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Thermal Contact Conductance of Composite Cylinders

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A review of the experimental and theoretical investigations of the heat transfer characteristics of composite cylinders is presented. While there are more studies devoted to thermal conduction through cylindrical contacts than there were six years ago, this phenomenon is rather underrepresented in the literature. Some studies are more theoretical in nature, dealing primarily with fundamental issues, and are broadly classified as general studies; others are experimentally based and are classified as applications studies. Tabulations of previously published correlations, and figures demonstrating the range of available data in these categories are also presented. Much of the available data for the general studies are presented in the form of a dimensionless thermal contact conductance as a function of dimensionless interface heat flux. A correlation for the prediction of the dimensionless contact conductance of composite cylinders has been proposed. Based on this review, it is evident that some areas of cylindrical thermal contact conductance have not been adequately investigated. There are still many gaps in our understanding of cylindrical contacts that need to be explored.

Nomenclature

A = correlation constant
a = inner radius
B = correlation constant
b = interface radius
C_i = model parameter
C_p = specific heat at constant pressure
C_v = specific heat at constant volume
c = outer radius
d = Vickers indention diameter, μm
d_o = tube outer diameter
E = elastic modulus
$F(\,)$ = eigenfunction
$f(\,)$ = interface heat flux distribution
fpi = number of fins per unit length
H = bulk microhardness
H_e = effective microhardness
H_m = Meyer hardness
h^* = dimensionless conductance
h_c = contact conductance
I = tube expansion, interference
k = conductivity
k_e = effective (geometric mean) conductivity
k_{g0} = gap gas thermal conductivity
k_n = eigenvalue
k_s = harmonic mean of thermal conductivity of joint materials
L = fin collar length
l = length
l_c = contact length
m = effective asperity slope (geometric mean)
m_i = asperity slope of surface i
N_{uc} = joint conductance
P = ambient pressure
P_{atm} = atmospheric pressure
P_c = contact pressure
Pr = Prandtl number
Q = thermal load
q'' = interface heat flux
q^* = dimensionless heat flux
R^* = overall thermal resistance without gap resistance
R_c = contact resistance, $1/h_c$
R_g = gap resistance
R_i = inside thermal resistance, including film and fouling
R_o = outside thermal resistance including film and fouling
r_i = inner radius of shell i
T = temperature
T_a = temperature at radius a, ambient bulk fluid temperature
T_b = temperature at radius b, fin base temperature
T_c = temperature at radius c
T_0 = fabrication temperature for fin and tube
t = fin thickness
u_c = initial interference at interface
Y = mean plane separation
$\boldsymbol{\alpha}$ = accommodation parameter
α_f = linear thermal expansion coefficient of fin material
α_t = linear thermal expansion coefficient of tube material
$\boldsymbol{\beta}$ = fluid parameter, coefficient of thermal expansion
ΔT_m = mean temperature difference across interface
δ_i = casing thickness for shell i
Λ = molecular mean free path
$\boldsymbol{\lambda}_i$ = thermal conductivity of inner tube
λ_o = thermal conductivity of outer tube
μ = parameter
ν = Poisson ratio
ρ = parameter
σ_i = surface roughness of material i
ϕ_a = contact spot half-angle
ϕ_b = half-angle > ϕ_a

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Introduction

WHILE studies dealing with thermal contact conductance between flat surfaces are common, conductance studies dealing with cylindrical contacts are limited and deal primarily with specific applications. Cylindrical contacts occur in such diverse applications as composite cylindrical tanks, space structures, power transmission lines, nuclear fuel elements, air conditioning systems, and pipelines. As a consequence, conduction through cylindrical contacts is an important phenomenon to understand.

Cylindrical joints behave differently than flat joints. For conductance through flat contacts, pressure (although dependent on interface heat flux, interface temperature level, heat flow direction, interstitial fluid, and the thermophysical properties of the materials that compose the joint) can be monitored and controlled independently of the heat flux through the interface, and results are often presented in terms of conductance as a function of interface pressure. For cylindrical contacts, in addition to the factors that influence interface pressure for flat surfaces, the interface pressure is also dependent on the initial degree of fit, and the differential expansion of the cylinders caused by the temperature difference at the interface and the temperature distribution within the individual cylindrical shells. Of the test parameters, only the heat flux through the joint may be independently controlled, therefore, the heat flux is far more important for cylindrical contacts than it is for flat contacts. As a consequence, the experimental thermal contact conductance results for cylindrical contacts are presented as a function of interface heat flux, a function of calculated interface pressure, or as a function of the temperature difference across one of the cylindrical shells.

Figure 1 illustrates the four different composite cylinder combinations of thick- and thin-shell geometries that can form a cylindrical interface, where a thin shell is defined (from a strength of materials point of view) as one whose thickness is either less than one-twentieth the nominal radius[1] or less than one-tenth the nominal radius.[2] Thin cylindrical shells are commonly found in such places as the cladding of superconducting wires, tension-wound finned tubes, and large diameter pipes. Thick cylindrical shells (including solid composite cylinders) may occur in such applications as nuclear fuel rods and composite pipes. In many studies the thickness of the cylindrical shell is not given and must be assumed. Unless otherwise specified, the studies reviewed in this article examine contacts formed by thick cylindrical shells. Figure 2 shows the nomenclature for radii and positive heat flux direction for a generic cylindrical joint.

In addition to surface irregularities such as waviness and roughness, conduction through cylindrical contacts involves other parameters such as out-of-roundness. Furthermore, placement of instrumentation for studies involving cylindrical contacts is more difficult than it is for flat contacts. It is probable that these additional complexities discourage investigations into thermal contact conductance of cylindrical contacts.

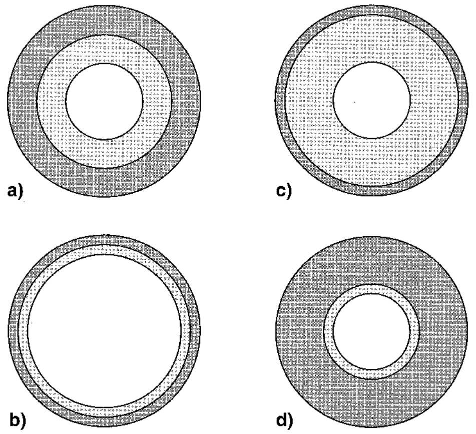

Fig. 1 Four different types of composite cylinders with inner–outer shell thickness combinations of a) thick–thick, b) thin–thin, c) thick–thin, and d) thin–thick.

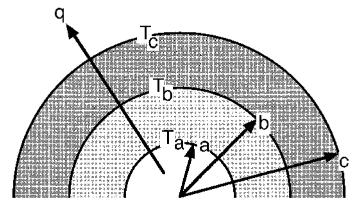

Fig. 2 Heat conduction through a thick-shell geometry, composite cylinder.

Nevertheless, conduction through cylindrical contacts is increasingly important and additional studies are warranted. This article provides an overall review of the analytical and experimental studies of the thermal contact conductance of composite cylinders and associated configurations, and provides a comparison of existing experimental data and correlations, insofar as possible.

Literature Review

A brief summary of selected investigations dealing with cylindrical thermal contact conductance is provided. Some studies are more theoretical in nature, dealing primarily with fundamental issues. Others are experimentally based, addressing applications. To facilitate this review, the studies have been categorized as either general studies or applications studies.

