy to the user busses. The power is generated by two planar solar arrays, each approximately 40 by 48 meters in size, for a total panel area of 3840 m^2. The solar arrays are on a two axis gimbal system for solar alignment.

The power processing and electrical energy storage components are physically located in the power module structure. The supplied voltage is 120 to 250 VDC unregulated voltage with nickel-hydrogen batteries used as a baseline for electrical storage. Figure 6 is a schematic diagram of the assumed power management system along with representative energy losses for the assumed system. This system was evolved by General Dynamics Corporation in the Multi-Hundred Kilowatt Power System study (Reference 2). The total losses in the power processing equipment (not including storage) is approximately 13 to 20%, all of which is assumed to be waste heat. Typical losses are shown in Table 1 for various power management system elements.

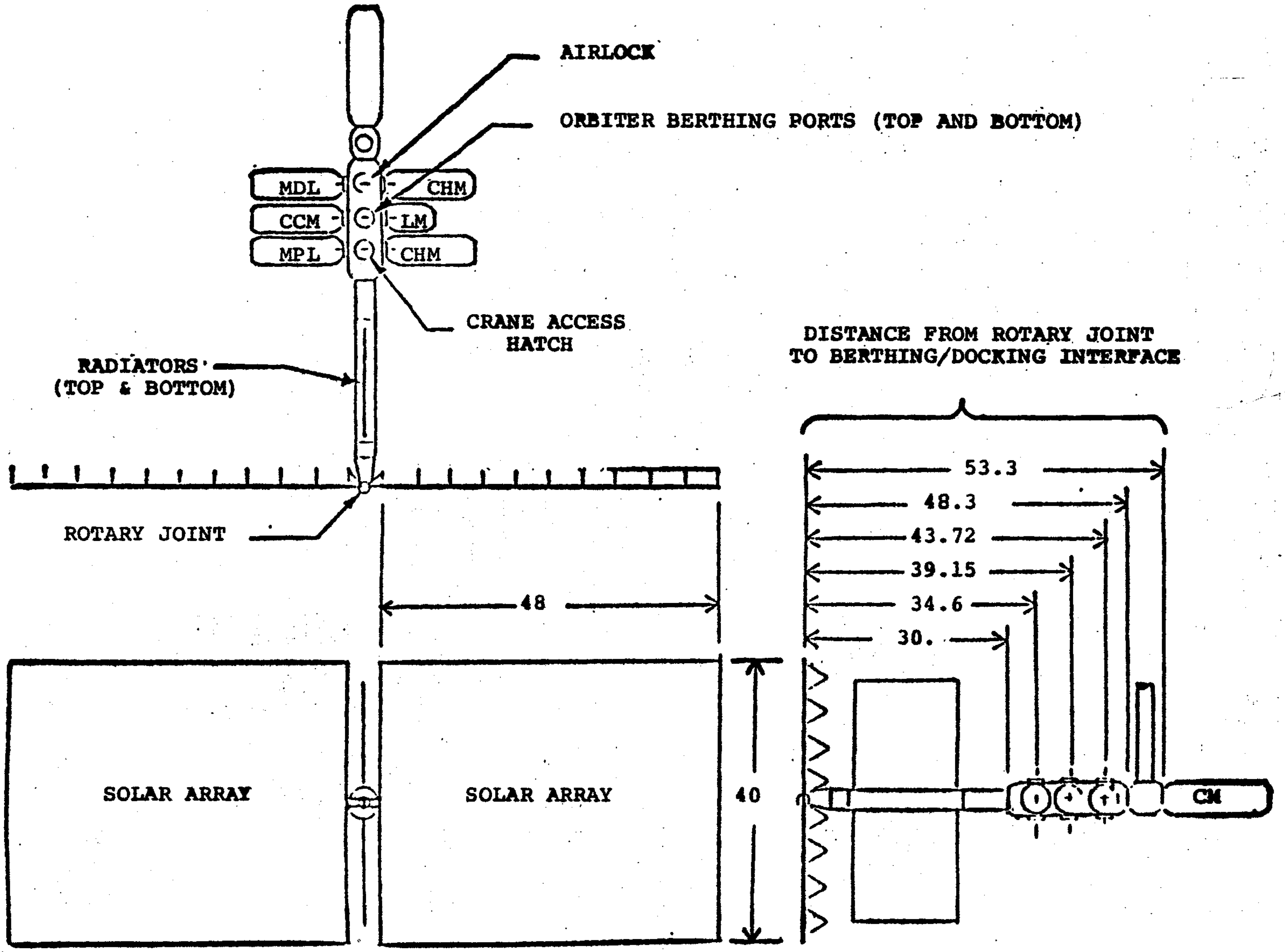


FIGURE 5 SPATIAL RELATIONSHIPS (DISTANCES IN METERS)

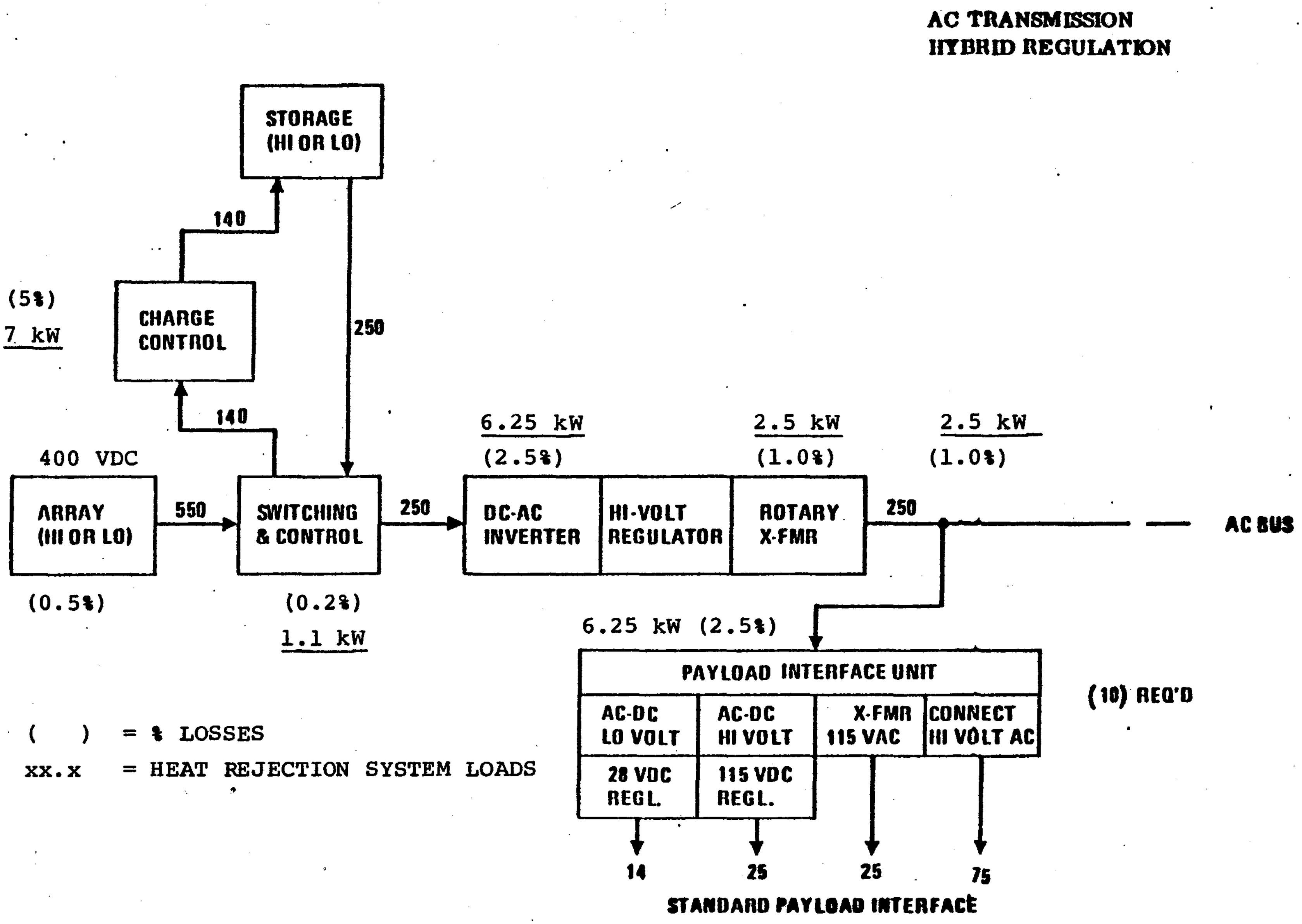


FIGURE 6 BASELINED POWER MANAGEMENT SYSTEM: AC TRANSMISSION, HYBRID REGULATION
(FROM GENERAL DYNAMICS CONVAIR DIVISION)

The electrical energy storage system candidates for the power system were: (1) nickel hydrogen batteries and (2) fuel cell/electrolyzer. Some of the requirements for the two approaches are given in Table 2. The energy conversion efficiency of the nickel-hydrogen battery was baselined to be 70% with 80% of the energy losses during discharge and 20% of the losses during the charge. For the fuel cell/electrolyzer approach, the conversion efficiency is approximately 58% with 83% of the losses during discharge and 17% of the losses during charge. The fuel cell approach has lower efficiency but, has the advantage of higher operating temperature. For this study, we baselined the nickel-hydrogen batteries.

3.4.2 Life Support System - The life support system for the 250 kW space platform was baselined to be an Advanced Integrated Life Support System. With this approach, shown in Figure 7, the CO_2 and water from the crew metabolism are processed to reclaim a large portion of the required oxygen and water. Hydrogen (from the food) is dumped overboard with this approach. Figure 8 includes the name of the processes planned for each function of the life support system and the various heat and mass transfer rates. Table 3 provides the thermal cooling loads required for the 20 man system. These data were provided by the Hamilton Standard Division of United Technologies, Inc.

3.5 Missions and System Requirements

The missions for an evolutionary, multipurpose space platform vary with time and cover a spectrum of activities from science and applications experiments and observations to construction of large space structures and in-orbit support of orbit transfer vehicles (OTV). The baseline Space Platform is configured to conduct experiments in physics and chemistry, materials processing and life sciences, to make observations in earth sciences and astronomy, and to construct large structures that support RF receiving and transmitting equipment and solar power collection and conversion equipment. The Space Station Systems Analysis studies have identified beneficial uses for Low Earth Orbit (LEO) space platforms in a range of inclinations from 28.5° to 55°, and in a range of circular orbit altitudes from 370 to 650 km (200 - 350 N.M.), as well as later applications in Polar Earth Orbit (PEO) and Geostationary Equatorial Orbit (GEO).

TABLE 1

POWER MANAGEMENT EQUIPMENT THERMAL REQUIREMENTS

<u>THERMAL LOSSES</u>

- Two-axis Gimbal DC Slip Rings : 1-1.2%
- Two-axis Gimbal AC Rotary Transformer : 1%
- Ni-H2 Charger : 3-6%
- High Voltage DC Regulator : 3-5%
- High Voltage DC-AC Inverter : 2-4%
- Conversion HVDC to 28 VDC : 8-12%
- DC System Diodes, Lines, & Distribution : 1.5-2%
- AC System Lines and Distribution : 1%

<u>TEMPERATURES</u>

- Nominal Avionics Range : 15°C - 60°C

<u>INTERFACES</u>

- Coldplates, etc., On Vehicle Side
- Special Concepts for High Density/High Power Electronics (Heat Pipes, etc.) on Equipment Side

TABLE 2

ENERGY STORAGE THERMAL REQUIREMENTS

NICKEL-HYDROGEN BATTERIES

- Energy Conversion Efficiency : 70%
- Operating Temperature: 10°C - 25°C
- Thermal Load Partition: 20% During Charge/80% During Discharge
- Trickle Charge: None for LEO
- Special Thermal Interfaces: Heat Pipes, Etc. on Batter Side of Interface

SOLID POLYMER ELECTROLYTE (SPE) FUEL CELL AND ELECTROLYZER

- Energy Conversion Efficiency : Fuel Cell - 65%
  Electrolyzer - 89%
  Water Pump - 10%
- Operating Temperature : Fuel Cell - 72°C
  Electrolyzer - 110°C
- Special Thermal Interfaces : Only on Fuel Cell/Electrolyzer Side of Interface

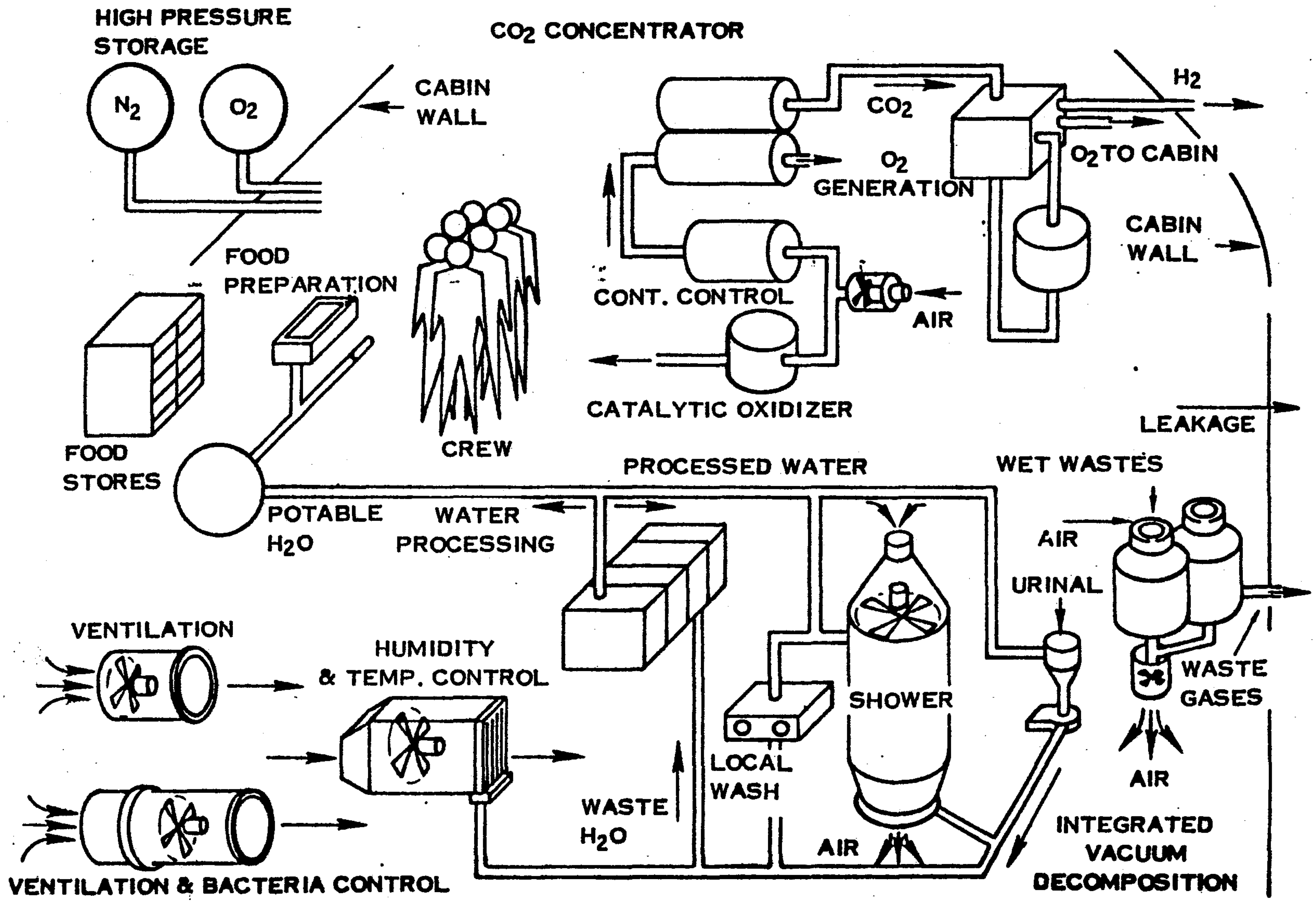
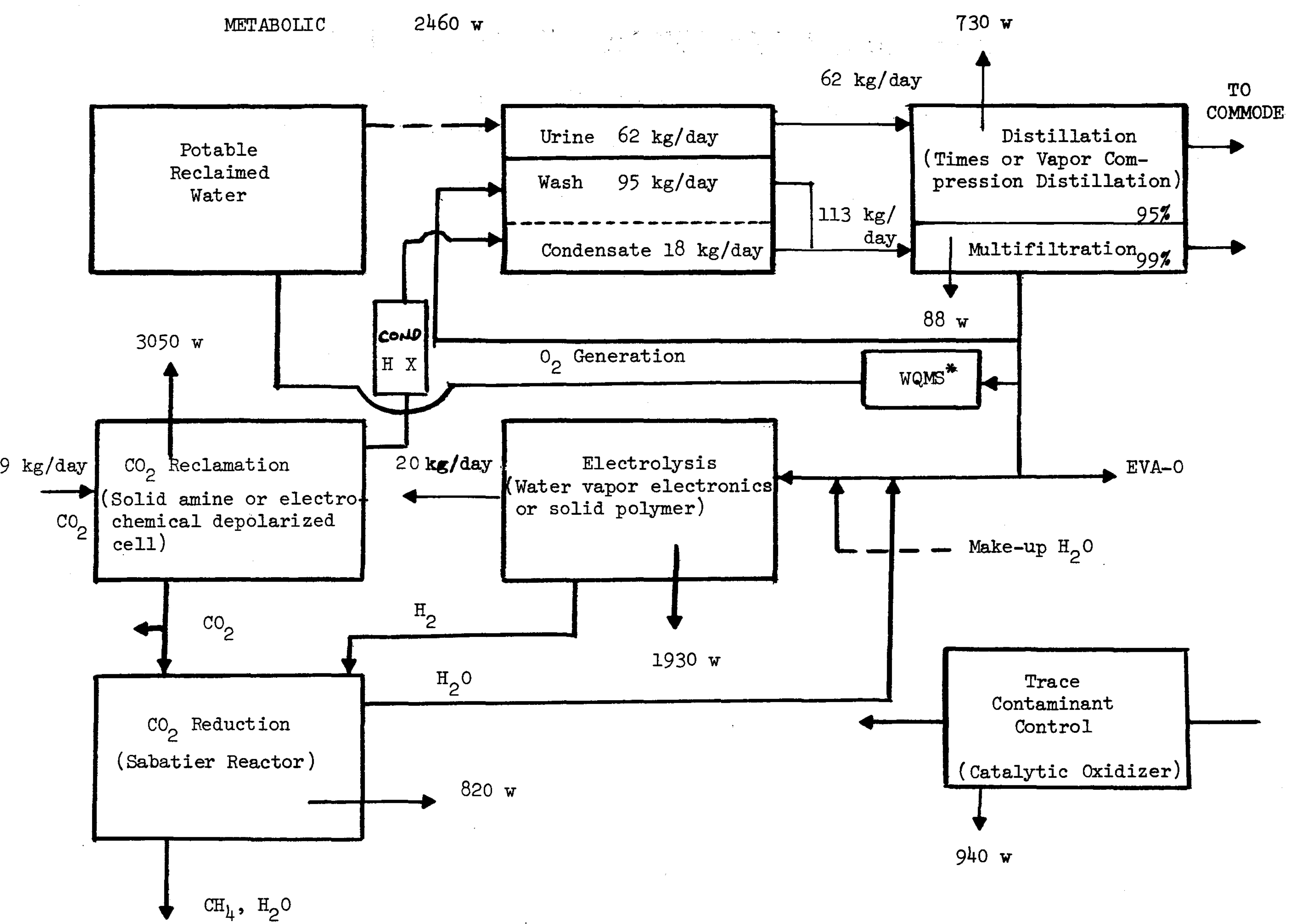


FIGURE 7 ADVANCED INTEGRATED LIFE SUPPORT SYSTEM



*Water Quality Management System

FIGURE 8 TYPICAL REGENERATIVE LIFE SUPPORT SYSTEM (KG/DAY) - 20 MEN

TABLE 3

TYPICAL ECS HEAT REJECTION LOADS
(20 MAN SYSTEM)

SUBSYSTEM	HEAT LOADS WATTS
Gas Storage and Pressurization	30
Trace Contaminant Control	940
Water Reclamation and Management	820
Temperature, Humidity, Ventilation (Coolant Inlet Temp - 7°C Max)	4,480
Crew Provisions (Coolant Inlet Temp - 7°C Max)	1,140
O_2 Generation	1,930
Waste Management	60
CO_2 Control	3,870
EC/LS Instrumentation	470
Crew Metabolic Load	2,460
TOTAL	16,200

The groundrules and assumptions for the Space Platform Thermal Management Study are given in Table 4. The operational time period is 1987 to 1990 Technology readiness for early 1990's missions. A 10 year system life in the meteoroid environment of space is the design goal. The orbital environments and penalties used in the study are also shown in Table 4, along with safety and interface requirements.

In the early operational phases of the Space Platform, stationkeeping impulse will be provided at 60 day intervals by the Orbiter. However, as the size and mass of the platform and structures under construction increase to large scale proportions, flight control subsystem elements will be added to enable the Space Platform to perform the stationkeeping function.

Experiment and construction activities are scheduled for around-the-clock operations. Three eight hour shifts will be worked, with overlap at each shift change, and a mid-shift break for eating and personal hygiene. The nominal assignment of crew duties is as follows:

Experiments and Construction	=	14
Housekeeping, Commun. & Data Mgt.	=	6
TOTAL		20 Crewmen

The tour of duty for each crewman is 180 days. Approximately one third of the crew is rotated each 60 days during routine logistics flights.

Propellant resupply of orbit transfer vehicles has been estimated to require 1000 metric tons per year by the early 1990's. The Space Platform will serve as the orbital depot where large, heavily insulated propellant storage tanks will be berthed. To eliminate boil-off losses, the storage facility will be equipped with reverse Brayton Cycle refrigeration equipment that will reliquify the hydrogen and oxygen gases and return them to the storage tanks. The daily processing load of the reliquification plant is estimated to be 137 kg of H_2 and 351 kg of O_2 per day. This requires cryogenic refrigeration heat loads of approximately 92 watts at 20^oK and 56 watts at 90^oK. The estimated power to drive the process is 50 kW. A heat rejection load of 51 kW was used in the study.

Two special experiments were identified by NASA for consideration in this study. These are 1) a microwave power transmission antenna test and

TABLE 4

SYSTEM REQUIREMENTS

TIMING AND GROWTH

- Baseline Concepts for 1987 Technology Readiness for Early 1990's Missions
- Alternate Higher Risk Concepts for 1990 Technology Readiness or Later
- Stepwise Growth of Platform and Subsystems
- Modular Heat Rejection System Preferred

LIFE, MAINTENANCE, AND RELIABILITY

- 10 Year Design Goal for Heat Rejection System as a Probability to be Determined from Trades
- Redundancy and Micrometeoroid Protection to Achieve Survivability
- Indefinite Life with Orbital Replacement and Maintenance
- Replaceability of Major Subsystem Elements
- Consider Fault Detection and Isolation

ENVIRONMENTS

- Orbit Altitude : 370 Km to 650 Km
- Orbit Inclination : 28.5° to 90°
- Orbit β Angle : 0° to 90°
- Micrometeoroids : NASA SP 8013
- Thermal : Consider Solar, Earth, Vehicle Interactions

PENALTIES

- Cost to Orbit : 1500 $/kg
- Power : 362 lb/kWe (100 to 1000 $/kW Range)
- Volume : Prefer Minimum Length in Shuttle Cargo Bay

SAFETY

- No Toxic Fluids In Inhabited Areas
- No Contact Temperatures Above 113°F
- No Flammability Hazards
- Consider Thermal Management of Emergency Power and Life Support Systems

INTERFACES

- Shuttle Orbiter Compatible for Transport and Deployment
  - C.G. Constraints
  - RCS Acceleration Loads and Bump Loads
  - RMS and EVA Capabilities and Timelines
  - Launch Loads
- Minimum Obstruction to Scientific Viewing Payloads Desired
- Minimum Aerodynamic Drag
- Avoid Unwanted Moments Due to Unfavorably Placed Deployed Masses
- Avoid Physical Interference with Gimballed Solar Arrays or Payloads
- Minimize Payload Contamination Threat Due to Fluid Leakage

2) a particle beam injection experiment. The microwave transmission antenna experiment may be built by the construction module and could be on an unmanned pallet. Figure 9 shows the projected efficiency train of the microwave transmission experiment and identifies the heat rejection load as 12.4 kW. The particle beam experiment is a series of 500 kW peak power pulses, each for a duration of a few seconds, but with a 10% duty cycle. We baselined an average power and heat rejection heat load of 50 kW for this experiment.

3.6 Thermal Management Requirements

Representative power requirements were developed for a typical day operation of the baseline space platform in Reference (2). The loads are summarized in Table 5 in terms of the average, maximum and minimum electrical power levels and the total daily energy consumed in each of the baseline space platform elements. However, it should be noted that the power expended in operating and controlling the power management system (PMS) itself is not included in these load figures. Also, when the Orbiter is docked to the cluster during resupply missions, an additional 14 kW will be drawn through the berthing module.

Table 6 lists the hour-by-hour load profile for each of the individual modules and for the Space Platform cluster as a whole (excluding the PMS internal loads).

FIGURE 9
MICROWAVE TRANSMISSION EXPERIMENT TRAIN EFFICIENCIES

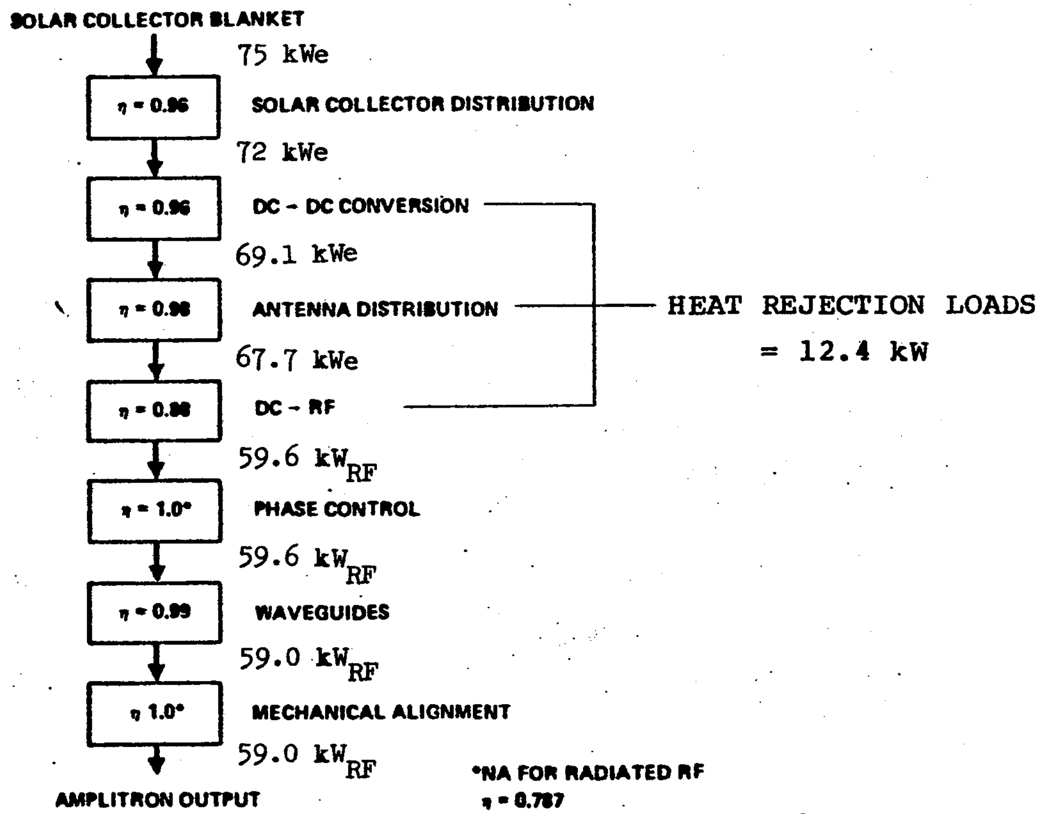


TABLE 5

BASELINE SPACE PLATFORM DESCRIPTION

MODULE	HEAT REJECTION LOADS (kW) PEAK	LOW	AVG	HEAT LOADS TEMPERATURES °C
Berthing Module	12.3	11.2	11.7	4.4 & 12.8 to 38
Power Module	98.6	62.8	85.1	12.8 to 38
Control Center Module	23.2	11.8	17.5	12.8 to 38
Crew Habitat Module #1	8.4	5.5	6.2	12.8 to 38
Crew Habitat Module #2	8.4	5.5	6.2	12.8 to 38
Multi-Discipline Lab	29.2	14.8	20.0	12.8 to 38
Logistics Module	2.	2.	2.	12.8 to 38
Unmanned Pallet	25.	10.	16.	12.8 to 43
Materials & Processing Lab	71	17	50	12.8 to 38 & 93
Construction Module	15.8	14.2	15	12.8 to 38
Crane Module	5.5	2.0	5.1	4.4 & 12.8 to 38
TOTALS	229.4	156.8	234.8	

TABLE 6

CLUSTER LOAD PROFILE – TYPICAL DAY

TIME, HOURS →		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
OPERATIONAL SHIFT: 1																									
2																									
3																									
LOCATION	CREW																								
MDL	1	14.8	16.8	24.0	16.2	16.2	22.6	19.4	21.8	29.2	18.2	19.0	15.4	21.4	23.	29.2	15.6	14.8	24.6	19.4	20.2	27.2	16.2	17.0	17.0
MPL	1	47.5	47.5	47.5	47.5	47.5	47.5	47.5	50.3	57.3	57.3	57.3	18.5	22.3	19.3	17.3	71.1	70.3	60.3	60.3	49.5	70.3	70.3	60.3	50.2
CM	1	15.	15.	15.	14.2	15.	15.	15.	15.8	15.	15.	15.	14.2	15.	15.	15.	15.8	15.	15.	15.	14.2	15.	15.	15.	15.6
CRANE	1	5.5	5.5	5.5	2.0	5.5	5.5	5.5	5.5	5.5	5.5	5.5	2.0	5.5	5.5	5.5	5.5	5.5	5.5	5.5	2.0	5.5	5.5	5.5	5.5
CCM	5-10	14.9	17.8	14.9	13.7	11.8	17.0	16.5	13.3	15.3	22.9	17.5	15.3	16.0	20.4	20.7	17.5	19.1	23.2	18.3	15.3	16.0	21.6	19.9	17.5
CHM #1	3-7	6.5	6.0	6.0	7.3	8.1	5.5	5.5	5.5	6.0	5.5	6.0	6.0	7.9	5.5	5.5	5.5	6.0	5.5	6.0	6.8	8.4	6.0	6.0	6.0
CHM #2	3-4	6.3	5.5	6.0	6.8	7.6	6.5	6.0	6.0	6.8	6.0	6.5	6.5	7.6	6.0	5.5	5.5	6.3	5.5	6.0	6.0	7.1	6.0	5.5	5.5
BM	1	11.2	11.2	11.2	12.2	12.3	12.3	12.2	11.2	11.2	11.2	11.2	12.2	12.3	12.3	12.2	11.2	11.2	11.2	11.2	12.2	12.3	12.3	12.2	11.2
LM	0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.0
PM (FCLSS)	0	8.2	8.4	11.1	8.1	8.3	8.7	8.5	8.6	9.6	9.0	9.0	6.6	7.5	7.6	7.6	9.5	9.5	9.6	9.2	8.4	10.2	9.8	9.2	8.5
TOTAL CLUSTER LOAD PROFILE		131.9	135.7	143.2	130.	134.3	142.6	138.1	140.	160.9	149.1	149.	98.7	117.5	118.6	120.5	159.2	159.7	162.4	152.9	136.6	174.	166.7	152.6	139.3

TOTAL DAILY AVG: 3,413.5 kW/Hr

AVG LOAD: 142.23 kW

LOW : 98.7 kW

HIGH: 174. kW

4.0 CONCEPT STUDIES

Studies were conducted to determine the best overall approaches for thermal management of the 250 kW Space Platform. These studies involved generating promising concepts and approaches; sizing, optimization and design analyses of each concept; performing cost analyses and constructing a trade matrix for comparison and selection of the best approach. The heat rejection and heat transport segments of the thermal management system were studied separately to obtain the best approach for each. The study assumptions are summarized below in Section 4.1. The heat transport studies are summarized in Section 4.2 and the heat rejection studies are discussed in Section 4.3.

4.1 Study Assumptions

Assumptions and groundrules for the study included the following:

1. A centralized heat rejection system located in the power module was assumed. This dictates a centralized heat transport system within the platform berthing and power module.
2. The thermal load for the berthing module was assumed to be 240 kW distributed over 13 docking ports. The maximum heat load allowed at each of the docking ports is 25 kW for 12 ports and 100 kW at one.
3. The power distribution and storage heat load is 87 kW located in the power module.
4. The heat acquisition temperatures were divided among the heat loads as follows:

Berthing Module:	75% @ $16^{o}C$ ($60^{o}F$)
	25% @ $4^{o}C$ ($40^{o}F$)
Power Module :	75% @ $16^{o}C$ ($60^{o}F$)
	25% @ $27^{o}C$ ($80^{o}F$)

5. The maximum heat transport distance was 46 meters (150 ft.).
6. The electrical power specific weight used in the study is 164 kg/kW (360 LBm/kW). For some high power using concepts, 45 kg/kW (100 LBm/kW) was also considered to provide a sensitivity comparison.

Cost studies were conducted as a part of the trade studies for the heat transport and heat rejection system. The RCA PRICE routine was utilized for this parametric analysis. Assumptions that were made for the cost studies were as follows:

(1) The assumed program schedule is:

- o Development Start : January 1988
- o Prototype Complete : January 1989
- o Development Complete : January 1990
- o Production Start : February 1991
- o Delivery : August 1992

(2) The year of economics is 1980 dollars.

(3) The year of technology is 1985.

(4) The total cost is prime contractor acquisition cost. No vehicle level tests, flight support or maintenance costs are included.

(5) PRICE routine complexity factors were based on historical cost data when available. Otherwise, component supplier costs estimates were used.

Table 7 shows the engineering and manufacturing complexity factors which were derived for the various components for input to the PRICE routine. Also shown are the platform factor inputs. Typical values for the manufacturing and engineering complexity factors are shown in Tables 8 and 9. The platform factor of 2.5 was used which indicates manned space.

4.2 Heat Transport Concept Trade Studies

The 250 kW Space Platform heat transport system must provide the following functions:

- o Collect or add heat as required at specified locations within the platform to maintain the various equipment at the required temperature.
- o Transport the space platform waste heat generated at the various modules within the platform to the centralized heat rejection system for removal.
- o Provide the interface between the heat transfer loop and the various payloads and the heat rejection.
- o Accommodate a wide variety of requirements for the various payloads and a changing payload mix.

TABLE 7

ASSUMPTIONS FOR COST ANALYSIS
OF
SPACE PLATFORM THERMAL CONTROL SYSTEM

- PRICE Routine Inputs

COMPONENT	ENGINEERING COMPLEXITY	MANUFACTURING COMPLEXITY	PLATFORM FACTOR
Radiator Panels	1.5	7.2	2.5*
Heat Pipes	1.172	6.5	2.5
Pump/Motor	.238	9.1	2.5
Accumulator	1.566	5.4	2.5
Temp Control Valve	.866	9.1	2.5
Temp Sensors	1.37	6.1	2.5
Heat Exchanger	0.865	9.1	2.5
Flex Hoses	1.633	5.2	2.5
Deployment Mechanism	1.361	6.1	2.5
Integration & Test	1.162	7.020	2.5

* Platform Factor of 2.5 is manned space

TABLE 8

TYPICAL VALUES FOR MANUFACTURING

MANUFACTURING COMPLEXITY - A factor to describe the product producibility, usually an empirically derived factor. It is a function of the material type, finished density and fabrication methods.

TYPICAL VALUES

Equipments	Typical Examples	** WSCF	1.0 * Ground	1.4 Mobile	1.8 Airborne	2.0 Space	2.5 Manned Space
Antennas	Small, Spiral, Horn, Flush, Parabolic	4	4.75	5.39	5.64	6.55-7.04	6.92-7.44
	Scanning Radar 10-40' Wide	8	5.3	5.4	5.5	—	—
	Phased Arrays (Less Radiators)	6-8	5.9	6.2	6.4	7.0	7.2
Engines & Motors	Automobile - 100 to 400 H.P.	25-35	—	4.30	—	—	—
	Turbo-Jet (Prime Propulsion)	25-35	—	—	6.6-7.9	—	—
	Rocket Motors	14-15	—	—	6.1-6.5	6.4-7.3	7.2-8.2
	Electric Motors	75-100	4.47	5.08	5.3	5.4-6.3	5.4-6.3
Drive Assemblies	Machined Parts, Gears, etc.	7-10	5.11-5.24	5.5	5.8	—	—
	Mechanisms w/Stampings (Hi Prod)	12	3.33-3.73	—	—	—	—
Microwave Transmission	Waveguide, Isolators, Couplers,	11-20	5.4-5.6	5.4-5.6	5.5-5.7	5.5-5.9	5.5-5.9
	Stripline Circuitry	9	5.7	5.8	5.9	6.0	6.1
Optics	Good (Commercial)	70-90	5.1	5.4	6.3	6.7	7.3
	Excellent (Military)	70-90	5.4	5.8	7.3	7.8	8.0
	Highest (Add 0.1 per 10% Yield)	70-90	5.9	6.8	8.0	8.3	8.5
Ordnance Fuze	Automated Production	14-20	—	4.3-4.65	4.3-4.65	—	—
	Small Production-Min. Tooling	14-20	—	5.11-5.33	5.11-5.33	—	—
Servo	Mech Drive & Coupling Networks	65-75	5.63	5.63-5.7	5.7-6.26	5.7-6.86	5.7-6.86
Tools	Machine Tools	25-30	4.45-4.52	—	—	—	—
Printed CKT Cards (Boards Only)	Paper Phenolic	83	4.1-4.3	4.1-4.3	4.1-4.3	4.1-4.3	4.1-4.3
	Glass Expoxy, Double Sided (Add 0.2 for 3 Layers & 0.05 for Addn'l) Add 0.1 for Plated-Thru Holes	110	5.3	5.3	5.3	5.3	5.3
Cabling	Multiconductor w/MS Connectors	40	4.9	5.0	5.0	5.1	5.2
	Same w/ Hermetically Sealed Connectors	40	5.1	5.2	5.2	5.3	5.3
Battery	Lead Acid	68-125	4.47	4.49	4.01	4.8-5.4	4.9-5.5
	Nickel Cadmium	75	5.39	5.83	6.73	7.63	8.38
Gyro	Inertial Platform Type	79	6.01	6.56	6.8	6.9-9.1	7.0-9.4

*Platform Factors

**Mechanical Density, LB/FT^3

TABLE 9

TYPICAL VALUES OF ENGINEERING COMPLEXITY

ENGINEERING COMPLEXITY - Used to scope development effort and to develop calendar time for first prototype.

TYPICAL VALUES

SCOPE OF DESIGN EFFORT	Extensive experience, with similar type designs. Many are experts in the field, top talent leading effort.	Normal experience, engineers previously completed similar type designs	Mixed experience, some are familiar with this type of design, others are new to job	Unfamiliar with design, many new to job
Simple modification to an existing design	.2	.3	.4	.5
Extensive modifications to an existing design	.6	.7	.8	.9
New design, within the established product line, continuation of existing state of art	.9	1.0	1.1	1.2
New design, different from established product line. Utilizes existing materials and/or electronic components	1.0	1.2	1.4	1.6
New design, different from established product line. Requires in-house development of new electronic components, or of new materials and processes	1.3	1.6	1.9	2.2
Same as above, except state of art being advanced or multiple design path required to search goals	1.9	2.3	2.7	3.1

These functions must be accomplished while meeting all the requirements specified under Section 3.0. In this study the heat transport system function does not include that of controlling temperatures within payloads and instruments such as batteries and other equipment. This function is considered outside the function of the heat transport system.

Studies were conducted with the objective of determining the best heat transport systems to meet the thermal management requirements of the 250 kW Space Platform. This section describes the concept studies conducted to generate the trade data necessary to evaluate the various concepts. Critical parameters which were determined for each concept included the weight, sizes and volumes of the various components and elements of the concepts, the interface approaches, and the cost. Other important trade parameters such as reliability, flexibility for growth and reconfiguration, development status, operational characteristics, (constructability and erectability), impacts to the vehicle (payload contamination, etc.) were evaluated on a relative basis as opposed to a quantitative basis. All of the concepts assume a redundant system for reliability purposes.

