NASA-CR-197178

NASw-4435

IN-54-CR
26154
51P

The Embodiment Design of the Heat Rejection System for the Portable Life Support System

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April 25, 1994

(NASA-CR-197178) THE EMBODIMENT DESIGN OF THE HEAT REJECTION SYSTEM FOR THE PORTABLE LIFE SUPPORT SYSTEM (USRA) 51 p

N95-12555

Unclas

G3/54 0026154

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The Portable Life Support System (PLSS) provides a suitable environment for the astronaut in the Extravehicular Mobility Unit (EMU), and the heat rejection system controls the thermal conditions in the space suit. The current PLSS sublimates water to the space environment; therefore, the systems loses mass. Since additional supplies of fluid ~~must~~ be available on the Space Shuttle, NASA desires a closed heat rejecting system. This document presents the embodiment design for a radiative plate heat rejection system without mass transfer to the space environment. This project will transform the concept variant into a design complete with material selection, dimensions of the system, layouts of the heat rejection system, suggestions for manufacturing, and financial viability.

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Scope and Limitations

Background and Clarification of Task

The Portable Life Support System (PLSS) is part of the Extravehicular Mobility Unit (EMU) that provides thermal control of the space suit environment and oxygen to the astronaut. A heat rejection system is used to cool the Space Suit Assembly (SSA). Fluid circulates through a Liquid Cooled Ventilation Garment (LCVG) worn by the astronaut to absorb heat from the astronaut's body, and the heated fluid is carried to the PLSS. Depending on the temperature of the fluid, the fluid takes a path to either the heating system or the cooling system.

The current system uses a heat exchanger to cool the circulating fluid and a sublimator that rejects heat to the environment. As the heated water circulates to the heat exchanger and sublimator, heat is transferred through the aluminum wall to the porous wall. Ice forms in the pores of the wall and is sublimated by the heat absorbed in the walls into a gas. Water vapor travels through the pores and into space (Vogt, p. 15). The heat transfer rate of the sublimator is dependent on the metabolic rate of the astronaut. As the work load increases, the amount of heat absorbed into the cooling fluid increases. As a result, the ice is sublimated more rapidly, and heat can be removed from the system at a greater rate (Vogt, p. 17). Since a portion of the fluid is evaporated to the space environment, fluid is depleted from the system. As a result, a new supply of fluid must be provided. Since additional supplies of fluid must be maintained and recharging is necessary, NASA desires a closed system heat rejecting unit that does not require regeneration between Extra-vehicular Activities (EVAs). A closed system decreases the amount of cooling materials that must be transported for each space mission. The task is to embody the heat rejection system for the PLSS. The system must meet the specifications and functional requirements provided to the design team. The specification sheet appears in Appendix A.

Previously, a design team developed two concept variants as solutions for the closed heat rejection system. One heat rejection system is a multiple radiative plate array, and the other system is a phase change heat sink. The first step in this project is to determine which of these two concept variants should be embodied for final design. This project will transform an abstract concept idea into a concrete design complete with material selection, dimensions of the unit, layouts of the heat rejection system, suggestions for manufacturing, and financial viability. In addition, more information is acquired to provide details of the design and to complete the unfinished tasks of the conceptual design of the heat rejection system.

The scope of this project includes the embodiment of the heat rejection plate but not the sub-systems which support the heat rejection system. These subsystems include the pump to circulate the fluid through the SSA and the PLSS, temperature sensors, malfunctioning signals, backup systems, valves, batteries, and the fluid transfer system between the pump and the heat rejection system. Since there is a separate system to provide heat to the astronaut during body temperature drops, the heating system will not be considered in the scope of this paper. Although perspiration introduces moisture into the system, a system to remove the moisture will not be included in this project.

Radiative Plate vs. Heat Sink

The two concept variants given for the PLSS system are shown in Figures 1 and 2. Figure 1 illustrates the radiative plate. The radiative plate consists of a flat plate or array of plates with multiple fins. The fins are attached to increase the surface area in order to provide more radiative heat transfer. The plate has tubes with a cooling fluid running through its interior. As the fluid passes through the plate, the heat that was absorbed in the fluid while traveling across the astronaut's body is convected from the fluid, conducted through the plate, and then radiated out into space. The phase change heat sink is shown in Figure 2. The heat sink contains a storage container packed with a bed of ice spheres. As the fluid coming from the LCVG flows through the packed bed, the heat stored in this fluid is transferred to the ice which cools the liquid.

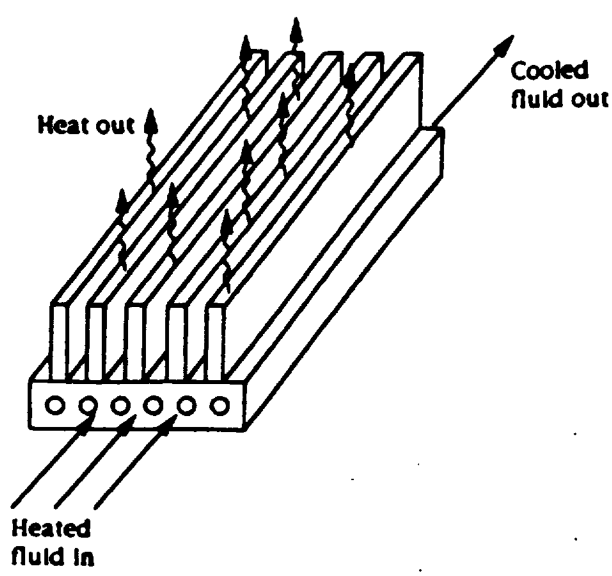


Figure 1. Radiative Plate Conceptual Design (Bourell)

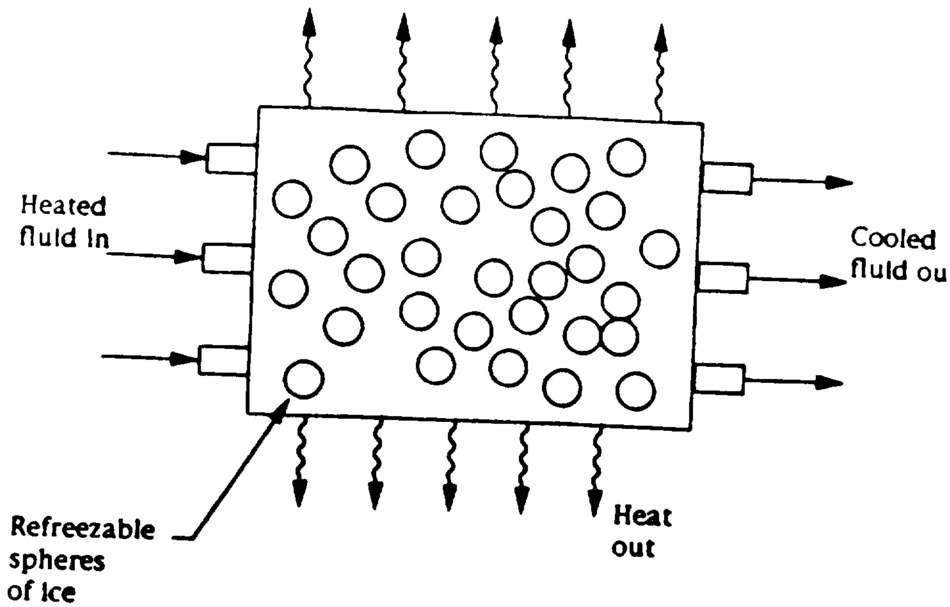


Figure 2. Heat Sink Conceptual Design (Bourell)

Several calculations were performed to determine which concept variant should be chosen (Appendix B). The amount of heat that must be removed is based on the metabolic rate of the astronaut. For the phase change heat sink, the mass of ice required to remove this quantity of heat was calculated. From the calculations, the required amount of ice exceeds the desired mass constraint that National Aeronautics and Space Administration (NASA) imposed by 70%. In order to determine if the radiative plate is feasible, the surface area that would be needed to remove the heat was calculated. It was determined there is a range for the surface area that fits within the dimension constraints. Based on these calculations, the radiative plate is the better choice for embodiment.

Several additional factors affected the decision between the embodiment of the radiative plate and ice spheres. The radiative plate contains fewer components compared to the heat sink. The heat sink would require a more complicated manufacturing process because the heat rejection unit must contain the ice balls, and the ice balls requires many steps to produce. In order to prevent the ice balls from moving around within the radiative system, the manufacturing processes will require tighter manufacturing tolerances. Due to the complexity of the design of the heat sink, the system is more likely to malfunction. The estimated cost for the heat sink would be much greater than the radiative plate because the heat sink requires more components, more detailed manufacturing process, more materials, and a regeneration method to refreeze the ice balls.

A decision matrix was created in order to verify which design should be embodied Appendix C shows the binary matrix which determines the weights for the categories the decision is based. Figure 3 illustrates the decision matrix which concluded that the radiative plate is the better choice; therefore, this concept variant will be taken through the embodiment stage.

	Capacity to Absorb Heat 43%	Time Constant 29%	Mass 14%	Complexity 14%	Total
Heat Sink	10 / 4.3	10 / 2.9	4 / 0.6	4 / 0.6	**8.4**
Radiative Plate	9 / 3.9	9 / 2.6	10 / 1.4	10 / 1.4	**9.3**

Figure 3. Decision Matrix for Concept Variants

Design Issues

There are several design issues which must be considered prior to the embodiment of the radiative plate. The peak metabolic rates and heat entering from the environment by absorption must be considered. When selecting materials, the radiative properties must be examined. The system should be able to maintain the atmosphere of the EMU between 294 K and 300 K (21° C and 27°C). This temperature range provides a comfortable environment for the astronaut (Barry, p. 372). In order to maintain the temperature, the system must not absorb more heat than is required to maintain the fluid between the comfortable temperature ranges. If the system absorbs more heat than needs to be radiated, a heating system will be required to work in conjunction with the cooling system. The circulating fluid should have adequate convective properties to absorb heat from the astronaut. The heat rejection system should be made of a material that can absorb the heat from the fluid and radiate the heat to the environment. The acceptable amount of time or time constant for heat to be released to the environment depends on the period of time before the environment of the EMU reaches a dangerous ambient temperature of 310 K

(Barry, p 372) The system must be a closed system with no mass loss to the environment, and the system should not require regeneration or recharging after the Extravehicular Activity (EVA).

Functional Description

Embodiment Determining Requirements

The following is a list of criteria that will be used to evaluate the embodiment and aid in the determination of final dimensions of the system based on the provided specifications:

- Heat rejection capacity
- Operation for the duration of EVA
- Geometrical constraints
- Impact strength
- Mass

The heat rejection capacity of the radiative plate will be analyzed in the embodiment of the heat rejection system because the primary function of the radiative plate is to reject heat from the fluid flowing through the plate. The radiative plate must reject the average metabolic rate for a six hour EVA and account for metabolic peaks during EVA. The material of the radiative plate must absorb the heat from the cooling fluid and store any heat that cannot be immediately radiated from the system. The material of the radiative plate must have thermal properties that will conduct and radiate the heat that is convected in the fluid. The important properties of the radiative plate material are the density, the specific heat, conductivity, emissivity, and absorptivity. The material must create a blackbody with high emissivity and low absorptivity (Purser, p. 145). If the radiative plate can accommodate these factors, the astronaut will not become overheated, and the heat rejection system will fulfill the functional requirements.

Since the longest EVA missions last for six hours, the radiative plate must reject the average metabolic rate for this time period. In addition, a heat rejection system that can be used in longer EVA missions required during space station construction is of interest to NASA.

Due to safety factors and ergonometric considerations, the PLSS cannot extend more than 0.17 m from the astronauts back. As a result, the radiative plate is subject to the size constraints of the PLSS. The heat rejection system can only use sixty percent or

0.11m^3 of the PLSS volume. The surface area of the radiative plate must be large enough to radiate the heat produced by the astronaut from the system.

Because impacts occur between space objects and micrometeoroids and other objects such as structures of the space shuttle, the radiative plate must be able to withstand impacts of 89 N. If the heat rejection system cannot withstand these impacts, the system may become damaged and may harm the astronaut.

