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ADVANCED SPACECRAFT THERMAL CONTROL TECHNIQUES

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Preliminary Design Office

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16. ABSTRACT

The problems of rejecting large amounts of heat have been significantly studied during the past decade. Shuttle Space Laboratory heat rejection uses 1 kW_e for pumps and fans for every 5 kW_t heat rejection. This is rather inefficient, and for future programs more efficient methods must be developed.

This review is based on a 1971 Grumman Aerospace Corporation study with slight changes and improvements. Two advanced systems were studied and compared to the present pumped-loop system. The advanced concepts are the air-cooled semipassive system, which features rejection of large percentage of the load through the outer skin, and the heat pipe system, which incorporates heat pipes for every thermal control function.

Other systems should be reviewed to find the most efficient heat rejection system for payloads in the 1990's. The system selected must use standardized components to reduce test requirements. Paints and other materials used for long-life duration must also be found.

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INTRODUCTION

The primary objective of a manned Space Station is to provide a self-contained facility capable of supporting a crew and to conduct multidisciplinary experiments and applications program for a minimum of 10 to 15 years. The scenario developed in this study for thermal control is each module will have its own heat rejection system and capability which will provide the greatest flexibility and reliability. No thermal interface is required between modules and, if one module fails, the crew can go to another module while the failure is being corrected.

Two advanced systems were studied and compared to the present pumped-loop system. The advanced concepts are the air-cooled semipassive system, which features rejection of a large percentage of the load through the outer skin, and the heat pipe system, which incorporates heat pipes for every thermal control function. Both advanced systems show significant weight and power consumption advantages over the state-of-the-art pumped-loop system.

AIR-COOLED, SEMIPASSIVE DESIGN

Anticipated thermal loads for various modules range from 40 000 to 120 000 Btu/hr. The large minimum load level makes the consideration of passive heat-rejection techniques attractive since it exists under all flight conditions. The inclusion of passive techniques could reduce the power and weight requirements of the active thermal control system. A simple way of accomplishing this latter goal is to decrease the insulation effectiveness between the pressure shell and skin and allow thermal radiation to accomplish the job. Low temperatures must be maintained on the outer skins by using low α_s/ϵ coatings (e.g., white paint, second surface mirrors).

The results of a computer analysis performed to evaluate this approach are depicted in Figure 1. As shown, the insulation loss is less than 10 000 Btu/hr for an effectiveness of 0.01; however, internal vehicle temperatures will exceed

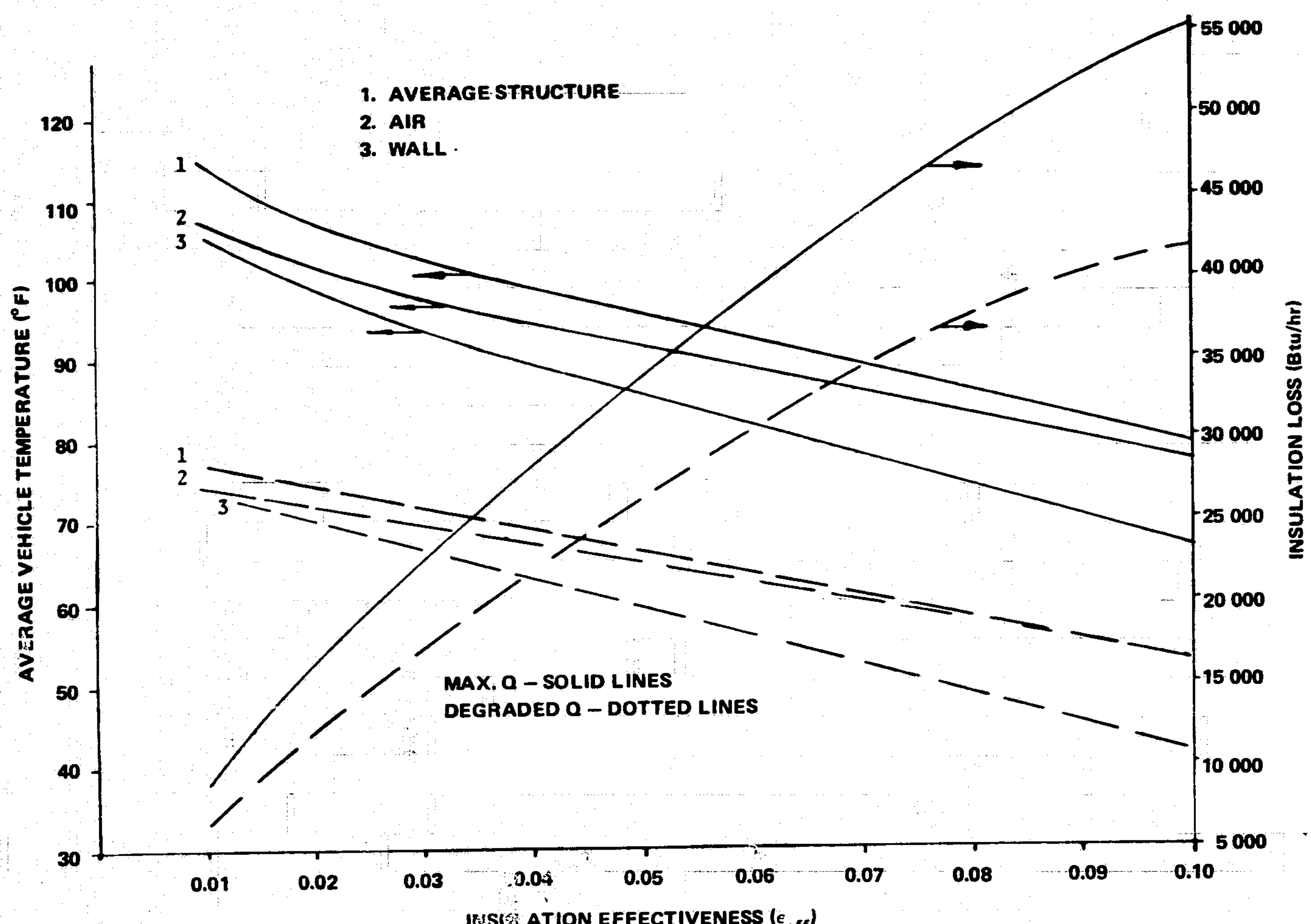

Figure 1. Effect of insulation on vehicle thermal behavior.

100°F for the maximum load case. For the same conditions, the minimum loads maintain the vehicle at approximately 75°F. If an effective insulation (i.e., effectiveness = 0.1) is incorporated into the design, the insulation loss increases to 55 000 Btu/hr, and the vehicle temperatures are a nominal 75°F for the maximum heat load. However, when the vehicle load drops, the corresponding insulation loss of 40 000 Btu/hr causes low vehicle temperatures of 50°F. The conclusion is that no single design value of ϵ_{eff} can provide acceptable temperature without including an active control in the system.

AIR-COOLED, SEMIPASSIVE SYSTEM

This design combines both active and passive cooling. For this system, part of the internal equipment load is handled by the air-conditioning system, while the remaining heat is rejected through the insulation. As the internal power dissipation varied, the air-cooling load would be varied to maintain acceptable vehicle temperatures. This design approach is called the semi-passive, air-cooled system.