Madhusudana et al.[3] reviewed studies of theoretical and experimental investigations of heat transfer in compound cylinders, categorizing these studies as application oriented (and appropriate to specific materials and tubing) or fundamental (utilizing idealized surfaces and geometries). They found that there are significant holes in the state of knowledge of heat transfer through cylindrical contacts, and that only a modest effort has been made to fill these gaps. They conclude that there are several important shortcomings in the literature: contact resistance is measured indirectly from total resistance and computed material resistances; contact resistance is assumed to be constant (neglecting the effects of differential expansion); fundamental studies consider idealized surfaces that bear little resemblance to engineering surfaces; and that application-oriented studies generally refer to specific conditions and materials. Furthermore, they recognize that there is a dearth of data that can be used to confirm theoretical models.

Hrnjak and Sheffield[4] produced a similar review of plate–fin studies. They identify thermal contact conductance between the fin and tube as a neglected phenomenon, as far as most heat exchanger studies are concerned. They concluded that only a limited amount of experimental data is generally available, and that many parameters important to current contact conductance theories are not reported.

General Studies

General studies of thermal contact conductance through cylindrical contacts consist of investigations dealing with composite cylinders and cylindrical shells. For the most part, the heat flow through composite cylinders is axisymmetric and radial. While some of these studies are for specific materials and applications, it is fairly easy to extend their results to more general cases. Appendix A provides a summary of conductance correlations obtained from the investigations reviewed in this section.

Brutto et al.[5] conducted a study of compound tubes in a vacuum environment. Their study was motivated by the problem of extracting heat from clad cylindrical nuclear fuel elements at interface heat fluxes between 3×10^5 and 9×10^5 W/m^2. The experimental thermal contact conductance results are presented as a function of the temperature of the water surrounding the fuel elements, and cannot be compared to other published results of theories. They found that the method of joining the cladding to the fuel had a significant effect on the thermal behavior of the interface, because of the existence of metallurgical bonding, plastic deformation of tube layers, and contact pressure caused by residual elastic strain. The authors suggest surface roughness at the interface as a contributing factor to the contact resistance. They also recommend that the cladding process (i.e., how the cladding is bonded to the fuel) be taken into account when making studies in this field, suggesting further that a metallic bond between the fuel and the cladding material is superior to a mechanical bond.

Cohen et al.[6] studied the conductance between cylindrical uranium–dioxide pellets and thin stainless-steel cladding. They performed in-pile experiments, measuring the centerline temperature of the fuel pellet from 100°C to nearly 1200°C. They obtained experimental thermal contact conductance data as functions of contact pressure, which were calculated from the relative thermal expansion of the pellet and the cladding. They identified operating factors that influence the interface pressure (including fuel cracking, initial clearance, and assembly method). Among the interesting phenomenon observed was the variation of thermal contact conductance with each startup and as a function of time during the operating cycle.

Williams and Madhusudana[7] identified some basic problems associated with studies of conductance in cylindrical contacts, and presented the experimental results of two geometries: 1) partial cylindrical contacts (included angle less than 20 deg) and 2) full cylindrical contacts. The experimental thermal contact conductance data were obtained over a range between 1×10^5 and 5×10^5 W/m^2, with both air and vacuum as interstitial gasses. Some of the problems they encountered included difficulty in obtaining truly cylindrical contacts and the facilitation of a uniform heat flux through the interface. They reported that partial cylindrical contacts behave very similarly to flat contacts, so much so that they may be simplified as such for many cases. Full cylindrical contacts were found to be very susceptible to small out-of-roundness deviations in the interfacial area. They recommend interference fits to enhance the conduction through the joint, in fact, the thermal resistance through such interfaces was too small to be accurately determined.

Novikov et al.[8] conducted a study of the heat transfer between coaxial cylindrical casings in vacuum, and determined that the contact pressure is strongly related to the ratio of thermal expansion coefficients, the initial stress state of the composite tube, and the thermal load. One of their conclusions is that when the thermal expansion coefficient of the outer casing is greater than that of the inner cylinder, the interface pressure will decrease and that contact resistance will increase with increasing, radially outward thermal load. Their model takes into account the thermophysical and mechanical properties of the cylindrical casings, but neglects surface conditions and imperfections. The experimental data used for their model development were not available for comparison.

Tam[9] and Hsu and Tam[10] conducted experiments in air with composite cylinders, varying the heat flux and surface roughness of the inner side of the interface. They identified phenomena that might explain the difference in the thermal contact conductance behavior of materials in cylindrical contacts and flat contacts. Predictions of contact conductance calculated by modified flat contact models[11,12] are much lower than experimental data in Refs. 9 and 10. They propose as a possible explanation the lateral expansion of flat contacts, which would reduce the thermally induced strain at the interface and increase the associated microcontact area. They also identify the effects of surface waviness, interfacial heat flux, oxidation, and radius of curvature as important parameters that should be accounted for in future correlations.

Madhusudana and Fletcher[13] present results of thermal contact conductance tests for cylindrical interface heat fluxes between 8×10^3 and 4.2×10^4. They report that joints with an interference fit exhibit negligible thermal contact resistance, and that the increase of the joint conductance in air compared to the increase in vacuum was of the same order of magnitude as the ratio of the thermal conductivity of air to the amount of initial clearance. They further assert that the primary factors of importance for predicting the thermal contact conductance are the initial fit, the differential expansion due to the temperature gradient within the cylindrical shells, and the differential expansion due to the temperature difference across the interface. They suggest that, as a result of the coupling between the contact resistance, interface pressure, and interface temperature difference, predictions of the contact conductance must be solved iteratively.

Wang and Nowak[14] conducted a theoretical analysis of the interface between duplex tubes where there is a sector at the interface where the tubes are not in contact with each other. Both isothermal and isoflux boundary conditions at the interface were studied through the use of an electrical analog tank and a computer model of a representative sector of the duplex tube. Results in the form of a predictive model for the contact resistance are presented, but no thermal contact conductance data are given in their study.

Madhusudana[15] presents a predictive model for the thermal contact conductance through a cylindrical joint based on material properties, cylinder geometry, surface finish, and initial degree of fit. He illustrates the influence of thermal load, material combination, and interstitial fluid on interface pressure and contact conductance. In general, the model predicts that contact pressure increases with heat load. One interesting result of his model is that a material combination with a lower effective conductivity (defined as the harmonic mean of the two material thermal conductivities) may result in a higher contact conductance than a material combination of a higher effective conductivity, depending on the direction of the heat flow. He concludes that there is a strong relation between the contact pressure and material combination, and between conductance and the properties of the interstitial medium (especially at low thermal loads). He also states that there is a weak relation between the contact pressure and the properties of the interstitial medium. However, there are no comparisons with experimental data.

Lemczyk and Yovanovich[16] offer a predictive model for the conductance of cylindrical contacts based on advanced models for conductance of flat contacts that considers contact pressure, microhardness, and surface roughness. The model is iterative in nature, and results are presented in the form of thermal contact conductance as a function of an estimated contact pressure. Good agreement is found with the data of Hsu and Tam,[10] although some surface properties (which were not provided), were estimated. Comparisons are made with modified flat contact models of Shlykov and Ganin,[12] Veziroglu,[17] and Ross and Stoute,[11] demonstrating that the Lemczyk–Yovanovich model[16] is more effective for prediction of the contact conductance of realistic (rough and wavy) surfaces than the other three models. They recommend that, in future models, the difference between axial and circumferential roughness be taken into account.

Madhusudana and Litvak[18] conducted an experimental study of conduction through a composite cylinder, focusing on the design, construction, and validation of an experimental test facility. The test facility consisted of a composite cylindrical shell (stainless-steel 303 and aluminum 2011), which was heated on the interior by hot fluid and cooled on the exterior by cool fluid. They present the thermal contact conductance as a function of temperature difference across the interior cylinder, although they also give heat rates (and through calculation, interfacial heat fluxes). There is no mention of the clearance between the cylindrical shells or the surface characteristics (roughness and waviness). They call attention to the effect of the interstitial fluid and emphasize that the interface pressure can only be estimated, and should not be used to present data. They recommend that theoretical models be refined to take into account large-scale irregularities (such as out-of-roundness and waviness) at the interface.

