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Research Article

On the Issue of Solving the Theoretical Problem of Heat Exchange in Pipes with Diaphragms Depending on the Prandtl Criterion in an Extended Range of its Variation

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Abstract

The calculation method was used to study the dependence on the Prandtl number in a wide range of its change: (Pr = ~10^{-3}÷~10^{+5}) of the distribution of the integral heat transfer during turbulent convective heat transfer in a pipe with a sequence of periodic protrusions of semicircular geometry based on the numerical solution of the system of Reynolds equations, closed by using the Menter shear stress transfer model, and the energy equation on multiscale intersecting structured grids. A general analysis of the obtained calculated data showed that for increased (Pr>1) Prandtl numbers, the maximum increase in relative heat transfer, which can be quite noticeable, occurs at low Reynolds numbers, large relative heights of turbulators, small relative steps between turbulators, and for reduced (Pr< 1) Prandtl numbers - for large Reynolds numbers, large relative heights of turbulators, large relative steps between turbulators. The minimum values of relative heat transfer for increased Prandtl numbers are observed at high Reynolds numbers for high turbulators with a large step between them, and for reduced Prandtl numbers - at medium Reynolds numbers for high turbulators with a large step between them.

Keywords: Modeling; Numerical; Channel; Pipe; Convective; The Menter Model; Turbulizer; Heat Exchange; Hydraulic Resistance; Prandtl Number

Introduction

The application of periodic protrusions on the walls of the surfaces being washed is a well-tested in practice method of vortex heat transfer intensification [7,8,26].

The intensification of heat transfer for the conditions of the flow of heat carriers in pipes with turbulators was and is carried out mainly by experimental methods [7,8,26], and theoretical studies are rather few, many of them are based on integral approaches [2,11,14,18,19].

At the present stage of research, the problems of aeromechanics and thermal physics of separated and vortex flows are increasingly being solved by the methods of multiblock computing technologies based on intersecting structured grids [22-25,28].

The present study is a logical continuation of the above computational methods [1,3-6,9-13,15-17] for the analysis of turbulent flow and heat transfer in pipes with semicircular flow turbulators (diaphragms) with different relative heights, steps for different coolant flow regimes in order to analyze in more detail the heat transfer intensification for coolants with different Prandtl numbers in a wide range of its change: Pr = ~10^{-3}÷~10^{+5}. Previously, this aspect was studied insufficiently and for a much narrower range of the Prandtl criteria.

Matemathematical and discrete models

In this work, using fully implicit finite-difference schemes on a centered non-uniform oblique grid, the system of Reynolds and energy equations written in natural variables is solved.

The SIMPLEC procedure is used to calculate the pressure field; the principle of splitting according to physical processes takes place. The convective terms are approximated using a quadratic upwind scheme.

Difference equations are solved using a highly efficient method of incomplete matrix factorization with accelerated convergence using the additive correction method.

The multiblock algorithm for solving the problem on intersecting grids of different scales, which has been tested in solving problems of vortex dynamics and heat transfer [22], is used to correctly describe turbulent heat transfer.

Using the zonal low-Reynolds Menter model [28], a description of turbulent transport is implemented. The study considered channels of a constant cylindrical cross section with eight turbulators located on the walls in the form of periodic diaphragms of a semicircular cross section.

The parameters changed in the following ranges: d/D = 0.98÷0.90; t/D = 0.25÷1, where t is the spacing of the turbulators; d is the diaphragm diameter; D is the pipe diameter; Re = 10^4÷$5\cdot10^5$ is the Reynolds number; Pr = 0.0038÷96432 (Pr = ~10^{-3}÷~10^{+5}).

Briefly, the calculation model can be characterized as follows. The three-dimensional computational domain under study has several sections, each of which consists of one ledge (Figure 1).

When calculating, it is assumed that with the considered number of turbulators, the turbulent flow becomes steady.

In the main part of the study, the calculation of heat transfer was carried out under the boundary condition on the wall of the first kind with a successive change in the Prandtl number in order to establish the pattern of changes in the intensification of heat transfer for various coolants.

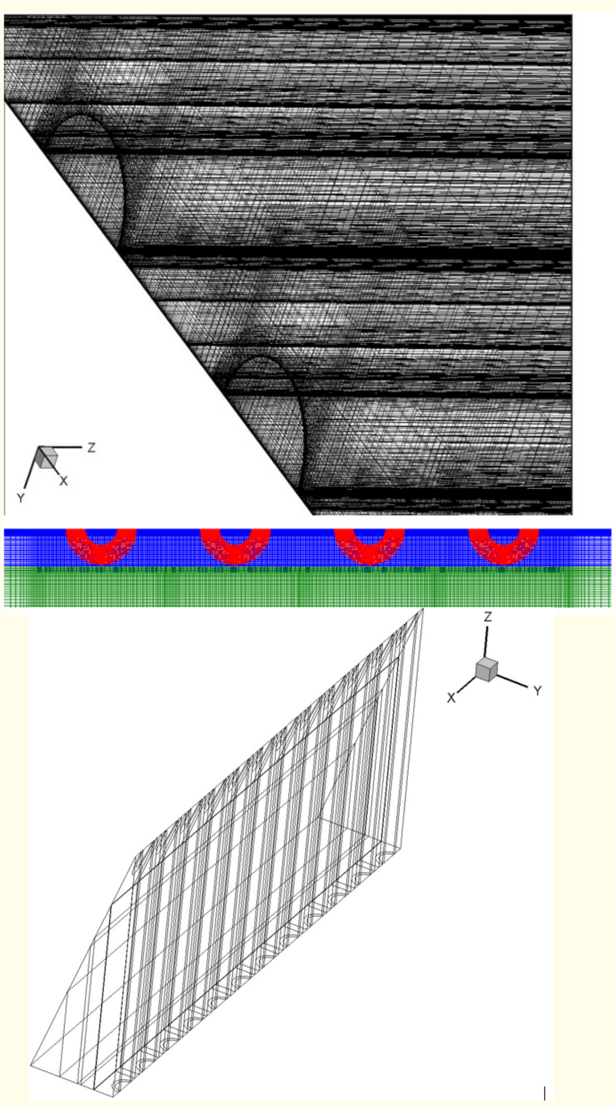

Figure 1: Calculation grid of the pipe, consisting of several sections with a semicircular turbulator located in the middle.

