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Research Article

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Theory of Heat Transfer in Straight Round Pipes with Square and Triangular Turbulators Under High Reynolds Criteria

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Abstract

Objectives: *To carry out mathematical modeling of the structure of vortex zones between periodic flow turbulators with a surface arrangement of triangular and square transverse profiles on the basis of multi-block computing technologies based on solutions of the Reynolds equations (closed by means of the Menter shear stress transfer model) and energy equations (on multi-scale intersecting structured grids) with high Reynolds criteria Re = 106 with an exhaustive analysis of the corresponding current lines.*

Method: *The calculations were carried out on the basis of a theoretical method based on the solution of the Reynolds equations by the factorized finite-volume method, which are closed using the low-Reynolds model of the Menter shear stress transfer, and the energy equation on multi -scale intersecting structured grids (FCOM).*

Result: *Mathematical simulations of the heat exchange process in straight and round pipes with turbulators with d / D = 0.95 ... 0.90 and t / D = 0.25 ... 1.00 square and triangular cross-sections at large Reynolds numbers (Re = 106) on a foundation with multi-block computing technologies, which are based on solutions of the Reynolds equations and energy equations in a finite-volume and factorized way. It is found that the relative intensification of heat transfer [(Nu / Nusm) | Re = 106] / [(Nu / Nusm) | Re = 105] in round pipes with square air turbulators for large Reynolds numbers (Re = 106), which may well be relevant in the channels used in heat exchangers, may be higher with a large-scale increment of hydraulic resistance than for slightly smaller numbers (Re = 105), for relatively high flow turbulators d / D = 0. 90 for the entire range under consideration for the parameter of the relative step between them t / D = 0.25 ... 1.00 a little more than 3%; for turbulators of triangular cross-section, similar indicators are approximately the same. For lower square turbulators with d / D = 0.95, this increase in relative heat transfer for large Reynolds numbers (Re = 106) compared to smaller numbers (Re = 105) does not exceed 6%; for triangular cross-section turbulators, similar indicators are slightly more than 4%.*

Conclusion: *According to the results of calculations based on the developed model, it is possible to optimize the intensification of double turbulators, as well as to control the process of heat transfer intensification. It is shown that for higher square turbulators and at higher Reynolds numbers, a slight increase in the relative Nusselt number Nu / Nusm is accompanied by a significant increase in the relative hydraulic resistance due to the very significant influence of return currents, which can flow directly on the turbulator itself to the greater extent, the higher the Reynolds number; for triangular turbulators, the above trend persists and even deepens.*

Keywords: Theoretical, Mathematical, Modeling, Turbulator, Pipe, Cross-Section, Triangular, Square, Semicircular, Diaphragm, Reynolds Criterion, Coolant, Menter Model

Introductory Part: Relevance of the Study of Heat Transfer Intensification

A well-known and quite practically tested method of tornado (vortex) intensification of heat transfer consists in the application of a system of turbulators on the walls of the washing channel surface with a periodic arrangement [1]. The study of the structures of flows with intensified flow, as a rule, was done experimentally, however, existing new computational studies in this direction were not very numerous, but only partially devoted directly to the study of the structure of intensified flows; a certain part of these methods (for example, partially works) apply exclusively averaged (integral) methods to the above problem. Recently, multi-block computing technologies have been intensively developed for solving the problem of vortex thermophysics and aeromechanics, based on an overlapping structured mesh [1-9]. This article is directly intended to study the structures of flows with an intensified flow in pipes with turbulators; heat transfer intensification is carried out at the expense of surface periodic turbulators of triangular and square profiles for high values (Re = 106) of Reynolds criteria, for which reliable experimental data have not been obtained at this stage.

Progressive Trends in Theoretical Research of Intensified Heat Exchange

Theoretically Scientific research for local (local) and integral (averaged) characteristics of flows and heat transfer in channels by turbulators of triangular and square cross-sections seem to be mainly promising in areas of development based on multi-unit computing technologies with specialized parallelized packages, the specialized directions of which can be summarized as follows way.

1. In the process of development of a specific multi-block computing technology, which is based on a multi-scale-intersecting structured grid set with the aim of highly efficient and refined solutions for a non-stationary 2-dimensional and for a non-stationary 3-dimensional problem with respect to convective heat transfer in round straight pipe with organized artificial roughness in the form of turbulators in homogeneous working media in wide enough ranges for the Reynolds criterion (Re = 104 ... 106) and for the Prandtl criterion (Pr = 0.7 ... 12) [3-6].

A distinctive feature of the existing versions of the packages should be recognized that the methodological basis was supplemented by the use of periodic boundary conditions, which made it possible to estimate the asymptotic indicators of intensified flow and heat transfer in pipes with discrete roughness [3-6].

Modification of the model allows you to increase the efficiency of computational operations for modeling, to carry out curvature correction on the current line. For a channel with protrusions, the following are subject to determination: the distribution of surface local and averaged thermal and power parameters (pressure, friction, heat flow, resistance to flow, hydraulic losses), profiles of velocity components, pressure, temperature, parameters describing flow turbulence (turbulence energy, turbulent viscosity, tensor components for Reynolds tension, dissipation, generation, etc.).

2. An initial system with differential equations in partial derivatives, i.e. the Reynolds, Navier-Stokes equations, are closed by specific modifications in terms of taking into account the curvatures of the current lines thanks to Menter's model for shear stress transfers. Relevant data about the governing equations, optimal boundary conditions can be found in [10].

The procedures for pressure corrections and for mass-average temperature corrections are applied, which are based on the periodicity of the original boundary conditions.

The methodologies for making decisions about fundamental equations are based on a pressure correction procedure that is based on a schematic decomposition into different physical processes.

The methodological foundations of promising computational tools is a multi-unit computational technology, which is based on the use of an intersecting multi-scale structured grid associated with the perception of specific elements of the structures of tornado (vortex) currents and distributions of temperature fields, which provides the necessary accuracy and increased efficiency, which is comparable to the application. n. adaptive grid.

3. Similar numerous studies of intensified heat transfer in a pipe and a channel with turbulators for smaller Reynolds criteria were carried out in [11-47].
4. Primary attention should be paid to both local and averaged parameters related to convective heat transfer, including the components (profiles) of velocities, losses for pumping the coolant, heat transfer averaged over the selected areas of the channel wall sections; calculated results with respect to turbulent characteristics in terms in the equations for pulsating turbulent energy (convective transport, generation, diffusion transport, dissipation,). In systems with external (external) flow with rectangular (square) protrusions, a similar method was previously used, for example, in the study [48].
5. The main direction for this article should be briefly named as follows: based on the analysis of relatively vortex zones in channel systems with triangular and square turbulators, reveal the level of relative intensification of heat transfer for increased (Re = 106) Reynolds criteria.

