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**Determination of Coefficients
for the Universal Laws
of Friction and Heat Transfer
for CFTL Calculations**

S. A. Hodge

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FOR THE UNITED STATES
DEPARTMENT OF ENERGY

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UNIVERSAL LAWS OF FRICTION AND HEAT TRANSFER
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CFTL CALCULATIONS**

S. A. Hodge

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DETERMINATION OF COEFFICIENTS FOR THE UNIVERSAL LAWS OF FRICTION AND HEAT TRANSFER FOR CFTL CALCULATIONS

S. A. Hodge

ABSTRACT

The friction factor and Stanton number for flow past a roughened surface are determined by the parameters A and $R(h^+)$ of the universal law of friction and A_H and $G(h^+)$ of the universal law of heat transfer. The methods to be used for determination of these parameters for the particular roughness to be used in the Core Flow Test Loop (CFTL) are presented. Examples are given concerning the application of these methods to both transitional and fully rough flow using experimental results taken from the literature.

INTRODUCTION

The Core Flow Test Loop (CFTL) currently under construction at Oak Ridge National Laboratory is a high temperature, high pressure system for the circulation of helium coolant in axial flow through a bundle of electrically heated fuel rod simulators (FRS) under steady state and transient conditions. The fuel rod simulators are typical of GCFR application, with the cladding surface roughened over the heated section.

The plan for the CFTL includes the testing of prototypic fuel bundle assemblies of 37, 61, and 91 rods. This testing sequence is intended to provide a basis for confirmation of the computational methodology and extension of these procedures to a full-sized 271-rod bundle.

A cross-section within the roughened portion of the first of the test bundles is shown in Fig. 1. This 37-rod bundle, designated AG-1, consists of 34 heated FRS plus three larger-diameter, unheated spacer grid support rods. This bundle construction comprises four types of flow channel. These are:

1. central channels, bounded by three heated FRS;
2. support rod channels, bounded by two heated FRS and an unheated spacer grid support rod;
3. wall channels, bounded by two FRS and the smooth duct wall; and
4. corner channels, bounded by one FRS and an interior angle of the duct wall.

An example of each of these four channel types is delineated in Fig. 1.

A profile of the roughness machined on the CFTL FRS cladding surface is shown in Fig. 2. This roughness has a trapezoidal shape, a pitch-to-height ratio of 12, and a width-to-height ratio of 3.5. The volumetric radius of a roughened rod is the radius which would prevail if the volume of the roughness elements were smeared evenly over the surface of each rod; for the CFTL, the diameter bounded by this surface is 7.84 mm for the fuel rod simulators and 9.44 mm for the spacer grid support rods.

CE CENTRAL CHANNEL
S SUPPORT ROD CHANNEL
W WALL CHANNEL
CO CORNER CHANNEL

DIRECTION OF FLOW IS
INTO THE DRAWING

SCALE 2:1

Fig. 1. Plan view of the first CFTL bundle.

Fig. 2. Profile of CFTL fuel rod simulator cladding roughness.

A most important parameter to the determination of the friction factor and Stanton number for flow past a particular roughened surface is the ratio of the velocity profile width to the rib height. The velocity profile width is defined as the separation distance between the volumetric surface of a roughened rod and the surface of zero net momentum transfer, or shear, in the adjacent flow. This distance varies among the four types of CFTL channel due to differing channel geometry. In the wall and corner channels, it is also a function of the flow since the surface of zero shear moves outward from the roughened rods toward the smooth duct wall as the Reynolds number increases.¹

Thus, there is a wide range of velocity profile widths which can exist in the various channels of the CFTL bundle under the planned steady state and transient test program. It is therefore essential to an accurate analysis of the CFTL flows that correlations expressing the dependence of friction factor and Stanton number upon velocity profile width be established experimentally for the particular CFTL roughness. The methods to be used in the establishment of these correlations are described in this work.

