Thermal Conductance of Cylindrical Joints—A Critical Review

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Presented here is a review of the theoretical and experimental studies conducted over the past 30 years of the heat transfer characteristics of compound cylinders. For the purpose of this review, such work has been broadly classified into two categories—application oriented studies and fundamental studies. A majority of the application oriented studies deal with heat transfer in finned tubes as used in heat exchangers and heat transfer in nuclear fuel elements. In all cases, the heat flow is considered to be radially outward. The results of this review indicate that the contact resistance in heat exchangers is generally measured indirectly and is often assumed to be constant. The correlations proposed usually apply only to specific types of tubing and prescribed materials. The fundamental studies, in general, consider idealized surfaces with no large-scale irregularities such as waviness or out-of-roundness. In most of the studies, the effect of the axial temperature gradient is not considered. Finally, it was found that there exists very little experimental data dealing with the fundamental aspects of heat transfer through cylindrical joints.

Nomenclature

a = radius, defined in Fig. 6
b = radius, defined in Fig. 6
c = radius, defined in Fig. 6
d = indentation diameter
d_e = diameter of expansion tool
d_h = hole or collar inside diameter
d_0 = tube outside diameter
E_i = modulus of elasticity for the inner material
E_0 = modulus of elasticity for the outer material
fpi = fin density
g_1 = temperature jump distance
g_2 = temperature jump distance
h_c = thermal contact conductance ($W/\text{cm}^2\,K$)
h_{s1} = average height of the surface roughness, surface 1
h_{s2} = average height of the surface roughness, surface 2
I = net tube expansion interference
k_{gas} = thermal conductivity of the gas
N = fins per unit length
R_b = bond resistance
R_c = contact resistance
R_f = fin resistance
R_g = gap resistance
R_i = inside film resistance
R_0 = measured overall resistance
R_T = total resistance
R_t = resistance of the tube wall
t = fin thickness
t_w = tube wall thickness
ΔT_i = temperature difference
Δu_f = radial displacement of the fin
Δu_t = radial displacement of the tube
α_1 = thermal coefficient of the core
α_2 = thermal coefficient of the casing
δ_A = differential expansion due to the temperature gradients caused by heat flow
δ_B = differential expansion caused by the temperature difference at the interface
δ_C = initial degree of fit
δ_{eff} = effective radial gap
δ_m = mean physical gap
ϕ_A = angle, defined in Fig. 6
ϕ_B = angle, defined in Fig. 6

Introduction

OVER the past 40 years, a substantial amount of research work, both theoretical and experimental, has been conducted on heat flow across plane joints. This has resulted in a considerable number of publications, the results of which have been summarized in various reviews and bibliographies.[1-7] Over the same period of time, however, the amount of research conducted on cylindrical joints has been quite limited. For example, over the decade spanning 1970–1980, there were only about half a dozen publications on cylindrical joints compared to some 150 on flat joints. This is both surprising and disappointing since cylindrical joints are as prevalent as flat contacts in engineering practice. Examples include conventional nuclear reactor fuel elements, shrink fit cylinders, and finned tubes of either the tension-wound or extruded types.

Before reviewing the previously published literature on cylindrical joints, it is instructive to consider those aspects which make cylindrical joints different from flat joints:

1) In a flat joint, the contact pressure is explicitly known and is usually chosen, in both theory and experiment, as the independent variable controlling the conductance. For a given cylindrical joint, however, the contact pressure is a function of the differential expansion of the two cylinders.[8,9] According to classical thermoelastic analysis, the differential expansion, in turn, is a function of the heat flux.[10] Therefore, the heat flux,

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rather than the indirectly estimated contact pressure, becomes the logical independent variable for the analysis of heat flow through cylindrical joints.

2) In addition to the surface irregularities encountered in flat joints, cylindrical surfaces have additional characteristics, e.g., out-of-roundness, which must be taken into consideration.

3) The location of thermocouples for measurement of temperature is generally more difficult for cylindrical joints than for flat joints.

It is probably due to the complexities associated with the three factors listed above that there have been comparatively fewer investigations of the heat flow across cylindrical joints.

The existing literature can be divided into two major categories, specific applications and fundamental studies of compound cylinders. The following is a review of the literature for each of these categories.

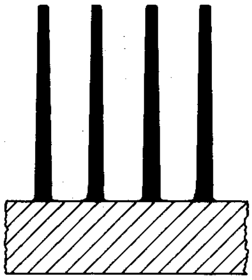

Fig. 1a Edge wound "I" fin.

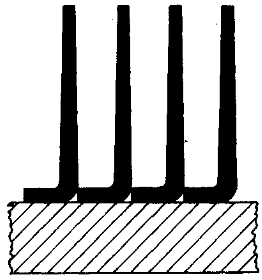

Fig. 1b *L*-foot fin.

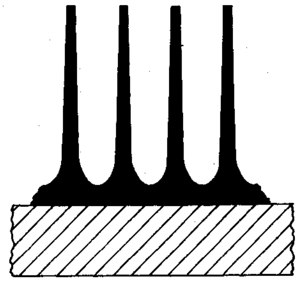

Fig. 1c Extruded fin or muff.

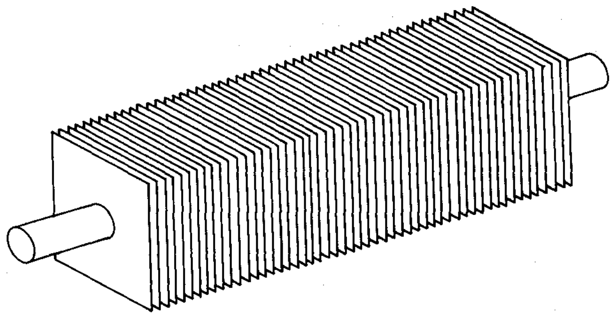

Fig. 1d Plate fin tube.

Cylindrical Joint Applications

Since the impetus for basic research must, ideally, arise from problems encountered in practice, it is best to first consider the research work conducted in regard to specific engineering applications. As far as cylindrical joints are concerned, the investigations may be considered in two main categories: 1) heat exchangers and 2) nuclear reactors.