Since this heat rejection system may be used in the future for planetary missions, the mass of the system should be small enough so that the weight of the system in gravity environments is not a burden on the astronaut carrying the PLSS. Increasing thermal capacitance by increasing the mass of the plate allows heat to be stored during peak metabolic periods. However, the mass should be constrained to prevent absorption of more heat than can be radiated for a certain metabolic rate. In addition, the cost of transporting the system to space increases as mass of the system increases.

There are several important factors in the embodiment that were not included in the specifications sheet shown in Appendix C. The time response of the system dictates how quickly heat can be rejected from the system and the time for steady state to be reached during metabolic peaks. During these peaks, the heat rejection system cannot reject all of the heat that has been absorbed. If the system cannot reach equilibrium, the plate will continue to heat up and the cooling fluid will heat up because the plate will no longer absorb heat. As a result, the system will no longer cool the astronaut. On the other hand, to prevent the astronaut from becoming cold, the radiative plate should not absorb too much heat. A power source for the system should be adequate to run the heat rejection system for six hours and operate the subsystems such as the pumps, valves, and thermocouples. The power source must be independent of the space shuttle because a power cord between the PLSS and the shuttle would restrict the distance range between the astronaut and the Space Shuttle. The pump should handle the mass flows required to produce the necessary flow rates and heat rejection rates, but the power requirement of the pump is limited by the capabilities of the power source. The pump needs to provide a sufficient pressure gradient to move the fluid through the system. The corrosive properties of the radiative plate should be considered. The radiative plate should be composed of a material that does not corrode while in contact with the cooling fluid. Corrosion affects the life and the reliability of the system, and corrosion could contaminate the fluid and the system.

Functional Description

The function structure given in Figure 4 gives the sub functions of the heat rejection unit for the PLSS. The heat rejection unit must accommodate each function either by individual devices or through function sharing. Some of these functions will be handled with subsystems that are not a part of the radiative plates and out of the scope of this project. These functions include collecting and storing energy, providing power to the back up system, measuring body temperature, and providing support.

The most important functions of the heat rejection system that must be taken into account are: transfer fluid, control fluid, transfer heat, absorb heat, and protecting the system from impacts and leaks. These subfunctions are imperative to the system because if these functions are not properly performed, the entire system will not be able to keep the astronaut cool. The system must transfer the fluid through the radiative plate and through the LCVG. This function is important because the number of pipes and the pipe diameter affect the mass and heat transfer capabilities of the system. Although it is not within the scope of this project, valves will have to be designed for this purpose. By controlling the mass flow of the fluid through the radiative plate, the amount of heat removed from the system can be controlled. As a result, the problem of removing too much or too little heat from the system is addressed, and the astronaut's body temperature can be maintained at a comfortable temperature range. The radiative plate must transfer enough heat from the cooling fluid to the exposed surface so the proper amount of heat can be radiated from the system to the space environment. The radiative plate also needs to be able to absorb enough heat to cool the astronaut at times of peak metabolic rates. A factor of safety is included in the calculations in order to accommodate uncertainties in the analysis. The thermal properties of the material determines the ability of the plate to absorb heat from the convecting cooling fluid.

Embodiment

Description of Embodied Design

According to Ullman's procedure for embodiment design, after determining the product from the conceptual design different materials and production techniques should be considered. The manufacturing methods will determine the tolerances for the design. Next, the spatial requirements and the geometry of the system must be considered. The radiative plate will consist of two plates connected to each other with tubes in between the plates for the cooling fluid to flow. The plate will require an insulator on the side facing the astronaut's back in order to prevent heat transfer to the astronaut. An external coating

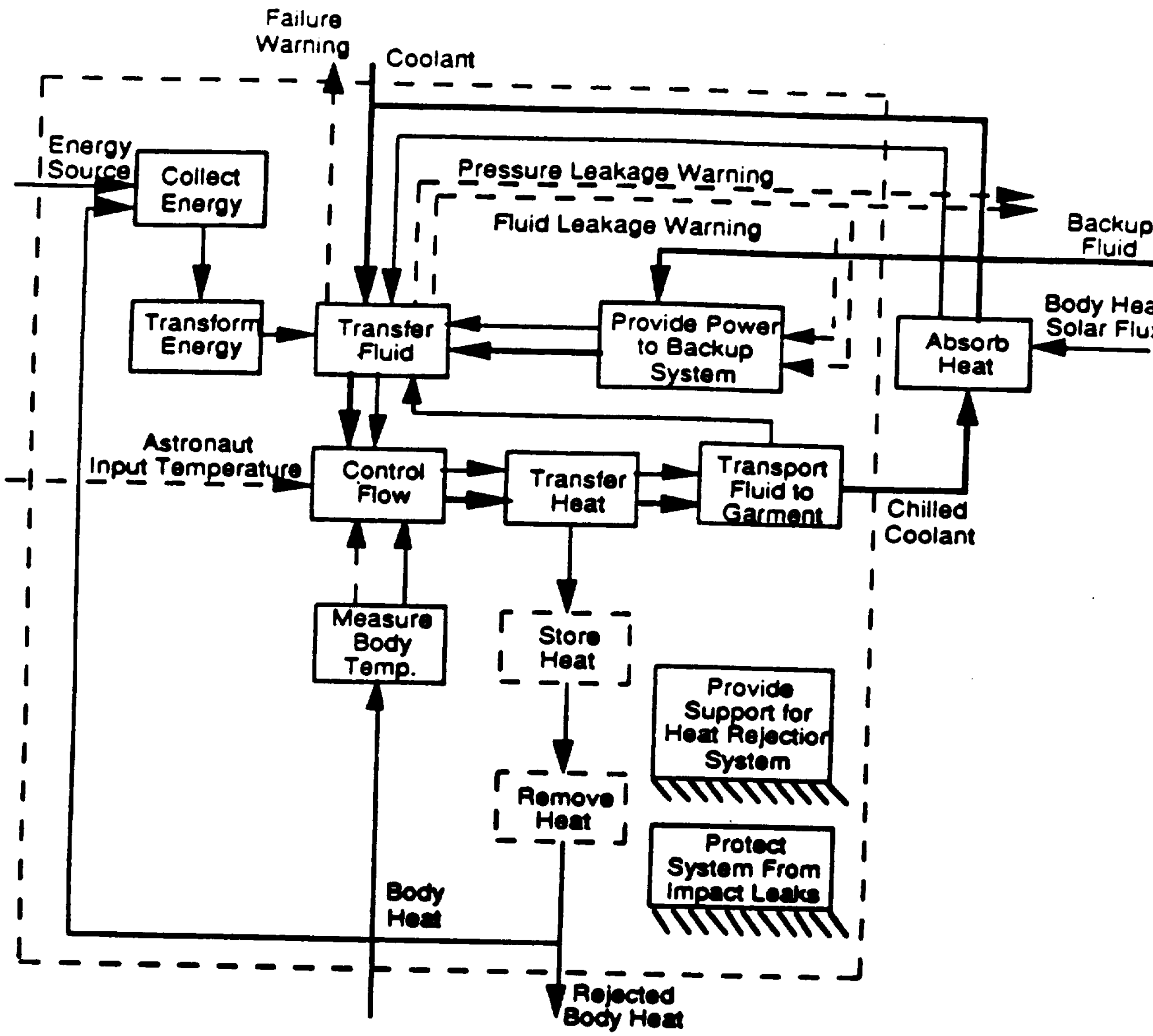


Figure 4. Function Structure (Bourell)

on the surface of the plate decreases the absorptivity of the radiative plate (Vliet). All dimensions must meet NASA's specifications. The radii of the fillets and edges should be included in the dimensioning of the plate. The properties of the plate, tubes, and fluid will be analyzed on a separate basis in order to maximize overall results. The heat transfer, mass flow rate, connection between the two plates, and reaction between the fluid and the plates, are examples of issues that must be considered because they effect the functions of the components.

Once functional requirements are addressed, the material selected, the fluid chosen, and the dimensions are made, the system is ready to be evaluated. The system is evaluated according to projected product performance with respect to customer requirements. For example, the amount of heat the system is capable of removing will be calculated and compared to the amount of heat that must be removed. After evaluation, the system will be refined until a quality design is produced.

An overall layout of the cooling system is shown in Figure 5. The figure illustrates how the radiative plate interacts with some of its subsystems. The mass flow rate in the radiative plate will be controlled by a subsystem containing a pump, valve, thermocouple, and a pressure gauge. The pump will be driven by a motor and powered by a battery. A device will be necessary to disperse the fluid into the pipes. Seals will be required to prevent leakage from the system. A method for diverting flow when the fluid does not need to be cooled will also be considered. For safety purposes, a signal that the system is malfunctioning should be provided to warn the astronaut to return to the Shuttle.

Calculations Showing Technical Feasibility

The dimensions for the design of the radiative plate are based on the amount of heat that has to be rejected by the system. By conservation of energy, the heat transferred from the astronaut's body to the liquid has to be transferred to the plate, and finally radiated to space.

The amount of heat that needs to be rejected from the system is based on the metabolic rate profile for a 6 hour mission as seen in Figure 6. The average metabolic rate is 245 Watts. Table 1 shows the differential fluxes of the astronaut in a hot or a cold environment. From this value, the average differential flux has to be subtracted, since the differential flux represents heat leaving the suit without the assistance of the heat rejection system. In space missions, sunrise occurs every 104 minutes with equal time of darkness and light; thus, the differential flux can be approximated as the average between the values for hot and cold conditions (Bourell, p. 2). In addition, a factor of safety of 1.5 is

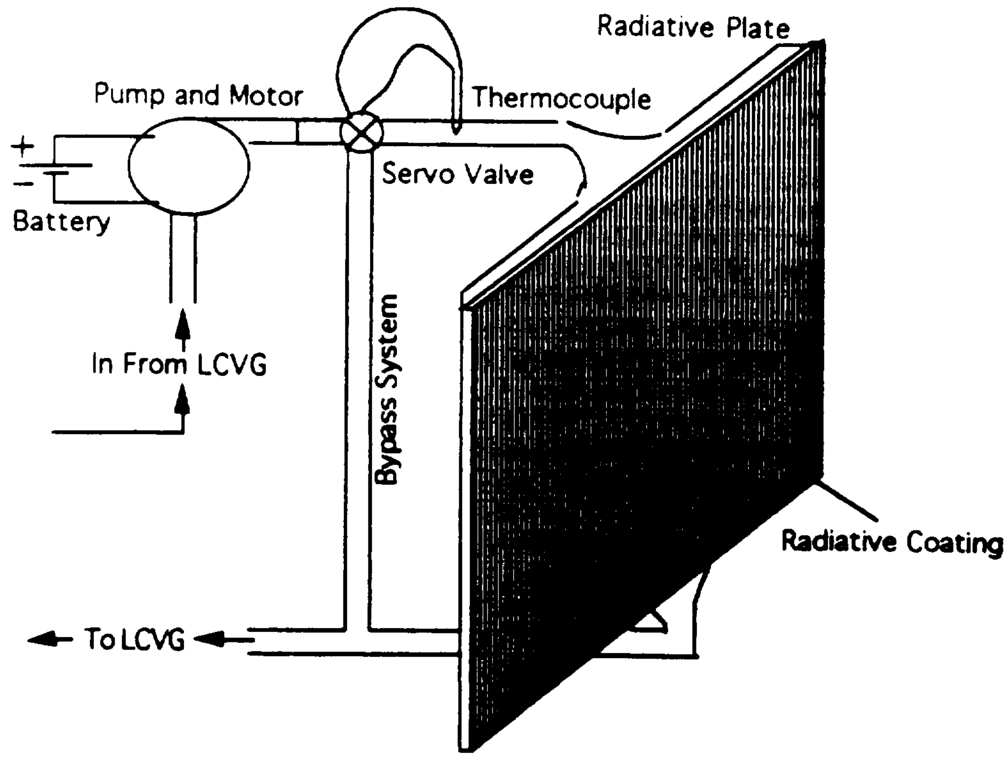


Figure 5. Overall layout for the embodied PLSS heat rejection unit.