Srinivasan and France[19] analytically studied heat transfer in prestressed duplex tubes. Their study, prompted by the erratic performance in the steam generator of an experimental nuclear reactor, suggested that the multiple operating conditions were explained by the time relaxation of the initial prestress. Analytical models showed that sufficiently low prestresses at the interface of the duplex tube would lead to nonunique solutions. This was explained by the shape of the thermal contact resistance curve assumed in their calculations. One suggested consequence of low prestress is the propagation of the noncontact region through the tube, leading to widespread separation of the layers of the tube. An increase in the temperatures throughout the tube and an increase in the heat flux in the contacting regions is also expected as a consequence of this separation. However, the results of the study cannot be compared with other published studies since the results are presented in arbitrary units.

Barber[20,21] also studied the nonuniqueness of analytic solutions for the temperature distribution in prestressed duplex tubes studies by Srinivasan and France.[19] Specifically, he examined the influence of the phenomenon on the axial temperature variation in the duplex tubes, and the stability of the temperature distribution solutions. He found that there is always an odd number of solutions, alternating between stable and unstable, and that the phenomenon is sensitive to a number of conditions, including the initial stress state of the composite tube, the temperature level at the interface, and the thermophysical properties of the tube materials.

Artyukhin et al.[22] constructed a computational algorithm to determine the thermal contact resistance between nuclear fuel pellets and cladding by solving the inverse heat transfer problem, with the aim of using the model for nonsteady experimental studies inside reactors. Their results suggest the optimal placement of temperature sensors within the fuel element and the means that may be used to analyze and process the data from nonsteady thermal experiments.

Danes and Simon[23] used a modification of the transient method used by Bourouga and Bardon[24] to determine the transient thermal contact resistance between two cylinders. They used several samples (constructed of an inner stainless-steel cylinder and an outer tin cylinder) and determined an exponential correlation dependent on temperature. They report a reduction in the measurement error of 50%. However, Bourouga and Bardon[24] cite an experimental uncertainty that is of the same order of magnitude as their measurement.

Application Studies

A majority of the application studies deal with heat exchangers, including the installation of fins or extended surfaces. Typical finned-tube configurations are shown in Fig. 3. Thermal contact resistance accounts for a significant portion of the thermal resistance in plate finned heat exchangers. Appendix B lists conductance correlations resulting from the investigations dealing with the thermal contact conductance of fin-tube interfaces. From a geometric viewpoint, tension wound, footed fins have a dual nature. At the foot, they might be expected to behave as thin cylindrical shells. At the point where the fin rises from the foot, they might be expected to behave as thick cylindrical shells. Therefore, analysis of the heat transfer through the fin–tube interface requires careful consideration of the geometry.

The effects of the thermal resistance of the fin–tube bond are of considerable interest. Kim[25] identified a major deficiency in the state of the art, specifically the lack of theoretical or empirical prediction techniques for the thermal contact resistance of all types of finned tubes. Kim[25] describes the objectives of a proposed study that was to investigate the effects of material properties, fin–tube geometry, and surface coating with the result of a correlation of joint conductance as a function of fin–tube geometry and mechanical properties.

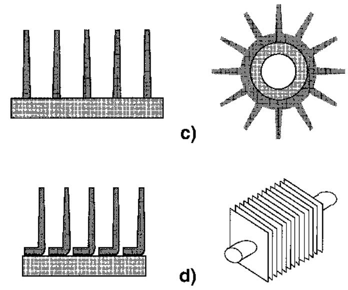

Fig. 3 Typical heat exchanger fin types: a) edge-wound *I* fins, b) edge-wound foot fins, c) extruded muff fins, and d) plate fins.

Dart[26] described a method of measuring the effect of fin bond on the heat transfer through the fin-tube contact and conducted a series of experimental tests to demonstrate the utility of the method. The method described in this seminal work was refined by Eckels[27,28] and later by Abuebid,[29] Sheffield et al.,[30] Ernest et al.,[31] Sheffield et al.,[32] Wood et al.,[33,34] Sheffield et al.,[35] and Sauer and Sheffield.[36] The method that uses geometric parameters to predict the heat transfer through the bond does not consider the effect of thermophysical properties.

Gardner and Carnavos[50] presented a model for the thermal contact resistance of the bond between various types of fins on tubing, based on finned-tube geometry (shown in Fig. 3), material properties, and fluid temperatures. They observed that the interface pressure (caused by elastic stresses in the tube and the fins) can be calculated with the method of Timoshenko and Goodier.[37] Experimental data include information from tests on tension wound fins and muff-type fins. They also proposed correlations to predict the gap resistance (bond resistance) as a function of operating conditions, material properties, and fabrication processes. They conclude that the bond-resistance of interference-fit finned tubes is negligible for air-cooled heat transfer equipment, but may be significant at extreme temperature level conditions. They also observe that the concentricity of the tube wall and the muff–fin base has little effect on the contact resistance, provided that the gap resistance is less than one-fifth the overall resistance, and the error introduced is less than 3%. They also observe that longitudinal variation of the interfacial diameters has a more dramatic effect on the thermal contact resistance.

Young and Briggs[38] and Kulkarni and Young[39] recognized that contact resistance can be a significant part of the overall thermal resistance in finned-tube heat exchangers. They produced design charts predicting the contact resistance as a function of air and fluid temperature operating conditions and classified them according to fin number and diameter. These charts assume an initial contact pressure and fabrication temperature, and were constructed from data for aluminum muff-type fins on steel tubes. Bond resistance (including contact resistance) is presented as a function of the temperature of the inside tube, which may not be uniform through the thickness or along the axis of the tube. Further, for different types of tubing or fin material, these charts may not be accurate, and they do not apply to different types of fin geometries such as tension wound fins.

Smith et al.[40] utilized a single tube test rig to characterize the performance of steel and aluminum tubes with aluminum tension-wound and footed fins. Both steady and cyclic operation tests were performed at tube temperatures between 330–375°C. They found that there was an increase in thermal resistance with tube wall temperature, and that thin fins deform with temperature (changing the pressure distribution of the interface).

Eckels[27,28] designed a test facility that used a large number of tube–fin interfaces in a tube coil to determine the average contact conductance between tubes and plate fins. The uncertainty of the thermal contact conductance values given by this technique were estimated to be on the order of 20%. Sheffield et al.[30,32,35] and Saver and Sheffield[36] used this technique in the studies reviewed here. In a second study, Eckels[28] utilized the previously designed test facility to obtain experimental values of the thermal contact conductance through a dry, mechanically expanded plate-finned tube, and produced a correlation that is dependent on geometry alone. He found that the values of temperature, mass flow rate, and the fin resistance were the most significant sources of error in his relation, and that the error in the results increases with heat flux (approximately 15% at 5.676 kW/m^2 K, and 30% at 11.3 kW/m^2 K).

Kuntysh et al.[41] studied the thermal contact resistance of L-footed fins. They used two different tube boundary conditions in their investigation: 1) constant tube wall temperature and 2) a varying tube temperature. They provide two power-law correlations of contact resistance as a function of interface heat flux. Their correlation does not take into account the common thermophysical properties and surface characteristics that are considered in more recent thermal contact conductance theories, nor are these parameters presented in their study. Without these parameters, their data (thermal contact resistance as a function of heat flux) are difficult to use in any subsequent modeling without assuming too much information. Further, their correlation does not explicitly have any dependence on the boundary conditions in tests used to generate the empirical data.