Heat Exchangers

Thermal contact resistance plays an important role in the heat transfer behavior of finned tubes. In particular, the performance of the three primary types of heat exchanger tubes are significantly affected by variations in thermal contact resistance:

1) Tension-wound tubes (see Figs. 1a and 1b)—In this type of finned tube, a strip of metal, usually aluminum, is spirally wound under high tension over a base tube of a material such as carbon steel, copper, or aluminum.

2) Extruded fin or "muff" tubes (see Fig. 1c)—An outer tube of large wall thickness, usually aluminum, is swaged over an inner base tube and extruded into high fins in one operation.

3) Plate fin tube (see Fig. 1d)—In this case, the joints between the fins and the tubes are usually formed by first inserting the tube into the fin stack and then mechanically expanding the tube into the plastic deformation zone until an interference fit is obtained between the tube and the fin collar. Such expansion is commonly achieved by forcing an expanding tool through the tube or by hydraulic expansion.

One of the earlier investigations of the thermal contact in plate fin tubes indicated that the joint between the fin and the tube was an important parameter in the characterization of heat exchanger performance.[11] In this method, the mechanically bonded tube was compared with a metallurgically bonded (soldered) tube assumed to have no contact resistance. The difference in the two overall resistances yields a value referred to as the bond resistance.

A detailed analytical and experimental study of interference fit finned tubes was conducted by Gardner and Carnavos.[12] The authors noted that, as the operating temperature increased, the fins expanded away from the tube wall due to the larger expansion coefficient. As the temperature was increased further, a successively greater proportion of the heat was transferred through the gas gap than through the actual metal-to-metal contact area. Eventually, a point was reached where the gap between the fin base and the tube wall was opened to such an extent that the heat transferred through metal-to-metal contact was deemed to be zero and all of the heat was assumed to pass through the entrapped fluid.

In this analysis, the radial displacements of the fin and the tube, Δu_f and Δu_t, respectively, were computed by means of thermoelastic equations. If the difference in these displacements, $\Delta u_f - \Delta u_t$, was positive, a radial gap would exist, and the contact pressure would be zero. Since the authors were mainly interested in the gap resistance, they did not consider negative values of $\Delta u_f - \Delta u_t$, which would serve to reduce this resistance. It would also appear that the gap resistance R_g was for a continuous annulus of gas and not that of individual pockets of gas surrounding the actual contact spots, as is usually assumed.

The experimental studies of Gardner and Carnavos[12] investigated finned tubes of three different types—mechanically embedded, tension-wound, and muff—produced by three different manufacturers. The sum of the outside resistance (film + fluid) R_0 and the contact resistance R_c was found by subtracting the computed values of inside film R_i, the tube wall R_t, and the fin resistance R_f from the measured overall resistance R_T, such that

$$R_T - (R_i + R_t + R_f) = R_0 + R_c \tag{1}$$

An equation was then developed relating $R_0 + R_c$ to air mass velocity. (In the experiments conducted, steam was used as the heating fluid in the tube and air as the cooling fluid over the fins.) At low temperatures the gap resistance was neglected, but at high temperatures the gap resistance was included.

$$R_T - (R_i + R_t + R_f) = R_0 + R_c + R_g \tag{2}$$

The results were tabulated from the various groups, except mechanically embedded tubes, for two different steam temperatures: 157°C (315°F) and 193°C (380°F).

The experimental values of $R_0 + R_c + R_g$ showed favorable agreement with the calculated values. The authors concluded that although the bond resistance may be negligible, gap resistance is a significant part of the total resistance when the service temperatures are sufficiently high.

The pioneering nature of this work is without question. The summary just presented, however, prompts several comments.

1) In deducing $R_c + R_0$ from R_T, several correlations, rather than actual measurements, are used to determine the remaining resistances, especially R_i.

2) It is assumed that R_c remains constant with temperature.

3) The experimental results are presented only for specific tubing at two different temperatures. Thus the results are not sufficiently general to be used by other workers.

Apart from these limitations, the work represents an important contribution to the study of heat transfer in finned tubing.

The heat transfer in bimetallic finned tubes of the muff type as used in air-cooled equipment has been analyzed.[13] The authors noted that although initial contact pressures of 24 to 30 MPa (3500–4300 psi) existed in the tubing at fabrication temperatures of 21°C (70°F), the pressure was reduced to zero at temperatures around 100°C (212°F), and an air gap was established.

The theory of Kulkarni and Young[13] was based on the results of Shlykov and Ganin[14] for contact and gap resistance, and contact pressure was calculated using the Gardner and Carnavos[12] relations. This theory was valid for predicting the bond resistance after a gap had been established, and the Shlykov and Ganin theory[14] was applicable when no continuous gap existed. The discontinuity between the two results at zero contact pressure was removed by assuming that, at zero contact pressure, the bond resistance could be expressed as

$$R_b = \frac{(h_{s1} + h_{s2})}{(2k_g)} \tag{3}$$

This expression is, in fact, part of the Shlykov and Ganin equation for the total contact resistance.

Kulkarni and Young[13] computed results for the contact and bond resistance as a function of the tube side fluid temperature over the range 20°C to 540°C (70°F to 1000°F). The film and fouling resistances (both inside and outside) were also considered. In all cases, an abrupt increase in the resistance between 100°C and 200°C (200°F and 400°F) was observed. Presumably, this range corresponds to the situation where the contact was broken and the gap established.

Overall this investigation is both logical and complete. It does not suffer from discontinuities or sweeping assumptions. It is based on the accepted literature available at the time, and the results are presented in a general and usable format. In view of these, it is surprising that it has not received wider acceptance.