In brief, the salient features of this approach are:

1. The rejection of a sizeable heat load passively through the insulation.

2. The elimination of toxic radiator fluids within the pressure shell.

3. Forced air-cooling of most electronic equipment.

4. The use of "cold walls" to actively cool batteries and power conditioning equipment which have high power densities.

AIR-COOLED CONSOLE AND RACK

A series of techniques were developed to cool the console and rack electronics. The air required to cool each box is drawn in from the cabin through self-contained inlet ducts as shown in Figure 2. After passing over the internal components, the air exits into the cabin housing from openings in the back of the box. The console housing itself serves as the exhaust header and must be reasonably sealed to prevent unwanted infiltration which would upset the flow balance. Equipment mounted on racks are similarly cooled with the air entering

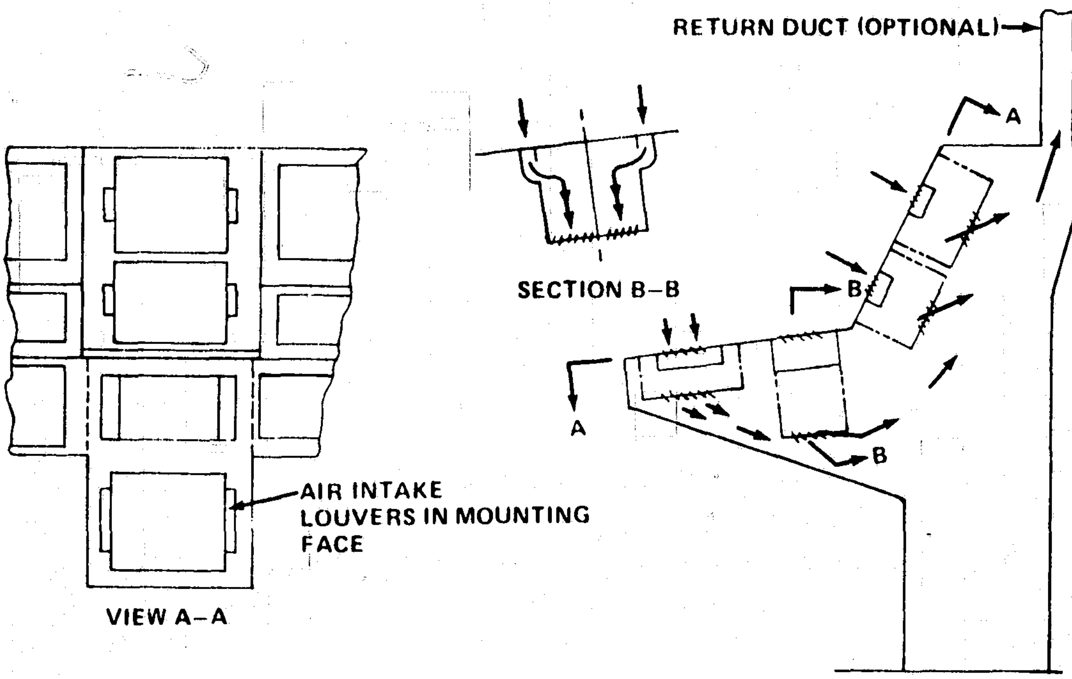

Figure 2. Air-cooled panel-mounted equipment.

uniformly over the face of the modules and passing over the equipment, before exiting into the return plenum. A return duct is used only when required by high exhaust temperatures. This is a well established practice in aircraft design [1].

"BASEBOARD" TYPE HEAT EXCHANGERS

All manned vehicles must include adequate cooling provisions at atmospheric temperature and humidity control. The air-cooled concept creates a special problem, since the absence of an internal loop precludes the use of state-of-the-art compact heat exchangers. Since the primary cooling fluid remains outside the pressure shell, the heat exchanger is most logically located at the wall.

Figure 3 illustrates the basic heat exchanger "hardware" design. As shown, the modular heat exchangers are simple-brazed assemblies which are fastened and bonded to the pressure shell.

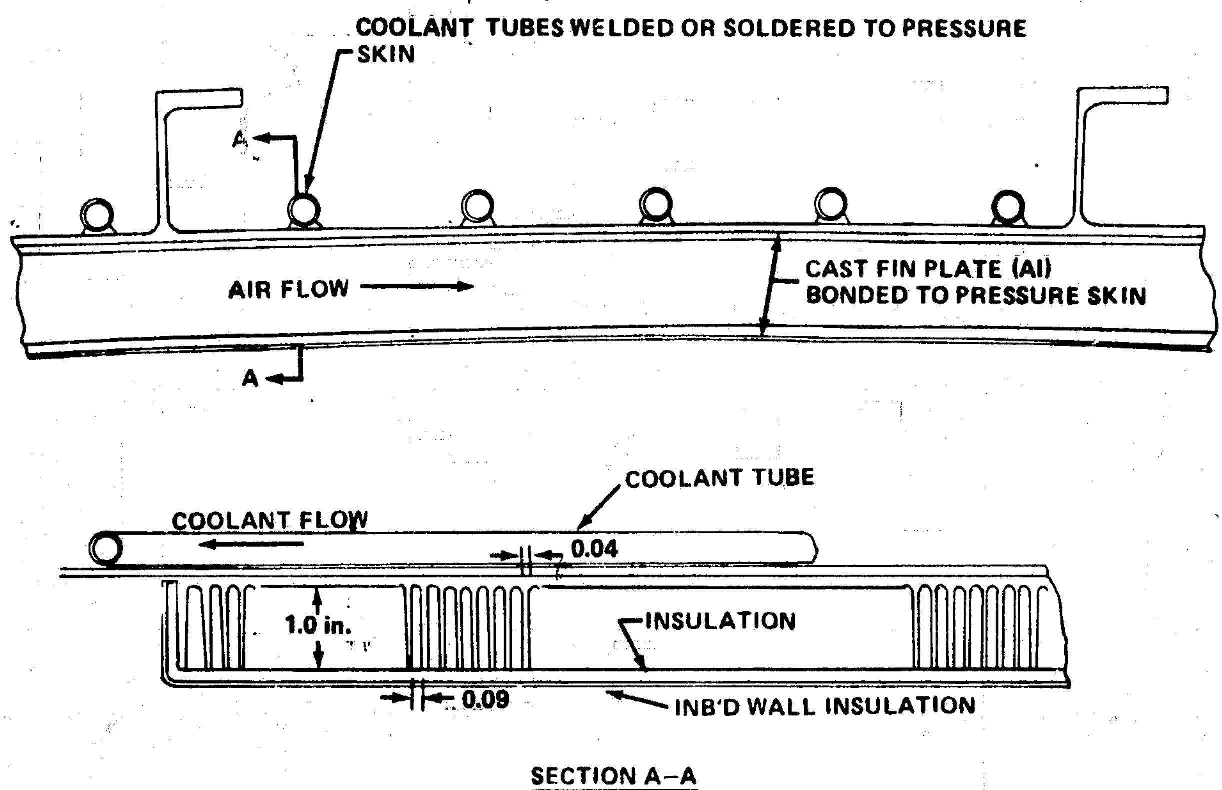

Figure 3. "Baseboard" heat exchanger design.

PRESSURE SHELL COLD WALL

In addition to the equipment which can be easily air cooled, there is a second group of low temperature hardware which is more of a problem. This equipment requires coolant air at approximately 40°F which would cause an undesirable growth in the size of the heat exchanger. An alternate cooling approach is to mount the equipment directly to the pressure shell and pass the thermal load directly to the structure and then to space. The rejection of the load could be through: (1) insulation, (2) thermal switches, (3) louvers, (4) heat pipes, or (5) a pump loop. Analysis indicates that the first three techniques are limited to relatively low power densities which can easily be air cooled.

High power density components, such as batteries and power conditioning equipment, operate most effectively at relatively low temperature levels ($\leq 50°F$)

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and, hence, require efficient cooling. The design evolved includes external fluid cold rails fastened to the pressure shell and internal structural members to conduct the thermal load from the boxes to the "cold wall." All of the connections are bolted, and the cabling is sandwiched in the space between the wall and box.

INTEGRATED SYSTEM DESIGN

Having selected specific cooling techniques to conform with the design philosophy, a totally integrated system was developed. Several preliminary designs were reviewed prior to evolving the final overall air-cooled, semi-passive space module thermal design. Among those considered were completely "closed" and "open" systems.

In general, cooling air is taken directly from the cabin into the experiment interior where it flows over the components and then exits back into the cabin. The warm air is drawn via return ducts to "baseboard" type heat exchangers where it is cooled and subsequently conditioned and returned to the cabin interior. Figure 4 schematically presents a module airflow circuit. As shown, the main blower pumps air into cabin where it picks up both a metabolic and sensible heat load. The air is subsequently drawn into the return duct for delivery to the heat exchangers and life support equipment. A bypass is provided to limit the total airflow to the heat exchangers and maintain the proper balance between the active and passive heat rejection systems.

HEAT PIPE SYSTEM DESIGN

An alternate thermal design features the use of heat pipes to transfer the thermal load from the individual sources to the space radiator. Although a single heat pipe could, in principle, transfer the load, design and maintainability considerations dictate the use of multiple heat pipes in series, since long heat pipes going directly from each source to the sink would prove unwieldy.

Figure 5 illustrates how the potentially large number of individual pipes is reduced. The source pipes from the loads (Q) are manifolded into intermediate headers which, in turn, transfer the heat to an internal header which serves as the sink. This internal (circumferential) header transfers the load through the pressure shell to the radiator panel headers. Finally, the panel headers distribute the load to the individual radiator pipes and fins for rejection to space.

