Heat Transfer to Water in Turbulent Flow in Internally Heated Annuli

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Nearly 900 values of local heat transfer coefficients were correlated for water flowing through long annuli 1/8, 1/4, and 3/8 in. wide, electrically heated at their inner surfaces and containing three spacer ribs. Both cosine and uniform lengthwise heat-flux distributions were employed. All heat transfer coefficients were computed for positions corresponding to (L/D_e) ratios larger than 150. Several methods of correlation were attempted and compared, especially with respect to the method of evaluating physical properties. The proportionality of the Colburn *j* factor to the Prandtl and Reynolds numbers with their usual exponents was verified, and the dependence of *j* upon D_2/D_1 was analyzed. There was no significant effect of cosine heat-flux distribution on the heat transfer coefficients. Evaluating physical properties at the usual film temperature gave the best correlation. A simplified dimensional equation for water at moderate temperatures and pressures was also developed.

During the course of an extensive experimental program on heat transfer and fluid flow characteristics of simulated nuclear fuel elements, large amounts of data have been collected that are of considerable general interest. Of the data presently available from this program, those related to nonboiling heat transfer to water in turbulent flow have been extracted and are presented, analyzed, and correlated in this paper. Data for water flowing in long internally heated annuli with nonuniform heat-flux generation with length and at high rates of heat transfer are of special interest in nuclear-reactor design. Data of this type, especially at heat-flux densities in the order of 1,000,000 B.t.u./(hr.)(sq. ft.), are not extensively reported in the literature.

The data available from these tests cover a wide range of heat-flux density and temperature difference and include uniform and nonuniform lengthwise heat-flux distribution and three different annulus sizes. The methods of measurement made it possible to compute true local heat transfer coefficients, and the wide range of temperature difference allowed consideration of the best method to account for physical-property temperature dependence across the fluid cross section. The effect of nonuniform as compared with uniform heat flux could also be examined.

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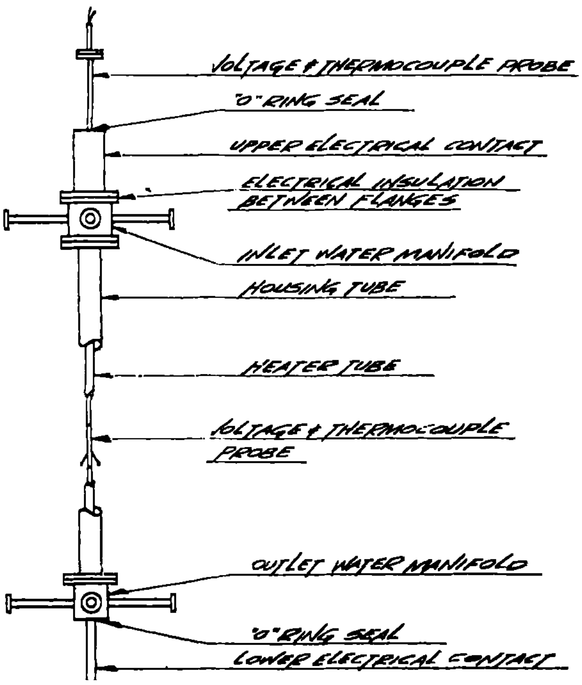

Fig. 1. Schematic of typical test section.

APPARATUS

Deionized water was pumped downward through the test section, then through a cooler and was returned to the pump. All associated piping and fittings were either 304 or 316 stainless steel. The test section, shown schematically in Figure 1, consisted of two coaxially placed aluminum tubes forming a concentric annulus over 10 ft. long.

The inner tube, which was either 2S or 61S aluminum of approximately 1-in. O. D., was heated by the passage of up to 20,000 amp. of d.c. through it and was cooled by water flowing in the annular space. The power for heat generation was supplied by a 5,000-h.p. motor-generator set. For essentially uniform heat generation, the heater tube used had a wall thickness of 0.035 in., and for a lengthwise cosine distribution* of heat generation, the I.D. was appropriately tapered, having its thinnest wall at the center. The tubes with tapered I.D. were extruded in two sections and then welded together at their thin ends. During operation the heater tube was pressurized internally with nitrogen.

