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Space Station Common Module Thermal Management: Design and Construction of a Test Bed

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ABSTRACT

The space station habitats, laboratories, and logistics modules will be uniformly designed as a large cylinder (approximately 15 x 35 feet) with a pressure vessel wall and high-efficiency thermal insulation. The interior and exterior environment will be such that a net quantity of heat must be rejected to space to prevent temperature buildup beyond a habitable range. Interior electrical equipment, refrigeration, pumps, experiments, and the workers' metabolism give rise to thermal energy loads, as well as the incoming solar energy.

In this project, a thermal test bed was designed, simulated, and planned for construction. The thermal system features interior and exterior thermal loads and interfacing with the central-radiator thermal bus.

Components of the test bed include: (a) Body mounted radiator loop with interface heat exchangers (6000 Btu/hr), (b) Internal loop with cabin air-conditioning and cold plates (3400 Btu/hr), (c) Interface heat exchangers to the central bus (13,000 Btu/hr), (d) Provisions for new technology including advanced radiators, thermal storage, and refrigeration. The apparatus will be mounted in a chamber, heated with lamps, and tested in a vacuum chamber with LN2-cooled walls (Sunspot I).

Simulation of the test bed was accomplished using a DEC PRO 350 computer and the software package TK!Solver. Key input variables were absorbed solar radiation and cold plate loads. The results indicate temperatures on the two loops will be nominal when the radiation and cold plate loads are in the range of 25% to 75% of peak loads. If all loads fall to zero, except the cabin air system which was fixed, the radiator fluid will drop below -100 F and may cause excessive pressure drop. If all loads reach 100%, the cabin air temperature could rise to 96 F. The mismatch between heat loads and heat removal capability is likely to be desirable when new technology is tested.

ACKNOWLEDGEMENTS

Because of my lifelong interest in rockets, space travel, and science fiction, the opportunity to work at Marshall Space Flight Center has been an overwhelming fulfillment of old dreams. The program hosts, Gerald R. Carr and Dina Conrad of UAH, and Leroy Osborn and Dr. Jim Dozier of NASA, by their kindness, patience, and consideration have made these ten weeks rewarding and enjoyable. Our tours and seminars, conducted by Gerald, Leroy, and Jim, were especially appreciated.

Day-to-day work was guided by Jim Owen, Dick Wegrich, Joe Sims, Bob Fisher, and Rick Bachtel. To them I offer heartfelt thanks. Friendly and informative discussions were had with Bill Till, Tim Page, and Farouk Huneidi. Laboratory Director Alex McCool and his staff, Paul Frederick, Bob Richmond, Randy Humphries, Harold Smyly, and Doug Lamb were helpful and kind beyond expectation. Finally, special thanks is due to Betty Golden, Jeanette Hall, and Bernice Bowling for helping with all the "little things" which kept my work running smoothly each day, and to Brenda Wade and Sue Madole who typed this report.

INTRODUCTION

Pressurized compartments in the space station, where astronauts live and work, will be subjected to various thermal loads. Internally, heat is dissipated to the atmosphere from all electrical and mechanical equipment, lights, experiments, and even human metabolism. Externally, the Sun and Earth will radiate energy to the station, while the outer walls will radiate to deep space. The external heat sources and sinks will vary on orbit and seasonally, as well as differences between individual compartments due to shading and view factor variability. Life support tolerances are such that an active thermal management system will be designed into each habitated unit or common module. Heat from the physical plant and various experiments will be transported by fluid (or liquid-vapor) loops to body-mounted radiators and to a central thermal bus. The central bus is an external loop which will transport heat to the central radiators where heat is rejected to space.

As phase B space station work is now commencing, there is a need for NASA in-house thermal management test bed facilities. The present work was initiated by J.W. Owen in 1984 to design and construct a common module test bed at Marshall Space Flight Center. The objective was to provide an early in-house test capability for thermal systems which support the MSFC work package. His approach was to integrate existing hardware, e.g., from skylab spare parts, into a test unit which simulates the essential functions of the common module thermal transport and heat rejection loops. Then, the system would be updated with new or advanced technology elements as they become available. The latter include body-mounted radiators, thermal storage, refrigeration, thermal bus, long-life fluid systems, heat transport across structural boundaries, advanced heat pipes, self-maintaining thermal surfaces, electrochromic panels, conjugating binary systems, microencapsulated phase-change material, metal hydrides, and conformal heat exchangers.

The test bed elements will be situated outside the Sunspot I thermal vacuum chamber. These elements include cabin air heat exchangers, cold plates, pumps, controls, data acquisition, and a central-bus interface heat exchanger, simulated by a facility water loop. Equipment and other interior loads will be simulated by electrical heaters. Inside the chamber, there will be two body-mounted radiators, radiator bypass and mixer control valve, and an interior-exterior interface heat exchanger. The radiators will be exposed to high vacuum, liquid nitrogen-cooled walls, and heater lamps. Plumbing provisions will be made for both interior and exterior advanced-technology elements. Specific plans, equipment details, and test procedures are summarized in an internal NASA report (1).

OBJECTIVES

The ultimate objectives of this project are to prepare a test bed and evaluate advanced thermal systems technologies for the space station common module. The immediate objectives of the present work are:

1. Design a thermal test bed representing the essential elements of the common module.

2. Specify and select equipment for the test bed.

3. Simulate the performance under various loads with calculations.

4. Prepare for construction and testing by making arrangements with supporting groups.

CONSTRUCTION AND SIMULATION

A. CONSTRUCTION AND TEST PLAN

Much of the equipment for the test bed is already in storage at MSFC. Leftover from ATM and skylab are pumps, valves, sensors, filters, and heat exchangers (1). Design specifications of these elements are available (2). Two radiators, double sided, 101.4 ft total area and several cold plates are also available. These were constructed by MDAC and TRW in earlier contracts. The general configuration of these elements is shown in figure 1. Infrared lamps are to be situated around the radiators to simulate variable orbital conditions. The lamps are to provide 0-17.3 w/sq ft controllable, evenly distributed heating. This is equivalent to 0-6000 Btu/hr insulation for the specified radiators. Inside the radiators a Coolanol 25 fluid will be used (Monsanto). The other loops will contain water. Interface heat transfer from the internal water to the thermal bus will be simulated by a facility water loop. General specifications and parameters for the test bed elements are shown in table I.

Requests for removing equipment from storage to building 4619 have been initiated. Elements will be selected and prepared for shipment to Lockheed, Huntsville. Miscellaneous items will be procured by Lockheed, then they will construct the apparatus and perform preliminary tests. Then, the test bed will be shipped to building 4619 to prepare for tests in the Sunspot facility when it is available.

