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DEVELOPMENT OF AN EMULATION-SIMULATION THERMAL CONTROL MODEL FOR SPACE STATION APPLICATION

By

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NASA Technical Officer
John B. Hall, Jr.
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DEVELOPMENT OF AN EMULATION-SIMULATION THERMAL CONTROL MODEL FOR SPACE STATION APPLICATION

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ABSTRACT

The program is aimed towards development of an improved capability to compare various techniques for thermal management in the "Space Station". The work involves two major tasks:

TASK I Complete development of a Space Station Thermal Control Technology Assessment program.

TASK II Develop and evaluate emulation models.

The overall computer program is now operating well. Additional emulation models are to be added to the program in the months ahead.

INTRODUCTION

Current planning for the orbiting space station calls for a dual-keel configuration as shown in Figure 1. The thermal control system (TCS) for the space station is composed of a central TCS and internal thermal control systems for the modules, shown in Figure 2, as well a service facilities and attached payloads (hereinafter referred to as experimental truss and resource modules). The internal TCS may be attached to the central TCS through a thermal bus.

The central TCS is composed of a main transport system which collects waste thermal energy from each of the modules and transports it through coolant lines to the main rejection system. The main rejection system, in turn, is composed of steerable, constructable radiator elements attached to the transverse booms of the space station structure.

The waste heat loads in the modules arise from electrical and electronic equipment as well as metabolic loads in the manned modules. These equipment and metabolic loads may be collected by the central TCS or they may be transported to small radiators mounted on the body of individual modules.

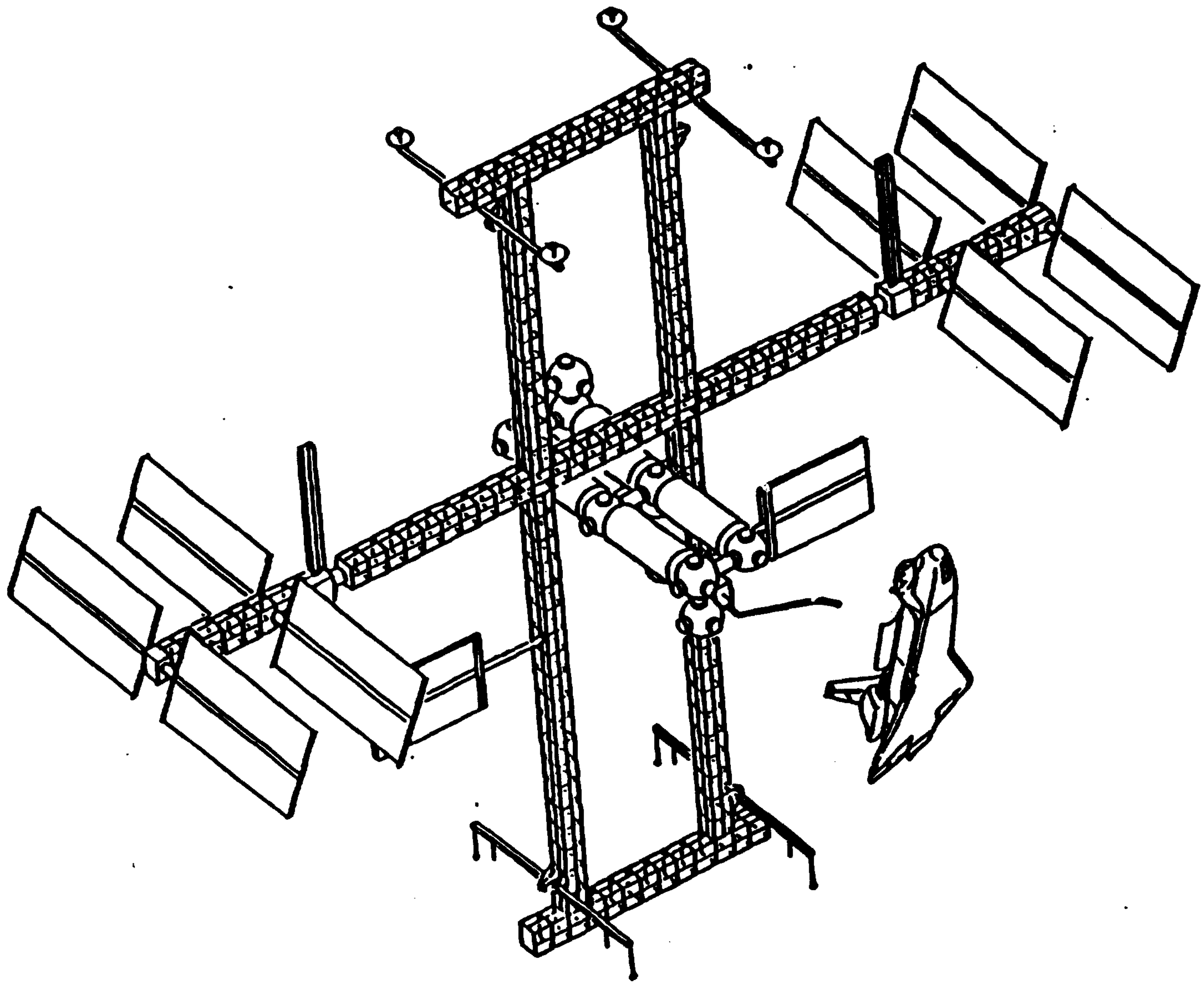


Figure 1. Space Station Configuration.

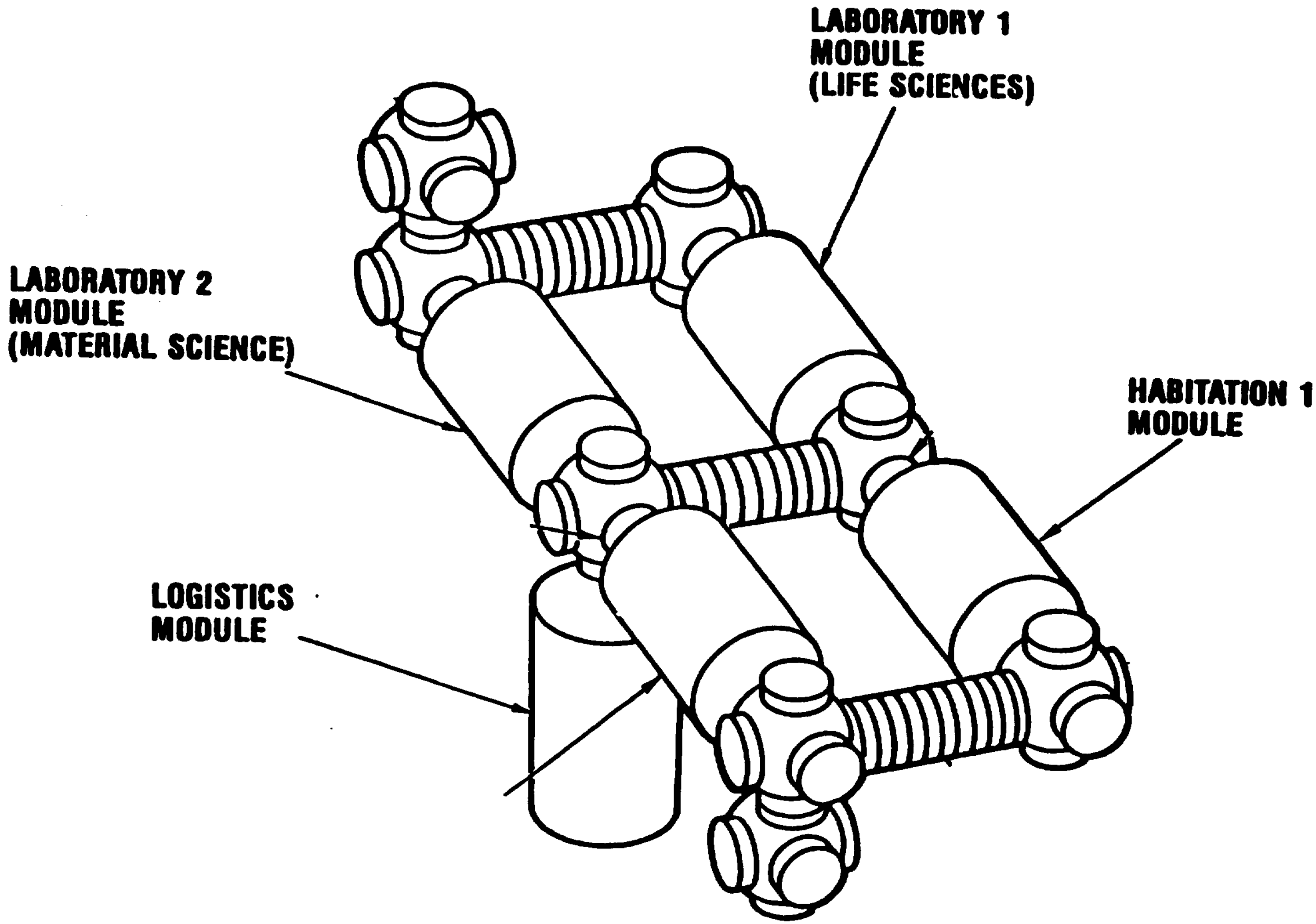


Figure 2. Station Modules.

