Heat Transfer Enhancement Techniques for Space Station Cold Plates

G. P. Peterson* and L. S. Fletcher†
Texas A&M University, College Station, Texas 77843

Two attachment techniques for mounting electronic equipment to Space Station cold plates were analyzed and compared using thin foils of lead, tin, aluminum, and copper to enhance the thermal contact conductance. The two techniques evaluated included a 70- × 70-mm bolted attachment technique and an attachment scheme using an inflatable bladder. The results indicate that, even in the presence of the metallic foils, the bolted technique results in large variations in the local thermal contact conductance over the surface of the cold plate, whereas the pressurized bladder yields more uniform local contact conductance values. The results indicated that the lead foil provided an enhancement factor of approximately 3, the tin foil an enhancement factor of approximately 1.5, the aluminum an enhancement factor of approximately 1.0, and the copper an enhancement factor of approximately 0.9.

Introduction

THE proposed Space Station active thermal control system consists of two major subsystems: a thermal loop to collect waste heat from the various subsystems and equipment modules and a large heat pipe radiator to dissipate the collected heat to space.[1] Within the thermal loop, several schemes for attaching the various subsystems and equipment modules to the cold plates are being considered. The present baseline design involves the use of 77, 5-mm-diam bolts. These bolts are spaced on a 70- × 70-mm matrix pattern over the surface of the 500- × 750-mm (19.6- × 29.5-in) cold plate. An alternative design utilizes a pressurized inflatable bladder. This bladder, when pressurized and placed behind the electronic equipment to be cooled, as shown in Fig. 1, insures contact between the base plate and the cold plate.

When any two metallic surfaces such as the electronic equipment base plate and the cold plate are brought together, as shown in Fig. 1, the actual area of contact is limited to a relatively few discrete points. These solid to solid contact points typically make up <5% of the apparent contact area and result from the individual surface roughnesses of the two contacting surfaces. Because of the reduction in area through which heat can be conducted, a thermal resistance, and, hence, a temperature drop, occur at the interface. This thermal resistance, or the inverse, the contact conductance, has been identified as a critical factor in the design and development of the Space Station thermal management system. As discussed in the European Space Agency (ESA) *Payload Accommodation Handbook*,[2] "the amount of heat transfer between a unit mounted to a cold plate and the working fluid within is chiefly a function of the thermal conductance or resistance of the interface."

For metallic interfaces such as those of interest here, heat can be transferred across the interface by conduction through the actual metal-to-metal contacts, by conduction through the substance in the gaps around the contacts, by radiation across the gaps, or a combination of the three. The obvious method of enhancing the thermal contact conductance is to increase the apparent contact pressure; however, this requires rigid support structures with prohibitive weights. Alternatively, the thermal contact conductance can be enhanced through the use of thermally conductive greases or thin metal foils. In both of these methods, the interstitial material, either the thermal grease or the metal foil material, flows into the gaps between the two surfaces and increases the actual contact area. As a result, significant increases in the thermal contact conductance can be obtained.

Because of the ease of application, thermal greases are the more desirable of these two techniques; however, at high temperatures and low surrounding pressures or in applications where the contact must be maintained over a long period of time, these greases can vaporize and/or migrate. Metal foils present an attractive alternative to thermal greases, but care must be exercised in the selection of combinations of materials that are both chemically stable and compatible; procedures must also be developed to insure that the foil is applied properly.

In order to determine the degree to which the thermal contact conductance between the cold plates and electronic equipment base plates can be enhanced, an experimental investigation was conducted in which four metallic foils were inserted at the interface.

Literature Review

Numerous investigations involving the use of thin metallic foils to enhance the thermal contact conductance at metallic junctions have been conducted over the past 20 years, and several comprehensive reviews of this work have been conducted.[3,4] In 1965, Koh and John[5] performed a systematic experimental investigation that studied the effect of thin metal foils of copper, aluminum, lead, and indium sandwiched between a pair of mild steel specimens. Although copper and aluminum have high thermal conductivities, the insertion of these foils resulted in a reduction in the thermal contact conductance when compared to the bare joint, whereas the lead

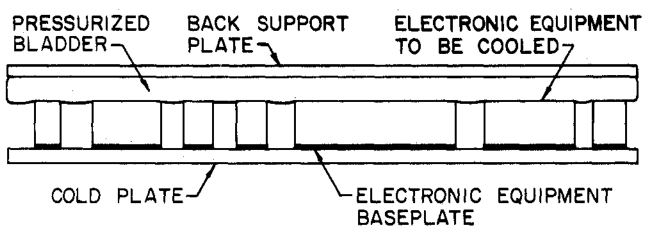

Fig. 1 Bladder attachment technique.