Stafford[42] and Sheffield et al.[43] used scanning electron microscopy to study the effects of expansion and other geometric parameters (such as waviness) on the characteristics of tube–fin interface surfaces. They determined that tube surface roughness decreases with decreasing fin number (fins per unit length) and increases with increasing tube expansion.

Christensen and Fernandes[44] studied the contact and fouling resistances in pneumatically expanded finned-tube heat exchangers, constructed of copper fins on copper, copper–nickel, or stainless–steel tubes. They presented order of magnitude results that suggest that the contact resistance is greater than the fouling resistance. They report an uncertainty in the contact resistance results that is of the same order as the contact resistance. Their thermal contact resistance data are not presented as functions of interface temperature, pressure, or heat flux. Further, they do not mention the influence of thermophysical properties or surface conditions on the thermal contact resistance.

Abuebid[29] and Sheffield et al.[30] used the method of Eckels[27] to determine the contact conductance of plate-finned, mechanically expanded tubes. Although Sheffield et al.[30] identify many parameters that are commonly accepted as being of importance in thermal contact conductance models, their correlation has no dependence on material properties, exhibits no influence of contact pressure or heat flux, and neglects surface roughness and out-of-roundness. The correlation is essentially a least-squares fit through data for a specific geometry.

Experimental studies investigating the thermal contact resistance of plate-finned tube coils in a vacuum were conducted.[31–36] In their apparatus, the plate fins of several tubes were joined to form tube sheets. The apparatus circulated cold water through the outer banks of tubes and hot water through a center bank of tubes in the coil. They assumed that the tube conduction resistance and contact resistance are the same for both the hot and cold tubes. However, since the contact resistance is dependent on the differential expansion of the contacting materials, it is indirectly dependent on the temperature.

Ernest et al.[31] investigated the quality of the mechanical bond between the fins and the tube, determining that a pullout test is an appropriate measure of the strength of the bond, and thus, the quality of the contact. The actual contact area, however, is dependent upon surface characteristics and fabrication process, therefore, such a pull test may not be appropriate for all configurations. Sheffield et al.[32] describe a technique developed to test plate–fin tube heat exchangers and to investigate the effects of varying geometry and manufacturing method.

Wood et al.[33,34] conducted experimental studies investigating the thermal contact resistance of plate-finned tube coils, and the effects of various tube coil parameters on the thermal contact resistance using the same apparatus of Sheffield et al.[30] The results seem to imply that there is no dependence of the contact conductance on temperature, and roughly half of the results presented are within 20% of the experimental correlation. In their second study, Wood et al.[34] conducted 32 experiments that examined the effects of the number of fins per inch, tube passes, fin conduction shape factor, fin–tube interference, and fin thickness on the contact conductance. They concluded that thermal contact conductance increases with interference, but decreases with tube diameter. Such conclusions are appropriate since interference fits increase the contact area and increased tube diameter usually involves more out-of-roundness and a nonuniform pressure distribution.

Sheffield et al.[35] and Sauer and Sheffield[36] conducted an experimental study investigating the thermal contact resistance of plate-finned tube coils in a vacuum. The apparatus was similar to that of their previous study. They again assumed that the tube conduction resistance and contact resistance are the same for both the hot and cold tubes. The correlation presented in this article extends the range of the one presented in their 1985 study.[30]

Nho and Yovanovich[45,46] experimentally studied the effects of surface condition on the thermal contact conductance of plate-finned tube heat exchangers. The method used was similar to that used by Sheffield et al.,[35] except that the fins appeared to be instrumented with more thermocouples. In addition to the same fin and tube parameters generally provided in this type of study, surface condition information including hardness, roughness, and asperity slope for both the tube and fin surfaces was provided. The hardness and conductivity of the oxide layer found on the contacting surfaces were also given, although no new correlation that uses this information was provided. They also examined the contact between the fin and the tube and found that the apparent contact area varies with the amount of expansion of the tube into the fin, causing the foot of the fin to either curl, buckle, or to displace the heel of the adjacent fin foot. Each of these factors affects the contact area, and thus, the conductance. The averaged conductance for each of the finned-tube units compares well with the model of Sheffield et al.,[35] being within the uncertainty of the model for most of the finned-tube units.

Egorov et al.[47] studied contact heat transfer resistance in finned-duplex tubes, which were manufactured by either drawing or pressing. They present results of contact conductance as a function of interfacial heat flux. They observed the effects of unstable contact between the tube layers, which was investigated in greater detail by Srinivasan and France[19] and Barber.[20, 21]

Matal et al.[48] studied duplex tubes for use in a liquid-sodium-heated, steam-generator component of a nuclear powerplant. The two tubes studied have either a set of grooves cut into the outer tube at the interface, or a set of three lands on the inner tube at the interface, to facilitate leak checking. Thermal contact resistance is given as a function of the temperature difference between the liquid sodium inside the tubes and the steam outside the tubes, and cannot be compared to other published results. Other results are given as a function of time. Many parameters are not reported, including surface finish, conductivity of the tube material, and interface heat flux. Further, in contrast to their figure relating thermal contact resistance to temperature difference, they claim to have perfect contact between the two tube layers.

Huang[49] used a finite element technique to model the flow of heat between tubes connected by a plate fin to determine the conduction resistance associated with the geometry. This method can be used in conjunction with experimental data for plate-finned tube heat exchangers to determine the resistance resulting from contact between the tubes and fins.[31–36] Previously, electrical analog techniques were used to determine the effect of the fin–tube geometry on the conduction resistance. The results of the method were compared with experimental data of Sheffield et al.[35] For 17 of the 32 datasets used for comparison, Huang[49] found that the average absolute error caused by measurement was 78%, and the average absolute error caused by the one-dimensional model was 68%. Further, his calculations for the same 17 datasets show that while the one-dimensional model used in the experimental study results in a contact resistance of approximately 20% of the total resistance, a modification of the one-dimensional model results in a thermal contact resistance of approximately 16% of the total resistance. A two-dimensional model results in a thermal contact resistance of approximately 11%, which indicates that analyses based on previous experimental studies have overestimated the fraction of the total thermal resistance caused by contact resistance by about 8%. Huang[49] concludes that his analysis would improve the experimental results because it does not assume a one-dimensional conduction between hot and cold tubes, and is able to avoid the error associated with that assumption.

Results and Discussion

To provide a comparison of the published correlations for the thermal conductance of cylindrical contacts, the models were categorized as shown in Appendices A and B. Appendix A compares correlations that are more fundamental in nature, and have general applicability. These correlations borrow heavily from flat contact models, and have (for the most part), the familiar hardness and roughness terms. Newer models for heat transfer through flat contacts may be modified to better account for out-of-roundness and longitudinal waviness. Appendix B compares the available correlations for conductance through cylindrical contacts found in finned tubes and tube collars. Much of this work is empirical, and thus applicable only to specific geometries and material combinations. Of particular interest is the absence of hardness and surface characteristics such as roughness or waviness terms in the correlations.

Figure 4 shows a comparison between the conductance data of Hsu and Tam[10] as a function of apparent contact pressure and the models of Ross and Stoute,[11] Shlykov and Ganin,[12] Vezeroglu,[17] and Lemczyk and Yovanovich.[16] Clearly, the model of Lemczyk and Yovanovich[16] (using the effective hardness) fits the data well, whereas the other models underpredict the data to varying degrees (between 20% for Ross and Stoute[11] and 60% for Vezeroglu[17]). For flat and near-flat contacts, pressure is often chosen as an independent experimental parameter. In the case of cylindrical contact conductance phenomenon, contact pressure cannot be easily measured, much less independently controlled.

