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Research Paper

Numerical Modeling of Heat Transfer and Hydrodynamics in Compact Shifted Arrangement Small Diameter Tube Bundles

Valery Gorobets[1], Viktor Trokhaniak[1], Yurii Bohdan[2], Ievgen Antypov[1]

[1] Department of Heat and Power Engineering, National University of Life and Environmental Sciences of Ukraine, Heroyiv Oborony st., 15, Kyiv, 03041, Ukraine, Emails: gorobetsv@ukr.net, trohaniak.v@gmail.com, ievgeniy_antypov@ukr.net

[2] Department of Vessel's Power Plants Operation, Kherson State Maritime Academy, Ushakova avenue, 20, Kherson, 73000, Ukraine, Email: bohdanyurii09@gmail.com

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Corresponding author: V. Trokhaniak (trohaniak.v@gmail.com)

Abstract. Numerical modeling of heat and hydrodynamics processes in the channels of compact small diameter tube bundles with different transverse shifted arrangement is carried out. The fields of velocities, temperatures, and pressures in the tube bundle channels were obtained, and their influence on heat transfer conditions and hydraulic losses were analyzed. The calculation of the thermohydraulic efficiency for different constructions of the tube bundles had been carried out, and their results with data of well-known tube bundles of different geometry are compared.

Keywords: Heat exchanger, Tube bundle, Thermohydraulic efficiency, Numerical modeling, Hydraulic losses.

1. Introduction

Toward creating new construction of heat exchangers, different factors play an important role. These include mass and size characteristics, the efficiency of heat transfer through heat surface, losses of pressure in channels each heat-carriers, and other parameters, which were previously investigated by Zhukauskas A.A. [1]. The same important part plays as an analysis tool for effectiveness evaluation of heat exchangers known as thermohydraulic efficiency criterion [2-8] that presents a ratio between heat exchanger heat output and power required to pump the heat-carriers through its channels. Another high-performance analysis tool for determining the efficiency of heat exchangers design is detailed numerical modeling of heat transfer and hydrodynamics processes in its channels [9-18]. Such modeling makes it possible to analyze the local hydraulic and thermal characteristics on the heat transfer surfaces. Moreover, it allows determining heat transfer surfaces thermal efficiency, as well as to evaluate the hydraulic losses that define the pump's power needful to pump heat-carriers.

The most common design of heat exchangers, which are mainly used in heat-power equipment are recuperative heat exchangers. According to their constructive differences, these heat exchangers are divided into two types shell and tube and plate. Each of these types has its benefits, and drawbacks and selection politics depend on their hydrodynamic and temperature operating modes.

Shell and tube heat exchangers, as a rule, have a staggered or inline arrangement of tubes in a bundle. The conditions of hydrodynamic flow and heat transfer for them were investigated in many papers [19-22]. However, hydrodynamics and heat transfer processes in channels of compact shifted arrangement small diameter tube in bundles are insufficient studied. It is worth noting that the use of such bundles in the shell and tube heat exchangers makes it possible to improve their mass-dimensional and cost characteristics were compared to well-known.

The purpose of the research is to develop a new design of compact shifted arrangement small diameter tube bundles and modeling of heat transfer and hydrodynamics processes in its channels. In addition, this paper contains the calculation of thermohydraulic efficiencies and Reynolds analogy factors for proposed tube bundles that had been made to evaluate their effectiveness.

2. Materials and Methods

Consider shifted arrangement small diameter tube bundles with transverse flow through the channels by Gorobets V.G. et al. [23]. The schematic configuration of the considered structure of small diameter tube bundles arranged in heat exchanger housing is presented in Fig. 1.

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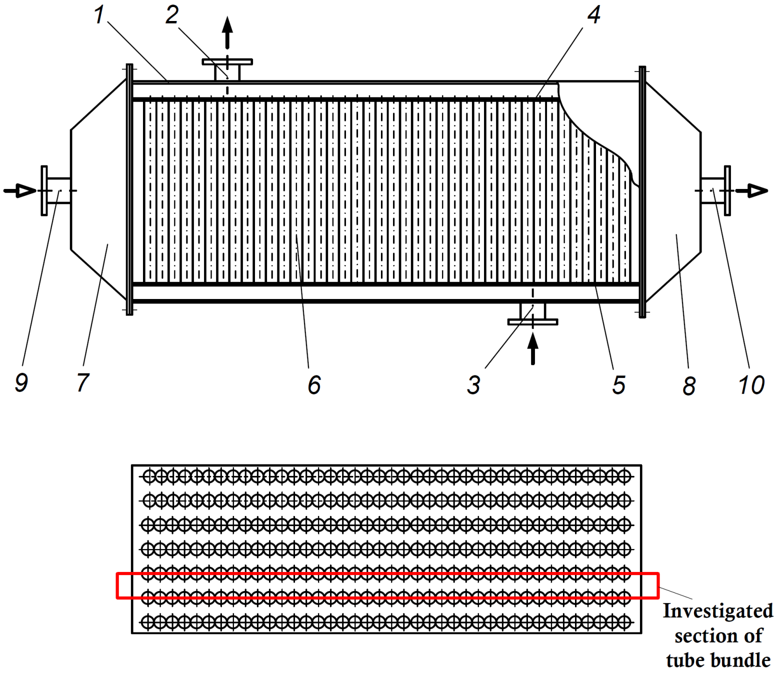


Fig. 1. Schematic configuration of the considered structure of small diameter tube bundles arranged in heat exchanger housing: 1- heat exchanger housing; 2,3 - inlet and outlet fittings for the first heat-carrier; 4,5 - tube plates; 6 - vertical tube bundle; 7,8 - end covers of the heat exchanger; 9,10 - inlet and outlet fittings for the second heat-carrier