A description of the test facility requirements and of the tests is found in reference (1). As indicated in figure 1, the radiators and radiator-to-water interface (HX1) will be situated inside the vacuum chamber. Cold plate loads and lamp power will be varied to simulate a range of internal and orbital conditions. As new technology concepts and hardware become available, they will be incorporated into the tests.

B. TEST BED FLOW SHEET

Nominal conditions, fluid flows, heat loads, and heat flows are shown in figure 2. Coolanol 25 is recommended for the radiator fluid. It has kinematic viscosity of 12 Centistokes and density of 0.91 g/cc at 0 F. Half-inch ID tubing is adequate for this service. However, at -50 F, it

TABLE I

Equipment Specifications

Radiators: Two units, 11 ft by 2 2/3 ft and 5 ft by 2 2/3 ft, double sided. Total area = 101.4 sq ft. Approximate heat rejections up to 7000 Btu/hr. Coolanol 25 fluid.

Heat Exchanger 1: HX1 consists of two units in series, originally designed for ground service to Airlock, ATM and suit cooling. Original rating was 17,700 Btu/hr each, at 183 lb/hr each (UA = 258 Btu/hr. - sq ft - F each). Present application: 1,700 Btu/hr at 183 lb/hr for each of two units. Part no. 52-83700-1202, ref. (2), pg. 25.0.

Heat Exchanger 2: HX2 consists of two cabin air heat exchangers in parallel water flow. Original rating was 680 Btu/hr each at 88 CFM each, 5 psia oxygen (UA = 48.1 Btu/hr - sq ft - F each). Present application: 1025 Btu/hr each. Part no.: 52-83700-1227, ref. (2), pg. 32.0.

Heat Exchanger 3: HX3 consists of 5 units connected in series. The units are the same as HX1 units above. The present application is 13,000 Btu/hr transfer from loop water (520 lb/hr) to facility water (550 lb/hr, 65 F). Facility water represents the central bus.

Cold Plates: Cold plate 1 is nominally 0.4 kw. Cold plates 2 through 5 are nominally 0.95 kw each, or a total of 3.8 kw. Several devices are potentially available, e.g., see reference (3) for a candidate. The cold plates must be fitted with electrical heaters which simulate equipment heat loads.

Pump Systems: This pump system provides pump, accumulator, low fluid and power indicators, and a fill port. The pumps are rated for "coolant" at 183 lb/hr with 175 psi pressure rise. Part nos. 52-83700-831, 833, and 869, ref. (2), pp. 19.0-21.3.

Radiator Mixer Valve: Details of the valve selection were included in a separate communication (5). Two options were considered: (a) Proportional-integral control using a proportional valve driven by a dc gear wound motor, and (b) stepper control using 5 parallel tubes fit with orifices of different openings and 4 on-off solenoid valves to select flow paths.

will thicken to 200 Cs and 0.95 g/cc which may make it unpumpable. The original fluid on skylab, Coolanol 15, is less viscous but it is no longer made. Water flow at 520 lb/hr and 70 F is handled by 1/4 or 3/8 inch ID tubing (6.8 and 3.0 ft/s, respectively).

The lamps surrounding the radiators are designed for 0-17.3 w/sq ft, which is 0-6,000 Btu/hr total incident radiation on the double surfaces. The lamps have a narrow profile and will not greatly affect outward radiation. An estimate of the total radiator rejection power is 2 kw or 7,000 Btu/hr. Part of the Coolanol will bypass the radiator, a fraction $\underline{f}$, and mix with the cooled fluid as shown in figure 1. The mixed outlet design temperature is 40 F. This will be accomplished by a controlled mixer valve to be discussed below. The rejection heat load on the radiator is that absorbed from the lamps and the heat picked up in heat exchanger 1 (HX1). The cabin air cooling load is roughly 2050 Btu/hr. This plus cold plate (CP-1) load totals 3420 Btu/hr, the nominal load for HX1 which is transferred to and rejected by the radiators. Cold plate 1 has nominally 0.4 kw load, while cold plates 2-5 have 0.95 kw each; i.e., the total load for cold plates 2-5 is 3.8 kw or 13,000 Btu/hr. The heat taken up in cold plates 2-5 is primarily rejected in HX3 to facility water. This heat exchanger represents the space station central bus interface.

An extra heat exchanger, HX on figure 1, is tentatively planned for the situation where heat loads are greater than the radiator rejection capability. Up to 3500 Btu/hr may be rejected in HX, in which coolant service at 20 F is required. This heat exchanger can also function as a ground service exchanger.

C. SIMULATION OF TEST BED PERFORMANCE

Equations describing heat transfer in the bed are given in table II. The 20 equations are listed in order of solution by TK!Solver:

Heat exchanger 2, Eqns. HX2 1-4

Radiator and control valve, Eqns. RAD 1-5

Heat exchanger 1, Eqns. HX1 1-4

Heat exchanger 3, Eqns. HX3 1-3

Cold plate 1, Eqns. CP1 1

TABLE II
Standard Model

	S	Rule
HX2	1	q2=114*(tw2-tw1)
	2	q2=fa*2*pa*0.6489*60*(ta1-ta2)/(ta1+460)
	3	q2=2*48.1*dt2
	4	dt2=((ta1-tw2)-(ta2-tw1))/ln((ta1-tw2)/(ta2-tw1))
RAD	1	qrn=wc*cpc*(tc2-tc3)
	2	qrr=qrn+qra
	3	tr=((qrr/ar/3.413-17.55)/0.17+35)
	4	tc1=2*tr-tc2
	5	f=(tc3-tc1)/(tc2-tc1)
HX1	1	q1=wc*cpc*(tc2-tc3)
	2	dt1=q1/516
	3	tw4=q1/114+tw1
	4	dt1=((tw4-tc2)-(tw1-tc3))/ln((tw4-tc2)/(tw1-tc3))
HX3	1	q3=ww5*cpw*(tw6-tw4)
	2	tf2=q3/wf+tf1
	3	q3=1290*((tw6-tf2)-(tw4-tf1))/ln((tw6-tf2)/(tw4-tf1))
CP1	1	tw3=qcp1/ww1+tw2
TEE	1	tw5=(ww1*tw3+ww2*tw4)/ww5
CP2-5	1	qcp=qcp2+qcp3+qcp4+qcp5
	2	qcp=ww5*cpw*(tw6-tw5)

C-2

Tee, Eqn. TEE 1

Cold plates 2-5, Eqns. CP2-5 1-2

The calculation scheme is to march through the equations in order and solve for one unknown in each equation. Since the equations are highly coupled, initial guesses of certain output variables are required. Also, input variables are required so that the problem is uniquely specified, i.e., total variables (39) minus input variables (19) equals equations (rules). A typical output list from TK!Solver is shown in table III. This case is for radiator and cold plate loads at 50% level, including:

qra	radiation absorbed by radiator
qcp 1	heat absorbed from cold plate 1
qcp 2, 3, 4, 5	heat absorbed from cold plate 2, 3, etc.

