Experimental studies and numerical modelling of heat and mass transfer process in shell-and-tube heat exchangers with compact arrangements of tube bundles

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Abstract. Shall-and-tube heat exchangers based on the bundles with in-line or staggered arrangements have been widely used in industry and power engineering. A large number of theoretical and experimental works are devoted to study of hydrodynamic and heat transfer processes in such bundles. In that, works the basic studies of heat and mass transfer for these bundles are found. However, heat exchangers of this type can have big dimensions and mass. One of the ways to improve the weight and dimensions of the shell-and-tube heat exchangers is to use compact arrangement of tube bundles. A new design of heat exchanger is proposed, in which there are no gaps between adjacent tubes that touch each other. Different geometry of these tube bundles with displacement of adjacent tubes in the direction of transverse to the flow is considered. Numerical modelling and experimental investigations of hydrodynamic, heat and mass transfer processes in such tube bundles has been carried out. The distribution of velocities, temperatures, and pressure in inter-tube channels have been obtained.

1 Introduction

One of the most common equipments that are used in all types of power plants and engines are heat exchangers. Today well known many types of heat exchangers and methods for improving of heat transfer, including those used for engines [1, 2].

It is known that for shell-and-tube heat-exchangers maximal heat transfer in tube bundles is observed in the case of a cross current of heat carriers, i.e. in the case of the transverse flow around such bundles. Results of researches of hydrodynamics and heat exchange of tube bundles (with the tubes of traditional circular cross-section), are described widely enough in the monograph [3]. A significant number of works devoted to the investigation of smooth-tube bundles with various geometry and arrangement in bundles [4, 5]. At the same time, the number of papers devoted to investigations of the processes of hydrodynamics and heat, mass transfer in compact smooth-tube bundles, with minimum intertube intervals, is very insignificantly.

Extensive studies of convective heat transfer of various transversely streamlined bundles of smooth cylindrical tubes with their inline arrangement were carried out by Zhukauskas et al. In that studies, the main laws of the change in heat transfer and hydrodynamics are studied.

Also researches of compact, transversely streamlined bundles of smooth tubes with the inline arrangement were carried out by Aiba [6, 7].

The conducted studies showed that the inline tube bundles with a transverse flow around it have a lower hydraulic resistance in comparison with it staggered arrangement, but they have more low coefficients of heat transfer on the surface. One way to increase the heat transfer coefficient is to use small diameter tubes.

In this paper, the investigation of the processes occurring in heat exchangers is aimed at increasing the efficiency of the compressed smooth-tube bundles with inline configuration and inline shifted configuration with a minimum longitudinal relative step equal to unity. Experimental research, numerical computer modelling and thermohydraulic calculations of processes of heat transfer and hydrodynamics, which occur in channels of compact bundles, are carried out.

2 Research description

2.1 Experimental plant

For realization of investigations the experimental co-generation plant is created with utilizer of waste heat of exhaust gases of new design [8], which installed on the

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line of exhaust gas channel of high speed reciprocating internal combustion engine (Figure 1).

Fig. 1. A general views of the experimental plant.

During realization of experiments collection and treatment of these measurable parameters come true by means of the automated computer complex.

Surface of heating of proposed heat exchanger has next descriptions: diameter and thickness of smooth tubes - $d \times \delta$ = 0.010 × 0.001 m; equivalent diameter of channel between rows of tubes - $D_{chan.eq}$ = 0.00967 m; length of tubes - l = 0.150 m; equivalent diameter of casing - $D_{cas.eq}$ = 0.150 m; total for two heating areas, heat-receiving surface - H = 1.602 m^2.

