DESIGN AND FABRICATION OF A RADIATIVE ACTIVELY COOLED HONEYCOMB SANDWICH PANEL

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INTRODUCTION

Design of structures to operate efficiently for long periods in the severe thermal environment encountered by hypersonic cruise aircraft requires careful selection of materials and structural concepts. In Reference 1 an actively cooled aluminum panel which absorbs all of the incident heat load was designed for hypersonic aircraft application. Hydrogen fuel was used as the heat sink to cool the structure to relatively low temperatures so that long life could be achieved. However, since cooling of the engines and the inlets requires a high percentage of the available heat sink it was doubtful that the remaining heat sink would be sufficient for airframe cooling. A solution to this problem is a radiative actively cooled panel which uses heat shields and insulation on the outer surface of the actively cooled structural panel. With this system, the outer surface operates at high temperatures which radiates an appreciable amount of the incident heat load back to the atmosphere and reduces the heat load that must be absorbed by the fuel. The program described here uses the honeycomb-sandwich actively cooled panel concept from Reference 1 with coolant passages in contact with the outer skin. However, the panel was optimized to be compatible with a radiative thermal protection system and the heat sink available for a representative hypersonic vehicle.

A primary purpose of the program was to compare the mass of a radiative actively cooled panel to the mass of a bare actively cooled panel designed to the same conditions and constraints. The approach was to design and optimize a 0.61 x 6.1-m (2 x 20 ft) full-scale panel which combines radiative and active cooling to control structural temperatures to levels compatible with use of lightweight materials and to fabricate a 0.61 x 1.22-m (2 x 4 ft) panel for performance testing by NASA.

Results of the design and optimization of the full-scale radiative actively cooled structural panel, including radiative concept selection, final configuration details, test panel description, and conclusions of the study are summarized herein.

RADIATIVE ACTIVELY COOLED PANEL (RACP) PROGRAM
(Figure 1)

Design of efficient structures for long time operation in the severe thermal environment experienced by hypersonic cruise aircraft is a difficult problem. Actively cooled structural panels have been proposed as one approach with high potential for structural mass reduction and cost savings. This concept uses a coolant circulating in a closed loop through the structure and then through a heat exchanger where the absorbed heat is transferred to hydrogen fuel enroute to the engines.

The problem is that cooling of the engines and inlets requires a high percentage of the available heat sink and there is not enough left to cool the structure. A solution to this problem is to use insulation and heat shields on the actively cooled panel (ACP) external surfaces to radiate most of the heat load back to the atmosphere.

The objectives of the program described herein were to demonstrate feasibility of the radiative actively cooled system by designing and fabricating a hypersonic aircraft panel and to compare the mass of this system with that of a base panel designed for the same conditions.

RADIATIVE ACTIVELY COOLED PANEL (RACP) PROGRAM

PROBLEM:

ACTIVE COOLING WOULD BE VIABLE EXCEPT HEAT LOAD $>>$ HEAT SINK

SOLUTION:

REDUCE HEAT LOAD VIA RADIATION

APPROACH:

HEAT SHIELD AND INSULATION ON ACTIVELY COOLED PANEL SURFACE

OBJECTIVE:

DEMONSTRATE RACP FEASIBILITY AND COMPARE WITH BARE ACP

Figure 1

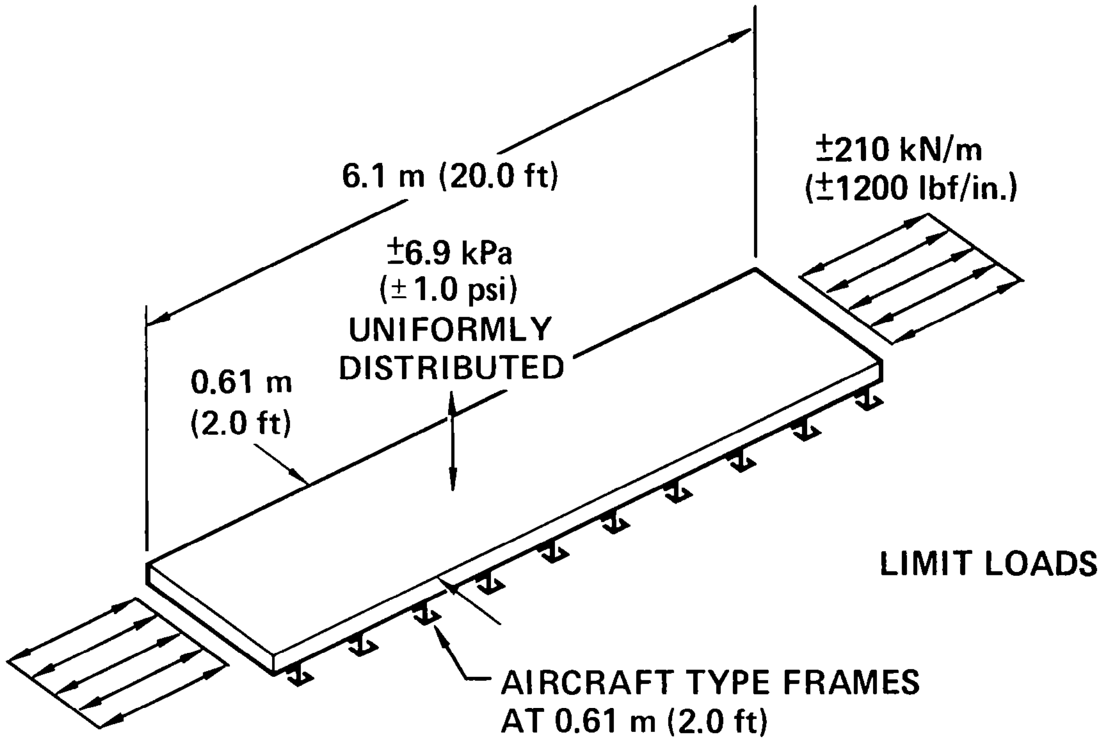
PANEL DESIGN CONDITIONS
(Figure 2)

This figure shows design conditions of greatest significance to the .61-m (2.0 ft) wide and 6.1-m (20.0 ft) long radiative actively cooled panel that was developed in this program. The design limit in-plane running loads and normal pressures are about the same magnitude as those experienced in the highly loaded areas of a hypersonic transport aircraft fuselage. The structure was designed to sustain ultimate load (1.5 times design limit load) combined with thermal loads without failure. In addition, the structure was designed to sustain, without failure or coolant leakage, 20 000 cycles of fully reversed design limit loads combined with thermal loads.

The design incident heat flux was equivalent to 136 kW/m^2 (12 $Btu/ft^2 sec$) to a 422 K ($300^{\circ}F$) surface temperature.

To assure realism, the panel edges and attachments to support frames were designed as in an airplane.

PANEL DESIGN CONDITIONS



DESIGN LIFE - 20000 CYCLES
REFERENCE HEAT FLUX - 136 kW/m^2 (12 Btu/ft^2 sec) AT
422K (300°F) SURFACE TEMPERATURE

Figure 2

DESIGN AND OPTIMIZATION APPROACH
(Figure 3)

This figure outlines the approach to design and optimization of the radiative actively cooled panel (RACP). Some of these activities were ongoing at the same time because several iterations, involving various engineering disciplines, were required to get the minimum weight solution. Each of these activities will be discussed in the presentation that follows.

Selection of actively cooled panel concept and structural materials was based on the results of the Reference 1 bare actively cooled panel program and the desire, in this program, to make a direct comparison between radiative and bare actively cooled panels.

A 60/40 mass solution of ethylene glycol/water was selected as the RACP coolant to assure compatibility with the NASA Langley Research Center test facility which was to be used for RACP panel testing. Reference 1 studies indicated lower system mass using a 60/40 mass solution of methanol/water coolant.

Trade studies and analyses were planned to select insulation materials and heat-shield concepts.

To determine minimum system mass, the minimum mass of each RACP component was determined as a function of cooling-system absorbed heat flux. By summing these masses vs absorbed heat flux, the minimum total mass and corresponding absorbed heat flux was determined and the details of the component parts were known.

Concurrent with the above activities, the joints and edges of all components were being designed and analyzed to minimize their mass.

DESIGN AND OPTIMIZATION APPROACH

- SELECT PANEL CONCEPT AND MATERIALS
- SELECT COOLANT AND INSULATION MATERIALS
- SELECT HEAT-SHIELD CONCEPT
- DETERMINE MINIMUM MASS vs ABSORBED HEAT FLUX
- DESIGN JOINTS AND EDGES FOR MINIMUM MASS

Figure 3

RADIATIVE ACTIVELY COOLED PANEL CONCEPT
(Figure 4)

This figure illustrates the radiative actively cooled panel concept as it was envisioned at the beginning of the program. It consisted of an actively cooled honeycomb-sandwich structure, an external heat shield (the concept was not yet decided on), and a layer of insulation.

The actively cooled panel concept is an all-aluminum, adhesive bonded, honeycomb sandwich with D-shaped coolant tubes (Dee tubes) nested in the honeycomb core against the outer skin. The coolant tubes are attached to manifolds which are located at the ends of the panel. These manifolds also incorporate provisions for attachment to adjacent panels. To provide adequate cooling in the area of the manifold, a split coolant passage was provided, as indicated in the figure. The coolant flows out to the manifold edges and then to the individual tubes. At the exit manifold the process is reversed.

Interface conductance of the adhesive joint between the panel outer skin and cooling passages was our primary concern. In the Reference 1 program, it was found that a bare panel subjected to 136 kW/m^2 (12 Btu/ft^2sec) needed high interface conductance, such as provided by solder, to avoid excessive temperature gradients and large mass penalties. With a much reduced heat flux resulting from addition of the heat shield and insulation, this panel performed well using the adhesive joint which has interface conductance of 1.65 kW/m^2K (290 $Btu/hr\ ft^2{}^\circ F$).

As indicated in section B-B, the insulation, heat shield and actively cooled panel are snugged up against each other. The primary reason for this is to minimize flow of boundary-layer air under the heat shield and insulation. Influx of boundary-layer air, if not planned for, could overheat the panel and overload the cooling system.

Several insulations were evaluated for this application. A fibrous, flexible, blanket-type insulation with a density of 256 kg/m^3 (16 lbm/ft^3) was selected because it was very efficient, flexible, and easily shaped to fuselage contour.

It will be noted that the heat shield is shown as a flat sheet in section B-B. Initially, the choice of heat shield concept was left open. Heat shield concept selection and optimization will be described later.

