Heat Transfer Augmentation in Gas-Cooled Channels

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PREFACE

This book is concerned with relationships governing high-rate, enhanced turbulent heat transfer in gas-cooled channels, derived as a part of an ongoing study of these problems conducted during the past decade in our Institute. It continues a cycle of study of local heat transfer and friction in smooth annuli under conditions when variability of physical properties of the working fluid must be taken into account.

This book presents results of detailed measurements of fluid-dynamic and thermal parameters on smooth and rough heat-emitting cylinders in axial flow, and heat-transfer and hydraulic-drag coefficients in rough annular and helical channels and in bundles of twisted tubes in axial flow. Data on heat transfer and drag obtained for rough annular channels are recalculated for conditions prevalent in bundles of rough tubes.

A place of significance is assigned in the book to the study of flow structure and turbulence in bundles of twisted tubes. This made it possible to explain a number of features specific to heat transfer in such systems. This information is necessary for improving the reliability of gas-cooled heat exchangers operating at high heat flux densities. The physical model of flow in bundles of twisted tubes is refined on the basis of flow-structure data.

The experimental arrangements and techniques are described. The results are correlated in a general form, which is suitable for practical application over a wide range of operating and geometric parameters. A part of the most typical experimental data is given in tabulated form.

The authors wish to express their heartfelt thanks to Prof. A. Žukauskas for his interest in all the details of the manuscript and for valuable advice, to A. Sakalauskas, who actively participated in investigating the flow structure in bundles of twisted tubes, and to E. Ušpuras, L. Burkoj, and all those who participated in preparing this book for publication.

The authors

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NOMENCLATURE

b	Width of roughness elements, helical channel, lobes of twisted tubes, space between tubes within a bundle, m
$c_f = 2\tau_w/\rho u_\infty^2$	Friction coefficient
c_p	Specific heat, J/kg · K
D	Diameter of the curvature of a coiled passage, m
d_0	Diameter of the cylinder, of the zero shear stress surface, describing a coiled tube, m
d_1, d_2	Diameters of washed surfaces of tubes comprising an annular or helical channel, m
d_e	Equivalent diameter of channel, m
E	Voltage across the anemometer wires, V
F	Net cross-sectional area, heating-wall surface, m^2
G	Working-fluid flow rate, kg/sec; coefficient
$G(k^+), G_1(k^+)$	Thermal functions of roughness
$H = \delta^*/\delta^{**}$	Shape factor
h	Height of helical passage, m
I	Current, A
k	Height of roughness projections, m
$k^+ = ku_*/\nu$	Dimensionless height of roughness projections at temperature T_f
l	Mixing length, length of channel, m
p	Pressure, Pa
Δp	Pressure differential, Pa
Q	Heat flux, W
q	Heat flux density, W/m^2
$R(k^+), R_1(k^+)$	Hydrodynamic functions of roughness
r	Tube radius, current radius, m
r_0	Radius of cylinder, of zero shear stress surface, m
$r_0^+ = r_0 u_*/\nu$	Dimensionless radius
s	Spacing between roughness elements, pitch of helical channel, of twisted tubes, m
s_1	Pitch of tubes within a bundle, m
T	Temperature, K

U	Potential drop, V
u, v, w	Velocity components, m/sec
u', v', w'	Velocity fluctuations, m/sec
u_0	Bulk velocity, m/sec
u_∞	Free-stream velocity, m/sec
u_r, u_τ	Radial and tangential velocity components, m/sec
$u_* = \sqrt{\tau_w/\rho}$	Friction velocity, m/sec
$u^+ = u/u_*$	Dimensionless velocity
$\mathbf{V}$	Total-velocity vector, m/sec
x, y, z	Cartesian coordinates
x	Length, distance from start of heating, m
x_0	Virtual start of turbulent boundary layer, m
y	Current distance along the radius, calculated from the surface of tubes or cylinder, m
$y^+ = yu_*/\nu$	Dimensionless distance
$y_R = r_0 \ln(r/r_0)$	Coordinate of axisymmetric boundary layer, m
$y_R^+ = r_0^+ \ln(r/r_0)$	Dimensionless coordinate
α	Heat-transfer coefficient, $\mathrm{W/m^2 \cdot K}$; angle, radius
δ	Thickness of boundary layer, of wall, m
δ^*	Displacement thickness, m
δ^{**}	Momentum thickness, m
ε_q	Eddy thermal diffusivity, $\mathrm{m^2/sec}$
ε_τ	Eddy viscosity, $\mathrm{m^2/sec}$
η	Efficiency
$\vartheta = T_w - T$	Temperature differential, K
$\vartheta_* = q_w/\rho c_p u_*$	Characteristic temperature, K
$\vartheta^+ = \vartheta/\vartheta_*$	Dimensionless temperature
$\varkappa$	Universal constant
$\Lambda = l/\delta_x$	Normalized mixing length
λ	Thermal conductivity, $\mathrm{W/m \cdot K}$
μ	Dynamic viscosity, $\mathrm{N \cdot sec/m^2}$
ν	Kinematic viscosity, $\mathrm{m^2/sec}$
ξ	Hydraulic drag coefficient
Π	Channel perimeter, m
ρ	Density, $\mathrm{kg/m^3}$
τ	Shear stress, $\mathrm{N/m^2}$
φ	Angle, degrees
$\Psi = T_w/T_f$	Temperature factor
$\mathrm{Fr}_M = s^2/d_0 d_e$	Modified Froude number