Statement of the Problem of Scientific Theoretical Research

It should be emphasized that the structures of turbulent flows in pipes, where it becomes necessary to intensify heat transfer, have been studied to a large extent both experimentally and theoretically, which led to the maximum increase in turbulent pulsation intensity for certain areas of flows, for which this gives the most effective intensification.

For the purpose of optimal application of the separation zone, it becomes necessary to know the mechanism of its interactions with the main turbulent flows, as well as the mechanisms of the processes directly in the separation zones, which are quite complex. Qualitatively, on the basis of experimental material, these processes were revealed to the following extent: there is a possibility of purposeful use of vortex zones in order to intensify heat transfer in channels [1, 2].

The main goal of this research study is to theoretically investigate the aforementioned vortex zones for a pipe with square and triangular ridges using factorized finite volume methods (FKOM-s), which have been successfully tested in order to calculate similar flows in studies, in which the main attention was paid to the calculations of the averaged characteristics of intensified fluxes and heat transfer [3-6].

The article investigates predominantly interesting cases for the use of periodically and superficially located protrusions of square and triangular profile sections in round and straight pipes, i.e. object of research; specifically: t / D = 0.25 ... 1.00; d / D = 0.94 ... 0.90; Re = 105 ... 106; Pr = 0.72; (d and D - smaller and larger internal diameters of pipes with protrusions, respectively; t - steps between protrusions) [1, 2].

The cross-sections and design grid for straight and round straight pipes with protrusions (ribs) with surface and transverse installation for flows with square, triangular and semicircular cross-sections are shown in Fig. 1.

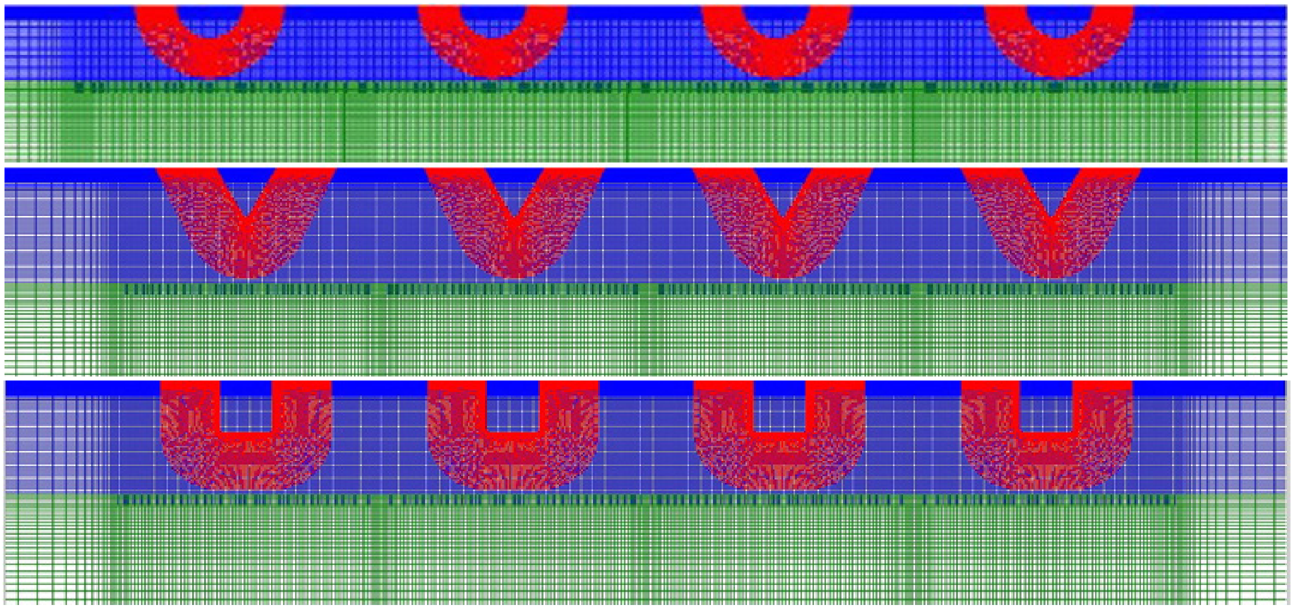


Figure 1: A grid of pipes, consisting of a number of sections with a median semicircular (upper figure), triangular (middle figure) and square (lower figure) turbulators, the inlet and outlet sections of which are smooth-tube.

In this article, protrusions of square and triangular profile sections were considered, since It is these protrusions that are most characteristic in the study of tornado (vortex) zones in order to intensify heat transfer.

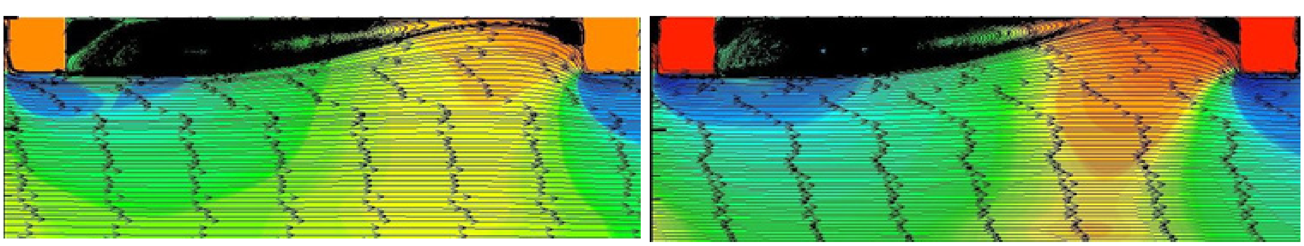
Research of Structures for Artificial Turbulized Flows in Canals with Turbulizers of Square and Triangular Profiles for High Reynolds Criteria Re = 106 And Their Relative Heights D / D = 0.90 ... 0.95 AND RELATIVE STEPS between h and t / D = 0.25 ... 1.00

In earlier studies, in terms of the flow structure, square turbulators with d / D= 0.94 and t / D = 1.00 at the maximum considered Reynolds number Re = 105 as qualitatively characteristic [49, 50].