RADIATIVE ACTIVELY COOLED PANEL CONCEPT

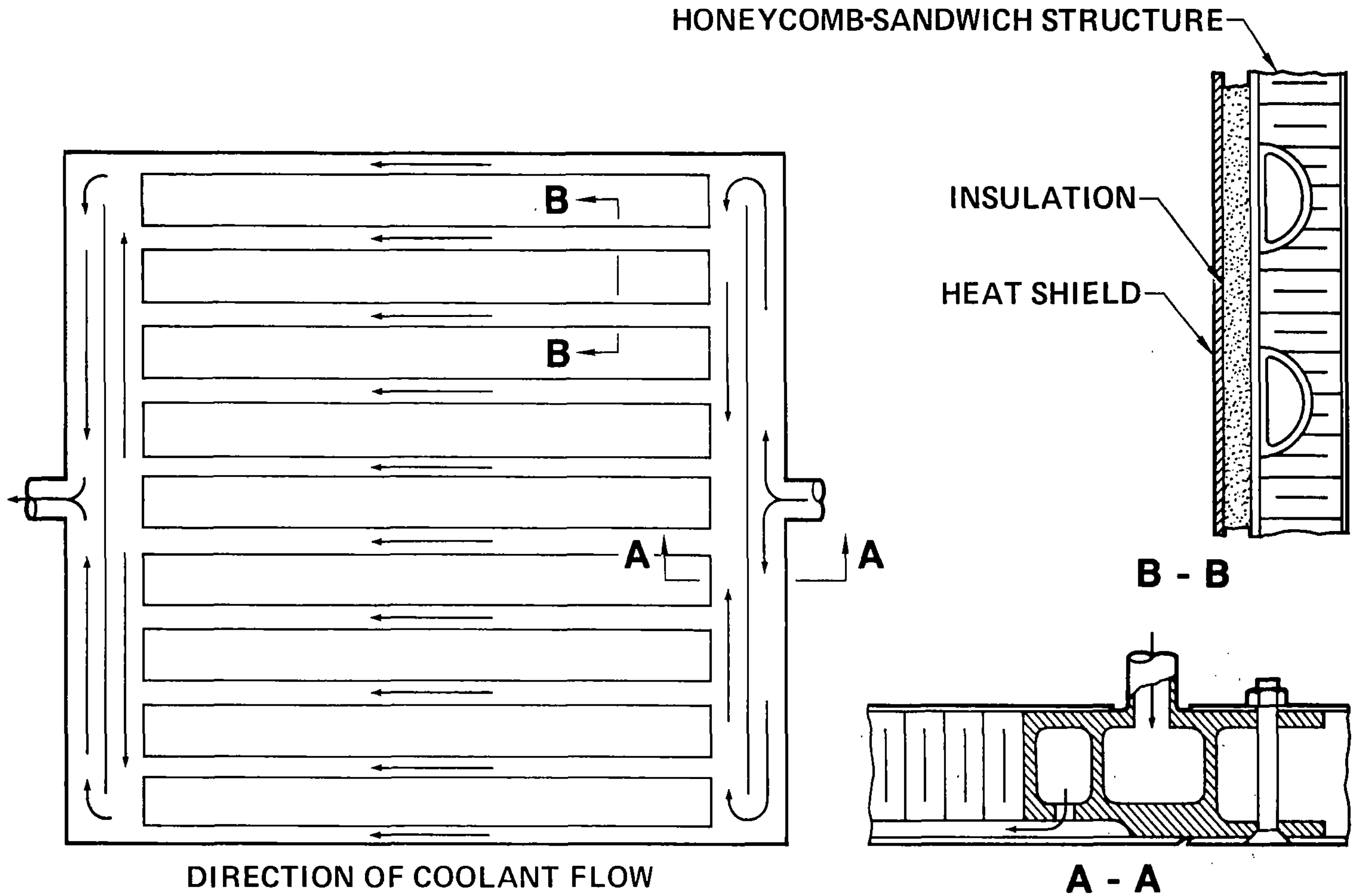


Figure 4

COOLING-SYSTEM AND INSULATION MASS
VS ABSORBED HEAT FLUX
(Figure 5)

This figure shows how active-cooling-system (ACS) absorbed heat flux affects active-cooling-system and insulation mass. Also shown is the variation of heat-shield temperature as a function of absorbed heat flux. With the heat-shield temperatures indicated here, it is evident that superalloys are required.

As the mass of the insulation is reduced, less of the aerodynamic heat load is radiated to space and more is absorbed by the active cooling system. As shown, the active-cooling-system mass, which includes all elements external to the panel as well as propellant for the pumps, is highly sensitive to increased heat flux. The lowest combined mass of active cooling system and insulation is obtained at an absorbed heat flux of 9.1 kW/m^2 (0.8 Btu/ft^2sec). The RACP system was optimized using a 60/40 mass solution of ethylene glycol and water as the coolant with an inlet temperature of 283 K (50^oF).

Of course, the effects of coolant temperatures and pressures and the variation of actively cooled panel and heat-shield mass with absorbed heat flux had to be determined before total system optimization was complete.

COOLING-SYSTEM AND INSULATION MASS vs ABSORBED HEAT FLUX

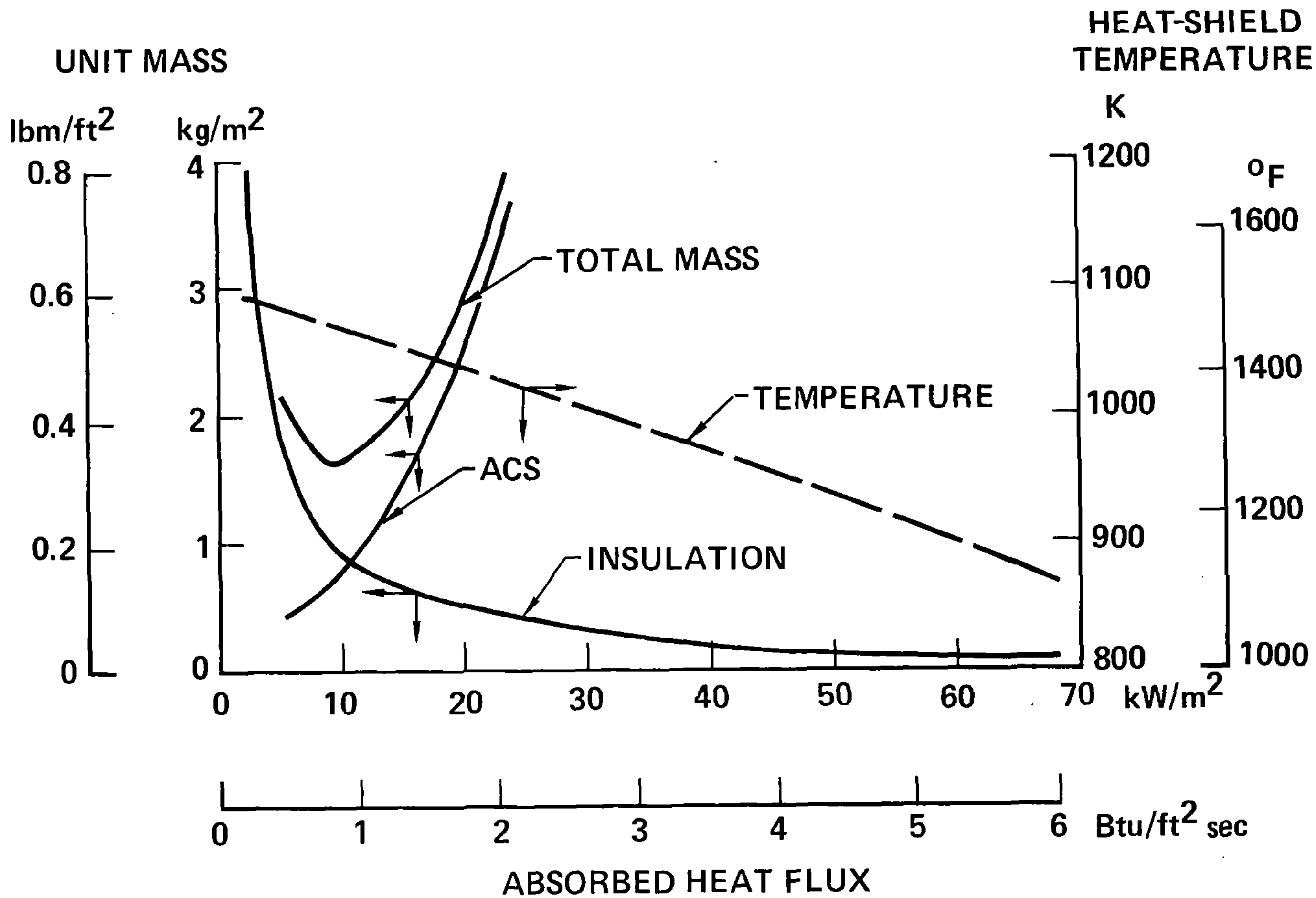


Figure 5

INCREASING PRESSURE REDUCES COOLING-SYSTEM MASS
(Figure 6)

As shown in the figure, a significant reduction (25%) in active cooling system mass was achieved by increasing the system pressure from 0.7 MPa (100 lbf/in^2) to 1.4 MPa (210 lbf/in^2). The figure also shows mass of individual components and identifies the reduction in residual coolant mass (as a result of smaller line sizes) as the driving element. At pressures above the design value of 1.4 MPa (210 lbf/in^2), no additional benefits are realized as the reduction in residual coolant mass is negated by a comparable increase in the mass of the pumps and auxiliary power system (APS) propellant, due to the increase in system pressure drop.

The data presented on this figure are for a 60/40 mass solution ethylene glycol/water coolant with inlet/outlet temperatures of 283 K/322 K (50^{o}F/120^{o}F). Varying inlet/outlet temperatures within practical limits, did not alter the conclusions presented here. Cooling-system masses were calculated using equations developed in this program and by other investigators in previous programs. A complete discussion of these equations is given in the program final report, Reference 2.

INCREASING PRESSURE REDUCES COOLING-SYSTEM MASS

60/40 MASS SOLUTION OF ETHYLENE GLYCOL AND WATER
COOLANT INLET/OUTLET TEMPERATURE = 283 K/322 K (50°F/120°F)

$P_s = \Delta P_{lines} + 414$ kPa (60 lbf/in.^{2})

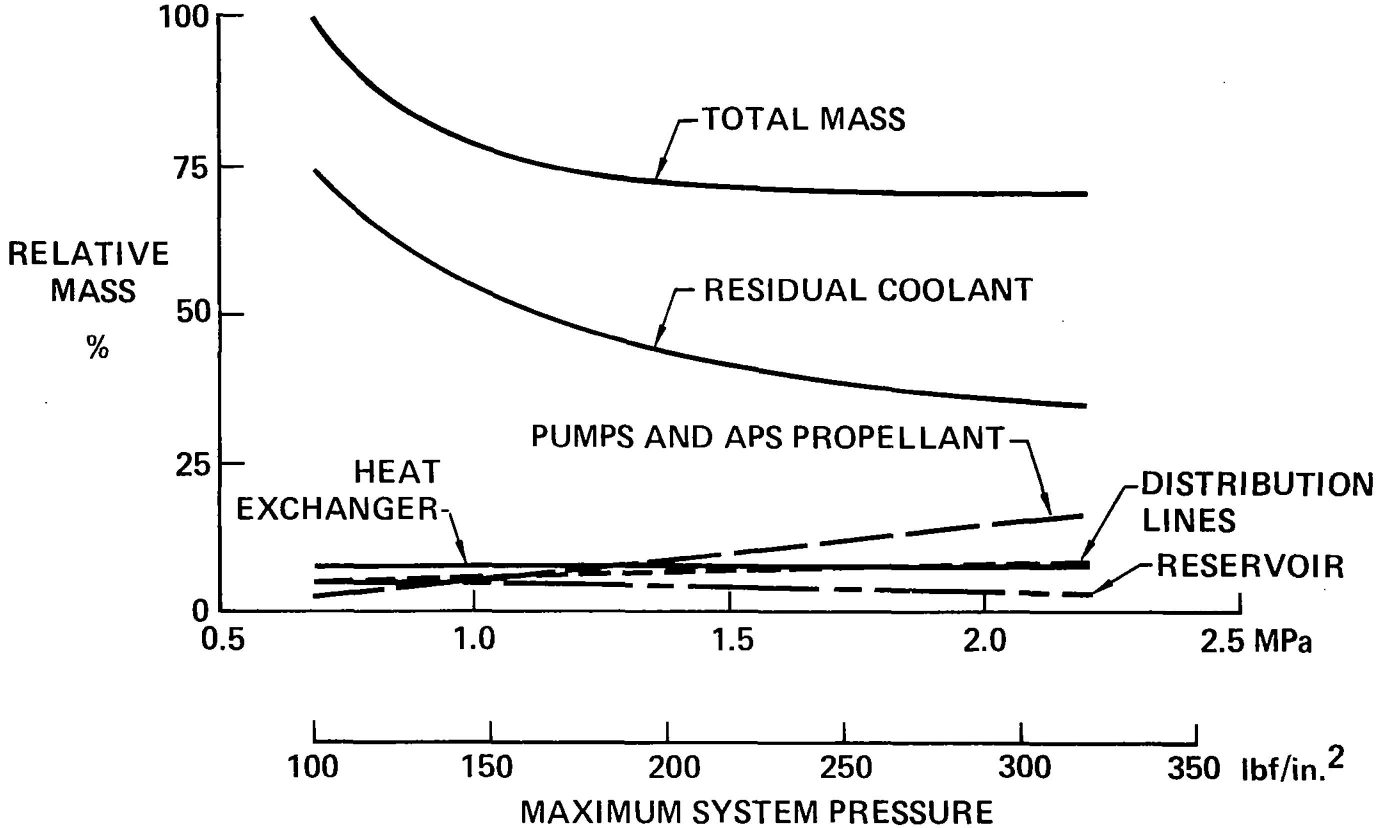


Figure 6

OUTER SKIN/TUBE/HEAT FLUX PARAMETRIC
(Figure 7)

Outer skin thickness as a function of tube diameter (with absorbed heat flux and tube pitch as parameters) is presented in the figure for a 422 K (300°F) maximum skin temperature. Analysis results were obtained assuming a uniform heat flux is absorbed by the outer skin and transferred via conduction in the outer skin, across the bond joint, and through the tube wall where it is then transferred via convection to the coolant.

