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ARTICLE

Theory of Heat Exchange in Pipes With Turbulators With d/D = 0.95 ÷ 0.90 And t/D = 0.25 ÷ 1.00, and Also in Rough Pipes, by Air With Great Reynold's Numbers Re = 106

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ABSTRACT

Mathematical modeling of heat exchange in air in pipes with turbulators with d / D = 0.95 ÷ 0.90 and t / D = 0.25 ÷ 1.00, as well as in rough pipes, with large Reynolds numbers (Re = 10^6). The solution of the heat exchange problem for semicircular cross-section flow turbulizers based on multi-block computing technologies based on the factorized Reynolds equations (closed using the Menter shear stress transfer model) and the energy equation (on multi-scale intersecting structured grids) was considered. This method was previously successfully applied and verified by experiment in [1-4] for lower Reynolds numbers. The article continues the computational studies initiated in [1-4,25-27].

1. Introduction

A known and very well tested in practice vortex method of heat transfer enhancement is the application of periodic protrusions on the wall surfaces lapped by [5] (Figure 1).

Investigation of the structure of an intensified flow mainly carried out by experimental methods [5,6], while the current design work on this subject are relatively few [1-4] and only partially devoted directly to the structure of an intensified flow; Some of the techniques (eg., a certain part of [4,7-9]) is used only integrated approaches to this problem.

In recent years intensively developing multi-block computational technology for solving the vortex aerodynamics and thermal physics, based on intersecting structured grids.

This work is devoted to the study directly heat at high Reynolds numbers in the tubes intensified periodically disposed surface turbulence semicircular cross section, since in this range there are no sufficiently reliable experimental data; for comparison the corresponding theoretical data for rough tubes [15-19,24,25-27].

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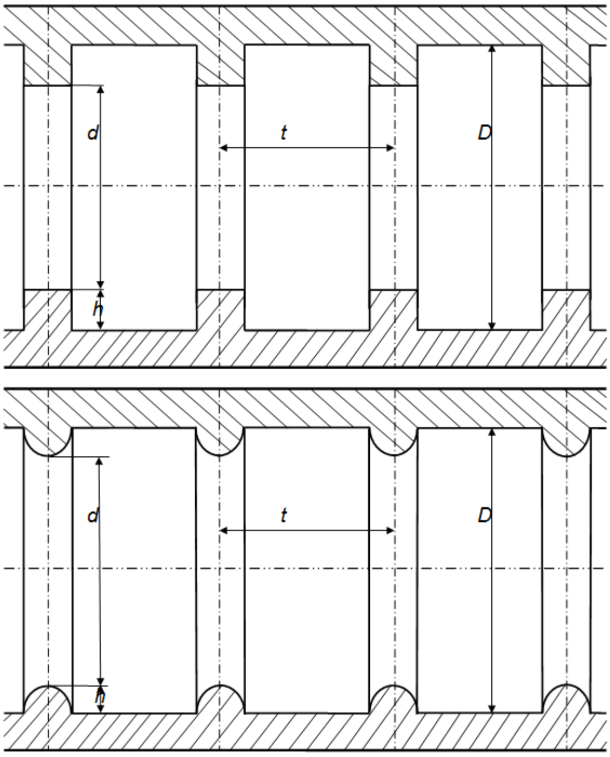

Figure 1. Cut a straight circular tube with a transverse surface located turbulators and flow square semicircular (lower panel) cross-sections

2. Perspective Directions of Development of Numerical Theoretical Study of Intensified Heat Exchange

Theoretical investigation of local flow parameters, and as averaging and heat transfer tubes with turbulators to be the most promising in the development direction based multiblock parallelized computational technologies specialized packages can be described target direction which follows:

(1) The development of multi-block original computing technology [1-4] based on different scales intersecting structured grids, for highly efficient and accurate solutions of two-dimensional and three-dimensional unsteady convective heat transfer problems in straight round pipes roughness organized in the form of protrusions in the homogeneous operating environment within a wide range Reynolds number (Re = $10^4 \div 10^6$) and Prandtl (Pr = 0,7÷12). Unlike previous packet embodiment [1-4] is that the methodology is supplemented using periodic boundary conditions, which allows to estimate asymptotic characteristics pipes with discrete roughness. Modification allowed to increase the computational efficiency of modeling, to realize the correction on the curvature of the streamlines. For tubes with turbulators are determined: surface distribution of local and integrated power and thermal characteristics (pressure, friction, heat fluxes, resistance to motion, the hydraulic losses), the profiles of the velocity, pressure, temperature and turbulence characteristics (energy of turbulence, eddy viscosity tensor components Reynolds stress generation, dissipation, and the like).

