Accession No.

SID 67-209

QUARTERLY REPORT NO. 3
(Nov 1966-Jan 1967)
A STUDY OF ELECTRONIC PACKAGES ENVIRONMENTAL CONTROL SYSTEMS AND VEHICLE THERMAL SYSTEMS INTEGRATION
(Contract NAS 8-20320)

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FOREWORD

This document is submitted by the Space and Information Systems Division (S&ID) of North American Aviation, Inc. to the George C. Marshall Space Flight Center (MSFC) of the National Aeronautics and Space Administration in accordance with Contract NAS 8-20320. This report summarizes the study activity conducted during the third quarter of the contract.

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ILLUSTRATIONS

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TABLES

1.0 INTRODUCTION AND SUMMARY

This study is directed toward defining environmental control system (ECS) concepts to meet the increased complexity and changing thermal conditioning requirements of the electronic equipment on the Saturn V vehicle for missions of durations varying from four and one-half hours to one-hundred eighty days. The study objective is to establish the optimum environmental control concepts for thermally conditioning individual electronic packages. Essential characteristics of the optimum environmental control systems are maximum practicable reliability, maximum operating range, minimum weight, minimum power, and minimum volume. The optimum systems are to be selected on the basis of future Saturn mission requirements, future trend in the design and development of the electronic equipment and being an integral part of the overall vehicle thermal system.

The results of the study effort will help to define the developments and advancements required in the technology of thermal control methods and systems. Critical areas or pacing items for development are to be identified. This should help to insure the necessary development to be accomplished and on a timely basis. Other results from the study include design guidelines for electronic packages to achieve optimum environmental control designs and for integrated ECS and vehicle thermal control systems.

Longer mission duration and greater heat dissipation requirements together with closer temperature regulation require thermal systems in which the control method is self regulating and is capable of operating under widely varying conditions. The simple passive system cannot meet these requirements. The study effort is directed toward the investigation of system concepts which can function reliably within widely varying conditions. These systems concepts, like the simple passive system, will depend upon rejecting waste heat to space, but the means of transporting heat from the source to the radiating surface and the surface itself will be quite different. The systems will be more complex than the pure passive approach and thus it becomes a challenge to achieve practical, simple and reliable systems. Many alternate concepts are to be investigated and the most promising ones are to be studied in depth.

SUMMARY

During this reporting period, Tasks 1, 2 and 3 have been completed and the major effort being devoted to Task 4 and 5. Figure 1 indicates the schedule position.

During the quarter, the major effort has been devoted to establishing the heat balances of the combined astrionic equipment and the thermal conditioning subsystem of the environmental control system for various heat loads, environmental conditions, values for surface finishes and other significant parameters. This is a continuing effort for which the results will be presented in a parametric form to provide a readily usable information for the selection of the system concept and operating conditions for given requirements.

In addition, the heat balance of a electronic package mounted on the instrument unit structure was investigated to determine the feasibility of a purely passive thermal control approach. The results to date indicate that for the orbital conditions that were considered, the passive approach does appear to be feasible for those electronic packages that have a wide temperature tolerance. Further analysis will be conducted on a parametric basis to establish the range of conditions for which the purely passive approach is feasible. In addition to the direct skin mounted arrangement, the case of the equipment mounted on the coldplate with no coolant flow is also being investigated.

The results of the above analyses will be used to define the heat rejection concepts for the various orbital conditions and mission durations. For the shorter mission duration up to 24 hours, the present expendable heat sink (sublimator) appears to be feasible. For the longer missions, the use of a recycle method which requires a space radiator as a heat sink appears necessary.

The mission and vehicle models to be used in the study were finalized which consists of various combinations of near earth and synchronous earth orbits launch and vehicle configurations. The vehicle configurations are basically the instrument unit attached to the S-IVB with various configurations mounted on the forward end of the instrument unit.

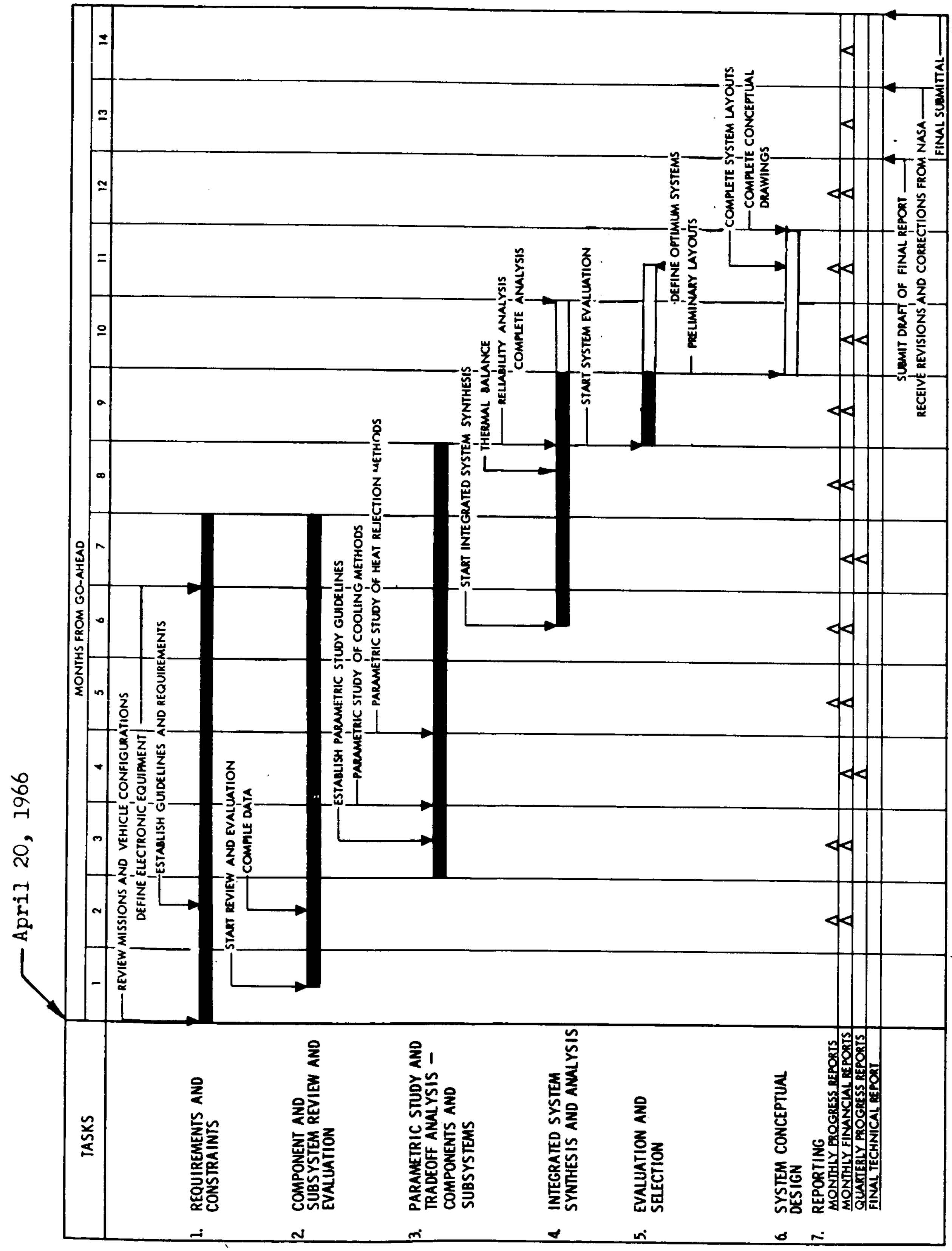


Figure 1. Program Schedule

2.0 CURRENT STUDY STATUS

2.1 MISSION/VEHICLE MODELS

The mission and vehicle models to be used in the study were finalized along with the mission profile. These models are considered to be representative of the future Saturn missions. Near earth circular orbits at 100 and 200 nautical miles, synchronous earth orbit, and orbital launch from a 200 nautical mile parking orbit were selected as the basic orbital characteristics, and these are illustrated in Figure 2. For each of these basic orbits, the orbital planes and the vehicle orientations to give maximum and minimum environmental heat loads are being considered. In addition, the case for which the vehicle longitudinal axis is in the direction of the velocity vector (or tangent to the flight path) is also included.

The present S-IV B/IU configuration was selected as the basic configuration with various configurations at the forward end of the instrument unit and the special case when the forward end of instrument unit is open and exposed to the space environment. The various configurations are illustrated in Figures 3 and 4. In Configuration D, Figure 3, the S-IV C is an uprated or modified S-IV B and is considered to be identical to S-IV B for purposes of this study. Figure 4 is an alternate configuration to Configuration A of Figure 3, in which the SLA panels with solar cells are shown in the open position. Since the backsides of the SLA panels act as radiators to maintain the solar cells at design temperature, the possible effects on the radiating IU structure are to be considered.

For the initial investigation, the near earth circular orbit at 200 nautical miles, 29° and 34° inclinations, and Configuration A of Figure 3 were selected for conducting the thermal control analyses.

2.2 SYSTEM CONCEPT

The approach to establishing the environmental control system concepts for the assumed mission models and mission durations up to 180 days has been to begin with the current Saturn V instrument unit design concept as the baseline and to investigate various feasible modifications to the baseline environmental control system to extend its useful life. Currently, the minimum modifications are being investigated to determine their feasibility. Various values for the coolant flow rate, coolant inlet temperature to the coldplates, emissivities and absorptivities are being used in the thermal analysis of the environmental control system to determine the suitable design values.

In order to identify possible modifications for more detailed investigations, the current environmental control system and the astrionic equipment have been reviewed from a thermal standpoint.

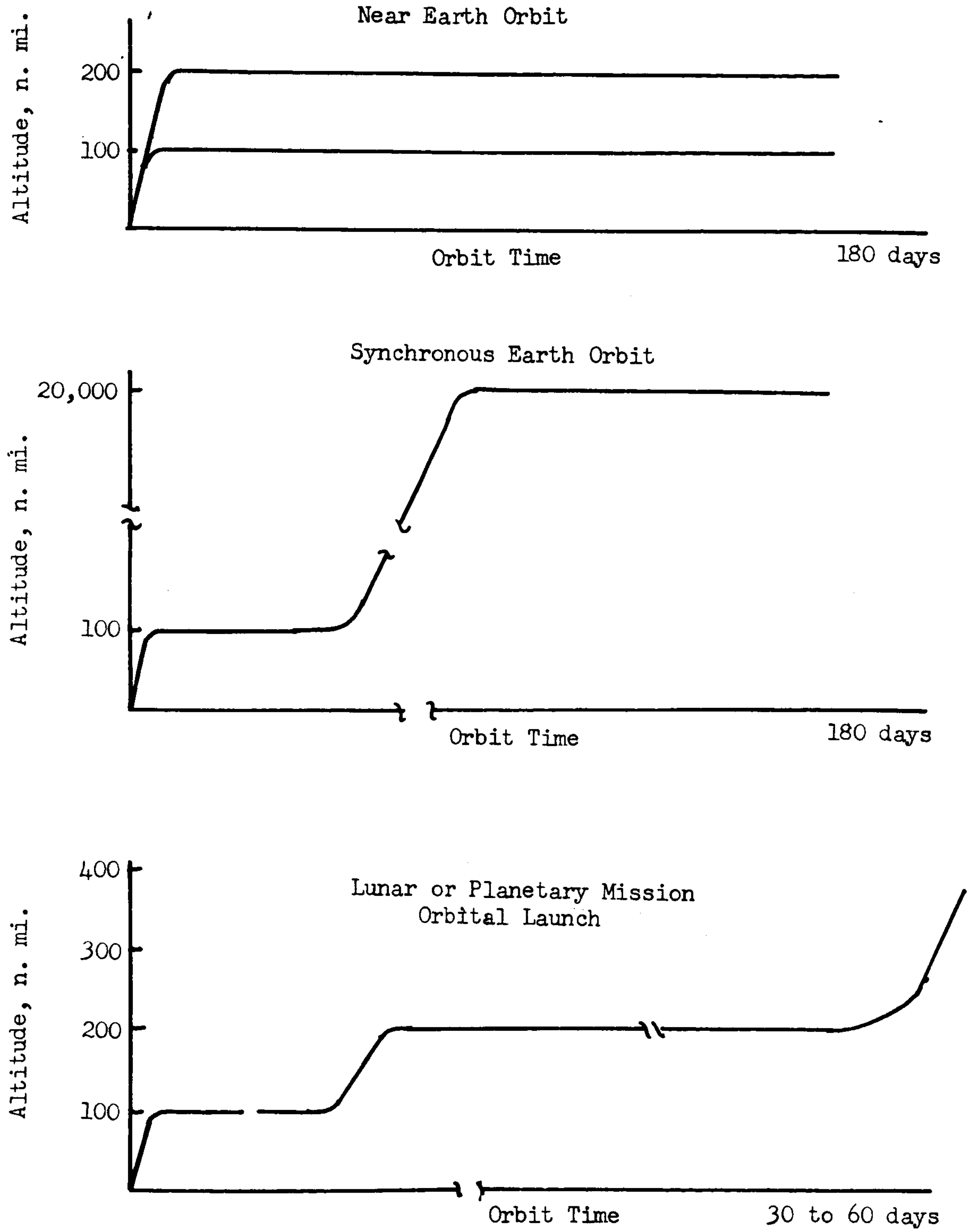


Figure 2. Basic Mission Profiles

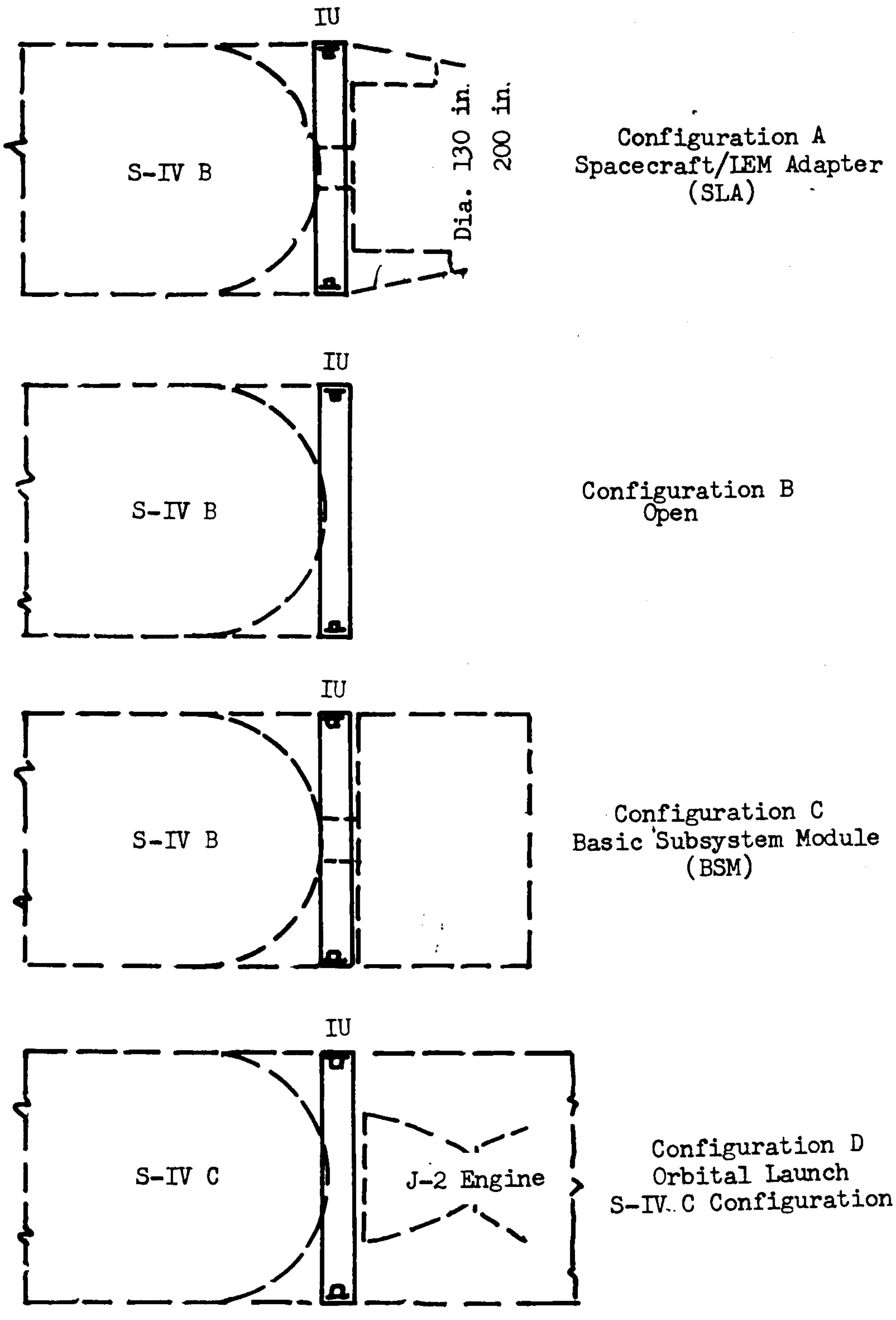


Figure 3. Vehicle Configurations

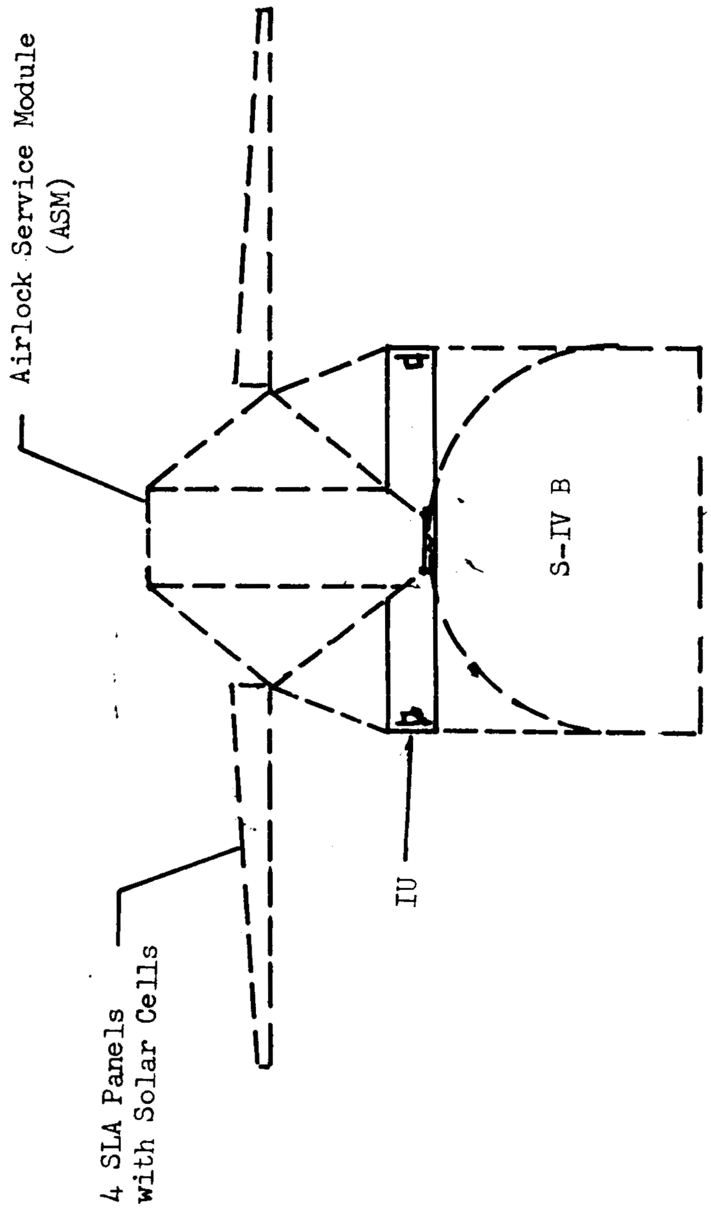


Figure 4. ASM/IU Configuration

The current or baseline environmental control system is illustrated in Figure 5, which shows the gaseous nitrogen supply subsystem and the thermal conditioning subsystem. Not shown in Figure 5 are the sixteen coldplates which are connected in parallel and the four pieces of equipment which are thermally conditioned internally or by circulating coolant in the package case. The review of the baseline system has been primarily directed in the thermal conditioning subsystems which is closed recirculating coolant loop which employs an expendable heat sink (water plus a sublimator) to reject excess waste heat. Since the coldplates are located around the inner periphery of the instrument unit and are mounted near the structure, the potentials of utilizing the coldplates as secondary radiators to reject heat or as in some cases as a heat source appear feasible. With the coldplates located around the inner periphery and since in most cases the environmental heat load varies around the periphery, it is possible there are conditions whereby a heat balance could be achieved in which the coolant return temperature would nearly equal the coolant inlet temperature so that no heat needs to be rejected or added by other means. Although such an ideal situation may be remote, it would certainly be advantageous to investigate and to establish design values which would approach the ideal case. On the basis of the review, the initial modifications to the environmental control system being investigated are: the coolant flow rates, coolant inlet temperatures, surface coatings of the instrument unit structure and the coldplates, and the conduction heat transfer through the coldplate mounting to the structure. The present configuration of the closed coolant loop, the methanol-water coolant, and the coldplate design are maintained for the initial investigation. Changes in these items will be investigated as additional possible modifications.

The present astrionic equipment located in the instrument unit was reviewed to determine if the operating temperature tolerance and heat loads may be modified and to establish possible grouping of equipment on the basis of temperature tolerance, or thermal interface of the individual package, and the possible combination of these two and heat load.

Figure 6 is a graphical presentation of the present Saturn V IU instrument's tolerable case temperature range as the ordinate versus the astrionic equipment heat load as the abscissa. These temperature range and heat load data are those provided by NASA-MSFC for this study.

In Figure 6 the equipment is plotted in the order of diminishing lower temperature limit. That is, the batteries, which have a lower operating temperature limit of +68 F, are plotted first, and the flight control computer and the control signal processor, which have a lower operating temperature limit of -67°F, are plotted last. Such a plot readily shows the heat loads associated with the temperature sensitivity of the equipment and allows grouping, on a thermal capability basis, of the equipments.

In order to minimize the number of groups, the tolerable temperature range of some packages is extended beyond the range specified by NASA. In these instances, it is believed that the present limits are

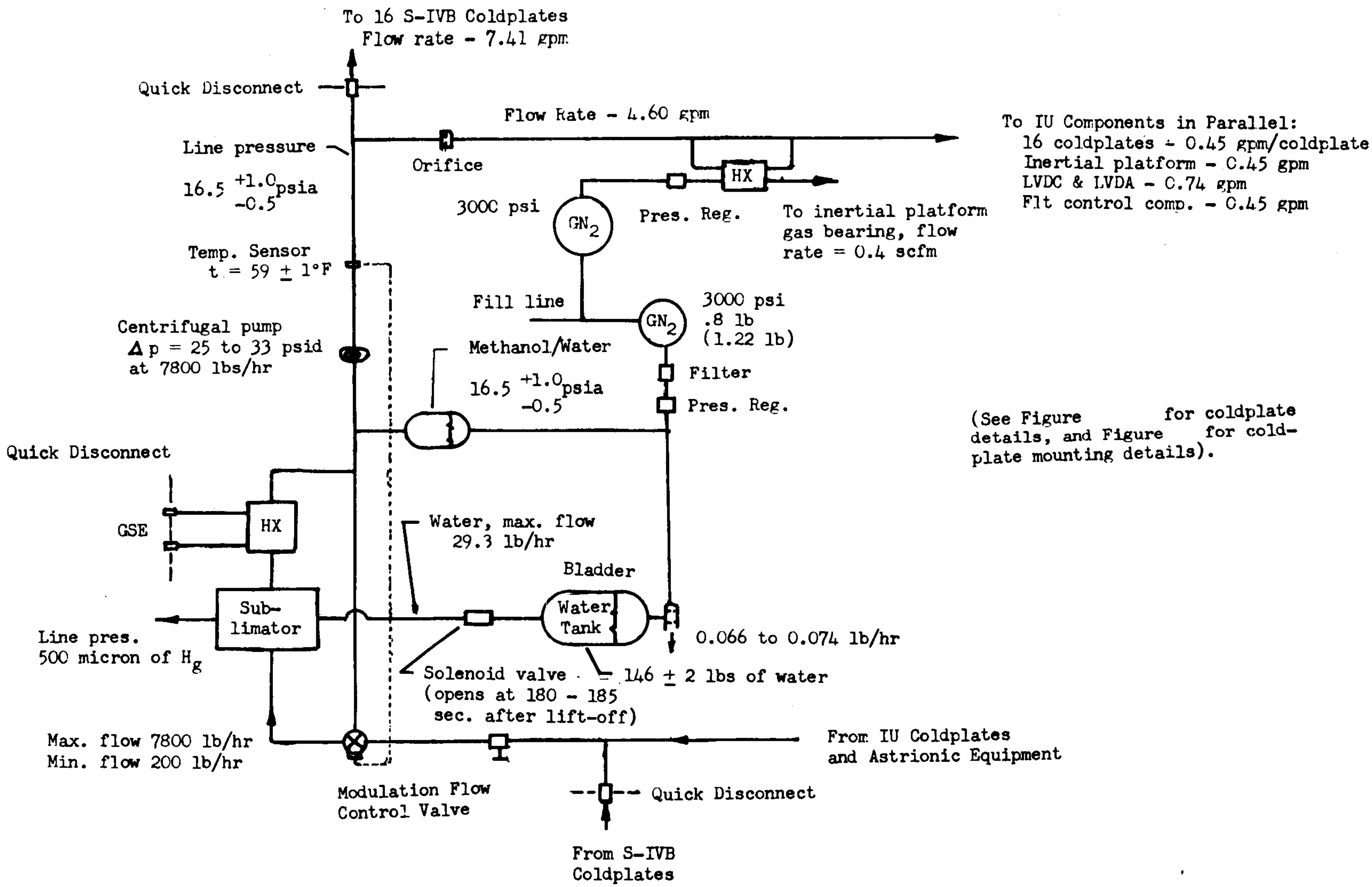
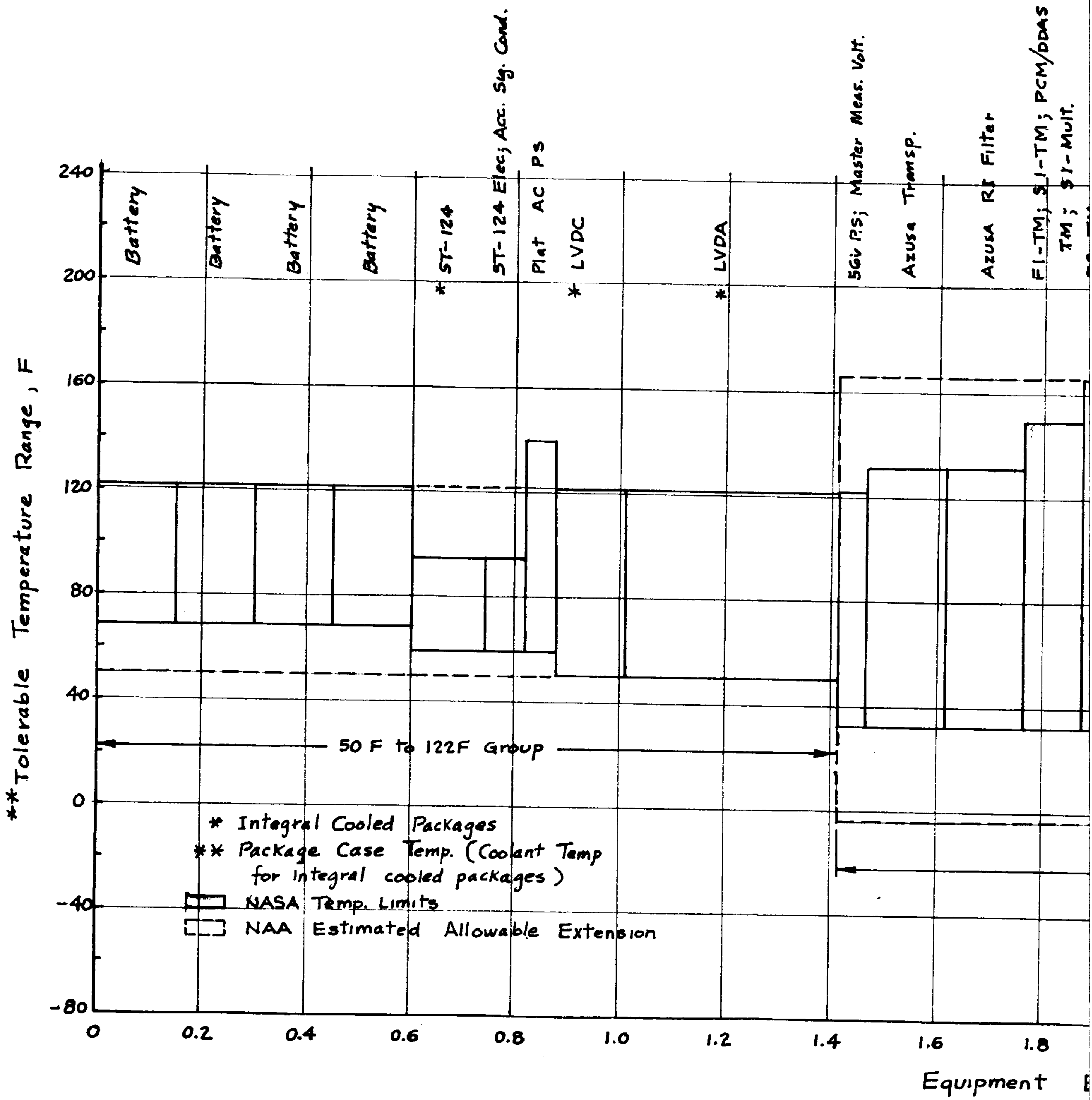