2.2 Numerical computer modelling

For the co-generation plant on the base of combustion engine with new design waste heat utilizer of exhaust gases the computer modelling of hydrodynamics and heat transfer processes, which occur in channels of investigated tube bundles is conducted. The goal of these calculation is to obtain the local distributions of the field of velocities, temperatures and pressures, and as well as obtaining of dependences of heat transfer characteristics from the dynamic and thermophysical parameters of heat-carriers, including local distributions of coefficients of heat transfer on the circumference of tubes 1-4th rows for the first section of bundle. Geometry of the prospected channel with the compact location of bundles with small diameter tubes is brought in Figure 2, that has next geometrical characteristics: values of transversal and longitudinal step of tubes $S_a \times S_b$ = 0.015×0.010 m; diameter of tubes d = 0.010 m; number of tubes in raw i = 42; the tube bundles consist of j = 9 rows, and the common number of tubes in bundles makes z = 378, thus bundles divide between itself a technological gap, as shown in Figure 2. In Table 1 and Table 2 the geometry parameters of arrangement of tube bundles with inline and shifted compact location of tubes are shown.

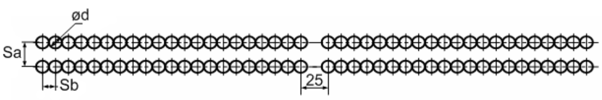

Fig. 2. Geometry of channel with the compact location of tube bundles.

The numerical computer modelling of processes of heat transfer and hydrodynamics in the investigated tube bundles was conducted on the basis of method of eventual elements by means of ANSYS Fluent software. Mathematical model of heat- and mass transfer processes, which take place in the investigated heat exchanger in two-dimensional formulation includes a system of Navier-Stokes equations, equation of energy transfer for convective flows [9] and standard k-ε model of turbulence [10].

Table 1. Geometry parameters of arrangement of tube bundles with the compact location of tubes.

No	Transversal step S_a, m	Longitudinal step S_b, m	Diameter of tube d, m	Number of tubes in raw, i
1	0.013	0.008	0.008	42
2	0.015	0.008	0.008	42
3	0.017	0.008	0.008	42
4	0.015	0.010	0.010	42
5	0.017	0.010	0.010	42
6	0.019	0.010	0.010	42
7	0.017	0.012	0.012	42
8	0.019	0.012	0.012	42
9	0.021	0.012	0.012	42

Table 2. Geometry parameters of arrangement of tube bundles with the shifted compact location of tubes.

No	Transversal step S_a, m	Shifting of tubes K, m	Diameter of tube d, m	Number of tubes in raw, i
1	0.15	0.001	0.010	40
2	0.15	0.002	0.010	40
3	0.15	0.003	0.010	40
4	0.15	0.004	0.010	40
5	0.15	0.005	0.010	40

Numerical calculations in the investigated channels of tube bundles were carried out for 9 constructions (Table 1) with the Reynolds number Re = 7085. As heat carriers are used exhaust gases (hot heat carrier) with a temperature at the inlet of 470 °C, which flow in the channels of the tube bundles. The temperature of the walls of the pipes was assumed constant, but changed for each section as the coolant moved (first section 75 °C, second 34 °C). Similar conditions take place, for example, in multi-pass heat exchangers.

In addition to the inline compact arrangements of tube bundles, numerical simulations were carried out for the more compact arrangements of tube bundle. The geometry parameters of tube bundles with a tube diameter d = 0.010 m is carried out in Table 2, which differ from those considered above by their more compact arrangement with shifting of adjacent tubes by some distance, while five variants of the bundle design are considered, with a shifting of tubes in the transverse direction K from 0.001 to 0.005 m. The calculations were carried out at a Reynolds number of 18,6·10^3. As the heat carriers, the exhaust gases (hot heat carrier) of the internal combustion engine with a temperature of 470 °C at the inlet to the heat exchanger, which flow in the intertubular channels, and the fresh water (cold heat carrier) moving inside pipes with inlet temperature in the heat exchanger equal to 20 °C.

Numerical modelling of heat transfer and hydrodynamics in channels with compact arrangement of tube bundles was performed for bundles that contain 40 tubes in a single row with a diameter of 0.010 m.