At the preliminary stage of the study, a modification of the multidisciplinary computer complex for numerical simulation of spatial separated flows and vortex heat transfer is carried out in order to adapt and refine the mathematical model of flow and vortex heat transfer in channels with turbulators, assuming the occurrence of spatial vortex structures responsible for the vortex intensification of heat transfer processes in the near-wall zones of pipes with turbulators.

In order to solve the problem of enhanced heat transfer, the computational three-dimensional grid was built in the same way: a two-dimensional grid was constructed in axial and radial coordinates, unfolded along the circumferential coordinate with a constant step. In order to achieve the required resolution in the vicinity of the obstacle, two-dimensional grids in the form of multi-tiered structured grids were used, and the obstacle was described on the most detailed grid with the highest spatial resolution.

The detailed grid was built into a coarser grid, which was used to describe the flow in the near wake of the obstacle, and the transition from the near-wall region to the axis was also carried out using intermediate grids, the purpose of which is to increase the longitudinal step of the grid in the area of the obstacle and change the resolution along the circumferential coordinate. In the future, we will not dwell on the details of directly model aspects of numerical calculations using this technique, since they were considered in [3,4,6,8,9,11,15,20,22-24,28].

Data for initial calculations

In the inlet section of the pipe section under consideration, a uniform flow with a thin boundary layer allowing for variation was considered; the turbulence parameters correspond to experimental tests in a pipe, assuming the turbulence scale is of the order of the pipe diameter, which is chosen as the characteristic dimension, and the degree of turbulence is assumed to be one and a half percent.

In the outlet section of the considered section of the pipe, "soft" boundary conditions are set, otherwise called the conditions for continuing the solution, which are characterized by extrapolation of parameters outside the calculation area.

On the pipe walls washed by the coolant, which are considered to be isothermal under the boundary conditions of the first kind and having a temperature higher or lower by a certain number of degrees with respect to the temperature of the oncoming flow, sticking conditions take place.

For the selected channel geometry, each individual problem from several sections is solved in two stages: first, the dynamic problem is solved, after which, for the previously calculated fields of the flow velocity components and turbulence characteristics, the thermal problem is solved for various Prandtl numbers (including for a wide range of its change $Pr = \sim 10^{-3} \div \sim 10^{+5}$).

In contrast to earlier scientific works [29,30], in this article, the calculations of enhanced heat transfer using this factorized control-volume method were carried out in a three-dimensional setting instead of a two-dimensional one (as in [1,3-6,9-19]) with an increase in the number of turbulators in the channel is up to 12, with a lower discrepancy (10^{-5}), which made it possible to significantly expand the calculated range for the geometric characteristics of the turbulators and for the defining Reynolds and Prandtl criteria: from Pr = 1÷20 to their limiting values for the coolants used in engineering Pr = 0.0038÷96432. Previously, in such a wide range of Prandtl criteria, calculations of enhanced heat transfer have not yet been carried out.

The convergence criteria for dynamic and thermal problems are determined by limiting the errors in calculating the Cartesian velocity components, and for the thermal problem, by limiting the magnitude of the increase in heat fluxes on the walls; within the framework of this work, the value of 10^{-5} was taken as the relative error.

Influence of the Prandtl number in a very wide range of its change **P**r = 0.0038÷96432 (**P**r = $\sim 10^{-3} \div \sim 10^{+5}$) on heat transfer in straight round pipes with periodically located surface flow turbulators of a semicircular cross section at various geometric and operating parameters.

The drag coefficient ξ and the averaged Nusselt number Nu for a tube with semicircular turbulators under turbulent convective heat transfer were determined in this work by a calculation method based on the numerical solution of the system of Reynolds equations closed using the Menter shear stress transfer model and the energy equation on different-scale intersecting structured grids.

The adequacy of the applied method is substantiated by the fact that earlier for comparison in [3-5,9-11,13,15,16], similar experimental data on heat transfer and hydraulic resistance for pipes with semicircular turbulators or diaphragms were used, where there was a good correlation of the theory and experiment.

Revealed in the previous theoretical works of the author (for example, in [3-5,9-11,13,15,16]), the adequacy of the implemented calculation model to the existing experimental data for local and average flow characteristics and heat transfer in pipes with turbulators determines its application for revealing the patterns of integral (averaged) heat transfer parameters in pipes with different Prandtl numbers (including, in a wide range of its change $Pr = \sim 10^{-3} \div \sim 10^{+5}$) depending on the channel geometry and the coolant flow regime. In this study, only the most common turbulators of a semicircular cross section, typical for pipes with diaphragms, are considered. For sharper turbulators, the convergence range of the problem can be noticeably narrower.

This issue seems important, since it is necessary to know for which Prandtl numbers (including for a very wide range of its change $Pr = \sim 10^{-3} \div \sim 10^{+5}$) a higher heat transfer intensification will take place depending on the determining parameters.

In earlier studies [29,30], calculations of enhanced heat transfer using this factorized control-volumetric method were carried out only for the most typical geometric and operating characteristics for pipes with turbulators (d/D = 0.92; 0.90; t/D = 0.25; 0.50; 1.00; Re = 10^4; 10^5) [7,8,26] for a relatively limited range of Prandtl numbers Pr = 1÷20.

Within the framework of this article the task is to study at a higher level and with higher accuracy of intensified heat transfer in pipes with semicircular turbulators for an extremely wide range of Prandtl number variation (Pr = 0.0038÷96432), i.e. for Prandl numbers of the order: $Pr = \sim 10^{-3} \div \sim 10^{+5}$.

