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Heat transfer and friction characteristics of spirally corrugated tubes for power plant condensers—2. A mixing-length model for predicting fluid friction and heat transfer

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Abstract—A mixing-length formulation modified in accordance with the suggestion of Rotta is used to predict the Fanning friction factor and the heat transfer coefficient on corrugated (roped) tubes. Several empirical parameters are required in this formulation and they are obtained via experimental data fitting. The experiments are carried out with water flow in corrugated tubes in the ranges $10^4 \leqslant Re \leqslant 6\times10^4$ and $2.2 \leqslant Pr \leqslant 3.4$. Simple correlations are suggested for the relations between the two most important empirical parameters—the dimensionless distance $\Delta\eta$ that shifts downward the reference wall and the cavity Stanton number, St_w, and the geometrical parameters of the ridges of the tubes. These correlations yield the mean value boundary layer profiles, Fanning friction factors and Nusselt (Stanton) numbers through the solution of the momentum and energy transfer equations for one-dimensional time-independent stabilized flow.

INTRODUCTION

In a recent paper [1] we studied theoretically the fluid friction and in-tube heat transfer for a stabilized turbulent flow of an incompressible fluid in a round tube with internally sand-grain-roughened walls at moderate Prandtl numbers. A mixing-length approach [2] was used in the formulation of Rotta [3] to generalize our earlier results [4, 5] for smooth wall tubes. Tables of Fanning friction factors and in-tube Nusselt numbers were presented in refs. [1, 4, 5] to compare our predictions to similar theoretical and/or experimental results published elsewhere.

The mixing-length approach for the study of stabilized turbulent flows with internal roughness depends essentially on the knowledge of a dimensionless quantity $\Delta\eta$ accounting for the influence of the roughness elements on the hydraulic and thermal characteristics of the flow (for smooth tubes $\Delta\eta = 0$). Formally the reference wall is shifted downward by a distance Δy and moves with a velocity ΔU in a direction opposite to the direction of the main flow. If $\Delta\eta$ is known as a function of the geometrical parameters of a particular type of wall roughness, then the procedure developed in our previous studies [1, 4] can easily be applied to predict the friction characteristics of the flow in a circular tube with its walls roughened correspondingly. Usually the dependence of $\Delta\eta$ on the roughness geometry is determined using experimental information [3]. Once the mean value boundary layer profiles are available, with additional assumptions for a turbulent Prandtl number [3] and a cavity Stanton number [6], the numerical solution of the energy equation can be carried out.

The purpose of this communication is to extend the method [3] for the prediction of $\Delta\eta$ for corrugated (roped) tubes [7–11], using data for the geometry of the helical ridges and our experimental information for the hydraulic behaviour of the flow. This information was discussed in detail in part 1 of the present study [12] where it was reported in a form compatible with that of other authors. An attempt is also made to understand better the influence of the ridge shape on the momentum transfer mechanism. Next, we extend the idea of a sand-roughened wall thermal resistance [1, 6] for the case of corrugated tubes [7–10]. Lastly, we report numerical results for the Fanning friction factor and the in-tube Stanton number for the tubes investigated. These results were obtained using the formalism and the slightly modified software described earlier [1, 4, 5]. Due to the absence of detailed information for the geometry of the corrugated tubes studied in refs. [7–10], a direct comparison for the friction factors between our predictions and similar results reported by other authors is not possible.

THE PHYSICAL MODEL

Despite the fact that several studies [7–10] were conducted on a variety of corrugated surfaces, a lack of sufficient knowledge about the flow mechanism over corrugated surfaces does not permit the prediction of the friction factors and the heat transfer rates by analytical methods. The similarity law concept, which was first developed by Nikuradse [13] to correlate the friction results inside sand-grain-roughened tubes, was applied later [6, 14] to heat

NOMENCLATURE

c_p	specific heat capacity [J kg^{-1} K^{-1}]
D_i	tube inside diameter [m]
e	roughness height [m]
h_i	in-tube heat transfer coefficient [W m^{-2} K^{-1}]
k	thermal conductivity [W m^{-1} K^{-1}]
L	mixing length [m]
p	pitch of corrugation [m]
q_w	heat flux [W m^{-2}]
r_o	tube radius [m]
s	cap height of the ridge [m]
t	cap width of the ridge [m]
T	temperature [K]
U	fluid velocity [m s^{-1}]
u_*	shear velocity, $(\tau_w/\rho)^{0.5}$ [m s^{-1}]
x	axial distance [m]
y	radial distance from the wall [m].