2.2 Analysis concept

To evaluate the heat exchange surface, from the energy point of view, the coefficient of thermohydraulic efficiency (M.V. Kirpichev's criterion) is used, which is defined as the ratio of the transferred amount of heat Q through the heat exchange surface to the total power N required for pumping heat carriers through the heat transfer surface on both sides (without taking into account the efficiency of the superchargers and drives) [11].

$$E = \frac{Q}{N} \qquad (1)$$

The amount of heat that is taken from the hot or transferred to the cold heat carrier is determined by the formula

$$Q = c_p G \Delta T \qquad (2)$$

Power required to pump the heat carrier:

$$N = \frac{\Delta p G}{\rho} \qquad (3)$$

3 Results and discussions

The results of the numerical calculations are given for one of the geometries under consideration (d = 0.008 m, Sa = 0.013 m) in Figure 2. The distribution of the velocity field in the channels of the tube bundle is shown in Figure 3, and in Figure 6 shows the distribution of the velocity vectors on a small section of the channel. The value of the average speed of exhaust gas in the narrowest cross section of the channel is 86±2 m/s (Figure 3).

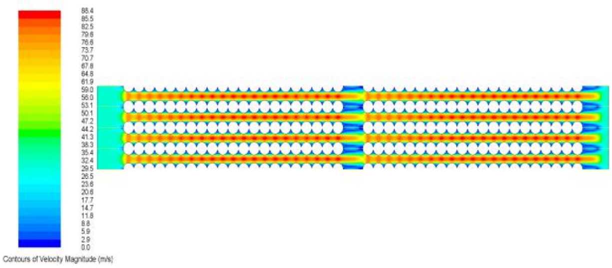

Fig. 3. Velocity of exhaust gases in the channels of tube bundles, m/s (Re = 7085).

Figure 4 shows the temperature distribution in the channels where the exhaust gas temperature at the outlet is 82±3 °C. Figure 5 shows the change in pressure in the flow of exhaust gases flowing along the channels of the tube bundles.

The results of numerical calculations for the channel with a shifting of adjacent tubes of 0.005 m are given in Figures 6-8. The distribution of the velocity of the heat carrier flow in the channel is shown in Figure 6.

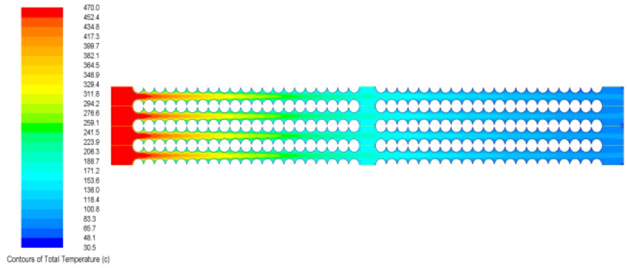

Fig. 4. Change of temperature in the channels of tube bundles, °C (Re = 7085).

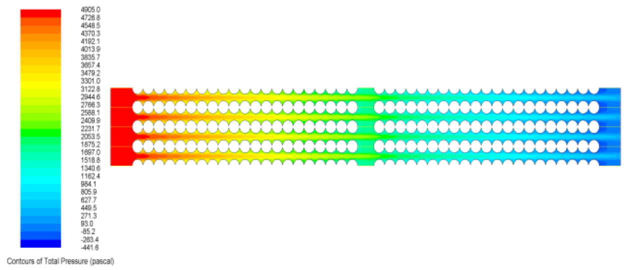

Fig. 5. Change of pressure in the channels of tube bundles, Pa (Re = 7085).

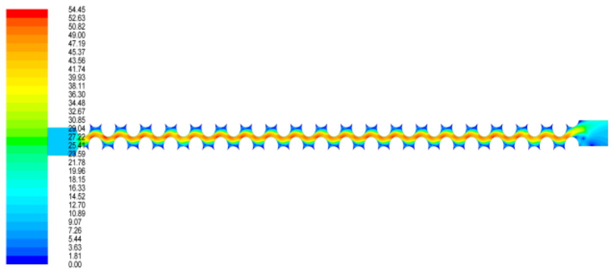

Fig. 6. Velocity of exhaust gases in the channels of tube bundles with shifting, m/s (Re = $18{,}6\cdot10^3$).