Figure 5. Baseline Environmental Control System


Battery
Battery
Battery
Battery
* ST-124
ST-124 Elec; Acc. Sig. Cond.
Plat AC PS
* LVDC
* LVDA
56v P.S; Master Meas. Volt.
Azusa Transp.
Azusa RF Filter
F1-TM; S1-TM; PCM/DDAS
TM; S1-Mult.
240
200
160
120
80
40
0
-40
-80
**Tolerable Temperature Range, F
50 F to 122F Group
* Integral Cooled Packages
** Package Case Temp. (Coolant Temp for Integral cooled packages)
NASA Temp. Limits
NAA Estimated Allowable Extension
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
Equipment

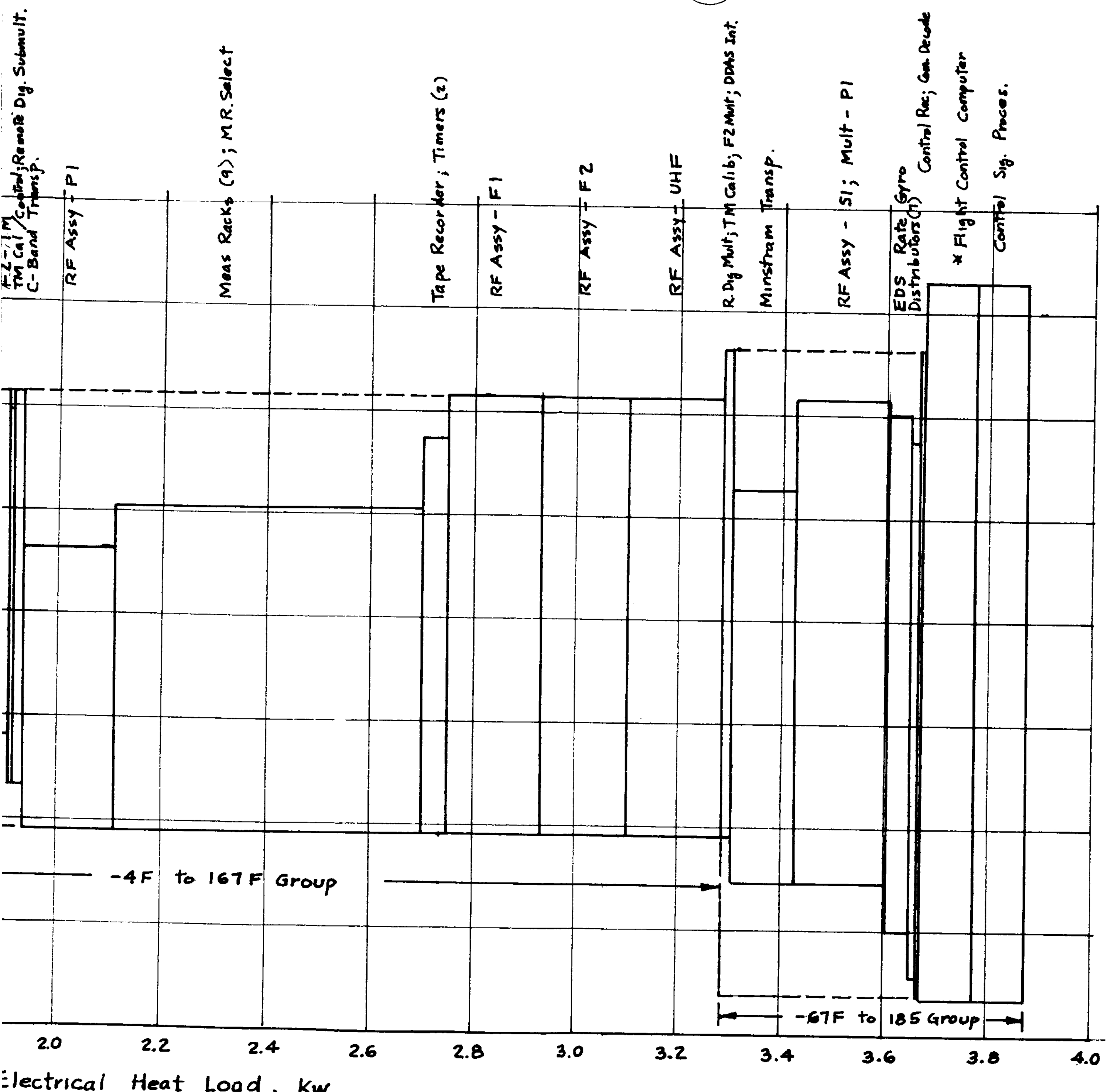


Figure 6. Tolerable Case Temperature Range

Table 1. Alternate Approaches For Thermal Control

Astrionic Equipment	Thermal Control Approaches for Mission Periods		
	Pre-launch	Ascent	Orbit
Group I	GSE HX	Active: Sublimator	Active: Sublimator and/or radiators plus coldplates as secondary radiators.
Group II	GSE HX	Closed liquid loop. GSE HX (Active: sublimator).	Closed liquid loop with coldplates as secondary radiators (radiation only).
Group II	GSE HX	Closed liquid loop with coldplates as secondary radiators (radiation only).	(No cooling by sublimator or radiator).
Group III	Included in Group II.	Included in Group II.	Passive (radiation only). Coldplate mounted.
Group III	Passive. IU skin mounted.	Passive. IU skin mounted.	Passive (radiation only). IU skin mounted.

conservative and that the wider temperature limits are tolerable by the equipment (or could be made tolerable with only minor thermal design improvements within the packages).

Thus, for the present Saturn V IU packages, three groups appear logical. These are: the 50 F to 122 F group (Group I), the -4 F to 167 F group (Group II), and the -67 F to 185 F group (Group III). Group I would, for all practical purposes, require an active ECS whereby a recirculating coolant under some temperature control is utilized. In Group II, again an ECS with a recirculating coolant system is required, however, temperature control would probably not be required. The recirculation of the coolant system in a closed liquid coolant loop utilizing the vehicle cold plates as radiators may be sufficient. The Group III equipment could perhaps be treated completely passively, where the thermal mass of the equipment and the vehicle coldplate or structure to which the equipment mount does not allow the equipment to exceed their tolerable limits.

One exception to a completely passive ECS in Group III is the integrally cooled flight control computer. Although from a temperature range tolerance standpoint this unit is in Group III, the need for a coolant flow would require that it be grouped with Group II.

In addition to the present astrionic equipment, a review of possible future design has been made which indicates a trend toward lower heat loads and integrally cooled design rather than external interface, such as a coldplate. The temperature tolerance range does not appear to change appreciably so that the three groupings indicated above appear to be a reasonable arrangement. The exceptions are those packages which are integrally cooled and thus are grouped according to design.

On the basis of the review of the current environmental control system and astrionic equipment, several alternate approaches to the ECS design appear feasible in addition to the present configuration. The alternates are summarized in Table 1 in accordance to the astrionic equipment grouping and mission periods. It is to be noted that the ECS concepts to be selected will be total concepts which encompass the three groups and the various mission periods on an integral basis.

As indicated in Table 1, Group I and Group II involve closed coolant loop with variable thermal control. Groups I and II are assumed to be in separate loops but can be thermally connected by a heat exchanger. For Group III, the basic approach is a passive means for thermal control. One alternate is to consider the equipment to be mounted on coldplates with provisions to modulate the coolant flow to the coldplate as required, with the possibility of no-flow to be tolerable. The other alternate is to consider the electronic package to be mounted directly on the inner surface of the instrument unit structure. This approach is inherently reliable and certainly deserves consideration. The investigation of the purely passive approach will require careful consideration of the various thermal environments, particularly the ascent heating.

In addition to the alternate approaches given in Table 1, various means to achieve thermal control on an individual package basis will be considered for those special cases or problems, as revealed during the detailed thermal analysis.

2.3 SYSTEM ANALYSIS

Detailed analysis of the various approaches to the environmental control concepts are underway and the results of the cases investigated during this reporting period are given in the following paragraphs. In addition, the results of the investigation into the determination of the tolerable temperature range for the coolant for both the coldplate mounted and integrally cooled equipment and the discussion on the problem of meteoroid penetration are presented.

2.3.1 Fluid Temperature Limits

The tolerable temperature ranges for the coolant or heat transfer fluid in the coldplates and the integrally cooled equipment were established for the present equipment for both the NASA specified equipment temperature tolerances and the NAA suggested modified temperature tolerances.

In establishing the tolerable temperature ranges, those equipment which required special consideration are identified and the approaches to bring the tolerable range in line with the other equipment are discussed.

Also, this investigation was concerned with trying to achieve a maximum tolerable range and the highest permissible upper temperature. The maximum possible tolerable range is desirable from the standpoint of simplifying the control or the maintainance of the coolant temperature within desired values. The upper permissible temperature should be as high as possible in order to make maximum utilization of radiation heat transfer to reject waste heat. This is significant because of the limited surface area available on the instrument unit for heat rejection. In Figure 7, the required heat transfer area in terms of radiation temperature, surface equilibrium temperature and heat dissipation are given. This graph shows that nearly all of the available area on the surface of the instrument unit is required to dissipate a 5 KW heat load, which is approximately the current heat load. Thus, in order to dissipate the expected heat loads, maximum utilization of the available surface area becomes mandatory. The use of surface area on adjoining structure is a possibility.

One additional consideration is the effect of fluid flow rate. By utilizing the minimum flow rate necessary, the pump power requirements would be reduced, along with a lower pressure drop.

To establish the tolerable temperature range, the integrally cooled equipment were treated separately from the coldplate mounted equipment. For the integrally cooled equipment, the temperature drop between the

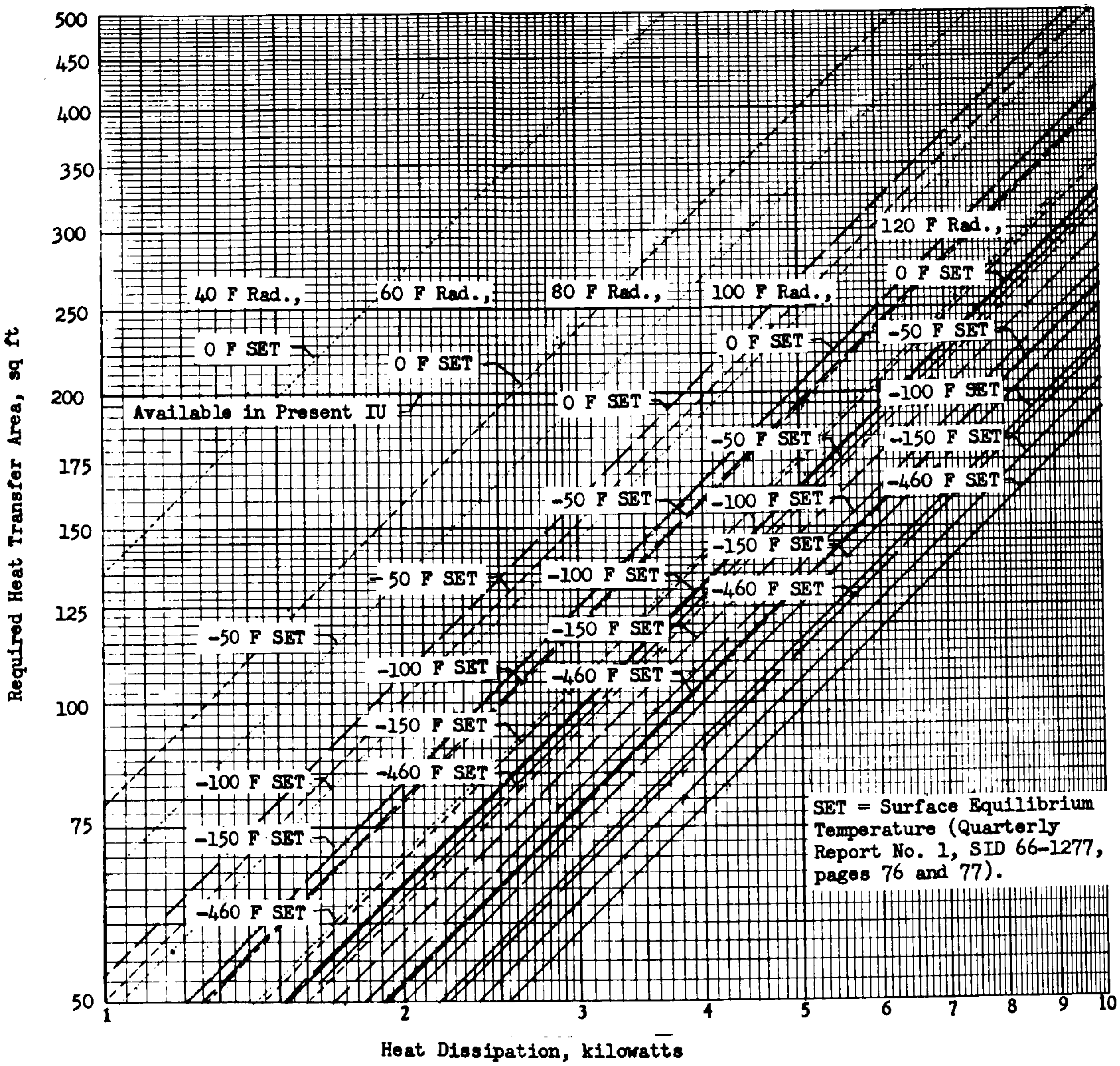


Figure 7. Heat Transfer Area Versus Heat Load

equipment and the coolant is relatively simple since there is a direct contact between the coolant and the interface surface. For the case of the coldplate mounted equipment, the determination of the temperature drop is more complicated because of the joint resistance and heat transfer along the coldplate surface. These are considered in the following discussion.

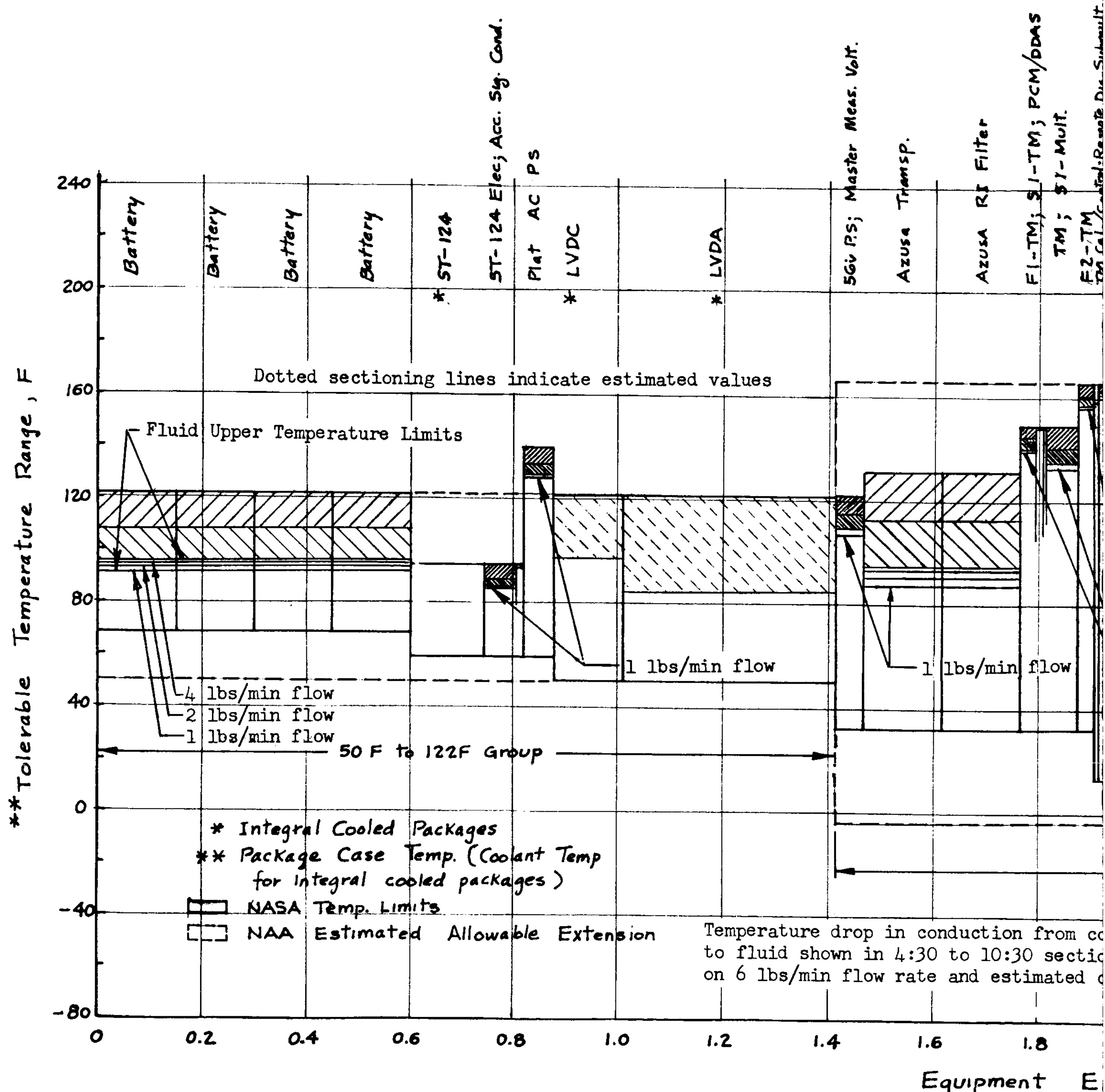
As estimated, joint contact resistance of $\frac{1}{4}$°F per watt of heat transfer was used in preparing Figures 8, 9, 10 and 11. The resistance between the joint and the fluid, based on an estimated joint diameter of contact, was determined from the four figures in Appendix A, which show $\Delta T/Q$ as a function of coldplate equivalent skin thickness, contact diameter and equivalent heat transfer coefficient.

Figure 8 shows the maximum permissible temperature of the heat transfer fluid based on the equipment temperature limits of NASA drawing numbers A50 M04001 and A50 M123434. The bottom of the cross sectioned areas is the first approximation of the maximum temperature permissible for the cooling fluid. This was based on a cooling fluid flow of 6 lb/min. per coldplate (the maximum design value). The bottom line is for a flow of 1 lb/min. The intermediate lines are for 2 and 3 lb/min. An examination shows that based on the indicated estimates, one piece of equipment (RF Assembly P1) requires a fluid temperature approaching 60°F. In addition to this piece of equipment, the stable platform would require temperature less than 80°F.

The two approaches to raising the upper permissible fluid temperature are the design modification of the equipment and the reduction in joint contact resistance. For the RF Assembly P1, careful treatment to reduce the contact resistance could possibly reduce the contact temperature drop sufficiently to permit the fluid temperature to increase to about 80°F.

Figure 8 shows the influence of the various flow rates, and in nearly all cases, there is relatively small temperature difference between the maximum and minimum flow conditions. Thus, the 1 lb/min. flow or even less may be suitable for most of the equipment. This suggests another possible grouping of the equipment on the basis of coldplate flow rate, in addition to the grouping previously indicated.

In addition to the RF Assembly P1 and ST-124 platform, other pieces of equipment which includes the ST-124 electronic assembly, accelerometer signal conditioner, launch vehicle data computer, launch vehicle data adapter, Azusa RF filter, Azusa transponder, and the batteries will influence the selection of the upper permissible temperature and flow rate. For example, the batteries, with increased mission time, could increase in number and have proportionally a lower amperage output. As a consequence, since I^2R is the internal power dissipation and this is proportional to temperature difference, a lower temperature difference between the case and the circulating fluid will exist. This temperature drop is inversely proportional to the square of the number of batteries. Thus, for increased mission time, the temperature tolerance for batteries



UPPER FLUID TEMPERATURE LIMITS USING METHANOL WATER CIRCULATING FLUID AND NASA SPECIFICATION EQUIPMENT TEMPERATURE LIMITS.