TURBULENT FLOW REGIMES

It was shown by Nikuradse² that turbulent flow past a roughened surface can be characterized by one of three flow regimes, depending on the nature of variation of the friction factor with the Reynolds number of the flow. At relatively low flow in the turbulent range, the friction factor varies with the Reynolds number in the same manner as in turbulent flow past a smooth surface, and the flow is said to be "hydraulically smooth." As the Reynolds number is increased, the thin region near the wall in which the flow field is dominated by the fluid viscosity narrows so that the roughness elements protrude into the main flow stream. This generates an increased frictional pressure loss by the mechanism of form drag. The degree of protuberance and, concomitantly, the friction factor increase with Reynolds number in this "transition region." At sufficiently high Reynolds numbers, the roughness elements are fully exposed to the main flow stream and the frictional pressure loss is virtually all due to form drag. Consequently, the friction factor becomes independent of the Reynolds number, and the flow characterized by a constant friction factor is termed "fully rough."

Each of these three turbulent flow regimes will exist within the roughened portion of the CFTL bundle at some time during the planned test schedule. For tests at low flow, it is anticipated that fully rough flow will exist in the central and support rod channels concomitant with transition region and/or hydraulically smooth flow in the wall and corner channels. While existing relations applicable to flow past smooth walls can be used for the analysis of hydraulically smooth flow, correlations for the variation of friction factor and Stanton number with velocity profile width remain to be developed for the particular CFTL roughness in both the transition and fully rough regimes of flow. It is also important to establish the Reynolds number range in which each of these correlations applies, that is, to establish the extent of the transition region.

FRICTION FACTOR CORRELATION

A friction factor for flow past roughened surfaces can be defined in a manner similar to that for smooth surfaces. The friction factor for rough surface flow is defined as the proportionality factor f_1 in the equation for the axially averaged shear stress τ_w at the rough wall,

$$\tau_w = f_1 \frac{\rho_1 \bar{u}_1^2}{2} \quad (1)$$

The terms ρ_1 and $\bar{u}_1$ in this equation represent the density and the spacially averaged velocity for the flow between the rough wall and the surface of zero shear in the adjacent flow.

The friction factor f_1 can be related to the velocity profile width Y_L in the adjacent flow by the universal law of friction for flow past roughened surfaces.¹ For flow in an annular geometry, such as that formed by axial flow over the surface of one of the rough rods in a bundle, this law can be written as the linear equation

$$\sqrt{2/f_1} = A \left[\ln(Y_L/h) - \frac{1}{2} - \frac{1}{2 + Y_L/r_1} \right] + R(h^+) \quad (2)$$

The term h in Eq. (2) represents the height of the roughness above the root, while r_1 is the volumetric radius of the rough rod. The parameters A and $R(h^+)$ are properties of the particular roughened surface and must be determined by experiment.

It has been previously demonstrated^{1,3} that the parameters A and $R(h^+)$ are independent of Reynolds number in fully rough flow. Mathematically, the parameter A is the gradient of the linear portion of the dimensionless velocity profile while $R(h^+)$ is the value of the dimensionless velocity at a point h units above the volumetric surface. However, recent experiments by Meyer and Vogel⁴ in the parallel plate geometry have shown that the gradient of the linear portion of the dimensionless velocity profile varies with the velocity profile width and can differ significantly from the slope A of the linear relation of Eq. (2). This discrepancy is probably due to the physical deviation of the dimensionless velocity profile from linearity both near the rough wall and near the surface of zero shear. The effect of these deviations on the average value of the dimensionless velocity, which is directly related to the friction factor, that is,

$$\bar{u}_1^+ = \sqrt{2/f_1} \quad (3)$$

will vary with the velocity profile width since a smaller fraction of the dimensionless velocity profile will be linear in a small channel with a short velocity profile.

Nevertheless, the linearity of Eq. (2) is not a function of velocity profile width for flow in an annular geometry.* Accordingly, for fully rough flow, the constants A and $R(h^+)$ can be determined for the CFTL roughness by the testing of a single prototype rod in a succession of annular geometries, each formed by placing the rough rod at the center of one of a series of smooth tubes of varying diameter. The friction factor (f_1) and the corresponding velocity profile width (Y_L) are determined for the flow adjacent to the rough rod in each of these annular geometries using experimentally measured values of the overall pressure drop and flow and application of the Dalle Donne-Meyer transformation.^{5,6} The slope A and intercept $R(h^+)$ of Eq. (2) can then be determined by plotting the friction factor term $\sqrt{2/f_1}$ as a function of the velocity profile width term

$$\left[\ln(Y_L/h) - \frac{1}{2} - \frac{1}{2 + Y_L/r_1} \right] \quad (4)$$

*Nor in rectangular channel flow as confirmed by Meyer and Vogel⁴ for the version of Eq. (2) applicable to parallel plate geometry.

