# Thermal Contact Conductance of Composite Cylinders: An Experimental Study

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**When two concentric cylinders are assembled with a slight interference fit and heat flows radially through such a joint, the contact at the interface may improve or deteriorate depending on several factors, including the direction of heat flow, the magnitude of the heat flux, and the thermomechanical properties of the materials of the two cylinders. This paper deals with the design and construction of an apparatus to measure the thermal contact conductance of cylindrical joints as a function of the heat flux. Results are presented for stainless steel → stainless steel and stainless steel → aluminum composite cylinders, for radially outward heat flow. It is noted that the results obtained are in reasonable agreement with those predicted by theory.**

## Nomenclature

$a$ = outer radius of outer cylinder
$b$ = interface radius
$c$ = inner radius of inner cylinder
$h$ = thermal contact conductance
$k$ = thermal conductivity
$L$ = length of cylinders
$q$ = rate of heat flow
$T$ = temperature
$\delta$ = mean gap thickness

*Subscripts*

$C$ = cold fluid
eff = effective
$f$ = gas
$H$ = hot fluid
$i$ = inner cylinder
$o$ = outer cylinder

## Introduction

THE existence of thermal contact resistance at a joint formed by two solids has long been appreciated. Over the past 40 years, extensive theoretical and experimental investigations have been conducted on heat flow across *flat* joints. The results have been summarized in various reviews and bibliographies.[1-6]

The equally important aspect of heat flow across cylindrical joints, however, has received comparatively little attention. In fact, over the decade spanning 1970 to 1980, there were only about a half-dozen works dealing specifically with cylindrical joints compared with more than 150 studies on flat joints. This situation is clearly undesirable since cylindrical joints are at least as commonplace as flat joints in engineering practice. Examples include nuclear reactor fuel elements, bimetallic finned tubes in which the fins are extruded on the core tube, and shrunk fit cylinders.

A review of previous investigations concerning heat flow across cylindrical joints indicates that such works may be classified into two broad categories:

1) *Experimental*. This category includes the works listed in Refs. 7–17. Of these, the works of Dudgeon and Prior[7] and Cohen et al.[10] were directed toward the specific application of nuclear fuel element heat transfer. Gardner and Carnavos[9] dealt with specific types of heat exchanger tubing. Brutto et al.[8] tested long coaxial tubes in vacuum only. Williams and Madhusudana[12] attempted direct measurement of contact conductance of stainless steel cylinders assembled with different degrees of initial fit.

Hsu and Tam[13] tested two pairs of aluminum alloy (2011-T351) and stainless steel (type 304) cylinders in air. They pointed out that their studies were rather exploratory and indicated the need for further experimental work on composite cylinders.

Experimental studies on the contact conductance of plate finned tube heat exchangers have been reported by Ernest et al.,[14] Sheffield et al.,[15,16] and Wood et al.[17] These papers resulted from research projects ASHRAE RP-295 and RP-446 sponsored by the American Society of Heating, Refrigerating, and Air-Conditioning Engineers with the specific purpose of investigating the interface heat-transfer characteristics of this type of heat exchangers.

2) *Theoretical*. In this category could be listed the works of Williams and Madhusudana,[12] Hsu and Tam,[13] Wang and Nowak,[18] and Madhusudana.[19,20] Of these, Refs. 12, 18, and 19 deal with joints in vacuum only.

Barber[21,22] presented theoretical studies on the heat-transfer characteristics of prestressed (shrunk-fit) duplex tubes. In these studies, it is considered that the thermal expansion of the tube changes the value of the pressure or gap-dependent thermal contact resistance. The surface characteristics that affect the resistance, however, are not identified separately.

Theoretical models for predicting thermal contact resistance in compound cylinders and finned tubes have also been presented by Lemczyk and Yovanovich.[23] The results for compound cylinders were presented in a form to facilitate comparison with the particular experimental work of Hsu and Tam.[13] The subject of bond resistance in high finned tubes has also been recently reviewed by Taborek[24] who notes the lack of progress in the standardization of test procedures for such tubes.