The outer tube had a relatively thick wall and was manufactured from 61S aluminum. Three different sizes of outer tubes were used giving annulus widths of ⅛, ¼, and ⅜ in. Concentric placement of the inner tube was obtained by the use of three silicone-glass laminate centering ribs held snugly and accurately in lengthwise broached slots in the wall of the outer tube. The ribs occupied about 5% of the cross-sectional area to flow in the annulus and, as shown in Figure 2, were of triangular cross section making nearly point contact with the heater surface. In most tests the ribs used were continuous along the entire annulus length, creating three distinct subchannels between them. In later tests the ribs were vented to allow for flow communication among subchannels. Other than the small effect on the circumferential heater-surface temperature discussed below, venting of the ribs was not important to the experimental results reported here.

Twelve combination voltage and thermocouple probes were placed inside the heater tube. These were attached to a long tube consisting of short alternate sections of mild steel and melamine or silicon-glass laminate, which extended into the heater tube through an 0-ring seal at the top end. The thermocouple junctions were held against the inner wall of the heater tube by spring tension and measured both the inner wall temperature and the voltage drop along the tube. The thermocouple tube could be rotated to allow the measurement of the circumferential temperature distribution.

*The term *cosine distribution* denotes nonuniform heat generation, with the maximum occurring at the center of the heating tube and then decreasing according to a cosine law on both sides of the center. This kind of distribution is frequently encountered in nuclear reactors.

Water flow rates were measured by sharp-edged orifices during the early part of the program and later by a Potter turbine type of flow-sensing element. Inlet and outlet stream temperatures were measured by resistance thermometers. All voltages, including thermocouple outputs and voltage drop along the heater tube, were measured by a self-balancing potentiometer with a voltage-dividing circuit and selector switches. Electrical current was measured with a large shunt and recording potentiometer.

EXPERIMENTAL DATA

The data recorded during a typical heat transfer run were as follows:

1. Voltage drop over the entire heated section and at twelve equidistant points along the inside of the heater tube.
2. Electrical current.
3. Flow rate.
4. Inlet and outlet temperatures.
5. Heater inner surface temperature at six equiangular positions, starting at a random position, around the circumference for each of the twelve equidistant points along the length at which the voltages were measured.
6. Pressure at various positions along the annulus length.

The accuracy of these measurements was sufficient to ensure a precision of heat transfer coefficients of better than 10%.

CALCULATION OF HEAT TRANSFER COEFFICIENTS

The local heat-flux density at any point along the heater was given by

$$(q/A) = \frac{3.413}{\pi D_1} I \frac{dE}{dL} \tag{1}$$

For runs with essentially uniform heat generation, the voltage gradient along the heater was nearly constant and therefore could be calculated with sufficient accuracy at any position by use of the incremental voltage change over adjacent measurements. For runs with cosine heat generation, the voltage gradient was not constant, and use of the above method would not be so accurate; however, the voltage distribution was nearly linear with sin ax, where x is the distance measured from the center of the heater (i.e., at maximum flux) and a is a constant determined by the design value of heat flux at the ends of the heater. Hence, for cosine distributions the measured voltages were related to sin ax rather than to the heated length directly, and the voltage gradients were computed as follows:

$$\begin{aligned}\frac{dE}{dL} &= a \cos ax \frac{dE}{d(\sin ax)} \\ &\simeq a \cos ax \frac{\Delta E}{\Delta(\sin ax)}\end{aligned} \tag{2}$$

The bulk water temperatures t_b were computed by adding the voltage-drop proportion of the over-all temperature rise to the measured inlet temperature.

The presence of the three centering ribs resulted in a circumferential nonuniformity of heater-wall temperature and introduced the question of what was the proper value of temperature difference for use in calculating a heat transfer coefficient. Since the temperature rises under the ribs were not very great, the problem was not a serious one but certainly deserved consideration. Several detailed circumferential temperature traverses were made, the results of a typical one being shown in Figure 3. This figure shows that wall temperatures were never actually uniform over any portion of the circumference. In addition to the circumferential variation of wall temperature, the presence of the ribs resulted in both a circumferential variation of local water temperature and of fluid velocity. The most reasonable value of water temperature to be used in computing the temperature difference would be the usual mixing cup or bulk temperature, and the most practical value of velocity to be associated with this temperature difference would be the average value calculated from the actual cross-sectional area and flow rate. Since the circumferential variation of wall temperature consisted of gradual changes extending nearly completely around the circumference, it was assumed that the best value of wall temperature would be the circumferential mean value. This value was determined by averaging the six measured inside-heater-wall temperatures and then correcting for wall drop. The wall-drop correction was calculated as a simple heat-conduction problem with longitudinal and circumferential conduction neglected, the latter being due to the nonuniformity of wall temperature. The circumferential variation in surface temperature averaged about 8°F. and was never more than 30% of $\bar{t}_s - t_b$, as shown in Figure 3, which is one of the more severe cases. The wall-temperature drop averaged about 7°F. and was never more than 15% of $\bar{t}_s - t_b$. The heat transfer coefficient was computed from