Greek symbols

β	helix angle [deg]
ε_h	eddy diffusivity of heat [m^2 s^{-1}]
ε_m	eddy kinematic viscosity [m^2 s^{-1}]
μ	dynamic viscosity [Pa s]
ν	kinematic viscosity [m^2 s^{-1}]
ρ	fluid density [kg m^{-3}]
τ	shear stress [Pa].

Dimensionless groups

e^+	roughness Reynolds number, $(e/D_i)Re(f/2)^{0.5}$
f	Fanning friction factor, $2\tau_w/(\rho U_m^2)$
$G(e^+, Pr)$	heat transfer roughness function, $[(f/2St-1)/(f/2)^{0.5}+R(e^+)]$
L^+	dimensionless mixing length, (Lu_*/ν)
Nu	Nusselt number, (h_iD_i/k)
Pr	Prandtl number, $(c_p\mu/k)$
Pr_t	turbulent Prandtl number, $(\varepsilon_m/\varepsilon_h)$
r_o^+	dimensionless radius of the tube, $(D_iu_*/2\nu)$
Re	Reynolds number, (U_mD_i/ν)
$R(e^+)$	momentum transfer roughness function, $\sqrt{(2/f)}+2.5\ln(2e/D_i)+3.75$
St	Stanton number, $[h_i/(\rho U_m c_p)]$
T^+	dimensionless temperature, $T/(q_w/\rho c_p u_*)$
U^+	dimensionless velocity, U/u_*
β_*	$\beta/90°$
θ	dimensionless temperature, $(T_w-T)/(T_w-T_m)$
η	dimensionless radial distance measured from the wall, (y/r_o)
$\Delta\eta$	dimensionless shift, $(\Delta y/r_o)$
Φ_*	dimensionless group, $(p-t)\cdot s/e^2$.

Subscripts

c	core region
e	at the top of the ridge
i	inside diameter
m	mean value
max	maximum value
r	rough tube
s	smooth tube
w	at the wall.

Superscript

H	constant wall heat flux.

transfer studies. The model [6, 14] is based on the heat–momentum transfer analogy, for a two-region flow model. This model yields the velocity distribution for the turbulence-dominated part of the flow near the wall as

$$U^+ = 2.5\ln(y/e)+R(e^+). \tag{1}$$

The friction factor can be obtained by integrating the velocity $U^+ = U^+(y)$, equation (1), over the entire cross-section of the tube. Utilizing the velocity defect law in the form $U_{max}^+ - U_m^+ = 3.75$, the similarity law for rough surfaces can be obtained in the form

$$R(e^+) = \sqrt{(2/f)}+2.5\ln(2e/D_i)+3.75. \tag{2}$$

Based on the assumption that the wall similarity law applies to both temperature and velocity profiles, in ref. [6] a heat transfer similarity law was proposed

$$G(e^+, Pr) = \frac{(f/2St-1)}{\sqrt{(f/2)}}+R(e^+). \tag{3}$$

The functions R and G, equations (2) and (3), represent the effect of the roughness on the momentum and heat transport in the neighbourhood of the wall. They are independent of the geometry and the size of the test channel and are functions of such local parameters as the roughness Reynolds number

$$e^+ = e/D_i\ Re\sqrt{(f/2)}$$

the geometry of the ribs p/e, β_* and the ridge shape for corrugated tubes. The values of R and G are usually determined from experimental information.