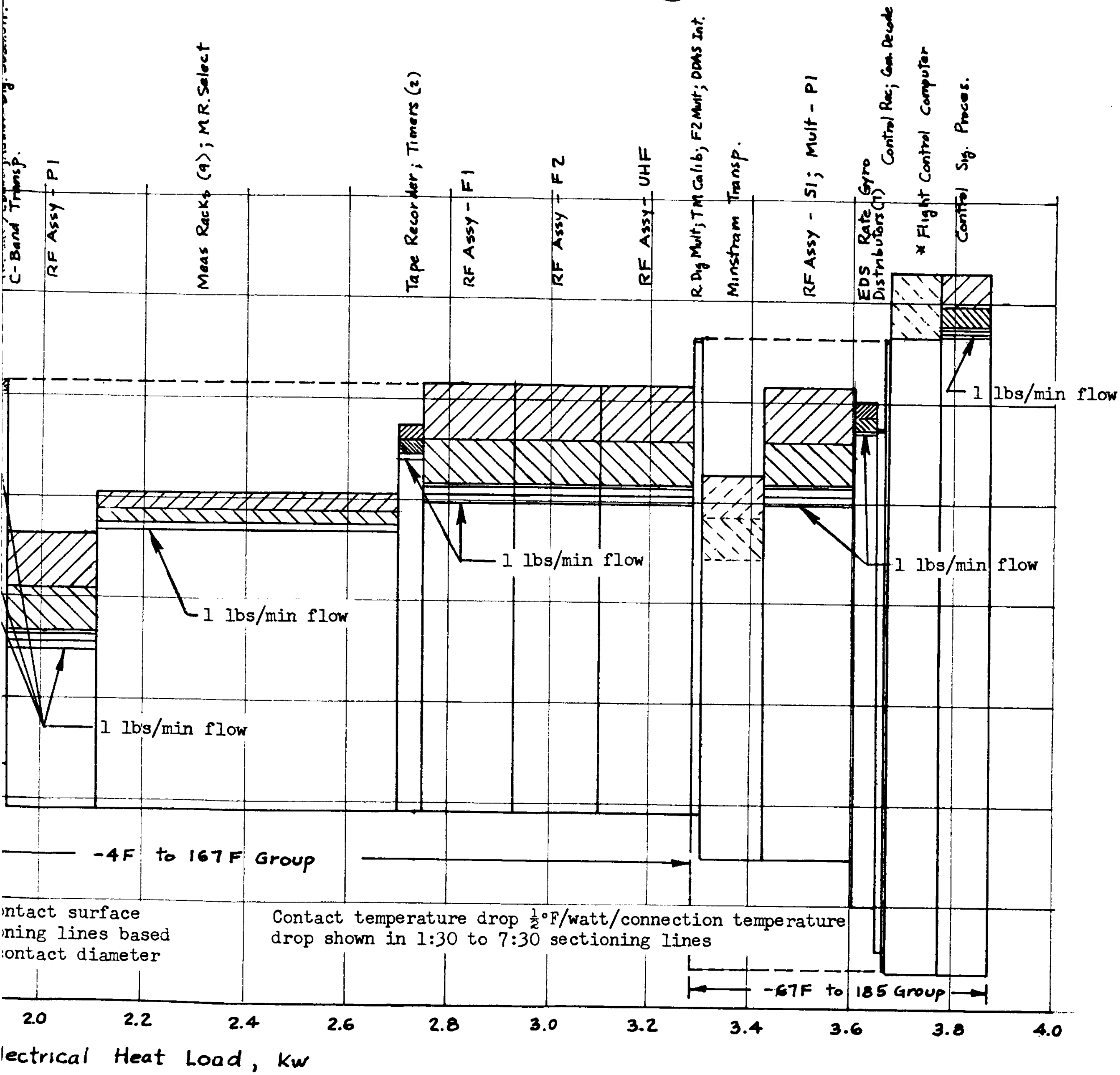
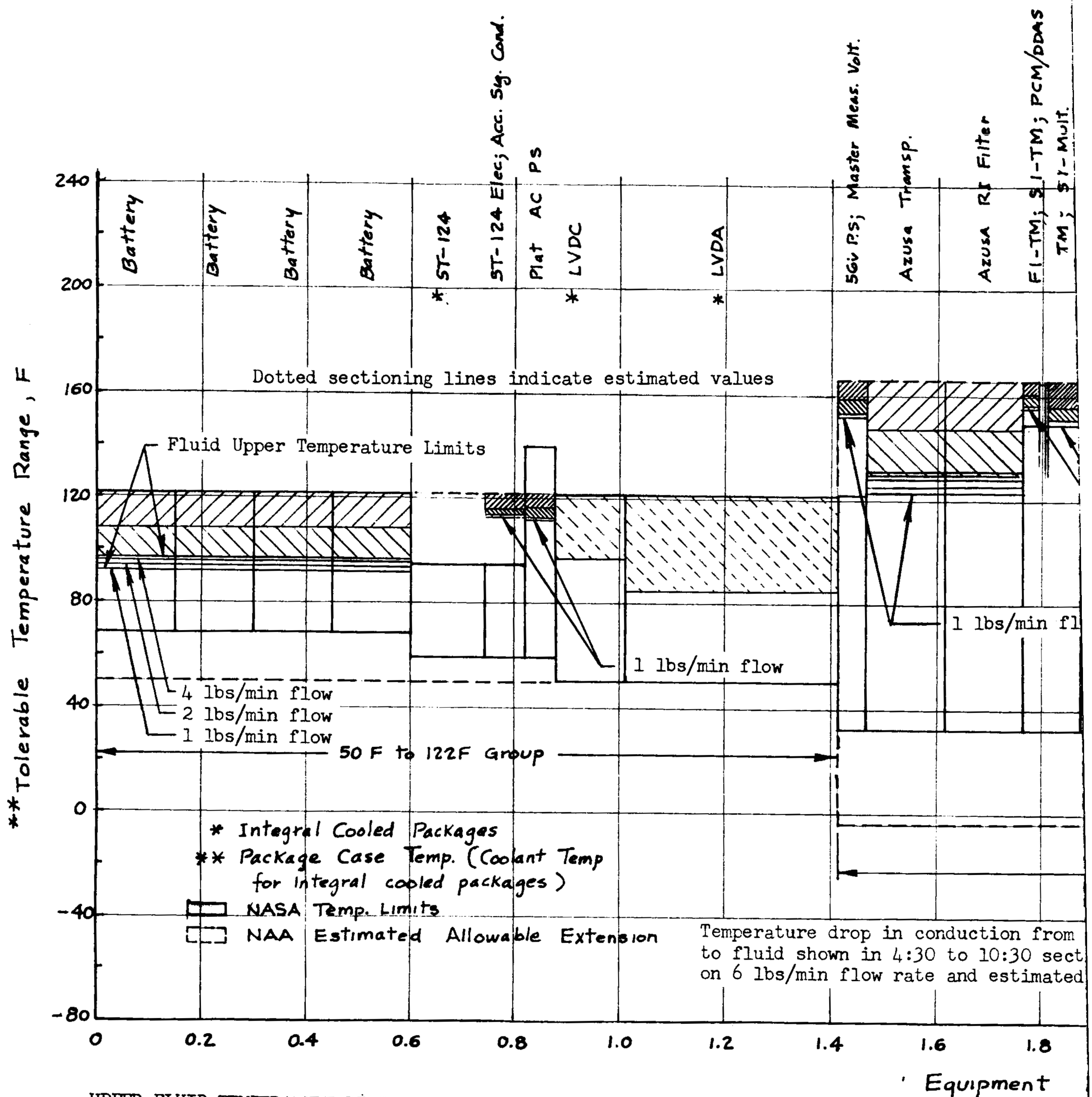


Figure 8. Upper Fluid Temperature Limits, NASA Specification Equipment Temperature Limits



UPPER FLUID TEMPERATURE LIMITS USING METHANOL WATER CIRCULATING FLUID AND NAA ESTIMATED ALLOWABLE EXTENSION OF EQUIPMENT TEMPERATURE LIMITS

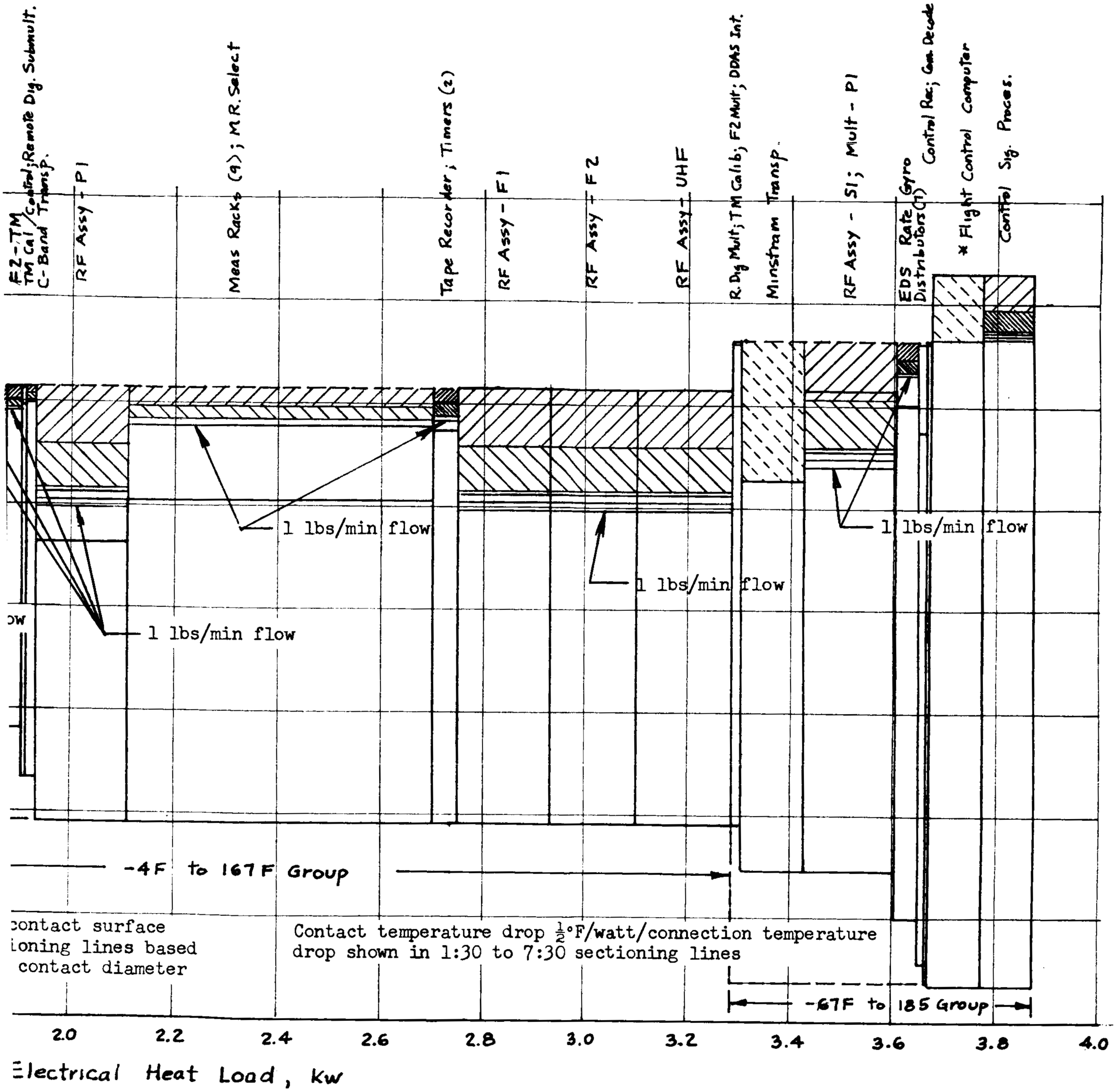
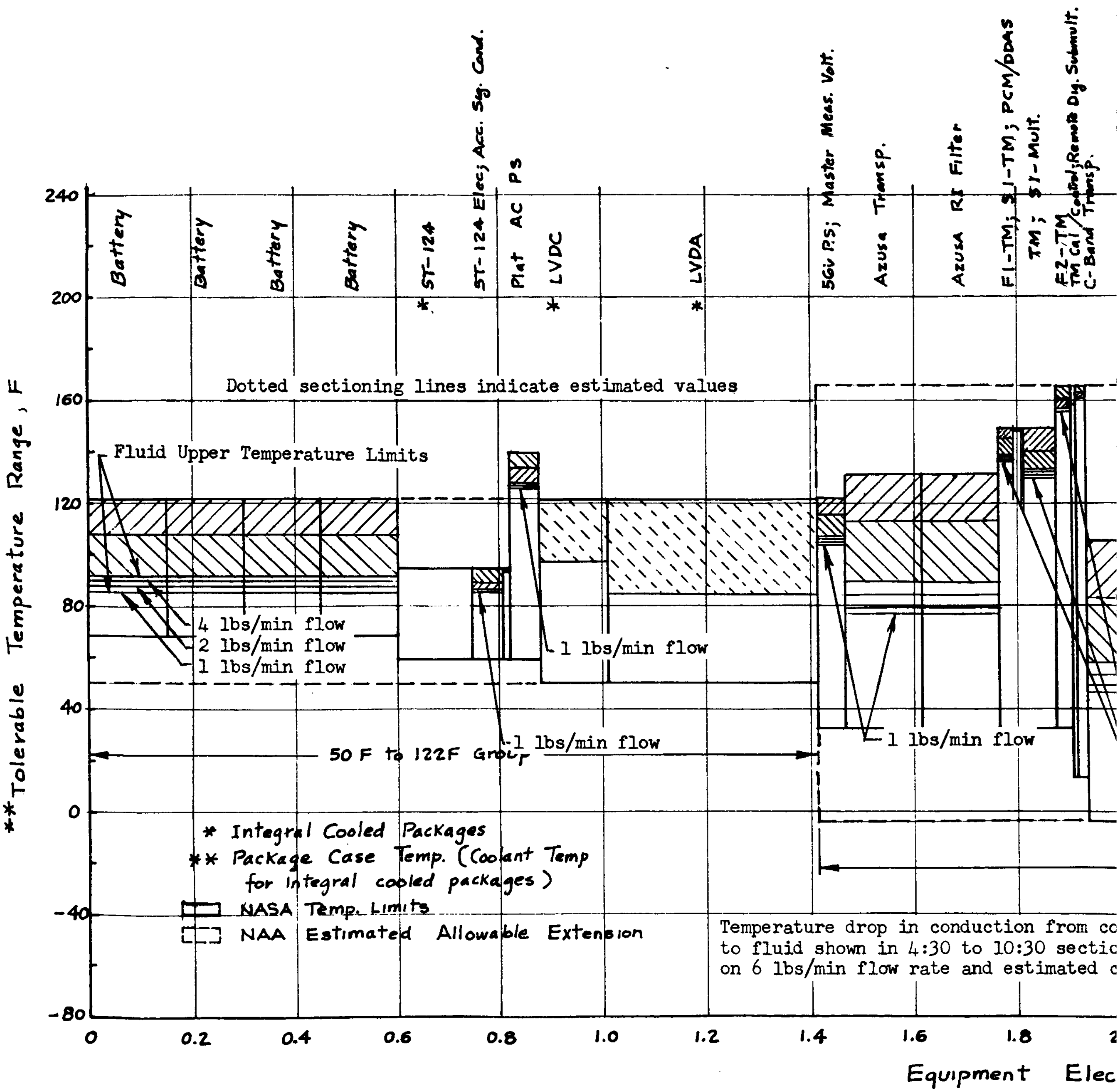


Figure 9. Upper Fluid Temperature Limits, NAA Modified Temperature Limits



UPPER FLUID TEMPERATURE LIMITS USING ORGANIC CIRCULATING FLUID AND
NASA SPECIFICATION EQUIPMENT TEMPERATURE LIMITS

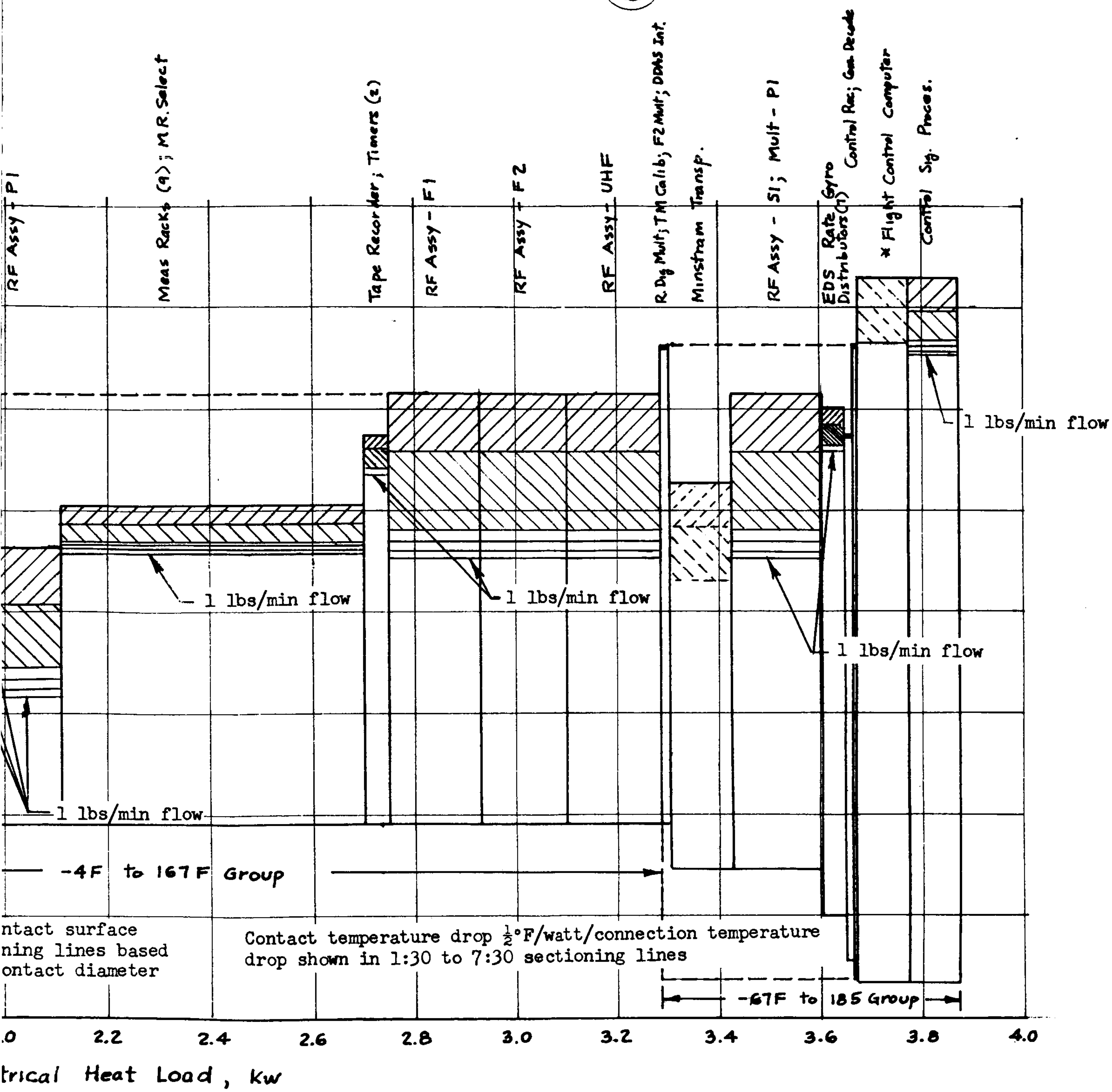
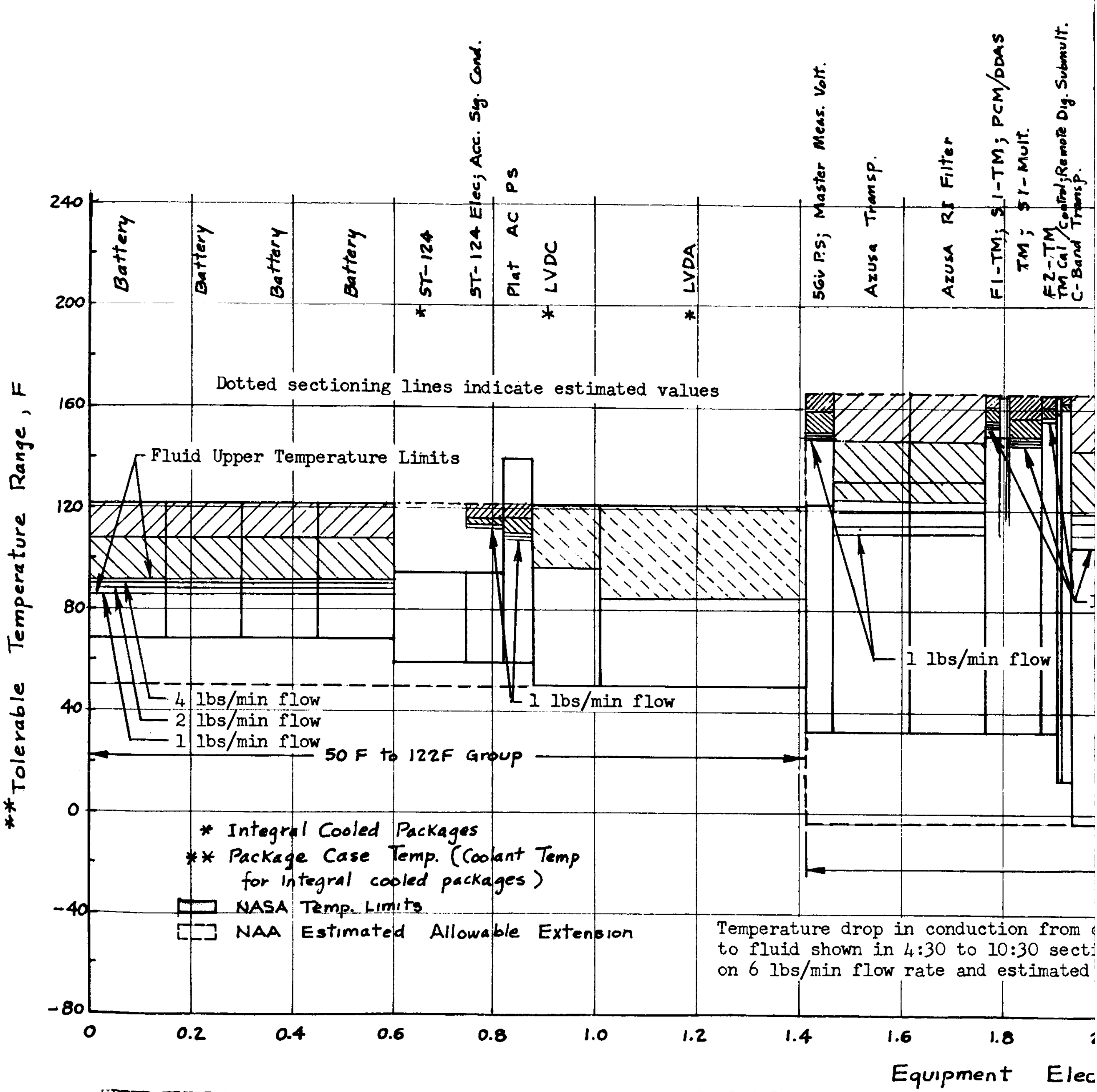


Figure 10. Upper Fluid Temperature Limits, Organic Circulating Fluid



UPPER FLUID TEMPERATURE LIMITS USING ORGANIC CIRCULATING FLUID AND NAA ESTIMATED ALLOWABLE EXTENSION OF EQUIPMENT TEMPERATURE

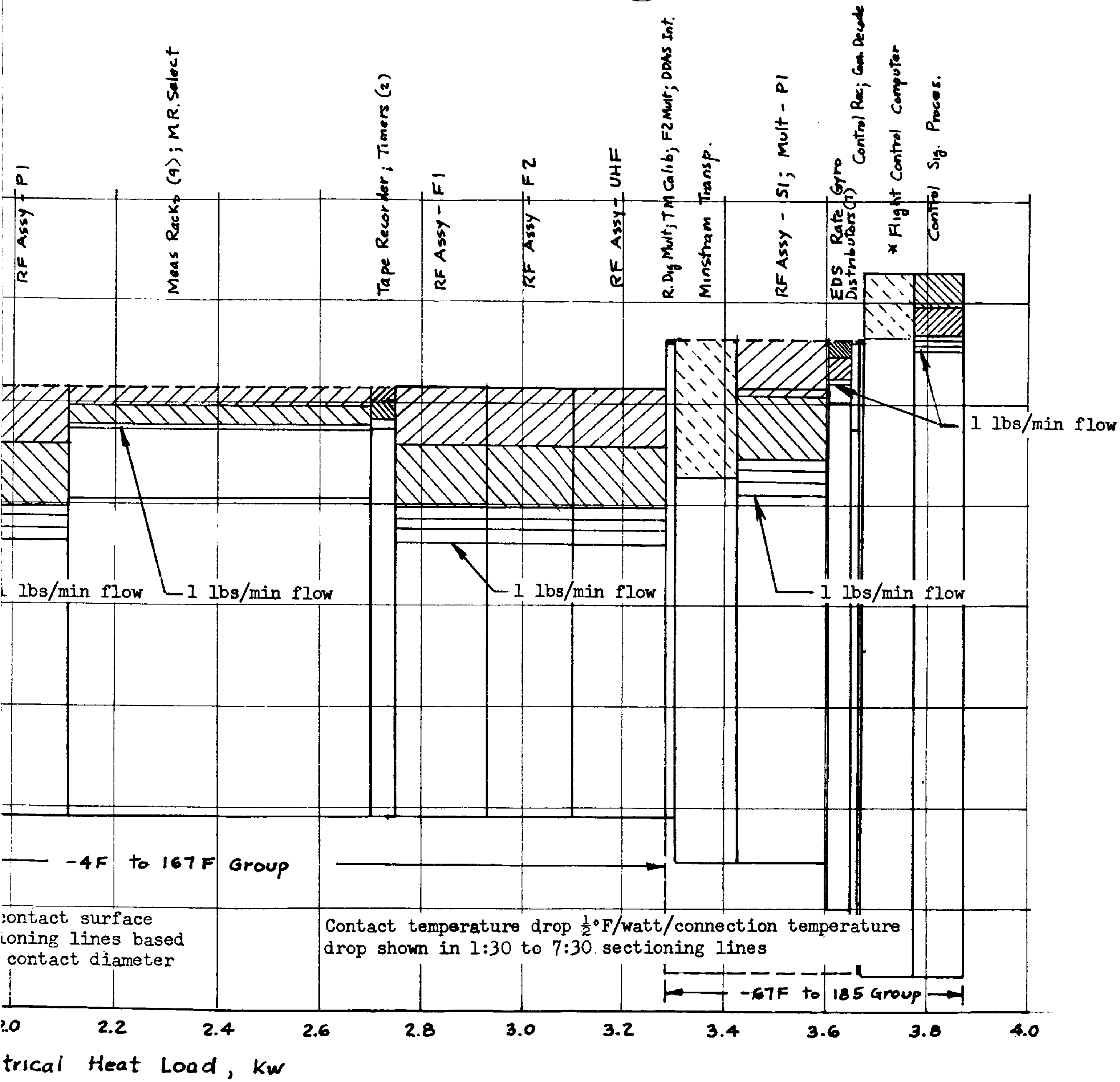


Figure 11. Upper Fluid Temperature Limits, Organic Circulating Fluid.

would appear to be less restrictive than shown.

One further consideration is the temperature rise of the cooling fluid as it passes through the coldplate and this is influenced by the equipment mounted on the individual coldplate.

On the basis of the above discussion, an upper permissible temperature limit of 80°F and a flow rate of 1 lb/min or possibly less, appear feasible for the present equipment.

