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HEAT PIPE THERMAL CONDITIONING PANEL

DETAILED TECHNICAL REPORT

Prepared Un[illegible]er Contract No. NAS8-28639 by

Donald W. Douglas Laboratories

McDonnell Douglas Astronautics Company

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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PREFACE

This document is the Detailed Technical Report submitted by the Donald W. Douglas Laboratories, Richland, Washington, under Contract NAS8-28639 (DCN 1-2-50-23615) and covers the period 28 June 1972 to 12 August 1973.

This program was monitored by the National Aeronautics and Space Administration's Marshall Space Flight Center, Huntsville, Alabama.

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SYMBOLS

K_p	Wick permeability (ft^2) (m^2)
A_w	Wick cross-sectional area $(ft^2)(m^2)$
r_c	Capillary radius of curvature (ft)(m)
ρ_ℓ	Liquid density $(lb/ft^3)(g/cm^3)$
h_{fg}	Latent heat of vaporization (Btu/lb)(j/kg)
σ	Surface tension (lb/ft)(kg/m)
μ_ℓ	Viscosity $(lb_m/ft/sec)$ (poise)
R_a	Bundle radius (ft)(m)
r_i	ndividual tube O.D. (ft)(m)
t	Screen thickness (ft)(m)
n	Number of tubes in multiple artery
ℓ_{eff}	$\ell_a + 1/2\ (\ell_e + \ell_c)$ (ft)(m)
K_m	Face-sheet thermal conductivity (Btu/hr/ft/°F)(w/cm/°K)
K_g	Adhesive thermal conductivity (Btu/hr/ft/°F)(W/cm/°K)
Q_a	Heat flux density $(Btu/ft^2)(w/m^2)$
t	Face-sheet thickness (ft)(m)
t'	Component base-plate thickness (ft)(m)
ℓ_1	Skin width (ft)(m)
w	Heat pipe contact width (ft)(m)
n	Effective film thickness factor
D	Heat pipe inside dia (ft)(m)
$\bar{K}$	Wick/liquid thermal conductivities (Btu/hr/ft/°F)(w/cm/°K)

Section 1
INTRODUCTION AND SUMMARY

Thermal control of electronic hardware and experiments on many of NASA's planned future space vehicles is critical to proper functioning and long life. Thermal conditioning panels (cold plates) are a baseline control technique in current conceptual studies. Heat generating components mounted on the panels are typically cooled by fluid flowing through integral channels within the panel. Replacing the pumped fluid coolant loop within the panel with heat pipes offers attractive advantages.

A heat pipe consists basically of a closed chamber with a capillary wick structure on the inner wall and a working fluid. Heat is transferred by evaporating the working fluid in a heating zone and condensing the vapor in a cooling zone. Circulation is completed by return flow of the condensate to the evaporation zone through a capillary structure. Heat pipes are nearly isothermal because the only temperature drops occur through the wall and wick in both the evaporator and condenser. Proper choice of materials yields a minimum temperature differential in the evaporator and condenser. For thermal conditioning panel applications, heat pipes provide high conductance for heat transfer to the panel edges where the heat can be rejected to a relatively modest and compact heat exchanger. The heat pipe offers a high degree of isothermalization, high reliability because of redundant heat pipe network design, light weight, and passive operation.

The objective of this program was to develop and verify a heat pipe thermal conditioning panel satisfying a broad range of future thermal control system needs on NASA spacecraft. From an initial study of spacecraft thermal requirements, design specifications were developed for a 30 x 30 in. (0.76 x 0.76 m) heat pipe panel.

Program goals included fabrication and performance verification of two heat pipe thermal conditioning panels satisfying or exceeding all thermal and mechanical constraints identified in the NASA spacecraft study. The fundamental constraint was a maximum 15°F (8.33°K) gradient from source to sink at 300 w input and a flux density of 2 w/in.^{2} (0.31 w/cm^2). The first prototype panel constructed met mechanical constraints but did not meet the design goal thermal gradient. However, modifications indicated by performance of the first panel were successfully integrated into the second panel; all design goals were met. Measured gradients were 10° to 15°F (5.55° to 8.33°K). Ultimate capacity of the panel is approximately 1 kilowatt at ΔT = 20°F (11.10°K) and 2 w/in.^{2} (0.31 w/cm^2). Panel weight is less than 20 lb (9.08 kg), and the panel will accept 100 lb (45.4 kg) of equipment with a 8g acceleration factor.

Section 2
DESIGN APPROACH

Several key features were identified in developing a thermal conditioning panel for spacecraft. An optimum design will satisfy thermal requirements of panel conductance and heat transport capacity while maintaining satisfactory mechanical strength, low weight, high reliability, and low fabrication cost. All factors were considered in the panel design. Figure 2.1 shows the thermal conditioning panel in an installation with mounted equipment modules and a heat exchanger attached along one edge.

To establish specific system constraints for panel design and mounting, and definition of general and detail specifications, equipment cooling requirements for a number of future NASA spacecraft were surveyed. Included in the study were Space Shuttle, Space Station, Space Tug, RAM, and SOAR.

2.1 SPACE SHUTTLE

The space shuttle is a transportation system for carrying personnel, cargo, and scientific payloads to and from low earth orbit.

The orbiter avionics system implements guidance and navigation, flight controls, data management, communications and navaids, avionics displays and controls, and software functions. A coolant fluid loop is required to absorb heat generated by a significant number of these electronic components, therefore requiring cold plates and/or cold rails. Components mounting on temperature controlled surfaces is similar to the methods used on previous space vehicles and aircraft.

2.2 SPACE STATION

NASA's space station program is designed to support earth surveys and the sciences of astronomy, astrophysics, biomedicine, biology, and space physics, as well as developing technology for space systems and operations. The space

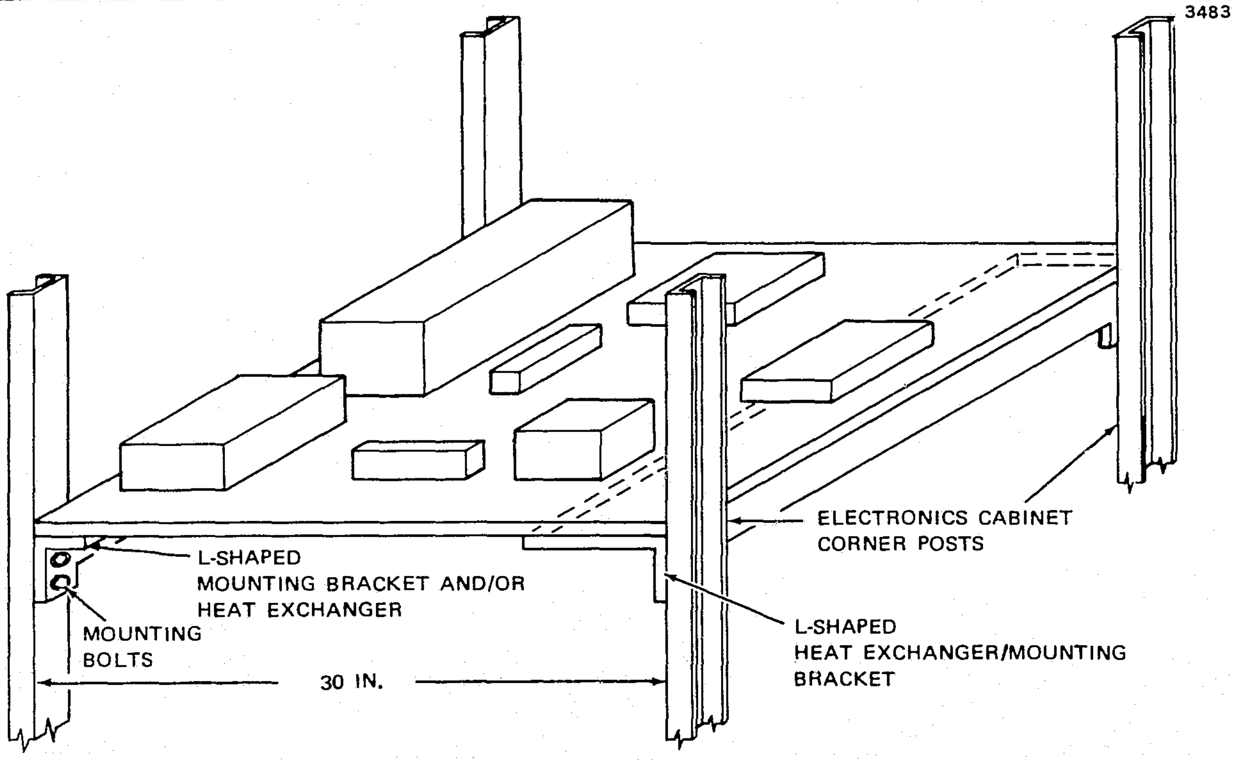

Figure 2-1. Heat Pipe Thermal Conditioning Panel Mounting Concept

station program supports these objectives by providing a long-lasting general-purpose facility in earth orbit. Dominating its design is the need to accommodate scientific personnel performing a broad range of experimental activities that may change markedly over the years. The design, therefore, emphasizes versatility for multipurpose use.

Present baseline equipment cooling is a book-shelf concept basically consisting of equipment racks with plug-in book-like modules. Equipment cooling is provided by an integral T-shaped cold plate which is part of the rack structure. Standardization of module sizes and materials is maintained to achieve relatively easy equipment installation and maintenance. A family of module sizes is provided to accommodate individual subsystem functional requirements; however, a basic module size 1.25 in. wide x 9.0 in. deep x 8.0 in. high (0.032 x 0.229 x 0.203 m) satisfies the majority of subsystem packaging requirements.

The heat dissipation limit for each basic module is 20 watts, with an average density of 0.4 w/in.^{3} (0.024 w/cm^3). Heat flux to the cold plate from each basic module is 1.77 w/in.^{2} (0.28 w/cm^2). If each book-like module were backed by a heat pipe thermal panel, a higher basic-module power density can be tolerated without exceeding critical centerline temperatures.

2.3 SPACE TUG

The space tug is the third stage of NASA's space shuttle vehicle and is designed to be delivered to low earth orbit in the shuttle payload bay. It will either deploy or retrieve earth orbiting payloads. Space tug is a highly efficient stage compatible with the shuttle orbiter and a variety of payloads and mission requirements. The space tug/space shuttle comprise the space transportation system (STS).

The principal source of heat which must be accommodated by the thermal control system for space tug is the fuel cell. The present baseline utilizes a radiator/condenser concept with pumped-loop distribution. Five sq ft (0.46 m^2) of radiator are required to provide coolant at 115° ±15°F (319° ±8.33°K) at the inlet to the fuel cells.

2.4 RESEARCH AND APPLICATION MODULES (RAM)

The research and application module system is a family of payload carrier modules that can be delivered to and retrieved from earth orbit by the space shuttle. RAM payload carriers will be capable of supporting diverse technological and scientific investigations and practical applications, primarily in areas requiring personnel participation for orbital performance, calibration, servicing, and updating. The experiment, mission, and programmatic requirements led to the evolution of three basic RAM system elements: pressurized RAMs, unpressurized RAMs, and free-flying RAMs. The overall objective of the RAM project is to provide versatile and economical payload carriers as laboratory and observatory facilities to compliment and supplement the space station and space shuttle in earth orbital research and applications activity.

2.5 SOAR

The shuttle orbital applications and requirements (SOAR) definition study identified shuttle mission applications, with emphasis on interface and design accommodation analyses for a representative range of shuttle-compatible payloads. The applications include a payload that remains attached to the shuttle payload bay throughout the mission. During the in-orbit phase, with the payload bay doors open, the shuttle radiator is deployed and the payload is operating. There is negligible heat exchange between the shuttle and payload structure. The heat to be controlled and dissipated is the heat generated by the operating equipmen in the payload. The payloads have heat loads proportional to the power each dissipates. The majority of the heat generated is rejected to the coldplates incorporated in the payload thermal control system. Potential thermal conditioning panel applications have been identified for pallet payloads and manned support modules (MSM). A coldplate thermal load totaling approximately 1300 watts is representative of the majority of the identified SOAR missions.

2.6 SUMMARY OF COLD PLATE REQUIREMENTS

Equipment cooling and mounting requirements for shuttle orbiter, space station, RAM, SOAR, and space tug have been identified and categorized. Representative panel load and sizing requirements for these applications are summarized in Table 2-1. Of these requirements, those for the shuttle orbiter are the most readily defined, the depth of design being most complete on this vehicle. The requirements established for shuttle are based on the MDAC design; however, these should be representative of the selected NAR design.

Table 2-1

THERMAL CONDITIONING PANEL SIZING REQUIREMENTS

Application	Coldplate Contact Area ($in.^2$)	(m^2)	Thermal Load (w)	Thermal Flux ($w/in.^2$)	(w/cm^2)	No. Panels	Panel Size (in.)	(m)
Shuttle Orbiter	260	(1.68)	269	1.0	(0.16)	1	17 x 17	(0.43 x 0.43)
	1569	(10.1)	1285	0.82	(0.13)	5	18 x 18	(0.46 x 0.46)
	199	(1.28)	132	0.66	(0.10)	1	15 x 15	(0.38 x 0.38)
RAM	9504	(61.3)	7226	0.76	(0.12)	25	20 x 20	(0.51 x 0.51)
SOAR	1807	(11.7)	1288	0.71	(0.11)	5	19 x 19	(0.48 x 0.48)
Space Tug	144	(0.93)	290	2.01	(0.31)	1	8 x 8	(0.20 x 0.20)
Space Station	11	(0.07)	20	1.81	(0.28)	1	9 x 1.25	(0.23 x 0.03)

2.7 CONDITIONING PANEL CONCEPTS

Among design concepts utilizing heat pipes to transfer heat and isothermalize, there are two fundamental approaches. In the vapor chamber approach, thermal conditioning panel faceplates not only transfer heat and mount components, but also contain the working fluid and vapor inside the panel. In the other approach, the working fluid/vapor system is contained within separate heat pipes within the panel and the faceplates transfer heat and mount components.