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*Professor of Mechanical Engineering, Department of Mechanical Engineering. Associate Fellow AIAA.

†Thomas A. Dietz Professor of Mechanical Engineering, Department of Mechanical Engineering. Fellow AIAA.

and indium foils (softer but with lower thermal conductivity) increased the conductance. As a result, it was concluded that foil hardness was of greater significance in determining the conductance than the foil thermal conductivity. In a second set of experiments,[5] the effect of foil thickness was investigated. The results of this investigation indicated that an optimum thickness existed. For surface roughness on the order of 4 μm rms, the optimum thickness of the foil was found to be 25 μm. Using foils of this thickness, the thermal contact conductance was three times the value for the bare metallic joint, whereas foils >100 μm thick, produced no measurable improvement in the contact conductance.

Cunningham[6] compared the contact conductance of both smooth and rough, bare aluminum/aluminum and magnesium/magnesium junctions, with junctions in which indium foils were present. The results of this work indicated that the use of the indium foil increased the thermal contact conductance significantly. Using aluminum specimens and indium foil, it was found that, as the surface roughness of the metal surfaces increased, so did the thermal contact conductance of the interface, indicating that a near optimum foil thickness had been used.

Mal'kov and Dobashin[7] performed a study of the thermal contact conductance of stainless steel interfaces with thin copper foils placed between metallic interfaces composed of two different stainless steel and one molybdenum alloy samples. In this investigation, it was observed that insertion of the copper foil increased the value of the contact conductance for the copper foil by a factor of 3–5 for contact pressures of 2.5×10^5–2.9×10^7 N/m^2. It was concluded that variations in the mean interface temperature had little or no effect on the contact resistance, either with or without the foil. Molgaard and Smeltzer[8] conducted a similar investigation involving gold foils over a temperature range of 50–300°C and a pressure range of 2.5×10^7–9×10^7 N/m^2. This work supports the previous conclusions of Moligåard and Smeltzer[8] and Mal'kov and Dobashin,[7] as does the investigation performed by Moore et al.[9]

In a more recent investigation, Yovanovich[10] studied the effect of several soft foils between Armco iron specimens and suggested that the enhancement performance of various foils could be ranked according to the ratio of the thermal conductivity to the hardness of the foil. It was demonstrated empirically that, the higher the value of this parameter, the greater the improvement in the contact conductance over a bare joint. Experimental evidence was presented that supported the conclusion that an optimum foil thickness exists and that it varies from different foils. Using this technique to determine the optimum foil thickness, Peterson and Fletcher[11] determined the effect of variations in the roughness of the contacting surfaces. The results indicated that, in addition to an optimum thickness, an optimum roughness exists. By coupling the optimum thickness and optimum roughness, the thermal contact conductance was improved by as much as a factor of 7 over the bare interface.

The results of these and other investigations[12–14] can be summarized as follows:

1) Foil hardness is of greater significance in determining the thermal contact conductance than foil thermal conductivity.

2) An optimum foil thickness exists and this thickness provides the maximum enhancement of the interface thermal conductance.

3) When an optimum foil thickness is used, as the surface roughness of the metal surfaces increases, so does the thermal contact conductance of the interface.

4) Small variations in the mean interface temperature, 20–30°C, have little or no effect on the magnitude of the thermal contact conductance.

5) The enhancement performance of various foils can be ranked either by hardness or by the ratio of the thermal conductivity to the hardness of the foil material.

