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HEAT TRANSFER INSIDE THE ROUGH-WALLED TUBULAR CHANNELS OF POWER PLANT ELEMENTS

ТЕПЛОВІДДАЧА ВСЕРЕДЕНІ ТРУБЧАСТИХ КАНАЛІВ З ШОРСТКОЮ СТІНКОЮ ЕЛЕМЕНТІВ ЕНЕРГЕТИЧНИХ УСТАНОВОК

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Abstract. Tubular channels with rough walls are now widely used for the formation of heat transfer surfaces of heat exchangers, which must meet the technical and economic requirements for power plants. Among these requirements, intensity and compactness are the main ones. The implementation of these requirements is possible with the intensification of heat transfer processes in a confined space. As shown by prior studies, the intensification of heat transfer on rough surfaces is based on a complex analysis of the effect of roughness in geometric parameters and thermophysical properties of heat agents on their efficiency. At the same time, the results of these studies are limited to the investigated heat agents, as well as the relative roughness heights. To compare the results of experimental studies on heat transfer, a generalized mathematical expression is derived on the basis of the Lion equation. It is intended for determining the heat transfer coefficient of rough surfaces taking into account the influence of velocity distribution on temperature drop in the turbulent core of the flow and the dependence of thickness of the viscous sublayer on the flow mode. The height is determined as the key factor of roughness. Surface roughness is divided into technical and regular. Comparison of the obtained solution to the available experimental data on heat transfer in technically rough pipes showed a difference in the results up to 10%, while in pipes with a regular roughness the divergence did not exceed 14%.

Keywords: roughness; tube; channel; intensification; heat transfer.

Анотація. Для порівняння результатів експериментальних досліджень тепловіддачі отримано узагальнений математичний вираз на підставі рівняння Лайона для визначення коефіцієнта тепловіддачі шорстких поверхонь з урахуванням впливу розподілу швидкостей на перепад температур в турбулентному ядрі потоку та залежності товщини в'язкого підшару від режиму течії. В якості визначального фактору шорсткості виділена її висота. Шорсткості поверхонь розділені на технічну і регулярну. Порівняння отриманого рішення з наявними експериментальними даними по тепловіддачі технічно шорстких труб показало відмінність в результатах до 10%, в трубах з регулярною шорсткістю розбіжність не перевищила 14%.

Ключові слова: шорсткість; труба; канал; інтенсифікація; тепловіддача.

Аннотация. Для сравнения результатов экспериментальных исследований теплоотдачи получено обобщенное математическое выражение на основании уравнения Лайона для определения коэффициента теплоотдачи шероховатых поверхностей с учетом влияния распределения скоростей на перепад температур в турбулентном ядре потока и зависимости толщины вязкого подслоя от режима течения. В качестве определяющего фактора шероховатости выделена ее высота. Шероховатости поверхностей разделены на техническую и регулярную. Сравнение полученного решения с имеющимися экспериментальными данными по теплоотдаче технически шероховатых труб показало различие в результатах до 10%, в трубах с регулярной шероховатостью расхождение не превысило 14%.

Ключевые слова: шероховатость; труба; канал; интенсификация; теплоотдача.

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Problem statement. Tubular channels with rough walls are now widely used for the formation of heat transfer surfaces of heat exchangers [1-4], which must meet the technical and economic requirements for power plants. Among these requirements, intensity and compactness are the main ones. Implementation of these requirements is possible with the intensification of heat transfer processes in a confined space. Existing ways of heat transfer intensification are based on the balance of heat transferred between heat carriers through the separating surface. They come down to increasing the heat flow while maintaining the equality of the products of the heat transfer coefficient to the area of its surface. The methods for achieving compactness are based on the reduction of weight and dimensions; they boil down to increasing the heat transfer area in a fixed volume while providing the conditions for minimizing the total energy consumption for the motion of heat carriers.

The resulting uncertainty between the intensification of heat transfer and the compactness of surfaces on the heat carrier side is the source of the scientific problem.

To solve this problem, heat exchangers should feature the heat transfer intensity necessary for ensuring their preset weight and dimensions, with a moderate increase in hydrodynamic resistance. An effective way to achieve this is to use rough surfaces [5]. The heat transfer efficiency is increased because of the shape and dimensions of the roughness, its spacing, as well as the ratio of the areas of rough and smooth surfaces. Simultaneous consideration of these parameters in heat transfer using a rough surface is complicated by a lack of knowledge about the determining effect of each of these factors, as well as their mutual influence. This is the basic disadvantage of such estimation of the effect of roughness indicators on the intensity of heat transfer, and its solution will be a partial solution to the scientific problem posed above.