Eckels[15] applied a semitheoretical approach to the investigation of the thermal resistance of plate fin tubes. The interfacial pressure and the contact conductance were related to the design parameters by

$$h_c = 256\left\{\frac{t/d_0}{\left[\dfrac{1}{t(fpi)}\right]^2 - 1}\right\}^{0.6422} \tag{4}$$

Based upon the experimental data from six coils, the probable error band of Eq. (4) was estimated to be 30% at 1.1352W/cm^2–K (2000 BTU/h-ft^2-°F) and diminished to 15% at 0.5676W/cm^2–K (1000 BTU/h-ft^2-°F). Eckels[15] found the most significant sources of error to be the temperature and the mass flow rate measurements along with the value of the fin resistance. It should be noted that the above correlation only applies to specific types of tubing with prescribed materials.

Christensen and Fernades[16] experimentally measured the thermal resistance and the inside fouling resistance of pneumatically expanded plate fin tubes. The tube materials were copper, cupronickel, and stainless steel, and the fins were made of copper. From the overall resistance, the computed tube side film resistance and the tube and fin material resistances were subtracted to obtain the combined contact resistance and the fouling resistance. By conducting similar tests in freon, the fouling resistance was eliminated thereby isolating the contact resistance.

The results indicate that the contact resistance was larger than the fouling resistance. However, the experimental errors were of the same magnitude as either of these resistances. Hence, the results cannot be viewed in absolute terms, but rather as a domain of possible values.

The results from a series of investigations[17-19] dealing with fin-to-tube joint characteristics including the thermal contact conductance of mechanically expanded plate fin tube heat exchangers has been reported in the literature. The first of these investigations dealt with the surface characteristics of a finned tube heat exchanger.[17] The majority of the work was exploratory in nature. It was found that, for a given fin number, the surface roughness of the fin collar increased as the expansion increased. This conclusion is somewhat surprising since the roughness would normally be expected to decrease with increasing expansion, especially since the authors state in the conclusions that "the ridges seen on the surface before expansion becomes plateaus after being expanded." In the second of these investigations, the mechanical bond between the tube and the fin was characterized.[18] It was found that the maximum force obtained during a tube pullout could be used as a good descriptor of such a bond. For thinner fins and higher fin numbers, the amount of interference was less influential than other parameters. At sufficiently high interferences, the fin thickness was the dominant factor affecting the pullout force. Since the heat exchanger industry is attempting to reduce fin thickness to improve performance and save material, a compromise is clearly necessary.

The third investigation[19] presented an empirical correlation of the contact conductance. An end view of the plate fin tube heat exchanger evaluated is shown in Fig. 2, together with a schematic of a single hot tube and the surroundings. Test coils contained three rows of tubes with 12 tubes per row. Hot water passed through the center row and cold water through the two outer rows. The technique was similar to that used by Dart[11] and Eckels[15]; although Dart and Eckels did not conduct their tests in a vacuum. For the "dry" tests, the vacuum was of the order of 40×10^{-3} mm Hg and for the "wet" tests, in which water was sprayed on the coil, the vacuum increased to 900×10^{-3} mm Hg.

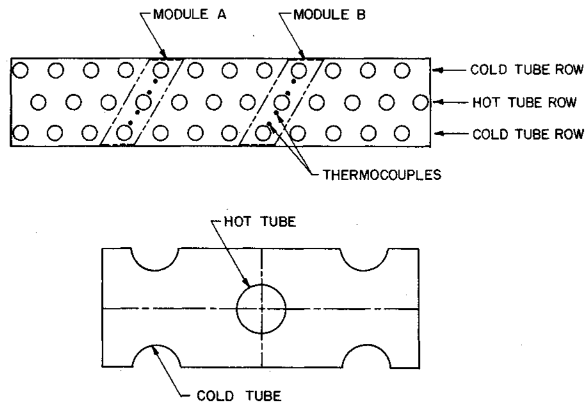

Fig. 2 The heat exchanger and mathematical model.[19]

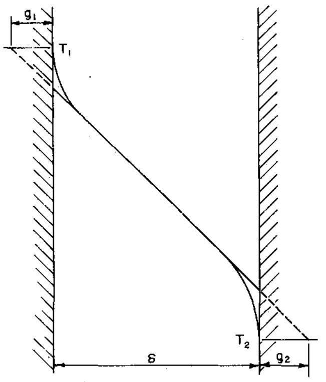

Fig. 3 Temperature jump distance.

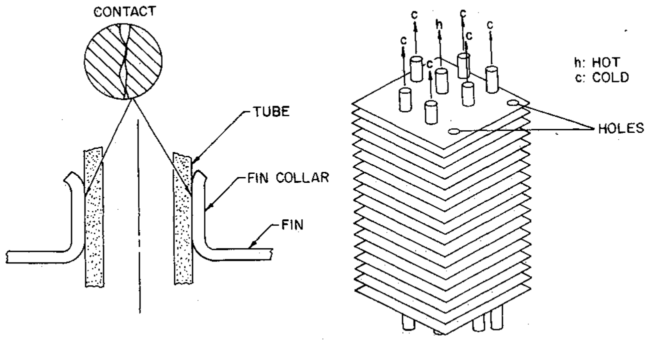

Fig. 4 Test core and element of interest.[26]

The authors assumed that the contact resistance remained the same for the hot and cold sides. This is not a valid assumption since the contact resistance depends on the differential expansion and, therefore, on the temperature. The authors further noted that the maximum gap height in the contact region was $\simeq 3.2 \times 10^{-3}$ mm (1.25×10^{-4} in.), and the mean free path for air "molecules" at 40×10^{-3} mm Hg is 6.6 mm. Therefore, it was concluded that for heat transfer to take place through the air trapped in the gap, the gap height must be at least 6.6 mm. As a result, it was assumed that no heat was transferred through the trapped air in the gap. This conclusion is contradictory to what has been previously reported. The smaller the gap, the greater the probability that the gas molecules will conduct heat by direct impact with the solid surfaces. Increasing the gap would have the opposite effect. It has been established[21] that the heat transfer coefficient for rarefield gas in small gaps may be approximated by

$$h_g = \frac{k_g}{\delta_m + g_1 + g_2} \tag{5}$$

where δ_m is the mean physical gap and g_1 and g_2 are the temperature jump distances (see Fig. 3), which depend on the mean free path. Hence, the greater the physical gap, the less likelihood of heat conduction through the gap. The experimental facility for measuring the contact conductance of plate finned tube heat exchangers is described in another reference[21] arising from the same investigation.