These variables are marked as "INPUT" on the computer screen and on the output list ("Comment Column"). Similarly marked is the variable tc3 which is the temperature set point for the control valve outlet downstream from the radiator, normally 40 F. Variables which are marked "Guess" require a G in the "St" column and a new guess in the "Input" column before each run. The "Comment" and "Unit" columns contain the nomenclature list. (HX1, 2, and 3 represent the heat transferred in the respective heat exchangers.)

Several types of relationships are represented in table II:

1. qi = w x cp x (tj - tk)

   Stream heat balance: Eqns. HX2-1,2, HX1-1,

   HX1-3, HX3-1,2, CP1-1, CP2-5-2

2. qi = U x A x dti

   (Where dti is log mean temperature difference)

   Heat exchanger design equation: Eqns. HX2-3,

   HX1-2, HX3-3

3. dti = ((ta1-tw2)-(ta2-tw1))/ln((ta1-tw2)/(ta2-tw1))

TABLE III
Standard Model Results
50% Heat Load

St	Input	Name	Output	Unit	Comment
--	-----	----	------	----	-------
	2050	q2		Btu/hr	cabin air load HX2
		tw2	60.702381	F	water out HX2
		tw1	42.719925	F	GUESS water out HX1
	88	fa		cfm	cabin air (1 unit)
	16	pa		psia	air inlet to HX2
		ta1	78.298013	F	GUESS air inlet to HX1
		ta2	68.232985	F	air out HX2
		dt2	21.309771	F	log mean del t, HX2
		qrn	3050.3357	Btu/hr	net rad ht (+ = out)
	366	wc		lb/hr	Coolanol flow
	.45	cpc		Btu/lb-F	Coolanol ht cap
		tc2	58.520557	F	GUESS inlet Cool. temp
	40	tc3		F	INPUT contr valve out
		qrr	6050.3357	Btu/hr	ht rej by rad
	3000	qra		Btu/hr	INPUT ht abs by rad
		tr	34.603342	F	avg rad temp
	101.4	ar		sq ft	total rad area
		tc1	10.686128	F	out rad Cool. temp
		f	.61281953	none	fract of flow bypass rad
		q1	3050.3357	Btu/hr	HX1
		dt1	5.9115034	F	log mean del t, HX1
		tw4	69.477256	F	water out HX3
		q3	6184.6643	Btu/hr	HX3
	520	ww5		lb/hr	water flow
	1	cpw		Btu/lb-F	ht cap water
		tw6	81.370841	F	GUESS water in HX3
		tf2	76.244844	F	facility outlet temp
	550	wf		lb/hr	facility water flow
	65	tf1		F	facil inlet temp
		tw3	66.711153	F	water out cp1
	685	qcp1		Btu/hr	INPUT cold plate 1
	114	ww1		lb/hr	water flow to hx1
		tw5	68.870841	F	water flow to cp2-5
	406	ww2		lb/hr	water flow to cp2-5
		qcp	6500	Btu/hr	cp2-5 total load
	1625	qcp2		Btu/hr	INPUT cp4 load
	1625	qcp3		"	INPUT cp3 load
	1625	qcp4		"	INPUT cp4 load
	1625	qcp5		"	INPUT cp5 load

Log mean temperature difference: Eqns. HX2-4, and HX1-4

4. qrr = qrr + qra

   Radiation balance: Eqn. RAD-2

5. tr = ((qrr/(3.413/ar) -17.55)/0.17+35)

   Radiator model for heat rejection (based on calculations in ref. (4)): Eqn. RAD-3

6. tr = 2 x tr - tc2

   Average radiator temperature: Eqn. RAD-4

7. f = (tc3 - tc1)/(tc2 - tc1)

   Stream mixing point balance: Eqn. RAD-5

   and

   tw5 = (ww1 x tw3 + ww2 x tw4)/ww5

   Eqn. TEE-1

8. qcp = qcp2 + qcp3 + qcp4 + qcp5

   Cold plate heat input sum: Eqn. CP2-5-1

The radiator model (Eqn. RAD-3) was obtained by curve fitting the results of a detailed calculation in ref. (4). The heat rejected, qrr, is simply a function of tr, the average radiator fluid temperature (Eqn. RAD-3). Emissivity and efficiency are included in the constants. This model is not directly sensitive to changes in coolant flow rate. But, it will predict very cold outlet temperature when the coolant bypass rate is around 90%, i.e., when loads are small. A better control policy for small loads may be to increase the radiator coolant flow and apply strip heaters at the inlet.

Design equations (Eqns. HX2-3, HX1-2, and HX3-3) contain an overall heat transfer parameter, UA, Btu/hr-F. This value was computed from performance temperature data (2). Since the test bed conditions differ from the original services for these exchangers, there should be some variability in UA values. No attempt was made to correct for flow and temperature level. These parameters will be more accurately determined from tests. Values for UA's in table II are:

HX2 UA = 2 x 48.1 Btu/hr -F

HX1 UA = 2 x 258 Btu/hr -F

HX3 UA = 5 x 258 Btu/hr -F

The simulation scheme was to allow heat exchangers in HX1 and HX3, q1 and q3, to vary with the level of radiation and cold plate loads. However, the cabin air load was assumed fixed, q2 = 2050 Btu/hr. The model will solve for cabin air temperature, ta1, and air outlet temperature, ta2. In this way, the value ta1 indicates whether or not the ECL standard is being met. Facility water input to HX3 is assumed constant at 550 lb/hr, 65 F. This will reasonably approximate the test situation but will not simulate a two-phase central bus. The model will require slight modification of the HX3 equations in the latter case.

Once a model is run on the computer, there are three ways to check for consistency:

1. qrn = q1

2. $0 \leq f \leq 1$

3. q2 + qcp1 + qcp = q1 + q3

If any of these equations are violated, then the simulation is in error.

During high-load situations, the radiator will not dissipate enough heat to retain tc3 at 40 F. A signal for this condition is a value of <u>f</u> outside its range of 0 to 1. When this happens, the model and the variables must be changed slightly. The model Eqn. RAD-5 is replaced by:

tc3 = tc1

The variable change is:

Input <u>0.0</u> for <u>f</u>

As a result of these changes, all Coolanol will flow through the radiator and the outlet temperatures (tc1 = tc3) will be calculated and will be above the set point of 40 F.