Latest research and publications analysis. To date, a number of works have covered the study of hydrodynamics and heat transfer on rough surfaces [5–8]. For the most part, these processes have been studied with regard to flat and cylindrical surfaces. As a result, criterial equations were obtained for determining the hydraulic resistance and heat transfer coefficients, and the comprehensive accounting of the quantities for the intensity of heat transfer processes is carried out according to the Reynolds analogy factor.

Publication [5] presents a detailed analysis of the efficiency of discretely rough channels obtained with the help of knurls, grooves or protrusions. It is shown that two fundamentally different approaches are used for the calculation of such channels.

The first approach is based on the following assumptions. The flow near the wall with discrete transverse protrusions varies significantly under the influence of the protrusions in comparison with the flow on a smooth surface. The dimensions of a protrusion are characterized by the height *h*. The rough surface can be replaced by a conventional effective flat surface (serving as the origin for the transverse coordinate *y*). The effect of protrusions on the flow is not regarded as local but rather as averaged along the flow. Under the influence of roughness, the velocity profile near the wall changes. However, it follows from the experiments that the logarithmic velocity profile and the Prandtl's formula for turbulent friction remain valid for the rough wall. The distribution of the mixing path magnitude across the boundary layer is the same both on the rough and smooth walls; the value of the constant χ in the velocity profile also remains the same.

For the heat transfer intensification using roughness, of practical interest is the state (mode) of its full manifestation, when the dynamic Reynolds number is $hw_*/\nu > 70$. In this expression: $w_* = \overline{w}\sqrt{\frac{\xi}{8}}$ is the dynamic velocity, which is a measure of turbulent pulsations, $\overline{w}$ is the average velocity of the heat carrier in the channel, with the resistance of $\xi = \frac{8}{\overline{k}^2}$ and the parameter $\overline{k} = \frac{1}{\chi}\ln\frac{R}{h} + 4.75$, while $\chi = 0.4$ is the universal constant of the semi-empirical turbulence theory, and R is the channel radius.

In this mode, the protrusions come out of the sublayer completely, and the main component of the profile resistance at their flowing around is the form resistance ($\xi = f(h/R)$ and $\xi \neq f(\mathrm{Re})$).

Taking into account the above assumptions, the velocity profile shall be described with a logarithmic dependence both on a smooth wall and in the boundary layer on a rough surface. On the basis of experiments in the steady-state turbulent flow mode, the dependence takes the following form [5]:

$$w = 2.5\ln\frac{R}{h}\left(1 - \frac{r}{R}\right) + 8.5 =$$

$$= 2.5\ln\frac{R}{h} + 4.75 + 2.5\left(1 - \frac{r}{R}\right) + 3.75\,, \qquad (1)$$

and the relative velocity distribution along the channel radius is represented as follows [5, 12]

$$\frac{w_x}{\overline{w}_x} = 1 + \frac{3.75}{2.5\ln\frac{R}{h} + 4.75} + \frac{2.5\left(1 - \frac{r}{R}\right)}{2.5\ln\frac{R}{h} + 4.75}\,. \qquad (2)$$

To calculate heat transfer in tubes with a sandy roughness under the full manifestation of roughness, the following relations are obtained based on the heat and momentum transfer analogy [5]:

$$G\left(h^+; \mathrm{Pr}\right) = \frac{(\xi/8\,\mathrm{St}) - 1}{\sqrt{(\xi/8)}} + B\left(\frac{w_* h}{\nu}\right), \qquad (3)$$

$$St = \frac{\xi/8}{(G - B)\sqrt{\xi/8} + 1},$$

where

$$B\left(\frac{w_* h}{\nu}\right) = 3.75 + \frac{2\sqrt{2}}{\sqrt{\varepsilon}} - 2.5\ln\frac{R}{h}\,, \qquad (4)$$

also, $St = \mathrm{Nu}/\mathrm{Re}\cdot\mathrm{Pr}$ is the Stanton number, Pr is the Prandtl number, $G(h^+;\ \mathrm{Pr})$ is the heat transfer function for a rough wall, which depends on the roughness height and geometry, as well as the Prandtl number; $h^* = hw_*/\nu$. For different roughness, the right-hand side of the latter formula takes on different values.