4.2.1 Approaches for Integration and Interfaces

One important issue to be addressed in the concept studies is how the centralized heat transport system interfaces with the heat loads and the thermal requirements of the individual modules. Three approaches were considered in the studies. These approaches are shown in Figures 10 thru 12. The approach shown in Figure 10 is a direct fluid connection approach in which the thermal control systems of the individual modules interface directly into the centralized heat transport loop with fluid connections. These interface fluid connections may be quick disconnects or more permanent type of connections that are applied on-orbit after docking. Only one of the two redundant loops are shown in Figure 10 for clarity. In this concept each module payload heat load is in parallel with the other module heat loads so that the individual modules are thermally isolated from one another. This approach permits the same temperature fluid to be available at each of the individual module heat loads. A means of control is available to each of the individual heat loads with a temperature control valve at the outlet of the heat load. This control may be locally self-contained or it may be a valve

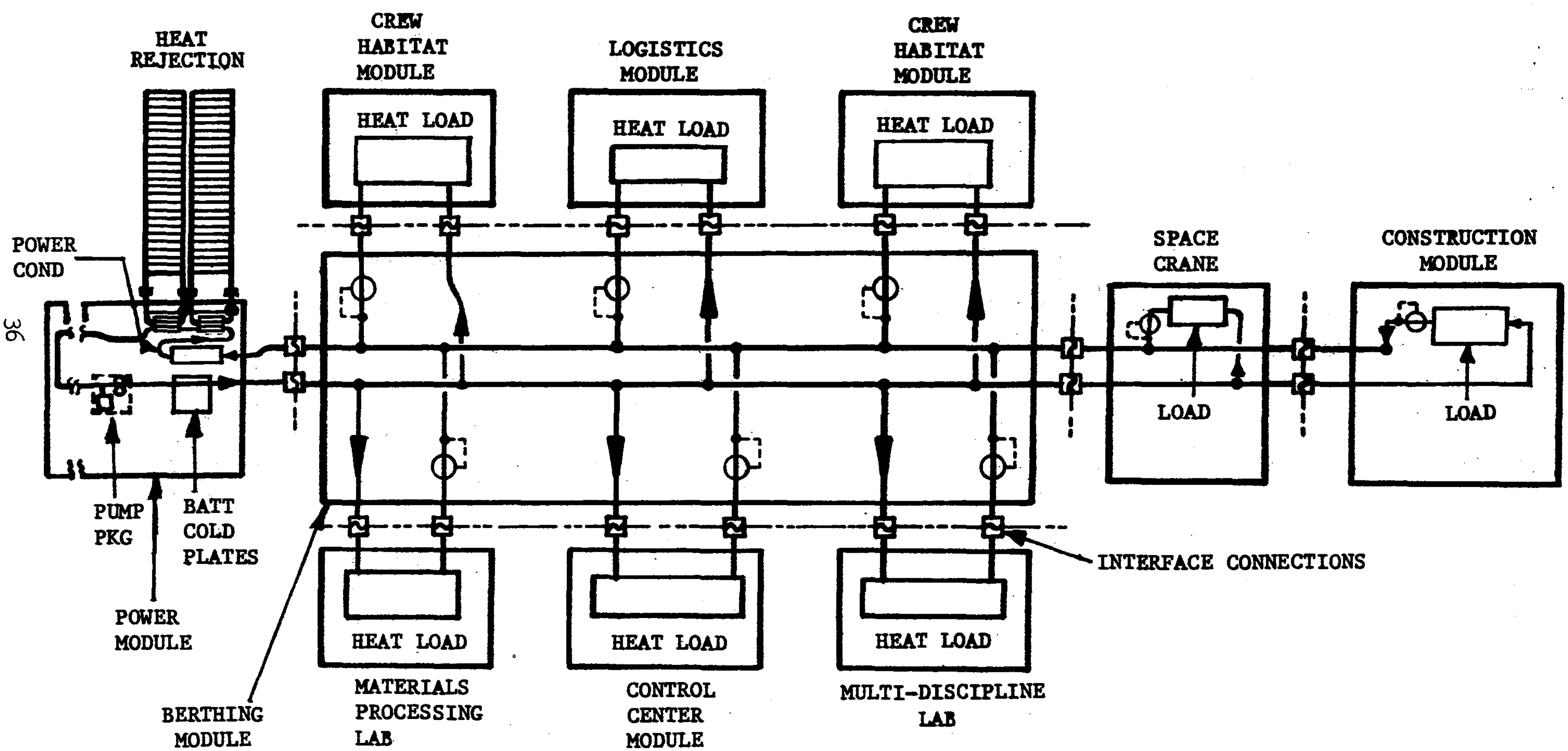


FIGURE 10 **DIRECT FLUID CONNECTION MODULE INTERFACE CONCEPT**

VOUGHT

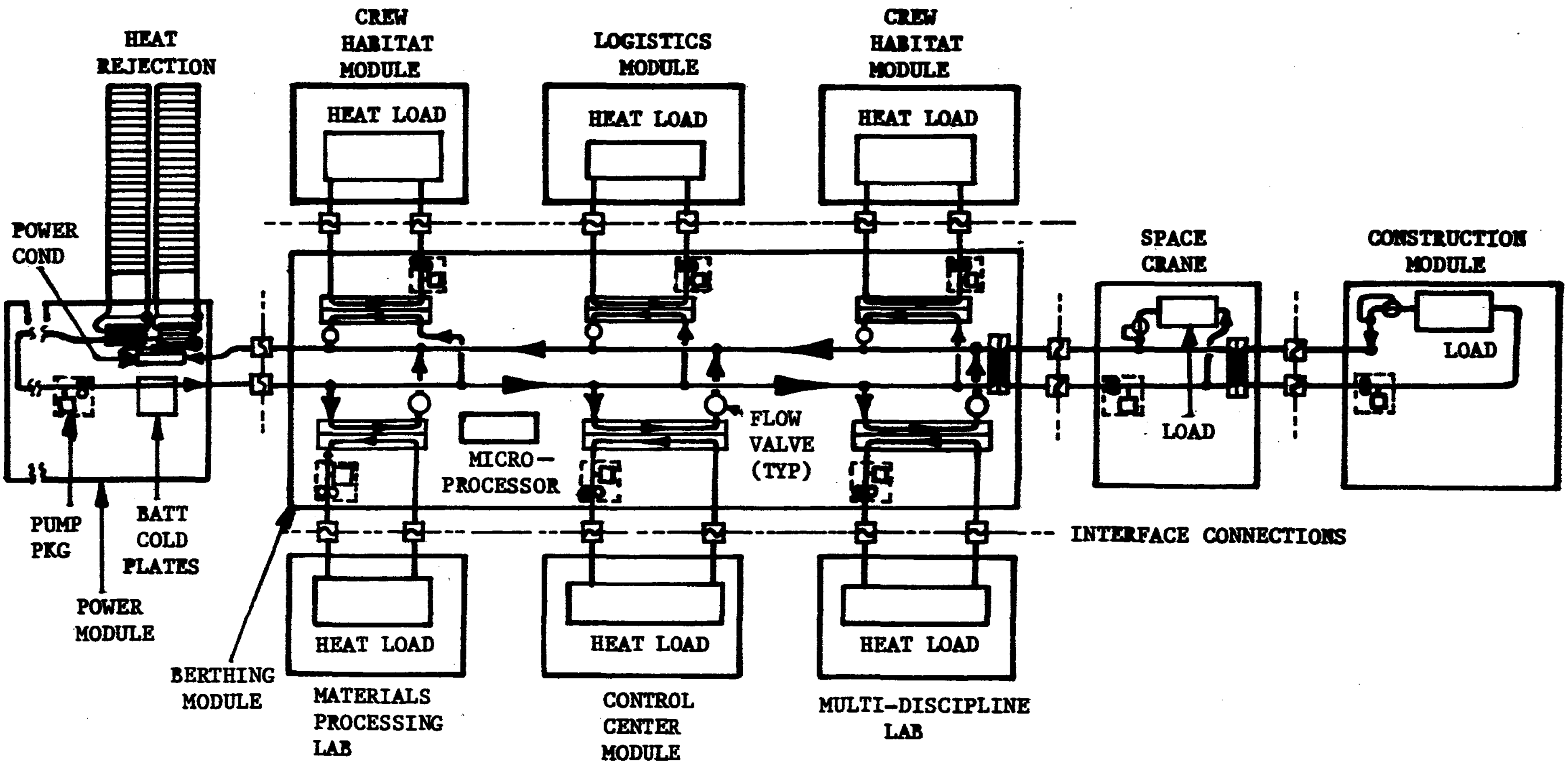


FIGURE 11 **MODULAR HEAT EXCHANGER /FLUID CONNECTION MODULE INTERFACE CONCEPT**

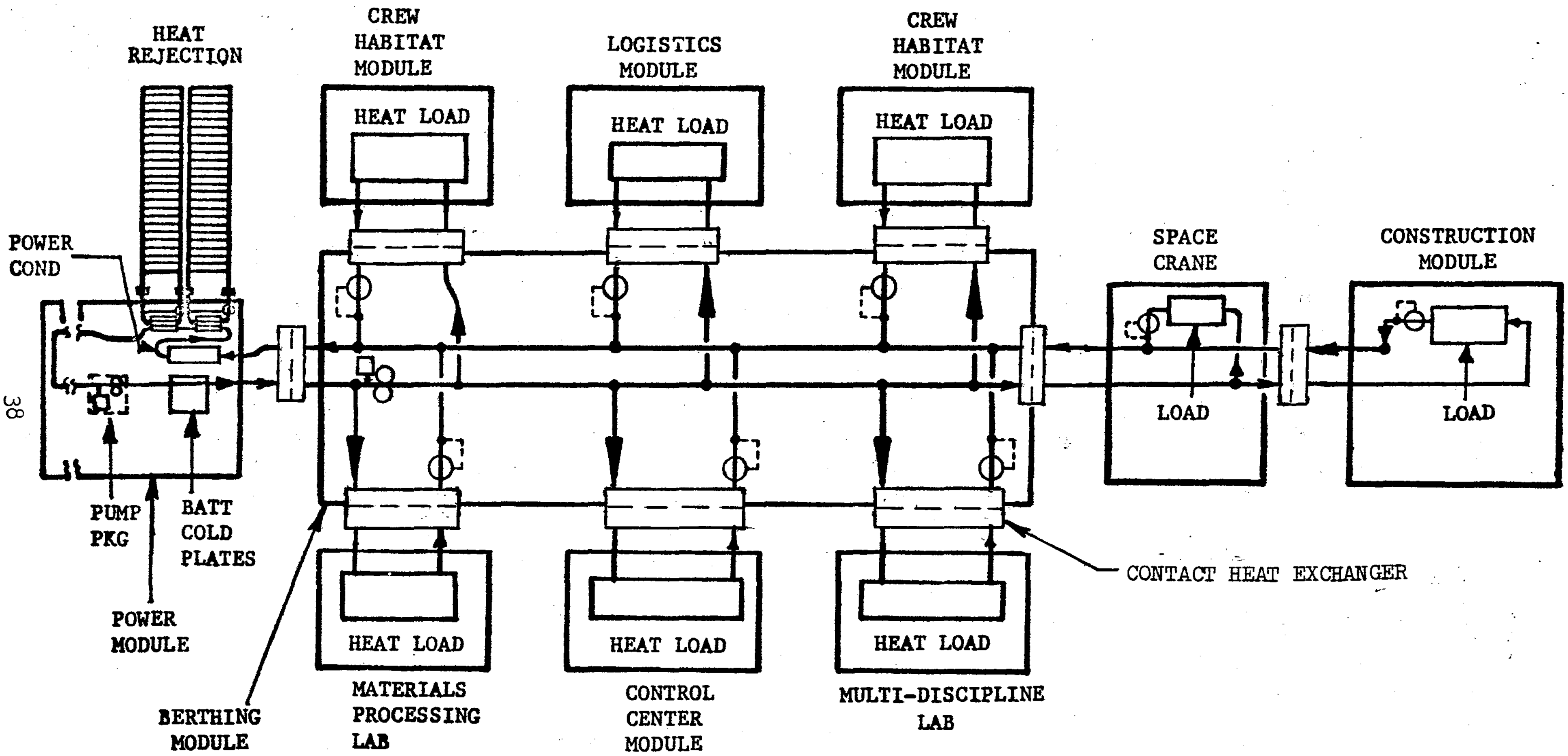


FIGURE 12 INTERFACE CONCEPT NO. 3: CONTACT HEAT EXCHANGER INTERFACE

operated by a microprocessor that maintains control at each of the heat loads. An advantage of the direct fluid connection approach is its flexibility, which permits a wide range of heat load control at each of the individual modules. The microprocessor, in combination with the direct fluid connection approach, could optimize and prioritize the heat loads in case of shortage of cooling capacity. It could also be reprogrammed easily to re-adjust control with changing requirements. The primary disadvantages of the direct fluid connection approach are the reliability aspect of the large number of disconnects connecting the modules. Also, the central loop is not self-contained and a failure in any system jeopardizes all.

Figure 11 illustrates a second approach in which the centralized loop provides cooling to individual modules with interface payload heat exchangers. Again, only one of the two redundant loops required for reliability is shown for clarity. With this approach each module will have its own independent thermal control loop which interfaces with the central loop heat exchanger via quick disconnects. A central loop heat exchanger would be located in the berthing module at each interface port. This approach has the advantage of having the centralized heat transport system contained within the berthing module and with no outside connections and flow loop independent of the payloads. The interface disconnects would be in the individual module loops and thus a failure at that point would not affect the other payload modules. A disadvantage of this approach is the natural limit in the amount of heat transfer possible through a given heat exchanger, thus limiting the flexibility of the system. This limitation can be overcome by sizing each heat exchanger large enough to surpass the desired limit but a weight penalty would result. Each payload would have controls with valves at the outlet on the berthing module side. This control could be either self-contained control or be monitored by a centralized microprocessor as with the previous concept.

A third approach, shown in Figure 12, would eliminate the need for the quick disconnects from both loops. With this approach the interface between the payload modules and the centralized heat transport system would be a contact heat exchanger located at the interface point. Half of the heat exchanger would be contained in the berthing module centralized heat transport loop and the other half would be contained on the payload module temperature control loop. Upon docking the two halves of the heat exchanger would be mated automatically. The primary disadvantage of this approach would be an increase in size of the heat exchangers due to the contact conductance (about 50% to 100% larger). However, it has real advantages in the operational and reliability aspects of the concepts.

Table 10 is a comparison of the advantages and disadvantages of the three approaches. The direct fluid connection approach has the advantages of high thermal efficiency, low weight, low cost and a high degree of flexibility. The disadvantages are poor reliability and the system interaction between modules. The second concept, the fluid-to-fluid heat exchanger concept, has the advantages of allowing flexibility in the design of the thermal control system of individual modules such as high temperature loops, etc., high reliability of central loop, current technology, and lightweight thermally efficient approach. The disadvantages of this approach are the low flexibility in maximum heat load at each docking port, the requirement for additional components on the individual thermal control systems for each module, and the requirement for quick disconnects in the individual module loop. The third concept, the contact heat exchanger, has the advantages of eliminating fluid connections totally, thus, improving reliability and providing more flexibility in the design of the thermal control systems for the individual modules. For instance, an all heat pipe system could be designed for the payload side of the thermal management system while a fluid loop is used on the space platform. Also, a higher temperature fluid loop using a different heat transport fluid could be used on the payload side of the thermal system. This approach simplifies the operation of docking. The primary disadvantage is higher temperature drop across the heat exchanger. It also has the disadvantages of being an undeveloped technology and requiring additional components in the fluid loops of the individual payload thermal control systems.

For the purposes of this study the second concept, which is the centralized fluid loop with heat exchangers at each port, is baselined. This concept was selected primarily because of its higher reliability and state-of-the-art technology. However, it should be pointed out that the contact heat exchanger approach is felt to be the superior of the approaches and is recommended for technology development.

TABLE 10

POWER SYSTEM/PAYLOAD FLUID INTERFACE CONCEPT SUMMARY

CONCEPT	ADVANTAGES	DISADVANTAGES
Direct Fluid Connection	• Best Thermal Efficiency • Lightest Weight • Low Cost • Best System Flexibility For Heat Load Allocations	• Potentially Lower Reliability Due to Fluid Connections • Variable Loop ΔP
*Fluid/Fluid Heat Exchanger	• Allows High Temp Payloads • Simplified Heat Rejection Control • Thermally Efficient • State-of-the-art Tech • Lightweight • Allows Independent P/L Loop Design	• Full Capacity Reqd At Each Port • Requires P/L Pump • Requires Quick Disconnects on Payload Side
Contact Heat Exchanger	• Eliminates Fluid Connections & Leakage Potential • Allows All Heat Pipe TCS for Payloads • Allows High Temp Payloads • Simplified Heat Rejection Control • Allows Independent P/L Loop Design	• Higher Temp Drop • Full Capacity Reqd At Each Port • Requires Developmt • Requires P/L Pump • Higher Weight

*Selected for studies

4.2.2 CONCEPT 1: Redundant Pumped Liquid Loop Concept (Reference Concept)

The reference concept for the heat transport system is shown schematically in Figure 13. This concept consists of a centralized pumped liquid loop which removes heat from the individual payload modules and vehicle heat loads and transports it to a central heat rejection system. The loop interfaces with the various heat loads and the heat rejection system interfaces with heat exchangers. The fluid for this pumped liquid loop is assumed to be water. The entire loop is contained within the Berthing Module and the Power Module. Any of the heat rejection system concepts studied in the next section can be used. However, the fluid for the heat rejection system will probably be one which can withstand the low temperatures required for heat rejection systems, such as Freon 21. Most of these low temperature fluids cannot be used in a habited environment because of toxicity. Different fluids will be needed in the cabins and the heat rejection system. This is the reason for the additional heat exchanger required in the Power Module between the thermal control loop and the heat rejection loop.

A heat exchanger is assumed to be located at each of the 13 ports available for docking on the Berthing Module. The requirements study indicated that the majority of the payload heat loads would fall into the 0 to 25 kW range except for a few special exceptions, such as a materials processing payload. For the purpose of sizing each heat exchanger was assumed to be 25 kW, except for one at 110 kW (for materials processing). Only one of the two redundant systems is shown in Figure 13. Each redundant system is capable of transporting the full heat load. Therefore, the redundant loop is a standby loop with standby pumps, accumulators, temperature control valves, and temperature sensors. Temperature control for each heat load is achieved by a temperature control valve which can either be microprocessor controlled or independently controlled. The temperature of the fluid supplied to each heat exchanger is assumed to be $4^{\circ}C$ in this concept. The fluid temperature in the return line is assumed to be $38^{\circ}C$.

A sizing analysis was performed for the system in which each major component (heat exchangers, lines, etc.) was optimized. Table 11 is a

FIGURE 13

CONCEPT 1: REFERENCE PUMPED LIQUID THERMAL BUSS

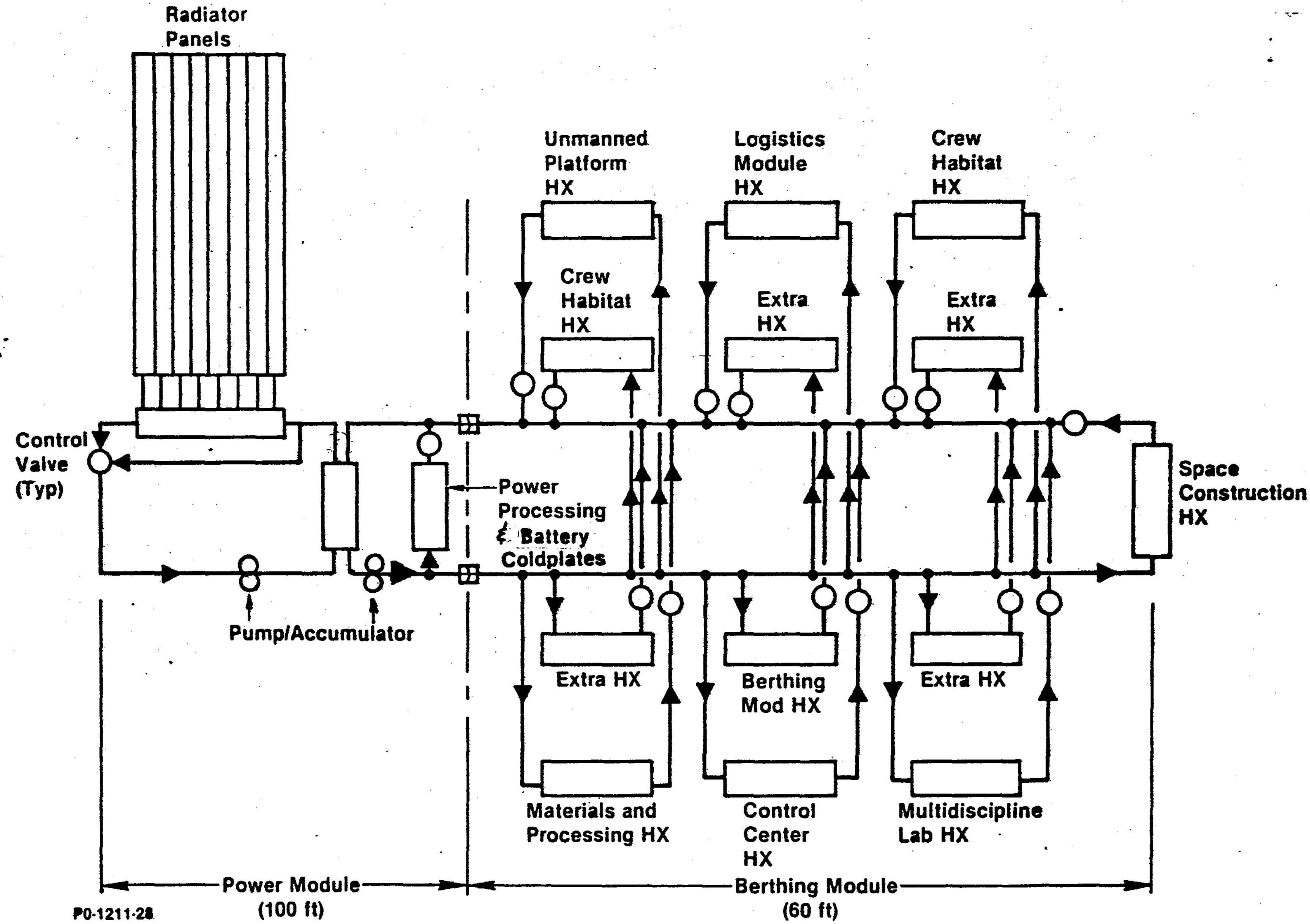


TABLE 11

SUMMARY OF PHYSICAL CHARACTERISTICS
REFERENCE CONCEPT (CONCEPT 1)
SINGLE LOOP PUMPED LIQUID

COMPONENT	WEIGHT (kg) (DRY)	DIMENSIONS	COMMENTS
Battery Coldplates	765	85.5 m^2	
Power Processing Coldplates	26.8	3.0 m^2	
25 kW Berthing Module H/X (12 required)	170 (14.2 ea)	94cm x 14cm x 14cm	
110kW Berthing Module Heat Exchanger	60.4	167.4cm x 23cm x 23cm	
Pump (4)	35.5		
Accumulator (2)	68.2 (Wet)		
Lines and Fittings	68.2	1.3 to 5.3 cm ID	
Radiator Subsystem Delta	0		Integral Manifold Heat Pipe Radiator
Power Equivalent Wt.	33.2		@ 160 kg/kW
Fluid Weight	522.7		Water
TOTAL SYSTEM	1750		

summary of the sizing analysis results. The weight is shown for each of the major components of this concept. The estimated total system weight for this heat transport system is approximately 1750 kg. The major weight elements are battery coldplates, fluid in the system and heat exchangers. "Radiator subsystem delta" weight, shown to be zero here, is a gain or loss to the radiator subsystem as a result of the approach as compared to the reference concept. Since this is the reference concept, the delta is zero. For other candidates it will be some other value.

A cost analysis was performed using the RCA PRICE routine for the Concept 1 heat transport system. The assumptions discussed in Section 4.1 were utilized. These include the complexity factor inputs given in Table 9. The costs analysis results are shown in Table 12. Costs numbers shown are in 1980 dollars, and include both development and production costs. Total cost for a 250 kW heat transport system was \$23.9 Million.

TABLE 12

HEAT TRANSPORT SYSTEM
REDUNDANT PUMPED LIQUID

250 KW

COST-THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Pump/Motor	753	606	1359
Accumulator	1054	32	1087
Temperature Control Valve	704	1816	2520
Coldplates	1712	9997	11709
Temperature Sensors	30	31	61
Heat Exchangers	531	1601	2132
Lines and Fittings	933	18	950
Integration and Test	3406	665	4071
			23900

4.2.3 CONCEPT 2: Pumped Liquid Loop With Multiple Radiator Controlled Temperatures

A flow schematic of the Concept 2 heat transport loop is shown in Figure 15. This concept is a pumped liquid heat transport loop similar to Concept 1 except that heat transport fluid is supplied to the module heat sources at more than one temperature level. Two fluid temperatures are available at each module. A selection valve provides the option for selecting either of the two temperature sources. Only one of two redundant systems are shown for clarity. For system sizing purposes, the two supply fluid temperatures were established as 4^{o}C and 13^{o}C. These were the temperatures that were found to be most prevalent in the Space Platform requirements. Approximately 75% of the heat load was required at 13^{o}C and approximately 25% of the heat load was required at 4.4^{o}C.

An analysis was conducted to optimize component sizes. The results are presented in Table 13. This multiple temperature system weighs approximately 1365 kg, 380 kg less than the reference concept (No. 1). The radiator is more effective than the reference concept due to the split temperature level control, and although the fluid system is heavier because of increased number of lines and fluid, the net weight savings is 380 kg. This system is also more complex than Concept 1, resulting in slightly lower reliability. The system shows a requirement for 85 sq. m of coldplates for the batteries and 3 sq. m of coldplates for the power processing equipment (the same as Concept 1). The 25 kW heat exchangers that are required at each of the 12 interface ports are approximately 94 x 14 x 14 cm. The larger heat exchanger required for the power processing heat load at 110 kW is 167 x 23 x 23 cm. The lines varied in size from 2.5 cm to 4 cm in diameter. The fluid is water for the heat transport system and Freon 21 for the radiator subsystem.

A cost analysis was conducted for Concept 2 using the RCA PRICE routine and with the assumptions discussed in Section 4.1. The estimated development and production costs are shown for each major component in Table 14. The resulting total cost is $24.6 Million dollars. This is close to the cost of the reference concept (Concept 1) being only $0.8 Million more.

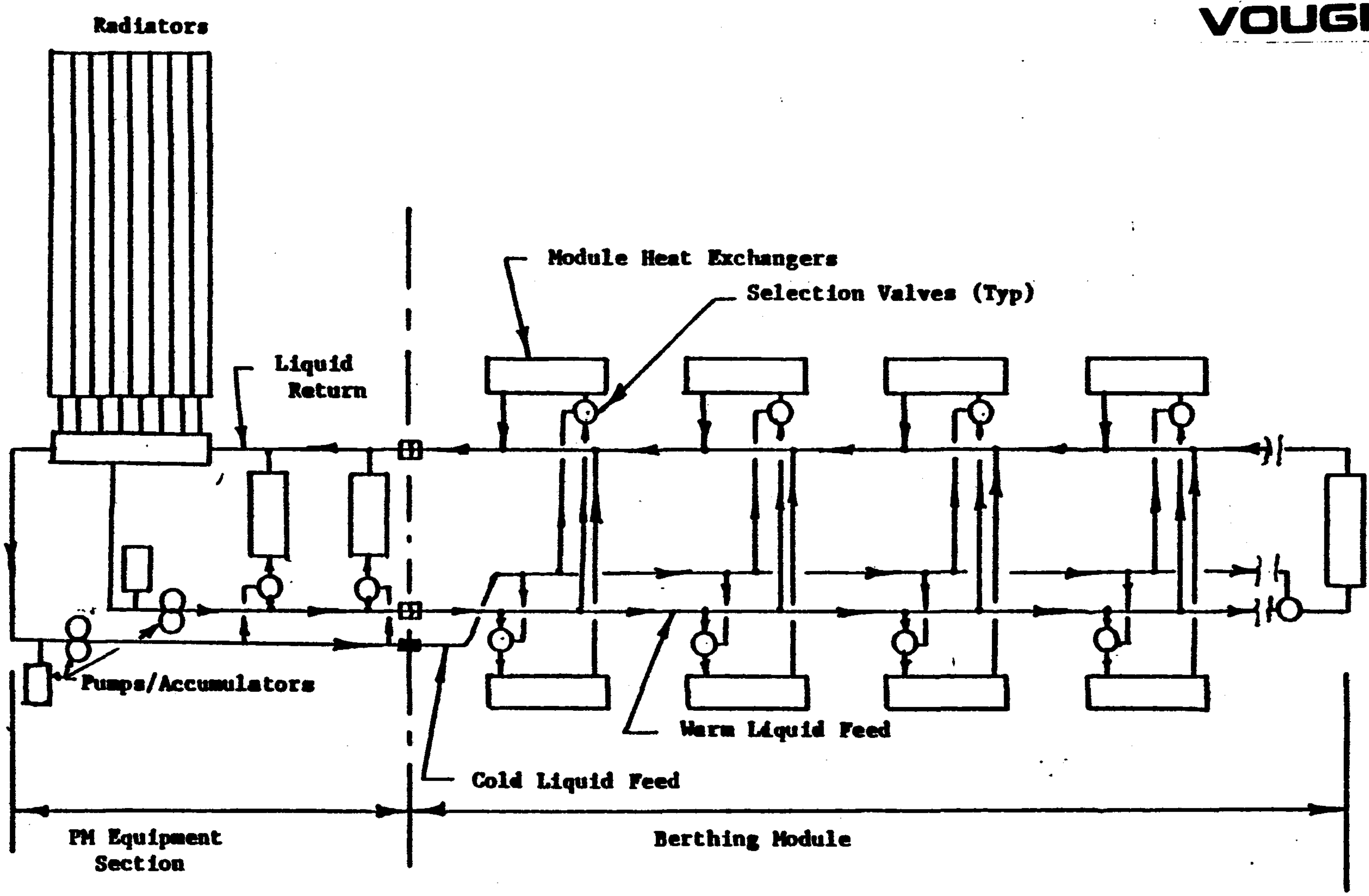


FIGURE 14 CONCEPT 2: SPLIT TEMPERATURE PUMPED LIQUID CONCEPT

TABLE 13

SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 2
SPLIT LOOP PUMPED FLUID 4°C & 13°C

COMPONENT	WEIGHT (KG) (DRY)	DIMENSIONS	COMMENTS
Battery Coldplates	763.4	85.5 m^2	
Power Processing Coldplates	26.8	3.0 m^2	
25 kW Berthing Module Heat Exchanger (12 Reqd)	169.6 (14.2 ea)	94cm x 14cm x 14cm	
110 kW Berthing Module Heat Exchanger	60.3	167.4cm x 23cm x 23cm	
Pump (4)	30.8		
Accumulator (2)	94.8		
Lines & Fittings	94.8	From 2.5cm to 4.5cm I.D.	
Radiator Subsystem Delta	-659.5		
Power Equivalent Weight	29.5		@ 164.2 Kg/kW
Fluid Weight	754.3		Water
TOTAL SYSTEM	1364.9		

TABLE 14

HEAT TRANSPORT SYSTEM
REDUNDANT PUMPED LIQUID, SPLIT RADIATOR TEMPERATURE

COST-THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Pump/Motor	1098	852	1950
Accumulators	1437	70	1507
Temperature Control Valves	484	1227	1711
Temperature Sensors	54	22	76
Lines and Fittings	1224	26	1250
Coldplates	1692	9892	11584
Heat Exchangers	525	1585	2110
Integration and Test	3698	704	4402
			24566

4.2.4 CONCEPT 3: Pumped Liquid Loop with Bottoming Refrigeration Unit

Concept 3, shown in Figure 15, consists of a pumped liquid loop with two temperatures (4°C and 13°C) available at each of the module heat loads similar to Concept 2. Concept 3 differs from Concept 2 in the means for achieving the lower temperature source. In this concept a small vapor compression refrigeration unit is utilized to lower the temperature of the 4°C portion of the loop. The refrigeration waste heat is rejected back into the higher temperature return loop. In this concept the radiator outlet temperature is 13°C as opposed to the 4°C required for the two previous concepts. This permits radiation at a higher temperature. The primary disadvantage of this approach is the power required for the refrigeration cycle and also its complexity and undeveloped nature for space application. Freon 12 was assumed to be the working fluid for the refrigeration loop. The heat transport fluid for this concept was water with Freon 21 for the heat rejection portion of the loop. Valves are available at each heat load heat exchanger to select the temperature required for that particular load.

A sizing analysis was conducted in which all major aspects of the system were optimized. Table 15 summarizes the physical characteristics for the components in Concept 3. The results indicate there is no weight savings achieved by using the refrigeration unit. While there is a savings of 491 kg for the radiator subsystem, the additional mass of fluid and the refrigeration loop subsystem results in an additional 210 kg for this concept over the reference concept. This concept would have advantages, however, in reducing sensitivity to radiator coating degradation due to the higher radiator temperature. This could possibily decrease the radiator weight still further or increase the radiator coating life reducing the maintenance required.

A cost estimate was performed for Concept 3 consistent with the assumptions discussed in Section 4.1. The development and production costs are shown in Table 16. The cost of this concept is $32.4 Million dollars. This is higher than the Reference Concept (Concept 1) by about the amount of the cost of the refrigeration unit ($7.7 Million dollars).

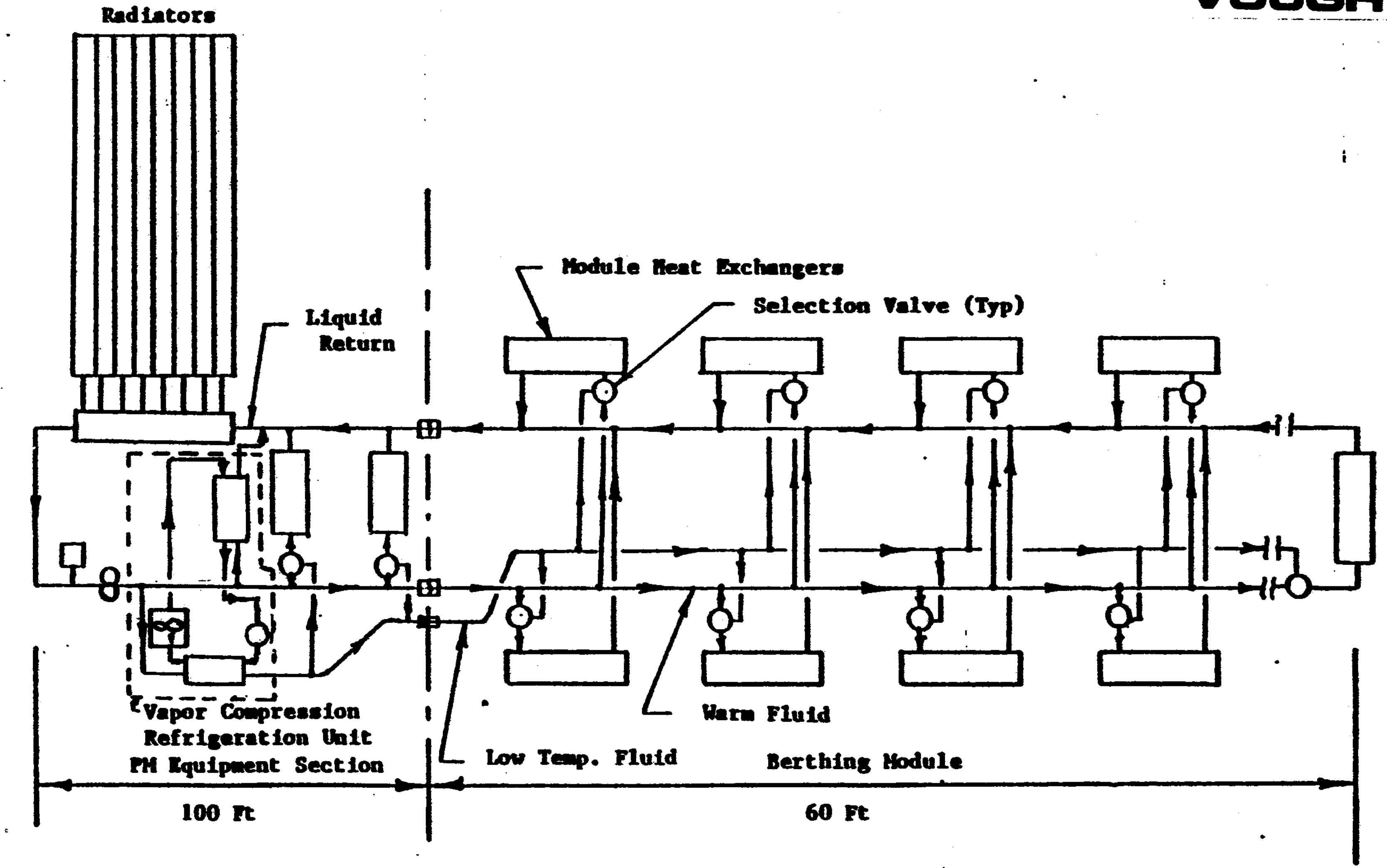


FIGURE 15 CONCEPT 3: REFRIGERATION ASSISTED SPLIT TEMPERATURE PUMPED FLUID CONCEPT

TABLE 15

SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 3
SPLIT LOOP REFRIGERATOR ASSISTED

COMPONENT	WEIGHT (KG) (DRY)	DIMENSIONS	COMMENTS
Battery Coldplates	763.4	85.5 m^2	
Power Processing Coldplates	26.8	3.0 m^2	
25 kW Berthing Module Heat Exchanger (12 Reqd)	169.6 (14.2 ea)	94cm x 14cm x 14cm	
110 kW Berthing Module Heat Exchanger	60.3	167.4cm x 23cm x 23cm	
Pump (4)	30.8		
Accumulator (2)	106.1		
Lines and Fittings	94.8	From 2.5cm to 4.5cm I.D.	
Radiator Subsystem Delta	-490.3		Integral Manifold
Power Equivalent Weight	29.5		
Fluid Weight	813.7		Water
	4.99		Freon 21
Refrigeration Loop Weight	207.3		
TOTAL SYSTEM	1817.1		

TABLE 16

HEAT TRANSPORT SYSTEM
REDUNDANT PUMPED LIQUID WITH BOTTOMING REFRIGERATION LOOP

COST-THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Pump/Motor	1098	352	1950
Accumulators	1437	70	1507
Temperature Control Valves	484	1167	1650
Temperature Sensors	56	16	72
Lines and Fittings	1245	27	1272
Coldplates	1692	9892	11584
Heat Exchangers	525	1585	2110
Refrigeration Unit	3866	3884	7750
Compressor/Motor	(639)	(649)	(1288)
Evaporator	(1411)	(1428)	(2838)
Condenser	(1587)	(1622)	(3209)
Temperature Control Valves	(229)	(185)	(415)
Integration and Testing	3792	718	4510
			32400

4.2.5 CONCEPT 4: Osmotic Heat Pipe System

The osmotic heat pipe is an advanced concept in heat pipes currently being developed by Hughes Aircraft Company for the Air Force. The concept utilizes the forces of osmosis for the pumping of liquid in the heat pipe. Capillary forces are utilized for this purpose in conventional heat pipes. The device holds the promise of providing the capability of an all-heat pipe thermal control system for large spacecraft. This could greatly improve the life and reliability characteristics of the system.

The preliminary sizing analysis on the concept using the best projections of currently available data indicated that the osmotic heat pipe would not be weight competitive. Based upon our preliminary analysis, the concept was not considered further. The device is still in the laboratory proof-of-principle stage of development and considerable Research and Development is needed to reduce the principle to a practical, competitive device.

Figure 16 illustrates the principle of the osmotic heat pipe. The major elements of a osmotic heat pipe are the membrane, two fluids, (one a solvent and the other a solution of the solvent and a solute) and two heat exchangers (a condenser section and an evaporator). The fluid is circulated in a closed loop with the membrane acting as the pumping unit. The solvent is on one side of the membrane and the concentration of the solvent and solute is on the other. Solvent migrates through the osmotic membrane into the solution of solvent and solute as a result of osmotic pressure. In the evaporator section, the solvent is evaporated, leaving the solute behind, and then migrates to the condenser section due to the pressure differences in the pipe. The solvent is condensed in the condenser section providing the liquid solvent on the upstream side of the osmotic membrane thus completing the cycle. The solute which was left behind at the evaporator section must migrate back against the direction of flow to the osmotic membrane.

Laboratory tests have been conducted on the osmotic heat pipe by the Hughes Aircraft Company with some promising results. High "deadhead" (no flow) pumping pressures on the order of tens of atmospheres can be achieved. However, the osmotic heat pipe is still in the laboratory stage and there are

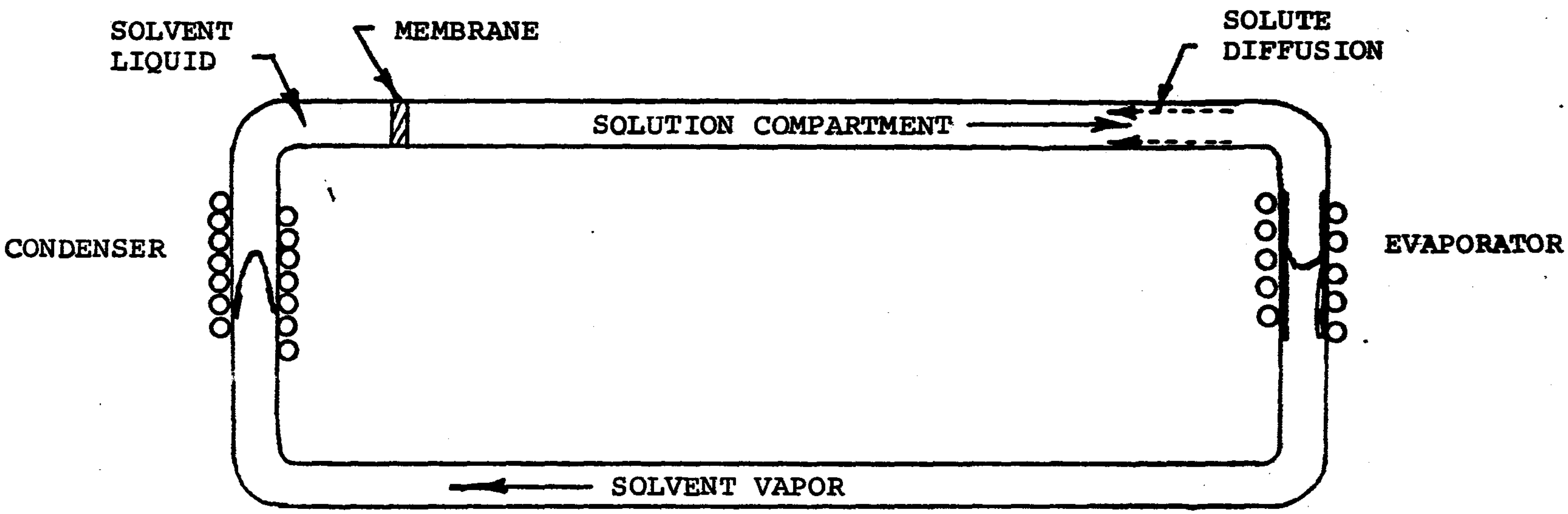


FIGURE 16
SCHEMATIC OF OSMOTIC HEAT PIPE

many questions to be answered in the areas of materials, life, weight, etc. Figure 17 shows a schematic of a heat transport system utilizing the osmotic heat pipe concept. In this approach an osmotic membrane is located just upstream of each heat load. The solvent and solute would be contained in the heat exchanger of each heat load. The heat load evaporates the solvent which flows back to the radiator system in the vapor phase and condenses there. The solvent is then circulated back through the return line to the osmotic membrane.