$$h = \frac{q/A}{\bar{t}_s - t_b} \tag{3}$$

When computed in this manner, heat transfer coefficients obtained from data taken with annuli having many small sections of the ribs cut out agreed with those obtained from annuli having continuous ribs. This observation supports the validity of the method.

Values of heat transfer coefficients were not computed for positions corresponding to an (L/D_e) less than 150 at the inlet end of the annulus and for positions closer than 1½ ft. from the outlet. This was done to avoid using inaccurate data in those regions of the tube where, with a cosine distribution, the heat generation was small. Entrance effects are damped out at considerably smaller (L/D_e) ratios. In addition, data points with heater-surface temperatures 10°F. higher than the local saturation temperature were excluded to eliminate the possibility of including local boiling values.

Ranges of calculated heat transfer coefficients, heat-flux density, bulk temperature, average heater-surface temperatures, mass flow rate, and velocity are given in Table 1.

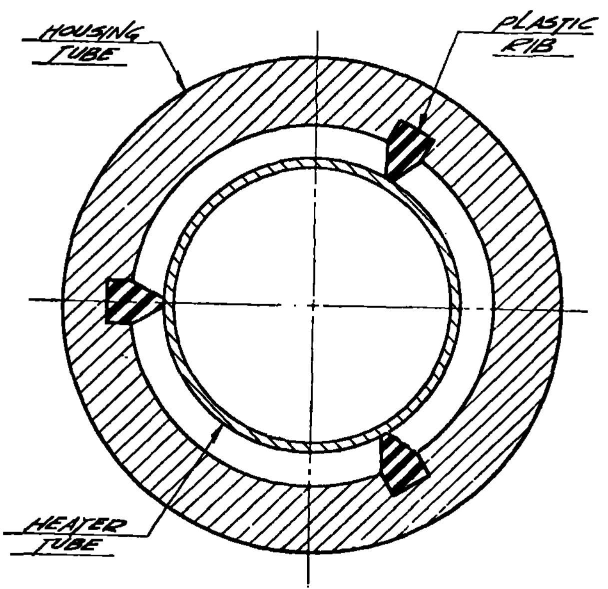

Fig. 2. Cross section of annular flow channel.

CORRELATION OF EXPERIMENTAL DATA

The Stanton, Prandtl, and Reynolds numbers were computed for each experimental point. Physical properties were initially evaluated at the arithmetic mean of the average surface and bulk temperatures, i.e., the usual film temperature t_f. For computing Reynolds numbers, the hydraulic equivalent diameter D_e was used.

$$D_e = \frac{4S}{L_p} \tag{4}$$

The influence of the centering ribs on both the cross-sectional area and the wetted perimeter was included in the calculation of D_e.

The usual dependence on the Prandtl number was verified by selecting data points at constant values of Reynolds numbers and observing that the Colburn j factor $(St)(Pr)_f^{2/3}$ was independent of Pr at constant Re. The logarithm of the j factor was plotted vs. the logarithm of the Reynolds number for representative samples of the data. This preliminary examination verified the well-established proportionality of j to $Re^{-0.2}$ and also indicated that the effects of the non-uniform heat-flux distribution on the heat transfer coefficient, as compared with the results with essentially uniform heat generation, were not significant. These results are illustrated in Figure 4, where a representative sample of the data for one annulus size is plotted. The ranges of the correlated data for each annulus size are summarized in Table 2.

Although deionized water of relatively high purity was used, operation with high values of heater-wall temperature over extended periods of time made possible the formation of a film of corrosion products. At the high rates of heat transfer investigated a very small amount of such film formation would have a significant effect on calculated heat transfer coefficients; therefore, it was desirable to check this possibility carefully, by plotting the quantity $jRe^{0.2}$ vs. run number for each series of runs made with the same heater tube. Except for a few runs made after the tube had been operating with considerable local and bulk boiling, no significant effect of corrosion-product film formation could be detected.