Published experimental data are shown in Fig. 5 to demonstrate the range of available information as a function of interface heat flux. While heat flux is of relatively little influence for flat and near flat contacts, it appears to be very influential in cylindrical contact conductance phenomenon. The conductance-interface heat flux data fall in families, which can be nondimensionalized.

Figure 6 presents dimensionless thermal contact conductance as a function of dimensionless interface heat flux. Heat flow is assumed to be radially outward through an interface formed by two thick cylindrical shells. The terms selected to form the dimensionless heat flux were chosen to form meaningful groupings:

$$h^* = \left(\frac{h_c R_e}{k_e m}\right)$$
$$q^* = \left(\frac{q_i'' \beta_e R_e}{k_e}\right)\left(\frac{E_e}{H_e}\right)\left(\frac{\beta_i}{\beta_o}\right)^3 \left[\frac{1}{2}\left(1 + \frac{P}{P_{\mathrm{atm}}}\right)\right]^2 \tag{1}$$

The dimensionless conductance is the familiar dimensionless term used in flat contact studies, where contact conductance is nondimensionalized by the ratio of the effective roughness to the product of the effective conductivity and asperity slope. The first term in the dimensionless heat flux is a nondimensionalization of the heat flux with effective (geometric mean) thermal expansion, effective surface roughness, and effective conductivity. The second term was chosen to account for the effects of effective hardness and elasticity, increasing hardness

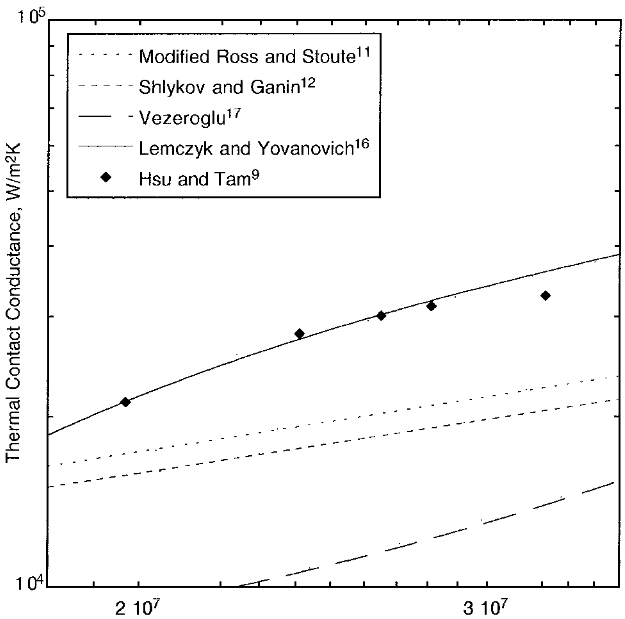

Fig. 4 Comparison between the experimental data of Hsu and Tam[10] for 4.826-μm Al2011 inside 1.65-μm SS304 and models of Ross and Stoute,[11] Shlykov and Ganin,[12] Vezeroglu,[17] and Lemczyk and Yovanovich.[16]

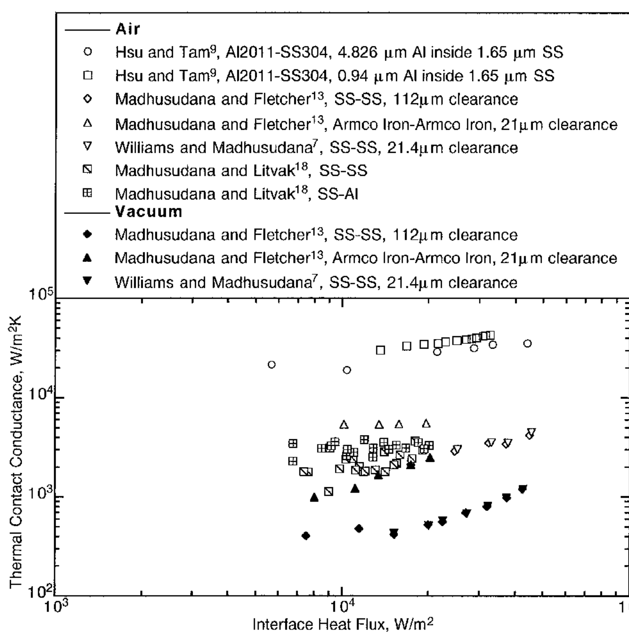

Fig. 5 Experimental thermal contact conductance data for cylindrical contacts as functions of interface heat flux.

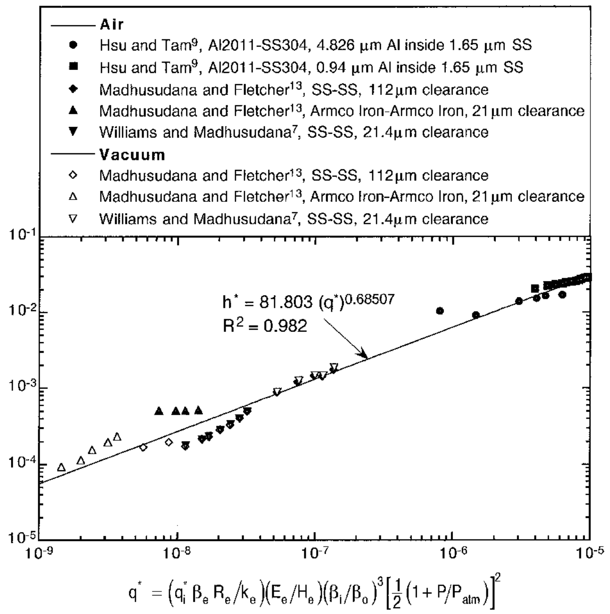

Fig. 6 Dimensionless thermal contact conductance data for cylindrical contacts as functions of dimensionless interface heat flux.

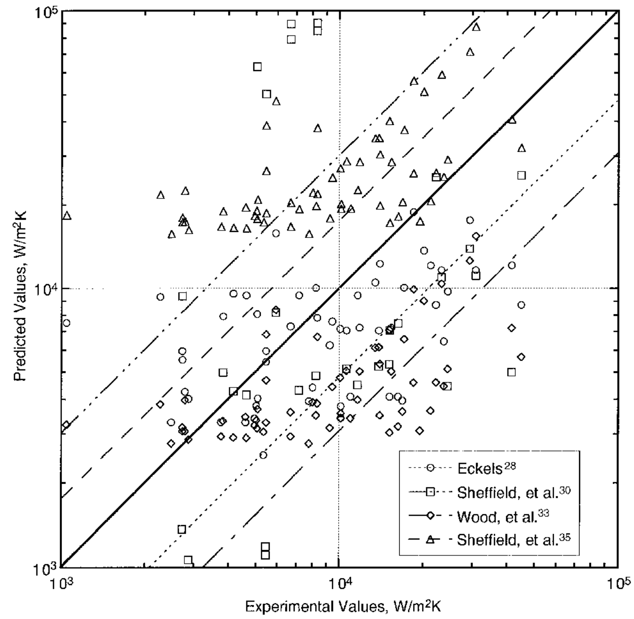

Fig. 7 Comparison of correlation predicted values of contact conductance with measured values for plate-finned tubes.

will diminish the conductance, whereas increased elasticity is expected to increase it. The ratio of the coefficients of thermal expansions was chosen to account for the effect of the differential thermal expansion of the two shells. If the inner shell expands faster than the outer shell, then the interface pressure will increase, and so will the conductance. The fourth term is included to represent the effects of an interstitial gas. Conductance is expected to be higher when there is an interstitial gas. The whole number exponents of the third and fourth terms were selected to align the data. Since the calculation of the interface pressures is beyond the scope of this study, no attempt is made to account for their influence. As predicted by Madhusudana,[15] the effect of interface fluid on the conductance is smaller than that of the relative expansion of the cylinders that make the interface. A power-law fit through the experimental data serves as a point of departure for further analytical studies:

$$h^* = 81.803(q^*)^{0.68507} \tag{2}$$

This correlation has a Pearson's r^2 value of 0.982, indicating a fairly good fit.

