SEPTEMBER 1973 MDC G4422

HEAT PIPE THERMAL CONDITIONING PANEL

EXECUTIVE SUMMARY REPORT

(NASA-CR-124453) HEAT PIPE THERMAL CONDITIONING PANEL Executive Summary Report, 28 Jun. 1972 - 12 Aug. 1972 (McDonnell-Douglas Astronautics Co.) 26 p HC $3.50 CSCL 20M G3/33

N73-33899

Unclas 15710

Prepared Under Contract No. NAS8-28639 by

Donald W. Douglas Laboratories

McDonnell Douglas Astronautics Company

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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by

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PREFACE

This document is the Executive Summary 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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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 operating 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 the 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, the heat pipe offers a high degree of isothermalization, high reliability because of redundant heat pipe network design, light weight, and passive operation.

Heat pipes isothermalize the panel and provide high lateral 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 thermal conditioning panel is lighter in weight because the conventional coolant loop is replaced by a lightweight aluminum extrusion filled principally with vapor.

The objective of this program was to develop and fabricate two working heat pipe thermal conditioning panels verify performance, and establish the design concept. The panels were designed and fabricated based on an analysis of several planned NASA space vehicles, in terms of panel size, capacity, temperature gradients, and integration with various heat exchangers and electronic components.

The practicability of a heat pipe thermal conditioning panel was conclusively shown. With the final heat pipe thermal conditioning panel, all program goals for thermal efficiency and heat transport capacity were met or exceeded.

Section 2
APPLICATIONS STUDY

To establish system constraints for panel design and defining the 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. 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.

The panel sizing requirements shown in Table 2-1 are based on equipment dimensions and a maximum power load of 300 watts per panel. The majority of thermal control requirements can be satisfied by a flat square panel configuration. One exception is the space station, which is currently using as base-line a book-like module concept. Consequently, the panel design evolving from this current study may not satisfy this application without modification.

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)

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Section 3
THERMAL-PANEL GENERAL SPECIFICATIONS

General specifications tentatively established by NASA for a heat pipe thermal conditioning panel have been evaluated based on the applications requirements summarized in Section 2; certain modifications to specifications were approved by NASA to better reflect flight requirements. Table 3-1 summarizes these specifications; modifications from originally specified values are thermal load maximum density, mounting surface temperature, and available sink temperature. The panel size that accommodates the majority of equipment components and satisfies the maximum heat load limit per panel of 300 w is 20 in. (0.51m) square; however, the 30-in. (0.76-m) square was maintained to accommodate the sublimator and to provide growth potential. Reviewing existing cooling requirements, the maximum thermal fluxes identified are about 2 w/in.^{2} (0.31 w/cm^2) on space tug. Relaxing the specification from 5 to 2 w/in.^{2} (0.78 to 0.31 w/cm^2) does not compromise the versatility of the system and adds some flexibility to the design options. The mounting surface temperature and available heat sink temperature upper limits were increased to 85°F (303°K) so as to be compatible with most of the vehicle coolant loops examined and still maintain equipment temperatures within reasonable limits.

Table 3-1

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

Section 4
DESIGN CONCEPTS

Several designs were considered before a choice was made on a configuration that embodied the most favorable compromise between low weight, cost, high thermal performance, and reliability.

4.1 PRELIMINARY CONCEPTS

Three conceptual approaches to meet the design specifications are shown in Figures 4-1, 4-2, and 4-3. The first (Figure 4-1) is a vapor chamber which, for unmanned locations, can be ammonia/aluminum; for manned areas, Freon/aluminum, water/copper, or possibly water/titanium are candidate systems. Vapor chamber designs result in maximum thermal performance but tend to be heavy, with difficulties in integrating fasteners, and unreliable because a single puncture or leak causes failure.

Figure 4-2 shows a design using interlinked U-shaped heat pipes surrounded by aluminum honeycomb. The honeycomb segments provide rigidity, at low weight penalty; honeycomb has an excellent strength-to-weight ratio. Front and back faceplates are aluminum sheet. Ammonia/aluminum can be used for unmanned areas and Freon/aluminum in manned areas. There is, however, some question as to whether water/copper or water/titanium can be readily integrated into an aluminum structure primarily because of varying thermal coefficients of expansion and the bond cure temperatures required. Thermal performance for this system is somewhat lower than a vapor chamber, but more than adequate for the applications considered. Redundancy of the pipes provides high reliability and a number of fastener designs and attachment techniques are possible.