Several case studies were run to encompass a range of conditions to be tested. Figures 3, 4, and 5 show the results of loop simulation with TK!Solver for radiation and cold plate loads of 0, 25, 50, 75, and 100%. Temperatures at various positions are listed, as well as the Q1, Q2, and Q3 (heat transferred in HX1, HX2, and HX3) values. Also, the calculated or specified value of f is given at the radiator bypass line. Loads of 75 and 100% were calculated as overloads with f specified as 0.0. Three closer-mesh runs, 55, 60, and 65% load, were run to converge on the load value which just drives f to zero (figure 5 shows 55% case, only). The effect of load on f is plotted in figure 6. The standard model is valid up to 63% load where the curve crosses f=0. Note on the bottom of figure 6, the 65% case causes f=-0.39 using the standard model. The negative result is physically impossible, indicating that the overload model must be used instead.

Both the standard and overload models were used to produce figure 7, which represents various loop temperatures vs. % load. TC1 and TC3 show radiator outlet and mixer valve outlets. These two converge at 63% load. Beyond this point, they physically must converge because the radiator bypass is closed. In the lower portion of figure 7 are plotted TW1, the lowest water temperature in the loop (inlet to cabin heat exchanger), TW6, the highest water loop temperature (outlet from cold plates 2-5), and TA1, the cabin air temperature inlet to HX2. TW1 and TW6 show no problem, but TA1 indicates an excessive cabin air temperature as loads increase beyond 63%.

Another problem, at low load, is indicated by the very low temperatures of TC1. The Coolanol fluid will become very viscous below -50 F and probably become a gel-like fluid. This problem might be avoided by operating the mixer valve to permit no more than 75% bypass, i.e., $f \leq 0.75$. A "lowload" model was constructed by simply setting f=0.75 (input) and removing 40 F as input for TC3 (i.e., remove 40 in table for TC3; and, input 0.75 for f in table III). This results in TC1 rising to -49 F at zero load and -27 F at 25% load, but water may freeze in parts of the loop and the cabin air outlet falls as low as 54 F (TA2). These difficulties could all be avoided by (a) heating the radiator inlet line with a strip heater when TC1 falls below about -25 F, or (b) providing another heat exchanger at the radiator inlet to use the high side of the central bus (facility inlet) to preheat the Coolanol.

The mixer valve control algorithm naturally results from loop simulation. A feasible policy is to use proportional-integral modes to control the valve position with the set point being TC3=40F. During overload, the set point cannot be reached, and the valve will be closed. At low loads, when TC1 drops below -25 F, turn on an electrical heater at a fixed power, e.g., 1 kw and position the valve at the maximum bypass setting of f=0.75. This will both prevent the Coolanol and the water loops from freezing, as well as keep the cabin temperature in a reasonable zone. Alternatively, the loop conditions probably should not be set below 25% radiation load since this is not a realistic orbital situation.

CONCLUSIONS AND RECOMMENDATIONS

Conclusions: The thermal test bed has been designed, planned, and preparations are in progress to construct and test it. Simulation results indicate that the loop is well behaved and existing skylab equipment should be adequate. The heater lamps and equipment simulators (attached to cold plates) are capable of overloading the loop on the hot side. Also, the radiator loop may freeze at very low lamp power.

Recommendations: Parameters used in the simulation should be substantiated in single unit tests, especially UA values for the heat exchangers, and the radiator performance. Measured parameters should be inserted in the loop simulation for more accurate prediction of overall performance. Details of mixer valve design, reported separately, present two alternatives, with the simplest approach being a proportional valve and P-I controller. Other test situations may be simulated later if desired. A transient model of the loop should be formulated and solved before final detailed design is completed.

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3. G.L. Brown, "Test Plan - Cold Plate ORU - Thermal Vacuum Test", TRW Report No. L015.16-83-02, February 18, 1983.

4. J.R. Schuster et al., "Space Station Thermal Integration Study," Final Report, NAS8-35624, Apr., 1985, pgs. 2-10 to 2-12.