$\mathrm{Nu} = \alpha d_e/\lambda$, $\mathrm{Nu}_x = \alpha x/\lambda$ Nusselt numbers at pertinent reference parameters

$\mathrm{Pr} = \mu c_p/\lambda$ Prandtl number

$\mathrm{Pr}_T = \varepsilon_\tau/\varepsilon_q$ Turbulent Prandtl number

$\mathrm{Re} = u_0 d_e/\nu$ Reynolds numbers at pertinent reference parameters

$\quad \mathrm{Re}_{r_0} = u_\infty r_0/\nu$

$\quad \mathrm{Re}_x = u_\infty x/\nu$

$\quad \mathrm{Re}_{\delta*} = u_\infty \delta^*/\nu$

$\mathrm{St} = \alpha/\rho c_p u_\infty$ Stanton number at pertinent reference parameters

$\quad = \mathrm{Nu}/\mathrm{Re}\,\mathrm{Pr}$

$Tu_\infty = \sqrt{\overline{u'^2}}/u_\infty$ Turbulence intensity

Subscripts

0	At the plate, for a smooth surface, for the channel as a whole
1	Inner tube, zone of annular channel
2	Outer tube, zone of annular channel
∞	In the free stream, in the stabilized heat-transfer region
f	In the flow
in	At the inlet
out	At the exit
s	For sand roughness
t	In the tube
tr	Upon transition from partial to complete manifestation of roughness
w	At the wall
$\Psi = 1$	At constant physical properties
$\overline{(\ )}$	Averaging

Other symbols are defined in the text.

1
Introduction

Engineering progress in the conversion of thermal or chemical energy into other forms involves searching for new methods of conducting these processes at higher temperatures, which allows for improvement of the efficiency of thermodynamic or thermochemical systems. In the majority of cases, the working fluids or coolants in such systems are gaseous, and exchange of thermal energy between them occurs through a solid wall. To be able to manufacture equipment with acceptable weight, size, and performance, one must significantly augment the heat-transfer processes, which is attained by producing large temperature differentials and also by various other methods of intentional enhancement of heat transfer: use of artificial roughness, turbulization or swirling of the flow, reducing the thickness of boundary layers or increasing the velocity gradient in them, etc. Enhancement of heat transfer in equipment such as gas-cooled nuclear reactors, all kinds of heat exchangers, and eventually high-temperature thermoelectrochemical systems of hydrogen production and direct conversion of chemical into electrical energy, is most effectively attained by using rough heating surfaces.

A great deal of attention has been paid to the enhancement of heat transfer under power-plant conditions. This is indicated by the large number of new and even reissued monographs[1-8] in which these problems are analyzed in extensive detail.

Particular attention has been given to operating reliability and operating life of high-temperature systems. The solution of this problem depends to a large extent on successful selection of the design and shape of the heating surface. In this respect, an advantage is held by heat exchangers, consisting of densely packed bundles of twisted tubes, which do not require special spacing grids or exhibit high vibration

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resistance and internal stiffness. As shown in some books,[3,4] the significant intertube mixing occurring in such heat exchangers brings about a rapid equalization of various flow and temperature maldistributions. This is very important when the heat in the unit is released in a highly nonuniform manner or in the presence of flow perturbations produced by inlet and discharge devices. Heat-transfer systems of circular elements of different diameter in axial flow and annular channels are extensively employed. Basically these are bundles of fuel rods of nuclear reactors and blankets of thermonuclear reactors, cooling channels of various industrial devices, etc.

High-temperature systems with gaseous working fluids operate as a rule at high-temperature differences, which makes it always necessary to make allowance for variation in physical properties in the boundary layer, which significantly complicates the calculation of heat transfer and drag.