A waffle pattern design is shown in Figure 4-3. This system meets specification requirements and allows maximum flexibility in fastener location, because holes can be drilled wherever the front and back faces are bonded together. Because the bonded area is larger than in Figure 4-2, the epoxy bond stresses are lower. However, location of the heat pipe pattern is much more restrictive, and component

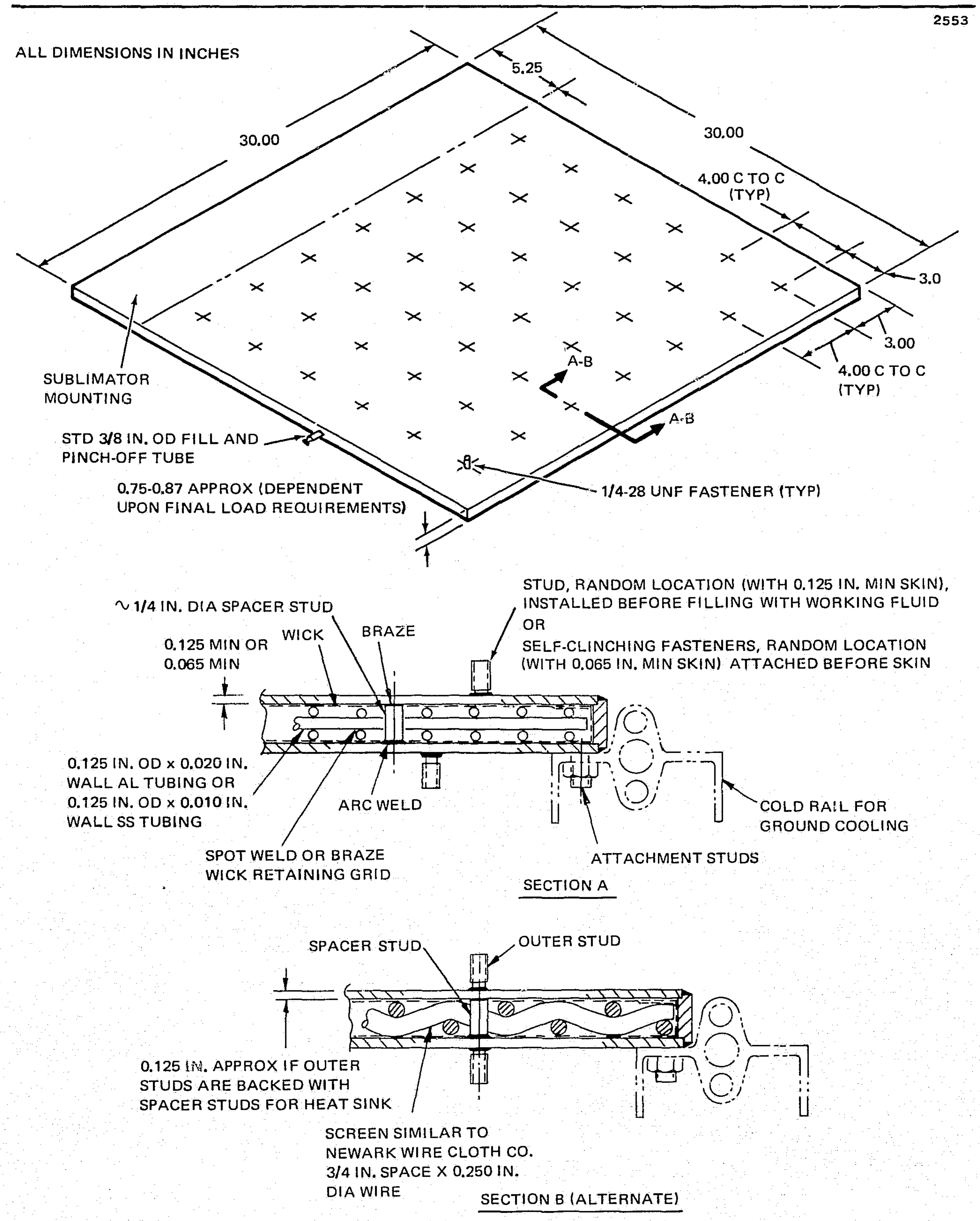

Figure 4-1. Vapor Chamber Thermal Conditioning Panel

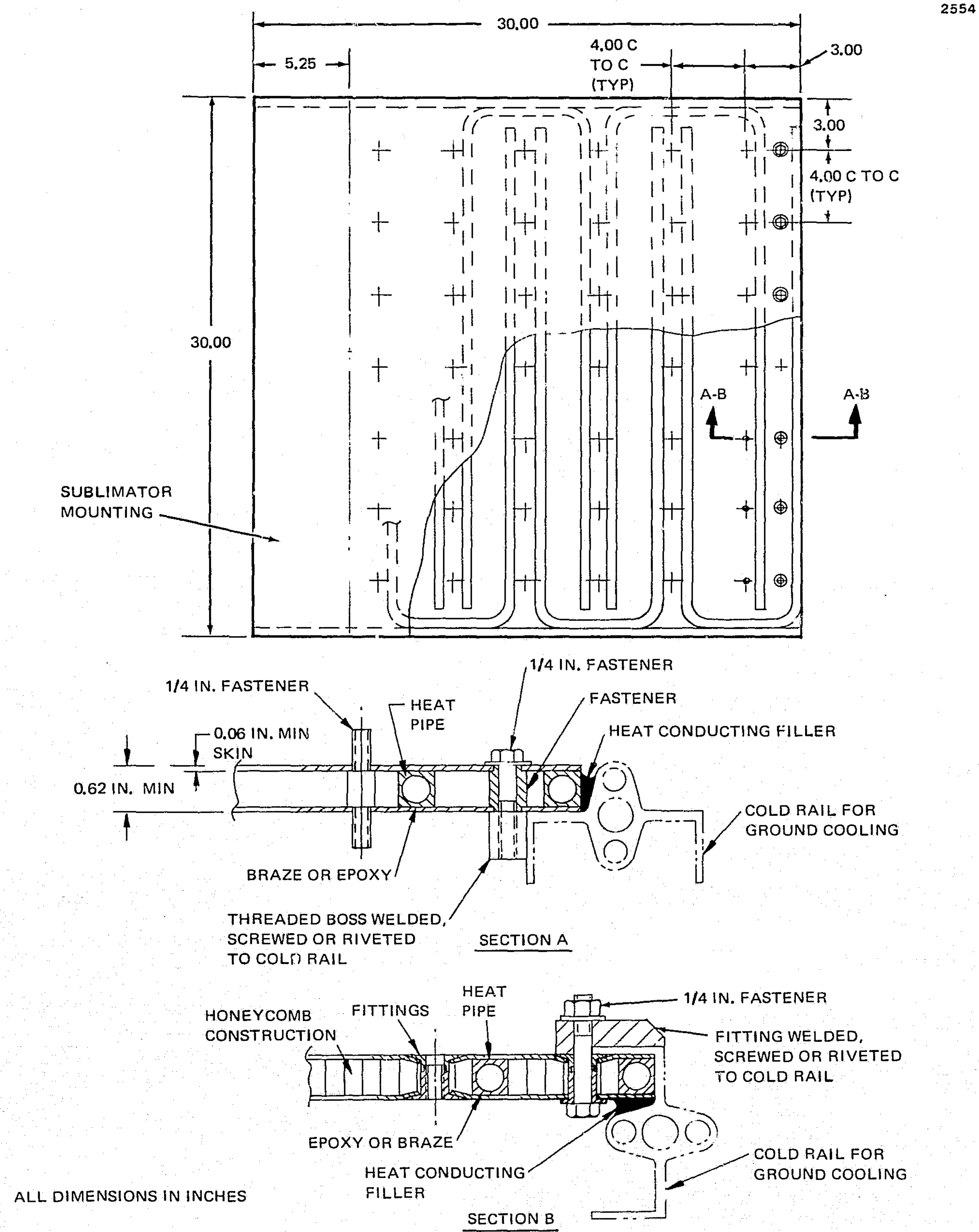