Several candidate technologies are being considered for acquiring the waste heat loads, for transporting the thermal energy between the acquisition and rejection systems, and for rejecting the waste heat to space. The analysis techniques described in the present paper were developed for use in evaluating reliability, weights, costs, volumes, and power requirements for configurations using different candidates and different mission parameters.

EVALUATION TECHNIQUES

The thermal control system analysis program permits the user to design and analyze a space station thermal control system. The space station is assumed to be composed of seven distinct modules, each of which may have its own metabolic heat loads and equipment heat loads. In each of the modules, the user may specify the total metabolic load and the size and locations of the equipment loads. The metabolic loads are assumed to be acquired by air-water heat exchangers, transported by pumped liquid water loops, and rejected to space by body-mounted radiators attached to each of the modules which have metabolic loads. Because the metabolic loop is local to a module it is called an autonomous loop.

Heat loads generated by equipment in each module are assumed to be acquired by cold plates. The user may choose among the following candidates technologies for the cold plates in each module:

1. Conductive cold plate
2. Two-phase cold plate
3. Capillary cold plate

In addition, the user may locate up to five cold plates (each having a different capacity) in a module, choose the cold plate operating teperature, and specify the working fluid (water, ammonia or Freon-11). The user also has the option to specify whether the equipment loop is to be integrated or autonomous. If the equipment loop is integrated, the heat from the equipment is transported from the cold plates to the main heat transport system for eventual rejection to space by the main rejection system. On the other hand, if the equipment loop is autonomous, the heat from the equipment is rejected to space by body-mounted radiators located on the module exterior. In this case the user may specify separate candidate technologies for heat transport and heat rejecton in the autonomous equipment loop.

The user may select from the following candidate technologies for the main heat transport system or the heat transport system for a module having an autonomous equipment loop:

1. Pumped liquid loop
2. Pumped two-phase loop
3. Two-phase pumped heat pipe
4. Capillary pumped heat pipe.

In addition, the user may choose the transport length and specify the working fluid.

For the main heat rejection system or the heat rejection system for a module having an autonomous equipment loop, the user may select from the following candidate technologies:

1. Heat pipe radiator
2. High capacity heat pipe radiator
3. Liquid droplet radiator

In addition, the user may choose the radiator surface temperature and the emissivity of the radiator surface.

The data base for the thermal control system analysis program is divided into three major parts: the mission model parameters file, the candidate data files, and the system configuration file. Each of these are discussed in the following paragraphs.

The mission model parameters file contains information which applies specifically to the mission or which applies to the space station as a whole. A sample mission model parameter file, as it appears to the user, is shown in Figure 3. When the program begins execution, the mission model parameter file is read from the data base. Any one or all of these parameters may be changed and used temporarily for assessment purposes or they may replaced in the data base. In the latter instance, they become the new mission model parameter file when program execution begins anew because only the most recently saved version of the mission model parameter file is retained in the data base.

The candidate data files contain generic information for each of the candidate technologies available for heat acquisition, heat transport, and heat rejection. The data base contains one file for each candidate. A sample candidate data file, as it appears to the user, is shown in Figure 4. The weights, volumes, times and costs shown in the figure are those for the specified candidate rating. If the candidate technology is used with a different rating, these values are scaled accordingly. When the program begins execution, the candidate data files are read from the data base. Any one or all of the values in these files may be changed and used temporarily for assessment purposes or they may be replaced in the data base. In the latter instance, they become the new candidate data files when program execution begins anew because only the most recently saved versions of the candidate data files are retained in the data base.

The system configuration file is used to describe the actual thermal control system for the space station. The configuration of each module is specified by choosing the acquisition candidate (e.g. conductive cold plate) to be used to acquire the equipment load and by choosing the equipment loop to

MISSION MODEL PARAMETERS

1.	M...MISSION DURATION, DAYS:	3650.00
2.	R...RESUPPLY INTERVAL,DAYS:	90.00
3.	NP..POWER PENALTY, LB/KW:	350.00
4.	NC..CONTROL PENALTY:	.00
5.	NP1.PROPULSION PENALTY:	60.00
6.	P...PROBABILITY OF METEROID PENETRATION, (0.920 TO 0.993):	.990
7.	CFA.TRANSPORTATION COST FACTOR, THOUSAND DOLLARS/LB:	1.60
8.	MR..MAINTENANCE COST FACTOR, THOUSAND DOLLARS/HR:	35.00
9.	IF..INTEGRATION COST FACTOR, %:	35.00
10.	PF..PROGRAMMATIC COST FACTOR, %:	70.00

DO YOU WISH TO CHANGE ANY VALUES (Y OR N)
DO YOU WISH TO REPLACE THE
MISSION MODEL PARAMETERS (Y OR N)

Figure 3. Mission Parameters.

CANDIDATE DATA
CANDIDATE NAME: CONDUCTIVE COLD PLATE

1.	CANDIDATE RATING, KW:	50.000
2.	WEIGHT OF SPARES FOR 90 DAYS, LB:	22.100
3.	VOLUME OF SPARES FOR 90 DAYS, FT3:	6.350
4.	WEIGHT OF CONSUMABLES FOR 90 DAYS, LB:	.000
5.	VOLUME OF CONSUMABLES FOR 90 DAYS, FT3:	.000
6.	RELIABILITY (0-8):	8.000
7.	TECHNOLOGY READINESS (0-8):	8.000
8.	PACING TECHNOLOGY PROBLEMS (0-8):	8.000
9.	90 DAY MAINTENANCE TIME, HR:	5.000
10.	NONRECURRING DESIGN, DEVELOPMENT, TEST AND CERTIFY, 1983 MILLION DOLLARS:	213.800
11.	SPARES AND CONSUMABLFS TO OPERATE FOR 90 DAYS, 1983 MILLION DOLLARS:	.240
12.	COST OF FLIGHT UNIT, 1983 MILLION DOLLARS:	4.800

DO YOU WISH TO CHANGE ANY VALUES (Y OR N)
DO YOU WISH TO REPLACE THIS CANDIDATE FILE (Y OR N)

Figure 4. Candidate Data.

be integrated (i.e. attached to the main transport and main rejection systems) or autonomous (i.e. attached to body-mounted radiators). In addition, the user may specify the configuration data illustrated in Figure 5 for each module. Figure 6 shows a schematic of a typical configuration for an integrated module.

Each system configuration file contains configuration details for all modules as well a specifications for the main heat transport and main heat rejection systems. A default system configuration is stored in the data base and is retrieved when the program begins execution. Any of the values in the system configuration file may be changed, and the new system configuration may be saved under a system name specified by the user. Up to 71 different system configurations can be stored in the data base at one time, and these may be recalled for later use by directing the program to retrieve a previously saved system configuration file.

The thermal control system analysis program uses the system configuration file, together with the mission model parameter file and the candidate data files, to assess the reliability, weight, volume and cost of the proposed thermal control system. The analysis produces the following output:

1. Acquisition assessment for each module
2. Summary acquisition assessment for all modules
3. Summary transport assessment for the main transport system
4. Summary rejection assessment for the main rejection system
5. Summary assessment for the entire thermal control system.