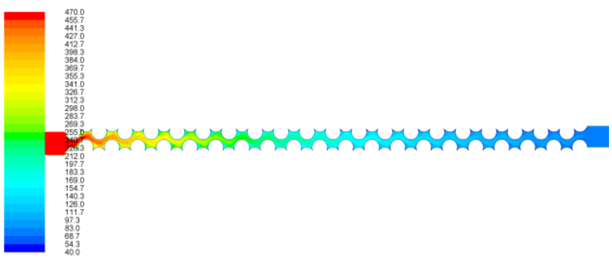

Fig. 7. Change of temperature in the channels of tube bundles with shifting, °C (Re = $18{,}6\cdot10^3$).

As we can see in Figure 6, at the upper point of the tube, the boundary layer separates, and at the junction of adjacent tubes, stagnant zones are observed. At separate points of the channel, the velocity of the exhaust gases reaches 55 m/s, and their average velocity in the narrow cross-section of the channel is about 45 m/s, which is greater than in the channels without shifting. Figure 7 shows the temperature distributions in the intertubular channel. The outlet temperature of the cooled gas is 78 °C, which is the lowest temperature among the output temperatures that are achieved in the channels of other structures.

Figure 8 shows the change in the pressure fields in the channel of the investigated structure. From the obtained pressure distributions it follows that the total pressure drop is about 15700 Pa and is unacceptable for heat exchanger intended for the co-generation plant, it is necessary to apply a small shifting of the tubes 0,001÷0,002 m not exceeding the boundary-permissible back pressure of the engine.

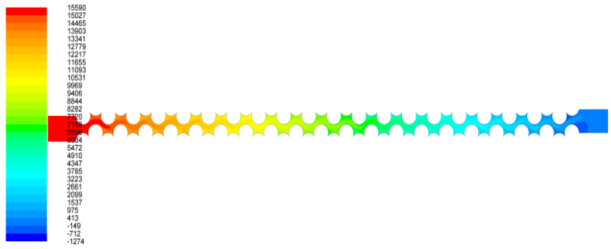

Fig. 8. Change of pressure in the channels of tube bundles with shifting, Pa (Re = 18,6·10³).

In the Figure 9 the dependences of the Nusselt number on the bundles geometry is showed.

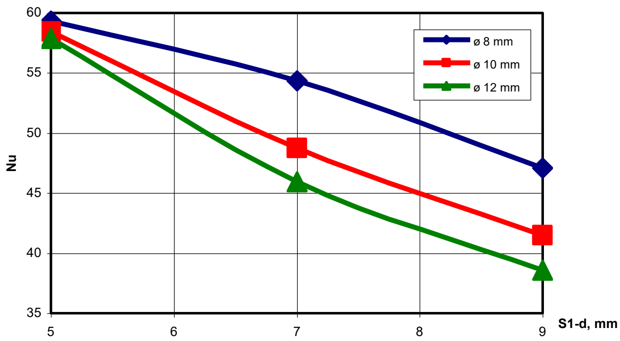

Fig. 9. Dependence of the Nusselt number Nu on the geometry of a tube bundles.

As a result of the analysis of the obtained dependences, it can be concluded that the most effective design for the intensity of heat transfer is a tube bundles with a transverse step of 0.005 m and a diameter of 0.008 m.

Figure 10 shows the dependence that characterizes the pressure loss of the heat exchange surface on the tube bundles of various geometries.

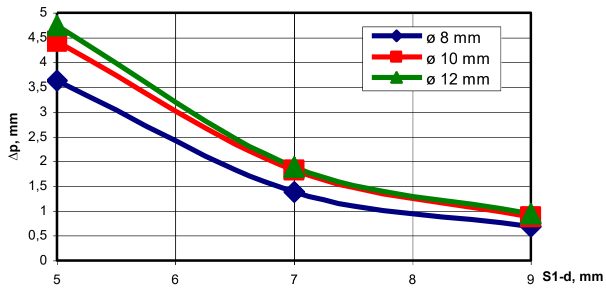

Fig. 10. Dependence of the pressure drop Δp on the geometry of a tube bundles.

As we can see in Figure 11 the thermohydraulic efficiency decreases with decreasing step and diameter of the tubes due to the growth of hydraulic resistances.

