Numerical Studies of Heat Transfer of Flue Gases in a Narrow Channel with an Improved Heat Exchange Surface

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Abstract: This article presents the results of a study of heat transfer on surfaces with spherical depressions. It provides a rationale for the relevance of the study and a brief overview of previously conducted studies. It specifies the factors that affect the intensity of heat transfer, and explains the mechanism of flow turbulence in a channel with spherical depressions and protrusions. It displays data that determine the parameters that are directly or indirectly included in the similarity criteria, initial and boundary conditions, as well as the geometric and thermal characteristics necessary to create a mathematical model. The mathematical model of heat exchange in channels with an improved heat exchange surface with applied spherical depressions and protrusions is considered, allowing to estimate the influence of the presence of local turbulent areas on the channel surfaces on the Nusselt number in comparison with a similar channel with smooth surfaces. Also, the results of numerical studies on the influence of turbulence on heat exchange characteristics of channels with an improved heat exchange surface are presented in the form of graphs.

1 INTRODUCTION

One of the main tasks of heat engineering for many years remains the task of studying the flow of viscous liquid and heat exchange in channels with spherical depressions and protrusions applied to the heat exchange surface. The practical significance of these studies is associated with the intensification of heat exchange in surface heat exchangers.

One of the solutions for increasing the efficiency of heat-using equipment is to increase heat transfer (heat transfer) from one heat carrier to another. This goal is achieved by various methods and techniques. The most commonly used method is turbulization of the heat carrier flow. With turbulized flow of liquid or gas, heat transfer is much more intense than with laminar flow. Turbulization of the flow can be achieved by using various devices located on the heat exchange surfaces. Fins of various shapes, nozzles, artificial roughness and much more can be used as turbulators. It should be borne in mind that the use of turbulators, although it helps to increase the efficiency of heat transfer, leads to an increase in resistance, and therefore to an increase in the pressure drop of the flow.

2 MATERIALS AND METHODS

To intensify heat exchange in channels with flat heat exchange surfaces, a method of flow turbulization is often used by applying spherical and other forms of irregularities to the heat exchange surface [1]-[4].

The use of turbulizing surfaces in the form of various dimples and protrusions is one of the methods of intensifying heat exchange in narrow channels [5]-[7].

The flow and heat exchange modes in channels, especially in narrow channels with turbulizing surfaces, differ from similar processes occurring in smooth channels.

The main factor influencing the intensity of heat exchange in such channels is the presence of spherical depressions and protrusions on the heat exchange surfaces, which serve to turbulize the flow.

When coolants flow around spherical depressions and protrusions located on the heat exchange surface, dynamic vortex structures arise, which are observed both in laminar and turbulent flow regimes. The resulting vortices contribute to an increase in heat transfer [8], [9].

When constructing a model and processing the experimental results in the form of criteria formulas, it is necessary to specify the determining parameters that are directly or indirectly included in the similarity criteria. In stationary problems of convective heat exchange, the determining parameters include:

- R_0 – determines (characteristic) size of the heat exchange region;
- T_0 – determines (characteristic) temperature in the heat exchange region;
- w_0 – determines velocity.

When creating a mathematical model of heat exchange on a surface with applied spherical depressions and protrusions, the geometric values of the depressions and protrusions were changed in the following ranges: h is the depth of the depression 0.1-20 mm; S_x is the longitudinal step, 10-50 mm; S_y is the transverse step, 5-35 mm; γ is the density of application of depressions and protrusions, 20-70%, and the thermophysical parameters of the flow took values in the following ranges: Re – 1000-50000; ν is the kinematic viscosity of flue gases 112x106 m^2/sec; ω_0 is the flow velocity of flue gases, 1-20 m/sec: *a* is the coefficient of thermal diffusivity of flue gases, = 183.2 m^2/sec, the geometric parameters of the channel were adopted as follows: d_k is the channel diameter, d_k= 4f/П; H – 30-60 mm; l – 35-85 mm; $d_л$ – recess diameter, 10-30 mm.