The considerable labor intensity of experimental studies intended for establishing the *B* and *G* functions with respect to a specific roughness geometry limits the use of dependences (3) and (4) for calculation of heat transfer and friction in tubular channels with rough walls.

The common disadvantage of publications employing an averaged approach to the description of the hydrodynamic flow pattern near a wall with discrete projections is that the notion of an averaged parallel flow near the wall does not reflect the actual flow pattern; moreover, the difference between the turbulent and viscous layers becomes blurred. The flowing around of a single protrusion and the flow between two adjacent protrusions are not considered; the actual detailed flow pattern near the wall is replaced win an averaged conditional one [5].

The second, more detailed approach to modeling the flow near a rough wall suggests the need to consider a detailed pattern of the flowing around of a single roughness element and the subsequent process of flow development on the way to the next element.

The basic model of turbulent flow in a discretely rough channel is the system of differential equations describing convective heat transfer.

The complexity of analytical solution of the differential equations for the turbulent boundary layer has necessitated the development of integral methods that simplify calculation of the boundary layer.

Using the assumptions related to the nature of distribution of the temperature head and velocity of the external flow along the wall, one of the options of the solution has been virtually reduced to the similarity equation for local heat transfer on a flat wall [5]

$$Nu_x = c \cdot \mathrm{Re}_x^n \cdot \left(w/w_\infty \right)^n,$$

where w and w_∞ are the velocities of the potential and incoming flows, respectively; $\mathrm{Re}_x = w \cdot X/v$ is the local Reynolds number; $Nu_x = \alpha_x \cdot X/\lambda$; $c = 0.0255$ for a turbulent boundary layer; $n = 0.8$; X is the distance from the beginning of the formation of a turbulent layer. By expanding the application of the model of flow on a plate behind the protrusion to the case of flow in a tube with transverse protrusions, the following formula for the average turbulent heat transfer in a tube has been obtained:

$$Nu / Nu_{\mathrm{sm}} = \left(l/t \right)^{0.2}, \tag{5}$$

where l is length of identical smooth and rough tubes, and t is the protrusion spacing.

It is noteworthy that the obvious drawback of dependence (5) is the absence of the dependence of heat transfer on the height of protrusions in the tube.

Paper [6] proposes the following analytical solution for the heat transfer of rough surfaces:

$$Nu = \frac{0.7 \cdot \mathrm{Re}_*}{1.87 \cdot \left(\dfrac{R}{\delta_{\mathrm{r}}} \right)^{0.315} + \beta \dfrac{1.25}{1 - \dfrac{\delta_{\mathrm{r}}}{R}} \left(\dfrac{1}{1 - \dfrac{\delta_{\mathrm{r}}}{R}} + \dfrac{F_{\mathrm{sm}}}{F_{\mathrm{r}}} \right)} \cdot$$

$$\cdot \frac{1}{\ln \left[1 + \dfrac{0.7 \cdot \mathrm{Re}_* \left(1 - \dfrac{\delta_{\mathrm{r}}}{R} \right) \dfrac{\delta_{\mathrm{r}}}{R}}{5 + 25.2 \left(1 - \dfrac{\delta_{\mathrm{r}}}{R} \right) \sqrt{\dfrac{F_{\mathrm{sm}}}{F_{\mathrm{r}}}}} \right] + 12{,}6 \sqrt{\dfrac{F_{\mathrm{sm}}}{F_{\mathrm{r}}}}}, \tag{6}$$

where R is the channel radius, mm; δ_{r} is the height of the roughness element, mm; β is the volume fraction of depressions in the laminar sublayer; $F_{\mathrm{sm}}/F_{\mathrm{r}}$ is the ratio of the smooth and rough surface areas.

For tubular recuperative heat exchangers, reduction of the heat resistance within the channel substantially depends on the rough surface.

The heat transfer patterns in such heat exchangers are defined by thermophysical properties of the heat carrier and the nature of its motion, as well as geometric parameters, shape and size of the surface roughness within the channels. The existing theoretical solutions to the problem of heat transfer in rough tubes have been obtained at additional assumptions that do not take into account the influence of velocity distribution in the turbulent core of the flow on temperature drop and the dependence of thickness of the viscous sublayer on the flow mode.