The data presented here are for a 60/40 mass solution of ethylene glycol/water coolant, a maximum outer surface skin temperature (T_{MAX}) of 422 K (300°F), and a Dee-tube wall thickness of 0.089 cm (0.035 in). Similar data were developed for other combinations of skin thickness, T_{MAX}, and tube wall thickness and were used in the optimization process.

As indicated, the thermodynamic requirements for heat transfer can be satisfied with various combinations of skin thickness and tube diameter. This benefits the strength analyst because he can now determine, for each of these combinations, the minimum-mass honeycomb core and inner skin that will satisfy the strength requirement. When that is done, he will know the minimum actively cooled panel mass for any absorbed heat flux. This is illustrated in the next figure.

OUTER SKIN/TUBE/HEAT FLUX PARAMETRIC
60/40 MASS SOLUTION OF ETHYLENE GLYCOL/WATER

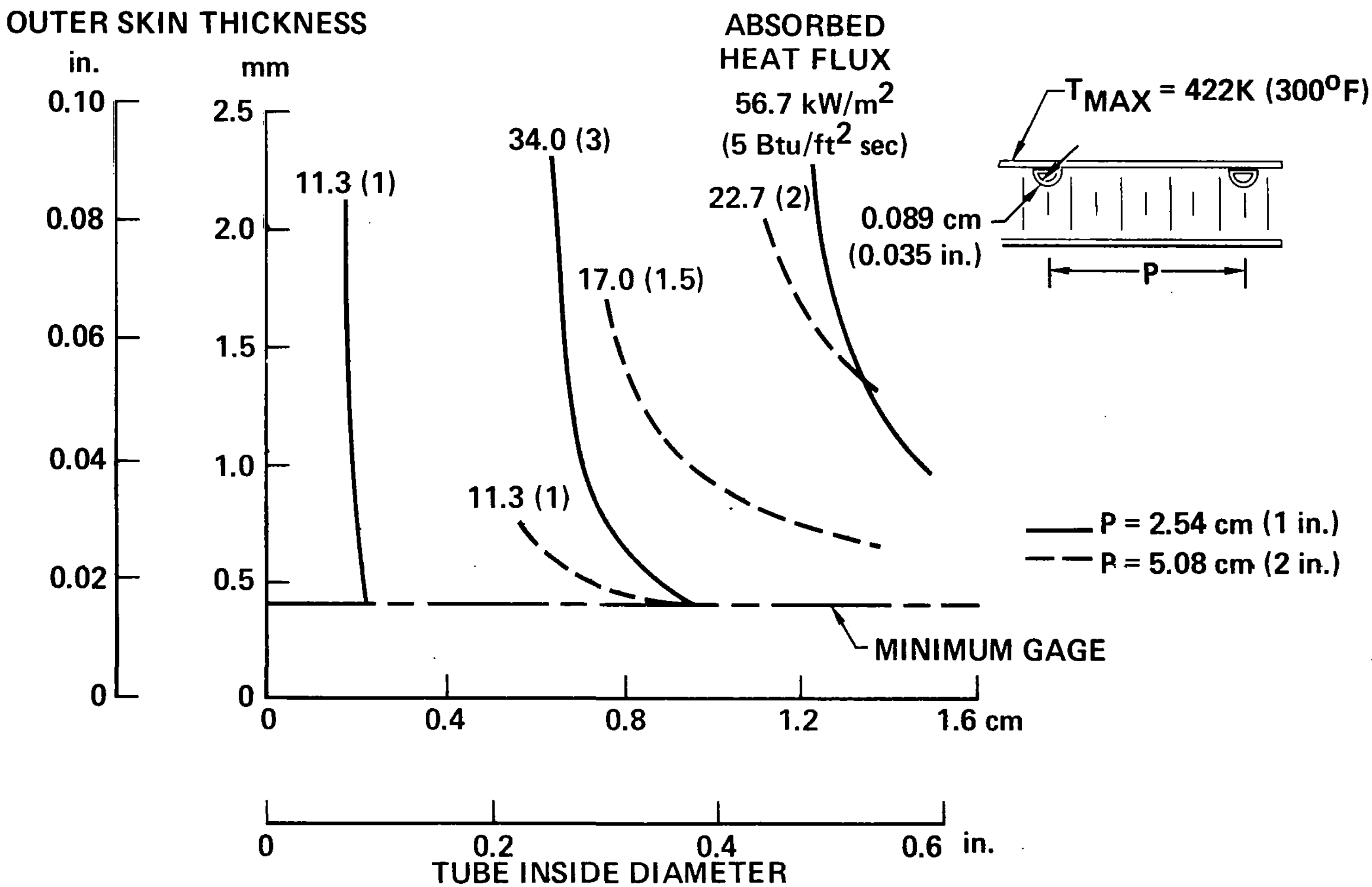


Figure 7

ACTIVELY COOLED PANEL MASS VERSUS ABSORBED HEAT FLUX
(Figure 8)

This figure shows how the optimized actively cooled panel plus residual coolant mass varies with absorbed heat flux for three different values of tube pitch. Each point on these curves represents a unique combination of inner and outer skin thicknesses, honeycomb core height and tube diameter that results in minimum mass for the specified materials, maximum outer skin temperature (T_{MAX}), and coolant temperature change (ΔT_c), and tube wall thickness (t_t). Similar plots, not shown, were developed for other combinations of T_{MAX}, ΔT_c and t_t. They showed that panel mass was not sensitive to significant variations of these parameters; except when T_{MAX} exceeded approximately 436 K (325°F).

These curves indicate that panel plus residual coolant mass approaches a minimum at low absorbed heat flux levels. The same trend was observed earlier for the insulation and active cooling system. They also show that mass is reduced by using smaller diameter coolant tubes. However, the minimum mass at an absorbed heat flux of 9.1 kW/m^2 (0.8 Btu/ft^{2}sec) is virtually the same for tube pitches ranging from 2.54 cm to 5.08 cm (1.0 in to 2.0 in). Based on this parametric study, the optimum coolant parameters, and basic actively cooled panel materials and dimensions were determined.

In order to size the structure for the specified design conditions (Figure 2), allowable operating stresses had to be determined for all of the panel structural elements. Following is a discussion of the approach to establishing these allowables.

ACTIVELY COOLED PANEL MASS vs ABSORBED HEAT FLUX

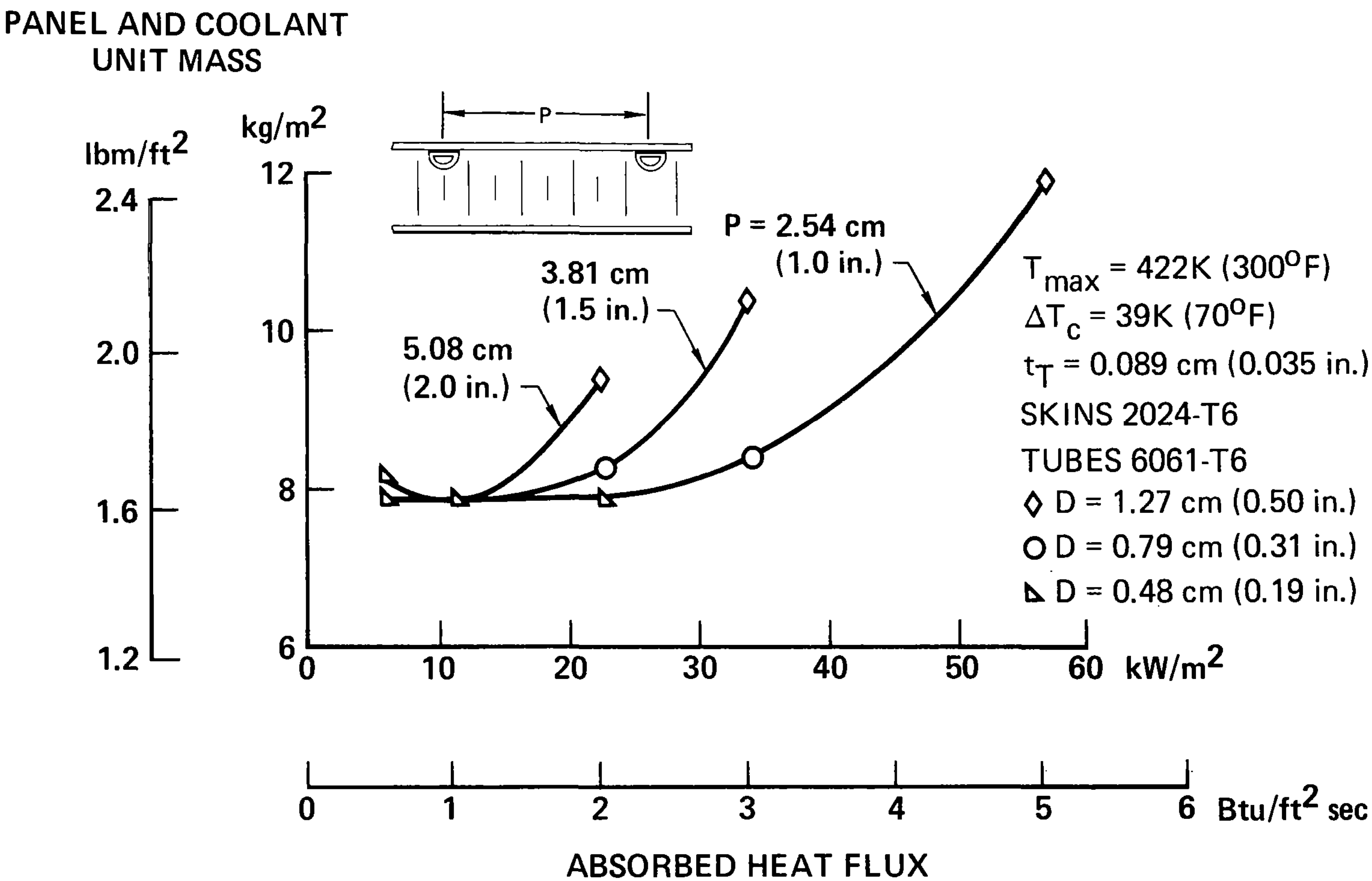


Figure 8

DEE-TUBE CRACK GROWTH
(Figure 9)

The actively cooled panel was designed to sustain a fully reversed mechanical loading, which was specified, combined with thermal loading, which could only be calculated after panel element temperatures and sizes were determined. Thus, establishment of allowable stress levels and sizing of the elements was an iterative process. Initially, allowable tension stresses were based on fatigue analysis and crack propagation analysis assuming fully reversed loading. Allowable compression stresses were set equal to the material's compression yield strength. Later iterations showed the compression allowables were O.K., but that the tension stresses needed revision to account for the change in stress ratio due to combining the fully reversed mechanical loads (Stress ratio = minimum stress/maximum stress = -1.0) with thermal stresses, which sometimes added and sometimes subtracted.