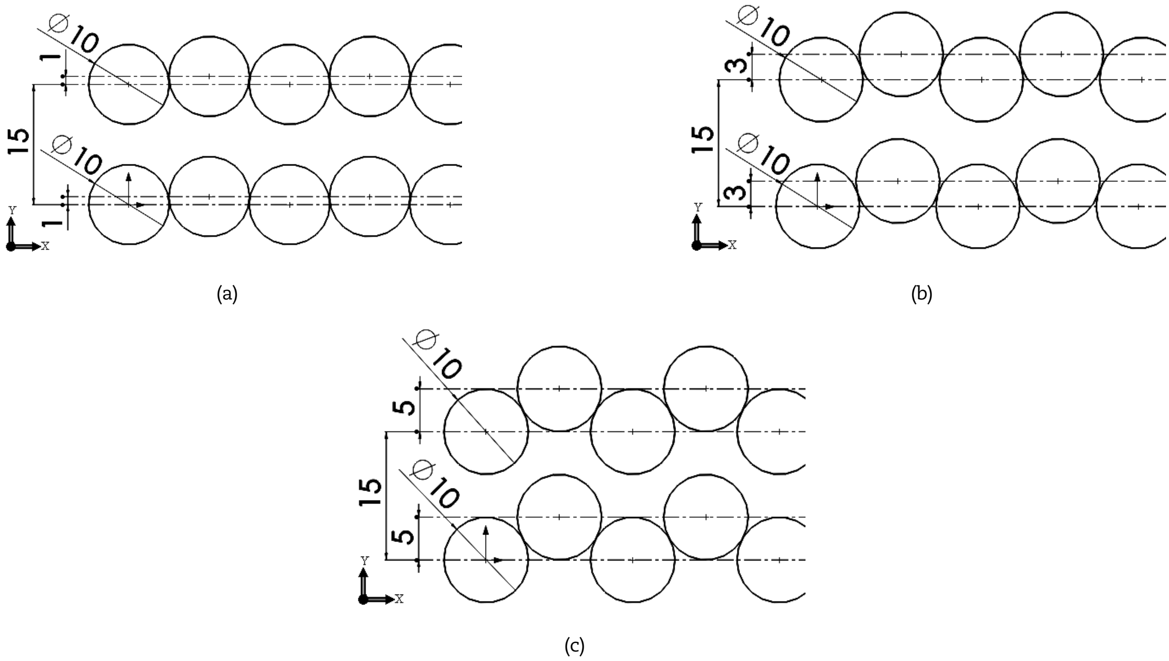


Fig. 2. Geometries of compact shifted arrangement small diameter tube bundles (top view): a – displacement of tubes by 1 *mm*, b – displacement by 3 *mm*, c – displacement by 5 *mm*

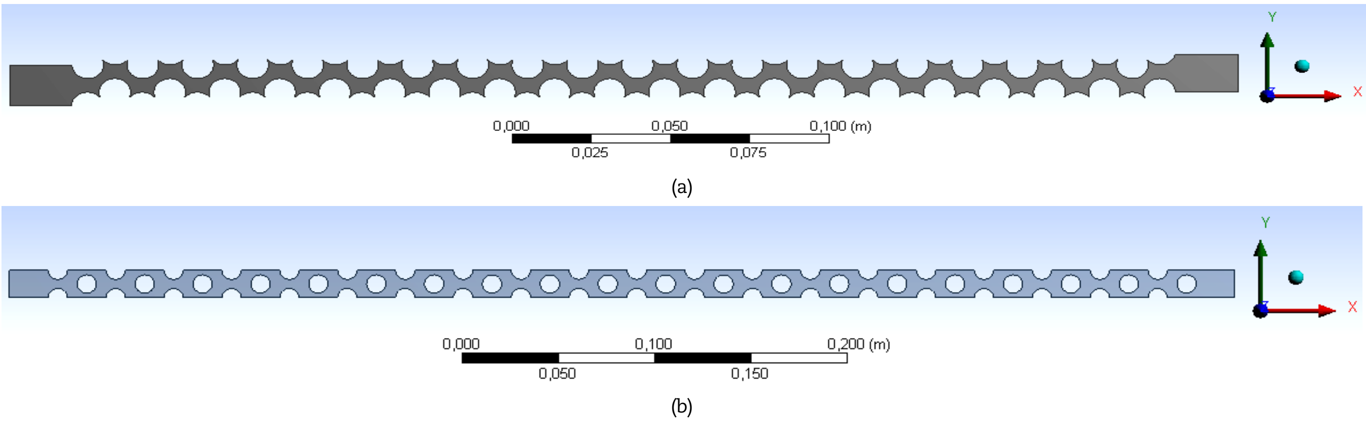


Fig. 3. (a) Geometry analytical models of compact shifted arrangement small diameter tube bundle and (b) a staggered arrangement tube bundle.

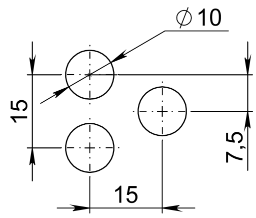
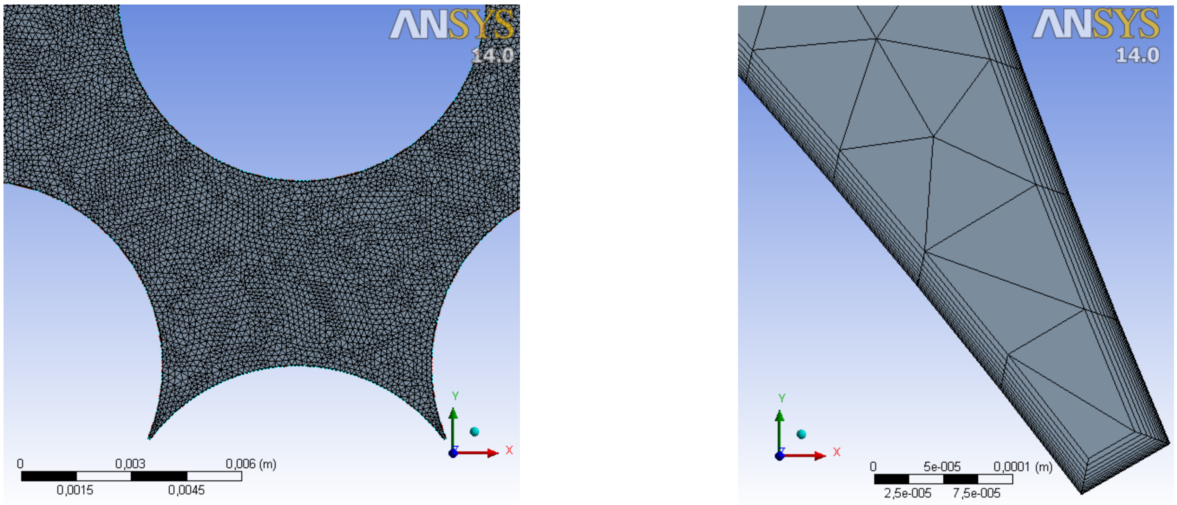


Fig. 4. The geometry of staggered arrangement tube bundle.



