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Thermal Conductance of Two Space Station Cold Plate Attachment Techniques

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Introduction

TWO attachment techniques for mounting electronic equipment to space station cold plates were evaluated and compared. The characteristics investigated include the temperature variations and the thermal contact conductance at the electronic equipment/cold plate interface. Numerous investigations of heat transfer at the interface of two contacting metallic surfaces have been conducted including investigations dealing with the extent to which the thermal contact conductance can be improved using greases,[1] thin metallic foils,[2] or vapor deposited coatings.[3,4] Several excellent reviews which summarize these results and the other recent literature have been published.[5,6] In addition to these fundamental reviews, the results of several investigations directed specifically at the thermal contact conductance between components bolted onto single-phase liquid cold plates have been presented.[7-9]

The two techniques evaluated in this investigation included the standard 500 × 750 mm bolted cold plate, which utilizes 77 5-mm-diam bolts, approximately 15 mm in length and spaced on a 70 × 70 mm matrix pattern as shown in Fig. 1, and a flexible pressurized bladder. In normal operation, the pressurized bladder would be placed behind the devices to be cooled, clamping the base plate of the devices to the cold plate and eliminating the need for bolts.

In order to determine the effectiveness and thermal characteristics of these two attachment techniques, an experimental investigation was conducted using a specially developed thermal test plate and a flight-ready cold plate provided by Marshall Space Flight Center. Measurements of the surface temperature variation and the thermal contact conductance at the interface of the test plate and the cold plate were made for both attachment techniques.

For the purpose of this investigation, the heat source was simulated by an electrically heated, 500 × 750 mm aluminum 6061-T6 plate 2.54 cm thick. A series of 77 5-mm bolt holes 12.7 mm deep were drilled and tapped in the aluminum plate on 70-mm centers to match the bolt matrix of the cold plate. After tapping the bold holes, the aluminum plate was machined and ground to an overall flatness deviation of 0.1 mm with an rms surface roughness of 0.0032 mm.

Four silicone rubber resistance heaters, each with a rated capacity of 2250 W, were installed on the backside of the thermal test plate with silicone adhesive. The power to each heater was measured by both a digital Wattmeter and a multimeter used to measure the voltage and current. The temperature of the cold plate was controlled by varying the temperature of water flowing through a constant temperature circulating bath at a flow rate of 0.13 kg/s (2 gal/min).

In order to determine the thermal contact conductance and temperature variation at the contacting surface, the regions around four different bolt holes were instrumented. Around each hole, a series of 1.2-mm-diam thermocouple wells were drilled from the backside of the thermal test plate at four different radii, as illustrated in Fig. 1. A Chromel-Alumel thermocouple (AWG-36) was inserted and packed into each thermocouple well with a thermally conductive epoxy. Small channels were machined on the backside of the thermal test plate for the thermocouple wires and sealed with epoxy. In addition to a three-dimensional temperature profile around each bolt, this combination of thermocouples provided a means by which the thermal test plate surface temperature could be obtained at each radial distance.

The inflatable bladder used in this investigation (constructed by ILC Dover of Frederica, Delaware) was proof tested to 0.2071 MPa (30 psi) but was only operated to a pressure of 0.138 MPa (20 psi). The bladder was equipped with a standard Schrader valve. Pressure in the bladder was

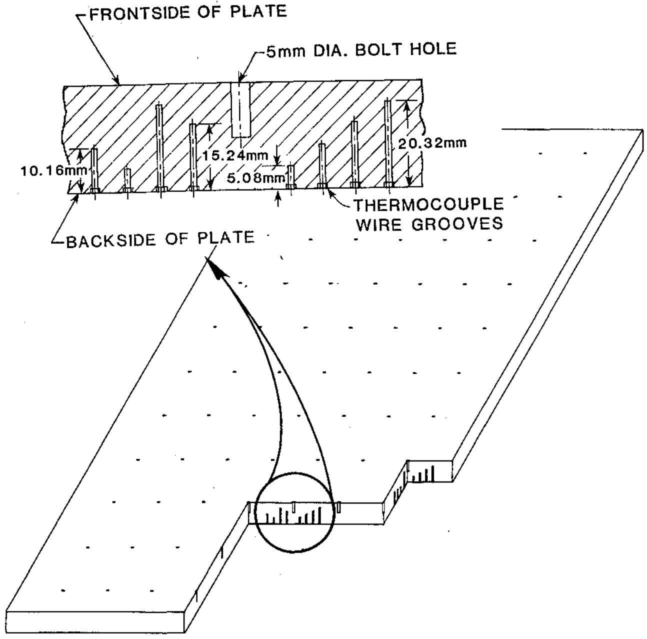

Fig. 1 Thermal test plate and thermocouple well locations.

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supplied by a hand pump and measured by a Magnehelic Series 200, 0-20 psi pressure gauge.