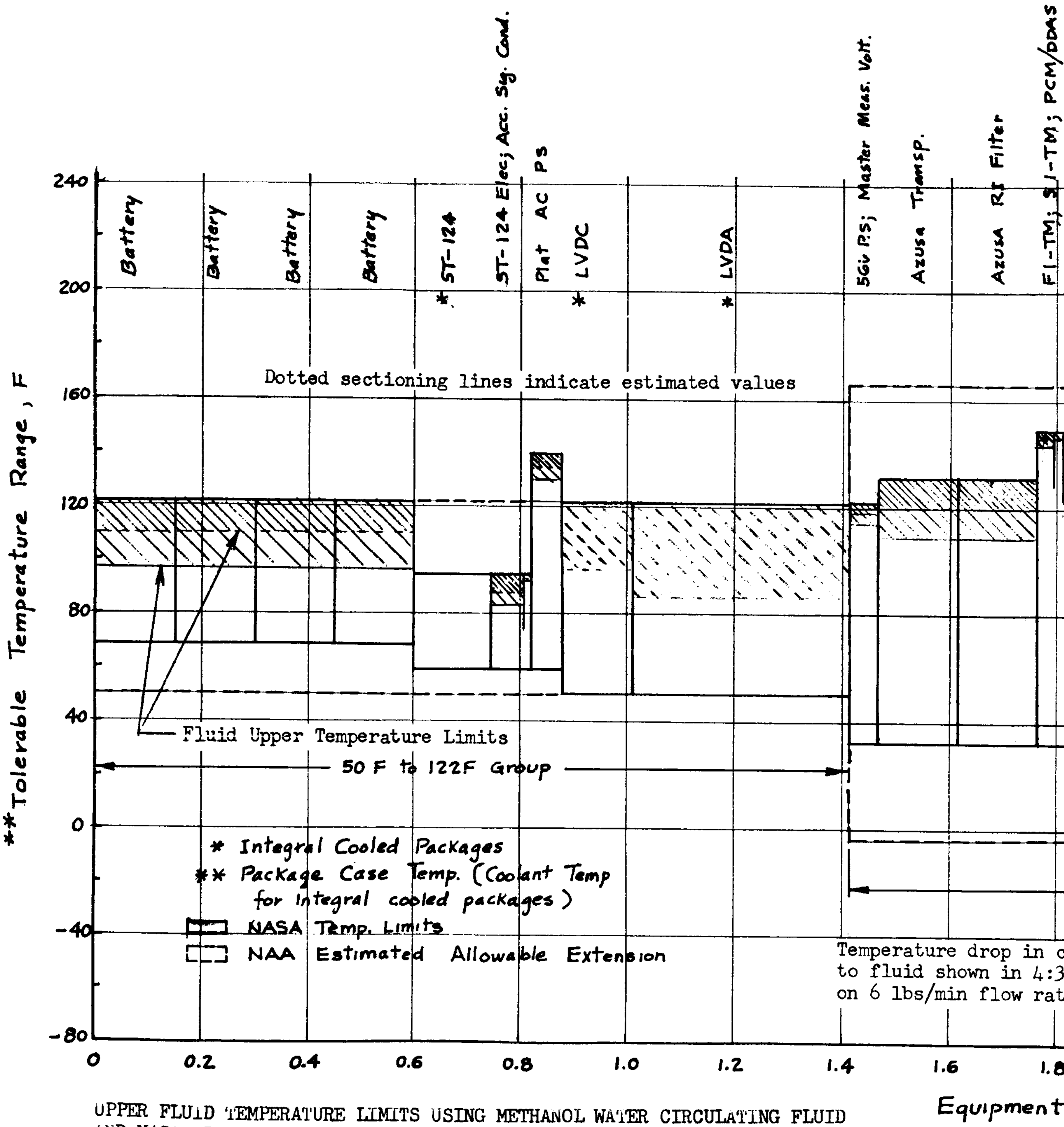
A more detailed analysis of the contact resistance and internally cooled equipment could result in further reductions in fluid flow and increases in maximum fluid temperature. Only the critical pieces of equipment needed to be examined. Of these, only the equipment which gives off a considerable amount of heat will be significantly affected by contact resistance, which is a function of size, force, and surface finish.

Figure 9 is similar to Figure 8 except for the NAA modified temperature limits. The maximum cooling fluid temperature limitations for the majority of equipment coldplates can probably be raised another 40°F upward and still require only a minimum flow. If a separate coolant loop for this equipment is utilized, coolant flow may be needed only for prelaunch and injection. Some equipment have wide temperature limits, so that mounted on a plate without coolant flow may be satisfactory.

Figure 10, which is similar to Figure 8, shows the effect of substitution of a hydrocarbon circulating fluid on temperature limitations. The hydrocarbon circulating fluid would eliminate any overcooling problems in the sublimator or in a radiator. It also has certain advantages for internally cooled equipment. However, the heat transfer coefficient decreases at comparable fluid flow rates. Figure 10 shows how much lower a fluid temperature would be required with a hydrocarbon fluid in place of the methanol-water solution. Slightly greater pumping power may also be required. Figure 11 is similar to Figure 9 except that a hydrocarbon circulating fluid is substituted for the methanol-water solution. This shows a slightly lower required fluid temperature.

Figure 12 is similar to Figure 8, except a better approximation for joint contact resistance was used. It is based on 65,000 lb/sq in bolt yield strength and an 18,000 lb/sq in for the softest structural material and 50% of maximum tightening. (See Figure 1B in Appendix B) This also shows the effect of thin dead soft aluminum washers for critical pieces of equipment. Some improvement in increasing the upper permissible fluid temperature is indicated for the RF Assembly P1.

A plot of the lower temperature limit is shown in Figure 13. The long dashed lines indicate the minimum fluid temperature based on NAA estimated allowable temperature limits for the individual pieces of equipment. It can be noted that the highest minimum temperature occurs for packages with very small electrical loads. Insulation between these extremely small heat rejection packages and the coldplate can improve



UPPER FLUID TEMPERATURE LIMITS USING METHANOL WATER CIRCULATING FLUID AND NASA SPECIFICATION EQUIPMENT TEMPERATURE LIMITS - IMPROVED ESTIMATE

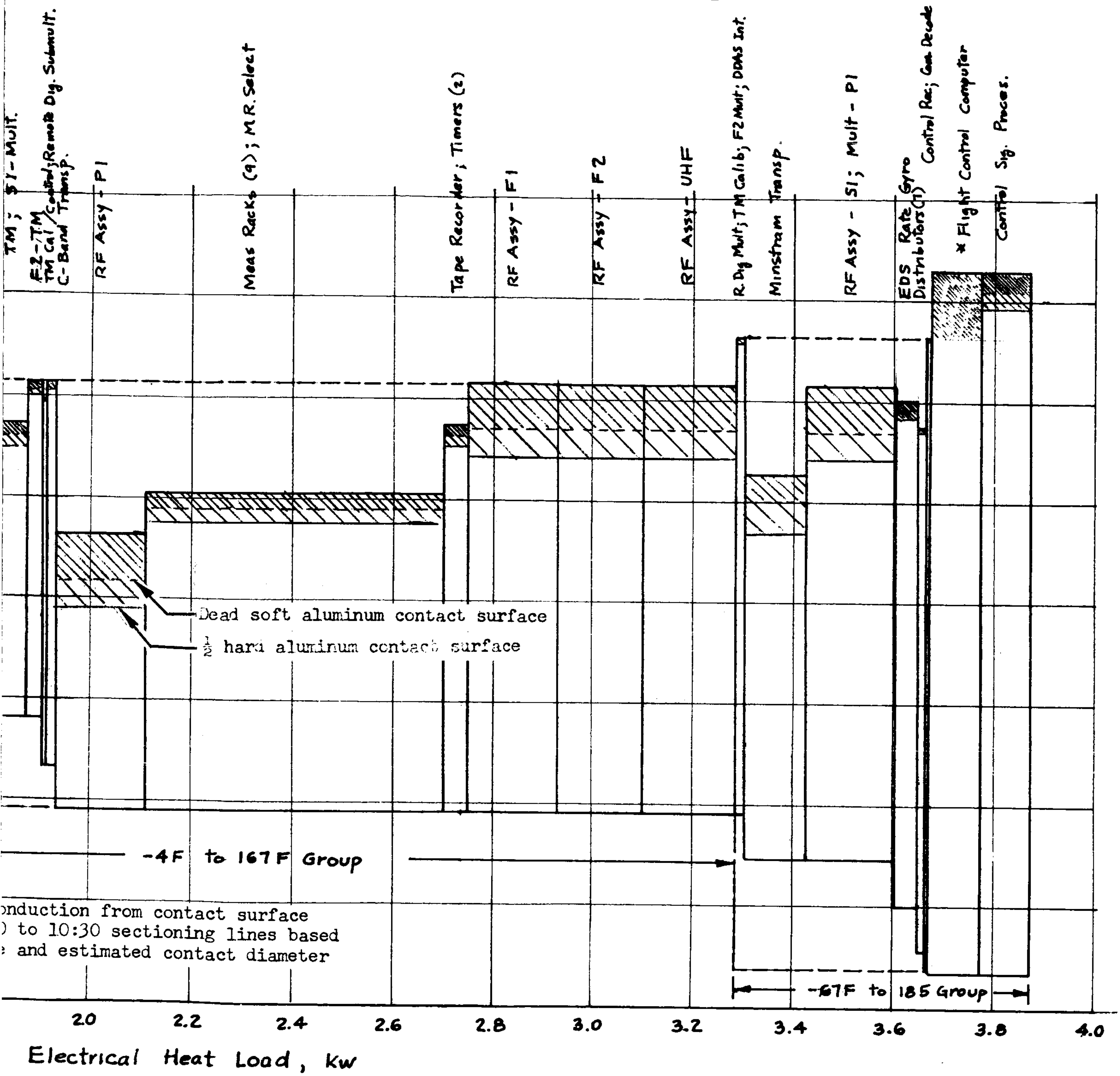
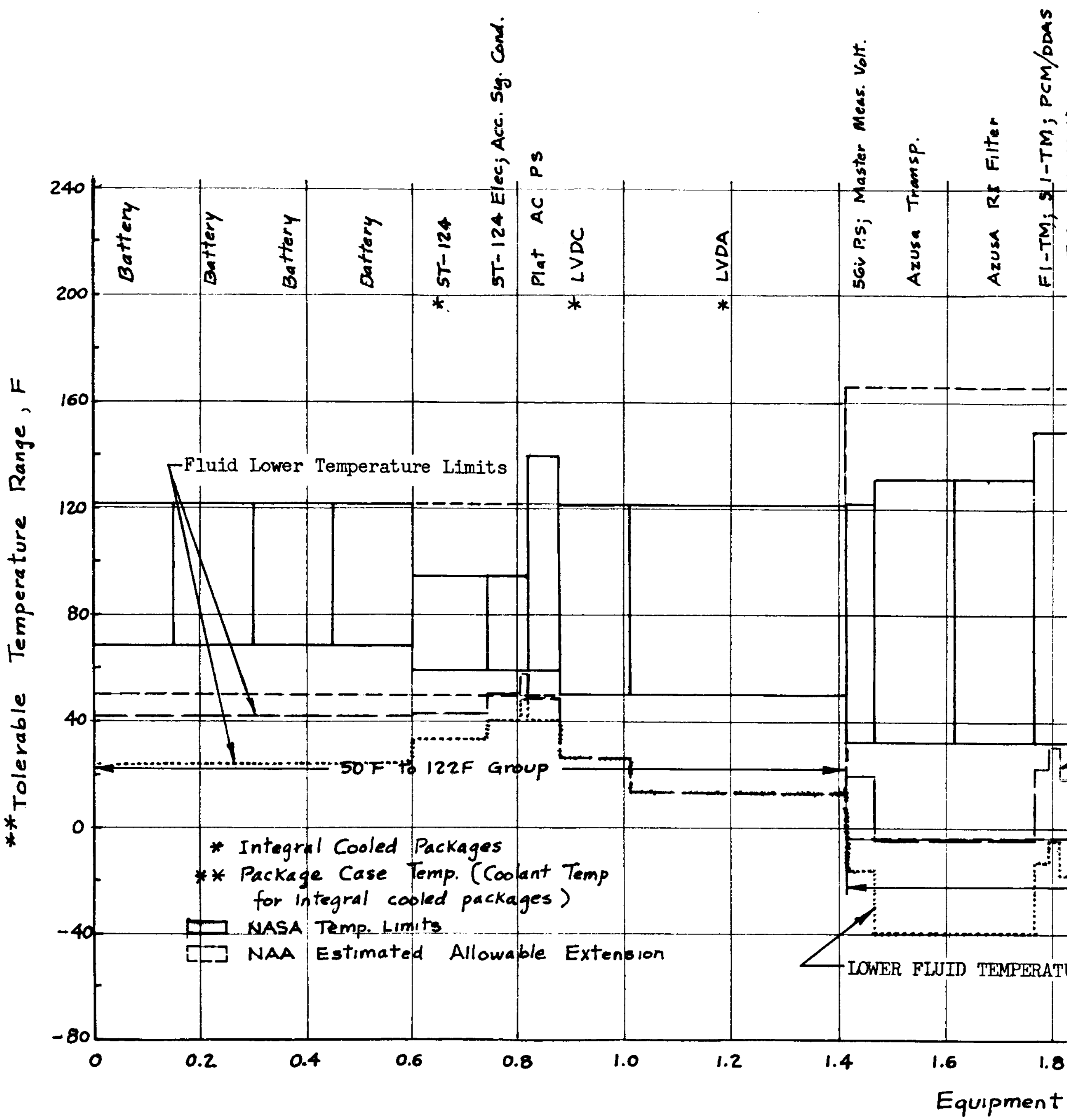


Figure 12. Upper Fluid Temperature Limits Improved Estimates



LOWER FLUID TEMPERATURE LIMITS USING METHANOL WATER CIRCULATING FLUID FOR BOTH NASA SPECIFICATION EQUIPMENT TEMPERATURE LIMITS AND NAA ESTIMATED ALLOWABLE EXTENSION OF EQUIPMENT TEMPERATURE LIMITS WITH NO INTERMEDIATE RESISTANCE OR FLOW CONTROLS

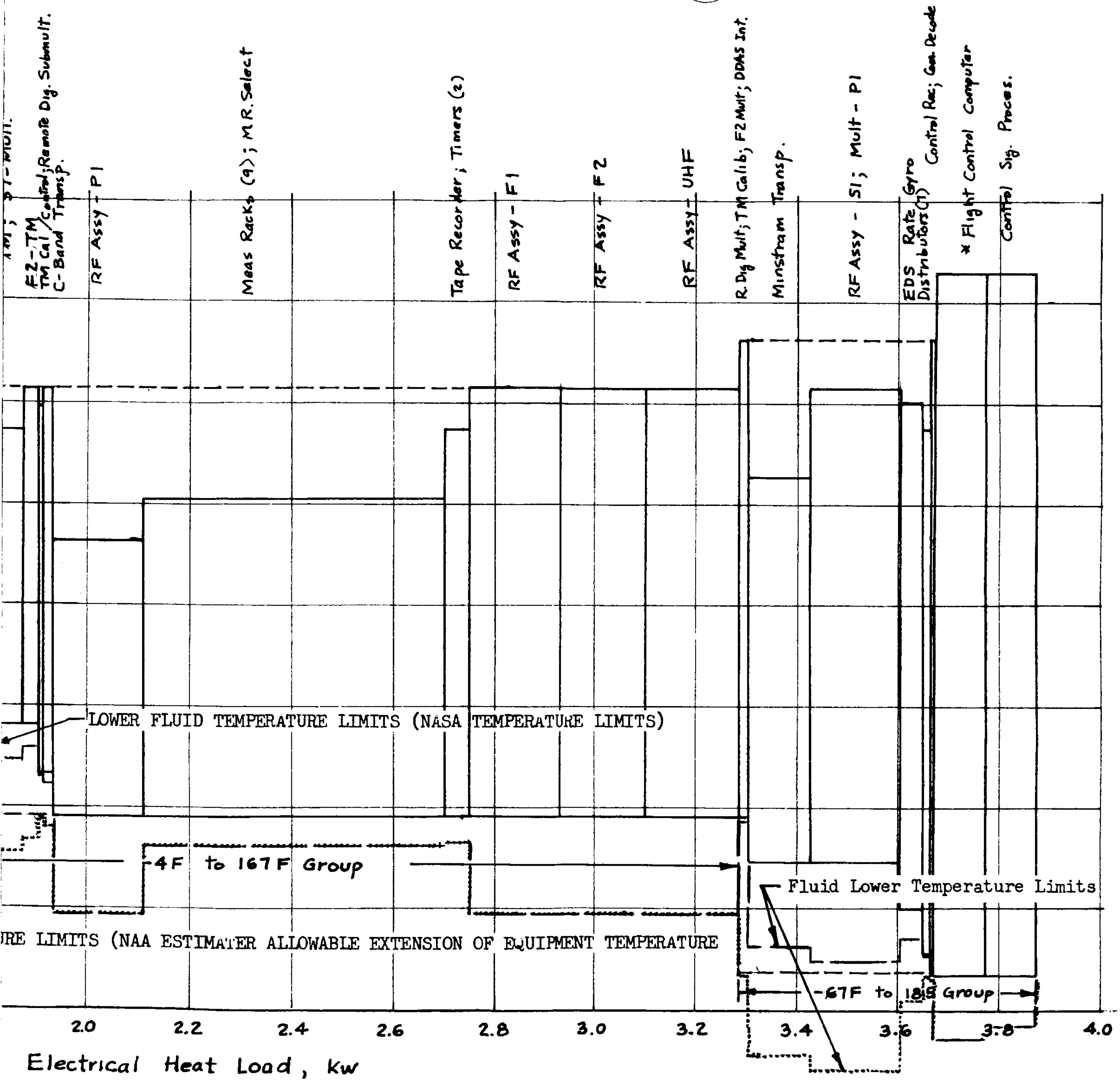


Figure 13. Lower Fluid Temperature Limits, NASA Specification Equipment Temperature Limits and NAA Modified Temperature Limits

the minimum temperature condition. Packages which always have a sizeable heat rejection can have their minimum temperature controlled by a thermostatic valve on the coldplate, set so that the valve closes some ten to twenty degrees Fahrenheit below the maximum permissible temperature. Packages which have a very low heat rejection can have their temperature controlled by being located on the downstream portion of the coldplate that is controlled by the presence of a large heat source and exit thermostatic valve. Very low heat rejection packages can also be controlled by electrical heat or Peltier cooling, with insulation between the package and the coldplate. Low fluid temperature can be controlled by effectively bypassing radiator area or eliminating cooling from capacitive cooling devices.

These graphs are useful for grouping equipment on coldplates, equalizing coldplate loads and setting fluid out temperature limits for individual coldplates.

2.3.2 Active Thermal Control

Heat balance for the current or baseline active thermal control method consists of a closed liquid loop with coldplates and integrally cooled packages were performed. For the initial investigation, the 200 nautical mile circular earth orbit and Configuration A of Figure 3 were selected. Six cases were completed and the various values and for each case are summarized in Table 2. As indicated in the table, three different coolant inlet temperatures and two different values for the IU outer skin absorptivities were used. The electronic package heat load of 150 watts was selected of the initial cases and was used for all sixteen coldplates. This assumption represents a thermally ideal case. The cases of different heat loads for various coldplates will be used in future investigations. The same internal thermal environment were used for all the cases. The temperature profile assumed for the S-IV B dome is given in Figure 14. The temperature of the structure on the forward end of the instrument unit was assumed at constant 40 F. Also, the IU skin temperature was assumed to be at 200 F at the point of injection into orbit.

The results of the heat balance for the six cases are presented in Figures 15 through 20 for 20 hours of orbit time.

The positive heat load indicates a net heat gain for the total closed loop and thus cooling would be required to bring the coolant temperature down to the design or selected value for the coolant inlet temperature to the coldplates and integrally cooled equipment. The negative heat load indicates a net heat loss and thus heating would be required to bring this coolant temperature up to the design or selected value.

In comparing these figures, the influence of the coolant inlet or design temperature and the absorptivity value of the IU outer skin become apparent. On the bases of these initial results, the lower value for absorptivity (0.18) for the IU outer skin appear to be desirable to

Table 2. Active Thermal Control Analysis Data

	200 n. mi. Earth Orbit, Launch June 21, 29° Incl., X Tan					
	Coolant Inlet 30°F		Coolant Inlet 50°F		Coolant Inlet 75°F	
	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
IU Structure						
Area/node, sq ft	8.5	8.5	8.5	8.5	8.5	8.5
Surface properties						
Outside: ,	0.18, 0.9	0.9, 0.9	0.18, 0.9	0.9, 0.9	0.18, 0.9	0.9, 0.9
Inside:	0.18	0.18	0.18	0.18	0.18	0.18
Coldplate						
Radiant interchange area, sq in	450	450	450	450	450	450
Emissivity,	0.18	0.18	0.18	0.18	0.18	0.18
Coolant flow, lb/hr	60	60	60	60	60	60
Electronic Package						
Radiant interchange area, sq in	450	450	450	450	450	450
Capacitance, Btu/°F	29.6	29.6	29.6	29.6	29.6	29.6
Joint Conductance, Btu/(hr)(F)	14.0	14.0	14.0	14.0	14.0	14.0
Case Emissivity	0.9	0.9	0.9	0.9	0.9	0.9
Heat Load, watts	150	150	150	150	150	150
S-IV B Dome						
Emissivity,	0.18	0.18	0.18	0.18	0.18	0.18
Temperature, F	Variable	Variable	Variable	Variable	Variable	Variable
FWD Structure						
Emissivity	0.18	0.18	0.18	0.18	0.18	0.18
Temperature, F	40	40	40	40	40	40
Area, sq in	20,100	20,100	20,100	20,100	20,100	20,100