(a) (b)

Fig. 5. Triangular type mesh of small diameter tube bundle in 2D: a – displacement of tubes by 5 *mm*, b – boundary layer.

The geometry of the arrangement of tube bundles with a diameter of tube d =10 *mm* is shown in Fig. 2, which differs from the traditional inline and staggered arrangement tube bundles. The primary difference is the fact that between adjacent tubes, there is no inter-tube gap, and in addition they are shifted in relation to each other at a certain distance.

2.1 Numerical modeling of heat transfer and hydrodynamics processes in the channels of investigated tube bundles

In the course of the work, the numerical modeling of heat transfer and hydrodynamics processes in the channels of shifted arrangement small diameter tube bundles is considered, and the estimation of the thermohydraulic efficiency of various geometries of these bundles is carried out.

Numerical modeling of the staggered arrangement tube bundle is carried out to test the adequacy of the model and compare it to known experimental data by Zhukauskas A.A. [1] for the further accurate modeling of proposed design tube bundles. The geometry of the staggered arrangement of a tube bundle is presented in Fig. 4.

The mathematical model based on Unsteady Reynolds Averaged Navier-Stokes equations commonly referred to as (URANS) for incompressible liquids by Khmelnik S.I. [24] and the equation of convective energy transfer. It is assumed that the flow in the inter-tubular channels is turbulent. For that reason, a standard k-ε turbulence model (KES) is used [25-27].

The calculations were carried out for five geometries with displacement in the transverse direction of adjacent tubes in longitudinal rows by an amount of 1,2,3,4 and 5 *mm* respectively (see Fig. 2, a, b, c).

All calculations are performed at the value of Reynolds number Re =18600, where $Re = w \cdot D_{chan.eq} / \nu$ – Reynolds number; w -; ν - $D_{chan.eq}$=4A/X- equivalent diameter of the inter-tube channels; A – sectional area, m^2; X– perimeter, m. As one heat-carrier was selected, air with a temperature +40 ˚C at the inlet, which flows through the channels and another heat-carrier – water which moves inside tubes and has a temperature at the inlet +10 ˚C. Similar conditions for the cooling heated air are taking place, for instance, in conditioning systems in the summer period for accommodations of different designations. The heat-carriers flow schematic is crossflow.

Meshes generated in ANSYS Workbench Meshing designer. Triangular type mesh in 2D utilized, as one can see in Fig. 5, a, b. For boundary layer creation Total Thickness method is utilized. The thickness of the 1st layer is $2\ 10^{-5}$ *m*, and layers quantity is 11 *pcs* as depicted in Fig. 5, b. Orthogonal quality meshes for all tube bundle geometries lay in a range from 0.458 to 0.592 of the orthogonal quality mesh metrics spectrum. That is an indication of good quality. Detail data of the generated meshes are presented in Table 1.

The accuracy of simulations (CFD-calculations) was estimated by comparing their results with the experimental data. Moreover, before performing the investigation CFD-calculations, the test calculations (verification) were carried out. That is for checking the developed CFD-model for adequacy, by comparison, the calculated characteristics with the known ones from the physical experiments [18, 28-30]. Test calculations were performed for the cases that are as close as possible to the subjects, which include all the physical processes that take place in the investigated tasks. The verification was carried out during the calculations with different meshes and with different turbulence models, so the calculated parameters of the calculation models were determined, which ensure the best match of the calculation results with the experimental data.

2.2 Estimation of the thermohydraulic efficiency of compact small diameter tube bundles

Estimation of heat transfer surface efficiency from the energy point of view uses the ratio of the quantity of heat transmitted through the heat transfer surface to the value of power required for feeding heat-carriers through the heat exchanger channels by Gorobets V.G. [2]:

$$E = Q/N \tag{1}$$

Table 1. Generated meshes data.

Tube bundle characteristic parameter*	1 (*mm*)	2 (*mm*)	3 (*mm*)	4 (*mm*)	5 (*mm*)	1.5×1.5
Mesh type	Triangles					
Size function	Proximity and Curvature					
Max face size, (*m*)	$1.5\cdot10^{-4}$	$1.5\cdot10^{-4}$	$1.5\cdot10^{-4}$	1.5·10-4	1.5·10-4	2.5·10-4
Max size, (*m*)	$3\cdot10^{-4}$	$2\cdot10^{-3}$	$3\cdot10^{-4}$	3·10-4	3·10-4	5·10-4
Inflation option	Total Thickness					
Number of layers, (*pcs*)	11					
Maximum thickness, (*m*)	2·10-5					
Nodes	252535	248841	242424	233504	218581	298103
Elements	408628	401120	383482	368754	337821	552782
Mesh metric Orthogonal Quality	0.458	0.514	0.505	0.499	0.471	0.592

*Displacement in *mm* for shifted arrangement tube bundles and multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundles.

Criterion E is called the thermohydraulic efficiency coefficient of the heat transfer surface. The quantity of heat Q that is taken from the hot or transferred to the cold heat-carrier can be determined by the formula:

$$Q = c_p G \Delta T \tag{2}$$

where $\Delta T = T_{in} - T_{out}$ – temperature difference of the heat-carriers in heat exchanger channels; T_{in}, T_{out} – the temperature of heat-carrier at the inlet and outlet of the channels, *K*; *G* – mass flow rate of heat-carrier, *kg/s*; c_p – heat capacity of heat-carrier, *kJ/(kg·K)*.

Power *N* required for feeding heat-carriers through the heat exchanger channels, which is included in Eq. (1) can be found from the expression:

$$N = \Delta p G / \rho \tag{3}$$

where Δp – differential pressure at the inlet and outlet of the channel, *kPa*; ρ is the density of the medium, *kg/m³*.