This figure shows fatigue crack propagation in a 6061-T6 Dee-tube with an initial flaw of the largest size that McDonnell Aircraft felt would escape detection during nondestructive inspection. The assumed flaw was a surface crack 0.46 mm (0.018 in) long and 0.23 mm (0.009 in) deep. It was determined that the crack would grow to the tube wall thickness in 23 000 cycles with a stress ratio of -0.09 and a maximum tension stress in each cycle of 163.27 MPa (23 680 lbf/in^2). This analysis was the basis for establishing the 6061-T6 Dee-tube wall thickness and allowable tension stress.

DEE-TUBE CRACK GROWTH

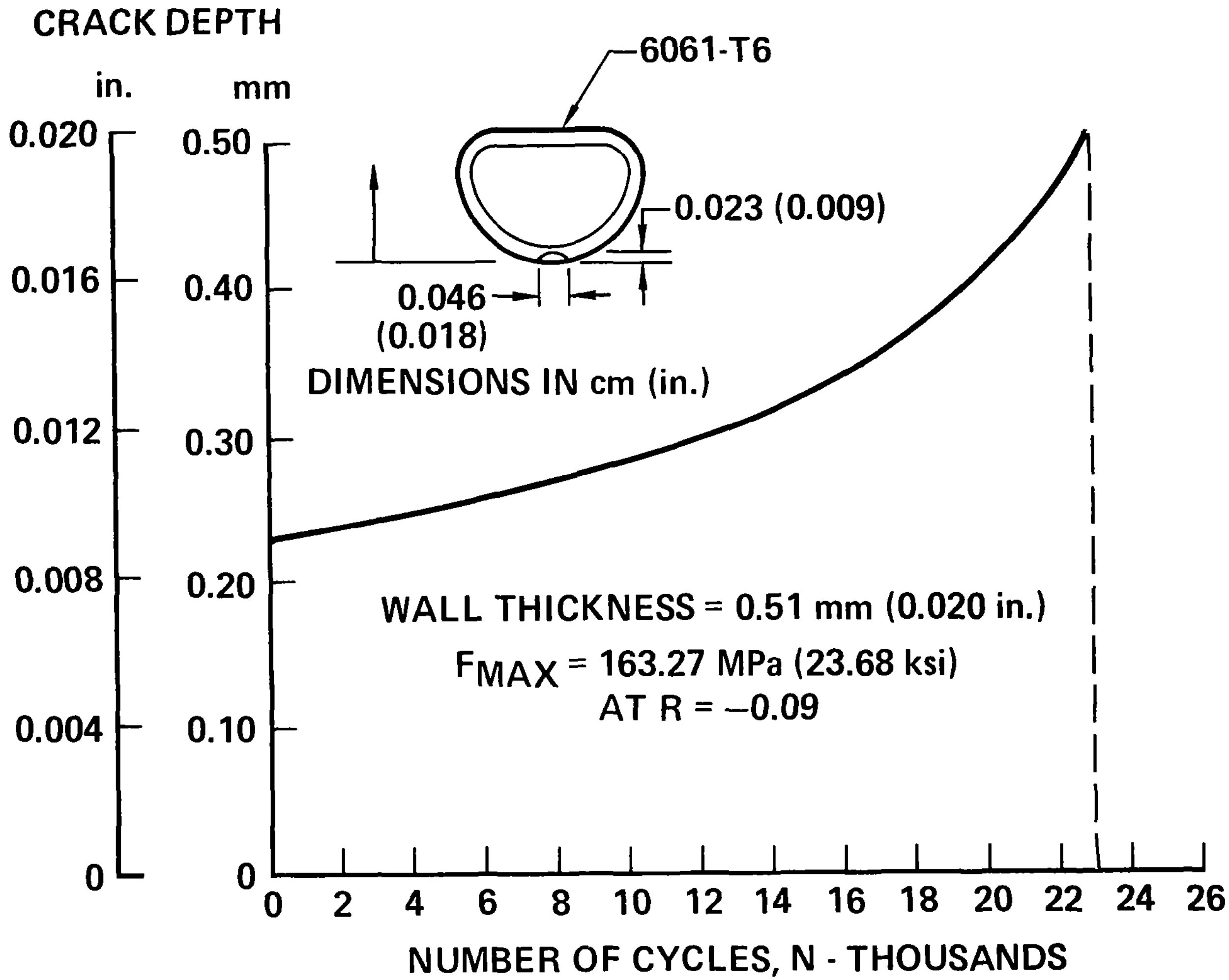


Figure 9

OPTIMIZED PANEL DETAILS AND COOLANT PARAMETERS
(Figure 10)

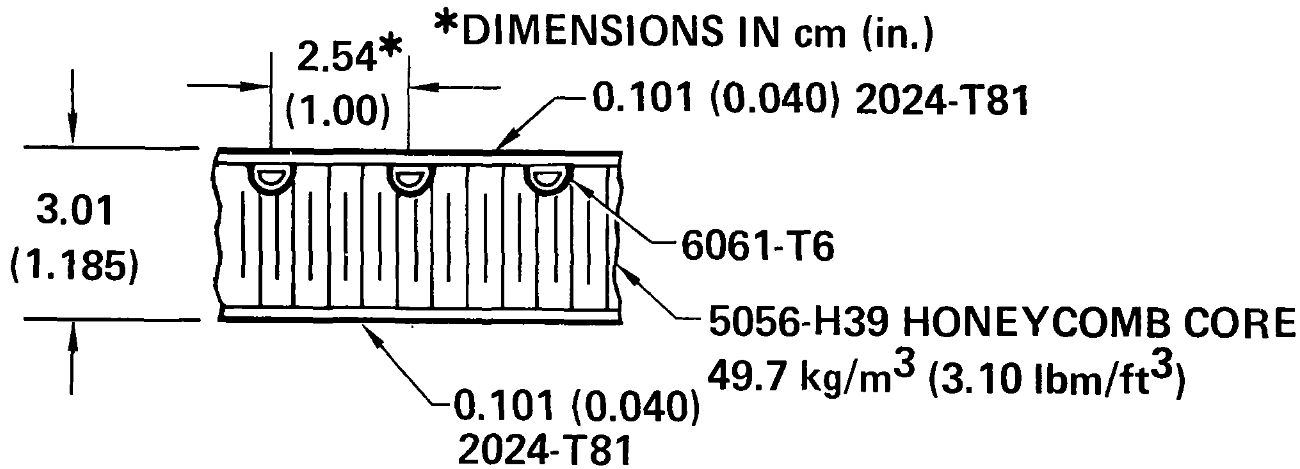
This figure identifies the optimized, actively cooled honeycomb-sandwich panel structural materials, geometry and coolant parameters. Both face skins are 0.101 cm (0.040 in) 2024-T81 aluminum. A low density 5056-H39 nonperforated core is used. Total depth of the sandwich is 3.01 cm (1.185 in).

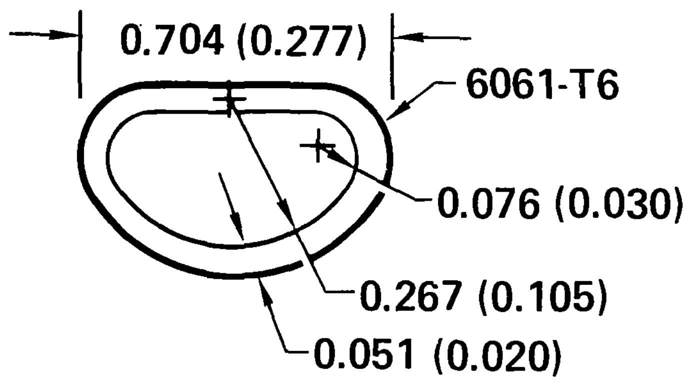
The 6061-T6 Dee-tube illustrated here is the configuration selected for the test panel. It is fabricated by flattening a round tube against an internal, removable mandrel. For production application, a drawn extrusion with a wider flat, hence more area for conduction, would be selected.

The skins, tubes and honeycomb were designed to be joined by adhesive bonding. An aluminum-filled, film-type adhesive was selected for use in the skin/tube joints and the skin/honeycomb joints. The aluminum-filled adhesives have better thermal conductivity than the nonfilled types and the system selected for this application can be used up to 491 K (425°F). To assure adequate heat transfer a maximum skin/tube bondline thickness of 0.25 mm (0.010 in) was selected. This was assured by using only one layer of adhesive in this joint. A foaming adhesive, capable of expanding to fill gaps, was used in the tube/honeycomb joint as the fit was not expected to be good enough for film-type adhesives.

The actively cooled panel was designed for a maximum operating temperature (midway between tubes near the exit manifold) of 422 K (300°F). The panel is cooled by pumping a 60/40 mass solution of ethylene glycol and water through the coolant passages at a mass flow rate of 62 g/sm^2 (46 $lbm/hr\ ft^2$) with an inlet coolant temperature of 283 K (50°F). The use of ethylene glycol/water as the coolant and the 283 K (50°F) inlet temperature was based on results from Reference 1. Including losses in the inlet and exit manifold, a pressure drop of 110 kPa (16 lbf/in^2) is experienced across the panel. The panel was designed for a maximum inlet pressure of 862 kPa (125 lbf/in^2) and a minimum exit pressure of 345 kPa (50 lbf/in^2).

OPTIMIZED PANEL DETAILS AND COOLANT PARAMETERS





T_{max} = 422K (300^{o}F)

T_{ci} = 283K (50^{o} F)

T_{co} = 329K (133^{o}F)

P_i = 456 kPa (66 lbf/in.^{2})

P_o = 345 kPa (50 lbf/in.^{2})

Figure 10

ATTACHMENT OF DEE TUBE TO MANIFOLD
(Figure 11)

This figure shows the method of attaching the Dee tubes to the manifold. Initially, the plan was to dip braze the 6061-0 Dee tubes to individual sockets in the 6061-0 manifold and then to solution treat this subassembly to the W condition before it is straightened, aged to T6 and bonded to the outer skin. Manufacturing felt that straightening and heat treatment of these large assemblies would require considerable development and would be expensive. Therefore, the approach shown here was adopted. It allows the tubes and manifold to be straightened and heat treated separately. Then they are joined together and to the outer skin by adhesive bonding.

The 6061-0 Dee tubes are cut approximately 2.5 percent shorter than ultimately required, and the ends are crimped and spot welded shut. The ends are then torch brazed to the recess of the machined 6061-0 aluminum tabs. After the coolant passage is electrical-discharge machined (EDM) as indicated in the inset, the tube/tab assembly is solution treated to the W condition, stretch straightened to final length, and aged to the T6 condition.

The manifolds were designed as machined 6061-T6 aluminum extrusions with welded-on end caps and inlet port. For the test panel, they were welded-up of several machinings. Holes were drilled in the tube/tab recesses to allow coolant flow.

An adhesive film between the manifold and the tab provides for load transfer and seals the joint against leakage. To prevent the adhesive from flowing into and blocking the passages during the bonding operation, rubber rods were inserted through small access holes drilled in the manifold surface opposite each coolant tube passage. After bonding, rods were removed and the access holes were plugged with Lee plugs.