For higher Reynolds numbers (Re = 106), the flow structure was studied for semicircular turbulators [51, 52]. For square turbulators, similar studies were carried out in [52]. The nature of streamlines for square and triangular turbulators will qualitatively differ from semicircular ones, therefore, it is relevant to study the flow structure for these conditions, based on the above-mentioned previous calculation works, in which the structures of the corresponding vortex zones were analyzed.

After the above analysis, one should proceed to a comparative analysis of vortex zones for triangular and square turbulators. with other conditions being equal, but for higher Reynolds numbers Re = 106 and Re = 105.

For this purpose, na fig. 2 shows a comparison of streamlines for a pipe with relatively low turbulators of triangular and square cross-sections at Re = 105 (upper figures) and Re = 106 (lower figures), d / D = 0.95, t / D = 0.25 for air ...



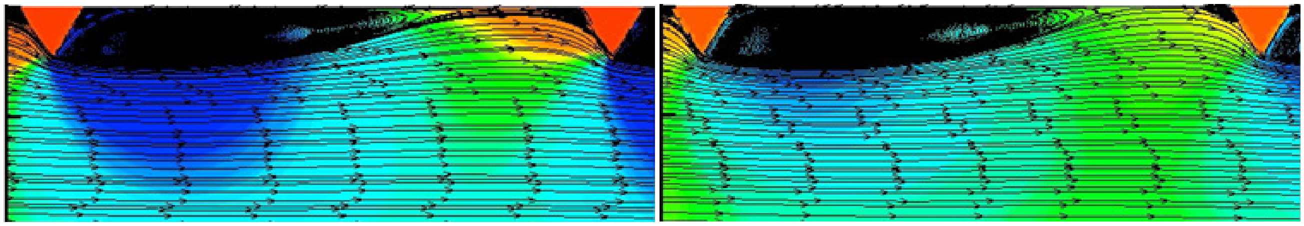


Figure 2: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections at Re = 105 (upper figures) and Re = 106 (lower figures); d / D = 0.95; t / D = 0.25 in air

Fig. 2 it is clearly seen that the points of attachment for both cases are located approximately at the same distance from the square turbulator; for them, the main vortices in both cases retain their external dimensions, but inside the main vortices a qualitative deformation occurs when the Reynolds criterion increases from Re = 105 to and Re = 106 (Fig. 2).

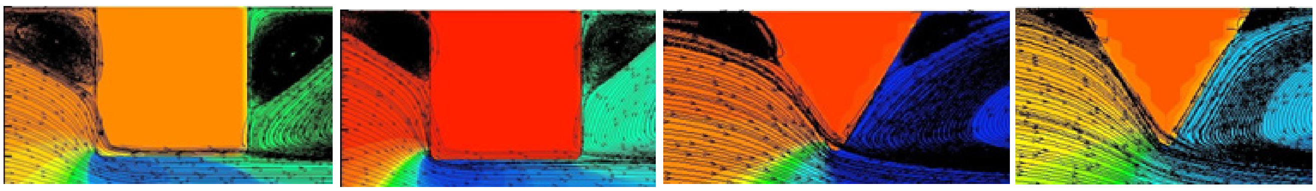


Figure 3: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections for corner vortices up to and behind the turbulator at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.95; t / D = 0.25 in air

For triangular turbulators, the attachment points are also located approximately at the same distance from the turbulators, but the qualitative deformation of the main vortex is even more pronounced than in square turbulators: it deforms more strongly when the Reynolds criterion is increased from Re = 105 to and Re = 106 (Fig. 2) both in the direction of the flow core and in the direction of the corner vortices. It is clear that the deformation of the main vortex into the main flow causes increased hydraulic resistance for triangular turbulators in relation to square ones.

The deformation of angular vortices is shown in Fig. 3, where it can be seen that the angular vortex after the square turbulator decreases with an increase in the Reynolds number, i.e. this corner vortex is squeezed out by the main vortex towards the wall. As for the angular vortex to the square turbulator (Fig. 3), it also decreases with an increase in the Reynolds number, but to a lesser extent; the separation point shifts with an increase in the Reynolds number somewhat closer to the turbulator. For triangular turbulators, such tendencies are even more pronounced (Fig. 3): angular vortices decrease even more, especially the vortex behind the triangular turbulator. With an increase in the relative pitch between individual turbulators of a square profile (t / D = 0.50), there is (Fig. 4) an increase in the main vortex into the main flow; the attachment point for cases with Re = 105 and Re = 106 is approximately at the same distance from the turbulator. For triangular turbulators, a similar trend persists (Fig. 4).

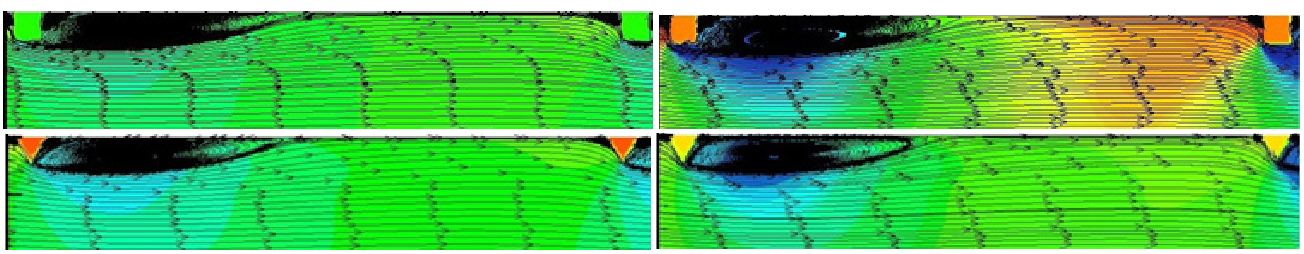


Figure 4: Comparison of current lines for pipes with turbulators of square and triangular cross-sections at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.95; t / D = 0.50 in air

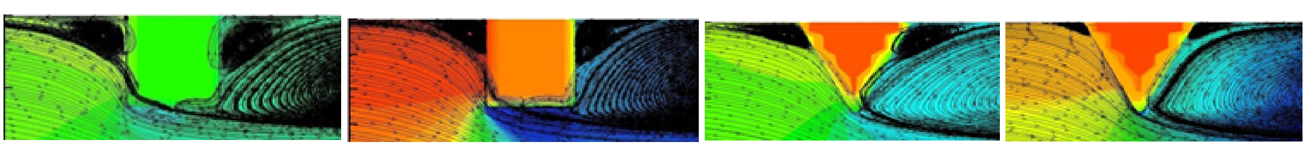


Figure 5: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections for corner vortices up to and behind the turbulator, as well as above the turbulator at Re = 105 (left drawings) and Re = 106 (upper figures); d / D = 0.95; t / D = 0.50 in air

