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Intensification of Heat Transfer by the Method of Artificial Roughness at a Liquid Film Flowing Down a Vertical Surface

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Summary

**Dedicated
to the memory of our colleagues and friends
Vazha Jamardzhashvili, Archil Gomelauri
and Jondo Rusishvili**

Used designations:

α – Heat transfer coefficient, W/(m² K);
a – Liquid thermal diffusivity coefficient, m²/c;
c_p – Specific heat capacity of liquid, kj/(kg K);
η – Dimensionless distance from the wall;
η_1 – Dimensionless distance to the outer boundary of the viscous sublayer;
η_2 – Dimensionless distance to the outer boundary of the buffer zone;
F – Heat transfer surface area, m²;
G – Intensity of irrigation of the heat transfer surface, m²/c;

h – Height of two-dimensional roughness elements, mm;
k_s – Height of J. Nikuradze roughness ("sandy") elements, mm;
L – Defining geometric dimension, m;
λ – Liquid thermal conductivity coefficient, W/(m K);
ν – Kinematic coefficient of liquid viscosity, m^2/c;
ξ – Hydraulic resistance coefficient;
q – Specific heat flux, W/m^2;
r_0 – Radius of pipe, m;
ρ – Liquid density, kg/m^3;
s – Step between roughness elements, mm;
s/h – Relative Step between roughness elements;
τ – Shear stress, N/m^2;
u – Velocity of liquid flow, m/c;
u_* – Dynamic velocity, m/c;
$\varphi = \frac{u}{u_*}$ – Dimensionless velocity, m/c;
y – Distance from heating surface, m.

Criteria:

$Ni = k_s u_*/\nu$ – Nikuradze number;
$Nu = \frac{\alpha L}{\lambda}$ – Nusselt number;
$Pr = \frac{\nu}{a}$ – Prandtl number;
$Re = \frac{u\,L}{\nu}$ – Reynolds number
Re_* – The Reynolds number at which the full manifestation of roughness begins.

Indexes: w – viscous; sm – smooth; r – rough; t – turbulent.

The monograph is dedicated to such a relevant problem as the intensification of the heat transfer process in power devices.

In modern installations, such as, for example, steam generators of thermal power plants, cooling systems of condensers and electric generators, metallurgical, aviation, rocket, space equipment, heating systems, etc., an extremely large role is assigned to increasing the intensity of heat transfer, because the efficient operation of the equipment and, most importantly, its compactness, depend on the latter.

The intensity of heat transfer from a solid surface to a liquid (gas) or steam, and vice versa, largely determines the intensity of the heat transfer process between two heat carriers through the solid heating surface mentioned above. Therefore, the study and development of heat transfer intensification methods are of great practical importance. Along with this, since the intensification of heat transfer is inextricably linked to the impact of the fluid flow on the boundary layer and the change in the structure of the flow caused by this, the investigation of the mentioned problem is of great theoretical importance.

The paper notes that the use of any method of intensifying heat transfer in a fluid flow necessarily leads to an increase in the energy spent on the movement of this fluid, so it is important to find a method that provides maximum heat transfer intensification with a minimum increase in hydraulic resistance. Taking this into account, one of the most effective methods for intensifying heat transfer can be considered the use of artificial roughness.

In the introductory part of the monograph, the issues of hydrodynamics and heat transfer of turbulent fluid flows are analyzed. The merits of those scientists who made the greatest contribution to the study of the regularities of hydrodynamics and heat transfer with liquid flow around on rough surfaces are highlighted.

The study of hydrodynamics of fluid flow in pipes with rough surfaces was founded by resaerches of H. Darcy, H. Bazin, R. Miss, L. Schiller and other scientists in the second half of the 19th century and the beginning of the 20th century.

Of particular note is the great contribution of I. Nikuradze to the study of the hydrodynamic regularities of fluid flow on rough surfaces. In the research conducted in the 30s of the last century at the Kaiser Wilhelm (now Max Planck) Institute in Göttingen, I. Nikuradze, among other important issues, experimentally identified three modes of roughness manifestation, which are determined by the value of the dimensionless complex[1] – $k_s u_*/\nu$.

The mentioned modes are:

1. Mode in which roughness is not manifested at all. During this mode, the roughness elements are completely immersed in the viscous sublayer:

$$0 \leq Ni \leq 5.$$

In this regime, the hydraulic resistance depends on the Reynolds number ξ=f(Re) and its absolute value is the same as in the case of smooth channels.