Thermohydraulic efficiency coefficient Eq. (1) is recorded, using the given Eqs. (2) and (3), as:

$$E = c_p \rho \Delta T / \Delta p \tag{4}$$

In order to compare the values of the energy efficiency of different types of heat transfer surfaces, in addition to thermohydraulic efficiency, the Reynolds analogy factor (FAR) is used, which is presented in the form [31-33]:

$$FAR = \frac{\overline{Nu}/\overline{Nu_0}}{\overline{\xi}/\overline{\xi_0}} \tag{5}$$

The given parameter makes it possible to compare the thermohydraulic efficiency of the investigated surface with a certain reference surface. A straight, smooth channel (denoted by the index "0") is chosen as the reference surface. Values of average Nusselt number $\overline{Nu_0}$ and hydraulic resistance coefficients $\overline{\xi_0}$ for the straight channels are calculated for the stabilized turbulent flow according to the known formulas [2, 31]. Values of average Nusselt number $\overline{Nu}$ and hydraulic resistance coefficients $\overline{\xi}$ for the investigated curvilinear channels are calculated at the same formulas and values of average Reynolds number as for straight channels.

3. Results and Discussion

Numerical modeling of both staggered and shifted arrangement tube bundles is performed under the same boundary conditions.

Fig. 6 depicted the velocity field in the channels of the staggered arrangement of the tube bundle in the range from 0 to 36 *m/s*. Figure 7 displays temperature changes along the channels of the staggered tube bundle varying in the range from +40 to +11.85 °C. Figure 8 shows that the total pressure drop is about 21500 *Pa*.

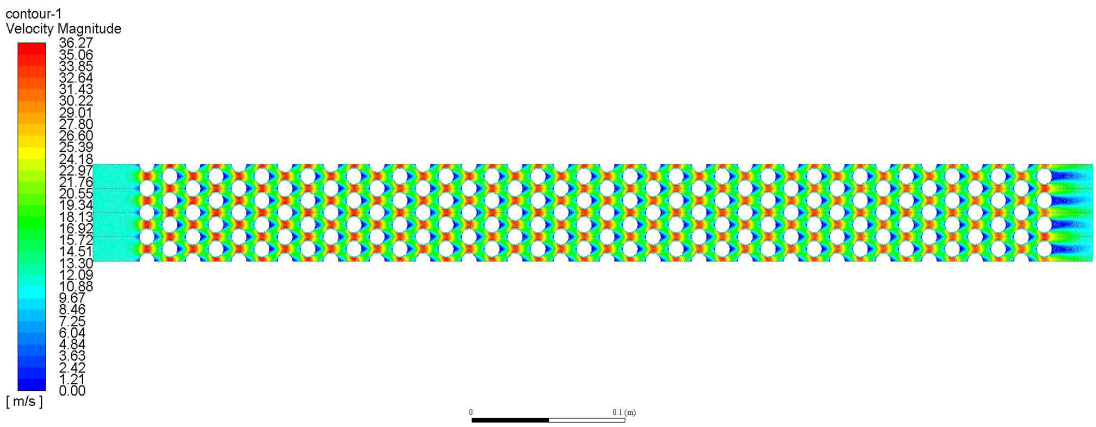


Fig. 6. Velocity field in the channels of the staggered arrangement of the tube bundle, *m/s*.

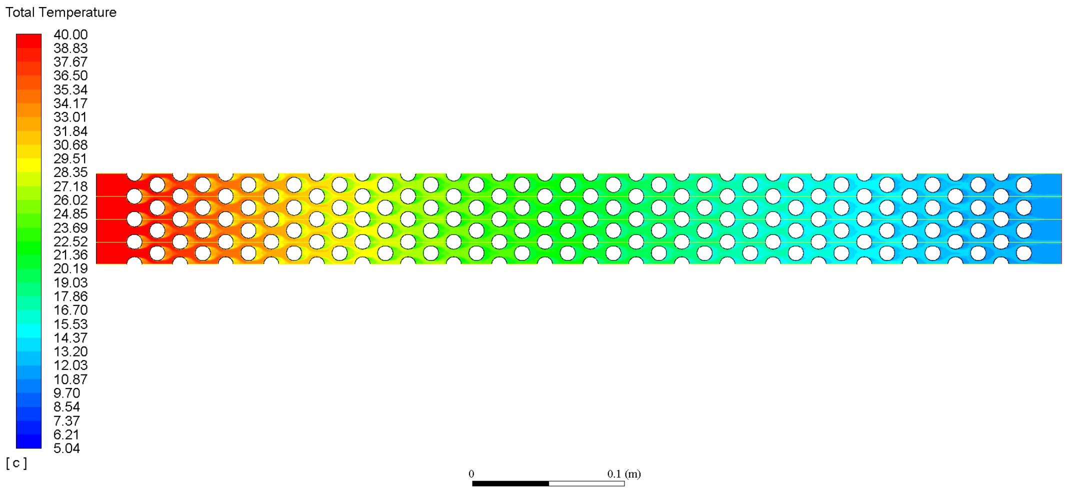


Fig. 7. Temperature field in the channels of the staggered arrangement of the tube bundle, °C.

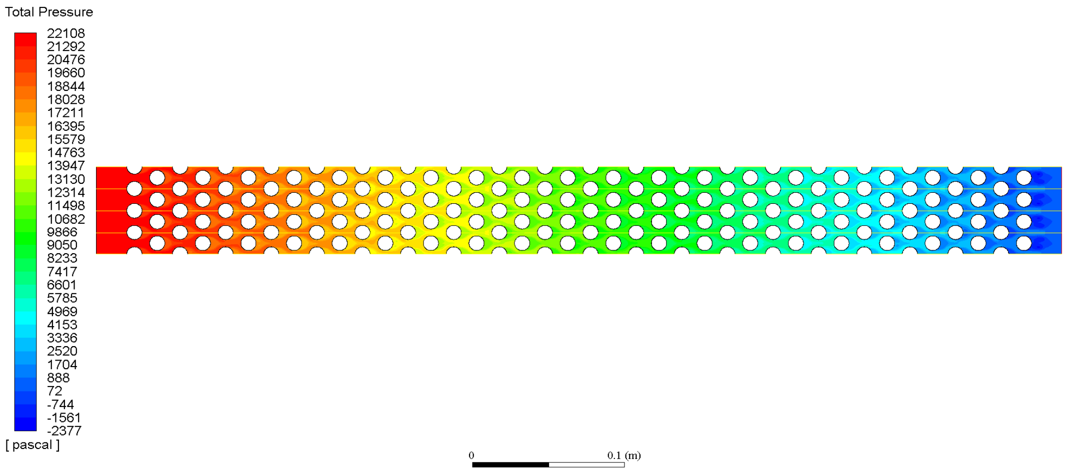


Fig. 8. Pressure field in the channels of the staggered arrangement tube bundle, *Pa*.