5. Private communication to J.W. Owen, EP44, Marshall Space Flight Center, AL

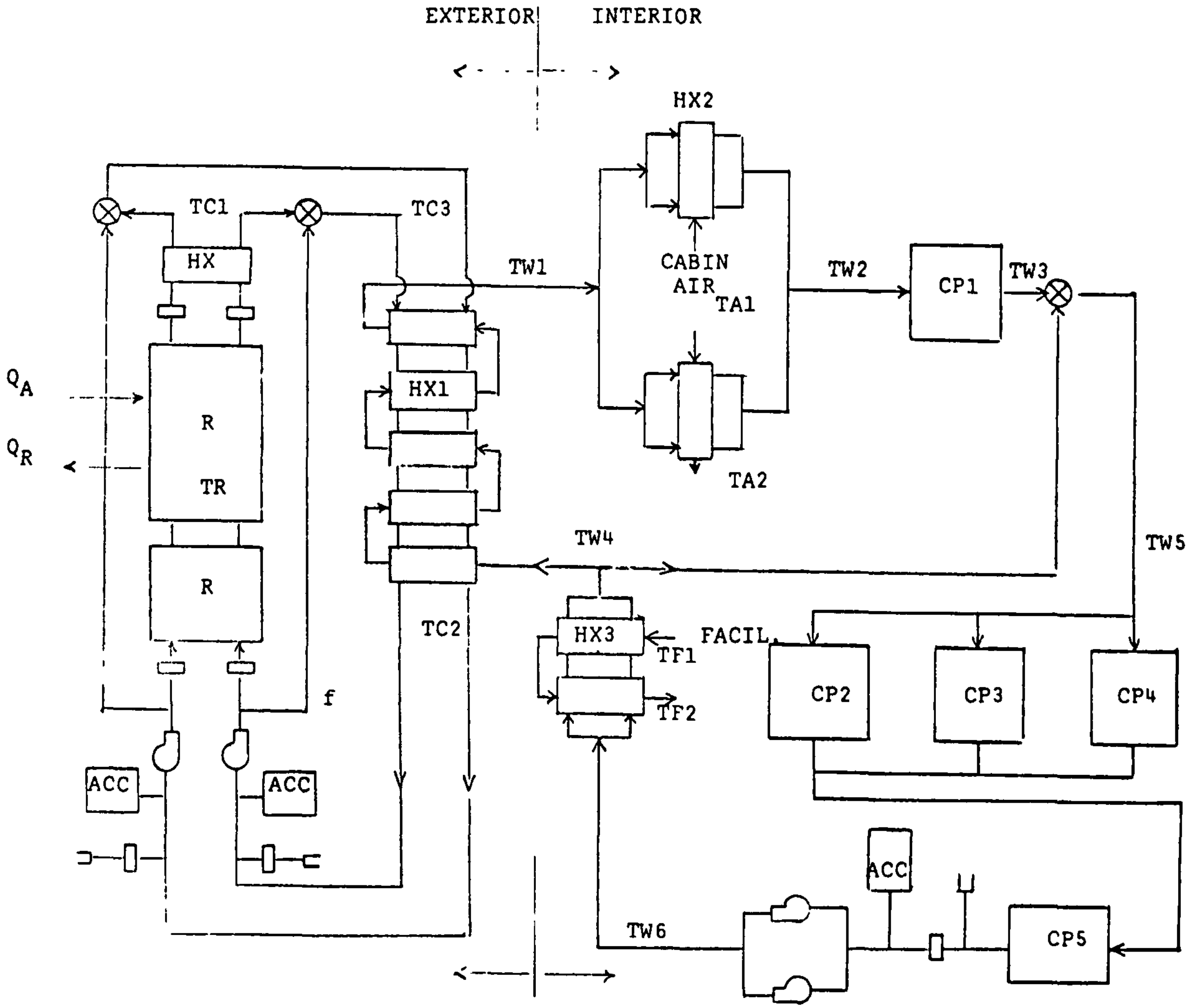


Figure 1. Test bed loop. Symbols correspond to Standard Model, table II.

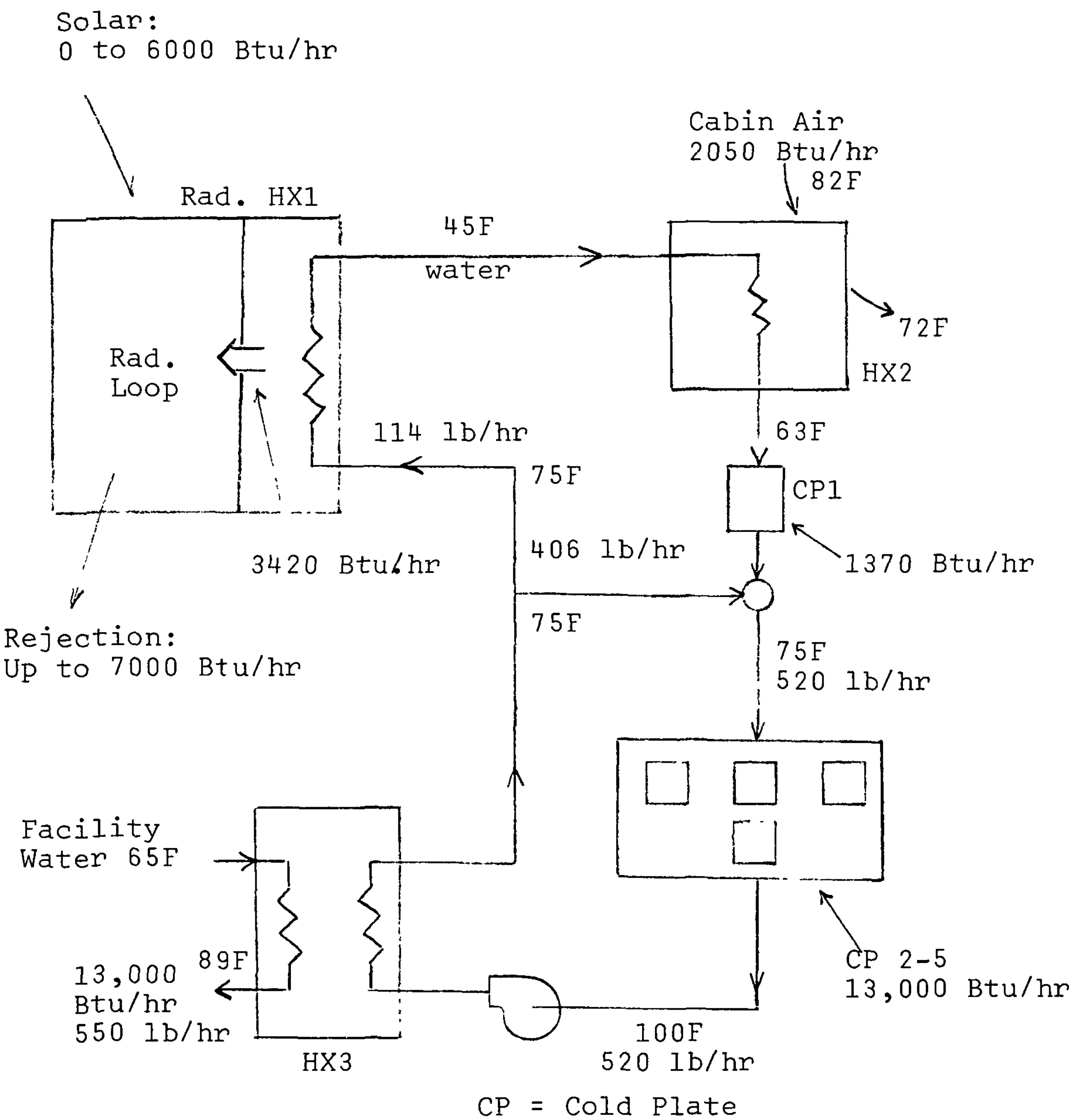


Figure 2. Example of conditions and heat flows in test bed.