ATTACHMENT OF DEE TUBE TO MANIFOLD

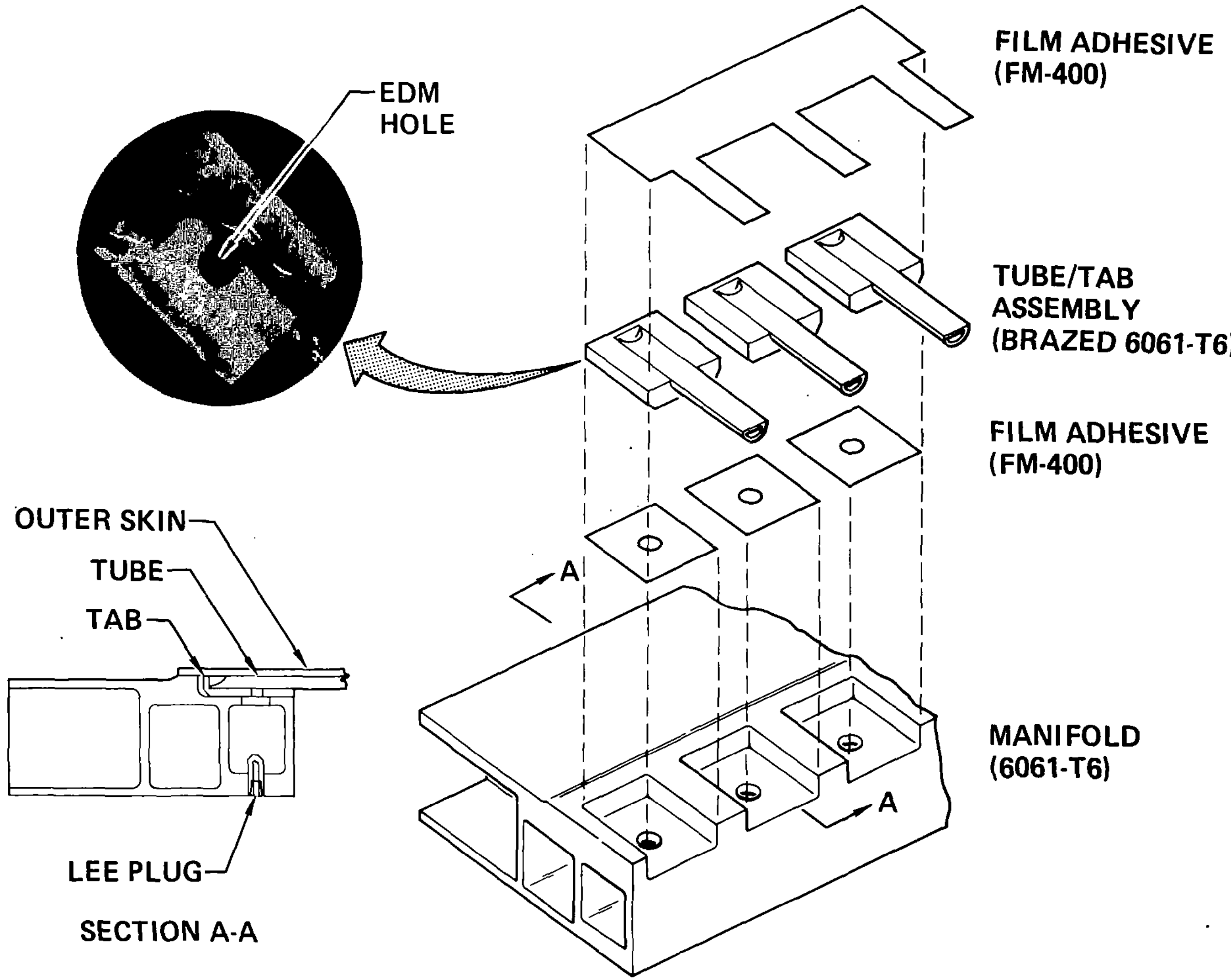


Figure 11

ACTIVELY COOLED PANEL JOINT DETAILS
(Figure 12)

Some of the actively cooled panel joint and edging details are illustrated in this figure. The longitudinal and transverse splices, with help from the frames and bulkheads, can handle any type of loading. This is one of the reasons the actively cooled panel, which was designed for a unidirectional longitudinal loading, can sustain an additional tranverse loading equal to 50 percent of the longitudinal loading with no mass increase.

The longitudinal splice plates are 0.082-cm (0.032 in) thick 2024-T81 aluminum. They are attached with permanent fasteners that pass through the panel. A potting compound in the honeycomb core prevents crushing during fastener installation.

The transverse splice is accomplished by the bulkhead and a 0.254-cm (0.10 in) thick outer splice plate. A double row of bolts through the splice plate, a solid pad in the manifold, and the bulkhead flanges complete the joint.

A RTV 560 adhesive is used in attaching the outer splice plates to the panel. This was necessary to improve heat transfer to the panel and, thus, avoid excessive splice plate temperatures.

Intermediate frames are attached to the panel by bolts that pass through the panel and between coolant tubes. Aluminum bushings are used in the panel to prevent core crushing when the bolts are torqued up. These are removable bolts because, as will be shown later, they are also used for heat shield attachment. The abestos phenolic washer under the platenut reduced heat transfer from the bolt to the panel and frame.

ACTIVELY COOLED PANEL JOINT DETAILS

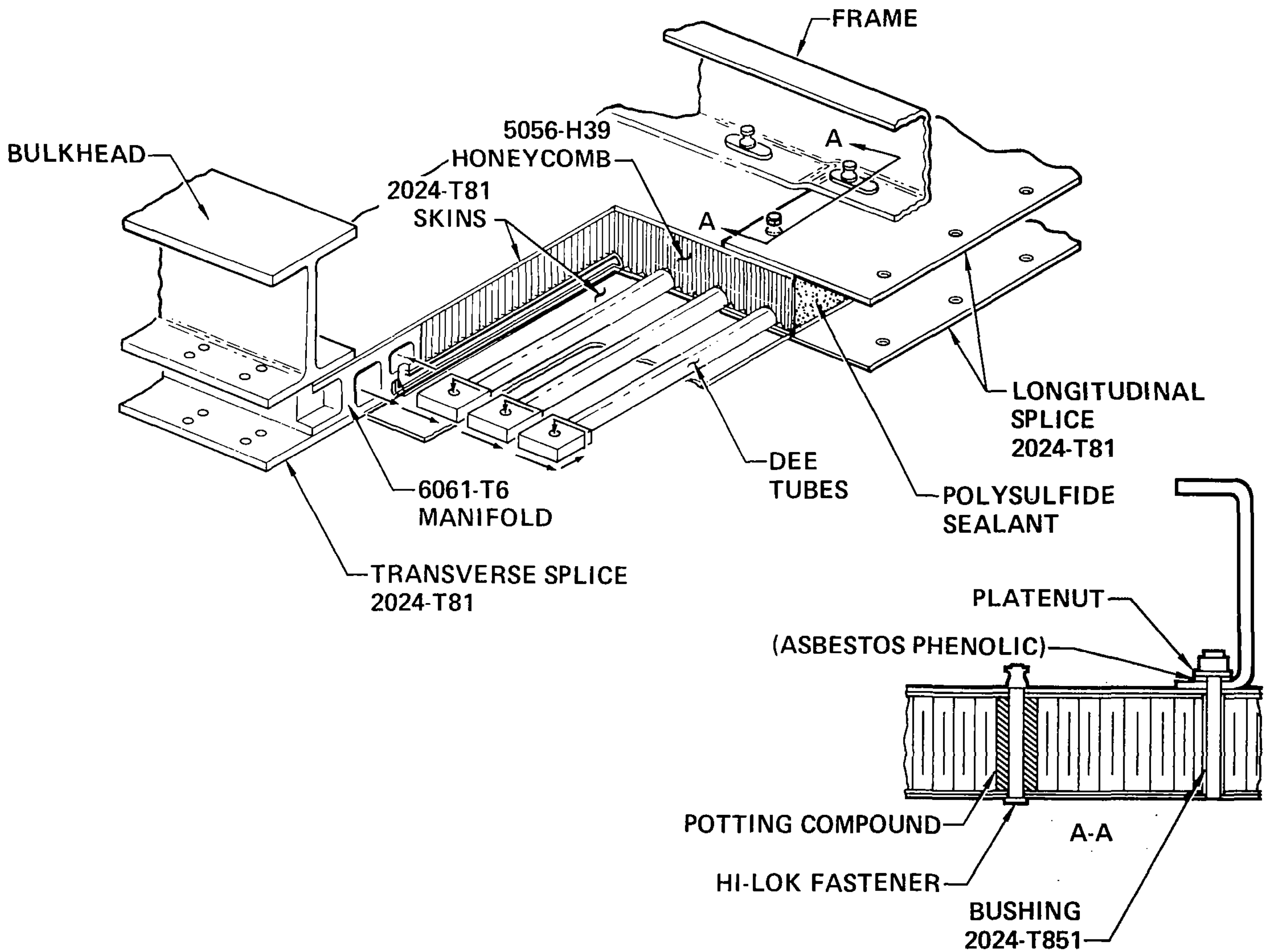


Figure 12

HEAT-SHIELD CONCEPT RANKING
(Figure 13)

Nine different radiation system concepts were evaluated to identify the concept to be used on the radiative actively cooled panel. This figure shows the four highest ranked concepts, based on consideration of mass, cost, producibility, inspectibility, maintainability, durability, volmetric efficiency, thermal/structural performance and integrity, advanced development needs, resistance to hot gas influx, and tolerance to overheating.

The screen-sandwich concept consisted of a layer of insulation encased in a very fine mesh screen and bonded directly to the structure. It ranked first but was not selected because it required manufacturing development, which was beyond the scope of the contract, and there was a good deal of uncertainty about its durability and maintenance requirements.

The corrugation stiffened beaded-skin concept ranked number two and was selected for use on the radiative actively cooled panel. Rene' 41 was selected in consideration of the 1080 K (1485°F) temperature experienced on the outer surface. The metallic heat shield seems to inspire greater confidence because of the amount of experience with it. The design adapted in this program has open beads and corrugations, which overlap those of adjacent heat shields, and has attachments to the panel at each flat. The beaded and corrugated skins are joined by spot welding and the Rene' 41 is oxidized to increase emissivity.

The preloaded dome is a Rene' 41 square plate which is formed to a spherical shape. This dome is then preloaded by means of a bolt at the center of the plate attached to the actively cooled panel. The optimized, domed heat shield was 20.3 cm (8.0 in.) square. An attractive feature of the design is that one heat shield would fit almost anywhere on the aircraft's surface. However, aerodynamic drag was the undoing of this design. It was estimated that the increase in fuel volume, aircraft size and engine performance required to maintain equal performance with this high drag, was equivalent to zero-drag domed heat shields that weighed 38.5 kg/m^2 (7.9 lbm/ft^2). The heat shield itself had quite low mass.

The Rene' 41 beaded skin concept ranked fourth. It is a very simple design. However, it was designed by stiffness requirements to prevent flutter and had significantly higher mass than the selected concept.

