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Heat transfer and friction of rough ducts carrying gas flow with variable physical properties

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Abstract—The paper presents the results of an experimental investigation into the local heat transfer and friction of gas (air) flow in annuli with the inner rough tube for the following ranges of parameters: $Re = 5 \times 10^3$–5×10^5, temperature factor $T_w/T_f = 1$–2.8, relative height $k/d_e = 0.0028$–0.021 and nearly optimum relative pitch $s/k \sim 10$ of rectangular and rounded trapezoidal roughness elements.

The correlations have been developed to calculate the heat transfer and friction of rough annuli with allowance for the effect of variable physical properties of a coolant and also the relationships for the hydrodynamic and thermal functions of roughness to determine the thermohydraulic characteristics of rough tube clusters.

It has been established that, in contrast to smooth tubes, the effect of variable physical properties of the gas on heat transfer in rough tubes depends on k/d_e and Re and diminishes with the increase of these parameters.

1. INTRODUCTION

ONE OF THE MOST efficient methods of turbulent convective heat transfer enhancement is artificial roughening of the heat-transfer surface.

The investigation of the trends in the effect of roughness on heat transfer and friction and the search for the most efficient configurations and optimum dimensions of roughness elements were the concern of a great number of experimental, and a number of theoretical, studies but almost all of these were carried out at constant physical properties of the coolant. Moreover, basically the zone of fully effective roughness was investigated, while the region of partial roughness and the process of transition from partial to fully rough conditions have been studied but little. The effect of roughness on the thermohydraulic characteristics in the conditions of variable physical properties of the coolant has hardly been investigated at all.

Increased power densities of modern heat-exchanging devices require higher heat-transfer rates and temperature differences, thus bringing the need for a rigorous allowance for the effect of variable physical properties of a coolant in specific operating conditions. This is especially true for high power-consuming systems in which gas coolants are employed.

Most conveniently the augmentation of heat transfer can be investigated in an annular passage with the rough core tube, since it is more easily roughened, and the existing techniques of 'separation' into the internal and external zones of the annular channel allow the results obtained to be extended to other, more complicated geometries.

Two trends can be distinguished in interpreting the results of investigations. The first is a direct generalization of the results in the form of the dependence of Nu or St and ξ on Re [1]. The other is the determination of the so-called hydrodynamic, $R(k^+)$, and thermal, $G(k^+, Pr)$, functions of roughness, which are interrelated and also related to the integral quantities ξ, St [2] and make it possible to assess the thermohydraulic characteristics in complex geometry channels.

The objective of this study was to experimentally investigate, in a wide range of operating and geometric parameters ($Re = 5 \times 10^3$–5×10^5, $\Psi = 1$–2.8, $k/d_e = 0.0028$–0.021, $s/k \sim 10$), the effect of artificial roughness on local heat transfer and friction for a turbulent air flow in annuli with the rough inner tube in the case of constant and variable physical properties of the gas; to transform the results obtained to fit fully rough channels (rough tubes, clusters of longitudinally streamlined rough rods).

2. METHOD OF INVESTIGATION

The present experimental investigations were conducted on two test sections, connected to an open-type wind tunnel, whose detailed description is given in reference [3]. Here, only some of their special features will be mentioned.

The local heat transfer was investigated at 10 cross-sections of the annular passage having the smooth outer tube with an o.d. of 29.75 mm, wall thickness of 0.8 mm and length of 2145 mm and interchangeable rough inner tubes 1600 mm long. The entry unheated section of hydrodynamic stabilization 545 mm long was smooth. All the calorimetric tubes were fabricated from 1Cr18Ni10Ti stainless steel and were heated by direct electric current.

The friction was investigated in an annular passage

NOMENCLATURE

b	width of roughness elements
c_p	specific heat at constant pressure
d_0	zero shear stress surface diameter
d_1, d_2	diameters of wetted surface of annulus tubes
d_e	equivalent diameter of annulus, d_2–d_1
$G(k^+), G_1(k_w^+)$	thermal functions of roughness
k	height of roughness elements
k^+	dimensionless height of roughness elements, ku_*/v
k_w^+	dimensionless height of roughness elements, $ku_*/v_w = k^+ v/v_w$
m	Re number exponent in the heat-transfer law
n	exponent of Ψ
q	heat transfer rate
$R(k^+), R_1(k^+)$	hydrodynamic functions of roughness
r_3, r_4	radii of rounding of roughness elements
s	roughness pitch
T	temperature
t^+	dimensionless temperature, $(T_w - T)\rho c_p u_*/q_w$
u	longitudinal velocity
u_f	mean mass velocity
u_*	dynamic velocity, $\sqrt{(\tau_w/\rho)}$
u^+	dimensionless velocity, u/u_*
x	distance from the start of heating
y	distance along the radius reckoned from the surface of tubes
y^+	dimensionless ordinate, yu_*/v
z	constant factor
α	heat-transfer coefficient
δ_1	rough wall thickness
λ	thermal conductivity
μ	dynamic viscosity
v	kinematic viscosity
ξ	friction factor
ρ	density
τ	shear stress
Ψ	temperature factor, T_w/T_f
Nu	Nusselt number, $\alpha d_e/\lambda$
Pr	Prandtl number, $\mu c_p/\lambda$
Re	Reynolds number, $u_f d_e/v$
St	Stanton number, $Nu/(RePr)$.