In a related investigation,[22] a similar problem was studied, and the following correlation for the contact conductance of plate fin tubes was proposed.

$$h_c = \exp\{6.092 + 2.889[(I\,fpi\;d/d_0)^{0.75}(t\,fpi)^{1.25}]\} \tag{6}$$

where the terms d_0, t, and fpi are the same as used in the Eckels correlation.[15] This correlation was found to predict the conductance values, within an error band of $\pm 20\%$, for approximately half the total number of coils tested.[22]

The effect of various parameters on the contact conductance of fin tubes has also been investigated.[23] In this investigation it was observed that the contact conductance increased with interference but decreased as the tube diameter increased. The authors found that the contact resistance was not significant and could not be predicted accurately enough as a variable independently evaluated from the air-side resistance to restructure the American Refrigeration Institute (ARI) coil testing standards (ARI 1981). However, they indicated that an increase in the contact resistance, over a period of time, could be used to determine when the tool wear had become critical and used as a guideline for manufacturing processes and tool replacement.

A more recent investigation[24] summarized the experimental test results for 31 typical coils by means of the correlation

$$h_c = \exp\{8.6379 + 0.1844\;[I\,N\,(d/d_0)^{0.75} \times (t\,N)^{1.25}]\} \tag{7}$$

The authors noted that the correlation was valid only for mechanically expanded copper tubes with aluminum fins. It was also noted that in many instances the thermal contact resistance plays a significant role in plate finned tube heat exchangers.

Later, Taborek[25] summarized the current status of the bond resistance and design temperatures for finned tubes and concluded

1) The temperatures recommended by previous investigators, were far too conservative due to the improved manufacturing techniques and quality control now obtainable.[26]

2) Many manufacturers rely on a "pull test" alone to measure the fin bond quality.

3) Some manufacturers perform simple comparative heat transfer tests on the coils as they come out of the firming machines to ensure quality control.

All three of these observations are in agreement with the conclusions of Wood et al.[22]

Taborek[25] also noted that over the past 20 years, little progress had been made in establishing orderly standards for testing or accepted practices which could serve as a guide to manufacturers and users. In addition, the currently recommended maximum bond temperatures for several types of finned tubes were summarized in Table 1.

Table 1 Maximum recommended bond temperatures

	USA	Europe
L-footed tubes	176°C (350°F)	150°C (300°F)
Extended fins	230°C (400°F)	250°C (480°F)

Nho and Yovanovich[27,28] reported experimental results for five different cores (finned tubes). The test core and element of interest are shown in Fig. 4. The tubes were constructed from a copper alloy, C12200, and the fin material was aluminum alloy, MR-160. Three different "bullet" sizes were used to obtain "reasonable" contact pressure levels, and three different collar lengths were used to obtain different fin densities. Tests were conducted in a vacuum environment.

The measured conductance values ranged from 2488 to 5460 W/m^2K and agreed well with the correlation proposed by Sheffield et al. An uncertainty analysis, however, was not presented for this experimental investigation.

A thermal mechanical model of a plate fin-tube heat exchanger was developed by Burgers.[29] In addition to the thermomechanical properties, the expansion interference and the differential expansion of the collar and the tubes, it was noted that the work hardening exponents of the materials also influenced the joint pressure. It was further noted that the increase of the thermal contact resistance with the age of a sample may not only be due to the buildup of oxide layers within the joint but also to the leaching of the oil out of the joint.

In summary the following conclusions can be made in regard to finned tube heat transfer characteristics:

1) The contact resistance is never directly measured; it is usually determined as the difference between the measured overall resistance and the estimated sum of resistances other than the contact resistance, as determined by a correlation, formula, or experiments on a metallurgically bonded tube.

2) Most of the results apply to specific types of tubing and are not sufficiently general.

3) The bond resistance is usually considered to be a small part (< 15%) of the overall resistance.

Nuclear Reactor Applications

Although the heat flow path in a conventional fuel element is radially outward between concentric cylinders, the majority of the experimental studies conducted involving the contact conductance in nuclear fuel and container materials make use of the flat joint between the ends of two cylinders whose axes are collinear.[30-33] The two studies discussed below are unusual in that they attempt to simulate the situation more realistically.

The work of Brutto et al.[34] deals with a comparison of different cladding processes such as cold drawing, hot drawing, and hydrostatic pressing. To avoid estimating convection resistances by approximate correlations, tests were conducted on a clad tube and a similar but unclad tube. Radially outward heat flow was achieved by internal heating with hot water (40 to 80°C) and external cooling by cold water. The difference, after correcting for the cladding tube wall resistance, resulted in a value for the contact resistance. This method resulted in an increase in the accuracy and eliminated systematic errors. In order to minimize the error in the measurement of the contact resistance, the fluid flow rates, both internal and external, were maintained as high as possible, and a copper tube was used to simulate the fuel rod. Since the contact resistance depends on the heat flux, the heat fluxes used in the experiments were of the same order of magnitude as in a reactor (300 to 900 kW/m^2). The results indicate that in this heat flux range, the contact resistance remained nearly constant except for the aluminum clad tube, which showed a decrease in resistance over the range of 40 to 80°C. The authors point out this method gives only an average value of contact resistance; it does not reveal isolated areas of poor contact.