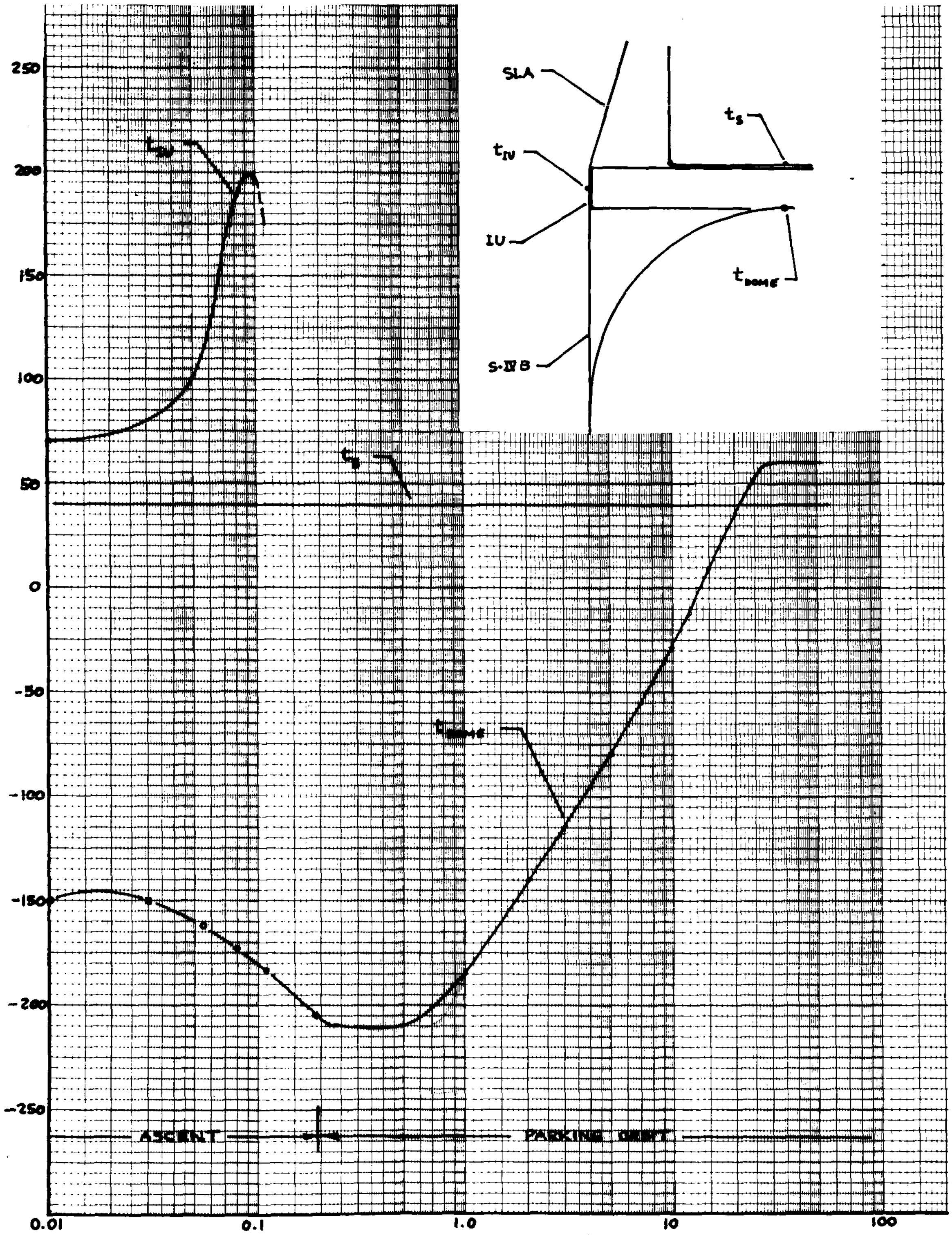


Figure 14. IU and Internal Structure Temperature

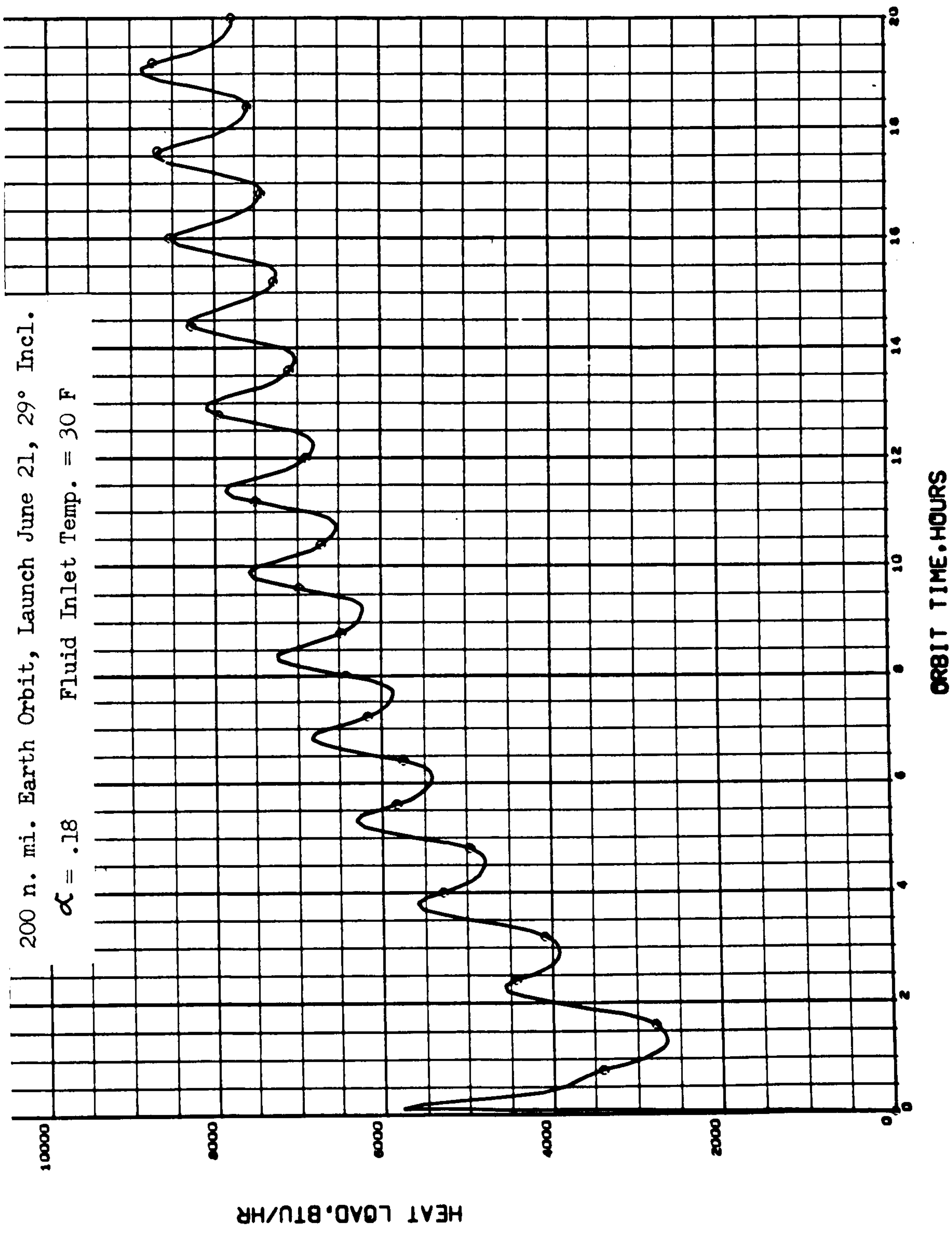


Figure 15. Heat Load Versus Orbit Time, Case 1

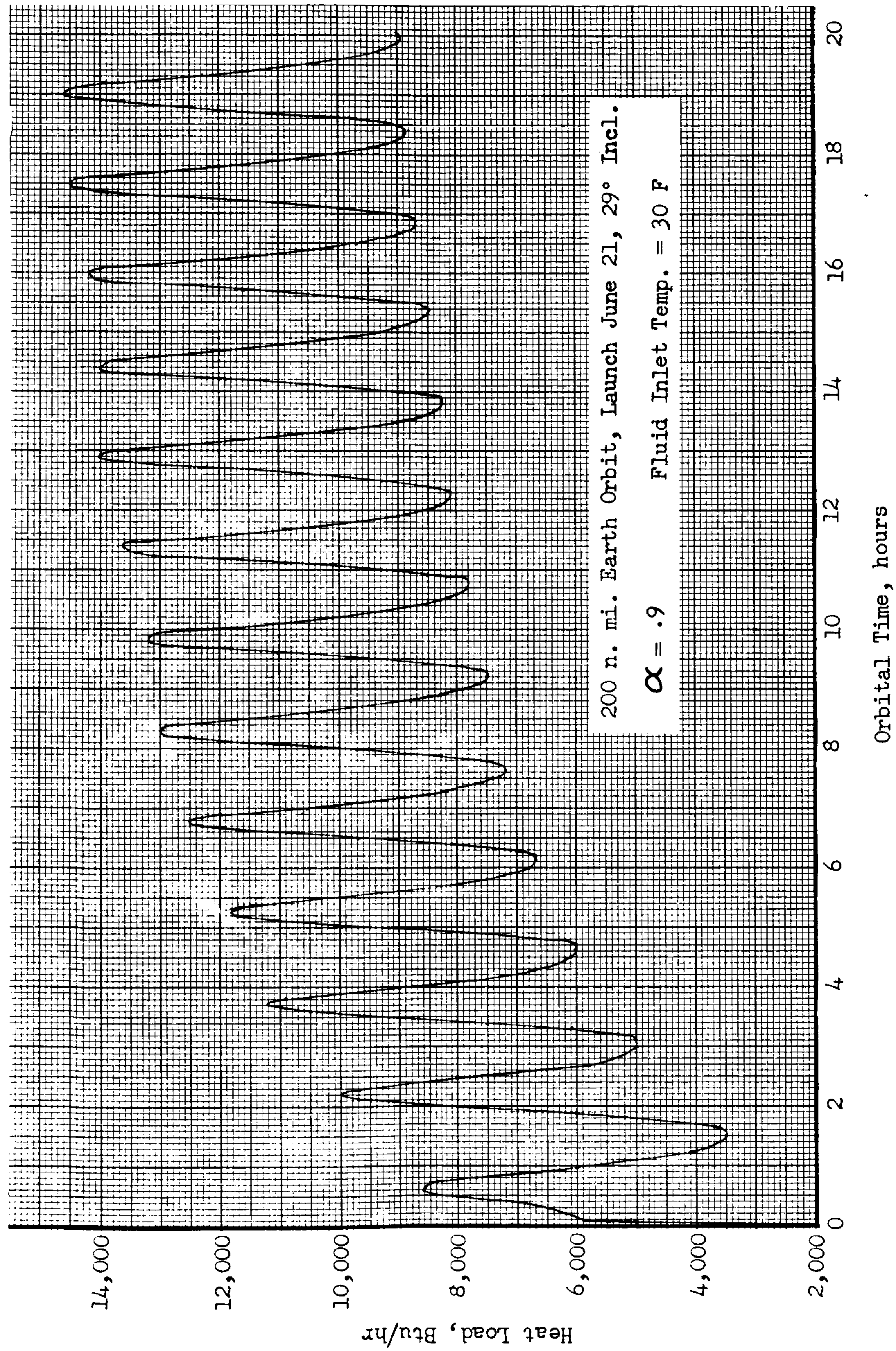


Figure 16. Heat Load Versus Orbit Time, Case 2

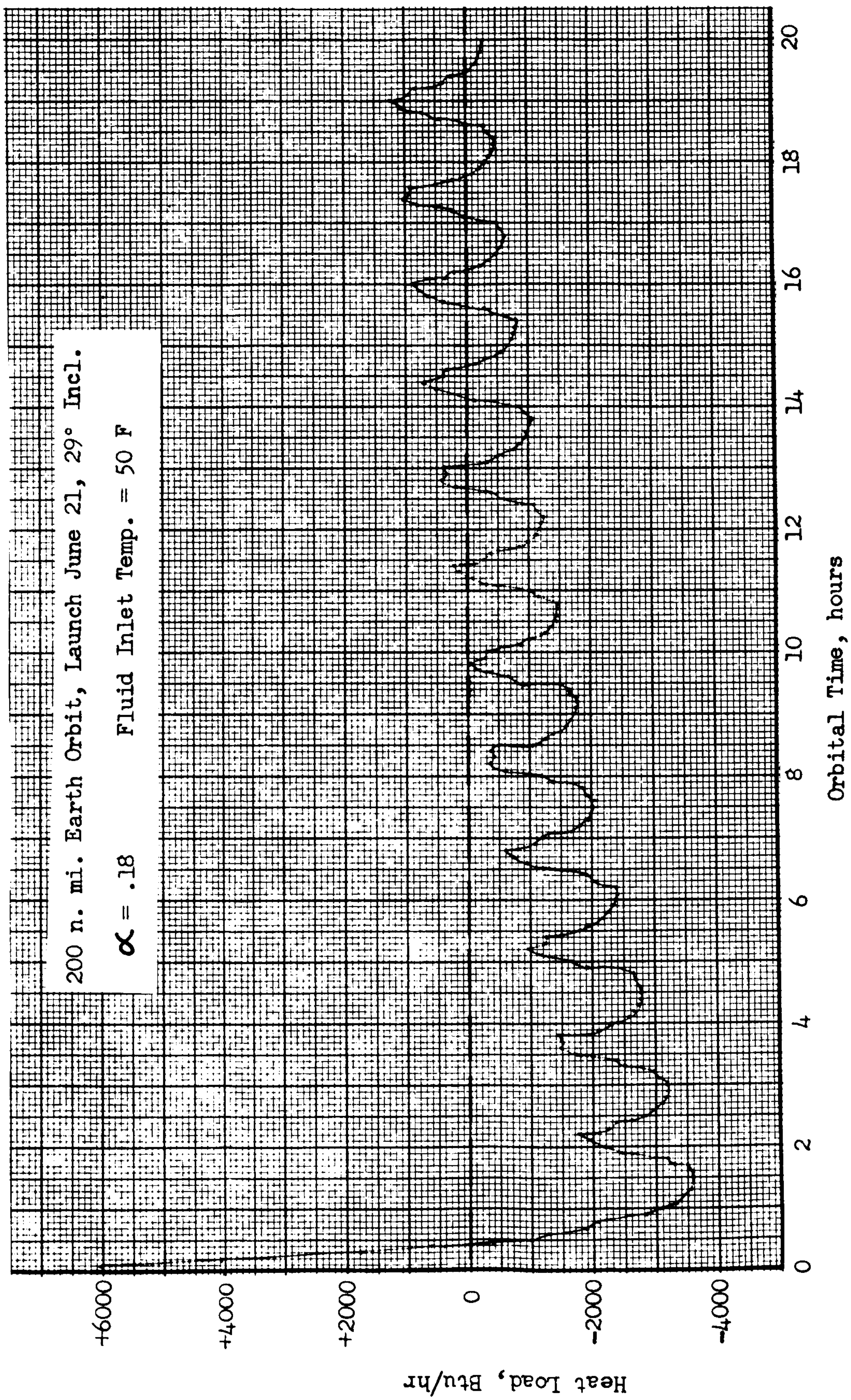


Figure 17. Heat Load Versus Orbit Time, Case 3

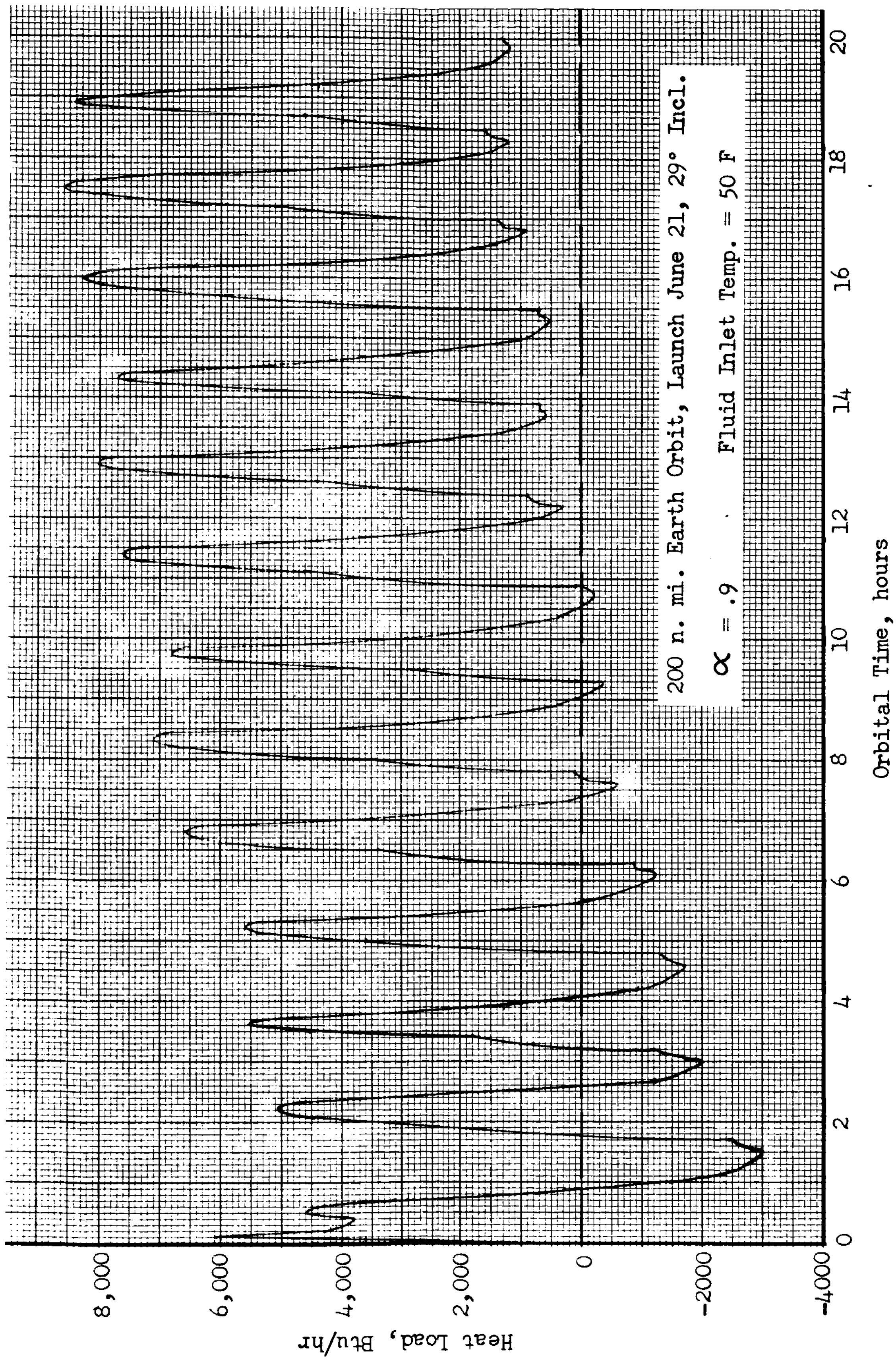


Figure 18. Heat Load Versus Orbit Time, Case 4

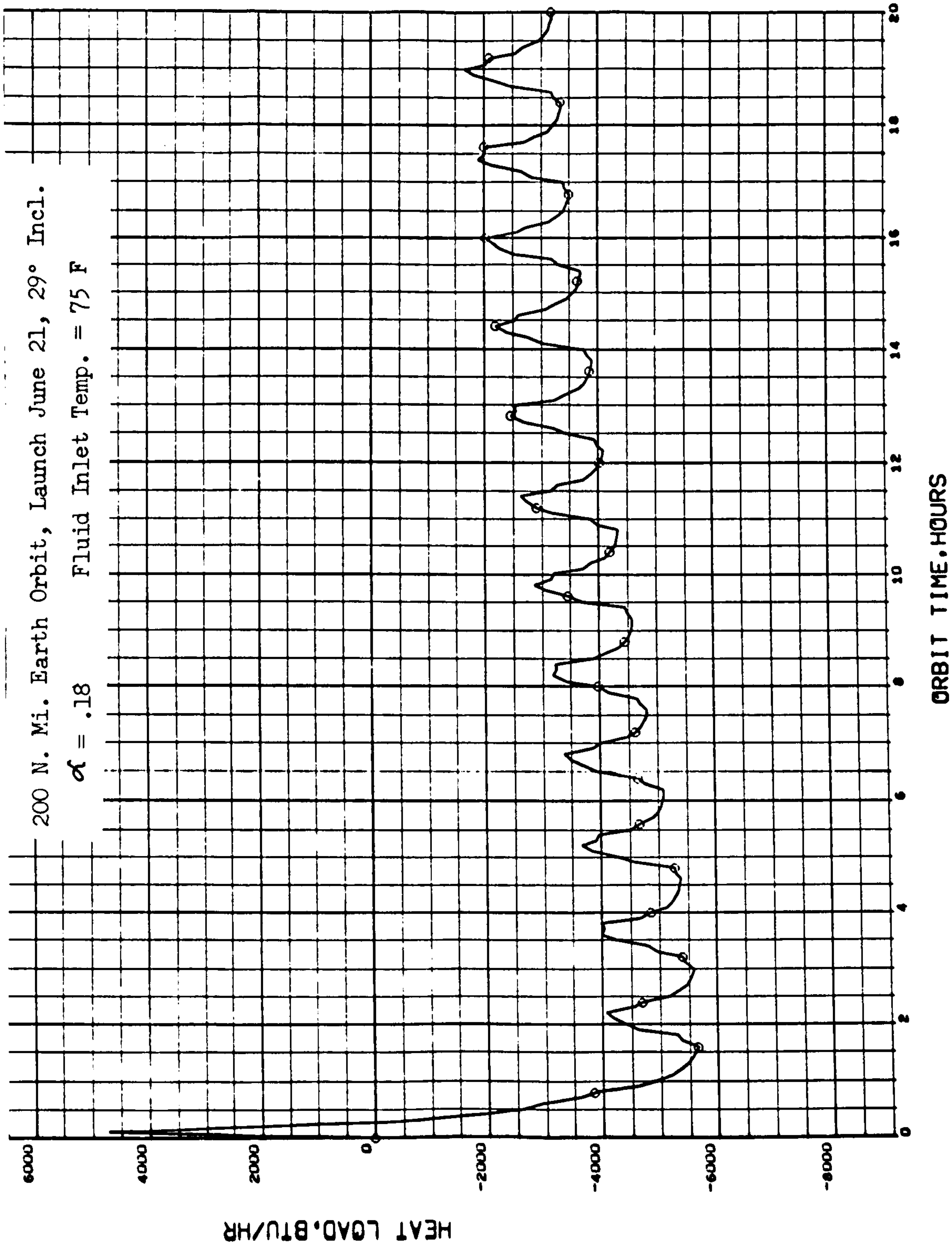


Figure 19. Heat Load Versus Orbit Time, Case 5

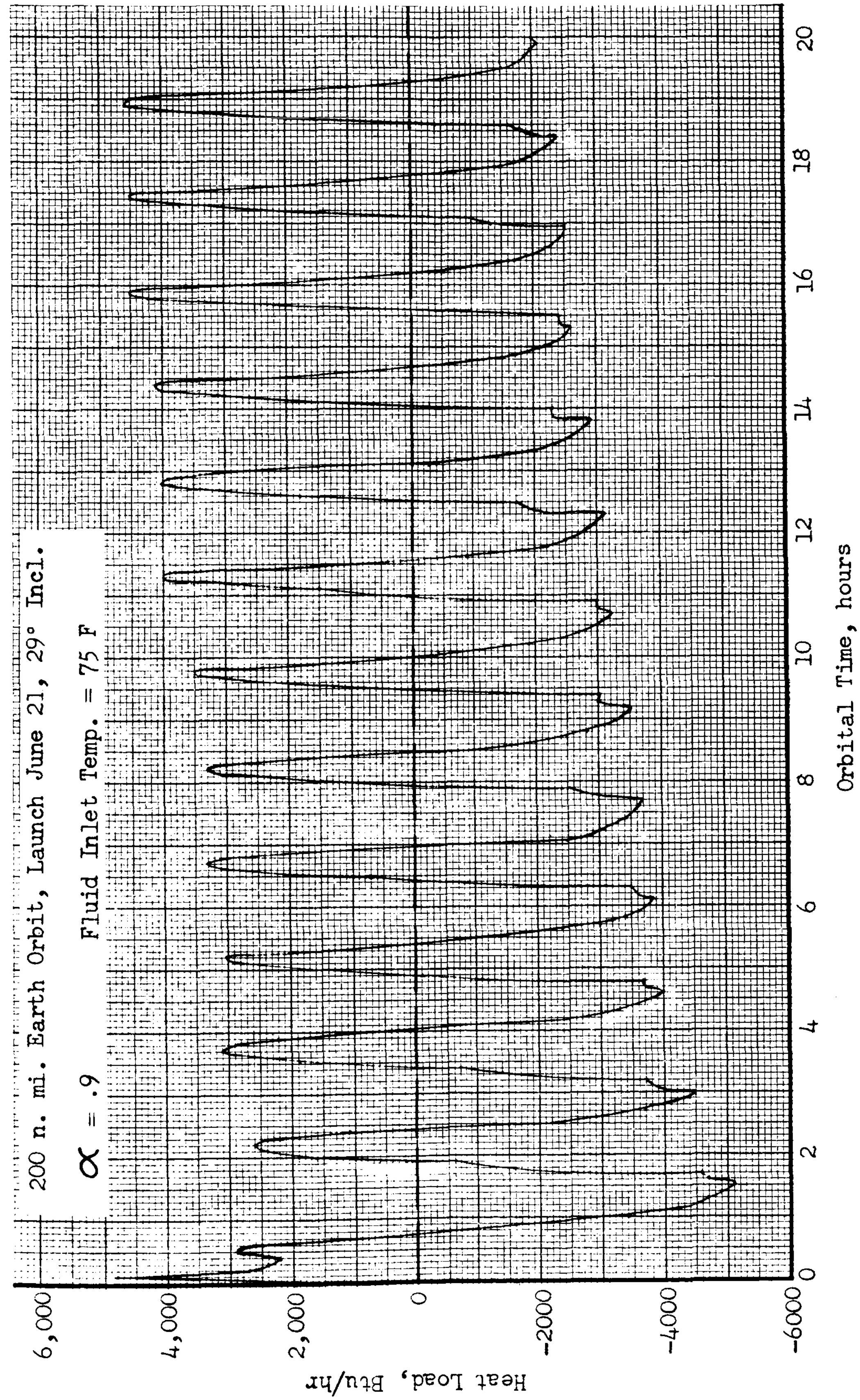


Figure 20. Heat Load Versus Orbit Time, Case 6

minimize the influence of the external environmental heat loads and thus reduce the heat load amplitude per orbit.

Figures 21, 22 and 23 are the heat loads for the orbital time period between 20 and 40 hours for Cases 1, 3 and 5. The figures indicate the heat loads are reaching equilibrium conditions, that is, the maximum and minimum values per orbit are approaching constant values. It is to be noted that for coolant inlet temperature of 30 F and 50 F, the coldplates as secondary radiators, are not sufficient to reject the waste seat and thus additional heat rejection method would be required. For coolant inlet temperature of 75 F, the opposite condition is indicated and heating would be required.

The variations in the heat load with orbital time suggests a range for the coolant temperature to minimize the heating or cooling requirements. During the first few hours in orbit, the coolant temperature between 30 and 50 F, possibly around 40 F, would be the lower limit and for the upper limit, a value between 50 and 75 F, possibly around 60 F, would be desirable. Before any temperature range can be selected, the equipment temperatures for the corresponding coolant temperatures needs to be considered.

For all the cases investigated, the equipment temperature continued to increase with orbital time and eventually reached an equilibrium condition. Table 3 illustrates the variation in the equipment temperature for the various IU locations and orbital time. It is to be noted that the equilibrium temperature attained may have exceeded the upper temperature limits for some of the equipment. Further analysis to be made is expected to give more data from which to draw more definite conclusions.

An additional item to be noted is the influence of the S-IV B dome temperature. The heat load variation with orbital time appears to have a similar profile to that of the dome temperature profile of Figure 14. Various approaches to minimizing the influence of the dome temperature are under consideration which include investigation of low emissivity values for the dome, the coldplate and the electronic packages, and the possible use of insulating materials. This will be done in conjunction with investigating alternate dome temperature profiles.

A general thermal network of the complete IU for closed liquid loop with coldplates and integrally cooled packages have been established to perform the complete heat balance with the aid of a digital computer.

