THE METHOD OF NUMERICAL MODELING OF HYDRODYNAMICS AND HEAT EXCHANGE IN A CHANNEL WITH DISCRETE ROUGHNESS

МЕТОДИКА ЧИСЛЕННОГО МОДЕЛИРОВАНИЯ ГИДРОДИНАМИКИ И ТЕПЛООБМЕНА В КАНАЛЕ С ДИСКРЕТНОЙ ШЕРОХОВАТОСТЬЮ

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Abstract: Basic methodology has been developed for a numerical modeling and heat exchange in a smooth channel and in a channel with a discrete roughness in the form of semispherical dimples. The methodology provides a possibility of computerized parametric calculations that adequately enough model the examined physical phenomena and allow to determine their characteristics being of practical interest. The efficiency of the examined discretely rough surface was estimated based on the coefficients of heat transfer and hydraulic resistance.

KEYWORDS: HEAT TRANSFER ENHANCEMENT, DISCRETE ROUGHNESS, COMPUTER MODELING, NUMERICAL CALCULATION.

1. Introduction

In modern heat-and-power devices based on alternative energy sources, heat-exchange equipment is mainly a major part and largely determines their overall technical and economic indicators. And in the future, one of the main ways to increase the efficiency of heat-and-power devices is to improve the heat exchange equipment, which can be realized by introducing the efficient methods of enhancing the heat transfer.

By means of heat transfer enhancement, the amount of heat transferred through a unit of the heat exchange surface increases, and a more favorable ratio between the transmitted heat quantity and the heat coolant pumping capacity is achieved.

For industrial use, the most promising is the heat transfer enhancement in the channels due to the artificial discrete roughness of the wall.

But the issue of the intensification and efficiency of this method remains open, since there is a little knowledge of the mechanism of these processes and changes, in particular for reliable calculation of the heat exchange and hydraulic losses therein, which are necessary for the design of power devices.

Therefore, the study of the processes of heat exchange and hydrodynamics of heat coolants in channels with discrete roughness and the development of a method for calculating them represent an actual task for the engineering practice.

The purpose of the present work is to develop a basic methodology for a computer modeling and numerical calculation of hydrothermodynamic parameters in a channel with discrete roughness in the form of spherical dimples.

2. Prerequisites and means for solving the problem

In recent years, interest has increased in the study of heat exchange and flow structure in channels in the presence of depressions in the form of hemispheres (dimples). It is explained by the fact that the dimples have proved to be an efficient intensifier of heat transfer when they are flown around [1]. Studies of the flow in channels with intensifiers in the form of dimples were examined in works [2-5].

In a number of works, it has been experimentally established that heat exchange surfaces with spherical depressions allow a significant increase (by 1.5-4.5 times) in heat exchange at a moderate growth in the hydraulic resistance. And at this stage, it is an efficient way of intensification from the standpoint of the ratio in the increments of heat transfer and hydraulic resistance.

The thermohydraulic characteristics of surfaces formed by spherical depressions depend on the shape of the dimples (with sharp edges or smooth outlines), their density on the surface, the longitudinal and transverse spacing of the dimples, their relative depth, the relative height of the channel. In addition to these parameters, the relative location of the dimples on adjacent channel surfaces has a considerable impact on the hydraulic resistance and heat transfer in heat exchangers.

At present time, no complete clarity exists about the ratios of the coefficients of heat transfer $\frac{Nu_2}{Nu_1}$ and resistance $\frac{\zeta_2}{\zeta_1}$ for surfaces with dimples [6].

In this paper, an attempt is made to perfect the basic methodology for numerical modeling of hydrodynamics and heat transfer in a channel with discrete roughness in the form of hemispherical dimples with a sharp entry edge.

The main phases of the basic methodology for the numerical solution of the problem under consideration can be represented as follows:

- Physical formulation of the study task.
- Development of a simplified model of the initial study subject.
- Construction of the geometric electronic model of the subject.
- Mathematical formulation of the problem, boundary and initial conditions.
- Selection of necessary CAD/CAE programs.
- Creation of a discrete calculation model, optimization of the computational grid.
- Development of a strategy for solution of the task:
  - formulation of calculation purposes and criteria for completion of the calculation;
  - methodology for monitoring and control of the calculation process;
  - technique for data visualization and processing of digital calculation data.
- Solution of check problems, comparison with known data, evaluation of the accuracy of solutions obtained.
- Interpretation of calculation data in order to optimize the properties of the subject under study.

