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

Dr.sc.ing. Sidenko N., Dr. sc.ing. hab. prof. Dzelzitis E.
Riga Technical University, Faculty of Civil Engineering, Institute of Heat, Gas and Water Technology.
Nsidenko@inbox.lv, Egils.Dzelzitis@rtu.lv

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 between the coefficients of heat transfer Nu_2/Nu_1 and resistance ξ_2/ξ_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-ε* model of turbulence.

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

Re_{inlet} = 5000 to 18000 at the initial fluid temperature t_{inlet} = 20°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 roughed pipes were constructed with the following geometric characteristics: inner pipe diameter d_b = 0.018 m, external diameter d_{out} = 0.022 m, length L = 1.6 m. As a discretely rough surface in the pipe, spherical depressions (dimples) with sharp entrance 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 φ = 120 °. Fig. 1.

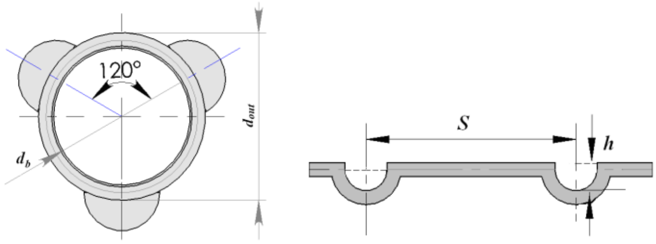


Fig.1. *Geometric model with discretely rough surface*

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°C. At outlet: pressure P_{outlet}=101235 Pa. The channel walls have physical properties of aluminum and are heated up to the temperature t_{wall}=105°C. Fluid flow is turbulent (Fig. 2.)

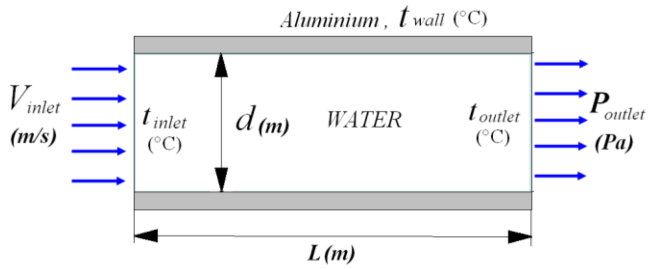


Fig. 2. *Boundary conditions*

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 ε in accordance with the known k - ε 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:

$$\frac{\partial \rho u_i}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_i u_j - \tau_{ij}) + \frac{\partial P}{\partial x_i} = F_i \qquad (1)$$

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j) = 0; \qquad (2)$$

$$\frac{\partial (\rho E)}{\partial t} + \frac{\partial}{\partial x_i}((\rho E + P)u_i + q_i - \tau_{ij}u_j) = F_i u_i + Q_H\,; \qquad (3)$$

$$\frac{\partial \rho k}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i k) = \frac{\partial}{\partial x_i}((\mu_l + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_i}) + S_k\,; \qquad (4)$$

$$\frac{\partial \rho \varepsilon}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i \varepsilon) = \frac{\partial}{\partial x_i}((\mu_l + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_i}) + S_\varepsilon\,; \qquad (5)$$

$$S_k = \tau^R{}_{ij}\frac{\partial u_i}{\partial x_j} - \rho\varepsilon + \mu_t P_B\,; \quad \rho = \frac{P}{RT}\,;$$

$$q_i = -(\frac{\mu_l}{\Pr} + \frac{\mu_t}{\sigma_c})c_P \frac{\partial T}{\partial x_i}\,; \qquad (6)$$

$$S_\varepsilon = C_{\varepsilon 1}\frac{\varepsilon}{k}(f_1 \tau^R{}_{ij}\frac{\partial u_i}{\partial x_j} + \mu_t C_B P_B) - C_{\varepsilon 2} f_2 \frac{\rho \varepsilon^2}{k}\,;$$