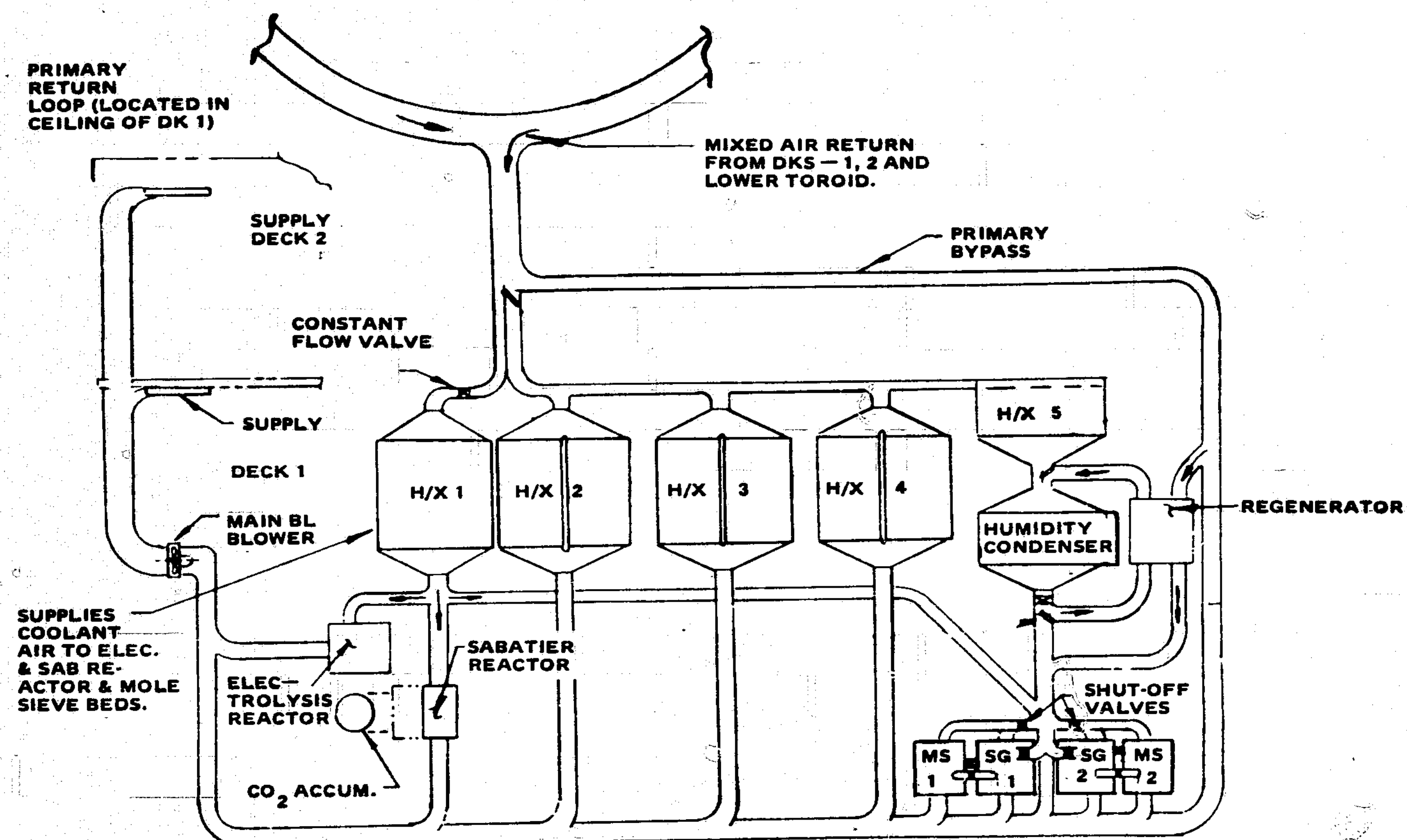

Figure 4. Air-loop schematic.

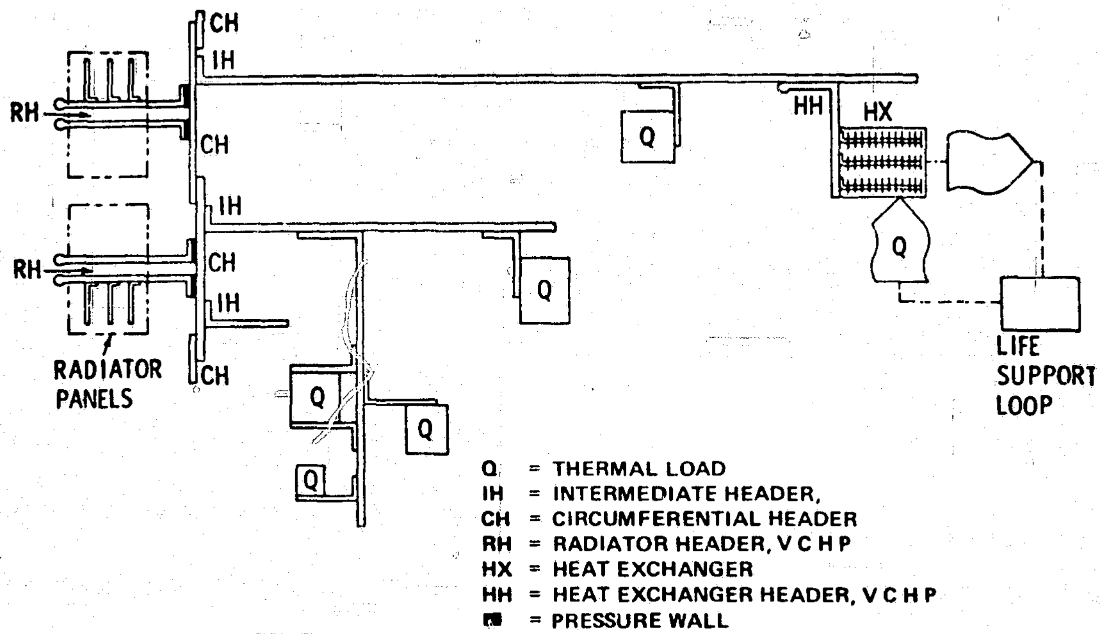

Figure 5. Heat pipe loop schematic.

In its simplest form, the system can be thought of as a thermal circuit made of heat pipes. By using the isothermal feature of heat pipes; all the pipes associated with a particular circumferential header are designed to operate at the nominal temperatures of that header. This includes the radiator header, so that all of the heat sources on that circuit are assured a "constant" temperature sink. The major control elements in the circuit are the variable conductance heat pipe (VCHP) headers for the radiators.

Since the system depends on heat being transferred through a series of coupled heat pipes, high heat-transfer rates through the joints are desirable. While welded or brazed joints would give these transfer rates, they are not used exclusively, since individual welded pipes are not easily replaceable and are not consistent with a maintainable design. The maintainable couplings are of the clamped or bolted flange type.

Typical contact conductance data [2-5] are presented in Figure 6 as the conductance coefficient versus the surface roughness. For the study, a conservative value of 500 Btu/ft^2-hr-°F was used for joint contact conductance (h_c). Additional assumptions are a 30 μin. surface finish and 10 psi contact pressure with a 1 mil indium foil shim. Note that for these conditions, Figure 6 shows h_c values over 2000 for vacuum and nearly double that for atmospheric pressure.