The modeling is based on numerical calculations based on the dependence proposed in [10], [11], [12]:

$$\frac{Nu}{Nu0} = (\ 1+0{,}50\ \gamma)\ K_f\ , \qquad (1)$$

where $K_f = \pi\ h^2 / 2\ S_x\ S_z$.

For a smooth surface, heat transfer is calculated using the following formula:

$$Nu_0 = 0.024\ Re_0^{\ 0.8}\ Pr^{\ 0}, \qquad (2)$$

where Pr = 0,61, criteria $Re_0 = \omega_0\ d/\ \nu$.

Having determined Nu_0 from (1), we find Nu and then α, the heat transfer coefficient for the improved surface:

$$\alpha_1 = Nu\ \lambda_{gas} / d_{К}, \qquad (3)$$

where λ_{gas} = 8.27 10-2, W/m deg, is the thermal conductivity coefficient of flue gases at t_{gas} = 700°C. In addition, calculations were carried out using the empirical equation proposed in [12], where the channel height H is taken into account in the geometric coefficient.

$$\frac{Nu}{Nu_0} = 1 + 4{,}4\ [\gamma(h/D)]^{0{,}8}/(\tfrac{H}{D})^{0{,}6}\ , \qquad (4)$$

where H is the channel height, m.

The analysis of (1) and (4) showed that (1) does not take into account the depression diameter D, while (4) does not consider the density of the applied depressions γ. Considering the significance of the parameters D and γ, the authors proposed the following (5) for calculating the heat transfer intensity, which accounts for the influence of both of these parameters.

$$\frac{Nu}{Nu_o} = 1 + (\gamma\ K_{f1})/\ Nuo, \qquad (5)$$

where $K_{f1} = (\pi(2hr - h^2))/(S\_x\ S\_z\ )$,
where r – radius of the depression, m.

3 RESULTS AND DISCUSSION

Numerical calculations for given geometric and thermophysical parameters yielded the following results, some of which are shown in Figure 1, Figure 2, Figure 3, Figure 4, Figure 5, Figure 6.

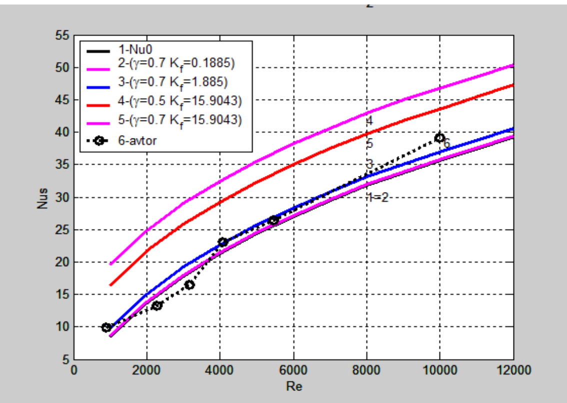

Figure 1: Dependence of the Nusselt number Nu on the Reynolds number Re at various values of γ.

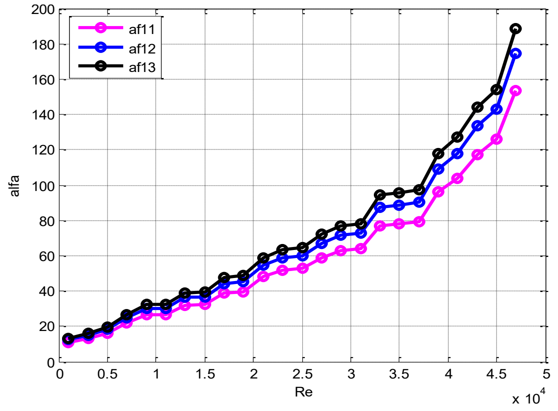

Figure 2: Dependence of the heat transfer coefficient α on the Reynolds number Re for different values of γ.