Studies of heat transfer and hydrodynamics in variously shaped gas-cooled channels have been ongoing in our Institute for a long time now. Initial attention was focused on the effect of variability of physical properties in the boundary layer on heat transfer and drag in flow over smooth surfaces. The results of these studies are correlated in the book by Vilemas *et al.*[9] Subsequent studies centered on investigating the specifics of enhancement of heat transfer on rough surfaces and in rough channels at high-temperature gradients in the boundary layer. In addition, the effect of the transverse curvature (diameter) of a cylinder in longitudinal flow on the structure of the boundary layer, heat transfer, and drag was investigated in detail under these conditions. The effect of free-stream turbulence on the structure of the boundary layer, whose thickness is commensurate with or larger than the diameter of the cylinder, was investigated in detail.

Direct studies of enhancement of heat transfer in fully rough ducts (in pipes with rough inner surface, in bundles of rough tubes in axial flow) involve great difficulties, in particular due to the difficulties and high cost of applying the roughness on the inner surface of pipes, complexity of designs of models of fuel-rod bundles, and difficulties in interpretation and correlation of results obtained with such bundles. Indirect methods for determining the heat-transfer and hydraulic-drag coefficients in completely rough ducts on the basis of data obtained in partially rough annular channels, formed by a smooth outer shell and rough inner pipe, have been developed during the past 15 years. For this reason, in this study enhancement of heat transfer in ducts was investigated primarily in such annuli.

The production of artificial roughness is one of the effective methods for the enhancement of convective heat transfer, particularly at high

heat loads, since the roughness elements turbulize the wall layer in which the bulk of the thermal resistance is concentrated.

The study of relationships governing the effect of roughness and the search for the most effective shapes and optimal dimensions of roughness elements are the subject matter of a large number of experimental and a number of analytic studies; however, most of them were performed at constant physical properties (small temperature differentials) of the working fluid.

Nikuradse's[10] classical study is one of the first investigations of hydraulic drag in pipes with sand roughness, and its results are employed up to the present when investigating other types of manmade roughness. Investigations performed over a wide range of Reynolds numbers and relative heights of roughness elements k_s/r provide insight into the relationships governing the interaction between the roughness and the boundary layer, and help establish typical modes of roughness manifestation. Three flow modes were found to exist, depending on the dimensionless height of sand roughness $k_s^+ = k_s u_* /\nu$ or the so-called roughness Reynolds number: (I) hydraulically smooth flow, when $k_s^+ < 5$, whereas $\xi = f(\mathrm{Re})$, when all the roughness projections are contained within the laminar sublayer and the hydraulic drag is the same as in a smooth pipe; (II) transition flow (flow with partial manifestation of roughness), when $5 \le k_s^+ \le 70$ and $\xi = f(k_s/r, \mathrm{Re})$, in which the increase in drag as compared with a smooth pipe is primarily caused by the shape of the roughness elements, which protrude partially from the laminar sublayer; and (III) completely rough flow, when $k_s^+ > 70$, and $\xi = f(k^+)$, when all the roughness elements protrude from the laminar sublayer, in which case the bulk of the resistance is controlled by the shape of the roughness projections and ξ is no longer a function of Re.

The flow modes observed in channels with manmade regular two- or three-dimensional roughness are similar, except that the transitions between the flow modes may be shifted slightly in one or another direction, depending on the degree of streamlining of the roughness elements. The shape of these elements also significantly affects the effectiveness of heating surfaces. A distinction is made between integral roughness, which conducts heat as a single entity with the wall, and applied roughness elements (Fig. 1.1). The roughness element shapes most extensively used in practice are the trapezoidal (b) and (g), rectangular (c), semicircular rolled (j) shapes, and also wound wires (l) and meshes (n). It is known that changing the shape of roughness elements while maintaining their height and pitch has little effect on the heat transfer and a significant effect on the hydraulic drag. Rectangular roughness elements exhibit the highest drag, whereas rounding off
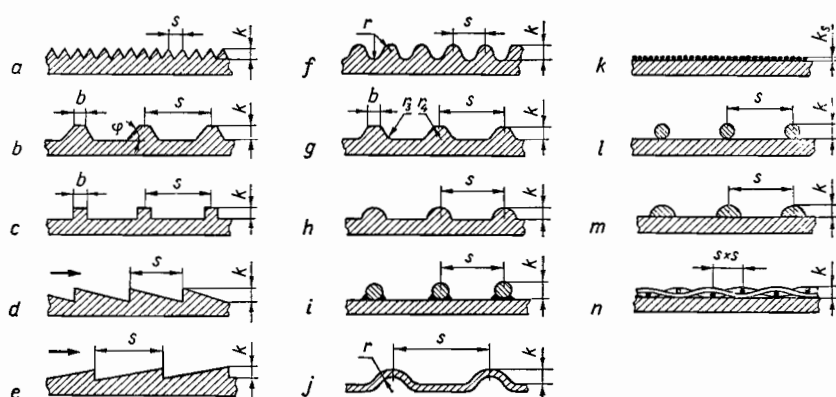
FIGURE 1.1. Principal types and shapes of roughness elements. (a)–(e) integral sharp edged; (f)–(j) integral rounded off; (k)–(n) applied.