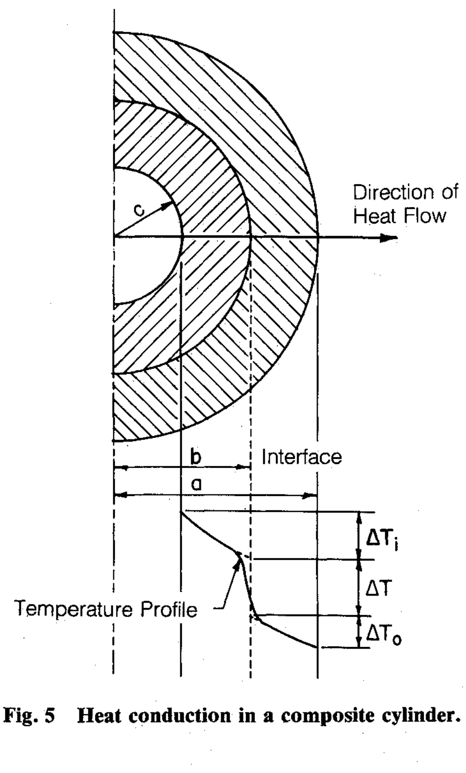

Fig. 5 Heat conduction in a composite cylinder.

The results of a series of in-pile experiments were reported by Cohen et al.[35] in which the effective thermal conductivity of stainless steel clad uranium dioxide rods during irradiation was measured.

For one sample within a zero initial clearance, results of contact conductance as a function of contact pressure were given. For this case, the contact pressure was estimated by formulas for a press-fit assembly after calculating the relative thermal expansions of the fuel UO_2 and the cladding, Stainless Steel 304. For the other specimens, with an initial clearance, the gap conductance was plotted as a function of the initial diametral gap. To theoretically estimate the gap conductance, the method of Cetinkale and Fishenden[36] was used and illustrated favorable quantitative agreement between the theory and experiment.

In addition, Cohen et al.[35] found that the gap conductance was insensitive to the gas medium within the sheath. For example, in the computation of the conductance it did not matter whether the gas was helium or a mixture of xenon and krypton. As demonstrated by Madhusudana and Fletcher,[37] this can be explained by means of a dependence of the accommodation coefficients on the molecular weights of the gases.

Cylindrical Joint Fundamental Studies

The fundamental studies of cylindrical joints consider heat flow through concentric cylinders in vacuum or in a conducting medium. In a majority of these studies, the direction of heat flow is radial and axisymmetric (see Fig. 5). These studies attempt to be as general as possible without regard to specific material or interstitial gas. It must, however, be noted that although the heat flow is two dimensional from a macroscopic point of view, it is three dimensional from a microscopic consideration since, in general, the actual contact spots are isolated areas and not uniform strips.

One of the first published studies of this phenomenon was by Williams and Madhusudana.[38] In this study, it was found that the interference and, therefore, the contact pressure between the two cylinders, consisted of three components, namely, the following:

1) δ_A—differential expansion due to the temperature gradients caused by heat flow, which can be calculated by the well-known thermoelastic equations,

2) δ_B—differential expansion caused by the fact that the two surfaces at the interface would be at two different temperatures because of the contact resistance, and

3) δ_C—the initial degree of fit.

The interactive nature of the conductance and the contact pressure, unique to cylindrical joints, was therefore noted. It was anticipated that because of the second component δ_B, the heat flux should be the controlling variable for a given pair of cylinders.

In the experiments conducted on stainless steel cylinders, the contact resistance was determined by measuring the radial temperature profile in the two cylinders. The results indicated that 1) the resistance decreased with heat flux, and 2) resistance decreased by nearly an order of magnitude when the vacuum was replaced by air.

Novikov et al.[39] presented a summary of experimental results for composite cylinders of different materials in a vacuum. Their results indicate that when $\alpha_2 < \alpha_1$, the contact resistance decreased with thermal load, i.e., heat flux. These results are in complete agreement with those of Williams and Madhusudana.[38]

Experimental results on interference fit compound cylinders were also reported by Hsu and Tam.[40] The inner cylinder was made of aluminum and the outer cylinder of Stainless Steel 304. All tests were conducted in air, as was the case previously, and the contact resistance was determined by direct measurement. The results were presented graphically for contact resis-

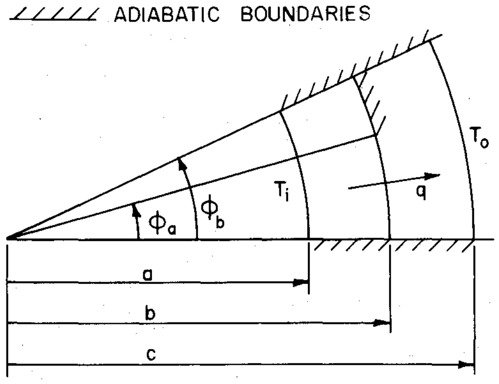

Fig. 6 The two-dimensional model of Wang and Nowak.[41]

tance as a function of heat flux as well as contact resistance as a function of estimated contact pressure. The measured results were considerably less than those predicted using the theory of Shlykov and Ganin.[14] The following points are also worth noting: 1) although the overall temperature drop across the two cylinders was approximately 30°C, the interfacial temperature drop was less than or equal to 1°C; and 2) the error bands on the data indicated an error of ±9%.

A two-dimensional theoretical analysis of the constriction resistance for radial heat flux in duplex tubes was presented by Wang and Nowak.[41] The heat conduction domain analyzed is shown in Fig. 6. Two cases—uniform heat flux and isothermal conduction—were considered at the contact land. The results for the two cases did not differ significantly (about 4-5%) and agreed well with results of a two-dimensional conduction sheet analog and the results of a numerical analysis. Results for only one set of parameters were presented, and there were no general tables or graphs covering a range of ϕ_A, ϕ_B, a, b, and c. Even if such data were presented, it is apparent that it could only be applied to two-dimensional contacts such as grooves or ridges in a vacuum.

A theory predicting the contact pressure and contact conductance as a function of the heat flux was presented by Madhusudana.[8] This presents a development of the ideas indicated in Williams and Madhusudana[38] and extends the ideas to include similar as well as dissimilar materials in vacuum; results were presented, for radially outward flow, in the form of graphs. Since the temperature drop ΔT_i across the inner cylinder, for example, is a measure of the heat flux, the plots showed pressure as a function of ΔT_i as well as h_c, contact conductance as a function of ΔT_i. Other parameters considered were the degree of initial fit, surface roughness, and the material combination. The following conclusions were drawn.