2. Transitional, i.e. mode of partial manifestation of roughness, during which the tips of the roughness elements are in the buffer layer:

$$5 \leq Ni \leq 70.$$

The coefficient of hydraulic resistance in the transient regime is a function of both the relative height of the roughness element and the Reynolds number:

$$\xi = f\left(\frac{k_s}{r_0}, Re\right).$$

3. Mode of full manifestation of roughness, during which the tips of roughness elements are on the outer boundary of the turbulent core of the liquid flow:

$$Ni > 70$$

In this mode, the coefficient of hydraulic resistance depends only on the height of the roughness elements k (automodel mode).

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[1] In 2008, at the VI International Forum on Heat and Mass Transfer in Minsk, one of the authors of the book, T. Magrakvelidze, presented to the forum participants a proposal to name the mentioned dimensionless complex after I. Nikuradze for his great merits in determining the regularities of turbulent flow in smooth and rough channels and write this dimensionless complex as follows way: $Ni = k_s u_*/\nu$.

The first systematic investigation, which studied the issues of heat transfer in a single-phase coolant flow in rough pipes, was carried out by W. Nanner in the 50s of the 20th century at the same Göttingen Institute. In this study, along with numerous experimental results, he proposed a physical model of the heat transfer process of rough surfaces and a calculation formula for the heat transfer intensity based on it.

According to the physical model of V. Nunner, the roughness of the heat transfer surface causes additional turbulence in the core of the coolant flow, and therefore, the reduction of the thermal resistance of the flow core. Therefore, according to V. Nunner's model, the effect of artificial roughness on heat transfer should be significant in the case of liquids with Pr<1 (liquid metals), which was not confirmed in subsequent experiments.

The inaccuracy of the physical model of V. Nunner was convincingly substantiated in the study of Academician V. Gomelauri, according to which, unlike Nunner, the vortices detached from the roughness elements, together with the turbulization of the flow core, cause a significant perturbation of the boundary layer (one might even say thinning), which leads to the intensification of heat transfer. Therefore, according to the model of V. Gomelauri, the effect of artificial roughness on the intensity of heat transfer should be significant for liquids with Pr>1. V. Gomelauri obtained a criterion equation for calculating the heat transfer coefficient of rough surfaces, which generalizes well the numerous experimental data. The opinions expressed by V. Gomelauri were later fully confirmed in the researches of both V. Gomelauri and his disciples, as well as other authors.

The theoretical analysis of hydrodynamics and heat transfer issues of surfaces with artificial roughness, which is mainly based on thermohydrodynamic analogy, is given in the works of various authors. Among them, the research of D. Dipprey and R. Sabersky can be distinguished, in which, on the basis of thermohydrodynamic analogy, the calculation formula for the heat transfer coefficient of surfaces with sandy roughness (Nikuradze roughness) was obtained, which well generalizes their own experimental data.

It should be noted that in the model of D. Dipprey and R. Sabersky, it is implied that the thermohydrodynamic analogy in rough pipes is valid for full hydraulic resistance. In addition, it is known that the thermohydrodynamic analogy is valid only for the frictional resistance, that is, in other words, the resistance of the shape of the roughness elements does not directly affect the heat transfer process. Thus, the agreement of the above-mentioned experimental data with the theory should be considered to be somewhat coincidental.

This contradiction was taken into account by V. Migai, but it must be said that the method proposed by him for separating the friction resistance from the total resistance cannot be considered perfect. In particular, according to the model of V. Migai, the shape resistance in pipes with rough surfase appears at Re = 5000 and does not depend on the height of the roughness elements, which is not confirmed experimentally.

An interesting method of separating the friction resistance from the full hydraulic resistance was proposed in the research of one of the authors of the monograph (T. Magrakvelidze). In this work, based on the opinions of S. Kutateladze, it is assumed that before the transition to the auto-model mode, the friction resistance is equal to the total hydraulic resistance, and after the transition to the auto-model mode, the resistance of the form of roughness elements appears. A calculation formula for frictional resistance in the automodel mode was proposed.

In modern devices, the heat transfer process can be carried out under the conditions of a liquid film flowing down on the heating surface. Such processes occur in condensers of thermal power plants, in nuclear power plants, in chemical-technological installations, etc. Because of this, the intensification of the heat transfer under the conditions of a liquid film flowing down on the heating surface is of great practical interest.