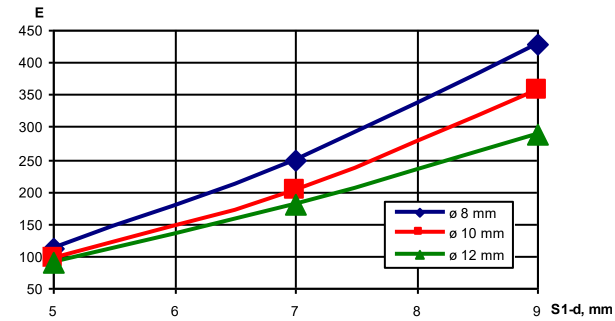

Fig. 11. Dependence of the thermohydraulic efficiency E on the geometry of a tube bundles.

Analyzing the obtained dependences from the point of view of the effect of geometry on the heat and hydrodynamic characteristics of a compact transversely streamlined tube bundle, it can be noted that the best characteristics are tube bundles with a transverse step of 0.013 m and a tube diameter of 0.008 m. In this case, a significant reduction in the temperature of the heat carrier at the outlet of the channel is achieved.

Figure 12 shows the change the Nusselt number, depending on the geometry of the tubes arrangement in the bundles with shifting. The abscissa indicates the displacement values of adjacent tubes equal to 1, 2, 3, 4 and 5 mm, respectively, and the ordinate represents the average values of the Nusselt number over the surface of the tube bundles. As a result of the analysis of the obtained dependence, it can be concluded that the most effective in terms of the intensity of heat exchange is the geometry of the bundle with shifting of the tubes by 5 mm. It should also be noted that when the tubes are shifted by 5 mm, the pressure difference in the channel will be maximum (Figure 13), because of this, when using such a geometry in the heat exchangers, the back pressure on the diesel exhaust exceeds the limit value of 5 kPa and will be to have a significant impact on the engine efficiency. Therefore, in order to intensify the heat transfer, the most suitable small shifting of 1 to 3 mm, which do not cause a significant increase in aerodynamic resistance of the bundle.

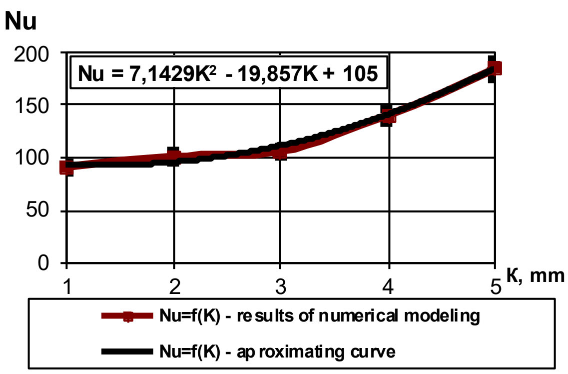

Fig. 12. Dependence of the Nusselt number on the shifting of a tubes in bundles.

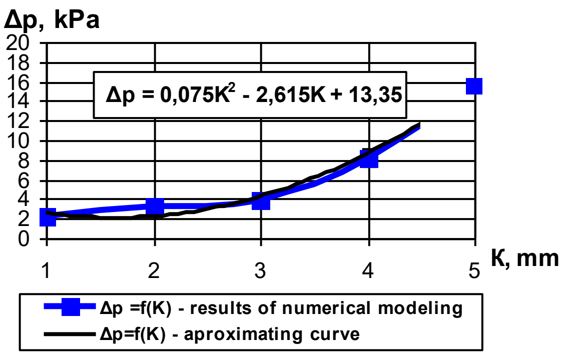

Fig. 13. Dependence of the pressure drop on the on the shifting of a tubes in bundles.

Figure 14 shows the dependence of the thermohydraulic efficiency E (formula 1) on the geometry of tube arrangement in the bundle with shifted tubes. As can we see in Figure 14, the thermohydraulic efficiency in such channels due to the growth of hydraulic resistance decreases with increasing shifting of adjacent tubes.

Analysis of the obtained dependences shows that from the point of view of the intensification of heat transfer processes on the surface of the tube bundles with an insignificant increase in the aerodynamic resistance, the design has better characteristics with a small shifting of 1 to 3 mm (Figures 12-14).