Several investigations have been conducted on cold plates similar to those of interest here. Schwartz[15] conducted an investigation to determine the thermal contact conductance at the interface of a flat test plate, 10 mm thick and stiffened at the edges to simulate a Spacelab computer mounting plate. Standard 5-mm-diameter bolts tightened to a torque of 2 and 3.2 N-m and located on a 70- × 70-mm bolt matrix were used to fasten the test plate and the cold plate together. Five different situations were evaluated: bare metallic surfaces in contact; bare metallic surface with a thin layer of Cho-Therm 1661, a silicone foil; bare metallic surfaces with a thin layer of Sigraflex, a graphite film; a thin layer of RTV-11, a cure in place elastomer; and bare metallic surfaces with a fill gas of Nitrogen. The results indicated that significant improvements in the thermal contact conductance in vacuum environments could be achieved with filler materials. Use of Cho-Therm 1661, Sigraflex, and RTV-11 resulted in increases of 400, 530, and 610%, respectively, at a torque of 2 N-m. Similar increases were observed for torques of 3.2 N-m.

In a separate investigation,[16] the maximum thermal contact conductance for a 500- × 750-mm coldplate with bare contacting surfaces was 1136 W/m^2-K. Since the computed equivalent gap thickness was almost an order of magnitude larger than the roughness of the contacting surfaces, the authors concluded that the surface flatness deviation was more important than the surface roughness in determining the thermal conductance values.

In a recent study by Peterson et al.,[17] the two attachment techniques discussed previously, one utilizing a 70- × 70-mm bolt pattern and one using an inflatable bladder, were evaluated and compared experimentally for surfaces without enhancement materials. The results indicated that the temperature distribution and the resulting contact conductance for the bolted attachment technique were highly nonuniform; whereas for the bladder attachment technique, the temperature and thermal contact conductance were almost constant. It was readily apparent that, even for relatively low bladder pressures (42 kPa), the bladder attachment technique resulted in a higher total contact conductance per unit bolt area than the bolted attachment technique.

Test Program

The objective of this investigation was to determine the extent to which the thermal contact conductance at the interface of a 500- × 750-mm single-phase liquid cold plate and a thermal test plate could be enhanced. Four different foils were evaluated for two different attachment techniques. Using a standard flight certified cold plate provided by Marshall Space Flight Center and a specially developed thermal test plate, measurements of the thermal contact conductance between the cold plate and test plate were made for the bare interface and for interfaces into which thin layers of lead, tin, aluminum, or copper were inserted. The two attachment techniques investigated were 1) the standard attachment technique[2] of 77, 5-mm-diam bolts located on 70-mm centers, and 2) an inflatable bladder (Fig. 1).

The experimental test facility and procedure are similar to that previously described by Peterson et al.[17] The heat source was simulated by a 500- × 750-mm heated aluminum 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 on the cold plate. Following the tapping of the bolt holes, the aluminum plate was machined to an overall flatness deviation of 0.1 mm and an rms surface roughness of 0.0032 mm.

To determine the thermal contact conductance and temperature distribution at the contact surface, a centrally located bolt hole was selected for instrumentation. A series of 16 thermocouple wells were drilled (number 56 drill) from the backside of the thermal test plate at four different radii from the bolt centerline, as illustrated in Fig. 2. At each radial distance, four different depth wells were drilled, 5.08, 101.6,

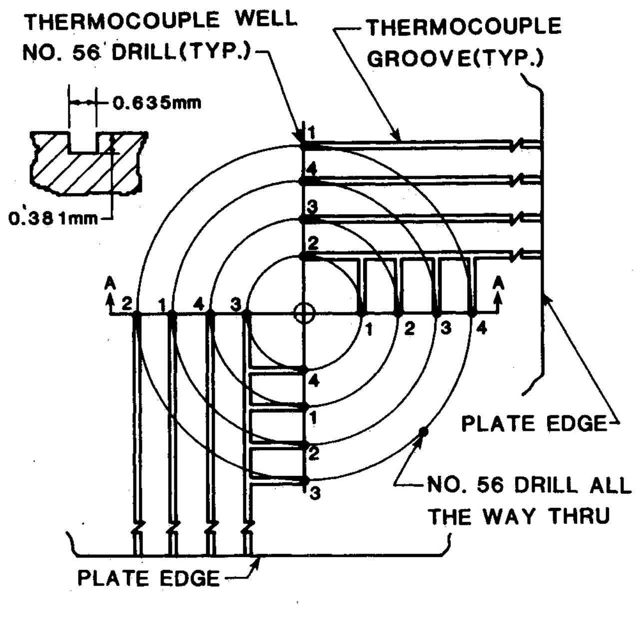

Fig. 2 Thermocouple well locations.