Because use of the bladder required a method by which the cold plate and aluminum plate could be held in contact, a test facility comprised of a wooden fixture supported by a series of 12 (6 on each side), 76-mm-wide × 0.64-m-long pieces of channel was constructed. These supports were held together by 12 19-mm-diam bolts. The thermal test plate, which was insulated from the container with 40 9.5-mm Teflon standoffs, and the cold plate were placed inside the wooden fixture. Although not necessary for the bolted configuration, all tests were conducted in this test facility to minimize variations in the experimental results.

During the experimental tests, the bolt torque was varied from 0.79 N-m (7 in.-lb) to 3.04 N-m (27 in.-lb) and the pressure in the bladder from 41.37 kPa (6 psi) to 130 kPa (19 psi). Prior to the tests, the thermal test plate and the cold plate were placed in contact and loaded to 130 kPa to reduce the effect of repeated loading. The plates were then separated, and the surfaces of both the cold plate and the thermal test plate were wiped clean with acetone and allowed to dry prior to each test.

The test plates were placed in contact, and 77 standard 5-mm steel bolts, 15 mm long, each with a 10-mm-diam washer, were tightened to the required torque in a predetermined pattern starting from the center and working outward. At each bolt torque, the power level and coolant temperature were adjusted to achieve an interface temperature of 300 K ± 5 K prior to recording data. For the bladder configuration, the bladder pressure was varied in 20 kPa (3 psi) increments, and again the interface temperature was allowed to stabilize between each test.

At each radial distance, a linear least-squares fit was used to extrapolate the measured temperatures to the test plate surface. Typically the extrapolated values were within ± 0.1 K of the temperature as measured by a thermocouple located at the surface of the test plate. The interface surface temperature of the cold plate was assumed to be the average temperature of the inlet and outlet working fluid. The temperature difference across the interface was calculated by subtracting the average cold plate temperature from the extrapolated aluminum plate surface temperature. This procedure was consistent with that used in previous investigations.[7-9]

The local thermal contact conductance, defined as

$$h_c = Q/(A\Delta T) \tag{1}$$

where Q is the thermal power, A is the cold plate area, and ΔT is the temperature drop at the interface, was calculated at four different radii using the extrapolated surface temperature of the thermal test plate and the average temperature of the coolant fluid. The overall experimental uncertainty associated with these computed values was the result of uncertainties in the location of the thermocouples, the temperature measurements, and the measured coolant temperature. Using root-sum-square method, the uncertainty was estimated to be ±8%.

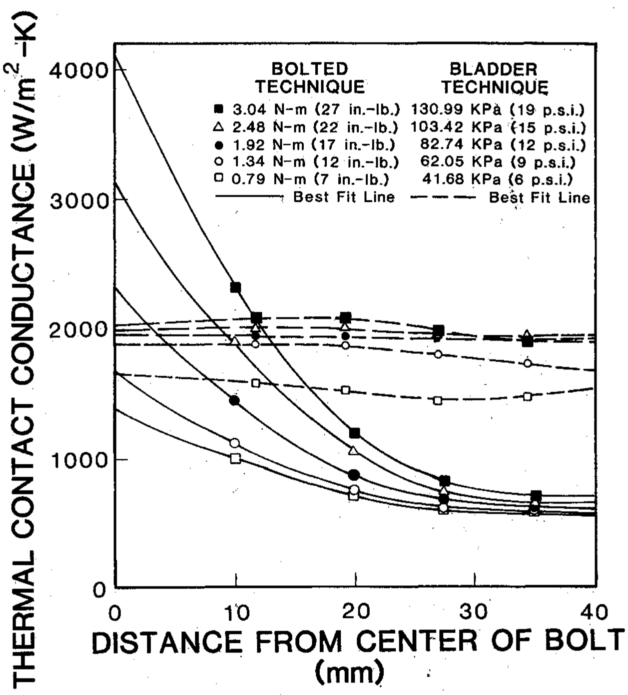

Fig. 3 Thermal contact conductance variation for the two attachment techniques.

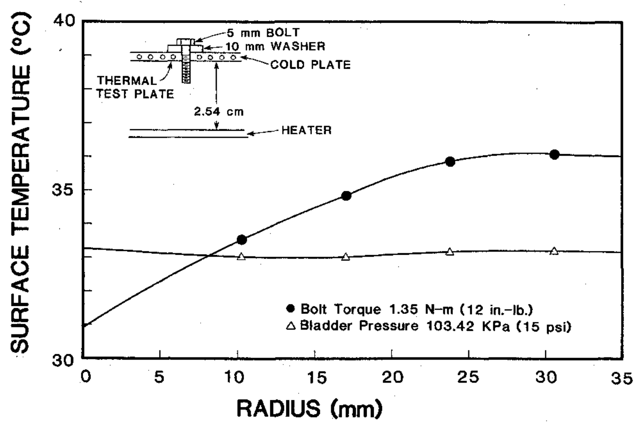

Fig. 2 Surface temperature variation for the two attachment techniques.