Solutions to the problem of studying enhanced heat transfer in pipes with semicircular turbulators for an extremely wide range of Prandtl number changes were carried out for the following characteristic points (for the coolant in the form of air, the calculations were carried out on the basis that it is the most common, i.e. for air, there are the most extensive experimental data, and is most suitable for verification of calculated data):

Pr = 0.0038 for potassium at 700°C (Pr = 0.0039 for sodium at 700°C);
Pr = 0.005 for potassium at 300°C (for sodium at 450°C);
Pr = 0.05 for lithium at 200°C;
Pr = 0.67 for monatomic gases;
Pr = 0.72 for air;
Pr = 1.00 for polyatomic gases;
Pr = 1.75 for water at 100°C;
Pr = 13.7 for water at 0°C;
Pr = 22.4 for ethylene glycol at 100°C;
Pr = 34.8 for transformer oil at 120°C;
Pr = 125 for ethylene glycol at 34°C (for transformer oil at 46°C; for glycerol at 100°C);
Pr = 328 for glycerol at 80°C;
Pr = 615 for ethylene glycol at 0°C;
Pr = 919 for glycerol at 60°C;
Pr = 11846 for glycerin at 20°C;
Pr = 96432 for glycerin at 0°C.

Characteristic values for regime and geometric parameters were chosen as follows: d/D = 0.90÷0.98; t/D = 0.25÷1.00; Re = $10^4 \div 5 \cdot 10^5$.

Relative heat transfer values Nu/NuGL for various Prandtl numbers, other things being equal, were calculated for isothermal flow with equivalent parameters both for pipes with turbulators and without them.

As a fundamental calculated relative simplex, one should choose the parameter, which shows how, all other things being equal, the intensified heat transfer for the current Prandl number differs from the intensified heat transfer for the single Prandtl criterion $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$.

The basis forsuch an analysis is the method of relative correspondence, which is widely used in studies of enhanced heat transfer [7,8,20,21].

From the analysis of physical processes of intensified heat exchange, it can be postulated that:

$$\lim_{Pr\to\infty}\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}=\frac{1}{(Nu/Nu_{ГЛ})|_{Pr=1}} \text{ and } \lim_{Pr\to 0}\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}=\frac{1}{(Nu/Nu_{ГЛ})|_{Pr=1}}.$$

The results of the calculation according to the proposed model for higher of the specified range of defining parameters are given on figure 2-7: also other things being equal, the results are given depending on the Prandtl number, where they are distributed for increased (Pr>1) decreased (Pr<1) Prandtl numbers (Figure 8-9 and table 1-2).

In the future, the presented data make it possible to analyze the effect on relative heat transfer (ceteris paribus) not only of the Reynolds number, but also of the relative height (by the parameter d/D) and the pitch between the turbulators (t/D).

Theoretical characteristic streamlines for various Reynolds and Prandtl numbers and channel geometry studied in the article, calculated using the proposed model

As an illustration of the calculated data obtained by this method, which are given inrice. 2-5, characteristic calculated streamlines (as well as isotherms, i.e. lines of constant temperatures) are given for pipes with relatively high and medium sizes of transverse annular turbulators of a semicircular cross section for the considered flow conditions for closed, semi-open and open depressions (classification according to [11- 19]).

On the figure 6 And figure 7 similar streamlines and isotherms are given for turbulators of lower relative heights (d/D = 0.96), from which it is clear that open troughs are typical for turbulizers of small heights.

Analysis of the streamlines makes it possible to qualitatively estimate which particular sublayer is turbulent, i.e. makes it possible to judge the nature of heat transfer intensification. For example, if vortex zones are thrown into the core of the flow, then the flow is intensified with a large increase in hydraulic resistance; the presence of stagnant zones indicates that heat transfer will deteriorate with an increase in hydraulic resistance; the location of the attach-

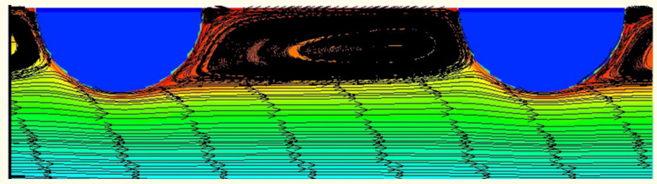

Figure 2: Estimated streamlines for flow in a pipe with turbulators with a semicircular cross section for heat transfer intensification for a closed cavity at Pr = 0.05; Re = 5·105; d/D = 0.90; t/D = 0.25.

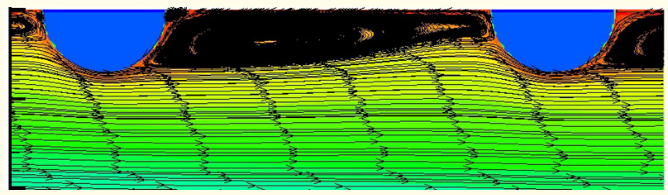

Figure 3: Estimated streamlines for flow in a pipe with turbulators with a semicircular cross section for heat transfer intensification for a semi-open cavity at Pr = 0.0038; Re = 5·105; d/D = 0.93; t/D = 0.25.

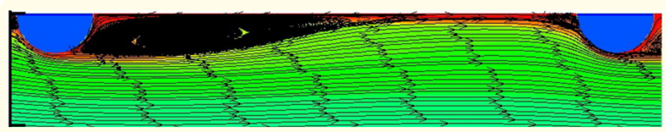

Figure 4: Estimated streamlines for flow in a pipe with turbulators with a semicircular cross section for heat transfer intensification for an open cavity at Pr = 13.7; Re = 104; d/D = 0.93; t/D = 0.50.

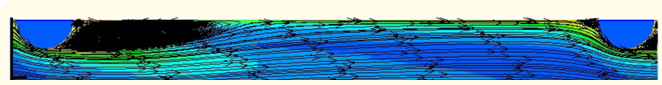

Figure 5: Calculated streamlines for flow in a pipe with turbulators with heat transfer intensification for an open cavity with a semicircular cross section at Pr = 919; Re = 105; d/D = 0.90; t/D = 1.00.

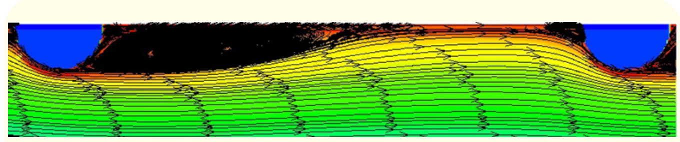

Figure 6: Calculated streamlines for flow in a pipe with heat transfer intensification with relatively low turbulators with a semicircular cross section at Pr = 0.67; Re = 105; d/D = 0.96; t/D = 0.25.