Figure 4-2. Heat Pipe Thermal Conditioning Panel

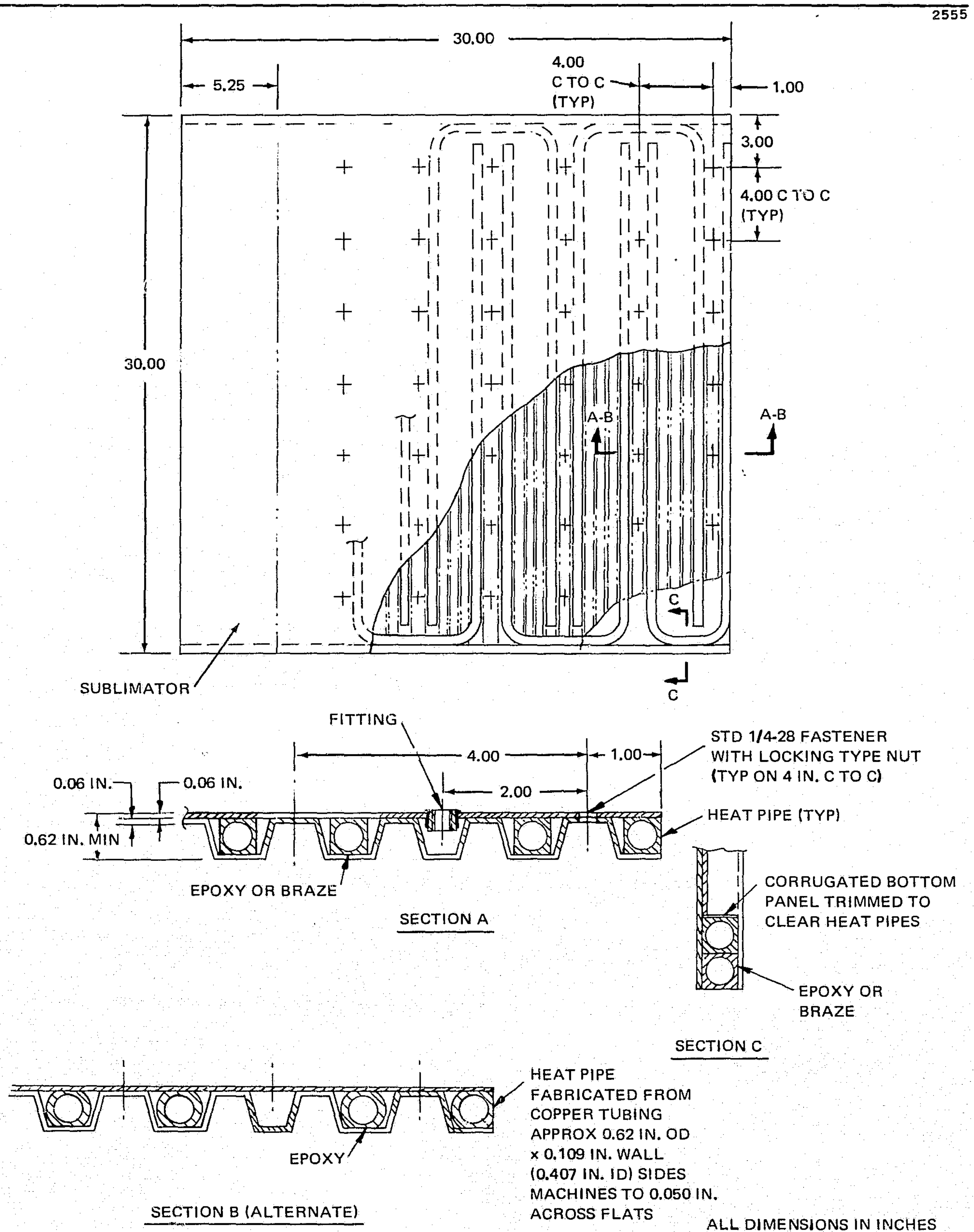

Figure 4-3. Heat Pipe Thermal Conditioning Panel

mounting on the underface is more difficult. A quantitative comparison of the three concepts is shown in Table 4-1.