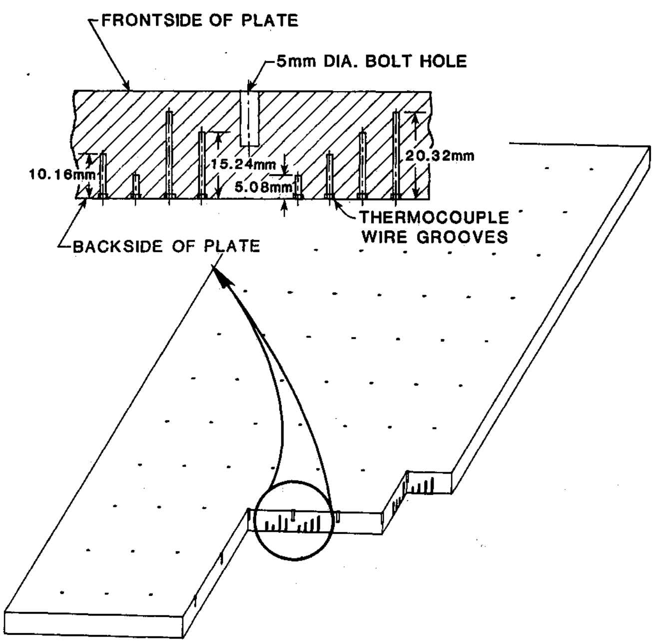

Fig. 3 Thermocouple well depth.

15.24, and 20.32 mm deep, as illustrated in Fig. 3. At each of the four radial distances, a thermocouple well was drilled all the way through. In each of these thermocouple wells, a Chromel-Alumel thermocouple (AWG-36) was inserted and packed in place with a thermally conductive epoxy. On the back side of the thermal test plate, small channels, 0.635 × 0.381 mm, were machined through, which the thermocouple wires were routed. These channels were filled with a thermally conductive epoxy in order to reduce any distortion of the temperature field. With this combination of thermocouples, the plate surface temperatures could be obtained at each radial distance and compared with the measured values. Using a one-dimensional technique to approximate the local thermal contact conductance at each radial location, a three-dimensional temperature profile around each bolt could be obtained and an overall thermal map of the test plate developed. Previous investigations[17] have indicated that, although the flow of heat is three-dimensional in nature, a one-dimensional approximation is acceptable provided that the variation is small and the thermocouples are in close proximity to one another.

Once the thermal test plate had been instrumented, a series of four silicone rubber resistance heaters were installed on the backside of the thermal test plate. Each heater was 125 mm wide and 750 mm long and had a rated capacity of 2250 W. These heaters were attached to the backside of the thermal test plate with a uniform coating of silicone adhesive. The power to each heater was measured independently by both a digital wattmeter and by measuring the voltage and current supplied by the individual power supplies.

The temperature of the cold plate was controlled by chilled water flowing through a constant temperature circulating bath at a flow rate of 0.13 kg/s (2 gpm). The temperature at the inlet and outlet of the cold plate as well as the temperatures of all of the thermocouples located in the thermal test plate were continuously monitored with an automated data acquisition system.

The inflatable bladder used in this investigation was constructed by the International Latex Corporation Dover of Frederica, Delaware. It was proof tested to 0.2071 MPa (30 psi), but was operated at a maximum pressure of 0.138 MPa (20 psi) for the present investigation. The bladder was equipped with a standard Schrader valve, and pressure in the bladder was supplied by a hand pump. The pressure was measured by a Magnehelic Series 200, 0-136 kPa pressure gauge. The accuracy of the gauge was specified by Magnehelic to be ± 0.4 psi.