This point of view was shared by many authors investigating corrugated tubes; however, different values for the functions R and G were reported, according to the geometry and the shape of the helical ridging [7–10]. Regardless of this the R and G functions facilitate the comparison of the experimental results reported. In ref. [11] we showed that the function $R = R(e^+)$, representing the effect of the roughness on the momentum transport in the region near the wall, could be related to the following geometrical parameters of the helical ridge (see Fig. 1) and the Reynolds number Re:

$$R = R(Re, e/D_i, \Phi_*, \beta_*) \tag{4}$$

where

$$\Phi_* = \frac{(p-t)s}{e^2} \quad \text{and} \quad \beta_* = \beta/90^\circ.$$

We expect that the same variables as in equation (4) will determine the dimensionless shift $\Delta\eta$ and we shall obtain a simple power law correlation for $\Delta\eta$ in the form

$$\Delta\eta = a_0\, Re^{a_1}(e/D_i)^{a_2}\Phi_*^{a_3}\beta_*^{a_4}. \tag{5}$$

Detailed information for the geometry of corrugated tubes studied from other authors particularly to calculate the complex Φ_* is lacking in the literature and we have to utilize the information from our experiments to obtain this correlation.

In the fully turbulent part of the wall region, the law of the wall for a uniformly roughened surface is similar to that for a smooth surface and the mean velocity distribution for smooth and rough surfaces differ with a constant value ΔU^+ [3], namely

$$\Delta U^+ = U_s^+ - U_r^+ = \Delta U^+(\Delta\eta). \tag{6a}$$

For a particular roughness the value of ΔU^+ can be obtained from

$$\Delta U^+ = 2.5 \ln\{0.5 Re\sqrt{(f/2)}\} + 1.75 - \sqrt{(2/f)} \tag{6b}$$

if the data for the friction factor f are already available.

To investigate the effect of the helical ridge roughness of the tube wall on the total thermal resistance of the fluid the latter can be decomposed into two components:

(a) the thermal resistance of the fluid surrounding the roughness elements in the region $0 \leqslant \eta \leqslant 2e/D_i$;

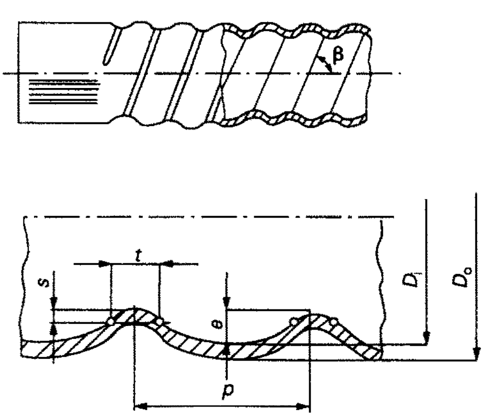

FIG. 1. Characteristic parameters of a spirally corrugated tube.

(b) the thermal resistance of the core region $2e/D_i < \eta \leqslant 1$.

Having this in mind we can describe the total thermal resistance of the fluid in the form of a Stanton number St, as

$$St^{-1} = \frac{\rho c_p U_m}{q_w}(T_w - T_m) = \frac{\rho c_p U_m}{q_w}(T_w - T_c)$$
$$+ \frac{\rho c_p U_m}{q_w}(T_c - T_m) = \frac{\rho c_p U_*}{q_w}(T_w - T_c)\sqrt{(2/f)}$$
$$+ St_c^{-1} = (T_w^+ - T_c^+)\sqrt{(2/f)} + St_c^{-1}$$

where St_c^{-1} is the core resistance which can be calculated using standard techniques [5, 15], while the wall layer resistance St_w^{-1} which depends essentially on the particular type of roughness is defined from

$$St_w^{-1} = \frac{T_w^+ - T_c^+}{\sqrt{(f/2)}} = g(e^+, Pr)\sqrt{(2/f)} = St^{-1} - St_c^{-1}. \tag{7}$$

It is calculated as the difference between the total thermal resistance and the core thermal resistance. The function $g(e^+, Pr)$ in equation (7) is similar to the function $G(e^+, Pr)$, equation (3), defined in ref. [6] and utilized in refs. [7–10, 14]. The two functions, however, could differ quantitatively since the function $g(e^+, Pr)$ reflects the fluid thermal resistance in the region $0 < \eta < 2e/D_i$, i.e. up to the top of the ridge, whereas the function $G(e^+, Pr)$ represents the thermal resistance of the region near the wall beyond which the shear stresses due to viscosity can be neglected completely.