It should be noted that the experimental requirements are relatively simple, in that the velocity profile need not be measured. The computational procedure for fully rough flow has been described in previous works;^{3,4} an example plot resulting from applications of this technique to the experimental results for rod No. 1 of the ten-rod test series of Dalle Donne and Meyer^{5,6} is shown in Fig. 3.

TRANSITION REGION FLOW

The parameter $R(h^+)$ is a constant in fully rough flow because the friction factor is constant; this is not the case in the transition region. The friction factor can be expressed as a function of the Reynolds number, but it is more common to characterize flow in the transition region as a function of the roughness Reynolds number h^+ , where

$$h^+ = \frac{hu^*}{\nu} = \frac{h}{D} \text{Re} \sqrt{f_1/2} . \quad (5)$$

The roughness Reynolds number is preferred because it is not a function of the hydraulic diameter. Thus, for single-rod tests in a succession of annular geometries with differing hydraulic diameters, the flow will become fully rough at different Reynolds numbers, but at the same value of h^+ , since the roughness height would not be changed in these tests.

Because the friction factor varies with the roughness Reynolds number in the transition region, it would be expected that both the slope A and the intercept $R(h^+)$ of Eq. (2) are functions of h^+ in this regime. This variation of the intercept has been long known; hence the appellation $R(h^+)$. However, experimental results from which the variation of the slope A in the transition region can be determined are limited. The only known set is the results of Dalle Donne and Meyer^{5,6} for test rod No. 1 of their ten rod test series. The portion of these results applicable to fully rough flow was used in the preparation of Fig. 3.

To prepare a graph of Eq. (2) in the transition region, it is necessary that each ordinate value $\sqrt{2/f_1}$ be plotted as a function of the abscissa value [Eq. (4)] which corresponds to the same roughness Reynolds number (h^+). To this end, it is useful to first separately plot the ordinate and the abscissa values as functions of h^+ .

A semilogarithmic plot of the ordinate values $\sqrt{2/f_1}$ as a function of h^+ for rod No. 1 of Dalle Donne and Meyer is shown in Fig. 4. Each of the four curves represents experimental results for a different annular geometry; the parameter Y_T which differentiates the curves is the annular gap, that is, the distance (mm) between the volumetric surface of the rough rod and the inner surface of the smooth outer tube in which the rod is centered. It can be seen that the friction factor (f_1) increases with h^+ in the transition zone, then becomes constant (within the limits of experimental error) as the flow becomes fully rough at an h^+ value of about 50.

The constants for the correlations shown in Fig. 4,

$$\sqrt{2/f_1} = C_1 + C_2 \ln(h^+) + C_3 [\ln(h^+)]^2 , \quad (6)$$

were obtained by a least-squares fit to the experimental results for values of h^+ less than 100 in each of the four annular geometries. These constants are listed in Table 1.

The abscissa values [Eq. (4)] are plotted as a function of h^+ on the semilogarithmic plot of Fig. 5. The modest but continuous increase in these values with h^+ is caused by the slight movement of the surface of

Fig. 3. The universal law of friction for rod No. 1 tested by Dalle Donne and Meyer (refs. 5 and 6).

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Fig. 4. Variation of ordinate values with roughness Reynolds number in the transition zone.

Table 1. Constants for the correlation of $\sqrt{2/f_1}$ with $\ln(h^+)$ per Eq. (6)

Annular gap Y_T (mm)	C_1	C_2	C_3
3.47	11.0827	-1.9737	0.2325
8.13	17.1878	-4.1908	0.5000
18.24	20.2508	-4.7451	0.5556
25.69	17.1520	-3.0943	0.3782

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Fig. 5. Variation of abscissa values with roughness Reynolds number in the transition zone.

zero shear outward from the rough rod toward the inner surface of the smooth tube as the velocity of the flow increases. It should be noted that this increase in the velocity profile width (Y_L) with the roughness Reynolds number occurs in fully rough flow as well, albeit at a reduced rate.