of the asperities reduces the drag, which is caused primarily by the reduction of the shape drag coefficient.

The velocity distribution in a rough duct is less filled than in a smooth duct, but it still remains logarithmic near the wall (Fig. 1.2), except that in this case this distribution is displaced by the amount $\Delta u / u_{*}$, which in general is a function of k^+, the type of roughness and the density of asperity distribution, and takes the form

$$u^+ = \frac{1}{\varkappa} \ln \frac{y}{k} + R(k^+). \qquad (1.1)$$

Here $\varkappa = 0.4$ is the universal constant in the distribution of the Prandtl mixing length $l = \varkappa y$, whereas $R(k^+) = 5.5 - A$ is a hydrodynamic roughness function, containing the constant 5.5 from the velocity distribution for smooth ducts, and the coefficient A, which depends on the kind of flow, the type of roughness, and its geometric parameters.

In completely rough flow $R(k_s^+) = 8.5$, whereas in hydraulically smooth and transition flow $R(k_s^+)$ is a function of k_s^+ (Fig. 1.3).

The hydraulic-drag coefficient can be obtained easily by integrating velocity distribution (1.1). In general this coefficient for differently shaped ducts can be expressed as

$$\sqrt{8/\xi} = 2.5 \ln(y_0/k) + R(k^+) - G, \qquad (1.2)$$

where y_0 is the distance from the wall to the zero shear stress curve
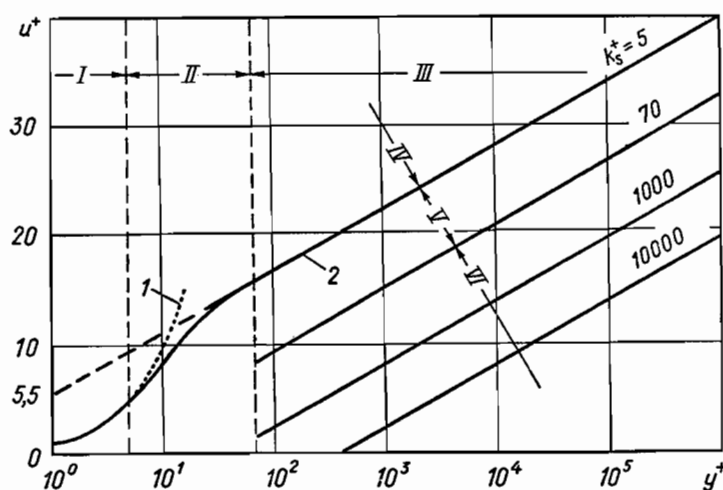
FIGURE 1.2. Logarithmic velocity distribution in turbulent flow in smooth and rough pipes according to Schlichting.[11] (I) Laminar sublayer; (II) transition region; (III) fully turbulent flow; (IV) hydraulically smooth flow; (V) transition flow (partial manifestation of roughness); (VI) completely rough flow; (1) $u^+ = y^+$; (2) $u^+ = 2.5 \ln y^+ + 5.5$.
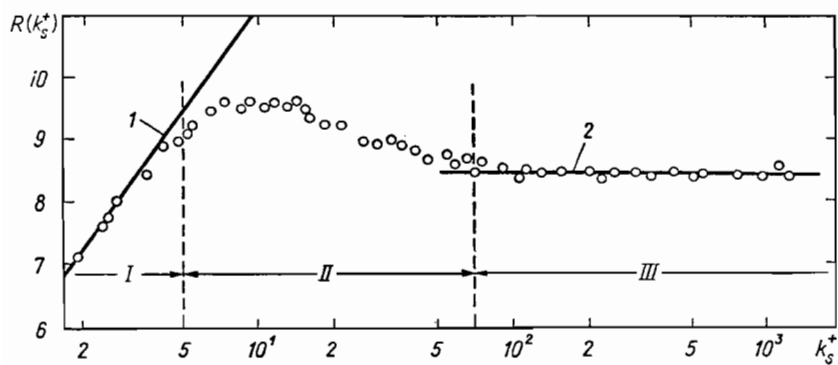
FIGURE 1.3. Distribution of $R(k_s^+)$ as a function of k_s^+ for sand roughness.[11] (I) Hydraulically smooth surface; (II) transition flow; (III) completely rough flow; (1) $R(k_s^+) = 2.5 \ln k_s^+ + 5.5$; (2) $R(k_s^+) = 8.5$.
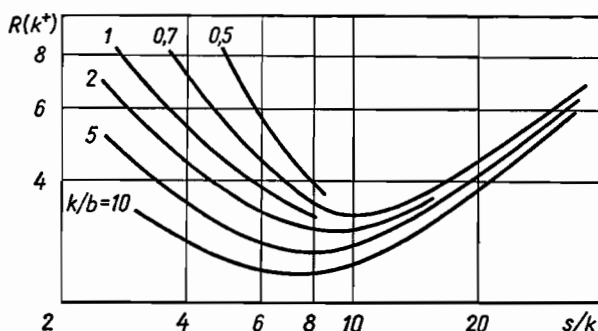
FIGURE 1.4. Characteristic distribution of $R(k^+)$ as a function of s/k and k/b in completely rough flow over rectangular roughness elements according to Maubach.[12]