Although a vapor chamber system has thermal advantages in less interfaces and maintains a high degree of isothermality because of the open structure, the generally disadvantages are reliability, weight, and cost.

For any panel configuration to be considered as a possible design, it must have proven reliability, performance, ease of fabrication, and successful operation. The vapor chamber inherently requires simultaneous assembly of many parts for bi-directional heat transfer. It is difficult to use reliable current-technology techniques to fabricate such a panel. With the large number of equipment mounting bosses required, it is difficult to fabricate a continuous wicking system and ensure the many leak-tight joints required in such a design. Although the concept of a vapor chamber has been explored for years, it has not been reduced to practice in a panel of this size. With high-pressure working fluids, vapor chambers are often excessively heavy because the inherent weakness of a rectangular pressure vessel necessitates thicker faceplates to obtain reasonably flat mounting surfaces.

A preliminary investigation was made with ammonia working fluid, evaluating the relative weight of a 30 x 30 in. (0.76 x 0.76 m) vapor chamber and a composite panel (i.e., separate heat pipes and face plates). Total panel weight for the vapor chamber was calculated, assuming pressure containment was the controlling factor on faceplate thickness. Figure 2-2 shows the weight and face-plate thickness of a 30 x 30 in. (0.76 x 0.76 m) vapor chamber as a function of the distance between supports. To stay within 0.010-in. (0.025 cm) TIR design goal on surface flatness, 0.005-in. (0.013 cm) outward bulging from internal pressure was allowed. On a weight basis, an 18-lb (8.16 kg) composite panel satisfying thermal requirements is equal to a vapor chamber which must have coupling bars at less than 1-in. (2.54 cm) intervals. That is, to satisfy dimensional

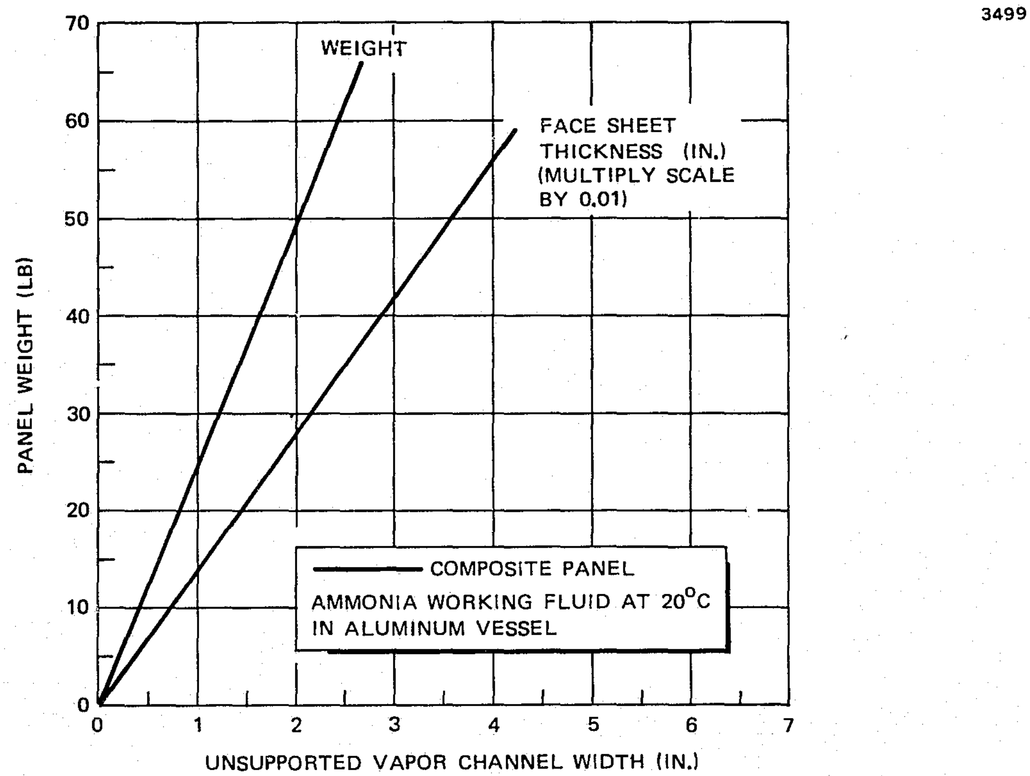

Figure 2-2. Vapor Chamber Weight and Face-Sheet Thickness

goals, a crossbar must firmly connect the two faces at less than 1-in (2.54 cm) intervals across the panel width. In addition, there is little allowance for excursions in temperature resulting from exponential vapor pressure dependence on temperature, whereas the composite panel has been proof-pressure tested to 1500 psig (10,353,168.21 n/m^2).

In summary, the vapor chamber suffers from a weight problem if ammonia or other high-pressure fluids are to be used (or if the panel is subjected to high temperatures), and appears to be more unreliable and costly in a panel such as this. The composite panel is a much more flexible design and was chosen as the best candidate design to meet the thermal conditions in Table 2-2.

Table 2-2

THERMAL PANEL GENERAL SPECIFICATIONS

	Original Specification	Application Study Recommendation
Size of Panel	30 x 30 in. (0.76 x 0.76 m)	30 x 30 in. (0.76 x 0.76 m)
Thermal Load		
Mounting Boxes	10 w	10 w
Max. Density	5 w/in.^{2} (0.78 w/cm^2)	2 w/in.^{2} (0.31 w/cm^2)
Max. Total per Panel	300 w	300 w
Mounting Surface Temperature	32° to 77°F (273° to 298°K)	32° to 85°F (273° to 303°K)
Temperature Gradient		
Across load areas	5°F (2.77°K)	5°F (2.77°K)
Between panel surface points at source and sink	15°F (8.33°K)	15°F (8.33°K)
Available Sink Temperature	32° to 70°F (273° to 294°K)	32° to 85°F (273° to 303°K)
Bolt Pattern	4 x 4 in. (0.10 x 0.10 m) centers	Adaptable
Component Mass	100 lb (45.4 kg) max	100 lb (45.4 kg) max

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Section 3
DETAILED DESIGN ANALYSIS

Detailed design of the two thermal conditioning panels constructed is discussed in this section. Section 3.1 considers factors in the design and selection of heat pipe wicking and working fluid; Section 3.2 presents detailed thermal models used in evaluating heat pipe configurations for the thermal conditioning panels.

3.1 HEAT PIPE MATERIALS EVALUATION

Independent of geometrical factors concerning heat pipe configuration within the panel, consideration was given to optimizing the heat pipe. This optimization included heat transport capacity, thermal gradients, and materials compatibility.

Anhydrous ammonia was selected as the best working fluid to use on the basis of high Figures of Merit*, high thermal conductivity, and most extensive compatibility data with both aluminum and stainless steel (Table 3-1). The only disadvantage with ammonia is toxicity, which precludes its use in manned areas. Table 3-2 presents toxicity information on various working fluids. Ammonia is very toxic, while the Freons, which have low Figures of Merit and thermal conductivity, are relatively non-toxic. The fabrication of a Freon working-fluid panel prototype for manned missions would however be desirable. Ammonia is an excellent working fluid with which to demonstrate the basic feasibility of a heat pipe thermal conditioning panel, and there are many future unmanned missions for which an ammonia panel may be desirable or necessary because of high heat transport demands, the necessity of minimizing temperature differentials, or operating at temperatures below 32°F (273°K).

Aluminum was chosen for the heat pipe because of high thermal conductivity and availability in a variety of extruded forms. The wicking material selected was Type 304 stainless steel. Stainless-steel screen is compatible with ammonia and is available in a fine mesh which provides maximizing capillary pumping.

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*Thermal Figure of Merit = $\frac{\sigma \rho H_{fg}}{\mu}$

Pumping Figure of Merit = $\frac{\sigma}{\rho}$

Table 3-1

LONG-TERM COMPATIBILITY TEST COMBINATIONS

Fluid	Material	Temperature (°F)	Temperature (°K)	Time (hr)	Test Mode
Methanol	Stainless Steel	175	352	30,000	Reflux
Water	Copper	175	352	30,000	Reflux
Ammonia	Aluminum/Stainless	120	322	8,350	Accelerated life
Ammonia	Aluminum/Stainless	60	289	43,800	PAC/OSO-G (ground + in-flight)
Ammonia	Aluminum/Stainless	70	294	43,600	Static
Ammonia	Aluminum/Stainless	-120/100	200/311	8,350	Eclipse simulation
Freon-22	Aluminum/Stainless	130	327	130	Reflux
Water	Copper	100	311	15,050	Reflux

Table 3-2

TOXICITY OF FLUIDS

Fluid	TLV*
Pyridine	5
Ammonia	50
Methanol	200
Acetone	1000
Ethanol	1000
Freon-21	1000
Freon-22	Comparatively non-toxic; exposure limit of 1000 ppm is generally accepted.

TLV*: Threshold Limit Value, ppm of air by volume at 25°C (298°K) and 1 atm (1.0132×10^5 n/m^2)

3.2 WICKING CONSTRUCTION

The transfer of heat in a heat pipe is limited by maximum axial heat flow and maximum radial heat flux. For low temperature heat pipes such as those for spacecraft thermal control, the most important limitation on maximum axial heat flow is the capillary pumping limit. This limit is a function of fluid properties, and is given, for perfect wetting, by

$$Q \cdot \ell_{eff} = \frac{2K_p A_w}{r_c} \left(\frac{\rho_\ell h_{fg} \sigma}{\mu \ell} \right) \qquad \text{(w-cm)} \qquad (3.1)$$

The multiple artery wicking arrangement used in the thermal conditioning panel heat pipes is shown in Figure 3-1. A large screened tube is closely packed with a number of smaller screened tubes. Each small tube acts as an artery with high axial permeability; the bundling of tubes provides high redundancy. If a section of one artery is blocked by noncondensable gas, the remaining arteries shunt fluid to the evaporator. In empirical testing of these structures in the presence of noncondensable gas, re-priming after emptying was very good.

The permeability K_p (Reference 1) of this arterial structure is

$$K_p = \frac{1}{8} \left[\frac{R_a^2 - nt(2r_i - t)}{R_a + n(2r_i - t)} \right]^2 \qquad (3.2)$$

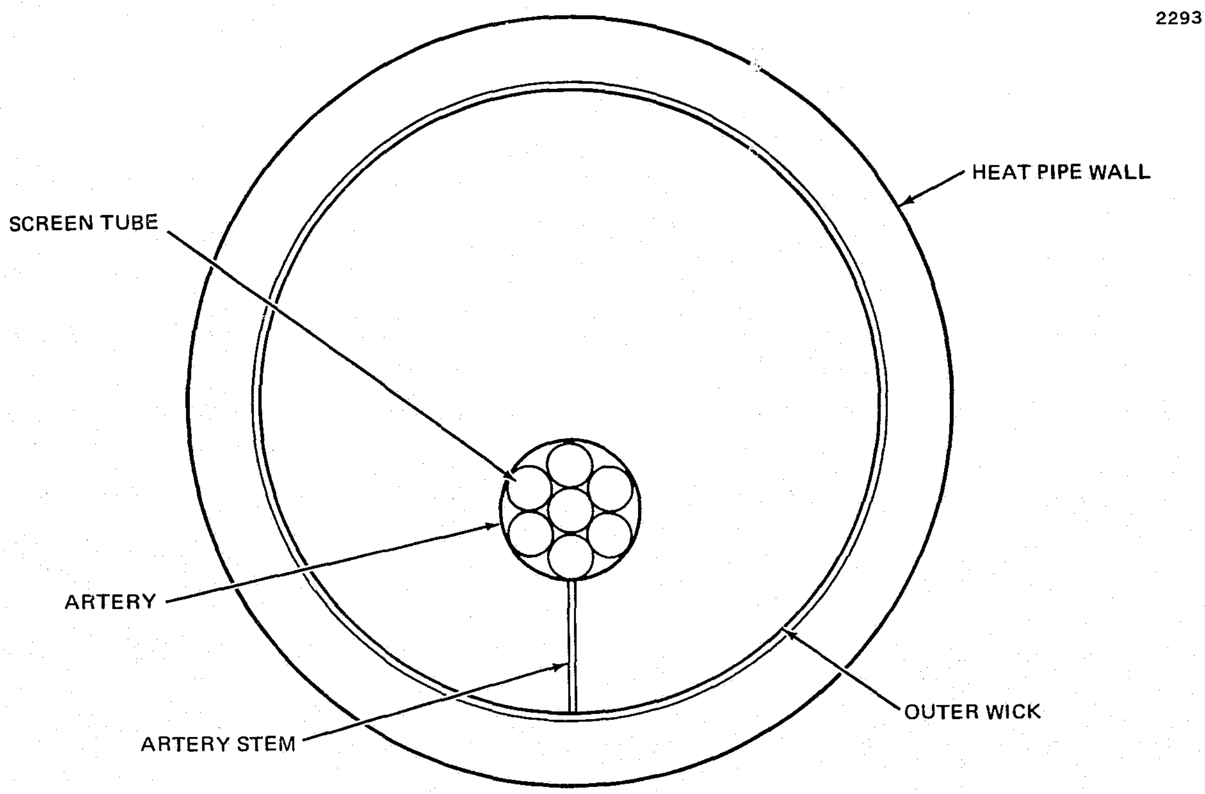

Figure 3-1. Multiple Artery Wick System

For the 5- and 7-tube arteries in the first and second panels, respectively, the transport capacities of an individual heat pipe (Equation 3.1), are 2500 and 1200 w-in. (6350 and 3048 w-cm). Additional tubes in the second panel artery bundle, within the same bundle diameter, reduce capacity while increasing fluid transport redundancy. Because the maximum demand on a single heat pipe in either panel is about 500 w-in. (1270 w-cm), there is a considerable operating safety margin.