(2) The original set of differential equations of - the Navier-Stokes equations and Reynolds closed by a modified taking into account the curvature of the streamlines, according to the approach Menter, shear transfer model. Initial information about the Governing equations and appropriate boundary conditions are contained in [13]. Are used based on the periodic boundary conditions original pressure correction procedure and the weight average temperature. Methodology of solutions of the original equations - based on the concept of splitting into physical processes pressure adjustment procedure. For problems with periodic boundary conditions apply pressure gradient correction procedure and the weight average temperature. Methodical basis of long-term calculation tool - multiblock computing technologies,

There should be more focus on the specific features characteristic of the periodic boundary conditions.

Periodic boundary conditions determine more optimal mesh tube construction (Figure 2). (In all figures the upper figure to compare ceteris paribus shown turbulators square cross section, and the bottom - a semicircular). The pipe is divided in several sections arranged in the middle of the baffle and the inlet and outlet of a smooth portions (see. Figure 2).

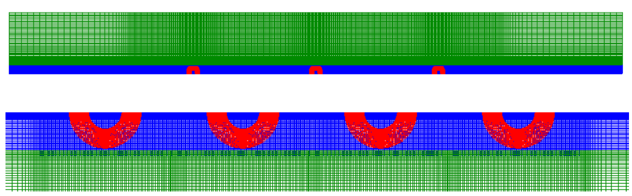

Figure 2. The mesh tube composed of several sections with a baffle located in the middle, smooth input and output portions; a periodic statement is considered only one section (semicircular baffles are shown in a larger scale)

In the periodic formulation is considered only one section, while it is generally necessary to use several sections ([1-4,7-12] reached the number of sections 12, and the same number of sections used for verification). more parietal region in the pipe is released to reduce the number of computational nodes (blue mesh) and less detailed axial (green). If this granularity changes in both the longitudinal and the circumferential directions (under application of three-dimensional case). Additionally, for three-dimensional calculation is introduced near the axis so called

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"Patch" that eliminates unnecessary mesh refinement near the axis. The latter circumstance, ceteris paribus, reduces the number of cells calculated by about half (this fact becomes even more important when three-dimensional calculation). You can even reduce the number of cells by applying the periodic conditions along the longitudinal axis, as inlet and outlet portions are eliminated and left one section.

In terms of hydrodynamics periodic task is set as the task of keeping a predetermined mass flow rate calculated for the unit at the input speed. In terms of heat exchange, depending on the selected boundary conditions for the temperature, there are two possibilities. For insulated walls problem is solved by assuming a constant average temperature in the inlet section. In a second - assumed known average temperature gradient calculated by the value of the heat flux to the walls. Naturally, the inlet temperature is not fixed. In addition to periodic full record of the current state of the problem in the program is able to perform at a specified interval sampling records with their accumulation in the file, which is especially important for use in solving time-dependent problems.

(3) The focus is on the local and integral characteristics of convective heat transfer, including the components of velocity and hydraulic losses at the selected average channel wall heat transfer area of the site, the results of calculation of the characteristics of a turbulent members equation for turbulent pulsations energy (generation, dissipation, convective and diffusive transport). For external flow rectangular protrusions similar approach has been applied, eg., In [14].

(4) The main direction of this work can be briefly described as follows: the method further verify the calculation of heat transfer in the tubes with turbulators (d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00) for extremely high Reynolds numbers which have been examined in the present experiments [5,6], the actual experimental data and theoretical data of other approaches [1-4,7-10]; and after verification of the conduct calculations for higher Reynolds numbers, where there are no reliable experimental data; computing received further compared with the corresponding values for rough pipes.

3. A brief Analysis of the Effect on Integral Flow Characteristics and Heat Transfer Tubes with Turbulators Intensified Flow Pattern for Relatively Low Reynolds Numbers Re = 104÷105

As a result of [20] Numerical calculations have been received local and integral flow and heat transfer characteristics in straight round pipes with semi-circular and square turbulators.

The value of relative coefficient of hydraulic resistanceξ/ξGLisξ/ξGL= 1.96 for a square pipe with Turborecuperators at Re = 104, d / D = 0,94, t / D = 1,00 when as averaging the relative heat exchange Nu / NuGL = 1.63 [1-4,7-10,20].

For turbulators semicircular cross section corresponding values ceteris paribus amount ξ/ξGL= 1.75 and Nu / NuGL = 1.56 [1-4,7-10,20], which is more optimal because secondary vorticity in the flow turbulators semicircular clearly smaller than square.

Further increase in the Reynolds number Re = 105 implements the following as averaging flow characteristics and heat transfer, which will be:ξ/ξGL= 4.61 and Nu / NuGL = 1.76 [1-4,7-10,20] (intermediate values of the above characteristics, an intermediate Reynolds number).