One important unresolved issue for the osmotic heat pipe is solvent/solute selection. Tests to date have been conducted using a water solution with sucrose or sodium chloride. However, neither of these two solutes appear to be suitable for use in a spacecraft because of the corrosive nature of the sodium chloride and the tendency for fermentation in the sucrose. Also important is the membrane material. The membrane material characteristics are critical to the heat pipe performance. The objective in evaluating candidate membrane materials is to obtain membranes with high solvent flowrates while allowing no solute leakage. In addition, the membranes should have capability for long life at temperatures of 50° to 70°C. Membranes developed todate have been for use in water purification and thus, they are water compatible at nominal temperatures. Some candidate materials that have been evaluated include cellulose acetate, polyethyleneimines, polyamides, polybenzimidazales (PBI), sulfonated polysulfone (SPS), and sulfonated polyfurfuryl alcohol (SPFA). Materials evaluations are currently being conducted by the Hughes Aircraft Company.

In addition to the material development, some areas in which development is needed are (1) containment and management of the solvent in the zero-g environment, (2) wicking of the solute from the evaporator back to the membrane without clogging or deposition of the solute, and (3) lower weight designs of the membranes.

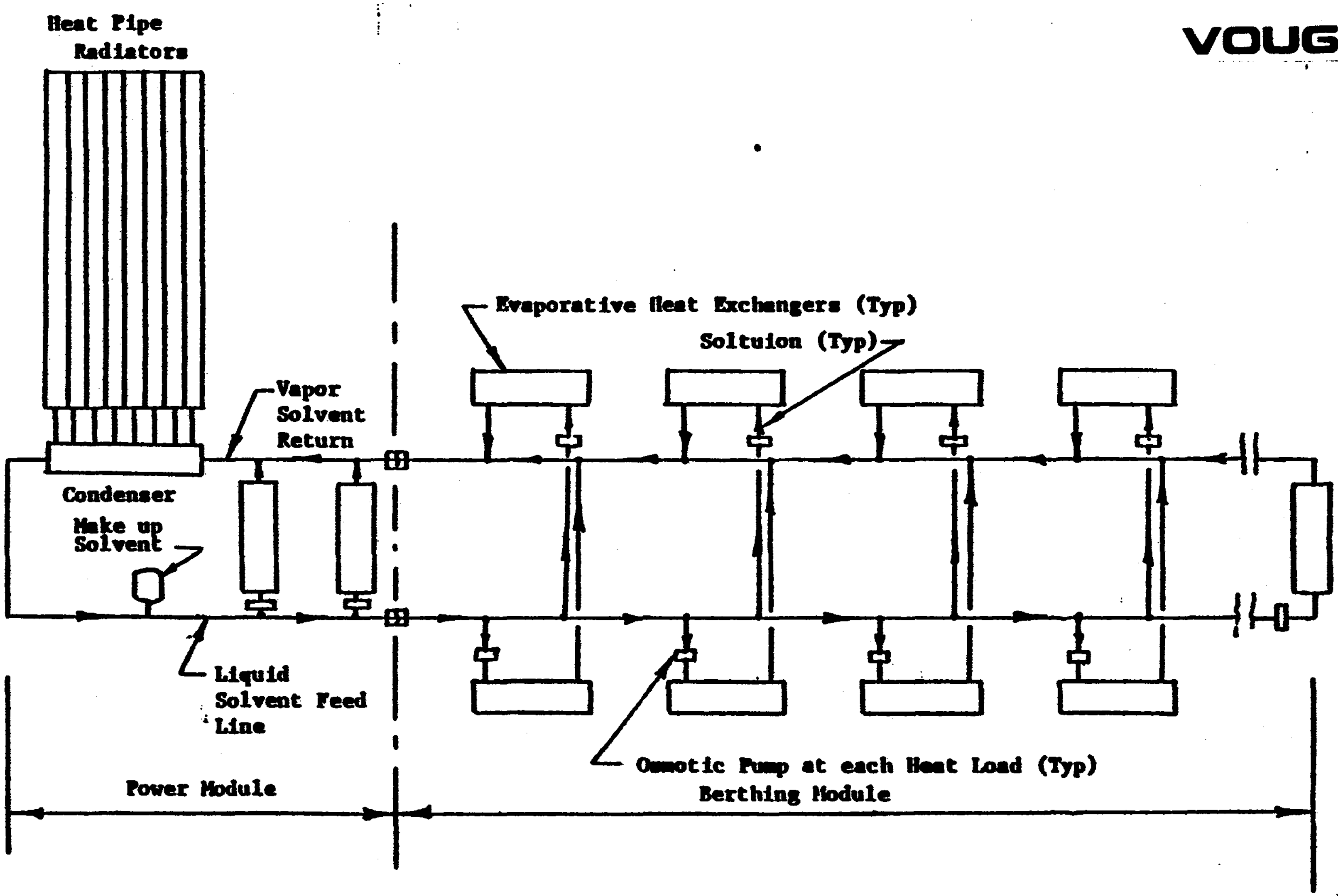


FIGURE 17 CONCEPT 4: OSMOTIC HEAT PIPE ALL HEAT PIPE CONCEPT

4.2.6 CONCEPT 5: Pump Driven Heat Pipe - Single Loop

Concept 5 is a two phase heat transport concept in which the heat transfer into or out of the loop is achieved by evaporation or condensation of a working fluid. The prime mover for the fluid is a pump located in the liquid portion of the loop. A schematic of the concept is shown in Figure 18. In this concept the heat load is removed from the individual modules through evaporative heat exchangers. Heat is added to the heat transport loop by evaporating the fluid in the loop evaporators. The vapor from the heat exchanger is fed into the vapor return line and returns to the radiator subsystem where it is condensed. The liquid that comes from the radiator subsystem is then circulated back to the heat loads with a liquid pump. As with all the heat transport concepts, redundant systems are assumed for reliability.

An important consideration in any of the two phase concepts is the choice of fluid. Some of the important properties of these fluids are:

- Safe for use in the cabin environment (manned cabins)
- Good thermal properties
- Low weight

A study was conducted for Concept 5 to determine the best fluid for use in the manned cabin environment. Ten fluids, listed in Table 17, were evaluated on the basis of minimum system weight. Table 17 shows the fluid properties for the candidate fluids. Also shown are the liquid and vapor pressure drop parameters, ζ_L and ζ_V . The higher the factor, the higher the pressure drop. These two parameters were determined through analysis to be indicators of system weight. Fluids were also compared on the basis of vapor pressure and safety in the cabin. Safety concerns are those of toxicity and flammability. Based on this cursory analysis of these candidate fluids, Freon 114 was determined to be the best fluid for this application and was used in the sizing analysis.

The results of the sizing analysis for concept 5, shown in Table 18, indicated that many of the components of the thermal control system could be smaller in size due to the high heat transfer rates in the condensing flow. However, because all the heat is rejected at the minimum system

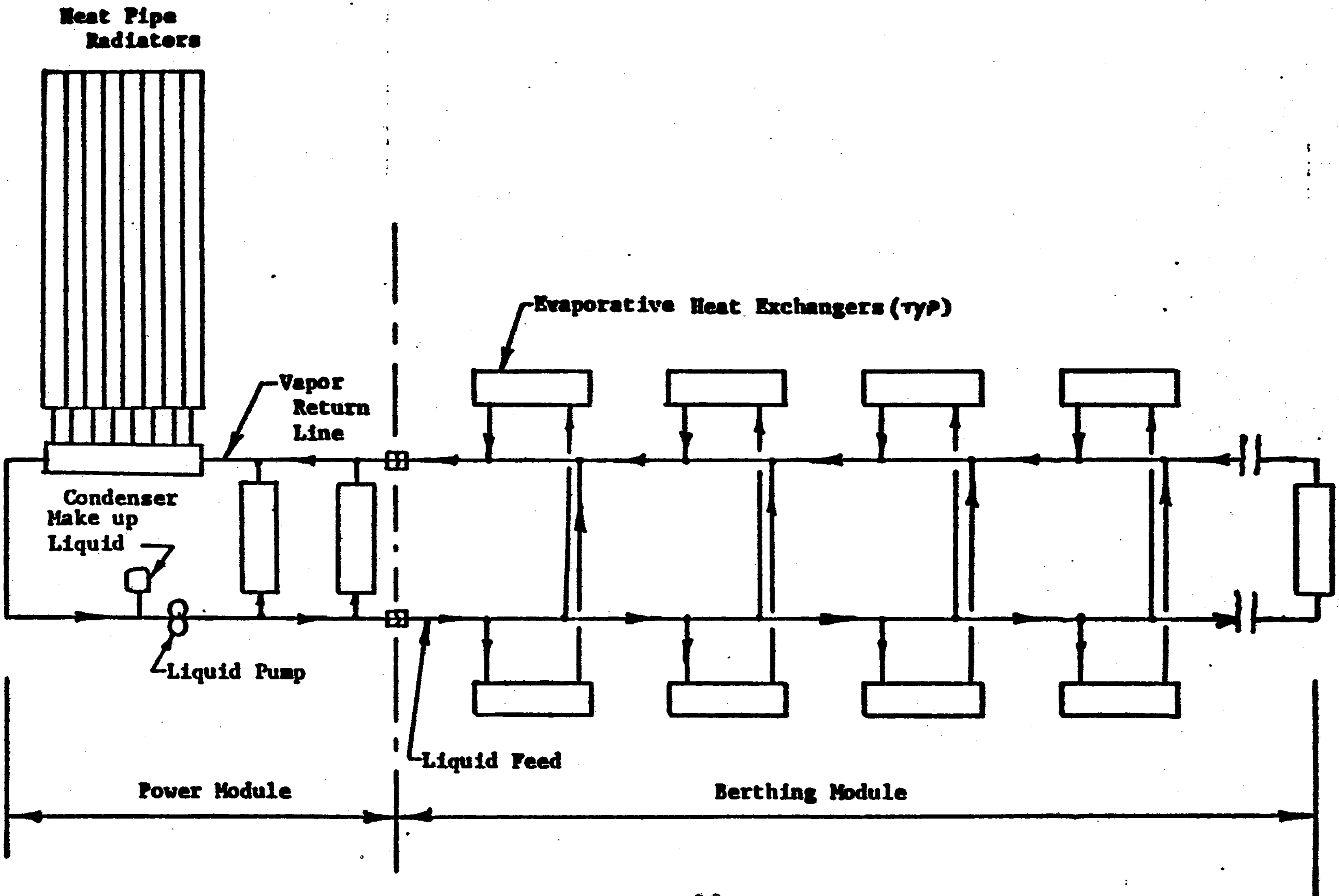


FIGURE 18
CONCEPT 5: PUMP DRIVEN ALL HEAT PIPE CONCEPT

TABLE 17

CANDIDATE FLUID COMPARISON FOR PUMP DRIVE HEAT PIPE

FLUID	T °F	ρ_V LBM/FT^3	λ BTU/LBM	μ_V LB/FT-HR	P_V PSIA	μ_L LBM/FT-HR	ρ_L LBM/FT^3	Cp BTU/LBM-°F	k_ℓ BTU/HR/FT°F	ζ_L* $\frac{\mu^{.25}}{\rho(\lambda)^{1.75}}$	ζ_V** $\frac{\mu^{.25}}{\rho(\lambda)^{1.75}}$
AMMONIA	40	.25	536.2	.02522	71.82	0.7204	39.64	1.115	.306	3.381×10^{-7}	2.668×10^{-5}
AMMONIA	60	.363	518.1	.02680	108.68	0.3627	38.44	1.137	.219	3.627×10^{-7}	1.991×10^{-5}
FREON 114	40	.501	58.435	.00024	15.08	.012156	94.1974	.231	.0375	2.8542×10^{-6}	2.0116×10^{-4}
FREON 114	60	.732	56.735	.00024	22.57	.010615	92.6870	.237	.0356	2.9527×10^{-6}	1.4498×10^{-4}
N-BUTANE	40	.200	163.8	.0001690	17.62	.00494	37.216	.540	.06549	9.498×10^{-7}	7.601×10^{-5}
N-BUTANE	60	.288	159.8	.0001757	26.00	.00446	36.443	.540	.06351	9.873×10^{-7}	5.566×10^{-5}
ISO-BUTANE	40	.306	149.7	.000170	26.48	.00510	35.93	.580	.06467	1.1609×10^{-6}	5.824×10^{-5}
ISO-BUTANE	60	.432	144.5	.0001771	38.04	.004565	35.11	.580	.06307	1.2293×10^{-6}	7.885×10^{-5}
ACETONE	40	.01947	236.32	.022403	1.624	.9288	50.069	.5086	.1038	1.3765×10^{-6}	1.395×10^{-3}
ACETONE	60	.03159	255.91	.020565	2.814	.81716	49.282	.5141	.1006	1.4656×10^{-6}	7.321×10^{-4}
METHANOL	40	.00414	519.16	.021595	.7265	1.83736	50.312	.5828	.1205	4.098×10^{-7}	1.621×10^{-3}
METHANOL	60	.00825	513.348	.02265	1.4252	1.52868	49.526	.5881	.1185	4.055×10^{-7}	8.494×10^{-4}
FREON 21	40	.23786	104.791	.02490	12.180	.86303	86.823	.24989	.06546	3.226×10^{-6}	4.866×10^{-4}
FREON 21	60	.34767	102.325	.02606	18.372	.94627	82.366	.2444	.06866	3.3186×10^{-6}	3.510×10^{-4}
FREON 11	40	.28094	78.7449	.002492	10.875	.010716	93.645	.205	.05606	1.6506×10^{-6}	3.821×10^{-4}
FREON 11	60	.32776	77.8843	.002516	13.485	.010184	93.021	.207	.05537	1.6725×10^{-6}	3.346×10^{-4}
PROPANE	40	7339	158.867	.000184	78.155	.00326	32.726	.6032	.06127	3.2723×10^{-6}	2.232×10^{-5}
PROPANE	60	1.0201	150.949	.001900	108.866	.00291	31.658	.6228	.05838	3.5708×10^{-6}	3.148×10^{-5}
WATER	40	.000409	1073.814	.0196114	.1217	3.7134	62.555	1.005	.3341	1.102×10^{-7}	4.542×10^{-3}
WATER	60	.000829	1059.60	.0213413	.2563	3.170	62.34	1.000	.344	1.088×10^{-7}	2.343×10^{-3}

* ζ_L – LIQUID PRESSURE DROP PARAMETER

** ζ_V – VAPOR PRESSURE DROP PARAMETER

TABLE 18

SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 5

COMPONENTS	WEIGHT (KG)	DIMENSIONS
Battery Coldplates	771.1	86.4 m^2
Power Processing Coldplates	27.2	3 m^2
25 kW Berthing Module Heat Exchanger (12 Required)	53.1 (4.43 ea)	18.5cm x 42.7cm x 42.7cm
110 kW Berthing Module Heat Exchanger	19.5	30.2cm x 42.7cm x 42.7cm
Pump	1.8	3 m^2
Accumulator (2)	68.0 (Wet)	
Lines and Fittings	181.4	2.4cm I.D. to 26.7cm I.D.
Radiator Subsystem Delta	2425.4	
Power Equivalent Weight	15.9	@ 164.2 kg/kW
Fluid Weight (Freon 114)	498.9	
TOTAL SYSTEM WEIGHT	4062.4	

temperature of $4^{o}C$ the, radiator subsystem weight becomes very large (2270 kg larger than Concept 1, the Reference Concept). Also, the lines for this system are much larger due to the vapor flow and are considerably heavier than the baseline system. Total weight for this concept is about 4100 kg, or more than twice as much as the Reference Concept (Concept 1).

In an attempt to reduce mass, dual temperature loops for the heat transport system were examined. One loop is at approximately $15^{o}C$, the other is at approximately $4^{o}C$. The heat load was approximately 75% at the $15^{o}C$ temperature and 25% at the $4^{o}C$ temperature. A schematic of the radiator portion of this approach is shown in Figure 19. A sizing and optimization study was conducted for this approach (Table 19). The lines and fittings weights and the radiator subsystem weight were reduced to approximately half of that required for the single temperature loop. The total system weight for the two temperature loop concept is approximately 455 kg heavier than Concept 1, but has the advantage of isothermal heat transfer. If higher temperature levels can be identified on the platform, three or more discrete loops could reduce the weight even further.

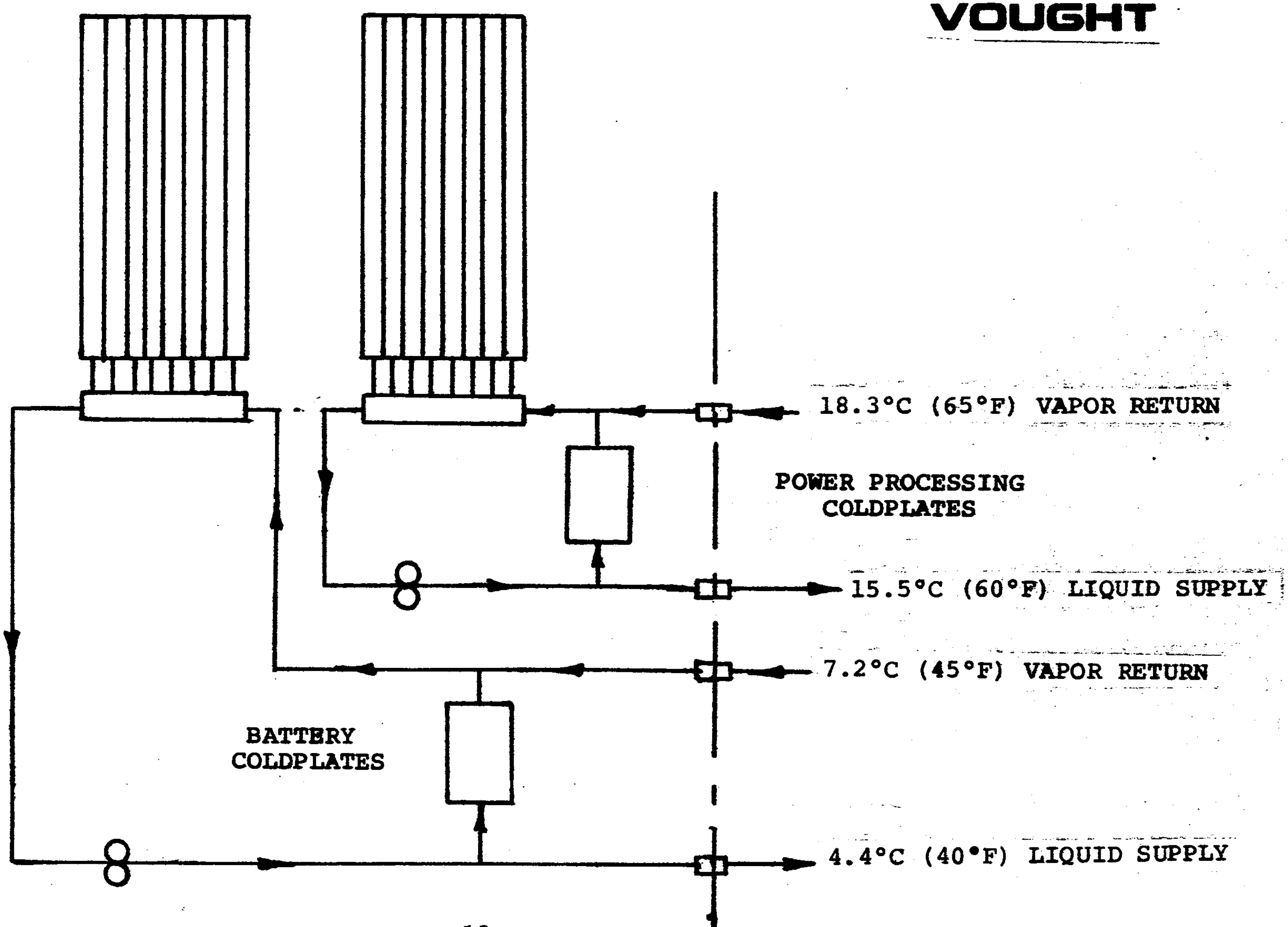


FIGURE 19
CONCEPT 5A: DUAL THERMAL PUMPED HEAT PIPE CONCEPTS

TABLE 19

SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 5A
ACTIVE PUMP DRIVEN ALL HEAT PIPE CONCEPTS

COMPONENTS	WEIGHT-KG	DIMENSIONS
Battery Coldplates	771.1 (Dry)	85.8 m^2
Power Conditioning Coldplates	27.2 (Dry)	2.98 m^2
25 kW Heat Exchangers (12 Reqd)	51.3 (Dry) (4.53 ea)	
110 kW Heat Exchanger	20.4 (Dry)	
Pump	3.6 (Dry)	
Accumulators (2)	68.0 (Dry)	
Lines and Fittings (15°C)	53.1 (Dry)	.02m - .06m ID x 60.96m Long
Lines and Fittings (4°C) Long	31.3 (Dry)	.01m - .04m ID x 60.96m
Radiator System Delta	1094.1	
Pump Power Equivalent Wt.	15.9	@ 164.2 kg/kW
Fluid Weight (Freon 114)	75.7	
TOTAL SYSTEM WEIGHT	2215	

4.2.7 CONCEPT 6: Compressor Driven Heat Pipe System

Concept 6 is another two phase heat transport concept. It contains a working fluid which is evaporated at the payload module heat load heat exchangers and condensed at the radiator portion of the heat transport loop. It is similar to Concept 5, except the prime mover is a compressor or fan located in the vapor portion of the heat pipe. Figure 20 is a schematic of this concept. The working fluid for this concept was assumed to be Freon 114. Because of the location of the compressor in the loop, the pressure tends to be highest at the condenser thus allowing the condensation to occur at higher temperatures and reducing the size of the radiator. (The opposite effect occurs for the pump driven concept where the lowest pressure in the loop is in the condenser section making the radiator large.) A major disadvantage of a compressor driven heat pipe concept is the large amount of power required by the compressor. A summary of the sizing analysis for Concept 6 is provided in Table 20. The results show that the radiator system is significantly lighter than the Reference Concept but a tremendous amount of weight (19500 kg) is required for the power system to drive the compressor, assuming the specific power weight is 165 kg/kW (360 LBm/kW based on current power system designs). The power system weight to drive the compressor is 5450 kg if the power specific weight of 45 kg/kW (100 LBm/kW) is assumed.

A modification to Concept 6, Figure 21, was analyzed in which cooling was provided at two temperature levels. The two temperature levels assumed were 16°C for 3/4 of the heat load and 4°C for 1/4 of the heat load. The liquid portion of the loop was common for the two systems, but the vapor portion was separate and at two different pressures. An expansion valve will be required at each heat load which could be set at either of the two temperature levels. The weight analysis (see Table 21) for this concept also indicates that a significant weight savings can be achieved over the single temperature level concept. The system is shown to be weight competitive with the Reference Concept, Concept 1, if the power system equivalent weight is 45 kg/kW. However, it is 4 times as heavy as the Reference Concept for a power equivalent weight of 160 kg/kW. Thus we see that this concept is very sensitive to power system weight.

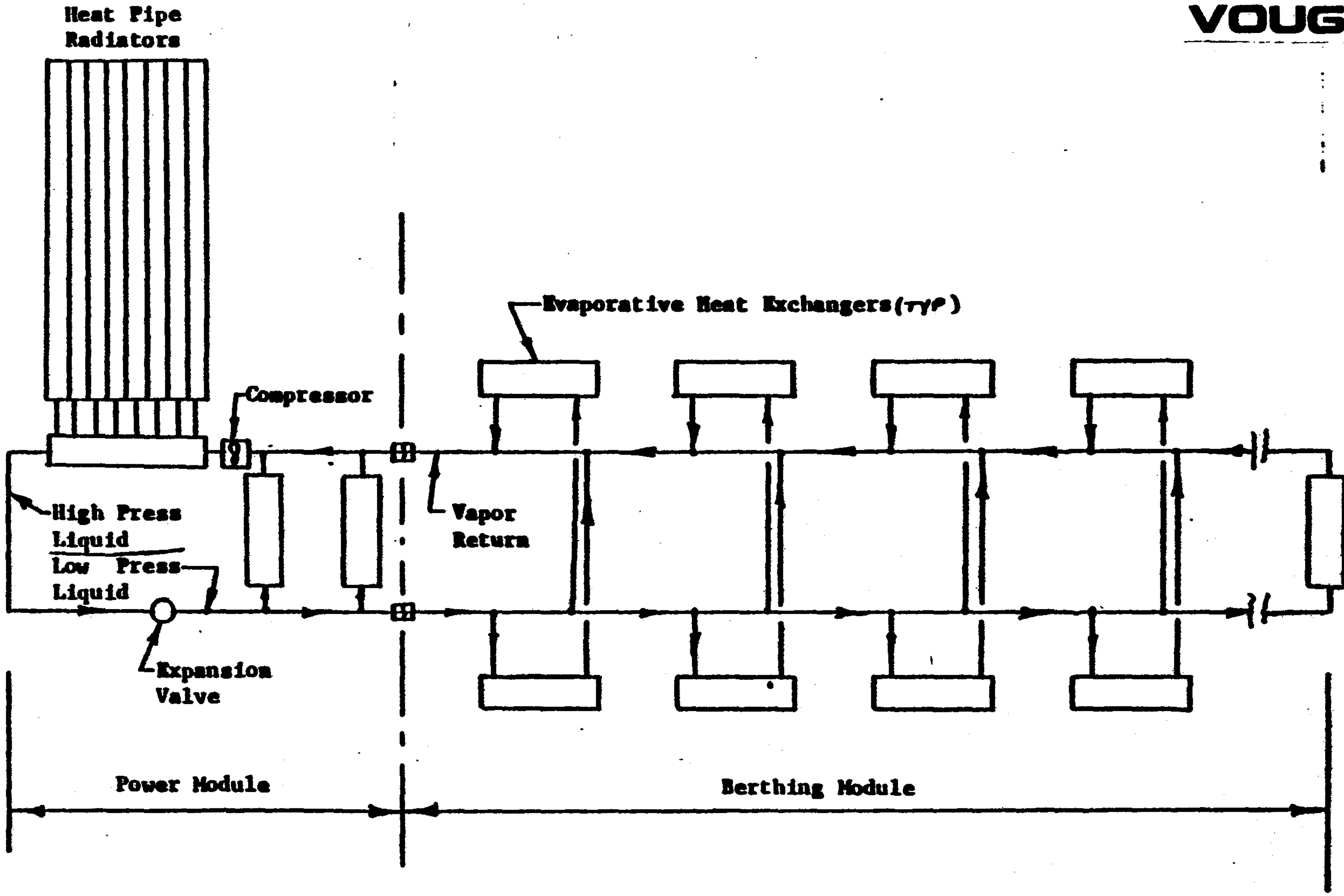


FIGURE 20 CONCEPT 6: COMPRESSOR DRIVEN HEAT PIPE/REFRIGERATION SYSTEM

TABLE 20

SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 6

COMPONENTS	WEIGHT (KG)	DIMENSIONS	COMMENTS
Battery Coldplates	771.1 (Dry)	85.84 m^2	
Power Processing Coldplates	27.2 (Dry)	2.98 m^2	
25 kW Berthing Module Heat Exchanger (12 Required)	712.6 (Dry) (59.4 ea)	.2057m O.D. Shell .8656m Long	
110 kW Berthing Module Heat Exchanger	104.8 (Dry)	.2718m O.D. Shell 1.018m Long	
Pump (Compressor, Centrif.)	95.3 (Dry)	-	
Accumulator (2)	None	-	
Lines and Fittings	148.8 (Dry)	.0356m I.D. -.0229m I.D. .1778m ID Vaporline	
Radiator Subsystem Delta	-1829.3	18.29m Fins, .2098m W, .864m Thick, .9144m HX Lengths	
Power Equivalent Wt.	19504.5 5352.4	None	@ 164.2 kg/kW @ 45.3 kg/kW
Fluid Weight	678.6	None	Water
TOTAL SYSTEM WEIGHT	20276 (164.2kg/kW) 6166 (45.3kg/kW)		

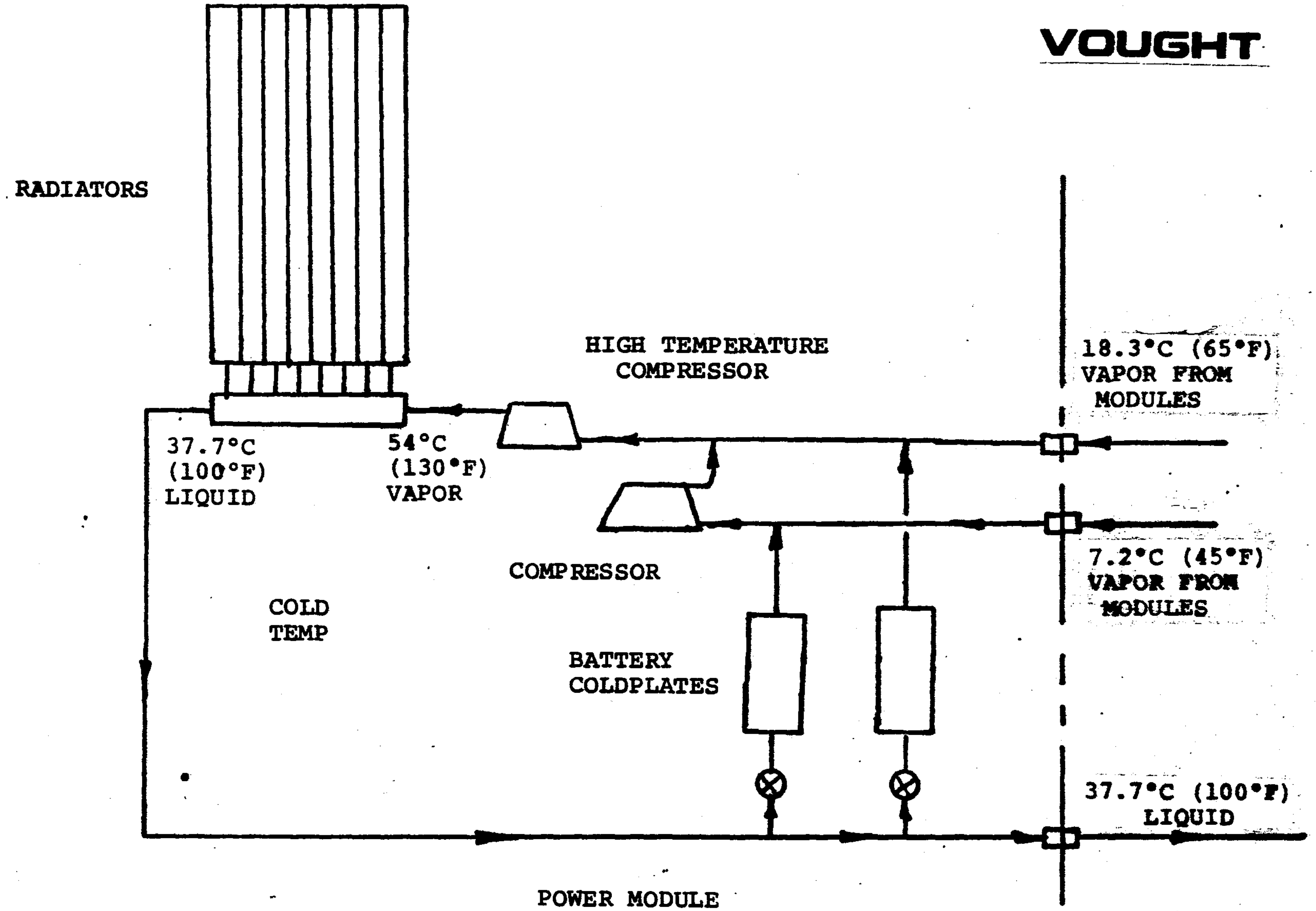


FIGURE 21
CONCEPT 6A: COMPRESSOR DRIVEN THERMAL BUS WITH DUAL TEMPERATURE CONTROL

TABLE 21
SUMMARY OF PHYSICAL CHARACTERISTICS
CONCEPT 6A
ACTIVE COMPRESSOR DRIVEN ALL HEAT PIPE CONCEPT

COMPONENT	WEIGHT-KG	DIMENSIONS
Battery Coldplates	771.1	84.84 m^2
Power Conditioning Coldplates	27.2	2.98 m^2
25 kW Heat Exchangers (12 Reqd)	55.8 (4.63 ea.)	
110 kW Heat Exchanger	20.4	
Compressor - 56.3 HP 6.8 HP	24.9 10.9	Centrifugal Centrifugal
Pipe and Fittings	25.3	5.3cm to 14.9cm ID
Radiator System Delta	-1829.3	
Pumping Power Equivalent Weight	7718.8 2131.9	@ 164.2 kg/kW @ 45.4 kg/kW
Fluid Weight (Freon 114)	408.2	
TOTAL SYSTEM WEIGHT	7282.9 kg @ 164.2 kg/kW 1696.0 kg @ 45.4 kg/kW	

4.2.8 Heat Transport Concept Trades and Selection

The heat transport system approaches were compared against a set of trade criteria. There were six major categories for criterion. These were: Potential for Benefit, Development Considerations, Operational Considerations, Impact to Vehicle, Performance Considerations, and Reliability and Life Considerations. The seven concepts which were evaluated are shown in Table 22. Table 23 shows the trade matrix which compares the concept for each of the trade criteria. The comparisons for each of these are discussed separately below.

Potential For Benefit

Cost analysis was only performed on the pumped liquid concepts. As expected, they were the lowest in cost because of their advanced development status. Of these three systems, Concept 1 had the lowest cost. There was very little increase in the cost for Concept 2, the split temperature pumped liquid loop. Concept 3, the Refrigeration Assisted Split Temperature Pumped Liquid Loop cost about 30% to 35% more due to the cost of the refrigeration system. A comparison of all seven concepts for the other criteria under this category shows no clearcut advantages. Thus, the Reference Concept (Concept 1) and the split temperature pumped liquid concept (Concept 2) appear to have the advantage in this category.

Development Considerations

Comparison of the concepts on the basis of this category shows a clear advantage for the more highly developed concepts, Concepts 1 and 2. Concept 3, the refrigeration assisted pumped liquid loop, was also rated fairly high. However, it cost an additional $4 Million to develop the refrigeration system. Based on a comparison of the development status, Concepts 1 and 2 are again clear winners.

Operational Considerations

No clear advantage appeared to exist for any of the concepts for this category.

Impacts

The primary differences between the concepts for the criteria in this category are their effect on radiator area. This area difference causes impacts in such criteria as orbital drag, moment-of-inertia, stowage volume,

TABLE 22

MOST PROMISING HEAT TRANSPORT CONCEPTS

1. REDUNDANT PUMPED LIQUID LOOP (REFERENCE CONCEPT)

2. REDUNDANT PUMPED LIQUID LOOP WITH SPLIT RADIATOR OUTLET TEMPERATURES

3. REDUNDANT PUMPED FLUID LOOP WITH BOTTOMING REFRIGERATION UNIT

5. PUMP DRIVEN HEAT PIPE, SINGLE LOOP

5A. PUMP DRIVEN HEAT PIPE, DUAL LOOP

6. COMPRESSOR DRIVEN HEAT PIPE, SINGLE LOOP

6A. COMPRESSOR DRIVEN HEAT PIPE, DUAL LOOP

TABLE 23
CONCEPT TRADE MATRIX FOR HEAT TRANSPORT
SYSTEMS RANKED CRITERIA

RANKED CRITERIA	ORDER OF PRIORITY	CONCEPT NO. 1	2	3	5	5A	6	6A
POTENTIAL FOR BENEFIT								
. Cost $M	1	23.9	24.60	32.4	-	-	-	-
. Operations	1	No	No	No	No	No	No	No
. Integration with Other Systems	2	Yes	Yes	Yes	Yes	Yes	Yes	Yes
. Growth and Reconfiguration	1	Good	Good	Good	Fair	Fair	Fair	Fair
. Autonomous Operation	1	Yes	Yes	Yes	Yes	Yes	Yes	Yes
. Reduced Impacts	2	No	No	No	No	No	No	No
. Long Life	1	Good	Good	Fair	Good	Good	Fair	Fair
DEVELOPMENT CONSIDERATIONS								
. Costs $M	1	9.1	10.2	14.2	-	-	-	-
. Lead Time	2	1 Yr	1 Yr	3 Yrs	5 Yrs	5 Yrs	7 Yrs	7 Yrs
. Evolutionary Capability	1	Good	Good	Good	Good	Good		
. Potential For Success	1	Excel	Excel	Good	Fair	Fair	Fair	Fair
. Technology Assessment	1	Dev	Dev	Undev	Lab Stage	Unproven Feas.	Unproven Feas.	Unproven Feas.
OPERATIONAL CONSIDERATIONS								
. Constructability Erectability	1	Fair	Fair	Fair	Good	Good	Good	Good
. Operational Constraints	2	Good	Good	Good	Fair	Fair	Fair	Fair
. EVA/RMS Replaceability	2	N/R	N/R	N/R	N/R	N/R	N/R	N/R
. Reconfiguration & Oper Versatility	1	Good	Good	Good	Good	Good	Good	Good
IMPACTS								
. Payload Contamin.	1	Minor	Minor	Minor	Minor	Minor	Minor	Minor
. Drag	2	Fair	Fair	Fair	Poor	Poor	Good	Good
. Moment of Inertia	2	Fair	Fair	Fair	Poor	Poor	Good	Good
. Payload Blockage	2	Fair	Fair	Fair	Poor	Poor	Good	Good
. Stowage Volume	2	Good	Good	Good	Poor	Poor	Poor	Poor
. Compatibility with alternate vehicle config.	1	Good	Good	Good	Fair	Fair	Fair	Fair

TABLE 23 (CONTINUED)

RANKED CRITERIA	ORDER OF PRIORITY	1	2	3	5	5A	6	6A
IMPACTS (CONT'D)								
. Modularity	2	Good	Good	Good	Good	Good	Good	Good
. Payload & Module Interfaces	1	Good	Good	Good	Good	Good	Good	Good
PERFORMANCE CONSIDERATIONS								
. Weight, kgw	1	1750	1360	3270	4060	2200	6100* to 45000	1700* to 16000
. Power, kW	2	.2	.2	.9	.1	.1	118	47
. Area, M^2	1	0	-90	-92	460	230	-370	-370
. Controllability	1	Good	Good	Good	Good	Good	Good	Good
. Space Environment Compatibility	2	Good	Good	Good	Good	Good	Fair	Fair
. Temp Range,	2	0-200	0-200	0-200	-45 to			
. Isothermal Heat Transfer	2	Fair	Fair	Fair	Excel	Excel	Excel	Excel
RELIABILITY & LIFE								
. Complexity and No. of pieces	1	Excel	Good	Fair	Good	Fair	Good	Fair
. Component Life	2	5 yrs	5 yrs	2-5yrs	5 yrs	5 yrs	2-5yrs	2-5yrs
. Maintainability & Health Monitor	1	Good	Good	Good	Good	Good	Good	Good
. Failure Modes	1	Poor	Poor	Poor	Poor	Poor	Poor	Poor

*For Specific Powers of 45 kg/kW and 164 kg/kW and payload blockage. Comparison of the concepts on these criteria shows the compressor assisted heat transport system, Concepts 6 and 6A, to be the best. Close second in the ranking were the pumped liquid concepts (Concepts 1, 2, and 3). Concepts 5 and 5A were ranked last due to the large radiator area requirements.

Performance Considerations

Many of the ranking criteria in this category can be quantified. These include weight, power, radiator area, and temperature range of operation. A comparison of the criteria for the concepts shows Concept 2, the dual-temperature pumped liquid loop, to have the lowest weight, low radiator area and wide temperature range of operation. It has only fair isothermal heat transfer capability. Concept 1, the Reference Concept, is the second best in this category. It is heavier than Concept 2 (10%) and has 90 m^2 more area out of 1000 m^2 (9%), but is otherwise comparable. Concept 5A, the pump augmented heat pipe with two temperature levels, is ranked third in the performance category. It has the disadvantages of significantly higher weight and area than Concepts 1 or 2 but has the advantage of good capability for heat transfer under isothermal conditions. Concept 3 is ranked fourth with a high weight (3270 kg) and a relatively low radiator area. The other three concepts (5, 6 and 6A) are considered impractical from a performance standpoint being either excessively heavy and/or requiring excessive power.

Reliability and Life Considerations

Under this category, Concepts 1, 2, 5 and 5A are ranked nearly equal with Concept 1, the Reference Concept, having a slight advantage and 5A having a slight disadvantage. The concepts with a refrigeration system (3, 6 and 6A) are ranked lower than those with pumps.