The analysis begins with a determination of the launch weight, launch volume, heat transfer surface areas and external power requirement imposed by the acquisition system for each module. These computations depend upon the acquisition candidate and module configuration and are performed in separate

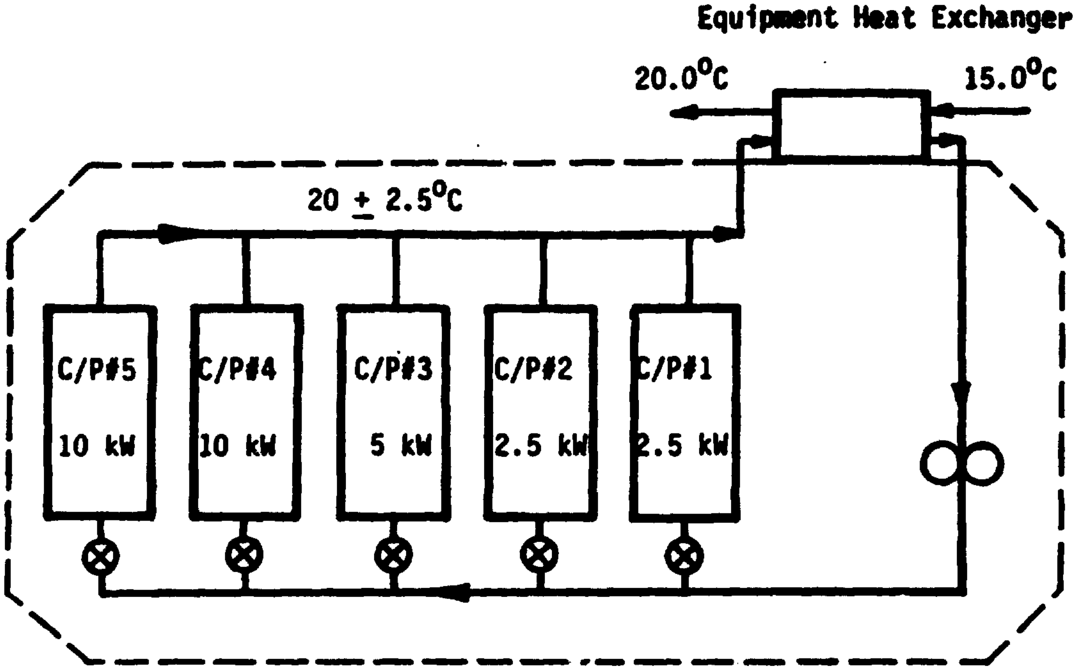
LOGISTICS MODULE

ACQUISITIO SUBSYSTEM: CONDUCTIVE COLD PLATE
TOTAL COLD PLATE CAPACITY, KW: 12.00

1.	NUMBER OF COLD PLATES:			3.00
2.	COLD PLATE OPERATING TEMPERATURE, C:			20.00
3.	METABOLIC LOAD, KW:			2.36
		CP #1	CP #2	CP #3
4.	HEAT REJECTION LOADS, KW:	4.00	4.00	4.00
5.	MAIN SUPPLY LINE LENGTHS, FT:	8.00	4.00	4.00
6.	BRANCH SUPPLY LINE LENGTHS, FT:	10.00	10.00	10.00
7.	MAIN RETURN LINE LENGTHS, FT:	8.00	4.00	4.00
8.	BRANCH RETURN LINE LENGTHS, FT:	10.00	10.00	10.00
9.	WORKING FLUID:			AMMONIA
	PIPE MATERIAL:			STAINLESS STEEL

DO YOU WISH TO CHANGE ANY VALUES (Y OR N)

Figure 5. Module Configuration Data.



TYPICAL MODULE EQUIPMENT LOOP

Figure 6. Typical Configuration for an Integrated Module.

subroutines - one for each of the candidate technologies. For example, acquisition system subroutines contain algorithms for sizing coolant lines for minimum weight, determining cold plate sizes and weights, computing pumping power required, determining thermal bus connection requirements, and computing the volume occupied by the acquisition systems. These computations depend upon the candidate technology employed (i.e. single phase or two-phase cold plates, etc.), working fluid, materials, and operating temperatures. For a rejection system candidate such as a heat pipe radiator, the candidate subroutine contains algorithms for assessing the performance of heat pipe elements which would be used to construct the radiator. In this case, parameters such as working fluid, material, radiator temperature, geometry and surface radiative properties may be selected and included in the design calculations.

The launch weight, launch volume, surface areas and power requirement computed in the candidate subroutine, together with the mission model parameters and candidate data file, are used to compute all of the other assessment information illustrated in Appendix I. The algorithms for these computations are detailed in Appendix II. A flow schematic illustrating the operation of the program as the user views it is shown in Figure 7. The following paragraphs describe several of the thermal models used in the candidate subroutines.

TWO-PHASE COLD PLATE MODELS

Two-Phase Cold Plates

The following assumptions are made for the two-phase cold-plate system (1).

1. Cold plate temperatures are to be maintained within 20 $\pm$ 2.5^oC.
2. Vaporization efficiency is 100 percent for the cold plates.

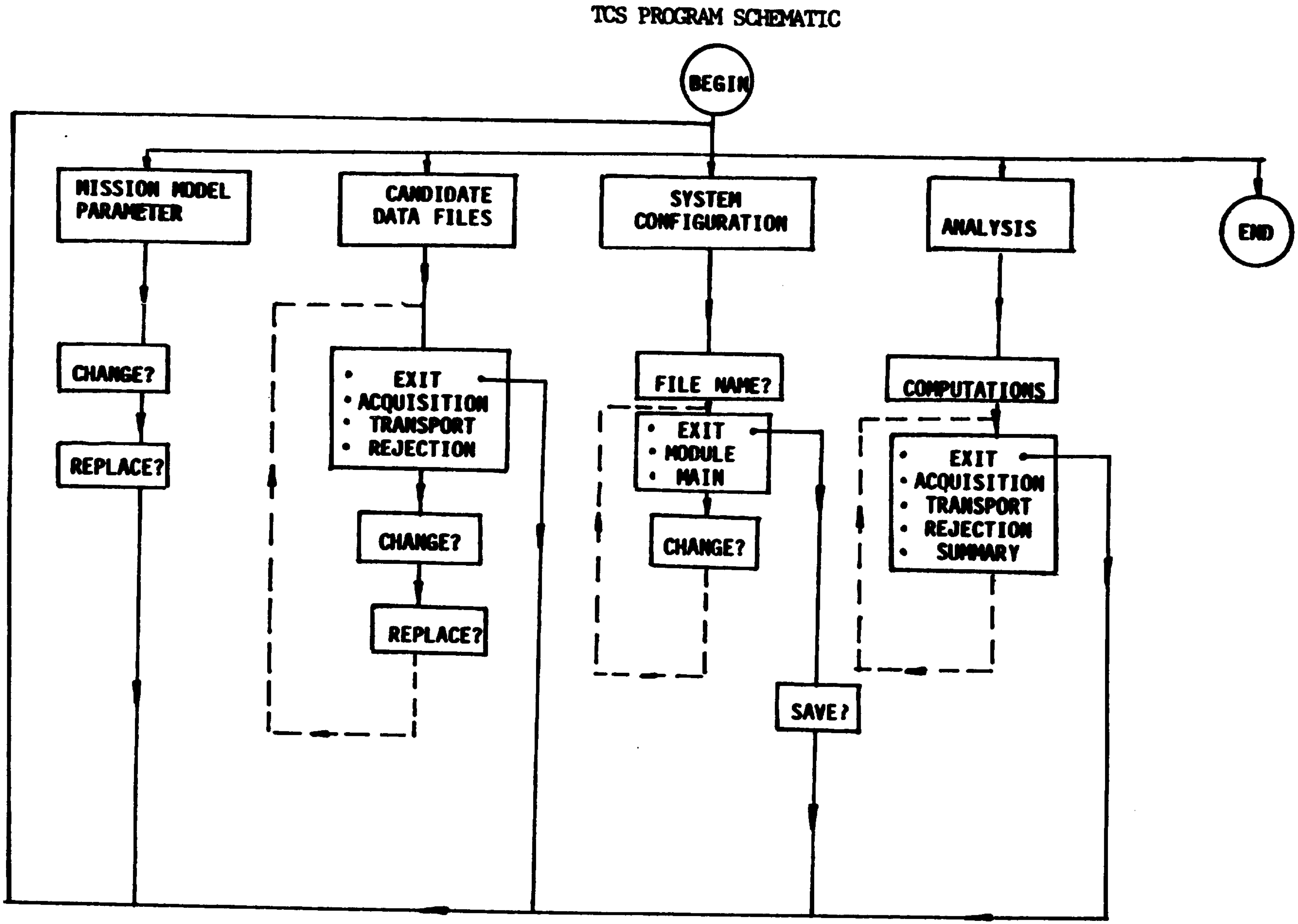


Figure 7. TCS Program Flow Schematic.