Figure 7 compares the relative fit of various plate-finned tube correlations with experimental data. Experimental data from the studies of Dart,[26] Eckels,[28] Sheffield et al.,[32] and Nho and Yovanovich[45,46] were compared with the values predicted by the correlations of Eckels,[28] Sheffield et al.,[30] Wood et al.,[33] and Sheffield et al.[35] at the same conditions. The horizontal axis indicates the measured conductance value, and the vertical axis indicates the conductance value predicted by the correlations. The data points in the figure are placed at the intersection of the experimental value and the predicted value for a specific set of conditions. A solid line at 45 deg to the horizontal indicates where the data points would fall if the experimental results were exactly predicted by a correlation. Straight lines parallel to the 45-deg solid line are a least-squares fit through all of the data points for each of the four correlations, and indicate the average overprediction or underprediction of the corresponding correlation. Lines that are above the solid line indicate that the corresponding correlations overpredict the experimental data, whereas lines that are below the solid line indicate that the corresponding correlations underpredict the experimental data. The patterns made by the data on this figure are also of interest. While the data are scattered over the entire domain of the plot, a careful analysis of the data indicates that conductance values for finned tubes are generally low.

Examination of Fig. 7 shows that the correlation of Eckels[28] underpredicts the data by approximately 50%. The pattern formed by the data points is a wideband across the center part of the figure. The correlation of Sheffield et al.[30] overpredicts the experimental values by approximately 200%. Also, the correlation both underpredicts and overpredicts the experimental data (as exhibited by the wide scatter of corresponding data points). Further, the values predicted by this correlation span more than two orders of magnitude, whereas the experimental data spans slightly more than one. Some of the data points for this correlation lie near the bottom of the figure, whereas others lie near the top. This wide range of predicted values may be a result of the large exponent of the fin thickness-tube diameter ratio. The correlation of Wood et al.[33] underpredicts the data by approximately 70%, and the associated data points exhibit a banded pattern across the lower part of the figure. The correlation by Sheffield et al.[35] overpredicts the data by an average of approximately 75%. The data points associated with the correlation exhibit a banded pattern across the upper part of the figure. The proposed correlations of Wood et al.[33] and Sheffield et al.[35] have limited utility at low values of thermal contact conductance, since both have a lower bound inherent in the form of the correlation.

Conclusions and Recommendations

Cylindrical contacts occur in many diverse applications. As a consequence, conduction through cylindrical contacts is an important phenomenon to understand. Cylindrical joints behave differently than flat joints. In flat contacts, interface pressure can be monitored and controlled independently of the heat flux through the interface, and thermal contact conductance results are often presented as a function of interface pressure. In cylindrical contacts, the interface pressure is dependent on the initial degree of fit and the differential expansion of the cylinders (which is because of the temperature difference at the interface and because of the temperature distribution within the individual cylindrical shells). As a consequence, the heat flux is far more important for cylindrical contacts than it is for flat contacts.

This article provides a comparative study of the thermal contact conductance in cylindrical contacts for selected materials and conditions. While there are more studies devoted to thermal conduction through cylindrical contacts than there were six years ago, this phenomenon is still rather underrepresented in the literature. There is a need for more quality experimental data. The available experimental data may not include all of the information required by current contact conductance models, including material designation, microhardness, or surface roughness and waviness (axially and circumferentially). Sufficiently complex models may be able to more adequately predict the behavior of specific finned-tube applications.

Based on this review, it is evident that some areas of cylindrical thermal contact conductance have not been adequately investigated, including the effects of macroscopic eccentricities such as out-of-roundness and waviness. There are still many gaps in our understanding that need to be explored.

Appendix A: General Cylindrical Conductance Correlations

Modified flat contact model, with some dependent on thermomechanical properties (Ross and Stoute[11]):

$$h_c = \frac{k_c P_c}{0.05H\sqrt{(\sigma_1^2 + \sigma_2^2)/2}}$$

Modified flat contact model, with some dependence on thermomechanical properties (Shlykov and Ganin[12]):

$$\frac{1}{R_c} = 4.2 \times 10^4 \frac{k_s P_c}{d_o(345 \text{ MPa})} + \frac{k_{g0}}{\sigma_1 + \sigma_2}$$

Modified flat contact model, with some dependence on thermomechanical properties (Veziroglu[17]):

$$N_{uc} = \frac{B^+ C^+}{\eta^+ \tan^{-1}[(1/C^+)\sqrt{1 - (1/N_{uc})} - 1]}$$

where

$$\begin{bmatrix} B^+ = 0.335(C^+)^{0.315(1_c/1)0.137} & \\ C^+ = (P_c/H_m)^{1/2} & \eta^+ = k_{g0}k_s \end{bmatrix}$$

Semitheoretical model for thin shells and power-law correlations for resistance as a function of instantaneous interface pressure, with some dependence on thermophysical properties (Novikov et al.[8]).

$P_c \leq$ yield stress:

$$R_c = \frac{(P_c - P_{c0})}{Q\alpha_1 r_1}\left\{\frac{r_1^{3/2}[3(1 - \nu_1^2)]^{1/4}}{E_1\delta_1^{3/2}} + \frac{r_2^2[3(1 - \nu_2^2)]^{1/4}}{E_2\delta_2 r_1^{1/2}\sigma_1^{1/2}}\right\}$$

Theoretical model for duplex tubes with spot contacts and an arbitrary heat flux, constant fluid temperatures, and steady, two-dimensional temperature distribution (Wang and Nowak[14]):

$$R_c = \frac{2k}{\pi\phi_a}\left\{\frac{1}{\boldsymbol{\lambda}_i}\sum_{n=1}^{\infty}\frac{1}{n}F(k_n\boldsymbol{\phi}_a)\left[\frac{(b/a)^{2k_n} - 1}{(b/a)^{2k_n} + 1}\right]\frac{\sin k_n\boldsymbol{\phi}_a}{k_n\phi_a}\right.$$
$$\left. + \frac{1}{\boldsymbol{\lambda}_o}\sum_{n=1}^{\infty}\frac{1}{n}F(k_n\phi_a)\left[\frac{(c/b)^{2k_n} - 1}{(c/b)^{2k_n} + 1}\right]\frac{\sin k_n\phi_a}{k_n\phi_a}\right\}$$

where $F(k_n\phi_a) = \int_0^{\phi_a} f(\phi)\cos(k_n\phi)\, d\phi$.