For turbulators semicircular cross section corresponding values ceteris paribus amount ξ/ξGL= 3.16 and Nu / NuGL = 1.64 [1-4,7-10,20], since the system for their return vortices much less pronounced and more deformed vortex core [1-4,7-10,20].

The hydraulic resistance values of the relative ratio are ξ/ξGL= 2.67 for square tubes with turbulators Re = 104, d / D = 0,94, t / D = 0,25 when as averaging the relative heat exchange Nu / NuGL = 1.80 [1-4,7-10,20].

For turbulators semicircular cross section corresponding values ceteris paribus amount ξ/ξGL= 2.00 and Nu / NuGL = 1.59 [1-4,7-10,20], since the reduced differences in the systems of vortex zones between the semi-circular and square turbulators [1-4,7-10,20].

The greatest relative heat exchange tubes with turbulators in the square cross-section for the given conditions occurs with t / D = 0,50 (for Re = 104, d / D = 0,94) - Nu / NuGL = 2,20 when ξ/ξGL= 3.08 [1-4,7-10,20]. For turbulators semicircular cross section corresponding values ceteris paribus amountξ/ξGL= 2.74 and Nu / NuGL = 1.87 [1-4,7-10,20] as secondary vortices to semicircular turbulators less than square.

Analysis of vortex zones between square turbulators shown that the higher turbulence at higher Reynolds numbers, a slight increase in the relative Nusselt number Nu/ NuGL accompanied by a significant increase in the relative hydraulic resistanceξ/ξGL because of the very significant impact of return flows, which may even impingement upon turbulator [1-4,7-10,20].

For turbulators semicircular cross section of impact of return vortices is smaller than that for square and there is a greater impact deforming the main vortex.

Thus, the flow resistance in the tubes with turbulators semicircular cross section smaller at other equal con-

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ditions, is less than in the tubes with turbulators square cross-section, resulting in a more optimal relationship between heat transfer and to intensify the hydraulic resistance [1-4,7- 10,20].

After vyshepredstavlennogo analysis for relatively moderate Reynolds number should go to the analysis data calculated for higher Reynolds numbers.

4. Summary Analysis of the Effect on Integral Flow Characteristics and Heat Transfer Tubes with Turbulators (d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00) Intensified Flow Structure for Large Numbers Re = Reynolds 106

Vysheprevedonny analysis of the effect on integral flow characteristics and heat transfer tubes with turbulators structure intensified flow with relatively small numbers Re = Reynolds $10^4 \div 10^5$ indicates that the best is to use turbulators semicircular cross section than rectangular.

Consequently, for higher Reynolds numbers Re = 10^6 (as well as for higher) is quite important to analyze all the flow and heat characteristics only for tubes with turbulators semicircular flow, and only an additional study should be undertaken for rectangular or square turbulators.

Paschitannye on the above procedure streamlines for Reynolds numbers Re = 10^6 with d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00 are shown in Figure 3-11.

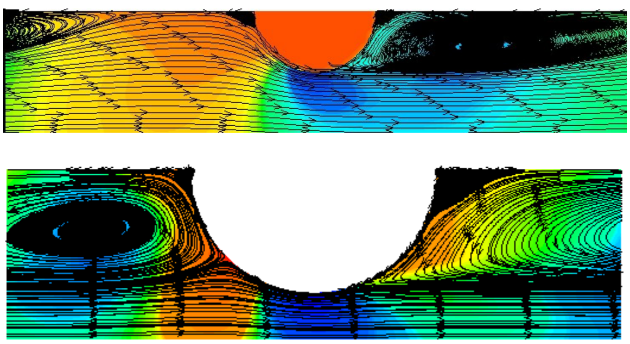


Figure 3. The flow lines for a tube with a semicircular cross-sectional turbulence at Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 0,25 for air

As shown by the calculated current line given in Figure 3-11, with increasing Reynolds number up to Re = 10^6 with d / D = 0,90 and t / D = 0,25 ÷ 1,00 turbulators on a semicircular cross-sectional growth additional corner vortices as to baffle and after turbulizer occurs in not very appreciable extent than when the conditions for Re = 10^5, resulting in not very significant increase in hydraulic losses.

For similar values at lower turbulators d / D = 0,95 corner vortices both before and after increasing turbulator qualitatively in the same manner as in the d / D = 0,90, but definitely smaller dimensions than in the case with d / D = 0,90.