Because use of the bladder required a method by which the cold plate and aluminum plate could be held in contact, a test fixture was constructed. This test fixture consisted of a wooden frame supported by a series of six, 76-mm-wide pieces of steel channel, 0.64 m in length, on both the top and bottom. These pieces of channel were held together by 12 19-mm-diam bolts. The thermal test plate, which was insulated from the container with 9.5-mm Teflon standoffs, along with the cold plate was placed inside of the wooden frame. Although not necessary for the bolted configuration, both the bladder and the bolted configuration were evaluated in this test fixture to eliminate any variations in the experimental results due to heat transferred through the test fixture. Also it should be noted that, although not necessary, the bolt holes were present for both test configurations. All tests were conducted using a carefully cleaned, dry interface in an atmospheric environment.

Test Procedure

The experimental tests were performed in such a manner so as to compare the relationship between the thermal contact conductance and the bolt torque, which was varied from 0.79 N-m (7 in.-lb) to 3.04 N-m (27 in.-lb), and the pressure in the bladder, which was varied 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 prestressed to 130 kPa, the highest pressure experienced during the course of the tests. This prestressing has been shown in previous investigations to be necessary to reduce the hysteresis that occurs as a result of repeated loadings. Because this hysteresis is the result of plastic deformation occurring during the initial loading, any tests conducted after the plastic deformation occurred would presumably have lower contact conductances. By preloading, the effect of repeated loadings could be reduced but not eliminated altogether.

Testing of the bolted configuration was performed first. Prior to testing, the surfaces of both the cold plate and the thermal test plate were wiped clean with acetone and allowed to dry. The test plates were then placed in contact, and the 5-mm bolts, each with a 10-mm washer, were installed to the required torque. The bolts were tightened in a predetermined pattern starting from the center and working outwards. Once all of the bolts had been tightened, the test bolt was loosened and retightened to the required torque. Three bolt torques were evaluated: 0.79 N-m (7 in.-lb), 1.92 N-m (17 in.-lb), and 3.04 N-m (27 in.-lb). At each bolt torque, the system was allowed to reach steady state (about 6 h) prior to recording the data. Once testing of the bolted configuration had been completed for the bare interface, the test setup was disassembled, cleaned, and reassembled for testing of the bladder configuration. The internal pressures evaluated for the bladder configuration were 41.68 kPa (6 psi), 82.74 kPa (12 psi), and 130.99 kPa (19 psi). Again, the system was allowed to reach steady state at each point before the recording of any of the temperature data. Once the baseline values had been established for the bare interface, a similar procedure was followed for each of the four foils shown in Table 1. An average interface temperature of 300 K ± 3 K was maintained for all tests.

Using the extrapolated surface temperature of the thermal test plate and the average temperature of the fluid entering and leaving the cold plate, an estimation of the thermal contact conductance at the interface, expressed as

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

could be computed at four different radii for each bolt. 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 the uncertainty technique presented by Kline and McClintock,[18] this experimental uncertainty was estimated to be ±8%.

Results and Discussion

The local thermal contact conductance was calculated using the temperature gradient in the aluminum plate as measured by the 16 thermocouples. Because of the way the thermocouples were located, it was possible to measure the temperature at four different depths for each of four different radial distances. 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 of the temperature, as measured by the thermocouple located at the surface of the test plate. This extrapolation required that the temperature distribution around the bolt be symmetrical.

The surface temperature of the cold plate was assumed to be the average temperature of the inlet and outlet working fluid. The ΔT across the interface was then calculated by subtracting the average cold plate temperature from the extrapolated aluminum plate surface temperature. Although not accounting for variations in temperature over the surface of the cold plate, this procedure was consistent with previous investigations and was used for comparative purposes.

The heat flux through the test plate was determined by monitoring the flow rate, which was typically 0.13 kg/s (2 gpm), and the temperature increase of the coolant across the cold plate. Using these measured values, the sensible heat of the coolant was calculated and compared with the input power, which was generally 6–8% higher due to external convection losses. Typical total heat flow values of between 2000 and 2700 W were used.

Figure 4 illustrates the temperature distribution for 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 distribution, whereas the bolted technique results in an increase in the test plate surface temperature. Because of the limited number of extrapolated temperatures available, the surface temperature appears to decrease sharply at R = O, however, it may be that this profile actually levels off to form an S-shaped curve. The temperature distributions for the other pressures and torques were similar in shape and magnitude.