3. Valves control the liquid flow to the cold plates.

4. Cold plate mass is 11.5 lbm/ft^2.

5. Cold plates are sized based upon an interface heat flux of 600 W/ft^2.

6. Pump package mass is 40 lbm.

7. Equipment loop heat exchanger mass is 10.6 lbm/ft^2.

8. Maximum allowable vapor line temperature drop is limited to 1.7°C.

With the cold plate capacity, $\dot{Q}$, specified, the mass flow rate of working fluid through the cold plate is calculated from

$$\dot{m} = \frac{\dot{Q}}{h_{fg}} \tag{1}$$

where h_{fg} is the latent heat of vaporization of the working fluid at a saturation temperature of 20°C (assumptions 1 and 2). The heat transfer surface area for each cold plate is given by (assumption 5)

$$A = \frac{\dot{Q}}{600\ W/ft^2} \tag{2}$$

and the cold plate mass is (assumption 4)

$$m_{cp} = (11.5\ lbm/ft^2)\ A \tag{3}$$

As the working fluid changes phase in the cold plate, the temperature of the working fluid remains relatively constant at the saturation temperature of 20°C. Furthermore the cold plate is designed for a high overall heat transfer coefficient, U. Since the cold plate temperature is related to the heat transfer rate by

$$\dot{Q} = UA(T_{cp} - T) \qquad (4)$$

the difference between the cold plate temperature and the saturation temperature of the working fluid can be kept small.

Two-Phase Loop Analysis

The analysis of the two-phase equipment cooling loop for a particular module assumes that the location and heat transfer capacity of each cold plate in the loop are given. This information for each module is stored in the data base and is accessible for the analysis of two-phase loops and other candidate technologies as well. The user of the analysis program may specify different cold plate capacities, select various working fluids for the two-phase loop, and change operating temperatures, if desired.

Liquid Supply Lines

The pipe sizes for the liquid supply line in the two-phase cold plate system are determined by minimizing the weight of the piping system (1). Each segment of pipe in the longest pipe run is optimized individually by minimizing the mass or weight of the segment which is determined from

$$\text{Mass} = M_i = \text{mass of pipe} + \text{mass of liquid} + \text{pump power penalty mass}$$

where

$$\text{mass of pipe} = \rho_{ss} L_i \pi (D_i + t_i) t_i$$

$$\text{mass of liquid} = \rho_\ell \pi D_i^2 \, L_i / 4$$

$$\text{pump power penalty mass} = M_p P_p$$

and the pump power is determined from

$$P_p = \frac{\dot{m}_i \Delta P_i}{\rho_\ell \eta_p}$$

The pressure drop for the segment of pipe is calculated from

$$\Delta P_i = \frac{8 L_i \dot{m}_i^2 f_i}{\pi^2 \rho_\ell D_i^5}$$

where the friction factor is

$$f_i = 0.316/Re^{1/4}$$

for turbulent flow (2) in smooth pipes and

$$f_i = 64/Re$$

for laminar flow (2), and the Reynolds number is

$$Re = \frac{4 \dot{m}_i}{\pi \mu_\ell D_i}$$

Thus

$$\Delta P_i = \frac{128 \mu_\ell L_i \dot{m}_i}{\pi \rho_\ell D_i^4}$$

and the pipe segment mass to be minimized is

$$M_i = \rho_{ss} L_i \pi (D_i + t_i) t_i + \rho_\ell \pi D_i^2 L_i/4 + M_p \frac{\dot{m}_i \Delta P_i}{\rho_\ell \eta_p} \quad (5)$$

The pipe thickness, t_i, is determined by the internal pipe diameter according to standard pipe and tube specifications.

The remaining pipe sizes for shorter runs are determined by the lengths, mass flow rates and the pressure drops required to match those dictated by the longest run of pipe.

The vapor line sizes in the two-phase cold plate system are selected consistent with the desire to limit the loss of stagnation pressure and stagnation temperature in the vapor return lines (1). The analysis of these losses is based upon adiabatic, compressible pipe flow with friction (3) as outlined below.

The vapor line diameter for each segment of the longest run in the vapor return line is chosen such that the stagnation pressure drop is less than, say, 2 percent of the stagnation pressure at the exit of the cold plate. The conditions at the inlet of the vapor line are denoted by the subscript 1 and the subscript 2 denotes the conditions at the exit, and we require that

$$P_{02}/P_{01} > 0.98 \tag{6}$$

where the zero subscript designates stagnation conditions.

The stagnation pressure ratio can be computed from

$$\frac{P_{02}}{P_{01}} = \frac{M_1}{M_2}\left[\frac{\left(1 + \frac{k-1}{2} M_2^{\,2}\right)}{\left(1 + \frac{k-1}{2} M_1^{\,2}\right)}\right]^{\frac{(k+1)}{2(k-1)}}$$

where

$M_1 = V_1/C_1$ is the Mach number

$C_1 = \quad kRT_1 g_c$ is the sonic velocity

k = is the ratio of specific heats for the vapor

R = is the gas constant for the vapor

The general procedure for determining the information necessary to calculate the stagnation pressure ratio is iterative in nature as outlined in the following.

1. Assume a pipe diameter D and calculate the inlet vapor velocity, V_1, from the known mass flow rate.
2. Calculate the inlet Mach number, M_1
3. Calculate the inlet Reynolds number, Re_1, determine the friction factor, f, for turbulent or laminar flow as dictated by the Reynolds number, and calculate $\bar{f}L/D)_{actual}$ from the given pipe length and assumed diameter.
4. Calculate the inlet stagnation temperature

$$T_{01} = T_1 + \frac{V_1^{\,2}}{2C_p}$$

and the inlet stagnation pressure

$$P_{01} = P_1 \left(\frac{T_{01}}{T_1} \right)^{k/(k-1)}$$

5. Calculate the quantity $\bar{f}L^*/D)_1$ at the inlet,

$$\frac{\bar{f}L^*}{D}\Big)_1 = \frac{1 - M_1^{\,2}}{k\, M_1^{\,2}} + \frac{k+1}{2k} \ln \left[\frac{(k+1)M_1^{\,2}}{2[1 + \frac{1}{2}(k-1)M_1^{\,2}]} \right]$$

and $\frac{\bar{f}L^*}{D})_2$ from

$$\frac{\bar{f}L^*}{D})_2 = \frac{\bar{f}L^*}{D})_1 - \frac{\bar{f}L}{D})_{actual}$$

6. Solve the following transcendental equation for the exit Mach number, M_2:

$$\frac{\bar{f}L^*}{D})_2 = \frac{1 - M_2^2}{kM_2^2} + \frac{k+1}{2k} \ln \left[\frac{(k+1)M_2^2}{2[1 + \frac{1}{2}(k-1)M_2^2]} \right]$$

7. Finally, compute P_{02}/P_{01} from Equation (6). If $P_{02}/P_{01} < 0.98$, choose a larger pipe diameter and repeat steps 1 through 6. If $P_{02}/P_{01} > 0.98$ choose a smaller pipe diameter and repeat steps 1 through 6. If $P_{02}/P_{01} = 0.98$, the assumed pipe diameter is adequate for this pipe segment.

When all vapor and liquid line diameters have been selected the wet and dry piping weights can be calculated and the pump size, power and weight can be determined. A schematic of the two-phase loop analysis subroutine is shown in Figure 8.

HIGH CAPACITY HEAT PIPE RADIATORS MODELS

A high performance heat pipe radiator using a series of heat pipes with combination slab and circumferential capillary structure is modeled for space station use in the temperature range of $310^{\circ}K$ to $366^{\circ}K$ ($100^{\circ}F$ to $200^{\circ}F$). A schematic of the capillary structure is shown in Figure 9. Axial transport of working fluid primarily occurs through the central slab while the circumferential structure distributes the fluid around the circumference in the heated and cooled sections.