Table 1. Differential Flux (Bourell)

Environmental Conditions	Avg. Orbital Fluxes (BTU/hr ft^2) Solar	Avg. Orbital Fluxes (BTU/hr ft^2) IR	Differential Flux (BTU/hr)
Hot	144	85.2	70
Typical Modules	128	52.0	155
Typical Truss	67	29.8	250
Cold	24	16.7	305

included to account for possible deviations from the typical metabolic rate profile The resulting value for the metabolic rate is 282 Watts.

The metabolic rate information is presented as heat (Watts), but for the purposes of the heat transfer calculations we need to transform this heat into a heat flux (W/m^2). To obtain a heat flux, the average metabolic rate is divided by the average human surface area. This surface area is 2.2 m^2 (Woodson, p. 707). The resulting average heat flux is 128 W/m^2, and this value is used for the general heat transfer analysis.

In order to keep the astronaut at a comfortable temperature, the temperature of the fluid leaving the heat exchanger has to be controlled. The temperature of the circulating fluid supplied to the LCVG can be related to the metabolic heat rate by the following equation, generated from experimental data by NASA-Johnson Space Center (Strumpf, p. 2).

$$T(^\circ F) = 74.12 + 0.67\left(\frac{MR}{1000}\right) - 5.52\left(\frac{MR}{1000}\right)^2 \qquad (1)$$

Where MR stands for the metabolic rate in Btu/hour. The results of this equation were converted to temperatures in Kelvin.

The required fluid temperature, derived from the metabolic rate profile (Figure 6) and Equation 1 is shown as a function of EVA time in Figure 7. This figure presents instantaneous temperature requirements, and neglects the system thermal lag. In addition, Figure 8 displays the inverse relationship of metabolic rate and circulating fluid temperature. As the astronaut's metabolic rate increases, the temperature of the cooling fluid must decrease to maintain comfortable temperature levels. For the metabolic rate of 282 Watts used in the calculations, the required fluid supply temperature is 294 K.

The next step in the process is to calculate the change in the temperature of the fluid as it flows through the LCVG. The analysis for this calculation is based on the assumption that the heat flux is constant throughout the path followed by the fluid in the LCVG. The expression used in the calculations is developed from an energy balance on a control volume for internal flow in a pipe (Incropera, p. 482).

$$\Delta T = \frac{q'' PL}{\dot{m} C_p} \qquad (2)$$

In this equation: q'' = constant heat flux (W/m^2)

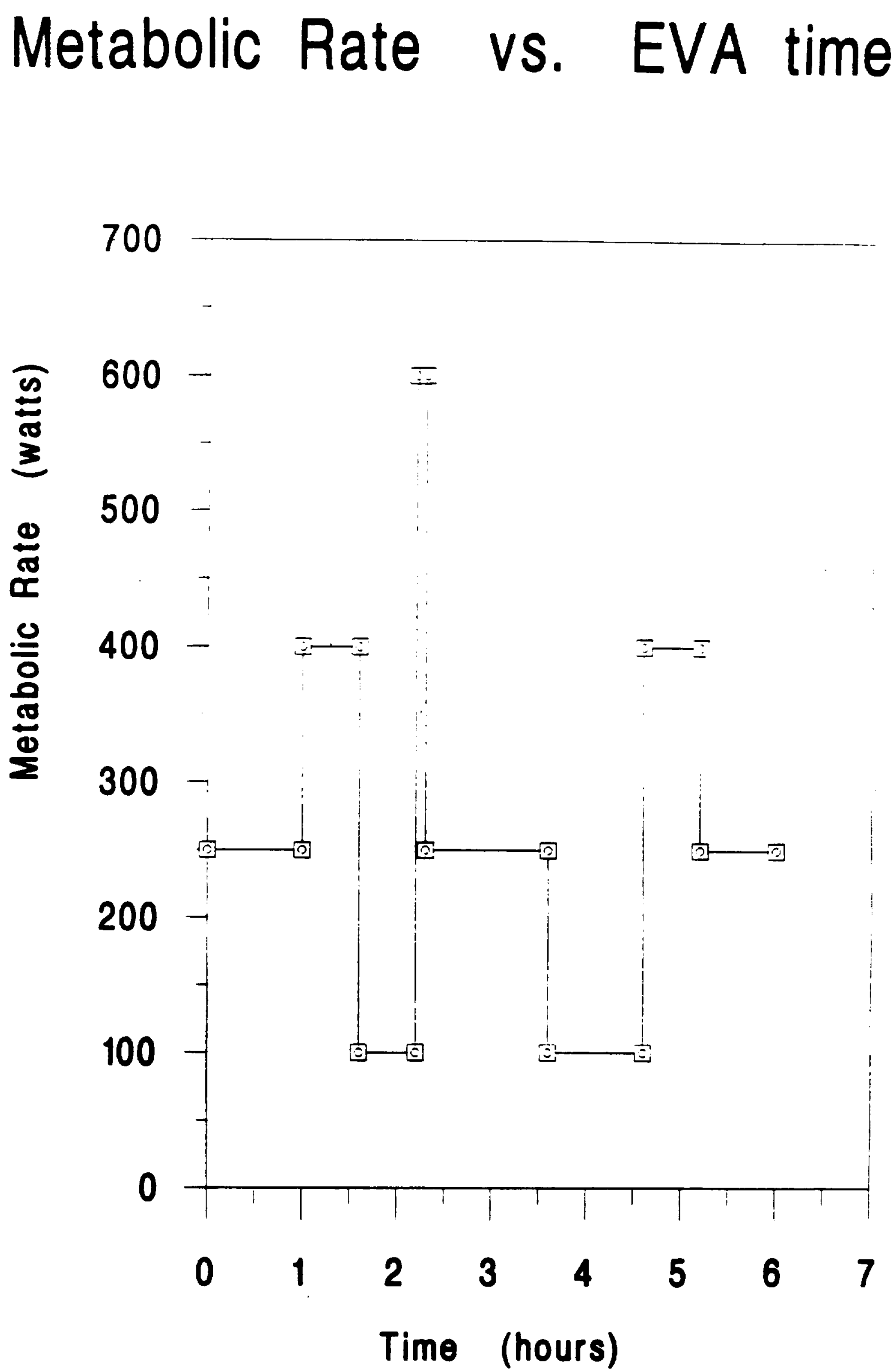


Figure 6. Metabolic Rate Profile (Bourell)

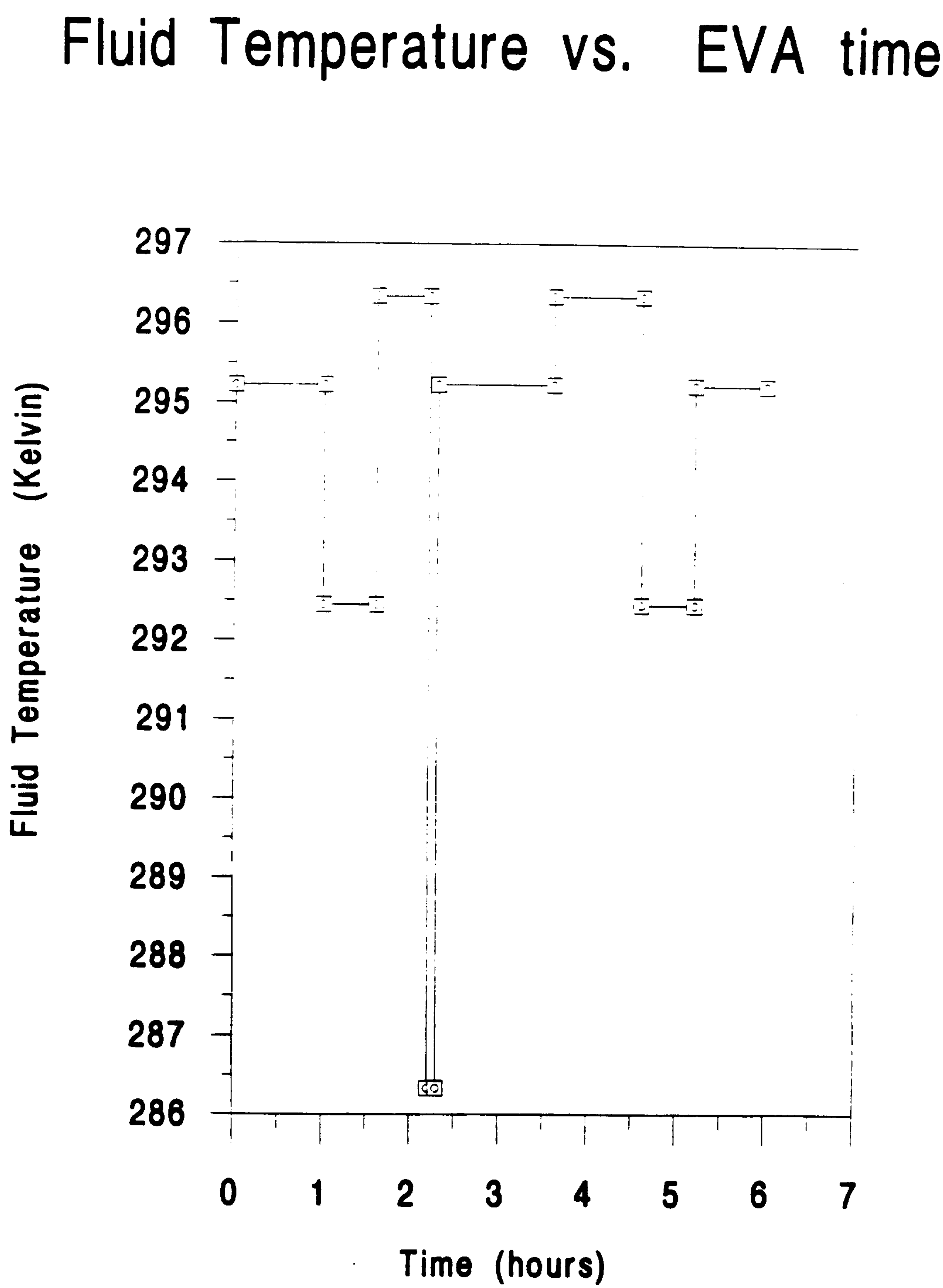
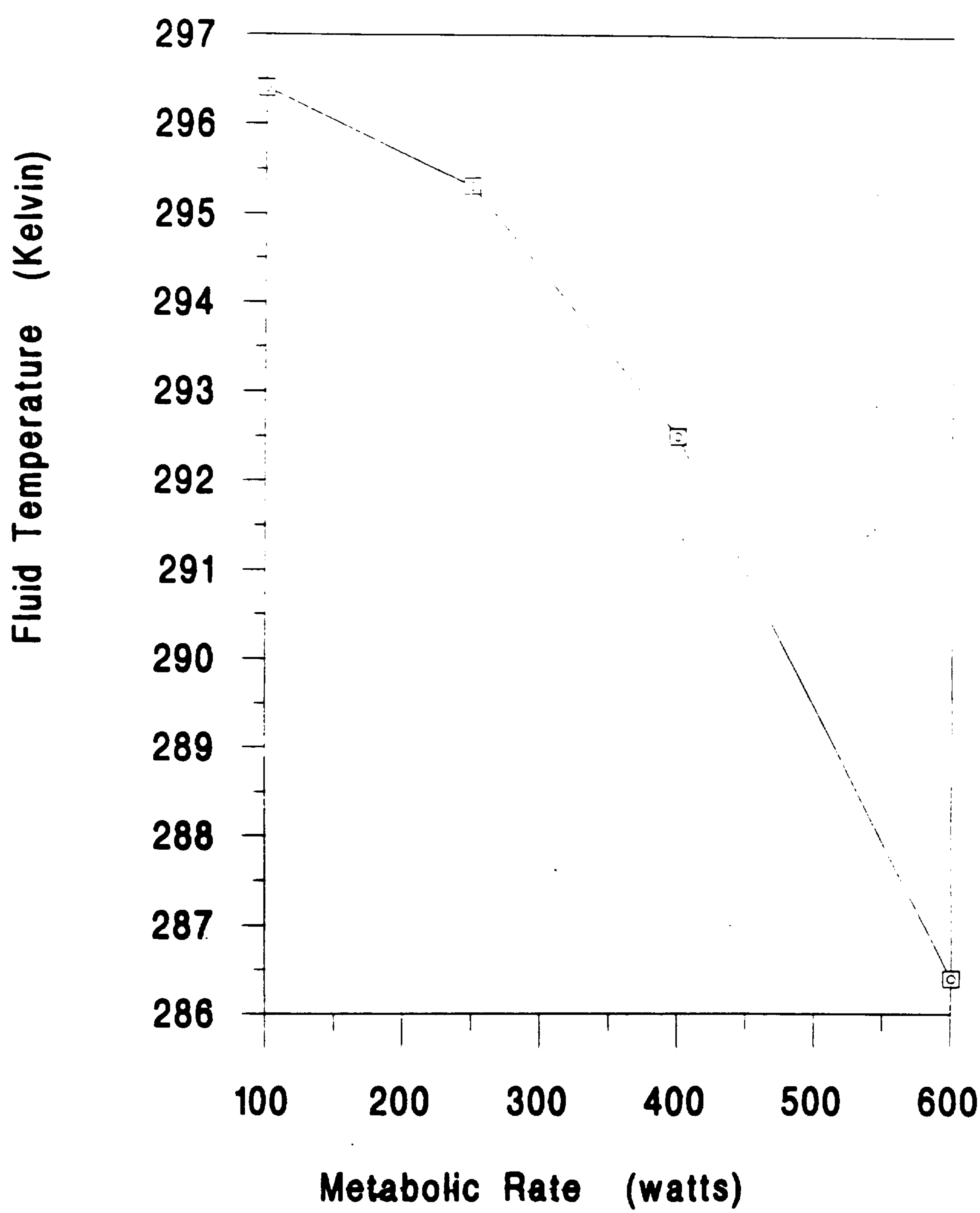