From the above it becomes clear that once $\Delta\eta$ and $g(e^+, Pr)$ are derived using experimental data for f and St, one can apply the procedures developed in refs. [1, 4, 5] to predict fluid friction and intube heat transfer for the case of corrugated tubes.

THE MATHEMATICAL MODEL AND ITS SOLUTION

The solution deals with heat transfer for the case of a fully developed turbulent flow in a corrugated tube having internal helical ridges with boundary conditions of uniform wall heat flux. The uniform wall temperature boundary condition is not studied in what follows because it is known [1, 16] that numerically the difference between the Nusselt (Stanton) numbers for uniform wall temperature and uniform wall heat flux is of the order of the experimental errors. The results to be reported are obtained if the following assumptions hold:

(i) the fluid is single-phase, incompressible and its physical properties are constant;
(ii) the transport processes are time independent;
(iii) the turbulent flow is both hydrodynamically and thermally stabilized;

(iv) the axial conduction and viscous dissipation in the fluid are neglected.

Momentum transfer

For the case considered the momentum equation yields the linear shear stress distribution as

$$\tau/\tau_w = 1-\eta \tag{8a}$$

where the shear stress is assumed to satisfy the constitutive equation

$$\frac{\tau}{\tau_w} = \frac{1}{r_o^+}\left(1+\frac{\varepsilon_m}{\nu}\right)\frac{dU^+}{d\eta}. \tag{8b}$$

Following ref. [1], for the wall region of the rough tube the eddy kinematic viscosity can be defined as

$$\frac{\varepsilon_m}{\nu} = r_o^+\left(\frac{L^+}{r_o^+}\right)^2\frac{dU^+}{d\eta}, \quad 0<\eta<\eta_w \tag{8c}$$

where

$$(L^+/r_o^+) = 0.4(\eta+\Delta\eta)\{1-\exp[-(\eta+\Delta\eta)r_o^+/26]\}. \tag{8d}$$

Obviously, to calculate the velocity $U^+(\eta)$ and the eddy kinematic viscosity $\varepsilon_m/\nu(\eta)$ from equation (8) one has to know the dependence of the dimensionless shift $\Delta\eta$ on the geometry of the corrugated tube. Now following ref. [3], we shall extend the model for sand-grain roughness to helical ridging roughness and we shall establish a relationship between $\Delta\eta$ and Re, e/D_i, Φ_* and β_*. For this purpose equations (8) are integrated from zero to an unknown upper limit $\Delta\eta$ for a given ΔU^+, thus

$$\int_0^{\Delta\eta}\left(\frac{dU^+}{d\eta}\right)d\eta-\Delta U^+ = 0 \tag{9a}$$

with

$$\frac{dU^+}{d\eta} = \frac{2r_o^+[1-(\eta+\Delta\eta)]}{1+\{1+4r_o^{+2}[1-(\eta+\Delta\eta)](L^+/r_o^+)^2\}^{0.5}}. \tag{9b}$$

The values for ΔU^+ in equation (9a) were obtained from equation (6b) using the experimental data for the friction factor coefficient f. Now $\Delta\eta$ can be calculated as the root of the (implicit) non-linear equation (9a). Figure 2 shows the calculated results for the case of helical ridging roughness. They can be approximated by (see also equation (5))

$$\Delta\eta = 108.5Re^{-0.58}(e/D_i)^{0.89}\beta_*^{2.14}\Phi_*^{-0.07} \tag{10}$$

in the ranges

$$10^4 < Re < 6\times10^4;\quad 0.017 < e/D_i < 0.047;$$
$$0.760 < \beta_* < 0.950;\quad 1.40 < \Phi_* < 5.90.$$

To obtain correlation (10) we used modifications of the QUADPACK routine QNG [17] to approximate the integral in equation (9a), of the function ZEROIN [18] to solve equation (9a) for $\Delta\eta$ for different values of ΔU^+ and the LINPACK routines DQRDC and DQRSL [19] to find the linear least squares fit for the data. The data set utilized to derive correlation (10) comprised 150 experimental points. The relative residuals for $\Delta\eta$ did not exceed $\pm10\%$ for all points.