Overall Rankings

A tabulation of the relative rankings of each heat transport concept is shown in Table 24 for each of the ranking categories discussed above along with the overall rankings. Based upon these evaluations, Concepts 1 and 2, the pumped liquid loops, are selected as the best approaches in every category. Concept 5A and Concept 3 are next in ranking with 5, 6 and 6A last.

Based upon this study it is concluded that the pumped liquid loop approach is superior to the two phased approaches as configured in these

TABLE 24

COMPARISON OF CONCEPT FOR EACH RANKING CATEGORY

RANKING CATEGORY	CONCEPT NO.						
	1	2	3	5	5A	6	6A
Potential for Benefit	1	1	2	2	2	2	2
Development Consideration	1	2	3	4	4	5	5
Operational Considerations	1	1	1	1	1	1	1
Vehicle Impacts	2	2	2	3	3	1	1
Performance Considerations	2	1	4	5	3	5	5
Reliability and Life	1	2	4	2	3	4	4
OVERALL RANKING	2	1	4	6	3	7	5

concepts. If two-phased concepts could be devised to show some significant benefits such as cost reduction, weight reduction, reliability/life, etc., this could change the trade results. However, for the concepts evaluated in this study, Concept 2, the split radiator outlet temperature concept, is selected as the recommended baseline.

4.3 Heat Rejection Concept Studies

Studies were conducted to identify the best concepts for rejection of the waste heat from the 250 kW Space Platform. These studies included evaluation of the radiation environment effects of different radiator location, concept sizing and optimization studies for deployed radiator arrays, sizing and optimization studies for constructable radiators, and utilization of individual module surfaces for heat rejection augmentation. Eight promising concepts were selected for evaluation of costs and additional trade studies.

A basic assumption of this study was that the heat rejection system is centralized. That is, all the waste heat from the Space Platform is collected by the heat transport system and brought to a central location for rejection. The centralized heat rejection system was assumed to be located on the Power Module portion of the Space Platform. Details of the individual studies are discussed in the following sections.

4.3.1 Panel Array Location and Thermal Environments

The 250 kW Space Platform was examined to determine attractive locations and orientations for the deployed radiator arrays. The primary criteria for determining a "good" radiator location were:

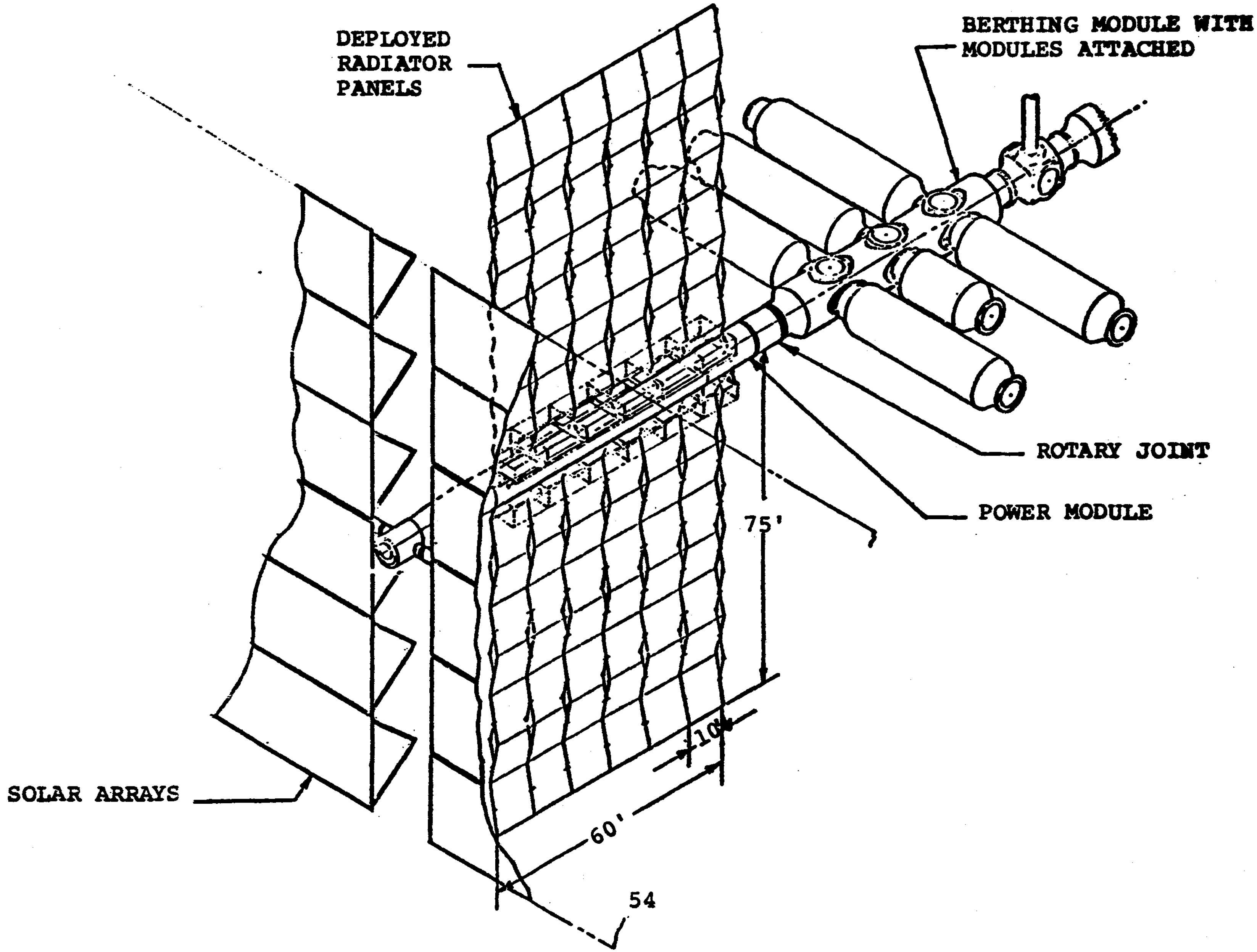
(1) Minimum viewing interference of the radiator panels with the spacecraft including payload viewing, solar array interference, etc.

(2) Low thermal flux from all radiant sources (sun, earth, spacecraft).

(3) Minimum complicating features such as rotary joints, disconnects, etc.

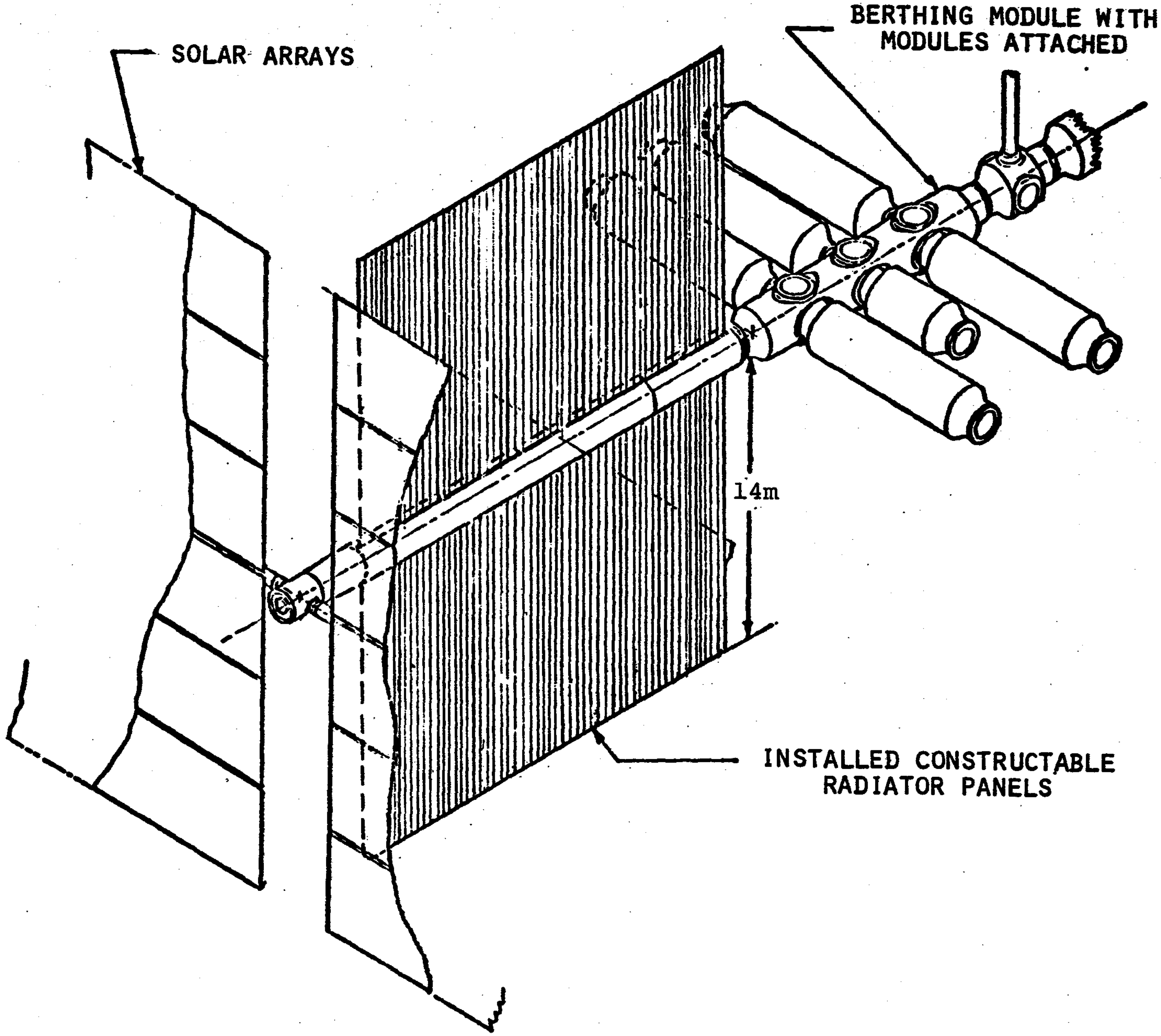
Figure 22 shows one candidate location. Figure 23 shows a different deployment concept, the space constructable radiator in the same location. The panels are located on the Power Module with an orientation that

FIGURE 22
DEPLOYED RADIATOR PANELS USING THE ATM SOLAR ARRAY TYPE DEPLOYMENT



VOUGHT

FIGURE 23
SPACE CONSTRUCTABLE RADIATOR PANELS WITH
INTERFACE INSIDE POWER MODULE



is always edge-to-sun. For this concept, the solar panels move relative to the radiator panels so varying amounts of the solar panels are seen by the radiators. The amount of energy radiated from the solar panels to the radiator panels varies depending on the solar panel location. Although the radiator panels would generally be oriented with their edge to the sun, some quantity of sunlight can impinge upon the radiator panels due to the small misalignment allowed for the solar arrays and this amount can have a significant impact.

Analyses were conducted to determine the equivalent sink temperature for the locations shown in Table 25. The mission assumptions for the studies were an orbital altitude of 417 km and beta angles (angles between the orbit plane and the earth-sun line) of 28.5°, 78°, and 90°. Also a number of solar array positions were considered. The radiators were assumed to be edge-to-sun for these analyses and the radiator coatings were assumed to be silver backed Teflon which has an $\alpha/\varepsilon = .11/.76$. The results of the studies conducted are shown in Table 25. It shows the effect of both the beta angle and the solar array positions on the peak sink temperature.

An alternate location for the radiators is shown in Figure 24. Here the fold-out or deployed radiator concept is shown with the radiators located on an arm between the solar array to the Power Module. The radiators are fixed relative to the solar arrays and thus always have the same view of the solar arrays. The radiators are assumed to be edge-to-sun at all times. The constructable radiator concept with automatic deployment is shown in similar orientation in Figure 25. Environment studies were considered for these orientations. The conditions analyzed were 417 km and beta angles of 28.5° and 90°. Silver backed Teflon was again assumed for this coating. For these studies a much lower sink temperature was observed than for the previous location. In these the sink temperature is approximately $-67.8^{\circ}C$ ($-90^{\circ}F$) for both 28.5° and 90° beta angles.

Based on these studies a range of sink temperatures was established for the parametric heat rejection system studies to follow. This range of temperatures was from -60 to $-20^{\circ}C$ (-80 to $-8^{\circ}F$) with an intermediate point of $-40^{\circ}C$ ($-40^{\circ}F$).

TABLE 25

RADIATOR ENVIRONMENT STUDIES
RADIATORS ON POWER MODULE

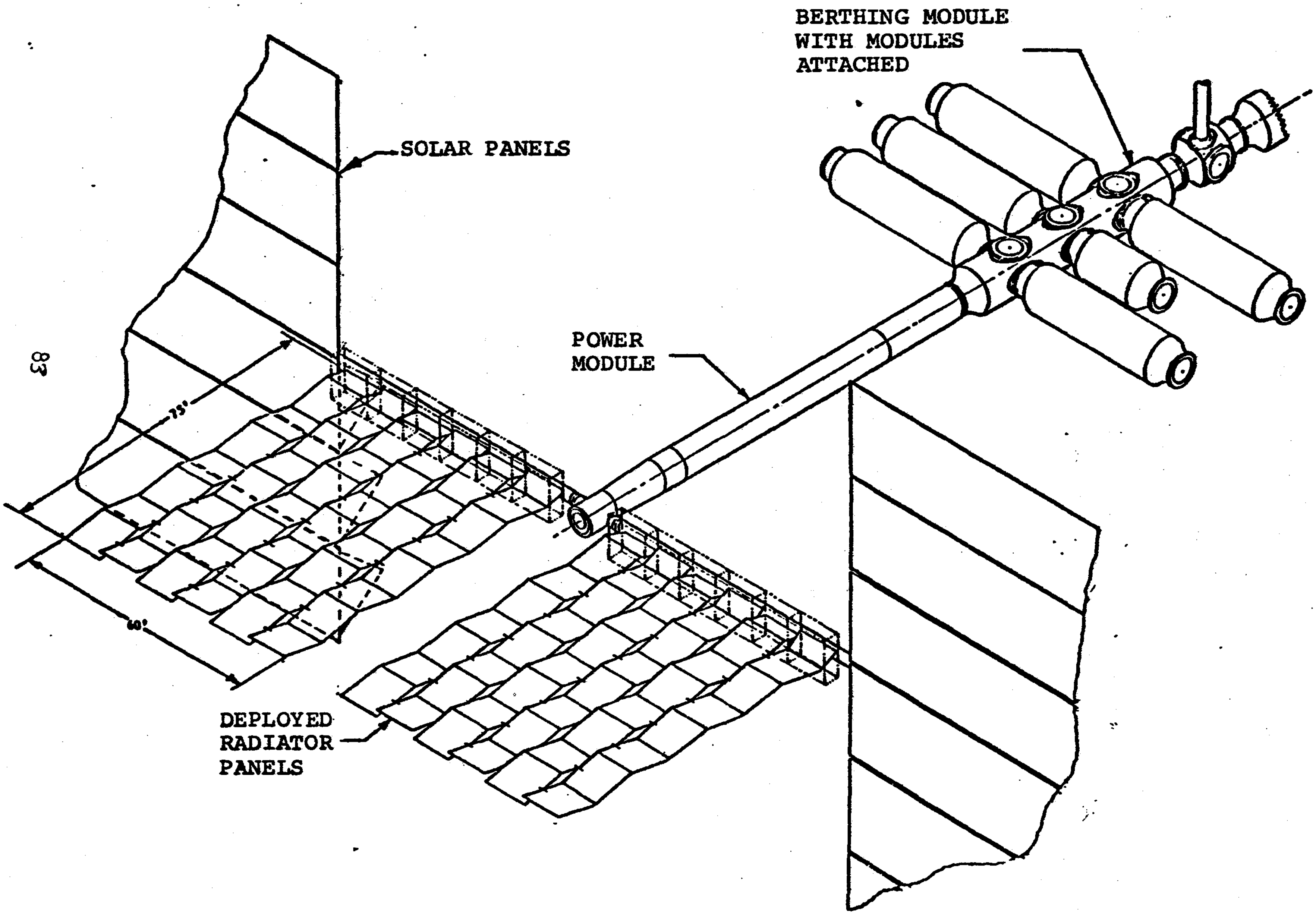
CONDITIONS

- 417 km Altitude
- β = 28.5°, Solar Arrays as Shown (Perpendicular to PM)
- β = 78°, Solar Arrays as Shown (Perpendicular to PM)
- β = 90°, Solar Arrays Rotated 90° (Parallel to PM)
- Radiators Edge to Sun
- Radiator Coatings: α/ε = .11/.76

RESULTS

β	SOLAR ARRAY POSITION	PEAK SINK TEMPERATURE - °F
28.5°	$\perp$ to PM	- 12.44
78°	$\perp$ to PM	- 20.0
90°	$\parallel$ to PM	- 76.6

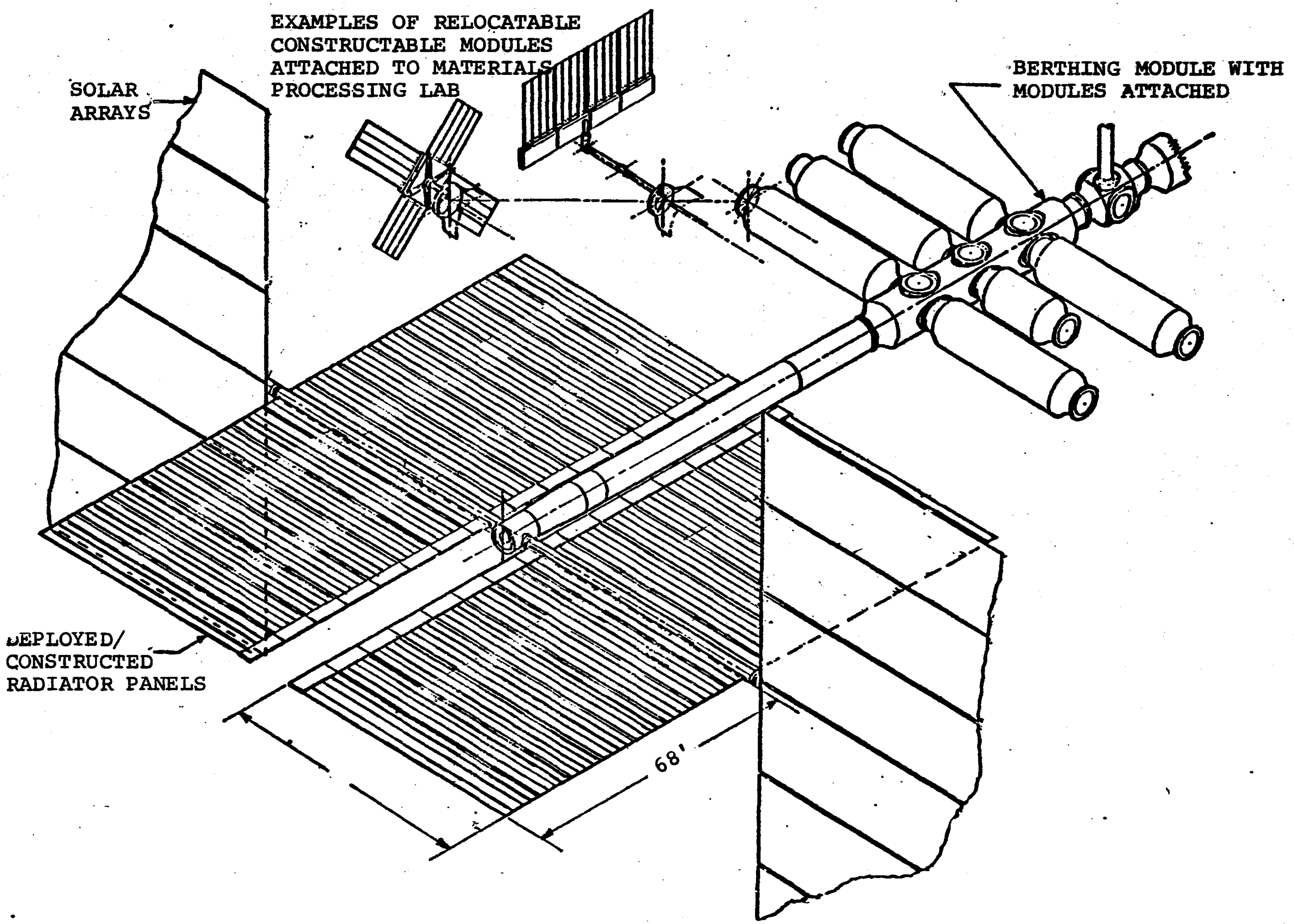
FIGURE 24
FOLD-OUT RADIATOR DEPLOYMENT FOR SUN ORIENTED RADIATORS



VOUGHT

FIGURE 25

SPACE CONSTRUCTABLE RADIATOR PANELS IN A SOLAR ORIENTED CONFIGURATION



4.3.2 Heat Rejection Concept Sizing and Optimization Studies

Concepts were evolved for deployed array radiator systems and each concept was designed, sized and optimized for comparison purposes. The concept studies were primarily conducted at the system level to determine the optimum approaches for achieving the required system reliability and life (0.99 probability of achieving a 10 year life). However, component designs were also optimized, particularly for the radiators. The optimum weight was determined parametrically for each concept as a function of heat load, radiator sink temperature and radiating temperature. These weights and sizes were utilized in cost and comparison trade studies to determine the best approaches.

Panel Design Concepts

Four panel design approaches were identified as promising candidates. These approaches have evolved in prior in-house and NASA studies for long life application. The four concepts were:

1) Pumped Fluid Radiator
2) Low-Technology or "Simple" Heat Pipe Hybrid Design
3) Integral Manifold Heat Pipe Hybrid Design
4) Deployed Constructable Radiator

Figure 26 shows a long life, high probability of success, low weight pumped fluid panel concept. This approach does not use heat pipes. The coolant fluid is distributed through the panel in the flow tube contained in the panels. The panel was designed in such a way as to achieve a high probability of success in a meteoroid environment, with low weight. Redundant fluid loops are assumed based on previous analyses for reliability purposes. Two separately manifolded systems are contained on each panel for the two separate fluid loops. Each fluid loop is capable of radiating the full load and thus the redundant loop is a standby or backup loop. Honeycomb construction was assumed for the panel concepts because it: is weight competitive, is a proven design, and is representative of current state-of-the-art. Figure 26 also shows a extrusion that is used for the tube in a pumped fluid panel design. This extrusion places the flow at the center

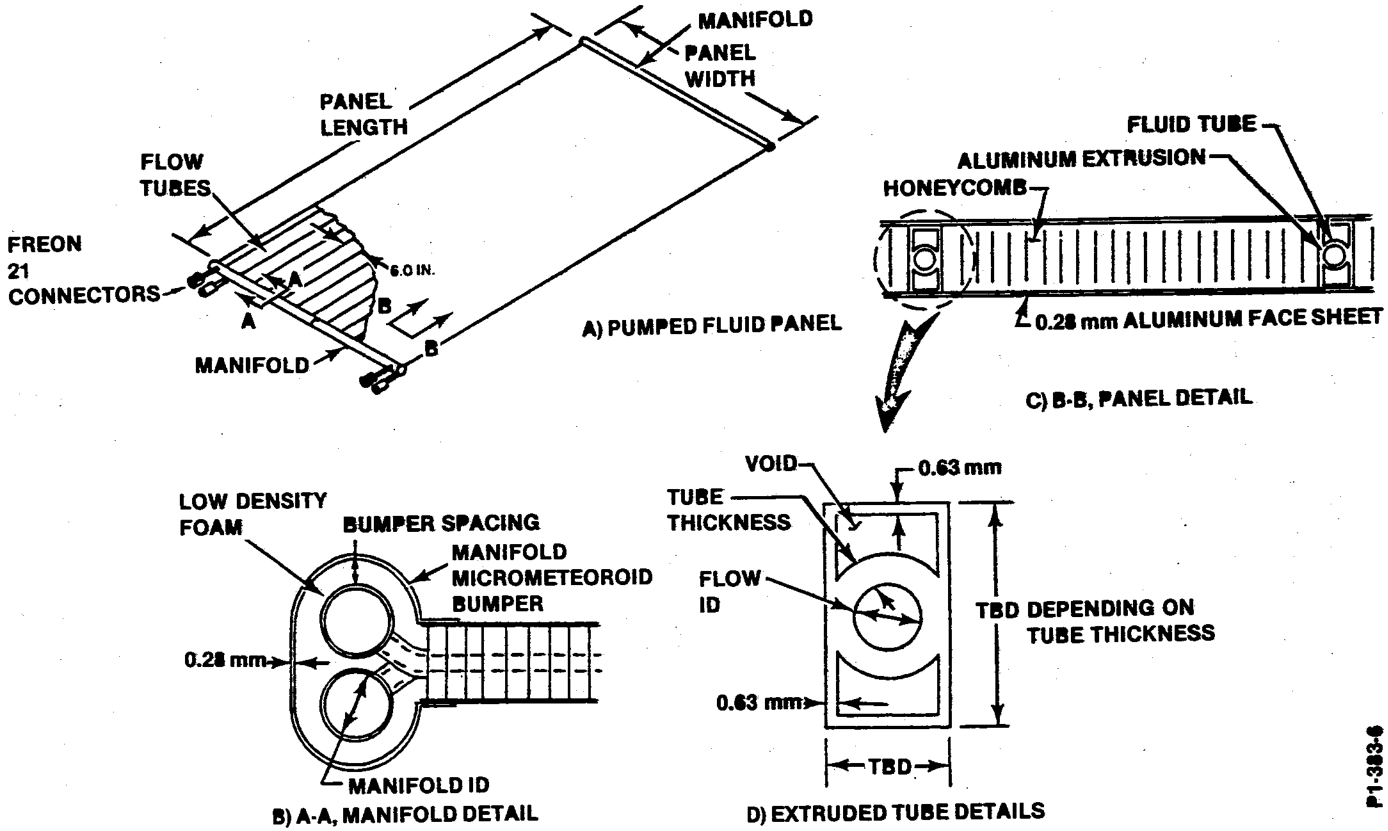


FIGURE 26 LONG LIFE PUMPED FLUID RADIATOR CONCEPT

of the panel and thus shields the tube from meteoroid penetration. The two facesheets of the panel act as bumpers to protect the panel tubes from meteoroid puncture.

Figure 27 shows a simple heat pipe panel design concept. This design utilizes a low technology heat pipe to spread the heat from the fluid that is contained in the heat exchangers onto the radiator panel. The fluid on the panel is contained in two compact heat exchangers, one for each individual loop. This panel again is of a honeycomb construction. The heat pipes would probably be an axial grooved, low watt-inch heat pipe.

Figure 28 shows a third approach for the radiator panel design. This is a high technology approach for a hybrid heat pipe/pumped fluid radiator panel. In this concept, fluid lines are contained within the center of the evaporator of the heat pipe and it flows through all the heat pipes on the panel at right angles. Each heat pipe is independent so that a puncture of the heat pipe that surrounds the fluid loop would result in the loss of only one heat pipe. This is an efficient design from a thermal standpoint because the heat pipe wick is in intimate contact with the fluid flow tube. The design of the condenser section of the heat pipe is a center core wick design. The internal flow tubes for the fluid loop are made from extruded, internally finned tube heat exchanger which contains internal fins to augment heat transfer.

Figure 29 shows the fourth radiator panel design approach, the constructable radiator. In this approach, the panel array is capable of automatic deployment, but, each individual panel (one for each heat pipe) can be removed and replaced if a failure in the heat pipe should occur. Two fluid loops are shown, each independent, and each capable of handling the full load. The heat pipe radiators are "plugged in" to cylindrical contact heat exchangers. These heat exchangers provide a loose fit for the heat pipes when they are initially plugged in. A clamping action is then provided by the contact heat exchanger, thus giving the contact force needed for good heat transfer contact conductance. The radiator segment shown in Figure 29 is a small, 4 kW segment of the total system. Each plug-in radiator panel is approximately 25 cm wide and 12 meters long after optimization.

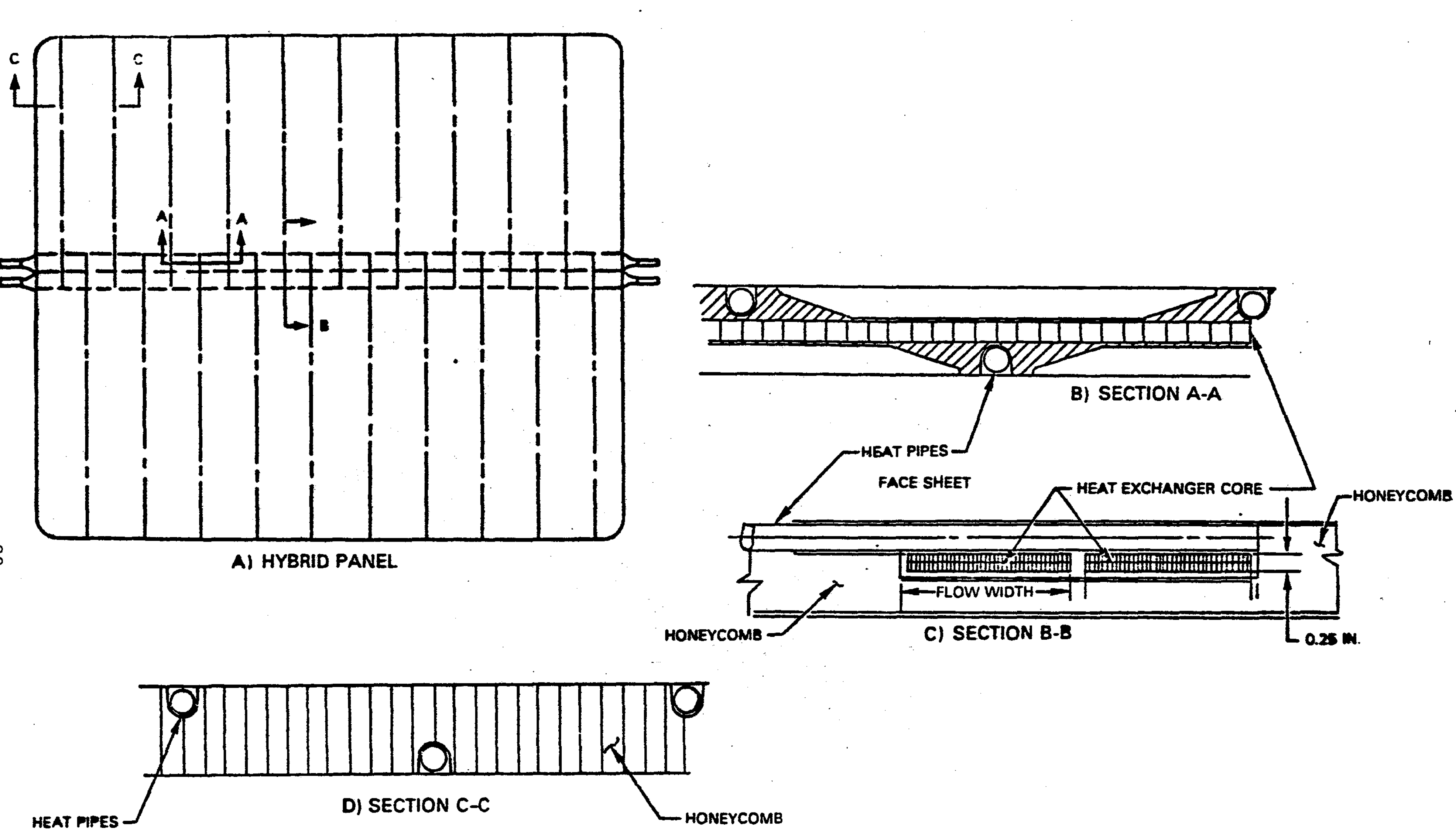


FIGURE 27. SIMPLE HEAT PIPE CONCEPT

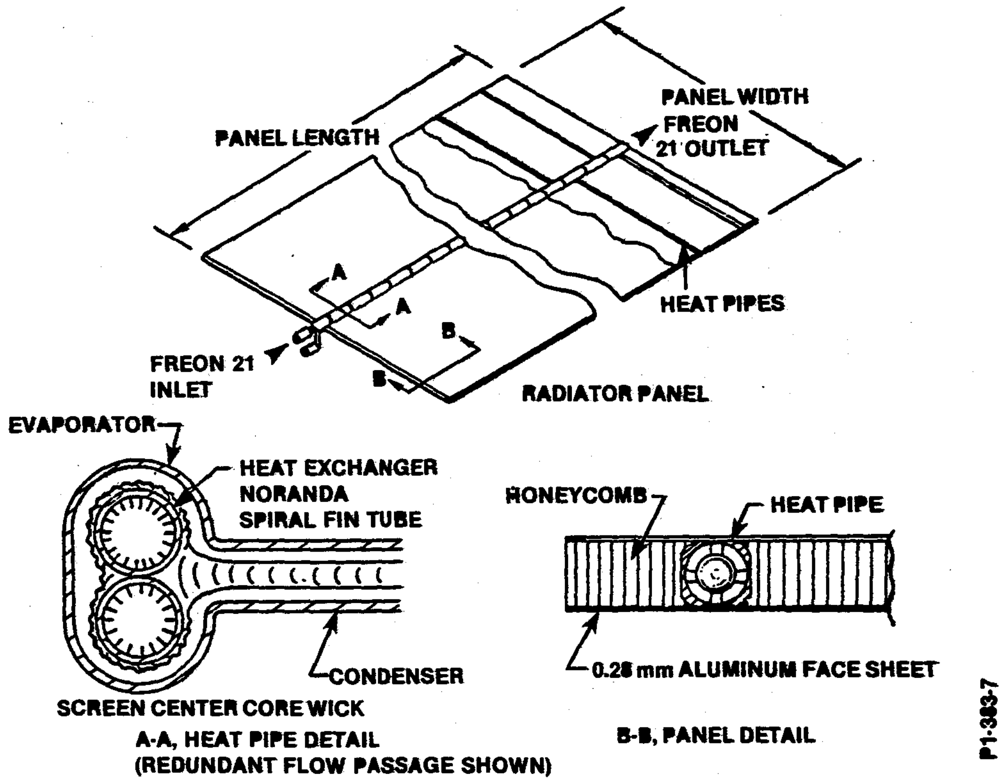


FIGURE 28 INTEGRAL MANIFOLD HEAT PIPE RADIATOR CONCEPT

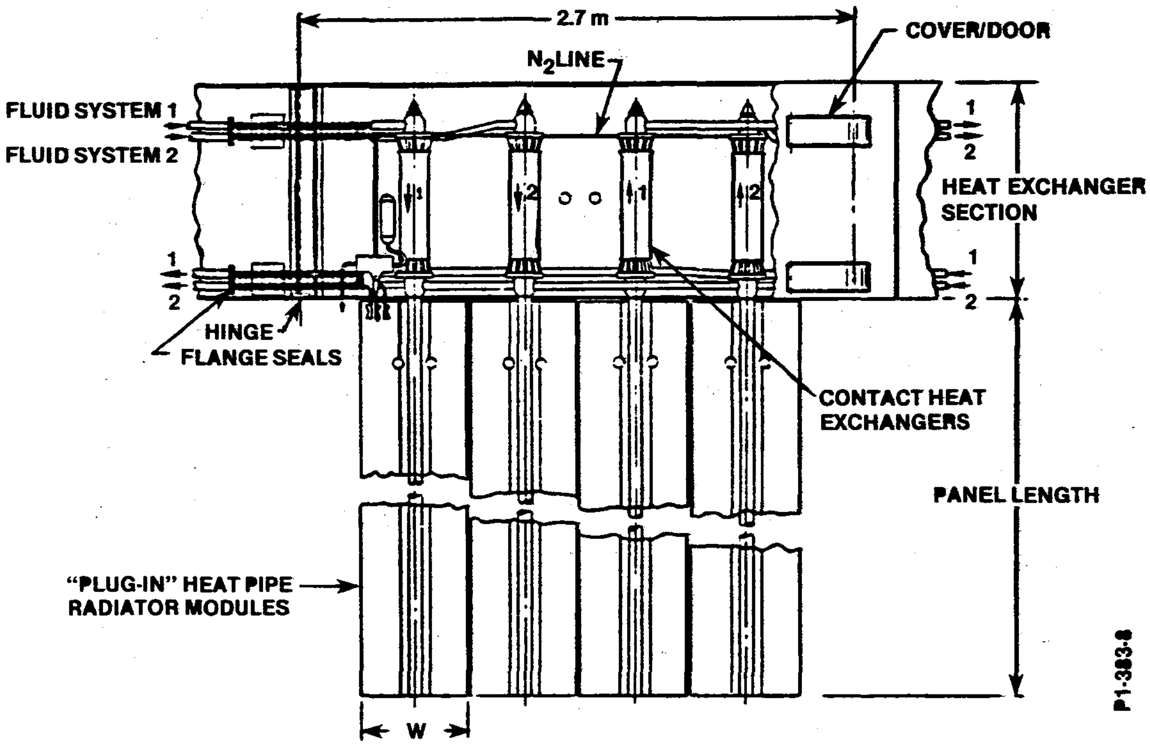


FIGURE 29 SPACE CONSTRUCTABLE RADIATOR

Figure 30 shows the effect of heat pipe diameter on radiator weight for both the low technology heat pipe and the integral manifold. The results shown were for a 16 kW heat load radiator system but are considered applicable to the higher heat loads as well. The figure shows that for the low technology heat pipe mass is minimum at approximately .95 cm diameter. The integral manifold heat pipe weight continues to decrease with decreasing heat pipe diameter. For this study, a heat pipe diameter of .95 cm was selected for both the low technology and integral manifold. A diameter smaller than .95 cm was not considered because the manufacturer (Hughes Aircraft) considered it to be difficult to make the integral manifold work at a smaller diameter . Also shown on Figure 30 are the required watt-inches for the heat pipe for the various diameters for both the integral manifold and low technology heat pipes. These watt-inches are those required based on an optimum radiator design.

Thermal Control System Reliability Study

The design reliability goal for the 250 kW Space Platform thermal control system was a probability of success of 0.99 for ten years. In order to determine the design meteoroid reliability (probability of no meteoroid penetration of a fluid passage) for the radiator panels, a study of coolant loop configurations was conducted to determine the required component and system redundancy. Figure 31 shows one coolant loop concept and the components included in the reliability study. Various redundancies in components and systems were analyzed to determine the subsystem reliability.

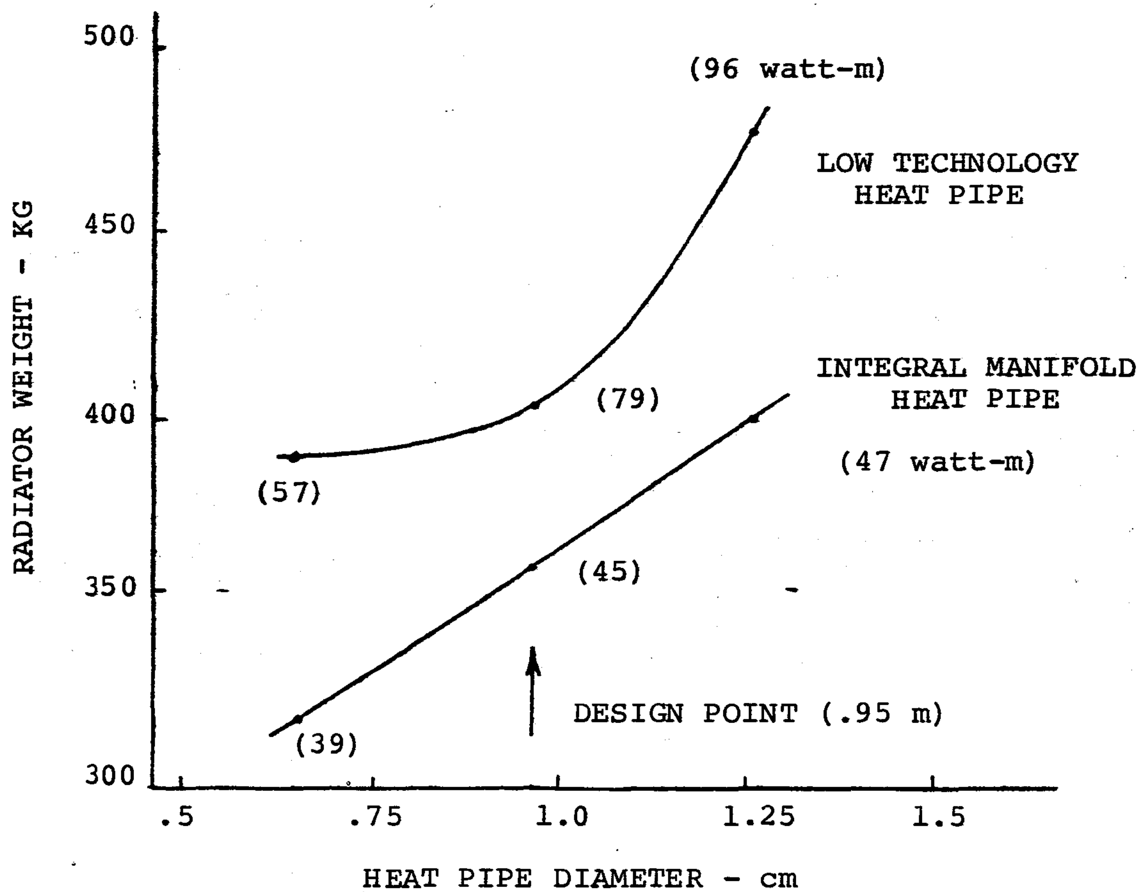
Table 26 shows the range of component failure rates and the resulting probabilities of success for (1) single loops with no redundant components, (2) single loops with redundant components, (3) redundant loops with no redundant components, and (4) redundant loops with redundant components in each loop. The probability of success (reliability) of the single loop was computed by the Poisson distribution function.

$$R = e^{-\Sigma \lambda t}$$

where λ is the failure rate and t is the mission time.