An illustration of a portion of the total network is given in Figure 24, which is typical for the electronic package mounted on the coldplate. A single node was assumed to be a reasonable accurate representation of the electronic package, the coldplate and for a segment of the IU structure. The IU structure was divided into 24 equal segments and each represented by a single node and each node connected to the adjoining nodes by thermal resistors. A single node simulation of the electronic package has been assumed, but a two-node model will be considered if it appears

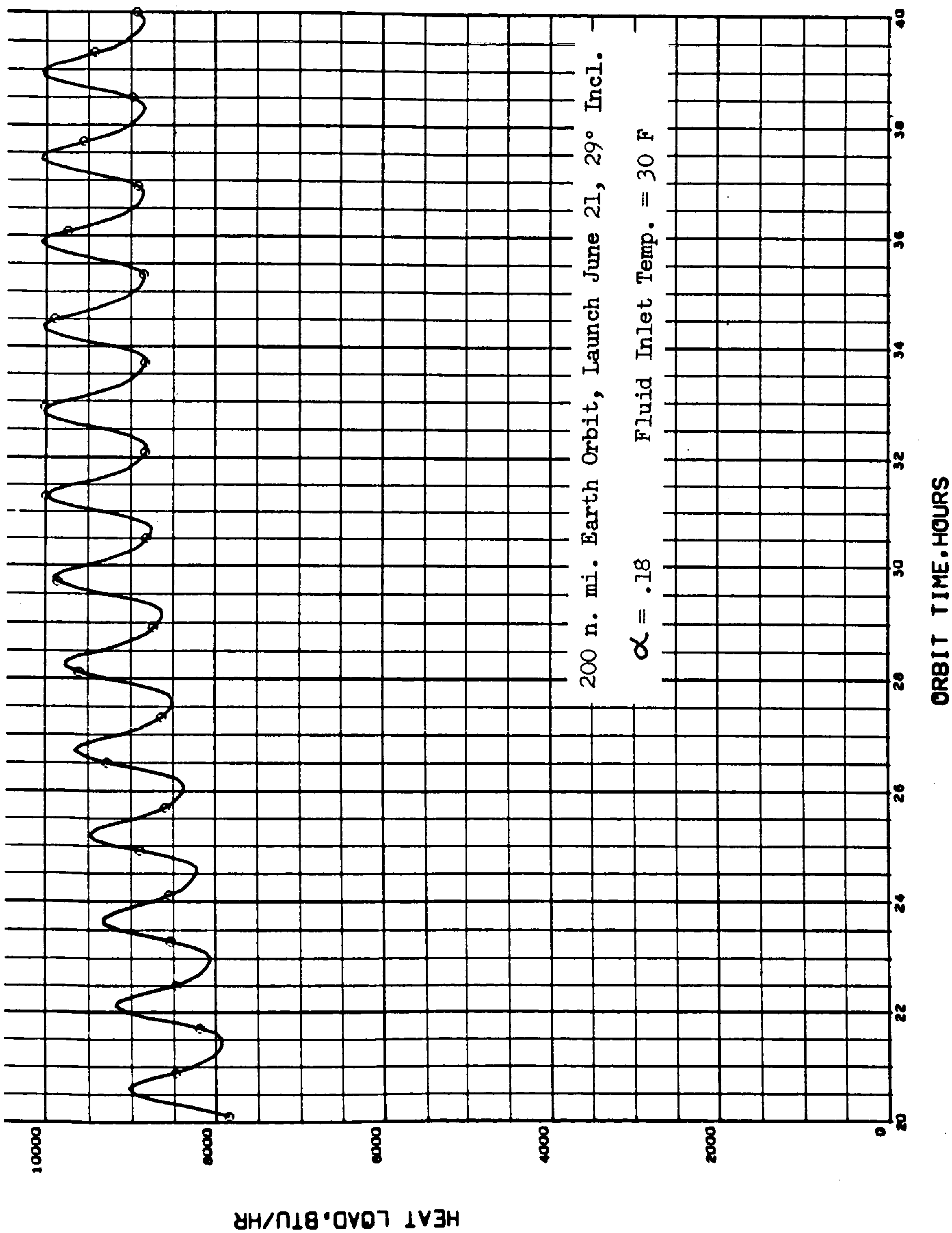


Figure 21. Heat Load Versus Orbit Time, Coolant Temperature 30F

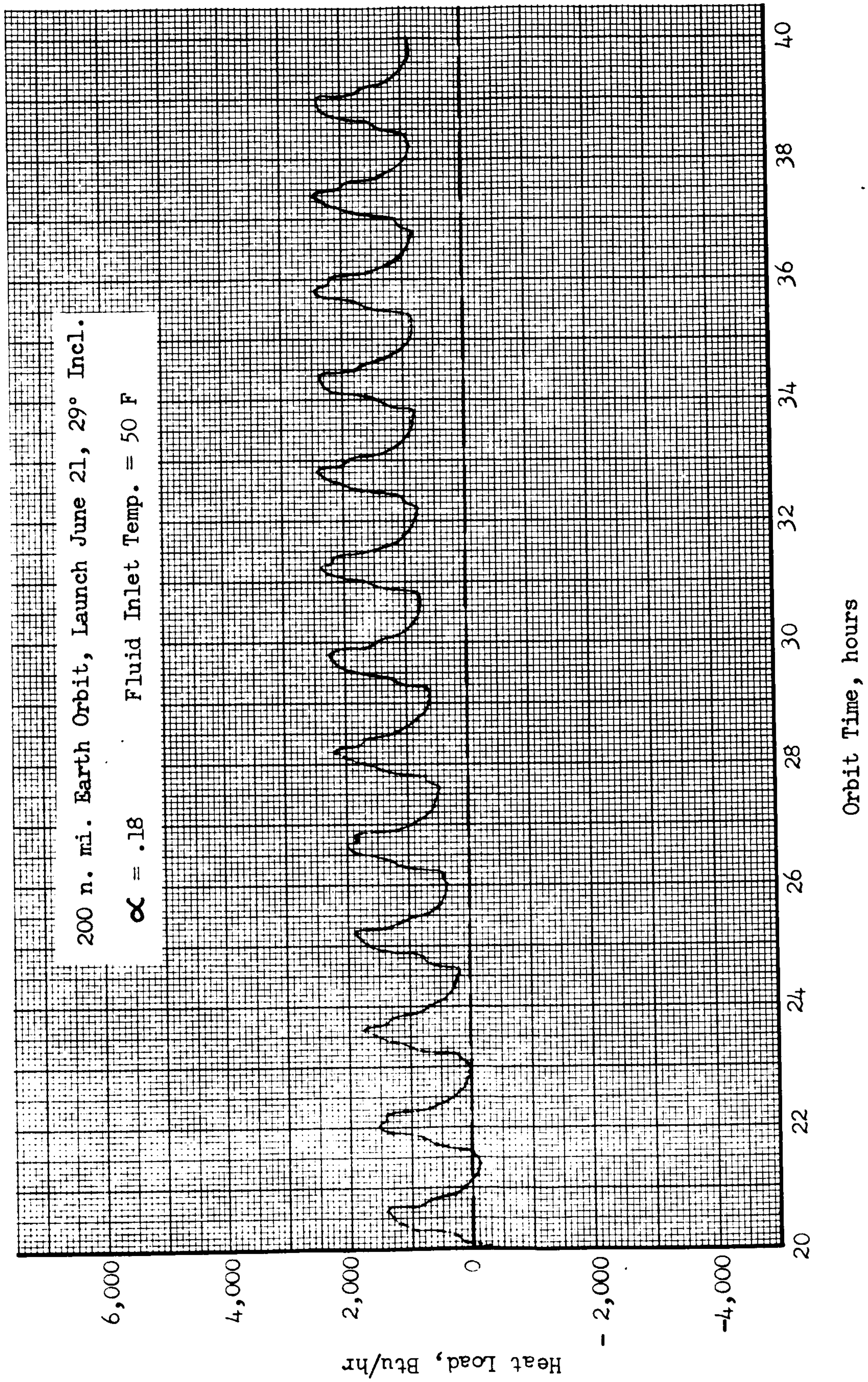


Figure 22. Heat Load Versus Orbit Time, Coolant Temperature 50 F

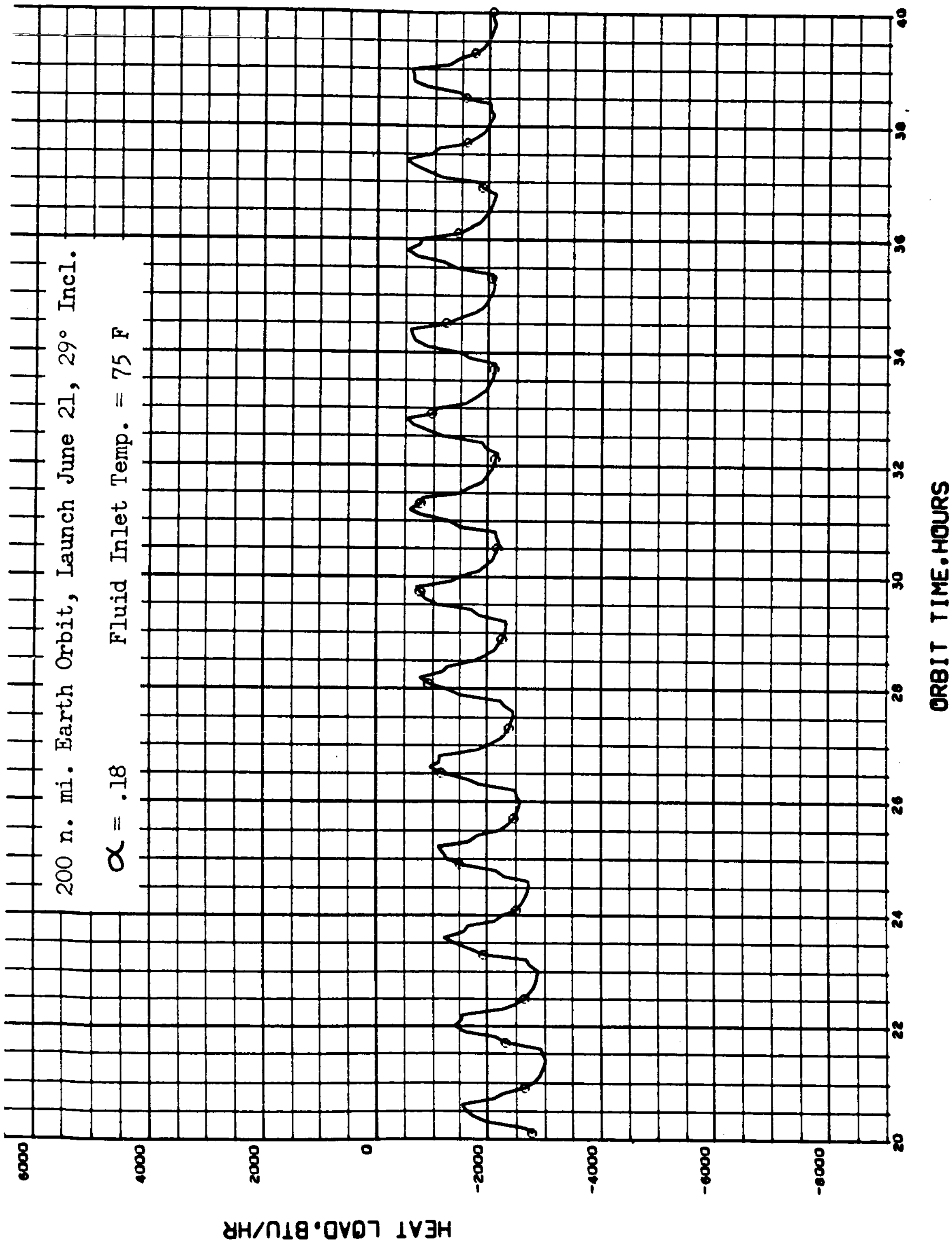


Figure 23. Heat Load Versus Orbit Time, Coolant Temperature 75F

Table 3. Equipment Temperature, Case 3
(Coolant Inlet, 50 F)

IU Location	Equipment	Equipment Temperature, F					
		Orbit Time, hours					
		0	1.5	10	20	30	40
1	Coldplate Mounted	50	60.3	74.6	80.7	85.2	85.8
2	Coldplate Mounted	50	60.3	74.37	80.5	85.0	85.6
3	(Blank)	—	—	—	—	—	—
4	Coldplate Mounted	50	60.3	74.4	80.5	85.0	85.6
5	Coldplate Mounted	50	59.3	71.7	77.90	82.3	83.0
6	(ECS)	—	—	—	—	—	—
7	(Imbilical)	—	—	—	—	—	—
8	(Access door)	—	—	—	—	—	—
9	Coldplate Mounted	50	57.4	66.7	72.5	77.4	77.9
10	Coldplate Mounted	50	57.6	67.2	72.9	77.8	78.3
11	Coldplate Mounted	50	57.6	67.3	72.6	78.0	78.0
12	Coldplate Mounted	50	57.6	67.3	72.6	78.0	78.0
13	Coldplate Mounted	50	57.8	67.9	72.9	78.5	78.3
14	Coldplate Mounted	50	57.8	67.8	72.9	78.5	78.2
15	Coldplate Mounted	50	57.5	66.7	72.0	77.4	77.3
16	Flt. Control Computer	50	37.8	45.2	52.1	57.9	57.8
17	Coldplate Mounted	50	57.1	65.4	71.0	76.1	76.3
18	Coldplate Mounted	50	57.1	65.5	71.1	76.2	76.5
19	LVDA/LVDC	50/50	61.9/46.4	70.9/54.0	75.5/59.6	79.1/64.0	79.3/64.1
20	Coldplate Mounted	50	56.8	64.5	70.5	75.2	75.9
21	ST-124 Platform	50	42.8	46.8	53.0	57.7	58.1
22	(GN_2 Supply)	—	—	—	—	—	—
23	Coldplate Mounted	50	59.1	71.2	77.6	81.9	82.6
24	Coldplate Mounted	50	59.3	71.9	78.0	82.6	83.6

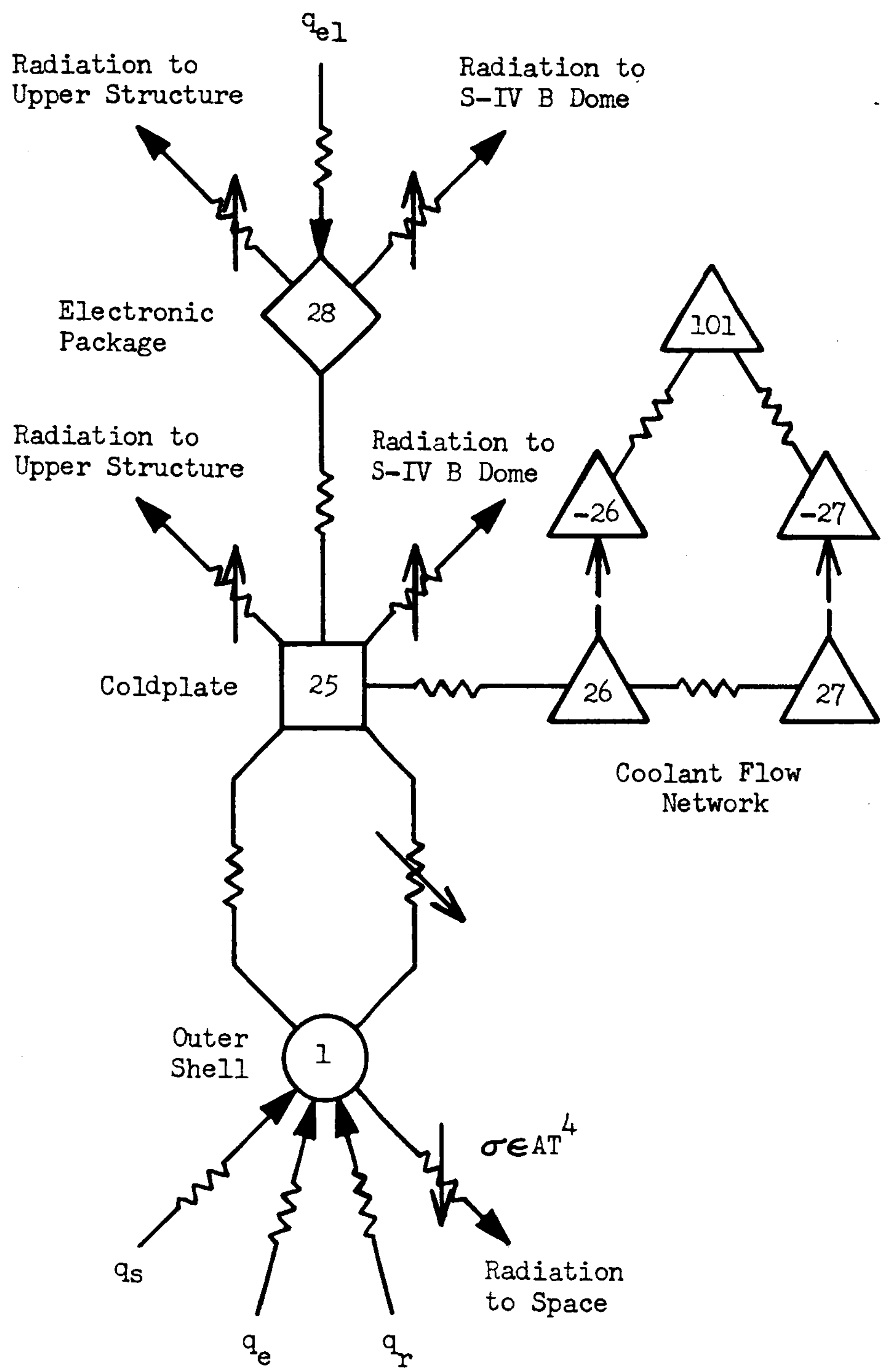


Figure 24. Thermal Network, Equipment Mounted on Coldplate

necessary for better accuracy. Also, it was assumed that the temperature difference between the IU and adjacent structures were sufficiently small and thus could be neglected.

As indicated in Figure 24, the external heat sources considered are: direct solar energy (q_s), planetary thermal radiation (q_e) and planetary albedo (q_r). The external heat sources were treated as separate components for easier application of the network for a range of values for emissivity (ϵ) and absorptivity (α) of the outer IU skin.

Internal heat sources consist of internal electronic package heat dissipation and radiative heat exchange between the packages and S-IV B dome and upper structure.

Computer output gave the temperature histories for given orbital condition for each node represented in the thermal network as well as net heat loss or gain of each coldplate or internally cooled package liquid cooling loop. The net heat loss or gain of total liquid cooling loop system was also computed directly.

2.3.3 Passive Thermal Control

Thermal control by pure passive means was investigated for which the electronic package was considered to be mounted on the inner surface of the IU structure. Four cases were investigated for a 200 nautical mile circular earth orbit at 34° inclination, and the longitudinal axis tangent to the flight path. For each of the cases, four different IU locations were selected for the skin mounted electronic package to determine the influence of the different environmental heat loads. At the point of injection into orbit, the skin temperature was assumed to be 200 F and the electronic package temperature to be 100 F. The results of the analysis are given in Figures 25 through 28, for the four cases.

For the four cases, the results indicate the possibility of passive thermal control for direct mounted equipment. Although the analysis was limited to 20 hours of orbit time, there is an indication that the temperature was approaching equilibrium condition. Further analysis for this configuration is planned after the case of the coldplate mounted with no coolant flow is investigated.

The various values used for the four cases are given in Table 4.

The thermal network simulating the electronic package mounted directly on the inner surface of the IU structure is illustrated in Figure 29. It was assumed that the heat transfer along the skin was radilly and uniformally so the IU structure was divided into five concentric rings (maximum radius, 18 in.). Each ring is represented by a single node. Node 2 of Figure 29 represents the disk on which the electronic package is mounted. For the initial analysis, electronic package mounted on the