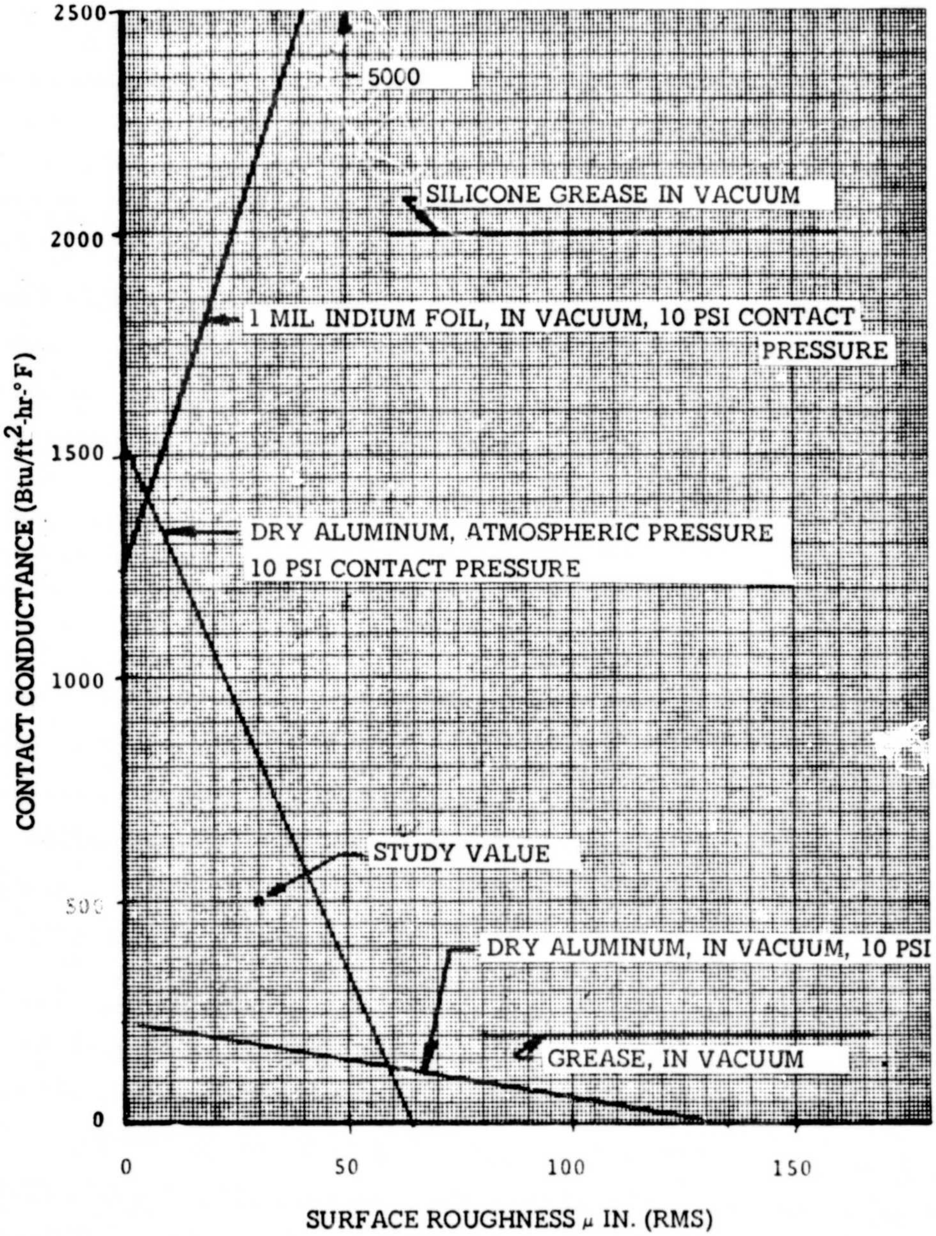

Figure 6. Typical contact conductance data.

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As noted, the heat pipe thermal control system of circuits depends on heat being transferred through a series of coupled heat pipes, where the isothermalizer (circumferential) header serves as a heat sink to groups of heat pipes in contact with it. Therefore, each circumferential header must operate at the nominal temperature of the source which requires the lowest sink temperature in order to maintain the design heat rejection.

In addition to load-carrying capability, the temperature drop from source-to-sink is a controlling factor in the design. All temperature drops include the applicable interface resistances in the circuit based on a conservative value of 500 for contact conductance. Thus, if we take as typical the Primary Control Center in order to handle the 9300-Btu/hr load, 93 pipes (0.75 in. OD) operating through a 27°F drop to the circumferential header are required. An additional 8°F drop is expected between the header and the radiator fin. Each module has at least one circumferential header. The temperature of the radiator panel associated with each module has been chosen to be 8°F lower than the lowest required header temperature. However, one of the benefits of this approach is that it lends itself to load segregation. For example, in the toroids, if the high and low temperature load groups are coupled to two different headers operating at selected temperatures, savings in the required radiator area result. By choosing to create a 65°F as well as a 35°F header in each toroid, a total of 385 ft^2 of radiator can be eliminated. Greater potential savings are possible, since there are loads operating at significantly higher temperatures (up to 500°F).

Heat Pipe Joints

During the initial formulation of the heat pipe thermal control system, it became apparent that some means of efficiently joining the pipes to one another was required. While early work assumed single long pipes from heat source to sink, detailed investigation showed this to be an extremely difficult approach to implement because of the large number of sources. Therefore, the first design task undertaken was a study of joint techniques. Some of the key factors considered are:

1. Attainment of sufficient thermal conductance through the joint to minimize source-to-sink temperature gradient: A large gradient will result in a low radiator temperature with an associated increase in required radiator area and weight.

2. Ease of maintenance so that replacement of failed pipes or upgrading of the system would be possible in flight.

3. Manufacturing and design flexibility so that the components can be manufactured "easily" and at reasonable cost.

Figure 7 covers the designs evolved for joining heat pipes. Figure 7(a) shows a "simple" bolted flange approach wherein the mounting or joining surface consists of a flanged saddle into which the tube is mounted. Figure 7(b) illustrates a variation of the basic flange which utilizes rectangular-shaped tubes to simplify the manufacturing problem and provide greater contact area.

Another approach to thermally joining the pipes involves clamping blocks illustrated in Figure 7(c). Some advantages of the blocks are that they do not require modification to the heat pipe and they may be interchangeable with many locations on the module, thus affording greater flexibility in design and application. An adaptation of the blocks is illustrated in Figure 7(d). To cut the weight and maintain the desirable thermal efficiency, the block is replaced with a solid "double saddle" which seats the two pipes to be joined. The contact pressure is applied with a clamp strap which squeezes the assembly together.

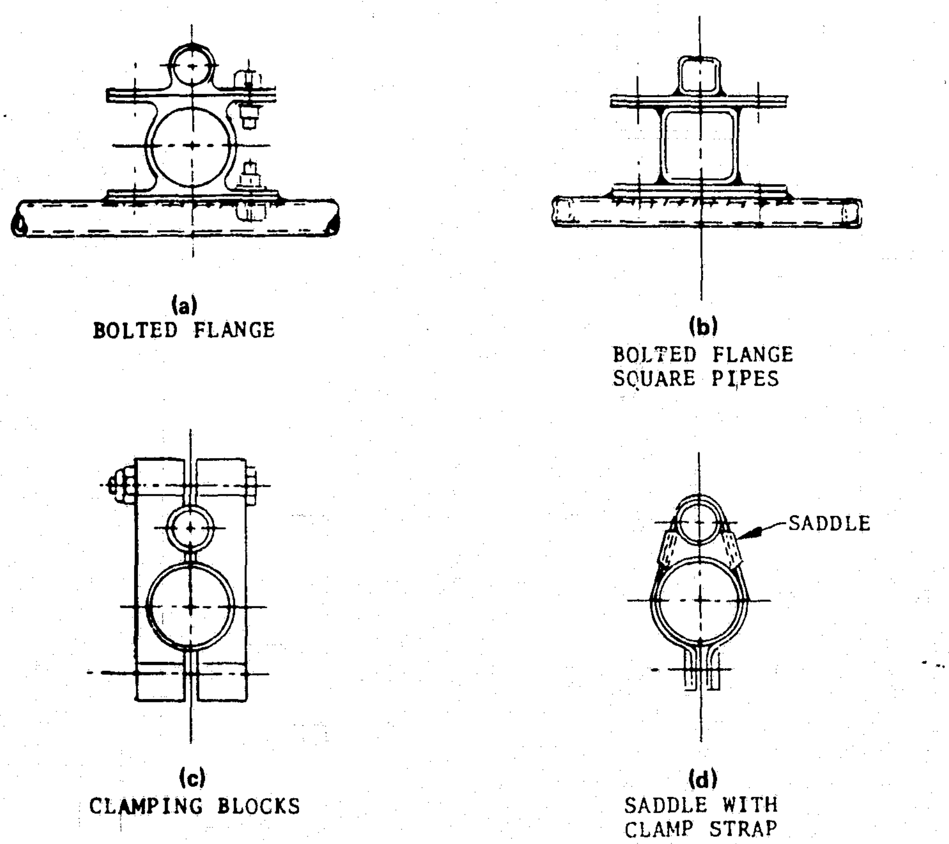


Figure 7. Heat pipe clamping techniques.