The thermal contact conductance was determined, as shown in Eq. (1), by computing the ratio of the computed heat flux to the temperature drop across the interface. The values listed in Table 2 were assumed to represent the local thermal contact conductance. Figures 5a–c illustrate the variation of the thermal contact conductance for the bolted technique for torques at 0.79 N-m, 1.92 N-m, and 3.04 N-m, respectively, as a function of the distance from the bolt centerline. As shown,

Table 1 Properties of the test foils

Foil type	Thickness, μm	Hardness, kg/mm²	Conduct, W/m-K	Modulus, kPa
Aluminum	30	27.0	204	68
Copper	40	80.0	384	110
Lead	75	4.0	35	18
Tin	100	5.3	60	41

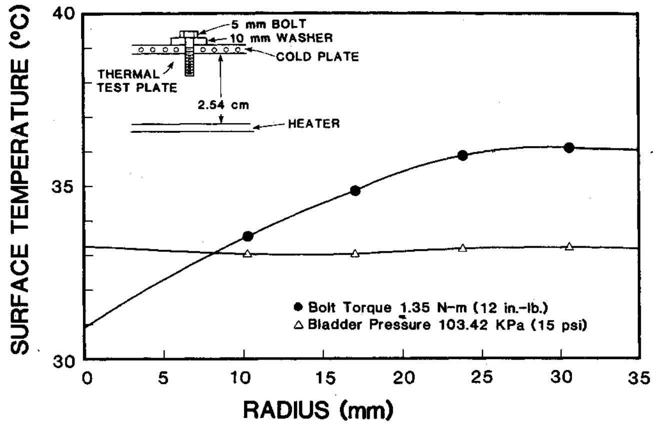

Fig. 4 Surface temperature variation for the two attachment techniques.

Table 2 Thermal contact conductance data[a]

Radius, mm	Bare	Lead	Tin	Aluminum	Copper
Bladder attachment technique					
11.76 (kPa)					
41.68	1579	4611	2213	1741	1310
82.74	1945	5621	2897	2132	1910
130.99	2086	6015	3026	2296	2101
19.51					
41.68	1520	4483	2180	1673	1473
82.74	1942	5546	2863	2114	1941
130.99	2084	5939	5001	2261	2089
27.25					
41.68	1443	4136	2146	1641	1471
82.74	1931	5536	2803	2097	1921
130.99	1999	5893	2903	2203	2059
35.00					
41.68	1460	4021	2130	1603	1446
82.74	1930	5481	2796	2131	1906
130.99	1913	5882	2920	2194	2051
Bolted attachment technique					
11.76 (N-m)					
0.79	989	2897	1480	1003	931
1.92	1324	3879	1962	1459	1301
3.04	2221	6603	3303	2493	2211
19.51					
0.79	646	2214	951	701	621
1.92	838	2395	1221	920	816
3.04	1126	3541	1603	1251	1103
27.21					
0.79	502	1879	742	536	501
1.92	602	2131	836	659	603
3.04	756	2546	1126	829	714
35.00					
0.79	459	1513	693	741	421
1.92	559	1841	818	602	546
3.04	631	2113	926	713	641

[a]All values in W/m²-K.