It may also be noted that the different problem of cylinders contacting planes has been addressed by Yovanovich[25] and McGee et al.[26]

This brief review of previous work reveals the need for further work on heat flow across cylindrical joints. Past

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**Table 1 Material properties**

| Property | Stainless steel 303 | Aluminum alloy 2011 | Air |
|---|---|---|---|
| Modulus of elasticity, MPa $\times 10^3$ | 200 | 70 | |
| Hardness, MPa | 3800 | 1400 | |
| Thermal conductivity, W/m-K | 16.5 | 142 | 0.0298 |
| Mean free path, $10^{-6}$ m | | | 0.064 |
| Poisson's ratio | 0.29 | 0.33 | |
| Coefficient of thermal expansion, $K^{-1} \times 10^{-6}$ | 17.3 | 23 | |

experimental research, with one or two exceptions, appears to have been carried out with specific applications in mind. Thus, there is a particular need for basic experimental research into heat flow across cylindrical joints formed of both similar and dissimilar materials. The work described in the present paper attempts to remedy this situation to some extent.

## Problem Statement

One significant conclusion from the results of investigations on flat joints is that, for a given pair of contacting surfaces, the thermal contact conductance varies as a simple power of the contact pressure. For most joints, this pressure is either explicitly known or can be estimated simply. It is usually assumed that the pressure is an independent variable for *flat* joints.

For cylindrical joints, however, the contact pressure depends on several factors, including the following:

1) initial degree of fit (clearance or interference)

2) differential expansion of the cylinders due to a) temperature gradients caused by heat flow and b) a temperature drop, $\Delta T$ at the interface due to imperfect contact (Fig. 1).

It may be noted that, for a given heat flux, the differential expansion depends on the following:

1) thermomechanical properties of the cylinder materials

2) interface medium (i.e., fluid such as air filling the interstitial gaps between the actual contact spots)

3) geometry of the cylinders

4) surface characteristics at the interface.

From the preceding, it may be appreciated that, for a given pair of cylinders operating in a given environment, the contact pressure depends on the heat flux only. Since the thermal contact conductance is a simple function of the contact pressure for a given pair of surfaces, a relationship can be derived between the heat flux and the conductance.[20] Thus, for cylindrical joints, the heat flux, rather than the contact pressure, is the appropriate independent variable. The temperature drop over the radial thickness of one of the cylinders is a measure of the heat flux, and sometimes it is convenient to use the temperature drop across the inner cylinder as the independent variable.

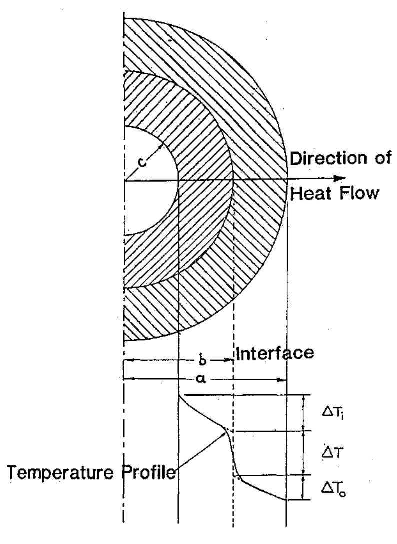


**Fig. 1 Heat conduction in a composite cylinder.**

It may be noted that, traditionally, the conductance is plotted as a function of the contact pressure. This is justifiable for *flat* joints where the contact pressure could be measured directly and is, quite rightly, considered as the independent variable. For cylindrical joints, however, the contact pressure is difficult, if not impossible, to measure directly; it can only be *estimated* and, therefore, cannot be chosen as the independent variable in the presentation of experimental results.

Experimentally, the contact conductance is determined as the ratio of the heat flux to the temperature drop $\Delta T$ at the interface.

**Table 2 Specimen details**

| | |
|---|---|
| Outside diameter, $2a$, mm | 50 |
| Interface diameter, $2b$, mm | 34 |
| Inside diameter, $2c$, *mm* | 25 |
| Length, $L$, mm | 150 |
| Surface roughness (range), $\mu$m | 1–2 |

## Scope of Present Work

This paper describes the design and construction of an experimental apparatus for measuring the thermal contact conductance of cylindrical joints in an environment of air at ambient pressure. It reports and discusses the test results for two typical joints, one of similar materials and the other of dissimilar materials.