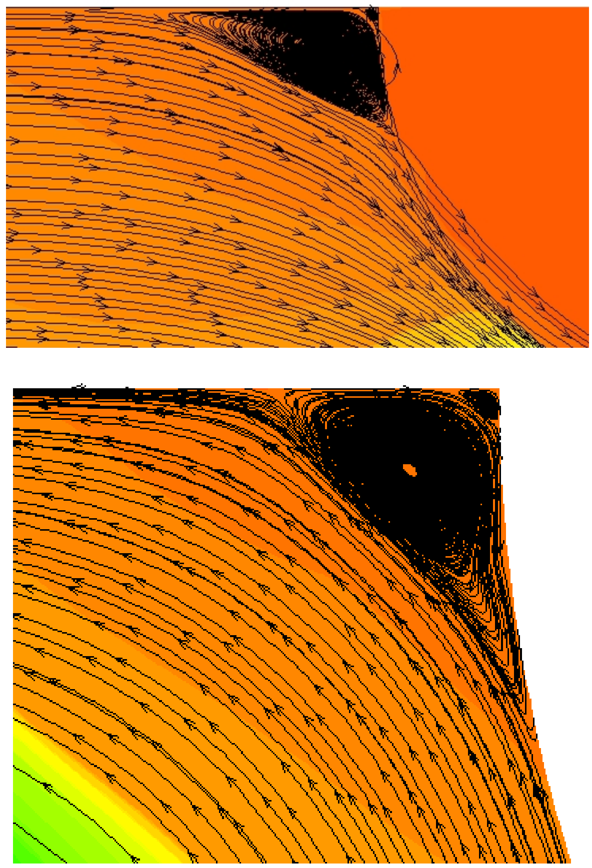
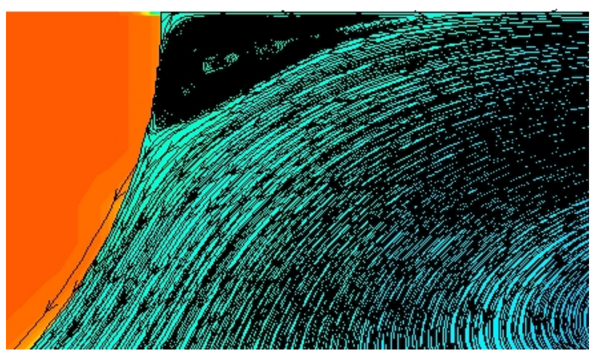


Figure 4. The flow lines of bend to vortex turbulator semicircular cross section with Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 0,25 for air, shown on a larger scale than in Figure 3

For turbulators with d / D = 0,90 a semicircular cross section with Re = 10^6 occurs further deformation and pulling the main vortex is clearly seen in Figure 3-11. The latter indicates not very significant increase of the hydraulic resistance for the semicircular turbulators with d / D = 0,90 when Re = 10^6, since under these conditions there is no generation of additional eddies and vortices of friction between them.



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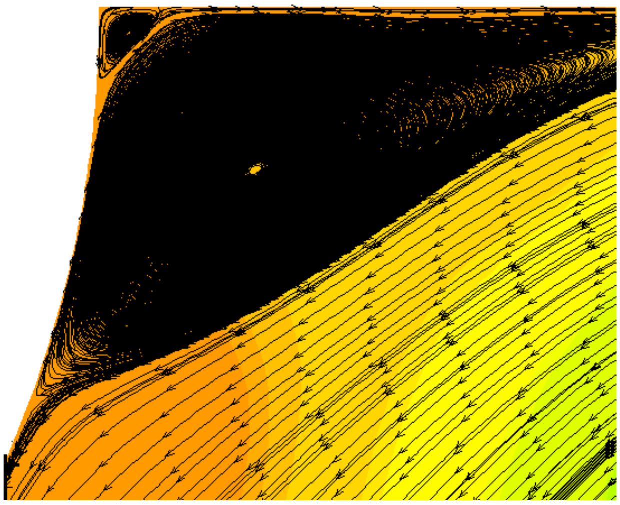


Figure 5. The flow lines for the angular vorticity of turbulizer semicircular cross section with Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 0,25 for air, shown on a larger scale than in Figure 3

For similar values at lower turbulators d / D = 0,95 aforementioned additional vortex generating even lower than when d / D = 0,90, and therefore the flow resistance in this case is even lower (Figure 3-11).

For the conditions at Re = 10+, and d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00 turbulence generation also occurs at the boundaries of vortex zones between them and the development of zones themselves decay after ejection .

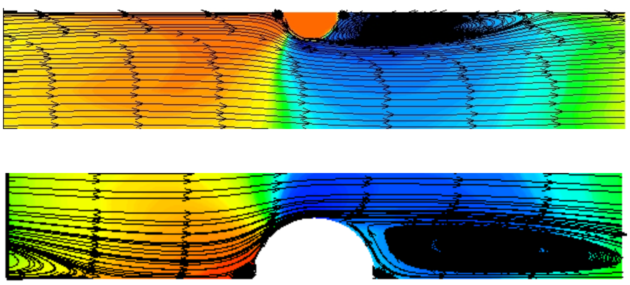
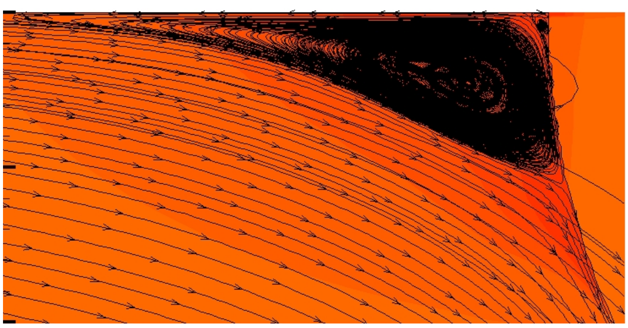