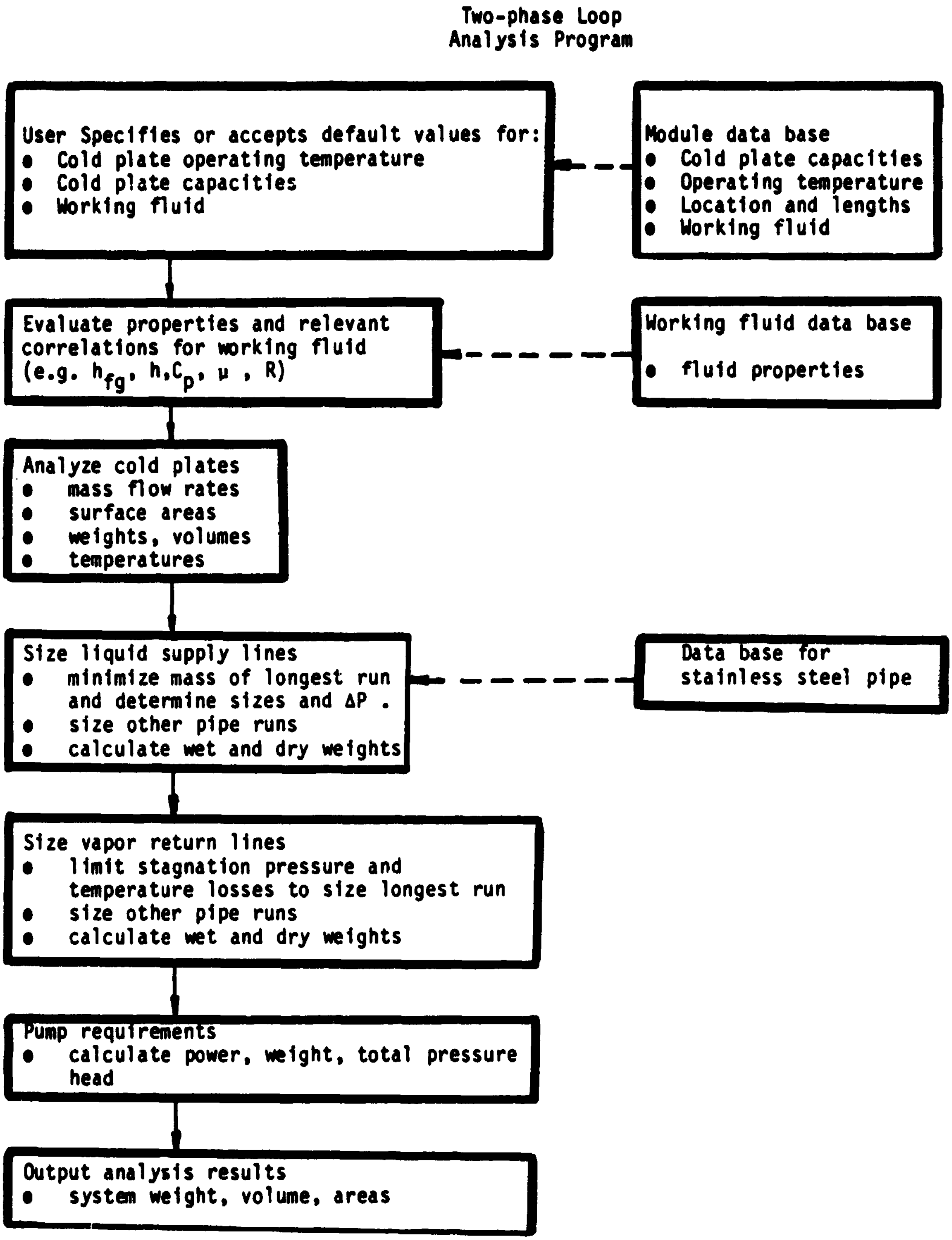


Figure 8. Schematic Two-Phase Loop Anaoysis.

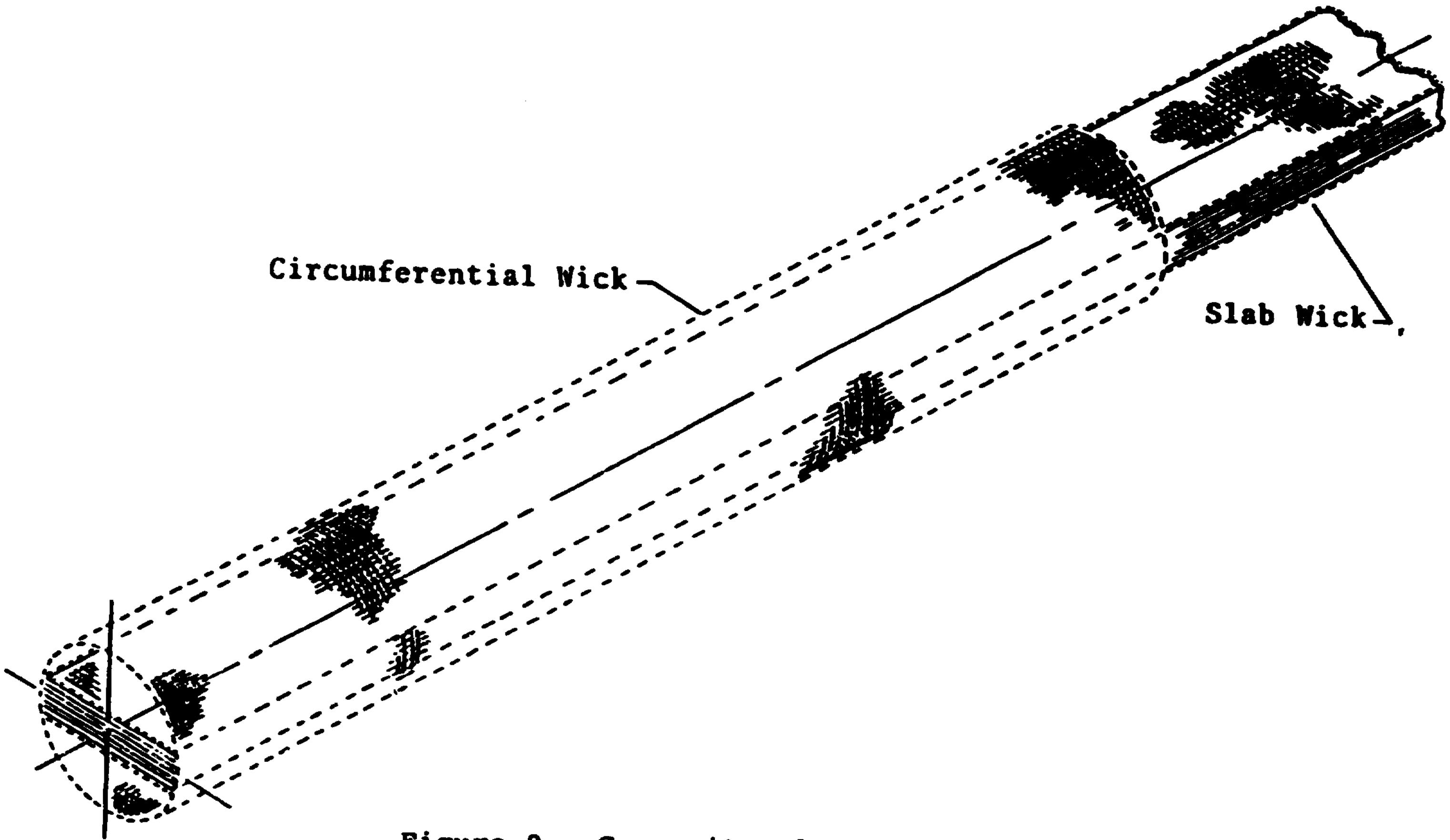


Figure 9. Composite Slab and Circumferential Capillary Structure at Evaporator.

Performances of various heat pipes to be used in a radiator panel are estimated from experimental studies performed at Georgia Tech, Reference (7) on a Refrigerant-11 heat pipe with slab capillary structure. The experimental heat pipe is described in Table I. It was found that this heat pipe could transport a maximum thermal energy of about 130 watts at 440^{o}K when operating with refrigerant-11 as a working fluid. Heat pipes to be used in a radiator for the space station may use other working fluids, may utilize different capillary structures, may be of different outside diameter and (or) length and may operate at different temperatures. All of these design parameters greatly affect heat pipe thermal transport capacity.

Writing momentum, energy and continuity equations for steady operation of the model heat pipe at capillary limited heat transfer and making the standard simplfying assumptions the following equation, from reference (8), is obtained.

$$\dot{Q}_{CL} = \frac{2N/r_p}{\dfrac{R\ell_{eff}}{b\delta_T} + \dfrac{K_C L}{4n_C\delta_C}\left(\dfrac{1}{\ell_e} + \dfrac{1}{\ell_c}\right) + \dfrac{8\mu_V\rho_L\ell_{eff}}{\pi\mu_L\rho_V r_V^4}}$$

where

$\dot{Q}_{CL}$ = Capillary limited heat transfer rate

$N = \dfrac{\sigma h_{fg}\rho_L}{\mu_L}$ = "Heat Pipe Number"

σ = surface tension of liquid

h_{fg} = heat of vaporization

ρ_L = liquid density

μ_L = liquid dynamic viscosity

r_p = pore radius at evaporator surface

TABLE I

Experimental Heat Pipe Details

Working Fluid	Refrigerant-11 (CCl_3F)
Container Material	Type 316 stainless steel
Total Heat pipe Length	80 cm
Evaporator Length	15.24 cm
Condenser Length	24.30 cm
Adiabatic Section Length	40.46 cm
Container Outside Diameter	1.91 cm
Container Inside Diameter	1.57 cm
Wick Material	Type 316 stainless steel
Central Composite Slab Wick	2 layers of 100 mesh screen around 4 layers of 40 mesh screen
Circumferential Wick	2 layers of 100 mesh screen
Cooling Jacket Material	Type 316 stainless steel
Cooling Jacket Outside Diameter	2.54 cm
Cooling Jacket Inside Diameter	2.21 cm
Coolant	General Electric Silicone Fluid, SF 1093 (50)

$$\bar{K} = \frac{\delta_T}{\frac{n_A \delta_A}{K_A} + \frac{n_B \delta_B}{K_B}}$$ = effective inverse permeability for slab based on approach velocity.