$$P_B = -\frac{g_i}{\sigma_B}\frac{1}{\rho}\frac{\partial \rho}{\partial x_i}\,; \qquad (7)$$

$$\tau_{ij} = \mu(\frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\frac{\partial u_l}{\partial x_l}\delta_{ij}) - \frac{2}{3}\rho k \delta_{ij}\,; \quad \mu = \mu_l + \mu_t\,; \qquad (8)$$

$$E = h + \frac{u^2}{2}\,;$$

$$\tau^R{}_{ij} = \mu_t(\frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\frac{\partial u_l}{\partial x_l}\delta_{ij}) - \frac{2}{3}\rho k \delta_{ij}\,; \qquad (9)$$

$$\mu_t = f_\mu \frac{C_\mu \rho k^2}{\varepsilon}; \;\; f_\mu = [1 - \exp(-0.025\frac{\rho\sqrt{k}\,y}{\mu_l})]^2 \cdot (1 + \frac{20{,}5\mu_l \varepsilon}{\rho k^2}), \qquad (10)$$

where u, P, ρ, T – velocity, pressure, density and temperature of fluid, R – gas constant, t - time, F_i – total force acting on the mass unit, E – total energy of fluid mass unit, Q_H – heat source per volume unit, q_i – diffusion heat flux, δ_{ij} – Kronecker symbol, τ_{ij} – viscous shear stress tensor, $\tau^R{}_{ij} \equiv -\overline{\rho u_i u_j}$ – stress tensor in Reynolds model, μ_l – dynamic viscosity coefficient, μ_t – turbulent viscosity coefficient, y – distance from solid wall, g_i – components of gravitational acceleration in direction x_i; $\sigma_c, \sigma_B, \sigma_k, \sigma_\varepsilon, C_B, C_\mu, C_{\varepsilon 1}, C_{\varepsilon 2}$ – empirical constants, c_P - specific thermal capacity at constant pressure, λ - thermal conductivity coefficient of fluid, $Pr = \mu c_p / \lambda$ - Prandtl number, parameters k, μ_t, ε are equal to zero for laminar flow; summing is made by subscripts $i = x, y, z;\ j = x, y, z$.

The equations (1-10) written out are rather general. When solving individual problems in the future, the values of constants are detailed as well as the dependent and independent variables.

For the numerical solution of the problem, the initial system of nonstationary Navier-Stokes equations with complementary equations describing the turbulent transport is discretized in both space in the computational area and time. As a result, the entire computational area is covered by a computational grid, the size and number of which cells are determined by the user or automatically. To discretize the differential equations and solve the resulting system of algebraic equations in the FlowSimulation program, the finite volume method is used. Depending on the type of problem, the satisfactory accuracy of the solution results required about 1,000,000 to 1,200,000 liquid and solid elements in this work.

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 $Pr = \mu C_p / \lambda$, were C_P – specific thermal capacity at constant temperature, λ - thermal conductivity coefficient of fluid, μ – 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]:

$$\mathrm{Re}_{1,2} = \frac{\rho_{1,2}\cdot V_0 \cdot d}{\upsilon_{1,2}} \qquad (11)$$

where Re - Reynolds number, d – equivalent pipe diameter (m), V_0 – mean velocity of fluid flow (m/s), $\mu_{1,2}$ – dynamic viscosity coefficient (Pa*s), $\rho_{1,2}$ – flow density (kg/m^3), 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_{inlet}= 0,3-1 m/s the Reynolds numbers Re_2 in a discretely rough pipe in relation to a smooth one grow approximately by 10-20%. Dependence of Reynolds numbers Re_1, Re_2 on the mean flow velocities V_{inlet} m/s for the examined channels is shown on Fig. 4.

Variation of number Re_2 value at the examined points is associated with the increase of temperature T_2°C in a discretely rough channel in comparison with temperature T_1°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_{inlet} Was also taken into consideration. Dependence diagrams of outlet temperature T1, T_2 versus mean flow velocity in a channel V_{inlet} are shown on Fig. 3.