The developed methodology should provide the possibility of computer parametric calculations that adequately simulate the examined physical phenomena and allow to determine their characteristics being of interest in practice.

The verification of the developed method for calculation of the coefficients of heat transfer and hydraulic resistance was carried out under forced turbulent flow of cold liquid (water) in smooth and discretely rough channels (pipes) with a relative length $L/d > 50$. Aluminum pipe walls were heated up to a certain constant temperature.

Computer modeling was performed using the selected package of CAD/CFD programs SolidWorks/FlowSimulation. The complete system of Navier-Stokes equations and the energy equation were solved using the $k-\varepsilon$ model of turbulence.

The velocity at the entrance to the channel varied within the $V_{\text{in}}$ range 0.3 m/s and 1 m/s. With the initial Reynolds numbers

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Re_{inlet} \approx 5000 \text{ to } 18000 \text{ at the initial fluid temperature } t_{inlet} = 20^\circ \text{C. The efficiency of the heat exchange surface was estimated from the coefficients of heat transfer and resistance.}

### 3. Solution of the examined problem

In the CAD program SolidWorks, three-dimensional computing models of smooth and discretely roughened pipes were constructed with the following geometric characteristics: inner pipe diameter \( d_0 = 0.018 \text{ m, external diameter } d_{wall} = 0.022 \text{ m, length } L = 1.6 \text{ m. As a}

discretely rough surface in the pipe, spherical depressions (dimples) with sharp edges and exit edges were chosen, which were located in an in-line arrangement with a relative spacing between the dimple axes \( S/h = 10 \) and the dimple position angle \( \varphi = 120^\circ \).

**Fig. 1. Geometric model with discretely rough surface**

The following were accepted as the boundary conditions: at the pipe inlet – flow rate within the \( V_{inlet} \) range 0.3 and 1 m/s at fluid temperature \( 20^\circ \text{C. At outlet: pressure } p_{outlet}=101235 \text{ Pa. The channel walls have physical properties of aluminum and are heated up to the temperature } t_{wall}=105^\circ \text{C. Fluid flow is turbulent (Fig. 2.} )

**Fig. 2. Boundary conditions**

The mathematical modeling of the medium motion and heat transfer used the non-stationary Navier-Stokes equations, the energy equation (the first law of thermodynamics), and the equation of state are used [8-9]. For turbulent flows, the initial equations are averaged by the Reynolds method and additional stresses due to turbulent fluctuations of the parameters are taken into account [9].

The obtained incomplete system of equations is closed with the help of complementary equations for the kinetic energy of turbulence \( k \) and the dissipation of turbulence energy \( \varepsilon \) in accordance with the known \( k - \varepsilon \) model of turbulence [9]. The system of equations for the conservation of impulse, mass and energy, which describes the turbulent, laminar and transient flows of a compressible fluid with heat exchange, can be represented in the following form:

\[
\begin{align*}
\frac{\partial(pu_i)}{\partial t} &+ \frac{\partial}{\partial x_j} \left( p(u_iu_j - \tau_{ij}) \right) + \frac{\partial P}{\partial x_i} = F_i \\
\frac{\partial \rho}{\partial t} &+ \frac{\partial}{\partial x_j} (\rho u_j) = 0; \\
\frac{\partial (\rhoE)}{\partial t} &+ \frac{\partial}{\partial x_j} ((\rhoE + P)u_j + q_i - \tau_{ij}u_j) = F_iu_j + Q_H;
\end{align*}
\]

Fig. 3. Flowpath with discretely rough surface

The obtained incomplete system of equations is closed with the aid of complementary equations for the kinetic energy of turbulence \( k \) and the dissipation of turbulence energy \( \varepsilon \) in accordance with the known \( k - \varepsilon \) model of turbulence [9]. The system of equations for the conservation of impulse, mass and energy, which describes the turbulent, laminar and transient flows of a compressible fluid with heat exchange, can be represented in the following form:

\[
\begin{align*}
\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_i u_j - \tau_{ij}) + \frac{\partial P}{\partial x_i} = F_i \\
\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j} (\rho u_j) = 0; \\
\frac{\partial (\rhoE)}{\partial t} + \frac{\partial}{\partial x_j} ((\rhoE + P)u_j + q_i - \tau_{ij}u_j) = F_iu_j + Q_H;
\end{align*}
\]

4. Results and discussion

The efficiency of the heat-exchange surface was evaluated by the coefficients of heat transfer and of resistance at points, of which values were averaged. The points were positioned on the horizontal axis along the pipe diameter with 0.5 mm spacing and at 1.5 m distance from the inlet section.