General theoretical cylindrical model; C_i are calculation parameters, with some dependence on composite cylinder geometry, material properties, gas properties, and surface characteristics (Madhusudana[15]):

$$h_c = \left(\frac{k}{\sigma}\right)\left[C_9\left(\frac{P_c}{H}\right)^{0.94} + \frac{k_{g0}}{C_{11}\sigma}\right]$$

where

$$P_c = \frac{E_1}{C_1(C_2 + \nu_2) + (C_3 - \nu_1)_i}\left(\alpha_1\Delta T\left\{(C_4 - C_5C_6C_7)\right.\right.$$
$$\left.\left. + \left[\frac{C_{10}(1 + C_7)}{2\,\ell n(b/a)}\right]\left[\frac{1}{C_9(P_c/H)^{0.94} + C_{11}(k_{g0}/k)}\right]\right\} + \frac{u_c}{b}\right)$$

Modification of standard conductance model for conforming surfaces, with calculated interface pressures. P_c is calculated through an iterative process that involves integration (Lemczyk and Yovanovich[16]): $1 \times 10^{-5} \leq P_eH_e \leq 1 \times 10^{-2}$; $2.34 \leq Y/\sigma \leq 4.26$; $0.14\ \mu\text{m} \leq \sigma \leq 14\ \mu\text{m}$; $9.33\ \mu\text{m} \leq \sigma/m \leq 40\ \mu\text{m}$; $0.015 \leq m \leq 0.35$; $0.001 \leq \Lambda_0/\sigma \leq 1.5$; $1 \leq \beta \leq 2$; $0.04\ \mu\text{m} \leq \Lambda_0 \leq 0.19\ \mu\text{m}$; $1 \times 10^{-4} \leq k_{g0}/k_s \leq 2 \times 10^{-2}$:

$$h_c = 1.25\frac{k_s m}{\sigma}\left(\frac{P_c}{H_e}\right)^{0.95} + \frac{k_{g0}}{Y + \alpha_a\beta\Lambda}$$

where

$$\beta = \left(\frac{2c_p/C_v}{c_p/c_v + 1}\right)\frac{1}{Pr}$$

Appendix B: Correlations for Conductance Through Cylindrical Contacts in Finned Tubes and Tube Sheets

For interference-fit finned-tubes, pressure based on elastic theory. Constant fluid temperatures, fin base, and tube wall are concentric and smooth, effect of fluid pressure is negligible (Gardner and Carnavos[50]).

$P < 0.43(t/d - t)^{0.5}$:

$$R_c = \rho\left\{(\alpha_f - \alpha_t)(T_h - T_0) - \mu p_{c0} - \left[\alpha_f\left(1 - \frac{r_0}{R^* + R_g}\right)\right.\right.$$
$$\left.\left. - \alpha_t\left(1 - \frac{r_i}{R^* + R_g}\right)\right](T_h - T_a)\right\}$$

Empirical curve fit, air–air heat transfer, coaxial circular find, and interface heat flux between 25–70 kW/m[2] (Kuntysh et al.[41]).

Constant tube wall temperature, between 350–365 K:

$$R_c = 0.552q_c^{-0.304}$$

Varying tube wall temperature, approximately linear from 345 to 510 K:

$$R_c = 0.004185q_c^{1.115}$$

Semitheoretical, apropos to a specific geometry and material combination, no temperature or material property dependence (Eckels[28])

$$h_c = 2.5556 \times 10^6\left\{\frac{t}{d_o}\left[\frac{1}{(fpi/t) - 1}\right]^2\right\}^{0.6422}$$

Mechanically expanded tubes and tube collars, minor dependence on material properties, no influence of contact pressure or heat load, apropos to a specific geometry and material combination, with no temperature dependence (Sheffield et al.[30]): $6.73\% \leq \text{error} \leq 18.35\%$; $d_o = 0.009525$ m; $6 \leq N \leq 20$; $0.001196\text{ m} \leq L \leq 0.004219$; and $0.000165\text{ m} \leq I \leq 0.000279$ m:

$$h_c = 2.815 \times 10^{23}\left(\frac{t}{d_o}\right)^{10.035}\left(\frac{I + 0.0005207}{L}\right)^{2.867}$$

Mechanically expanded, plate-finned tubes—copper tubes and aluminum fins, mechanically expanded tubes, apropos to a specific geometry and material combination, with no temperature dependence; nearly one-half of the coils tested fell within 20% of this correlation (Ref. 33): 0.00635 m $\leq d_o \leq$ 0.015875 m; 6 $\leq fpi \leq$ 18; 0.000114 m $\leq t \leq$ 0.000231 m; 0.000076 m $\leq I \leq$ 0.000191 m:

$$h_c = 5.67826 \exp\left\{6.092 + 2.889\left[\left(\frac{I fpi\, d}{d_o}\right)^{0.75} (t\, fpi)^{1.25}\right]\right\}$$

Mechanically expanded, plate-finned tubes, with minor dependence on material properties, no influence of contact pressure or heat load—least-squares fit for data apropos to a specific geometry and material combination, and no temperature dependence (Ref. 35): 0.009525 m $\leq d_o \leq$ 0.015875 m; 6 $\leq fpi \leq$ 18; 0.000102 m $\leq t \leq$ 0.000231 m; 0.000076 m $\leq I \leq$ 0.000191 m:

$$h_c = 5.67826 \exp\left\{7.828 + 2.889\left[\left(\frac{I fpi\, d}{d_o}\right)^{0.75} (t\, fpi)^{1.25}\right]\right\}$$

References

[1]Cook, R. D., and Young, W. C., *Advanced Mechanics of Materials*, 1st ed., Macmillan, New York, 1985.

[2]Shigley, J. E., *Mechanical Engineering Design*, 2nd ed., McGraw–Hill, New York, 1972.

[3]Madhusudana, C. V., Fletcher, L. S., and Peterson, G. P., "Thermal Conductance of Cylindrical Contacts—A Critical Review," *Journal of Thermophysics and Heat Transfer*, Vol. 4, No. 2, 1990, pp. 204–211.

[4]Hrnjak, P. S., and Sheffield, J. W., "Contact Resistance in Plate Finned Tubes Heat Exchangers: State-of-the-Art," *Proceedings of the 9th International Heat Transfer Conference*, Vol. 5, Hemisphere, New York, 1990, pp. 9–14.

[5]Brutto, E., Casagrande, I., and Perona, G., "Thermal Contact Resistance Between Cylindrical Metallic Surfaces," *Energia Nucleare*, Vol. 6, No. 8, 1959, pp. 532–540.

[6]Cohen, I., Lustman, B., and Eichberg, J. D., "Measurement of Thermal Conductivity of Metal Clad Uranium Rods During Irradiation," U.S. Atomic Energy Commission, Bettis Atomic Power Lab. Rept., WAPD-228, West Mifflin, PA, Aug. 1966.

[7]Williams, A., and Madhusudana, C. V., "Heat Flow Across Cylindrical Metallic Contacts," *Heat Transfer 1970*, Elsevier, Amsterdam, 1970, Paper Cu 3-6.

[8]Novikov, I. I., Kokorev, L. S., and Del'vin, N. N., "Experimental Heat Exchange Between Coaxial Cylinders in Vacuum," *Atomnaya Énergiya*, Vol. 32, No. 6, 1972, pp. 474, 475.

[9]Tam, W. K., "On Thermal Contact Resistance of Compound Cylinders," M.S. Thesis, Dept. of Mechanical Engineering, Univ. of Manitoba, Winnipeg, MB, Canada, 1976.

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

[11]Ross, A. M., and Stoute, R. L., "Heat Transfer Coefficient Between UO_2 and Zircaloy-2," Atomic Energy of Canada, Rept. CRFD-1075, Ottawa, ON, Canada, 1962.

[12]Shlykov, Y. P., and Ganin, Y. A., "Thermal Resistance of Metallic Contacts," *International Journal of Heat and Mass Transfer*, Vol. 7, No. 8, 1964, pp. 921–929.

[13]Madhusudana, C. V., and Fletcher, L. S., "Thermal Conductance of Cylindrical Contacts," *Proceedings of the 8th Canadian Congress of Applied Mechanics*, Univ. of Moncton, Moncton, NB, Canada, June 1981, pp. 755, 756.