In the core region there will be no correction for the wall roughness and the eddy kinematic viscosity distribution is the same as in ref. [1] (see also ref. [20])

$$\varepsilon_m/\nu = 0.07044r_o^+[1-(1-\eta)^2]\{1+2.345(1-\eta)^2\} \tag{11}$$

in the range $\eta_w \leqslant \eta \leqslant 1$.

The computational procedure which yields the velocity profile $U^+(\eta)$ and the values for r_o^+ and η_w from equations (8), (10) and (11) was discussed in detail in ref. [1]. Once the velocity distribution is available, the Fanning friction factor f can be predicted from

$$f = 0.5\left\{\int_0^1(1-\eta)U^+(\eta)\,d\eta\right\}^{-2}. \tag{12}$$

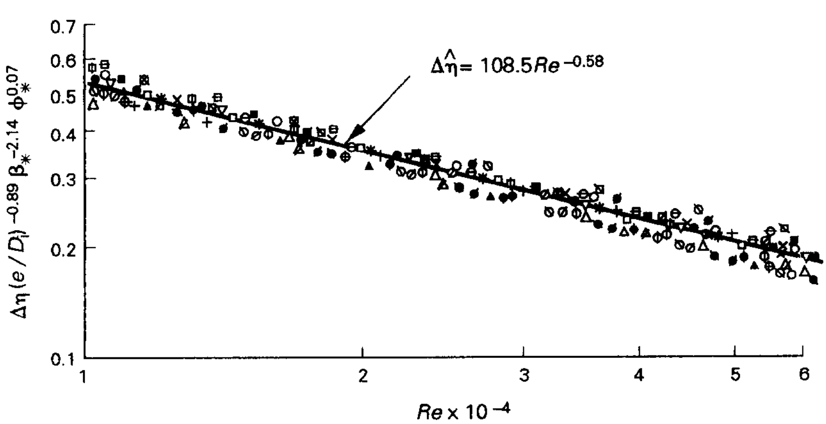

FIG. 2. Variations of $\Delta\eta(e/D_i)^{-0.89}\beta_*^{-2.14}\Phi_*^{0.07}$ with Reynolds number.

Heat transfer

With the above assumptions the energy equation in the case of a constant wall heat flux boundary condition is [21]

$$\frac{\mathrm{d}}{\mathrm{d}\eta}\left\{(1-\eta)\left[1+\frac{Pr}{Pr_t}\frac{\varepsilon_m}{\nu}\right]\frac{\mathrm{d}\theta}{\mathrm{d}\eta}\right\} + U^+(\eta)(f/2)^{0.5}(1-\eta)Nu = 0 \quad (13a)$$

with

$$\eta = 0, \theta = 0; \quad \eta = 1, \quad \frac{\mathrm{d}\theta}{\mathrm{d}\eta} = 0. \quad (13b)$$

Taking into account the correction for the roughness $\Delta\eta$, the turbulent Prandtl number Pr_t is defined [1, 3, 21] as

$$Pr_t = \begin{cases} 0.909B^+/26, & \eta = 0 \\ 0.909\dfrac{1-\exp[-(\eta+\Delta\eta)r_o^+/26]}{1-\exp[-(\eta+\Delta\eta)r_o^+/B^+]}, & 0 < \eta \leqslant \eta_w \\ 0.909, & \eta_w < \eta \leqslant 1 \end{cases} \quad (13c)$$

$$B^+ = Pr^{-0.5}\sum_{k=1}^{5} C_k(\log Pr)^{k-1}, \quad 0.02 < Pr < 15 \quad (13d)$$

and $C_1 = 31.96$, $C_2 = 28.79$, $C_3 = 33.95$, $C_4 = 6.33$, $C_5 = -1.186$.