HEAT-SHIELD CONCEPT RANKING

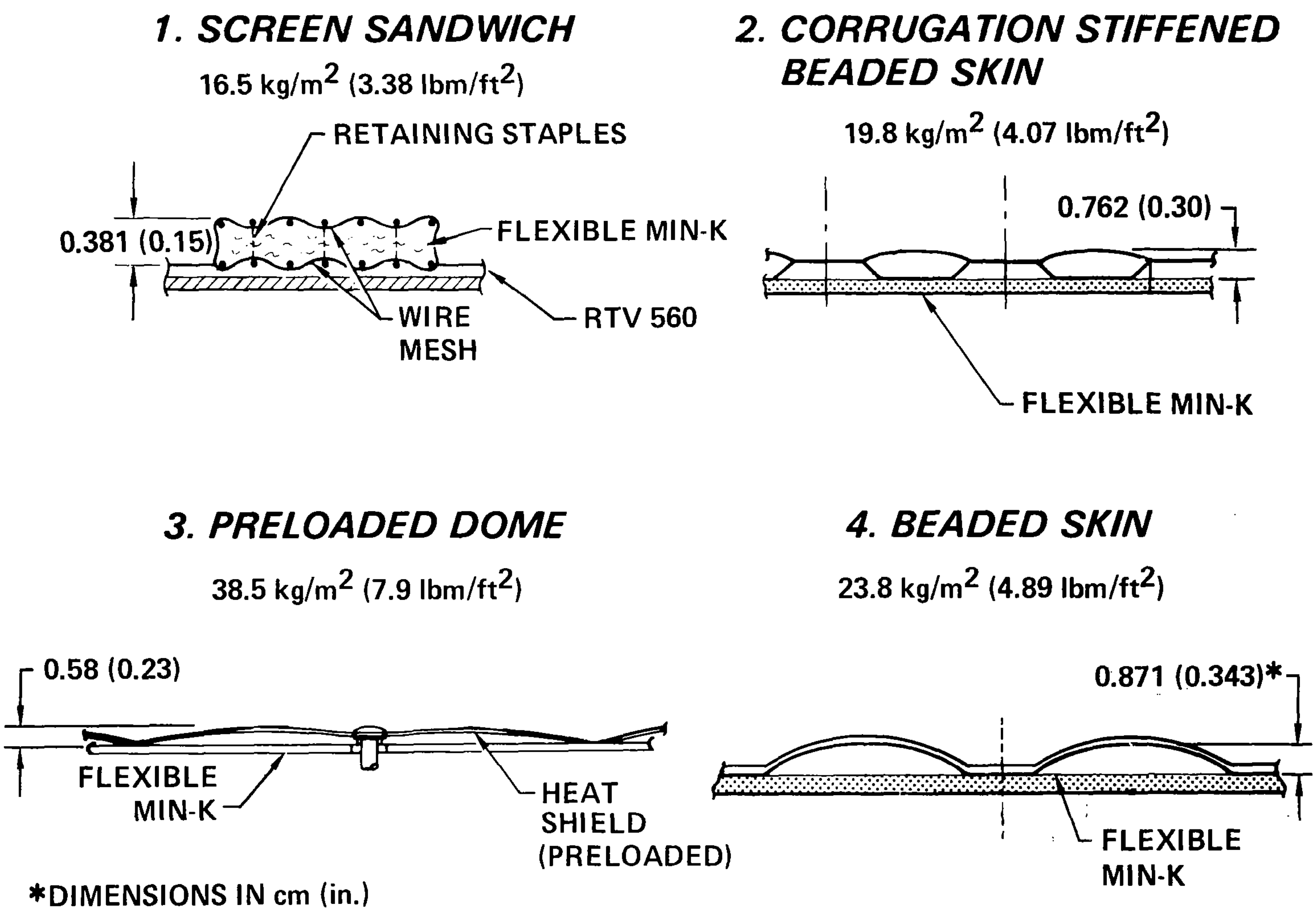


Figure 13

RENE' 41 HEAT-SHIELD MASS VS SUPPORT SPACING
(Figure 14)

All elements of the heat shield were optimized for minimum mass, except that skin thickness was not allowed to be less than indicated on this figure. The heat shield was designed to be restrained against lateral thermal expansion but not against longitudinal expansion.

Lateral expansion had to be accommodated by bending of the beads and corrugations. This sized the beaded skin but was not significant to the corrugations which were much less stiff. Minimum-mass outer skin was achieved with a 5.08-cm (2.0 in) bead pitch, 0.025-cm (0.010 in) skin thickness and 0.318-cm (0.125 in) crown on the corrugation.

With the minimum-mass beaded skin described above, the height of a minimum gage corrugation was varied as required to provide adequate bending strength for various heat-shield support spacings. Thus, the effect of support spacing on heat-shield mass could be determined. The results are shown on this figure. The heat-shield attachment mass is included in all cases.

It will be noted that significant mass reduction is achieved with support spacings of 30.4, 40.6, 45.8, and 50.8 cm (12, 16, 18 and 20 in). This is due to the fact that some of the frame attachments can also be used as heat-shield attachments.

Based on the above trade study, the selected heat-shield support spacing was 30.4 cm (12.0 in). The size of the heat shield was also studied, and it was determined that they could be 61.0 cm (24.0 in) long with a row of supports at each end and a row in the middle. Heat-shield width was set, somewhat arbitrarily, at 61.0 cm (24.0 in) because that was the width of the actively cooled panel. They could well be made wider.

RENE' 41 HEAT-SHIELD MASS vs SUPPORT SPACING

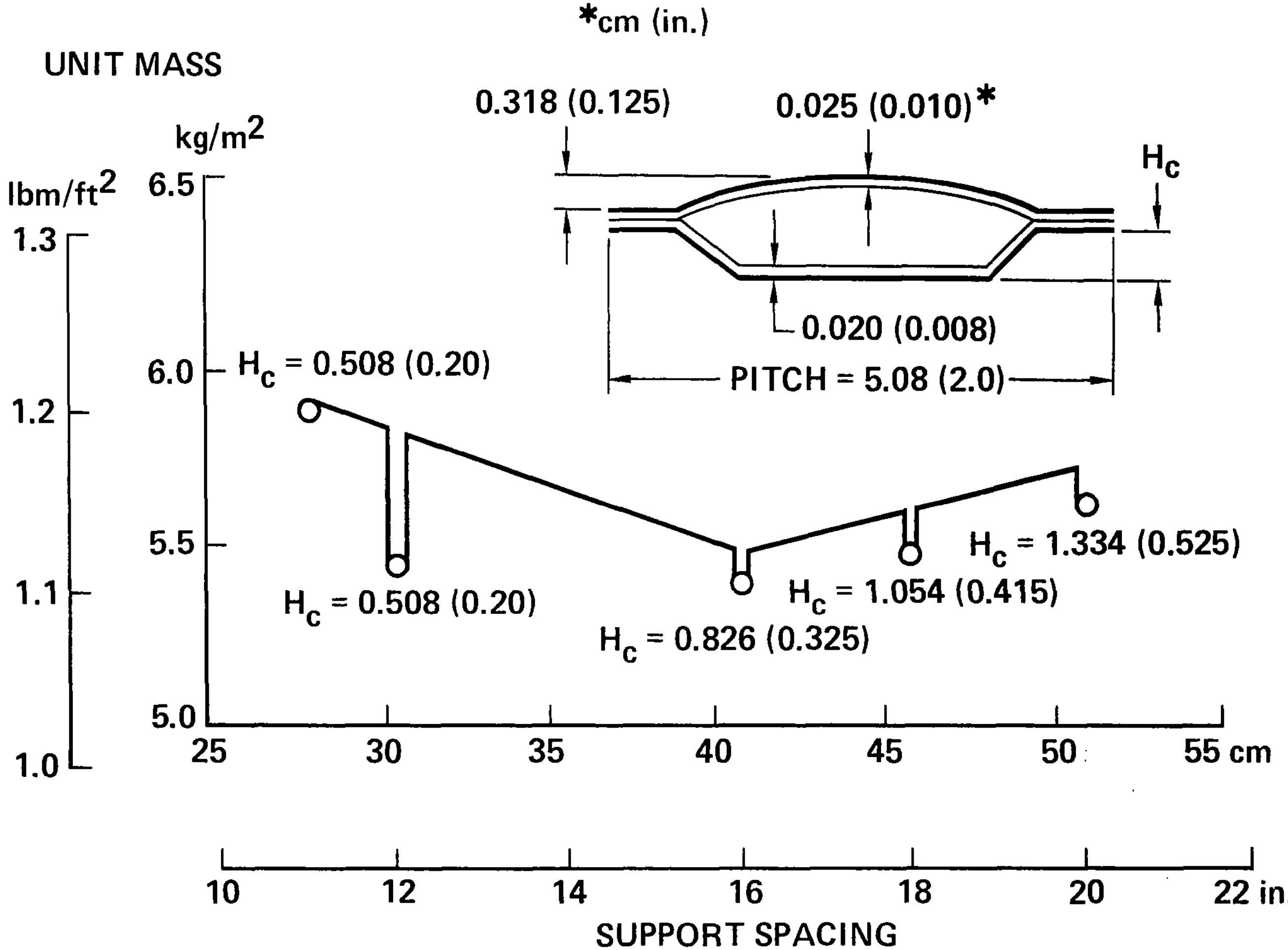


Figure 14

HEAT-SHIELD MASS VS ABSORBED HEAT FLUX
(Figure 15)

This figure shows how heat-shield mass varies with absorbed heat flux. It also shows that increasing bead/corrugation pitch increases mass.

The increase in heat-shield mass at low absorbed heat flux results from reduced material properties at higher temperatures. The Rene' 41 properties drop rapidly at temperatures above 1060 K (1450°F) which is experienced when the absorbed heat flux is less than 11 kW/m^2 (1.0 Btu/ft^{2}sec). At lower temperatures, the change in material properties is much less dramatic.

HEAT-SHIELD MASS vs ABSORBED HEAT FLUX

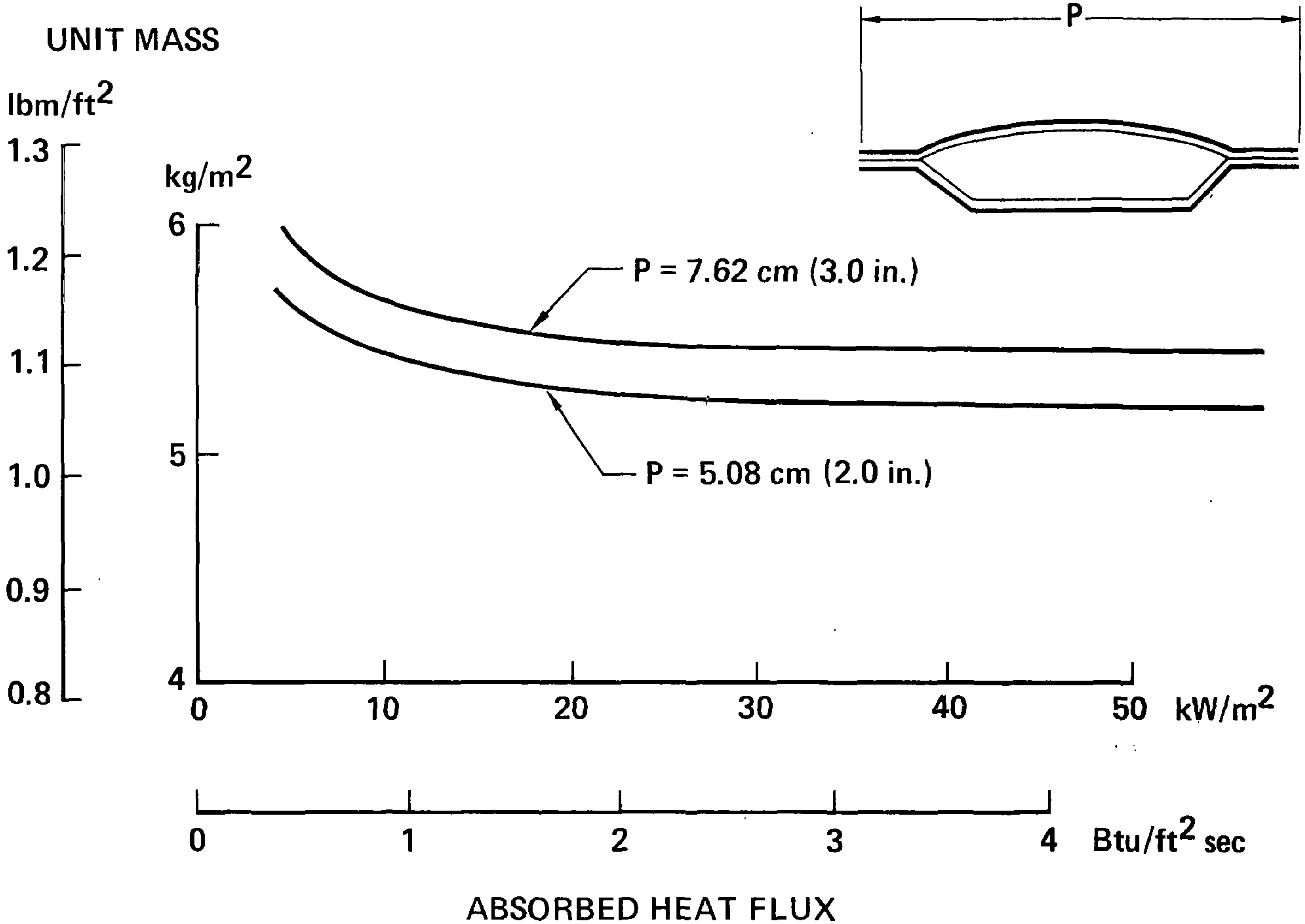


Figure 15

HEAT-SHIELD AND INSULATION DETAILS
(Figure 16)

This figure shows additional heat-shield and insulation details and the provisions for attaching them to the actively cooled panel. The method of determining the optimized heat-shield geometry, shown here, was discussed previously.