Figure 8 shows some approaches to making multiple pipe joints. Note the various combinations of blocks, saddles, flanges, and clamp straps.

Two broad conclusions were drawn:

1. For cases where many small pipes were to be permanently attached to a large one, the use of rectangular cross sections was best from a weight and manufacturing standpoint. An acceptable alternate would use a rectangular shape for the large pipe with round smaller pipes.

2. For single pipe-to-pipe joints, the tube shape is not critical.

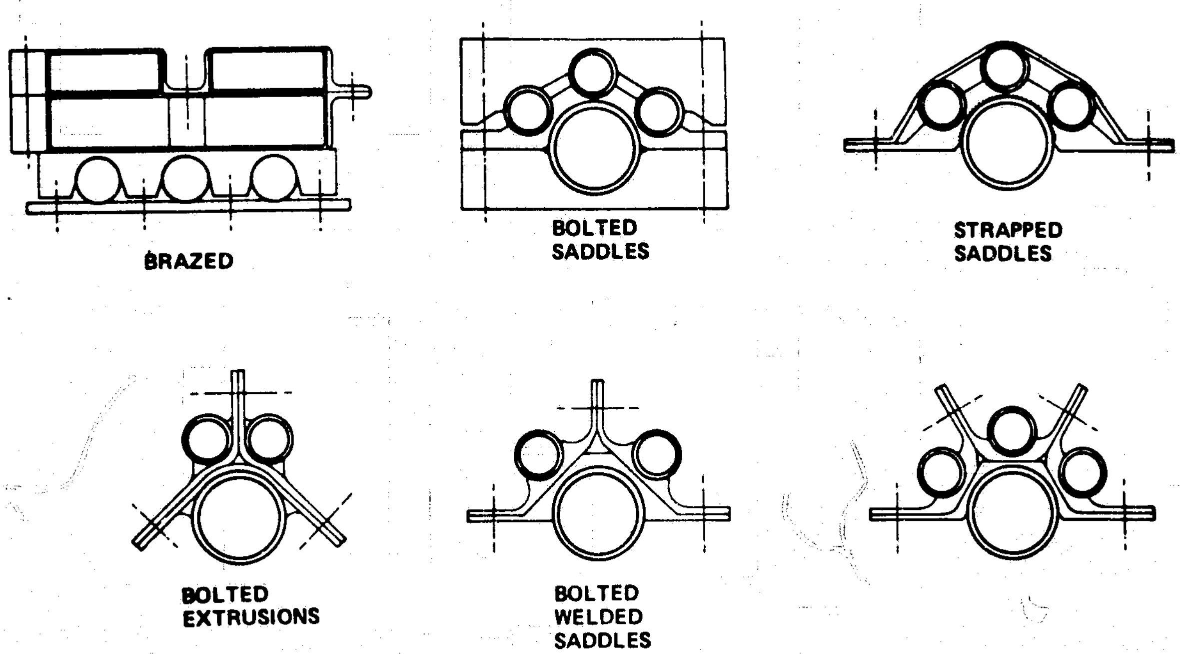


Figure 8. Multiple heat pipe joints.

Heat Pipe Exchangers

A relatively unique requirement for the heat pipes is that they provide atmospheric cooling, i.e., replace the usual air to fluid heat exchanger in the environmental control system (ECS). Thus, the second design task undertaken was the conceptualization of a heat pipe heat exchanger for this application. The finned heat pipe evaporators shown in Figure 9 are arranged in "packs" in

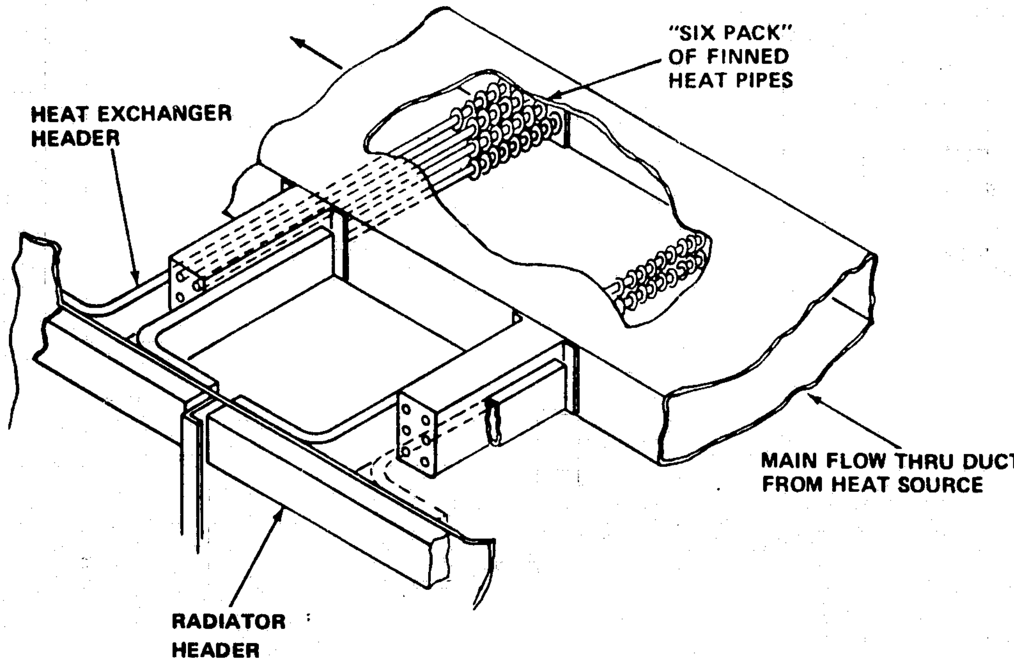


Figure 9. Heat pipe heat exchanger.

the duct. Each pack of six small pipes has two rectangular pipes which serve as intermediate headers between the heat exchanger in the duct and the external isothermalizer header. The replaceable unit is the "6 pack," and this can be further broken down by making the individual pipes in the unit replaceable. The number of modular packs used is determined by the design load in the particular duct. Variations of the modules can be devised to provide coverage for any cross-sectional duct area required.

Heat Pipe Cold Rails

Where the electronic equipment thermal loads cannot be handled by conduction or convection, it becomes necessary to provide an active coolant "loop." For the heat pipe thermal control concept, the cooling of electronic modules is achieved by mounting them on heat pipe cold rails.

Figure 10 shows the electronic packages as flat "card" modules with cooling fins on two edges. These fins slide into slots in the cold rail made up of machined blocks around the cooling pipes. This concept is fully maintainable in that the pipes or blocks can be replaced as required by failures or changes in cooling requirements.