and G is a geometric parameter which is a function of the cross-sectional shape of the duct. For a circular pipe $y_0 = r$ and $G = 3.75$, for a flat slot (with width h) $y_0 = h/2$ and $G = 2.5$, and for an annular duct $y_0 = r_0 - r_1$ and $G = (3.75 + 1.25 r_0/r_1)/(1 + r_0/r_1)$.

The hydraulic drag in completely rough flow in circular pipes with sand roughness is given by the expression

$$\xi = \frac{8}{[2.5 \ln(r/k_s) + 4.75]^2}. \tag{1.3}$$

Roughness employed for enhancement of heat transfer in heat exchangers, particularly in gas-cooled nuclear reactors, consists primarily of transverse projections of different shape (Fig. 1.1). The type of roughness most investigated is that in the shape of rectangular projections, whose principal dimensions were optimized in order to achieve maximum enhancement of heat transfer [obtain the minimum value of $R(k^+)$]. It was found that $R(k^+)$ is a function both of the relative pitch s/k of the roughness elements and the relative width k/b (Fig. 1.4), i.e., for each type of roughness $R(k^+) = f(s/k, k/b)$. According to Han et al.,[13] the optimum value of s/k at $0.3 < k/b \leq 15$ is given by the expression

$$(s/k)_{\text{opt}} = 9.9(k/b)^{-0.345} \tag{1.4}$$

and ranges from 7 to 12.

According to Bauman and Rehme,[14] the hydraulic drag of surfaces with rectangular transverse projections for the case of completely rough flow can be determined from Eq. (1.2), in which the roughness function

is calculated from the expressions

$$R(k^+) = 0.97(s/k)^{0.53} \quad \text{at} \quad s/k \geq 10, \tag{1.5}$$

$$R(k^+) = 4.45(s/k)^{-0.13} \quad \text{at} \quad s/k \leq 10. \tag{1.6}$$

Dipprey and Sabersky[15] showed on the basis of the analogy between the friction drag and heat transfer (at $Pr_T \approx 1$) that the temperature distribution in the wall region is governed by a universal logarithmic law, analogous to the velocity distribution

$$\vartheta^+ = 2.5 \ln(y/k) + G(k^+, Pr), \tag{1.7}$$

where $G(k^+, Pr)$ is the thermal function of roughness, controlled by the behavior of k^+ and Pr. It then follows from the Reynolds analogy that the Stanton number can be obtained from the expression

$$St = \frac{\xi/8}{1 + \sqrt{\xi/8}\,[G(k^+, Pr) - R(k^+)]}. \tag{1.8}$$

Dipprey and Sabersky found for completely rough flow over sand roughness that $R(k_s^+) = 8.48$, and expressed the thermal roughness function as

$$G(k_s^+, Pr) = 5.19(k_s^+)^{0.2}\, Pr^{0.44}. \tag{1.9}$$

Hudina[16] recommends that the thermal function for completely rough flow over transverse projections of different shape be determined from the expression

$$G(k^+, Pr) = 4.5(k^+)^{0.24}\, Pr^{0.44}, \tag{1.10}$$

which is valid at $25 < k^+ < 300$.