Heat-shield end details are illustrated on the right. The corrugations on the heat-shield trailing edges, and the beads on the leading edges, are trimmed back to allow adjacent panels to overlap and expand longitudinally without binding. Slotted holes at the overlays are also required at the attachment bolts to provide for thermal expansion.

The insulation is only 0.381 cm (0.15 in) thick. It is packaged in a stainless steel foil.

The heat-shield stand-off posts are an integral part of the insulation package. The heat-shield attachments pass through the stand-off posts and the actively cooled panel as the next figure illustrates.

HEAT-SHIELD AND INSULATION DETAILS

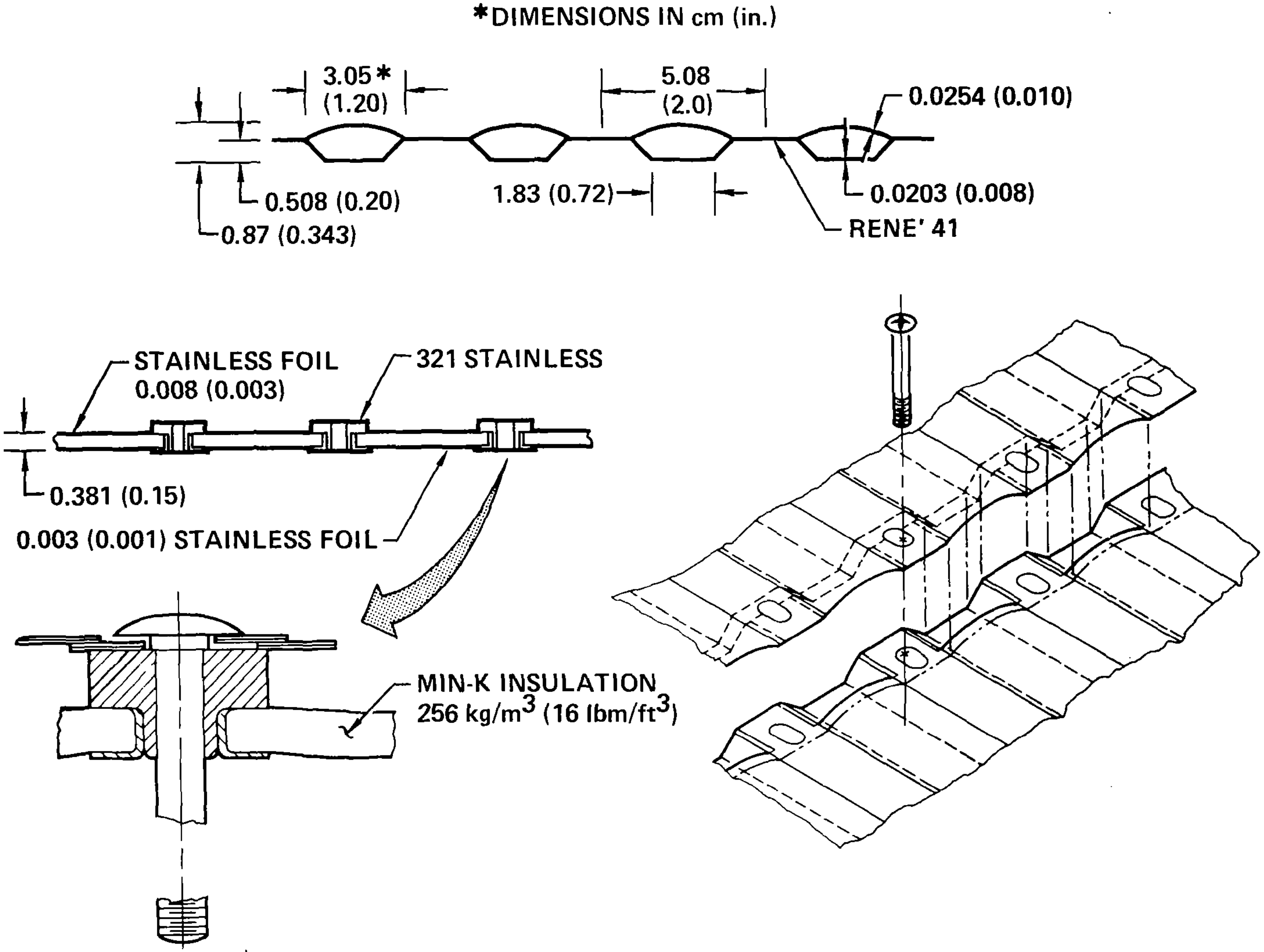


Figure 16

MAXIMUM TEMPERATURES AT LONGITUDINAL SPLICE
(Figure 17)

This figure shows the temperatures that exist in one of the hottest spots on the panel, which is at the edge of the panel near the exit manifold. The figure also shows more of the heat-shield attachment detail and the coolant tube that is located outside the longitudinal-splice attachment line to provide better cooling.

Maximum panel temperature is on the outer splice plate at the panel edge. Maximum heat-shield temperature is on the top of the bead.

The heat-shield stand-off post and the bolt act almost like direct heat shorts to the panel. This was accounted for by slightly increasing insulation thickness so that the absorbed heat flux accounts for heat shorts as well as heat passing through the insulation.

MAXIMUM TEMPERATURES AT LONGITUDINAL SPLICE

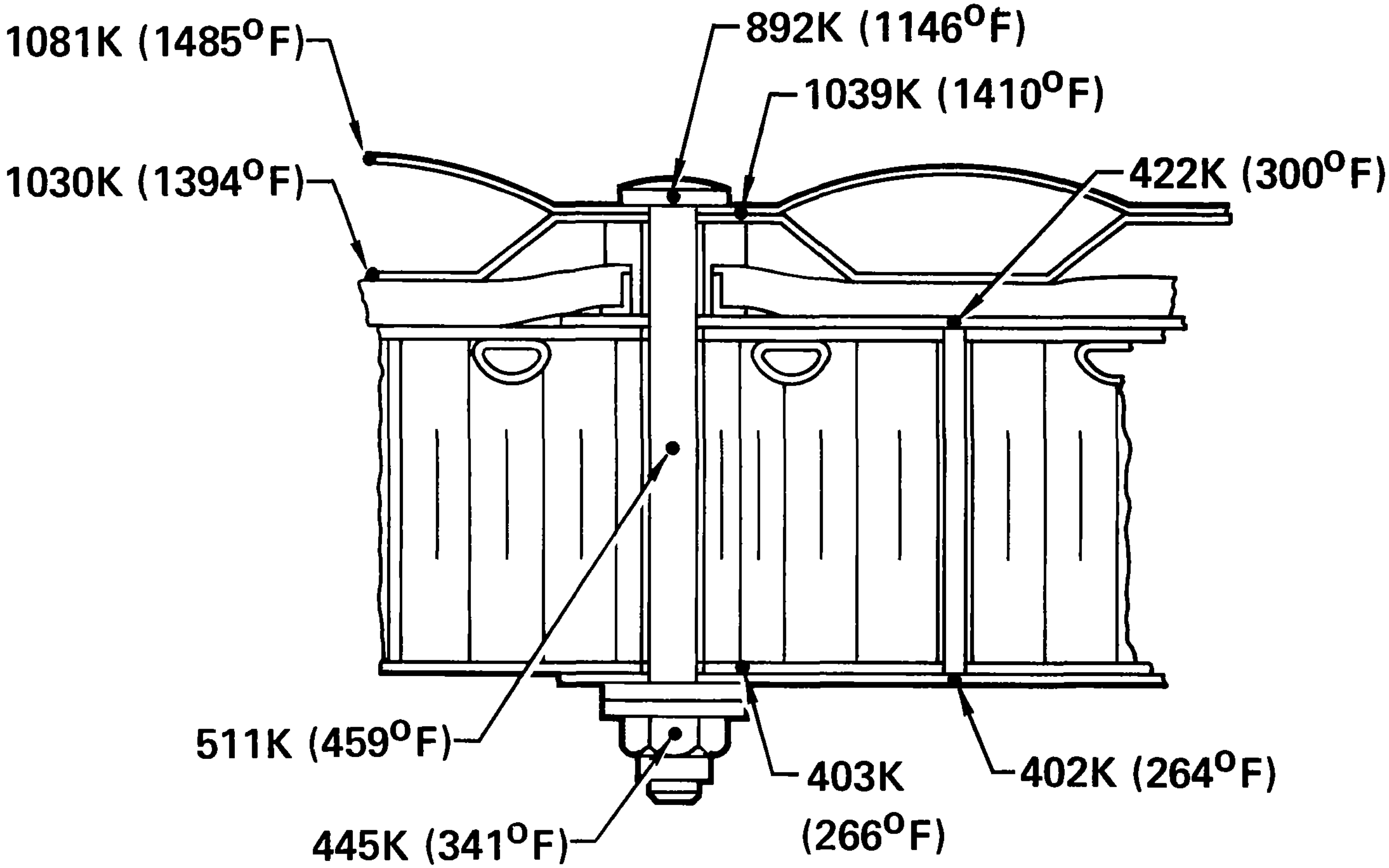


Figure 17

RACP SYSTEM MASS IS LESS THAN FOR BARE ACP
(Figure 18)

This figure shows that the total mass of a radiative actively cooled structural system is less than that of a bare actively cooled system designed for the same conditions.

There is not much difference in actively cooled panel mass for the two systems. This is not due to an insensitivity of panel mass to absorbed heat flux. It is more likely due to a difference in methods of attaching the Dee tubes to the outer skin. The bare panel, with absorbed heat flux of 136 kW/m^2 (12 Btu/ft^{2}sec), had tubes soldered to the skin. The radiative actively cooled panel (RACP), with an absorbed heat flux of 9.1 kW/m^2 (0.8 Btu/ft^{2}sec), had tubes adhesively bonded to the skin. With bonded joints, the RACP could not absorb 136 kW/m^2 regardless of tube spacing and coolant mass flow.

The combined mass of the radiation and cooling system of the RACP is slightly less than the mass of the cooling system for the bare panel.