Subscripts

1	inner tube, internal zone of annulus
2	outer tube, external zone of annulus
f	in flow
sm	smooth
tr	transition from partially to fully rough flow region
w	at wall
Ψ	= 1 at constant physical properties.

with a smooth outer tube 30 mm in o.d., having the wall thickness of 1 mm, and with interchangeable rough inner tubes of the first test section. For the static pressure tapping, three holes 0.4 mm in diameter were drilled at each of the ten cross-sections of the outer tube and at the entrance to the annular channel.

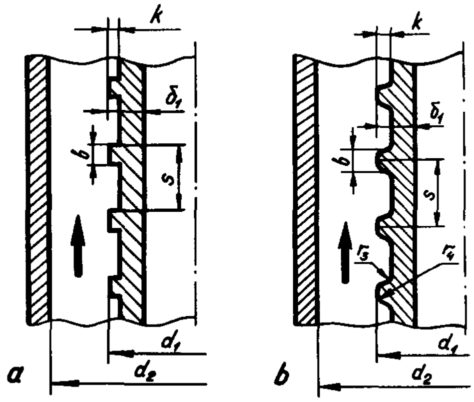
The measurements of all electric signals were made with the aid of the automatic measuring data-acquisition system 'SOLARTRON'. The static pressure drops were measured by an inductive differential pressure transducer PD-1.

Two types of rough surfaces were investigated. In four channels with the tube diameters ratio $d_1/d_2 \sim 0.42$, the roughness had rectangular profile (roughness of type I, Fig. 1a). The roughness in the form of circular grooves was produced on a smooth tube mechanically. In two channels with $d_1/d_2 \sim 0.35$, the roughness had rounded trapezoidal configuration (roughness of type II, Fig. 1b). The roughness in the form of a continuous helical line was made on a smooth tube by an electrochemical method. The rough surfaces of each type differed by the height of two-dimensional roughness elements at the nearly optimum (as regards heat-transfer enhancement) ratio $s/k \sim 10$ (see Table in Fig. 1). For a more exact comparison of the results obtained, the investigations were carried out in one smooth channel with $d_1/d_2 \sim 0.43$ which is geometrically similar to the channels with the type I roughness.

The experimental and data handling procedures were similar to those used earlier for smooth annular channels [3]. The treatment of experimental data was reduced to the determination of the numbers Nu, Re, Pr and also of the friction factor ξ, to derivation of correlations of the type $Nu = f_1(Re)$, $Nu/Nu_{\Psi=1} = f_2(\Psi)$, $\xi = f_3(Re)$ and to generalization of the results of investigation. The reference temperature was taken to be the local mean-mass flow temperature T_f, the reference velocity to be the local mean-mass velocity u_f, and the reference dimension to be the equivalent diameter of the annulus $d_e = d_2 - d_1$.

When treating the data, the allowance was made for: variation of the geometric dimensions due to thermal expansion, radiative heat-transfer, heat losses, longitudinal heat flows in the tube walls, temperature drops across the walls. The investigations were carried out at the boundary conditions close to where q_w is a constant.

The effect of variability of the gas physical properties in each channel for each Re and relative length x/d_e was taken into account through the temperature factor Ψ in



Channel No.	Type of roughness	d_2, mm	d_1, mm	k, mm	s, mm	b, mm	δ_1, mm	s/k	k/d_e	(s-b)/k	k/b
1	rectangular	28,15	11,91	0,046	0,6	0,15	0,84	13,0	0,0028	9,8	0,3
2			11,92	0,12	1,0	0,2	0,88	8,3	0,0074	6,7	0,6
3			11,83	0,21	2,0	0,4	0,85	9,5	0,013	7,6	0,5
4			11,52	0,34	4,0	0,41	0,72	11,8	0,021	10,6	0,8
5	rounded trapezoidal		9,97	0,1	1,0	0,2	0,95	10,0	0,0055	8,0	0,5
6			9,98	0,05	0,5	0,15	0,42	10,0	0,0028	7,0	0,3
7	smooth		12,12	—	—	—	0,33	—	—	—	—

FIG. 1. Configuration and basic geometric characteristics of surfaces: *a*, rectangular; *b*, rounded trapezoidal.

the form of the relation

$$Nu/Nu_{\Psi=1} = \Psi^n, \tag{1}$$

where $Nu_{\Psi=1}$ was calculated from

$$Nu_{\Psi=1} = c_{\Psi=1} Re^m Pr^{0.6}, \tag{2}$$

while the constant $c_{\Psi=1}$ for each heat transfer surface, Re and x/d_e were determined experimentally by the extrapolation of the test data correlated by

$$c = \frac{Nu}{Re^m Pr^{0.6}} = f(\Psi) \tag{3}$$

up to $\Psi = 1$. The exponents m of Re were determined from the experimental data obtained at small Ψ ($\Psi_{max} \sim 1.2$).

When determining the friction factor ξ, both the wall friction and the momentum change-induced flow friction were taken into account.

A detailed analysis of the experimental errors has shown that, depending on the operating conditions, the r.m.s errors in the determination of Re amounts to 1.8–2.5%, of Nu to 1.6–5.5% and of ξ to about 5%.