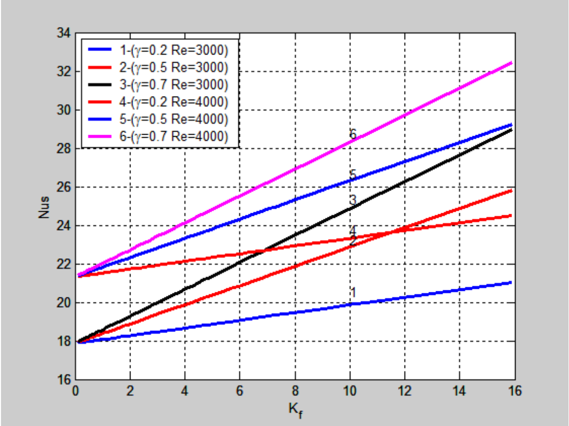

Figure 3: Dependence of the $\frac{Nu}{Nu0}$ ratio on the Reynolds number (Re) for different values of γ and dk.

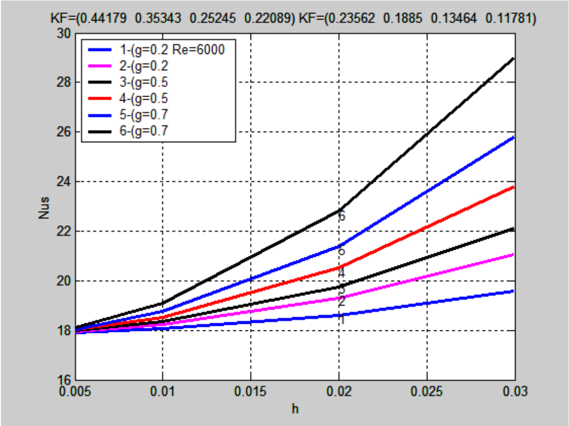

Figure 4: Graph of the dependence of $\frac{Nu}{Nu_0}$ on h, the dimple depth, at different dimple densities γm=0.2; 0.5; 0.7 and at various Kf values (Re= 6000).

Analysis of the obtained calculated curves showed that as the depth of the turbulence-inducing dimples and the area of the surface with such dimples increase, the intensity of heat transfer from the heat carrier to the surface also increases. A similar pattern of the influence of dimple depth on heat transfer intensity at different values of the improved (dimpled) surface area fraction is observed for various Reynolds numbers (Re).

The analysis of the calculated curves presented in Figure 4 shows that an increase in the density of turbulence-inducing dimples (g_m) leads to an increase in heat transfer intensity. Moreover, as the geometric coefficient Kf increases, the enhancement of heat transfer becomes more significant. It should be noted that the nature of the influence of dimple density on heat transfer intensity remains consistent for different Reynolds numbers (Re).

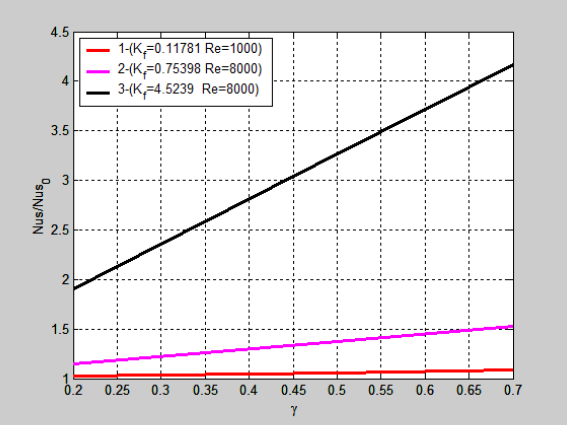

Figure 5: Dependence of $\frac{Nu}{Nu_0}$ on the dimple density γm at different values of Kf=0.11781, 0.7539, 4.5239 (Re= 1000, 8000).

Figure 6 presents graphs of the calculated values showing the influence of the geometric coefficient K_f on the intensity of heat transfer. The analysis showed that as the fraction of the surface improved with turbulence-inducing dimples increases, the heat transfer intensity also increases. An increase in the Reynolds number (Re) likewise leads to an increase in heat transfer intensity.