## Design Considerations

### Test Specimens

Materials for the test specimens were selected to meet the following criteria. They should

1) be materials commonly used in heat-transfer engineering,

2) have good machinability so that accurate specimens could be manufactured,

3) and cover a wide range of thermomechanical properties.

Aluminum alloy 2011 and 18 : 8 stainless steel (type 303) appeared to meet most of the requirements and were used in all of the experiments. The material properties, together with those of air, are listed in Table 1.

Dimensions of the specimens had to satisfy the following considerations:

1) The specimens could be made from readily available bar stock.

2) The diameters should be large enough so that the bore could facilitate the installation of inner-surface thermocouples and allow adequate flow of the heating fluid.

3) The diameters should be small enough to limit the overall size of the test rig both from the point of view of cost and ease of handling.

4) The length-to-diameter ratio should be high enough to minimize end effects; however, specimens that were too long had to be avoided in the interest of accuracy of manufacture—to minimize the effects of unavoidable tapering, etc., during the machining process.

The nominal dimensions of the specimens, arrived from the preceding considerations, are listed in Table 2. Also listed in the table is the range of surface roughness (CLA) values of the tested specimens. Since a light interference fit was required initially, the tolerance specified on the interface diameter of all of the specimens corresponded to American National Standards Institute locational interference fit, class LNI.[27]

For the specimens under test, the actual measured interference on the diameter ranged from +0.06 to −0.06 mm, i.e., about ±2 parts in 1000. Because of manufacturing limitations, as anticipated earlier in this section, higher accuracy could not be obtained.

#### Method of Heat Transfer to and from the Specimens

It is clearly essential that the specimens be heated uniformly over the surface of the bore and cooled uniformly over the outer periphery of the larger cylinder. This is ideally achieved by arranging the surfaces to be in direct contact with fluids at the appropriate temperatures. Indeed, previous experience had shown that semiconductor heaters, although suitable in many respects, did not provide uniform heating. Therefore, it was decided to use hot water circulated from a constant temperature bath (Thermomix by Braun) to provide the required uniform heating at the inner surface. The specimens were cooled by means of cold water supplied from a constant head overhead tank. Both flow rates could be controlled by means of valves and measured either by means of float-type meters or directly. Since the outer surface temperature of the larger cylinder depends on the cold water temperature, the heat flux could be varied simply by altering the hot water temperature and, therefore, the overall temperature drop across the cylinders. Of course, the heat flux could also be varied at any given temperature setting by altering the flow rates.

#### Test Chamber and End Flanges

The test chamber should:

1) Locate the specimens
2) Provide connections to the hot and cold streams
3) Provide an annular space for the circulation of cold water
4) Provide a facility for drawing out the thermocouple cables associated with the temperature measurements of the cylinders

In the final design, the fittings for the hot water line were located centrally on the end flanges, whereas the cold stream was connected to the sides of the chamber. The flanges have grooves to locate the O-rings to prevent leakage from the chamber to the outside and to prevent intermixing of the hot and cold streams. One end flange also located the stainless steel tubes through which the thermocouple cables were drawn out for connecting to the measurement system.

To avoid alternative heat flow paths and any electrochemical action that would exist if a metal chamber and connections were used, a PVC chamber was used together with cast nylon end flanges and reinforced nylon (Nylon 66) fittings.

The chamber could be mounted horizontally or vertically. However, it was soon realized that, in the horizontal configuration, there is no guarantee that the thermocouples in the hot and cold lines are completely submerged in the fluid, especially at low flow rates. Consequently, a vertical configuration was adopted in all of the tests. The test chamber and connections are shown in Fig. 2.

#### Measurement of Temperature

It was decided to measure the temperatures on the free surfaces of the cylinders, i.e., at radii $a$ and $c$, rather than within the cylinders because the location of thermocouples inside the cylinder walls would 1) involve drilling deep holes of very small diameter and 2) result in disturbing the heat flow paths.