FIGURE 30
EFFECT OF HEAT PIPE DIAMETER ON RADIATOR WEIGHT

16 kW HEAT LOAD, $T_{IN} = 38°C$, $T_{OUT} = 4°C$, $T_S = -40°C$



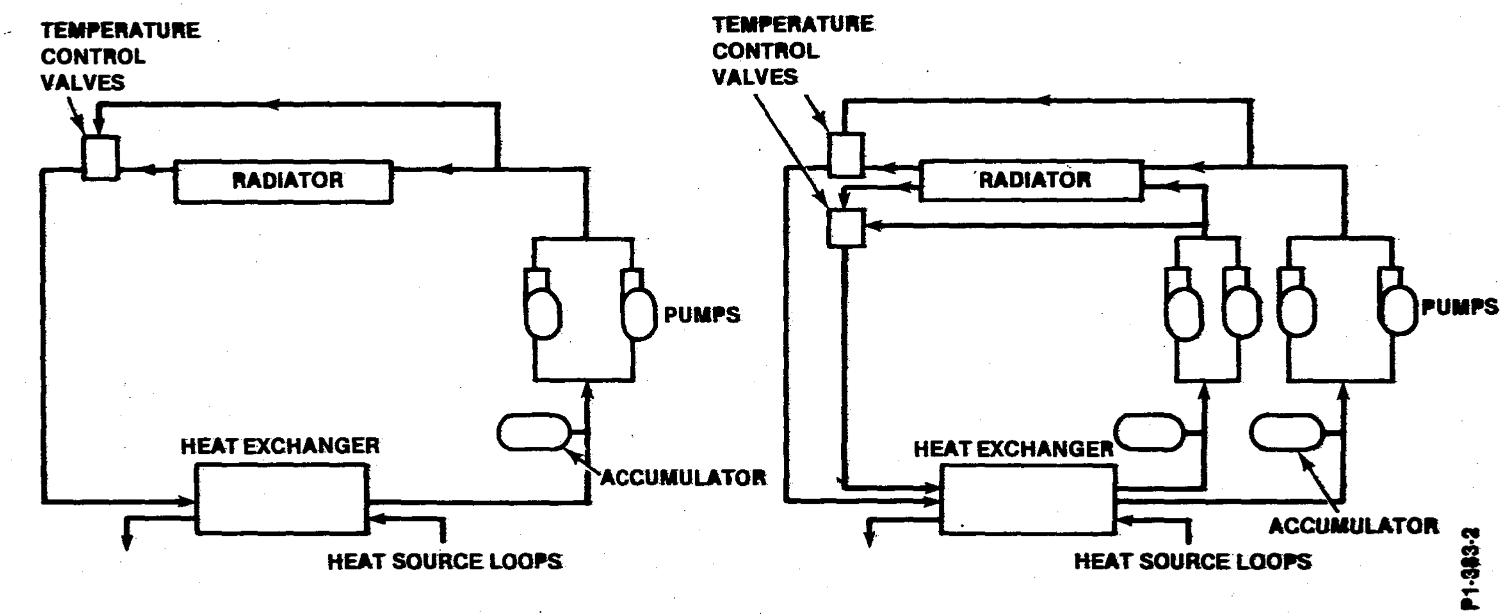


FIGURE 31 CANDIDATE SUBSYSTEM HEAT TRANSPORT LOOPS

TABLE 26 FLUID LOOP RELIABILITY CHARACTERISTICS

COMPONENT FAILURE RATE SUMMARY

COMPONENT	FAILURE RATE, λ FAILURES PER 10^6 hr	REDUNDANT COMPONENT FAILURE RATE, λ FAILURES PER 10^6 hr
Rad Panel Struct Integrity (8 Panels)	0.8 - 1.6	
Rad Panel Meteoroid	0.585	
Pump/Motor/Inverter	1.39 - 4.48	0.0439* - 0.4082*
Accumulator/Filter	0.14 - 0.30	0.00085 - 0.00389
Temp Control Valve	0.34 - 0.52	0.00498 - 0.0116
Fill Drain Valve, Pair	0.05	
Temperature Sensor**	1.50	0.27
Lines/Fittings	0.05	
Heat Exchanger	0.20	
	5.1 - 9.09	2.00 - 3.175
Single Loop Probability of Success (10 Years)	0.640 - 0.45	0.84 - 0.76
Redundant Loop Probability of Success	0.86 - 0.68	0.965 - 0.92

P1-383-17

* Switch System Reliability = 0.99 to 0.98

** Required for Health Monitoring Only

	PROBABILITY OF SYSTEM SUCCESS			
	SINGLE COMPONENTS		REDUNDANT COMPONENTS	
	ONE LOOP	TWO LOOPS	ONE LOOP	TWO LOOPS
Probability of No Micrometeoroid Puncture = 0.95	0.54 ± 0.10	0.77 ± 0.09	0.80 ± 0.04	0.94 ± 0.02
Probability of No Micrometeoroid Puncture = 0.99 to 1.0	0.57 ± 0.10	0.79 ± 0.09	0.83 ± 0.04	0.96 ± 0.02

P1-383-18

The reliability for the redundant loops was calculated by the relation

$$R_{RL} = R_S \left(R^2_{SL} + 2(1-R_{SL})\, R_{SL}\right)$$

where R_{RL} is the redundant loop reliability, R_S is the reliability of the failure detection and switch system and R_{SL} is the single loop reliability.

The failure rate data of Table 26 shows an assumed radiator panel micrometeoroid penetration reliability of 0.95. This value was selected as a "best" balance between system weight impact and reliability impact. Figure 32 shows the effect of different micrometeoroid reliabilities on the redundant standby loop reliability. Improving the micrometeoroid probability of penetration from 0.05 to 0.001 (probability of no micrometeoroid penetration from 0.95 to 0.999) will have little effect on the Thermal Control Subsystem reliability. The high side of the subsystem reliability will increase from .965 to .976 when the micrometeoroid probability of no penetration is increased from .95 to 1.0. However, the system probability of failure increases very rapidly with increase in probability of micrometeoroid penetrations above .05. Figure 33 shows the effect that micrometeoroid penetration has on system weight for three of the radiator panel concepts. This analysis is for a 32 kW subsystem. The figure shows little variation on system weight for the two heat pipe systems (LTHP = low technology heat pipe and IM = integral manifold). However, the pumped fluid (PF in Figure 33) system weight varies considerably with micrometeoroid penetration probability. The best balance between the two effects in Figure 32 and 33 was judged to be a micrometeoroid penetration probability of 0.05. This value was used in the study. It should be pointed out that this probability is for only one of the redundant loops.

Two concepts were considered for achieving the desired overall system reliability. One approach was to achieve the results with a single heat rejection subsystem with no required component and system redundancy. The other approach is to divide the heat rejection system into a number of smaller subsystems and then provide system oversizing (extra subsystems) to

FIGURE 32

EFFECT OF RADIATOR METEOROID RELIABILITY ON THERMAL CONTROL SYSTEM RELIABILITY

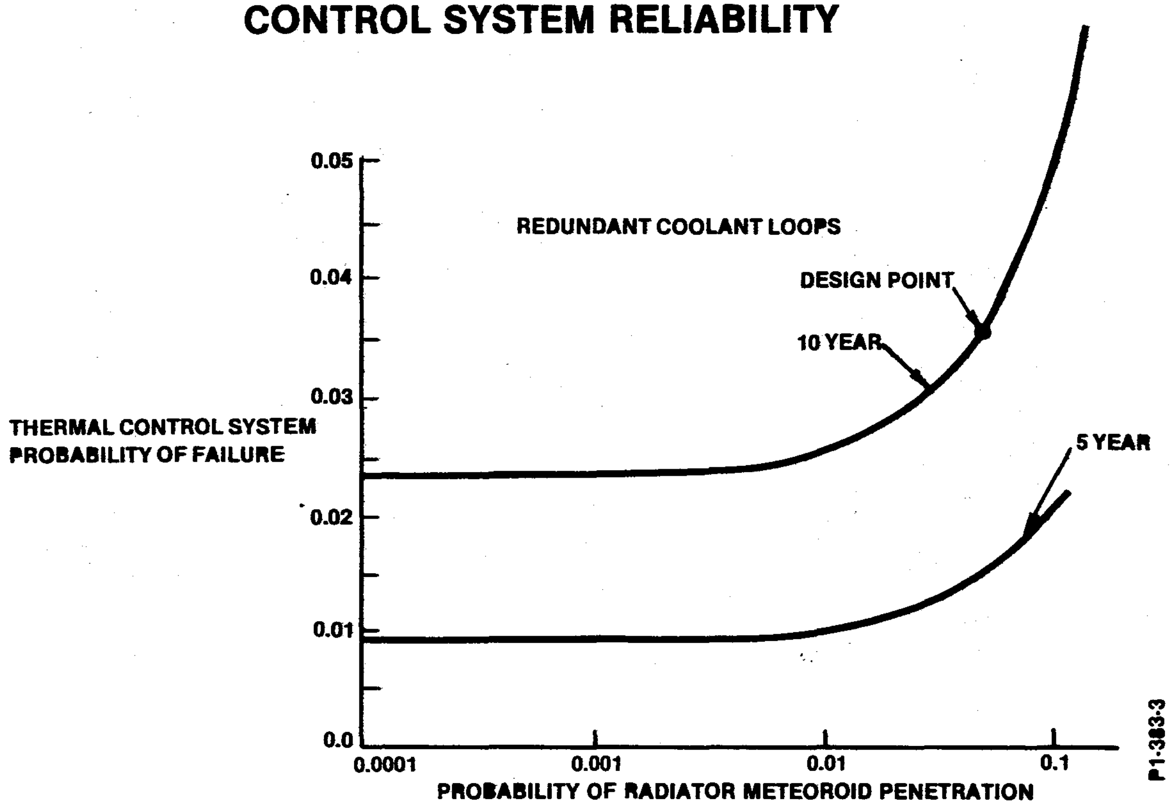
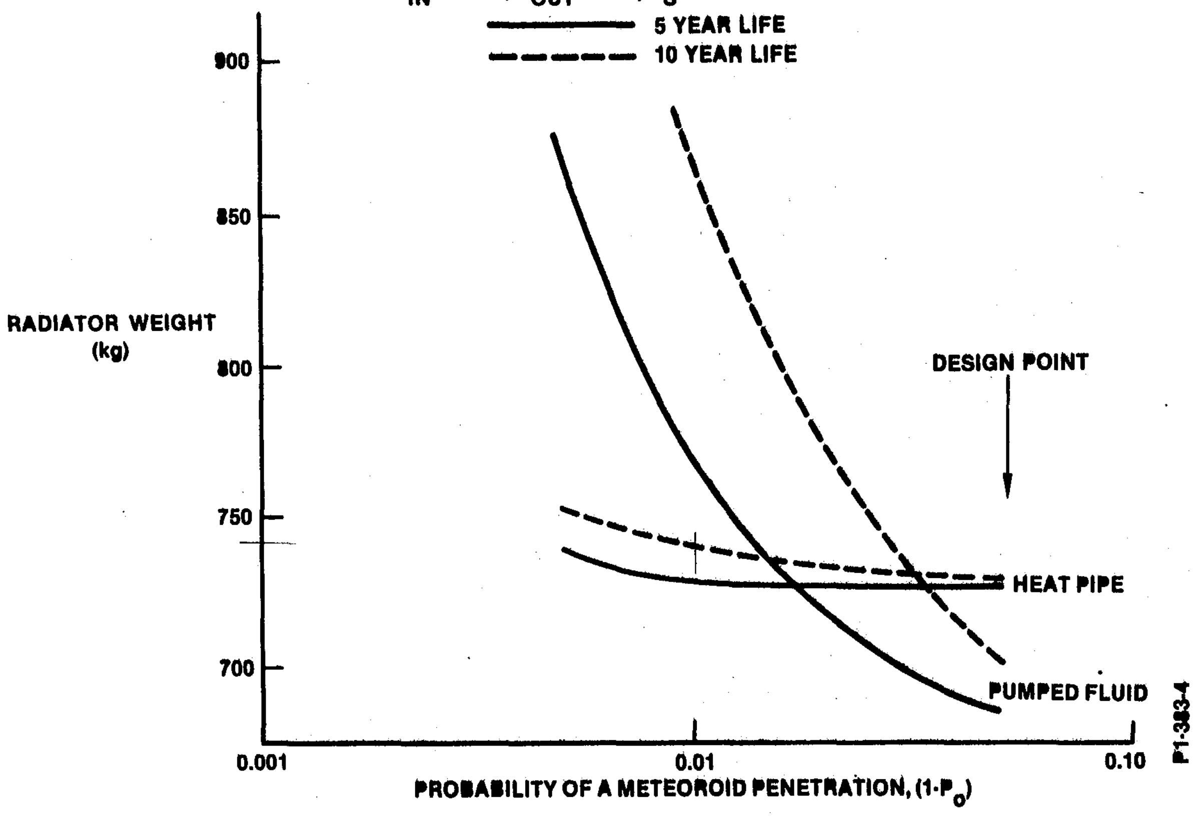


FIGURE 33

EFFECT OF METEOROID PROBABILITY ON RADIATOR WEIGHT

32 kW HEAT LOAD

$T_{IN} = 38°C$, $T_{OUT} = 4°C$, $T_S = -40°C$



achieve the desired reliability. The use of multiple heat rejection loops offers two advantages. First, the radiator micrometeoroid protection requirements are reduced for smaller independent radiator loops. The micrometeoroid penetration probability varies directly with radiator area; the probability of no micrometeoroid penetration for a given bumper configuration is a function of e^{-A}. The second advantage is that the system reliability can be increased above the individual heat rejection loop reliability by oversizing. Thus, a system made up of smaller, less reliable heat rejection subsystems is potentially lighter weight than a single high reliability heat rejection system. The amount of oversizing required to achieve a given system reliability is given by

$$P_S = \sum_{i=r}^{N} \binom{N}{i} P_{SS}^{\ i} (1-P_{SS})^{N-i}$$

where: P_S = system probability of success

PSS = subsystem probability of success

N = total number of subsystems

r = required number of subsystems

$$\binom{N}{i} = \frac{N!}{i!(N-i)!}$$

Figure 34 presents the solution of the above equation for various subsystem probabilities.

The pumped fluid radiator panels are designed with bumpered meteoroid protection of the fluid tubes and manifolds to provide a reliability of 0.95. The hybrid panels are designed with bumpered meteoroid protection of the coolant loop/heat pipe interface to provide a reliability of 0.95. In addition, the number of heat pipes are increased to allow for loss of heat rejection capability due to meteoroid penetration of the heat pipes. The amount of heat pipe oversizing is determined by the above equation where the

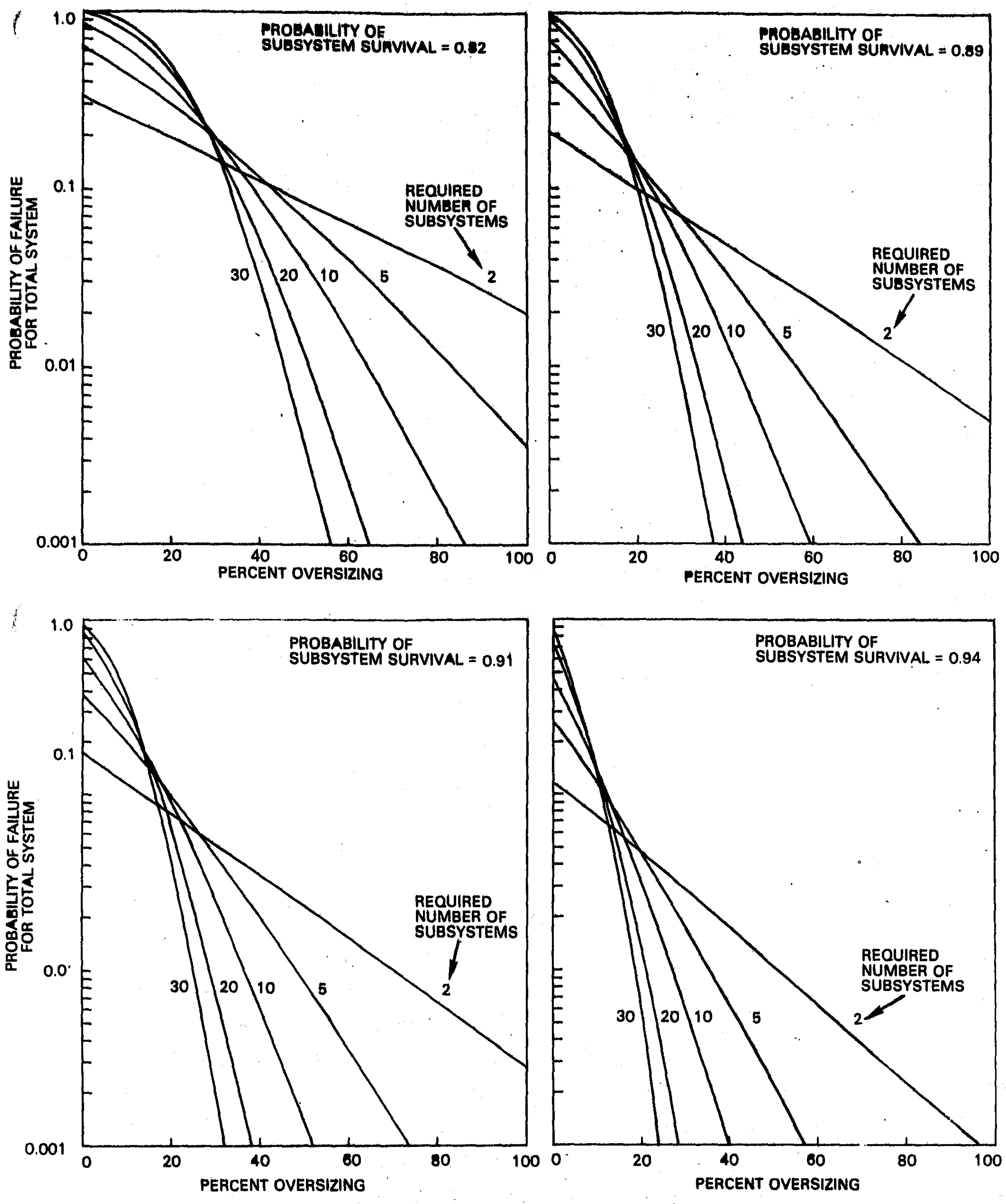


FIGURE 34 OVERSIZING TO ACHIEVE SYSTEM SURVIVABILITY

subsystem probability (P_{SS}) becomes the probability of meteoroid penetration of each heat pipe and r becomes the required number of heat pipes.

Deployment Mechanisms

For the purposes of the trade studies, concepts were needed for deployment mechanisms. Preliminary designs have been conducted at Vought in prior in-house efforts for two deployment approaches. Figure 35 illustrates the deployment design for radiator panel Concepts 1, 2 and 3 (pumped fluid, low technology heat pipe and integral manifold). This design is a scissors mechanism which can deploy 8 to 10 radiator panels each with dimensions of about 2 by 4.6 for about 150 m^2 to 190 m^2 of radiating surface. The panels make one arm of the scissors mechanism and an I-beam makes the other scissors arm. The base of this design is such that the panels can be rotated $\pm$ 30^{o} for solar avoidance. The same design was sized for the 25 kW Power System under study for NASA-MSFC (Reference 3) which deploys 169 m^2 and the total deployment and rotating base weight was estimated at 480 pounds. The deployment mechanism was scaled to meet the requirement for each design.

The second deployment mechanism, which was designed for use with the space constructable radiator, is shown in Figure 36. This is a cable-motor-spring approach in which the panels are deployed by spring hinges and retraction is accomplished by a tension cable. The tension is also used to "lock" the panels into position when deployed. The cable is attached to the mid-point of the outermost panel, passes through pivotal cable-eyes at the mid-point of each of the other panels and is wrapped on a motorized cable drum. Torsion springs at each panel hinge force the panel stack to extend when the stowage latch is released. A counter torque would be applied by the drum motor to control the deployment rate. When deployed the panels are locked into position.

Parametric Weight Analysis of Panel Concepts

Radiator panel concepts were optimized in order to obtain fair and meaningful trade comparisons. This analysis was performed on Concepts 1, 2 and 3 (pumped fluid, low technology hybrid and integral manifold hybrid). Parametric data providing weight optimized panels for different radiator heat loads, operating temperatures and environment temperatures are required for

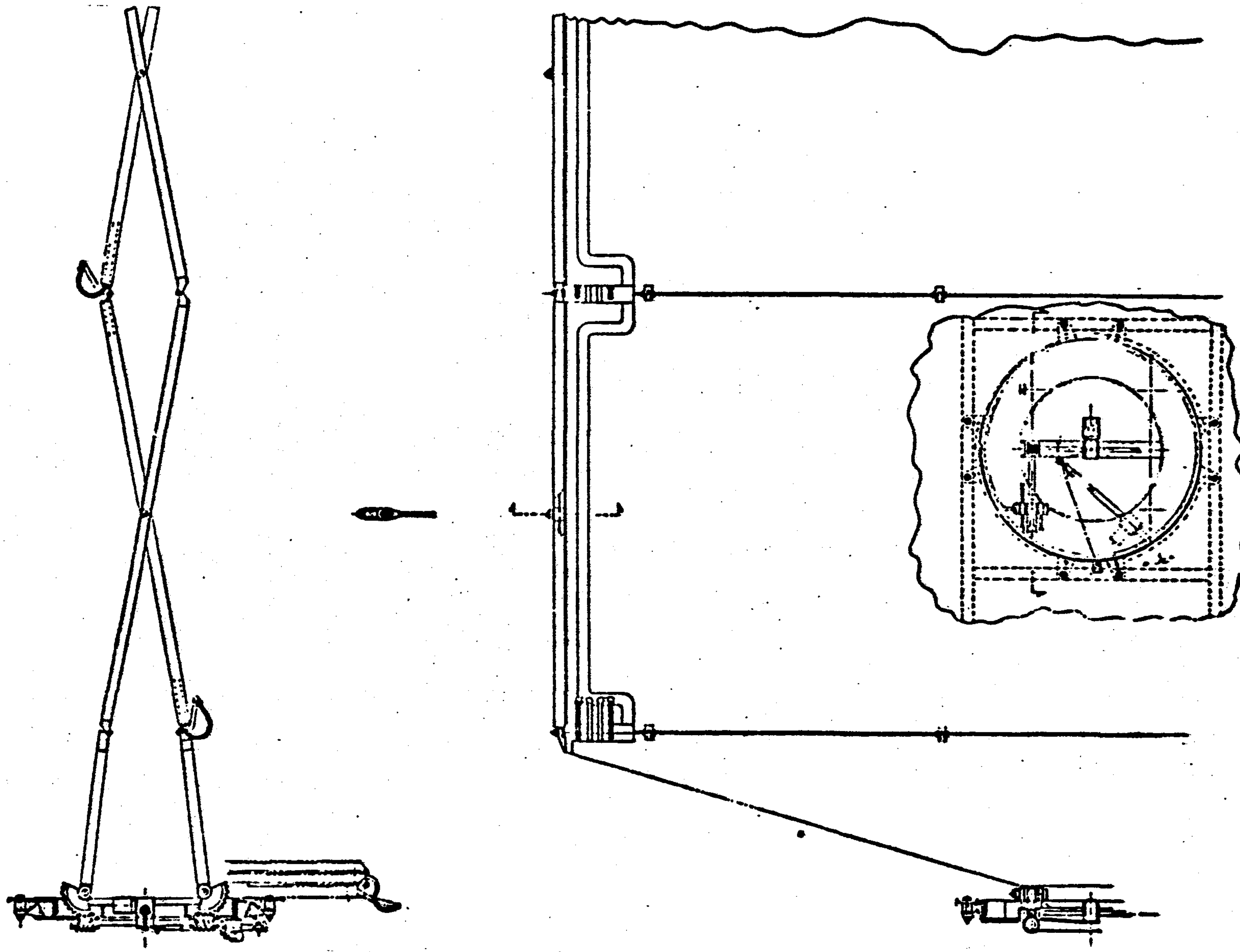


FIGURE 35 VOUGHT RADIATOR SCISSORS DEPLOYMENT MECHANISM WITH PANEL ORIENTATION

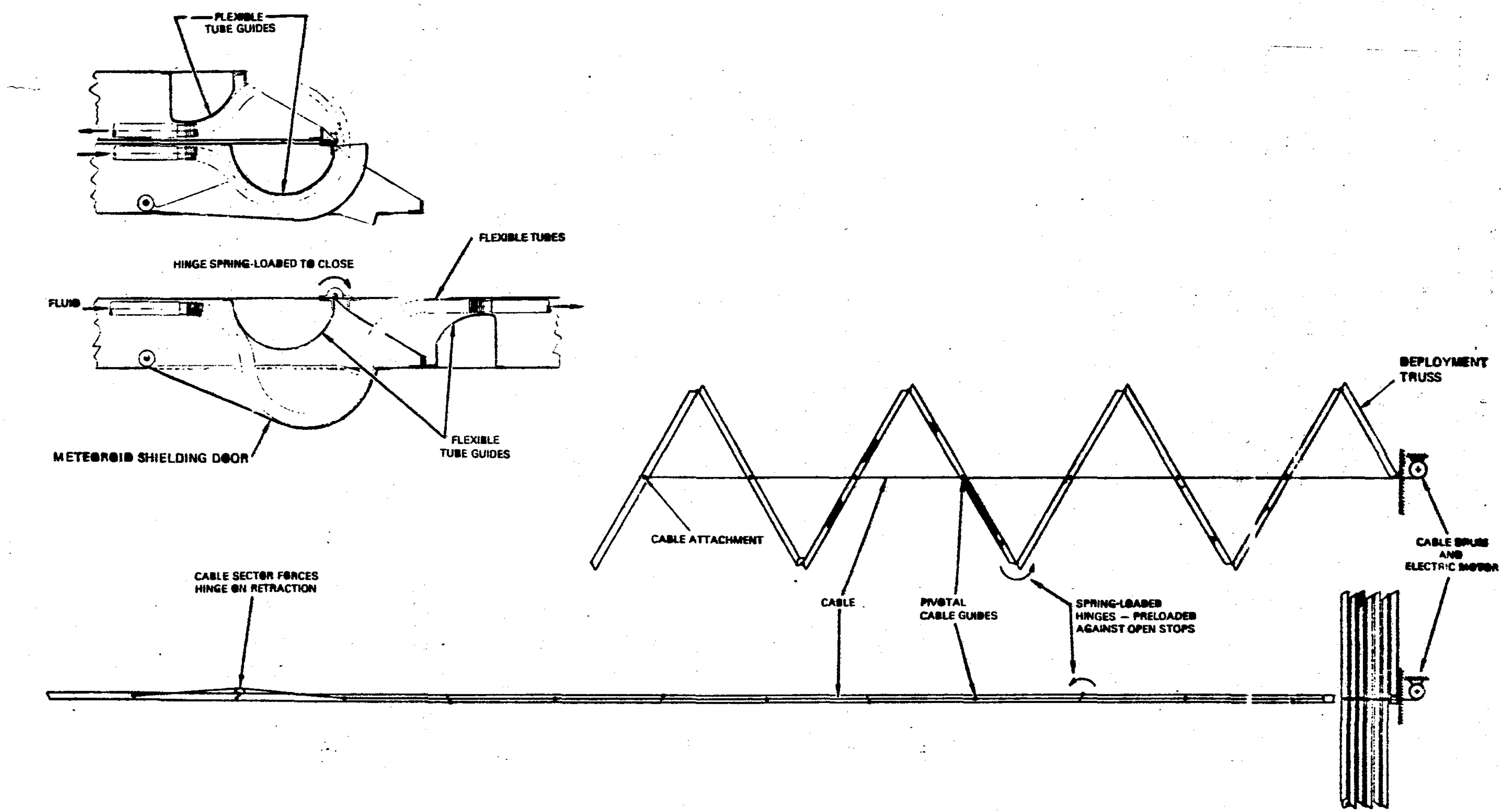


FIGURE 36 RADIATOR DEPLOYMENT CONCEPT SIMPLE DEPLOYMENT AND RETRACTION MECHANISM

each concept. Specialized computer routines were used for the parametric weight optimization of both the pumped fluid and hybrid concepts.

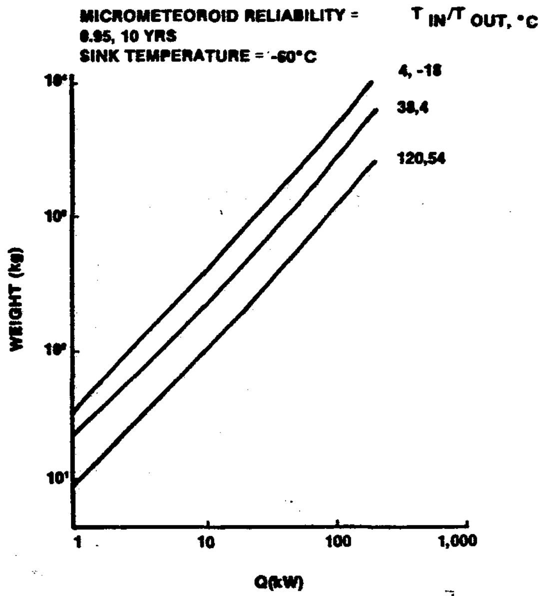
The items included in the weight of the pumped fluid radiator are facesheets, honeycomb, bonding adhesive, panel thermal control coatings, flow tube extrusions, manifolds, Freon 21, and equivalent pumping power penalty. The tube extrusion dimensions were determined based on a bumper distance (facesheet to tube outside surface) of 5.7 mm. This basic dimension plus the computed tube inside diameter and tube thickness required for meteoroid protection determines the extrusion dimensions and the honeycomb thickness. The facesheet thickness that resulted in the minimum weight was also determined. A minimum thickness of .25 mm was specified for manufacturing ease and for most cases this limit was used by the computer routine.

The hybrid panel weight included the facesheets, honeycomb, bonding adhesive, panel thermal control coating, heat pipe, heat pipe fluid, coolant loop manifold and heat exchanger, Freon 21 and equivalent pumping power penalty. Weights of aluminum-ammonia heat pipes with a wall thickness of 0.9 mm were used for all cases except the high operating temperatures. Aluminum-acetone heat pipe weights were used for the high temperature (20^{o}C) case.

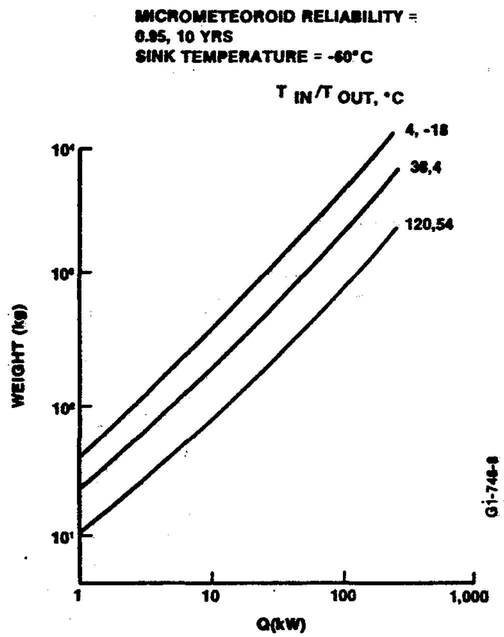
The optimized panel weights are given in Figures 37, 38, and 39. The optimum panel weights are shown parametrically over a heat load range from 1 to 250 kW, three radiator temperatures and radiation sink temperatures of -60^{o}C, -40^{o}C, and -20^{o}C. These panel weights were utilized as one element in the system weight optimization study (different than the panel optimization study) which determined the optimum subsystem size as discussed below.

<u>Optimum Subsystem Size Study</u>

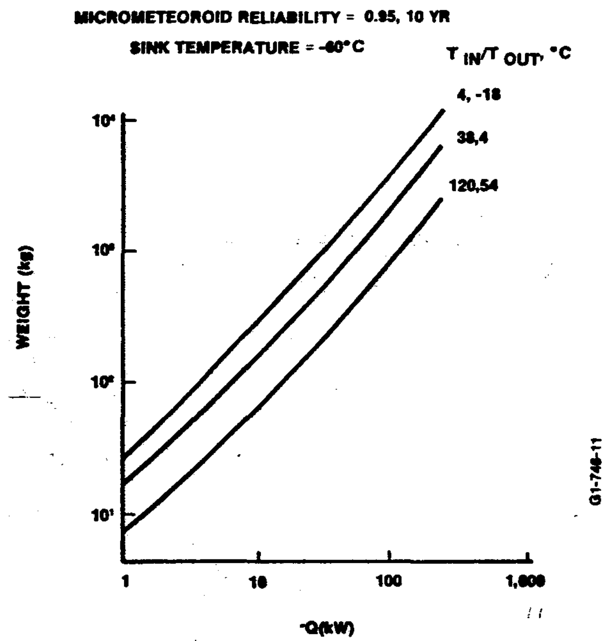
Using the results of the studies discussed above, the weight optimum system was determined for each heat rejection concept. The optimum subsystem size and corresponding number of subsystems was determined for system heat loads of 50 kW through 350 kW; for sink temperatures of -60^{o}C, -40^{o}C, and -20^{o}C; and for three radiator temperatures (4^{o}C inlet, -18^{o}C outlet; 38^{o}C inlet, 4^{o}C outlet; and 120^{o}C inlet, 54^{o}C outlet). This optimization study was performed for the pumped fluid, low cost



a) Integral Manifold Optimized Panel Weights



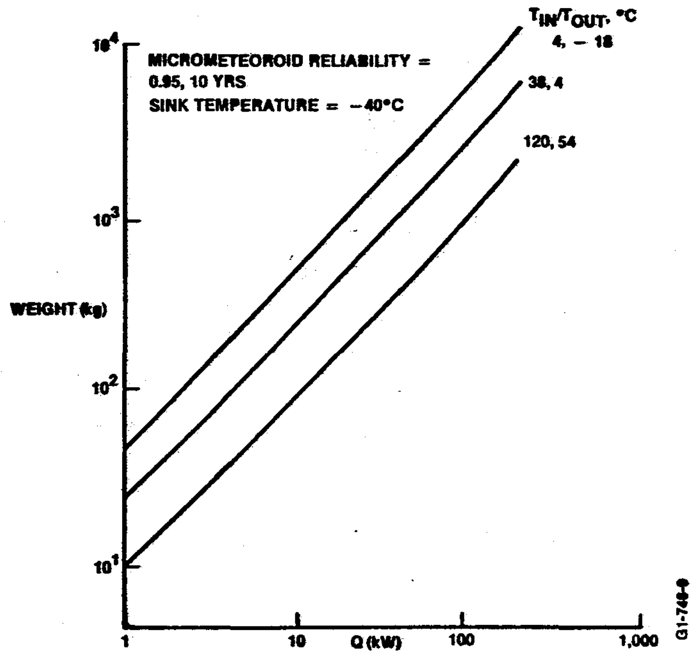
b) Low-Technology Heat Pipe Optimized Panel Weights



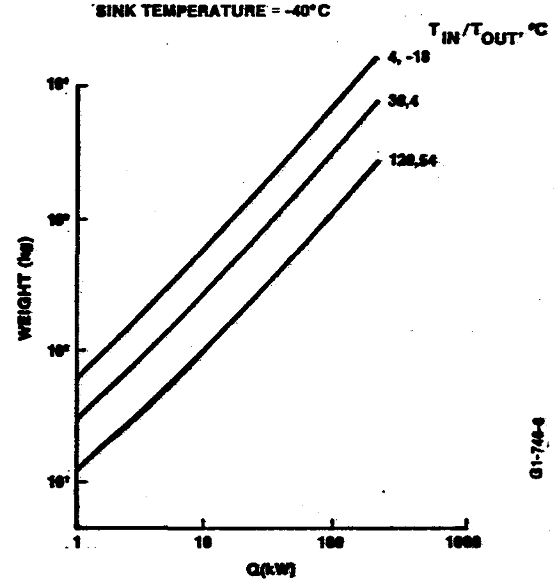
c) Pumped Fluid Optimized Panel Weights

FIGURE 37

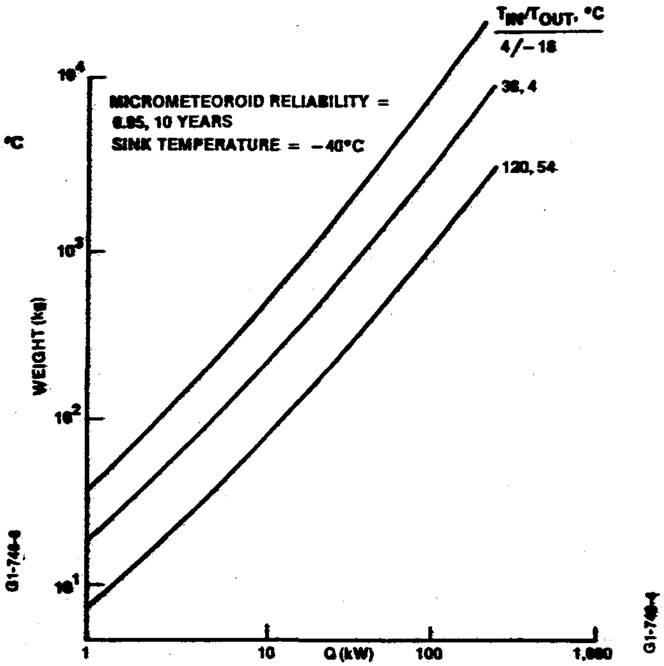
OPTIMIZED PANEL WEIGHTS FOR -60°C SINK TEMPERATURE



a) Integral Manifold Optimized Panel Weights



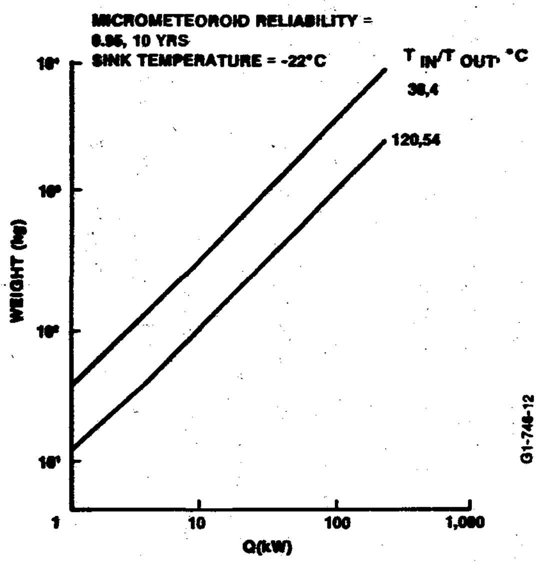
b) Low-Technology Heat Pipe Optimized Panel Weights



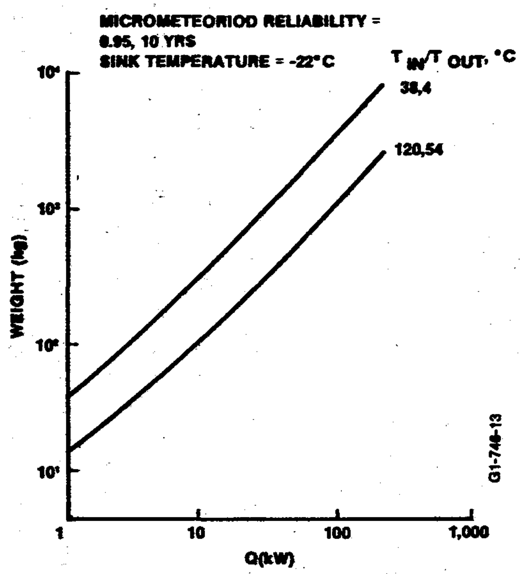
c) Pumped Fluid Optimized Panel Weights

FIGURE 38

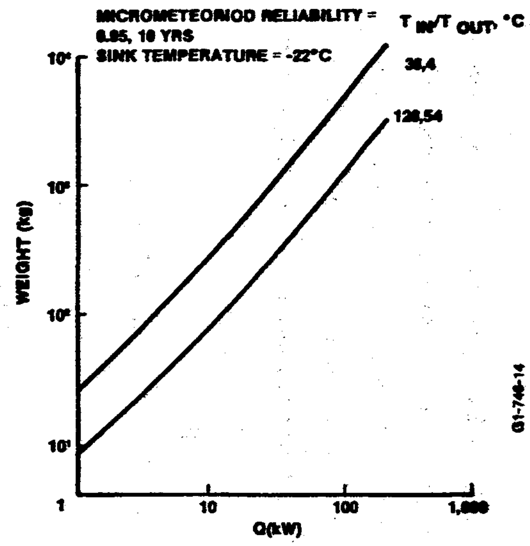
OPTIMIZED PANEL WEIGHTS FOR -40°C SINK TEMPERATURE



a) Integral Manifold Optimized Panel Weights



b) Low-Technology Heat Pipe Optimized Panel Weights



c) Pumped Fluid Optimized Panel Weights

FIGURE 39
OPTIMIZED PANEL WEIGHTS FOR -22°C SINK TEMPERATURE

heat pipe hybrid, and the integral manifold hybrid. The system weights for these studies included the panel weights discussed above and the additional components required for a closed loop for each subsystem. These components included the pumps, accumulators, temperature control valves and heat exchangers. The following were used to estimate the component weights:

Heat Exchanger	: 0.9 kg
Pump	: 2.5 kg
Accumulator	: .605 x fluid weight
Tubing Per Loop	: 18 kg
Temperature Control Valve	: 2 kg