Based on the relevance of the issue, and taking into account the insufficient study of the problem, the authors of the monograph decided to conduct large-scale experiments, for which an appropriate experimental setup was created.

The experimental part of the monograph is dedicated to the description of an experimental setup for studying heat transfer when a liquid film flows down surfaces, conducting experiments, and analyzing the obtained results.

Experiments were performed on a device operating in both open and closed circuits. The experiments were carried out on water in an open circuit and on alcohol and water distillate in a closed circuit. A vertically placed stainless steel pipe was used as a test area in the experiments. Experiments were also carried out on a vertically placed stainless steel flat plate. In both cases, the test area was heated by directly passing a low-voltage alternating current through it.

The flow rate of the liquid in the circuit, the power supplied to the test zone, the temperature of the liquid and the test area were measured with modern high-class measuring instruments. In addition, the well-established thermocouple measurement method was used to measure the tube wall temperature, and a non-contact method was used to measure local temperatures of the flat plate wall using a FLUKE Ti60+ infrared camera.

As a result of these experiments, a number of interesting issues of heat transfer intensification by the method of artificial roughness have been established. Part of the experimental data in logarithmic coordinates (Nu, Re) is shown in Fig. 1.

The experimental data on the graph are averaged with dashed lines. The entire line I corresponds to the formula of K. Chun and R. Seban for the laminar-wave regime of the liquid film flow and line II correspond to D. Labuntsov's formula for the transition zone from wave to turbulent regime of the liquid film flow.

It can be seen from the graph that the experimental data for smooth surfaces are in good agreement with the formulas of K. Chun, R. Seban and D. Labuntsov.

The graph also shows that in the laminar flow regime, the roughness of the heating surface practically does not affect on the intensity of heat transfer. But, laminar film turbulence is clearly observed at much lower Reynolds values than for smooth surfaces.

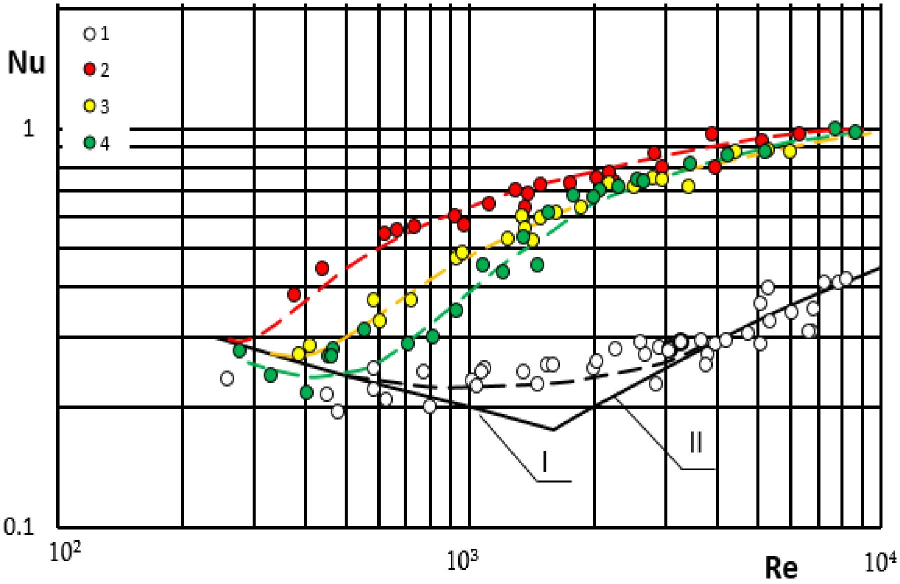

Fig. 1. The influence of the height of roughness elements on heat transfer, Pr=9-10:

1 - smooth surface;

Rough surfaces, s/h=10: 2 - h=1mm; 3 - h=0.5mm; 4 - h=0.3mm;

I - according to the formula of K. Chun and R. Seban; II - according to the formula of D. Labuntsov.

The mentioned turbulence in the case of a large height of the roughness elements occurs at lower Reynolds numbers than in the case of a low height of the roughness elements.

It should be noted that during the transition from laminar to turbulent regime, artificial roughness leads to a significant increase in the intensity of heat transfer. At the same time, the degree of intensification increases with an increase in the height of the roughness elements.