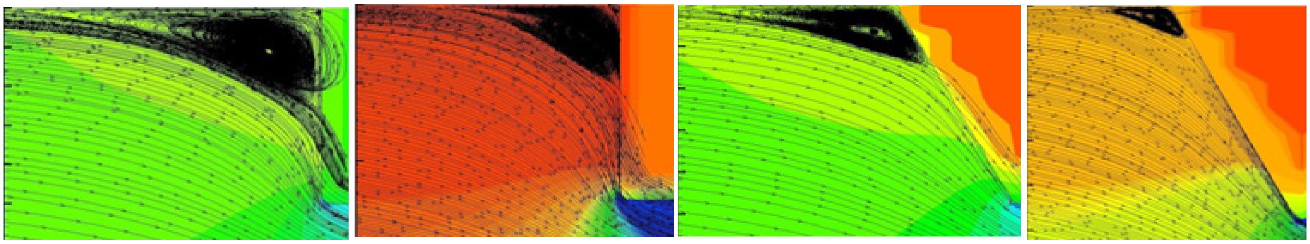


Figure 6: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections for angular vortices up to a turbulator at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 0.50 in air

As can be seen from Fig. 5, the vortex above the square turbulator with the increment of the Reynolds criterion is shifted against the flow, which causes a greater exit of the main vortex into the main flow.

The angular vortex up to the square turbulator (Fig. 6) with an increase in the Reynolds criterion to Re = 106 becomes already much smaller than for the case with Re = 105, and the separation point is based noticeably closer to the turbulator, even to a greater extent than for the similar case with t / D = 0.50. For the angular vortex after the square turbulator (Fig. 7), its decrease takes place, and the point of attachment is located closer to the turbulator by the increment of the Reynolds criterion.

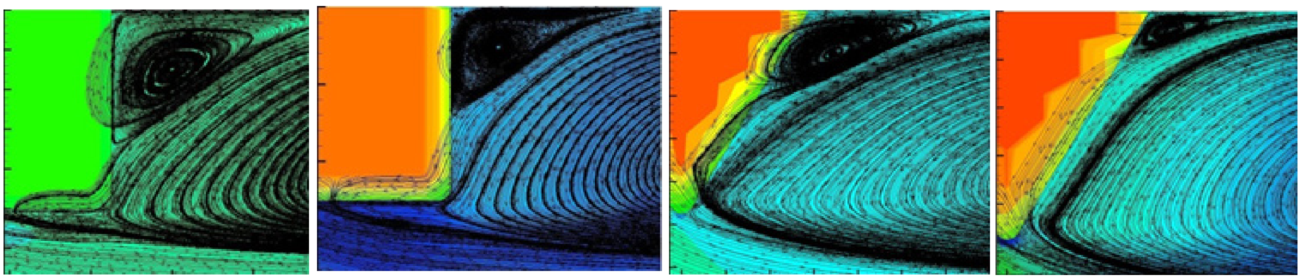


Figure 7: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections for corner vortices behind the turbulator, as well as above the turbulator at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 0.50 in air

For triangular turbulators, the above tendency even intensifies (Fig. 6) and (Fig. 7): angular vortices are squeezed out by the main vortex to an even greater extent than for square turbulators.

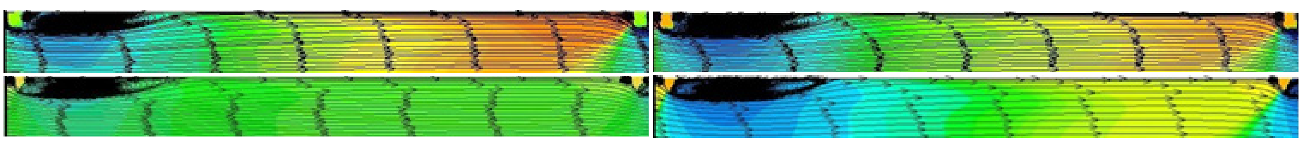


Figure 8: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 1.00 in air

For larger relative(dimensionless) steps between turbulators of a square profile in Fig. 8 shows a comparison along the current lines for pipes with relatively low turbulators of square and triangular cross-sections at Re = 105 (upper figures) and Re = 106 (lower figures), d / D = 0.95, t / D = 1.00 for air

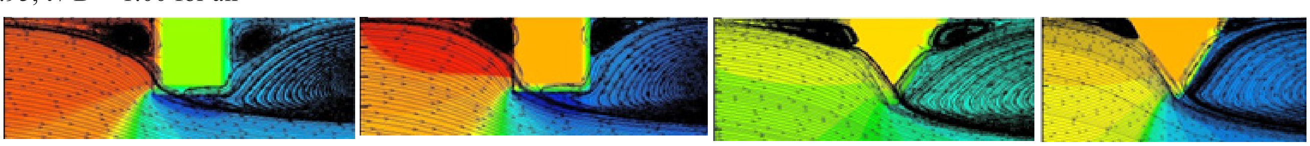


Figure 9: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections up to the turbulizer, behind the turbulator, and also above the turbulator at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 1.00 in air

As can be seen from Fig. 8 that the points of attachment for both cases are located approximately at the same distance from the triangular and square turbulators, but the main vortex will deform with the increment of the Reynolds criteria, which can be seen in Fig. 9, which shows a comparison along the lines of currents for pipes with turbulators of square and triangular cross-sections up to the turbulator, behind the turbulator, and also above the turbulator. As the Reynolds criteria increase to Re = 106, the main vortex will move towards the flow core (Fig. 8) and (Fig. 9). Before the square turbulator, the separation point shifts towards the flow, and the height of the angular vortex decreases even more (Fig. 10).

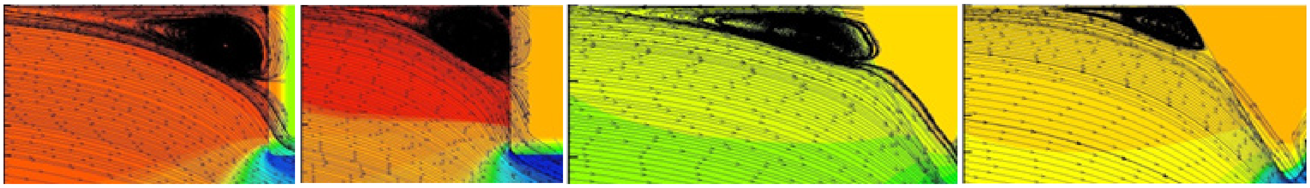


Figure 10: Comparison of current lines for a pipe with turbulators of square and triangular cross-section to the turbulator at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 1.00 in air

For triangular turbulators, the above tendency only intensifies: the separation point is located even closer behind the turbulizer, and the vortex height becomes even smaller (Fig. 10).