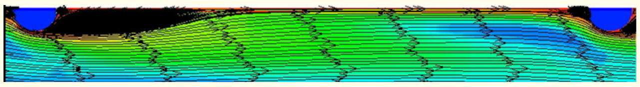

Figure 7: Calculated streamlines for flow in a pipe with heat transfer intensification with relatively low turbulators with a semicircular cross section at Pr = 328; Re = 5·105; d/D = 0.96; t/D = 0.50.

ment point of the turbulent boundary layer is important, since it is there that the maximum increase in heat transfer takes place while minimizing hydraulic resistance, etc., etc.

A detailed analysis of the characteristic streamlines was previously carried out in [1-6,9-11,13-15,19,29,30], therefore, within the framework of this article, limited material is presented that partially verifies the results of calculations.

The general differences in the patterns of change in relative heat transfer depending on the Prandtl number are substantiated by the fact that at low Reynolds numbers, the height of the turbulator is less than the height of the near-wall layer [7,8,20,21,26], and at high Reynolds numbers it is greater. The latter causes turbulization of only the core of the flow, increasing only the hydraulic resistance, almost without increasing the heat transfer.

Analysis of the calculation results according to the proposed model of the relative heat transfer parameter for various Reynolds numbers Re = 10^4÷**5·10**^{5} for increased (Pr>1÷96432) Prandtl numbers $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$.

Calculation results according to the proposed model of the relative heat transfer parameter for various Reynolds numbers Re = 10^4÷$5\cdot10^5$ for increased (1<Pr<96432) Prandtl numbers for d/D = 0.90÷0.98; t/D = 0.25÷1.00 are given in $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$.Table 1.

In Table 1 the calculation results of the proposed model of the

Re	d/D	t/D	Pr										
			1	**1,75**	**13,7**	**22,4**	**34,8**	**125**	**328**	**615**	**919**	**11846**	**96432**
Re = 10^4	d/D = 0,90	t/D = 0,25	1	1,106	1,361	1,396	1,423	1,468	1,473	1,465	1,453	1,412	1,349
Re = $5\cdot10^4$	d/D = 0,90	t/D = 0,25	1	0,994	1,004	1,034	1,068	1,188	1,263	1,292	1,301	1,183	0,818
Re = 10^5	d/D = 0,90	t/D = 0,25	1	0,968	0,929	0,950	0,978	1,082	1,148	1,173	1,178	0,937	0,649
Re = $5\cdot10^5$	d/D = 0,90	t/D = 0,25	1	0,939	0,848	0,863	0,884	0,934	0,895	0,835	0,792	0,626	0,606
Re = 10^4	d/D = 0,90	t/D = 0,50	1	1,065	1,209	1,229	1,242	1,270	1,274	1,267	1,258	1,169	1,100
Re = $5\cdot10^4$	d/D = 0,90	t/D = 0,25	1	0,985	0,900	0,891	0,890	0,917	0,944	0,949	0,951	0,903	0,731
Re = 10^5	d/D = 0,90	t/D = 0,25	1	0,972	0,808	0,792	0,789	0,811	0,835	0,847	0,852	0,761	0,612
Re = $5\cdot10^5$	d/D = 0,90	t/D = 0,25	1	0,932	0,738	0,7281	0,7283	0,747	0,740	0,718	0,700	0,607	0,594
Re = 10^4	d/D = 0,90	t/D = 1,00	1	1,033	1,106	1,122	1,131	1,178	1,213	1,223	1,222	1,133	1,033
Re = $5\cdot10^4$	d/D = 0,90	t/D = 1,00	1	0,966	0,873	0,863	0,859	0,840	0,819	0,8069	0,8068	0,889	0,845
Re = 10^5	d/D = 0,90	t/D = 1,00	1	0,962	0,799	0,780	0,771	0,772	0,783	0,790	0,791	0,728	0,613
Re = $5\cdot10^5$	d/D = 0,90	t/D = 1,00	1	0,891	0,667	0,650	0,644	0,647	0,643	0,629	0,616	0,550	0,539
Re = 10^4	d/D = 0,93	t/D = 0,25	1	1,113	1,418	1,461	1,492	1,539	1,533	1,520	1,508	1,435	1,280
Re = $5\cdot10^4$	d/D = 0,93	t/D = 0,25	1	1,004	1,009	1,018	1,032	1,106	1,165	1,193	1,206	1,124	0,879
Re = 10^5	d/D = 0,93	t/D = 0,25	1	0,979	0,925	0,930	0,943	1,017	1,076	1,101	1,107	0,914	0,667
Re = $5\cdot10^5$	d/D = 0,93	t/D = 0,25	1	0,934	0,834	0,845	0,861	0,903	0,880	0,836	0,801	0,662	0,644
Re = 10^4	d/D = 0,93	t/D = 0,50	1	1,068	1,229	1,261	1,292	1,361	1,387	1,386	1,377	1,266	1,159
Re = $5\cdot10^4$	d/D = 0,93	t/D = 0,50	1	0,973	0,852	0,841	0,838	0,862	0,890	0,902	0,905	0,839	0,671
Re = 10^5	d/D = 0,93	t/D = 0,50	1	0,970	0,840	0,829	0,827	0,854	0,880	0,893	0,896	0,808	0,635
Re = $5\cdot10^5$	d/D = 0,93	t/D = 0,50	1	0,919	0,701	0,688	0,685	0,692	0,683	0,660	0,642	0,557	0,544
Re = 10^4	d/D = 0,93	t/D = 1,00	1	1,036	1,116	1,138	1,164	1,227	1,278	1,302	1,308	1,241	1,130
Re = $5\cdot10^4$	d/D = 0,93	t/D = 1,00	1	0,985	0,904	0,891	0,885	0,902	0,923	0,934	0,937	0,888	0,763
Re = 10^5	d/D = 0,93	t/D = 1,00	1	0,975	0,817	0,801	0,791	0,795	0,812	0,820	0,822	0,749	0,644
Re = $5\cdot10^5$	d/D = 0,93	t/D = 1,00	1	0,906	0,698	0,68443	0,680	0,68442	0,682	0,670	0,658	0,591	0,579
Re = 10^4	d/D = 0,96	t/D = 0,25	1	1,103	1,432	1,481	1,517	1,580	1,591	1,583	1,572	1,414	1,279
Re = $5\cdot10^4$	d/D = 0,96	t/D = 0,25	1	1,006	1,013	1,027	1,041	1,103	1,150	1,173	1,182	1,033	0,789
Re = 10^5	d/D = 0,96	t/D = 0,25	1	0,979	0,933	0,938	0,951	1,012	1,061	1,0768	1,0771	0,900	0,646
Re = $5\cdot10^5$	d/D = 0,96	t/D = 0,25	1	0,941	0,806	0,808	0,819	0,846	0,825	0,788	0,760	0,637	0,621
Re = 10^4	d/D = 0,96	t/D = 0,50	1	1,060	1,269	1,310	1,347	1,445	1,491	1,497	1,493	1,420	1,318
Re = $5\cdot10^4$	d/D = 0,96	t/D = 0,50	1	0,995	0,975	0,986	0,997	1,051	1,097	1,121	1,132	1,051	0,831