[14]Wang, J. C. Y., and Nowak, E. S., "A Study of Thermal Construction Resistance at the Interface of Double Tubes," *Proceedings of the 7th International Heat Transfer Conference*, Vol. 2, Hemisphere, New York, 1982, pp. 87–92.

[15]Madhusudana, C. V., "On Heat Flow Across Cylindrical Contacts," *Proceedings of the 8th International Heat Transfer Conference*, Vol. 2, Hemisphere, New York, 1986, pp. 651–658.

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

[17]Veziroglu, T. N.,"Correlation of Thermal Contact Conductance Experimental Results," Vol. 20, Progress in Astronautics and Aeronautics, AIAA, New York, 1967, pp. 879–907.

[18]Madhusudana, C. V., and Litvak, A. L., "Thermal Contact Conductance of Composite Cylinders: An Experimental Study," *Journal of Thermophysics and Heat Transfer*, Vol. 4, No. 1, 1990, pp. 79–85.

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

[20]Barber, J. R., "Non-Uniqueness and Stability for Heat Conduction Through a Duplex Heat Exchanger Tube," *Journal of Thermal Stresses*, Vol. 9, No. 1, 1986, pp. 69–78.

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

[22]Artyukhin, E. A., Nenarokomov, A. B., Tryanin, A. P., Utenkov, S. A., and Yakovlev, V. V., "Identification of Contact Thermal Resistances in Nuclear Reactor Fuel Elements. 1. Development of Algorithms," *Inzhenerno-Fizicheskii Zhurnal*, Vol. 60, No. 3, 1991, pp. 478–487.

[23]Danes, F. E., and Simon, G., "Study of Thermal Contact Resistance in a Closed Cylindrical Geometry," *Science of Heat: Progress and Actual Challenges*, Vol. 2, Societé Française des Thermiciens 90: Science of Heat Colloquium: Progress and Actual Challenges, Nantes, France, 1990, pp. 217–220.

[24]Bourouga, B., and Bardon, J. P., "Mesures des Resistances Thermiques Superficielle et de Contact au Moyen de la Constante de Temps Fondamentale," *International Journal of Heat and Mass Transfer*, Vol. 35, No. 2, 1992, pp. 361–373.

[25]Kim, H. C., "Fin Bond—ASHRAE Project," Interoffice Memo, Gulf Western Manufacturing, Bohn Group, Feb. 28, 1978.

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

[27]Eckels, P. W., "Design of a Contact Conductance Experiment for Plate Finned Tube Heat Exchangers," Westinghouse Research Labs., Document 73-1E9-HEATA-M3, Pittsburgh, PA, 1974.

[28]Eckels, P. W., "Contact Conductance of Mechanically Expanded Plate Finned Tube Heat Exchangers," Westinghouse Research Labs., Document 77-1E9-SURCO-P1, Pittsburgh, PA, 1977.

[29]Abuebid, M. A., "A Thermal Contact Conductance Correlation for Mechanically Expanded Finned Tube Heat Exchangers," M.S. Thesis, Dept. of Mechanical and Aerospace Engineering, Univ. of Missouri, Rolla, MO, 1984.

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

[31]Ernest, T. L., Sheffield, J. W., and Sauer, H. J., Jr., "Finned Tube Contact Conductance: Characterizing the Integrity of the Mechanical Bond," *ASHRAE Transactions*, Vol. 91, Pt. 2, 1986, pp. 85–98.

[32]Sheffield, J. W., Sauer, H. J., Jr., and Wood, R. A., "An Experimental Method for Measuring the Thermal Contact Resistance of Plate Finned Tube Heat Exchangers," *ASHRAE Transactions*, Vol. 93, Pt. 2, 1987, pp. 776–795.

[33]Wood, R. A., Sheffield, J. W., and Sauer, H. J., Jr., "Thermal Contact Conductance of Finned Tubes: A Generalized Correlation," *ASHRAE Transactions,* Vol. 93, Pt. 2, No. 3103 (RP-446), 1987, pp. 786–797.

[34]Wood, R. A., Sheffield, J. W., and Sauer, H. J., Jr., "Thermal Contact Conductance of Finned Tubes: The Effect of Various Parameters," *ASHRAE Transactions*, Vol. 93, Pt. 2, 1987, pp. 798–810.

[35]Sheffield, J. W., Wood, R. A., and Sauer, H. J., Jr., "Experimental Investigation of Thermal Conductance of Finned Tube Contacts," *Experimental Thermal and Fluid Science*, Vol. 2, 1989, pp. 107–121.

[36]Sauer, H. S., and Sheffield, J. W., "Finned-Tube Contact Conductance Studies," *ASHRAE Journal*, Vol. 29, Nov. 1987, p. 48.

[37]Timoshenko, S. P., and Goodier, J. N., *Theory of Elasticity*, McGraw–Hill, New York, 1951.

[38]Young, E. H., and Briggs, D. E., "Bond Resistance of Bimetallic Finned Tubes," *Chemical Engineering Progress*, Vol. 61, No. 7, 1965, pp. 71–79.

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

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

[41]Kuntysh, V. B., Toporkova, M. A., and Girishin, V. P., ''Heat Exchange in Tube Banks with L-Shaped Profiled Fins,'' *Energomashinostroenie*, Vol. 7, No. 4, 1983, pp. 1–6.

[42]Stafford, B. C., ''Investigation of Important Surface Characteristics of a Finned Tube Heat Exchanger,'' M.S. Thesis, Dept. of Mechanical and Aerospace Engineering, Univ. of Missouri, Rolla, MO, 1983.

[43]Sheffield, J. W., Stafford, B. D., and Sauer, H. J., Jr., ''Finned-Tube Contact Conductance: Investigation of Contacting Surfaces,'' *ASHRAE Transactions*, Vol. 91, Pt. 1, No. 3242, 1985, pp. 442–452.

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

[45]Nho, K. M., and Yovanovich, M. M., ''Effect of Oxide Layers on Measured and Theoretical Contact Conductances in Finned Tube Heat Exchangers,'' A. L. London Symposium on Compact Heat Exchangers, Stanford Univ., Stanford, CA, March 1989.

[46]Nho, K. M., and Yovanovich, M. M., ''Measurement of Contact Resistance in Finned Tube Heat Exchangers,'' *ASHRAE Transactions*, Vol. 95, Pt. 1, 1989, pp. 370–378.

[47]Egorov, E. D., Nekrasov, M. I., and Pikus, V. Y., ''Investigation of Contact Heat-Transfer Resistance in Two Layer Finned-Tubes,'' *Energomashinostroenie*, Vol. 13, No. 6, 1989, pp. 17–19.

[48]Matal, O., Tomas, Z., Simo, T., and Konecny, A., ''Experimental Investigation of Duplex Tubes for Sodium Heated Steam Generator,'' *Proceedings of the 4th International Topical Meeting on Nuclear Reactor Thermal-Hydraulics*, Vol. 2, American Nuclear Society, LaGrange Park, IL, 1989, pp. 1402–1406.

[49]Huang, M. M., ''Contact Resistance of Round Tube and Plate Heat Exchangers Using Finite Element Analysis,'' M.S. Thesis, Dept. of Mechanical and Manufacturing Engineering, Univ. of Brighton, Brighton, England, UK, 1994.

[50]Gardner, K. A., and Carnavos, T. C., ''Thermal Contact Resistance in Finned Tubing,'' *Journal of Heat Transfer*, Vol. 82, Nov. 1960, pp. 279–293.