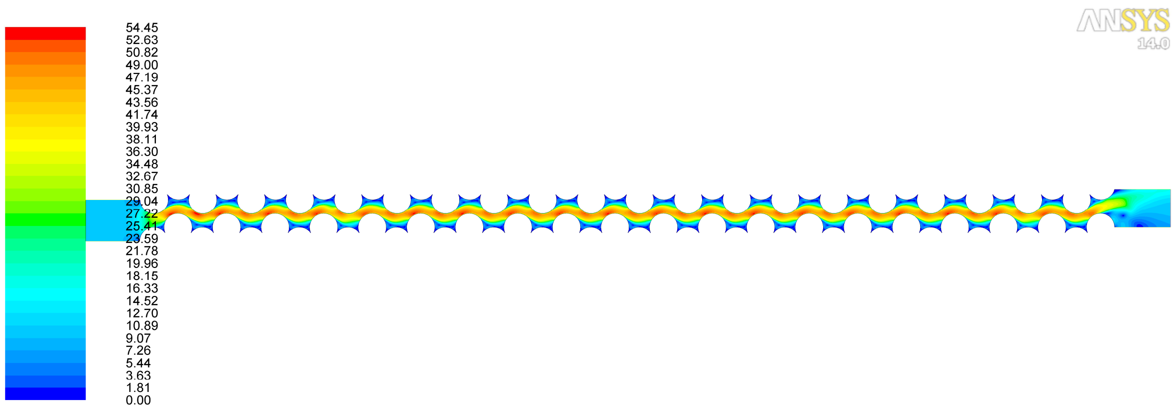


Fig. 9. Velocity field in the shifted arrangement tube bundle channel, m/s.

The calculation results for a channel with a shift of adjacent tubes by 5 mm are shown in Figs. 9-13. The velocity field in the shifted arrangement tube bundle channel is shown in Fig. 9 and in Fig. 10 – the distribution of velocity vectors in a separate magnitude element of the shifted arrangement tube bundle channel. As can be seen from Fig. 10 at the upper point of the tube, there is a separation of the boundary layer, and at the junction of adjacent tubes, there are stagnant zones. At some points of the channel, the air velocity reaches 55 *m/s,* and the average air velocity in the narrow cross-section of the channel is about 45 *m/s* (see Fig. 10).

Figure 11 displays the air temperature distributions in the channel. The temperature of the cooled air at the outlet of this channel is +10.1 *°C*. This is the lowest outlet temperature compare to temperatures that are reached in the channels of other tube bundles geometries with a displacement of 1,2,3 and 4 *mm*. This is due to the highest values of heat transfer coefficient for the bundle with tube displacement of 5 *mm* compared with the bundles for other displacements. Figure 12 shows the pressure field in the channel of the investigated tube bundles geometry. From the obtained pressure field follows that the total pressure drop is about 15700 *Pa*. In addition, Fig. 13 depicts the streamlines in the channel.

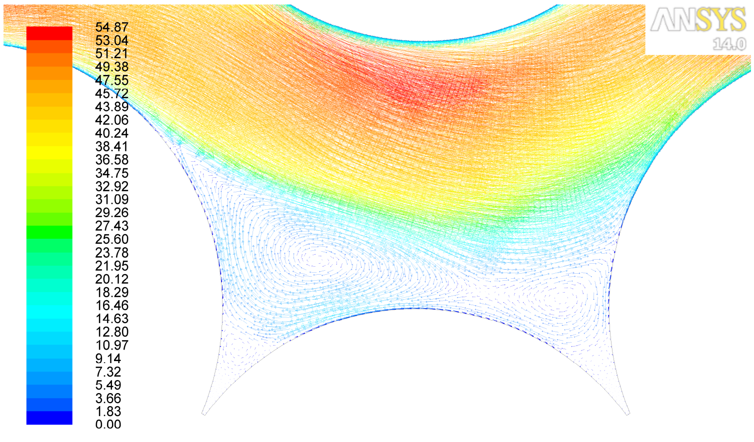


Fig. 10. Distribution of the velocity vectors in a separate magnitude element of the shifted arrangement tube bundle channel, *m/s*.

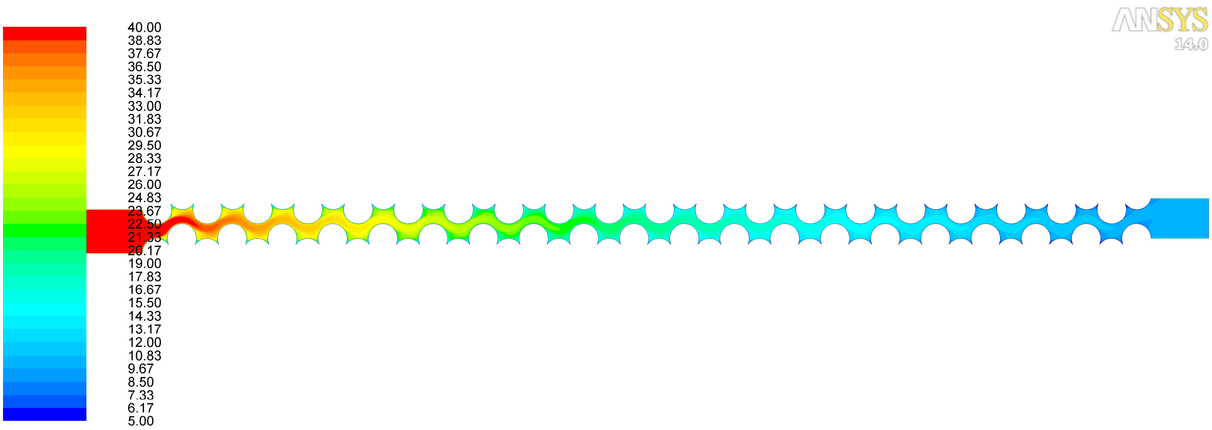


Fig. 11. Temperature field in the shifted arrangement tube bundle channel, °*C*.