Figure 7. Temperature vs. EVA Time

Fluid Temperature vs. Metabolic Rate



P = perimeter of each LCVG tube (m)
L = length of each LCVG tube (m)
m = mass flow rate (kg/sec)
Cp = specific heat of the fluid (J/kgK)

The change in fluid temperature through the LCVG is very important. In order to keep the astronaut at a comfortable temperature during steady state conditions, the temperature drop of the fluid through the heat rejection unit has to equal the increase in the fluid temperature in the LCVG.

Using the temperature change in the LCVG, the amount of heat that must be transferred to the radiative plate can be calculated. The convection correlation used in the analysis corresponds to internal laminar flow through a circular tube (Incropera, p. 508). Laminar flow can be assumed if the Reynolds number is under 2300, which represents the transition point to turbulent flow. Calculations made using the highest possible flow rate (150 kg/hour) yield a Reynolds number of 400. Since this number is well below the transition Reynolds number, laminar flow can be assumed. These calculations are presented in Appendix D.

The amount of heat transferred from the fluid to the plate is calculated using the general convection equation.

$$q = hA(\Delta T) \qquad (3)$$

where:

q = heat (watts)
h = convection coefficient (W/m^2K)
A = pipe internal area (m^2)
ΔT = temperature change (K)

The convection coefficient is dependent on the Nusselt number and the properties of the fluid. The Nusselt number used in the calculations is 4.36. This value corresponds to laminar internal flow with constant heat flux. The internal area is based on the number of pipes and the internal diameter of each pipe.

$$h = \frac{Nu \cdot k}{D} \qquad (4)$$

$$Area = \frac{\pi}{4} D^2 \cdot L \cdot N \qquad (5)$$

In these expressions:

k = thermal conductivity of fluid (W/mK)
D = internal diameter of pipe (m)
L = pipe length (m)
N = number of pipes

The amount of heat transferred from the fluid to the plate has to be radiated to space in order to keep the fluid at a comfortable temperature. If the heat radiated from the plate is less than the incoming heat from the fluid, the result will be an increase in the temperature of the plate. The following equation was used to calculate how much heat can be radiated out of the plate.

$$q = \varepsilon \sigma A \left(T_s^4 - T_\infty^4 \right) \qquad (6)$$

In this expression:

ε = emissivity
σ = Stephan-Boltzman constant (5.67 E -08 $W/m^2 K^4$)
A = surface area of plate (m^2)
Ts = surface temperature of plate (K)
T∞ = ambient temperature (K)

When calculating the radiating area, only the side exposed to the environment is considered. The side of the plate facing the interior of the PLSS is insulated; hence, heat is transferred through the exposed side only. The calculation of the effective radiating area includes the external surface of the pipes (Equation 7). The radiating area is irregular, resulting in a larger area than if the plate was flat. The increase in radiating area has a direct relationship to the number of pipes in the heat rejection unit. The surface temperature of the plate is calculated using the log mean temperature which approximates the surface temperature based on the inlet and outlet temperatures. This calculation is expressed by Equation 8. Finally, the ambient temperature is approximated at 125 K by the average temperature of the atmosphere at the altitude of shuttle missions (Barry, p. 10).

$$A = L \cdot \left(W + ND\left(\frac{\pi}{2} - 1\right) \right) \qquad (7)$$

$$T_S = \frac{T_{out} - T_{in} \cdot e^{\frac{-\pi DLh}{mCp}}}{1 - e^{\frac{-\pi DLh}{mCp}}} \qquad (8)$$

For this calculation:

L = length of heat rejection unit (m)
W = width of heat rejection unit (m)
N = number of pipes
D = diameter of pipes (m)
Tout = plate outlet temperature (K)
Tin = plate inlet temperature (K)
h = convection coefficient (W/m^2K)
m = mass flow rate (kg/sec pipe)
Cp = specific heat of fluid (J/kgK)

The final calculation deals with the mass of the radiative plate, this calculation plays an important role since NASA has established a mass constraint for the heat rejection system. The mass calculation is based on the volume of the plate and the density of the material.

$$\text{Mass (kg)} = \text{length} * \text{width} * \text{thickness} * \text{density} \qquad (9)$$

The equations presented above give the analytical basis for the iteration procedure used to determine the design parameters.

Optimization Process

The optimization process for embodiment design is an iterative process used to obtain optimal values for important design parameters. Optimization involves balance of trade-offs to find the design which fulfills all of the specified functional requirements. For example, increasing the dimensions of the radiative plate will result in higher heat rejection capabilities, but also increases the mass of the system. By optimizing the system, a balance between characteristics of the system can be obtained.

Ranges of acceptable values for design parameters must be defined before the iterative process can be completed. The goal of the optimization process is to obtain dimensions that will yield higher values of radiation out of the plate than the heat transferred to the plate by convection from the cooling fluid. If the heat into the radiative plate is higher than the heat out of the plate, the plate will absorb the excessive heat causing the plate to rise in temperature. The value of the heat radiated out of the system should be approximately 50 Watts above the incoming convected heat. This value will accommodate peaks in the metabolic rate as well as atypical metabolic profiles. The second goal in the iterative process is to find a range for the operating mass flow rate of the heat rejection system. By supplying a range of acceptable mass flow rates, the system can accommodate changes in the metabolic rate of the astronaut.

Four design parameters can be varied to optimize the heat rejection unit. The remaining variables are either constants or have assumed values. The four design parameters are: the pipe diameter, the plate thickness, the number of pipes, and the mass flow rate. The pipe diameter, while not important to the convective capabilities of the system, affects the effective radiating area and the mass of the system. Varying the pipe diameter also affects the pressure drop within the system, but these pressure changes are negligible compared the pressure drop experienced by the fluid in the LCVG (Norrell). Pressure calculations are presented in Appendix E. Due to the small thickness of the radiation plate and the high conductivity of the material, the temperature at the inside and outside of the plate can be considered equal. In other words, the conduction through the plate is instantaneous. The plate thickness is used to calculate the internal stresses due to the pressures in the system. For the plate thicknesses examined in Appendix F, internal stresses were found to be well under the maximum allowable shear stresses. The number of pipes in the system is directly proportional to the amount of heat that can be absorbed into the system by convection. The mass flow rate will be defined by a range of acceptable values. Within the defined range of flow rate values, the difference between radiation and convection will be maintained at 50 Watts.

Step 1 The first iteration varies the mass flow rates and the number of pipes in the heat rejection system. The values for the mass flow rate ranged between 25 and 150 kg/hr as established in the specifications. The number of pipes varied between 20 and 40 based on the approximated number of tubes in the LCVG. The results for this iteration can be found in Table 2. The results of the iteration show that the mass flow rate must remain between 60 and 120 kg/hr and the number of pipes must be between 30 and 40. Within these ranges, the desired difference between convection and radiation can be maintained.

Table 2. Iteration Step 1

Iteration procedure (iteration number 1)						
Metabolic Rate		282	W			
Human Surface Area		2.2	m^2			
Heat Flux		128.2	W/m^2			
LCVG diameter		0.004	m			
LCVG length		1.85	m			
LCVG inlet temp.		294	K			
Pipe length		0.813	m			
Nusselt number		4.36				
Pipe diameter		0.006				
		Effective				
Flow Rate	LCVG Tout	Plate Wid.	# Pipes	Inter. Area	q conv.	q rad.
kg/hour	K	m		m^2	W	W
30	297.65974	0.9157433	20	0.306494	484.48	283.04
60	295.79095	0.9157433	20	0.306494	242.24	282.03
90	295.16801	0.9157433	20	0.306494	161.49	281.39
120	294.85655	0.9157433	20	0.306494	121.12	280.98
150	294.66967	0.9157433	20	0.306494	96.895	280.71
30	297.65974	0.9928008	35	0.536364	847.83	307.23
60	295.79095	0.9928008	35	0.536364	423.92	306.93
90	295.16801	0.9928008	35	0.536364	282.61	306.56
120	294.85655	0.9928008	35	0.536364	211.96	306.27
150	294.66967	0.9928008	35	0.536364	169.57	306.05
30	297.65974	1.0698583	50	0.766234	1211.2	331.1
60	295.79095	1.0698583	50	0.766234	605.59	331.02
90	295.16801	1.0698583	50	0.766234	403.73	330.83
120	294.85655	1.0698583	50	0.766234	302.8	330.63
150	294.66967	1.0698583	50	0.766234	242.24	330.46

Step 2 The mass flow rate for the second iteration was kept between 85 and 105 kg/hr while the number of pipes was varied between 30 and 40. The results of the iteration are illustrated in Table 3. At flow rate values under 90 kg/hr, the convection into the system approaches or exceeds the outgoing radiation. When 40 pipes are used, the convection into the system exceeds the radiation. As a result, the optimum flow rate is 100 kg/hr because this flow allows a maximum range of plus or minus 10 kg/hr within which the system can still operate at the specified levels. The optimum number of pipes was determined using the graphs presented in Appendix G. These graphs illustrate the difference between convection and radiation for variable flow rates. The optimum number of pipes is 35 since this number of pipes provides the desired difference between convection and radiation regardless of the mass flow rate.

Additional engineering decisions had to made regarding pipe diameter, the thickness of the plate, external coating, and material choice. Length and width of the radiative plate and their tolerances were also determined. Pipe diameter and plate thickness were chosen simultaneously because both parameters need to be maximized for structural purposes without exceeding the mass constraint. As a result, the plate thickness is 0.0015 m and the pipe diameter is 0.006 m. The external coating of the radiative plate serves two purposes: maximizing the outgoing radiation and minimizing the incoming infrared flux. NASA Design Guide for Pressurized Gas Systems provides a list of commonly used coatings and their heat transfer properties (p. XIII-11). From this list, Silicone Alkyd Enamel + Titanium Oxide was chosen. The emissivity of this coating is 0.95 while the absorptivity is 0.18. The length and width of the radiative plate are constrained by the maximum dimensions given in the specification sheet. In order to maximize the radiative area, the maximum allowable dimensions of 0.813 m x 0.813 m were used.