The constants for the correlation of Eq. (4), that is,

$$\ln(Y_L/h) - \frac{1}{2} - \frac{1}{2 + Y_L/r_1} = C_4 + C_5 \ln(h^+) + C_6 [\ln(h^+)]^2 \quad (7)$$

obtained by a least-squares fit to the curves of Fig. 5 are listed in Table 2.

Table 2. Constants for the correlation of Eq. (4) with $\ln(h^+)$ per Eq. (7)

Annular gap Y_T (mm)	C_4	C_5	C_6
3.47	0.7212	0.1927	-0.0165
8.13	1.3433	0.2968	-0.0281
18.24	2.1228	0.2925	-0.0263
25.69	2.5627	0.2480	-0.0217

Derived ordinate values can be plotted against the corresponding values on the abscissa at selected values of the roughness Reynolds number h^+ , using these least-squares polynomial fits for both the ordinate and the abscissa terms of Eq. (2). The result is shown in Fig. 6 for values of h^+ at even intervals of ten from 10 to 100.

It can be seen in Fig. 6 that the plotted points for values of h^+ of 50 and above fall on top of one another for each of the four annular geometries used in the testing. This confirms that fully rough flow exists for the particular roughness of this rod at values of h^+ greater than 50.

In the transition region, that is, for values of h^+ less than 50, the points plotted in Fig. 6 representing the three smaller annular gaps used in the testing preserve the linearity of Eq. (2). These points indicate that the slope A of Eq. (2) increases markedly as the value of h^+ decreases in the transition region, while the value of the intercept $R(h^+)$ increases to a much smaller extent, with the relative position of the points preserving linearity. However, the points representing the largest annular geometry do not conform to this pattern. The basis for this nonconformity can be seen on Fig. 4, where the two plotted points representing flow in this largest annulus at values of h^+ less than 30 do not reflect the degree of increase in values of $\sqrt{2/f_1}$ at low flows suggested by the experimental results for the smaller annuli.

The variation of the slope A and wall intercept $R(h^+)$ with h^+ represented by the linear relations shown on Fig. 6 are given in Table 3. These listed values are derived from the experimental results for flow in the three smaller annuli only.

Since the slope of the dimensionless velocity profile for flow past a smooth wall is known to be 2.50,¹ it would seem that the slope A should approach this value as the roughness Reynolds number decreases through the transition region, approaching the regime of hydraulically smooth flow. Thus it is noteworthy that a value as high as 2.85 is indicated at h^+ of 10 in Table 3. It may be invalid to extrapolate the results plotted in Fig. 4 to an h^+ value as low as 10.

The continued linearity of Eq. (2) in the transitional region as indicated by Fig. 6 suggests that the two nonconformal points plotted for low flow in the largest annular geometry represented in Fig. 4 might be the result of experimental error. This question awaits resolution. It is intended that forthcoming single-rod tests in annular geometry with the CFTL prototype rod will concentrate on the transition region. Resolution of this question is important because much of the CFTL bundle testing will involve transition region flow.

Fig. 6. The universal law of friction for rod No. 1 in the transition region (data from refs. 5 and 6).

Table 3. Variation of the slope A and intercept $R(h^+)$ of Eq. (2) in the transition region

h^+	A	$R(h^+)$
10	2.85	4.74
20	2.33	4.55
30	2.14	4.48
40	2.04	4.45
≥ 50	2.00	4.37

HEAT TRANSFER CORRELATION

The universal law of heat transfer for flow in an annular geometry can be written in the form¹

$$\frac{\sqrt{f_1/2}}{St_1} = A_H \left[\ln(Y_L/h) - \frac{1}{2} - \frac{1}{2 + Y_L/r_1} \right] + G(h^+) . \quad (8)$$

This law expresses the ratio of the term $\sqrt{f_1/2}$ to the Stanton number St_1 applicable to the region of flow between the volumetric surface of the rough rod and the surface of zero shear. This Stanton number is defined by the equation

$$St_1 = \frac{q_w}{(T_w - T_{B1}) \rho_1 C_p \bar{u}_1} , \quad (9)$$

where T_{B1} is the bulk temperature of the coolant in this region. It should be noted that the bracketed term on the right side of Eq. (8) is the same as that appearing in the universal law of friction for an annular geometry, Eq. (2).