The findings under consideration were obtained during the experimental studies of heat transfer in tubes with artificial roughness in the form of knurls, holes, and protrusions [4–8]. Application of these findings is limited by the presence of the parameters of the geometric arrangement of roughness in the above-mentioned dependences, for example, $F_{\mathrm{sm}}/F_{\mathrm{r}}$ (dependence (6)) or s/d, where s is the roughening element spacing, and d is the tube diameter [8]. A change in these parameters is difficult to identify during the operation of a heat exchange surface.

It is recommended to implement an integrated approach to the analysis of the efficiency of the heat transfer process using the Reynolds analogy factor [9, 10, 11]:

$$FAR = \frac{Nu / Nu_0}{f / f_0}, \tag{7}$$

It characterizes the ratio of heat transfer intensification to the increase in pressure losses. The Nusselt number Nu_0 and the hydraulic resistance coefficient f for a smooth-walled channel are taken as the basic values for equation (7).

In accordance with the recommendations of publication [12], the values of the base quantities can be set as follows:

$$Nu_0 = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4},$$
$$f_0 = 0.079 \mathrm{Re}^{-0.25}.$$

Following the recommendations of [8], the resistance coefficient for a rough channel shall be calculated as follows:

$$f = \frac{8}{\overline{k}^2}, \tag{8}$$

where

$$\overline{k} = 2.5 \ln \frac{R}{\delta_{\mathrm{r}}} + 4.75, \tag{9}$$

R is the internal radius of the channel, and δr is the roughness height.

Thus, the analysis has shown that the intensification of heat transfer on rough surfaces is based on a comprehensive analysis of the effect of geometric parameters of roughness and thermophysical properties of heat carriers on the heat transfer efficiency. At the same time, the findings are limited to the heat carriers under study and relative roughness height values (δ_{r}/R). For a more accurate analysis and generalization of the results of studies on the heat transfer in heat exchangers with rough channels, it is necessary to employ the Lion equation in order to obtain a mathematical expression for determining the heat transfer coefficient of rough surfaces. It would take into account the influence of velocity distribution on temperature drop in the turbulent core of the flow, as well as the dependence of thickness of the viscous sublayer on the flow mode. At the same time, it is feasible to use the roughness height as the key factor, the roughness being practically distributed in the form of knurls, holes, and protrusions.

THE ARTICLE AIM is to obtain a generalized dependence of the influence of roughness height on heat transfer intensity taking into account hydrodynamic resistance. This comes down to solving the following problems:

1) determine the dependence of the effect of the roughness dimensions on the heat transfer coefficient of tubular channels;

2) substantiate the reliability of the obtained solution through comparison with the findings of other authors;

3) establish the special features of the influence of the roughness dimensions on the integrated efficiency of heat transfer.

Basic material.

Mathematical formulation of the problem. The computational diagram of heat transfer in a tube with rough walls is shown in Fig. 1.

According to the provisions of the hydrodynamic theory of heat transfer [12] for the turbulent mode of the heat carrier flow, the problem under consideration can be reduced to the energy equation for an axisymmetric stationary flow:

$$t_{R-\delta}-\bar{t}_1=\frac{4\cdot q_{st}\cdot(R-\delta)}{\lambda}\int_0^{1-\frac{\delta}{R}}\frac{\left[\int_0^{\frac{r}{R}}\frac{w_x}{\bar{w}_x}\frac{r}{R}\,d\left(\frac{r}{R}\right)\right]^2}{\left(1+\frac{\lambda_g}{\lambda}\right)\left(\frac{r}{R}\right)}\,d\left(\frac{r}{R}\right)=$$

$$=\frac{4\cdot q_{st}\cdot R\cdot\left(1-\frac{\delta}{R}\right)}{\lambda}\int_0^{1-\frac{\delta}{R}}\frac{\left[\int_0^{\frac{r}{R}}\frac{w_x}{\bar{w}_x}\frac{r}{R}\,d\left(\frac{r}{R}\right)\right]^2}{\left(1+\frac{\lambda_g}{\lambda}\right)\left(\frac{r}{R}\right)}\,d\left(\frac{r}{R}\right),\qquad(10)$$