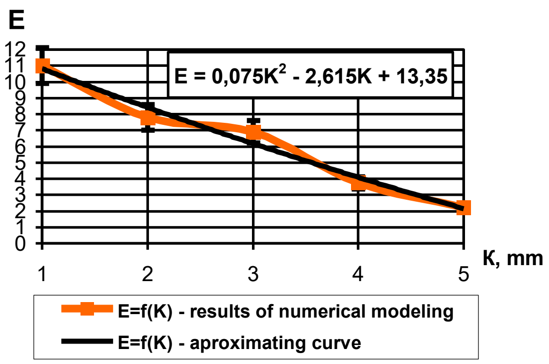

Fig. 14. Dependence of the thermohydraulic efficiency on the shifting of a tubes in bundles.

Intensification of heat transfer in bundle with the compact arrangement of tubes in comparison with an inline bundle is achieved by influencing the heat transfer conditions of next factors: 1) reduction of transverse distance (step) between the nearby rows of tubes, that results in the increase of velocity of external flow at the identical expense of heat carrier in the heat exchangers of the known constructions; 2) creation of longitudinal alternating pressure gradients, which also leads to intensification of heat transfer processes (analog of channels with variable cross-section or "corrugated" channels of type diffuser-confuser).

Results of investigations of the flow and heat transfer in the bundles of new design heat exchanger were estimated by comparing the obtained experimental data with the results of numerical simulation and known experimental data obtained by other researchers. The results of the researches are presented in the form of graphical dependence in Figure 15 for Reynolds numbers in the range Re = 3000 ÷ 6000 and Prandtl number Pr = 0.73.

Figure 15 shows the experimentally determined dependence of Nusselt number averaged over the surface $\overline{Nu}$ on Reynolds number Re.

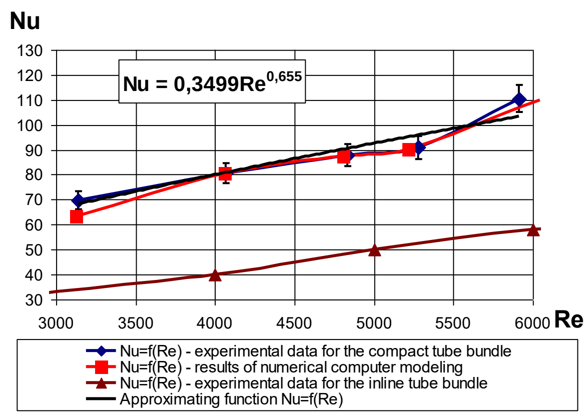

Fig. 15. Dependence of average Nusselt number $\overline{Nu}$ on Reynolds number Re: ♦ – experimental data for the compact tube bundle; ▲ – experimental data for the inline tube bundle; ■ – results of numerical computer modelling.

Figure 15 also shows the experimental dependence $\overline{Nu}$ = f(Re), obtained for the inline tube bundle in the work of Belov and Kudryavtsev [12]. The comparison shows that for the same values of Reynolds number, the intensity of heat transfer for a compact tube bundle is almost in 2 times greater than intensity of heat transfer on the surface of the inline tube bundle.

4 Conclusions

1. Experimental data have been obtained for heat and mass transfer processes in channels of compact tube bundles, which are represented in the form of the dependences of the heat transfer characteristics on the dynamic and thermophysical parameters of heat carriers.
2. A comparative analysis of the thermohydraulic efficiency for channels with different transverse steps of the tube in bundles has been carried out and it is shown that the structures developed are sufficiently effective at

a significant reduction in the mass-dimensions of the heat-exchange surface.

3. Numerical computer modelling of heat and mass transfer processes in channels with transverse flow past extremely compressed tube bundles with inline arrangement and absence of a gap between adjacent tubes in the direction of the heat carrier motion is carried out.

4. The new design of heat exchangers with the compact tube bundles is proposed, which has high efficiency, low aerodynamic and hydraulic resistance, and Nusselt number on the tube surface is approximately in 2 times higher than their values for a bundle with a traditional inline arrangement of tubes.

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