Figure 6. The flow lines for a tube with a semicircular cross-sectional turbulence at Re = 106; d / D = 0,95 and d / D = 0,90; t / D = 0,50 for air



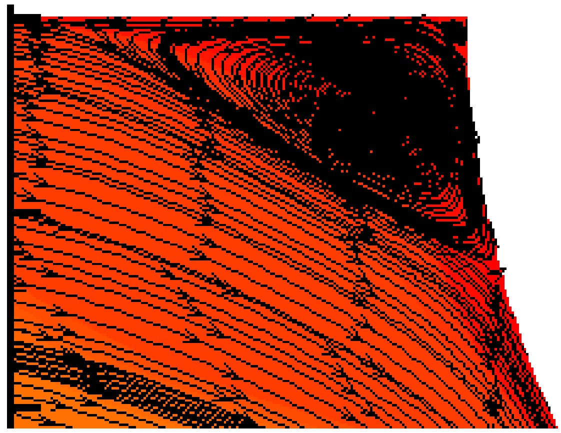
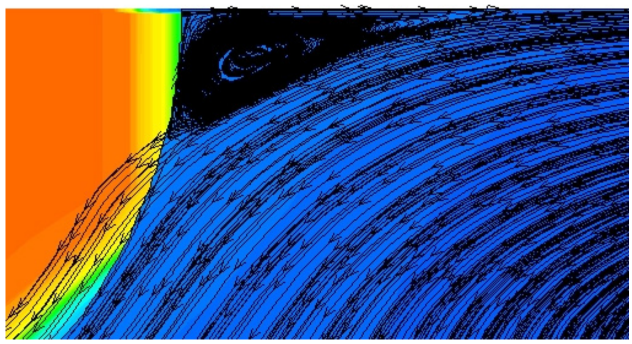


Figure 7. The flow lines of bend to vortex turbulator semicircular cross section with Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 0,50 for air, shown on a larger scale than in Figure 6



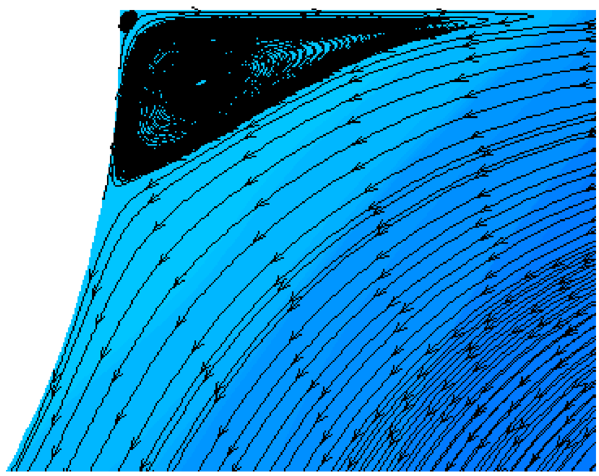
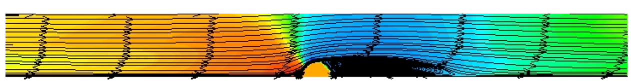


Figure 8. The streamlines bend vortex turbulator for a semicircular cross section with Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 0,50 for air, shown on a larger scale than in Figure 6



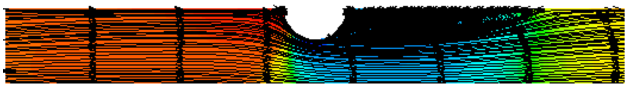


Figure 9. The flow lines for a tube with a semicircular cross-sectional turbulence at Re = 106; d / D = 0,95 and d / D = 0,90; t / D = 1,00 for air