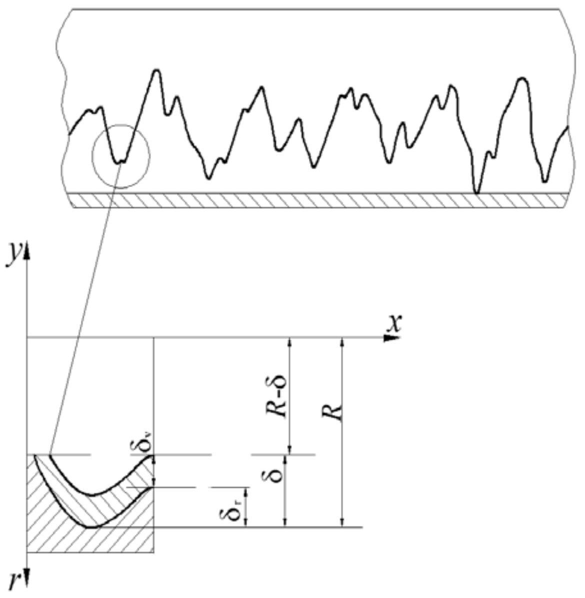

Fig. 1. Rough surface element and computational diagram of the problem

In this expression, w_x is the axial component of the heat carrier velocity in the boundary layer, q_{st} is the heat flow density at the surface, R is the internal radius of the channel, x and r are the current axial and radial coordinates, while δ_r and δ_v are the dimensions of the roughness and thickness of the viscous sublayer, respectively.

Solution of the problem. In formula (10), the multiplier $\left(1+\frac{\lambda_g}{\lambda}\right)$ is the ratio of the turbulent heat transfer coefficient to the heat conductivity coefficient. It is expressed through the values of the turbulent Prandtl number and the turbulent dynamic viscosity as follows:

$$1+\frac{\lambda_g}{\lambda}=1+\frac{\rho\cdot c_p\cdot\varepsilon_q}{\lambda}=1+\frac{\varepsilon_q}{\varepsilon_s}\Pr\frac{\mu_T}{\mu}=$$

$$=1+\beta\Pr\frac{\mu_T}{\mu}=1+\frac{\Pr}{\Pr_T}\frac{\mu_T}{\mu},\qquad(11)$$

The multiplier $\frac{1}{\Pr_T}$ is a function of the relative channel radius $\left(\frac{r}{R}\right)$ and can be estimated as follows [12]:

$\frac{r}{R}$	1	0.8	0.5	0.2	0.1
$\frac{1}{\Pr_T}$	1.02	1.1	1.2	1.4	1.5

The turbulent viscosity parameter $\frac{\mu_T}{\mu}$ as part of equation (11) can be represented as follows:

$$\frac{\mu_t}{\mu}=\frac{w_*^2\cdot\rho}{\mu\cdot\frac{d}{dr}\left[w_*\left(\frac{1}{\chi}\ln\left(\frac{R-r}{\delta}\right)+B\right)\right]}=$$

$$=\frac{w_*^2\cdot\rho}{\mu\cdot w_*\cdot\frac{1}{\chi}\cdot\frac{\delta}{R-r}\cdot\left(-\frac{1}{\delta}\right)}=-\frac{w_*\cdot 2R}{\frac{\mu}{\rho}}\frac{\chi}{2}\left(1-\frac{r}{R}\right)=$$

$$=-\frac{\mathrm{Re}_{*d}}{2}\cdot\chi\cdot\left(1-\frac{r}{R}\right)-1.\qquad(12)$$

Considering the effect of velocity distribution on temperature drop in the turbulent core of the flow and the dependence of the viscous sublayer thickness on the flow mode, the results of solving equation (10) with account for expressions (1), (2), (11), and (12) are reduced to a dimensionless form in order to obtain the criterial dependence for the calculation of heat transfer coefficients inside tubular channels with rough walls. It is provided below:

$$Nu_d=\frac{\alpha\cdot 2R}{\lambda}=\frac{q\cdot 2R}{\Delta t\cdot\lambda}=$$

$$=\frac{0.2\,\mathrm{Re}_{*d}}{2\left(1-\frac{\delta}{R}\right)\left\{PAR1\right\}+\frac{1}{2}\ln\left[PAR2\right]}.\qquad(13)$$

where $PAR1$ and $PAR2$ are the complex parameters that depend on the geometric dimensions of the roughness and are estimated in the process of solving the problem as follows:

$$PAR1 = -A_4 \ln\frac{\delta}{R} - \frac{1}{4}A_6 \ln\frac{\delta}{R}\ln^2\left[1-\frac{\delta}{R}\right]^2 +$$

$$+\ln\left(1-\frac{\delta}{R}\right)\left\{\left(1-\frac{\delta}{R}\right)\left[1-\frac{1}{2}A_6+\frac{1}{2}A_6-\frac{1}{2}A_6\right]+\right.$$