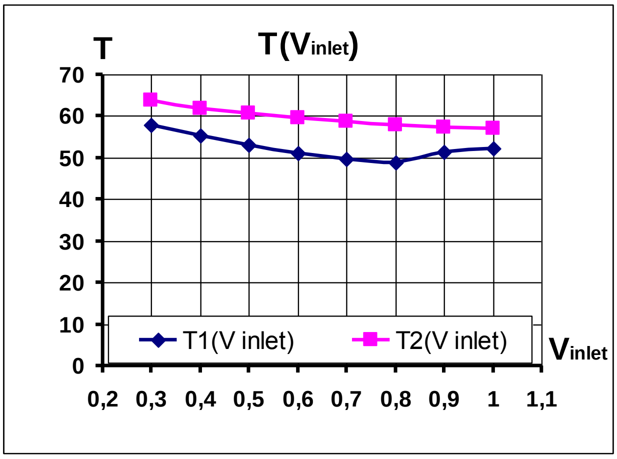


Fig. 3. *Dependence diagrams of temperature T°C on mean velocity V_{inlet} m/s of flows T_1 (V_{inlet}) u T_2(V_{inlet}).*

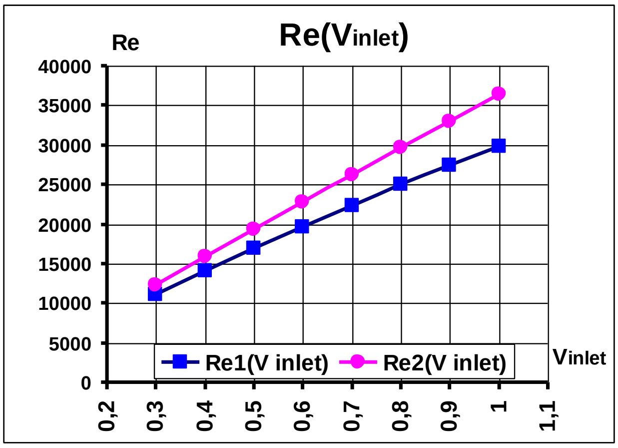


Fig. 4. *Dependence diagram of Reynolds numbers Re_1, Re_2 on mean flow velocities V_{inlet} m/s for smooth and discretely rough tube.*

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).

$$Nu_{1,2} = 0{,}021\cdot \mathrm{Re}_{water}^{0,8}\cdot \mathrm{Pr}_{water}^{0,43}\cdot\left(\frac{\mathrm{Pr}_{water}}{\mathrm{Pr}_{wall}}\right)^{0,25}\cdot\varepsilon_{1,2} \qquad (12)$$

where, Nu_1 – heat transfer coefficient for a smooth channel, Nu_2 – for a discretely rough channel, Pr – Prandtl number.

ε_1 – correction for the initial flow section for a smooth pipe, of which value is equal to 1 at L/d >50.

ε_2 – correction taking into account the growth in the heat transfer coefficient as a consequence of artificial roughness:

$$\varepsilon_2 = 1{,}04\cdot \mathrm{Pr}_{water}^{0,04}\cdot \exp\left[0.85\cdot f\,\frac{(s/h)_{opt}}{s/h}\right] \qquad (13)$$

where $\left(\frac{s}{h}\right) \le \left(\frac{s}{h}\right)_{opt}$ $\qquad f\left(\frac{s}{h}\right) = \left(\frac{s/h}{(s/h)_{opt}}\right)$