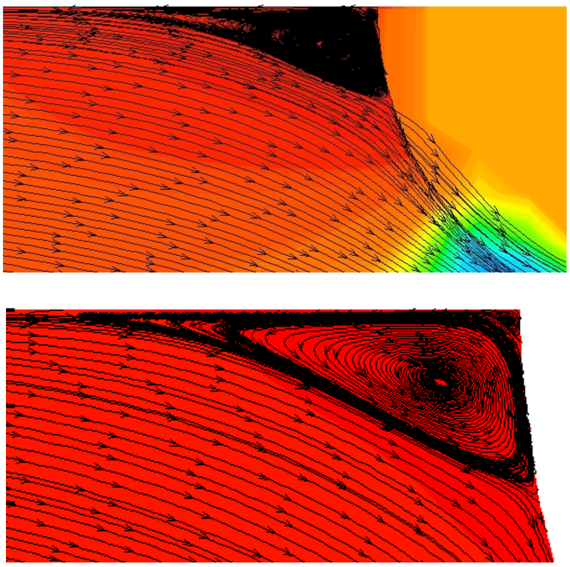
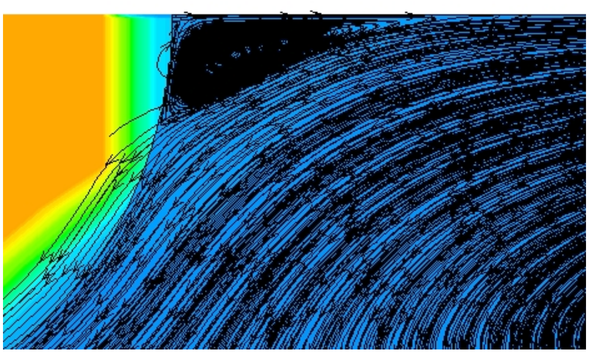


Figure 10. The streamlines bend vortex turbulator to semicircular cross-section with Re = 10+; d / D = 0,95 and d / D = 0,90; t / D = 1,00 for air, shown on a larger scale than in Figure 9

For turbulators semicircular cross section with Re = 10^6, and d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,0 also occurs no development, integration and disintegration of secondary vortices, considered in [7-10,20] and their deformation; greatest deformation undergoes a large vortex (Figure 3-11).

Last further indicates that the flow resistance at Re = 10^6, and d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,0 is not in such a great extent, if there was a system of the above secondary vortices, for example, turbulence of square cross-section. For lower turbulators with d / D = 0,95 secondary vortices is smaller than for higher with d / D = 0,95 (Figure 3-11).



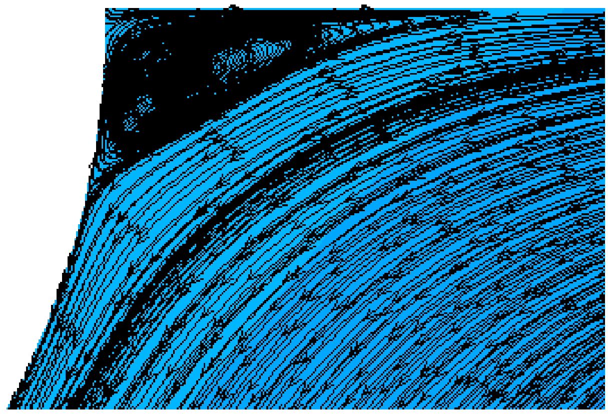


Figure 11. The streamlines bend vortex turbulator for a semicircular cross section with Re = 10^6; d / D = 0,95 and d / D = 0,90; t / D = 1,00 for air, shown on a larger scale than in Figure 9

The above analysis indicates that even at relatively high Reynolds number Re = 10^6, and d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,0 large vortex does not decompose, but only deformed, with deformation may occur both in the direction of the turbulator and in the direction of flow of the core.

Consequently, at high Reynolds numbers Re = 10^6, and d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,0 intensification of heat transfer can be increased without a very large increase in hydraulic resistance in the application of turbulators semicircular cross section, unlike vortex sharp outlines cross-sectional profile.

5. Analysis Calculated Data on the Heat Transfer in the Tubes with Turbulators (d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00) of Semicircular Cross Section for Large Numbers Re = Reynolds 10^6

Before calculating the intensified heat transfer for high Reynolds numbers, we must first analyze the correlation calculated values for heat exchange with the experimental data for the largest experimental Reynolds numbers in air (d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1 00) [5,6]. To this end, in Figure 3-11 were shown streamlines as between turbulators and for angular and vortex before and after the projections for a tube with a semicircular cross-sectional turbulence at Re = 10^6; d / D = 0,95 ÷ 0,90; t / D = 0,25; 0.50; 1.00 for air.

Calculated data for Intensified heat exchange tubes in the U-shaped turbulence in air for d / D = 0,90, t / D = 0,25 ÷ 1,00, Re = 2 · 10^5 ÷ 4 ·10^5 are compared with corresponding experimental data in Table. 1 and Table. 2; for comparison, the same shows similar data obtained on the four-layer flow model [4,7-10,21] as well as the corresponding

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data for rough tubes [15-19,24,25-27].