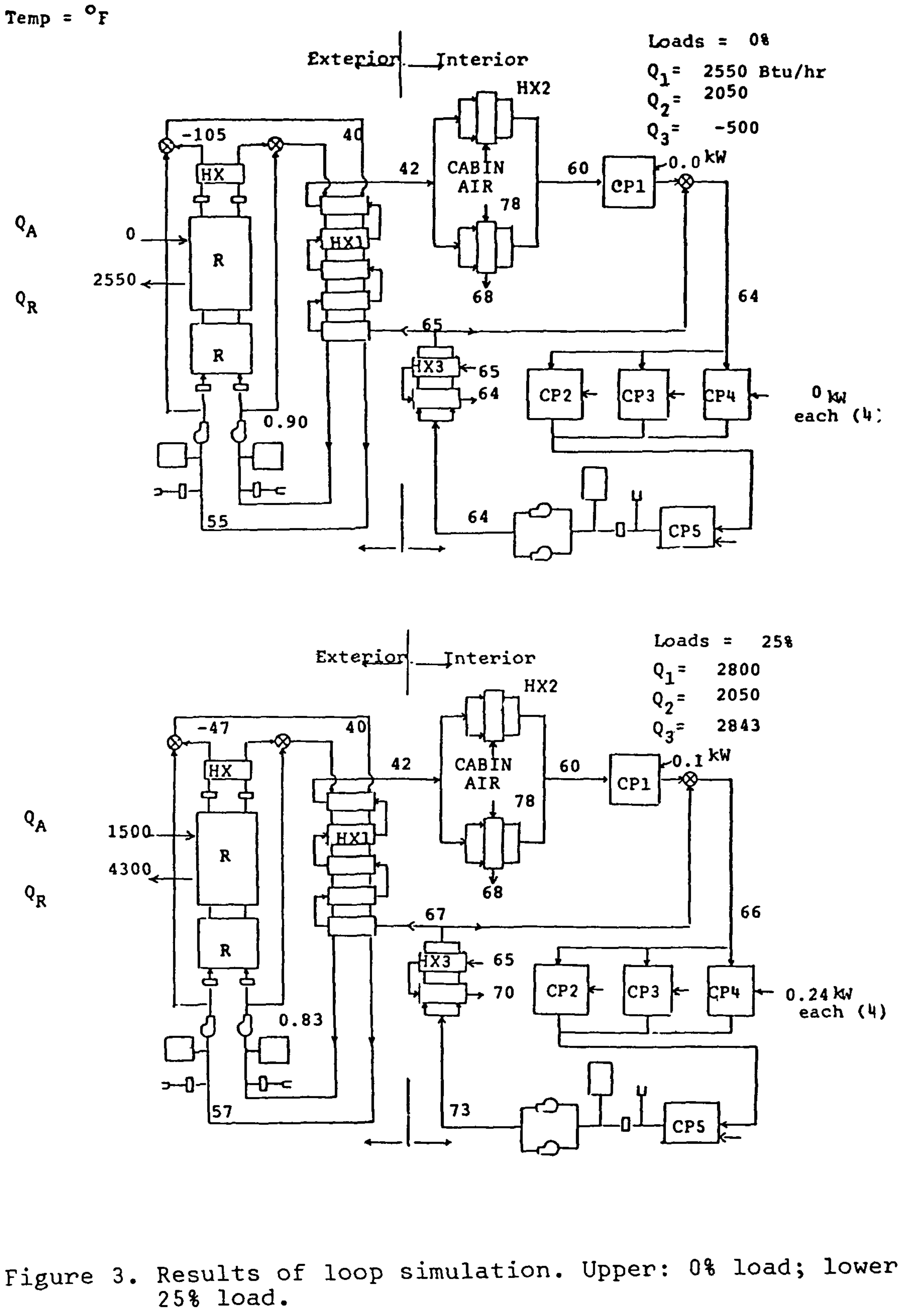


Figure 3. Results of loop simulation. Upper: 0% load; lower: 25% load.

Temp = °F

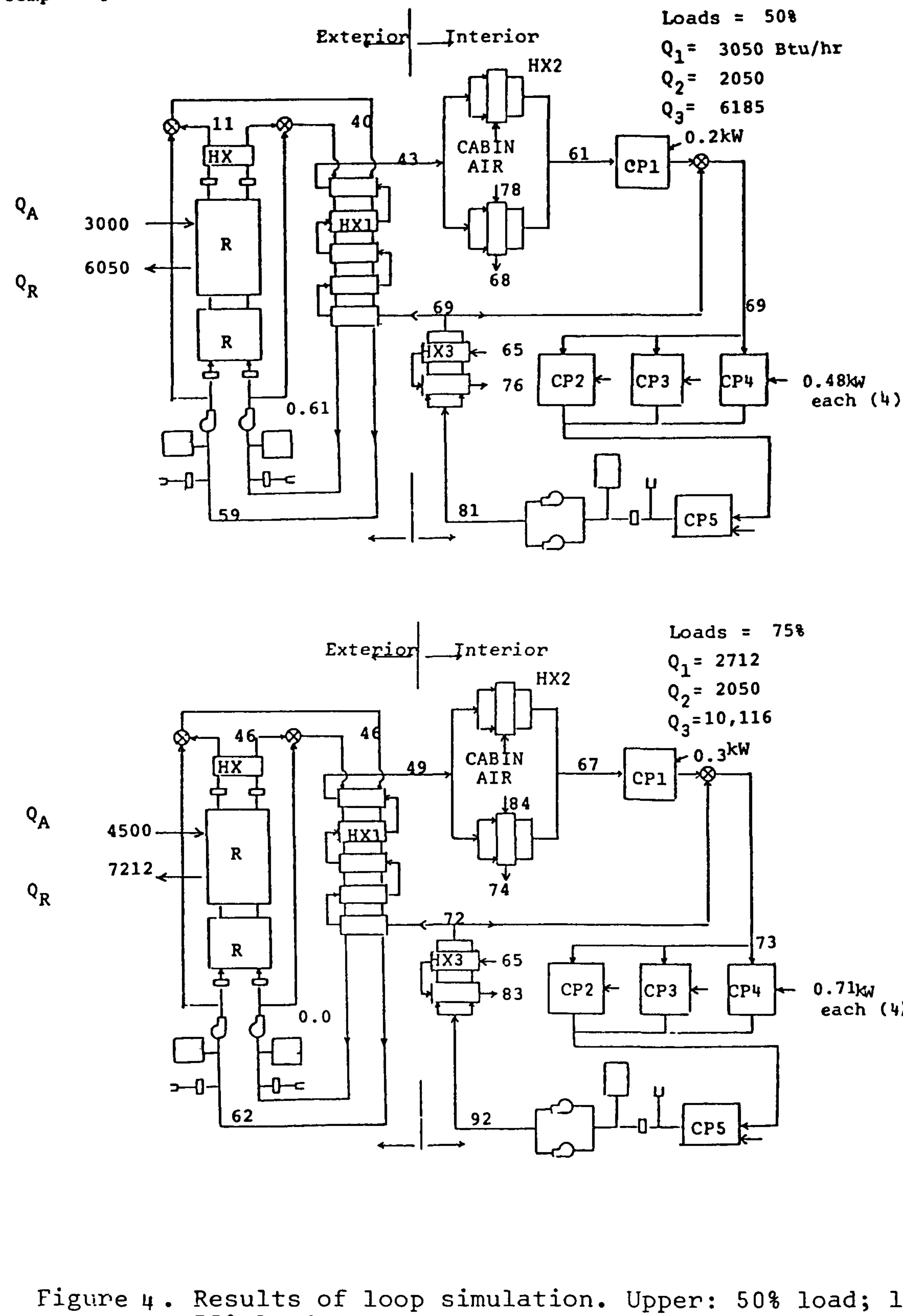
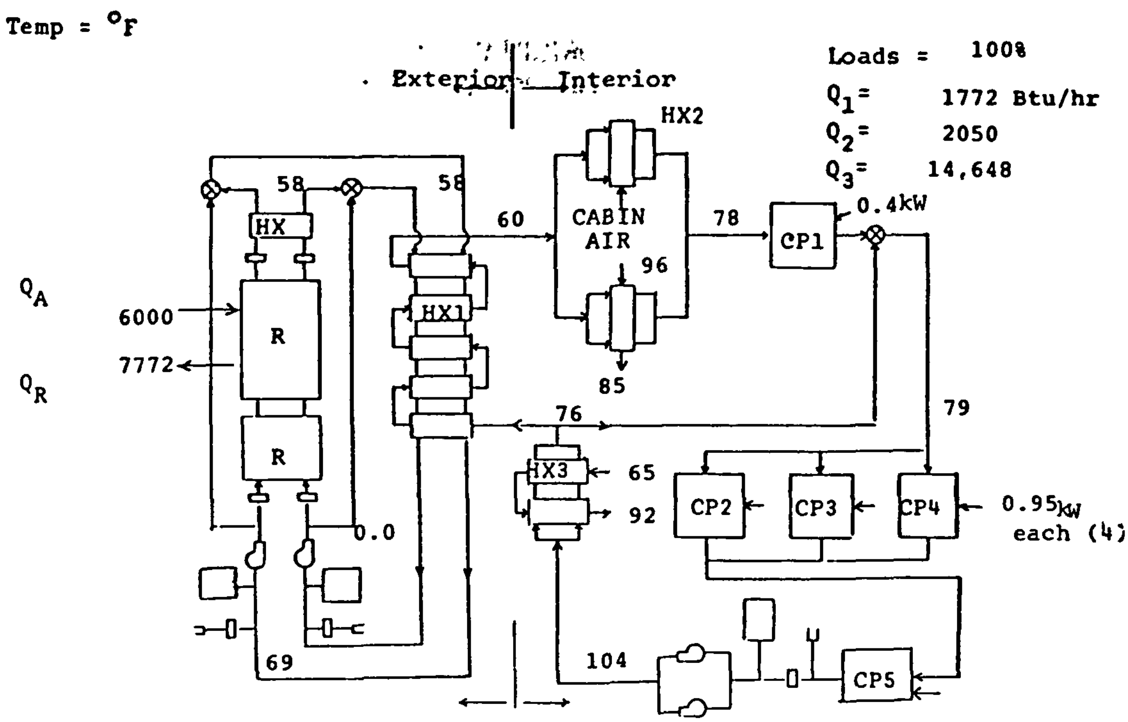