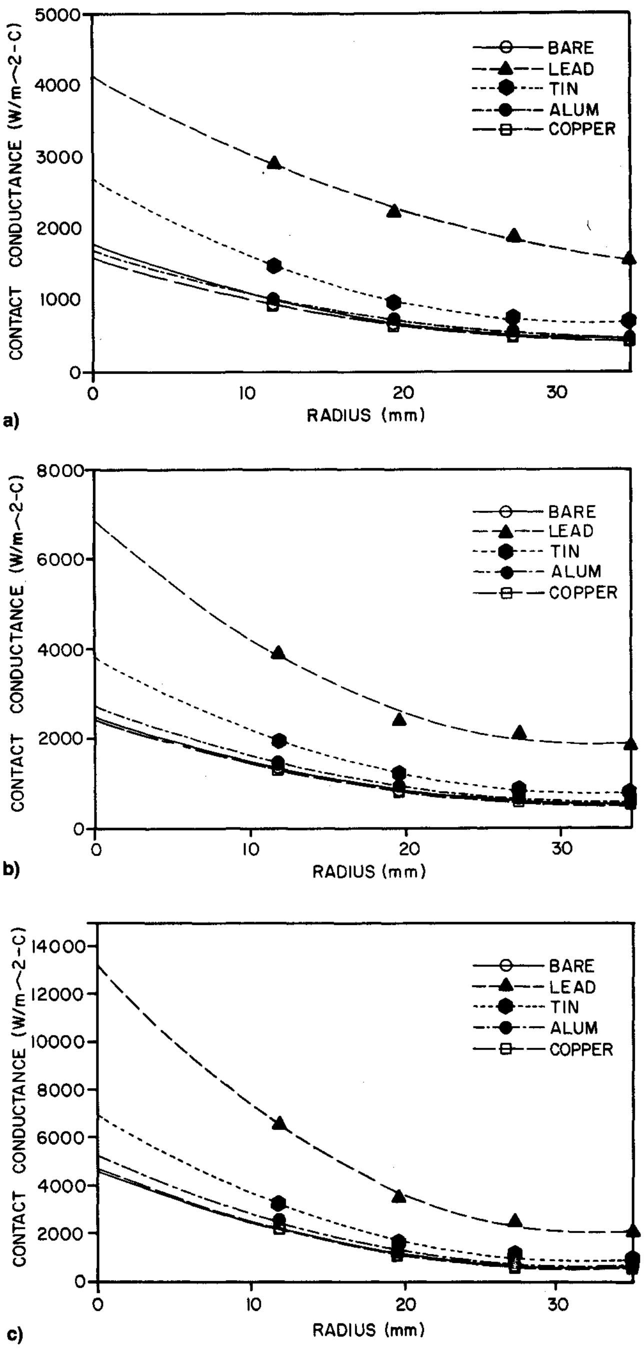

Fig. 5 Thermal contact conductance variation for the bolted attachment technique: a) 0.79 N-m; b) 1.92 N-m; c) 3.04 N-m.