3. EXPERIMENTAL RESULTS

The investigations have shown that heat transfer in rough annular channels increases differently over the studied range of Re numbers. The experimental data in the form of the dependence $Nu = f(Re)$ can be approximated with sufficient accuracy in logarithmic coordinates by broken lines (Fig. 2). Depending on k/d_e and Re, three characteristic zones can be clearly distinguished: (1) the zone of 'hydraulically smooth' flow region in which heat transfer practically coincides with that in a smooth channel, when its certain inappreciable increase can be attributed to the extension of the heat-transfer surface; (2) the zone of partial rough flow region in which the heat transfer rate depends markedly on the relative height k/d_e, i.e. the heat transfer increases with k/d_e and the law of Nu variation with Re differs significantly from this law for smooth channels; (3) the zone of fully rough flow region in which heat transfer ceases to increase with k/d_e and its maximum augmentation, attained on transition from the partially to fully rough region, practically does not change with a further increase of Re.

As is seen from Fig. 2, in the course of transition from the partially to fully rough region a clear kinking in the $Nu = c \cdot Re^m$ curves is observed for all of the channels except for 1 and 6. The Re_{tr} number, at which a change in the slope of the approximating straight lines is observed, decreases with an increase in the relative height k/d_e. For channels 1 and 6 with the minimum k/d_e, equal to 0.0028, the transition takes place at $Re > 5 \times 10^5$.

In the case of fully rough flow conditions ($Re > Re_{tr}$), heat transfer increases twice in the stabilized heat transfer zone as compared with heat transfer in smooth annular channels, and its rate is practically independent of k/d_e. A certain slight scattering of the

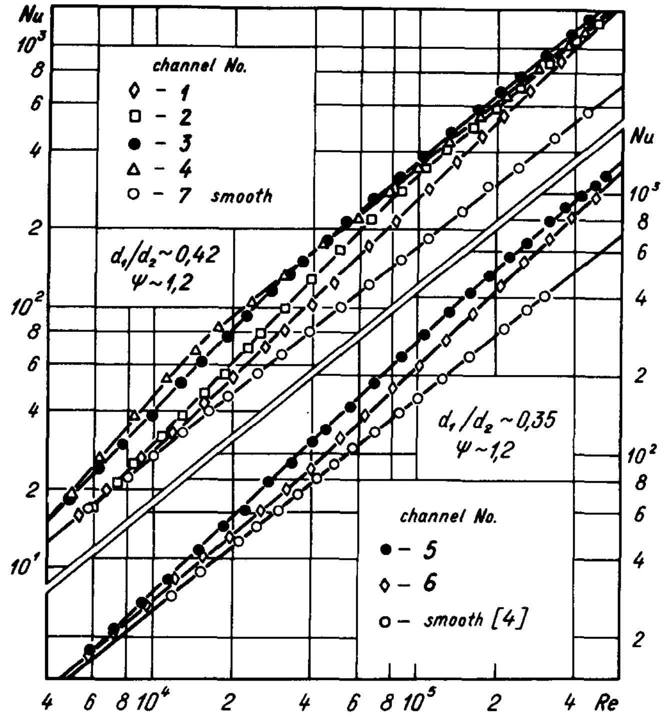


FIG. 2. The effect of roughness on heat transfer in the stabilized heat transfer zone at $\Psi \sim 1.2$.

experimental points in this zone is presumably associated with differently extended heat transfer surfaces and with the absence of complete similarity between the geometric characteristics of the roughness elements. The exponent m of Re in this case is close to its value for smooth channels (0.8) and in the case of partially rough conditions it is higher (~ 1).

The configuration of the roughness elements does not practically influence the heat-transfer rate. Thus, in the present experiments the maximum increase in the heat-transfer rate for the both types of roughness in the zone of fully rough conditions is almost identical (Fig. 2).

Despite an additional boundary-layer turbulization, the effect of variability of the gas physical properties on heat transfer in rough channels is significant. For

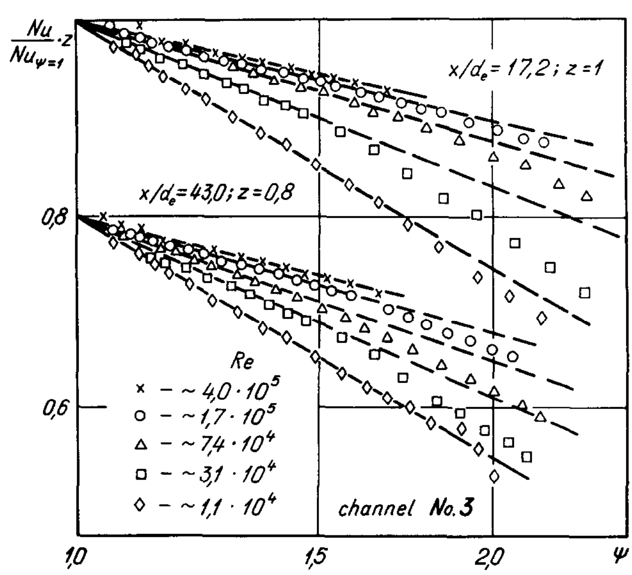


FIG. 3. The effect of temperature factor on heat transfer at different values of Re.