Table 1. Calculated data relative to air heat exchange N u / NuGL for round tubes with turbulators calculated from the theory developed in, for high Reynolds numbers Re = 106 with d / D = 0,90 and t / D = 0,25÷1.00; and comparative analysis of respective calculated values with the experimental data [5-6] for Re = 2-105÷4-105, and the data obtained by four-layer model of the turbulent boundary layer [6-10,21], and appropriate values for rough tubes obtained by the superposition theory of turbulent viscosity [15-19,24]

d / D	***t*/ D**	**Pr**		**Re**		
				2-10^5	**4-10^5**	**10^6**
0.90	0.25	0.72	Orifice tube; experiment [5-6]	2.88	3.08	-
0.90	0.50	0.72	Orifice tube; experiment [5-6]	2.77	2.92	-
0.90	1.00	0.72	Orifice tube; experiment [5-6]	2.40	2.47	-
0.90	0.25	0.72	Orifice tube; theory [6-10,21] (4-layer model)	2.87	3.12	3.58
0.90	0.50	0.72	Orifice tube; theory [6-10,21] (4-layer model)	2.74	2.98	3.32
0.90	1.00	0.72	Orifice tube; theory [6-10,21] (4-layer model)	2.36	2.43	2.72
0.90	0.25	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	2.90	3.13	3.60
0.90	0.50	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	2.70	2.93	3.29
0.90	1.00	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	2.44	2.46	2.77
0.90	-	0.72	rough pipe; theory [15-19,24] (Superposition viscosity)	2.48	2.75	3.19

As seen from Table. Table 1. 2, the calculated data on heat transfer to the air in the tubes with turbulators semicircular cross-section obtained by the generated in this paper the theory in very good agreement with the existing experiment for the maximum Reynolds numbers for the latter (Re = $4 \cdot 10^5$ [5,6]) and also for the somewhat lower Reynolds numbers (Re = $2 \cdot 10^5$ [5,6]).

Moreover, the data obtained by the proposed theory, and in good agreement with theoretical data obtained from an independent four-layer model of the turbulent boundary layer [4,7-10,21]; however, only as averaging the heat exchange, while for Low- model data allow to calculate and local heat.

Thus, in this study developed theoretical Low- method can be considered for the largest verified experimentally investigated in air Reynolds numbers Re = $4 \cdot 10^5$ [5,6] for d / D = 0,90, t / D = 0,25 ÷ 1 00, which it justifies its use and for higher Reynolds numbers the data pipe dimensions.

Table 2. Calculated data relative to air heat exchange Nu / NuGL for round tubes with turbulators calculated from the theory developed in, for high Reynolds numbers Re = 10^6 with d / D = 0,95 and t / D = 0,25÷1.00; and comparative analysis of respective calculated values with the experimental data [5-6] for Re = $2\text{-}10^5 \div 4\text{-}10^5$, and the data obtained by four-layer model of the turbulent boundary layer [6-10,21], and appropriate values for rough tubes obtained by the superposition theory of turbulent viscosity [15-19,24]

d / D	***t*/ D**	**Pr**		Re		
				2-10^5	4-10^5	**10^6**
0.95	0.25	0.72	Orifice tube; experiment [5-6]	2.37	2.45	-
0.95	0.50	0.72	Orifice tube; experiment [5-6]	2.24	2.28	-
0.95	1.00	0.72	Orifice tube; experiment [5-6]	1.82	1.75	-
0.95	0.25	0.72	Orifice tube; theory [6-10,21] (4-layer model)	2.38	2.51	2.66
0.95	0.50	0.72	Orifice tube; theory [6-10,21] (4-layer model)	2.23	2.31	2.35
0.95	1.00	0.72	Orifice tube; theory [6-10,21] (4-layer model)	1.84	1.81	1.80
0.95	0.25	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	2.39	2.48	2.65
0.95	0.50	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	2.19	2.29	2.37
0.95	1.00	0.72	Orifice tube; theory [1-3,11,12,20] (Menter model)	1.84	1.79	1.77
0.95	-	0.72	rough pipe; theory [15-19,24] (Superposition viscosity)	2.09	2.31	2.66

As shown by the calculated data Intensified heat exchange tubes in the U-shaped turbulence in air for d / D = 0,90, t / D = 0,25 ÷ 1,00, Re = 10^6 are presented in Table. 1, the relative heat exchange Nu / NuGL increases even more compared with smaller values of the Reynolds number, which is naturally accompanied More greater increase hydraulic resistance.

As shown by the calculated data Intensified heat exchange tubes in the U-shaped turbulence in air for d / D = 0,95, t / D = 0,25 ÷ 1,00, Re = 10^6 are presented in Table. 2, the relative heat exchange while increasing Nu / NuGL increases the Reynolds number as compared with smaller values of the Reynolds number much lower than at higher

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turbulizer with d / D = 0,95 and t / D = 0.25 and t / D = 0, 50, and does not occur when d / D = 0,95 and t / D = 1,00 a relative increase heat transfer. Unlike similar cases with d / D = 0,90 heat rise when d / D = 0,95 accompanied by a much smaller increase in the hydraulic resistance that is caused by a decrease in the generation of additional vortex formation in the latter case (see. **Figure 3-11**).