The circumferential wicking on the heat pipe wall is also stainless screen; the minimum capillary pumping radius r_c is approximately 0.0025 in. (0.0064 cm). Circumferential wicking thickness is approximately 0.011 in. (0.028 cm).

3.3 DETAILED THERMAL MODELING

Configurations for the two thermal conditioning panels are shown in Figures 3-2 and 3-3. Configuration selection is discussed in Sections 3.4 and 3.5.

The thermal gradients within the heated area are shown by the model in Figure 3-4. In cross-section, a thermal cell is $\ell_1 + \ell_2 + w$ wide and ℓ_s long. Maximum

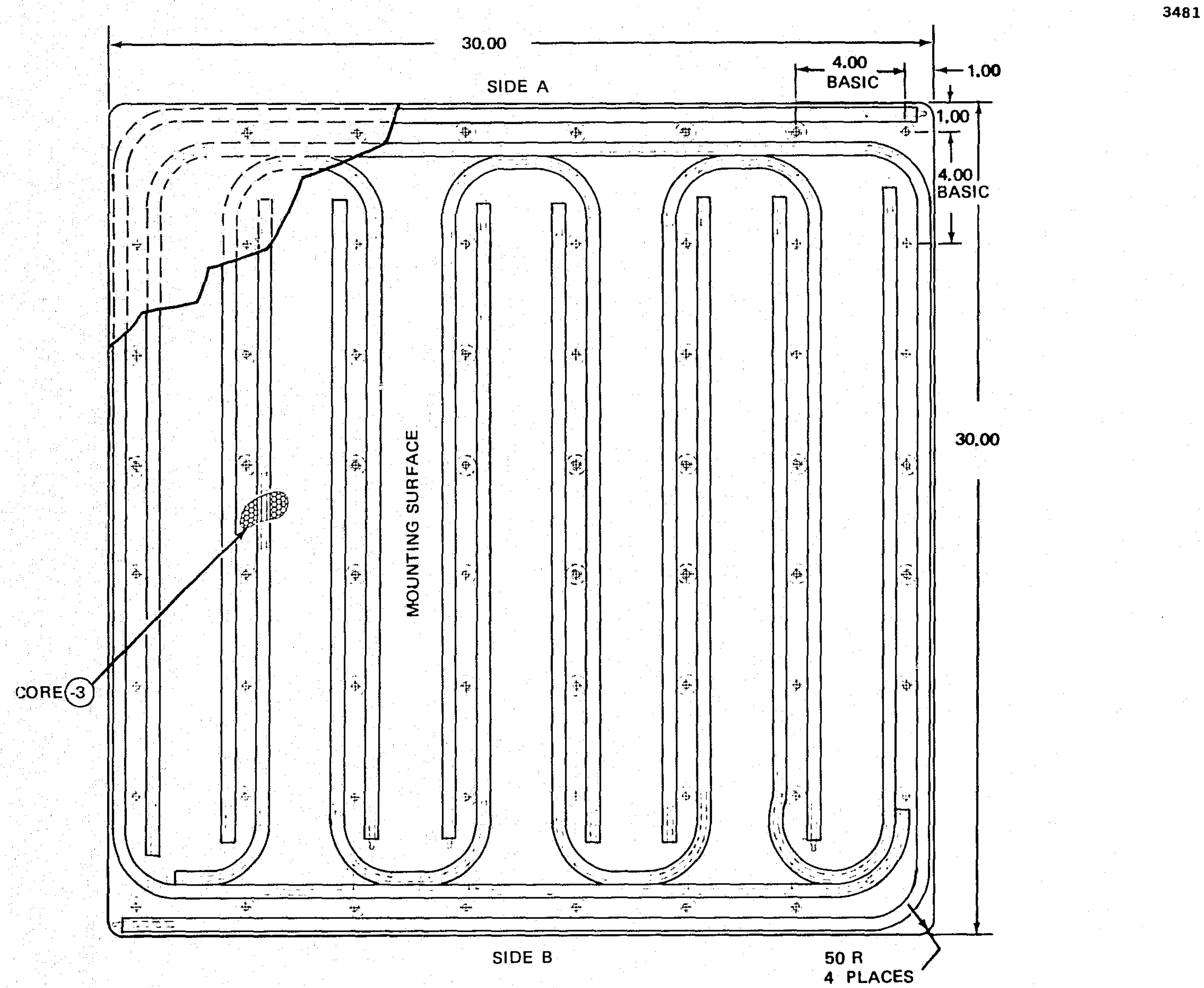

Figure 3-2. Thermal Conditioning Panel No. 1

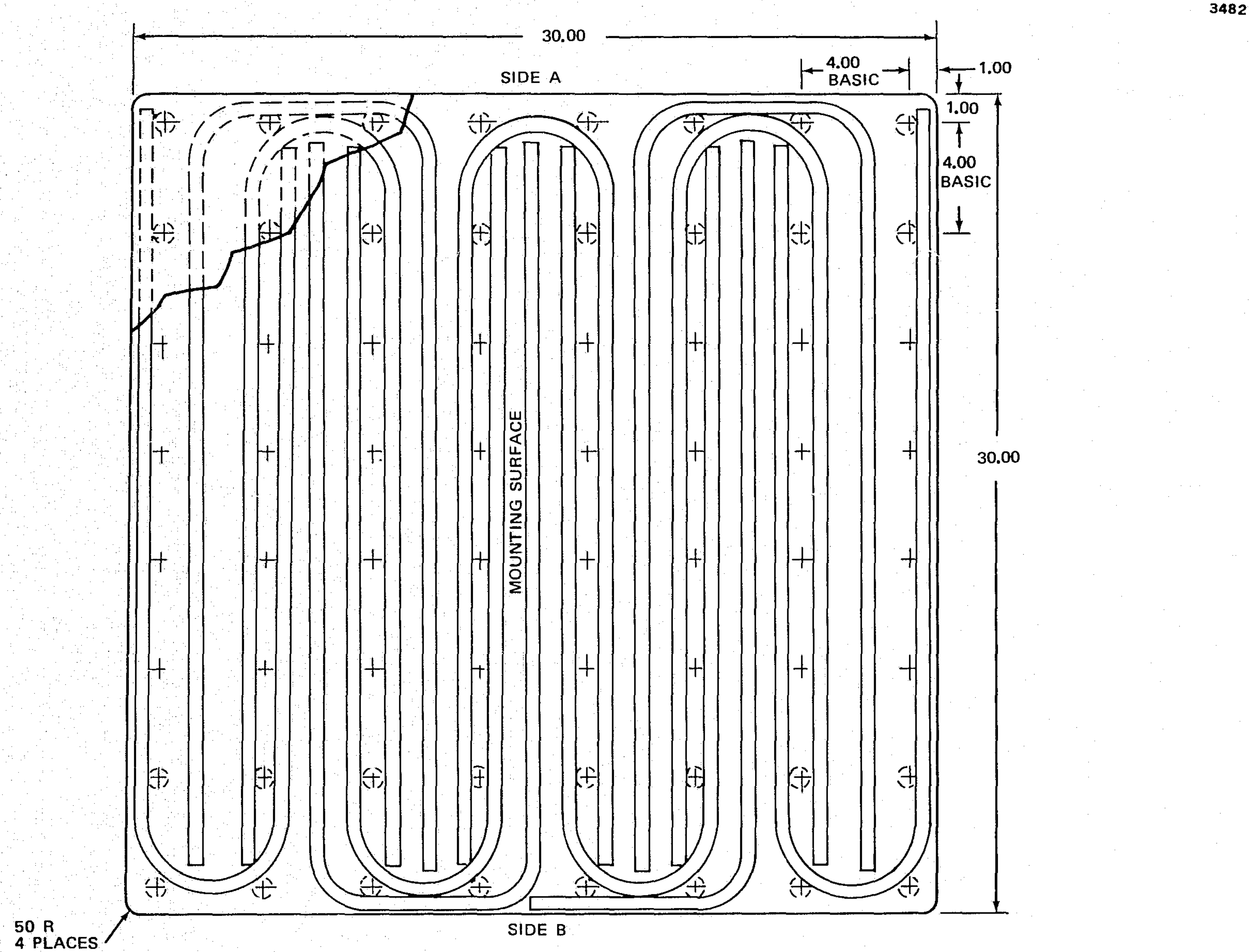

Figure 3-3. Thermal Conditioning Panel No. 2

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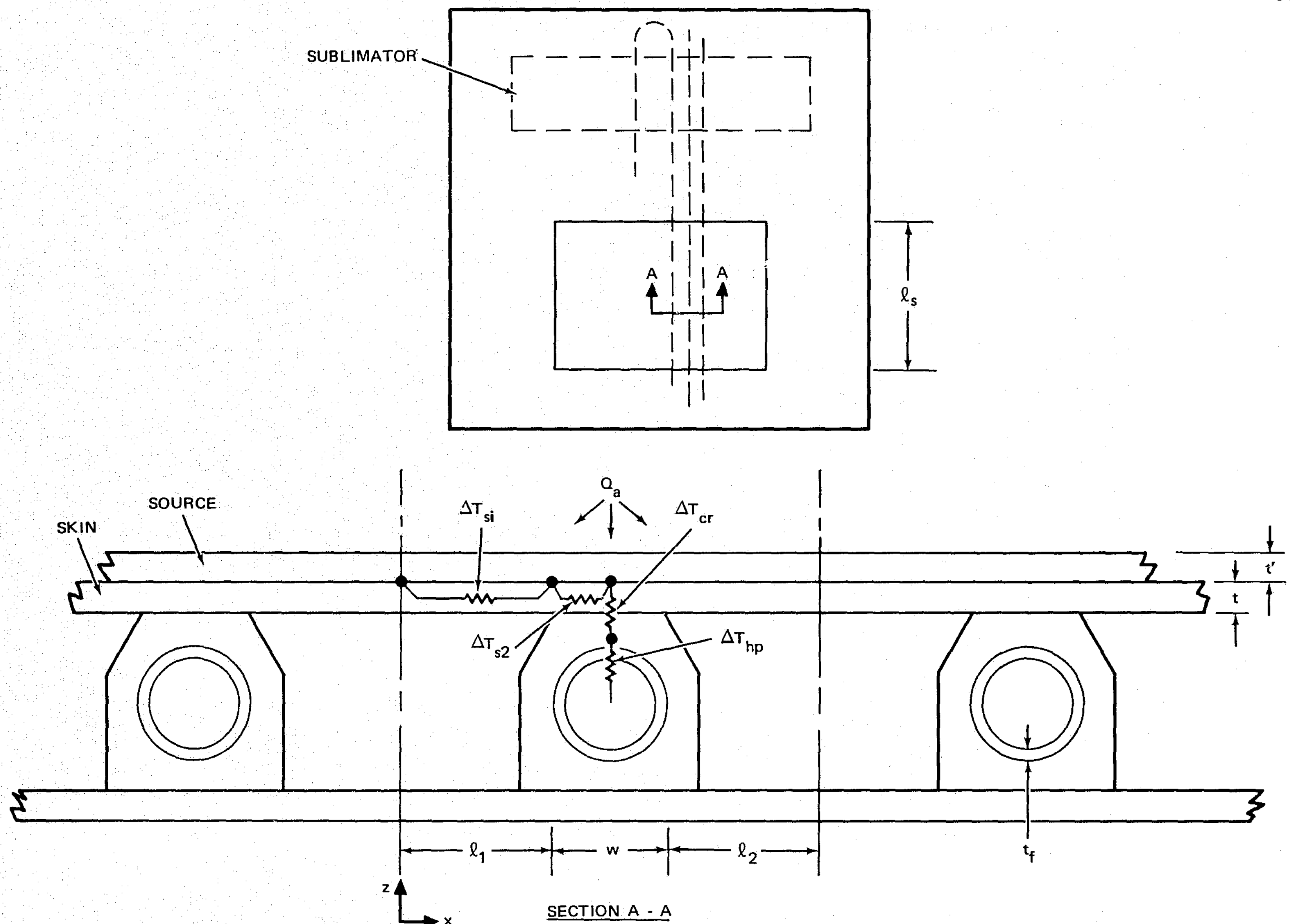

Figure 3-4. Thermal Model

surface temperature occurs at a cell boundary. Thermal resistances from a boundary to the heat pipe vapor core consist of 2 skin ΔTs, (ΔT_{s1} and ΔT_{s2}) a contact resistance ΔT_{cr}, and a heat pipe ΔT_{hp}.

$$\Delta T_{max} = \Delta T_{s1} + \Delta T_{s2} + \Delta T_{cr} + \Delta T_{hp} \tag{3.3}$$

Assuming at any x-position the panel face, t, and heat source face, t', are at a uniform temperature and composition, then the skin thermal drop ΔT_{s1} is obtained by solving the one-dimensional differential heat conduction equation with the following conditions

1. $dt/dx = O, \quad x = O$
2. Heat injection rate $= Q_a$
3. $-K_m d^2T/dx^2 = Q_a/(t + t')$

The gradient ΔT_{s1} is given by

$$\Delta T_{s1} = Q_a \ell_1^2 / 2K_m (t + t') \tag{3.4}$$

If uniform heat rejection is assumed over the heat pipe contact width w, then a similar analysis yields

$$\Delta T_{s2} = \frac{Q_a \ell_1 w}{4 K_m (t + t')} \tag{3.5}$$

Through the adhesive-filled gap of thickness t_g, gradient ΔT_{cr} is

$$\Delta T_{cr} = Q_a \left(1 + \frac{2\ell_1}{w}\right) \frac{t_g}{K_g} \tag{3.6}$$

The predominant temperature drops in a heat pipe occur through the fluid films covering the wall in the evaporator and condenser sections. Using the heat conduction law, the evaporator film temperature drop for a saturated wick is

$$\Delta T_{fe} = \frac{2 Q_a \ell_1 \left(1 + \frac{w}{2\ell_1}\right) (t_f \cdot n)}{\pi \bar{K} D} \tag{3.7}$$

The effective thermal conductivity $\bar{K}$ of the fluid film is often close to the bulk fluid conductivity because wicking often has high porosity and low conductivity.