The relationships governing heat transfer and drag are different in transition and in completely rough flow. Thus the curve of $St = f(Re)$ peaks, whereas the behavior of $\xi = f(Re)$ becomes self-similar with respect to Re (Fig. 1.5). As seen from the figure, the hydraulic drag increases in direct proportion to the height of the roughness elements, whereas the heat-transfer coefficient increases only to a certain limit. Hence, in addition to the optimum relative pitch $(s/k)_{opt}$ there also exists an optimum relative height of roughness elements $(k/y_0)_{opt}$, at which the maximum heat-transfer coefficients have a minimum increase in hydraulic drag corresponding to them. According to Sheriff and Gumley,[18] who investigated the heat transfer and drag of an annular
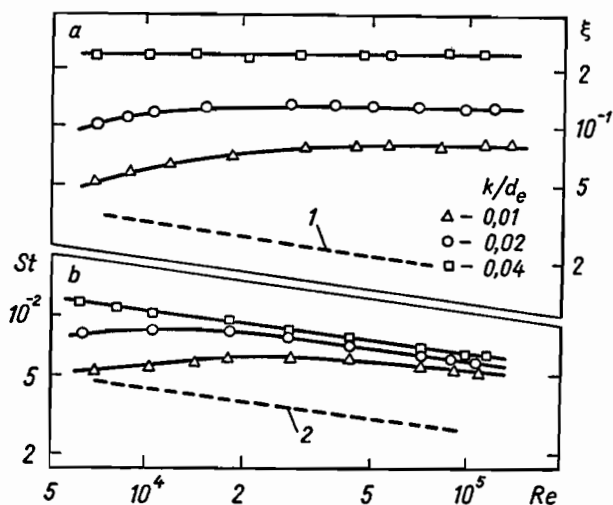
FIGURE 1.5. Hydraulic-drag (a) and heat-transfer (b) coefficients for a rough pipe at different k/d_e according to Webb *et al.*[17]; Pr = 0.71, $s/k = 10$; curves 1 and 2 are for a smooth pipe.

duct with rough inside pipe at $s/k = 10 = $ const and k/d_e varying from 0.001 to 0.02, the degree of enhancement of heat transfer is uniquely defined by the parameter $k^+ = (k/d_e)\mathrm{Re}\sqrt{\xi/8}$. Since St as a function of k^+ peaks at $k^+ \approx 35$, the optimum relative height can be obtained from the expression

$$k_{\mathrm{opt}}^+ = (k/d_e)_{\mathrm{opt}}\,\mathrm{Re}\sqrt{\xi/8} \approx 35, \qquad (1.11)$$

whence

$$(k/d_e)_{\mathrm{opt}} \approx \frac{35}{\mathrm{Re}\sqrt{\xi/8}}. \qquad (1.12)$$

One of the parameters representing the relative enhancement of heat transfer as compared with the relative increase in hydraulic drag is the efficiency

$$\eta = \frac{\mathrm{St}/\mathrm{St}_0}{\xi/\xi_0}. \qquad (1.13)$$

It was shown by a number of investigators[2,5,17] on the basis of analysis of a large volume of experimental data that the value of η at
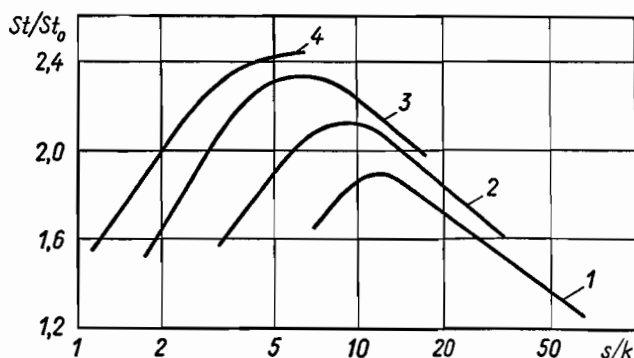
FIGURE 1.6. Effect of s/k on the enhancement of heat transfer in annular ducts at $Re = 10^4$ according to Feurstein and Rampf.[19] k/d_e: (1) 0.0083; (2) 0.017; (3) 0.035; (4) 0.054.

the optimum value of s/k is a function of k^+, k/d_e and Pr. On the other hand, each relative height k/d_e has an optimum value of s/k between 7 and 12 corresponding to it, at which the increase in the heat-transfer coefficient is at maximum as compared with a smooth channel. The variation in this coefficient within these limits of s/k is slight; however, as k/d_e is decreased, the maximum of St/St_0 is observed to shift somewhat toward higher s/k (Fig. 1.6). Although the hydraulic-drag coefficient is also at maximum at this value of s/k, the heat-transfer efficiency is then the highest.