Appendix H illustrates calculations for thermal expansion, insulation, and impact energy. Thermal expansion calculations were made for the inside diameter of the pipes and for the length and width of the plate. The expansion inside the pipes was found to be less than the manufacturing tolerances. The thermal expansion for the length and width were found to be 0.447 mm, which is higher than the manufacturing tolerance. As a result, considerations regarding the difference in thermal expansion between the plate and the mounting material must be addressed. The impact force of micrometeoroids is given as 89 N. This value is well under the maximum allowed impact energy for aluminum which is 3300 N. From the insulation calculations, it can be concluded that a layer of insulation as thin as 2 cm. will prevent heat losses back into the system.

Table 3. Iteration Step 2

Iteration procedure (iteration number 2)						
Metabolic Rate		282	W			
Human Surface Area		2.2	m^2			
Heat Flux		128.2	W/m^2			
LCVG diameter		0.004	m			
LCVG length		1.85	m			
LCVG inlet temp.		294	K			
Pipe length		0.813	m			
Nusselt number		4.36				
Pipe diameter		0.006				
Flow Rate	LCVG Tout	Effective Plate Wid.	# Pipes	Inter. Area	q conv.	q rad.
kg/hour	K	m		m^2	W	W
85	295.2413	0.967115	30	0.459741	256.49	298.41
90	295.16801	0.967115	30	0.459741	242.24	298.34
95	295.10244	0.967115	30	0.459741	229.49	298.28
100	295.04343	0.967115	30	0.459741	218.01	298.22
105	294.99003	0.967115	30	0.459741	207.63	298.16
85	295.2413	0.9928008	35	0.536364	299.23	306.62
90	295.16801	0.9928008	35	0.536364	282.61	306.56
95	295.10244	0.9928008	35	0.536364	267.74	306.51
100	295.04343	0.9928008	35	0.536364	254.35	306.45
105	294.99003	0.9928008	35	0.536364	242.24	306.4
85	295.2413	1.0184867	40	0.612988	341.98	314.75
90	295.16801	1.0184867	40	0.612988	322.98	314.7
95	295.10244	1.0184867	40	0.612988	305.98	314.65
100	295.04343	1.0184867	40	0.612988	290.69	314.61
105	294.99003	1.0184867	40	0.612988	276.84	314.56

Aluminum, Beryllium, Magnesium, and Titanium were examined to determined the best choice for the plate. A binary matrix in Appendix I was used to establish weights given to different properties of the materials. A decision matrix was then used to determine which metal had the optimal properties as seen in Table 4. A similar procedure was used to determine the best fluid for the system. The binary matrix is presented in Appendix J and the decision matrix is shown in Table 5.

Figure 9 incorporates all of the previously discussed parameters and shows the final dimensions and tolerances for the heat rejection unit. The parts, materials, and respective quantities are presented in Table 6.

How the Embodied Design Satisfies the Specified Functions

Because the specified functions are critical to the operation of the heat rejecting unit, the system must accommodate these functions in the design. The specified functions are: transfer fluid, control fluid flow, transfer heat, absorb heat, and protect the system from impacts and leaks.

The cooling fluid circulates through the pump and valve system after the fluid has absorbed the heat from the astronaut's body. Once the fluid exits the pump and valve system, the fluid is directed through the thirty five pipes in the radiative plate. The pipes transfer the fluid through the radiative plate so that heat may be absorbed into the plate and rejected into space.

Using a system composed of thermocouples, pressure gages, a positive displacement pump, and a servo valve, the mass flow rate through the radiative plate can be controlled. Depending on the temperature of the fluid and the atmosphere inside the EMU, the mass flow rate can be varied to control the amount of heat that can be rejected from the system. A variable mass flow rate accounts for the changing metabolic rates of the astronaut. As the mass flow rate is increased, the heat rejection capacity of the system increases.

After the cooling fluid takes the excess heat from the astronaut's body, the fluid is transported to the heat rejection unit and through the thirty five pipes in the radiative plate. As the fluid flows through these pipes, the heat is absorbed from the fluid into the plate. The conductivity of the aluminum plate allows the heat removed from the fluid to be transported to the outer surface, and the heat is radiated from the plate into the space environment. As a result, heat is transferred from the astronaut's body through the system to the environment by the radiative plate.

Since the radiative plate is made of aluminum, the plate has storage ability and can still absorb heat during metabolic peaks. Energy storage during metabolic peaks arises

	Conductivity 20%	Specific Heat 13%	Density 15%	Corrosiveness 20%	Safety 25%	Mechanical Properties 7%	Total
Aluminum	9 / 1.8	6 / 0.8	8 / 1.2	8 / 1.6	9 / 2.3	9 / 0.6	8.4
Beryllium	9 / 1.8	10 / 1.3	9 / 1.4	9 / 1.8	4 / 1.0	8 / 0.6	7.7
Magnesium	8 / 1.5	10 / 1.3	10 / 1.5	2 / 0.4	2 / 0.5	7 / 0.5	5.6
Titanium	3 / 0.6	9 / 1.2	5 / 0.8	7 / 1.4	5 / 1.3	7 / 0.5	4.6

Table 4. Decision Matrix for the Material

	Reactiveness and Corrosion 30%	Safety 45%	Viscosity 12.5%	Coefficient of Thermal Expansion 12.5%	Total
Water	7 / 2.1	10 / 4.5	4 / 0.5	9 / 1.1	8.2
Alcohol	5 / 1.5	8 / 3.6	5 / 0.6	5 / 0.6	6.4
Glycerol	4 / 1.2	4 / 1.8	8 / 1.0	5 / 0.6	4.6
Saline Solution	3 / 0.9	9 / 4.1	4 / 0.5	7 / 0.9	6.3
Carbon Dioxide	5 / 1.5	5 / 2.3	6 / 0.8	2 / 0.3	4.8
Freon	3 / 0.9	2 / 0.9	9 / 1.1	3 / 0.4	3.3

Table 5. Decision Matrix For Fluids

Item	Material / Specifics	Quantity
Plate	Aluminum 1100	2
Fluid	Demineralized Water	
Additional Parts Needed for Subsytems		
Thermocouple		
Servo Valve	Moog Valve Company	1
Positive Displacement Pump	Mass Flow Rate 25 - 150 kg/hour Pressure = 100 - 250 kPa	1
Battery	Silver - Zinc 42 Watts	1
Seal		
Motor	30 Watts, 115 VAC, 1.4 A	1
Porportional Controller (PPC)		1

(Cole-Parmer Instrument Company, p. 832)
(Schmidt)

Table 6 : Parts List

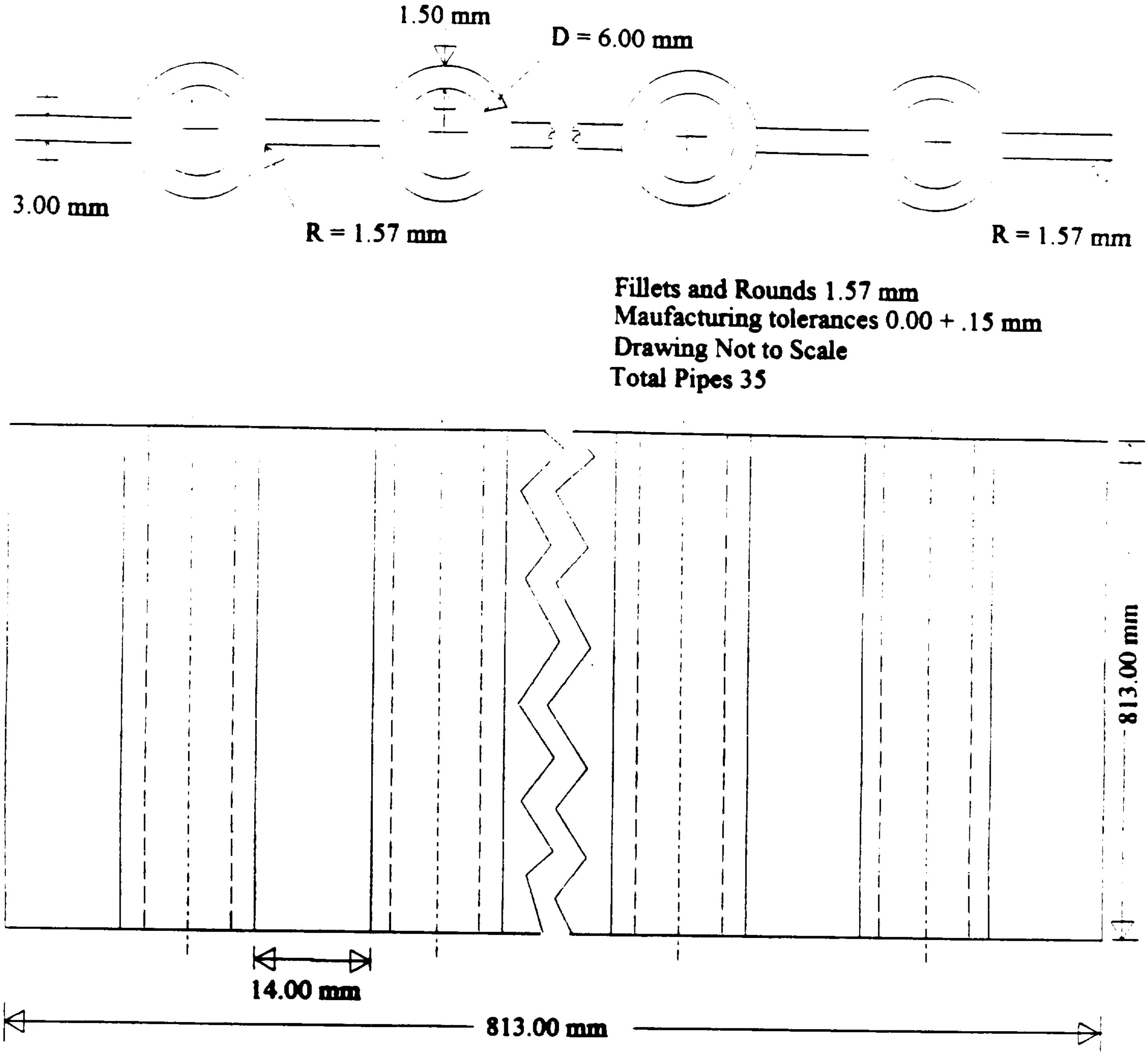


Figure 9. Dimensioned Drawing for the Radiative Plate

from the combination of a temperature increase of the plate and the specific heat of aluminum.

Advantages and Disadvantages of the Radiative Plate

The radiative plate has advantages over the current heat rejection system. Since the radiative plate has a thickness of 9 millimeters, the radiative plate only consumes five percent of the entire volume of the PLSS. Since the volume of the PLSS will be less than the maximum volume given in the specification list, the mass of the heat rejection system will be minimized. If the mass is minimized, the weight of the plate in the presence of gravity may be small enough to enable the system to be used in future planetary missions. The pumping system of the radiative plate can produce variable mass flow rates to handle metabolic rate changes. By varying the mass flow rate, the amount of heat rejected from the system can be controlled. As a result, the atmosphere inside the suit can be maintained between the comfortable levels of 294 K and 300 K. Because the radiative plate is not a flat surface, the surface area is increased by about twenty percent from .813 to .95 meters. Therefore, more heat can be radiated while the size constraints of the radiative plate are met. The radiative plate has only a few components. As a result, production costs are kept low because there are less parts to manufacture. In addition, the plate is subject to less wear because only a few components are subjected to the environment. For the same reason, maintenance is lowered, and reliability of the system is increased. The calculations for the time constant of the system is presented in Appendix K. The time constant of the system was calculated to be 0.854 seconds. This number means that the system will accommodate rapidly to changes in the metabolic rate.

The radiative plate system also presents some disadvantages. Because the plate is on the outside of the PLSS rather than enclosed, the system is subjected to hazards of the space environment such as micrometeoroids. Although the plate has been made to withstand the impact of micrometeoroids up to 89 N, larger impacts could damage the system. Since the radiative plate must be coated to reflect cosmic radiation and prevent excessive solar heating, radiation from the plate is slightly hindered.