The preliminary examination of the data indicated that equations of the type

$$(St)(Pr)^{2/3}(Re)^{0.2} = \text{constant} \tag{5}$$

would be suitable for correlation. Least square values of the constant were determined from all the data evaluating physical properties in several ways.

With physical properties evaluated at the film temperature, the constant was found to have a slight dependence on annulus width, and the average of the standard deviations* for each annulus size was 8% with a maximum of 10% for the ⅛-in. annulus. These deviations are probably very close to the precision of the measurements. With physical properties evaluated at the bulk temperature, the constant showed a similar dependence on annulus width, with an average standard deviation of 11% and a maximum of 12% for the ¼-in. annulus. Hence a slightly better correlation is obtained when physical properties are evaluated at the film temperature.

Another method for taking into account variations of physical properties was suggested by Sieder and Tate (*4*). The ratio of the viscosity at the surface temperature to that at the bulk temperature is included as an additional dimensionless term in Equation (5), all other physical properties being evaluated at the bulk temperature.

The data were also correlated by use of the Sieder and Tate recommendation, by determining the least square values of the constant for each annulus size in the equation

$$(St)(Pr)_b^{2/3}(Re)_b^{0.2}\left(\frac{\mu_s}{\mu_b}\right)^{0.14} = \text{constant} \tag{6}$$

As with the bulk and film temperature correlations, the constant was found to have a slight dependence on annulus width. The average of the standard deviations for each annulus size was 9% with a maximum of 11% for the ⅛-in. annulus. A correlation was also attempted on the assumption that the exponent on the viscosity ratio was unknown and on the calculation of both the exponent and the constant by least squares. This correlation resulted in inconsistent values of both the constant and the exponent and was therefore discarded. Thus it appears that the use of the viscosity ratio does not give so good a correlation as the use of the film temperature but gives a slightly better correlation than that where the physical properties are evaluated at bulk temperature.

The results of the three methods of correlation are summarized in Table 3.

The small but consistent increase of $(St)(Pr)^{2/3}(Re)^{0.2}$ with annulus width is in agreement with the findings of other investigators. In general, the heat transfer coefficient is found to increase as the ratio of the outer to the inner diameter of the annulus increases. The range of this ratio for the experimental data correlated here is not large enough to justify the establishing of a definite mathematical dependence on D_2/D_1. (For the ⅛-in. annulus, $D_2/D_1 = 1.232$; for the ⅜-in. annulus, $D_2/D_1 = 1.694$). Therefore, the form of equation chosen to correlate the data with annulus size was obtained by examining the results of other investigators whose range was much larger. In particular, the results of Monrad and Pelton (*3*) and of Wiegand (*5*) indicate that the heat transfer coefficient could be assumed proportional to the square root of the ratio of the diameters for a range of the ratio from 1 to 10. With this assumption the following correlating equations were obtained:

$$(St)(Pr)_b^{2/3}(Re)_b^{0.2}\left(\frac{D_1}{D_2}\right)^{1/2} = 0.0244 \pm 15\% \tag{7}$$

$$(St)(Pr)_b^{2/3}(Re)_b^{0.2}\left(\frac{\mu_s}{\mu_b}\right)^{0.14}\left(\frac{D_1}{D_2}\right)^{1/2} = 0.0222 \pm 13\% \tag{8}$$

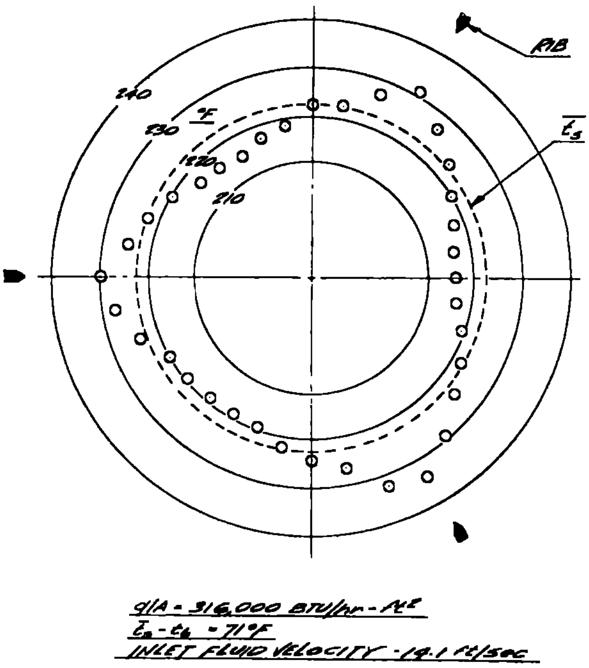

Fig. 3. Typical circumferential wall-temperature traverse.