Re = 10^5	d/D = 0,96	t/D = 0,50	1	0,974	0,9084	0,9076	0,912	0,957	0,993	1,005	1,006	0,890	0,715
Re = $5{\cdot}10^5$	d/D = 0,96	t/D = 0,50	1	0,938	0,800	0,795	0,797	0,810	0,802	0,781	0,762	0,666	0,650
Re = 10^4	d/D = 0,96	t/D = 1,00	1	1,028	1,147	1,182	1,217	1,329	1,399	1,426	1,432	1,355	1,254
Re = $5{\cdot}10^4$	d/D = 0,96	t/D = 1,00	1	0,991	0,953	0,957	0,967	1,005	1,039	1,056	1,062	1,014	0,888
Re = 10^5	d/D = 0,96	t/D = 1,00	1	0,976	0,9032	0,901	0,9029	0,925	0,949	0,959	0,961	0,883	0,773
Re = $5{\cdot}10^5$	d/D = 0,96	t/D = 1,00	1	0,947	0,827	0,817	0,816	0,819	0,816	0,804	0,791	0,716	0,703
Re = 10^4	d/D = 0,98	t/D = 0,25	1	1,016	1,069	1,081	1,091	1,164	1,276	1,416	1,481	1,643	1,569
Re = $5{\cdot}10^4$	d/D = 0,98	t/D = 0,25	1	1,015	1,047	1,069	1,095	1,204	1,287	1,322	1,338	1,220	0,953
Re = 10^5	d/D = 0,98	t/D = 0,25	1	0,985	0,957	0,972	0,995	1,096	1,158	1,178	1,138	1,009	0,716
Re = $5{\cdot}10^5$	d/D = 0,98	t/D = 0,25	1	0,946	0,848	0,855	0,867	0,899	0,890	0,857	0,829	0,694	0,680
Re = 10^4	d/D = 0,98	t/D = 0,50	1	1,012	1,054	1,065	1,078	1,142	1,215	1,264	1,325	1,472	1,456
Re = $5{\cdot}10^4$	d/D = 0,98	t/D = 0,50	1	1,005	1,004	1,014	1,028	1,100	1,159	1,203	1,218	1,158	0,982
Re = 10^5	d/D = 0,98	t/D = 0,50	1	0,982	0,929	0,934	0,946	1,015	1,065	1,081	1,087	0,996	0,825
Re = $5{\cdot}10^5$	d/D = 0,98	t/D = 0,50	1	0,951	0,849	0,851	0,858	0,882	0,883	0,867	0,850	0,756	0,742
Re = 10^4	d/D = 0,98	t/D = 1,00	1	1,006	1,039	1,049	1,058	1,104	1,154	1,185	1,200	1,293	1,243
Re = $5{\cdot}10^4$	d/D = 0,98	t/D = 1,00	1	0,999	0,979	0,980	0,985	1,025	1,066	1,088	1,098	1,073	0,974
Re = 10^5	d/D = 0,98	t/D = 1,00	1	0,984	0,923	0,920	0,924	0,961	0,994	1,008	1,013	0,974	0,860
Re = $5{\cdot}10^5$	d/D = 0,98	t/D = 1,00	1	0,961	0,868	0,866	0,870	0,886	0,891	0,882	0,872	0,802	0,790

Table 1: Calculated results of the relative heat transfer simplex for different Reynolds criteria Re = $10^4 \div 5{\cdot}10^5$ with increased (Pr>1÷96432) Prandtl criteria for projections with different relative heights d/D = 0,90÷0,98 and different relative steps of turbulators t/D = 0,25÷1,00.

relative heat transfer parameter $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ for various Reynolds numbers Re = $10^4 \div 5{\cdot}10^5$ for increased (Pr>1÷96432) Prandtl numbers for turbulators of various heights d/D = 0.90÷0.98 with different steps between them t/D = 0.25÷1.00.

General analysis of the data presented in Table 1, shows that the maximum increase in relative heat transfer $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ is observed at low Reynolds numbers (Re = 10^4): about +60% for turbulators relative to average heights (d/D = 0.93 and d/D = 0.96) with a small step between them (t/D = 0.25) in the region of Pr≈125÷615; about +65% for turbulators of relatively low heights (d/D = 0.98) with a small step between them (t/D = 0.25) around Pr≈10^4.

Slightly smaller values, a little less than +50%, take place for turbulators of relatively large heights (d/D = 0.90) with small steps between them at low Reynolds numbers (Re = 10^4); increases of the order of +40% take place for turbulators with d/D = 0.96 with a step t/D = 1.00 at Re = 10^4 at Prandtl numbers Pr≈$10^2 \div 10^3$.

The minimum values of relative heat transfer $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ are observed at high Reynolds numbers (Re = $5{\cdot}10^5$): -(40÷45)% for high and medium turbulators (d/D = 0.90 and d/D = 0.93) with large and medium pitch between them (t/D = 1.00 and t/D = 0.50) in the area Pr≈$10^4 \div 10^5$.

The general nature of the dependence of the relative heat transfer $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ on the Prandtl number for small Reynolds numbers is that it first increases, reaching a maximum, and then steadily decreases up to the maximum values of the Prandtl number. For high Reynolds numbers, there is a steady decrease in the relative heat transfer, first sharply (up to Pr≈15), and then quite insignificant up to the limiting values of the Prandtl numbers. For average Prandtl numbers, the nature of the dependence will be intermediate.