Special attention should be paid, when designing heat exchange surfaces with turbulizers, to the parameters of dimple arrangement on the surface — that is, to analyzing the influence of the longitudinal and transverse pitches of the dimples, which determine the dimple density coefficient. The influence of the arrangement parameters (transverse and longitudinal pitches) on the heat transfer intensity was investigated numerically (Figures 7 and 8).

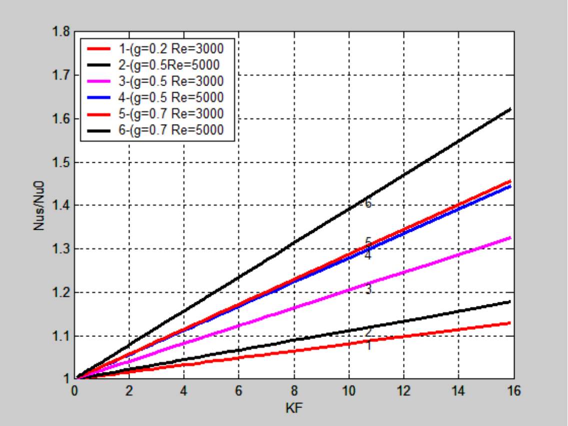

Figure 6: Graph of the dependence of heat transfer intensity $\frac{Nu}{Nu_0}$ on the geometric coefficient Kf at different dimple densities γ = 0.2; 0.5; 0.7 and Re = (3000, 5000).

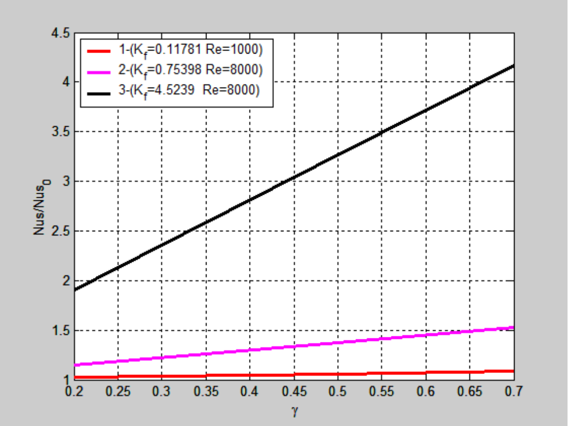

Figure 7: Graph of the dependence of $\frac{Nu}{Nu_0}$ o on Sz at γ = 0.2; 0.5; 0.7 and Re = 6000.

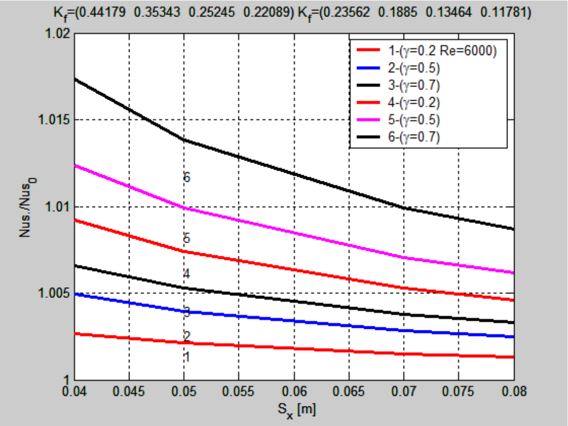

Figure 8: Graph of the dependence of $\frac{Nu}{Nu_0}$ on Sx at γ = 0.2, 0.5, 0.7 (Re = 6000).

The analysis showed that as the geometric parameters of the arrangement of turbulence-inducing depressions increase, the intensity of heat transfer decreases. This is due to the overall reduction of the geometric coefficient Kf and the decrease in the proportion of the enhanced surface.

The authors conducted a comparative analysis of the results obtained using the proposed (5) with the results of studies by other researchers [11], [12]. Figures 9, 10, and 11 present several graphs illustrating this comparative analysis.