$$+\frac{1}{2}A_6\sum_{n=1}^{\infty}\frac{\left(1-\frac{\delta}{R}\right)^n}{n^2}-\frac{1}{8}A_6\left(1-\frac{\delta}{R}\right)^2+\frac{1}{18}A_6\left(1-\frac{\delta}{R}\right)^3+$$

$$\left.+\frac{1}{8}A_6\right\}+\ln^2\left(1-\frac{\delta}{R}\right)\left\{-\frac{1}{4}A_6\left(1-\frac{\delta}{R}\right)+\right.$$

$$\left.+\frac{3}{8}A_6\left(1-\frac{\delta}{R}\right)^2+\frac{1}{12}A_6\left(1-\frac{\delta}{R}\right)^3\right\}+\left[1-\frac{\delta}{R}\right]\left\{-3A_4-\right.$$

$$\left.-\frac{1}{2}A_6\right\}+\left[1-\left(\frac{\delta}{R}\right)^2\right]\left\{\frac{3}{2}A_4\right\}+\left[1-\left(\frac{\delta}{R}\right)^3\right]\left\{-\frac{1}{3}A_4\right\}-$$

$$-\frac{1}{2}A_5\sum_{n=1}^{\infty}\frac{1}{n^2}\left(1-\left(\frac{\delta}{R}\right)^n\right)+$$

$$+\frac{1}{2}A_5\sum_{n=1}^{\infty}\frac{1}{n(n+2)}\left(1-\left(\frac{\delta}{R}\right)^{n+2}\right)-$$

$$-\frac{1}{2}A_5\sum_{n=1}^{\infty}\frac{1}{n(n+3)}\left[1-\left(\frac{\delta}{R}\right)^{n+3}\right]+$$

$$+\frac{1}{2}A_6\sum_{n=1}^{\infty}\frac{\left(1-\frac{\delta}{R}\right)^n}{n^3}-\frac{1}{2}A_6\sum_{n=1}^{\infty}\frac{1}{n^3}+\left(1-\frac{\delta}{R}\right)^2\frac{3}{16}A_6+$$

$$+\left(1-\frac{\delta}{R}\right)^3\frac{1}{54}A_6.$$

The input geometric parameters are estimated as follows:

$$A_4 = \left(0.25+\frac{3.125}{2.5\ln\frac{R}{\delta_r}+4.75}\right)^2;$$

$$A_5 = 2\left(0.25+\frac{3.125}{2.5\ln\frac{R}{\delta_r}+4.75}\right)\times$$

$$\times\left(0.25+\frac{3.125}{2.5\ln\frac{R}{\delta_r}+4.75}\right),$$

$$A_6 = \left(\frac{2.5}{2.5\ln\frac{R}{\delta_r}+4.75}\right)^2;$$

$$PAR2 = \frac{1-\frac{\delta_r}{R}}{1-\frac{\delta_r+2R\frac{C}{\mathrm{Re}_{*d}}}{R}},$$

where $C = 70$ for a flow with the full manifestation of roughness [12].

Analysis of the solution and substantiation of the conditions for its application. For the sake of analysis, the roughness of tubular channels is divided into two groups, irregular (or technical) and regular. The former of roughness is caused by the peculiarities of manufacture and operation of tubular channels, while the latter is discrete roughness applied to the tubes in the form of holes, knurls, etc.

Fig. 2 compares the solution obtained for the first type of roughness. Calculations were made for the roughness element height of $\delta_r = 0.5$ mm and the tube diameter $d = 22$ mm.

Analysis of the comparison results shows that the Nusselt numbers calculated according to expression (13) agree satisfactorily with the results of [6, 8] in the indicated range of the Reynolds numbers at the δ_r/R values from 0.045 to 0.091. With the roughness element spacing up to 10 mm, the difference in the results does not exceed 10%.

Fig. 3 compares the results of the solution obtained with the results of publication [13]. The discrepancy between the experimental data and the data obtained with the analytical dependence does not exceed 14% with $\delta_r/R = 0.67...0.9$. Thereat, the maximum discrepancy is observed in a relatively “narrow” channel ($H/d = 1.11$) with the Re values over 35000.

A similar discrepancy between the theoretical solution and the published data is also observed for the tubes with regular roughness in the form of ring rolls, which is demonstrated in Fig. 4.

Fig. 5 provides the dependence of the *FAR* indicator on the Re number that is obtained at the relative roughness heights δ_r/R ranging from 0.09 to 0.27.