The general nature of the dependence of relative heat transfer on the Prandtl number for the same heights of turbulators, but for different relative steps between the turbulators, shows that the Prandtl number affects it the most for large turbulators and for small Reynolds numbers, this effect is least of all for low turbulators with large relative steps between them with large Reynolds numbers (in some areas it is practically not observed). For intermediate values, there are intermediate values.

The above conclusion $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ can be drawn by redistributing the data in table 1 for the same relative heights of the turbulators, but with different relative steps between the turbulators.

The general nature of the dependence of the relative heat transfer on the Prandtl number for the same steps between the turbulators, but for different relative heights of the turbulators shows that the Prandtl number, as a rule, affects it the most for large turbulators and for small Reynolds numbers, this effect is least of all for low turbulators with large relative steps between them with large Reynolds numbers. For intermediate values, there are intermediate values. The above conclusion $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ can be drawn by redistributing the data in Table 1, for the same relative steps between the turbulators, but for different relative heights of the turbulators.

Thus, the influence of the increased Prandtl number on the relative intensification of heat transfer is significant and depends both on the geometry of the channel (relative heights and steps of the protrusions) and on the coolant flow regime (Reynolds number): it can either increase by almost two thirds or decrease about half in the considered range.

Analysis of the calculation results for the proposed model of the relative heat transfer parameter for various Reynolds numbers Re = 10^4÷**5·10**^{5} for reduced (Pr<1÷0.0038) Prandtl numbers $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$.

Calculation results according to the proposed model of the relative heat transfer parameter $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ for various Reynolds numbers Re = 10^4÷$5{\cdot}10^5$ for reduced (0.0038<Pr<1) Prandtl numbers for d/D = 0.90÷0.98; t/D = 0.25÷1.00 are given in table 2.

General analysis of the data presented in Table 2, shows that the maximum increase in relative heat transfer $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ is observed in the region of Pr≈0.05 at high Reynolds numbers (Re = $5{\cdot}10^5$) for average relative steps between turbulators (t/D = 0.50): +17% for high turbulators (d/D = 0.90) and +22% for turbulators with d/D = 0.93; changing the values of the relative steps leads to a decrease in the relative heat transfer, and a decrease in the relative heights of the turbulators leads to an even greater decrease.

The minimum decrease in relative heat transfer is observed

Re	d/D	t/D	Pr					
			0.0038	0.005	0.05	0.67	0.72	1
Re = 10^4	d/D = 0,90	t/D = 0,25	0.566	0.559	0.569	0.920	0.934	1
Re = $5{\cdot}10^4$	d/D = 0,90	t/D = 0,25	0.511	0.515	0.699	0.995	0.996	1
Re = 10^5	d/D = 0,90	t/D = 0,25	0.528	0.541	0.823	1.018	1.015	1
Re = $5{\cdot}10^5$	d/D = 0,90	t/D = 0,25	0.684	0.723	1.094	1.046	1.038	1
Re = 10^4	d/D = 0,90	t/D = 0,50	0.716	0.682	0.634	0.944	0.955	1
Re = $5{\cdot}10^4$	d/D = 0,90	t/D = 0,25	0.589	0.589	0.748	1.0044	1.0041	1
Re = 10^5	d/D = 0,90	t/D = 0,25	0.593	0.606	0.878	1.018	1.015	1
Re = $5{\cdot}10^5$	d/D = 0,90	t/D = 0,25	0.768	0.814	1.170	1.054	1.045	1
Re = 10^4	d/D = 0,90	t/D = 1,00	0.751	0.716	0.696	0.968	0.974	1
Re = $5{\cdot}10^4$	d/D = 0,90	t/D = 1,00	1.004	1.013	1.014	1.024	1.020	1
Re = 10^5	d/D = 0,90	t/D = 1,00	0.647	0.667	0.947	1.025	1.021	1
Re = $5{\cdot}10^5$	d/D = 0,90	t/D = 1,00	0.840	0.883	1.090	1.076	1.065	1
Re = 10^4	d/D = 0,93	t/D = 0,25	0.686	0.670	0.630	0.920	0.934	1
Re = $5{\cdot}10^4$	d/D = 0,93	t/D = 0,25	0.606	0.599	0.708	0.988	0.991	1
Re = 10^5	d/D = 0,93	t/D = 0,25	0.590	0.593	0.803	1.009	1.008	1
Re = $5{\cdot}10^5$	d/D = 0,93	t/D = 0,25	0.696	0.733	1.099	1.049	1.040	1
Re = 10^4	d/D = 0,93	t/D = 0,50	0.699	0.680	0.661	0.946	0.956	1
Re = $5{\cdot}10^4$	d/D = 0,93	t/D = 0,50	0.897	0.865	0.868	1.016	1.013	1
Re = 10^5	d/D = 0,93	t/D = 0,50	0.623	0.633	0.881	1.018	1.015	1
Re = $5{\cdot}10^5$	d/D = 0,93	t/D = 0,50	0.967	1.005	1.218	1.062	1.051	1
Re = 10^4	d/D = 0,93	t/D = 1,00	0.863	0.822	0.751	0.970	0.975	1
Re = $5{\cdot}10^4$	d/D = 0,93	t/D = 1,00	0.726	0.717	0.817	1.006	1.005	1
Re = 10^5	d/D = 0,93	t/D = 1,00	0.715	0.724	0.931	1.016	1.013	1
Re = $5{\cdot}10^5$	d/D = 0,93	t/D = 1,00	0.844	0.880	1.136	1.057	1.050	1
Re = 10^4	d/D = 0,96	t/D = 0,25	0.741	0.737	0.710	0.934	0.945	1
Re = $5{\cdot}10^4$	d/D = 0,96	t/D = 0,25	0.627	0.620	0.723	0.986	0.989	1
Re = 10^5	d/D = 0,96	t/D = 0,25	0.609	0.613	0.808	1.008	1.007	1
Re = $5{\cdot}10^5$	d/D = 0,96	t/D = 0,25	0.708	0.740	1.078	1.045	1.037	1
Re = 10^4	d/D = 0,96	t/D = 0,50	0.826	0.817	0.770	0.959	0.966	1
Re = $5{\cdot}10^4$	d/D = 0,96	t/D = 0,50	0.703	0.697	0.804	0.996	0.997	1
Re = 10^5	d/D = 0,96	t/D = 0,50	0.682	0.687	0.879	1.013	1.0108	1
Re = $5{\cdot}10^5$	d/D = 0,96	t/D = 0,50	0.792	0.825	1.111	1.046	1.038	1
Re = 10^4	d/D = 0,96	t/D = 1,00	1.066	1.014	0.870	0.980	0.983	1
Re = $5{\cdot}10^4$	d/D = 0,96	t/D = 1,00	0.802	0.793	0.881	1.0009	1.0010	1
Re = 10^5	d/D = 0,96	t/D = 1,00	0.775	0.779	0.935	1.012	1.0106	1