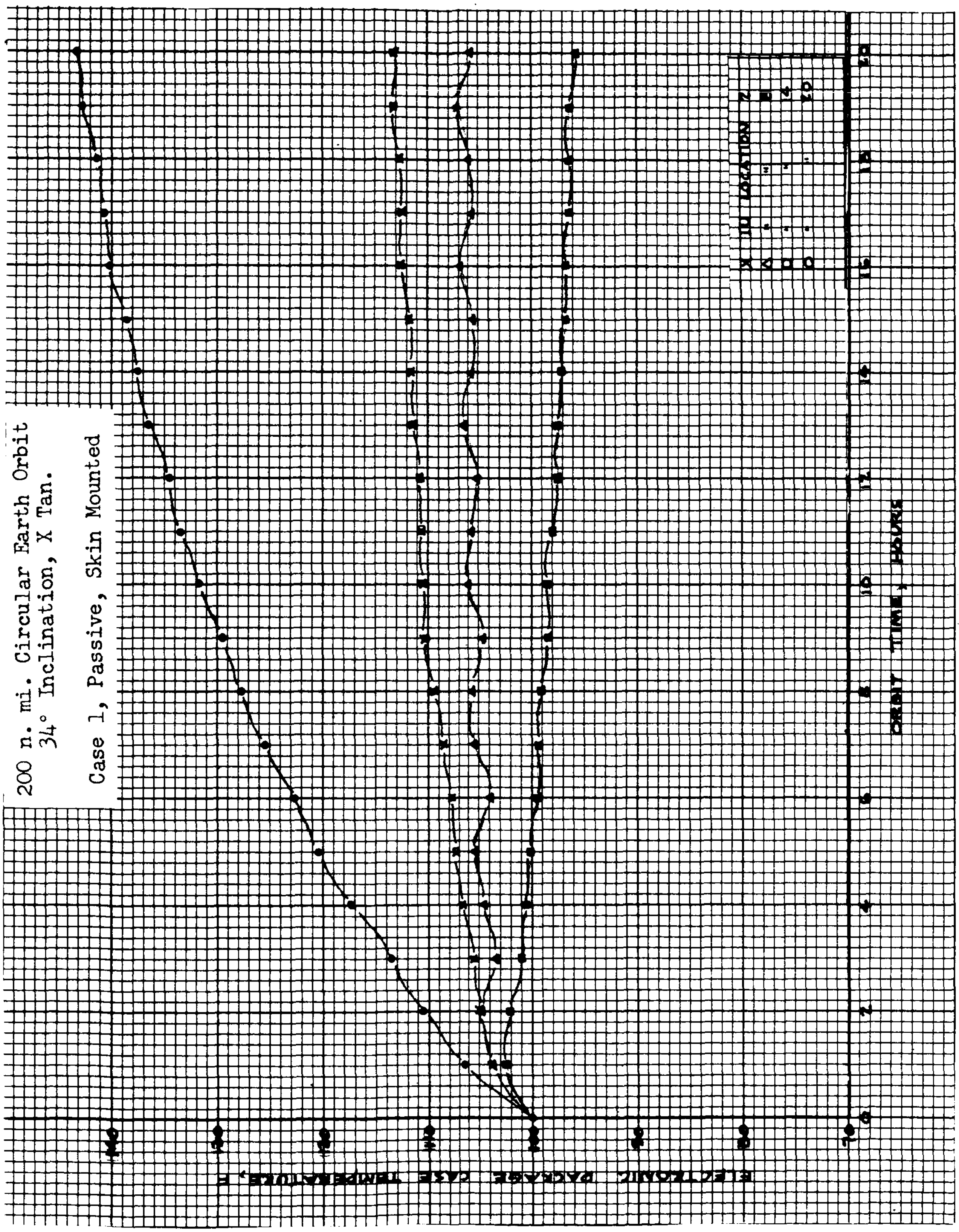


Figure 25. Passive Control, Skin Mounted, Case 1

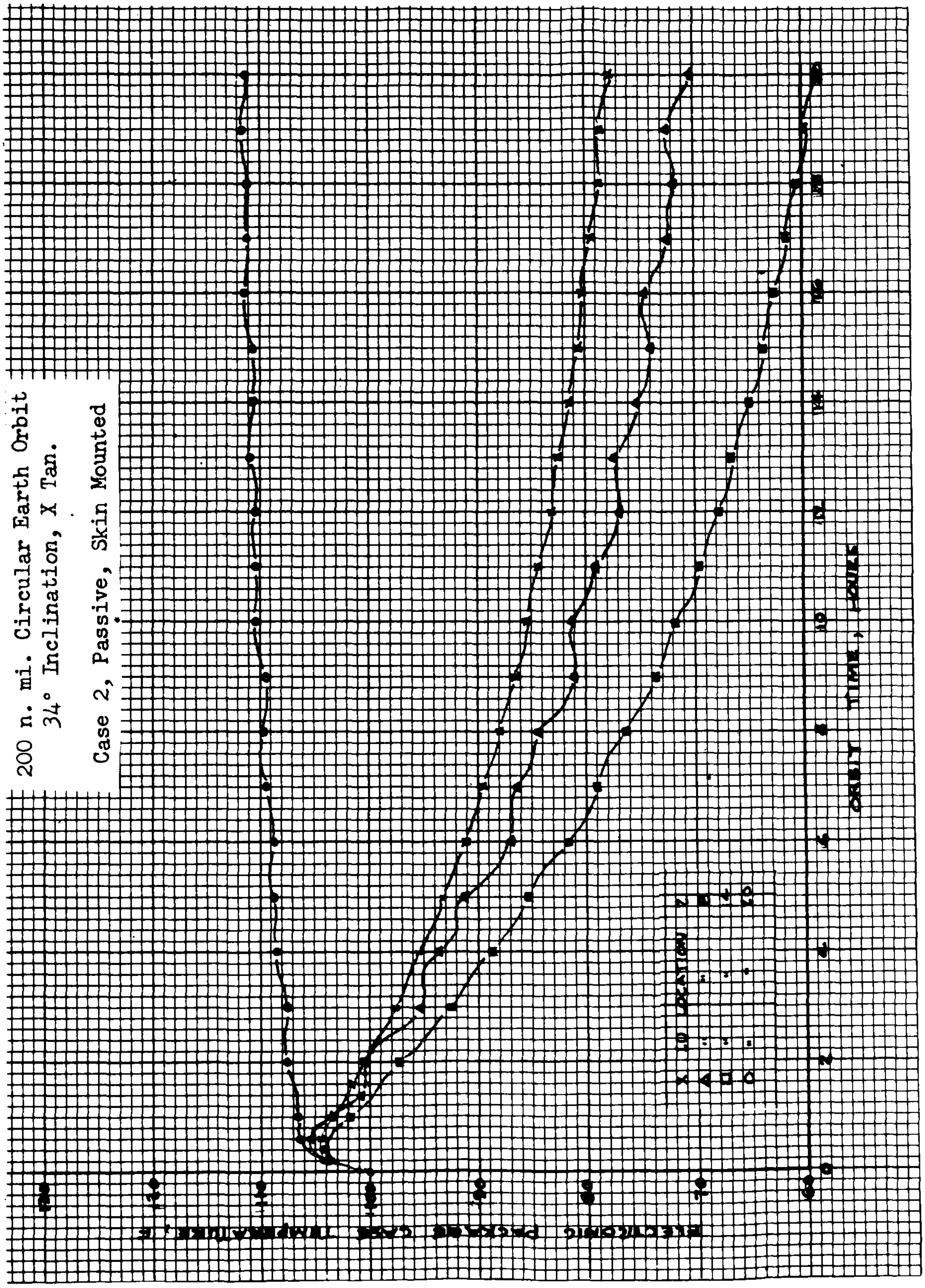


Figure 26. Passive Control, Skin Mounted, Case 2

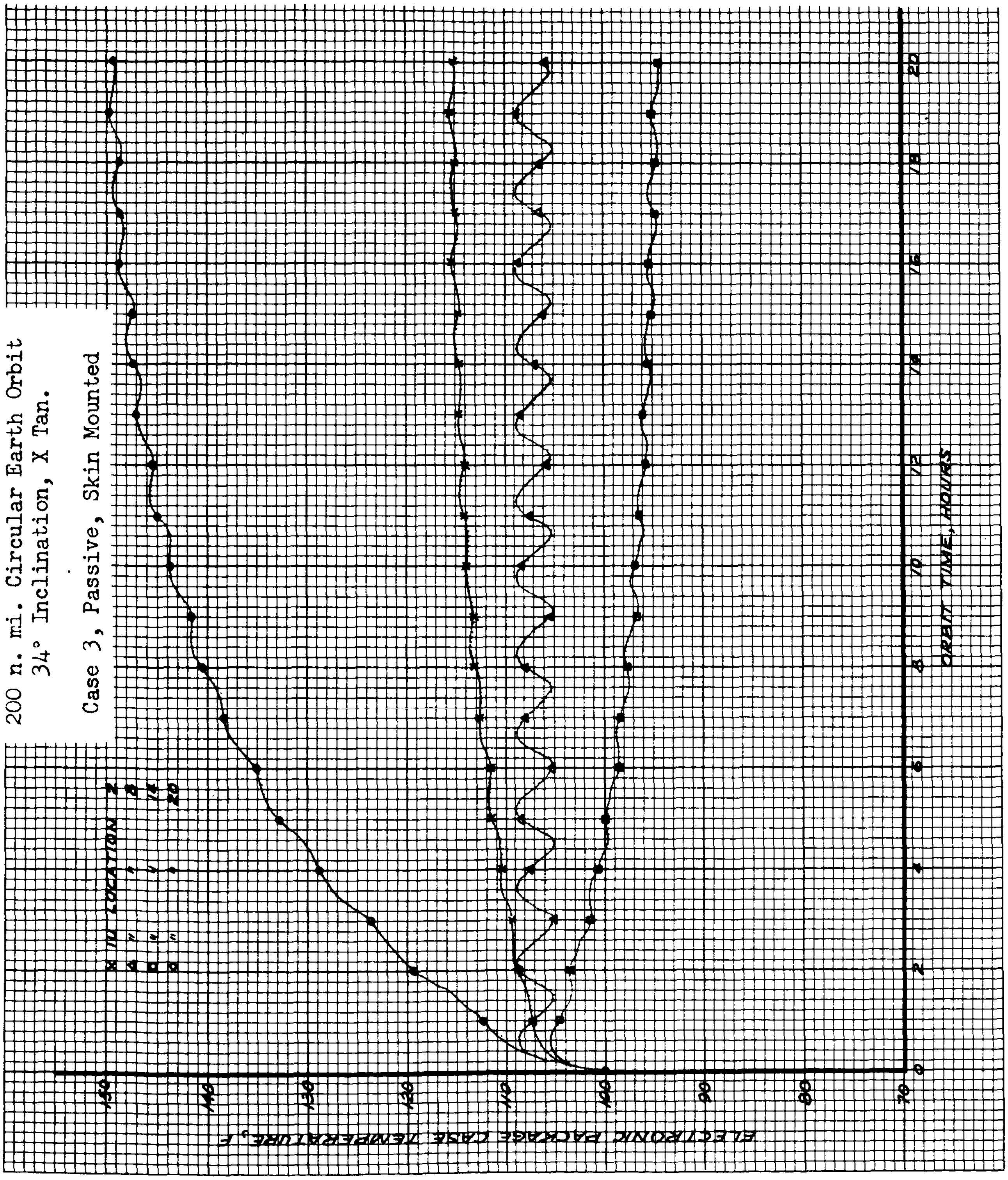


Figure 27. Passive Control, Skin Mounted, Case 3

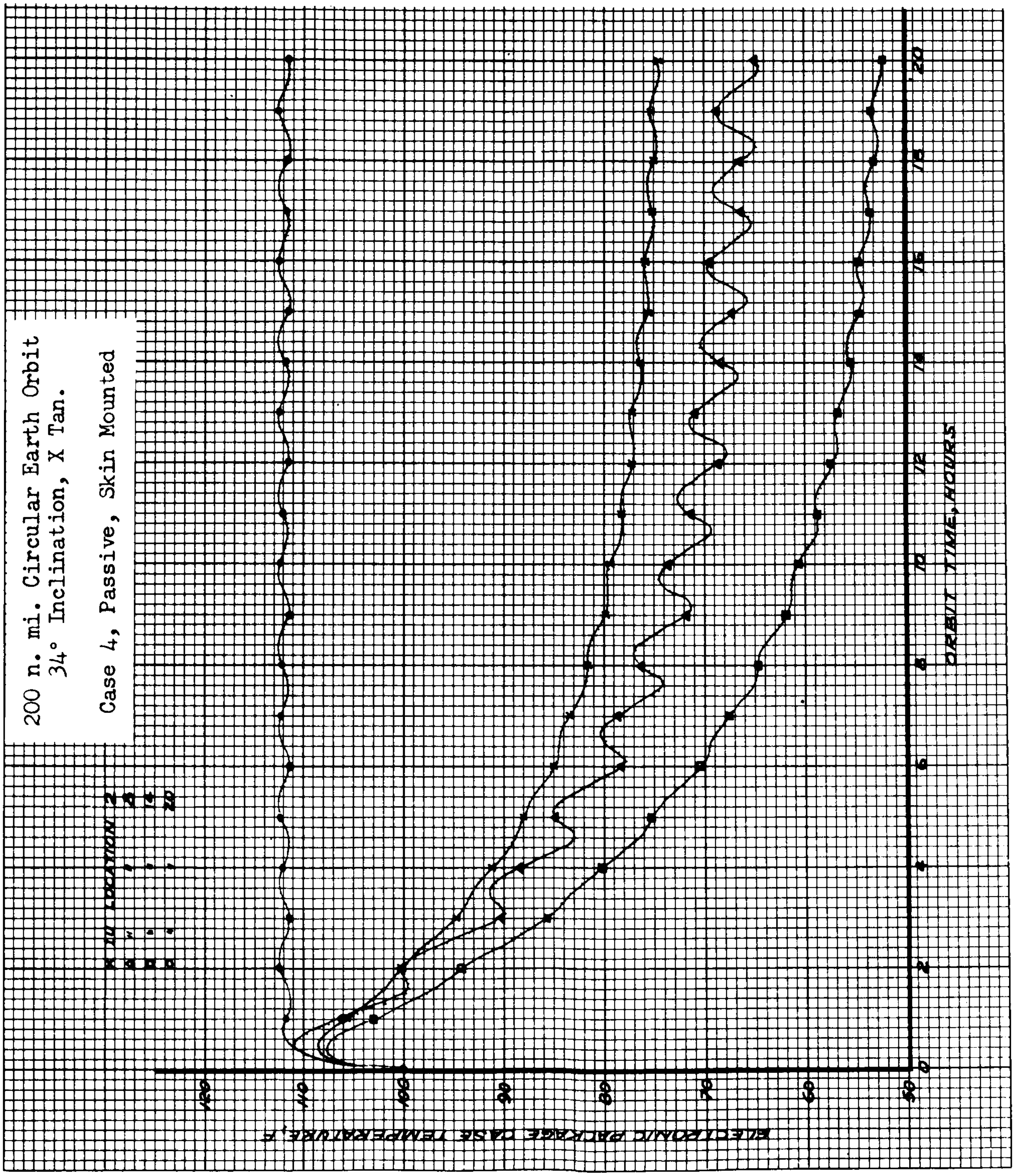


Figure 28. Passive Control, Skin Mounted, Case 4

Table 4. Passive Thermal Control Analysis Data

	Electronic Package $W \bar{C}_p = 29.6$		Electronic Package $W \bar{C}_p = 14.8$	
	Case 1	Case 2	Case 3	Case 4
Equivalent IU Skin Thickness, in.	0.05	0.10	0.05	0.10
Equivalent Skin Thermal Conductivity, k, Btu/(hr)(ft)(F)	100	100	100	100
Surface Property, ϵ α	0.90 0.18	0.90 0.18	0.90 0.18	0.90 0.18
Joint Conductance Equipment-to-Skin	27.32	54.64	27.32	54.64
Initial Skin Temp. F	200	200	200	200
Initial Electronic Package Temp. F	100	100	100	100
Equipment Heat Load, q_{el}, Btu/hr	400	400	400	400
Electronic Package Size, in.	10 x 12	10 x 12	10 x 12	10 x 12

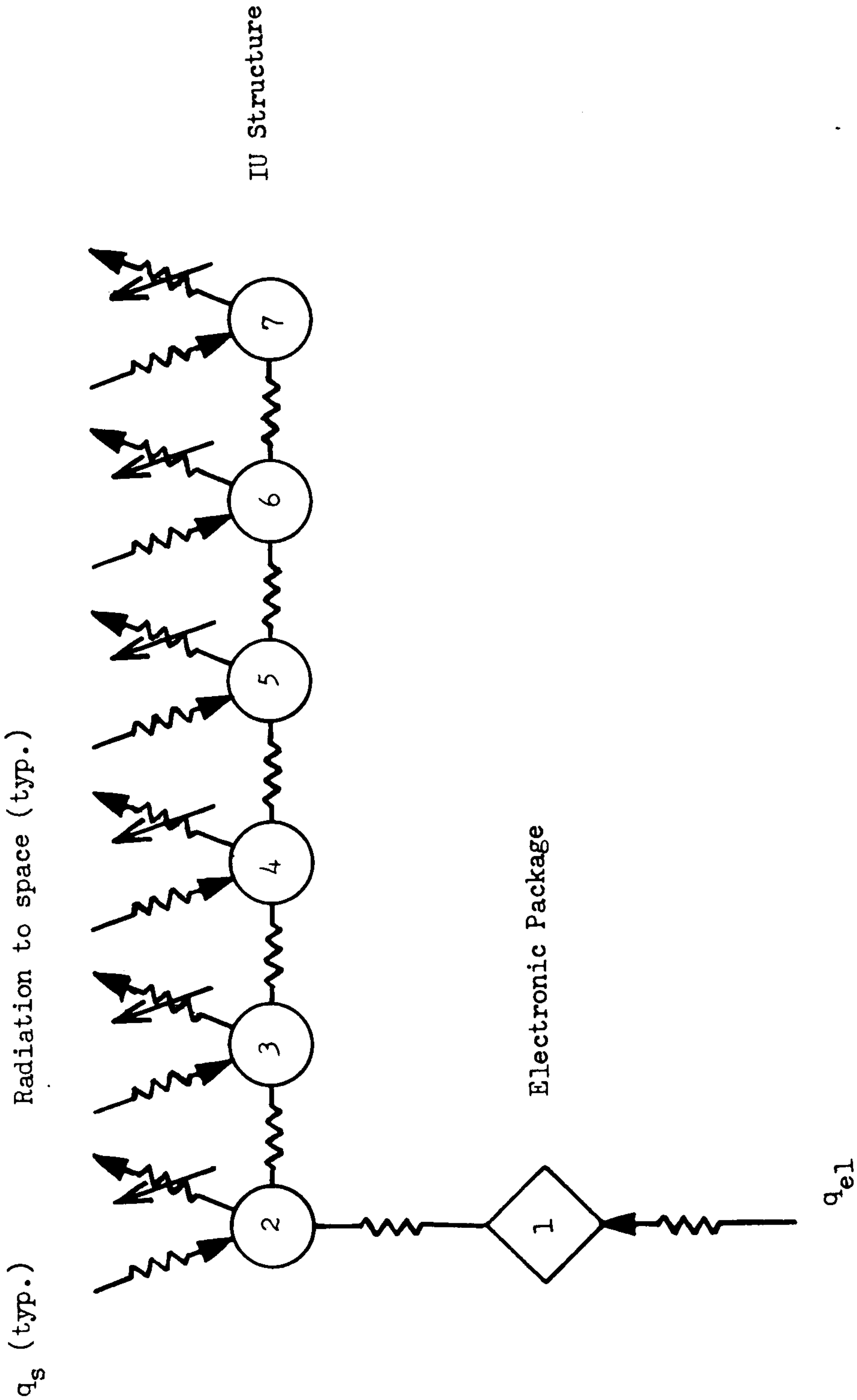


Figure 29. Thermal Network, Equipment Skin Mounted

structure was considered to be isolated from other thermal environments, such as the S-IV B dome. For future analysis, a more complete thermal environment will be considered.

2.3.4 Problem of Meteoroid Penetrations

The potential problem of meteoroid damage to the environmental control system must be considered and properly dealt with in order to achieve the high reliability required for long duration missions. Although much of the environmental control equipment are located within the instrument unit and thus assumed to have sufficient protection, there is a possibility during certain mission periods that the forward end of the instrument unit may be exposed directly the space environment. Also, for long duration missions, space radiators appear necessary and being located on the external surface, are vulnerable to meteoroid damage.

When the forward end of the instrument unit is exposed to the space for a period of time, such as assembly or disassembly in space, the most vulnerable components appear to be the coldplates. The coldplates with their very large fluid cooled surface areas may be considered to be exposed to half the normal flux of meteoroid particles. Figure 30 shows that the probability of meteoroid penetration becomes a critical item in a very short time. It also shows that there is an increasing probability of meteoroid puncture of the coldplate with time even when protected by the present IU structure and other surrounding structure. The most severe assumptions possible from existing data were used in preparation of Figure 30. Even if conservative by an order of magnitude, a redesign of coldplates may be necessary for the longer duration missions or alternate means for protection be provided.

The other possible component which would be vulnerable is the fluid duct which is located around the periphery of the instrument unit. For the portection of the ducts and coldplates, a meteoroid shield could be incorporated with minimum weight penalty. Figure 31 illustrates the possible weight increase of the total sixteen coldplates with meteoroid protection as a function of mission duration and probability of penetration.

Since the fluid passages or ducts are the most critical parts of the radiator vulnerable to possible meteoroid puncture, various methods for providing protection have been proposed by various investigators.

The use of protective armour or a meteoroid shield for the ducts have been suggested. Another approach is to use redundant radiators, but this may not be practical in most cases because of area limitations. As modifications, the use of redundant, parallel tubes could be considered, which will involve a completely separate radiator fluid loop.

Figure 32 shows the effect of increasing reliability and mission duration on the weight of the radiator with protective armour. It is important to note that the larger the required heat rejection the greater the weight per unit heat rejection. This important consideration result from two

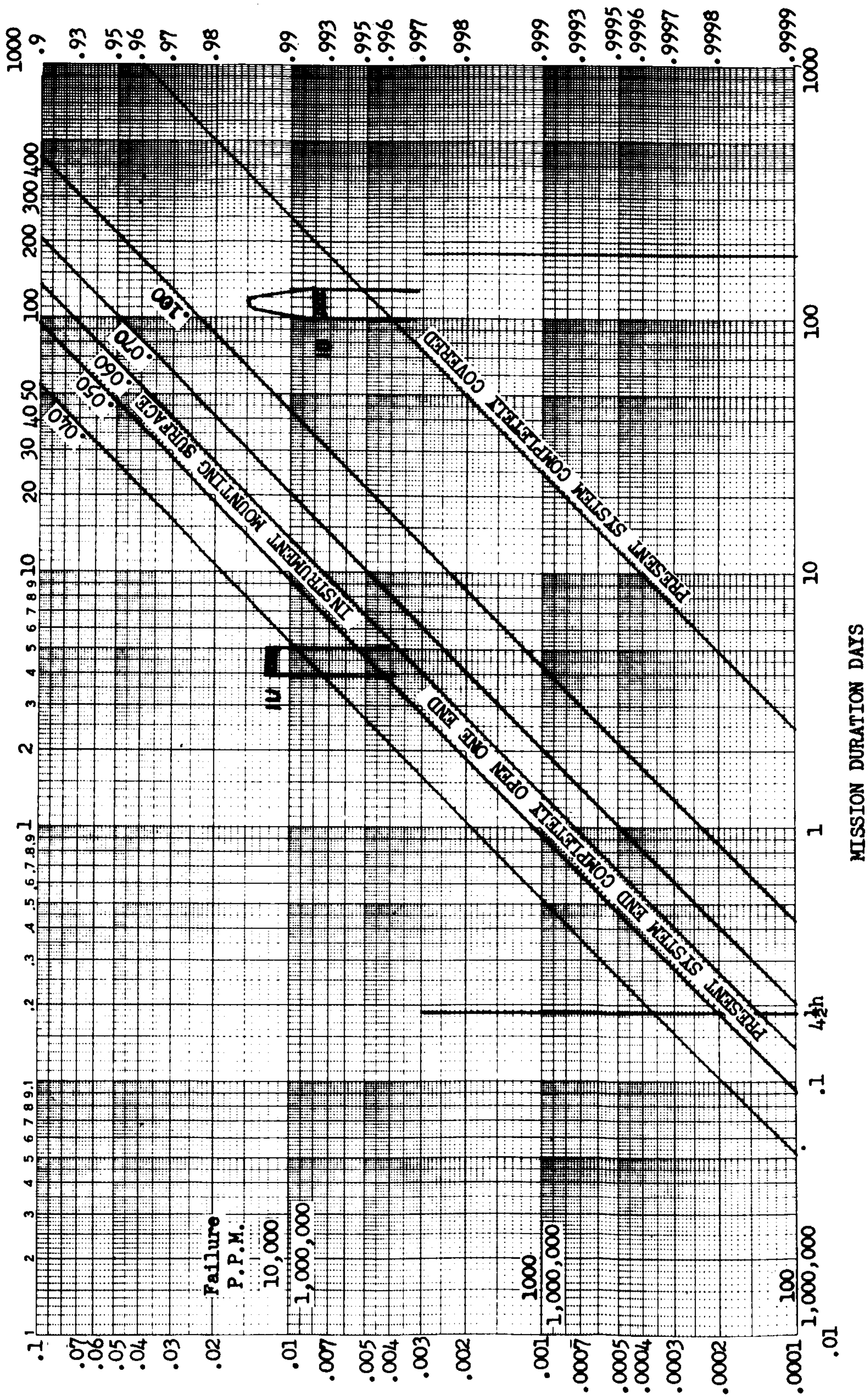


Figure 30. Probability of Meteoroid Penetration of Coldplate

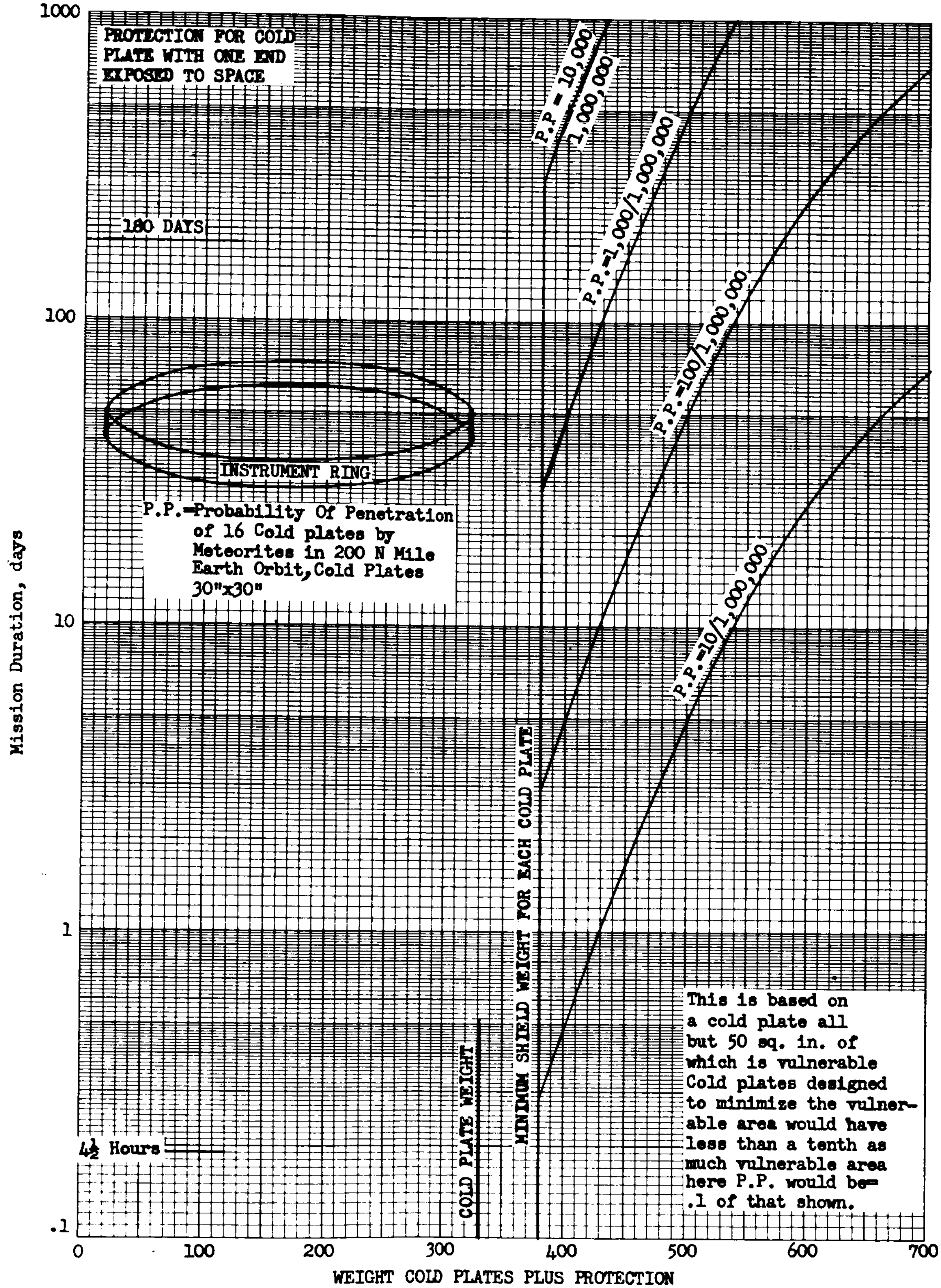


Figure 31. Coldplate Weight With Meteoroid Protection

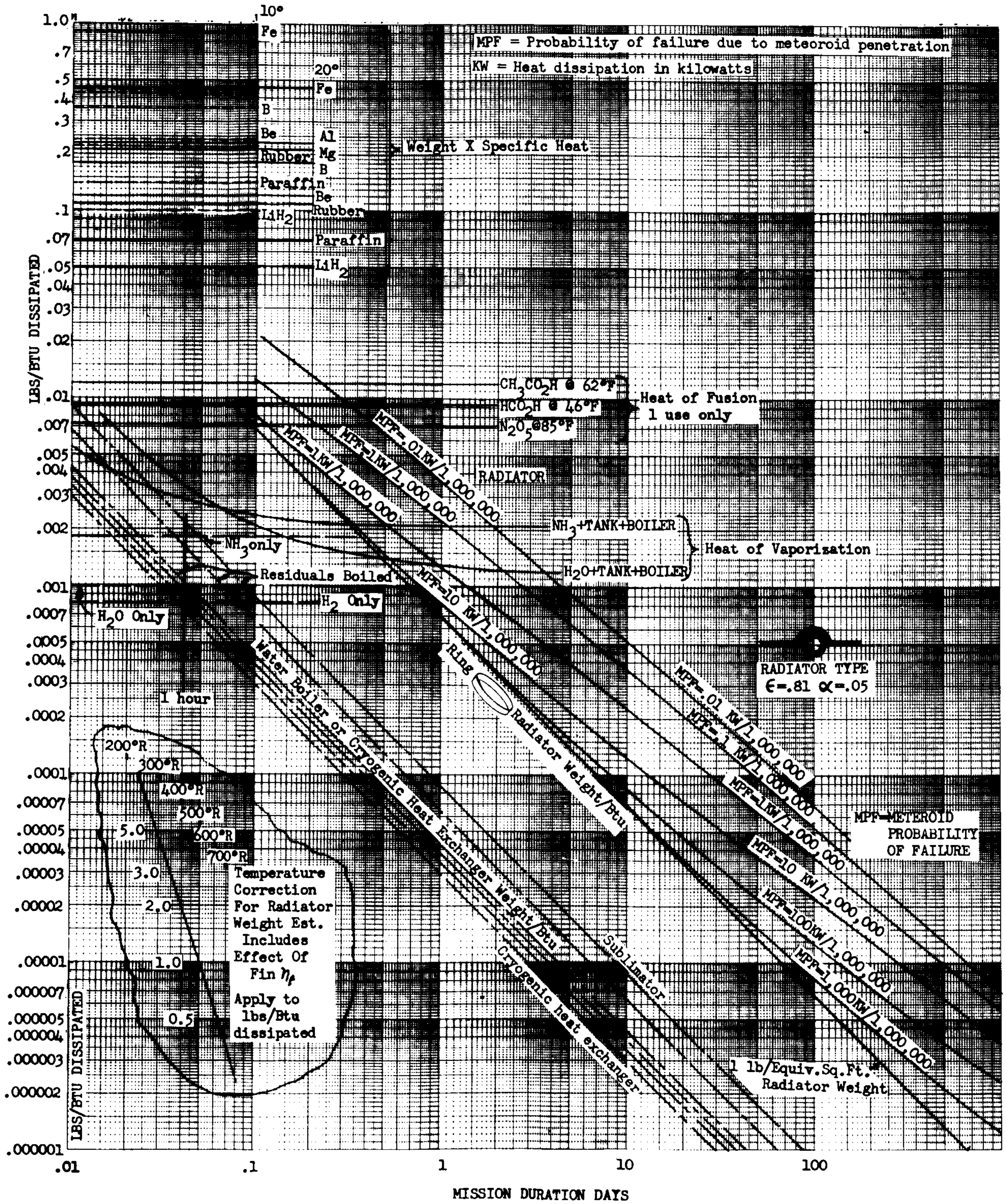


Figure 32. Heat Sink Weight Estimates

facts: first, the heat rejection area is directly proportional to the heat rejection and secondly, the probability of penetration is directly related to the area. It is to be noted, that even though the radiator weight increases with mission duration, it still results in a minimum weight method for heat rejection.

3.0 FORECASE FOR NEXT QUARTER

Since next reporting period is the final quarter, the major effort will be devoted first, to completing the technical investigation and secondly, to the preparation and submittal of the final technical report.

The technical investigation will be the continuation and completion of the combined astrionic equipment and environmental control system analysis to provide the necessary data in parametric form. The various orbits and vehicle orientation, IU/vehicle configurations, equipment heat loads, and IU compartment thermal environments will be considered. In addition to these, a range of values for the coolant flow rate, coolant temperature and surface properties will be investigated. The result of this analysis will provide the basis for the evaluation and selection of the system concepts.

Various system concepts for extending the operational life of the current ECS concept (the baseline system)will be evaluated and selected primarily on the basis of performance, weight and volume. Other consideration will include the relative cost and system reliability in regards to recommended modifications to improve the baseline system to meet the operation time of $4\frac{1}{2}$ hours to 180 days. The selected system concepts shall represent a logical evolution from the current concept.

A final technical report will be prepared and submitted in accordance with contractual requirements.

APPENDIX A

TEMPERATURE DROP IN COLDPLATE FROM CONTACT SURFACE TO FLUID

Heat is transferred from the contact face of a coldplate to the fluid thru the metal surface of the coldplate. The contact resistance has been shown to be small for properly tightened bolted joints (Reference 1A). The referenced article also points out the contact is essentially circular. However, for the heat to be transferred along the metal and then by convective coefficient to the heat transport fluid, there is an appreciable temperature drop. In order to determine the temperature drop, a computer program was used.