δ_T = total thickness of slab

n_A = number of layers of fine mesh in slab

n_B = number of layers of coarse mesh in slab

δ_A = thickness of a single layer of material A

δ_B = thickness of a single layer of material B

K_A = inverse permeability for material A based on approach velocity

K_B = inverse permeability for material B based on approach velocity

ℓ_{eff} = effective length of liquid path in slab

b = width of slab

K_c = inverse permeability for material at evaporator and condenser surfaces based on approach velocity

L = average distance traveled by liquid in circumferential capillary structure at evaporator or condenser (approximately 45^o arc)

n_c = number of layers of capillary material on circumference

δ_c = thickness of a single layer of material C

ℓ_e = axial length of evaporator section

ℓ_c = axial length of condenser section

μ_v = dynamic viscosity of vapor

ρ_v = density of vapor

r_v = hydraulic radius of vapor space

In the denominator of this equation the three terms are related to flow resistance in the central slab, the circumferential capillary structure and the vapor region, respectively. For the present design flow resistance is much larger in the slab than in the circumferential structure or the vapor region. Thus, approximately

$$\dot{Q}_{CL} = \frac{2N}{\frac{r_p \, R \, \ell_{eff}}{b\delta_T}}$$

and

$$\dot{Q}_{CL_{II}} = \dot{Q}_{CL_I} \frac{N_{II}}{N_I} \frac{R_I}{R_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{\ell_{eff,I}}{\ell_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

where subscript I refers to a known performance and known design parameters and II refers to predicted performance when new design parameters are chosen. The width of the slab is assumed constant.

Let us assume that design heat transport capability is one-half of maximum transport capability.

$$\dot{Q}_D = 1/2 \; \dot{Q}_{CL}$$

and

$$\dot{Q}_{D_{II}} = \dot{Q}_{D_I} \frac{N_{II}}{N_I} \frac{R_I}{R_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{\ell_{eff,I}}{\ell_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

As an example consider the prediction, from a measured value for R-11 at 440^{o}K, of design heat flux for a heat pipe with ammonia at 310^{o}K with different capillary structure and different length as shown in Table II.

Table II

PARAMETER	CASE I	CASE II
Working Fluid	R-11	Ammonia
Temperature	$440^{0}K$	$310^{0}K$
Slab Capillary Structure	2 layers 100 mesh +4 layers 40 mesh	4 layers 400 mesh +5 layers 30 mesh
Circumferential Capillary Strucuure	2 layers 100 mesh	2 layers 400 mesh
$\bar{K}$ ($\frac{1}{m^2}$)	0.829×10^{9}	0.696×10^{9}
$r_p(m)$	7.88×10^{-5}	1.91×10^{-5}
Heat Pipe Length (ft)	2.62	50
Effective Transport Length (ft)	1.98	25
Heat Pipe Number (w/m^2)	1.7×10^{9}	5.6×10^{10}
$S_T(m)$	2.79×10^{-3}	3.41×10^{-3}
$\dot{Q}_{CL}(kW)$	0.130	2.03
$\dot{Q}_D(kW)$	0.065	1.015

We now consider the design of the radiator. Assume the following values for design parameters

Heat load 50 kW

Steerable radiator with thermal storage

Absorptivity, $\alpha_s = 0.30$

Emissivity, $\varepsilon = 0.78$

Heat pipe fluid at $100^{o}F$

Radiator average surface temperature $75^{o}F$

Area 2,500 ft^2

Material aluminum

Figure 10 shows a radiator constructed from a series of 50 foot heat pipes and fin panels. Assuming each heat pipe is 3/4 in. outside diameter and 5/8 in. inside diameter and 50 ft. long the metal weight will be about 8 lbm and the working fluid will weigh about 1.5 lbm for a total weight of 9.5 lbm per pipe. The panel width and weight per panel are given by the following expressions:

$$w_p(\text{in}) = \text{panel width} = \frac{631}{N_p}$$

where

N_p = number of heat pipes in 50 kW radiator

$m_p(\text{lbm})$ = weight per panel

$$= \frac{600}{N_p}[631 - N_p(0.75)](0.0625)(0.1) + 9.5$$

where fin thickness is taken to be 1/16 in. For example for 200 pipes (and 200 panels) in a 50 kW radiator the weight per panel would be 18.5 lbm and total radiator weight would be 3,700 lbm. The volume of the unit would be approximately

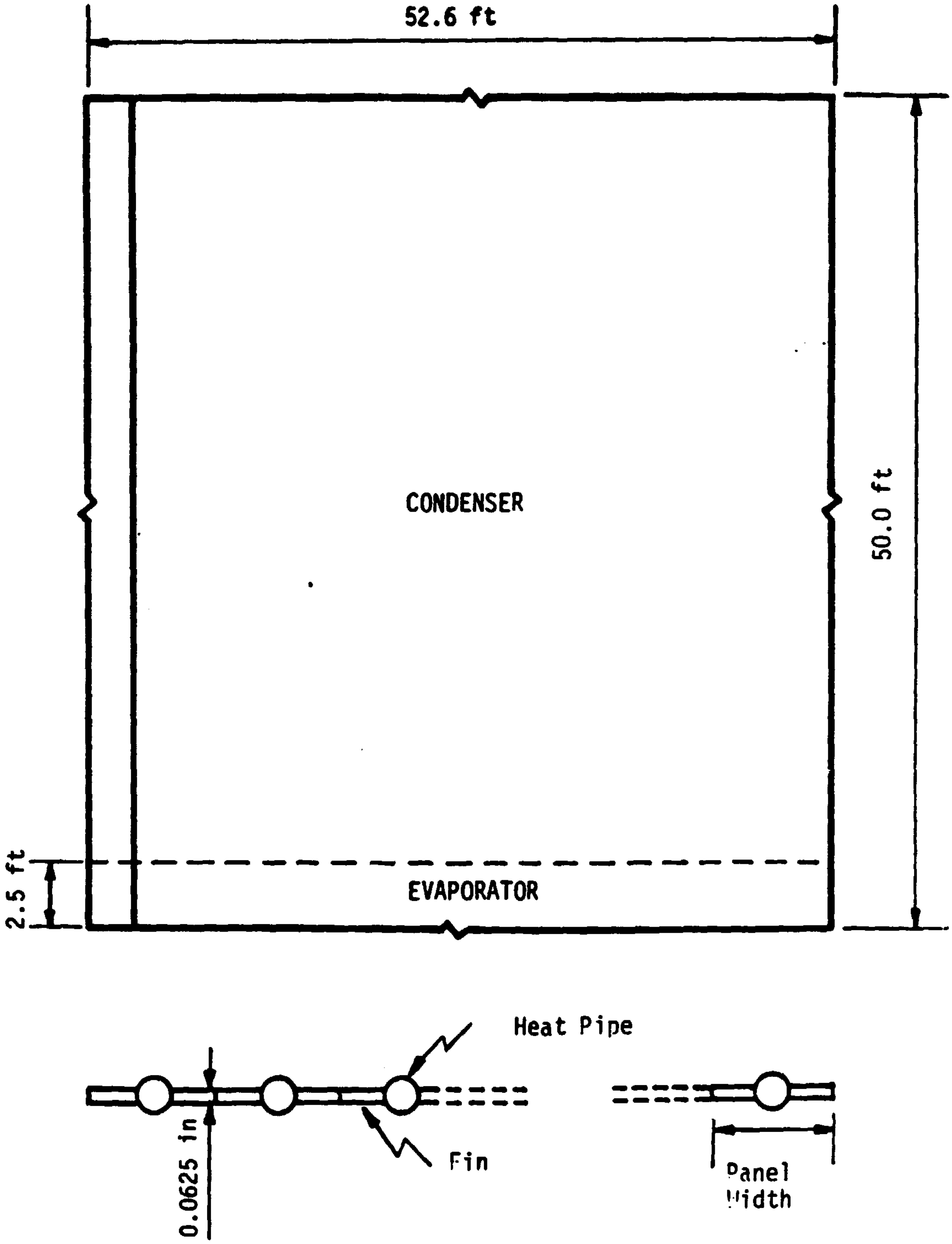


Figure 10. Heat Pipe Radiator.

$$50 \text{ ft} \times 52.6 \text{ ft} \times 0.0625 \text{ ft} = 164 \text{ ft}^3 \text{ for } 50 \text{ kW}$$

Table III shows the results of choosing among several different working fluids and working fluid temperatures. Values for various parameters used in computing values listed in the table are given below the table. Design heat transport per pipe (taken to be one half of capillary limitation) ranges between about 1 kW for ammonia at 310^{o}K to about 0.18 kW for R-11 at 366^{o}K. While total radiator weight varies between 2,580 lbm for ammonia at 310^{o}K to 4,090 lbm for R-11 at 366^{o}K.