The solution of the heat transfer problem, equations (13), is a particular case of the problem discussed in detail in ref. [1]. Part of this procedure is applied to compute the heat transfer coefficient Nu^H in tubes with helical ridging, calculating the values of Lyon's integral [15]

$$(Nu^H)^{-1} = 2\int_{2(e/D_i)}^{1} \frac{\left(\int_{1-\eta}^{1}(1-\eta)U^+(\eta)\,\mathrm{d}\eta\right)^2}{(1-\eta)\left(1+\frac{Pr}{Pr_t}\frac{\varepsilon_m}{\nu}\right)}\mathrm{d}\eta. \quad (14)$$

Many authors [9, 10, 22–24], indicate the insensitivity of the G-function to the variation of the shape and the geometrical parameters of the turbulence promoters, since the function $R(e^+)$ involved in the computation of the G-function already takes into account the hydraulic characteristics of the flow and the roughness parameters. We accept those arguments and suggest a power law relationship for the g-function in the form

$$g(e^+, Pr) = b_0(e^+)^{b_1}Pr^{b_2}. \quad (15a)$$

As discussed in an earlier report [25], the values of the constants b_1 and b_2 in equation (15a) can be found applying Huntley's dimensional analysis [26] without any experimental information. The values derived from the model [25] are $b_1 = 0.20$ and $b_2 = 0.60$. To verify the model [25] the variation of $g(e^+, Pr)Pr^{-0.60}$ with e^+ was investigated, and Fig. 3 shows this variation where the final correlation obtained is

$$g(e^+, Pr) = 1.96(e^+)^{0.265}Pr^{0.6}, \quad 20 < e^+ < 300. \quad (15b)$$

RESULTS AND DISCUSSION

The Fanning friction factors calculated by equation (12) were compared with 346 experimental points obtained from 25 corrugated tubes tested. Two hundred and ninety-one points show a relative difference of less than ±10% and for the remaining 56 points the relative difference is ±10–15%. The numerical results for the friction factors together with the experimental data of some corrugated tubes tested are presented in Table 1.

The heat transfer coefficients (transformed as Stanton numbers) were calculated using equations (13)–(15). Three hundred and twenty-four calculated points were compared with those measured from the experiments. Except 19, all points show a relative difference of less than ±10% and the maximum relative difference is less than ±15%. Part of the results

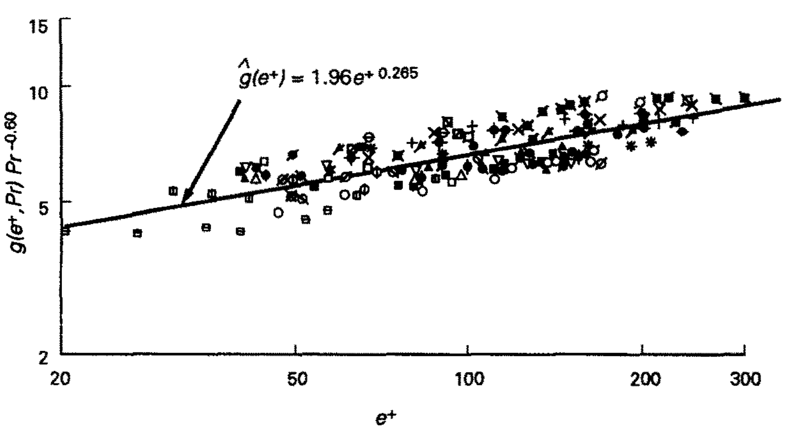

FIG. 3. Variations of $g(e^+, Pr)Pr^{-0.60}$ with roughness Reynolds number (e^+).