Table 4-1
QUANTITATIVE COMPARISON OF CONCEPTS

Concept	Cost	Reliability	Weight	Thermal Performance
Vapor Chamber	2	3	3	1
Heat Pipe/Honeycomb	1	1	1	2
Heat Pipe/Waffle Structure	2	1	1	2

4.2 SELECTED DESIGN

Two heat pipe thermal conditioning panels were constructed as shown in Figure 4-2, in accordance with program requirements shown in Table 3-1. The working fluid selected was anhydrous ammonia; aluminum extrusion and 304 stainless steel screen wicking were used. The condensate return wicking is in a multiple-artery configuration which gives good redundancy and adequate heat transport capability (~2500 w-in.; ~6350 w-cm).

Actual engineering designs for the first and second panel are shown in Figures 4-4 and 4-5. Both incorporate interlinked U-shaped heat pipes. On any bolt line, if one heat pipe fails, the heat load will be transferred to the other heat pipe, preventing a thermal excursion. The panels are symmetrial and can be operated equally well with heat exchangers on either end A or B, or both, depending on thermal requirements. Figure 4-6 shows a hypothetical heat exchanger/mounting concept that allow complete use of the 30 x 30 in. (0.76 x 0.76 m) panel faces for component mounting.

Design differences between the first and second panel reflect changes which decrease thermal gradients between the heat sources and sinks. The number of heat pipes per unit width has been increased on the second panel, and the header heat pipes shown in Figure 4-4 have been removed. Heat transfer to the sinks is directly from the primary heat pipes. With closer spacing of heat pipes in the second panel, the honeycomb material has been removed without impairing rigidity, thereby considerably reducing fitup cost, lightening the panel slightly because the faceplates are reduced in thickness from 0.060 to 0.040 in. (0.15 to 0.40 cm) with no penalty in increased thermal gradients.