To determine said coefficients, the following values were identified at each examined point: density, viscosity, velocity, temperature, Prandtl number \( \text{Pr} = \mu C_p / \lambda \), were \( C_p \) – specific thermal capacity at constant temperature, \( \lambda \) – thermal conductivity coefficient of fluid, \( \mu \) – dynamic viscosity coefficient.
Based on the results of the numerical calculation, the Reynolds number (Re) at the examined points was calculated from the known dependence [11]:

\[ \text{Re}_{1,2} = \frac{\rho_0 V_0 d}{\mu_1} \]  

(11)

where Re - Reynolds number, d - equivalent pipe diameter (m), \( V_0 \) - mean velocity of fluid flow (m/s), \( \mu_1 \) - dynamic viscosity coefficient (Pa*s), \( \rho_1 \) - flow density (kg/m³), subscripts throughout the paper have the following meaning: 1 - smooth, 2 - discretely rough (tubes).

The result has demonstrated that for mean flow velocities in the examined range \( V_{\text{inlet}} \), the Reynolds numbers \( \text{Re}_2 \) in a discretely rough pipe in relation to a smooth one grow approximately by 10-20%. Dependence of Reynolds numbers \( \text{Re}_1, \text{Re}_2 \) on the mean flow velocities \( V_{\text{inlet}} \) m/s for the examined channels is shown on Fig. 4.

Variation of number \( \text{Re}_2 \) value at the examined points is associated with the increase of temperature \( T_1°C \) in a discretely rough channel in comparison with temperature \( T_2°C \) in a smooth pipe \((T_2°C - T_1°C)\). Accordingly, temperature variation is accompanied by changes in physical properties of fluid, such as density and viscosity, which are included in the nondimensional velocity parameter Re. Variation of the mean flow velocity \( V_0 \) at the examined points compared to the mean initial velocity \( V_{\text{inlet}} \) was also taken into consideration. Dependence diagrams of outlet temperature \( T_1, T_2 \) versus mean flow velocity in a channel \( V_{\text{inlet}} \) are shown on Fig. 3.

![Fig. 3. Dependence diagrams of temperature \( T°C \) on mean velocity \( V_{\text{inlet}} \) m/s of flows \( T_1(V_{\text{inlet}}) \) and \( T_2(V_{\text{inlet}}) \).](image)

Variation of number \( Nu_2 \) value at the examined points is associated with the increase of temperature \( T_1°C \) in a discretely rough channel in comparison with temperature \( T_2°C \) in a smooth pipe \((T_2°C - T_1°C)\). Correspondingly, temperature variation is accompanied by changes in physical properties of fluid, such as density and viscosity, which are included in the nondimensional velocity parameter Re. Variation of the mean flow velocity \( V_0 \) at the examined points compared to the mean initial velocity \( V_{\text{inlet}} \) was also taken into consideration. Dependence diagrams of outlet temperature \( T_1, T_2 \) versus mean flow velocity in a channel \( V_{\text{inlet}} \) are shown on Fig. 3.

![Fig. 4. Dependence diagram of Reynolds numbers \( \text{Re}_1, \text{Re}_2 \) on mean flow velocities \( V_{\text{inlet}} \) m/s for smooth and discretely rough tube.](image)

Validity check of solution of the problem under consideration was carried out by comparison of theoretical solution [7, 10, 11] to the results of numerical experiment with use of dependencies [12, 13].

\[ Nu_{t,2} = 0.021 \cdot \text{Re}_0^{0.8} \cdot \text{Pr}_0^{0.43} \left( \frac{\text{Pr}_{\text{water}}}{\text{Pr}_{\text{wall}}} \right)^{0.25} \cdot \varepsilon_1 \]  

(12)

where, \( Nu_t \) – heat transfer coefficient for a smooth channel, \( Nu_2 \) – for a discretely rough channel, \( Pr \) – Prandtl number.

\( \varepsilon_1 \) – correction for the initial flow section for a smooth pipe, of which value is equal to 1 at \( L/d > 50 \).

\( \varepsilon_2 \) – correction taking into account the growth in the heat transfer coefficient as a consequence of artificial roughness:

\[ \varepsilon_2 = 1.04 \cdot \text{Pr}_0^{0.04} \cdot \exp \left[ 0.85 \cdot f \left( \frac{s/h}{s/h_{\text{opt}}} \right) \right] \]  

(13)

where \( s/h \leq s/h_{\text{opt}} \)

\[ f \left( \frac{s/h}{s/h_{\text{opt}}} \right) = \left( \frac{s/h}{s/h_{\text{opt}}} \right)^{2.1} \]

where \( s \) – distance between axes of dimples, \( h \) – inner radius of dimple \((\text{Fig. } 1)\), \( s/h_{\text{opt}}\) = 13±1

Comparison between the results of theoretical solution and numerical calculation for a smooth channel has demonstrated a satisfactory matching not exceeding the allowable range 5-10%.