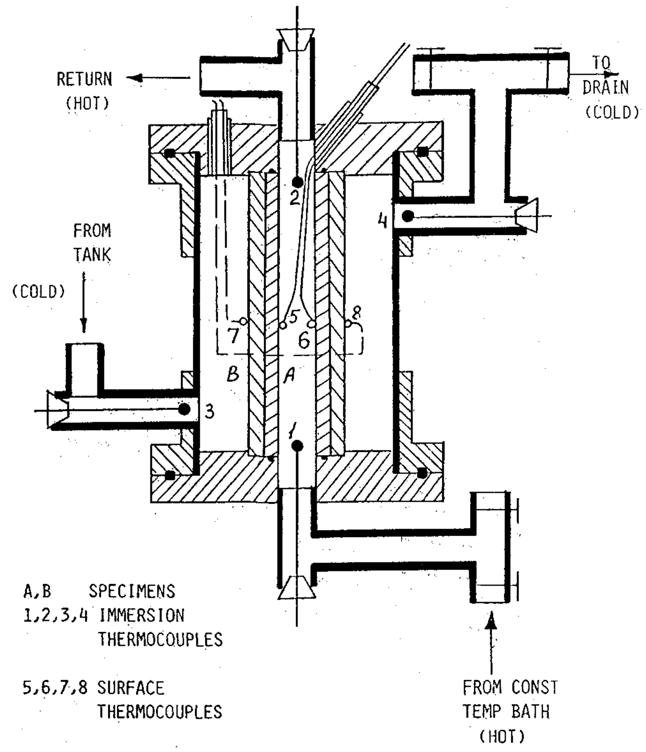


**Fig. 2 Test chamber and connections.**

Two thermocouples were used at each radius so that the average value could be used. In addition, thermocouples were located at the inlet and outlet of the hot and cold streams, resulting in a total of eight thermocouples.

In the early part of the test program, patch-type stick-on thermocouples were used for the surface temperature measurements. But these had a tendency to get detached during the tests. Also, the installation of these on the bore surface proved to be difficult. Hence, these thermocouples were replaced by 0.2-mm PTFE-insulated thermocouples cemented to the surface using DEVCON aluminum-filled epoxy mixed with an equal amount of hardener. In the preliminary tests, flexible thermocouples were also used to measure the temperatures of the hot and cold lines. However, the exact location of the thermocouple bead was difficult to ascertain with these thermocouples. Eventually, therefore, mineral-insulated immersion-type thermocouples of 1.5 mm o.d. (supplied by Pyrotenax) were used together with mounting glands. With these types of thermocouples, the exact location of temperature measurement could be easily determined and controlled.

All thermocouples used were of K-type (chromel-alumel). A digital data acquisition system incorporating two thermocouple amplifiers, a 12-bit analog-to-digital converter, and an Apple IIe computer was used for data reduction and analysis.

## Experimental Details

Before assembly, the diameters of the test cylinders were measured in three locations—at either end and at midlength. The three measurements generally agreed to better than 0.02 mm, indicating the accuracy of manufacture. The surface roughness was measured in a Talysurf surface analyzer for both contacting surfaces.

After assembling the cylinders, the four thermocouples were attached—two on both the inner and outer surfaces of the assembly. All thermocouples were located at midlength of the cylinders. The four thermocouples as well as the end surfaces

**Table 3 Ranges of test parameters**

| Parameter | Range | |
|---|---|---|
| | SS→SS | SS→AL |
| 1) Temperature drop in the inner cylinder, $\Delta T_i$, °C | 2–6 | 2–6 |
| 2) Temperature drop in the outer cylinder, $\Delta T_o$, °C | 4–10 | 3–7 |
| 3) Additional temperature drop at the interface, $\Delta T$, °C | 4–7 | 2.5–6.5 |
| 4) Temperature of cold fluid, $T_c$, °C | 15–18 | 15–18 |
| 5) Temperature of hot fluid, $T_H$, °C | 50–80 | 50–80 |
| 6) Heat flow rate, $q$ | | |
| at 0.0671 $s^{-1}$, W | 120–300 | 145–320 |
| at 0.0451 $s^{-1}$, W | 118–250 | 108–245 |
| 7) Inside surface temperature, °C | 35–60 | 35–60 |
| 8) Outside surface temperature, °C | 25–38 | 25–38 |