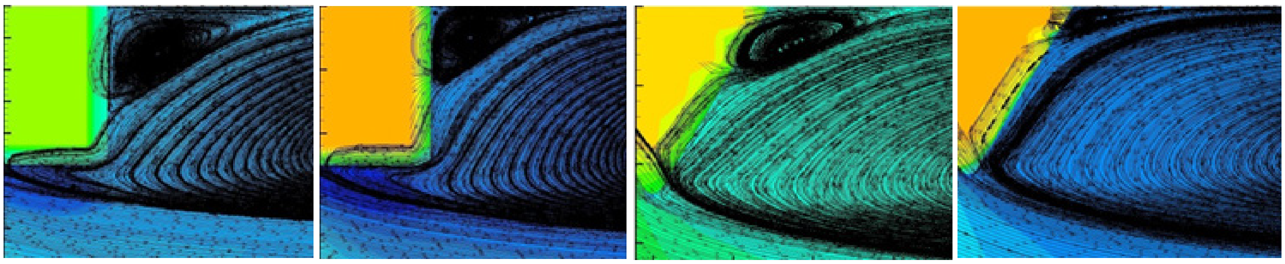


Figure 11: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections up to the turbulizer, behind the turbulizer, as well as for a part of a large vortex at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 1.00 in air

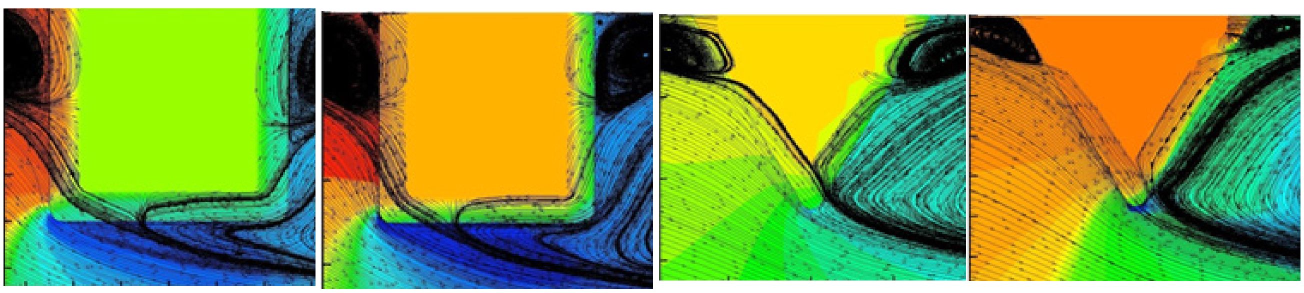


Figure 12: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections above the turbulator at Re = 105 (left figures) and Re = 106 (rightsdrawings); d / D = 0.95; t / D = 1.00 in air

The angular vortex after the square turbulator is even more squeezed out by the main flow to the wall with the increment of the Reynolds criteria (Fig. 11). The vortex above the square turbulator (Fig. 12) is noticeably displaced against the flow, which leads to the fact that the main vortex goes more towards the flow core (see Fig. 12) - the latter circumstance causes an increase in hydraulic resistance. For turbulators with a triangular transverse profile, the extrusion of the corner vortex by the main vortex after the turbulator becomes even greater (Fig. 11). The exit of the main vortex into the cores of the main flows at Re = 106 for turbulators of a triangular profile is comparable to the square ones (Fig. 12), therefore, their hydraulic resistances become comparable.

FROMA comparison along the lines of currents for pipes with relatively high turbulators of triangular and square cross-sections at Re = 105 (upper figures) and Re = 106 (lower figures), d / D = 0.90, t / D = 0.25 for air is given in Fig. 13.

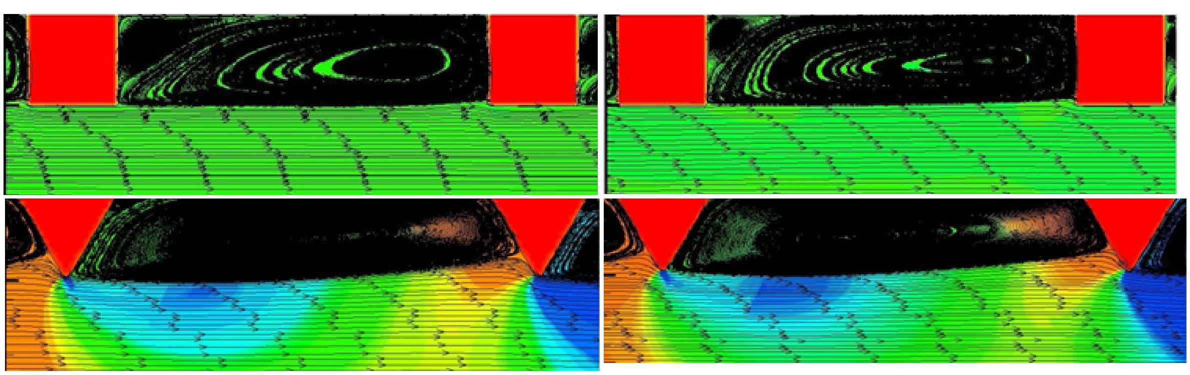


Figure 13: Comparison of current lines for pipes with turbulators of square and triangular cross-sections at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 0.25 in air

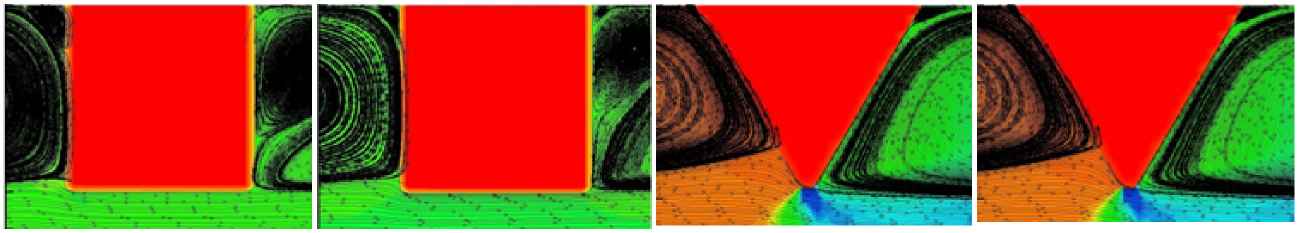


Figure 14: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections above the turbulator at Re = 105 (left figure) and Re = 106 (right figure); d / D = 0.90; t / D = 0.25 in air

The deformation of the main vortex for square turbulators with an increment of the Reynolds criteria also occurs due to the extrusion of the angular vortex behind the turbulator. For taller square turbulators, the angular vortex reaches larger sizes than for low ones; for turbulators of a triangular profile, the above tendency is less pronounced (Fig. 13 and Fig. 14).