The parameters A_H and $G(h^+)$ of Eq. (8) are properties of the rough surface and must be determined by experiment.* Since the friction factor f_1 is constant in fully rough flow but the Stanton number St_1 monotonically decreases as h^+ increases in this region,¹ Eq. (8) shows that $G(h^+)$ must monotonically increase in fully rough flow. Conversely, in the transition region both f_1 and St_1 increase with h^+ so that $G(h^+)$ will be relatively constant there. These considerations show that the variation of $G(h^+)$ with the roughness Reynolds number is markedly different than the behavior of the parameter $R(h^+)$ of the friction law, Eq. (2).

As in the case for the friction law, experimental results from which the variation of the slope A_H of Eq. (8) can be determined are limited. Dalle Donne and Meyer^{5,6} performed heat transfer tests for two of the rods in their ten rod test series, rods No. 8 and 10, but only two annular geometries were used for each rod in these heat transfer experiments. It is obvious that the linearity of Eq. (8) cannot be proven with only two points available. Nevertheless, with the assumption that Eq. (8) is linear, these results can be used to demonstrate the proposed method of analysis.

The method involves graphical determination of the slope A_H and intercept $G(h^+)$ of Eq. (8) by construction of straight lines connecting the points representing the flow in the two annular geometries used for each rod in the heat transfer experiment. Since the wall intercept $G(h^+)$ varies with the roughness Reynolds number, care must be taken to ensure that each pair of connected points represents results obtained at the same value of h^+ .

This method for determination of A_H and $G(h^+)$ is similar to that employed for determination of the analogous quantities for the friction law in the transition region. However, there is an important additional restriction on the utilization of the graphical procedure; this is due to the dependence of the coolant physical properties upon temperature. Since experimentally determined friction factors and Stanton numbers are affected by a variation of coolant temperature across the annulus, comparable experimental results in the different annular geometries must have been obtained at approximately the same coolant bulk temperature and wall-to-bulk temperature ratios (T_w/T_B). The maximum wall temperatures (T_{wm}) and the range of wall-to-bulk temperature ratios for the portion of the experimental results of Dalle Donne and Meyer for rods 8 and 10 which do meet this criterion are listed in Table 4.

Table 4. Compatible maximum wall temperatures and range of wall-to-bulk temperature ratios from the experimental results of Dalle Donne and Meyer

Rod No.	Annular gap Y_T (mm)	T_{wm} (°C)	T_w/T_B
8	8.63	160	1.16-1.22
	18.74	160	1.30-1.32
10	8.69	160	1.21-1.23
	18.80	160	1.30-1.33

The left side of Eq. (8) can be written in the form

$$\left(\frac{T_w - T_{B1}}{T_w - T_B} \right) \frac{(T_w - T_B) \rho C_p \bar{u}}{q_w} \frac{u_1^*}{\bar{u}} \frac{u_1^*}{u^*} \quad (10)$$

which is equivalently written as

$$\left(\frac{T_w - T_{B1}}{T_w - T_B} \right) \frac{\sqrt{f/2}}{St} \frac{u_1^*}{u^*} \quad (11)$$

where f and St are the friction factor and Stanton number for the overall annulus. These quantities are directly determined from experimental data, that is, no transformation is involved. The form of Eq. (11) is convenient because the ratio of friction velocities is easily evaluated using the relation¹

$$\frac{u_1^*}{u^*} = \left(\frac{\beta^2 - \alpha^2}{\alpha(1 - \alpha)} \right)^{1/2} \cdot \left(\frac{\mu}{\rho_1} \right)^{1/2} \quad (12)$$

The parameter α in Eq. (12) is the radius ratio of the annular geometry, while β represents the ratio of the radius of the surface of zero shear to the radius of the smooth outer tube in which the rough rod is

*It should be noted that the parameter $G(h^+)$ is a function of the Prandtl number and therefore differs for the same roughness in different fluids.¹

centered, and is calculated by the Dalle Donne-Meyer transformation.⁵ For most practical cases, the density ratio of Eq. (12) will be sufficiently close to unity to be neglected.