Re = $5\cdot10^5$	d/D = 0,96	t/D = 1,00	0.873	0.901	1.099	1.037	1.031	1
Re = 10^4	d/D = 0,98	t/D = 0,25	0.971	0.978	0.960	0.990	0.992	1
Re = $5\cdot10^4$	d/D = 0,98	t/D = 0,25	0.730	0.727	0.787	0.983	0.987	1
Re = 10^5	d/D = 0,98	t/D = 0,25	0.702	0.703	0.840	1.004	1.004	1
Re = $5\cdot10^5$	d/D = 0,98	t/D = 0,25	0.749	0.775	1.055	1.039	1.032	1
Re = 10^4	d/D = 0,98	t/D = 0,50	1.005	1.003	0.968	0.991	1.001	1
Re = $5\cdot10^4$	d/D = 0,98	t/D = 0,50	0.822	0.818	0.864	0.993	0.995	1
Re = 10^5	d/D = 0,98	t/D = 0,50	0.794	0.795	0.907	1.008	1.007	1
Re = $5\cdot10^5$	d/D = 0,98	t/D = 0,50	0.837	0.860	1.071	1.033	1.028	1
Re = 10^4	d/D = 0,98	t/D = 1,00	1.007	1.001	0.983	0.996	0.997	1
Re = $5\cdot10^4$	d/D = 0,98	t/D = 1,00	0.887	0.885	0.919	0.997	0.998	1
Re = 10^5	d/D = 0,98	t/D = 1,00	0.868	0.869	0.948	1.007	1.006	1
Re = $5\cdot10^5$	d/D = 0,98	t/D = 1,00	0.904	0.922	1.062	1.026	1.021	1

Table 2: Calculated results of the relative heat transfer simplex for different Reynolds criteria Re = $10^4\div5\cdot10^5$ with reduced (0,0038<Pr<1) Prandtl criteria for projections with different relative heights d/D = 0,90÷0,98 and different relative steps of turbulators t/D = 0,25÷1,00.

in the region Pr≈0.0038 at average Reynolds numbers (Re = $5\cdot10^4\div10^5$): -46% for high turbulators (d/D = 0.90) with a large step between them (t/D = 1.00); with a decrease in the relative heights of the turbulators up to d/D = 0.90÷0.96, the decrease in the relative heat transfer is approximately 40%; a further decrease $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ in the relative height of the turbulator leads to smaller reductions in the relative heat transfer. A decrease of the order of -30% occurs at d/D = 0.90 and t/D = 1.00; d/D = 0.93 and t/D = 0.50; d/D = 0.96 and t/D = 0.25 for Prandtl numbers Pr≈0.05.

An increase in relative heat transfer at low Prandtl numbers occurs at high Reynolds numbers (Re = 5·104) and Prandtl numbers (Pr≈0.05): about +20% for turbulators of medium heights (d/D = 0.93) and medium steps between them (t/D = 0.50); an increase of the order of +15% occurs at d/D = 0.93 and t/D = 1.00; an increase of the order of +10% occurs at d/D = 0.93 and t/D = 0.25, as well as at d/D = 0.96 and t/D = 0.50.

The general nature of the dependence of relative heat transfer on reduced Prandtl numbers for the same heights of turbulators, but for different relative steps between turbulators, shows that the Prandtl number affects it the most at small Reynolds numbers and small steps between turbulators; this influence is much less on low turbulators with high Reynolds numbers. For intermediate values, there are intermediate values. The above conclusion can be drawn by redistributing $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ the data in Table 2 for the same relative heights of the turbulators, but with different relative steps between the turbulators.

The general nature of the dependence of relative heat transfer on reduced Prandtl numbers for the same steps between the turbulators, but for different relative heights of the turbulators shows that the Prandtl number, as a rule, affects it the most for small turbulators (d/D = 0.96 and d/D = 0.98) and for average Reynolds numbers, this influence is least of all at large Reynolds numbers. The above conclusion can be drawn by redistributing $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ the data in Table 2 for the same relative steps between the turbulators, but for different relative heights of the turbulators.

Thus, the effect of a reduced Prandtl number on the relative intensification of heat transfer is also noticeable, but to a somewhat lesser extent than for an increased Prandtl number, and depends both on the channel geometry (relative heights and steps of the protrusions) and on the coolant flow regime (Reynolds number): it can either increase by more than one-fifth or decrease close to half in the considered range.

Generalizing analysis of the calculation results according to the proposed model of the relative heat transfer parameter for the entire considered range of Prandtl numbers **P**r = 0.0038÷96432 $\frac{Nu/Nu_{ГЛ}}{(Nu/Nu_{ГЛ})|_{Pr=1}}$ (**P**r = $\sim10^{-3}\div\sim\mathbf{10}^{+5}$).

The above analysis shows that for increased (Pr>1) Prandtl numbers, the maximum increase in relative heat transfer, which can be quite noticeable, occurs at low Reynolds numbers, large relative heights of turbulators, small relative steps between turbulators, and for reduced (Pr<1) Prandtl numbers - for large Reynolds numbers, large relative heights of turbulators, large relative steps between turbulators. The minimum values of relative heat transfer for increased Prandtl numbers are observed at high Reynolds numbers for high turbulators with a large step between them, and for reduced Prandtl numbers - at medium Reynolds numbers for high turbulators with a large step between them.