**Table 4 Test results for stainless steel→stainless steel joint**

| $T_H$, °C | $q$, W | $\Delta T_{tot}$, K | $(\Delta T_i + \Delta T_o)$,[a] K | $\Delta T$, K | $\Delta T_i$, K | $h$,[b] W/m²-K |
|---|---|---|---|---|---|---|
| | | *Flow rate of hot water = Flow rate of cold water = 0.067 liters/s* | | | | |
| 50 | 122 | 9.6 | 5.3 | 4.3 | 2.4 | 1775 |
| 58 | 175 | 12.1 | 7.6 | 4.5 | 3.4 | 2431 |
| 66 | 225 | 14.7 | 9.8 | 4.9 | 4.3 | 2869 |
| 74 | 288 | 17.4 | 12.5 | 4.9 | 5.6 | 3675 |
| 82 | 305 | 19.7 | 13.3 | 6.4 | 5.9 | 2975 |
| | | | *Repeat tests* | | | |
| 50 | 157 | 11.9 | 6.8 | 5.1 | 3.0 | 1925 |
| 58 | 190 | 14.9 | 8.3 | 6.6 | 3.7 | 1800 |
| 66 | 227 | 17.8 | 9.9 | 7.9 | 4.4 | 1794 |
| 74 | 255 | 17.1 | 11.1 | 6.0 | 4.9 | 2656 |
| 82 | 281 | 19.9 | 12.2 | 7.2 | 5.4 | 2438 |
| | | *Flow rate of hot water = Flow rate of cold water = 0.045 liters/s* | | | | |
| 50 | 118 | 9.2 | 5.1 | 4.1 | 2.3 | 1800 |
| 58 | 144 | 14.4 | 6.3 | 7.9 | 2.8 | 1138 |
| 66 | 184 | 13.7 | 8.0 | 5.7 | 3.6 | 2019 |
| 74 | 209 | 16.1 | 9.1 | 7.0 | 4.0 | 1870 |
| 82 | 248 | 17.9 | 10.8 | 7.1 | 4.8 | 2181 |
| | | | *Repeat tests* | | | |
| 50 | 138 | 8.8 | 6.0 | 2.8 | 2.7 | 3081 |
| 58 | 165 | 11.5 | 7.2 | 4.3 | 3.2 | 2400 |
| 66 | 179 | 13.8 | 7.8 | 6.0 | 3.5 | 1863 |
| 74 | 193 | 16.1 | 8.4 | 7.7 | 3.7 | 1806 |
| 82 | 243 | 17.8 | 10.6 | 7.2 | 4.7 | 2112 |

[a]Measured values (see discussion after Table 3). [b]Based on interfacial area $A = 2\pi lb = 16 \times 10^{-3}$ m².

were given a thin coating of silastic cement to prevent the ingress of moisture.

The four surface thermocouples and the four immersion thermocouples were all then calibrated with reference to a previously calibrated platinum resistance digital thermometer system (2180A RTD) over the range of temperatures (20–90°C) to be encountered in the tests. The calibration equation for each thermocouple was then written into the data processing program so that the temperatures displayed were as accurate as could be obtained by the available instrumentation.

The cylinder assembly was then placed in the chamber, the heating and cooling lines with the flow meters connected, the immersion thermocouples placed in the fluid lines, and the thermocouples connected to the measurement system.

Before the start of each experiment, the system was bled to make sure that no air bubbles were trapped within. It was also noted that the cooling water thermocouples should be placed in the lines as close as possible to the chamber but not within the chamber itself. This was to avoid any fluctuations due to circulation present as a result of temperature gradients.

## Experimental Results and Discussion

In any given set, the fluid flow rates were held constant and the hot water temperature was varied in steps of about 8°C to give at least five different values of heat fluxes over the temperature range available. At each setting, it took about 1 h for the system to reach steady-state conditions. Two combinations of materials were tested—stainless steel→stainless steel (SS→SS) and stainless steel→aluminum (SS→AL). In each case, two flow rates were used: 0.067 and 0.045 liters/s, each flow rate yielding a set of five readings. The higher flow rate was used to extend the heat flux range.