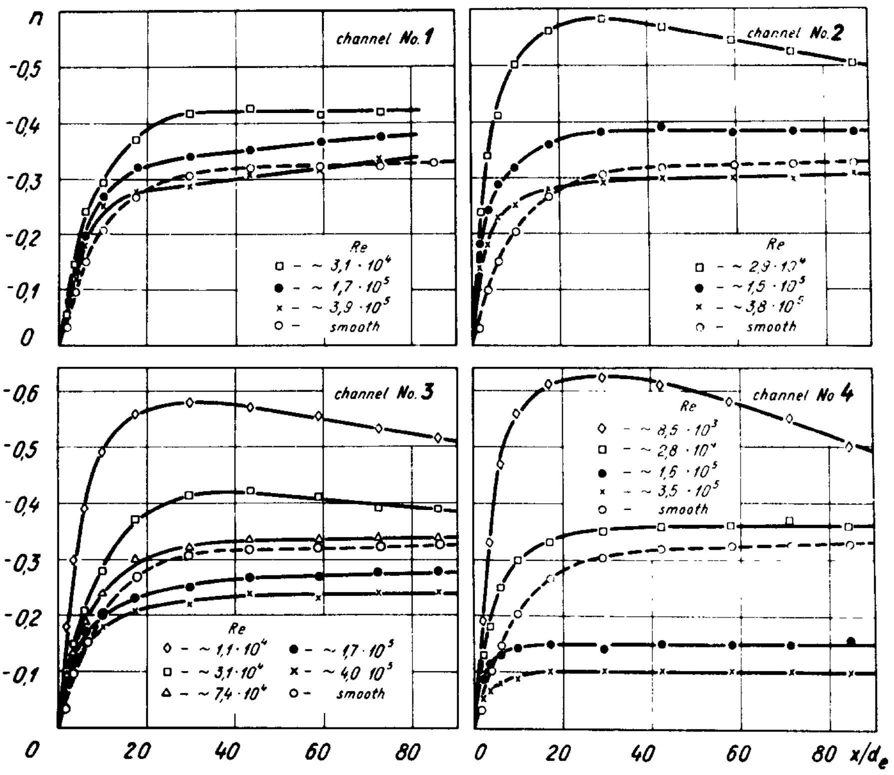


FIG. 4. Variation of the exponent n at Ψ along the channel length.

example, in rough channel 3, with Ψ changing from 1 to 1.9, the heat transfer in the partially rough zone decreases by about 50% and in the fully rough zone, by about 25%. Moreover, a little change is observed in the *Re* exponent m, and the beginning of the fully rough flow region shifts to the side of great *Re*'s with an increasing Ψ.

In contrast to smooth channels, the influence of Ψ on the local heat transfer in rough annular channels depends heavily on *Re* (Fig. 3). With an increasing *Re*, the influence of Ψ decreases. In spite of the fact that $Nu/Nu_{\Psi=1} = f(\Psi)$ is nonlinear in logarithmic coordinates, it is possible to neglect with sufficient accuracy the nonlinearity in the region $\Psi \leqslant 1.8$, to approximate the experimental data by expression (1) (dashed lines in Fig. 3) and to determine the exponent n at Ψ.

The influence of the variability of the gas physical properties on heat transfer along the length of a rough channel, just as in smooth ducts, increases for each *Re* up to a certain stabilized value. In Fig. 4, a change in the exponent n at Ψ along the length of the channels is shown. The exponent n increases sharply along the inlet zone of the annular channel, i.e. with an increase in the thermal boundary layer thickness, and it practically remains the same in the stabilized heat transfer zone.

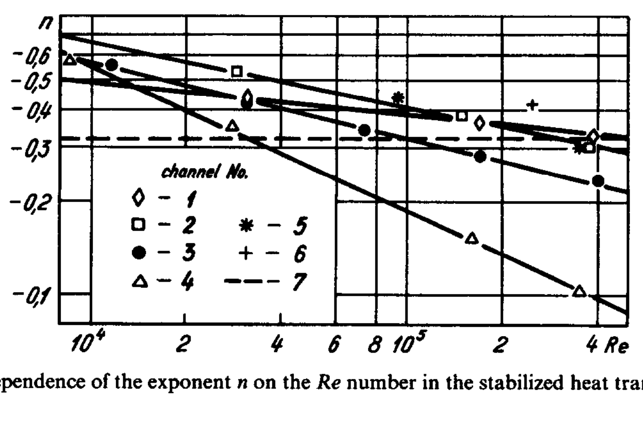


FIG. 5. Dependence of the exponent n on the *Re* number in the stabilized heat transfer zone.

Only at small Re, the value of n does not become stabilized, which apparently is due to flow laminarization at high heat fluxes.

The mean values of n at Ψ in the stabilized heat-transfer zone ($x/d_e > 30$) depending on Re can be approximated for each rough channel by the expressions of the type $n = n_0 \cdot Re^{-p}$ (Fig. 5). As is seen from this figure, the influence of Re on n is the more appreciable, the greater is k/d_e. Moreover, at small Re's (in the zones of partially rough flow conditions) the influence of the variability of the gas physical properties on heat transfer in rough channels is stronger than in smooth channels, while at large Re's (in the zone of fully rough flow conditions) it is weaker. A decrease in n at Ψ with an increasing Re is explained by higher flow turbulization in the wall layer, which is also indicated by high Re exponents m in the heat transfer law as compared with the exponents for smooth channels (Fig. 2). The influence of Re, k/d_e and x/d_e on the exponent n is correlated within $\pm 10\%$ by

$$n = -(0.29 + 0.03e^{51\sqrt{(k/d_e)}})Re^{-24k/d_e}(1 - e^{-0.16x/d_e}). \tag{4}$$

The effect of the geometric parameter d_1/d_2 can be taken into account in the first approximation in the same way as for smooth annular channels [4].