The following values for parameaters define a base design.

Ga. Tech heat pipe

50 kW

2500 ft^2(each side) - reference (4)

Radiator surface temperature 297^{o}K

Material - aluminum

Heat pipe I.D. - 0.625 in.

Heat pipe O.D. - 0.75 in.

Fin thickness - 0.0625 in.

Heat pipe length - 50 ft.

Capillary structure - 2 layers 400 mesh on circumference, 4 layers 400 mesh + 5 layers 30 mesh in slab.

Evaporator length 2.5 ft.

Condenser length 47.5 ft.

Working fluid ammonia

Working fluid temperature 310^{o}K

Design heat transfer per pipe 1.02 kW

Number of panels 50

Panel width per pipe 12.24 in.

TABLE III

HEAT PIPE WORKING FLUID AND TEMPERATURE

Parameter	R-11 310°K	R-11 366°K	Methanol 310°K	Methanol 366°K	Ammonia 310°K	Ammonia 366°K	Acetone 310°K	Acetone 366°K
$\dot{Q}_{CL}$(kW)	0.440	0.367	1.54	1.61	2.03	0.660	1.10	0.918
$\dot{Q}_{D}$(kW)	0.220	0.184	0.770	0.805	1.015	0.330	0.550	0.459
Number of Pipes for 50 KW	229	275	65	62	49	153	92110	
Panel Width Per Pipe (in)	2.62	2.18	9.23	9.68	12.24	3.92	6.52	5.45
Weight Per Panel (lbm)	16.5	14.9	41.3	43.0	52.6	21.4	31.1	27.1
Total Radiator Weight (lbm)	3,780	4,090	2,690	2,660	2,580	3,270	2,870	2,990
Radiator Volume (ft^3)	156	156	156	156	156	156	15[illegible]56	

Heat Load - 50 kW
Radiator Surface Area (per side) - 2,500 ft^2
Radiator Average Surface Temperature - 75°F
Material - Aluminuim
Heat Pipe I.D. - 0.625 in
Heat Pipe O.D. - 0.75 in
Fin Thickness - 0.0625 in
Heat Pipe Length - 50 ft
Capillary Structure - 2 layers 400 mesh on circumference, 4 layers 400 mesh
+ 5 layers 30 mesh in slab
Evaporator Length - 2.5 ft.
Condenser Length - 47.5 ft.

Weight per panel 52.6 lbm

Total raditor weight (exclusive of heat exchanger) 2,580 lbm

Radiator volume (exclusive of heat exchanger) 156 ft^3

Absorptivity, $\alpha_s = 0.30$

Emissivity, $\epsilon = 0.78$; ratio $\alpha_s/\epsilon = 0.385$

$\bar{K}_I$, effective inverse permeability of slab, 0.696×10^9 ($1/m^2$)

r_{p_I} pore radius at evapoarator, 1.91×10^{-5} m

$\ell_{eff,I}$, heat pipe effective length, 25 ft.

N_I, heat pipe number, 5.6×10^{10} W/m^2

$\delta_{T,I}$, slab total thickness, 3.41×10^{-3} m

The following equations may be used to predict areas and weights for a particular candidate from known values for the base design.

A. Design Heat Transport Per Pipe

$$\dot{Q}_{D_{II}} = \dot{Q}_{D_I} \frac{N_{II}}{N_I} \frac{\bar{K}_I}{\bar{K}_{II}} \frac{r_{p_I}}{r_{p_{II}}} \frac{\ell_{eff,I}}{\ell_{eff,II}} \frac{\delta_{T_{II}}}{\delta_{T_I}}$$

where subscripts I and II refer to the base case and case to be computed, respectively.

B. Number of Panels

$$N_p = \frac{\dot{Q}}{Q_{D_{II}}}$$

where $\dot{Q}$ = radiator rating (kW)

C. Radiator Surface Area

$$\frac{A_{II}}{A_I} = \frac{\dot{Q}_{II}}{\dot{Q}_I} \frac{\epsilon_I}{\epsilon_{II}} \frac{F_{\alpha II}}{F_{\alpha I}} \left(\frac{T_I}{T_{II}} \right)^4$$

where $F_\alpha = 1 + 0.5\ (\alpha_s - 0.20)$, adapted from reference (5) page 525

$F_{\alpha I} = 1 + 0.5\ (0.30 - 0.20) = 1.05$

Since

$A_I = 2500\ ft^2$ $\qquad$ $F_{\alpha I} = 1.05$

$\dot{Q}_I = 50\ kW$ $\qquad$ $T_I = 297^oK$

$\varepsilon_I = 0.78$

then

$$A_{II}(ft^2) = \left(\frac{\dot{Q}_{II}(kW)}{50} \right) \left(\frac{0.78}{\varepsilon_{II}} \right) \left(\frac{F_{\alpha II}}{1.05} \right) \left[\frac{297}{T_{II}(^oK)} \right]^4$$

D. Radiator Width

Assuming a length of 50 ft. for each panel, the radiator total width is given by

$$W_R(ft) = \frac{A_{II}(ft^2)}{50}$$

E. Width Per Panel

$$W_p(ft) = \frac{W_R(ft)}{N_p}$$

F. Weight Per Panel

$$m_p(lbm) = \frac{600}{N_p} [12\ W_R - N_p(0.75)](0.0625)(0.1) + 9.5$$

G. Total Radiator Weight (excluding heat exchangers)

$$m_R(lbm) = 600\ [12\ W_R - N_p(0.75)](0.0625)(0.1) + 9.5\ N_p$$

H. Total Radiator Volume

$$V_R(ft^3) = (50)(W_R)(0.0625)$$

These equations have been incorporated into a candidate subroutine in the thermal control system analysis program.

INPUT DATA REQUIRED:

Radiator rating (kW)

Radiator average surface temperature (^oK)

Heat pipe working fluid

Heat pipe operating temperature (^oK)

Working fluid transport number (W/m^2)

Number of layers of course mesh in slab, layer thickness and mesh inverse permeability

Number of layers of fine mesh in slab, layer thickness and mesh inverse permeability

Pore radius for mesh in evaporator (m)

Effective transport length for working fluid (ft)

Emissivity of radiator surface

Absorptivity of radiator surface

OUTPUT

Number of panels in radiator

Heat transport per panel

Radiator surface area

Radiator width

Weight per panel

Total radiator weight

Total radiator volume

SUMMARY

The orbiting space station being developed by the National Aeronautics and Space Administration will have many thermal sources and sinks as well as requirements for the transport of thermal energy through large distances. The station is also expected to evolve over twenty or more years from an initial design. As the station evolves, thermal management will become more difficult. Thus, analysis techniques to evaluate the effects of changing various thermal loads and the methods utilized to control temperature distributions in the station are essential. The analysis techniques described in the present paper consist of developing a data base for a particular station design and set of operating conditions and using simiulation equations for the various thermal components in the station to compute a new data base for different station designs, operating conditions, and mission parameters. A systems analyst using these techniques can evaluate the effects on mission costs, weights, volumes, and power requirements of changing mission requirements and station thermal operation.

CONCLUSIONS

Analysis techniques including a user-friendly computer program, have been developed which should prove quite useful to thermal designers and systems analysts working on the space station. The program uses a data base and user input to compute costs, sizes and power requirements for individual components and complete systems. User input consists of selecting mission parameters, selecting thermal acquisition configurations, transport systems and distances, and thermal rejection configurations. The capabilities of the program may be expanded by including additional thermal models as subroutines.

REFERENCES

1. "High Efficiency Automated Thermal Control System Study: Systems Preliminary Design Performance Analysis and Specifications - Cabin Thermal Bus," Shuttle Integration and Satellite Systems Division, Rockwell International Corporation, SSD84-0075 (May 1984).

2. Olson, R. M., Essentials of Engineering Fluid Mechanics, Fourth Ed., Harper & Row, New York, 1980.

3. Shapiro, A. H., The Dynamics and Thermodynamics of Compressible Fluid Flow, Vol. 1, The Ronald Press Co., New York, 1958.

4. Sadunas, J., and A. Lehtinen, "H.E.A.T. Contract Preliminary Design Review," Rockwell International to NASA JCS, August 16, 1984.