The results indicate a relatively high efficiency of heat transfer growth, *FAR* > 0.5. Moreover, each curve corresponding to a particular roughness height features a local maximum. This is explained by the fact that a further increase in the flow velocity increases its turbulence and makes the laminar sublayer thinner. Besides, the roughness element protrudes into the turbulent core, which creates additional resistance and reduces the efficiency of heat transfer.

The obtained results allow substantiating the flow mode inside tubes in order to achieve the maximum efficiency of heat transfer. Thus, in the case under consideration, such a heat transfer efficiency is achieved at Re ≈ 6500 (with the roughness element height $\delta_r = 0.001$ m). If δr increases, the maximum efficiency is observed at Re ≈ 6000.

Fig. 6 compares the calculated values of the Reynolds analogy factor (7) with the reference data for other methods of heat transfer intensification [11]. The

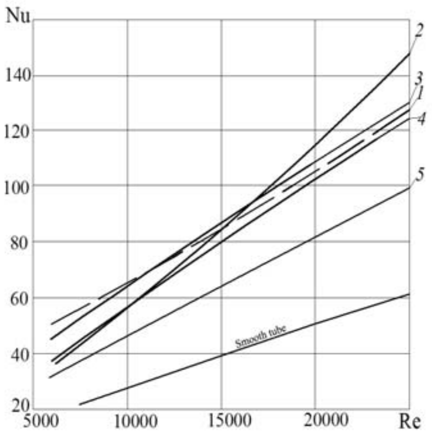

Fig. 2. Comparison of the obtained solution to the reference data for irregular roughness: *1* — calculation by dependence (13); *2* — calculation for dependence (6) [6]; *3* — results given in [8] with a roughness spacing $S = 5$ mm; *4* — results given in [8] with $S = 10$ mm, *5* — results given in [8] with $S = 20$ mm

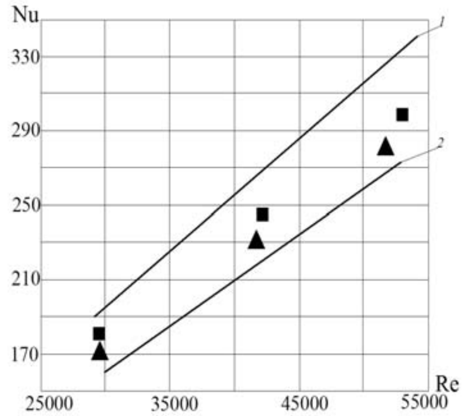

Fig. 3. Comparison of the obtained solution with the reference data for regular roughness in the form of a hole (*H* stands for the channel height, *d* is the hole diameter): *1*, *2* — calculation by dependence (13) for $H/d = 1.11$ and 1.49, respectively; ■, ▲ — results given in [13] for $H/d = 1.11$ and 1.49, respectively

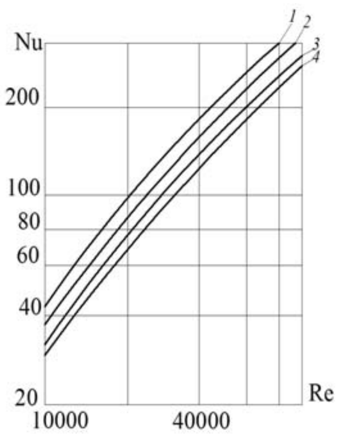

Fig. 4. Comparison of the obtained solution with the reference data for regular roughness in the form of a knurl (*d* stands for the internal diameter of the knurl, *D* is the internal diameter of the tube): *1*, *3* — calculation by dependence (13) for $d/D = 0.9$ and 0.95 ($\delta_r/R = 0.1$ and 0.05), respectively; *2*, *4* — results given in [14] for $d/D = 0.9$ and 0.95, respectively