Therefore, to minimize ΔT for the panel, a fluid with high conductivity is desirable. The film thickness t_f is nominally equal to the wicking thickness in both the condenser and evaporator because the modest capillary demand on the multiple artery condensate return structure. However, other thermal tests show inhomogeneities in wick thickness and extrusion gradients give an effective film thickness factor n of 1.18.

Condenser film drop is evaluated taking the total heat transferred, Q, and dividing by the total length of condenser-intercepted heat pipe. The unit-length heat dissipation factor Q_ℓ is substituted into the heat conduction equation to give

$$\Delta T_{fc} = \frac{Q_\ell (t_f \cdot n)}{\pi \bar{K} D} \tag{3.8}$$

Within the condenser sections, ΔT_{s1} and ΔT_{s2} can generally be disregarded because heat exchanger mass and thermal characteristics overwhelm surface effects.

To verify modeling validity, calculated values of the thermal gradient for the second panel are compared in Table 3-3 with experimental values for configuration C2 and the sublimator heat sink at 300 watts and 2 w/in.^{2} at 60°F (0.31 w/cm^2 at 289°K). Configurational details are given in Section 3.7.1.

Table 3-3

CALCULATED AND EXPERIMENTAL THERMAL GRADIENTS

	ΔT_{s1}	$+ \Delta T_{s2}$	$+ \Delta T_{cr}$	$+ \Delta T_{hp}$	$= \Delta T$(calc)	ΔT(exp)
Evaporator	0.64	0.28	1.42	3.88	6.22°F	
Condenser	-	-	2.14	5.84	7.98	
Total					14.2°F (7.89°K)	14.1°F (7.83°K)

Agreement is satisfactory. The bond-line thickness for the second panel was nominally 0.0035 in. (0.0089 cm), but 25% of the panel gradient appears across the Deltabond 154 adhesive. Table 3-4 summarizes thermal conductivities for various bonding agents. The bonding agent for the second panel has the relatively high conductivity, compared with other common industrial adhesives.

Table 3-4

THERMAL CONDUCTIVITIES OF THERMAL JOINT COMPOUNDS

Compound	Thermal Conductivity (Btu/hr-ft-°F)	(w/cm/°K)
Eccobond 56-C (unpolymerized)	1.04	0.0180
Eccobond 56-C (polymerized)	0.44	0.0076
Devcon F (AI putty)	0.50	0.0087
DC-340 Heat Sink Compound	0.43	0.0074
Dow Corning 732 RTV	0.14	0.0024
30% 732 RTV/70% Cu Powder	0.45	0.0078
Dow Corning Vacuum Grease	0.11	0.0019
30% Dow Corning Vacuum Grease/70% Cu Powder	0.45	0.0078
Honeywell Heat Conducting Compound No. 107408	0.23	0.0040
Thermon T-5 (John H. Marvin Co., Inc.)	0.19	0.0033
AF 126-2	0.10	0.0017
Deltabond 154	0.67	0.0116
Eccobond 285	0.87	0.0151

3.4 INITIAL PANEL DESIGN

Several heat pipe configurational patterns for the panels were evaluated to obtain a design with high conductance, multidirectional heat transfer capability, and functional redundancy. Figure 3-2 shows the design selected for the first heat pipe thermal conditioning panel. U-shaped heat pipes from sides A and B intermesh to form a redundant network of thermal sinks through the panel width. If any one heat pipe fails in this configuration, heat pipes on both sides of the failed pipe take up the load. Header heat pipes around the edges of the panel transfer heat from the array pipes (oriented from A to B) to cold rails mounted on any panel edges. Nine heat pipes were used, with 0.063-in. (0.16 cm) facesheets and aluminum honeycomb between heat pipes. Honeycomb was used for additional rigidity because of the large open spaces between adjacent heat pipes. The heat pipe extrusion is shown in Figure 3-5.

The first panel did not meet the design goal because of excessive temperature gradients at maximum power and flux density. Extensive tests and calculations established that the cause of this high gradient was contact resistance between the heat pipes and faceplates and the header and array heat pipes. Using an

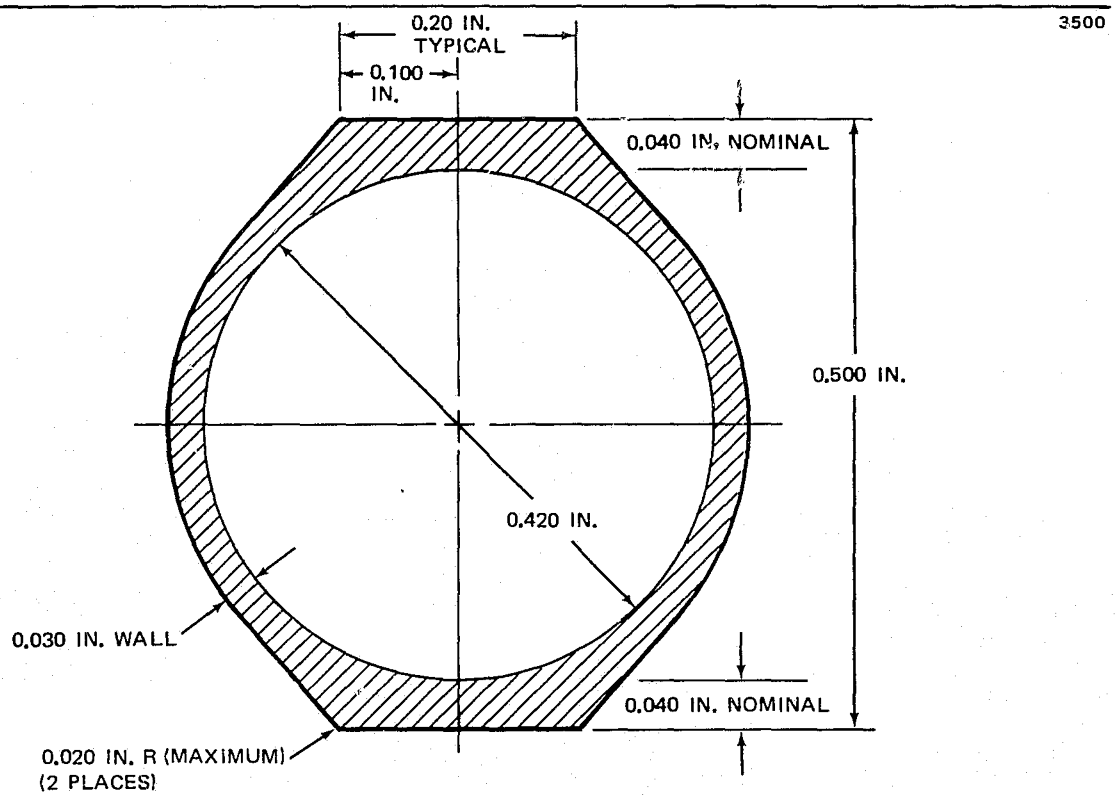

Figure 3-5. Panel No. 1 Extrusion

eddy-current detector, which senses gaps between metallic surfaces, it has been established that large areas of the panel have gaps from 0.001 to 0.010 in. (0.003 to 0.025 cm) filled with AF 126-2 low-conductivity bonding agent between the heat pipes and the plates. From the heat conduction equation, 0.001-in. (0.003 cm) of epoxy adhesive produces a 7°F (3.89°K) temperature gradient at rated power. In confirmation, heat sources placed on various sections of the panel substantiate eddy-current measurements by indicating larger gradients over large-gapped areas. A high thermal drop across the DC-340 heat sink compound used in mounting heat sources and sinks must be discounted because small exposed areas of the panel near the source and sink centers indicate within 1°F (0.56°K) of the temperatures on the source and sink surfaces. Analytical calculations show that the gradient is much too large to attribute to fluid film within the heat pipe. All evidence, analytical, thermal, and non-thermal, indicates inadequate contact as the source of the high gradient.

3.5 FINAL PANEL DESIGN

The design of the second panel (Figure 3-3) reflects changes to generally improve thermal performance and minimize contact resistances. Header heat pipes have been eliminated so that heat transfer is directly from heat source to heat sink. Some freedom in placement of the cold rails has been sacrificed to achieve thermal gradient improvements, because cold rails in this design are placed on edges A and B to intercept as many heat pipes as possible. However, this is not a severe constraint. A general comparison of the two panels is given in Table 3-5.

For both mechanical and thermal reasons, the extrusion used on panel No. 1 has been replaced with a square extrusion with 0.50-in. (1.27 cm) faces (Figure 3-6). The square extrusion does not twist as much on forming, and the contact area per heat pipe increases because the square extrusion has a contact face twice as wide as the extrusion used in panel No. 1. To further increase contact area, the array spacing has been modified to halve the pipe-to-pipe spacing. This decreased spacing allows the use of a thinner 0.040 in. (0.102 cm) faceplate which compensates for the heavier square extrusion.

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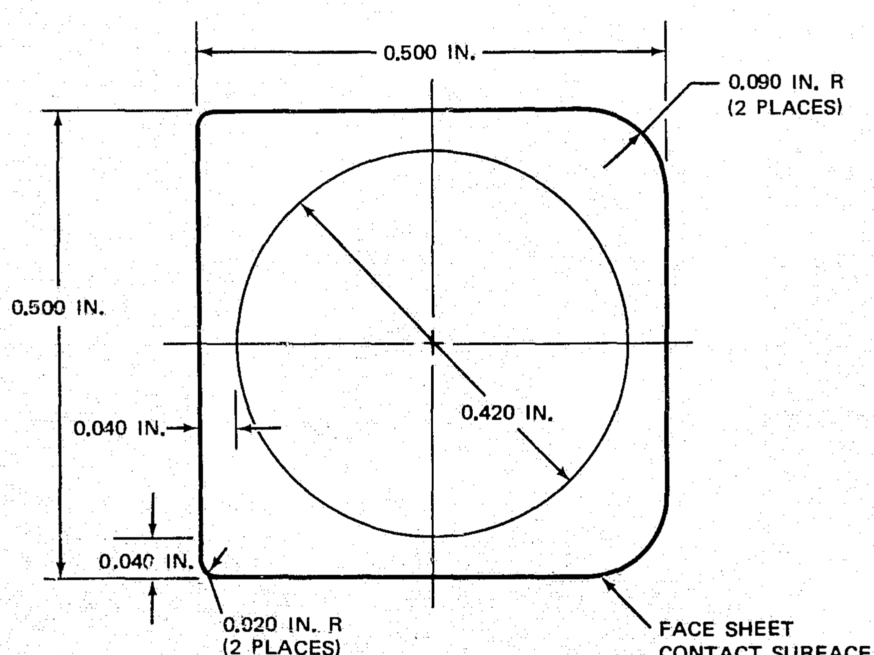

Figure 3-6. Panel No. 2 Extrusion

The interaction of heat pipe spacing, thermal gradient, panel stiffness, and panel weight is shown in Figure 3-7. A computer program was written to solve the thermal equations discussed in Section 3.3 for panel weight and stiffness as a function of heat pipe spacing, with thermal gradient held less than or equal to some maximum value. As heat pipe spacing is increased, panel weight first decreases, then increases rapidly because of increased faceplate thickness to compensate for the increased skin ΔT and higher heat fluxes at each heat pipe. Minimum weight follows the lower curve in Figure 3-7. For example, at $\Delta T = 12.5°F$ (6.94°K) a heat pipe spacing of 1.16 in. (0.029 m) is possible with only a 13.5-lb (61.29-kg) panel. For smaller pipe-to-pipe spacings, the ΔT is less than 12.5°F (6.94°K) but the weight is higher because of more extrusion. To maintain a 12.5°F (6.94°K) gradient at spacings larger than 1.16 in. (0.029 m) faceplate thickness must increase very rapidly to maintain the gradient below that value because heat pipe and contact resistances are the dominant gradients, as shown in Table 3-3. This creates a rapid weight increase as shown by the steep vertical curve. While it is desirable to minimize weight, the minimum weight panel also has the lowest rigidity. Moment of the inertia is defined by $\int Z^2 dA$ about the centroidal axis, where Z is the distance normal to the panel faces measured from the extrusion equator, and dA is the elemental area. The rigidity of a panel is rapidly increased as a faceplate thickness increases (broken curve, Figure 3-7), because the faceplates are far from the extrusion equator, in analogy with the horizontal members of an I-beam.

For the second panel, a thermal-mechanical compromise was required to satisfy thermal requirements and provide adequate rigidity so that epoxy-bond stress levels are within safe limits and deflection under load is not objectionable. Center-to-center heat pipe spacing is approximately 1.3 in. (0.033 m), gradients are below 15°F (8.33°K), and panel rigidity (0.24 in.^{4}; 9.99 cm^4) is slightly higher than for an equivalent solid extruded panel (0.23 in.^{4}; 9.57 cm^4). That is, the stiffness of this composite panel is somewhat higher than the stiffness of a panel where the center-to-center spacing is 0.5 in. (1.27 cm), implying no gap between heat pipes and the utilization of sixty 0.50-in. (1.27 cm) extrusions of the type shown in Figure 3-6. This panel strength results principally from the I-beam effect of the 0.040 in. (0.102 cm) facesheets.