The first term on the left side of Eq. (11) is a correction factor for the difference between the coolant bulk temperature (T_B) for the overall annulus and the area-averaged coolant temperature (T_{B1}) of the flow between the rough rod and the surface of zero shear. These temperatures will not differ significantly for values of the ratio T_w/T_B less than about 1.5 since the distance between the rough rod and the surface of zero shear is the major part of the annular gap and the temperature profile is relatively flat between the surface of zero shear and the wall of the smooth tube. Therefore, this temperature correction can be neglected without introducing significant error for flow conditions such as those listed in Table 4. For more extreme wall-to-bulk temperature ratios, this term can be determined iteratively, since the dimensionless temperature profile in the annular gap is defined by the parameters A_H and $G(h^+)$.^{1,5}

With the temperature correction factor of Eq. (11) neglected, Eq. (8) can be written in the form

$$\frac{\sqrt{f/2}}{St} \frac{u_1^*}{u^*} = A_H \left[\ln\left(\frac{Y_L}{h}\right) - \frac{1}{2} - \frac{1}{2 + Y_L/r_1} \right] + G(h^+). \quad (13)$$

To find the slope A_H graphically, values of the ordinate [on the left side of Eq. (13)] are plotted as a function of the corresponding values of the bracketed abscissa term on the right side. The corresponding values to be plotted must be applicable to flow within the two annular geometries at the same roughness Reynolds number so that the associated values of the wall intercept $G(h^+)$ are the same.

The ordinate terms of Eq. (13) from the results of Dalle Donne and Meyer for rod 8, obtained under the test conditions listed in Table 4, are plotted as a function of the roughness Reynolds number in Fig. 7. The two straight-line approximations on this figure are least-squares fits to the plotted points representing flow in each of the annular geometries, over the ranges indicated. Figure 8 is a similar plot representing the test results for rod 10.

Values for the ordinate [on the left side of Eq. (13)] can be obtained for flow past test rods 8 or 10 at the same value of h^+ in the two different annular geometries by using the applicable linear least-squares fit equations indicated on Figs. 7 or 8. The points calculated in this manner are plotted in the form of Eq. (13) at several values of h^+ in Figs. 9 and 10 for rods 8 and 10 respectively. The values on the abscissa for these points were obtained at the desired increments of h^+ by a least-squares fit to the experimental results for rods 8 and 10 (not shown, but analogous to those shown in Fig. 5).

The slopes and ordinate intercepts of the straight lines shown on Figs. 9 and 10 are the values of A_H and $G(h^+)$ of Eq. (13) for the applicable test rod at the indicated roughness Reynolds numbers. These quantities are listed as functions of h^+ in Table 5.

Mathematically, the slope A_H of Eq. (13) is identical to the gradient of the dimensionless temperature profile.¹ Previous investigators^{7,8} have investigated the dimensionless temperature profile in tubes by means of direct experimental measurement of the temperature distribution across the flow. These measurements in tubes have not indicated a dependence of the gradient A_H upon the roughness Reynolds number. Accordingly, it seems reasonable to conclude from the results presented in Table 5 that the slope A_H of Eq. (13) is ~ 2.35 for rod 8 and 2.05 for rod 10 and that these slopes should be considered to be independent of the roughness Reynolds number h^+ in this region.

The values of $G(h^+)$ calculated from the experimental results for rod No. 8 of Dalle Donne and Meyer using Eq. (13) with a constant value of 2.35 for the slope A_H are shown plotted as a function of h^+ in Fig. 11. As before, the plotted values are limited to those derived from the results of the tests

Fig. 7. Variation of ordinate values with roughness Reynolds number for flow with heat transfer.

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Fig. 8. Variation of ordinate values with roughness Reynolds number for flow with heat transfer.

Fig. 9. Universal law of heat transfer for rod No. 8 tested by Dalle Donne and Meyer.

Fig. 10. Universal law of heat transfer for rod No. 10 tested by Dalle Donne and Meyer.

Table 5. Slopes A_H and ordinate intercepts $G(h^+)$ for the straight lines of Figs. 9 and 10.