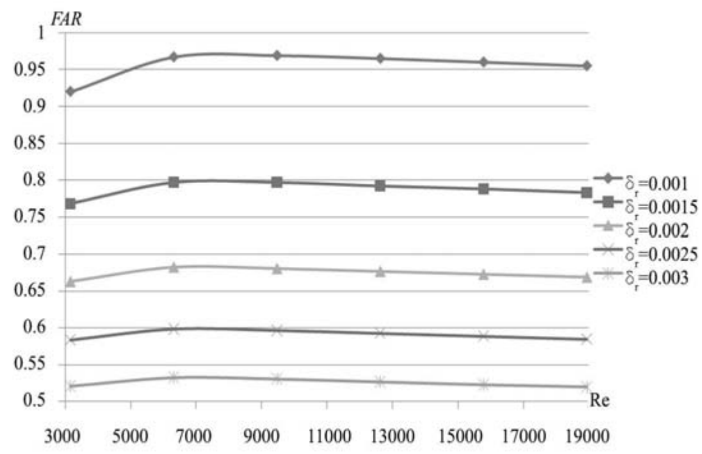

Fig. 5. Dependence of the Reynolds analogy factor *FAR* on the Reynolds number for various roughness heights δ_r, m ($\delta_r/R = 0.09$ to 0.27).

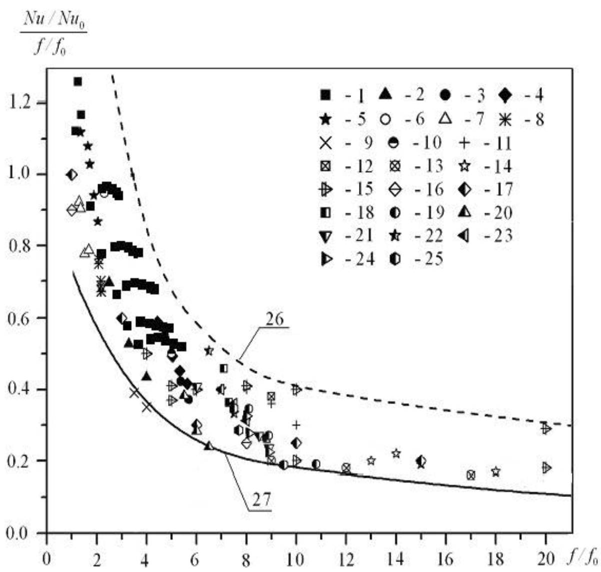

Fig. 6. Comparison of the obtained values of the Reynolds analogy factor with the data for different methods of heat exchange intensification: *1* — annular transverse protrusions in the tube; *2* — spiral projections in the tube; *3* — oblique, continuous protrusions in a square channel; *4* — beveled, split protrusions in a square channel; *5* — spherical protrusions, rectangular channel; *6* — spherical protrusions and depressions, tube; *7* — small spherical protrusions, tube; *8* — spherical protrusions, rectangular channel; *9* — internal grooves; *10* — spherical projections on a flat surface; *11* — 60° continuous and split ribs; *12* — 90° split ribs; *13* — alternating spherical protrusions and depressions; *14* — internal spiral grooves; *15* — "close" contact of protrusions with the opposite wall; *16, 17* — wire spiral inserts in a circular channel; *18* — delta-shaped vortex generators (located towards the flow); *19* — continuous V-shaped ribs (60°); *20* — continuous ribs perpendicular to the flow (90°); *21* — delta-shaped vortex generators (directed along the flow); *22* — V-shaped split ribs (60°); *23* — 60° split ribs; *24* — 90° split ribs; *25* — continuous ribs at 60° to the flow; *26* — surface with spherical depressions at low Reynolds numbers; *27* — ribbed surface at large Reynolds numbers, ■ — results of this study

analysis points at their comparatively high efficiency ($0.5 < FAR < 0.97$) at a moderate increase in hydraulic resistance ($f/f_0 \leq 5.5$). According to the physical flow pattern, the obtained results are similar to spherical protrusions in a rectangular and circular channel, which confirms the reliability of the results.

CONCLUSIONS.

1. A generalized theoretical solution is obtained for the problem of heat transfer intensification in rough-walled tubular channels of power plant elements. The ratio of the roughness height to the channel radius (δ_r/R) is assumed the key parameter in this solution.

2. As proved by the comparison of the solution with the reference data, the discrepancy in the obtained results does not exceed 10% for technical roughness in the range of the relative roughness height δ_r/R from 0.045 to 0.091. For regular roughness, it does not exceed 14% with the relative roughness height δ_r/R ranging from 0.67 to 0.9.

3. As established by the calculation results, increase in the flow velocity entails a local maximum of the efficiency of the *FAR* number, which is caused by thinning of the laminar sublayer and protrusion of the roughness element into the turbulent core.

4. The obtained results allow determining the relative roughness height in order to enhance the heat transfer intensification.

5. Further research will be focused on the development of appropriate rough-walled tubular channels for the power plant elements.

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