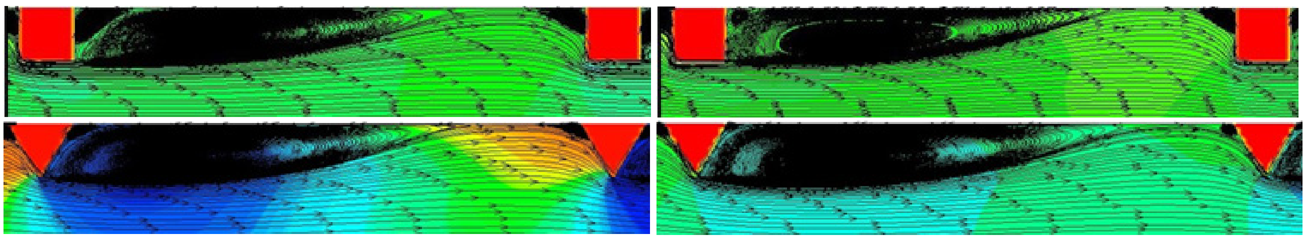


Figure 15: Comparison of current lines for pipes with turbulators of square and triangular cross-sections at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 0.50 in air

Extrusion of corner vortices by the main vortex is more pronounced for high turbulators than for low ones - for triangular turbulators it is even more pronounced than for square ones (Fig. 14).

The angular vortex before the turbulator for high square turbulators is squeezed out almost completely, in contrast to the low ones (Fig. 14); for triangular turbulators this extrusion is even more pronounced (Fig. 14). No vortices arise above the square turbulator for both small and large turbulators (Fig. 3) and (Fig. 14).

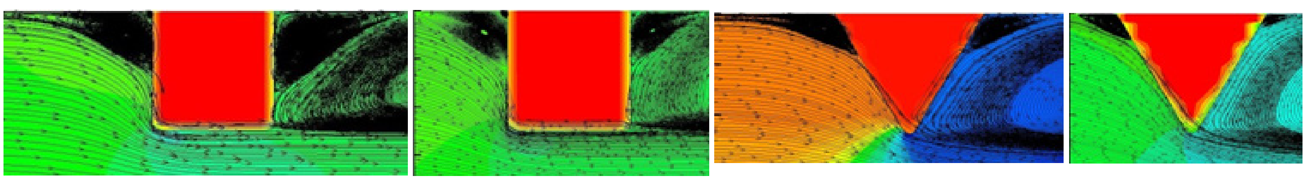


Figure 16: Comparison of current lines for pipes with turbulators of square and triangular cross-sections up to the turbulizer, behind the turbulator, and also above the turbulator at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 0.50 in air

The connection of the main vortex to the subsequent triangular turbulator occurs much closer to the wall than in square turbulators, in which this connection is located almost at the upper point of the subsequent square turbulator (Fig. 14).

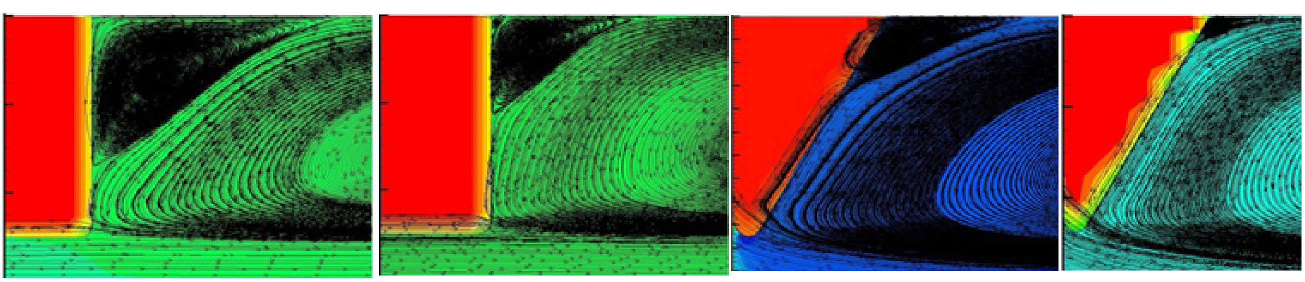


Figure 17: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections behind the turbulizer, as well as above the turbulizer at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 0.50 in air

When increase in relative (dimensionless) steps between turbulators (t / D = 0.50) for high square turbulators also takes place (Fig. 15) an increase in the main vortex into the main flow, and the attachment point for the cases with Re = 105 and Re = 106 is at approximately the same distance from the turbulator. For turbulators of a triangular profile, this trend persists (Fig. 15).

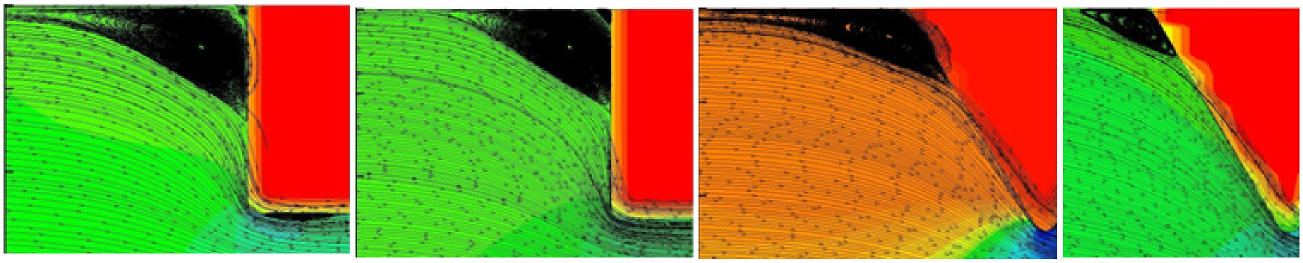


Figure 18: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections to the turbulizer at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 0.50 in air

The angular vortex behind the square turbulator (Fig. 16) and (Fig. 17) deforms to a large extent with the increment of the Reynolds criteria, and the break-off point is located much closer to the square turbulator, and almost no vortex above the square turbulator is observed (compare Fig. 16 and fig. 17 with fig. 5). For triangular turbulators, the angular vortex behind the turbulator with an increment of the Reynolds criterion decreases very significantly, much more than for square turbulators (Fig. 16) and (Fig. 17).