*By standard deviation is meant the statistical quantity defined as the square root of the variance. The variance is defined as the sum of the squares of the deviations from the mean divided by one less than the number of observations. For a fairly large number of observations (say $\geq$ 50), the standard deviation defines the region containing approximately two-thirds of all observations.

TABLE 1

RANGES OF EXPERIMENTAL DATA

	Annulus size, in.		
	1/8	1/4	3/8
Bulk temperature t_b, °F.	74–261	68–182	77–191
Average heater-surface temperature $\bar{t}_s$, °F.	106–315	101–290	119–279
Heat-flux density $q/A \times 10^{-3}$, B.t.u./(hr.)(sq. ft.)	40–1,125	61–1,055	115–1,090
Mass flow rate, lb./sec.	0.7–9.0	2.8–10.0	4.6–16.5
Inlet velocity V, ft./sec.	4–45.0	6–24	7–24
Heat transfer coefficient h, B.t.u./(hr.)(sq. ft.)(°F.)	1,200–12,700	1,500–6,700	2,000–7,300
$\Delta t = \bar{t}_s - t_b$, °F.	13–169	16–198	32–153
Number of experimental values	486	238	146
Number of different test assemblies	7	7	2

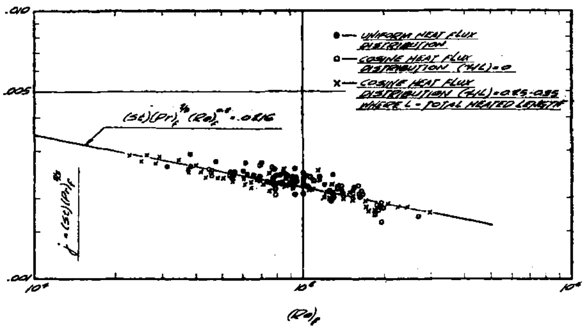

Fig. 4. j vs. Re_f for 1/8-in. annulus.

$$(St)(Pr)_f^{2/3}(Re)_f^{0.2}\left(\frac{D_1}{D_2}\right)^{1/2} = 0.0200 \pm 10\% \qquad (9)$$

Equation (7) evaluates physical properties at the bulk temperature, Equation (9) at the film temperature, and Equation (8) employs the Sieder and Tate viscosity correction. The indicated percentage of deviations were conservatively estimated by adding the average of the percentage of standard deviations for each annulus size to the average of the deviations introduced by use of the diameter ratio.

DISCUSSION

When physical properties are evaluated at the bulk temperature alone, no allowance is being made for the variation of these properties across the fluid cross section. These variations influence both momentum and heat exchange within the fluid. The Sieder and Tate viscosity correction attempts to take into account variations of viscosity but ignores other properties. The evaluation of physical properties at the film temperature approximately allows for changes across the fluid cross section because an average temperature is assumed to apply. Since the use of a film temperature allows for variations of all physical properties, it might be expected to result in the best correlation. In support of this expectation, the constants of the correlating equations and their standard deviations decrease in value as more of the physical-property variations across the fluid cross section are taken into account. An examination of Table 3 shows that the differences among the constants and the differences among the standard deviations are significant. Larger differences among the constants cannot be expected since with water at moderate pressures the temperature variation of the combination of physical properties involved is small. The differences among standard deviations are considered significant, as the same experimental data were used for the three methods of correlation.