Thus (See. Table. 1 and Table. 2), an intensification of heat at high Reynolds numbers (of the order of Re = 10^6) may be even higher than for lower Reynolds numbers (of the order of Re = $4 \cdot 10^5$) for a relatively high turbulence flow (about d / D = 0,90), but this requires significantly increase gidropoteri. For lower turbulators (about d / D = 0,95) intensification of heat transfer at high Reynolds numbers (of the order of Re = 10^6) not always higher (e.g., for larger steps between turbulators (about t / D = 1,00)), than for lower Reynolds numbers (Re = the order of $4 \cdot 10^5$), but it is achieved for small and medium-sized steps between turbulators (about t / D = 0,25 and t / D = 0,50) at least tangible gidropoteryah,

Terms with high Reynolds numbers in the channels with moderate flow rates are realized when modes with lower values of the kinematic viscosity. For example, the air kinematic viscosity appreciable reduction will take place at high pressures [22,23], therefore investigated the flow regimes with high Reynolds numbers can be considered relevant.

Obtained by Intensified nizkoreynolsovoy model for heat transfer in data pipes with turbulence correspond to physical representations of implemented processes [5,6].

Independent verification data Low- Menter model for d / D = 0.95 ÷ 0.90, t / D = 0.25 ÷ 1.00, Re = $2 \cdot 10^5 \div 4 \cdot 10^5$ air may also serve similar data obtained four-layer model of the turbulent boundary layer [4,7-10,21] (Table. 1 and Table. 2), which gives similar results, but the multilayer model proved less (although it has a greater range of application) than Low- model.

As the analysis presented for comparison of the heat transfer data for rough tubes (Table. 1 and Table. 2) for the high Reynolds numbers Re = 106, the relative heat transfer in rough pipes is close to the relative heat transfer in the tubes with turbulators with t / D = 0,50 when d / D = 0,90 and t / D = 0,25 when d / D = 0,95.

Previously, in [15-19,24,25-27], it was found that as the Reynolds number relative heat exchange in rough pipes is close to the relative heat transfer in the tubes with turbulators with lower relative pitch between t / D turbulators. Consequently, even when the Reynolds number increases bólshem up to Re = 10^6, this tendency is maintained, which is confirmed by the data of Table. 1 and Table. 2 the air flow conditions in the tubes with turbulators with d / D = 0,95 ÷ 0,90, t / D = 0,25 ÷ 1,00.

For additional verification of the data by Intensified heat transfer in the tubes with turbulators for high Reynolds numbers Re = 10^6 obtained by operation generated in this method, similar calculations were performed by the method used previously in [1-4,7-12] .

As shown by calculations for the heat exchange sections 12, turbulence of the method [1-4,7-12], the difference between it and generated in this work method is in the order (3 ÷ 4)%, but the new method converges more quickly to two orders of magnitude with increasing time accuracy of the main parameters to 10-4 for the method [1-4,7-12] to 10-5 for this method.

The above demonstrates the reduction method [1-4,7-12] with respect to the method developed in this research study.

Conducted in this work successful modeling of heat exchange in the air in the tubes with turbulators with d / D = 0,95 ÷ 0,90, t / D = 0,25 ÷ 1,00 Low-Menter based model at high Reynolds numbers up to Re = 10^6 determines a perspective modeling heat transfer in the tubes with turbulators this method and at higher Reynolds numbers.

6. Conclusions

The article was held in the mathematical modeling of the heat transfer tubes with turbulators with d / D = 0,95 ÷ 0,90 and t / D = 0,25 ÷ 1,00 semicircular cross section at high Reynolds numbers (Re = 10^6) based multiblock computing technologies based on the decision of factorized finite volumetric method of Reynolds equations and the energy equation.

B Articles it was found that the intensification of heat transfer in air for high Reynolds numbers (Re = 106), which may be relevant in the channels may be higher when tangible increase hydraulic resistance than a smaller number (Re = $4 \cdot 10^5$) for relatively high flow turbulators d / D = 0,90 for the whole considered range of the relative pitches between t / D = 0,25 ÷ 1,00, exceeding the values for rough tubes; while at lower turbulators with d / D = 0,95 occurs a certain increase in the relative heat transfer for large Reynolds numbers (Re = 10^6) compared to the smaller numbers (Re = 4×10^5) only at small distance between turbulators with t / D = 0 25, approaching the values for rough pipes, while increasing relative pitch between turbulators (i.e., when t / D = 0,25 ÷ 1,00) increase above substantially does not occur.

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