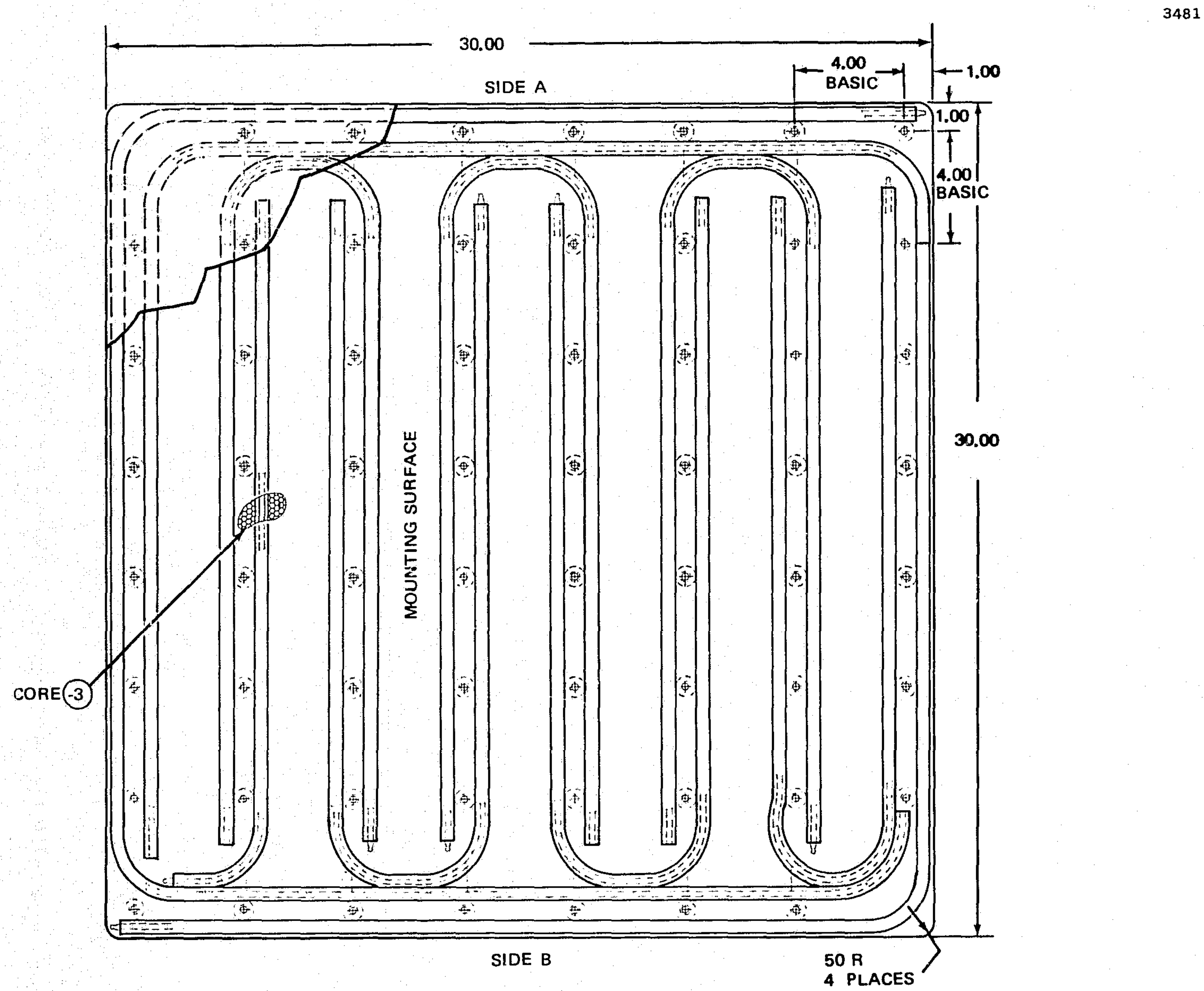

Figure 4-4. Thermal Conditioning Panel No. 1

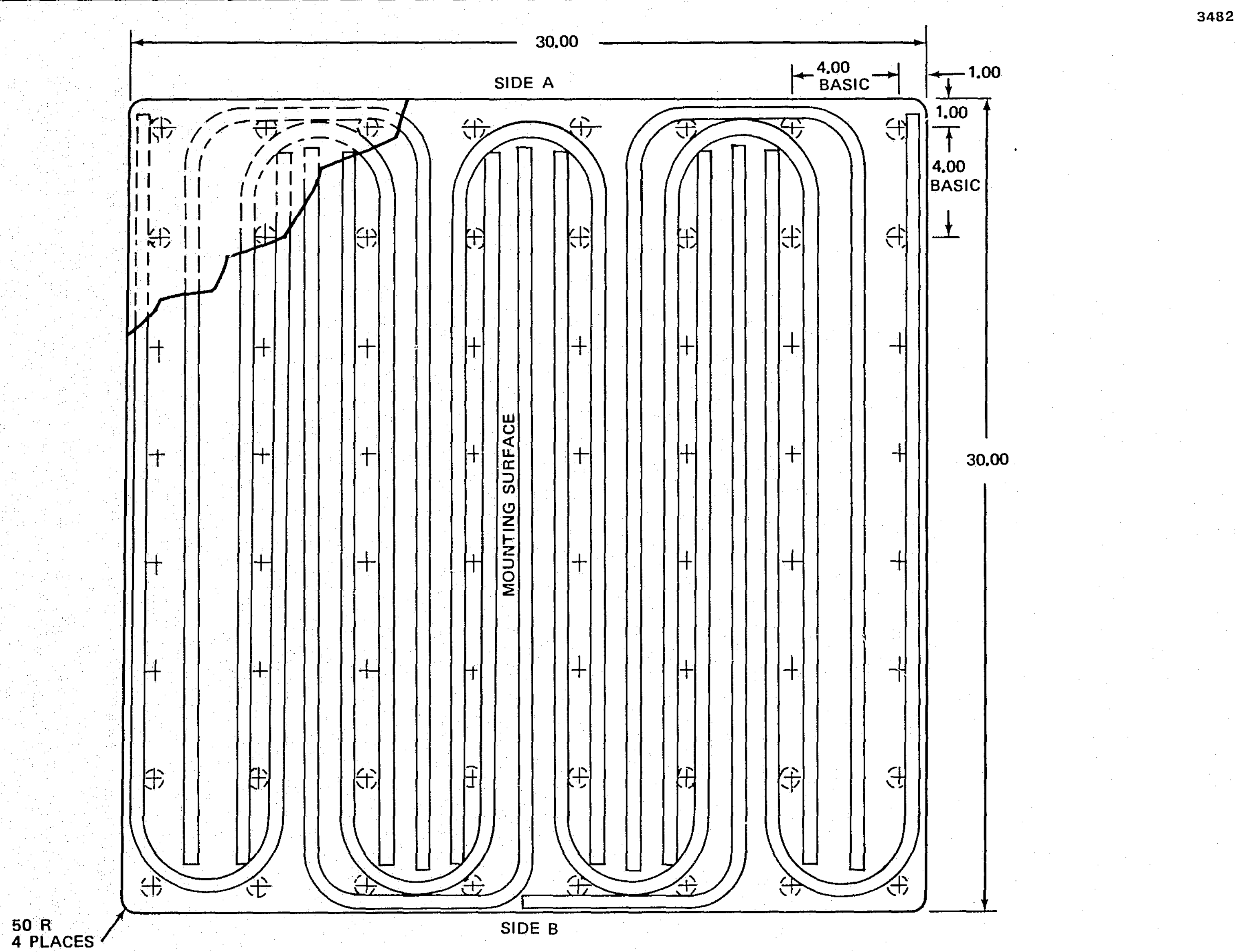

Figure 4-5. Thermal Conditioning Panel No. 2

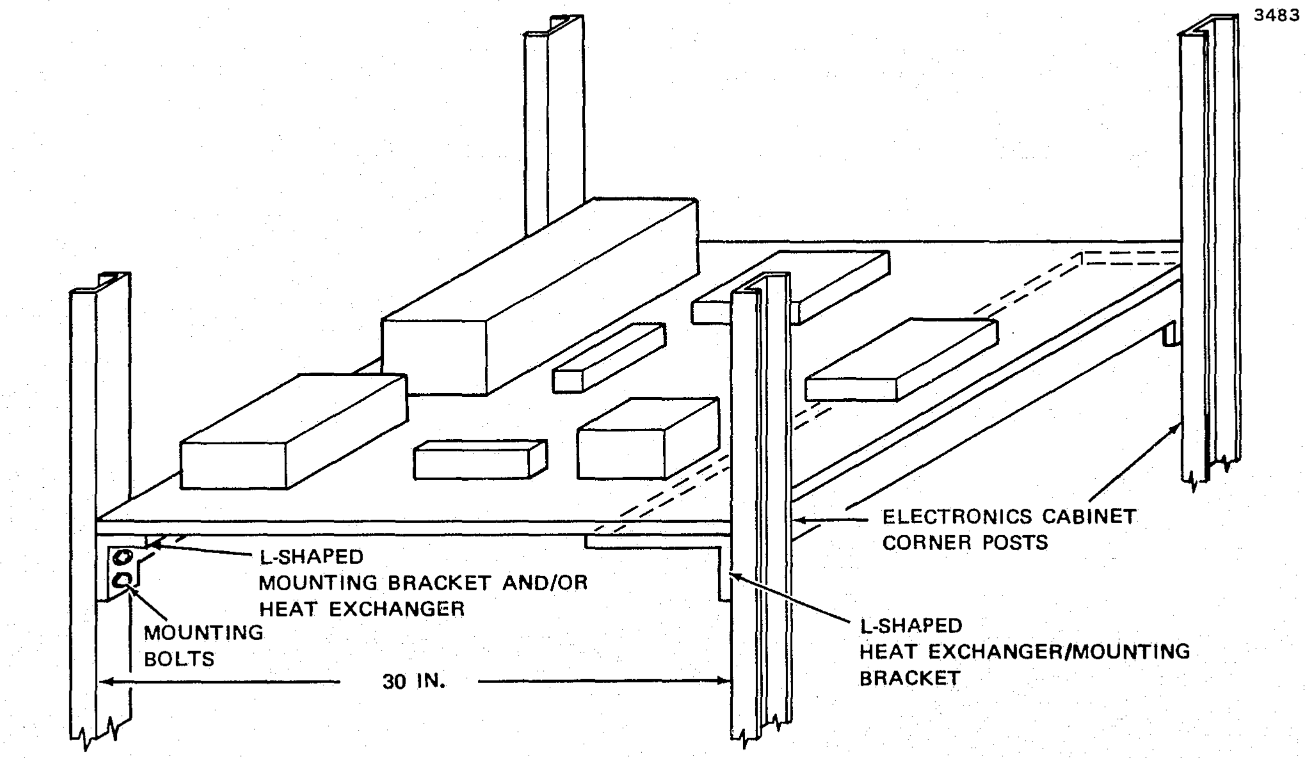

Figure 4-6. Heat Pipe Thermal Conditioning Panel Mounting Concept

Section 5
THERMAL CONDITIONING PANEL PERFORMANCE

Tables 5-1 and 5-2 present thermal and structural characteristics for each panel, including goals for critical design features. As shown, the first panel satisfied all thermal criteria with the exception of the thermal gradient for 300 watts input at 2 w/in.^{2} (0.31 w/cm^2). The second panel, utilizing data gained from construction of the prototype panel, satisfied or exceeded all thermal design goals.

In particular, the design goal of a 15°F temperature gradient from source to sink was achieved with the use of only one cold rail of the type shown in Figure 4-6. Two cold rails reduce the panel surface temperature gradient between source and sink to approximately 10°F (5.6°K). With a spot heat flux of 5 w/in.^{2} (0.78 w/cm^2) over two mounting positions, only an 11°F (6.1°K) gradient is measured. These high performance levels are in agreement with theoretical performance calculations (Reference 1). Using the long source designated S2 in Reference 1, the conductance of this panel is approximately 15 times the conductance of an aluminum panel of equal weight.