Design Recommendations

Manufacturing

The recommended manufacturing process for the radiative plate is stamping. Stamping is a process which uses a sheet of specified material known as the stock and shapes the sheet by cutting, forming, and bending. A die with the predetermined geometry of the desired shape is pressed against the stock so that the die geometry is mirrored on the sheet metal. Stamping is a cold working process since it is done at temperatures below the melting point of the stock material. Stamping is commonly used to make radiators such as those found in cars. If the stock material has a large difference between the tensile strength and the yield strength, forming characteristics are good (Trucks, p.150). The tensile strength and yield strength of aluminum are 83 MPa and 31 MPa, respectively which gives a wide range of 52 MPa for stamping processes. Tolerances for stamping processes are between 10 and 15 percent of the stock thickness. Fillet radii and rounds are also functions of the stock thickness. For stock thickness of 0.0015 m, the minimum bending radius is 0.00078 m and the minimum round radius is 0.00157 m.

For the radiative plate, stamping will be done in three stages: stamping the first side, stamping the second side, and spot welding the two pieces together. The two sides of the plate will be joined by spot welding at one inch intervals between the pipes. Spot welding allows for the joining of the two sheets without adding another material into the system which contributes to the weight and increases potential safety hazards.

Cost Analysis

A cost analysis for the production of the radiative plate was conducted. The analysis does not include any investment costs such as engineering hours or management costs since those investments have already been made at this point in the design stage. The cost for the radiative plate is based on the cost of the material stock, the energy cost of bending and cutting the metal, and the cost of spot welding the two plates together. The resulting cost of the plate is $28.80 per plate. This cost does not include the cost of designing and producing the die to form the sheet metal. The cost of producing a die ranges between $20,000 and $50,000.

Areas for Further Study

The subsystems that are dependent on the heat rejection system should be studied to ensure proper connections, geometries, and interactions of the entire system. After the water leaves the pump, the fluid needs to be separated equally to flow through the 35 pipes. One method to split the flow is a plenum chamber. A back up power system is needed in case the primary system fails. The back up system should allow enough time for the astronaut to safely return to the Shuttle bay before a dangerous temperature is reached. A warning signal should be provided to warn the astronaut that the system has failed. The signal that the system is malfunctioning can be reliant on pressure changes that might indicate a leak in the system and on the temperature of the fluid that indicates if the astronaut is being properly cooled. The manner in which the plate is attached to the PLSS must be considered. By virtue of our design, there are two flat plates extending from the outer tubes. Two runners with bolts can be attached to the PLSS as seen in Figure 10. The spacing of the bolts must be determined and the number of bolts must be determined. Seals will be used on the system and its connection to the PLSS in order to prevent leaks. Since most seals are made of elastomers, creep and thermal expansion will be problems in determining what type of seal can be used. To determine the type of seals, further research will be required. More information on an insulator pad that will prevent heat from radiating back into the EMU should be obtained.

Conclusions and Future Work

Current systems used to regulate the temperature inside the EMU suit are mass consuming and require regeneration. Mass loss constrains the time of EVA's because mass must be resupplied. A closed system which does not require any recharging or resupplying of material is desired. The radiative plate embodied in this paper is a closed system since the cooling fluid is conserved throughout the system. As a result, the radiative plate does not require a regenerated process which increases maintenance time between EVA. In addition, the duration of EVA's can be extended because the system is not constrained by loss of mass. The radiative plate must release the heat energy absorbed from the astronaut's metabolic emissions at a rate suitable to control the temperature inside the EMU suit. By increasing the duration of EVA possible for the astronaut, the system will be applicable in the construction of the space station and in future planetary missions.

Before the final blue prints of the radiative plate and its subsystems can be made, all design teams must meet to coordinate the subsystems with the radiative plate. Allowances for proper interfaces between subsystems such as the pump and the valve must be designed.

The next step in the development of the radiative plate will be communication with the manufacturer regarding building of a prototype. After the manufacturer evaluates the design from a production perspective, the design will likely need adjustments before it can be manufactured. Once the final changes have been made, the prototype can be built and tested.

In the future, the design will have to be adapted for planetary missions. Modifications may include the addition of fins to aid in convective heat transfer in an environment with an atmosphere. In addition, since gravitational forces may be present, the mass of the system will have to be reduced so that the astronaut will not be burdened by the weight of heat rejection unit.

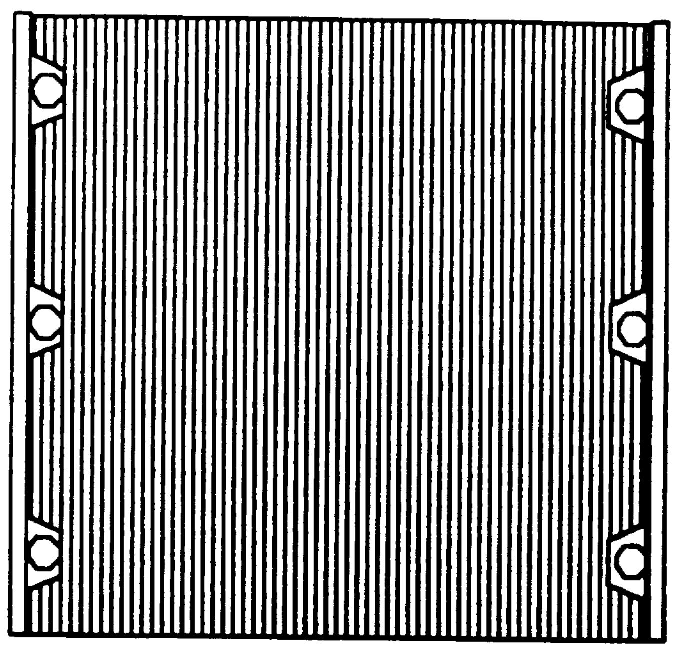


Figure 10. Possible mounting scheme for heat rejection plate. There are two rails on either of the vertical sides that run parallel to the flow pipes. Six mounting bolts secure the plate to the rails which are fastened to the PLSS.

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Appendix A
Specification Sheet

NASA/USRA ME366J	Specification For: Closed Unit Heat Rejection System	Page: 1 of 2

Changes	D/W	Requirements	Rspnsbl	Verify
		1. Function		
	D	Provide a suitable heat rejection.		
	D	Maintain temperature during all levels of metabolic activity.		
	D	Provide monitored use for 6 hrs. duration.		
	D	Reject heat to space environment during both day and night time activities.		
		2. Geometry		
	D	Total surface area of 1 m^2 (10.8 ft^2).		
	D	Portable Life Support System (PLSS) not to exceed 81.3 cm x 81.3 cm x 17.8 cm (32" x 23" x 7") (for connection to Space Suit Assembly (SSA)).		
		3. Forces		
	D	Withstand launch loads up to 3g.		
	D	Withstand direct impact of 89 N (20 lbs)		
	W	Minimize mass: Upper bound of 11.4 kg (25 lbs)		
		4. Energy		
	D	Cooling requirements: 250 W average metabolic rate for 6 hours 1500 W-hr total metabolic heat rejection Peak metabolic rate of 600 W for 6 minutes Peak metabolic rate of 400 W for 36 minutes (See Figure: Metabolic Rate Profile)		
		5. Material		
	D	Working fluid must be non-toxic, nonflammable, and non-corrosive.		
	D	System pressure from 100 to 250 kPa (14.5 to 36.3 psi).		
	D	System mass flow rate from 25 to 150 kg/hr (1.7 to 10.3 slug/hr).		
		6. Signals		
	W	Minimize physical input of astronaut to the heat rejection system (number of steps or expended energy).		

(Bourell)

<table>
<tr><td>NASA/USRA
ME366J</td><td colspan="2">Specification
For: Closed Unit Heat Rejection System</td><td colspan="2">Page: 2 of 2</td></tr>
<tr><td>Changes</td><td>D/W</td><td>Requirements</td><td>Rspnsbl</td><td>Verify</td></tr>
<tr><td></td><td>D</td><td>7. <u>Safety</u>
Fail-safe system to maintain space suit environment if part (or all) of the heat rejection system fails.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>No sharp exposed surfaces, edges, or corners.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>Apply overall safety factor of 1.5 for heat rejection system.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>8. <u>Operation</u>
Orbital fluxes:
75.7 to 453.9 W/m^2 (24 to 144 Btu/hr ft^2) for solar radiation.
52.6 to 268.6 W/m^2 (16.7 to 85.2 Btu/hr ft^2) for infrared radiation.
(See Table 1: Station Suit Heat Leaks)</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>Operate in zero (micro) gravity environment.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>No mass transfer out of the cooling system.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>Operate in zero pressure environment.</td><td></td><td></td></tr>
<tr><td></td><td>D</td><td>9. <u>Production</u>
Number of units:
Development - 1
Flight - Less than 15</td><td></td><td></td></tr>
<tr><td></td><td>W</td><td>Total prototype cost less than 1 million dollars.</td><td></td><td></td></tr>
<tr><td></td><td>W</td><td>Lifetime of unit:
One year on orbit.
52 EVA's at 1 EVA per week before ground maintenance.
15 year total lifetime.</td><td></td><td></td></tr>
</table>

Appendix B

Mass of Packed-Bed of Ice Spheres

Metabolic Rate	282	Watts
Latent Heat	330000	Jouls/kg
Total energy out	6091200	Jouls
Mass	18.45818	kg
Mass constraint	11.2	kg

The minimum mass for the ice heat sink is 64% over the mass constraint established by NASA

Appendix C
Binary Matrix for Concept Variants

Category	Capacity to Absorb Heat	Time Constant	Mass	No. of Components	**Total**
Capacity to Absorb Heat	X	1	1	1	**3**
Time Constant	0	X	1	1	**2**
Mass	0	1	X	0	**1**
No. of Components	0	0	1	X	**1**

Appendix D

Reynolds Number Calculation

The Reynolds number has to be calculated in order to determine if the flow inside the pipe is turbulent or laminar. The transition point between laminar and turbulent flow is 2300. Using the highest mass flow rate will result in the highest Reynolds number. If such number is under 2300, the flow can be assumed to be laminar.

Assumptions:
- mass flow rate (Q) = 150 kg/hour
- number of pipes (N) = 35
- pipe diameter (d) = 0.005 mm
- kinematic viscosity (ν) = 1.007 E -07 (m^2/sec)

$$V(m/s) = \frac{Q}{A \cdot 3600 \cdot 1000 \cdot N}$$

This results in a velocity of 0.10 m/sec.

$$\mathrm{Re} = \frac{Vd}{\nu}$$

The resulting Reynolds number is 400, which indicated that the flow can be considered laminar.