It may be argued that the inaccuracies introduced by evaluating physical properties at the bulk temperature alone are not large enough to justify the more difficult methods of calculation. This is especially true with the usual industrial applications, where, for example, significant resistance to heat transfer due to fouling must be allowed for by little better than guesswork. This is clearly not the case with a device such as a nuclear reactor, where heat-flux densities are at least an order of magnitude larger than was previously usual and surface temperatures must be known as accurately as possible. In addition, a correlation of experimental data as obtained here based on bulk temperature alone can be expected to introduce further inaccuracies when used with both heating and cooling and with fluids considerably different from water at moderate temperatures and pressures. Thus, a correlation which includes the effect of variation of physical properties across the laminar layer and into the turbulent core is desirable not only for improved accuracy, but also for more general applicability.

Although the Sieder and Tate viscosity correction appears to be widely used, to our knowledge it has never been shown to be preferable to evaluation of physical properties at the film temperature. The two methods are certainly not equivalent, although in some ranges and with some fluids they may be nearly so. Valid applications of a Sieder and Tate type of correlating equation are limited to regions where, of the physical properties involved, only the viscosity changes significantly with temperature. A correlation based on the film temperature offers more generality and for this reason is preferred by the authors.

A bulk temperature correlation is sometimes preferred by the engineer because it does not require the iterative calculations usually necessary with both the film-temperature and viscosity-ratio correlations. A viscosity-ratio correlation is sometimes preferred to a film-temperature correlation because the repetitive calculations with the former involve only a simple ratio. The authors suggest that this is false economy. Proper use of a film-temperature correlation can offer the same convenience as the others while still retaining its advantage of wider generality. For example, for those cases where Δt is small or where knowledge of the heat transfer coefficient to better than 15% is not necessary, an average Δt can be assumed over the entire range of application and used to evaluate an approximate film temperature; that is,

$$t_f = t_b \pm \tfrac{1}{2}\Delta t_{avg} \qquad (10)$$

where the choice of sign depends upon whether the fluid is being heated or cooled. In this way a film-temperature correlation can be easily converted into what is essentially the equivalent of a bulk-temperature correlation applicable in the particular range of interest.

The dependence of the heat transfer coefficient on annulus width and radius has been accounted for by evaluating Reynolds number by use of the usual hydraulic definition of equivalent diameter and assuming the heat transfer coefficient proportional to the square root

TABLE 2

RANGES OF CORRELATED DATA

	Annulus size, in.		
	1/8	1/4	3/8
St $(h/c_bG)\times 10^3$	0.7–1.7	0.7–1.7	0.8–1.7
$(Pr_b$ $(c_p\mu/k)_b$	1.4–6.6	2.2–7.2	2.1–6.1
$(Pr)_f$ $(c_p\mu/k)_f$	1.4–5.2	1.7–5.7	1.6–5.0
$(Re)_b$ $(D_eG/\mu)_b$ $\times 10^{-3}$	27–269	38–164	36–233
$(Re)_f$ $(D_eG/\mu)_f$ $\times 10^{-3}$	30–303	40–207	48–390
(μ_b/μ_s)	1.1–2.7	1.2–3.8	1.5–3.2
t_f, °F.	90–271	82–225	93–235

TABLE 3

RESULTS OF VARIOUS METHODS OF CORRELATION

Annulus size, in.	Physical Properties at Bulk Temperature: $(St)(Pr)_b^{2/3}(Re)_b^{0.2}$	Standard deviation	Physical Properties at Film Temperature: $(St)(Pr)_f^{2/3}(Re)_f^{0.2}$	Standard deviation	Sieder and Tate Viscosity Ratio: $(St)(Pr)_b^{2/3}(Re)_b^{0.2}(\mu_s/\mu_b)^{0.14}$	Standard deviation
1/8	0.02530	±0.00260	0.02159	±0.00220	0.02350	±0.00259
1/4	0.03013	±0.00374	0.02380	±0.00153	0.02704	±0.00254
3/8	0.03344	±0.00300	0.02657	±0.00170	0.03009	±0.00222

of the diameter ratio (D_2/D_1). This method of correlation is justified by the work of Monrad and Pelton (*3*) and of Wiegand (*6*). Although the latter proposed that this method be used for a range of the diameter ratio from 1 to 10, the present authors are of the opinion that there has not been sufficient experimental evidence to justify the method for values of the ratio larger than 3. The reason for this—which is also a reason why it is not a serious limitation—is that there is very little practical interest in annuli with larger diameter ratios; therefore, Equation (9), which evaluates physical properties at the film temperature and accounts for annulus radius by use of the square root of the diameter ratio, is the recommended correlation for heat transfer from the inner surface of annuli to turbulent flowing fluids.