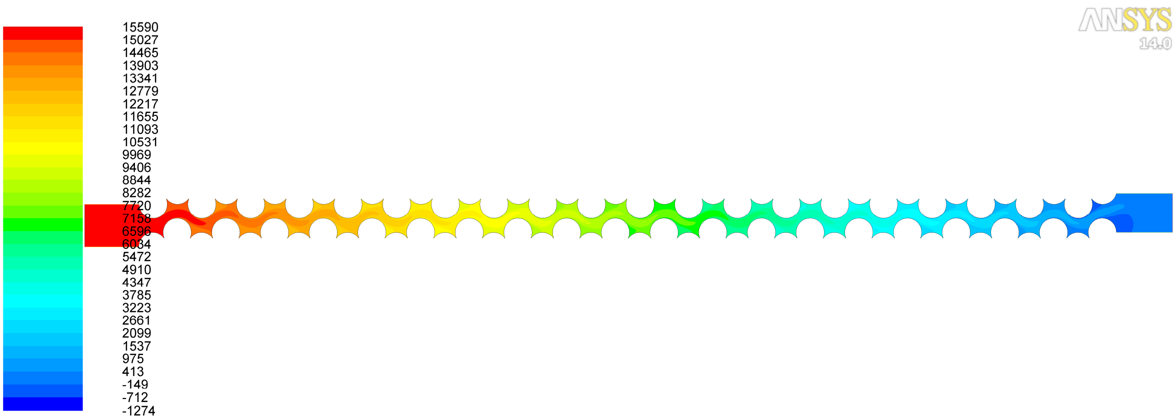


Fig. 12. Pressure field in the shifted arrangement tube bundle channel, *Pa*.

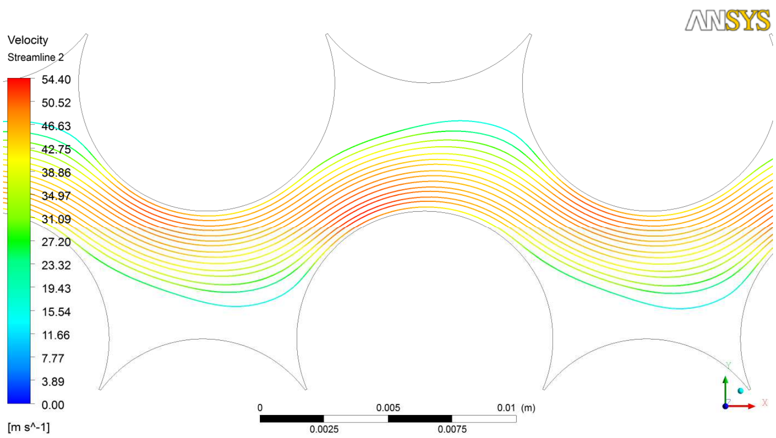


Fig. 13. Streamlines of airflow the shifted arrangement tube bundle channel, *m/s*.

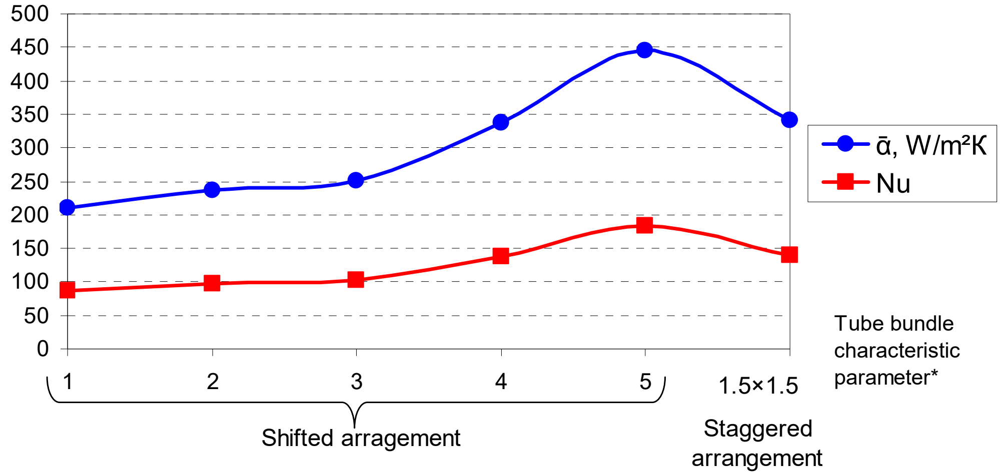


Fig. 14. Dependence of the Nusselt number and the heat transfer coefficient on the tube bundle characteristic parameter.

*Tube displacement in *mm* for shifted arrangement tube bundles and nondimensional multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundles.

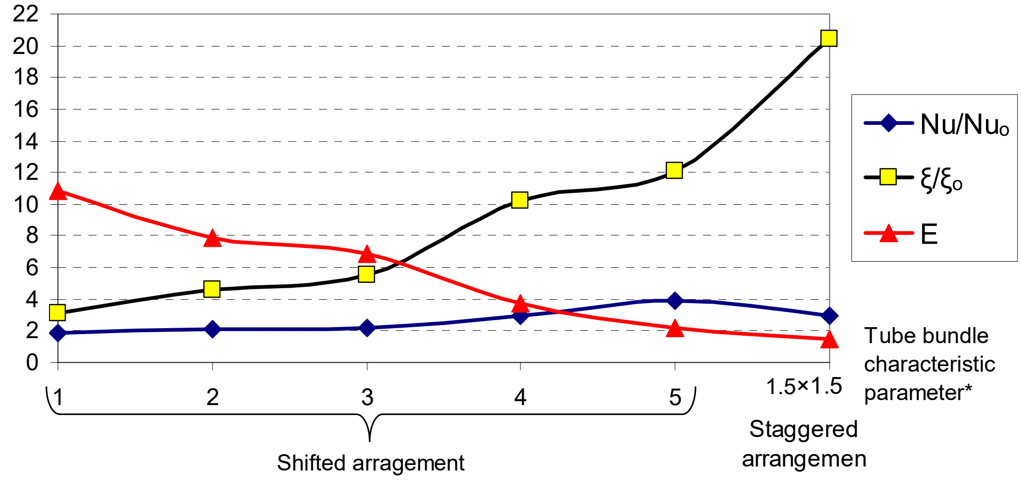


Fig. 15. Dependences of thermohydraulic efficiency, Nusselt numbers ratios, and hydraulic resistance coefficients on the tube bundle characteristic parameter.