Using the above values, the component weights in kilograms for each redundant subsystem was estimated by

$$W_{comp} = (.605\ W_{fl} + 45.0 + 0.9\ Q_{SUB})\ N_{SUB}$$

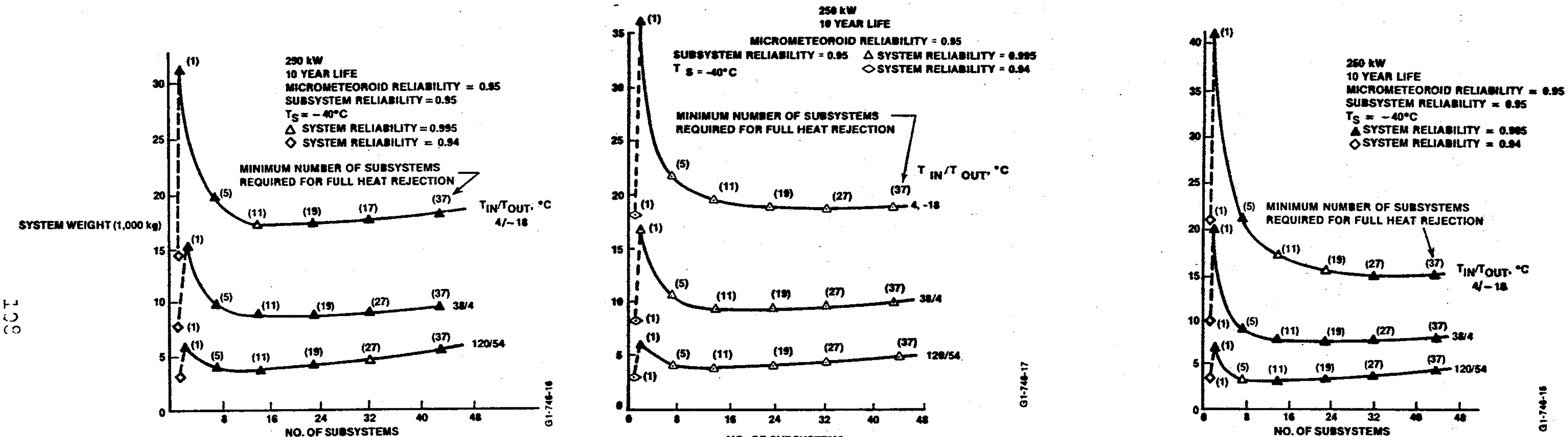
where: W_{comp} = total subsystem component weight (including redundant components)

W_{fl} = total system fluid weight (Freon 21)

Q_{SUB} = subsystem heat load

N_{SUB} = number of subsystems

Two approaches were considered in the subsystem size/reliability study: the single subsystem and the multiple subsystem. With the single subsystem approach, one loop is sized for the total system heat rejection and reliability is accomplished by component and loop redundancy. With the multiple subsystem approach, reliability is accomplished by dividing the heat load among several smaller subsystems and then providing extra subsystems. Figure 40(a) shows the effect of subsystem size on system weight for the integral manifold approach for a sink temperature of -40°C. The lowest weight approach for each temperature condition is the single subsystem. However, the probability of success for the single subsystem approach is only 0.92 to 0.96 whereas, the multiple subsystem approach reliability is 0.995. The lowest weight approach for the multiple subsystem is with approximately 11 subsystems required, 14 subsystems total. Thus, the optimum subsystem size for the multiple subsystem approach would be about 22.73 kW and three extra subsystems would be required to achieve the required reliability. For a radiating temperature of 38°C fluid inlet and 4°C fluid outlet, the optimum subsystem weight is 8800 kg for the multiple subsystem compared to 7600 kg for the single subsystem. A similar effect is observed for the low



a) Total System Weight vs No. of Subsystems for Integral Manifold Radiator Systems

b) Total System Weights vs No. of Subsystems for Low-Technology Heat Pipe Radiator System

c) Total System Weight vs No. of Subsystems for Pumped Fluid Radiator System

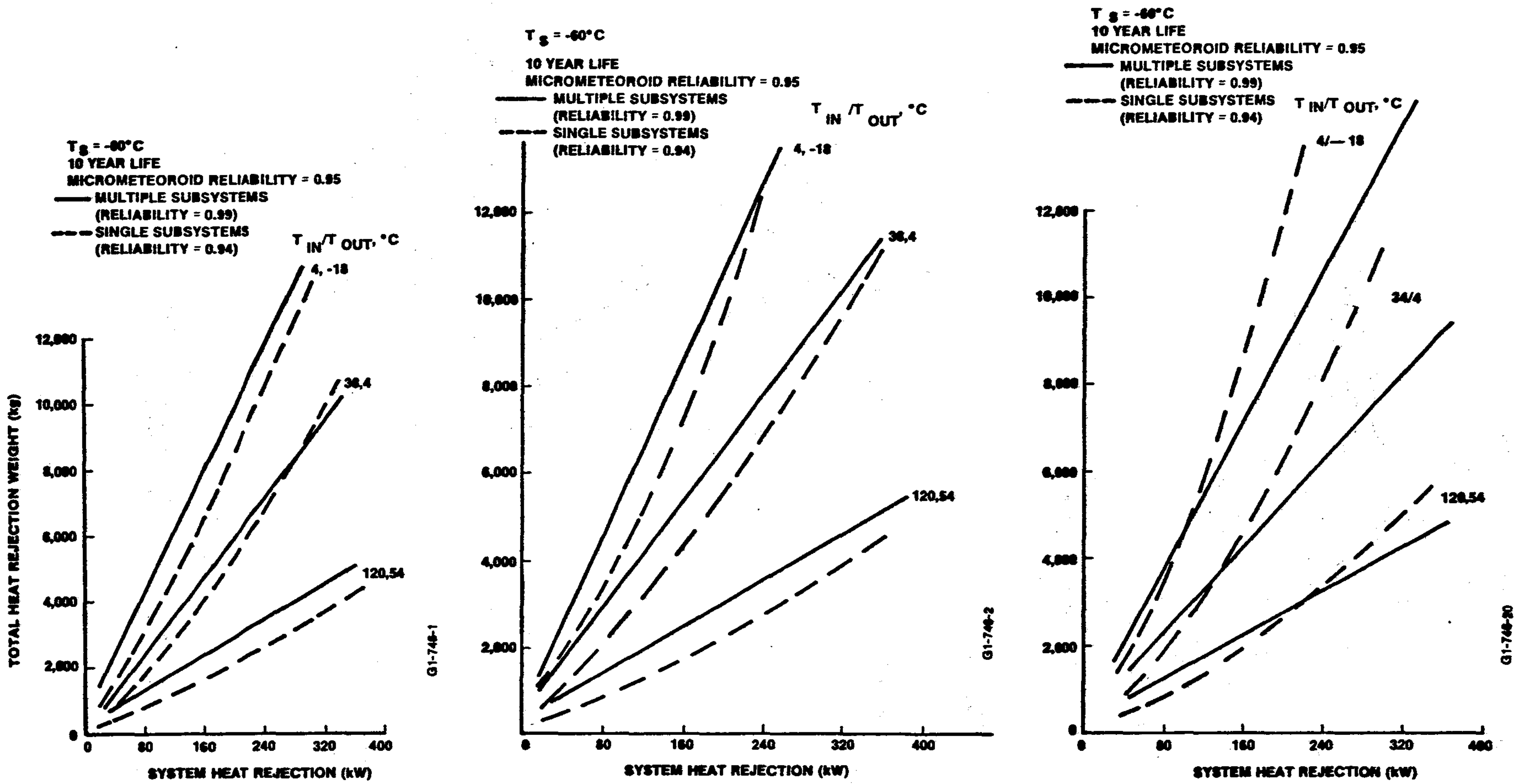
FIGURE 40

SUBSYSTEM SIZE OPTIMIZATION AT −40°C SINK TEMPERATURE technology heat pipe approach shown in Figure 40(b). However, the low technology heat pipe weight is higher than the integral manifold weight across the entire range of variables by about 10 to 15%.

The pumped fluid subsystem size study, shown in Figure 49, shows a slightly different effect. For this concept, the multiple subsystem approach which has the highest reliability is also lower in weight. The single subsystem weight is 9980 kg compared to 7711 kg for the multiple subsystem for the 38°C inlet, 4°C outlet radiation temperature case.

The optimum heat rejection system weights from the subsystem size study are shown in Figures 41, 42 and 43. Both the single subsystem and multiple subsystem weights are shown. The results show the following general trends. The low technology heat pipe concept is generally heavier than the pumped fluid or integral manifold for the complete range of heat loads, radiating temperatures and sink temperatures. When the integral manifold and the pumped fluid approaches are compared on an equal reliability basis (0.99 system reliability) the pumped fluid system is lighter in all cases by about 10 to 15%. However, if the lowest weight approach is considered, whether it be multiple or single subsystems, the pumped fluid concept is lowest weight for heat loads less than about 50 to 80 kW and for heat loads greater than about 250 to 350 kW. The largest difference between the two systems at -40°C sink temperature and 38/4°C radiation temperature is at 130 kW heat load (see Figures 42(a) and 42(b)). At that heat load, the integral manifold approach is about 635 kg or 18% lower weight than the pumped fluid approach (3450 kg compared to 4080 kg). However, at the same heat load, the multiple subsystem integral manifold approach is about 540 kg or 13% heavier than the pumped fluid.

The pumped fluid system weights have one characteristic that is different than the two hybrid systems. At the lower heat loads, the single subsystem is lower weight and at higher heat loads, the multiple subsystem is lower weight. Examining the pumped fluid -40°C sink temperature case (Figure 42(c)) at the nominal radiator temperature of 38/4°C shows the single subsystem to be 540 kg or 40% lower weight than the multiple subsystem (1270 kg vs 1815 kg) at 50 kW. At 130 kW the two approaches are of equal weight. At 250 kW, the multiple subsystem approach is 2270 kg or 30% lower



a) Integral Manifold System Weights

b) Low-Technology Heat Pipe System Weights

c) Pumped Fluid System Weights

FIGURE 41

OPTIMUM HEAT REJECTION SYSTEM WEIGHTS FOR -60°C SINK TEMPERATURE

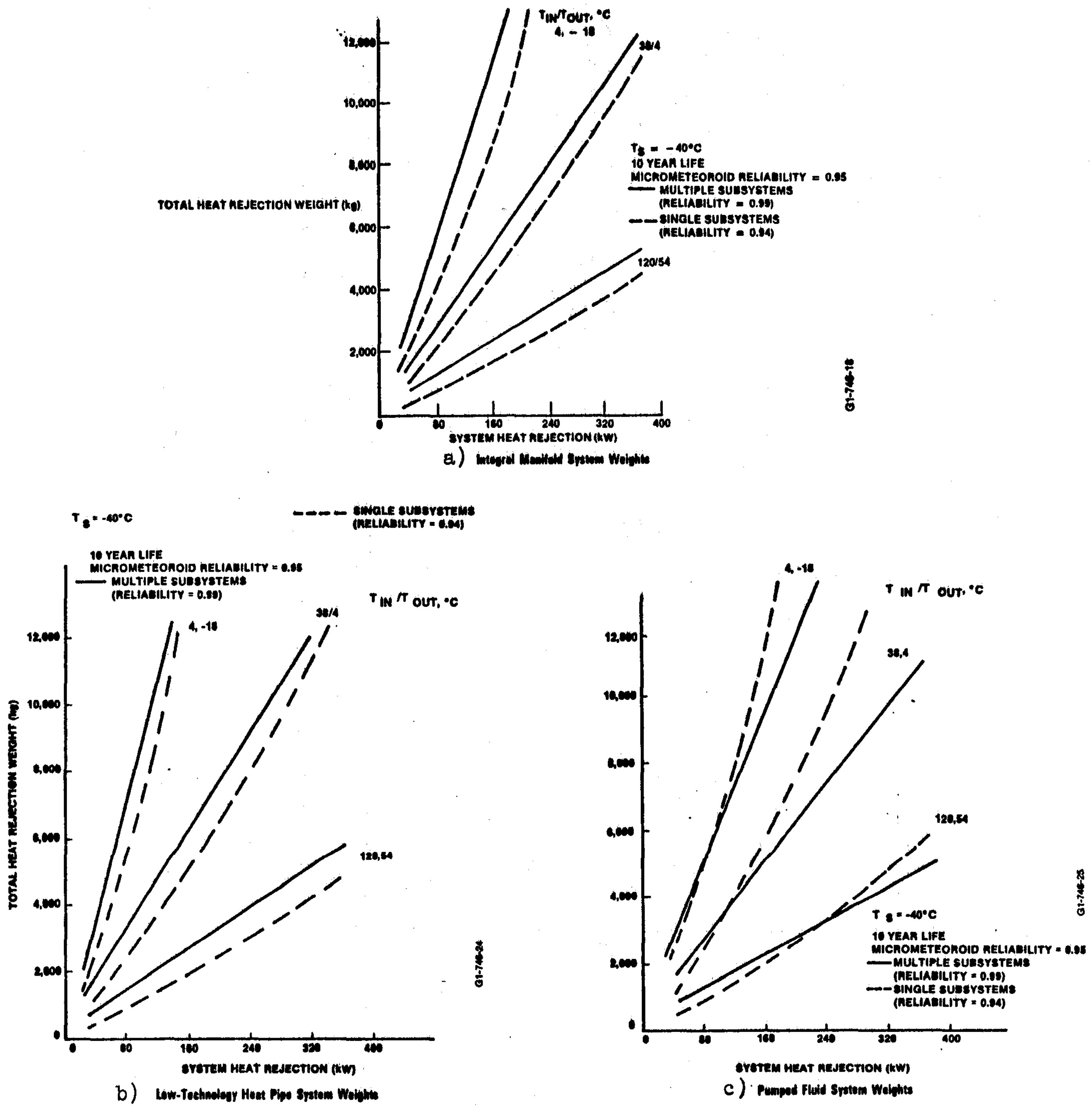


FIGURE 42

OPTIMUM HEAT REJECTION SYSTEM WEIGHTS
FOR -40°C SINK TEMPERATURE

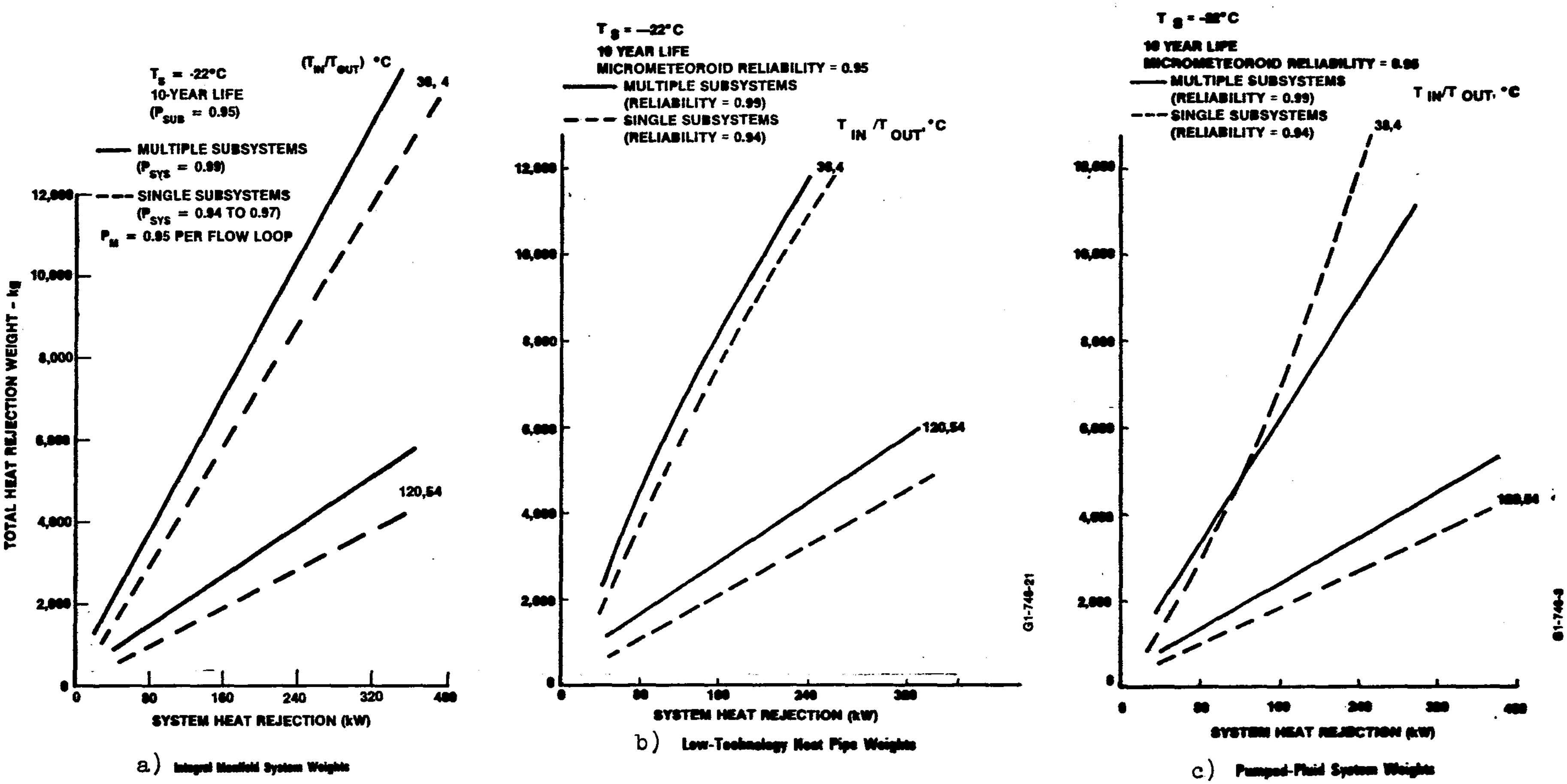


FIGURE 43

OPTIMUM HEAT REJECTION SYSTEM WEIGHTS FOR -22°C SINK TEMPERATURE weight than the multiple subsystem approach.

The subsystem size optimization study discussed above applies only to the pumped fluid, integral manifold and low technology heat pipe approaches. It does not apply to the constructable and deployed constructable radiator approaches discussed in the next section. A different approach is required for these systems.

Constructable Radiator Studies

The space constructable radiator is a new and advanced radiator concept currently under study by NASA. This approach is characterized by numerous small radiator panels, each of which can be easily installed or removed from the radiator system without breaking fluid connections. Two approaches were considered for the constructable radiators approach. In one the constructable radiator is automatically deployed on-orbit but the panels may still be removed and replaced if a failure occurs. Figure 29 is an example of a deployed constructable radiator. It shows two independent redundant fluid loops flowing through the heat exchanger section of the constructable radiator. The heat pipe radiators are plugged into cylindrical heat exchangers which transfer heat from the fluid loop to the radiator panels by contact conduction. With this approach the radiators can be unplugged from the system by reducing the contact pressure and pulling the radiator panel out. The segment of the radiator system shown in Figure 29 is one 4 kW submodule of the deployed system which consists of several submodules. Each radiator panel is approximately 1 kW in size and dimensions are on the order of 25 cm wide and 12 cm long, although these dimensions are determined in optimization studies as we will discuss later. The total deployed radiator system is illustrated in Figure 25.

The alternate constructable approach differs in that the radiator panels are not automatically deployed on-orbit. With this approach heat exchangers are contained within the Power Module of the 250 kW Space Platform. This approach allows the entire fluid loop system to be contained within the structure of the Power Module of the Space Platform. Radiator panels must be assembled by EVA or by a remote manipulator system. Considerable on-orbit construction would be required for this approach. We have called the second approach a "constructed" radiator system. Figure 44 shows a schematic of the heat exchanger arrangement within the Power Module Structure for the constructed radiator system. Different configurations were considered but the arrangement shown with 8 parallel flow paths was the lightest weight approach. Since the heat exchangers are contained within the structure, micrometeoroid protection is not required.

The deployed constructables and space constructed radiator systems were sized and weight optimized. The weights are plotted in Figures 45 and 46. These are optimized system weights and are shown as functions of heat load, radiator temperature, and sink temperature. Little difference was found between the deployed and assembled space constructable radiators. When compared with the conventional panel radiators discussed in Section 3.4.2 the constructable panel is heavier for heat loads less than 120 kW and lighter for heat loads above 120 kW. However, the differences were generally less than 10%.

Heat Rejection System Type Applicability

A map was constructed which shows the operating range best suited for each heat rejection system approach. The map given in Figure 47 shows that the single subsystem pumped fluid system is the lowest weight heat rejection system for heat loads below about 40 to 50 kW. The integral manifold heat pipe radiator is lowest weight for a wide range of heat loads from about 50 kW up to 100 to 200 kW depending on the radiating temperature. The space constructable radiator is lowest weight for higher heat loads. If the space constructable radiator should not succeed in the technology development required, a multiple subsystem pumped fluid loop approach is better than integral manifold above 200 to 350 kW depending on the temperature.

4.3.3 Heat Rejection from Module Surfaces

Studies were conducted to evaluate the use of the extensive area of the individual space platform modules to augment the thermal control system. Various concepts were investigated for utilizing this exterior surface to reduce the radiator size. Figure 48 shows one such concept. In this concept the individual cabin contains an air circulation system in which the warm air from the cabin flows through double walls in the cabin and it

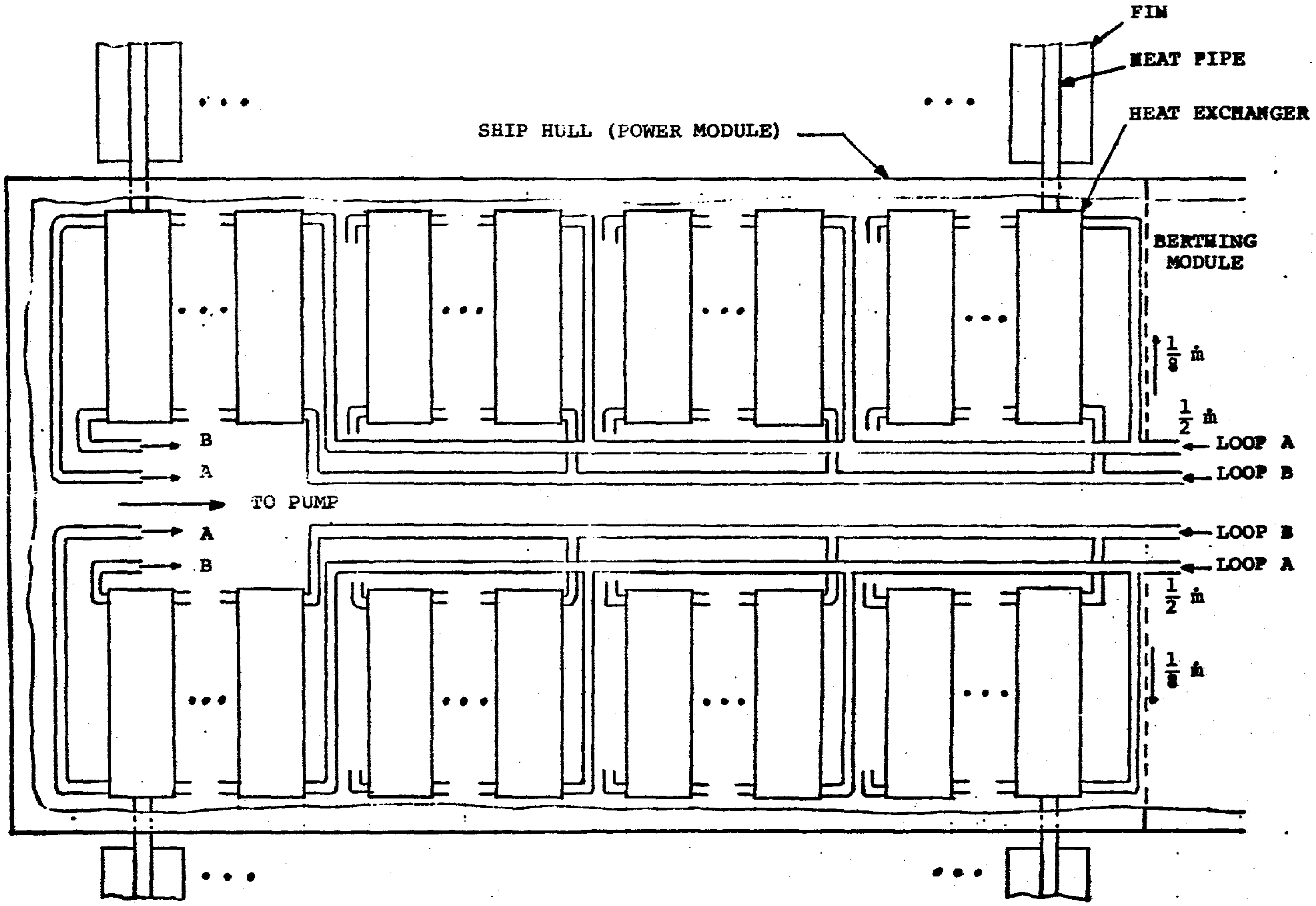


FIGURE 44
SPACE CONSTRUCTED RADIATORS - EIGHT PARALLEL HX SETS DESIGN
BASELINE FLOW ARRANGEMENT

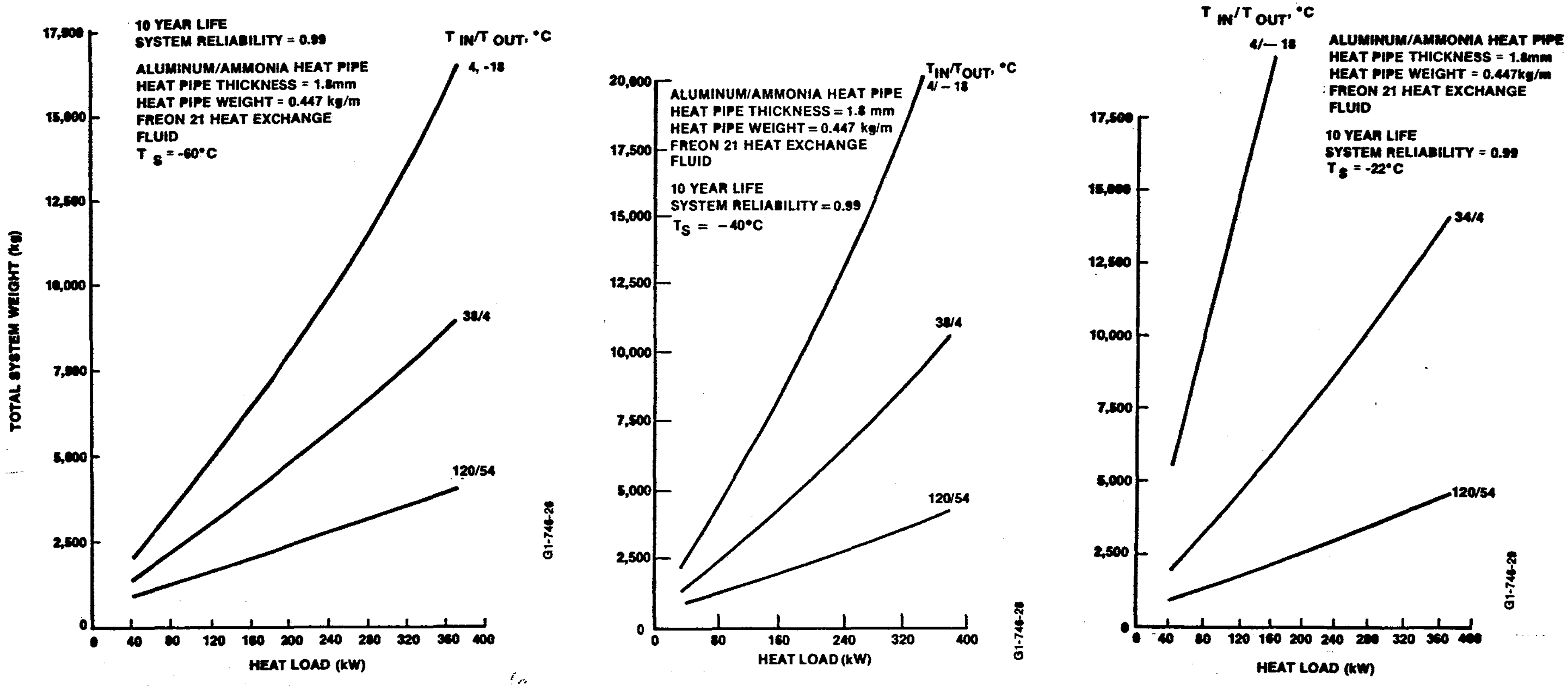


FIGURE 45

SPACE CONSTRUCTABLE RADIATOR WEIGHTS AT DIFFERENT RADIATOR SINK TEMPERATURES

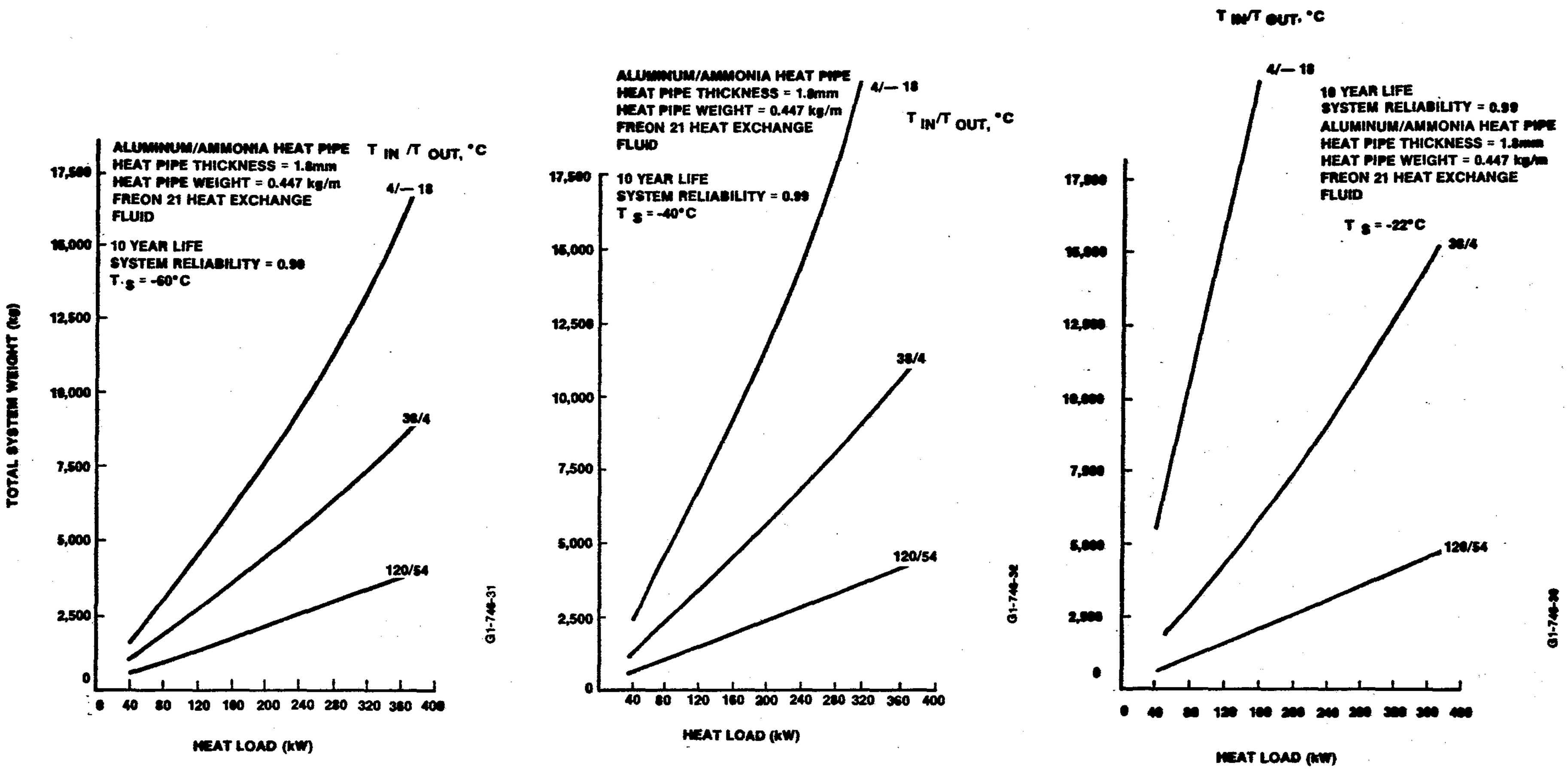


FIGURE 46

SPACE CONSTRUCTED RADIATOR OPTIMUM WEIGHTS AT DIFFERENT RADIATOR SINK TEMPERATURES

FIGURE 47 HEAT REJECTION TYPE APPLICABILITY MAP

VOUGHT

$T_S = -40°C$

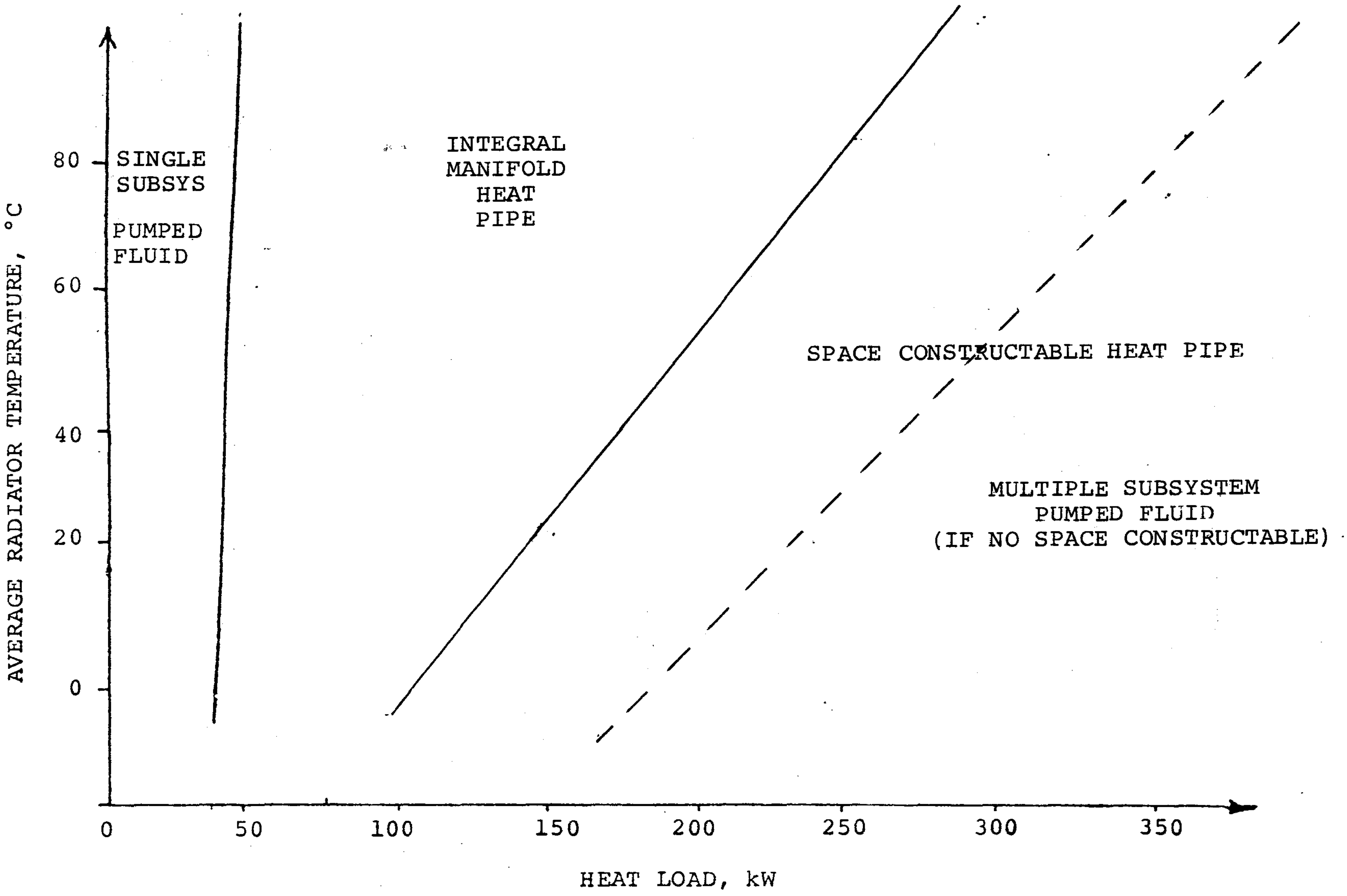
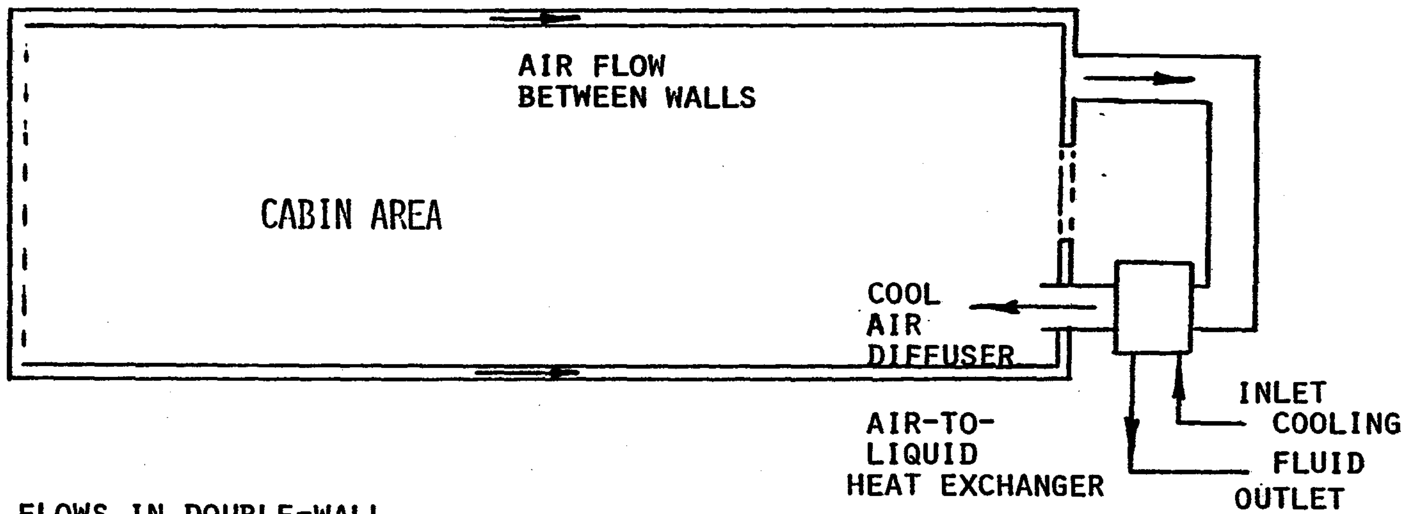


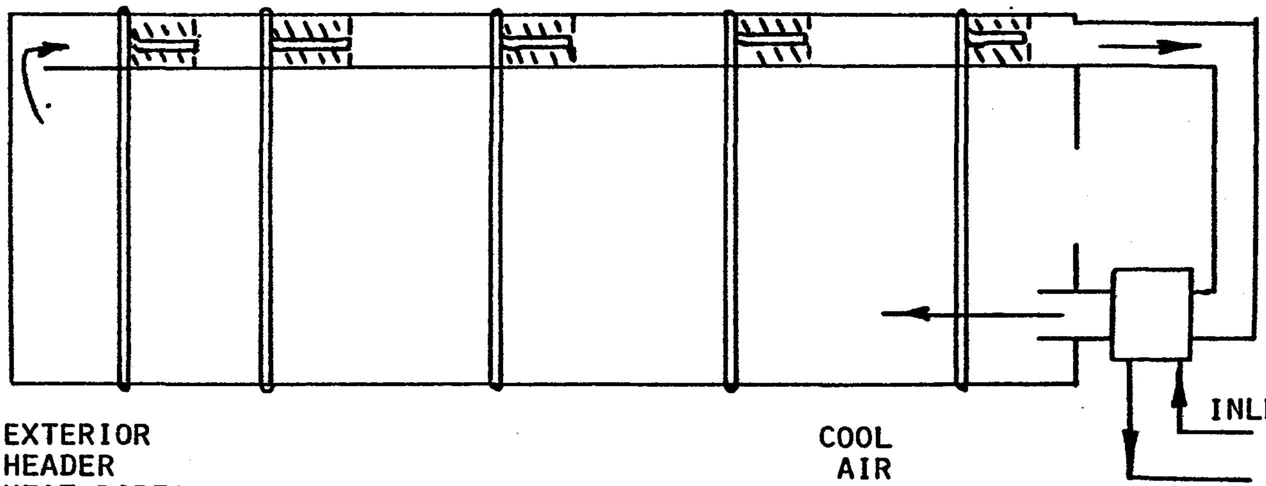
FIGURE 48
CABIN WALL HEAT REJECTION CONCEPT NO. 1
DOUBLE-WALL DUCTED AIR FLOW

TYPICAL MODULE OF SPACE PLATFORM



- CABIN AIR FLOWS IN DOUBLE-WALL PASSAGE & LOSES HEAT DUE TO RADIATION FROM OUTER WALL
- INCREASES FILM COEFFICIENT & TEMPERATURE OF AIR IN CONTACT WITH RADIATING SURFACE
- REDUCES HEAT PICKED UP BY COOLING FLUID
- CONTROL COULD BE ACHIEVED BY A BYPASS AIR DUCT OR BY USE OF CONTROLLABLE TRANSVERSE HEAT PIPES BETWEEN AIR PASSAGE & OUTER WALL.