In an attempt to correlate data over a larger range of the diameter ratios, Davis (*1*) recommends computing the Reynolds number by use of the inside diameter of the annulus in place of the equivalent diameter and inclusion of the diameter-ratio term with an exponent of 0.15. This method of correlating with annulus dimensions was not used by the authors because there does not appear to be sufficient evidence for its validity.

For the limiting case of $D_2/D_1 = 1$, Equation (9) should also apply to heat transfer from infinitely wide parallel plates. Correlations applicable to such systems have not been found in the literature, and therefore no comparisons could be made. For the case of tubes, the Colburn equation as recommended by McAdams (*2*) has a constant equal to 0.023, or about 15% higher than the constant of Equation (9). This difference is probably a result of only one of the flow boundaries of the annular channel being heated, whereas the constant 0.023 was determined mainly from experiments in tubes where the entire flow boundary is heated.

Computations by Stein (*5*) comparing the heat transfer coefficient for heat flow only from one side of the channel to that for equal heat flow from both sides predicts the 15% difference obtained experimentally in the same range of Prandtl and Reynolds numbers.

SIMPLIFIED CORRELATION FOR WATER

If the combined physical properties in Equation (9) are considered to be a linear function of temperature, then a simplified correlating equation can be obtained for a particular fluid. A correlation of this form was obtained for water as follows:

$$h = 97.7(1 + 0.0066t_f)\frac{V^{0.8}}{D_e^{0.2}}\left(\frac{D_2}{D_1}\right)^{1/2} \tag{11}$$

The velocity V is based on a density of 62.3 lb./cu. ft. For the temperature range of 90° to 270°F. this equation differs from Equation (9) by a maximum of only 3%. It is applicable, of course, at only moderate pressures.

SUMMARY

Heat transfer coefficients for the turbulent flow of water in internally heated annuli with spacer ribs at high rates of heat transfer with both uniform and nonuniform heat-flux distribution were successfully correlated by the usual dimensionless equations. The following equation gave the best correlation:

$$(St)(Pr)_f^{2/3}(Re)_f^{0.2}\left(\frac{D_1}{D_2}\right)^{0.5} = 0.0200$$

Evaluation of physical properties at the bulk temperature alone or use of the Sieder and Tate viscosity correction did not result in as good a correlation, although the differences were small. It is suggested that the equation above is also valid for heat transfer in annuli without spacer ribs and from infinitely wide plates ($D_1/D_2 = 1$) with only one side being heated.

ACKNOWLEDGMENT

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NOTATION

a = constant, defined in text
C_p = specific heat at constant pressure, B.t.u./[lb. (mass)](°F.)
D_e = hydraulic equivalent diameter, ft. $D_e = 4S/L_p$
D_1 = inner diameter of annulus, ft.
D_2 = outer diameter of annulus, ft.
E = voltage along heated length of tube, volts
G = mass velocity, lb.(mass)/(hr.)(sq. ft.)
h = local heat transfer coefficient, B.t.u./(hr.)(sq. ft.)(°F.)
I = electrical current, amp.
j = $(St)(Pr)^{2/3}$, dimensionless
k = thermal conductivity, B.t.u./(hr.)(ft.)(°F.)
L = heated length measured from inlet, ft.
L_p = wetted perimeter, ft.
Pr = Prandtl number ($C_p\mu/k$), dimensionless
q/A = heat flux density, B.t.u./(hr.)(sq. ft.)
Re = Reynolds number (D_eG/μ), dimensionless
S = cross-sectional area to flow, sq. ft.
St = Stanton number (h/C_pG), dimensionless
t_b = mixing-cup mean or bulk temperature, °F.
t_f = film temperature, $\frac{1}{2}(t_b + \bar{t}_s)$, °F.
t_s = heat transfer surface temperature, °F.
$\bar{t}_s$ = average heat transfer surface temperature, °F.
V = Inlet velocity, ft./sec.
Δt = temperature difference, $\bar{t}_s - t_b$, °F.
μ = viscosity, lb. (mass)/(ft.)(hr.)
x = heated length measured from position of maximum heat flux, ft.

Subscripts

b = fluid properties evaluated at bulk temperature, t_b
f = fluid properties evaluated at film temperature, t_f
s = fluid properties evaluated at surface temperature, $\bar{t}_s$

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