The effect of roughness on heat transfer in the turbulent regime is also important. In this case, the mentioned effect practically does not depend on the height of the roughness elements.

The influence of different types of roughness (pyramidal, two-dimensional, recesses, combined) on the intensity of heat transfer under the conditions of liquid film flow down on a vertical surface has been studied on this device. It should be noted that among the studied surfaces, the most effective in terms of heat transfer intensification turned out to be surfaces with two-dimensional and combined roughness.

Particular attention is paid to the results, according to which the use of artificial roughness leads to an increase in the intensity of heat transfer by a factor of 3 or more under conditions of a transitional regime of liquid film flow from laminar to turbulent.

The monograph presents the calculation formula for the intensity of heat transfer, adopted by the authors to generalize the experimental data obtained for a vertical pipe with the two-dimensional roughness surface and which is a modification of the well-known formula by D. Labuntsov (for smooth surfaces):

$$Nu = \frac{0.175\, Pr^{1.2}(Re/1600)}{Pr^{0.35}+0.9\,[(Re/1600)^{0.8}Pr^{0.5}-1]}\ . \tag{1}$$

The comparison of the formula (1) with the experimental data obtained by us is presented in Figure 2. As can be seen from the graph, formula (1) is in good agreement with the experimental data, which indicates that D. Labuntsov's formula, with appropriate corrections, can be successfully used even for rough surfaces.

Very interesting results were obtained when investigating the effect of artificial roughness on heat transfer in condition of a water film flows down the surface of a vertical flat plate. In particular, it is of great interest to determine the local values of the intensity of heat transfer between the roughness elements, which became possible due to non-contact temperature measurement at many points between the roughness elements of the heating surface.

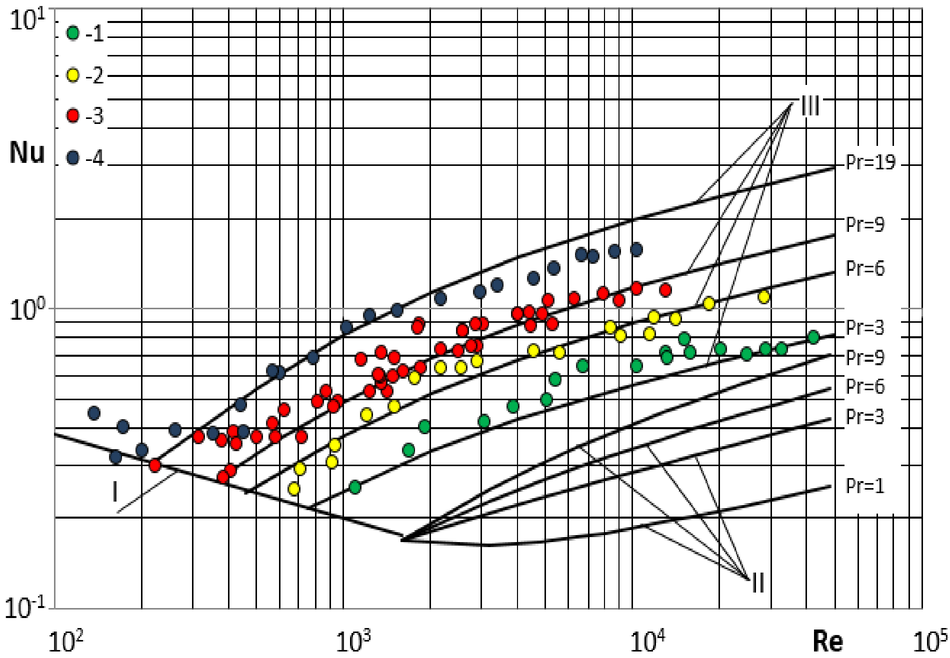

Fig. 2. Dependence of the intensity heat transfer on the Reynolds number:

Rough surfaces, h=0.5, s/h=10: 1 – Pr = 3; 2 – Pr = 6; 3 – Pr = 9; 4 – Pr = 19;

I - according to the formula of K. Chun and R. Seban; II - according to the formula of D. Labuntsov.

In Figure 3 is shown the dependence of the local heat transfer coefficients on the distance between the roughness elements.

As can be seen from the Fig.3, the local values of the heat transfer coefficients are relatively low directly on the roughness elements and in their surrounding area.