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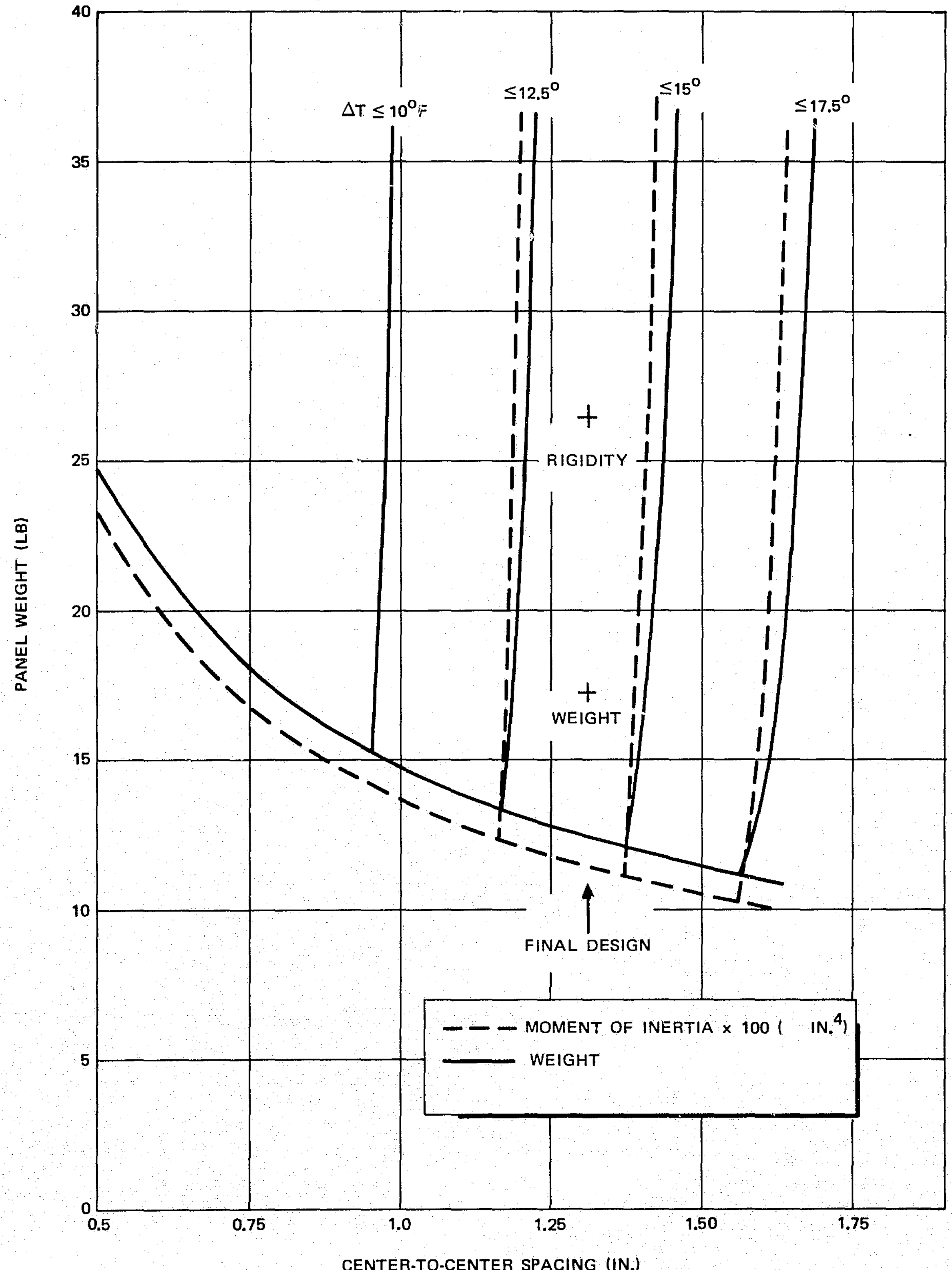

Figure 3-7. Effect of Heat Pipe Spacing on Panel Weight

3.6 FABRICATION TECHNIQUES

Poor thermal performance of the first panel is related in part to fabrication techniques. Excessively thick bondlines, attributable to fit-up of the heat pipes against the facesheets, was the major problem. The poor contact results in part from extrusion thickness increases produced during bending and forming, which increase extrusion height 0.005 in. (0.0127 cm) on the inside of bend radii.

The problem was compounded during bonding. The importance of maintaining surface flatness was underestimated. In addition, extrusion thickness varies with position, and it appears that the honeycomb material used in the first panel may have been limiting compression of the facesheets against the heat pipes.

The decision was therefore made on the second panel to use a more dimensionally stable extrusion, eliminate the honeycomb if possible, lap the heat pipe faces to ensure optimum contact, and use a higher conductivity adhesive for bonding. Sections 3.6.1 and 3.6.2 discuss manufacture of the second heat pipe thermal conditioning panel.

3.6.1 Heat Pipe Manufacturing

All necessary material and finished component parts for heat pipe fabrication were purchased from MDAC qualified sources. Upon receipt, each was inspected. All purchase records and inspection results were maintained for complete material traceability.

For the thermal conditioning panel heat pipes, Type 6063 aluminum alloy extrusions were first cut to length, prepared for welding, and cleaned. A Type 304 stainless-steel wick was then drawn into the extrusion and trimmed to length. Each heat pipe was then bent to shape. The heat pipes received a final cleaning followed by a check for nonvolatile residue. End plugs were then fitted into the heat pipes and a TIG weld made at each end to complete the heat pipe. Welds were made in compliance with MIL-W-8604 by certified welding personnel. Each weld was radiographically inspected in at least two orientations in accordance with MIL-STD-453.

Heat pipes were then worked to remove any twists and large sinusoidal variations from flatness, as indicated on a surface table. One face of each heat pipe was lapped using 220- and 600-mesh (7 and 2.8 x 10^{-5} μm) carbide abrasive slurry on a piece of 5/16-in. (0.794 cm) float glass mounted on the surface table. Lapping was terminated when the heat pipe face was flat within 0.0015 in. (0.0038 cm), as indicated by a flat feeler gage. Extrusion thickness was measured and recorded. After lapping, each pipe was proof-pressure tested to 1500 psig (1.04 x 10^6 n/m^2) and a helium leak test was performed at 10^{-8} std ml/sec sensitivity.

Heat pipes passing all quality assurance tests were evacuated, flushed with a purge charge of ammonia, and filled with a precise charge of high-purity ammonia. After charging, the filler tube was cold crushed followed by a TIG seal weld at the feathered edge. The pinch-off closures were radiographically checked and leak tested for ammonia evolution to 10^{-6} std ml/sec.

3.6.2 Panel Fabrication

One face sheet and all fastener blocks were chemically cleaned in accordance with MDAC specifications and bonded in place using Delta-Bond 154 aluminum-filled epoxy and hardener type C, which allows a 4-hr use time. The epoxy was polymerized at 150°F (339°K) for 15 hours. The heat pipes and face sheet were again chemically cleaned and the heat pipes were similarily bonded. Surface flatness was ensured by use of a vacuum frame which pressed the face-plate-heat pipe assembly against a surface table. Total force on the panel was about 1800 lb (8172 kg). After bonding of the heat pipe array to the first face, the unbonded faces were lapped as an assembly with 220 and 600 mesh (7 and 2.8 x 10^{-5} μm) abrasive in the same way as the individual pipes were lapped. Lapping continued until all surfaces were flat and parallel within 0.0015 in. (0.0038 cm) TIR. The average amount of material removed on each heat pipe face was 0.0020 in. (0.0051 cm).

The second facesheet was bonded using identical materials and techniques as the first side except that a 0.062 in. (0.157 cm) rubber blanket was placed between the facesheet and table to ensure good contact. After curing at 150°F (339°K) for 15 hours, the panel was checked dimensionally.

All dimensions were within specifications. The adhesive bond thickness varies from 0.002 to 0.005 in. (0.005 to 0.013 cm) over the panel; nominal bond thickness is about 0.0035 in. (0.0089 cm).

3.7 ACCEPTANCE TESTING

The heat pipe thermal conditioning panel was tested with a variety of heat source and mounting configurations. Heat source and sink configurations selected for testing reflect variations in component and heat sink mounting.

3.7.1 Heat Sources and Sinks

Heat sources used in testing are shown in Figure 3-8. Each of the larger sources has a surface area of 150 in.^{2} (0.9675 m^2) and at 2 w/in.^{2} (0.31 w/cm^2), 300 w are applied to the panel. The smaller spot heat source is used at a flux density of 2.75 w/in.^{2} (0.43 w/cm^2) to simulate a single high-flux source. To maintain uniform heat injection, the heater elements are 0.094 in. (0.239 cm) wide Inconel flat resistance wires spaced 0.15 to 0.175 in. (0.38 to 0.445 cm) center-to-center on 1/16 in. (0.159 cm) sheet aluminum. The heat rejection systems, a simulated

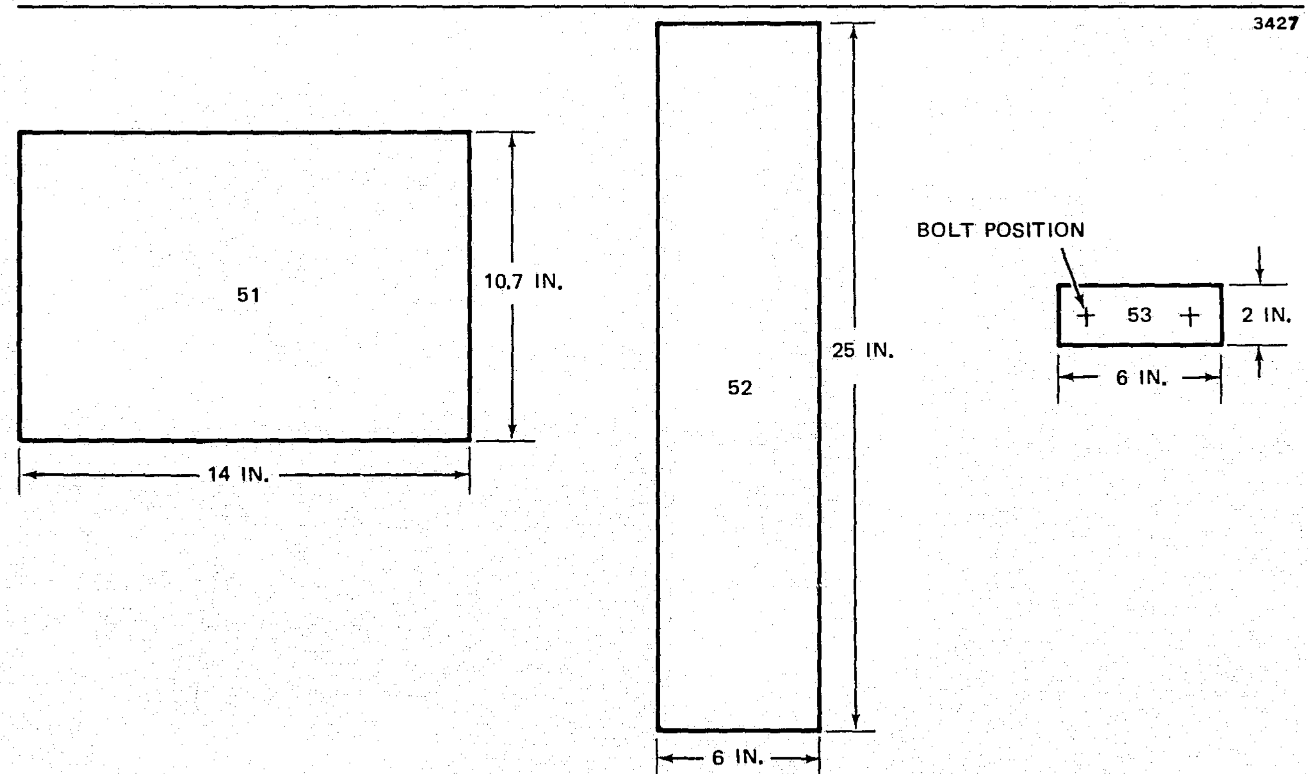

Figure 3-8. Heat Source Profiles

water sublimator, and cold rails, are shown in Figures 3-9 and 3-10. Figure 3-10 also shows placement of the cold rails and sublimator on the thermal conditioning panels. The first panel was tested with the 5 x 21 in. (0.127 x 0.533 m) sublimator simulator and two of the cold rails shown in Figure 3-9. Each cold rail extended about 3.0 in. (0.076 m) onto the panel at edges A and B, on top and bottom of the panel. The second panel had one cold rail 6.1 in. wide and 30.0 in. long (0.155 x 0.76 m) on one side of edge A only, leaving the other surface free for component mounting. Although panel width intercepted by the second cold rail is only slightly more than the sum of rail widths for the first panel, lower contact resistances within panel No. 2 eliminated the need for both top and bottom rails.

Figure 3-10 shows seven heat input configurations (C1 through C7) selected for rigorous thermal characterization. Configuration C1 through C5 are noted for 300 w at 2 w/in.^{2} (0.31 w/cm^2) while C6 is used at a spot heat source up to 5 w/in.^{2} (0.78 w/cm^2); C7 is a distributed heat source at 2/3 w/in.^{2} (0.1 w/cm^2). Tests were also run at elevated and reduced temperatures, as well as vertical and horizontal orientations, to obtain a complete thermal representation of the panels. In general, the thermal panel gradient has been characterized at three inputs, 100, 200, and 300 watts for each heat source configuration C1 through C5. For each heat source configuration the heat sinks used are summarized in the text and figures of section 3.7.3.