*Tube displacement in *mm* for shifted arrangement tube bundles and nondimensional multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundles.

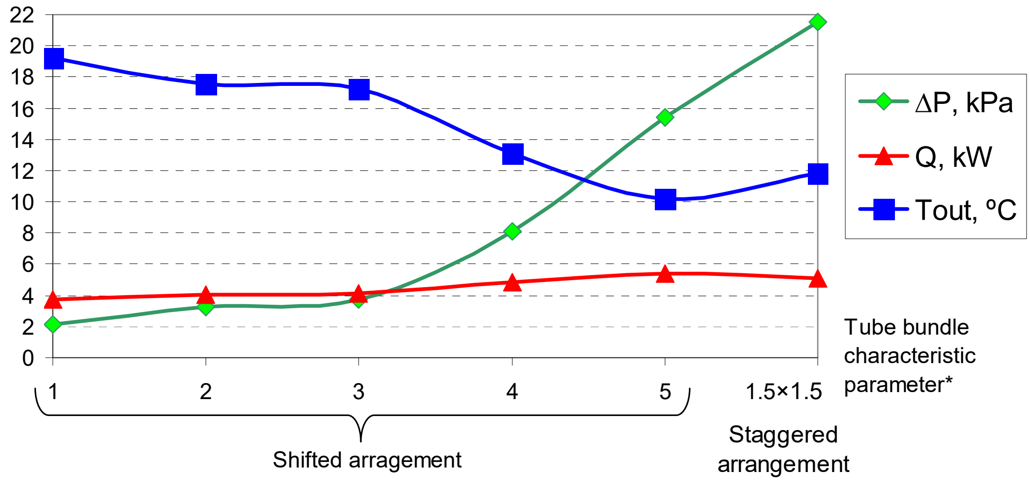


Fig. 16. Dependences of pressure drop *Δp*, the total quantity of heat *Q*, and the channel outlet heat-carrier temperature T_{out} at on the tube bundle characteristic parameter.

*Tube displacement in mm for shifted arrangement tube bundles and nondimensional multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundles.

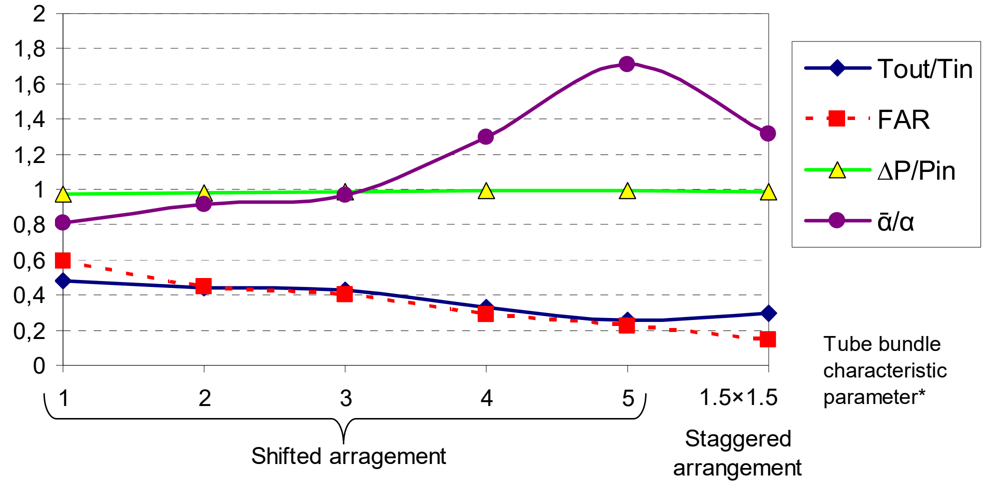


Fig. 17. The Reynolds analogy factor (FAR) and the dimensionless values of the outlet and inlet temperature ratios, pressure drop and heat transfer coefficients on the heat transfer surface of the investigated curvilinear channels of compact shifted arrangement tube bundles.

Table 2. Comparison of the thermal and hydraulic characteristics of compact shifted arrangement and staggered arrangement tube bundles.

Tube bundle characteristic parameter*	T_{in}, (°C)	T_{out}, (°C)	ΔP, (*kPa*)	ᾱ, (*W/m² K*)	Nu	FAR	E
1 *mm*	40	19.1964	2.171465	211.69249	87.03885	0.59321	10.87623
2 *mm*	40	17.5905	3.225801	238.60588	98.10449	0.44816	7.886554
3 *mm*	40	17.2233	3.76125	252.71292	103.9047	0.39935	6.874661
4 *mm*	40	13.1045	8.106872	338.54631	139.1957	0.28762	3.766335
5 *mm*	40	10.1402	15.44429	447.97947	184.1899	0.22097	2.194884
6 *mm*	40	7.3657	47.09984	690.26084	283.8056	0.12932	0.786588
Staggered arrangement	40	11.85	21.51949	325.88387	133.9894	0.14578	1.485043

* Displacement in *mm* for shifted arrangement tube bundles and multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundle.

*Tube displacement in *mm* for shifted arrangement tube bundles and nondimensional multiplication of transverse and longitudinal pitch-to-diameter ratios for staggered arrangement tube bundles.

Numerical modeling makes it possible to analyze the conditions of hydrodynamic flow and heat transfer processes in the investigated tube bundle channels and to determine which of these geometries are most effective.

Figure 14 displays the dependences of the Nusselt number and the heat transfer coefficient on the tube bundle characteristic parameter. Abscissa axis presents tube bundles characteristic parameter for shifted tube bundles where the displacement of adjacent tubes is marked respectively by 1, 2, 3, 4, and 5 *mm* and for staggered arrangement tube bundle nondimensional multiplication of transverse and longitudinal pitch-to-diameter ratios. Average values of heat transfer coefficient $\bar{\alpha}$ and Nusselt number $Nu=\bar{\alpha}d/\lambda$ on the heat transfer surface of tube bundles are located along the ordinate axis, where λ is the thermal conductivity of heat-carrier. As a result of the analysis of the obtained dependencies, one can conclude that the most effective heat transfer on the heat transfer surface of the tube bundle corresponds to the bundle with a tube displacement of 5 *mm*.