Results and Discussion

Figure 2 illustrates the temperature distribution for one of the instrumented locations with a uniform bladder pressure of 103.42 kPa (15 psi) and a bolt torque of 1.35 N-m (12 in.-lb). As shown, the bladder attachment technique yields a nearly uniform test plate surface temperature, and the bolted technique results in significant variations. The temperature distributions for the other pressures and torques were similar in shape and magnitude.

The computed thermal contact conductance values, determined by Eq. (1), were assumed to represent the local thermal contact conductance. Figure 3 illustrates the variation of these computed values for the bolted configuration as a function of the distance from the bolt centerline along with the results obtained for the pressurized bladder. As shown, the bolted attachment technique resulted in a rapid decrease in the thermal contact conductance at the interface, moving outward radially from the bolt centerline, due to the decrease in the contacting pressure. These results support the conclusion presented previously by other investigators that the pressure distribution in a bolted joint is a strong function of the distance from the bolt centerline.[10,11] The bladder attachment technique resulted in a more uniform distribution of the thermal contact conductance, which was essentially constant with respect to the radial distance. The slight decrease in the measured contact conductance is believed to be due to the presence of the bolt holes in both the cold plate and the thermal test plate. The experimental results obtained from the other three locations yielded similar results for both the bolted and bladder attachment techniques; however, the edge effects complicated the data reduction process considerably.

Using a least-squares curve fit technique, an equation was obtained for each of the curves illustrated in Fig. 3. Assuming symmetry around each bolt hole and defining a unit cell as a 70 × 70 mm region with the bolt hole located in the center, these equations were integrated with respect to the area to obtain the individual bolt conductance. The results of this proce-

Table 1 Experimental thermal contact conductance values

		Integrated values[a] (W/K)	Contact conductance (W/m^2-K)
Bolted technique			
(N-m)	(in.-lb)		
0.79	7	4.19	855
1.35	12	4.83	985
1.92	17	5.58	1139
2.48	22	6.52	1331
3.04	27	7.79	1589
Bladder technique			
(kPa)	(psi)		
41.37	6	7.35	1500
62.05	9	8.92	1820
82.74	12	9.49	1937
103.42	15	9.78	1995
130.99	19	9.90	2021

[a]Least-squares curve fit integrated over 70×70-mm area.

dure are given in Table 1. The average contact conductance for each case can be calculated by dividing these numbers by the area of the unit cell.

As shown, the average contact conductance values for the bolted attachment technique ranged from 855 W/m^2-K for a bolt torque of 0.79 N-m to 1589 W/m^2-K for a bolt torque of 3.04 N-m. These values are slightly higher than those reported previously in the literature.[7-9] For the bladder attachment technique, the average contact conductance values ranged from 1500 W/m^2-K for a bladder pressure of 41.37 kPa to 2021 W/m^2-K for a bladder pressure of 130.99 kPa. It is important to note from Table 1 that a bladder pressure of approximately 41.37 kPa resulted in a thermal contact conductance approximately equal to that obtained for a bolt torque of 3.04 N-m.

Conclusions

This investigation compared the thermal contact conductance resulting from two different cold plate attachment techniques to determine which attachment technique could provide the highest contact conductance at the interface and the most uniform interface temperature profile. The experimental results indicate that the temperature distribution and the resulting contact conductance for the bolted attachment technique are highly nonuniform; whereas for a bladder attachment technique, the temperature and thermal contact conductance are reasonably constant. This difference is the result of the pressure distributions imposed by the two attachment techniques.

Although it is readily apparent that the bladder attachment technique results in a higher total contact conductance per unit bolt area than the bolted attachment technique, it should be noted that significant structural supports are required for the pressurized bladder system.

Acknowledgments

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Conjugate Mixed Convection-Conduction of Micropolar Fluids on a Moving Vertical Cylinder

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Nomenclature

B = material parameter, L^2/j
C_s = specific heat of the cylinder
j = microinertia per unit mass
K_v = vortex viscosity coefficient
k = thermal conductivity of the fluid
L = length of the cylinder
Δ = material parameter, K_v/μ
γ = spin gradient
λ = material parameter, $\gamma/(j\mu)$

I. Introduction

THE problem of conjugate convection-conduction on a continuous moving cylinder has many technical applications. Well-known examples are rolling rods drawn from a die or fibers from an orifice. Among the earlier theoretical contributions, only a few papers are concerned with the conjugate problem.[1-3] The theory of micropolar fluid, formulated by Eringen[4] is expected to explain the non-Newtonian fluid flow behavior in certain fluids such as polymeric liquid, ferro liquid, and liquid with suspension. The purpose of this study is to investigate numerically the conjugate mixed convection-con-

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