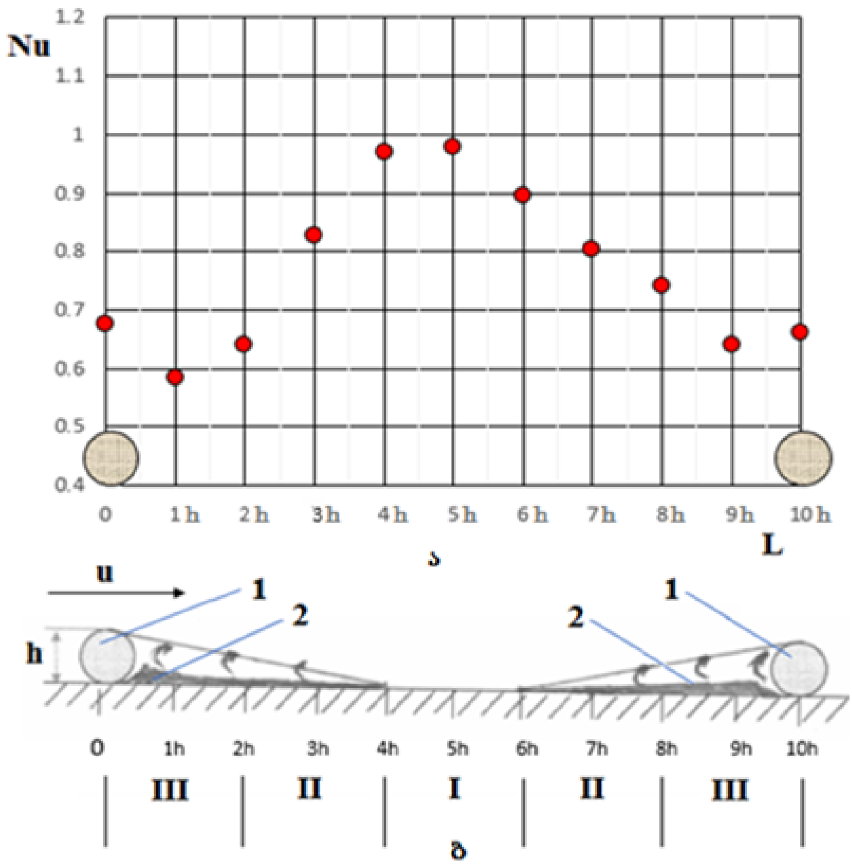

Fig. 3. Values of local heat transfer coefficients on tops of the roughness elements and between them, Re=1 972:

1 - roughness element; 2 - viscous sublayer.

As the distance L from the roughness element increases, the value of the heat transfer coefficient first increases, reaches a maximum at approximately L=(4-5)h, and then decreases again.

The obtained results give us a reason to assume that the intensity of heat transfer is maximum in zone I due to turbulence of the viscous boundary sublayer in zone (4–6)h as a result of the impact of vortices detached from the top of roughness elements. To the left and right of this zone (zones II) the renewal of the viscous boundary sublayer begins and, accordingly, the intensity of heat transfer decreases. In zones III (before and after the roughness elements), we have the so-called “dead zones”, in which the heat transfer intensity is minimal, although the heat transfer intensity in them is still high relative to a smooth surface due to the vortex nature of the fluid movement.

The results presented in Fig.3 are of great importance for determining the heat transfer mechanism of rough surfaces.

As far as we know, the first experiments to determine the local values of the intensity of heat transfer between roughness elements in the process of heat transfer of rough surfaces were carried out by V. Gomelauri, R. Kandelaki and M. Kipshidze in the conditions of turbulent fluid flow and heat transfer in a horizontally located channel with a rectangular cross section. In the mentioned experiments, the heating surface was a thin stainless steel flat plate glued to the bottom face of the channel and which was heated by passing a low-voltage alternating current through it. In these experiments, to determine the local values of the wall temperature and, consequently, the intensity of heat transfer between the roughness elements, the method of measuring the wall temperature using thermocouples was used.

The results obtained in this study fully proved the validity of the mechanism proposed by V. Gomelauri. But, at the same time, it should be noted that the placement of many thermocouples on the surface of the heating wall would undoubtedly cause a certain change in the temperature field in the heating wall especially when the wall is heated by passing low voltage alternating current directly through it. However, it should also be taken into account that at that time there were practically no other, more advanced means of measuring the temperatures of the heating surface.