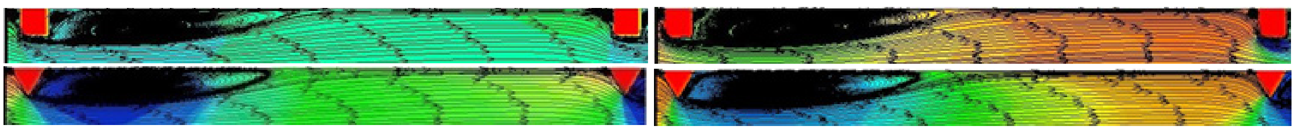


Figure 19: Comparison of current lines for pipes with turbulators of square and triangular cross-sections at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 1.00 in air

The above does not increase the hydraulic resistance in the case of square turbulators, in contrast to the case when a vortex above the turbulator is generated.

The angular vortex up to the square turbulator (Fig. 18) slightly decreases, but the separation point is located noticeably closer to the flow direction with the increment of the Reynolds criteria (Fig. 18). For turbulators of a triangular profile, this tendency is even more pronounced: the angular vortex to a triangular turbulator decreases significantly, and the separation point is located even closer to the direction of the flow than for turbulators of a square profile (Fig. 18).

For larger relative (dimensionless) steps between relatively high turbulators of triangular and square cross sections, Fig. 19 shows a comparison of streamlines for pipes at Re = 105 (upper figures) and Re = 106 (lower figures), d / D = 0.90, t / D = 1.00 for air.

Here, the deformation of the main vortex also takes place with the increment of the Reynolds criteria due to the extrusion of the angular vortex behind the square turbulator. For triangular turbulators, this extrusion is even more enhanced - at Re = 106, the angular vortex is not observed at all.

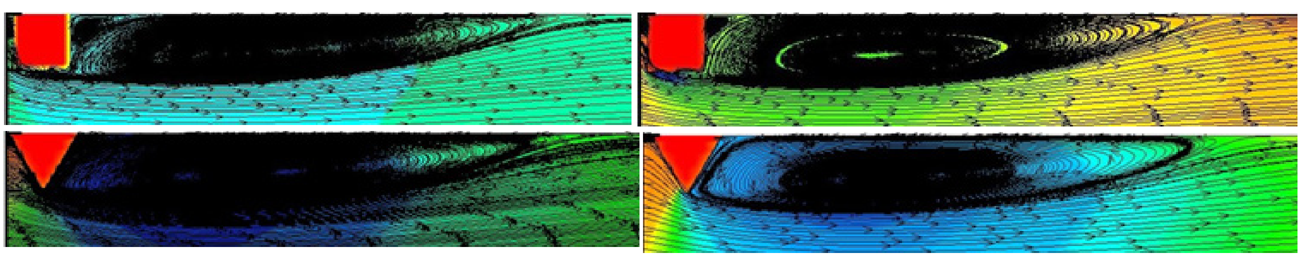


Figure 20: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections behind the turbulator at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 1.00 in air

As can be seen from Fig. 20, the points of flow attachment behind the square turbulator for the case with higher Reynolds numbers will already be located definitely closer to the turbulator than for the case with lower Reynolds numbers, i.e. will move against the direction of flow, which leads to a decrease in the intensification of heat transfer. For triangular turbulators, the above trend will continue (Fig. 20).

The angular vortex behind the square turbulator (Fig. 21) at higher Reynolds numbers is much less than at lower ones, which is due to its squeezing out due to the main flow. In addition, the point of attachment of the angular vortex behind the square turbulizer is also noticeably displaced against the direction of the main flow, which somewhat increases the hydraulic resistance. Behind the triangular turbulizer, the angular vortex is generally leveled with an increment of the Reynolds criteria up to Re = 106 (Fig. 21); is squeezed out almost completely by the main vortex, which somewhat reduces the hydraulic resistance.

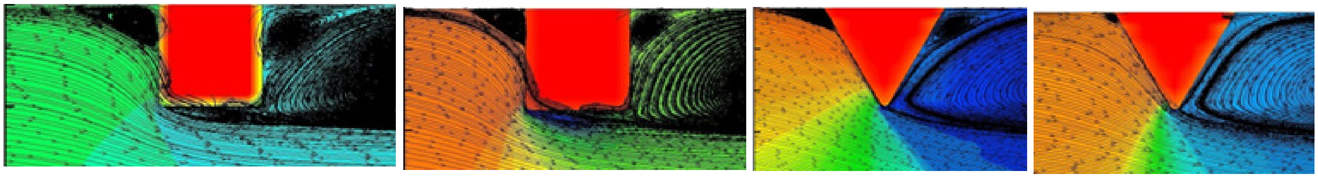


Figure 21: Comparison of current lines for pipes with turbulators of square and triangular cross-sections behind the turbulator, before the turbulator, and also above the turbulator at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 1.00 in air

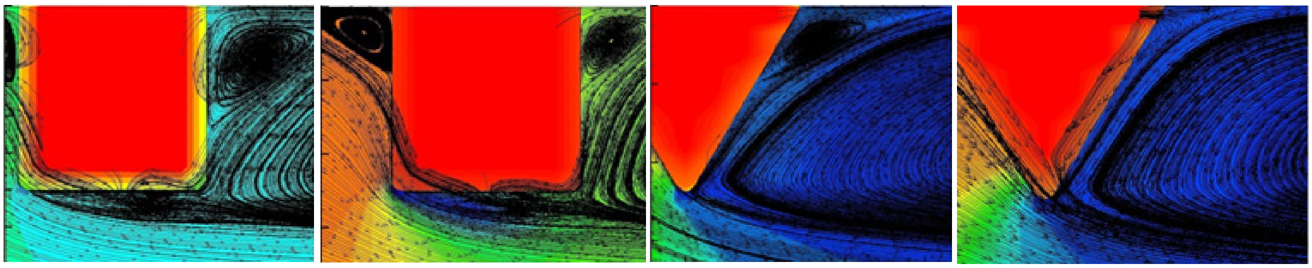


Figure 22: Comparison of current lines for a pipe with turbulators of square and triangular cross-sections behind the turbulator and above the turbulator at Re = 105 (left figures) and Re = 106 (right figures); d / D = 0.90; t / D = 1.00 in air

The vortex above the square turbulator (Fig. 22) slightly increases in size, shifts slightly against the flow direction and spreads towards the flow core, which causes a noticeable increase in hydraulic resistance. For turbulators of triangular transverse profiles, with an increment of the Reynolds criteria up to Re = 106, a greater introduction of the main vortex into the flow core occurs (Fig. 22), the streamlines are located at a greater angle to the flow axis (Fig. 22), which increases the flow resistance of the channel.