1) Contact pressure and contact conductance increase with heat flux.

2) Contact conductance is higher for $\alpha_i > \alpha_0$.

3) Contact conductance is higher for $E_i < E_0$.

4) The p is dependent on the surface roughness also, and hence contact conductance is not a simple function of the surface roughness unlike the situation in a flat joint.

The above work was further extended by Madhusudana[9] to include the heat transfer through the fluid contained in the interstitial gaps. The following additional conclusions were drawn:

1) Contact pressure is only a weak function of the fluid contained in the gaps.

2) If $(\alpha/E)_i > (\alpha/E)_0$, then large contact pressures are developed even for small heat fluxes.

3) Material combination with lower effective thermal conductivity $[2k_i k_o/k_i + k_o]$ can, in fact, exhibit higher contact conductance than one with much higher effective conductivity, provided $(\alpha/E)_i > (\alpha/E)_0$. This is in striking contrast to the case for flat joints.

4) Contact conductance is significantly influenced by the interstitial medium (although p is not) especially at low heat fluxes.

It should be noted that the two theories just discussed assume an idealized interface with no large-scale surface irregularities such as out of roundness or waviness. Further, the gas conductance is calculated using a simple correlation of the form $(k_g/\delta_{\text{eff}})$. The effect of an interstitial medium should be explored further taking into account the accommodation coefficients which are applicable to particular gas/surface combinations.

An analysis for predicting contact resistance in compound cylinders and finned tubes was presented by Lemczyk and Yovanovich.[42] This analysis utilized a logic and approach similar to the earlier work of Williams and Madhusudana[38] and Madhusudana.[8,9] Lemczyk and Yovanovich[42] utilized the results of Hsu and Tam[40] for comparison with their theory; although the values of Vickers hardness differed from those reported by Hsu and Tam. The thermal resistance was presented as a function of an estimated parameter (i.e., the contact pressure) rather than a measured value (i.e., the radial heat flux); therefore it is difficult to compare the results with other investigators. A similar analysis was developed by Lemczyk and Yovanovich[42] for plate fin tubes in which the tube was considered to be in a state of plane strain and the fin in a state of plane stress with air as the interstitial gas.

The Lemczyk and Yovanovich[42] investigation included several novel features, such as convection boundary conditions for the free surfaces of the cylinders and the effects of surface roughness at the tube/fin interface.

Results of an experimental program on the contact conductance of composite cylinders were presented by Madhusudana and Litvak.[43] The material combinations tested were Stainless Steel 303→Stainless Steel 303 (SS→SS) and Stainless Steel 303→Aluminum 2011 (SS→AL). The interstitial medium was air with the heat flow radially outward. The initial fit corresponded to ANSI locational interference fit, Class LN1. The contact conductance was estimated by comparing the results of the tests on compound cylinders with that of a solid cylinder tested at identical conditions. For the SS→AL combination, the overall resistance for a hypothetical compound cylinder with no interface resistance was computed and compared with that measured in the actual composite cylinder. The uncertainty in contact conductance was estimated to be from 15 to 18%. The results were plotted for contact conductance as a function of ΔT_i. The scatter in the experimental results was of the same order of magnitude as the uncertainty. The agreement between experimental data and theory of Madhusudana[9] was also of the same order.

All of the studies considered this far assume no axial temperature gradients. Srinivasan and France[44] however, in their study of heat transfer in prestressed duplex tubes used in the steam generators of an experimental breeder reactor (EBR), considered such a gradient. Initially, it was assumed that an arbitrary longitudinal temperature distribution between the inner and outer tubes existed. The interface condition along the length (i.e., the contact pressure or gap width) was determined as a function of the longitudinal temperature distribution, and hence the thermal resistance could be calculated. With the thermal resistance specified at several nodes along the length of the tube, the heat transfer was determined, and new values of the temperature difference were determined and compared with the original distribution. If the two distributions did not coincide, the procedure was repeated with the most recent values of temperature difference.

It was found that when the initial contact pressures were relatively high, 13.79 MPa (2000 psi), the different initial trial distributions always converged to a unique final distribution. When the pressure was lower, 10.34 MPa (1500 psi), the different initial distributions did not necessarily converge to a single final solution. In fact, there were three distinct solutions: the first corresponding to good contact along the entire length of

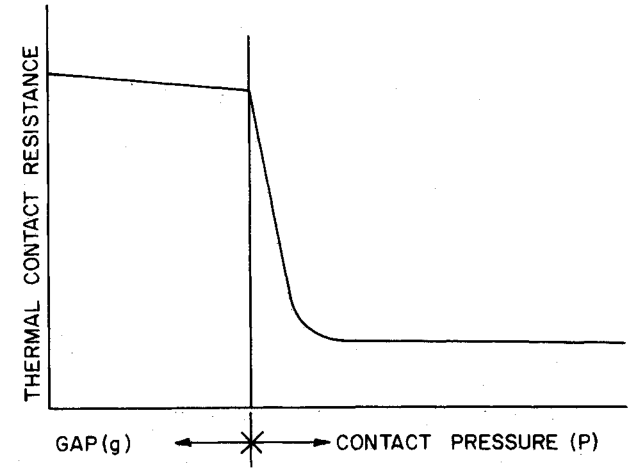

Fig. 7 Variation of thermal contact resistance by Srinivasan and France[44] and Barber.[45,46]

the tube, the second corresponding to 94% contact, and the third corresponding to contact over only 33% of the length (i.e., separation over 2/3 length). The reduction in heat-transfer capacity was of the right order of magnitude to account for the degradation of heat transfer experimentally observed.