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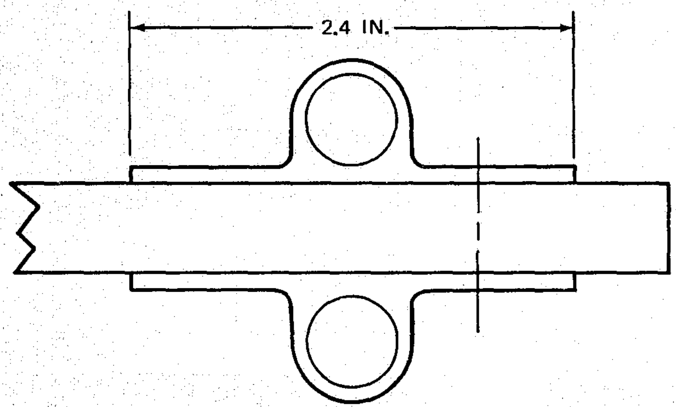
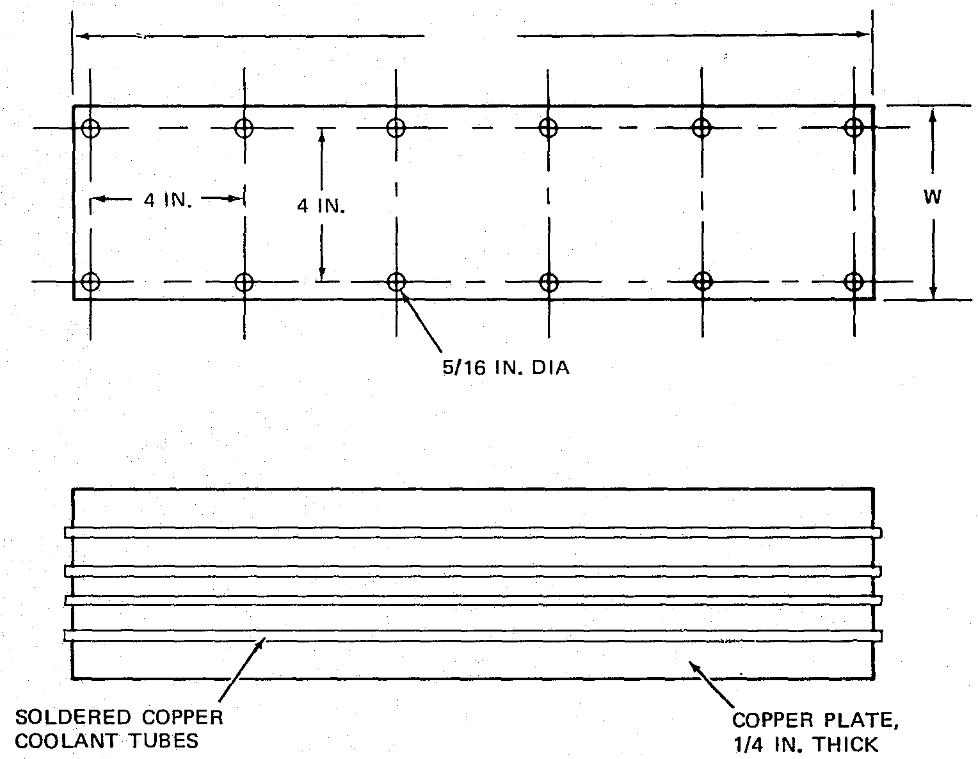


COLD RAIL, PANEL NO. 1



SUBLIMATOR HEAT SINK

DIMENSIONS

	L	W
SUBLIMINATOR	21 IN.	5.0 IN.
PANEL NO. 2 COLD RAIL	30 IN.	6.1 IN.

Figure 3-9. Simulated Heat Links

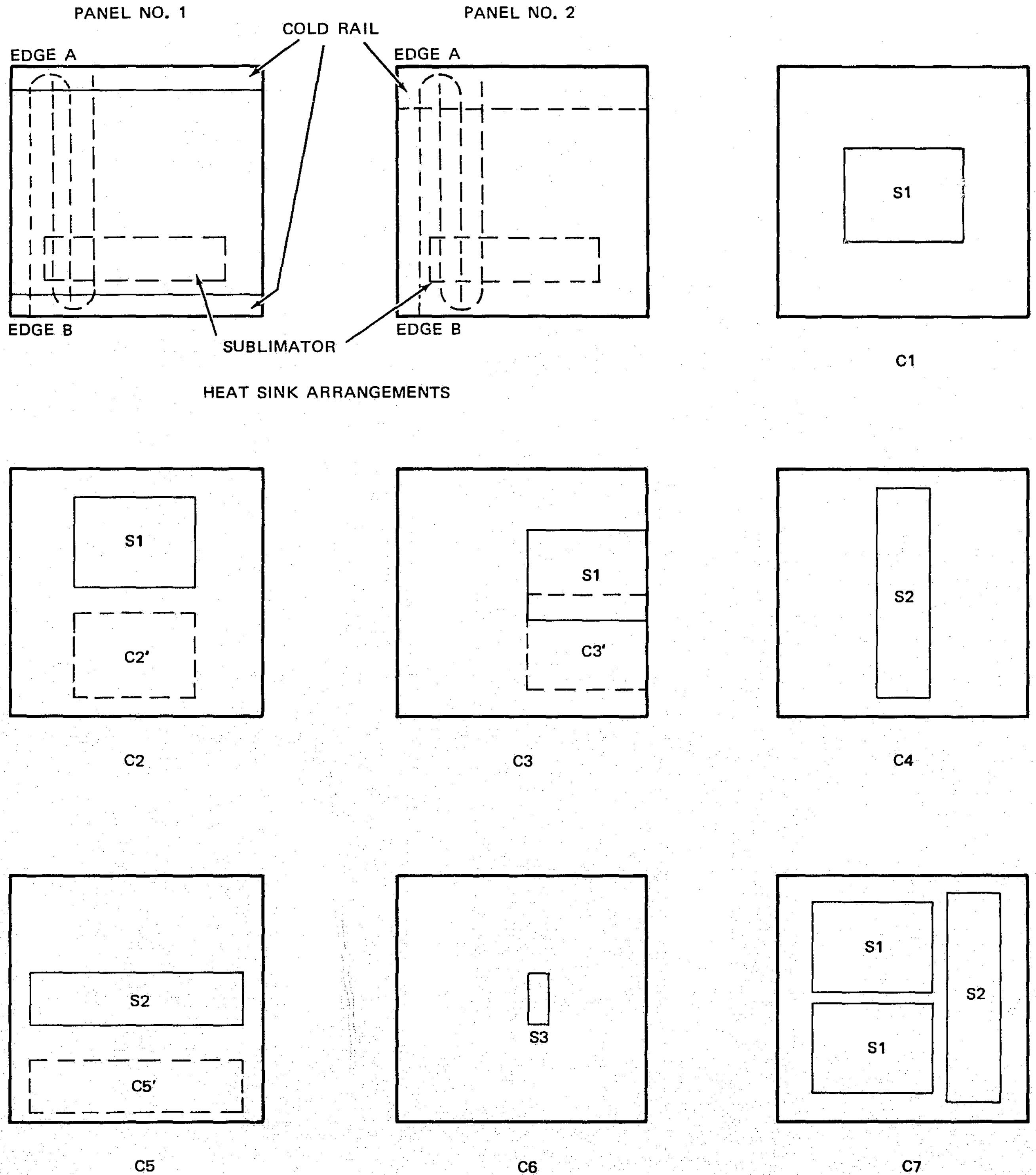


Figure 3-10. Heat Sink/Source Test Arrangements

Because only one rail was used on the second panel, some tests could be made more severe, that is, the heat source could be placed further from the heat sink. Test configurations (C2; C3; and C5') for the No. 2 panel only are also shown in Figure 2-10. For thermal testing, a minimum covering of 2 in. (0.051 m) of fiberglass insulation was used over all source, sink, and panel surfaces.

3.7.2 Data Reduction

Figures 3-11 through 3-17 present data on the two panels as a function of configuration, orientation, sink conditions, and sink temperature. Thermal gradients plotted as a function of heat input are mean gradients, that is, the average source panel surface temperature has been subtracted from the average panel temperature under the heat sink. The heat sinks had small holes with thermocouples embedded in contact with the panel at three positions over the length of the sink. No sink temperature corrections were necessary except for configuration C3, where two sink thermocouples monitored coolant inlet temperature and only the thermocouple directly in line with C3 responded. In that case, only the one sink thermocouple was considered indicative of surface temperature at a heat rejection zone.

The heat sources had only one thermocouple in contact with the plate. The remainder were on the heat source surface. It was empirically determined that at 2 w/in.^{2} (0.31 w/cm^2), there was a 1°F (0.56°K) average gradient between the source surface and panel surface. Average panel surface temperature was obtained by averaging source surface temperatures, minus the contact ΔT, with the one surface contact temperature. There was generally 5 to 10 thermocouples on the heat source.

The gradient across the heat source was taken as the maximum difference between heat source surface temperatures. Nominal values at 2 w/in.^{2} (0.31 w/cm^2) and 300 watts were from 3° to 5°F.

3.7.3 Thermal Test Results

The effect of configuration on performance is shown in Figures 3-11 and 3-12 for the first and second panels. Gradients associated with 150 sq in. (0.968 sq m) sources are grouped together for panel No. 2; the spot heat source C6 and extended heat source C7 are identified separately. The gradients associated with panel No. 1 are quite high, indicative of the contact resistance problems already discussed.