The non-contact measurement method proposed by the authors of the monograph, which provides for the measurement of local values of the temperature at any point of the heating surface , obviously excludes the error that may be caused by external disturbances during the measurements.

The theoretical part of the monograph is devoted to the application of the principles of thermohydrodynamic analogy to the issues of heat transfer and turbulent flow around rough surfaces. A number of theoretical studies have been devoted to the heat transfer issues of rough surfaces. These theoretical investigations in one form or another are based on the principles of thermohydrodynamic analogy.

As is known, the use of the thermohydrodynamic analogy for the theoretical analysis of the heat transfer process gave quite good results in the case of smooth surfaces. At the same time, the extension of this analogy to the case of surfaces with roughness elements is associated with extremely difficult problems. Moreover, according to some authors, in the case when there is separation of the boundary layer from the surface and its subsequent connection, the application of the principles of thermohydrodynamic analogy is not justified. However, according to the authors of some studies, the use of thermohydrodynamic analogy may be justified in the case of surfaces with closely spaced roughness elements.

An analysis of numerous experimental and theoretical studies has shown that when flowing around surfaces with roughness elements, a boundary layer is formed along this surface, the structure of which, in the general case, can be determined by the same parameters as in the case of flowing around a smooth surface. However, it should be taken into account that the presence of roughness elements on the surface leads to the formation of vortices that break off from the edges of the roughness elements and cause a disturbance of the boundary layer and, consequently, a decrease in its thickness.

In general, a rough surface can be considered as a combination of hills and depressions (Fig. 3). It can be assumed that when a turbulent flow flows around such a surface, as in the case of a smooth surface, viscous (quasi-laminar) and buffer (transitional) layers are formed in the immediate closeness of the wall. These layers are formed not only in the recesses between the roughness elements, but also on the tops of the elements themselves. It is clear that the thicknesses of these layers will be uneven along the surface. In particular, the thickness of both viscous and buffer layers will be smaller near the tops of the elements than in the recesses between the elements. Nevertheless, in the case of a relatively small height of roughness elements, it can be assumed that viscous and buffer layers with the so-called "effective thickness"[2] are formed along the entire surface.

It can be assumed that a vortex core is also formed between the roughness elements, which does not cover the entire heat transfer surface, but only a part of this surface between the roughness elements. In the mentioned model, of course, the presence of a turbulent flow core is also taken into account. (Fig. 4).

We are considering a turbulent flow in a cylindrical pipe. Heat is transferred from the pipe wall to the fluid flow.

Due to the increase in surface area caused by the creation of roughness:

$$\frac{F_{sm}}{F_r} = n; \ \ q_r = q_{sm}\, n; \ \ \tau_r = \tau_{sm}\, n, (n < 1). \tag{2}$$

[2] It should be noted that the presence of viscous and buffer layers is somewhat conditional and introduced to simplify the mechanism of turbulent flow.

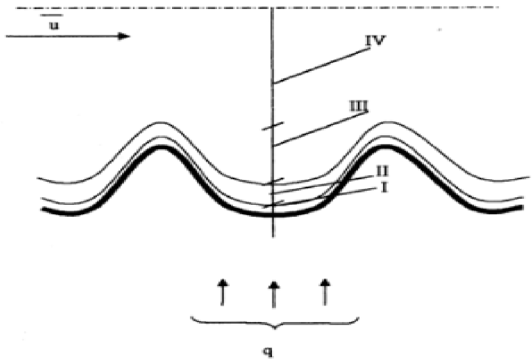

Fig. 4. Scheme of the physical model of the process of heat transfer of a rough surface:

I - viscous layer; II - buffer (transition) layer; III - vortex core between roughness elements; IV - turbulent core of the flow.