FIGURE 49
CABIN WALL CONTROLLED HEAT LEAK CONCEPT NO. 2
HEAT PIPE-TO-AIR HEAT EXCHANGER IN AIR DUCT



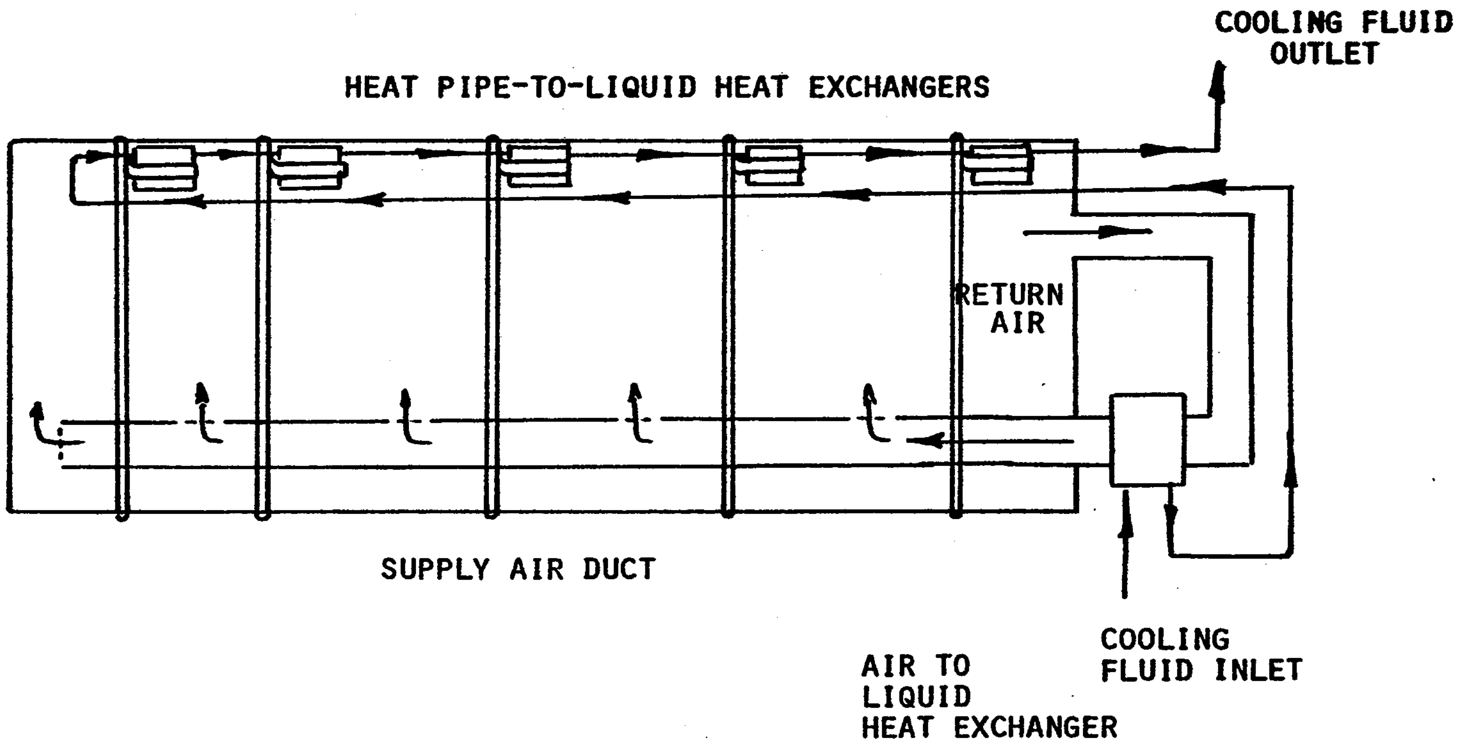
- WARM CABIN AIR REJECTS SOME OF ITS HEAT TO MODULE EXTERIOR VIA HEAT PIPES
- LIQUID COOLING FLUID PROVIDES FINAL TEMPERATURE CONTROL
- ADDITIONAL CONTROL SUCH AS VCHP'S OR AIR BYPASS MAY BE REQUIRED

rejects heat to space. The cooled air then flows through a fluid to gas heat exchanger during which the air temperature is reduced still further before it flows back into the cabin. The heat that is removed from the cabin air by the centralized fluid system is transported back to the radiator system for rejection. The majority of the cabin heat could be lost to the environment by the cabin walls (up to 70%) and thus a sizeable reduction in the centralized radiator system would occur using this approach. Figure 49 shows a second concept for rejecting heat from the cabin air to the cabin exterior surface. In this approach the warm air again flows through the cabin area to pick up the cabin heat load. The air then flows through a duct which contains air-to-heat pipe heat exchangers. These heat exchangers transfer heat from the cabin air to heat pipes which penetrate the cabin wall and conduct heat from the heat exchanger to the exterior surface of the wall. In addition to these primary heat pipes which transmit the heat out to the cabin surface, smaller heat pipes would be required which interface with the larger heat pipes to spread the heat over the cabin exterior surface. In effect, heat pipe radiators are built into the cabin surface. The air is cooled as heat is rejected to the exterior surface of the cabin wall. This cooler air then flows into fluid-to-gas heat exchangers and the remaining cabin air heat removal is accomplished. The heat removed is then transported to the centralized system and rejected. The net effect is a reduction in the size of the deployed radiator system.

A third concept for utilizing the cabin exterior surface is shown in Figure 50. This concept is essentially an all liquid concept in which the heat removal from the cabin air is performed in an air-to-liquid heat exchanger contained within the central fluid loop. The central fluid loop flows through liquid-to-heat pipe heat exchangers which transfer the heat to the exterior walls of the cabin.

A weight estimate was only made for Concept No. 3, the fluid-to-heat pipe concept. For each 18 m by 4.5 m diameter module, 157 m^2 of area (60%) was assumed to be available. A radiation sink temperature of $-34^{\circ}C$ was estimated along with an average radiation temperature of $4^{\circ}C$ ($13^{\circ}C$ internal temperature with a $9^{\circ}C$ temperature drop). This results in

FIGURE 50 CONTROLLED CABIN HEAT LEAK CONCEPT NO. 3 FLUID-TO-HEAT PIPE HEAT EXCHANGER



- COOLING LIQUID HEAT PICKED UP IN AIR-TO-LIQUID HEAT EXCHANGER IS PARTIALLY REJECTED VIA CABIN WALL HEAT PIPE PRIOR TO FLUID RETURNING TO CENTRAL THERMAL CONTROL SYSTEM.
- OUTER HEAT PIPE PANELS COULD BE REMOVABLE
- CONTROL COULD BE VIA CONTROLLABLE HEAT PIPES OF FLUID BYPASS

the ability to reject 18.3 kW of the heat load from each 18 m long module. The radiator weight for each module was estimated at 385 kg for 3.8 cm heat pipes spaced at 6 inches. The weight of the fluid-to-heat pipe "plug-in" type heat exchanger was estimated to be 195 kg per module. Thus, the total radiator weight with the capability to reject 18.3 kW was estimated at 580 kg.

Examination of the surface area available on the baseline 250 kW power system modules indicated that approximately 1390 to 1490 m^2 are available for radiators (6 modules at 4.5 m dia. by 15 m length, one at 4.5 m dia. by 18 m and one at 4.5 m dia. by 9 m with 60% of the area available). However, over 2140 m^2 are required to reject 250 kW of heat. Thus, it appears that sufficient area is not available for rejecting all the heat.

Weight estimates were made for the Integral Manifold (single subsystem) and the space constructed radiator systems assuming that half the heat was rejected via radiators mounted on the external modules and half by the centralized radiator. Also, a weight estimate was made for rejection of all the heat by module mounted radiators, assuming sufficient area could be found. The weight estimates were as follows:

1) Single subsystem hybrid integral manifold heat pipe central system augmented by 50% of heat rejection from cabin exterior surface	7570 kg
2) Space constructable radiator augmented by 50% of heat rejection from cabin exterior surface . .	7711 kg
3) All heat rejection from module surfaces	7940 kg

These results indicate that a slight weight savings can be realized for the integral manifold radiator system (approximately 680 kg or 8%) but a weight increase of 770 kg results for the space constructable radiator. The advantage of using the cabin surface for heat rejection is the reduction in the deployed radiator area which could block the view of instruments and payloads. However, there are some disadvantages to the body mounted radiator approach. One disadvantage is the fact that the heat rejection is much more sensitive to degradation of the thermal control coating properties because there is a greater likelihood of the radiator being radiated by the sun for extended periods of time. Also, since there is more solar flux, the degraded properties would have a bigger impact on performance. Another disadvantage would be the additional launch weight that the radiators would add to each module. While the individual weights are slightly less for a given vehicle configuration, the total launched weight will become greater with multiple launches of a given payload if the heat rejection system is carried on the payload.

4.3.5 Heat Rejection System Parametric Cost Analysis

Cost analysis was performed for each of the following heat rejection system concepts:

Concept	Heat Load
1) Pumped Fluid	
a) Single Subsystem	25 to 250 kW
b) Multiple Subsystem	25 to 250 kW
2) Integral Manifold Heat Pipe	
a) Single Subsystem	25 to 250 kW
b) Multiple Subsystem	25 to 250 kW
3) Low Technology Heat Pipe	
a) Single Subsystem	25 to 250 kW
b) Multiple Subsystem	25 to 250 kW
4) Space Constructed Radiator	250 kW
5) Space Constructable Radiator	250 kW
6) Single Subsystem Integral Manifold/Body Mounted Heat Pipe (50%/50%)	250 kW
7) Space Constructed Heat Pipe Radiator/Body Mounted Radiator (50%/50%)	250 kW
8) All Body Mounted Radiator	250 kW

The assumptions for the cost analysis were those discussed in Section 3.1 which included January 1988 Development Start, January 1989 Prototype Complete, January 1990 Development Complete, February 1991 Production Start and August 1992 Delivery. The year of economics is 1980 dollars and the year of technology is 1985. The PRICE routine was used for the analysis. The complexity factors used are summarized in Section 3.1. The heat rejection system cost analyses assumed the heat rejection system included

TABLE 27

25 kW PUMPED FLUID RADIATOR COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	MULTIPLE (4, 8.33 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	2299	638	2938	2196	406	2603
Heat Pipes	-	-	-	-	-	-
HR Loop Pump	175	560	735	925	140	1065
HR Loop Accumulator	329	9	338	482	6	488
Temperature Control Valve	153	178	331	120	46	166
Temperature Sensor	89	16	105	70	10	80
Heat Exchanger	405	496	901	810	490	1300
Lines and Fittings	198	12	210	198	8	205
Deployment Mechanism	1191	84	1275	1680	38	1718
Integration and Test	722	87	809	884	44	928
TOTALS			7642			8553

TABLE 28

50 kW PUMPED FLUID COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (7,8.33 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	2299	1022	3321	2131	873	3004
Heat Pipes	-	-	-	-	-	-
HR Loop Pump	175	980	1155	1025	160	1185
HR Loop Accumulator	329	14	343	721	10	732
Temperature Control Valve	153	283	436	120	46	166
Temperature Sensor	89	18	107	70	10	80
Heat Exchanger	405	792	1197	810	790	1600
Lines and Fittings	198	20	218	198	17	215
Deployment Mechanism	1191	133	1324	1680	72	1752
Integration and Test	722	128	850	1627	89	1716
TOTALS			8951			10450

TABLE 29

100 kW PUMPED FLUID COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (9,12.5 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	2690	1920	4610	2734	1865	4598
Heat Pipes	-	-	-	-	-	-
HR Loop Pump	175	1080	1255	1125	280	1405
HR Loop Accumulator	421	24	445	1050	17	1067
Temperature Control Valve	153	349	502	120	46	166
Temperature Sensor	89	19	108	70	10	80
Heat Exchanger	545	1400	1945	1090	1400	2490
Lines and Fittings	202	35	237	202	31	233
Deployment Mechanism	1216	167	1383	1730	152	1882
Integration and Test	975	209	1184	3070	185	3256
TOTALS			11669			15177

TABLE 30

250 kW PUMPED FLUID COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (23,13.1 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	1765	4721	6480	3039	5854	8893
HR Loop Pump	196	1749	1945	45	94	139
HR Loop Accumulator	340	55	395	1784	68	1852
Temperature Control Valve	63	344	407	382	308	690
Temperature Sensor	51	24	74	51	9	60
Heat Exchanger	317	1238	1552	1296	1333	2629
Lines and Fittings	342	15	357	632	34	666
Deployment Mechanism	2721	873	3594	2908	731	3640
Integration and Test	7566	756	8322	6734	544	7278
TOTALS			23100			25800

TABLE 31

25 kW INTEGRAL MANIFOLD HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (4,8.33 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	2865	651	3516	2439	403	2842
Heat Pipes	600	418	1018	600	297	897
HR Loop Pump	175	560	735	925	140	1065
HR Loop Accumulator	329	9	338	482	6	488
Temperature Control Valve	153	178	331	120	46	166
Temperature Sensor	89	16	105	70	10	80
Heat Exchanger	405	496	901	810	490	1300
Lines and Fittings	198	12	210	198	8	205
Deployment Mechanism	1191	84	1275	1680	38	1718
Integration and Test	731	93	833	1044	54	1099
TOTALS			9262			9860

TABLE 32

50 kW INTEGRAL MANIFOLD HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (5,12.5 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	3260	1105	4365	2761	789	3550
Heat Pipes	600	756	1356	600	589	1189
HR Loop Pump	175	700	875	1025	160	1185
HR Loop Accumulator	421	14	435	721	10	731
Temperature Control Valve	153	214	367	120	46	166
Temperature Sensor	89	17	106	70	10	80
Heat Exchanger	545	853	1398	1090	850	1940
Lines and Fittings	198	14	212	198	8	205
Deployment Mechanism	1216	102	1318	1680	72	1752
Integration and Test	975	203	1178	2080	125	2205
TOTALS			11610			13000

TABLE 33

100 kW INTEGRAL MANIFOLD HEAT PIPE COST ANALYSIS

$T_{IN} = 38,\ T_{OUT} = 4,\ T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (9,12.5 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	3260	1812	5072	3041	1583	4624
Heat Pipes	600	1361	1961	600	1183	1783
HR Loop Pump	175	1080	1255	1125	280	1405
HR Loop Accumulator	421	23	444	979	16	995
Temperature Control Valve	153	349	502	120	46	166
Temperature Sensor	89	19	108	70	10	80
Heat Exchanger	545	1400	1945	1090	1400	2490
Lines and Fittings	202	23	225	202	31	233
Deployment Mechanism	1216	167	1383	1680	118	1798
Integration and Test	975	209	1184	3616	196	3812
TOTALS			14079			17386

TABLE 34

250 kW INTEGRAL MANIFOLD HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (21,12.5 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	3274	5996	9269	2331	4422	6753
Heat Pipes	126	199	326	403	671	1074
HR Loop Pump	373	1362	1735	269	334	603
HR Loop Accumulator	467	56	523	1891	74	1965
Temperature Control Valve	78	205	284	343	5	347
Temperature Sensor	54	19	73	51	9	60
Heat Exchanger	487	1544	2031	1296	1333	2629
Lines and Fittings	282	8	290	1383	36	1420
Deployment Mechanism	2278	499	2777	3135	800	3933
Integration and Test	8022	827	8849	7302	640	7942
TOTALS			26150			26700

TABLE 35

25 kW LOW TECHNOLOGY HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oF$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	MULTIPLE (4,8.33 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	3148	814	3962	2567	519	3086
Heat Pipes	100	240	340	100	189	289
HR Loop Pump	175	560	735	925	140	1065
HR Loop Accumulator	329	9	338	482	6	488
Temperature Control Valve	153	178	331	120	46	166
Temperature Sensor	89	16	105	70	10	80
Heat Exchanger	405	496	901	1090	496	1586
Lines and Fittings	198	12	210	198	8	205
Deployment Mechanism	1191	84	1275	1680	38	1718
Integration and Test	863	101	964	1245	63	1307
TOTALS			9161			9990

TABLE 36

50 kW LOW TECHNOLOGY HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (5,12.5 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	4102	1402	5504	2810	1012	3822
Heat Pipes	100	450	550	100	476	576
HR Loop Pump	175	700	875	1025	160	1185
HR Loop Accumulator	421	14	435	721	10	731
Temperature Control Valve	153	214	367	120	46	166
Temperature Sensor	89	17	106	70	10	80
Heat Exchanger	545	853	1398	1090	850	1940
Lines and Fittings	198	14	212	198	8	205
Deployment Mechanism	1216	102	1318	1620	72	1692
Integration and Test	1123	154	1277	2166	113	2279
TOTALS			12042			12676

TABLE 37

100 kW LOW TECHNOLOGY HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	MULTIPLE (9,12.5 kW) SUBSYSTEMS DEVELOPMENT	MULTIPLE (9,12.5 kW) SUBSYSTEMS PRODUCTION	MULTIPLE (9,12.5 kW) SUBSYSTEMS TOTAL	SINGLE SUBSYSTEM DEVELOPMENT	SINGLE SUBSYSTEM PRODUCTION	SINGLE SUBSYSTEM TOTAL
Radiator Panels	4102	2302	6403	3176	1961	5138
Heat Pipes	100	810	910	100	1079	1176
HR Loop Pump	175	1080	1255	1125	280	1405
HR Loop Accumulator	421	23	444	979	16	995
Temperature Control Valve	153	349	502	120	46	166
Temperature Sensor	89	19	108	70	10	80
Heat Exchanger	545	1400	1945	1090	1400	2490
Lines and Fittings	202	23	225	202	31	233
Deployment Mechanism	1216	167	1383	1620	124	1744
Integration and Test	1123	235	1358	3870	233	4103
TOTALS			14533			17530

TABLE 38

250 kW LOW TECHNOLOGY HEAT PIPE COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE (14,22.7 kW) SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels	3094	5733	8827	2338	4596	6934
Heat Pipes	126	199	326	296	447	743
Pump	373	1362	1735	442	343	785
Accumulator	467	56	523	1679	62	1741
Temperature Control Valve	78	205	284	319	252	571
Temperature Sensor	54	19	73	51	9	60
Heat Exchanger	487	1544	2031	1296	1333	2629
Lines and Fittings	282	8	290	434	19	453
Deployment Mechanism	2278	499	2777	2886	719	3604
Integration and Test	7802	805	8607	6919	616	7535
TOTALS			25470			25050

TABLE 39

250 kW SPACE CONSTRUCTED RADIATOR

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

	MULTIPLE SUBSYSTEMS			SINGLE SUBSYSTEM		
COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels Heat Pipes	-	-	-	1689	3661	5350
HR Loop Pump	-	-	-	448	347	795
HR Loop Accumulator	-	-	-	1217	39	1256
Temperature Control Valve	-	-	-	325	1044	1370
Temperature Sensor	-	-	-	51	14	65
Heat Exchanger	-	-	-	464	1865	2329
Lines and Fittings	-	-	-	2072	57	2129
Integration and Test	-	-	-	6166	776	6942

TOTAL 20236

TABLE 40

250 kW CONSTRUCTABLE RADIATOR COST ANALYSIS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^oC$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	MULTIPLE SUBSYSTEMS			SINGLE SUBSYSTEM		
	DEVELOPMENT	PRODUCTION	TOTAL	DEVELOPMENT	PRODUCTION	TOTAL
Radiator Panels Heat Pipes	-	-	-	1627	3568	5195
HR Loop Pump	-	-	-	448	347	795
HR Loop Accumulator	-	-	-	1175	38	1213
Temperature Control Valve	-	-	-	177	516	692
Temperature Sensor	-	-	-	51	14	65
Heat Exchanger	-	-	-	463	1853	2316
Lines and Fittings	-	-	-	1942	52	1994
Deployment Mechanism	-	-	-	3959	565	4524
Shielding	-	-	-	999	75	1073
Integration and Test	-	-	-	9391	851	10242
TOTAL						28109

TABLE 41

250 kW SYSTEM WITH 125 kW SINGLE SUBSYSTEM INTEGRAL MANIFOLD AND 125 kW BODY MOUNTED HEAT PIPE RADIATOR

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Body Mounted Radiator Heat Pipes/ Panel	248	644	892
Blower/Motor	123	304	427
Duct	708	65	774
Temperature Control Valve	116	282	399
Temperature Sensors	51	14	64
Heat Exchanger: Fluid-to-Heat Pipe	94	2352	2446
Heat Exchanger: Fluid-to-Air	126	305	430
Lines and Fittings	331	9	340
Deployed Radiators	2331	2509	4840
Radiator Heat Pipes	403	671	1074
Pump/Motor	269	334	603
Accumulators	1891	74	1965
Temperature Control Valves	191	143	334
Temperature Sensors	51	9	60
Heat Exchangers	752	734	1489
Lines and Fittings	1243	30	1273
Deployment Mechanism	111	454	565
Integration and Test	6867	783	7650
TOTAL			25625

TABLE 42

250 kW SYSTEM WITH 125 kW SPACE CONSTRUCTED RADIATOR AND
125 kW BODY MOUNTED HEAT PIPE RADIATOR

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40^{\circ}C$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Body Mounted Heat Pipe Panels	248	644	892
Blower/Motor	123	304	427
Duct	708	65	774
Temperature Control Valves	116	282	399
Temperature Sensors	51	14	64
Heat Exchanger: Fluid-to-Heat Pipe	94	2352	2446
Heat Exchanger: Fluid-to-Air	126	305	430
Lines and Fittings	331	9	340
Heat Pipe-Fin Panels	975	2100	3075
Pump/Motor	379	291	670
Accumulators	842	27	869
Temperature Control Valves	177	515	692
Temperature Sensors	51	14	65
Heat Exchangers	265	1075	1340
Lines and Fittings	1557	39	1595
Integration and Test	5870	810	6680
TOTAL			20758

TABLE 43

250 kW ALL BODY MOUNTED HEAT PIPE RADIATORS

$T_{IN} = 38$, $T_{OUT} = 4$, $T_S = -40°C$

COST - THOUSANDS OF 1980 DOLLARS

COMPONENT	DEVELOPMENT	PRODUCTION	TOTAL
Body Mounted Heat Pipe Panels	223	2519	2742
Blower/Motor	491	325	816
Duct	684	2	685
Temperature Control Valves	116	282	399
Temperature Sensors	46	11	57
Heat Exchangers: Fluid-to-Air	126	305	430
Heat Exchanger: Fluid-to-Heat Pipe	91	5238	5329
Lines and Fittings	126	305	430
Pump Motor	51	298	349
Integration and Test	6352	1108	7460
TOTAL			18560

the radiator panels, heat pipes and the entire closed pumped fluid loop including all the components.

The results of the heat rejection system cost analysis are summarized in Tables 27 through 43. The cost for each concept is broken down by component and by development and production costs. The costs for three concepts, the pumped fluid, integral manifold heat pipe, and the low technology heat pipe are analyzed parametrically over a range of heat loads from 25 to 250 kW and for both the multiple subsystem and single subsystem approaches. The other five concepts are only analyzed for the 250 kW heat load. All concepts except the body mounted radiators were also analyzed for different radiation sink temperatures for the 250 kW heat loads.

The results of the pumped fluid heat rejection system cost analyses are given in Tables 27 through 30. The results show the cost for the multiple subsystem varying from $7.6 million dollars for a 25 kW system to 23.1 million dollars for a 250 kW system. The single subsystem costs were approximately 12% higher for all heat loads, ranging from 8.5 million dollars at 25 kW to 25.8 million dollars at 250 kW. The specific energy rejection cost decrease from 300 to 340 $/kW at 25 kW to 90 to 100 $/kW at 250 kW.

The results of the integral manifold heat pipe hybrid heat rejection system cost analysis are given in Tables 31 through 34. The multiple subsystem costs range from 9.3 million dollars for a 25 kW system to 26.2 million dollars for a 250 kW system. The single subsystem costs range from $9.9 million to $26.7 million for the 25 to 250 kW heat load range, about 2 to 7% higher than the multiple subsystem cost. The integral manifold costs are 13 to 20% higher than the pumped fluid system costs.

The low technology heat pipe costs are shown in Tables 35 through 39. The multiple subsystem costs for this approach range from $9.2 million at 25 kW to $25.5 million at 250 kW. The single subsystem costs range from $10 million at 25 kW to $25 million at 250 kW which is comparable to slightly lower than the integral manifold approach.

The space constructed radiator cost for a 250 kW system is shown in Table 39 to be $20.2 million. This is almost $2 million, or 12%, less than the multiple subsystem pumped fluid cost of $23.1. However, it must be pointed out that the $20.2 million cost for the space constructed radiator does not include deployment, whereas the pumped fluid cost does. Table 40 shows that the cost of the automatically deployed space constructable radiator is $28.1 million or almost $8 million higher. The additional costs are $4.5 million for a deployment mechanism and about $3.5 million additional integration and test cost.

The cost analysis results for 250 kW heat rejection systems augmented by body mounted heat pipe radiators are shown in Tables 41, 42, and 43. Table 41 shows the cost of a system with 50% (125 kW) of the heat load rejected by a single subsystem integral manifold heat pipe system and 50% (125 kW) rejected by heat pipe panels on the cabin walls. The projected cost of this system is $25.6 million compared to $26.7 million for a 250 kW integral manifold system. Table 42 shows the cost of a space constructed radiator with half the heat load rejected by body mounted heat pipe radiator panels to be $20.8 million compared to $20.2 million for a 250 kW space constructed radiator panel. Table 43 shows the cost of an all body mounted heat pipe radiator system to be $18.6 million. Thus, system costs are not affected much by the use of body mounted heat pipe systems unless all heat is rejected in that manner.

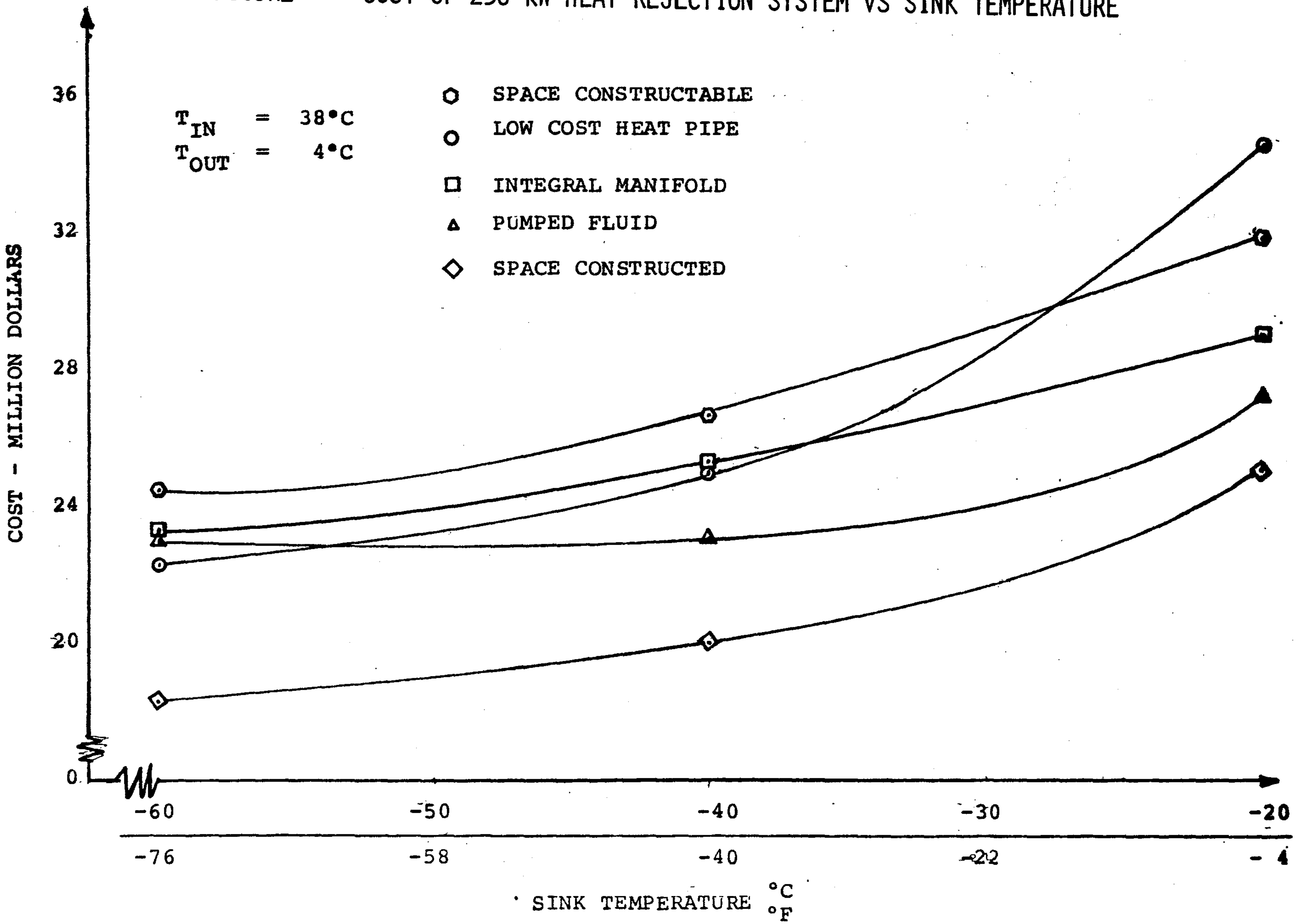
Cost analyses were also performed parametrically for a range of radiator sink temperatures for the multiple subsystem pumped fluid, single subsystem integral manifold, single subsystem low technology heat pipe, space constructed radiator and the automatically deployed space constructable radiator. The results of this parametric study are shown in Figure 51.

4.3.6 Heat Rejection System Concepts Trades and Analysis

Eight of the concepts studied were selected for further evaluation and trades. These concepts are tabulated in Table 44. The trade criteria for additional evaluation were grouped under the following six major categories: Potential for Benefit, Development Considerations, Operational Considerations, Impact to Vehicle, Performance Considerations, and Reliability and Life Considerations.

Table 45 shows the trade matrix which evaluates each trade criteria for each concept. The concept comparison for each major category are

FIGURE 51 COST OF 250 kW HEAT REJECTION SYSTEM VS SINK TEMPERATURE



144

TABLE 44

MOST PROMISING HEAT REJECTION SYSTEM CONCEPTS

1. MULTIPLE SUBSYSTEM PUMPED FLUID, RIGID DEPLOYMENT
2. MULTIPLE SUBSYSTEM HYBRID, INTEGRAL MANIFOLD HEAT PIPE
3. SINGLE SUBSYSTEM HYBRID INTEGRAL MANIFOLD HEAT PIPE RIGID DEPLOYMENT
4. SPACE CONSTRUCTED RADIATOR (NO DEPLOYMENT)
5. SINGLE SUBSYSTEM HYBRID INTEGRAL MANIFOLD HEAT PIPE, RIGID DEPLOYMENT AUGMENTED BY SEMI-PASSIVE BODY MOUNTED HEAT PIPE RADIATORS
6. SPACE CONSTRUCTED RADIATOR AUGMENTED BY BODY MOUNTED HEAT PIPE RADIATOR PANELS
7. AN ALL BODY MOUNTED RADIATOR SYSTEM
8. SPACE CONSTRUCTABLE RADIATOR (AUTOMATICALLY DEPLOYED)

TABLE 45 CONCEPT TRADE MATRIX FOR HEAT REJECTION SYSTEMS RANKED CRITERIA

RANKED CRITERIA	ORDER OF PRIORITY	CONCEPT NO.							
		1	2	3	4	5	6	7	8
Potential for Benefit									
o Cost $M	1	23.1	26.2	26.7	20.2*	25.6	20.8*	18.6**	28.1
o Operations	1	Fair	Fair	Fair	Good	Good	Good	Good	Good
o Integration with Other Systems	2	Good	Good	Good	Good	Good	Good	Fair	Good
o Growth & Reconfig.	1	Good	Good	Fair	Good	Good	Good	Poor	Good
o Autonomous Oper.	1	Good	Good	Good	Good	Good	Good	Excel	Good
o Reduced Impacts	2	None	None	Fair	None	Good	Good	Good	None
o Long Life	1	Good	Good	Good	Good	Good	Good	Good	Good
Development Considerations									
o Costs $M	1	13.4	15.4	18.4	12.4	15.9	11.9	8.3	20.2
o Lead Time	2	Good	Fair	Fair	Poor	Fair	Poor	Fair	Poor
o Evolutionary Capability	1	Good	Good	Fair	Excel	Good	Excel	Good	Excel
o Potential for Success	1	Excel	Excel	Good	Fair	Fair	Fair	Fair	Fair
Operational Considerations									
o Constructability Erectability	1	Good	Good	Fair	Excel	Poor	Fair	Poor	Excel
o Operational Constraints	2	Good	Good	Good	Good	Fair	Fair	Fair	Good
o EVA/RMS Replaceability	2	Fair	Fair	Poor	Excel	F/G	Good	F/G	Excel
o Reconfiguration & Opera. Versatility	1	Good	Good	Poor	Excel	Fair	Good	Fair	Excel

* Does not include construction costs.
** Much more dependent on configuration.

TABLE 45 (CONTINUED)

RANKED CRITERIA	ORDER OF PRIORITY	CONCEPT NO. 1	2	3	4	5	6	7	8
Impacts									
o Payload Contamin.	1	Fair	Fair	Fair	Good	Good	Good	Good	Fair
o Drag	2	Poor	Poor	Poor	Poor	Good	Good	Excel	Poor
o Moment of Inertia	2	Poor	Poor	Poor	Poor	Good	Good	Excel	Poor
o Payload Blockage	2	Poor	Poor	Poor	Poor	Good	Good	Excel	Poor
o Compatibility with Alternate Vehicle Configuration	1	Excel	Excel	Good	Good	Fair	Fair	Poor	Good
o Modularity	2	Good	Good	Poor	Excel	Fair	Good	Good	Excel
o Payload & Module Interfaces	1	Good	Good	Good	Good	Fair	Fair	Fair	Good
Performance Considerations									
o Weight, kg	1	7760	8760	7600	6940	7570	7700	7940	7080
o Deployed Area, M^2 (Planform)	1	950	1180	850	1160	420*	600*	-0-**	1190
o Controllability	1	Good	Good	Good	Good	Good	Good	Good	Good
o Space Environment Compatibility	2	Good	Good	Good	Good	Good	Good	Good	Good
o Temperature Range ^{o}C	2	120 to -120	85 to -75	85 to -75	85 to -75	85 to -75	85 to -75	85 to -75	85 to -75
Reliability and Life									
o Complexity and No. of Pieces	1	Poor	Poor	Good	Good	Poor	Poor	Good	Good
o Component Life	2	Good	Good	Good	Good	Good	Good	Good	Good
o Maintainability & Health Monitoring	1	Good	Good	Good	Good	Fair	Fair	Fair	Good
o Failure Modes	1	Good	Good	Fair	Good	Good	Good	Good	Fair

* Requires about 1080 M^2 of area on module outer surfaces.
** Requires about 2150 M^2 of area on module outer surfaces (only about 1490 M^2 are available).

discussed separately below.

Potential for Benefit

Primary differences under this category occur in the costs, operational benefits and reduced impacts. The lowest cost approaches are (1) the all body mounted approach (Concept 7) at $18.1 million, (2) the space constructed radiator which require orbital assembly (Concepts 4 and 6) at $20 million and $21 million respectively (the costs do not include orbital assembly) and (3) Concept 1, the pumped fluid concept at $23 million. Concepts 2, 3, and 5, the integral manifold heat pipe concepts all cost abut $26 million, while the automatically deployed space constructable radiator cost is $28 million. Operational advantages are shown for Concepts 4, 6 and 8 ,the space constructable concepts, because of their ease of maintenance, growth and potential reconfiguration. Concepts 5, 6, and 7, the body mounted radiator concepts, have an operational advantage in the reduced deployed radiator area. This also gives Concepts 5, 6, and 7 good ranking for the "Reduced Impacts" criteria.

When all the automatically deployed, centralized systems are compared for this category (i.e., Concept 1, the pumped fluid; Concepts 2 and 3, the integral manifold multiple and single subsystems; and Concept 8, the space constructable radiator), Concept 1 appears to have the highest potential. It has the lowest cost by about 3 to 5 million dollars or 13 to 20%. The operational benefits of Concept 8 do not justify the 20% increase in cost. Concept 8, the automatically deployed space constructable radiator, is ranked second. Concepts 3 and 2 are next in rank. Concept 4 is difficult to compare directly since it is not automatically deployed. However, if orbital assembly could be shown to cost less than $3 million, it might rank No. 1. Concepts 5, 6, and 7 show that the utilization of local module surfaces for heat removal offers the benefit of reduced impacts to the payload viewing without any increase in cost.

Development Considerations

Comparison of the automatically deployed centralized concepts for this category shows the pumped fluid concept (#1) having the advantage due to its advanced stage of development. Second is the Multiple Subsystem Integral

Manifold (#2) which has better growth capability and lower development cost than the single subsystem integral manifold (#3). Concept 3 is ranked third and Concept 8 is ranked fourth. Concept 7 shows a reduction in development cost of approximately $7 million from Concept 2 if orbital construction is used instead of automatic deployment. This also shows up with Concept 6.

Operational Considerations

Comparison of the centralized, automatically deployed heat rejection concepts for this category shows the best concept to be No. 8, the deployed space constructable radiator. It excels in all criteria. Concepts 1 and 2, the multiple subsystem pumped fluid and integral manifold are second in ranking, with Concept 3, the single subsystem integral manifold third. Concept 4 has a ranking equal to Concept 1. The body mounted concepts (5, 6, and 7) generally ranked poorer than the centralized for this category.

Impacts

There is little difference among the centralized, automatically deployed concepts for this category. The body mounted concepts are ranked much better than the centralized systems because of reduced drag, moment of inertia and payload viewing blockage. However, these are not as compatible with alternate vehicle configurations.

Performance Considerations

The comparison of the automatically deployed, centralized systems for this category shows the single subsystem integral manifold to be the first choice with low weight and deployed area. A close second is the pumped fluid concept (#1) which has a slight advantage in its operational temperature range. Third choice is the deployed constructable (#8) which is lowest weight of all but requires 25 to 40% more area. The multiple subsystem is the least desirable of the four with the largest weight and also the largest deployed area. The space constructed radiator (no deployment) ranks about equal to the constructable radiator - i.e., no additional advantage in this category. The body mounted radiator concepts show little or no weight advantage but have the obvious advantage of reduced deployed radiator area.

Reliability and Life

Comparison of the four centralized, automatically deployed panels

TABLE 46

RANKING ORDER FOR HEAT REJECTION CONCEPTS
FOR EACH MAJOR RANKING CATEGORY

RANKING CATEGORY	CONCEPT NO.							
	1	2	3	4	5	6	7	8
POTENTIAL FOR BENEFIT	2	5	3	1	3	1	6	4
DEVELOPMENT CONSIDERATIONS	1	4	5	2	5	2	3	6
OPERATIONAL CONSIDERATIONS	2	2	4	1	4	3	4	1
VEHICLE IMPACTS	1	1	2	1	3	3	4	2
PERFORMANCE CONSIDERATIONS	3	5	2	4	1	1	6	4
RELIABILITY AND LIFE	3	3	2	1	4	4	2	2

for this concept shows the highest ranking being the single subsystem integral manifold (#3) and the space constructable radiator concept (#8). The two multiple subsystem approaches (#1 and #2) are also about equal for second ranking. The space constructed radiator with no deployment (#4) ranks among the top for this category. The body mounted concepts rank lowest for this category.

Overall Ranking

Table 46 shows a summary of the ranking of each concept for the major categories discussed above. Based upon our evaluations of all the ranking criteria and applying judgements as to their relative importance, the following conclusions are reached from the trade study.

1) The highest ranking approach is Concept 4, the space constructed radiator approach. This selection must be qualified by the fact that the construction costs have not been included in the evaluation. However, if the construction costs are found to be less than $3 million, this selection will stand.
2) The second highest ranking concept is essentially a tie between Concept 1, the pumped fluid multiple subsystem approach and Concept 3, the single subsystem integral manifold. The pumped fluid approach has the edge in cost, development status, modularity, and flexibility. The integral manifold approach has the advantages of weight, deployed area, and considerable reduction in system complexity. Our judgement is that the performance and reliability advantages of the integral manifold approach are more important than the cost and flexibility approaches of the pumped fluid system. Thus, our selection for second ranking is Concept 3, the integral manifold with the pumped fluid being a very close third.
3) The use of body mounted radiators on the individual modules offers promise of reducing the deployed radiator area by as much as 50% with little impact in weight or cost. System simplicity and operational flexibility are reduced however.

In addition, launch weight for multiple launches and solar degradation of the thermal coatings would be significant disadvantages. Thus, the use of body mounted radiators are not recommended.

4) The use of the deployed space constructable radiator does not appear attractive because of the high cost of developing and integrating the large deployment systems. Additional studies are needed to determine the best deployment method for space constructable radiators.

5) The multiple subsystem hybrid integral manifold approach is not competitive due to excessive weight and large area. The best approach for the hybrid system for the 250 kW heat load is the single subsystem.

Additional observations concerning the heat rejection system concepts can be made based on the trade studies.

1) The low technology heat pipe approach is not weight competitive with the integral manifold or pumped fluid approaches (about 12% heavier than integral manifold).

2) The single subsystem pumped fluid approach is much heavier than the multiple subsystem (about 30%) at 250 kW heat load. Multiple subsystem approaches are lower in weight for heat loads above 80 to 100 kW for the nominal radiator temperature. Thus, for large pumped fluid concepts, multiple subsystem approaches must be used.

3) The parametric weight study found that single subsystem pumped fluid systems are advantageous for system sizes less than 60 kW. The single subsystem integral manifold concept has a weight advantage between about 60 and 160 kW. Space constructable radiator approaches have the advantage above 160 kW.