Significant temperature differentials in the boundary layer are observed at high heating loads in gas-cooled power plants even when employing rough surfaces. This applies in the first place to gas-cooled nuclear reactors. However, very little data are available on heat transfer on rough surfaces with highly varying physical properties. Only several investigators attempted to make allowance for this effect, whereas in the majority of cases design work is done on the assumption that the effect of the temperature factor is the same as in smooth duct.

Given the differences in the manner in which roughness affects the thermohydraulic parameters, depending on the previously mentioned flow modes as well as the effect of variability of physical properties and roughness on the wall layer, there are many reasons to believe that the effect of variability of properties on heat transfer in rough channels differs highly from that in smooth ducts.

Lyakhov and Kugay[20] investigated experimentally the effect of variability of properties of air on the heat-transfer and drag coefficients in a pipe whose inner surface was formed of closely wound 0.7-mm-diameter wire ($k \approx 0.35$ mm). They found that at $Re = (1.86 - 4) \cdot 10^4$ and

$\Psi = T_w/T_f = 1.2 - 4.0$ the temperature factor affects both the heat transfer and the hydraulic drag in the same manner:

$$\frac{\text{Nu}}{\text{Nu}_{\Psi=1}} = \frac{\xi}{\xi_{\Psi=1}} = \Psi^{-0.64}. \tag{1.14}$$

Dalle-Donne and Meerwald[21] conducted a semiempirical study of heat transfer and hydraulic drag of annular channels with triangularly threaded inner pipe. They employed 15 such pipes with different roughness heights and pitches at wall temperatures to 1470 K. They showed that variability of physical properties has virtually no effect on the drag in the rough channel, whereas heat transfer in completely rough flow is affected by the roughness in the same manner as in a smooth channel. However, in a later study[22] of heat transfer and drag in annular ducts with rectangular roughness elements, it was recommended that the effect of nonisothermicity on heat transfer be taken into account by taking the temperature factor (Ψ) with a power exponent of -0.5. For smooth annular ducts, the same investigators[23] recommended a much lower value of this exponent which, in addition, varies along the channel

$$n = -(0.25 + 0.0018x/d_e). \tag{1.15}$$

We also wish to note two analytic studies[24,25] concerned with the effect of the variability of physical properties on heat transfer and drag in rough pipes. Kutateladze[24] employed the principles of the semiempirical theory of turbulence, and obtained analytically data on the effect of the temperature factor in flow of gas in a pipe with sand roughness which are somewhat different than in the case of a smooth pipe. Whereas the effect of nonisothermicity in smooth channels is incorporated using the expression

$$\frac{\text{Nu}}{\text{Nu}_{\Psi=1}} = \frac{\xi}{\xi_{\Psi=1}} = \left(\frac{2}{\sqrt{\Psi}+1}\right)^2, \tag{1.16}$$

the equation to be used in channels with sand roughness is

$$\frac{\text{Nu}}{\text{Nu}_{\Psi=1}} = \frac{\xi}{\xi_{\Psi=1}} = \left[\frac{2}{(\sqrt{\Psi}+1)(1+\xi_{\Psi=1})} + \frac{\xi_{\Psi=1}}{1+\xi_{\Psi=1}}\right]^2. \tag{1.17}$$

Wassel and Mills[25] analyzed theoretically turbulent flows in smooth and rough pipes heated by gas flowing through them, and drew an important and interesting conclusion that the effect of Ψ on the drag

and heat transfer in rough pipes is more pronounced than in smooth channels, and depends on the geometry of the asperities and the Reynolds number. However, up to now no systematic experimental work was done to confirm their conclusions. The available data on the subject are contradictory and do not yield sufficiently reliable information on the effect of variability of physical properties of gaseous working fluids on heat transfer and friction in rough channels.

No studies whatsoever were performed on the changes in the structure of highly nonisothermal boundary layers on a rough surface, which makes it impossible to derive valid models of turbulent transport under these conditions. A small-diameter cylinder in longitudinal flow is most suitable in this respect for investigating the structure of boundary layers. For such a cylinder it is easy to produce high heat flux densities under laboratory conditions and to construct test sections. It is convenient to measure the velocity and temperature distributions in the forming boundary layer. In our Institute we accumulated a great deal of experience in investigating the heat transfer and friction of a smooth cylinder in axial flow placed in a large-diameter circular pipe. It was decided that the accumulated data be employed for detailed investigation of heat transfer, friction, velocity, and temperature distributions in the highly nonisothermal boundary layer that forms on smooth and rough cylinders in axial flow, and that a single procedure be employed for determining the effect of different factors on heat transfer. Such studies allow more comprehensive investigation of the inlet regions of annular ducts, which is particularly important for improving the designs of various heat exchangers with short flow passages.