Figure 15 shows the ratios of Nusselt numbers Nu, hydraulic resistance coefficients in the inter-tubular channel ξ , and the corresponding values in a smooth channel (indicated by the index "0"). Figure 15 also shows the thermohydraulic efficiency E as a function of the tube bundle characteristic parameter (see eq. (4)).

Dependences of Nusselt numbers ratios and hydraulic resistance coefficients on the tube bundle characteristic parameter (as can be seen in Fig. 15) rises with increasing displacement between the adjacent tubes. In the investigated curvilinear channels with a maximum displacement of adjacent tubes by 5 *mm,* an increase in the heat transfer coefficient on the tube surface of compact bundles is almost four times upper as compare to a straight channel. At the same time, as can be seen from Fig. 15 the value of hydraulic resistance increases eight times which leads to decreases in the thermal and hydraulic efficiency of tube bundles.

Figure 16 shows dependencies of the cold heat-carrier temperature changes in the outlet of the channel, the pressure drop, and the quantity of heat transmitted through investigated heat transfer surfaces on the tube bundle characteristic parameter. The dependences obtained in the analysis reveals that the temperature of cold heat-carrier in the outlet of the curvilinear channel (displacement 5 *mm*) is significantly reduced. The total quantity of heat transferred from the hot heat-carrier to cold heat-carrier increases simultaneously. The pressure drop of this channel increases up to 7 times (see Figs. 15 and 16). At once, the pressure drop increases only 0.157 *bar* that does not lead to a significant increase in the pump or fan power used for feeding heat-carriers.

Calculated values of the FAR, dimensionless values of the outlet and inlet temperature ratios, pressure drop, and heat transfer coefficients on the heat transfer surface of the investigated curvilinear channels of compact shifted arrangement tube bundles are shown in Fig. 17.

During analyzing obtained results for compact shifted arrangement tube bundles with a displacement of adjacent tubes in the range from 1 to 5 *mm,* some interest was in the further investigation of other displacements of adjacent tubes, for instance, 6 *mm*. The purpose of the further displacement of adjacent tubes in 6 *mm* is to define reasonable limits of pressure drop and heat transfer characteristics.

The thermal and hydraulic characteristics of compact shifted arrangement tube bundles with a displacement of adjacent tubes in the range from 1 to 6 *mm* and for a 1.5×1.5 staggered arrangement tube bundle is given in comparative Table 2.

As indicated in Table 2, the displacement of adjacent tubes by 6 *mm* or more leads to a significant increase of pressure drop in the tube bundles and substantially exceed the pressure drop even in the staggered arrangement of the tube bundle. In this case, the value of thermohydraulic efficiency E and parameter FAR also decreases and becomes smaller in comparison to the staggered tube bundle. Therefore, the design of compact bundles with a displacement of adjacent tubes 6 *mm* or more is inefficient.

Table 3. Comparison of tube bundles' total length depending on the characteristic parameter.

Tube bundle characteristic parameter	1 (*mm*)	2 (*mm*)	3 (*mm*)	4 (*mm*)	5 (*mm*)	1.5×1.5
Tube bundle total length, *m*	0.43805	0.43212	0.42204	0.40744	0.38695	0.655

The analysis of the obtained dependences (see Figs. 14-17) shows that from the point of view of the intensification heat transfer processes, the best characteristics have a construction with a displacement of tubes by 5 *mm* (see Fig. 2, c). The averaged values of the heat transfer coefficient $\bar{\alpha}$ for this tube bundle heat transfer surface are in 2.1 times greater than the corresponding values of $\bar{\alpha}$ for the construction with a displacement of tubes by 1 *mm*. In this case, a significant decrease in the temperature of the cold heat-carrier at the channel outlet is achieved. Hydraulic losses with an increase in the displacement value between adjacent tubes are increasing, but in the absolute value of a significant drop in total pressure, there is no observed, and this does not lead to a significant increase of the pump power for feeding the heat-carriers. Since the growth rate of hydraulic resistances exceeds the growth rate of heat transfer coefficients, the parameter of the FAR is somewhat reduced for the geometry of the channel with a large displacement of the tubes.

Table 3 presents comparisons between tube bundles' total length for different tube bundle characteristic parameters.

The compactness of the proposed tube bundles can reduce the total length of the tube bundle by 0.387 *m* in comparison with the length of the traditional staggered tube bundle, which is 0.655 *m*.

4. Conclusion

1. Computational modeling of heat transfer and hydrodynamics processes in compact shifted arrangement small diameter tube bundles of different geometries using the software ANSYS Fluent is carried out. The fields of velocities, temperatures, and pressure in the investigated channels are obtained. The conditions of the hydrodynamic flow in the channels were analyzed and estimated the heat transfer intensity between the hot and cold heat-carriers through the heat transfer surface.

2. Comparative analysis of the Nusselt numbers, the hydraulic resistance coefficients, the thermohydraulic efficiencies, and Reynolds analogy factors for compact shifted arrangement small diameter tube bundles with different geometries configurations have been carried out. It is shown that the intensification of the heat transfer processes in the channels of the tube bundles is accompanied by the growth of the hydraulic resistance. However, the absolute values of the total pressure drop are small and do not lead to a significant increase in the pump power for feeding heat-carriers in heat exchanger channels.

3. It is shown that the proposed constructions of compact shifted arrangement small diameter tube bundles give an opportunity of 1.45-1.65 times to reduce the size.

Author Contributions

V. Gorobets provided conceptualization and initiated the project; V. Gorobets and V. Trokhaniak described materials and methodology; V. Trokhaniak carried out simulation; Yu. Bohdan examined model verification; V. Gorobets, V. Trokhaniak, Yu. Bohdan and I. Antypov made formal analysis; The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed, and approved the final version of the manuscript.

Conflict of Interest

The authors declared no potential conflicts of interest with respect to the research, authorship and publication of this article.

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ORCID iD

Valery Gorobets https://orcid.org/000-0003-1180-4509
Viktor Trokhaniak https://orcid.org/000-0002-8084-1568
Yurii Bohdan https://orcid.org/000- 0002-3178-1941
Ievgen Antypov https://orcid.org/000- 0003-0509-4109

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