RACP SYSTEM MASS IS LESS THAN FOR BARE ACP

COMPONENTS	RACP		ACP	
	kg/m^2	lbm/ft^2	kg/m^2	lbm/ft^2
ACTIVELY COOLED PANEL				
SKINS	5.86	1.20	3.76	0.77
COOLING PASSAGES	0.78	0.16	2.73	0.56
HONEYCOMB	1.42	0.29	1.32	0.27
MANIFOLDS	0.78	0.16	0.64	0.13
CLOSEOUTS	0.63	0.13	1.76	0.36
ADHESIVES	1.95	0.40	2.10	0.43
FASTENERS, ETC.	1.02	0.21	0.49	0.10
PANEL MASS	12.44	2.55	12.80	2.62
RADIATION SYSTEM				
HEAT SHIELDS	4.34	0.89	–	–
INSULATION PACKAGES	1.86	0.38	–	–
FASTENERS, ETC.	1.07	0.22	–	–
RADIATION SYSTEM MASS	7.27	1.49		
COOLING SYSTEM				
COOLING INVENTORY	0.59	0.12	1.86	0.38
PUMPING PENALTY	0.01	–	0.34	0.07
DISTRIBUTION SYSTEM	1.76	0.36	8.64	1.77
COOLING SYSTEM MASS	2.36	0.48	10.84	2.22
TOTAL MASS	22.07	4.52	23.64	4.84

Figure 18

BONDED OUTER SKIN AND DEE-TUBE ASSEMBLY
(Figure 19)

A 0.61m (2 ft) x 1.22m (4 ft) radiative actively cooled test panel has been fabricated and delivered to NASA Langley Research Center for tests. This figure shows the partially completed actively cooled panel after the outer skin, tubes and manifolds had been bonded together. This was accomplished in one bonding operation.

After bonding, the cooling system was pressure checked to locate leaks and a Thermovision (IR scanning system) check was run to be sure all of the coolant passages were clear. Thermovision indicated one tube was blocked. The Lee plugs (see Figure 11) were removed and the adhesive blocking the coolant was removed. The panel was replugged, and Thermovision confirmed that all passages were clear and flow was uniform across the panel.

After that, the honeycomb core, with machined recesses for the tubes, and the inner skin were bonded in place. Foaming adhesive was used between the tube and core and between the manifold and core.

BONDED OUTER SKIN AND TUBE/MANIFOLD ASSEMBLY

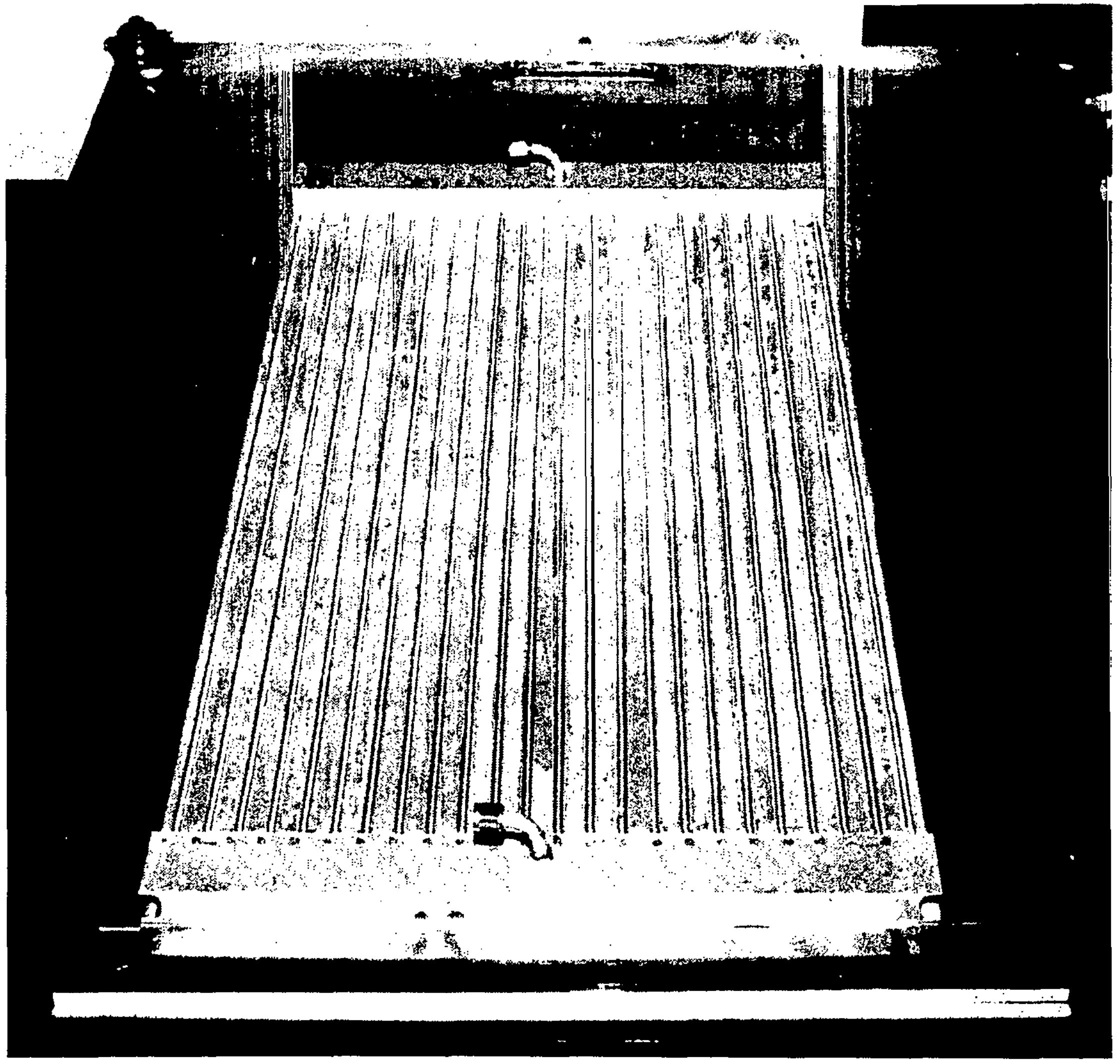


Figure 19

BONDED ACTIVELY COOLED PANEL
(Figure 20)

This figure shows the actively cooled panel after bonding. A drilling fixture was used to locate all fastener holes in the panel. The same fixture was then used to locate the mating holes in the heat shields and insulation packages. This was done to make sure the holes lines up properly.

BONDED ACTIVELY COOLED PANEL

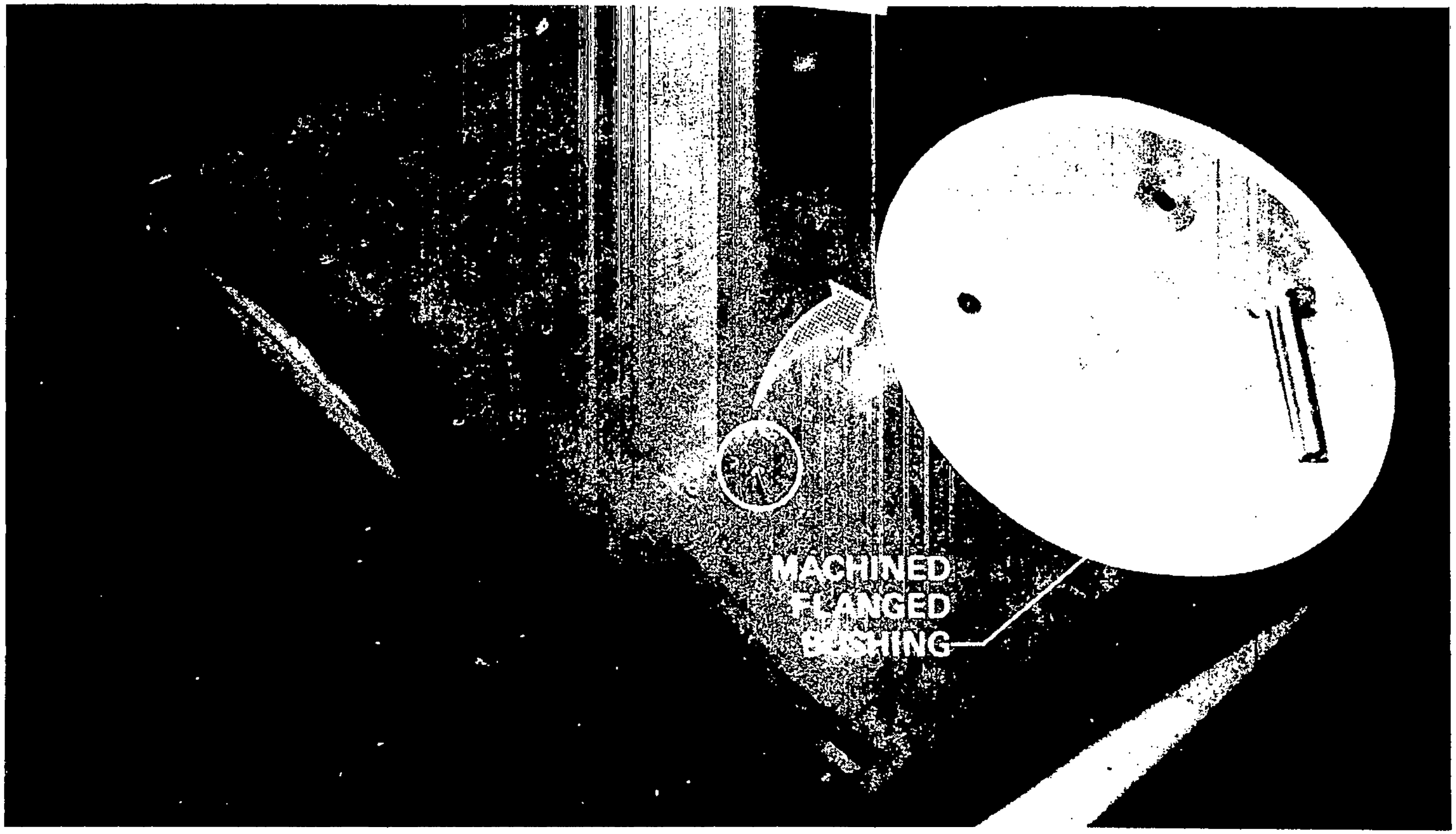


Figure 20

CONCLUSIONS
(Figure 21)

The primary objectives of the RACP program were achieved. Specific conclusions regarding sensitivity of component masses to geometric and coolant parameters were described earlier. General conclusions, which could significantly affect future systems design and R and D plans are summarized on this figure.

This program proved the feasibility of the radiative actively cooled structure approach for hypersonic cruise aircraft. It permits matching of the heat load to the available heat sink. Further, this match can probably be achieved by applying radiative active cooling to just part of the airframe surface.

The RACP designed in this program is seven percent lighter than a bare ACP designed for the same conditions. It is reasonable to expect that a similar conclusion would be reached using other actively cooled panel concepts.

The honeycomb-sandwich actively cooled panel is more difficult (complex) to fabricate than a conventional honeycomb panel. This is due to integration of the manifolds and cooling tubes. This was expected. A comparison of the fabricability of this RACP concept with alternate concepts for hypersonic cruise aircraft would be more meaningful.

CONCLUSIONS

- RACP PERMITS MATCHING HEAT LOAD AND AVAILABLE HEAT SINK
- RACP IS 7% LIGHTER THAN ACP
- RACP FABRICATION MORE DIFFICULT THAN CONVENTIONAL HONEYCOMB SANDWICH

Figure 21

REFERENCES

1. Koch, L. C.; and Pagel, L. L.: High Heat Flux Actively Cooled Honeycomb Sandwich Structural Panel for a Hypersonic Aircraft. NASA CR-2959, 1978.

2. Ellis, D. A.; Pagel, L. L.; Schaeffer, D. M.: Design and Fabrication of a Radiative Actively Cooled Honeycomb Sandwich Structural Panel for a Hypersonic Aircraft. NASA CR-2957, 1978