Vilemas et al.[9] presented the results of a measurement of velocity and temperature in the boundary layer that forms on the initial zone of a smooth annular duct, and of the effect of the variability of physical properties and surface curvature on heat transfer and friction. However, certain important factors either were not considered at all or were estimated only approximately. For example, the velocities and temperatures in the boundary layer, as well as the friction factors, were determined only on a 15.5-mm-diameter cylinder, for which the ratio of the thickness of the boundary layer and the radius of the cylinder is somewhat higher than unity, for which reason the effect of surface curvature was slight and was estimated only approximately. In addition, in determining the effect of curvature on heat transfer the thickness of the boundary layer was calculated from an expression for a flat plate, which also introduced a certain error. The velocity and temperature distributions measured over the depth of the boundary layer were worked up on the basis of the coordinate $y^+ = yu_*/\nu$ of the plane flow, which makes no allowance for the effect of curvature on the structure of the axisymmetric boundary layer. For this reason the expressions for

the laws of the wall and for the excess velocity were not general. The authors only touched upon the questions of effect of free-stream turbulence on the heat transfer and friction, and did not perform systematic studies of these factors. The free-stream turbulence in their studies was determined by the strain-gauge method on the basis of the measured longitudinal fluctuations of dynamic pressure, and its effect on friction was incorporated only approximately and was not included in expressions for the heat-transfer coefficient. It was assumed in the formulas recommended by them for calculating the friction factor

$$c_f = 0.329\big(\lg \mathrm{Re}_{x-x_0}\big)^{-2.45}\Psi^{-0.3}(1 + \delta_0/r_0)^{0.14} \qquad (1.18)$$

and the heat-transfer coefficient

$$\mathrm{St} = 0.183\,(\lg \mathrm{Re}_x)^{-2.45}\Psi^{-0.25}(1 + \delta_0/r_0)^{0.14}, \qquad (1.19)$$

that the effect of the temperature factor on these quantities is constant over the length of the cylinder and independent of the latter's radius. Subsequent studies[26,27] demonstrated that the effect of Ψ is a function of both these dimensions.

In this book, we present results of new studies of the structure of the boundary layer on cylinders in axial flow at high heat fluxes. We investigate the details of formation of an axisymmetric boundary layer on small-diameter cylinders at high-temperature differentials. We also determine such important characteristics as the integral thicknesses of the boundary layer, the shape factor, the turbulent Prandtl number, and the mixing length. On the basis of these results we suggest new predictive equations for the heat-transfer coefficients and friction factors of both smooth and rough cylinders in axial flow. These equations incorporate the effect of the cylinder geometry, free-stream turbulence, and the temperature factor.

Such a comprehensive investigation of the structural features of the mechanism of transport in a highly nonisothermal boundary layer on a cylinder in axial flow make it possible to understand better the principal factors governing the rate of heat transfer in annular ducts, and to more optimally investigate the enhancement of heat transfer in gas-cooled channels at high heating loads. In connection with this, we investigate the local heat transfer and hydraulic drag in annular ducts with a rough inside pipe at the most optimal geometric parameters and operating conditions including transition and completely rough flow. We determine the effect of variability of physical properties of the gas on the local heat transfer and friction. Data on these parameters in partially rough annular ducts were converted into data for bundles of rough tubes in longitudinal flow.

FIGURE 1.7.   General view of two-, three-, and four-lobe helically shaped tubes.

Vilemas *et al.*[9] also published data on heat transfer and hydraulic drag in annular channels with a helically shaped inside pipe with different helix pitches and different cross-sectional shapes (Fig. 1.7). In this book, we present results of comprehensive studies of heat transfer, hydraulic drag, and turbulent structure of longitudinal flow over bundles of such tubes. A great deal of attention is given to investigating the local characteristics over the perimeter and length of the tubes, clarifying the factors responsible for the observed specifics of flow over these surfaces, and making practical recommendations for their utilization. A physically validated model of flow in bundles of twisted tubes is developed.

All these comprehensive analytic and experimental studies were performed over a wide range of operating conditions and geometric parameters, and served for obtaining generalized predictive equations suitable for practical use.

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