With allowance for the temperature factor and for the effect of Pr, just as for smooth channels, the heat transfer in annular channels with rectangular roughness in the zone of fully rough flow conditions can be correlated by

$$Nu = 0.029Re^{0.84}Pr^{0.6}\Psi^n, \tag{5}$$

and in the zone of partially rough flow conditions at $k/d_e \geqslant 0.0025$ by

$$Nu = (0.0053 - 0.14\, k/d_e)Re^{0.95 + 7k/d_e}Pr^{0.6}\Psi^n, \tag{6}$$

where n is determined from equation (4). It should be noted that in the zone of fully rough flow the present data, reduced to the condition of constant physical properties ($\Psi = 1$), are rather well represented by Gomelauri's correlation [5], transformed for the case of gas coolants, and agree satisfactorily with the results of ref. [6] obtained for geometrically similar rough annular passages in a narrower range of Re numbers.

On the basis of the experimental data, reduced to the condition $\Psi = 1$, a correlation has been obtained to determine the place of transition from partially to fully rough flow conditions (Re_{tr} numbers). At $k/d_e \geqslant 0.005$, the Re_{tr} numbers are well correlated by

$$Re_{tr} = \exp(12.2 - 126\, k/d_e). \tag{7}$$

The present data on the transition agree with the results of references [7, 8] in which heat transfer and friction were investigated in rough annular passages and rough tubes. The agreement is rather good though different roughnesses were investigated: wire [7] and rectangular ribs [8]. It seems that the configuration of roughness elements insignificantly influences both the heat transfer and Re_{tr}.

The friction in rough annular channels increases with k/d_e and with the impairment of streamlining of roughness elements (Fig. 6). Thus, at almost the same height of roughness elements (channels 1 and 6, 2 and 5), the friction in channels with rounded trapezoidal roughness is smaller than in channels with rectangular roughness. The absence of self-similarity with respect to Re in the zone of fully rough flow conditions is associated with the fact that the wall of the outer tube of the annular channel was smooth and that in experiments the overall friction caused by the both walls was determined.

The experimental data agree qualitatively with the results of refs. [6, 9] obtained in rather narrow ranges of Re. Certain quantitative discrepancies are attributed to the absence of complete similarity between the geometric characteristics of channels and of roughness elements. The data for a smooth annular channel are satisfactorily represented by Filonenko's well-known correlation [10] for smooth tubes.

The data on the friction of all the channels studied in the zones of partially and fully rough flow conditions are generalized within $\pm 4\%$ by the following relations:

for channels with rectangular roughness

$$\xi = (0.053 + 1.85\, k/d_e)Re^{-0.07}, \tag{8}$$

and for channels with rounded trapezoidal roughness

$$\xi = (0.063 + 5k/d_e)Re^{-0.11}. \tag{9}$$

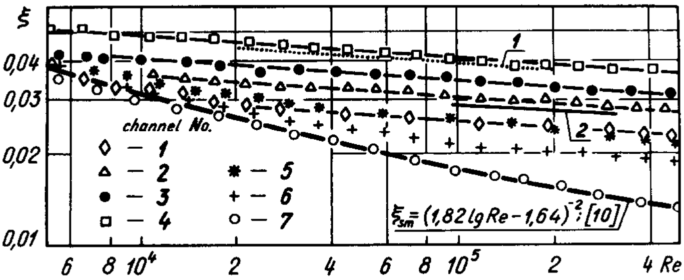


FIG. 6. The friction of rough annular channels. 1, data of ref. [6]: $k = 0.5$ mm, $s/k = 10$, $k/d_e = 0.0083$, $d_1/d_2 \sim 0.53$; 2, data of ref. [9]: $k = 0.1$ mm, $s/k = 10$, $k/d_e = 0.011$, $d_1/d_2 \sim 0.52$.

In rough annular channels one also observed the influence of the variability of gas physical properties on the friction, although this influence has not been discovered in a smooth channel (Fig. 7). The effect of Ψ decreases with an increasing *Re*, and vanishes completely at a certain value of the latter. Since in nominal operating conditions of gas coolant flow through heat-releasing fuel elements of nuclear reactors *Re* is usually above 5×10^4, when the effect of Ψ on ξ is already insignificant, then in practical calculations this influence can be neglected.

A simultaneous analysis of the data on heat transfer and friction in rough channels shows that, due to the absence of influence of the relative height k/d_e on the heat-transfer rate in the zone of fully rough flow, it is possible to use the minimum height of roughness elements in operating conditions provided that a heat-exchanging apparatus operates in the region $Re > Re_{tr}$, since the smaller is the height of asperities, the smaller is the friction. This also attests to the maximum values of the efficiency index St^3/ξ obtained for the region of transition from partially to fully rough flow conditions.