4) In cost analyses for 250 kW systems, the space constructed radiator is lowest cost followed by pumped fluid and integral manifold.

Based upon the evaluation and trade studies for the 250 kW heat rejection systems, the baseline selection is as follows:

1) The single subsystem integral manifold hybrid heat pipe system is selected as the near term baseline (1987 to 1990 technology). The multiple subsystem pumped fluid is a close second and is selected as an alternate.
2) The space constructed radiator system is selected as a high technology approach which offers promise of significant advantages. It is selected as a post 1990 technology alternate offering significant payoff.

Description of The Selection Baselines

A technical description of the selected baseline, the single subsystem integral manifold is presented in Table 47. Table 48 shows the description of the competing pumped fluid multiple subsystem approach. The advanced technology (space constructed) approach which offers significant gains is described in Table 49.

TABLE 47

DESCRIPTION OF SELECTED BASELINE: 1985-1987 TECHNOLOGY
INTEGRAL MANIFOLD RADIATOR SYSTEM

250 kW HEAT LOAD, T_{IN} = 311°K (100°F), T_{OUT} = 278°K (40°F)

PANEL LENGTH	:	7.1 m (40.1 Ft)
PANEL WIDTH	:	1.9 m (6.11 Ft)
PANEL THICKNESS (HONEYCOMB)	:	1.27 cm (.50 In.)
HEAT PIPE LENGTH	:	.93 m (3.05 Ft)
NUMBER OF PANELS	:	64
NO. OF HEAT PIPES PER PANEL	:	98
NO. OF HEAT PIPES TOTAL	:	6272
NO. OF HEAT PIPES EXTRA	:	200
HEAT PIPE PERFORMANCE REQD	:	29.21 w-m (1150 w-in)
HEAT PIPE DIAMETER	:	.95 cm (.375 in)
FLOWRATE TOTAL (FREON 21)	:	26205.3 kg/hr (57772.70 LBm/HR)
FLOWRATE PER PANEL	:	8.2 kg/hr (18.05 LBm/HR)
TOTAL PLAN AREA	:	844.1 m^2 (9085.74 Ft^2)
PLAN AREA PER PANEL	:	13.2 m^2 (141.967 Ft^2)
TOTAL WEIGHT	:	8356 kg (18375 LBm)
WEIGHT PANEL	:	84.1 kg (185.34 LBm)

TABLE 48

PUMPED FLUID MULTIPLE SUBSYSTEM RADIATOR SYSTEM

250 kW HEAT LOAD, T_{IN} = 311°K (100°F), T_{OUT} = 278°K (40°F), T_{SINK} = -40°F

NO. OF SUBSYSTEMS TOTAL	: 14
NO. OF SUBSYSTEMS REQUIRED	: 11
NO. OF EXTRA SUBSYSTEMS	: 3
HEAT REJECTION PER SUBSYSTEM	: 22.7 kW
TOTAL SYSTEM WEIGHT	: 7940 kg (17500 lbs)
TOTAL RADIATOR WEIGHT	: 6820 kg (15050 lbs)
TOTAL COMPONENTS WEIGHT	: 1120 kg (2460 lbs)
TOTAL SYSTEM DEPLOYED PLAN AREA	: 1000 m^2 (10800 ft^2)
PANEL LENGTH	: 7.9 m (25.76 ft)
PANEL WIDTH	: 2.3 m (7.5 ft)
PANEL THICKNESS	: 2.5 cm (1.0 in)
TUBE SPACING	: 14.3 cm (5.62 in)
TUBE INTERNAL DIAMETER	: 0.305 cm (0.12 in)
MASS FLOW PER SUBSYSTEM	: 3280 kg/hr (7225 lb/hr)
MANIFOLD DIAMETER	: 0.94 cm (0.37 in)
HEADER DIAMETER	: 2.3 cm (0.90 in)
TUBE WALL THICKNESS	: 0.366 cm (0.144 in)
FIN THICKNESS (1 in. HONEYCOMB)	: 0.079 cm equiv. (0.031 in equiv.)
FIN EFFECTIVENESS	: 0.90
PRESSURE DROP (SUBSYSTEM)	: 124 kPa (18 psi)
AREA PER SUBSYSTEM (PLAN FORM)	: 72 m^2 (773 ft^2)
NO. OF PANELS PER SUBSYSTEM	: 4
WEIGHT PER SUBSYSTEM	: 570 kg (1250 lbs)
PANEL WEIGHT PER SUBSYSTEM	: 490 kg (1075 lbs)
COMPONENT WEIGHT PER SUBSYSTEM	: 80 kg (175 lbs)
FLUID	: Freon 21

TABLE 49

DESCRIPTION OF ALTERNATE CONCEPT: 1990+ TECHNOLOGY
SPACE CONSTRUCTED RADIATOR SYSTEM

250 kW HEAT LOAD, T_{IN} = 311°K (100°F), T_{OUT} = 277.7°K (40°F)

PANEL LENGTH	:	12.23 m (40.1 ft)
PANEL WIDTH	:	22.54 cm (8.88 in)
PANEL THICKNESS	:	0.054 cm (.021 in)
HEAT PIPE LENGTH	:	12.77 m (41.89 ft)
HEAT PIPE THICKNESS	:	0.178 cm (.070 in)
HEAT REJECTION PER PANEL	:	0.951 kW
NUMBER OF PANELS	:	432
NUMBER OF EXTRA PANELS	:	72
HEAT PIPE PERFORMANCE REQUIRED	:	6072 w-m (239,000 w-in)
HEAT PIPE DIAMETER	:	2.5 cm (1 in)
TOTAL PLAN AREA	:	1191 m^2 (12820 ft^2)
TOTAL WEIGHT	:	7030 kg (15,500 lbs)
HEAT EXCHANGER LENGTH	:	.47 m (1.56 ft)
HEAT EXCHANGER DESIGN	:	0.89 cm ANNUGULAR FLOW, REDUNDANT PASSAGE CONTACT HEAT EXCHANGER
HEAT EXCHANGER UA	:	0.096 kW/°C (183 BTU/hr-°F)

5.0 TECHNOLOGY ASSESSMENT

The results of the concept studies were examined to determine the development required to provide technology readiness for the selected baseline system design at the appropriate dates. To begin this effort, an assessment was made of the current state-of-the-art in thermal management. The thermal management system functions were divided into the following categories: heat transport, heat rejection, connectable thermal interfaces, rotating thermal joints, thermal storage, refrigeration, and temperature control. This state-of-the-art assessment is summarized in Table 50. Also included in Table 50 are the currently available methods for meeting the thermal management functions and the current state-of-the-art performance for these methods. An approximation of the SOA life for these approaches is also shown. Table 51 projects the anticipated state-of-the-art requirement for the various thermal management concepts for an early 1990 Space Platform launch. Comparison of Tables 51 and 50 provides an estimate as to the advancement in the state-of-the-art needed in the next 10 years.

The recommended technology advancements to fill the 1990 technology gap are tabulated in Tables 52 and 53 along with comments relative to the expected payoff. Table 52 gives the technology advancement required for the heat transport systems and the heat transport system interfaces. Areas requiring development include extending the state-of-the-art of pumps and heat pipes, developing contact heat exchangers for integration into docking ports, and 360° rotation, no leak, long life fluid swivel. The technology required to handle evaporating and condensing flow in the environment of zero-gravity is required to permit the designing of refrigeration heat pumps to meet the needs of isothermal instruments, payload subsystems and for low temperature requirements. Zero gravity compressors are also needed for this application. A 5000 to 10,000 watt-hr inline thermal storage canister will be required to support payloads and experiments requiring high energy pulses.

The technology advancements required for the heat rejection system are shown in Table 53. Heat pipe-to-fluid heat exchanger technology must be advanced to support the advantages that the single subsystem hybrid heat pipe approach offers over the more complex multiple subsystem pumped

TABLE 50 CURRENT STATE-OF-THE-ART IN THERMAL MANAGEMENT

TMS FUNCTION	CURRENT METHODS FOR FUNCTION	CURRENT PERFORMANCE	LIFE
HEAT TRANSPORT	PUMPED LIQUID HEAT PIPE	UNLIMITED (CURRENT SYSTEM 500,000 w-m) 2540 w-m	2-1/2 YRS. 10 YEARS
HEAT REJECTION	RADIATING PANELS	150 w/m^2 (15°C) 30 w/kg	5 YRS. WITH DEGRADATION
CONNECTABLE THERMAL INTERFACES	QUICK DISCONNECTS IN FLUID LINE	0.68 cc SPILLAGE VOL. ΔP=0.6kPå @ 0.3 kg/s	500 CYCLES
ROTATING THERMAL JOINTS	FLEXIBLE HOSES WHICH ALLOW ONLY LIMITED MOVEMENT	LESS THAN ONE ROTATION	10,000 CYCLES 180° CYCLES
THERMAL STORAGE	FUSEABLE MATERIAL WITH HEAT EXCHANGER	20 watt-hr-kg 30 kW-Hr/m^3	UNKNOWN
REFRIGERATION (ROOM TEMP)	THERMOELECTRIC MECHANICAL	COP 0.5 COP 2	INDEFINITE - NOT DEMO. IN SPACE
TEMPERATURE CONTROL	TEMPERATURE CONTROL VALVE VARIABLE CONTROL HEAT PIPE	± 1.67°C(± 3°F) THERMAL ± 2.78°C (± 5°F)	2-1/2 YEARS INDEFINITE

TABLE 51
PROJECTED STATE-OF-THE-ART REQUIRED IN 1985-87 FOR A
1990 SPACE PLATFORM LAUNCH

TMS FUNCTION	CANDIDATE METHOD	REQUIRED PERFORMANCE	LIFE
HIGH CAPACITY HEAT TRANSPORT	PUMPED LIQUID ADVANCED "HEAT PIPE"	500,000 TO 5M w-m (20×10^6 TO 200×10^6 w-in)	> 10 YEARS
HEAT REJECTION	RADIATING PANELS	180 w/m^2 (15°C) 50 w/kg	10 YRS WITH ACCEPTABLE DEGRADATION
INTERMEDIATE HEAT TRANSPORT	PUMPED LIQUID HEAT PIPE	50,000-500,000 w-m (2×10^6 to 20×10^6 w-in)	> 10 YEARS
CONNECTABLE THERMAL INTERFACES	QUICK DISCONNECTS CONTACT HEAT EXCHANGERS	. NO SPILLAGE VOLUME ΔP=7kPa @ 0.4 kg/s . hc = 2800 w/m^2-°K	500 CYCLES
ROTATING THERMAL JOINTS	FLEXIBLE HOSE THERMAL SLIP RINGS	4 MILLION 180° CYCLES 2 MILLION REVOLUTIONS	4 MILLION 180° CYCLES 2 MILLION ROTATIONS
HEAT EXCHANGERS	. FLUID-TO-FLUID . FLUID-TO-HEAT PIPE . HEAT PIPE-TO-HEAT PIPE	OVERALL VALUES OF 474 w/m^2	10 YEARS
THERMAL STORAGE	FUSIBLE MATERIAL WITH HEAT EXCHANGER	50 watt-hr/kg 60 kW-hr/m^2	> 10 YEARS
REFRIGERATION (ROOM TEMP)	MECHANICAL REFRIGERATION THERMOELECTRIC	COP 2.5 TO 3.0 COP 1.0	10 YEARS

TABLE 52 TECHNOLOGY DEVELOPMENT REQUIRED FOR HEAT TRANSPORT SYSTEMS & INTERFACES

TECHNOLOGY ITEM	PAYOFF
• INCREASE PROVEN PUMP LIFE BY A FACTOR OF 4 (FROM 2-1/2 YEARS TO 10 YEARS)	• REDUCE ORBITAL MAINTENANCE BY A FACTOR OF 4 & SYSTEM WEIGHT (LESS REDUNDANCY)
• INCREASE PUMP CAPACITY BY ORDER OF MAGNITUDE	• REDUCE PUMP ASSEMBLY COMPLEXITY BY AN ORDER OF MAGNITUDE
• CONTACT HEAT EXCHANGERS FOR INTEGRATION INTO DOCKING PORT (FLUID-TO-FLUID, FLUID-TO-HEAT PIPE & HEAT PIPE-TO-HEAT PIPE)	• IMPROVE SYSTEM RELIABILITY & POTENTIAL FOR FLUID LEAKAGE ON DOCKING BY ELIMINATION OF FLUID DISCONNECTS; PERMIT DOCKING OF MODULES WITH ALL HEAT PIPE SYSTEMS INTO CENTRALIZED FLUID LOOPS
• 360° ROTATION, NO LEAK, LONG LIFE FLUID AND HEAT PIPE SWIVELS	• PROVIDES ORIENTATION FREEDOM OF DOCKED MODULES & EXPERIMENTS ON MODULES WHILE UTILIZING CENTRALIZED THERMAL MANAGEMENT SYSTEM
• ZERO GRAVITY FLUID MANAGEMENT UNDER TWO PHASED FLOW CONDITIONS (CONDENSING AND EVAPORATION) AND HEAT TRANSFER IN HEAT HX	• PERMITS DESIGNING & BUIDLING CONDENSING AND EVAPORATING HEAT EXCHANGERS FOR TWO PHASED THERMAL BUSS AND FOR VAPOR COMPRESSION REFRIGERATION SYSTEMS-PROVIDES ISOTHERMAL COOLING OR HEATING SOURCES
• ZERO GRAVITY, LONG LIFE COMPRESSORS FOR USE IN VAPOR COMPRESSION SYSTEMS	• PERMITS LOCALIZED ISOTHERMAL COOLING OR HEATING-ALL CAN REDUCE RADIATOR PROJECTED AREA ALTHOUGH SOLAR ARRAY INCREASES
• 5000 TO 10,000 WATT-HRS INLINE THERMAL STORAGE CANISTER	• REQUIRED FOR HIGH ENERGY PUSLING EXPERIMENTS SUCH AS PARTICLE BEAM INJECTION; WILL REDUCE SIZE AND WEIGHT OF THERMAL MANAGEMENT SYSTEM BY APPROXIMATELY 10%
• INCREASE MAXIMUM HEAT TRANSPORT CAPABILITY OF HEAT PIPES BY 2 TO 3 ORDERS OF MAGNITUDE FROM 2500 w-m TO 500,000/5,000,000 w-m	• PERMITS AN ALL HEAT PIPE SYSTEM WHICH REDUCES HEAT TRANSPORT SYSTEM COMPLEXITY & IMPROVES SYSTEM RELIABILITY
• DEVELOP A ROTATING THERMAL SLIP RING	• IMPROVES SYSTEM RELIABILITY FOR A THERMAL SYSTEM REQUIRING ROTATING JOINTS

TABLE 53 TECHNOLOGY DEVELOPMENT REQUIRED FOR HEAT REJECTION SYSTEMS

TECHNOLOGY ITEM	PAYOFF
• EFFICIENT LIGHTWEIGHT FLUID-TO-HEAT PIPE PANEL HEAT EXCHANGER	• REDUCTION IN RADIATOR SYSTEM WEIGHT BY APPROXIMATELY 10%
• RADIATOR PANEL COATING WITH 10 YEAR END-OF-LIFE THERMAL PROPERTIES OF $\alpha/\varepsilon \leq 0.2$ AND AN ε OF > 0.9	• REDUCTION IN RADIATOR AREA & WEIGHT OR REDUCTION IN MAINTENANCE REQUIRED BY AS MUCH AS A FACTOR OF 3
• RADIATOR DEPLOYMENT TECHNOLOGY EFFICIENT IN WEIGHT AND STOWED VOLUME	• REDUCTION IN STOWED VOLUME AND WEIGHT
• FLUID-TO-HEAT PIPE CONTACT HEAT EXCHANGERS FOR USE ON SPACE CONSTRUCTABLE RADIATORS • INCREASE HEAT PIPE HEAT TRANSPORT CAPABILITY BY ONE TO TWO ORDERS OF MAGNITUDE	. 20% SAVINGS IN WEIGHT . IMPROVED EASE OF MAINTENANCE . IMPROVED GROWTH POTENTIAL . RECONFIGURATION CAPABILITY . MODULARITY . REDUCED PAYLOAD CONTAMINATION THREAT . 15% REDUCTION IN COST

fluid. Heat rejection system weight reductions of about 10% can also be achieved with this technology on currently known concepts. Advanced concepts would show higher payoffs. An improved radiator coating is needed which has the ability to withstand long duration exposure in the space environment without degradation. This coating should also increase the emissivity from the current value of 0.76 to 0.90. The coating cost should also be reduced. Technology for a radiator deployment system which is low weight and efficiently stores the retracted radiator system is required. Technology is needed to support the space constructable radiator to achieve the many potential benefits this concept offers. The primary technologies needed are the contact heat exchanger technology and the heat pipe technology. Orbital assembly technology will also be needed.

A preliminary schedule of technology development to meet the 1990 250 kW Space Platform launch is shown in Figure 52. It shows milestones in achieving the desired results for thermal transport, contact heat exchangers, thermal joints, thermal storage, and radiator development.

FIGURE 52
THERMAL MANAGEMENT TECHNOLOGY DEVELOPMENT TO
SUPPORT A 1990 250 KW SPACE PLATFORM

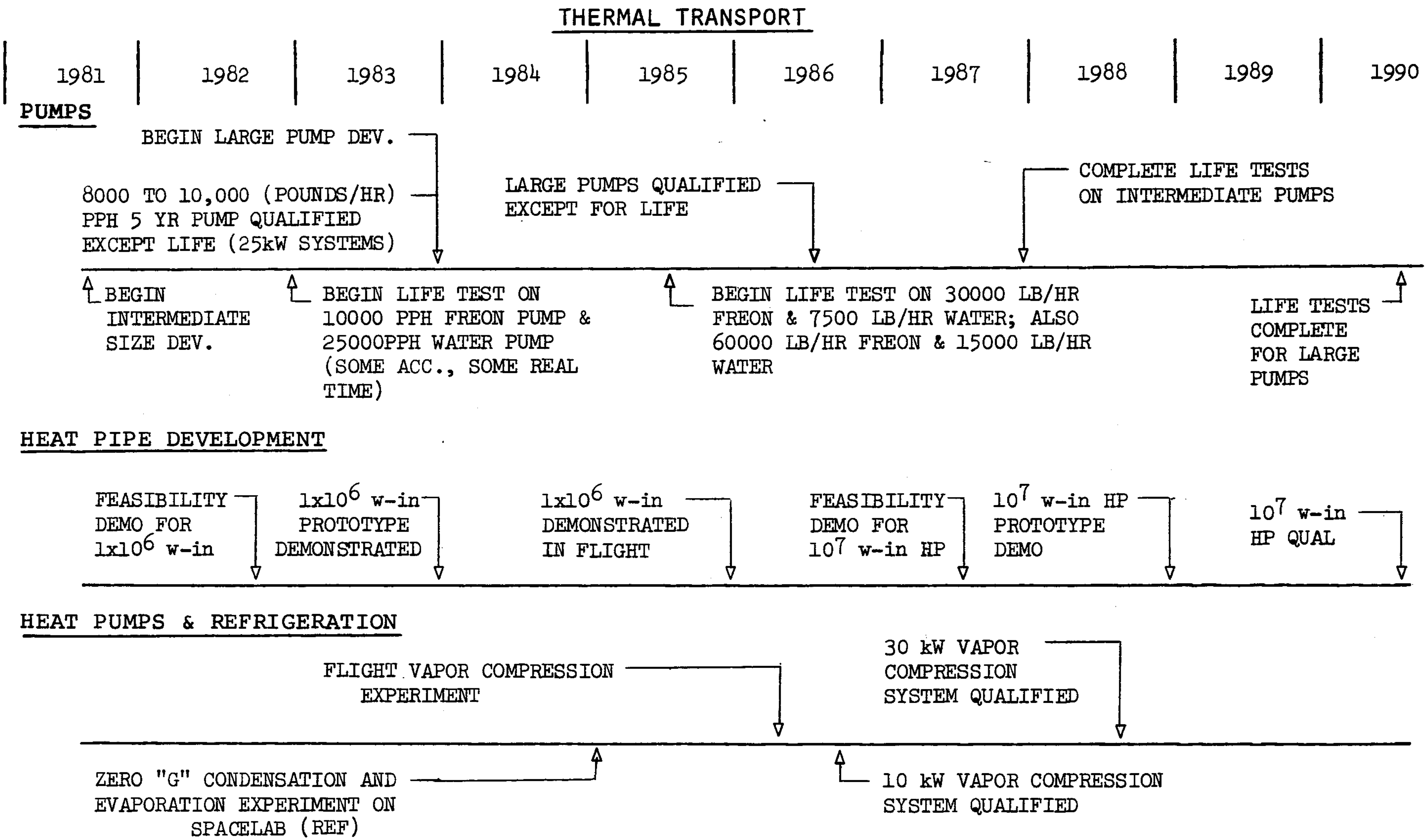


FIGURE 52 (CONT'D)

THERMAL TRANSPORT (CONT'D)

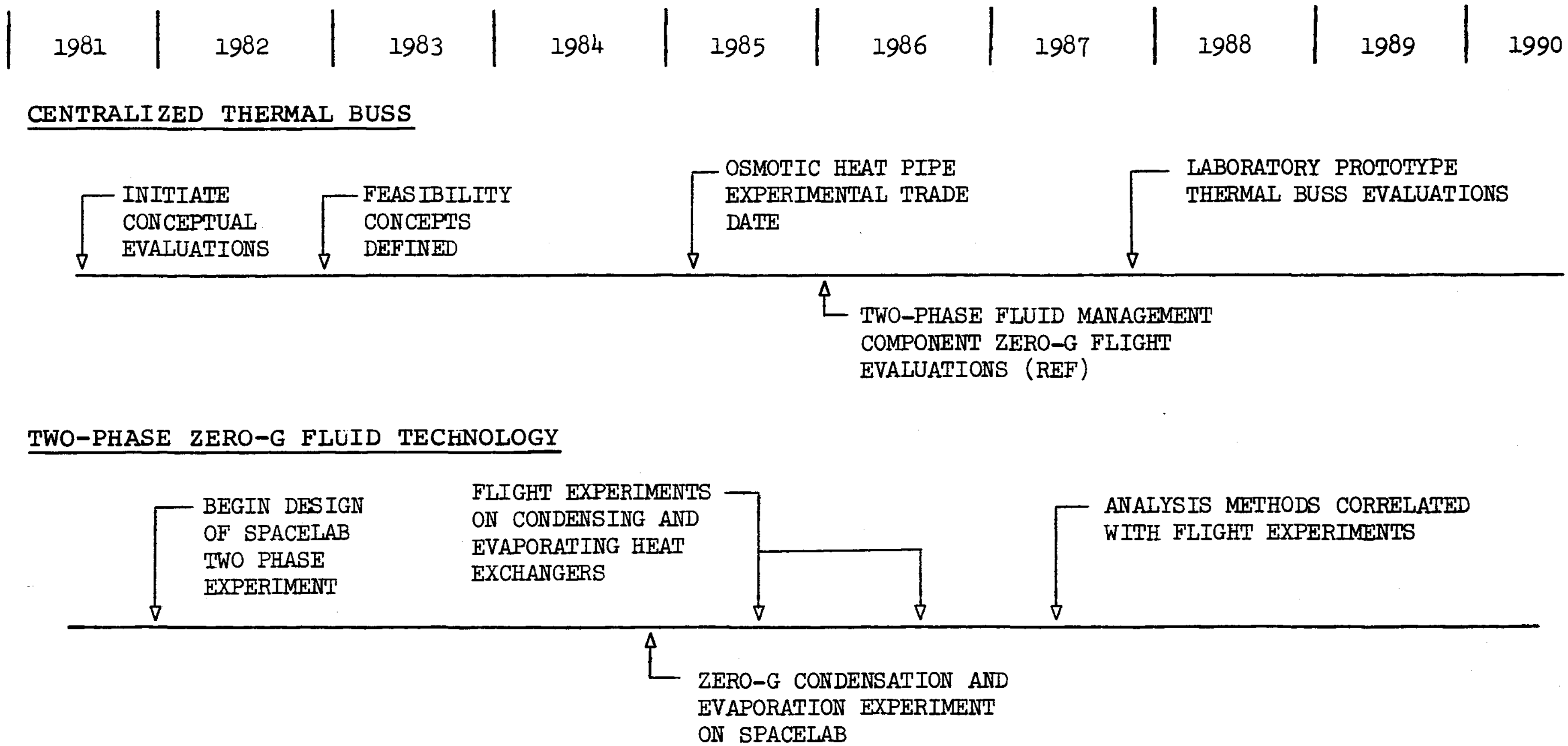


FIGURE 52 (CONTINUED)

CONTACT HEAT EXCHANGERS

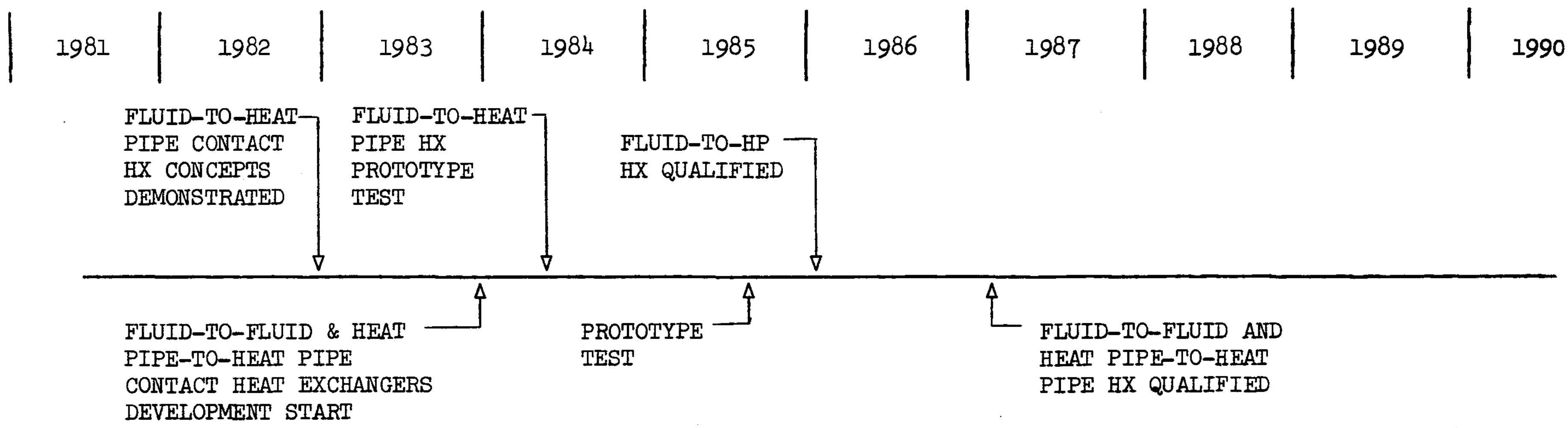


FIGURE 52 (CONTINUED)

THERMAL INTERFACES

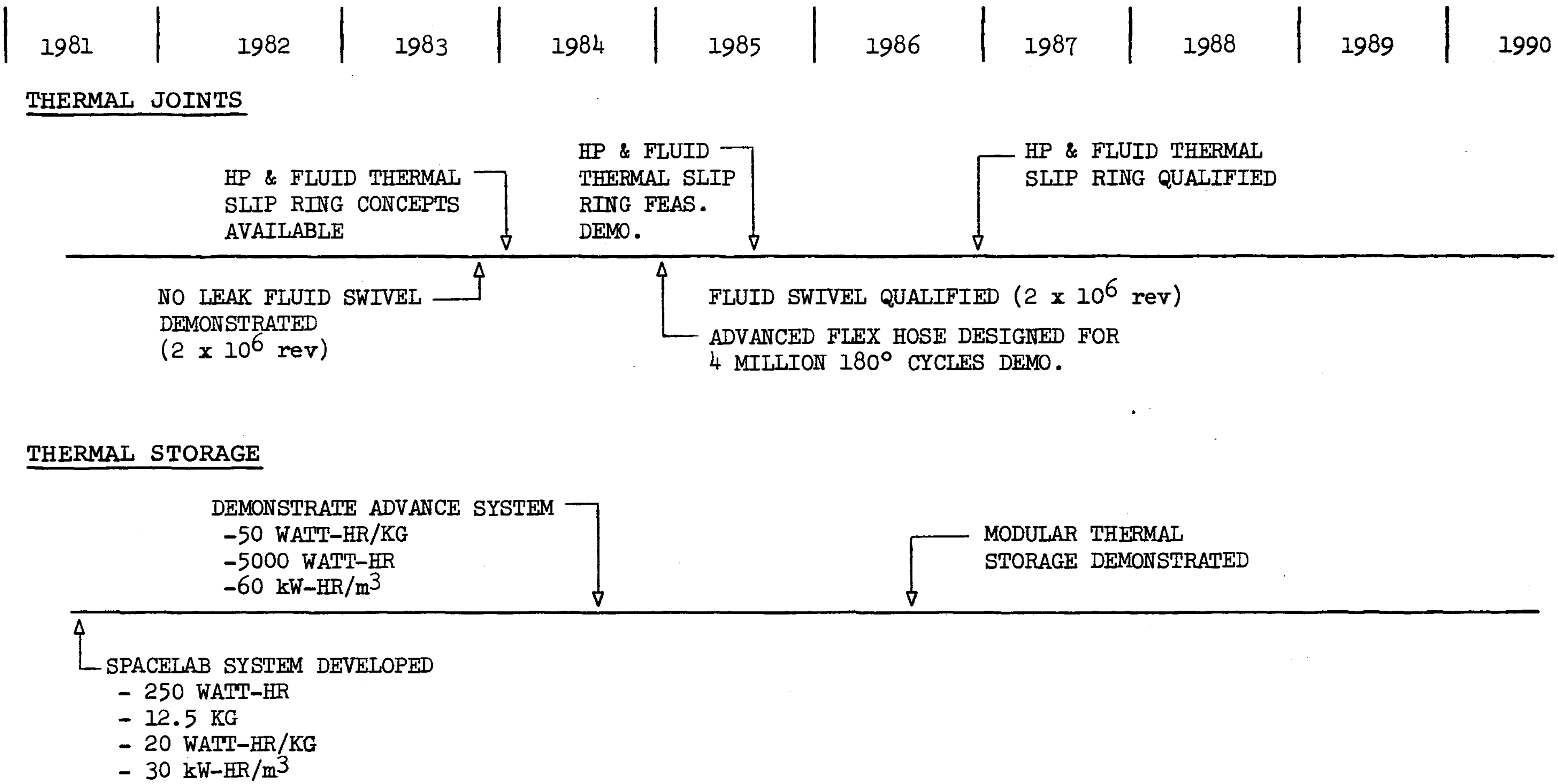
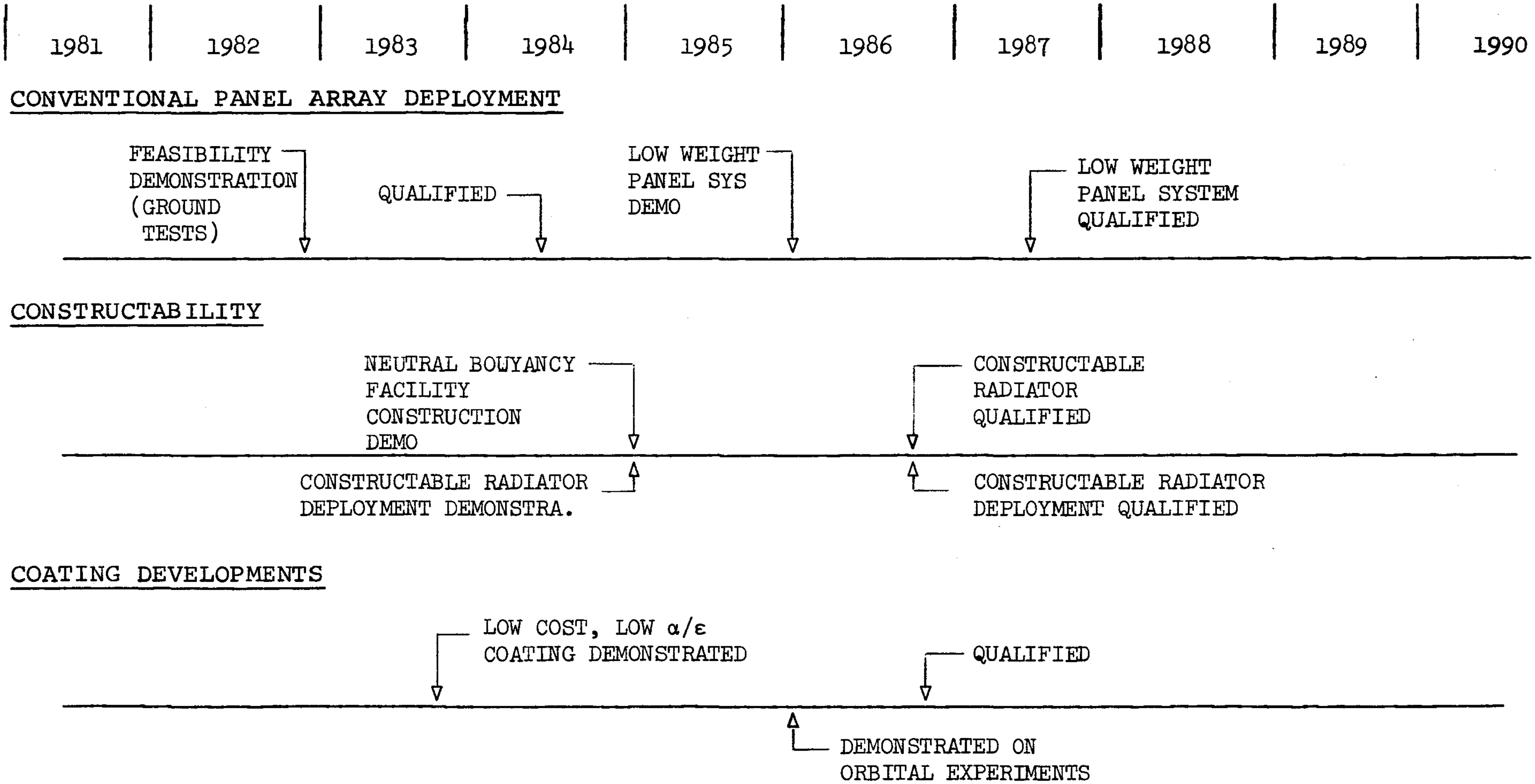


FIGURE 52 (CONCLUDED)

RADIATOR DEVELOPMENT



6.0 CONCLUSIONS

Based on the study results, the following conclusions have been reached relative to the overall thermal management of future large space platforms.

Heat Rejection

1) System weight will not be a deciding factor for the Heat Rejection System - The optimum weight for the three systems considered are all within a 10% range. This is considered within the ability to predict the system weights.

2) Use of heat pipe radiator panels permits the use of a single subsystem approach - Heat pipe radiator panels have the advantage of making the single subsystem approach weight competitive for large, long life systems. This greatly simplifies the heat rejection system, reducing the number of components by an order of magnitude. Multiple subsystem approaches are required for pumped fluid systems.

3) Constructable Radiators are weight competitive - Future systems can utilize the advantages in maintenance and flexibility that the space constructable radiator system offers while remaining weight competitive, especially for systems larger than 160 kW.

4) The multiple subsystem approach has reliability advantages - The multiple subsystem approach with oversizing is inherently more reliable than a single subsystem approach. Very high reliabilities (0.99 for 10 years or greater) are much easier to achieve with this approach.

5) The costs of the pumped fluid and heat pipe heat rejection approaches are within 10% at $23 to $25 Million.

6) Baseline Selection - The integral manifold hybrid heat pipe concept is selected as the baseline heat rejection system, primarily because it permits the use of the simpler single subsystem approach.

7) High Technology Alternate Selection - The space constructable radiator concept is selected as a high technology alternate. It offers significant advantages in modularity, growth, assembly and maintenance while remaining weight competitive. This concept is dependent on the development of a high performance heat pipe.

Heat Transport System Studies

The following are concluded for the Heat Transport System:

1) Baseline Selection - The pumped liquid loop (single phase) is selected for the heat transport system baseline water is selected for the working fluid where manned cabins are involved.

2) Multiple, discrete temperature level heat transport systems significantly reduce weight and radiator area.

3) Two Phase Thermal Buss - Two phase thermal buss approaches which offer the advantages of isothermal heat transfer require multiple descrete temperature loop to be weight competitive. Finding a safe two-phase fluid for operation in the cabin environment appears to be a problem.

4) Osmotic Heat Pipe - The osmotic heat pipe approach, which is still in the laboratory stage, requires more development before meaningful weight and cost projections can be made.

Heat Acquisition and Interfaces

The acquisition temperature for the heat loads for the 250 kW space platform is approximately 16°C (60°F) for 75% of the 250 kW heat load and 4°C (40°F) for 25% of the user heat load and between 16°C and 27°C for the power module. A thermal heat load of 25 kW at each docking port of the berthing module of the 250 kW space platform will satisfy all but special payloads such as space processing.

The best approach for the interface between the centralized heat transport loop and the payloads is a contact heat exchanger at the docking interface. This approach permits automated thermal system mating on docking with no breaking of fluid connections. It also allows more flexibility for the thermal control system design on the payload side. The contact heat exchanger interface approach has the disadvantage of lower overall heat exchanger performance due to the contact conductance.

Fundamental Technology Base

Discrepancies were identified between the current thermal management technology and the technology needed for the 250 kW space platform. The technology advancement needed for thermal management systems are summarized below.

Fluid Pump - The pump capacity for both Freon and water systems must be increased by an order of magnitude. A demonstrated pump life of 10 years is needed.

Two Phase Systems - The heat transport capability of heat pipes and other two phase systems should be increased by 2 to 3 orders of magnitude. This capability permits an all heat pipe thermal control system providing systems with reduced complexity and improved reliability.

Technology needed to support spaceborn vapor compression systems is needed. This includes a better understanding of the evaporating and condensing flow in zero gravity conditions and technology for a zero gravity long life vapor compressor.

Interfaces - Technology for a thermally efficient contact heat exchanger suitable for integration into a docking port is needed. This includes fluid-to-fluid, fluid-to-heat pipe, and heat pipe-to-heat pipe contact heat exchangers.

Rotating thermal slip ring technology is needed to support articulating payload requirements. This technology includes no leak, long life fluid swivels and advanced, thermal slip rings which permit heat pipe interfaces.

Heat Rejection System Technologies

The technology advancements required for the heat rejection system for the 250 kW space platform include an efficient fluid-to-heat pipe heat exchanger for the hybrid-heat pipe radiator, efficient deployment technologies, and technology to support the space constructable radiator. Also, radiator thermal coating advances are needed to provide a coating with low solar absorptance, high emissivity, and long life (low susceptability to degradation). The technologies needed to support the space constructable radiator include an efficient fluid-to-heat pipe contact heat exchanger and a heat pipe with an order of magnitude higher heat transport than available with current technology.

7.0 RECOMMENDATIONS

This study has addressed the subject of spacecraft thermal management for large, multi-hundred kilowatt space platforms which are expected to be launched in the early 1990's. Based on the study, recommendations can be made regarding the thermal management technology advancements that will be needed during the next 10 years. These recommendations are summarized below for the various thermal management functions.

Heat Transport Systems

The recommended thrust for heat transport systems is to develop the technology for high capacity, long life heat transport systems with 1000 times more capacity (watt-inches) by the end of the 1980's. This effort should be a parallel effort of pumped fluid, heat pipe, and other methods such as vapor compression and pump assisted heat pipes. Exploratory development for advanced, large thermal busses for multi-hundred kilowatt space platforms should be initiated in the near future. This should include such advanced concepts as the osmotic heat pipes, pump and compressor assisted heat pipes, and advanced conventional heat pipes. These concepts should be explored to the extent that sufficient performance data are developed to support quantitative assessments of the potential of the concepts.

A concentrated effort must be made in the next 10 years to develop the technology for managing two phase, single component, condensing/evaporating fluid in the environment of zero gravity. This technology is a necessity for the design of vapor compression refrigeration systems which will be needed. Technology is also needed for a long life vapor compressor which will operate in zero gravity. In addition, work is needed to develop liquid management techniques in heat exchangers and system two-phase flow channels. System development is needed to make a number of different sizes available to designers.

Heat Rejection Systems

The technology development for large, long life heat rejection systems should follow a two-pronged path. For the intermediate term (1987 to 1990), hybrid heat pipe panels technology must be developed. This includes efficient, lightweight heat pipe-to-fluid loop approaches, lightweight panel fins, high emissivity coatings, and efficient deployment mechanisms. In addition, more systems studies are needed to determine the best components and systems redundancy approaches for high reliability.

For the longer term, the constructable radiator technology must be developed. This includes the increase in heat pipe heat transport by an order of magnitude and the development of a connectable, contact "plug-in" heat pipe-to-fluid heat exchanger. Lightweight, low cost fin technology is also needed for this.

Interfaces

The projected thermal control system interface requirements for the 1990's include connectable thermal joints, articulating joints, and the ability to handle high peaking power payloads. The recommended technology advancements in these areas include the development of an advanced, 360° rotation, thermal slip ring which could accommodate either a heat pipe or pumped fluid heat transport system with no fluid leakage. An early version of this could be a zero leakage fluid swivel. A large thermal storage canister, on the order of 5000 to 10000 watt-inches, will be needed to support the 1990's thermal control systems.

8.0 REFERENCES

(1) "Conceptual Design Study - Science and Applications Space Platform (SASP) - Mid-Term Briefing", McDonnell Douglas Astronautics Co., dated March, 1980.

(2) "Study of Power Management Technology for Orbital Multi-100 kW Applications - Final Review", General Dynamics Corporation, dated 13 December 1979.

(3) "Study of Thermal Control Systems for Orbiting Power Systems, Final Review dated 15 May 1980.