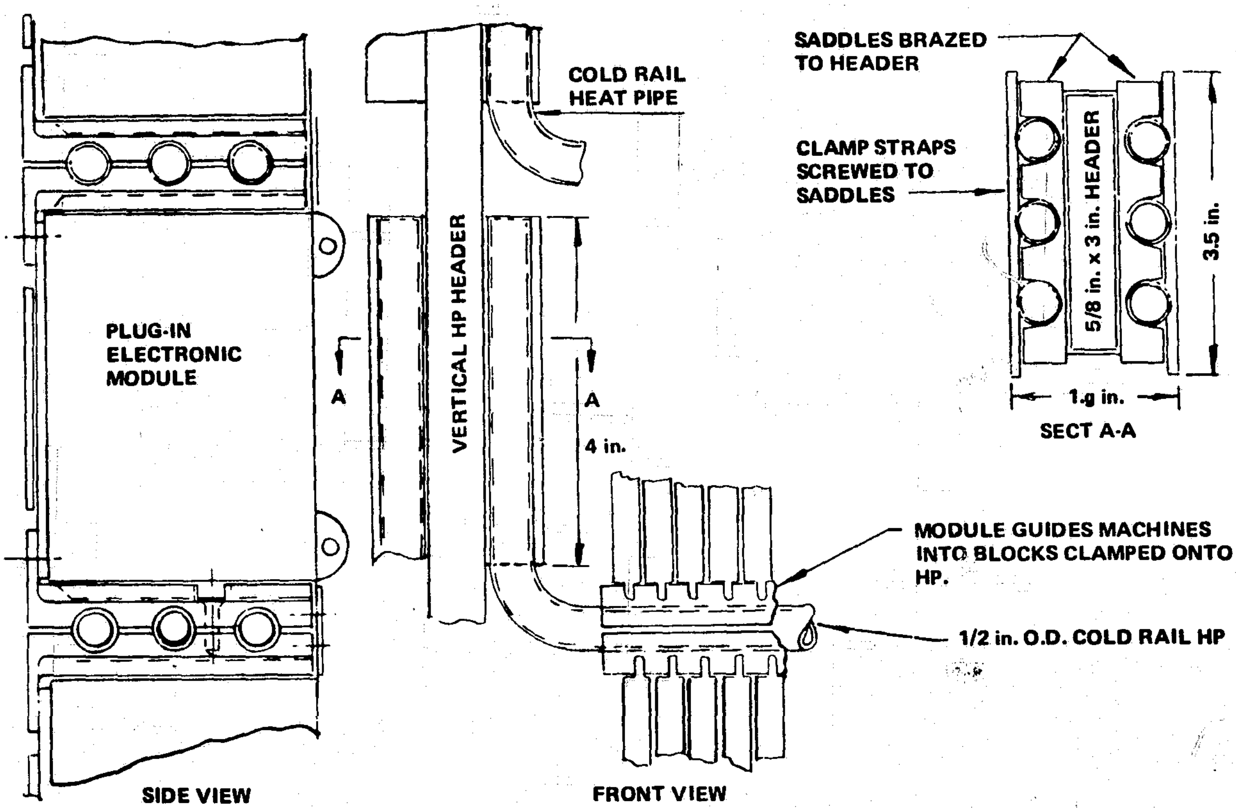


Figure 10. Heat pipe cold rails for card modules.

Heat Pipe Cooling of Control Console

The individual console boxes are shown in Figure 11 which also presents one approach to heat pipe cooling. Based on the box power, the number and size of the cooling pipes were chosen with boxes under 3 W conductively cooled by structure. Using the box dimensions, the availability of sufficient evaporator mounting area was verified and pipe routing was fixed. This design uses the smallest number of top console headers [6] and, therefore, has the least complex plumbing system. The price for this simplicity is paid for by the fact that failure of a header causes a loss of cooling on the (one to four) boxes attached to it. The pipe-to-box and pipe-to-pipe couplings may be detachable or permanent; sufficient working area is available so that the decision can be made on the basis of other considerations.

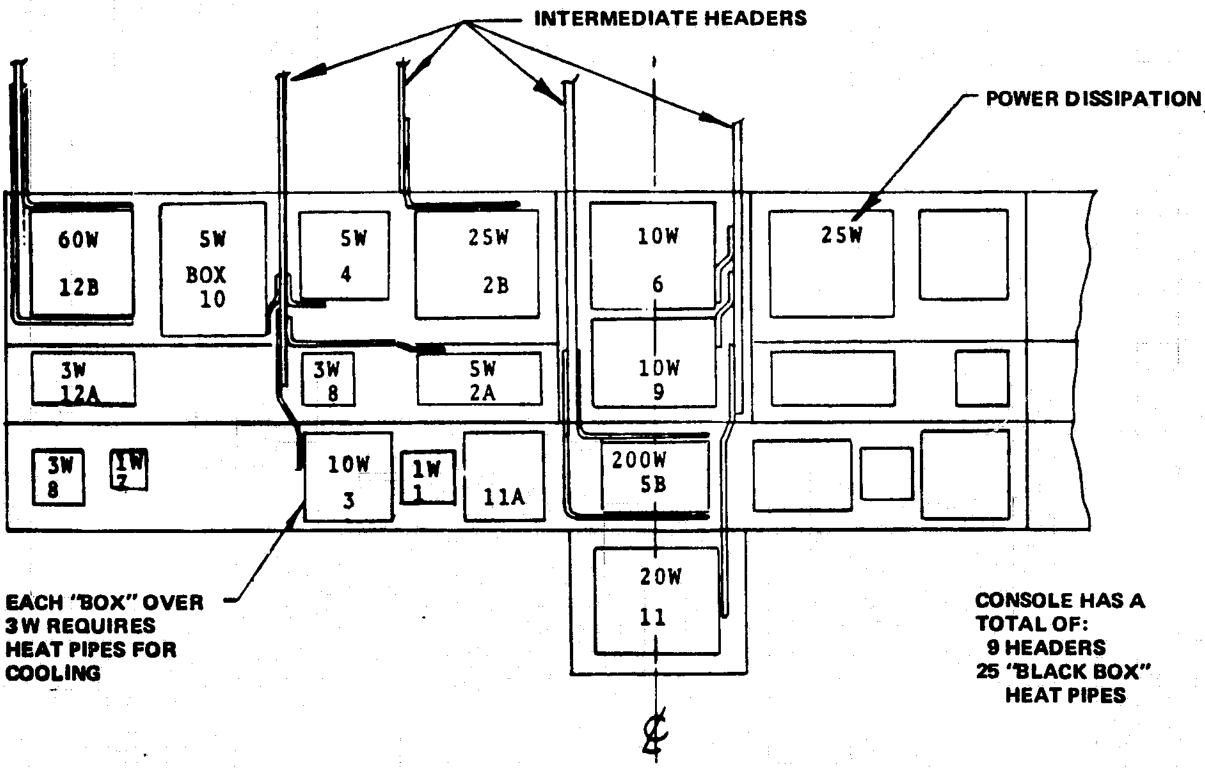


Figure 11. Heat pipe cooling of panel-mounted equipment.

The individual component pipes are small and relatively short (less than 1 in. OD and most less than 3 ft long) while the headers are larger, and preferably rectangular in cross section (1×2 in. area).

Heat Pipe Cooling of Equipment Racks

The lower part of the console will contain equipment in a typical "rack" configuration. The boxes will tend toward standardized sizes and shapes and will not require frequent viewing or access, except for failure analysis and repair. Thus, the designs will be equally applicable to full-length (approximately 6 ft high), free-standing racks.

Looking at Figure 12(a), we must think of the horizontal gap between boxes (2.8 in.) as the cold rail. Each box is mounted to two pipes which feed into a single header. The pipes can either provide redundant capacity or share the total load. The individual pipes are less than 1-in. OD, and the headers are