Therefore, based on the analysis of streamlines, it can be summarized that at large Reynolds numbers Re = 106 and d / D = 0.95 ... 0.90 and t / D = 0.25 ... 1.0 for square turbulators, unlike semicircular ones, the increase in the relative intensification of heat transfer is rather small, since there is a displacement of the separation point against the flow, and it is accompanied by a relatively large-scale increase in hydraulic resistance, due to the fact that the main vortex is significantly deformed and spreads into the core of the flow, in including by increasing the vortex above the turbulator [51-56]. For triangular turbulators, the above tendency will continue, but to a lesser extent than for square turbulators, i.e. triangular turbulators in this sense will occupy an intermediate position between semicircular and square turbulators.

Influence of Structures with Vortex Zones Between Periodic Flow Turbulators and the Surface Arrangement of Kbadpathogo and Triangular Transverse Profiles on the Integral (Averaged) Characteristics of the Flow and Heat Exchange (RE = 106; D / D = 0.90 ... 0.95; T / D = 0.25 ... 1.00)

As a result of the performed numerical calculations, local (local) and averaged (integral) characteristics of flow and heat transfer in straight circular horizontal pipes with turbulators of square and triangular transverse profiles were obtained earlier [1-9; 57-61] for Reynolds numbers Re≤105.

The main goal of this work is to identify those aspects that have not been previously disclosed for Reynolds numbers up to Re≤106.

Calculations have shown that the relative intensification of heat transfer Nu / NuГЛ for round and straight pipes with relatively low square turbulators for d / D = 0.95, t / D = 0.25 at Re = 106 is almost 5.6% higher than at Re = 105, other things being equal; for t / D = 0.50, this relative intensified heat transfer is higher by about 4.6%; for t / D = 1.00, this increase in relative heat transfer is even smaller and is only about 3.7%. For triangular turbulators, these indicators are, respectively: 4.1%, 3.5% and 3.1%, i.e. it is definitely lower than for square turbulators.

The calculated, relative intensification of heat transfer Nu / NuГЛ for round and straight pipes with relatively high square turbulators for d / D = 0.90, t / D = 0.25 at Re = 106 is almost 3.1% higher than at Re = 105 with other things being equal; for t / D = 0.50, this relative intensified heat transfer is higher by about 2.4%; for t / D = 1.00, this increase in relative heat transfer is even smaller and is only about 1.7%. For triangular turbulators, these indicators are, respectively: 3.1%, 1.7% and 0.2%, i.e. it is definitely lower than for square turbulators, and it is even lower than the corresponding figures for relatively low turbulators.

The above parameters of heat transfer intensification for square turbulators are definitely less than for semicircular turbulators [14, 19]); for triangular turbulators, these parameters are even lower, despite the fact that in relation to a smooth pipe with other equal conditions, the heat transfer will be greater.

Thus, for the considered conditions for square turbulators (Pr = 0.72, Re≤106, d / D = 0.95 ... 0.90, t / D = 0.25 ... 1.00), an increase in the relative intensified heat transfer Nu / NuHL with an increase in the Reynolds number from Re = 105 to Re = 106 with other conditions being equal is relatively small, especially for open depressions (classification generated in [4, 7-9, 12, 13]) at t / D = 1.00. For triangular turbulators, the above tendency persists, but the indicators (Nu / NuГЛ) | Re = 106 for triangular turbulators are slightly higher than for square turbulators.

As can be seen from the above pictures of vortex zones for turbulators of this type, the prevalence of momentum transfer over heat transfer at high Reynolds numbers Re = 106 is quite large, since the return flows are quite significant, the main vortex leaves the flow cores, which significantly increases the hydraulic resistance. For triangular turbulators, the above tendency is less pronounced than for square turbulators: return flows are less pronounced, they can even be almost completely squeezed out; the exit of the main vortex into the cores of the flows is also quite significant, which also determines the high hydraulic resistance.

Consequently, in the article, based on the analysis of vortex (tornado) zones between turbulators of square and triangular profiles, it was shown that for higher square turbulators and at higher Reynolds numbers, a slight increase in the relative Nusselt criterion Nu / NuГЛ is accompanied by a significant increase in the relative hydraulic resistance due to a very significant the influence of return flows, which can flow directly onto the turbulator itself, to the greater extent, the higher the Reynolds number; for triangular turbulators, the above tendency continues and even deepens.

Main Findings

The article carried out mathematical modeling of the heat exchange process in straight and round horizontal pipes with turbulators with d / D = 0.95 ... 0.90 and t / D = 0.25 ... 1.00 triangular and square transverse profiles at large Reynolds numbers (Re = 106) on a foundation with multiblock computing technologies, which are based on solutions using a finite-volume and factorized procedure of Reynolds equations, as well as energy equations.

It was established in the article that the relative intensification of heat transfer [(Nu / NuГЛ) | Re = 106] / [(Nu / NuГЛ) | Re = 105] in round pipes with square turbulators in air for large Reynolds numbers (Re = 106), which may well be relevant in channels used in heat exchangers, may be higher with a scale increase in hydraulic resistance than for slightly smaller numbers (Re = 105), for relatively high flow turbulators d / D = 0.90 for the entire range under consideration for the parameter of dimensionless steps between them t / D = 0.25 ... 1.00 a little more than 3%; for turbulators of triangular transverse profiles, similar indicators are approximately the same.

With lower square turbulators with d / D = 0.95, this increase in relative heat transfer for large Reynolds numbers (Re = 106), compared with lower numbers (Re = 105), does not exceed 6%; for turbulators of triangular transverse profiles, similar figures are slightly more than 4%.

For the entire investigated range under consideration, the relative intensification is higher for small steps than for large steps under similar identical conditions for both square and triangular turbulators.

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