Figure 4. Results of loop simulation. Upper: 50% load; lower: 75% load.



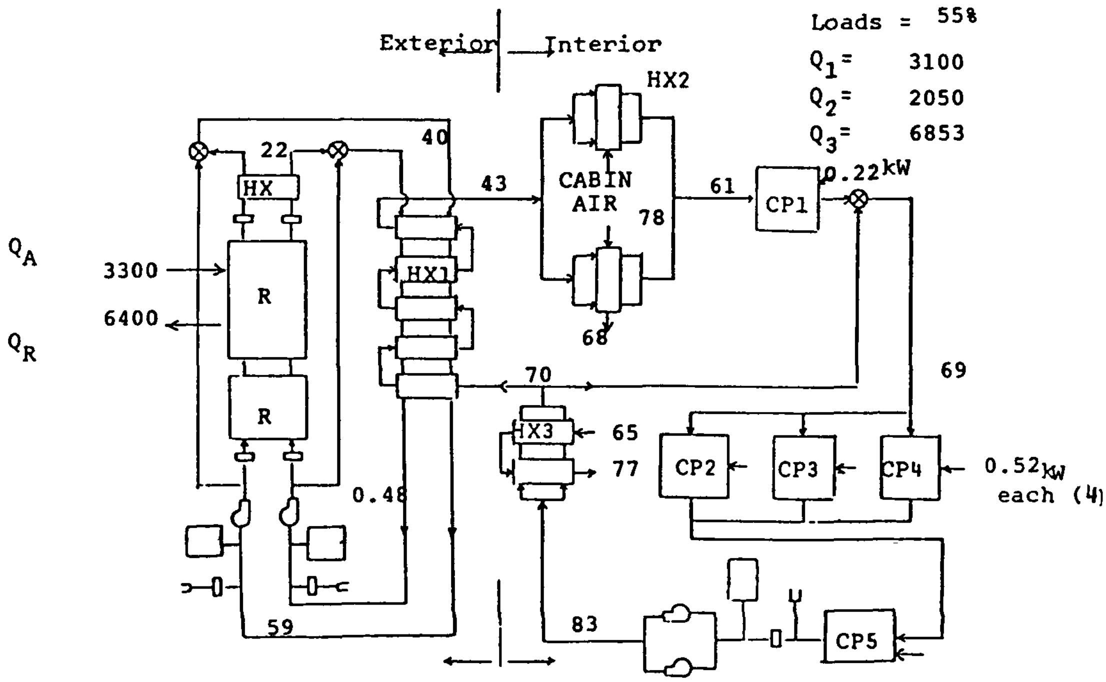


Figure 5. Results of loop simulation. Upper: 100% load; lower: 55% load.