In case of discretely rough channel at Reynolds numbers \( \text{Re}_2/\text{Re}_1 = 1.1-1.2 \), which corresponds to the flow velocity range \( V=0.3-1 \) m/s, physical properties of medium included in the left-hand member of equation (12) do not have a significant impact on the value of number \( Nu_t \).

Dependences of numbers \( Nu \) on the relative Reynolds number \( \text{Re}_2/\text{Re}_1 \) for the examined channels are shown on Fig. 5.

![Fig. 5. Dependence of Nusselt number \( Nu \) on relative Reynolds number without correction \( \varepsilon \), \( Nu_1 \) – theory, \( Nu_2 \) – numerical calculation (smooth channel), \( Nu_3 \) – numerical calculation (discretely rough channel).](image)

![Fig. 6. Dependence of relative numbers \( Nu_2/Nu_1 \) on Reynolds number \( \text{Re}_2/\text{Re}_1 \).](image)
The study further examines the impact of the resistance coefficient on the heat exchange processes. The occurrence of perturbing factors in the presence of the vortex structures on the heat transfer surface leads to the appearance of dimples, the large-scale dynamic vortex structures that are observable at different fluid flow regimes. The depressions in the form of dimples, the large-scale dynamic vortex structures, example, shown on Fig. 7 where when fluid flows around the pumpover of fluid.

The results of the resistance coefficient calculation have shown that \( \xi_2/\xi_1 \) in the examined range of numbers \( Re_2/Re_1 = 1.1-1.22 \) it is expedient to apply the considered rough surface from the standpoint of advantage in power input to the pumpover of fluid.

\[
\frac{\xi_2}{\xi_1} < C_2^{3,44} \tag{15}
\]

The methodology allows also to obtain the visualized pictures for the analyses of processes being of practical interest, such as, for example, shown on Fig. 7 where when fluid flows around the depressions in the form of dimples, the large-scale dynamic vortex structures appear observable at different fluid flow regimes. The presence of the vortex structures on the heat transfer surface leads to destruction of the boundary layer and, accordingly, enhances the heat exchange processes. The occurrence of perturbing factors in the flow influences both the flow hyrodynamics and the heat exchange.

The visualized pictures of flow-around of the dimples in a channel with a discrete roughness in the cross section are on Fig. 7a for flow velocities: a) \( V_{\text{inlet}} = 0.5 \text{ m/s} \), b) \( V_{\text{inlet}} = 0.9 \text{ m/s} \); in the cross section c) \( V_{\text{inlet}} = 0.5 \text{ m/s} \), d) \( V_{\text{inlet}} = 0.9 \text{ m/s} \).

\begin{align*}
\text{a) } V_{\text{inlet}} &= 0.5 \text{ m/s}. \\
\text{b) } V_{\text{inlet}} &= 0.9 \text{ m/s}. \\
\text{c) } V_{\text{inlet}} &= 0.5 \text{ m/s}. \\
\text{d) } V_{\text{inlet}} &= 0.9 \text{ m/s}. 
\end{align*}

**Fig. 7.** Pictures of flow-around of the dimples in the cross section: a) \( V_{\text{inlet}} = 0.5 \text{ m/s} \), b) \( V_{\text{inlet}} = 0.9 \text{ m/s} \); in the cross section: c) \( V_{\text{inlet}} = 0.5 \text{ m/s} \), d) \( V_{\text{inlet}} = 0.9 \text{ m/s} \).

5. Conclusion.

1. The developed basic methodology for numerical modeling has allowed to obtain results satisfactorily coinciding with known theoretical calculations thus confirming its validity.
2. The methodology provides the possibility of computer parametric calculations that adequately enough model the examined physical phenomena and allow to determine characteristics being of practical interest.
3. It is established that variation of temperature at the outlet of a discretely rough channel leads to the change in the value of Re number in relation to a smooth pipe.
4. It is demonstrated that the intensity of heat transfer is significantly influenced by the channel geometry and not by physical properties of the fluid.
5. The expediency is determined of applying the examined rough surface from the standpoint of advantage in power input to pumpover of fluid.

6. References