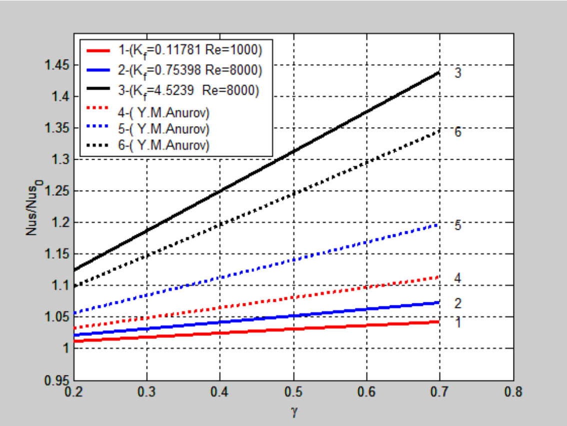

Figure 9: Dependence of the heat transfer intensity $\frac{Nu}{Nu_0}$ on the density of the applied depressions γ.

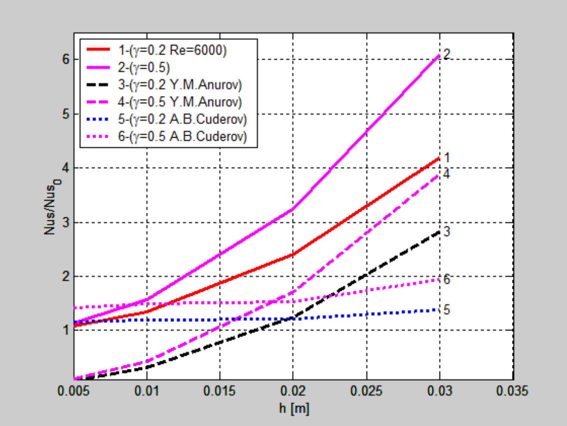

Figure 10: Dependence of the heat transfer intensity $\frac{Nu}{Nu_0}$ on the depth of the turbulence-inducing depressions h.

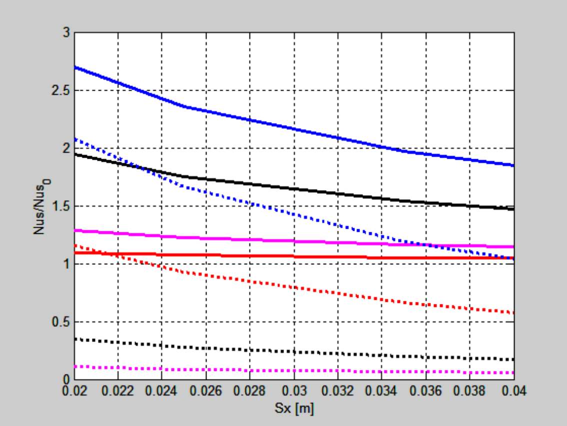

Figure 11: Dependence of the heat transfer intensity $\frac{Nu}{Nu_0}$ on the longitudinal pitch Sx of the depressions arrangement.

4 CONCLUSIONS

The analysis of the obtained dependencies showed that the flow turbulization achieved by applying turbulence-inducing depressions on the heat exchange surface enhances heat transfer. Increasing the depth of the turbulence-inducing depressions (h) together with the density of the applied depressions (γ) contributes to up to a twofold increase in heat transfer, although at certain S_x values, a slight decrease in heat transfer is observed.

An increase in the density of turbulence-inducing depressions at a constant geometric coefficient (K_f) also intensifies heat exchange, though to a lesser extent.

Changes in the geometric dimensions of the depressions likewise affect the heat transfer intensity; with larger dk values, large vortices form in the depression area, leading to a predictable increase in flow turbulence and consequently enhanced heat transfer. Overall, flow turbulization using depressions allows for an increase in heat exchange intensity, which in turn can reduce the initial consumption of organic fuel.

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