These five readings, for each flow rate, correspond to five different values of the inner fluid temperature and, therefore, five different heat flux values.

Table 3 gives the range of measured test parameters for both pairs of cylinders (SS→SS and SS→AL).

Full details of the experimental results are given in Tables 4 and 5.

To determine the thermal contact conductance of the SS→SS joint, identical tests were carried out on a solid

**Table 5 Test results for stainless steel → aluminum joint**

| $T_H$, °C | $q$, W | $\Delta T_{tot}$, K | $(\Delta T_i + \Delta T_o)$,[a] K | $\Delta T$, K | $\Delta T_i$, K | $h$,[b] W/m²-K |
|---|---|---|---|---|---|---|
| | | | *Flow rate of hot water = Flow rate of cold water = 0.067 liters/s* | | | |
| 50 | 145 | 6.1 | 3.2 | 2.9 | 2.8 | 3125 |
| 58 | 166 | 7.6 | 3.6 | 4.0 | 3.2 | 2594 |
| 66 | 206 | 9.4 | 4.5 | 4.9 | 4.0 | 2625 |
| 74 | 267 | 11.1 | 5.8 | 5.3 | 5.2 | 3125 |
| 82 | 322 | 13.1 | 7.0 | 6.1 | 6.2 | 3300 |
| | | | *Repeat tests* | | | |
| 50 | 147 | 6.0 | 3.2 | 2.8 | 2.8 | 3281 |
| 58 | 176 | 7.7 | 3.8 | 3.9 | 3.4 | 2819 |
| 66 | 205 | 9.6 | 4.5 | 5.1 | 4.0 | 2513 |
| 74 | 295 | 11.6 | 6.4 | 5.2 | 5.7 | 3544 |
| 82 | 309 | 13.0 | 6.7 | 6.3 | 6.0 | 3062 |
| | | | *Flow rate of hot water = Flow rate of cold water = 0.045 liters/s* | | | |
| 50 | 108 | 5.3 | 2.4 | 2.9 | 2.1 | 2286 |
| 58 | 151 | 5.9 | 3.3 | 2.6 | 2.9 | 3605 |
| 66 | 191 | 7.3 | 4.2 | 3.1 | 3.7 | 3792 |
| 74 | 224 | 8.8 | 4.9 | 3.9 | 4.3 | 3562 |
| 82 | 247 | 10.0 | 5.4 | 4.6 | 4.8 | 3334 |
| | | | *Repeat tests* | | | |
| 50 | 108 | 4.3 | 2.4 | 1.9 | 2.1 | 3458 |
| 58 | 136 | 5.7 | 3.0 | 2.7 | 2.6 | 3098 |
| 66 | 167 | 7.1 | 3.6 | 3.5 | 3.2 | 3008 |
| 74 | 206 | 8.6 | 4.5 | 4.1 | 4.0 | 3123 |
| 82 | 233 | 9.9 | 5.1 | 4.8 | 4.5 | 3012 |

[a]Calculated values (see discussion after Table 3). [b]Based on interfacial area $A = 16 \times 10^{-3}$ m².

stainless cylinder of overall dimensions the same as the composite cylinder. Let,

$$q = \text{heat flow rate}$$

$$\Delta T_{tot} = \text{measured temperature drop}$$

$$= \Delta T_i + \Delta T + \Delta T_o \tag{1}$$

But

$$\Delta T_i + \Delta T_o = \frac{q}{2\pi L k_i} \ell n \frac{b}{c} + \frac{q}{2\pi L k_o} \ell n \frac{a}{b} \tag{2}$$

For the SS → SS joint, therefore,

$$\Delta T_i + \Delta T_o = \frac{q}{2\pi L k_{ss}} \ell n \frac{a}{c} \tag{3}$$

For this joint, therefore, $\Delta T_i + \Delta T_o$ could be obtained by making measurements on a solid stainless steel cylinder of outer radius $a$ and inner radius $c$ under identical conditions. For the SS → AL joint, however, Eq. (2) must be used. Once $\Delta T_i + \Delta T_o$ has been estimated, then, from Eqs. (1),

$$\Delta T = \Delta T_{tot} - (\Delta T_i + \Delta T_o)$$

Finally, the contact conductance is obtained as

$$h = \frac{\text{heat flux at interface}}{\Delta T} = \frac{q/(2\pi L b)}{\Delta T} \tag{4}$$

For the SS → AL joint, there is no solid specimen with which the results could be compared directly. Hence, the additional temperature drop was computed by subtracting the calculated temperature drop in an SS → AL composite cylinder with zero contact resistance from the measured temperature drop in the actual composite cylinder.