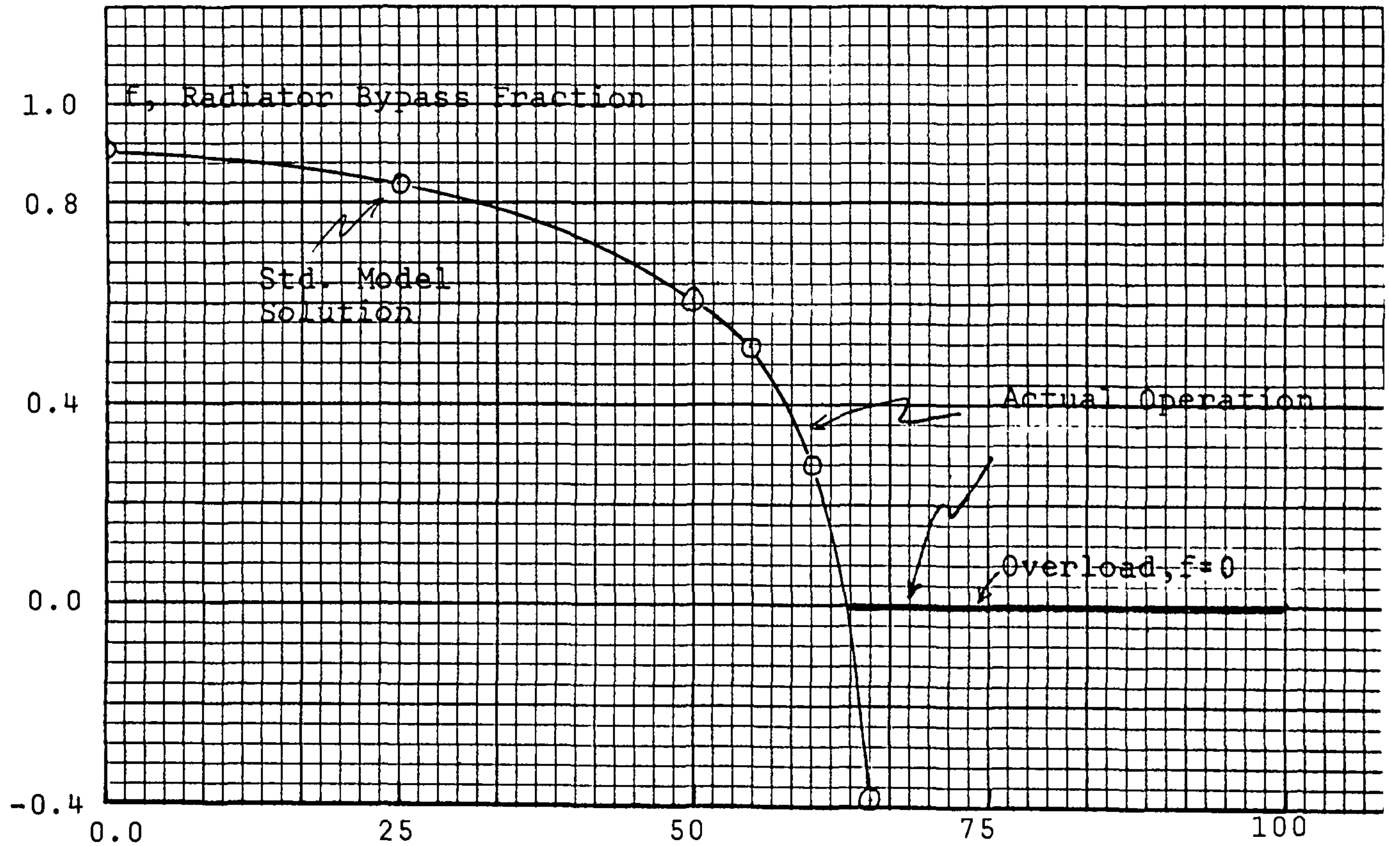


Figure 6. Standard Model solutions for f vs. % heat load.

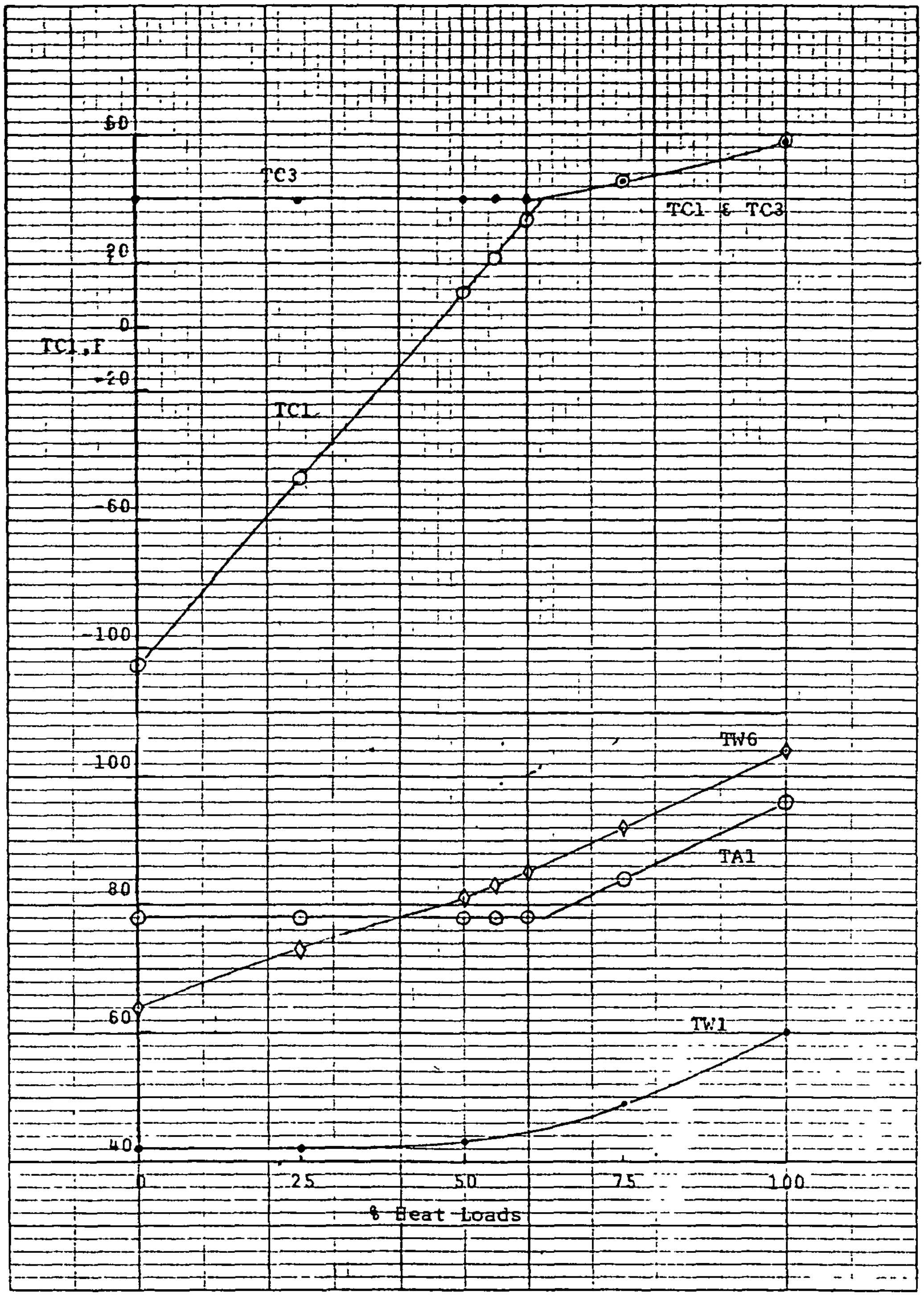


Figure 7. Typical loop temperatures vs % heat load.