4. DATA TRANSFORMATION TO FIT FULLY ROUGH CHANNELS

At present the most commonly used method of 'separation' into the internal rough and external smooth zones in annular channels is that suggested by Dalle Donne and Meyer [11]. It is based on determination of the position of zero shear-stress surface, which separates the zones of an annular channel, from the crossing-over of the adopted logarithmic laws for the Nikuradse velocity distribution in the internal rough and external smooth zones with allowance for their interaction

$$u_1^+ = 2.5 \ln \frac{y_1}{k} + R(k^+), \tag{10}$$

$$u_2^+ = A_s \ln y_2^+ + 5.5. \tag{11}$$

Similar to the logarithmic velocity distribution law in the internal rough zone, the logarithmic law of temperature distribution is assumed for temperatures which is operative throughout the entire annular gap between the rough and smooth walls of the channel

$$t^+ = 2.5 \ln \frac{y_1}{k} + G(k^+). \tag{12}$$

The results of experimental investigation of the heat transfer and friction of rough annular channels in the stabilized heat transfer zone under the conditions close to those with constant physical properties ($\Psi_{max} \sim 1.2$) have been used, after iterative solution of a certain system of equations [11], to obtain the following thermohydraulic characteristics: the position of the zero shear-stress surface β ($\beta = d_0/d_1$), Re_1, Re_2, ξ_1, ξ_2, St_1, the hydrodynamic, $R(k^+)$, and thermal, $G(k^+)$, functions of roughness.

As is seen from Fig. 8, after the data have been transformed, the friction in the internal rough zone of the channel is much higher, and in the external smooth zone is much smaller, than the friction in the whole of the annular channel. The calculated values of *Re* also differ: at the same cross-section in the internal zone Re_1 is higher, and in the external zone Re_2 is smaller, than Re_0 for the whole of the annulus. As was expected, in the case of fully rough flow conditions in the internal rough zone, the self-similarity of friction with respect to *Re* has been obtained which was not observed for the whole of partially rough annulus. The present data on friction in the internal rough zone agree well with the results obtained in rough tubes [8] and also with the data on the 'separation' into the zones in geometrically similar rough annuli [12].

After the transformation of the data, heat transfer in the internal rough zone differs from that in the annular channel as a whole much less than does the friction, and this is mainly attributed to variation of *Re* (Fig. 9). In some channels, the increase of heat transfer in this zone, as compared with heat-transfer for the whole of rough annulus, amounts only to 10–20%.

For the comparison of thermohydraulic characteristics and their estimation in rough tubes and in longitudinally streamlined clusters of rough rods, the hydrodynamic, $R(k^+)$, and thermal, $G(k^+)$, functions of roughness are incorporated into the logarithmic laws of velocity and temperature distributions, equations (10) and (12), respectively, modified with allowance for the boundary-layer thickness, effect of *Pr* and at the

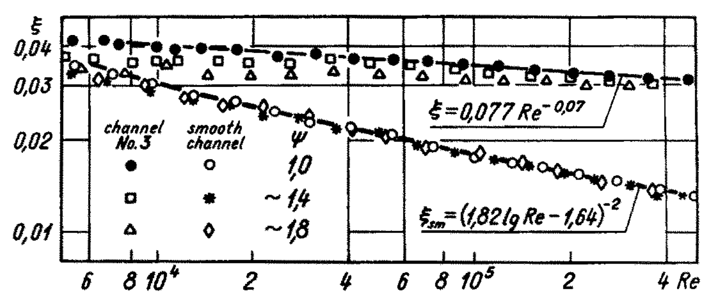


FIG. 7. The influence of temperature factor on friction.

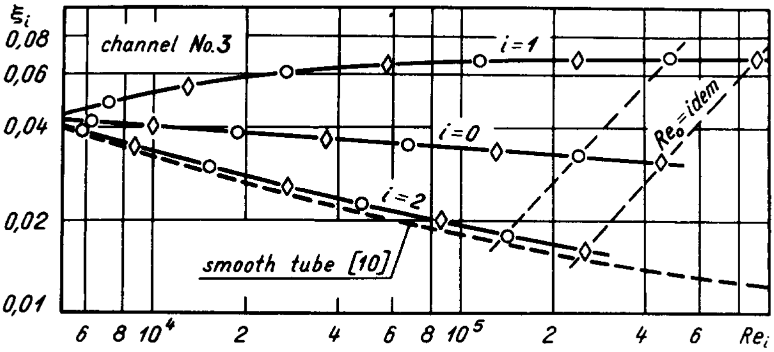


FIG. 8. The friction in separate zones of a rough channel: $i = 0$, the whole of the channel; $i = 1$, internal zone; $i = 2$, external zone.

temperature factor using the relations [11]:

$$R_1(k^+) = R(k^+) - 0.4 \ln\left(\frac{200k}{d_0 - d_1}\right), \quad (13)$$

$$G_1(k_w^+) = G(k_w^+) \Big/ \left[\left(\frac{200k}{d_2 - d_1}\right)^{0.053} Pr^{0.44}\Psi^{0.5}\right]. \quad (14)$$

The hydrodynamic roughness function $R_1(k^+)$ depends not only on the dimensionless height of roughness elements k^+—the parameter determining the boundaries of different zones of the effect of roughness, but also on the configuration and relative pitch of roughness elements (Fig. 10). In a hydraulically smooth zone ($k^+ < \sim 5$), the present data are close to the relation $R_1(k^+) = 2.5 \ln k^+ + 5.5$, and then, with an increase of k^+, $R_1(k^+)$ decreases approaching the stabilized value in the zone of fully rough flow. The more rounded the roughness elements are, the higher is the function $R_1(k^+)$. The familiar $R_1(k^+)$ distribution as a function of k^+ for the Nikuradse sand-grain roughness [14] agrees qualitatively with the present data.