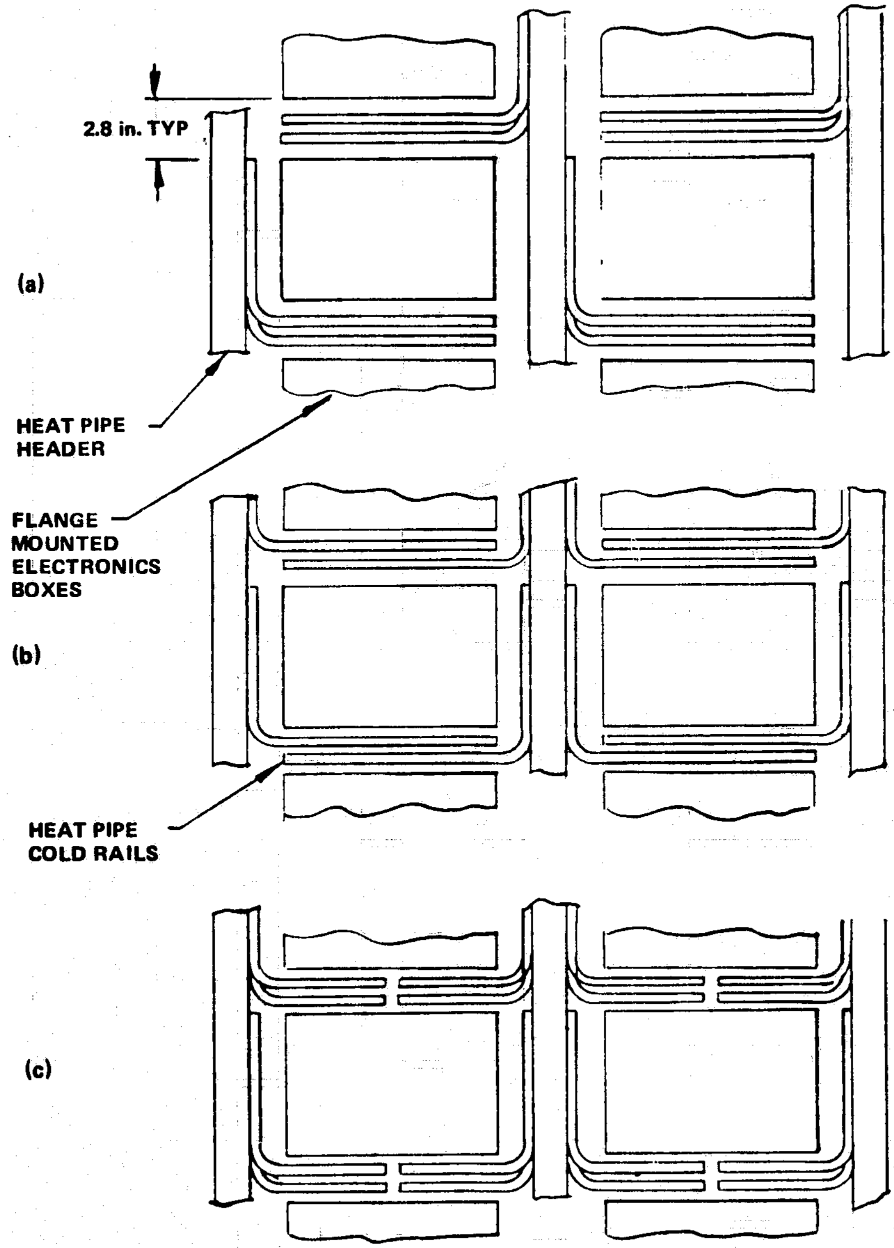
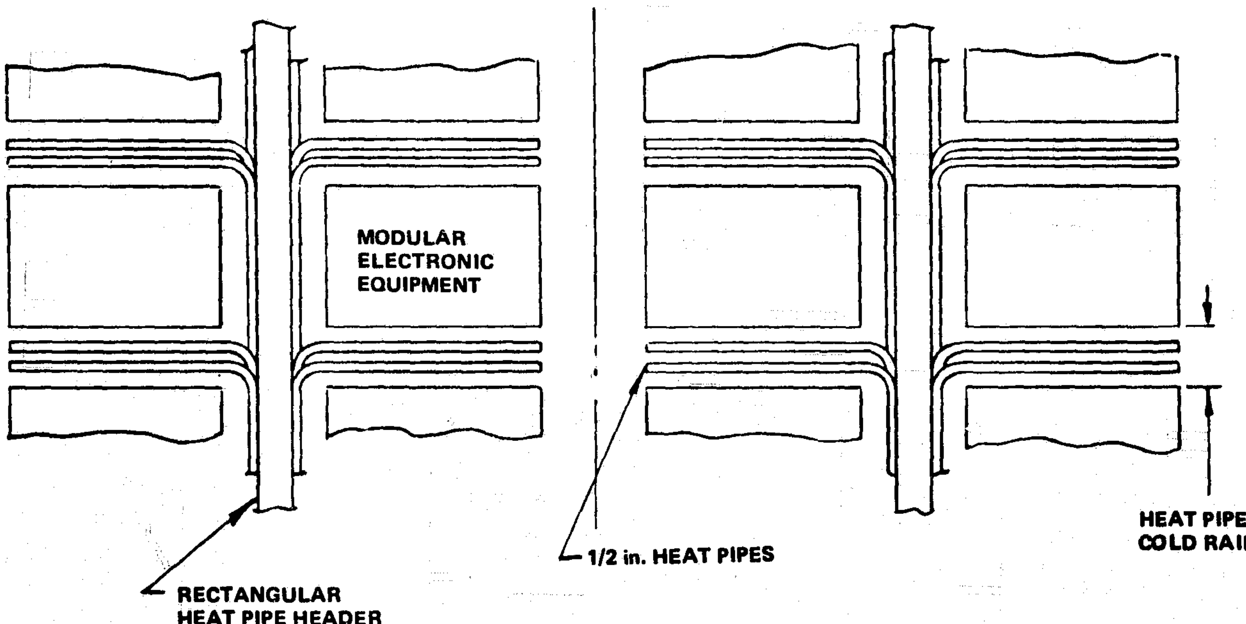


Figure 12. Heat pipe rack cooling designs.

nominally 1-1/2 in. OD or 7/8 in. × 2 in. rectangular cross sections. Figure 12(b) shows a similar approach, except that the pipes are staggered; i.e., each cold rail feeds two headers. The advantage of this scheme is that a header failure cuts cooling equally to both sides of a box rather than causing an entire rail to go dead. Figure 12(c) changes the concept by doubling the number of heat pipes to decrease the effect of a single pipe failure. Figure 13 shows a simple system which reduced the number of headers by a factor of two. In gaining this simplicity, we change a complete loss of cooling to the boxes on a header in the event of its failure. By varying the number and placement of the heat pipes, the desired failure capacity can be obtained. For each individual case, trends similar to those illustrated in Figure 14 can be developed.



- TYPICAL INSTALLATION FOR LOWER PORTION OF CONTROL CONSOLE AND LARGE RACK MOUNTED EQUIPMENT

Figure 13. Alternate heat pipe rack cooling techniques.

Heat Pipe Radiators

The remaining hardware item to be discussed is the heat pipe radiator. It is the most complex part of the system in both configuration and operation. It is the largest assembly, most susceptible to environmental damage and most inaccessible. The radiator must be extremely reliable over the widest combination of load conditions and operating temperatures.

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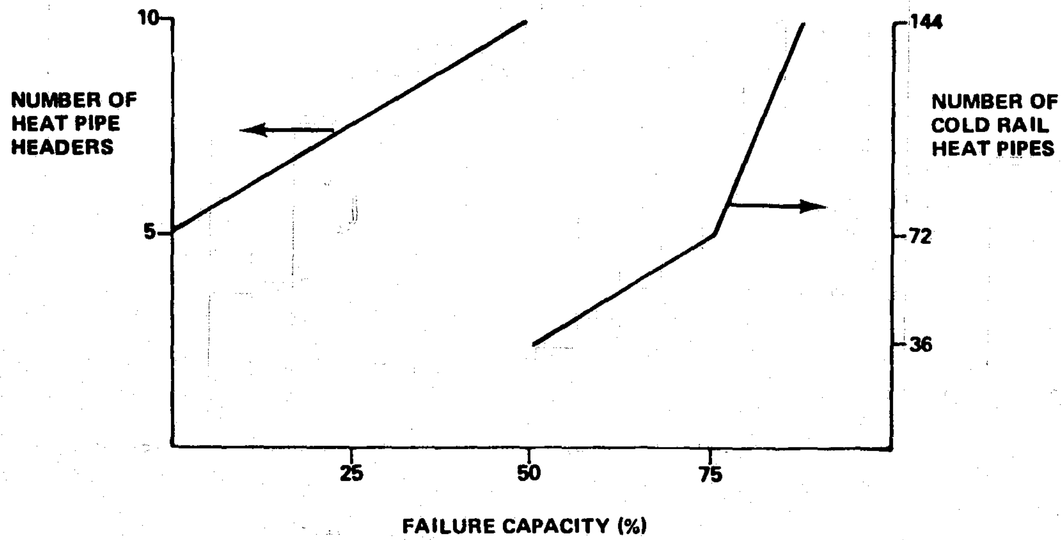