Appendix E

Pressure Head Losses

FLUID CONVECTION ANALYSIS

Water:

Properties at 275 K

Viscosity	0.001652	Ns/m^2
Cp	4.211	kJ/kgK
k (cond)	0.574	W/mK
v (spc. V)	0.001	m^3/kg
Pr	12.22	

Pipe D (m)	Pipe L (m)	Plate Width (m)	Number Pipes n	m dot total (kg/hr)	m dot total (kg/s)	m dot/pipe (kg/s)	Re	f	hd loss (Pa)	hd loss (psi)
0.003	0.813	0.813	90	25	0.00694444	7.6876E-05	19.75005	3.240497	51.9356	0.007531
0.004	0.813	0.813	68	25	0.00694444	0.0001025	19.75005	3.240497	21.91033	0.003177
0.005	0.813	0.813	54	25	0.00694444	0.00012813	19.75005	3.240497	11.21809	0.001627
0.006	0.813	0.813	45	25	0.00694444	0.00015375	19.75005	3.240497	6.49195	0.000941
0.007	0.813	0.813	39	25	0.00694444	0.00017938	19.75005	3.240497	4.088225	0.000593
0.008	0.813	0.813	34	25	0.00694444	0.000205	19.75005	3.240497	2.738791	0.000397
0.009	0.813	0.813	30	25	0.00694444	0.00023063	19.75005	3.240497	1.923541	0.000279
0.01	0.813	0.813	27	25	0.00694444	0.00025625	19.75005	3.240497	1.402261	0.000203

Pipe D (m)	Pipe L (m)	Plate Width (m)	Number Pipes n	m dot total (kg/hr)	m dot total (kg/s)	m dot/pipe (kg/s)	Re	f	hd loss (Pa)	hd loss (psi)
0.003	0.813	0.813	90	150	0.04166667	0.00046125	118.5003	0.540083	311.6136	0.045184
0.004	0.813	0.813	68	150	0.04166667	0.00061501	118.5003	0.540083	131.462	0.019062
0.005	0.813	0.813	54	150	0.04166667	0.00076876	118.5003	0.540083	67.30853	0.00976
0.006	0.813	0.813	45	150	0.04166667	0.00092251	118.5003	0.540083	38.9517	0.005648
0.007	0.813	0.813	39	150	0.04166667	0.00107626	118.5003	0.540083	24.52935	0.003557
0.008	0.813	0.813	34	150	0.04166667	0.00123001	118.5003	0.540083	16.43275	0.002383
0.009	0.813	0.813	30	150	0.04166667	0.00138376	118.5003	0.540083	11.54124	0.001673
0.01	0.813	0.813	27	150	0.04166667	0.00153752	118.5003	0.540083	8.413567	0.00122

Appendix F

Internal Pressure Calculations

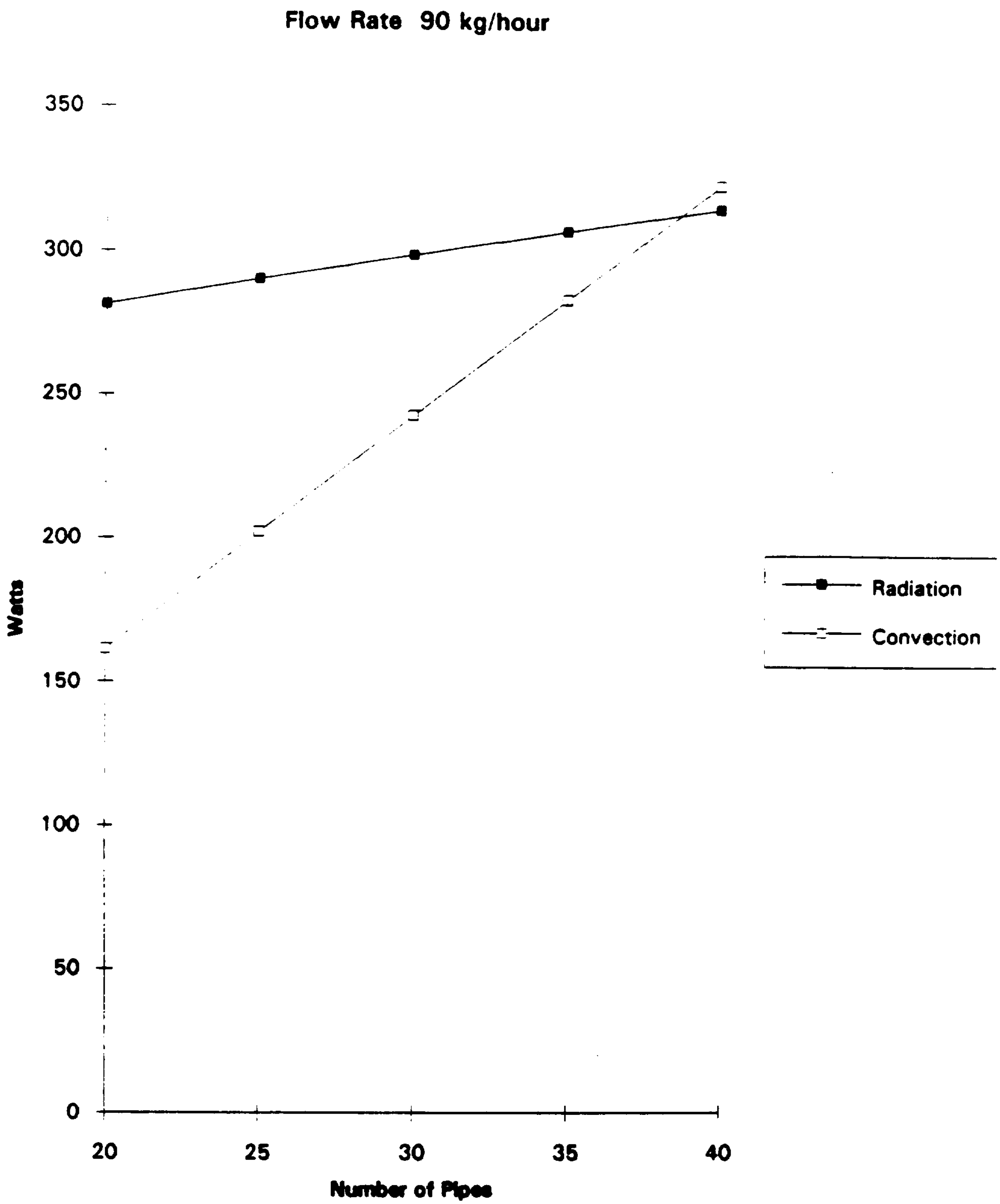
r (int) (m)	thick. (m)	pressure KPa	Max. Shear Stress (KPa)	Material	Yield Strength	Allowable Shear
0.003	0.0015	250	125	Aluminum	20000	6666.667
0.003	0.002	250	93.75	Aluminum	20000	6666.667
0.0015	0.0015	250	62.5	Aluminum	20000	6666.667
0.0015	0.002	250	46.875	Aluminum	20000	6666.667
0.003	0.0015	250	125	Mag. Alloy	80000	26666.67
0.003	0.002	250	93.75	Mag. Alloy	80000	26666.67
0.0015	0.0015	250	62.5	Mag. Alloy	80000	26666.67
0.0015	0.002	250	46.875	Mag. Alloy	80000	26666.67
0.003	0.0015	250	125	Titanium	400000	133333.3
0.003	0.002	250	93.75	Titanium	400000	133333.3
0.0015	0.0015	250	62.5	Titanium	400000	133333.3
0.0015	0.002	250	46.875	Titanium	400000	133333.3

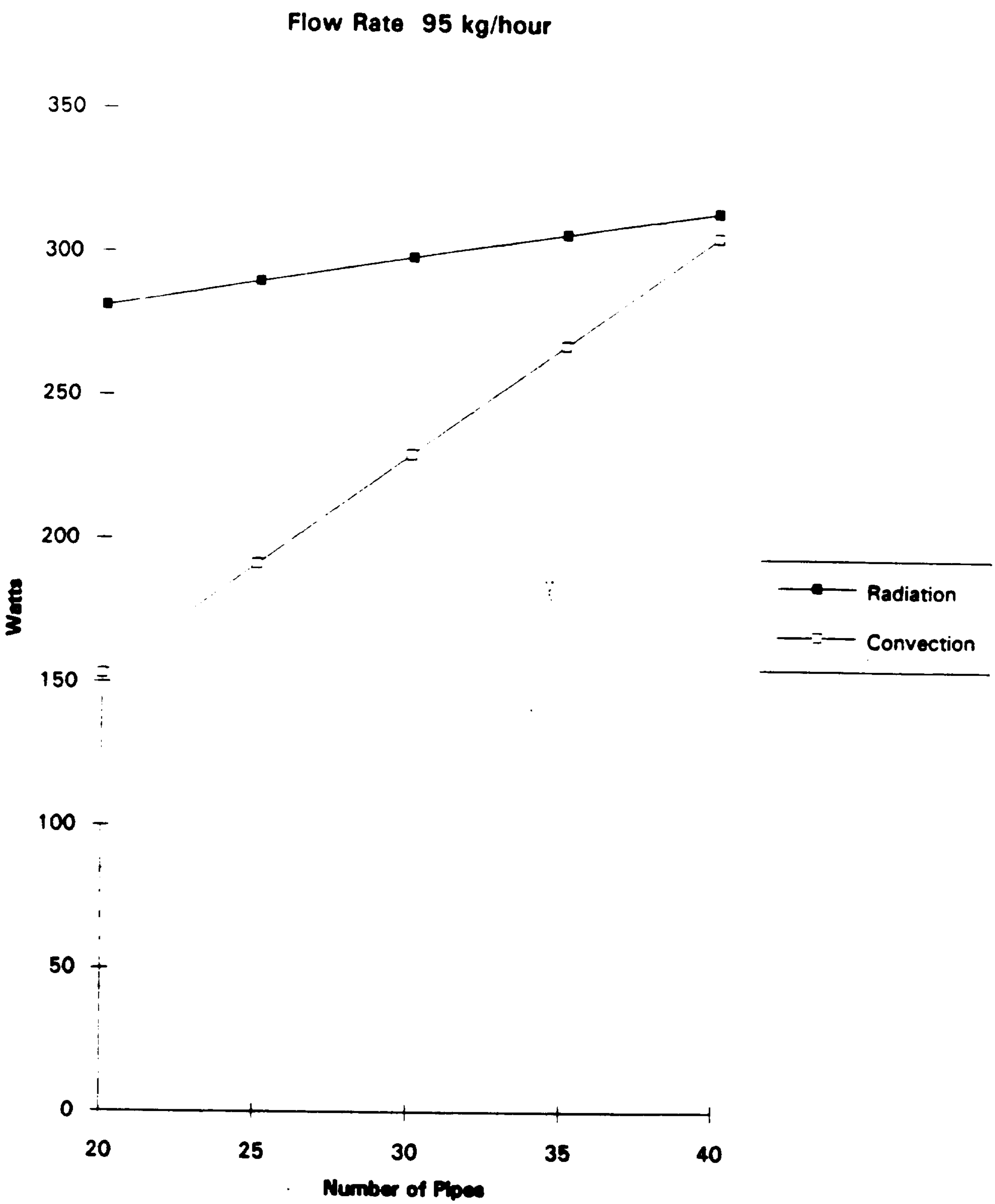
Material Properties (all S.I. units)

Material	Density	Yield Strength KPa	Conduct.	Cp
Aluminum	2700	20000	237	903
Beryllium	1850	N/A	200	1825
Magnesium	1740	80000	156	1024