Mechanical characteristics of the panels are in agreement with design goals. The top component mounting surface is flat within 0.019 in. (0.025 cm) on each panel; and the second panel in addition has only a 0.003 in. (0.008 cm) average deviation from planarity on the top face. The bottom surface of both panels can serve as a mounting and heat transfer surface for the cold rail, leaving the upper surface available for component mounting.

Each panel weighs about 18 lb (8.2 kg) and withstands well over 8 g with a full 100-lb (45.4 kg) component load.

If the support edges A and B rest on pivots, the panel will deflect 0.0106 in. (0.03 cm) per 100 lb (45.4 kg) of load, while with fixed edges, i.e., with the panel mounted in a frame, the deflection is 0.0021 in./100 lb (0.0053 cm/45.4 kg).

Inserts used to mount equipment are 3/4-in. (1.9 cm) aluminum plugs drilled, tapped, and installed with 1/4-28 UNF Helicoil steel threads. In associated testing, it was found that 2160 lb (548.6 kg) of tension was necessary to break the plug-faceplate epoxy bond; > 90 ft-lb (122 j) of torque was required to break the bond by twisting. Neither of these mechanical stresses are expected during the use of these panels.

Table 5-1

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 5-2

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

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Section 6
CONCLUSIONS

All lightweight, strong, and effective heat pipe thermal conditioning panel has been developed and fabricated. Thermal capacity is equal to or in excess of requirements for five planned NASA space vehicles: Shuttle Orbiter, RAM, SOAR, Space Tug, and Space Station. During performance verification using three different heat source shapes in 5 different configurations, at a heat flux of 2 w/in.^{2} (0.31 w/cm^2) and 300 net watts input, thermal gradients of 10° to 15°F (5.55° to 8.33°K) between source and sink were typical, and 5°F (5.55°K) between points on the heat source itself. The heat pipe approach to thermal conditioning panel design has been verified as a viable and attractive alternative to forced fluid flow cooling. Advantages over forced flow cooling include a high degree of isothermalization, high reliability because of redundant heat pipe network design, light weight, and passive operation. In relation to a solid metal plate, the final panel has an equivalent conductance 15 times that of an equal-weight aluminum sheet.

The panel is adaptable to a number of different sink arrangements, and highly planar surfaces on top and bottom allow maximum flexibility in component mounting. With standard mounting techniques, the panel will withstand more than 8 g of acceleration with a 100-lb (45.4 kg) component load.

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Section 7
REFERENCES

1. E. W. Saaski. Detailed Technical Report - Thermal Conditioning Panel, Contract No. NAS8-28639 (CPFF). McDonnell Douglas Report MDC G4421, September 1973.