For rectangular roughness in the zone of fully rough flow in the ranges $(s-b)/k = 6.3$–160 and $k/b = 0.086$–5, the following correlation has been suggested [11] which determines the hydrodynamic function within $\pm 15\%$

$$R_{1\infty} = 1.04\left(\frac{s-b}{k}\right)^{0.46} - \left[2 + \frac{7}{(s-b)/k}\right] \lg \frac{k}{b}. \quad (15)$$

The present data are also satisfactorily described by this relation. Extending the generalization of $R_1(k^+)$ also to the zone of partially rough flow at $k^+ > \sim 5$, the following relation has been obtained

$$R_1(k^+) = R_{1\infty} + 4.3 \exp(-0.8 \lg^2 k^+), \quad (16)$$

where $R_{1\infty}$ is determined from formula (15).

For rounded trapezoidal roughness at $k^+ > \sim 5$, the following relation can be recommended

$$R_1(k^+) = R_{1\infty} + 12 \exp(-2.4 \lg k^+), \quad (17)$$

where $R_{1\infty} = 5.8$ for the roughness in channel 5 and $R_{1\infty} = 7.3$ for the roughness in channel 6.

The thermal function of roughness $G_1(k_w^+)$ in the zone of partially rough flow does not actually depend on the dimensionless height k_w^+, while in the fully rough zone, this dependence is obvious (Fig. 11). Practically, the configuration of the roughness elements also does not influence the $G_1(k_w^+)$ function distribution. The present

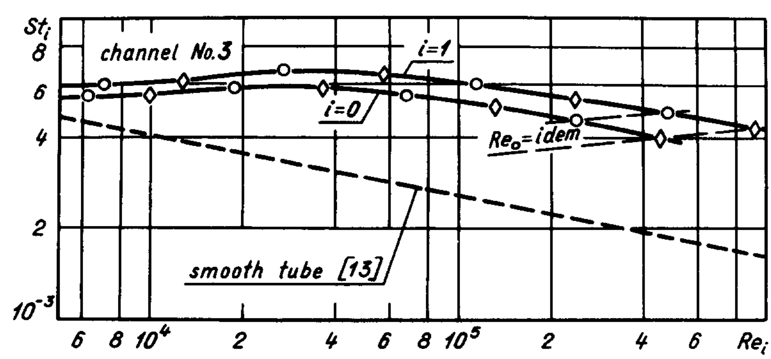


FIG. 9. Heat transfer in separate zones of a rough channel: $i = 0$, the whole of the channel; $i = 1$, internal zone.

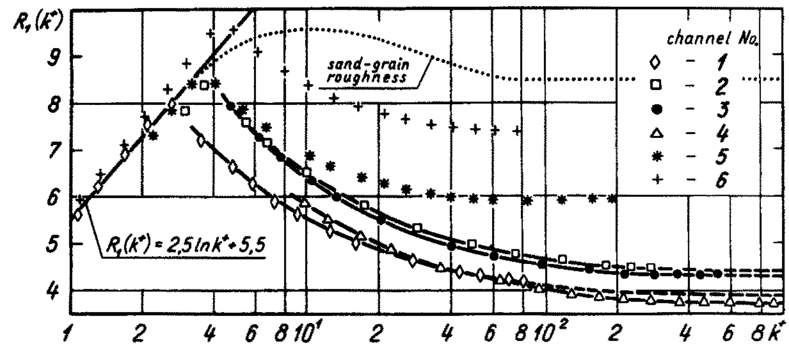


FIG. 10. The hydrodynamic function of roughness $R_1(k^+)$ vs the dimensionless height k^+.

data for the zone of partially and fully rough flow ($k_w^+ > \sim 5$) for all of the roughness configurations are correlated within $\pm 10\%$ as

$$G_1(k_w^+) = 7.8 + 0.15 \exp(1.6 \lg k_w^+). \tag{18}$$

Once the hydrodynamic and thermal functions of roughness are known then, after simple transformations [2, 11], it is possible to determine the friction and heat transfer in rough longitudinally streamlined tube bundles.

5. CONCLUSIONS

(1) It has been established experimentally that the influence of the variability of gas physical properties on heat transfer in rough channels depends on Re and k/d_e. With the increase of these parameters this influence decreases and does not actually depend on the configuration of roughness elements. The effect of the temperature factor on heat transfer along the channel length increases for each Re up to a certain stabilized value. The effect of Ψ on friction has been detected only at smaller Re.

(2) With an increasing k/d_e, the region of transition from partially to fully rough flow shifts to the side of smaller Re. The configuration of roughness elements does not practically influence this transition; equation (7) has been derived to determine the transition Re number. In the zone of partially rough flow conditions the heat-transfer rate depends markedly on k/d_e, while in the zone of fully rough conditions the heat transfer is practically independent of both k/d_e and configuration of roughness elements and is twice that in smooth channels.

(3) The optimum operating parameters for the augmentation of heat transfer are the smaller possible height of surface roughness elements, provided that a heat-exchanging device operates in the region of the transition from partially to fully rough flow conditions.