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

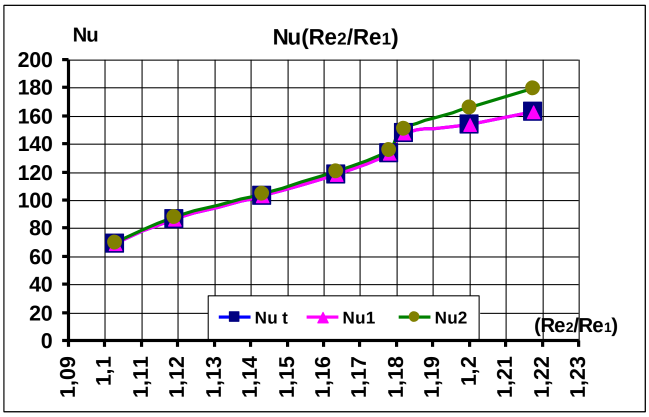


Fig. 5. *Dependence of Nusselt number Nu on relative Reynolds number without correction ε. Nu_t – theory, Nu_1 – numerical calculation (smooth channel), Nu_2 – numerical calculation (discretely rough channel).*

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 Re_2/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_2.

Dependencies of numbers Nu on the relative Reynolds number Re_2/Re_1 for the examined channels are shown on Fig. 5.

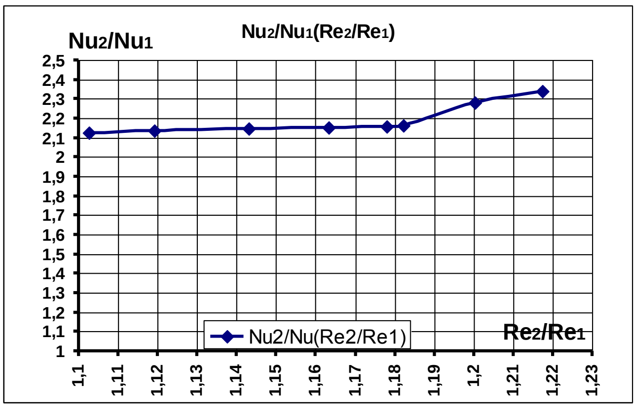


Fig. 6. *Dependence of relative numbers Nu_2/Nu_1 on Reynolds number Re_2/Re_1.*

The result obtained with consideration of correction for artificial roughness ε_2 (13) has demonstrated that the heat transfer coefficient Nu increases in relation to the smooth pipe approximately by two times. That is, the intensity of heat transfer is significantly influenced by the channel geometry and not by physical properties of the fluid. The obtained results of numerical calculations do not contradict the theoretical studies. [10, 11].

Dependence diagram of relative value Nu_2/Nu_1 versus Re_2/Re_1 with consideration of correction ε_2 is shown on Figure 6.

The study further examines the impact of the resistance coefficient $\xi_{1,2}$ in the channels (14):

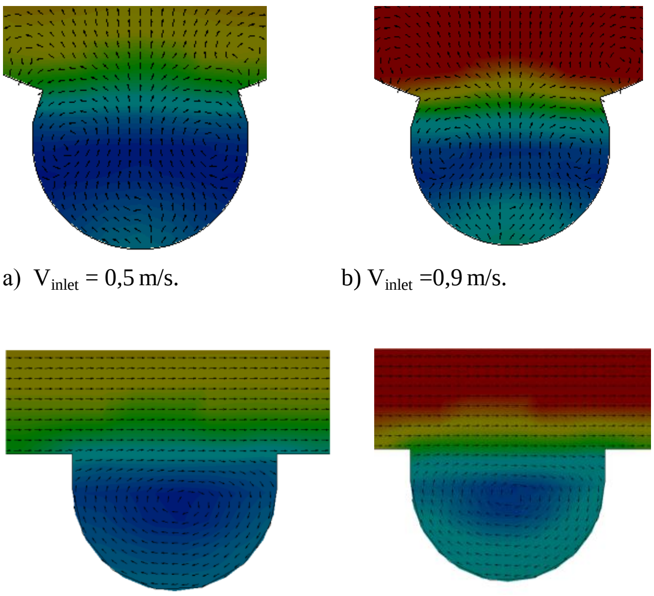
$$\xi_{1,2} = \frac{1}{(1{,}82 \cdot \lg Re_{1,2} - 1{,}64)^2} \qquad (14)$$