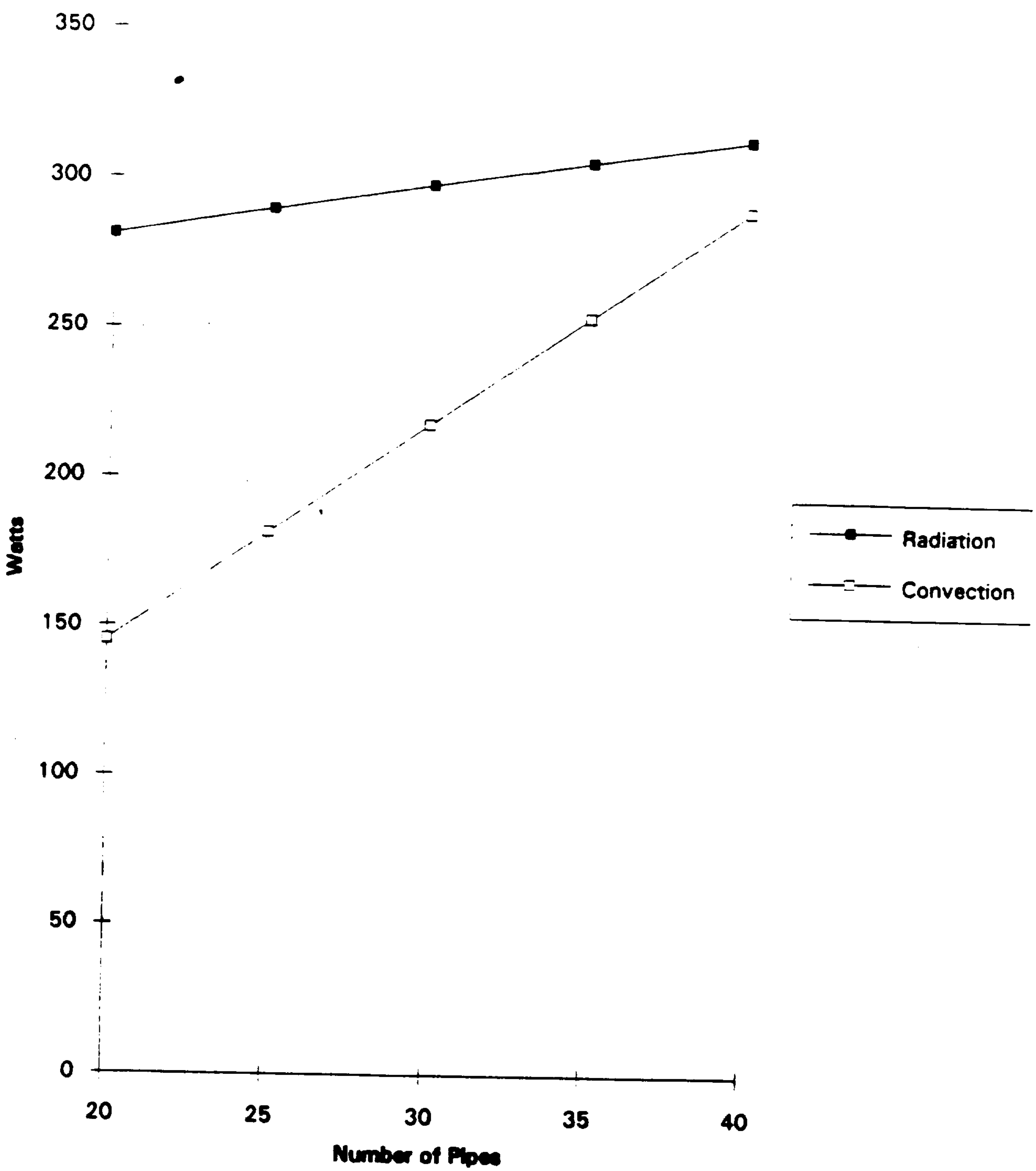
Apendix G

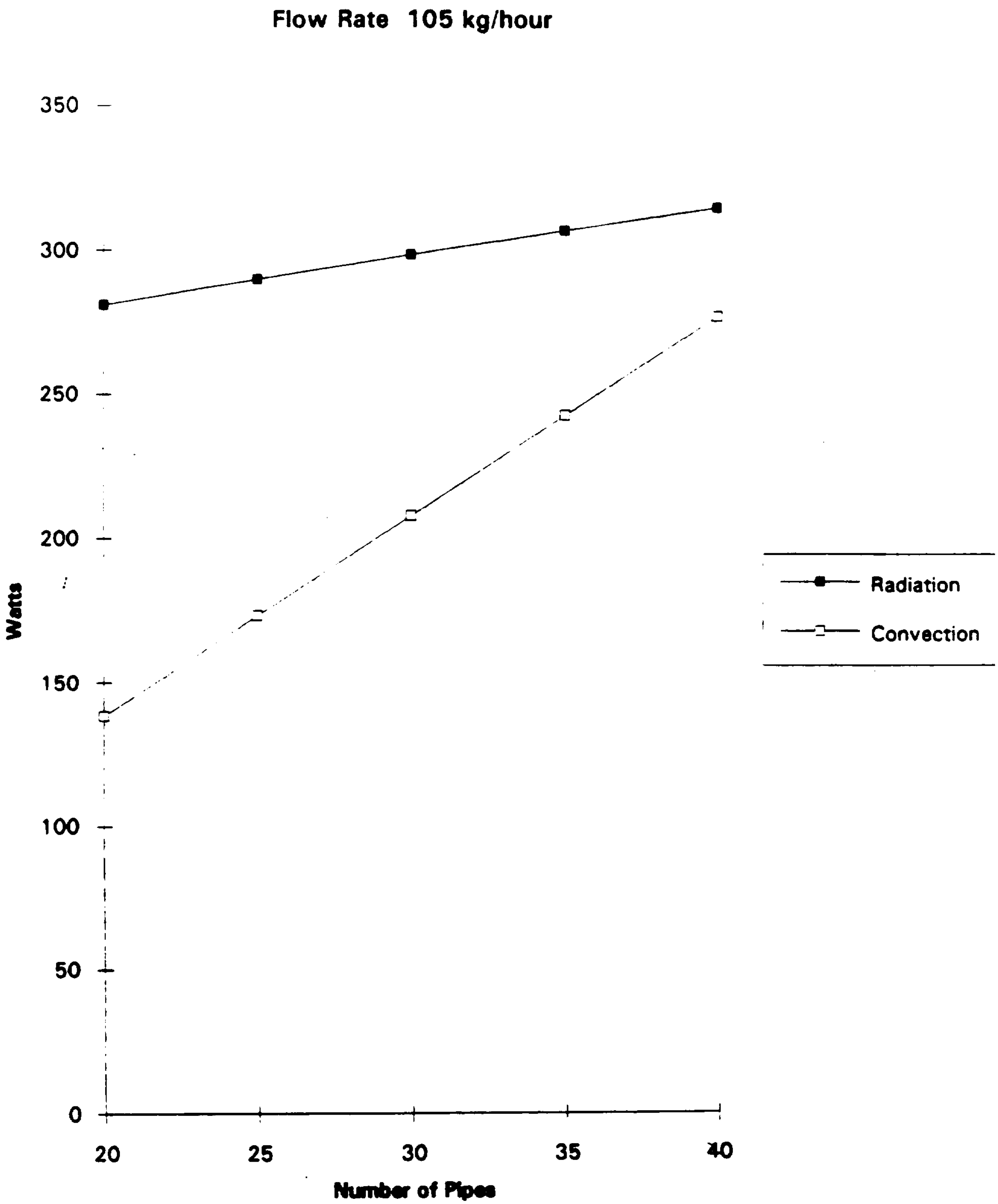
Number of Pipes vs. Flow Rate




Flow Rate 95 kg/hour
350
300
250
200
150
100
50
0
Watts
20
25
30
35
40
Number of Pipes
Radiation
Convection

Flow Rate 100 kg/hour




Flow Rate 105 kg/hour
350
300
250
200
150
100
50
0
Watts
20
25
30
35
40
Number of Pipes
Radiation
Convection

Appendix H
Additional Calculations

Thermal Expansion

Thermal expansion can be a problem because there is a possibility of the diameter of the pipes changing which may cause the mass flow rate to change and the type of flow may change from laminar to turbulent. Thermal expansion may also effect the dimensions for the connection to the PLSS. Calculations for thermal expansion are based on the equation:

$$dL = \alpha * L_o * dT$$

Where dL = change in length
L_o = initial length
dT = change in temperature
α = coefficient of thermal expansion

Length to be Determined	Coefficient of Thermal Expansion	Initial Length	Change in Temperature	Thermal Expansion
Outer Diameter	$22 \times 10^{-6}/°C$	9.00 mm	25 K	.00495 mm
Inner Diameter	$22 \times 10^{-6}/°C$	6.00 mm	25 K	.00330 mm
Width/Length	$22 \times 10^{-6}/°C$	813.00 mm	25 K	.447 mm

The thermal expansion for the diameter of the pipes is less than the $\pm$ .15 mm. However, the thermal expansion for the width and length of the plates is larger than the tolerance. The thermal expansion must be taken into account when making measurements for the connection with the PLSS.

Impact Strength

One specification for the system is that it must withstand an impact of 89 N. To determine if the radiative plates can withstand this force, the maximum impact force Aluminum can withstand was calculated:

Impact Strength x Time in Contact/Length

The length of the impact is estimated to be the diameter of a micrometeriod, .01mm. The contact time is estimated to be .001 sec. The impact strength for Aluminum is 33.0 J/sec. These values give an impact force of 3300 N, this is much greater than the 89 N specified, therefore the radiative plates can withstand the impact.

Insulation

The purpose of insulating the inner wall of the heat rejection system is to keep heat from going back to the astronaut. In the calculations, glass fiber insulation is used. The thermal conductivity of this material is 0.043 W/mK. Glass fiber is used since this material will not outgas in space. In order to decide on a thickness for the insulation, the heat transferred across the insulation layer has to be minimized. The heat transfer through the insulation can be calculated with the following expression:

$$q = \frac{k}{L} A \cdot (T_{out} - T_{in})$$

k = 0.043 W/mK
L = 0.813 m
A = 0.78 m^2
Tout = 300 K
T in = 285 K

Thickness	Heat Transferred
0.005 m	27.95 Watts
0.01 m	13.97 Watts
0.02 m	6.98 Watts

If an insulation of 2 cm. is used, only 7 Watts are transferred back to the astronaut. This value is approximately 2% of the heat radiated out of the system.

Appendix I
Binary Matrix for the Material

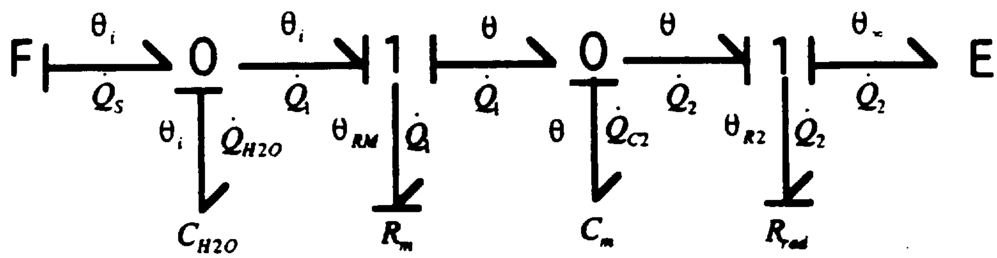
Category	Conductivity	Specific Heat	Density	Corrosiveness	Safety	Mechanica Properties	**Total**
Conductivity	X	1	1	0	0	1	**3**
Specific Heat	0	X	1	0	0	1	**2**
Density	0	0	X	1	0	0	**1**
Corrosiveness	1	1	0	X	0	1	**3**
Safety	1	1	1	1	X	1	**5**
Mechanical Properties	0	0	1	0	0	X	**1**

Appendix J
Binary Matrix for Fluid Considerations

Category	Reactiveness and Corrosion	Safety	Viscosity	Coefficient of Thermal Expansion	**Total**
Reactiveness and Corrosion	X	1	1	0	**2**
Safety	0	X	1	0	**1**
Viscosity	0	1	X	0	**1**
Coefficient of Thermal Expansion	1	1	1	X	**3**

Appendix K
Time Constant Analysis for the Radiative Plate

To obtain the time response of the radiative plate, bond graph techniques are used to model the heat exchanger system and to determine the time constant of the system. In this analysis, the LCVG is assumed to be 100% efficient. All of the bond graph elements are assumed to be ideal and linear. The following is the bond graph of the heat exchanger system:



For this thermal system, the constitutive relations are:

$$Q_{H2O} = C_{H2O} \qquad\qquad \theta_{RM} = R_m \dot{Q}_1$$
$$Q_{C2} = C_m \theta \qquad\qquad \theta_{R2} = R_{rad} \dot{Q}_2$$

The junction equations are given by:

$$\dot{Q}_S = \dot{Q}_{H2O} + \dot{Q}_1$$
$$\dot{Q}_1 = \dot{Q}_2 + \dot{Q}_{C2}$$
$$\theta_i = \theta_{RM} + \theta$$
$$\theta = \theta_\infty + \theta_{R2}$$

From the above bond graph and the junction and constitutive relations, the following state equations are derived for this second order system:

$$\dot{Q}_{H2O} = \dot{Q}_S + \frac{Q_{H2O}}{C_{H2O}R_m} - \frac{Q_{C2}}{C_m R_m}$$

$$\dot{Q}_{C2} = \frac{Q_{H2O}}{C_{H2O}R_m} - \frac{Q_{C2}}{C_m}\left(\frac{1}{R_m} + \frac{1}{R_{rad}}\right) + \frac{\theta_\infty}{R_{rad}}$$

Arranging the state equations in the form of:

$$\tau\dot{Q} + Q = f(t)$$

enables the time constant to be easily determined from the state equations. For the first state equation, the time constant was determined to be 0.854 seconds. The time constant for the second equation was determined to be 0.590 seconds. The larger time constant of 0.854 seconds dominates the system response. Since the system is considered to be at steady state after five time constants, the system will reach steady state after a disturbance in just 4.27 seconds.