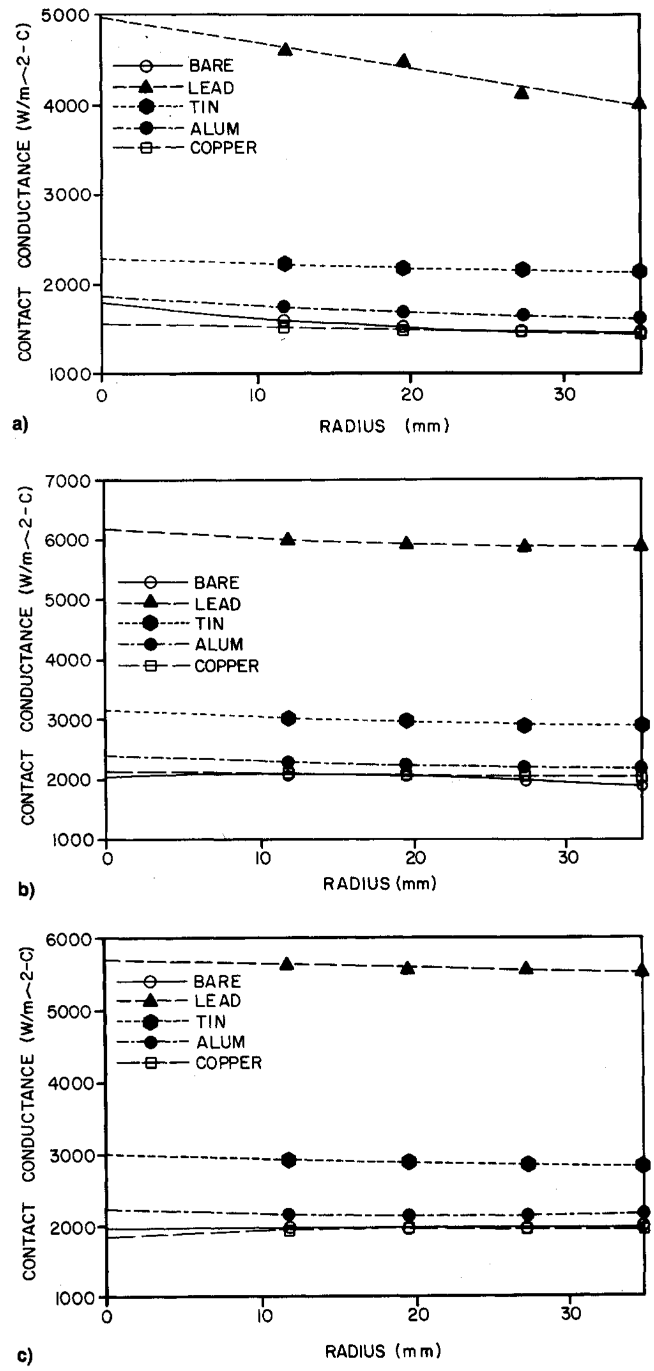

Fig. 6 Thermal contact conductance variation for the bladder attachment technique: a) 41.68 kPa; b) 82.74 kPa; c) 130.99 kPa.

a rapid decrease in the thermal contact conductance of the interface occurs due to the variation in the contact pressures. These results support the conclusions previously presented by other investigators[17,19,20] that the pressure distribution in a bolted joint is a parabolic function of the distance from the bolt centerline.

Figures 6a–c illustrate a similar comparison for the bladder configuration for pressures of 41.68, 82.74, and 130.99 kPa. As shown, this attachment technique results in a more uniform distribution of the thermal contact conductance, which is essentially constant with respect to the radial distance from any bolt hole.

Using a least squares curve fitting technique, an equation was obtained for each of the curves illustrated in Figs. 5 and 6. Assuming symmetry around each bolt hole and defining a unit cell as a 70- × 70-mm area with the bolt located in the center, these expressions were integrated with respect to the area to obtain the individual conductance values.

As shown in Table 2, local contact conductance values for the bolted attachment technique range from 459 W/m^2-K for a bolt torque of 0.79 N-m and a radius of 35 mm for the bare surface to 6603 W/m^2-K for a bolt torque of 11.76 N-m using a lead foil. For the bladder attachment technique, the local contact conductance values ranged from 1443 W/m^2-K for a bladder pressure of 41.37 kPa to 6015 W/m^2-K for a bladder pressure of 130.99 kPa.

The average contact conductance for each case was calculated by dividing the values shown in Table 2 by the area of the unit cell. The results of this procedure are listed in Table 3. These integrated values are slightly higher than those reported previously for cold plates of the size tested here but slightly lower than results from 2.54-cm-diam test specimens.[17] The latter is most likely due to the increased difficulty of maintaining a high quality, uniform surface finish over a surface area of 0.375 m^2. Overall, the integrated value indi-

Table 3 Integrated contact conductance values

		Contact Conductance, W/m²-K				
		Bare	Lead	Tin	Alum	Copper
		Bolt technique				
0.79 N-m	7 in.-lb	603	1993	899	629	573
1.92 N-m	17 in.-lb	764	2396	1110	835	751
3.04 N-m	27 in.-lb	1050	3323	1544	1173	1037
		Bladder				
41.37 kPa	6 psi	1500	4049	2167	1664	1475
82.74 kPa	12 psi	1937	5534	2829	2119	1917
130.99 kPa	19 psi	2021	5920	2950	2227	2069

cated that the lead foil provided an enhancement factor of approximately 3, the tin foil an enhancement factor of approximately 1.5, the aluminum an enhancement factor of approximately 1.0, and the copper an enhancement factor of approximately 0.9. As in a previous experiment, a bladder pressure of 41.68 kPa resulted in a thermal contact conductance approximately equal to that obtained for a bolt torque of 3.04 N-m.

Conclusions

It was the goal of this investigation to determine the extent to which the thermal contact conductance for two different attachment techniques could be enhanced through the use of thin metal foils. The experimental results indicate that the temperature distribution and the resulting contact conductance for the bolted attachment technique are highly nonuniform; whereas for the bladder attachment technique, the temperature and thermal contact conductance are almost constant. In addition, the integrated values of the thermal contact conductance indicated that the lead foil provided an enhancement factor of approximately 3, the tin foil an enhancement factor of approximately 1.5, the aluminum an enhancement factor of approximately 1.0, and the copper an enhancement factor of approximately 0.9 for the conditions evaluated in this investigation.

It should be emphasized that, although the bladder attachment technique results in a higher total contact conductance per bolt area than does the bolted technique, significant structural supports are required for use of a pressurized bladder.

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