### Uncertainty Analysis

The uncertainty in the measured value of the conductance is due to the uncertainties in the measurement of the additional temperature drop $\Delta T$ and the heat flux. Each of these will be considered in turn and then the overall uncertainty will be estimated.

For the SS → SS joint, the overall temperature drop measured across the composite cylinder ranged from 10 to 20°C for the high flow rate and from 9 to 18°C for the low flow rate. For SS → AL joint, the corresponding ranges were from 6 to 13°C and from 4.5 to 10°C, respectively. The temperature measurements were estimated to be accurate to 0.2°C. It is noted, however, that the additional temperature drop is estimated as the difference between two temperature differences. Therefore, the law of error propagation as applicable to algebraic sums,[28] the uncertainty increasing by a factor of $\sqrt{2}$ each time a difference is taken, must be applied twice to get the uncertainty in this measurement. Thus, the uncertainty in $\Delta T$ would be on the order of 2–4% for SS → SS and 3–9% for the SS → AL joint.

The temperature difference measured in the fluid stream were on the order of 2°C. The uncertainty in the temperature difference measurement is estimated to be $0.2\sqrt{2}$°C. The uncertainty in the value of thermal conductivity is assumed to be 5%. Errors (estimated to be less than 1%) in the measured values of the interfacial area and the flow rates are comparatively small to cause any changes in the overall uncertainty. Thus, the overall uncertainty in the heat flux measurement is determined to be about 15%.

Combining the two uncertainties, the uncertainty in the conductance is calculated to range from 15 to 18%.

### Discussion

Figure 3 shows the results for the SS → SS joint. The results represent four different series of experiments conducted at two different flow rates. The conductance values range from about 2000 to 3000 W/m²-K. Given the statistical nature of the contact problem and if we reject two points that clearly lie outside the bulk of data, the scatter is not very large. Some scatter, at least, is due to unavoidable fluctuations in the fluid stream temperatures.

A theoretical analysis for the prediction of the thermal contact conductance in composite cylinders has been pre-

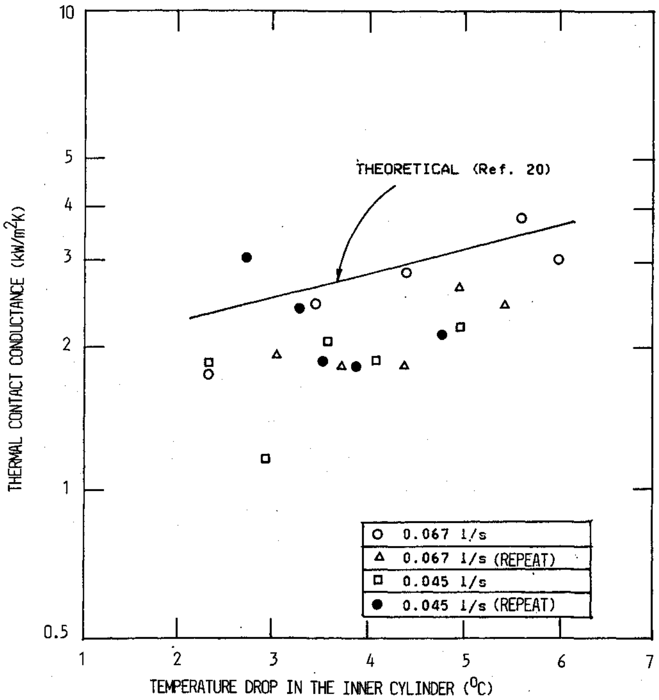


Fig. 3 Results for SS→SS joint.