5. "Space Station Reference Configuration Description," NASA JSC, August 1984.

6. Sadunas, J., and A. Lehtinen, "H.E.A.T. Contract Critical Design Review Data Package", NAS9-16782, Rockwell International for NASA JSC, December 7, 1984.

7. Chang, W. S., "Heat Pipe Startup from the Supercrictical State," Ph.D. Thesis, Georgia Institute of Technology, March 1981.

8. Colwell, Gene T., "Prediction of Dryogenic Heat Pipe Performance," Annual Report for 1975 on NASA Grant NSG-2054, by Georgia Tech to NASA Ames and NASA Goddard, February 1, 1976.

APPENDIX

Appendix I
ASSESSMENT ALGORITHMS

Acquisition Assessment Algorithms for Individual Modules

A. Reliability, Technology Readiness and Pacing Technology Rating for Integrated modules

R_i	$R_{c,a}$
TR_i	$TR_{c,a}$
PT_i	$PT_{c,a}$

For autonomous modules

R_i	Minimum $(R_{c,a}, R_{c,t}, R_{c,r})$
TR_i	Minimum $(TR_{c,s}, TR_{c,t}, TR_{c,r})$
PT_i	Minimum $(RT_{c,a}, PT_{c,t}, PT_{c,r}$

B. Metabolic load

$ML_i = ML_i$ from system configuration file, $i = 1,\ldots,n$

C. Acquisition load

$$AL_i = \sum_{j=1}^{p} (CP_j)_i \; ; \; i = 1,\ldots,n$$

ML_T = sum of AL_i for integrated modules

$ML_R = ML_T$

D. Resupply consumables

$$RC_i = RC_m + (WS_a + WC_a) \; *\left(\frac{AL_i}{CR_a}\right)\left(\frac{RI}{90}\right) \quad \text{for integrated modules}$$

$$RC_i = RC_m + \Big[\sum_{k=e,t,r} (WS_k + WC_k)/CR_k\Big](AL_i)\left(\frac{RI}{90}\right) \quad \text{for autonomous modules}$$

$$RC_k = (WS_k + WC_k)\left(\frac{ML_k}{CR_k}\right)\left(\frac{RI}{90}\right) \; ; \quad k = T,R$$

E. Resupply volume

$$RV_i = RV_m + (VS_a + VC_a)\left(\frac{AL_i}{CR_a}\right)\left(\frac{RI}{90}\right) \text{ for integrated modules}$$

$$RV_i = RV_m + \Big[\sum_{k=a,t,r} (VS_k + VC_k)/CR_k\Big](AL_i)\left(\frac{RI}{90}\right) \text{ for autonomous modules}$$

O. Rejection surface area

RSA_i = RSA_m + rejection surface area for autonomous module (or main rejection system) computed in candidate subroutine; i = autonomous modules and R.

Note: The following costs are FY83 million dollars.

P. Cost of design, development, test and evaluate

$CDTE_i$=($DDTE_a$)/(number of modules having sam acquisition candidate) i = 1,...,n

$CDTE_k$ = ($DDTE_k$)/(number of modules having same k candidate + 1) k = T,R

Q. Cost of flight unit, spares and consumables for initial launch

$$CFU_i = [FU_a + (CSC_a)(\frac{RI}{90})](\frac{AL_i}{CR_a}) \; ; \; i = 1,\ldots,n \text{ (Note 1)}$$

$$CFU_R = [FU_k + (CSC_k)(\frac{RI}{90})] (\frac{ML_k}{CR_k}); \; k = T,R$$

R. Cost of spares and consumables to operate over mission

$$CSC_i = (CS_a)(\frac{MD}{RI} - 1)(\frac{AL_i}{CR_a}); \; i = 1,\ldots,n \quad \text{(Note 1)}$$

$$CSC_k = (CS_k)(\frac{MD}{RI} - 1)(\frac{ML_k}{CR_k}); \; k = T,R$$

S. Integration cost

CI_i = ($CDTE_i$ + CFU_i)(ICF/100); i = 1,...,n and T,R

T. Programmatic cost

CPR_i = ($CDTE_i$ + CFU_i)(PCF/100); i = 1,...,n and T, R

U. Transportation costs for a spares and consumables over mission

$CTSC_i$ = (RC_i)($\frac{MP}{RI}$ - 1)(TCF/1000); i = 1,...n and T,R

V. Transportation cost for flight unit, spares and consumables to operate over initial resupply interval

$CTFU_i$ = (RC_i + LW_i)(TCF/1000); i = 1,...,n and T, R

Note 1:Includes only acquisition system for integrated modules; includes acquisition, transport and reject systems for autonomous modules.

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W. Cost of maintenance for mission

$$CMM_i = (MT_i)\left(\frac{MD}{RI} - 1\right)\left(\frac{MCF}{1000}\right); \; i = 1,\ldots,n \text{ and } T,\ R$$

X. Life cycle cost for mission

$$CLC_i = (CDTE_i + CFU_i + CCS_i + CI_i + CPR_i + CTSC_i + CTFU_i + CMM_i) \; ; \; i = 1,\ldots,n \text{ and } T,R$$

II. Summary Assessment Algorithms

A.
$$\begin{Bmatrix} R_A \\ TR_A \\ PT_A \end{Bmatrix} = \begin{Bmatrix} \text{Minimum } (R_i;\ i = 1,\ldots,n) \\ \text{Minimum } (TR_i;\ i = 1,\ldots,n) \\ \text{Minimum } (PT_i;\ i = 1,\ldots,n) \end{Bmatrix}$$

$$\begin{Bmatrix} R_o \\ TR_o \\ PT_o \end{Bmatrix} = \begin{Bmatrix} \text{Minimum } (R_k;\ k = A, T, R) \\ \text{Minimum } ({}_{R}k;\ k = A, T, R) \\ \text{Minimum } (R_k;\ k = A, T. R) \end{Bmatrix}$$

B. $$ML_A = \sum_{i=1}^{n} ML_i \quad ; \quad ML_o = ML_A$$

C. AAL = Sum of AL_i for autonomous modules
IAL = Sum of AL_i for integrated modules

D. through X.

$$\text{Value}_A = \sum_{i=1}^{n} \text{Value}_i$$

$$\text{Value}_o = \text{Value}_A + \text{Value}_T + \text{Value}_R$$

Nomenclature

AAL	autonomous acquisiton load, kW
ACDF	acquisition candidate data file
AL	acquisition load, kW
ASA	acquisition surface area, ft^2
CDTE	cost of design, development, tst and evaluation, million $
CFU	cost of flight unit, spares, and consumables for initial launch, million $
CI	integration cost, million $
CLC	life cycle cost for mission, million $
CP	cold plate load, kW
CR	candidate rating, kW, from ACDF
CS	cost of spares and consummables for 90 days from ACDF, million $
CSC	cost of spares and consummables to operate over mission, million $
CSI	control system impact, lb
CSP	control system penalty, lb/kW, from MMPF
CTFU	transportation cost for flight unit, spares and consummables to operate over initial resupply interval, million $
CTSC	transportation cost for spares and consummables over mission, million $
DDTE	design, development, test and evaluate cost from ACDF, million $
FU	flight unit cost for initial launch cost from ACDF, million $
IAL	integrated acquisition load, kW
ICF	integration cost factor, %, from MMPF
LV	launch volume, ft^3
LW	launch weight, lb
MCF	maintenance cost factor, k$/hr, from MMPF
MD	mission duration, days, from MMPF
ML	metabolic load, kW
MMPF	mission model parameter file
MT	maintenance time over resupply interval, hr
PCF	programmatic cost factor, %, from MMPF
PR	power required, kW
PRSI	propulsion system impact, lb
PRSP	propulsion system penalty, lb/kW, from MMPF

PSI	power system impact, lb
PSP	power system penalty, lb/kW, from MMPF
PT	pacing technology rating
R	reliability
RC	resupply consumables, lb
RI	ressuply interval, days, from MMPF
RMT	90-day maintenance time, hr, from ACDF
RSA	rejection surface area, ft^2
RV	ressuply volume, ft^3
TCF	transportation cost factor, k$/lb from MMPF
TR	technology readiness
VC	volume of consumables from 90 days, ft^3, ACDF
VS	volume of spares for 90 days, ft^3, ACDF
WC	weight of consumables for 90 days, lb, from ACDF
WS	weight of spares for 90 days, lb, from ACDF

Subscripts

a	acquisition candidate
A	total acquisition system
c	candidate data file value
i	module i
j	cold plate
m	metabolic loop
n	number of modules
o	overall assessment
p	number of cold plates
r	rejection candidate
R	main rejection system
t	transport candidate
T	main transport system