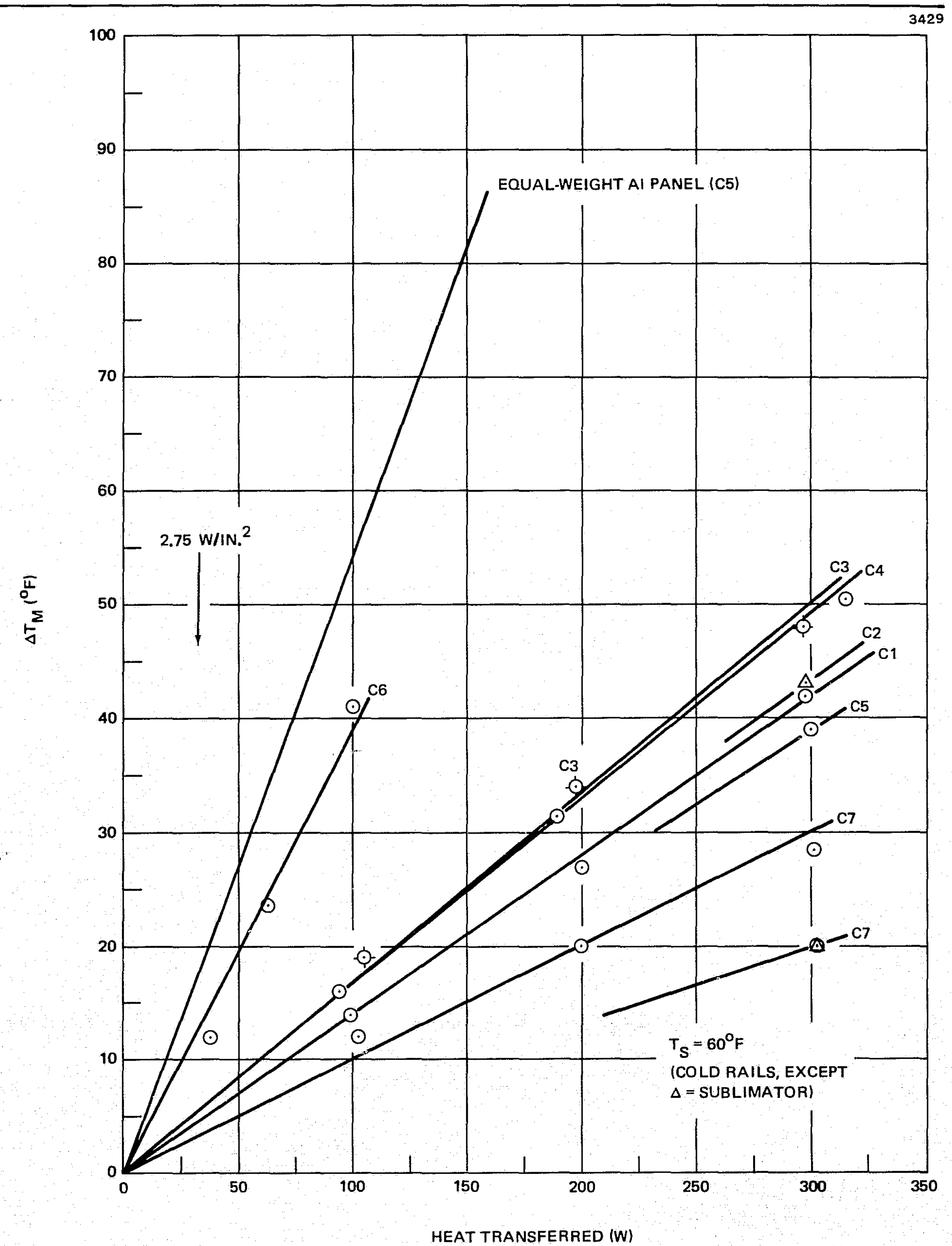


Figure 3-11. Effect of Configuration on Performance of Panel No. 1

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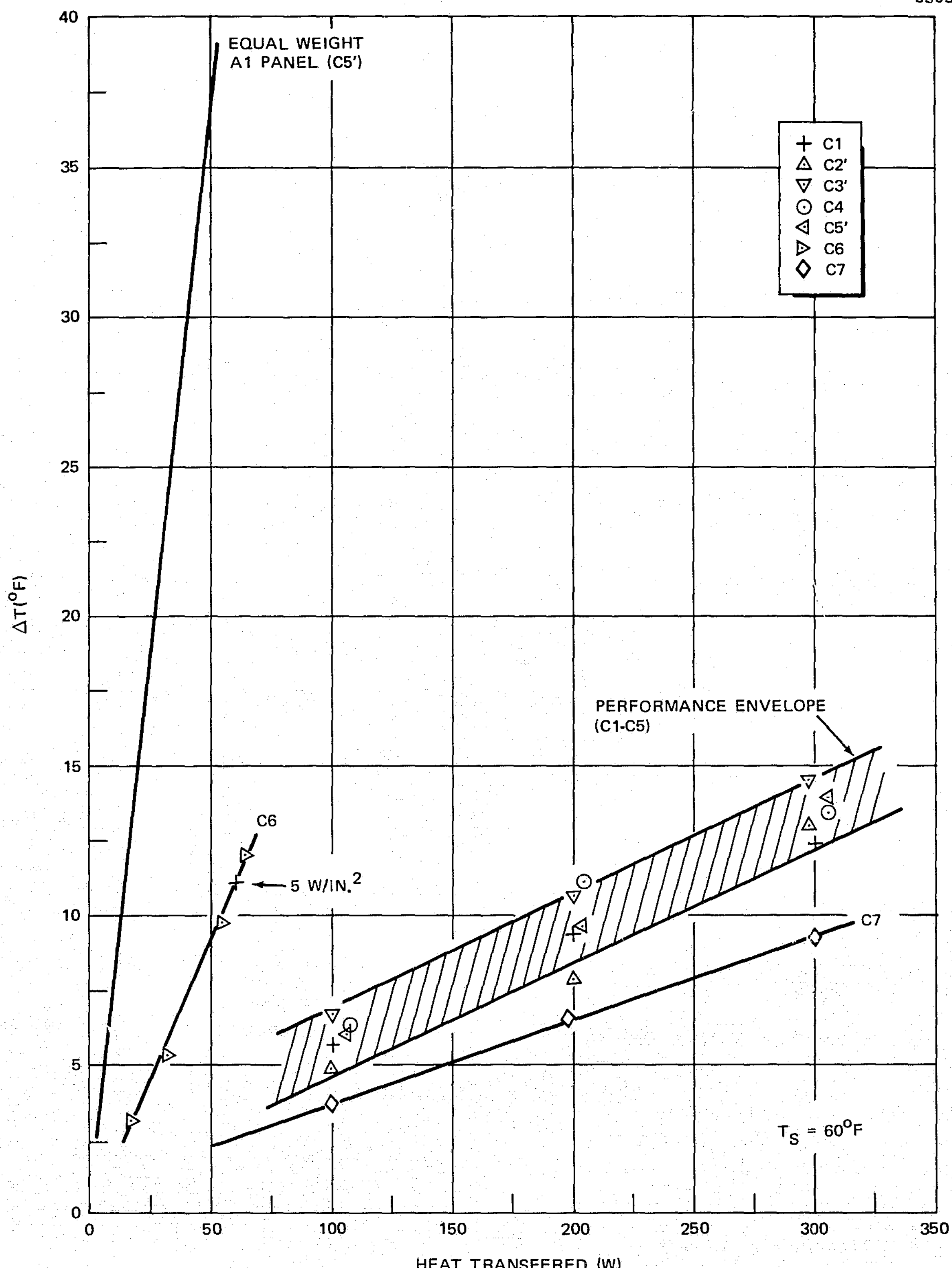

Figure 3-12. Effect of Configuration on Performance on Panel No. 2

Grouped sources for panel No. 2 have a mean ΔT of 13.5° ±1.25° F (7.50° ± 0.69°K) at 300 watts, for an effective thermal resistance of 0.045°F/w (0.025°K/w). The high flux density source (C6) specification was 2.75 w/in.^{2} (0.43 w/cm^2) and at 5 w/in.^{2} (0.78 w/cm^2) the original NASA guideline for the spot source, the gradient was still within the 15°F (8.33°K) specification at ΔT = 11.1°F (6.17°K). The source C7, which models a panel with a disturbed array of power-dissipation electronic modules, had a ΔT of 9.5 F (5.28°K) at 300 watts, for an effective thermal resistance of 0.032°F/w (0.018°K/w).

The thermal gradient for an equal-weight aluminum panel is also shown for configuration C5'. The aluminum plate has a thermal resistance 15 times higher than the heat pipe thermal conditioning panel.

The effect of panel orientation is shown in Figures 3-13 and 3-14. Because of the heat pipe orientation, the panel operates well in the vertical position when the heat pipes are essentially horizontal but operation suffers when the pipes are vertical and fluid drains to the bottom of the pipes. As is typical of a screen wick heat pipe, adverse tilting of the heat pipes one inch from horizontal produces little effect on performance.

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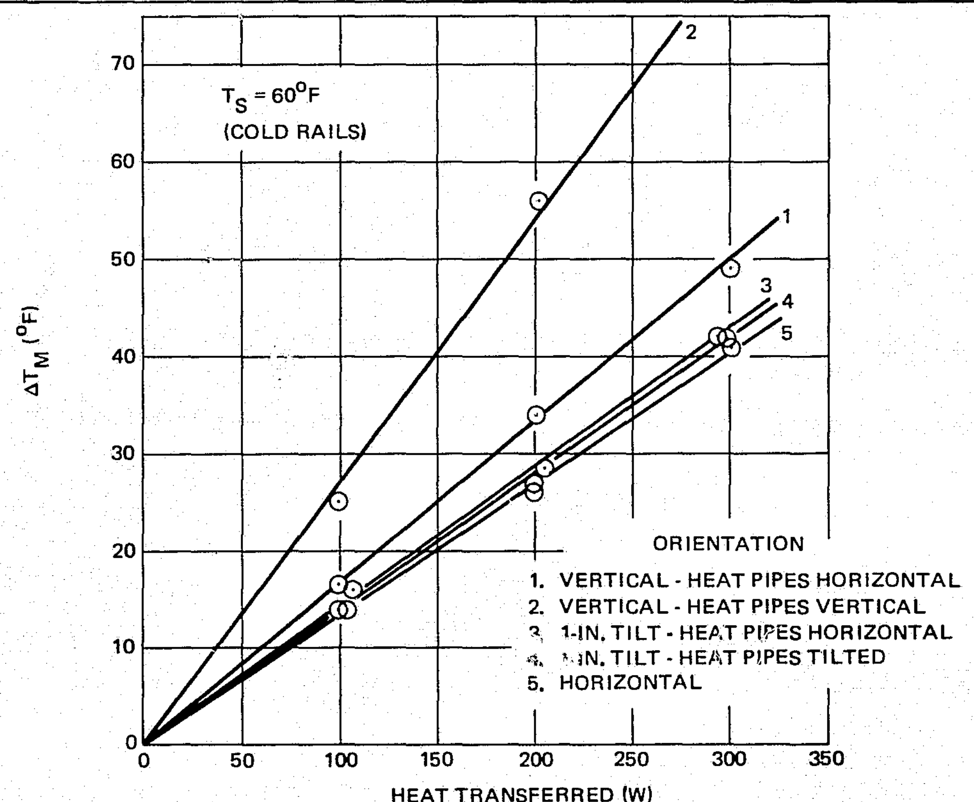

Figure 3-13. Effect of Panel Orientation on Panel No. 1 Performance (C1)

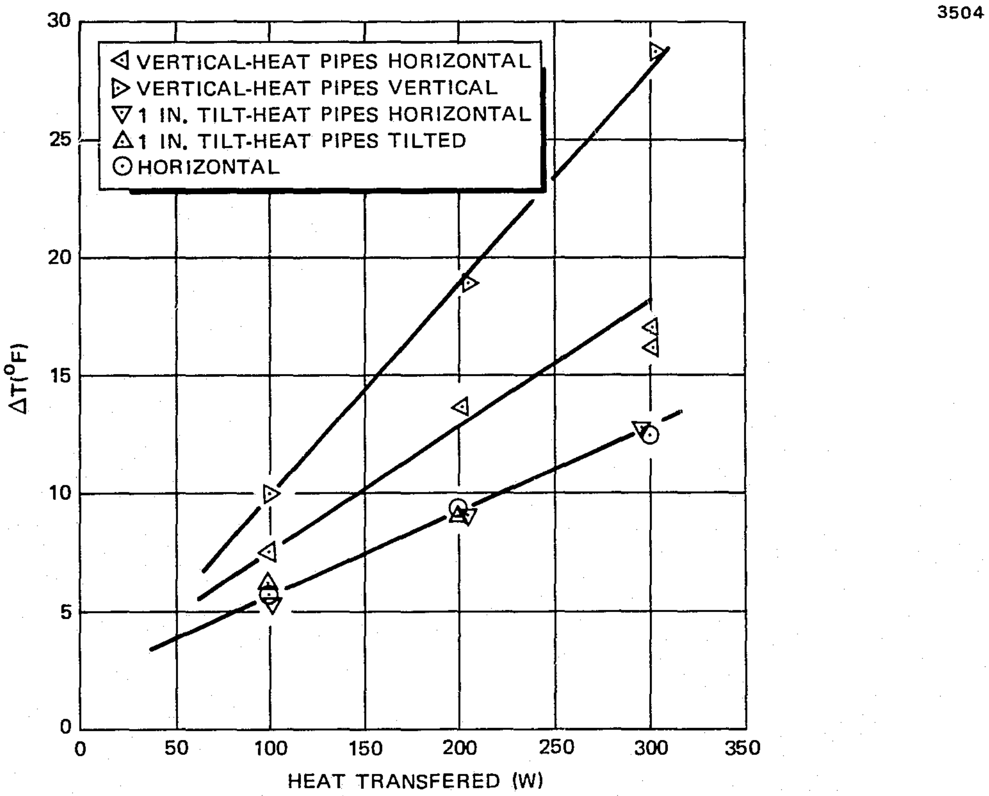

Figure 3-14. Effect of Panel Orientation on Panel No. 2 Performance (C1)

The effect of sink conditions on performance is shown in Figure 3-15. With source C2', the ΔT with one cold rail or one sublimator is from 13.5° to 14°F (7.50 to 7.78°K) at 300 watts. Use of the cold rail and sublimator decreases the ΔT to 10.3°F (5.72°K). Similarly, use of one cold rail with the C7 source produces a ΔT = 9.5°F, and two cold rails (one 6 in. wide and one of 5 in. wide (15.24 x 12.70 cm) on the opposite edge) decreases the ΔT to 6.3°F (3.50°K) at 300 watts.

This behavior is entirely consistent with calculations in Section 2.3 for the C2 configuration. The use of a sublimator and cold rail effectively doubles the heat rejection area or halves the condenser ΔT because the cold rail and sublimator are quite similar for the second panel. If this is factored into the values calculated and tabulated, the ΔT with two sinks is 10.2°F ($6.21°F + \frac{7.98}{2}$), or 5.67°K in good agreement with measurement. A similar argument can be made for the C7 configuration, which changed by 1/3 when the second rail was added.

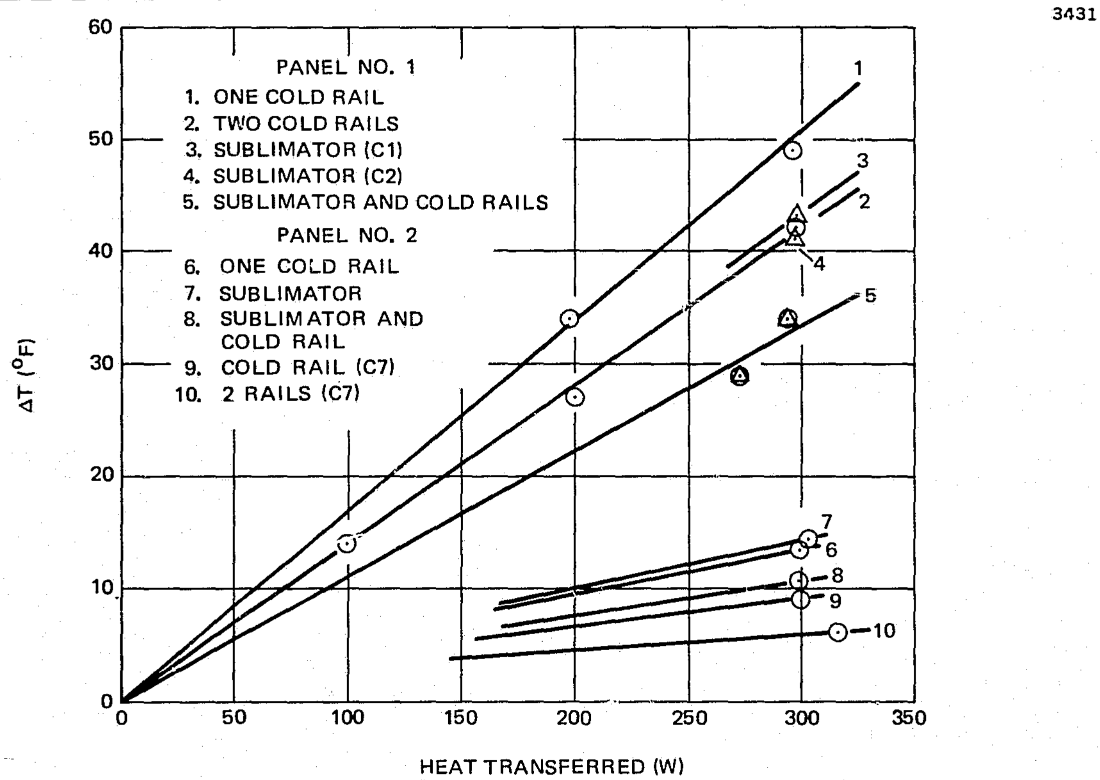

Figure 3-15. Effect of Sink Conditions on Panel Performance (C2 and C2')

The effect of sink or panel temperature is shown in Figure 3-16 over the design goal operating range. From 32° to 80°F (273° to 300°K), the ΔT decreases by about 1°F (0.56°K) for panel No. 2, and approximately by the same percentages (5% to 10%) for panel No. 1

Considerable excess transport capacity is designed into the panel. By exceeding the ΔT requirements, burnouts were attempted (Figure 3-17). Panel No. 1 burned out at 700 watts, or more than double the design goal. Panel No. 2 exceeded capacity of the coolant supply used at 900 watts, without burnout. A conservative estimate of burnout capacity is well over one kilowatt. Even at one kilowatt, if two cold rails 5 to 6 in. (0.127 to 0.152 m) wide were used, the ΔT is on the order of 20°F (17°K) for configuration C7. Table 3-6 summarizes general thermal properties of the thermal conditioning panel.