The authors indicated that the mechanism responsible for nonuniqueness was related to the nature of the contact resistance curve (see Fig. 7) assumed in their calculations. If the time variation of boundary conditions during the ascent-to-power run of the plant were such that the contact pressure at the interface throughout the length of the tube was in the flat region on the right side of the curve, then the steady-state value would correspond to that obtained for full contact. On the other hand, if a transient event caused a low contact pressure over some part of the tube, the resulting increase in contact resistance would in turn cause an increase in ΔT. This change in temperature difference would cause further loss of contact and ultimately lead to the separation of tubes over either a shorter or longer part of their length.

The multiple solutions, however, did not exist at the higher temperatures. As a result, it was postulated that there was some critical initial pressure below which multiple solutions would be obtained. This conclusion was confirmed by the theoretical investigations of Barber,[45,46] who found that multiple solutions existed only if the initial value of the interference was less than a minimum value.

Conclusions

The current review reveals that there are significant gaps in the current state of knowledge and understanding of heat flow through cylindrical joints. Over the past 30 years, a very moderate effort and modest progress has been made by the heat-transfer community to explore the rather unique heat-transfer behavior of composite cylinders.

Although a considerable amount of the reviewed work dealt with heat transfer in finned tubes, such studies, in general, seem to suffer from the following shortcomings.

1) Contact resistance is invariably measured indirectly by subtracting computed film resistance, etc. from the overall resistance.

2) Contact resistance is usually assumed to be constant. In particular the change in contact resistance caused by temperature related differential expansion is often neglected. It is also not sufficient to merely consider the average effect over the appropriate temperature range (see p. 282 of Ref. 12).

3) Studies usually refer to specific types of tubes and materials. The correlations proposed usually apply to a restricted class of heat exchanger coils and are not generally applicable.

The fundamental studies discussed herein also have several drawbacks. In particular, they consider idealized surfaces with no waviness or out of roundness. In general, they do not consider axial temperature gradients. Finally, experimental data with which the theories can be confirmed is also very limited.

References

[1]Wong, H. Y., "A Survey of Thermal Conductance of Metallic Contacts," Atomic Energy Commission, Rept., 1973, London, 1986.

[2]Mianges, M. L., "Thermal Contact Resistance, Volume I—A Review of the Literature," Wright-Patterson Air Force Base, Ohio, Technical Rept., Air Force Materials Lab., AFML-TR65-365, April 1966.

[3]Williams, A., "Heat Transfer Across Metallic Joints," *Mechanical and Chemical Engineering Transactions,* The Institution of Engineers, Australia, Paper No. 2305, 1968, pp. 247–254.

[4]Gex, R. C., "Thermal Resistance of Metal-to-Metal Contacts, an Annotated Bibliography," Lockheed Mission and Space Division, Sunnyvale, CA, Rept. S8-61-30, 1961.

[5]Moore, C. J., Atkins, H., and Blum, H. A., "Subject Classification Bibliography of Thermal Contact Resistance Studies," American Society of Mechanical Engineers, New York, Paper 68-WA/HT-18, 1968.

[6]Madhusudana, C. V., and Fletcher, L. S., "Contact Heat Transfer—The Last Decade," *AIAA Journal,* Vol. 24, March 1986, pp. 510–523.

[7]Fletcher, L. S., "Recent Developments in Contact Conductance Heat Transfer," *Journal of Heat Transfer,* Vol. 110, 1988, pp. 1059–1070.

[8]Madhusudana, C. V., "Contact Heat Transfer Between Coaxial Cylinders of Similar or Dissimilar Materials," *Proceedings of the ASME-JSME Thermal Engineering Joint Conference,* American Society of Mechanical Engineers, New York, Vol. 3, 1983, pp. 317–322.

[9]Madhusudana, C. V., "On Heat Flow Across Cylindrical Joints," *Proceedings of the Eighth International Heat Transfer Conference,* Vol. 2, American Society of Mechanical Engineers, New York, 1986, pp. 651–658.

[10]Timoshenko, S., and Goodier, J. N., *Theory of Elasticity,* 3rd ed., McGraw-Hill, New York, 1970, pp. 443–449.

[11]Dart, D. M., "Effect of Fin Bond on Heat Transfer," *ASHRAE Journal,* Vol. 1, May 1959, pp. 67–71.

[12]Gardner, K. A., and Carnavos, T. C., "Thermal Contact Resistance in Finned Tubing," *Transactions of ASME, Journal of Heat Transfer,* Vol. 82, Nov. 1960, pp. 270–293.

[13]Kulkarni, M. V., and Young, E. H., "Bimetallic Finned Tubes," *Chemical Engineering Progress,* Vol. 62, July 1966, pp. 68–71.

[14]Shlykov, Yu. P., and Ganin E. A., "Thermal Resistance of Metallic Contacts," *International Journal of Heat and Mass Transfer,* Vol. 7, 1964, 921–929.

[15]Eckels, P. W., "Contact Conductance of Mechanically Expanded Plate Finned Tube Heat Exchangers," American Society of Mechanical Engineers, New York, Paper 77-HT-24, 1977.

[16]Christensen, R. N., and Fernandes, H. V., "Contact and Fouling Resistances in Finned Tube Heat Exchangers," American Society of Mechanical Engineers, New York, Paper 83-HT-39, 1983.

[17]Ernest, T. L., Sheffield, J. W., and Sauer, H. J., "Finned Tube Contact Conductance: Characterizing the Integrity of the Mechanical Bond," *ASHRAE Transactions,* Vol. 91, Part 2A, 1985, pp. 85–99.

[18]Sheffield, J. W., Stafford, B. D., and Sauer, H. J., "Finned Tube Contact Conductance: Investigation of Contracting Surfaces," *ASHRAE Transactions,* Vol. 91, Part 1, 1985, pp. 85–99.

[19]Sheffield, J. W., Abu-Ebid, M., and Sauer, H. J., "Finned Tube Contact Conductance: Empirical Correlation of Thermal Conductance," *ASHRAE Transactions,* Vol. 91, Part 2A, 1985, pp. 100–117.

[20]Kennard, E. H., *Kinetic Theory of Gases,* McGraw-Hill, New York, 1938, pp. 312–315.