The results of the resistance coefficient calculation have shown that ξ_2/ξ_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.

$$\frac{\xi_2}{\xi_1} < \varepsilon_2^{\;3{,}44} \qquad (15)$$

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-scaly 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 hydrodynamics 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_{inlet} = 0.5 m/s, b) V_{inlet} = 0.9 m/s; in the cross section c) V_{inlet} = 0.5 m/s, d) V_{inlet} = 0.9 m/s.



a) V_{inlet} = 0,5 m/s. b) V_{inlet} =0,9 m/s.

d) V_{inlet} =0,9 m/s. c) V_{inlet} =0,5 m/s.

Fig. 7. *Pictures of flow-around of the dimples in the cross section: a) V_{inlet} = 0.5 m/s, b) V_{inlet} = 0.9 m/s; in the cross section: c) V_{inlet} = 0,5 m/s, d) V_{inlet} = 0,9 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

[1] Халатов А.А. Теплообмен и гидродинамика около поверхностных углублений (лунок) – Киев: НАН Украины, Институт технической теплофизики, 2005. -59 с.
[2] Mahmood G. I., Hill M. L., Nelson D. L., Ligrani P. M., Moon H.-K., Glezer B. Local heat transfer and flow structure on and above a dimpled surface in a channel. Transactions of the ASME. Journal of Turbomachinery, 2001,vol. 123, iss. 1, pp. 115 - 123. DOI: 10.1115/1.1333694
[3] Mahmood G. I., Ligrani P. M. Heat transfer in a dimpled channel: Combined influences of aspect ratio, temperature ratio, Reynolds number, and flow Structure. Intern. J. of Heat and Mass Transfer, 2002, vol. 45, iss. 10, pp. 2011-2020.
[4]. Kiselev N.A., Burtsev S.A., Strongin M.M. A procedure for determining the heat transfer coefficients of surfaces with regular relief. Measurement Techniques, 2015, vol. 58, no. 9, pp. 1016-1022. DOI: 10.1007/s11018-015-0835-7
[5]. Leontiev A.I., Dilevskaya E.V., Vinogradov Yu.A., Yermolaev I.K., Strongin M.M., Bednov S.M., Golikov A.N. Effect of vortex flows at surface with hollow-type relief on heat transfer coefficients and equilibrium temperature in supersonic flow. Experimental Thermal and Fluid Science, 2002, vol. 26, iss. 5, pp. 487-497. DOI: 10.1016/S0894-1777(02)00157-7
[6] А.Г. Лаптев, Н.А. Николаев, М.М. Башаров Методы интенсификации и моделирования тепломассообменных процесов.Справочное пособие.- М.: «Теплотехник», 2011 335 с.
[7] Д.Н. Попов, С.С. Панаиотти, М.В. Рябинин Гидромеханика, Москва, Издательство МГТУ им.Н.Э. Баумана, 2002, 383 стр.
[8] Chang P. Separation of Flow //New York, Pergamon Press, 1971.
[9] Алямовский А.А. и др. SolidWorks 2007/2008. Компьютерное моделирование в инженерной практике.// БХВ– Петербург, Санкт – Петербург, 2008.
[10] Гомелуари В.И. Влияние искуственной шероховатости на конвективный теплообмен. «Труды Института физики АН ГССР», Тбилиси, 1963, т.9,с 111-145.
[11] Михеев М.А. Михеева И.М. Основы теплопередачи. – М.: Энергия.1977. 344с.

This work has been supported by the European Regional Development Fund within the Activity 1.1.1.2 "Post-doctoral Research Aid" of the Specific Aid Objective 1.1.1 "To increase the research and innovative capacity of scientific institutions of Latvia and the ability to attract external financing, investing in human resources and infrastructure" of the Operational Programme "Growth and Employment" (No Nr.1.1.1.2./VIAA/1/16/093).