Taking into account (2), the shear stress equation, which is actually a simplified form of the equation of motion, for rough surfaces can be written as follows:

$$\tau_w \, n \, (1 - y/\, r_0) = \rho \, (\nu + \nu_t) \frac{du}{dy} \tag{3}$$

And the heat flow equation, which is obtained from the energy equation, for rough surfaces is written as follows:

$$q_w \, n \, (1 - y/\, r_0) = -\rho \, C_\rho (\nu/Pr + \nu_t/Pr_t) \frac{dt}{dy} \,. \tag{4}$$

If we introduce notations:

$$u_* = \sqrt{\tau_w n/\rho} \, , \; \varphi = \frac{u}{u_*} \; \; and \; \; \eta = \frac{y u_*}{\nu}, \tag{5}$$

Based on equations (2) and (3), we obtain a formula for calculating the intensity of heat transfer, which for moderate values of the Prandtl number has the following form:

$$Nu = \frac{0.5\sqrt{n^{-1}}\left(1+\frac{1.75}{Pr+8}\right)\mathrm{Pr}\,Re\,\sqrt{\xi/2}}{\eta_1 Pr+\eta_1 ln\left(1+\frac{\eta_2-\eta_1}{\eta_1}Pr\right)+A_1-2.5B_1} \qquad (6)$$

where, $A_1 = \frac{K\,Ni\,Pr}{1+0.165\,Ni\,Pr}$; $B_1 = \sqrt{n^{-1}}ln\left(\frac{Ni}{\eta_0}+\frac{\eta_2}{\eta_0}\right)$.

according to the presented model:

$$\eta_1 = 5\,(\xi_0/\xi)^{2/3}; \qquad \eta_2 = 30(\xi_0/\xi)^{2/3}$$

ξ_0 is the coefficient of hydraulic resistance in the case of a smooth surface and is calculated by the formula:

$$\xi_0 = (logRe)^{-2.49} \qquad (7)$$

ξ is the coefficient of frictional resistance for the pipe with rough surface.

According to the model, when $Re < Re_*$, $\xi = \xi_f$, and in the mode of full manifestation of roughness, when $Re \geq Re_*$,

$$\xi = \xi_f\left(\frac{logRe_*}{logRe}\right)^{2.49}.$$

ξ_f and Re_* are determined according to Nikuradze's experiments.

When Ni= 0, ξ= ξ_0 the formula (6) takes the form of the well-known Batchelor-Martinel-Carman formula, which is valid for smooth surfaces in the case of moderate values of the Prandtl number of the coolant.

Figure 5 shows the comparison of formula (6) with the experimental data of D. Dipprey and R. Sabersky for smooth and rough pipes in the logarithmic coordinates of Nu=f(Re).

The foregoing indicates that the thermohydrodynamic analogy can be successfully used in processing the fundamental issues of flow and heat transfer around rough surfaces.

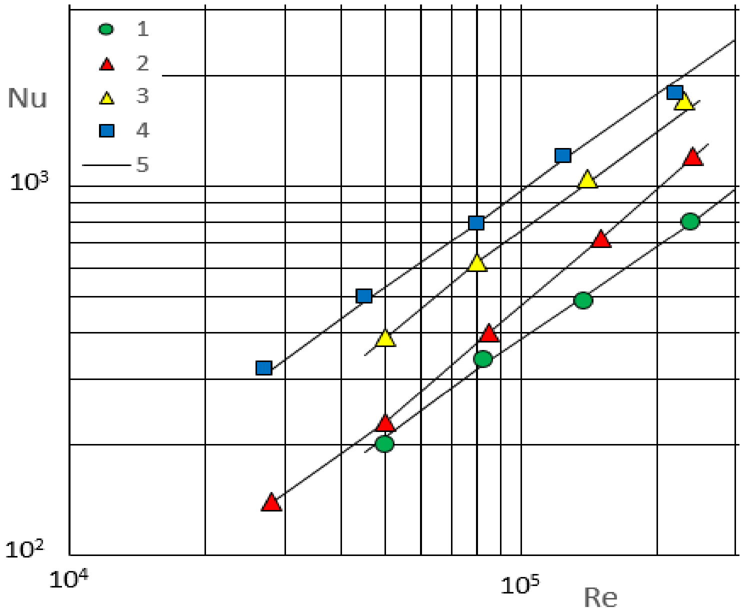

Fig. 5. Dependence of the intensity of heat transfer on the Reynolds number, Pr=2.79:

Experimental data of D. Dipprey and R. Sabersky: 1 - smooth surface;
Rough surfaces: 2 - $h/r_0 = 0.0048$; 3 - $h/r_0 = 0.0276$; 4 - $h/r_0 = 0.0976$;
5 - according to the formula (6).

Based on the results obtained by the authors, in the monograph is made conclusion that the use of the artificial roughness method in heat exchange devices for various technical purposes will significantly increase the efficiency of these devices, and at the same time, due to their compactness, this will help save expensive metal materials.

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