Figure	Rod No.	h^+	A_H	$G(h^+)$
9	8	100	2.36	15.59
		200	2.38	19.02
		300	2.37	21.36
		400	2.35	23.20
		500	2.33	24.72
10	10	100	2.05	18.29
		200	2.03	21.60
		300	2.00	23.82
		400	1.96	25.53

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Fig. 11. Correlation for $G(h^+)$ using slope determined graphically for rod No. 8.

described in Table 4, for which the characteristic temperatures of the flows past each rod in the two different annular geometries are compatible. Figure 12 is a similar plot for the results applicable to test rod No. 10.

While this method for determination of the parameters A_H and $G(h^+)$ can be used with the results of the forthcoming single-rod tests of a CFTL prototype rod, it is evident that additional experimental investigation of heat transfer from a roughened rod in annular geometry is necessary. In particular, application of this method to experimental results for flow at compatible temperatures and wall-to-bulk temperature ratios in three or more annular geometries would greatly increase the level of confidence in the values of A_H and $G(h^+)$ determined by this graphical method.

It should also be noted that the method described here is limited in application to the portion of the experimental results for which the linear relations shown on Figs. 7 and 8 are obtained. For flow at lower values of h^+ , estimates for the value of the quantity

$$\frac{\sqrt{f/2}}{St} \cdot \frac{u_1^*}{u^*} \quad (14)$$

would have to be taken directly from these figures.

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Fig. 12. Correlation for $G(h^+)$ using slope determined graphically for rod No. 10.

SUMMARY

Several different velocity profile widths will exist simultaneously in the different channels within the roughened portion of the CFTL bundle. A successful analysis of these channel flows depends upon prior determination of the behavior of the friction factor and Stanton number as functions of the velocity profile width for the particular CFTL roughness. The behavior of the friction factor can be specified by experimental investigation of the parameters A and $R(h^+)$ of Eq. (2); similarly, the behavior of the Stanton number is specified if the variation of A_H and $G(h^+)$ of Eq. (13) is known.

The variation of these four parameters with the roughness Reynolds number h^+ can be experimentally determined by the testing of a single prototypic CFTL roughened cladding surface in a series of annular geometries. The annular geometries for these tests are formed by placing the roughened surface at the center of each of a sequence of smooth tubes of differing diameter; the Dalle Donne-Meyer transformation is used for calculation of both the location of the surface of zero shear in the flow and the friction factor associated with the rough surface. The velocity and temperature profiles need not be measured across the flow.

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NOMENCLATURE

A	Slope of Eq. (2); mathematically equivalent to slope of dimensionless velocity profile
A _H	Slope of Eq. (13); mathematically equivalent to slope of dimensionless temperature profile
C ₁ , C ₂ , C ₃	Constants in Eq. (5)
C ₄ , C ₅ , C ₆	Constants in Eq. (6)
C _p	Specific heat at constant pressure
D	Equivalent diameter
f	Fanning friction factor
G(h ⁺)	Heat transfer roughness parameter for a particular fluid
h	Height of roughness element
h ⁺	Roughness Reynolds number ($= \frac{hu^*}{\nu}$) based on the viscosity at the temperature at the wall
q _w	Heat flux at the wall
r	Radius
Re	Reynolds number
R(h ⁺)	Roughness parameter for frictional pressure loss
St	Stanton number
T _B	Bulk coolant temperature
T _w	Temperature at the wall
T _{wm}	Maximum measured temperature along the rough wall
$\bar{u}$	Spacially averaged axial velocity
u [*]	Friction velocity ($= \sqrt{\tau_w / \rho}$)
$\bar{u}^+$	Average value of dimensionless velocity ($= \bar{u} / u^*$)
Y _I	Velocity profile width, from volumetric surface of rough wall to surface of zero shear
Y _T	Annular gap, distance in annular test geometry from volumetric surface of rough wall to smooth surface

Greek Symbols

α	Radius ratio of annulus ($= r_1 / r_2$)
β	Ratio of radius of surface of zero shear in annular test geometry to radius of smooth tube wall ($= r_0 / r_2$)
ρ	Density
τ	Shear stress at the wall
ν	Kinematic viscosity

NOMENCLATURE (continued)**Subscripts**

0	Pertaining to the surface of zero shear
1	Pertaining to the rough surface, or to the zone of flow in annular geometry between the rough surface and the surface of zero shear
2	Pertaining to the smooth tube wall in annular test geometry

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