A thermal model was set up to represent the transfer of heat from a surface contact of some particular diameter thru the coldplate to the flowing fluid. This is illustrated in Figure 1A. The plate surface is divided into an inside diameter (to which the heat will be added) and a number of concentric diameters, each larger than the previous one by a factor 1.111. The areas between the concentric circles are all assigned separate nodes. The area inside the inner circle and the fluid also are assigned node numbers. Node 1, the largest node, is 20.00 in. outside diameter, and 18.0 in. inside diameter. Node 2 is 18.0 in. outside diameter, and 16.2 in. inside diameter. Only the nodes considered significant were used in the calculation. This relationship is continued for each successive node.

Heat is added to the inner circle node. This is transferred to the fluid thru a conductance which is determined by the convective coefficient and heat transfer area and to the next larger "sheet" node whose conductance is determined by the plate thickness, circumference of the circle which separates the nodes. The difference in average radii of adjacent nodes, and the conductivity of the material (aluminum, $k = 100$). From the next larger "sheet" node the heat is transferred to the fluid and a still next large sheet node. This is continued until the temperature difference between node and fluid is negligible.

By impressing a unit heat flux on the central diameter, the program calculated the temperature of the successive ring nodes. Since both conduction and convection are direct functions of the temperature difference, it is necessary only to calculate the temperature of the nodes with the sink at 0 F to obtain the $\Delta T/Q$ for various metal surface thicknesses and heat transfer coefficients.

The results of the calculations are given in Figures 2A through 5A, for the temperature drop in the coldplate from contact surface to fluid.

Reference:

1B. Aron, Walter, and Gerold Colombo, "Controlling Factors of Thermal Conductance Across Bolted Joints in A Vacuum Environment." ASME Paper No. 63-NA-196, presented at the Winter Annual Meeting, Philadelphia, Pa. November, 1963.

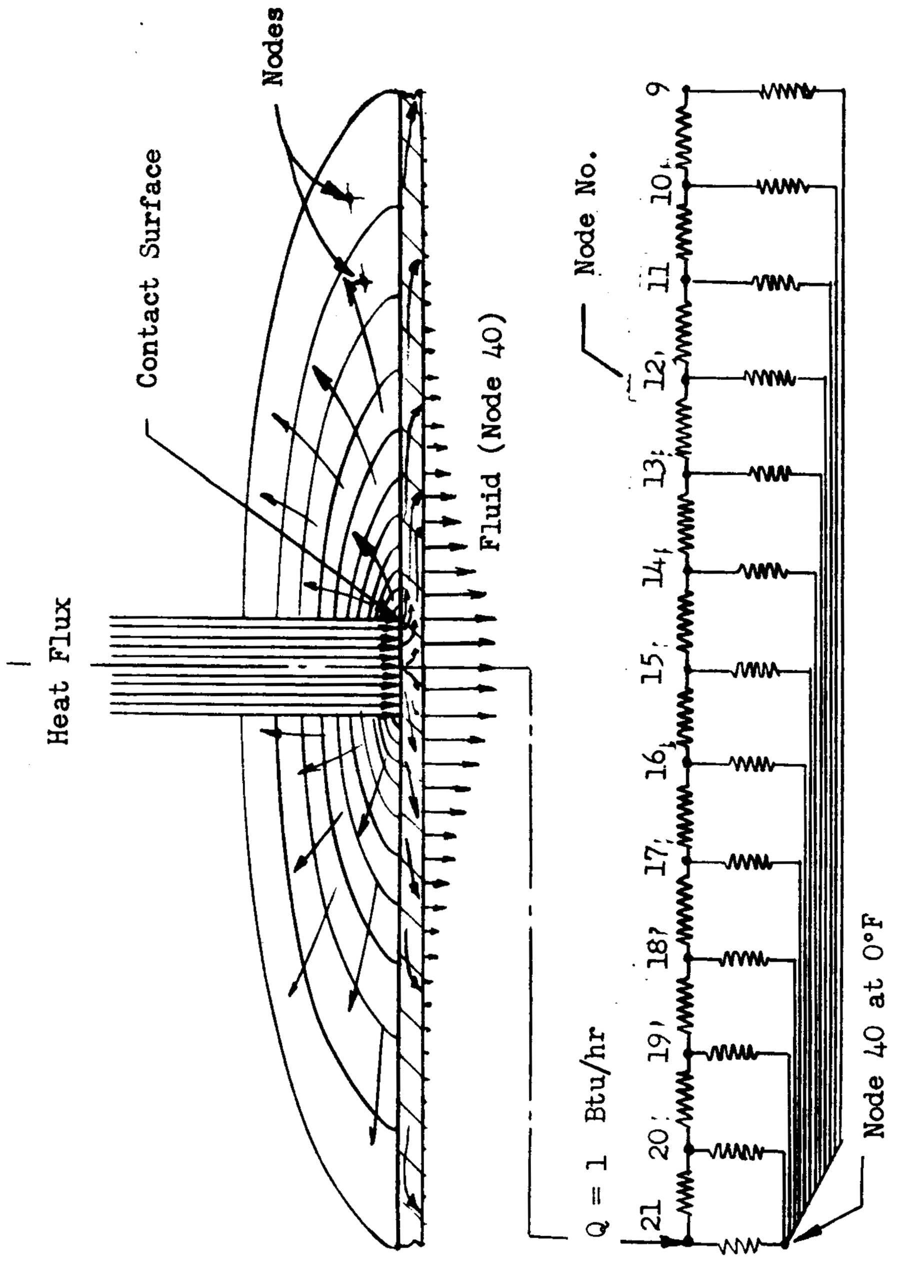


Figure 1A. Thermal Network of Heat Transfer from Coldplate to Fluid

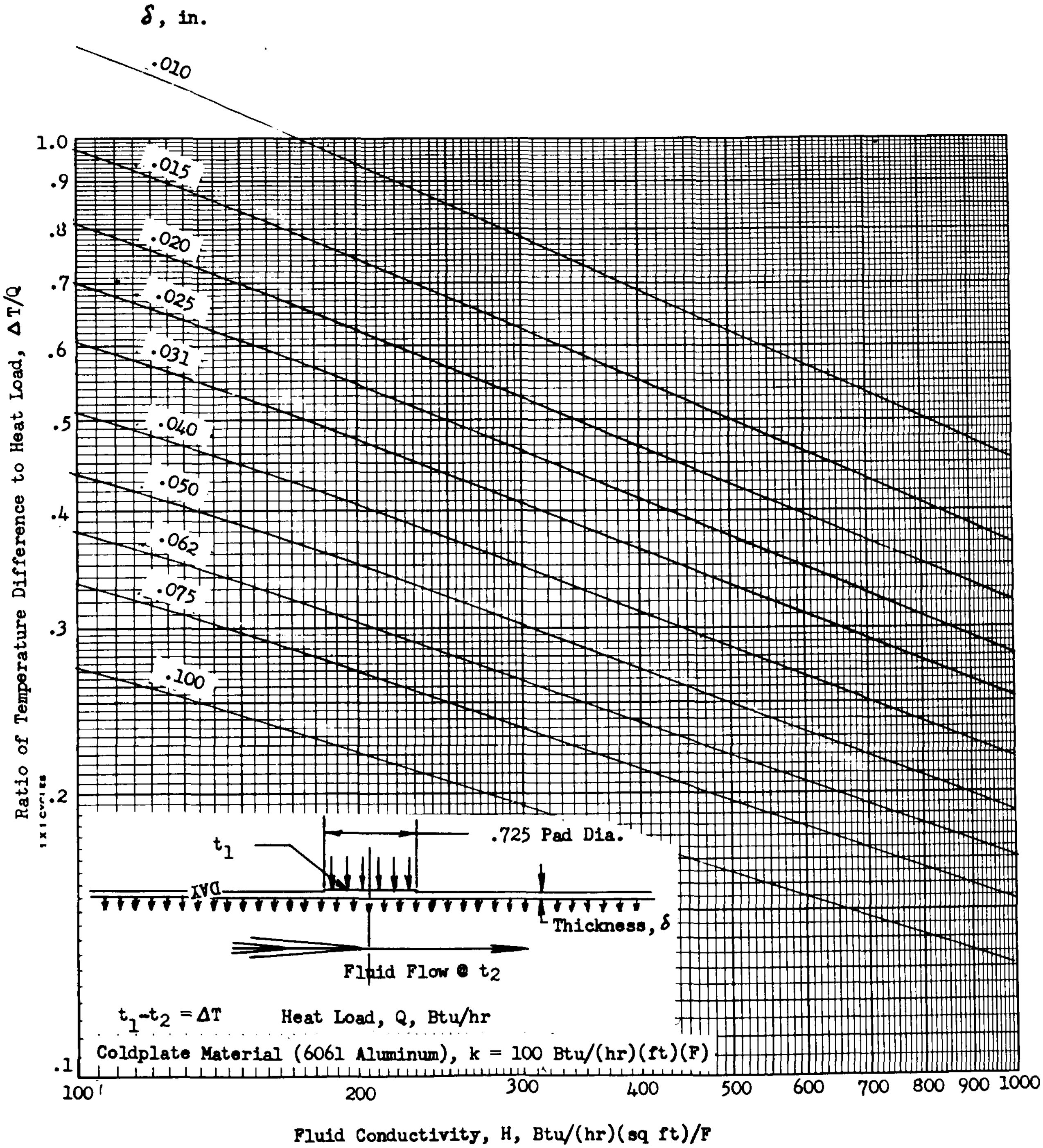


Figure 2A. Coldplate Temperature Drop, Pad Face to Fluid (.725 in. Pad. dia.)

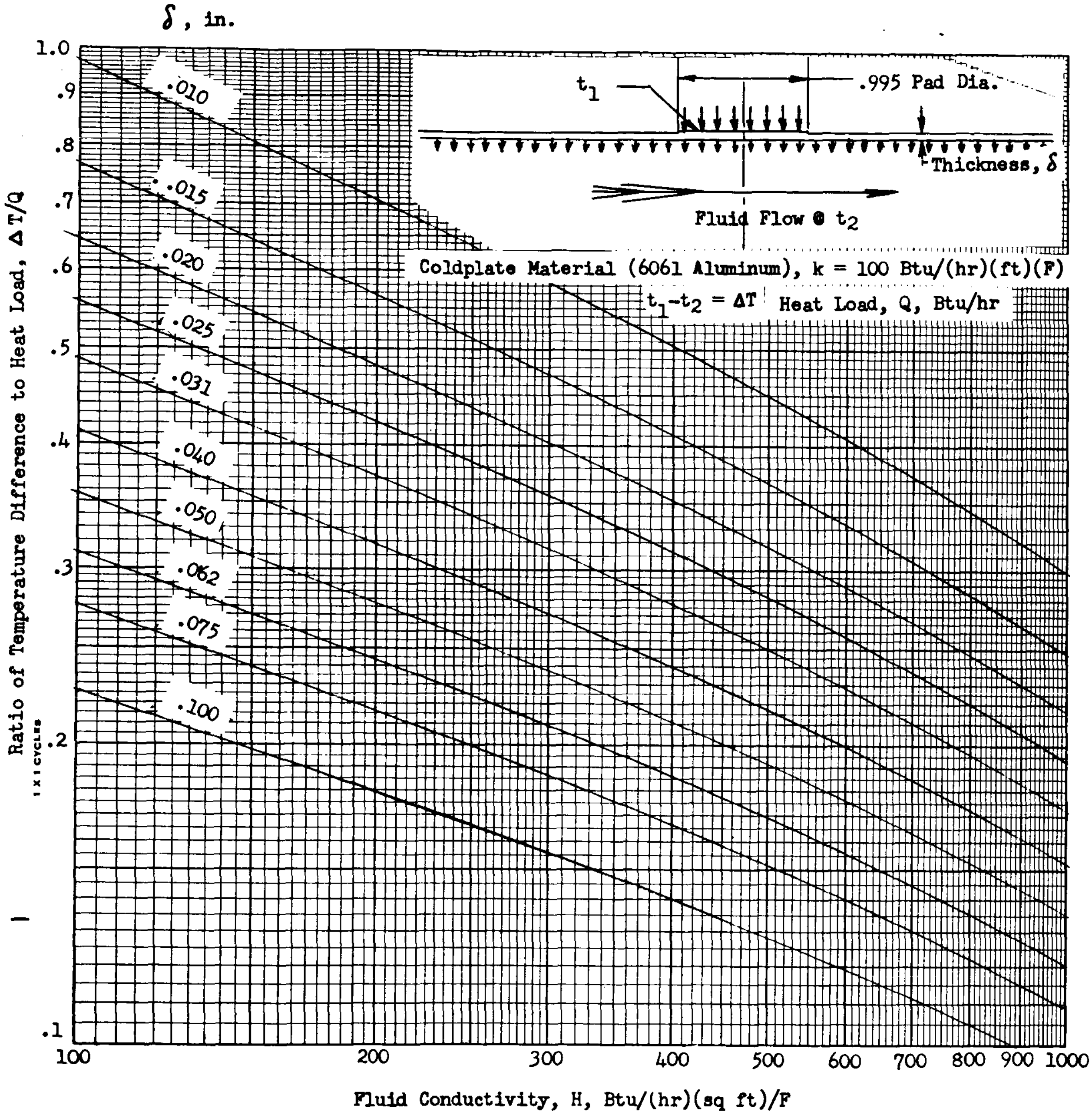


Figure 3A. Coldplate Temperature Drop, Pad Face to Fluid (.995 in. Pad. dia.)

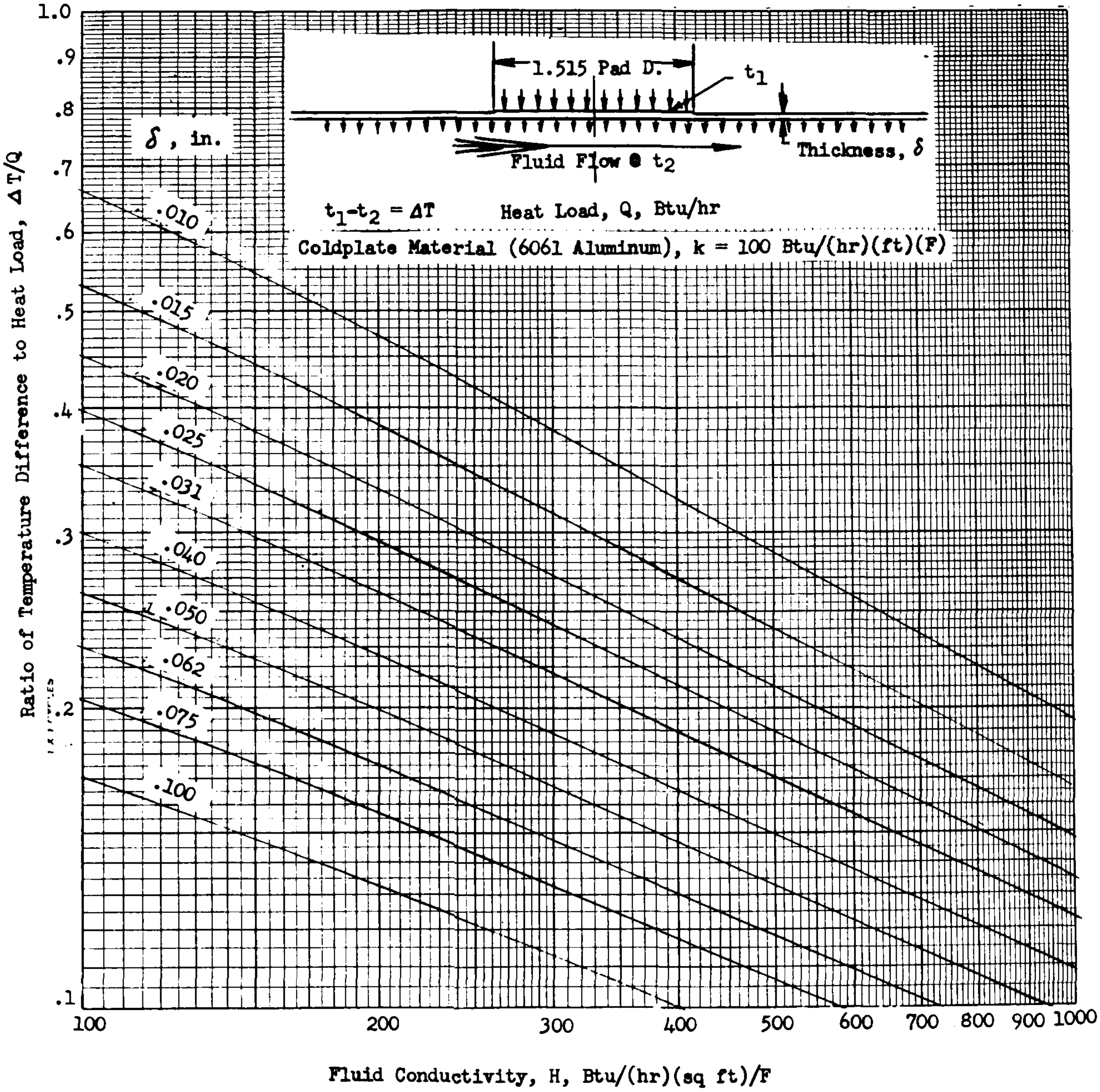


Figure 4A. Coldplate Temperature Drop, Pad Face to Fluid (1.515 in. Pa. dia.)

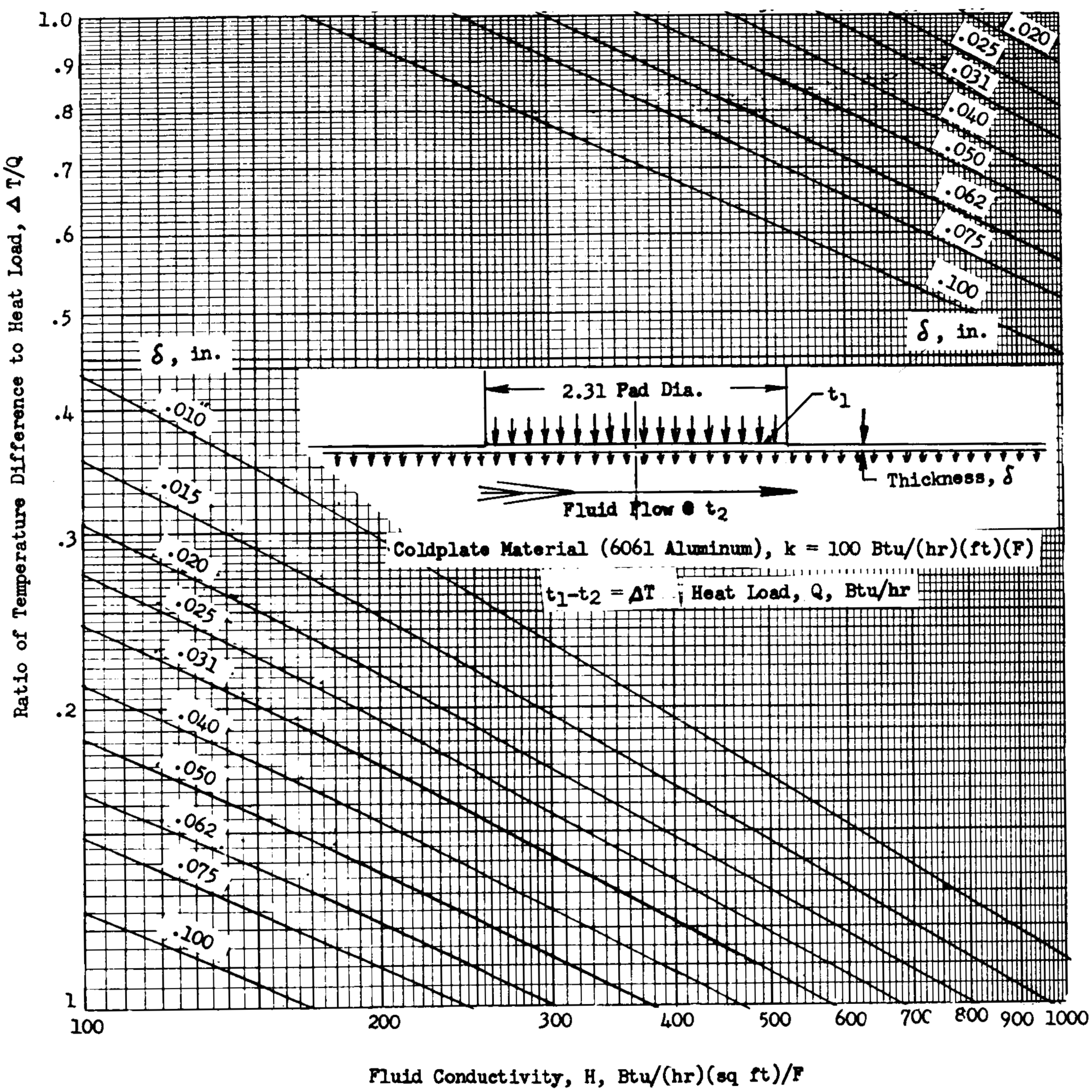


Figure 5A. Coldplate Temperature Drop, Pad Face to Fluid (2.31 in. Pad. dia.)

APPENDIX B

JOINT CONTACT RESISTANCE

The major part of joint contact resistance is due to the greatly restricted area of heat transfer around the bolt and the temperature drop in the surrounding surface material. A better approximation of the total resistance can be obtained by estimating the true contact diameter d_{te}.

This diameter $d_{te} = d_b \sqrt{(T\% x Syb/Sys)+1}$

where d_B = bolt diameter (fine series)

T% = % of maximum tightening torque

Syb = yield strength of bolt

Sys = yield strength of material at junction surface interface

This is shown in nomographic form in Figure 1B. This is used in conjunction with Figures 2A through 5A of Appendix A to determine the $\Delta T/Q$ for each of the bolted joints of the electronic package to the coldplate. The approximation also points the way to lower losses since the lower the yield strength of material at the junction interface, the larger the true contact diameter and lower the $\Delta T/Q$ for any given surface thickness and fluid heat transfer coefficient. The limit, of course, is a yield strength at the junction interface of 0 (a liquid). Here only the pad dimensions would limit the true contact diameter.

Dead soft 1100 aluminum has a very low tensile strength and used as an intermediate washer would cut down the temperature drop considerably. The vapor pressure is very low so it would hold up in a vacuum. Vacuum grease (possibly filled with flakes of high conductivity metal) or the metal gallium both have suitable vapor pressures and would be suitable for obtaining contact for the entire black box.

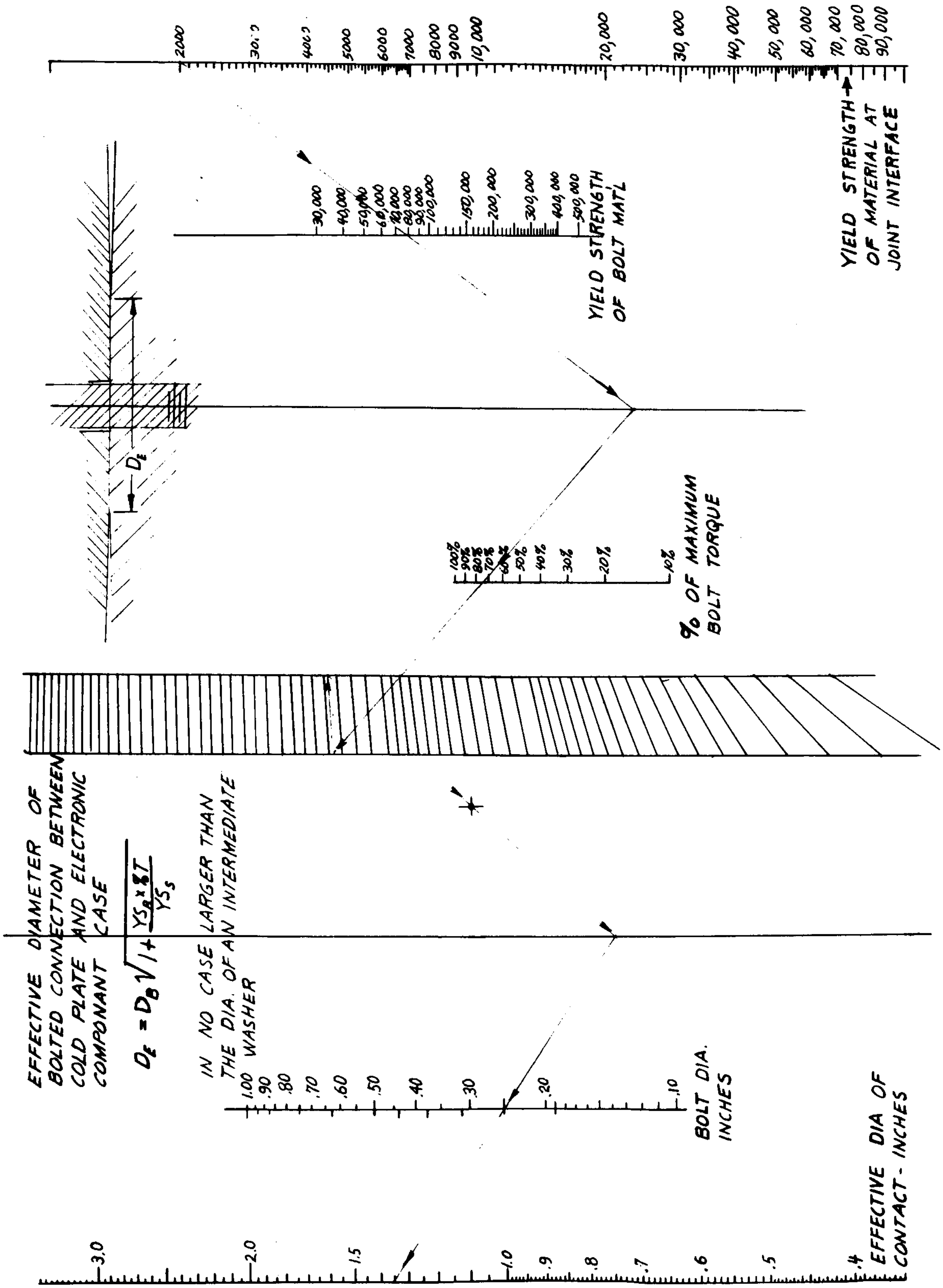


Figure 1B Effective Diameter of Bolted Connection