[21]Wood, R. A., Sheffield, J. W., and Sauer, H. J., "Thermal Contact Conductance of Finned Tubes: A Generalized Correlation," *ASHRAE Transactions,* Vol. 93, Part 2, 1987, pp. 786–796.

[22]Wood, R. A., Sheffield, J. W., and Sauer, H. J., "Thermal Contact Conductance of Finned Tubes: The Effect of Various Parameters," *ASHRAE Transactions,* Vol. 93, Part 2, 1987, pp. 798–809.

[23]Sheffield, J. W., Sauer, H. J., and Wood, R. A., "An Experimental Method for Measuring the Thermal Contact Resistance of Plate Finned Tube Heat Exchangers," *ASHRAE Transactions,* Vol. 93,

Part 2, 1987, pp. 776–785.

[24]Sheffield, J. W., Wood, R. A., and Sauer, H. J., "Experimental Investigation of Thermal Conductance of Finned Tube Contacts," *Experimental Thermal and Fluid Sciences,* Vol. 2, No. 1, 1989, pp. 107–121.

[25]Taborek, J. "Bond Resistance and Design Temperatures for High-Finned Tubes. A Reappraisal," *Heat Transfer Engineering,* Vol. 8, No. 2, 1987, pp. 26–34.

[26]Smith, C. E., Gunter, A. Y., and Victory, S. P., "Fin Tube Performance," *Chemical Engineering Progress,* Vol. 62, No. 7, 1966, pp. 57–67.

[27]Nho, K. M., and Yovanovich, M. M., "Measurement of Contact Resistance in Finned Tube Heat Exchangers," *ASHRAE Transactions,* Vol. 95, Part 1, Paper No. 3242, 1989.

[28]Nho, K. M. "Measurement of Contact Resistance in Finned Tube Heat Exchangers," M.S. Thesis, University of Waterloo, Waterloo, Ontario, Canada, 1987.

[29]Burgers, J. G. M., "Thermal Mechanical Models for Plate Fin-Tube Heat Exchangers," M. S. Thesis, University of Waterloo, Waterloo, Ontario, Canada, 1988.

[30]Rapier, A. C., Jones, T. M., and McIntosh, J. E., "Thermal Conductance of Uranium Dioxide/Stainless Steel Interface," *International Journal of Heat Mass Transfer,* Vol. 6, 1963, pp. 397–416.

[31]Sanderson, P. D., "Heat Transfer from Uranium Fuel to the Magnox Can in a Gas-Cooled Reactor," *Proceedings of International Developments in Heat Transfer Conference,* American Society of Mechanical Engineers, New York, 1961, pp. 53–64.

[32]Ross, A. M., and Stoute, R. L., "Heat Transfer Coefficient Between UO2 and Zircaloy-2," Atomic Energy of Canada Ltd., Ontario, Rept. CRFD-1075, 1962.

[33]Madhusudana, C. V., "Experiments on Heat Flow Through Zircaloy-2/Uranium Dioxide Surfaces in Contact," *Journal of Nuclear Materials,* Vol. 92, 1980, pp. 345–348.

[34]Brutto, E., Casagrande, I., and Perona, G., "Thermal Contact Resistance Between Cylindrical Metallic Surfaces," *Energia Nucleare,* Vol. 6, Aug. 1959, pp. 532–540.

[35]Cohen, I., Lustman, B., and Eichenberg, J. D., "Measurement of Thermal Conductivity of Metal Clad Uranium Rods during Irradiation," U.S. Atomic Energy Commission, Rept. WAPD-228, Aug. 1960.

[36]Cetinkale, T. N. and Fishenden, M., "Thermal Conductance of Metal Surfaces in Contact," *Proceedings of General Discussion on Heat Transfer,* Institution of Mechanical Engineers, London, 1951, pp. 271–275.

[37]Madhusudana, C. V., and Fletcher, L. S., "Gas Conductance Contribution to Contact Heat Transfer," AIAA Paper 81-1163, 1981.

[38]Williams, A. and Madhusudana, C. V., "Heat Flow Across Cylindrical Metallic Joints," *Heat Transfer 1970*, Elsevier, Amsterdam, Paper Cu 3-6, 1970, pp. 372–384.

[39]Novikov, I. I., Kokorev, L. S., and Delvin, N. N., "Experimental Investigation of Contact Heat Exchange Between Coaxial Cylinders in Vacuum," *Soviet Atomic Energy,* Vol. 33, June 1972, p. 559.

[40]Hsu, T. R., and Tam, W. K., "On Thermal Contact Resistance in Composite Cylinders," AIAA Paper 79-1069, June 1979.

[41]Wang, J. C. Y., and Nowak, E. S., "A Study of Thermal Constriction Resistance at the Interface of Double Tubes," *Proceedings of the Seventh International Heat Transfer Conference,* Vol. 2, Hemisphere, New York, 1982, pp. 87–92.

[42]Lemczyk, T. F., and Yovanovich, M. M., "New Methods and Methodology for Predicting Thermal Contact Resistance in Compound Cylinders and Finned Tubes," *Heat Transfer Engineering,* Vol. 8, No. 2, 1987, pp. 35–48.

[43]Madhusudana, C. V., and Litvak, A. L., "Thermal Contact Conductance of Composite Cylinders: An Experimental Study," AIAA Paper 88-2659, June 1988.

[44]Srinivasan, M. G., and France, D. M., "Non Uniqueness in Steady Heat Transfer in Prestressed Duplex Tubes—Analysis and Case History," *Journal of Applied Mechanics,* Vol. 52, June 1985, pp. 257–262.

[45]Barber, J. R., "Non-Uniqueness and Stability for Heat Conduction Through a Duplex Heat Exchanger Tube," *Journal of Thermal Stresses,* Vol. 9, 1986, pp. 69–78.

[46]Barber, J. R., "The Influence of Interface Thermal Contact Resistance on the Heat Transfer Performance of Prestressed Duplex Tube," *International Journal of Heat and Mass Transfer,* Vol. 29, No. 5, 1986, pp. 761–767.