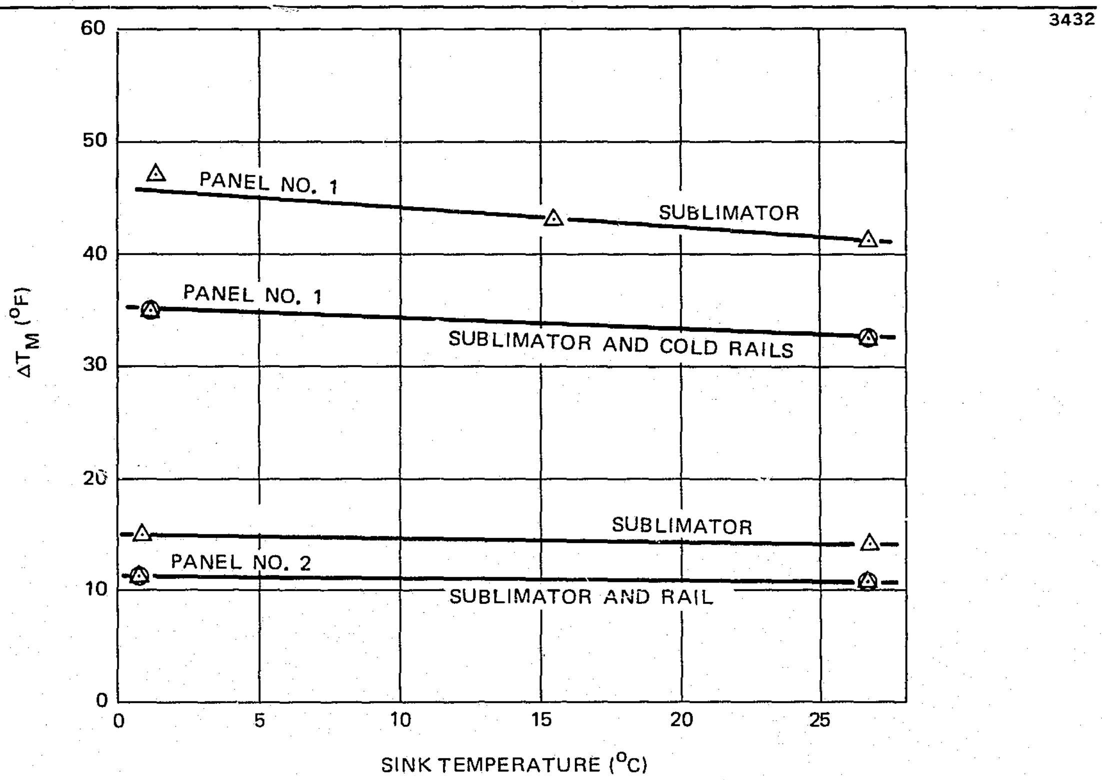

Figure 3-16. Effect of Sink Temperature on Panel Performance (C1 at 300 w)

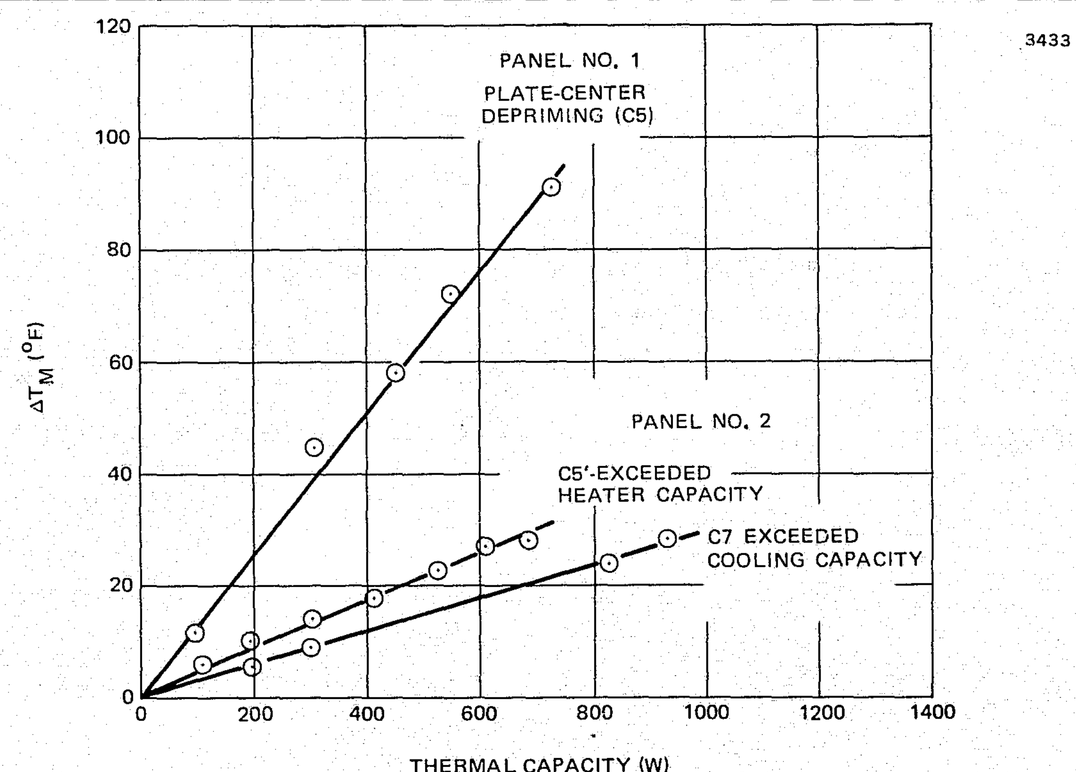

Figure 3-17. Panel Burnout Capacity

3.8 MECHANICAL CHARACTERISTICS

To identify a thermal conditioning panel load limitation, an experiment was performed which used a single 30 in. (0.76 m) long U-shaped heat pipe extrusion bonded between two 0.40 in. (0.102 cm) facesheets, 10 in. wide and 30 in. long (0.254 x 0.76 m) the edges of the U were 8 in. (0.203 m) apart. The panel section was set on parallel 1/2-in. (1.27 cm) diameter bars, 28 in. (0.71 m) apart, a line load was applied perpendicular to the panel top face half-way between the supports, and deflection of the panel face was measured as a function of load. At each load, the bond line was examined for failure at the points of maximum stress. No failure was observed with loads to 200 lb (908 kg) although centerline deflection was over 0.20 in. (0.508 cm). By allowing for the difference in flexural rigidity of this specimen and the thermal panel, the thermal panel will not fail with a static 800-lb (3632 kg) line load at the panel center with support available at the panel edges perpendicular to the heat pipe axes.

Panel strength was not measured with line support on opposite edges parallel to the heat pipes, and the absence of any continuous metal members in that orientation places more stress on the adhesive bonds. Therefore, for heavy component applications, it is recommended that the panel be mounted on sides A and B to take advantage of inherent strengths in the panel. In addition, if a rigid cross-member ran under the panel, limiting panel deflection to some small amount such as 0.020 in. (0.508 cm) or less, panel strength is significantly enhanced as many adhesive bond failures are directly attributable to excessive deflection. Mounting the panel rigidity along edges A and B also inhibits failure by limiting deflection under load. Table 3-7 summarizes mechanical properties of the thermal conditioning panels.

Table 3-5

PANEL COMPARISON

	Panel No. 1	Panel No. 2
Dimensions	30 x 30 x 0.625 in. (0.76 x 0.76 x 0.016 m)	30 x 30 x 0.583 (0.76 x 0.76 x 0.015 m)
Number of heat pipes	9	11
Extrusion	Figure 3-5	Figure 3-6
Total length of heat pipe	44.8 ft (13.66 m)	52.0 ft (15.85 m)
Center-to-center heat pipe spacing	2.75 in. (0.070 m)	1.66 in. (0.042 m)
Total extrusion weight	4.5 lb (20.43 kg)	7.8 lb (35.41 kg)
Face-plate thickness	0.062 in. (0.157 cm)	0.040 in. (0.102 cm)
Mounting inserts	62	64
Honeycomb	Yes	No
Bonding agent	AF 126-2	Deltabond 154
Nominal panel weight	18.3 lb (83.082 kg)	17.6 lb (79.904 kg)
Nominal gradients at 300 watts	40.0°F (22.22°K)	14.0°F (7.78°K)

Table 3-6

THERMAL PERFORMANCE CRITERIA

	Design goal	Panel No. 1	Panel No. 2
Maximum component heat load	300 w	700 w	900 w
Panel surface temperature gradient from source-to-sink at 2.0 w/in.^{2} and 300 w (0.31 w/cm^2)	15°F (8.33°K)	40°F (22.22°K)	10° to 15°F (5.55 to 8.33)
Maximum gradient between load	5°F (2.77°K)	5°F (2.77°K)	5°F (2.77°K)
Panel surface temperature gradient from source-to-sink at spot flux of 2.75 w/in.^{2} (0.43 w/cm^2	15°F (8.33°K)	12.5°F (6.94°K)	11.1 at 5 w/in.^{2} (6.17°K at 0.78 w/cm^2)
Mounting surface temperature	32° to 85°F (273° to 303°K)	0° to 120°F (255° to 322°K)	0° to 120°F (255° to 322°K)
Startup time to 90% of final ΔT at 200w input	N.S.	Not measured	15.0 min

*N.S. = Not Specified

Table 3-7

STRUCTURAL PERFORMANCE CRITERIA

	Design goal	Panel No. 1	Panel No. 2
Panel size	30 x 30 in. (0.76 x 0.76 m)	30 x 30 x 0.625 in. (0.76 x 0.76 x 0.016 m)	30 x 30 x 0.583 in. (0.76 x 0.76 x 0.015 m)
Bolt pattern	4 x 4 in. (0.10 x 0.10 m) centers	4 x 4 in. (0.10 x 0.10 m) centers	4 x 4 in. (0.10 x 0.10 m) centers
Fasteners	1/4-28 UNF-2B threads	1/4-28 UNF Helicoil in 0.75 in. dia x 0.5 in. (1.43 cm dia x 1.27 cm) spool	
Surface flatness			
Top	0.010 in. (0.025 cm) TIR	0.010 in. (0.025 cm) TIR	0.009 in. (0.02 cm) TIR (0.003 in. (0.008 cm) TIR avg)
Bottom	0.020 in. (0.050 cm) TIR	0.020 in. (0.050 cm) TIR	0.012 in. (0.030 cm) TIR
Component loading	100 lb (45.4 kg)	100 lb (45.4 kg)	100 lb (45.4 kg)
Static g-load	8 g	8 g	8 g
Panel weight	15 lb (6.81 kg)	18.3 lb (8.31 kg)	17.6 lb (7.99 kg)
Centerline deflection uniform load, supported at edges A and B only			
Simple supported	N.S.*	-	0.0106 in./100 lb (0.0269 cm/45.4 kg)
Fixed edges	N.S.*	-	0.0021 in./100 lb (0.0053 cm/45.4 kg)
Flexural rigidity (EI)	N.S.*	-	2.68 (10^6) lb-in.^{2} (7.84 (10^6) kg-cm^2)
Insert strength			
Tension	N.S.	-	2160 lb (548.64 kg)
Torque	N.S.	-	>90 ft-lb (122 j)

*N.S. = Not Specified

Section 4
CONCLUSIONS

The feasibility of a heat pipe thermal conditioning panel has been demonstrated conclusively. All thermal design goals as identified by future NASA space needs have been met or exceeded. Thermal gradients at rated power and thermal flux density are 10° to 15°F (5.54 to 8.33°K) for most configurations, and as low as 6.3°F (3.54°K) with a uniformly distributed load of 300 watts at 2/3 w/in.^{2} (0.1 w/cm^2). The ultimate thermal capacity of the heat pipe panel is estimated to exceed 1 kilowatt at a gradient of about 20°F (11.08°K). Panel capability exceeded the heat rejection capacity of laboratory coolant systems at 900 watts.

Mechanical strength of the panel is adequate to withstand over 8 g acceleration with a 100-lb (45.4 kg) uniform component load. Securing panel edges to a rigid frame enhances rigidity of the panel and improves ability to withstand acceleration.

Surface flatness of the final panel top face-plate is within 0.003 in. (0.0076 cm) average and 0.009 in. (0.023 cm) maximum. Surface flatness of the lower face is within 0.012 in. (0.031 cm) TIR. Both surfaces allow effective mounting of electronic components, experiments, and heat exchangers.

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Section 5
REFERENCES

1. G.D. Johnson, E.W. Saaski, Arterial Wick Heat Pipes, ASME Winter Meeting, New York, November, 1972, ASME 72 WA/HT-36.