The general nature of the dependence of the relative intensification on the Reynolds number is that, most often, an increase in the relative heat transfer at elevated Prantl numbers occurs at small steps between the turbulators, and a decrease occurs at large steps between the turbulators. For low Prandtl numbers, an increase in relative heat transfer occurs most often at large steps between the turbulators, and a decrease - at small ones.

For increased Prandtl numbers, the increase in relative heat transfer can be quite significant - several times higher than for reduced ones; and the decrease is about the same.

For overall clarity, Figure 8 and Figure 9 nine graphs based on Table 1 and Table 2 respectively. On the Figure 8 and Figure 9 the areas of increase and decrease in relative heat transfer are clearly visible depending on the Prandtl numbers, which were analyzed in this article. The above graphs confirm the conclusion that the influence of the Prandtl number on the relative enhanced heat transfer can be quite significant.

Partial (since the range of theoretical data is much wider than the experimental range) experimental confirmation of the given theoretical data was given in [7,8,20,21,26], where the authors' experiments, experiments [27], as well as regularities for the limiting heat transfer [7,8,20,21,26]. Another partial confirmation of the obtained calculated numerical dependences is the data of analytical solutions for enhanced heat transfer obtained from a modified four-layer model of a turbulent boundary layer [2,11,18,19]. In addition, classical works on enhanced heat transfer [7,8,26] indicate that there are no reliable experimental data, but it is assumed that artificial turbulence of liquid metal flows should, as a rule.

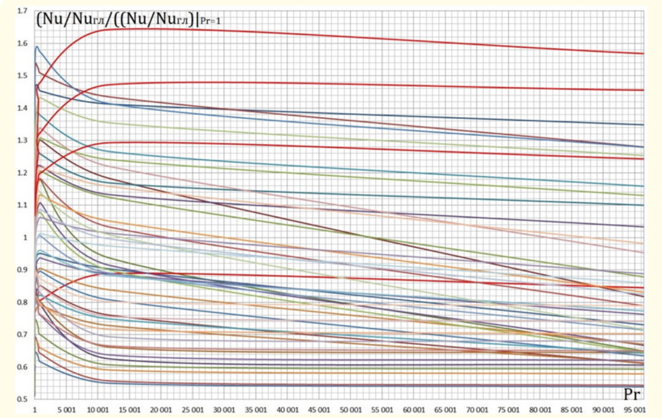

Figure 8: Graphs of relative heat transfer as a dependence on the increased values of the Prandtl criterion.

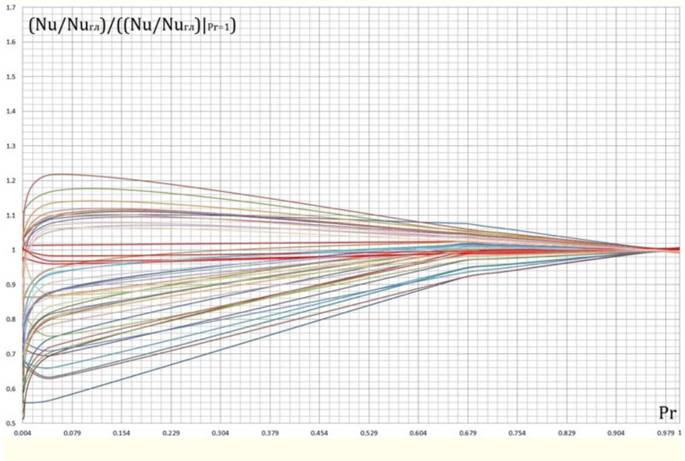

Figure 9: Graphs of relative heat transfer as a dependence on the reduced values of the Prandtl criterion.

The above analysis indicates that the theoretical data are fully consistent with the existing experimental material, significantly overlapping the range of the latter's defining parameters. Theoretical data made it possible to reveal the patterns of relative heat transfer depending on the Prandtl number in those areas where there are no reliable experimental data yet that allow predicting the ranges of increase and decrease in intensified heat transfer.

Main Conclusions

- The calculation method developed and used in this study, based on the solution of the Reynolds equations by the finite volume method, closed using the Menter shear stress transfer model and the energy equation on different-scale intersecting structured grids, made it possible to calculate with acceptable accuracy the relative heat transfer in pipes with semicircular ring turbulators for coolants with different Prandtl numbers.
- In the study, an analysis was made of the calculated dependences of the relative heat transfer on the Prandtl number Pr for various values of the relative height of the turbulator h/D, the relative step between the turbulators t/D, for various values of the Reynolds number Re, other things being equal, which showed qualitative and quantitative changes in the calculated parameters.
- The calculations carried out in the work showed that with an increase in the Prandtl number at low Reynolds numbers, at first there is a noticeable increase in the relative heat transfer, and then the relative heat transfer changes less, and for small steps it increases, for medium steps it almost stabilizes, for large steps - slight decrease.
- At high Reynolds numbers, the relative heat transfer decreases with an increase in the Prandtl number with its further stabilization.
- Analytical substantiation of the obtained calculated regularities is that at small Reynolds numbers, the height of the turbulator is greater, and at large - less than the height of the near-wall layer, therefore, only the core of the flow is turbulized, which leads only to an increase in hydraulic resistance and to a non-increase in heat transfer.
- The theoretical data obtained in the work determined the

laws of relative heat transfer in a wide range of Prandtl numbers, including those areas where there is no experimental material yet.

- For increased (Pr>1) Prandtl numbers, the maximum increase in relative heat transfer, which can be quite noticeable, occurs mainly at low Reynolds numbers, average relative heights of turbulators, and small relative steps between turbulators; and for reduced (Pr<1) Prandtl numbers - for large Reynolds numbers, large relative heights of turbulators, large relative steps between turbulators. The minimum values of relative heat transfer for increased Prandtl numbers are observed at high Reynolds numbers for high and medium turbulators with a large and medium pitch between them, and for reduced Prandtl numbers - at medium Reynolds numbers for high turbulators with a large pitch between them.
- For increased Prandtl numbers, the increase in relative heat transfer can be quite significant - several times higher than for reduced ones; and the decrease is about the same.

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