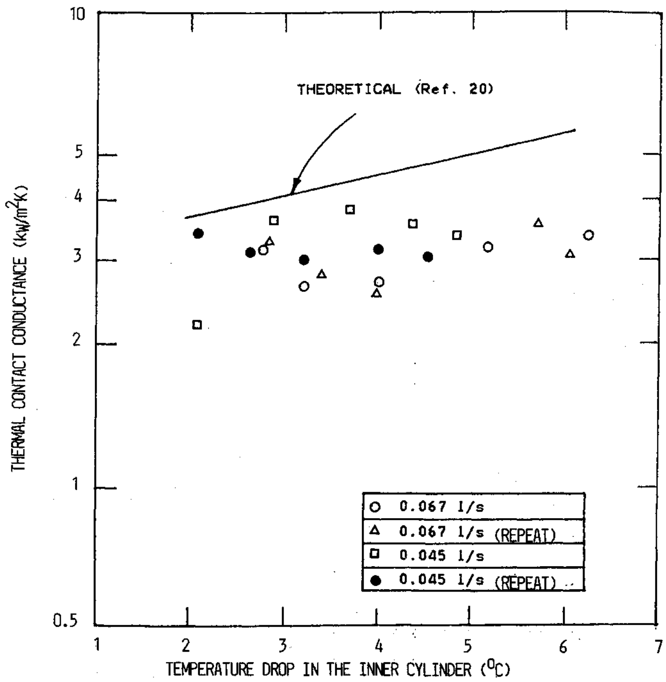


Fig. 4 Results for SS→AL joint.

sented in Ref. 20. This theory considers that:

1) The cylinders could be of similar or dissimilar materials.

2) The cylinders may be in a conducting medium or vacuum.

3) For a given pair of cylinders (materials, geometry, and surface characteristics), the contact pressure depends on the following: initial degree of fit, differential expansion due to heat flow, and differential expansion due to the temperature drop at the interface.

Since the last two factors here depend on the heat flux, the contact pressure and, hence, the contact conductance also depend on the heat flux. Thus, a closed-form equation is derived relating the geometry, thermomechanical properties and the surface characteristics of the cylinders, and the heat flux to the contact (solid spot and gas-gap) conductance. This equation is easily solved to obtain the contact conductance (and contact pressure, if needed) for any value of heat flux for a prescribed pair of cylinders.

This theory indicates that

1) For the range of heat flux considered, the interface pressure would be on the order of 1–4 MPa.

2) For these relatively small values of contact pressure, the conductance is predominantly through the interstitial medium—in this case, air—and should have values ranging from 2500 to 3500 W/m$^2$-K, as indicated in Fig. 3.

It is clear that the experimental values are close to those predicted. It is to be noted that the theoretical values are based on an approximate correlation between the surface roughness and the mean effective gap for calculating the gas conductance. Specifically, the correlation used is of the following form:

$$\text{gas conductance } h_f = k_f/\delta_{\text{eff}}$$

According to several investigators,[29,30] $\delta_{\text{eff}}$ is about three times the rms value of the surface roughness heights; the theory also does not take into account large-scale irregularities such as waviness or out-of-roundness, both of which tend to increase resistance.

The results for the SS→AL joint are depicted in Fig. 4. Again, these represent four different series of experiments conducted at two different flow rates. It can be readily seen that the scatter is quite small comapred with the average value of contact conductance. The measured conductance values range from 2500 to 3800 W/m$^2$-K compared with the theoretical estimates[20] of 3900–5900 W/m$^2$-K. In addition to the remarks made in connection with the discussion of the results for the SS→SS joint, it may further be noted that the theoretical results are based on an assumed constant value (0.1) for the slope of the surface profiles. A change in this value could materially affect the predicted value of conductance.

## Conclusions and Recommendations

An experimental rig has been designed and constructed to study the heat flow across cylindrical joints in air. The results for stainless steel→stainless steel and stainless steel→aluminum joints are presented. The conductance values range from 2000 to 3000 W/m$^2$-K for the stainless steel→stainless steel joint and from 2500 to 3800 W/m$^2$-K for the stainless steel→aluminum joint. These results are in fair agreement with theoretical predictions.

Further work in this area should include increasing the range of temperatures and heat fluxes and refinement of the theoretical model to include large-scale irregularities at the interface.

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