Table 1

Tube No.	Re	$f_{exp.}$	$f_{calc.}$	Tube No.	Re	$f_{exp.}$	$f_{calc.}$	Tube No.	Re	$f_{exp.}$	$f_{calc.}$
11	14400	0.0216	0.0209	18	15900	0.0316	0.0321	26	14500	0.0238	0.0217
	36800	0.0191	0.0194		36900	0.0282	0.0275		36200	0.0202	0.0200
	59900	0.0174	0.0184		53100	0.0271	0.0254		58900	0.0190	0.0189
12	16400	0.0193	0.0213	19	13700	0.0277	0.0280	27	13700	0.0286	0.0270
	33500	0.0178	0.0200		32200	0.0244	0.0248		35000	0.0258	0.0238
	57900	0.0162	0.0180		55900	0.0232	0.0226		58800	0.0240	0.0219
13	14500	0.0226	0.0239	21	19300	0.0232	0.0229	29	15400	0.0264	0.0262
	36500	0.0190	0.0217		33400	0.0218	0.0216		33500	0.0231	0.0235
	59400	0.0179	0.0203		53200	0.0207	0.0204		55500	0.0220	0.0219
15	15600	0.0214	0.0221	23	13600	0.0230	0.0250	31	16200	0.0234	0.0231
	39300	0.0191	0.0202		37300	0.0202	0.0222		35400	0.0210	0.0214
	59500	0.0174	0.0192		58500	0.0184	0.0208		53100	0.0200	0.0203
17	14800	0.0260	0.0272	24	14300	0.0215	0.0222	32	15800	0.0341	0.0321
	36800	0.0241	0.0240		36700	0.0196	0.0203		33600	0.0305	0.0280
	58100	0.0228	0.0221		57600	0.0181	0.0193		55800	0.0280	0.0252
33	15400	0.0143	0.0136	34	17200	0.0185	0.0181	35	14400	0.0363	0.0343
	31900	0.0130	0.0128		35700	0.0160	0.0173		29700	0.0332	0.0300
	53700	0.0120	0.0125		56100	0.0152	0.0166		49600	0.0305	0.0268

Table 2

Tube No.	Re	Pr	$St_{calc.}$	$St_{exp.}$	Tube No.	Re	Pr	$St_{calc.}$	$St_{exp.}$
13	13900	3.23	0.00466	0.00446	21	15100	3.25	0.00451	0.00453
	40300	2.46	0.00426	0.00460		39200	2.57	0.00414	0.00445
	55200	2.20	0.00416	0.00452		51200	2.36	0.00407	0.00415
15	13700	3.17	0.00419	0.00425	24	14000	3.21	0.00444	0.00424
	24600	2.77	0.00437	0.00439		27500	2.72	0.00434	0.00450
	49100	2.19	0.00425	0.00465		51300	2.21	0.00420	0.00446
17	14000	3.24	0.00506	0.00458	26	15700	3.10	0.00442	0.00428
	37100	2.55	0.00464	0.00424		26400	2.75	0.00431	0.00456
	52800	2.23	0.00447	0.00418		51300	2.25	0.00407	0.00427
18	15800	3.12	0.00560	0.00509	29	17100	2.88	0.00488	0.00478
	39300	2.43	0.00502	0.00511		27400	2.59	0.00466	0.00469
	52200	2.18	0.00478	0.00491		49700	2.18	0.00443	0.00470
19	13400	3.21	0.00507	0.00476	31	13700	3.17	0.00471	0.00433
	28800	2.67	0.00473	0.00452		39000	2.46	0.00441	0.00448
	53400	2.20	0.00446	0.00434		52700	2.18	0.00425	0.00453
32	16300	3.03	0.00568	0.00515	33	13900	3.14	0.00301	0.00306
	27200	2.66	0.00536	0.00506		41300	2.40	0.00316	0.00337
	52000	2.17	0.00484	0.00516		51600	2.22	0.00309	0.00350

are summarized in Table 2. Taking into account the experimental error in the measurements, this agreement should be considered as fairly good.