(4) The correlations have been derived to calculate the heat transfer [equations (5) and (6)] and friction [equations (8) and (9)] in rough annular channels with allowance for the effect of the variability of gas physical properties and also the relations for the hydrodynamic [equations (16) and (17)] and thermal [equation (18)] functions of roughness to estimate the thermohydraulic characteristics in longitudinally streamlined bundles of rough tubes.

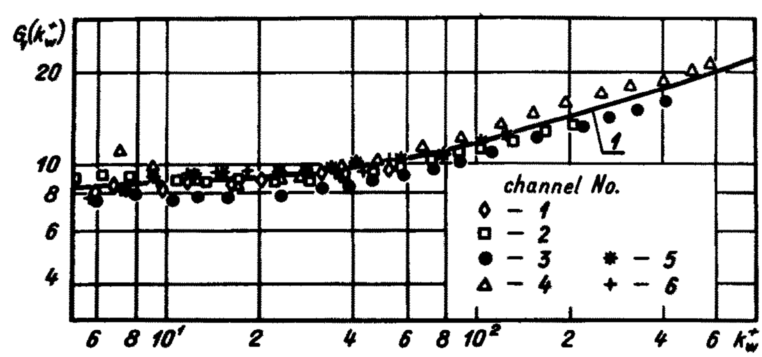


FIG. 11. The thermal function of roughness $G_1(k_w^+)$ vs the dimensionless height k_w^+. 1, correlation (18).

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TRANSFERT THERMIQUE ET FROTTEMENT DANS LES CANAUX RUGUEUX TRANSPORTANT UN GAZ A PROPRIETES VARIABLES

Résumé—On présente les résultats d'une étude expérimentale du transfert thermique local et du frottement pour un gaz s'écoulant dans un espace annulaire avec le tube intérieur rugueux, pour des domaines de paramètres suivants : $Re = 5 \times 10^3$–5×10^5, rapport des températures $T_w/T_f = 1$–2,8, hauteur relative $k/d_e = 0{,}0028$–0,021 et pas proche de l'optimum $s/k \sim 10$ pour des éléments de rugosité rectangulaires et trapézoïdaux arrondis. Les formules sont développées pour calculer le transfert thermique et le frottement en tenant compte des effets de variation des propriétés physiques d'un réfrigérant et aussi les relations pour les fonctions hydrodynamiques et thermiques de rugosité pour déterminer les caractéristiques thermohydrauliques de grappes de tubes rugueux. On établit que, contrairement au cas des tubes lisses, l'effet des propriétés physiques variables du gaz sur le transfert thermique dans les tubes rugueux dépend de k/d_e et de Re et il diminue quand ces paramètres augmentent.

WÄRMEÜBERGANG UND DRUCKABFALL IN RAUHEN ROHREN, DIE VON EINEM GAS MIT VARIABLEN PHYSIKALISCHEN EIGENSCHAFTEN DURCHSTRÖMT SIND

Zusammenfassung—In der Arbeit werden die Ergebnisse einer experimentellen Untersuchung des örtlichen Wärmeübergangs und Druckabfalls einer Gas-(Luft-)Strömung in einem Ringspalt um ein rauhes Rohr für die folgenden Parameterbereiche untersucht: $Re = 5 \times 10^3$–5×10^5, Temperaturfaktor $T_w/T_f = 1$–2,8, relative Spalthöhe $k/d_e = 0{,}0028$–0,021 und nahezu optimale Teilung $s/k \sim 10$ von rechteckigen und gerundeten trapezförmigen Rauhigkeitselementen. Die Beziehungen wurden entwickelt, um Wärmeübergang und Druckabfall von rauhen Ringspalten bei variablen physikalischen Stoffeigenschaften und den Einfluß der Rauhigkeit zu berechnen. Damit soll das thermohydraulische Verhalten von rauhen Rohranordnungen bestimmt werden. Es wurde festgestellt, daß im Gegensatz zu glatten Rohren der Einfluß variabler physikalischer Eigenschaften des Gases auf den Wärmeübergang in rauhen Rohren von k/d_e und Re abhängt, und daß der Einfluß abnimmt, wenn diese Parameter größer werden.

ТЕПЛООБМЕН И ГИДРАВЛИЧЕСКОЕ СОПРОТИВЛЕНИЕ ШЕРОХОВАТЫХ КАНАЛОВ ПРИ ТЕЧЕНИИ ГАЗА С ПЕРЕМЕННЫМИ ФИЗИЧЕСКИМИ СВОЙСТВАМИ

Аннотация—Представлены результаты экспериментального исследования местной теплоотдачи и гидравлического сопротивления кольцевых каналов с внутренней шероховатой трубой при течении газа (воздуха) в интервале чисел $Re = 5 \times 10^3$–5×10^5, температурного фактора $T_w/T_f = 1$–2,8, относительной высоты $k/d_e = 0{,}0028$–0,021 и при близком к оптимальному относительном шаге $s/k \sim 10$ элементов шероховатости прямоугольной и закругленной трапецеидальной формы. Получены обобщающие зависимости для расчета теплоотдачи и гидравлического сопротивления шероховатых кольцевых каналов с учетом влияния переменности физических свойств теплоносителя, а также зависимости для гидродинамической и тепловой функций шероховатости, позволяющие определить теплогидравлические характеристики в сборках шероховатых труб. Установлено, что в отличие от гладких каналов влияние переменности физических свойств газа на теплоотдачу в шероховатых каналах зависит от k/d_e и Re и с увеличением этих параметров уменьшается.