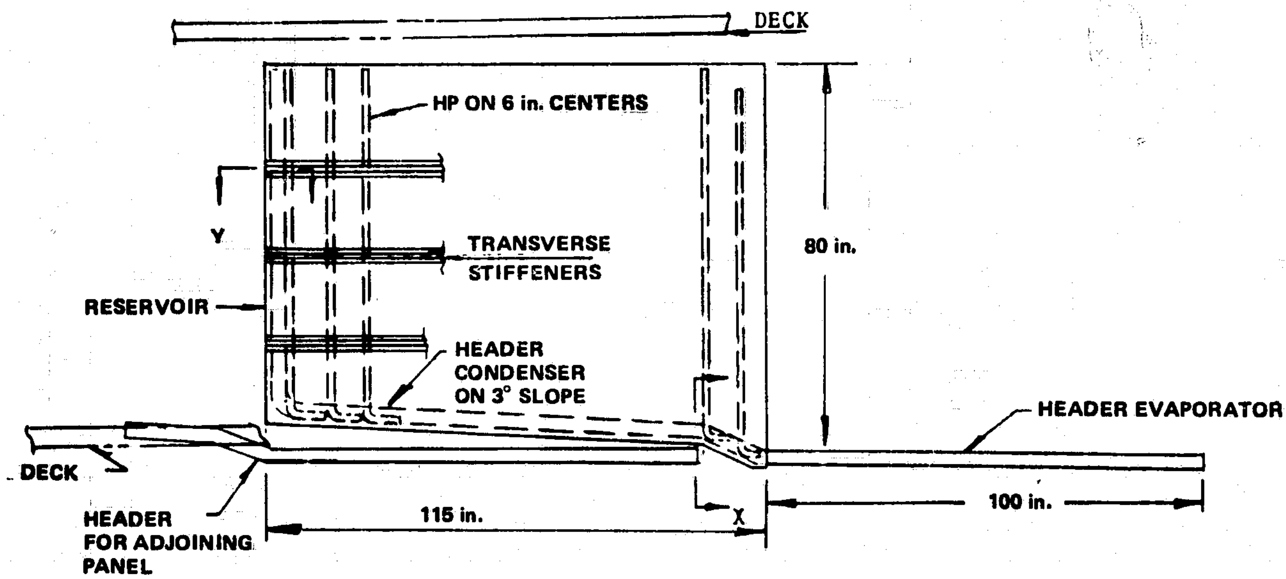
CAPACITY VALUE IS FOR THE FAILURE OF A SINGLE COLD RAIL PIPE OR HEADER; IT ONLY APPLIES TO THE BOXES ASSOCIATED WITH THE FAILED PART.

Figure 14. Heat pipe failure impact on system capacity.

The application of heat pipes to space radiators significantly reduces the effects of meteoroid penetration on performance. Since the radiator panel is made up of numerous individual heat pipes, a puncture in one pipe results in a minor thermal performance penalty. Therefore, extra meteoroid shielding is not required on the heat pipe panel.

The decision was made to eliminate the use of round tubes on the radiator panels because of the difficulties in attaining the required precise alignment between the individual pipes, their saddle blocks, and the header.

Figure 15 shows an overall view of the panel and associated headers, in position, on the pressure shell. The tubes are nominally 6 in. apart, and the headers are slanted so that the evaporators are always lower in the gravity field (when it exists) than the condenser. This slant, combined with the desire to be able to remove a single panel without disturbing adjacent ones, results in a loss of about 3 percent of the usable surface area. The individual pipes are arranged so that those which operate under gravity conditions have the headers down.



- RECTANGULAR TUBES w/ 0.020 in. FIN
- WEIGHT 0.58 lb/ft^2 PLUS 0.37 lb/ft^2 FOR STRUCTURAL SUPPORT AND METEOROID PROTECTION

Figure 15. Heat pipe radiator assembly.

The basic control mode for the radiator panels involves the use of a single VCHP to feed the load into each panel. The use of a VCHP as a header provides the built-in automatic control required to handle the range of internal loads. The operation of a VCHP has been discussed in numerous publications. In effect, this design makes each radiator panel a constant temperature sink, capable of handling a wide range of waste heat loads.

Integrated Design — Heat Pipe Thermal Control

The integrated system design drawings [7] include typical items of equipment and their locations in the station. All of the hardware was drawn to scale to assure that adequate volume exists and that the design geometry did not impair operation.

The self-wicking heat pipe design [6] provides instantaneous start-up and eliminates transient response problems. The longest pipes are nominally less than 10 ft long. Even the circumferential headers encircling the shell are made up of shorter segments to avoid very long pipes having inherent

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manufacturing, operating, and maintainability problems. Since the circumferential headers inside the pressure shell operate near the dewpoints, insulation is used to prevent local condensation. The final design incorporates approximately 252 individual internal heat pipes and a total of three circumferential headers feeding 848 ft^2 of radiator divided into 24 separate panels.

SYSTEM COMPARISON

In order to make a valid thermal control system weight comparison between the three concepts, the weight summary categorizes the hardware under the headings, internal cooling, external cooling, and insulation and is presented in Table 1.

The air-cooled systems appear to be the lightest when either a pumped-loop or heat pipe radiator is used. However, if the semipassive feature is eliminated, the air-cooled system weight is increased by approximately 1700 lb, which considerably diminishes its advantage. In addition, the air-cooled equipment requires lower packing and power densities which may result in increased box volumes and weights. A quantitative evaluation of this effect requires detailed equipment designs which were not available. The heat pipe system also shows a large weight advantage when compared to the pumped-loop design and compares favorably with the air-cooled concepts.

Additionally, a comparison was made of the power required to operate the pumps, fans, etc. associated with each of the systems. The pumped-loop system requires approximately 3000 W and the air-cooled design requires approximately 2700 W. The power savings of the air-cooled design may be attributed to the semipassive heat rejection feature. Since the heat pipe design eliminates the need for circulating a coolant, its power requirements are only 300 W. There is a weight penalty due to the additional solar arrays, batteries, and conditioning equipment. An overall conversion ratio of 0.6 lb/W was used [8]. Thus, when the power supply weight penalty is included with the system weight, as it should be since the systems are in continuous operation and their power requirements could not be used for alternate functions, the two advanced concepts show larger advantages.

The combined hardware and power system weights are summarized in Table 2. As shown, the heat pipe system has a 3915 lb total weight saving when compared to the pumped-loop system, while the air-cooled systems yield a nominal 3000 lb savings. A comparison of alternate thermal control systems is given in Table 3.

TABLE 1. THERMAL CONTROL CONCEPTS [WEIGHT SUMMARY WITH EQUALIZED POWER LEVELS (100 000 Btu/hr)]

Item Concept		Pump-loop	Heat Pipe	Semipassive
Internal Cooling	Subtotal	3242	1518	2608
Ducting		500	280	1030
Fans		180	50	132
Heat exchangers		636	319	244
Pumps		90		66
Valves plumbing		1376		272
Cold plates			210	
Water and Freon		397		102
Heat pipes			55	
Headers			449	
Joints			55	
Insulation			40	
Mounts and supports		63	60	50
External Cooling	Subtotal	3290	2530	2140[d] 1630[e]
Radiator core		2476		
Pumps		80		
Freon and reservoir		647		
Header heat pipes			305	
Panel heat pipes			484	
Fins			896	
Bumper			129	
Support structure			540	
Mounts, supports, hardware		87	166[a]	
Insulation	Subtotal	497[b]	497	100
	Total (lb)	7029	4535	4236[c] 3726[c]

a. Based on requirements generated by full-scale vibration tests. Pumped-loop value seems too low.
b. Grumman design figure is 300 lb less than footnote c data.
c. Add 1745 lb for eliminating semipassive feature.
d. Pumped-loop radiators
e. Heat pipe radiators.

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TABLE 2. SUMMARY OF COMBINED HARDWARE AND POWER SYSTEM WEIGHTS

Concept	Total Weight	Weight Saving
Heat pipe	4835 lb	3915 lb
Air-cooled (semipassive)		
1 Heat pipe radiator	5246 lb	3504 lb
2 Pumped-loop radiator	5756	2994
Pumped-loop	8750	0

TABLE 3. COMPARISON OF ALTERNATE THERMAL CONTROL SYSTEMS

Factor / System	Radiator Puncture	Control	Noise	Complexity
Heat pipe	No repair required Minor capacity loss	No external devices or sensors	No noise	No moving parts, fluid connection or controllers
Pumped-loop	Switch to redundant loop	Sensor feed-back with electro-mechanical control	Rotating machinery	Multiple electro-devices, fluid-filled plumbing connections, sensors, and controls
Semipassive air-cooled	Switch to redundant loop	Sensor feed-back with electro-mechanical control	Rotating machinery	Fluid ext to press shell only

CONCLUSIONS

From the conventional pumped-loop system, two alternate thermal control systems have evolved. The heat pipe system is an advanced technique which takes maximum advantage of an operationally simple device. The air-cooled system uses an existing technology in a heretofore unfamiliar application. The hardware feasibility of each of these designs has been demonstrated. Salient features of these designs are compared in Table 3.

From a system's design standpoint, the implementation of these innovative concepts to future spacecraft design should prove to be of significant benefit. They merit consideration as alternatives to the pumped-loop system.

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APPROVAL

ADVANCED SPACECRAFT THERMAL CONTROL TECHNIQUES

By Carl H. Fritz

The information in this report has been reviewed for security classification. Review of any information concerning Department of Defense or Atomic Energy Commission programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.

This document has also been reviewed and approved for technical accuracy.

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