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CARACTERISTIQUES DE TRANSFERT THERMIQUE ET DE FROTTEMENT POUR DES TUBES CORRUGUES EN SPIRALE POUR CONDENSEURS DE CENTRALE THERMIQUE—2. UN MODELE DE LONGUEUR DE MELANGE POUR PREDIRE LE FROTTEMENT FLUIDE ET LE TRANSFERT THERMIQUE

Résumé—On utilise une formulation de longueur de mélange modifiée pour s'accorder à la suggestion de Rotta afin de prédire le facteur de frottement et le coefficient de transfer dans les tubes corrugués. Plusieurs paramètres empiriques sont nécessaires dans celle formulation et ils sont obtenus à l'aide des données expérimentales. Les expériences sont faites avec de l'eau dans des tubes corrugués dans les domaines $10^4 \leqslant Re \leqslant 6 \times 10^4$ et $2{,}2 \leqslant Pr \leqslant 3{,}4$. Des formules simples sont suggérées pour les relations entre les deux paramètres les plus importants, à la distance adimensionnelle $\Delta\eta$ et le nombre de Stanton St_w de la cavité, ainsi que les paramètres géométriques. Ces formules donnent les profils des couches limites, les coefficients de frottement et les nombres de Nusselt (Stanton) à travers les solutions des équations de la quantité de mouvement et de l'énergie pour un écoulement monodimensionnel, indépendant du temps.

WÄRMEÜBERGANG UND DRUCKABFALL AN SPIRALFÖRMIG GERILLTEN ROHREN FÜR KRAFTWERKSKONDENSATOREN—2. EIN MISCHUNGSWEGMODELL ZUR BESTIMMUNG VON REIBUNG UND WÄRMEÜBERGANG

Zusammenfassung—Es wird ein Mischungswegmodell, welches entsprechend den Vorschlägen von Rotta modifiziert wurde, zur Berechnung des Widerstandsbeiwerts und des Wärmeübergangskoeffizienten in gerillten Rohren verwendet. Dabei wird eine Reihe empirischer Parameter benötigt, die durch Anpassung an Versuchsdaten gewonnen werden. Die Versuche werden in wasserdurchströmten, gerillten Rohren im Bereich der Reynolds-Zahl von 10^4 bis 6×10^4 und der Prandtl-Zahl von 2,2 bis 3,4 durchgeführt. Es werden einfache Korrelationen für die Beziehung zwischen den beiden wichtigsten Parametern angegeben—dem dimensionslosen Abstand $\Delta\eta$, der sich von oben nach unten entlang der Bezugswand und mit der Stanton-Zahl des Hohlraums St_w verändert sowie den geometrischen Parametern der Rohrrillen. Diese Korrelationen liefern die mittleren Grenzschichtprofile, die Widerstandsbeiwerte und die Nusselt-(Stanton-) Zahl durch die Lösung der Impuls- und Energiegleichung für eindimensionale stationäre stabilisierte Strömung.

ХАРАКТЕРИСТИКИ ТЕПЛОПЕРЕНОСА И ТРЕНИЕ ТРУБ СО СПИРАЛЬНЫМ ОРЕБРЕНИЕМ В КОНДЕНСАТОРАХ ЭНЕРГЕТИЧЕСКИХ УСТАНОВОК—2. ИСПОЛЬЗОВАНИЕ МОДЕЛИ ДЛИНЫ СМЕШЕНИЯ ДЛЯ РАСЧЕТА ГИДРОДИНАМИЧЕСКОГО ТРЕНИЯ И ТЕПЛОПЕРЕНОСА

Аннотация—Формулировка длины смешения, модифицированная в соответствии с предположением Ротта, используется для расчета коэффициентов трения и теплопереноса в оребренных (с навивкой) труб. При этом требуются несколько эмпирических параметров, которые получены посредством обобщения экспериментальных данных. Эксперименты проводились с потоком воды в оребренных трубах в диапазонах $10^4 \leqslant Re \leqslant 6 \times 10^4$ и $2{,}2 \leqslant Pr \leqslant 3{,}4$. Предложены простые соотношения между безразмерным расстоянием $\Delta\eta$, числом Стэнтона для полости St_w и геометрическими параметрами, характеризующими оребрение труб. Эти соотношения позволяют определить профили средних величин в пограничном слое, коэффициенты трения и числа Нуссельта (Стэнтона) с помощью решения уравнений количества движения и энергии для одномерного стационарного устойчивого течения.