Проведено експериментальне дослідження теплообміну та гідравлічного опору при поперечному обтіканні п'ятирядного пучка труб із поверхневими заглибинами у формі усіченого конусу. Отримані залежності щодо коефіцієнтів теплообміну та числа Ейлера від числа Рейнольдса, запропоновані критеріальні співвідношення. Визначено фактори інтенсифікації теплообміну, підвищення опору та параметру аналогії Рейнольдса

Ключові слова: пучок труб, поперечне обтікання, поверхневі заглибини, коефіцієнт теплообміну, число Ейлера

Выполнено экспериментальное исследование теплообмена и гидравлического сопротивления при поперечном обтекании пятирядного пучка труб с поверхностными углублениями в форме усеченного конуса. Получены зависимости по коэффициентам теплообмена и числу Эйлера от числа Рейнольдса, предложены критериальные соотношения. Определены факторы интенсификации теплообмена, повышения гидравлического сопротивления и параметра аналогии Рейнольдса

Ключевые слова: пучок труб, поперечное обтекание, поверхностные углубления, коэффициент теплообмена, число Эйлера

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EXPERIMENTAL STUDY OF HEAT TRANSFER AND HYDRAULIC RESISTANCE AT CROSS FLOW OF TUBE BUNDLE WITH INDENTATIONS

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1. Introduction

Interest to the study of heat transfer and hydraulic resistance of tube bundles with surface indentations is due to the need to improve heat exchangers for power, chemical technology, metallurgy and other industries. Heat transfer augmentation, as a rule, is accompanied by an outstripping growth of hydraulic losses. Therefore, various methods are used to improve the thermal and hydraulic efficiency of heat exchangers. The heat transfer augmentation through the use of various types of roughness, namely the use of depressions on the outer surface of the pipes is of particular interest. In addition to the heat transfer augmentation, indentations lead to a displacement of the separation point downstream, as well as to acceleration of the laminar-turbulent transition of the boundary layer, which makes it possible to reduce the drag. As a result of the use of indentations, conditions for the anticipated growth of heat transfer in comparison with the growth of hydraulic resistance are created. Thus, for given values of the effectiveness of heat transfer and hydraulic resistance, smaller heat exchange surface, dimensions and lower materials consumption of the heat exchange equipment are provided.

2. Literature review and problem statement

Available publications [1–8] consider the cross flow of both a single tube and tube bundles with surface indentations in a wide range of Reynolds numbers. In [1–5], devoted to the cross flow of a single tube, the main emphasis is on the study of hydrodynamic regimes, the point of flow separation, the frequency of vortex shedding, the dependence of the drag resistance on the Reynolds number. In the works devoted to tube bundles, hydraulic losses, heat transfer and thermal and hydraulic efficiency were investigated.

The experiments performed in [1] on the study of the aerodynamics of a cross flowing tube with indentations (spherical shape) on the surface in the Reynolds number range $2\cdot10^4...3\cdot10^5$ show that the effect of dimples on aerodynamics turned out to be almost the same as at the flow around a golf ball, investigated by the same authors in [9]. The shallow (h/D=0.1) dimples located on the surface shift the separation point further downstream and reduce the critical Reynolds number. After reaching Re_{cr}, the C_D value does not increase, but remains approximately at the same level.

In [2], the aerodynamics of cross flow of a single tube in the Reynolds number range $(1.39...4.76)\cdot10^5$ was studied. Both deep (h/D=0.35) and shallow (h/D=0.08) spherical indentations were used. Studies have shown that the use of indentations on the surface of the cylinder results in a shift upstream of the laminar-turbulent transition point, and a downward displacement of the separation point.

Work was also carried out to study the cross flow of a single tube with indentations of other shapes. In [3], hydraulic characteristics of the flow of a circular tube with cylindrical symmetrical and asymmetrical indentations were investigated. Studies have shown that in the region of small

Reynolds numbers (up to 10^4), the presence of indentations on the surface reduces the length of the flow separation zone behind the tube in comparison with the case of a smooth tube, and this effect is intensified with increasing depth of the dimples.

In [4], cross flow over a cylinder with indentations and protrusions of a hexagonal shape in the Reynolds number range of $3.14 \cdot 10^4 ... 2.77 \cdot 10^5$ was investigated. Visualization showed that the separation displacement occurs at lower Reynolds numbers, in comparison with the critical Reynolds number on a smooth cylinder.

In [5], the drag of a single tube with cylindrical indentations was investigated in the range of the Reynolds number $7.43 \cdot 10^3 ... 1.80 \cdot 10^4$. Two different values 20 % and 40 % of the filling density of the surface by depressions were studied. A decrease in drag by 8 % was obtained at the lowest Reynolds number, and by 15 % – at the highest Reynolds number.

The nature of the flow over the tubes in the bundle differs from the flow around a single tube, due to the mutual influence of neighbouring tubes. Studies on the heat transfer and resistance at the cross flow of tube bundles with surface indentations were mainly aimed at creating industrial heat exchangers with improved thermal and hydraulic performance. Since the heat transfer apparatuses are characterized by low flow velocities, the investigations were carried out, mainly, at low Reynolds numbers ($10^3 ... 3 \cdot 10^4$).

In [6], the hydraulic resistance of a five-row tube bundle of a staggered arrangement with semi-spherical (h/D=0.5) surface indentations was experimentally investigated. Three variants of filling the surface of the tubes with indentations – with a factor of increasing the surface of 1.2; 1.31 and 1.41 are considered. The Reynolds number was determined from the velocity in the minimum cross section. Studies have shown that in the range of the Reynolds number $10^3 < Re < 3 \cdot 10^4$, the hydraulic resistance of the tube bundle with indentations increased by 8...14 % compared to the smooth-tube bundle.

In [7], cross-flow of six-row tube bundles of both in-line and staggered arrangement with shallow spherical indentations (h/D=0.12) in the Reynolds number range $6 \cdot 10^3 ... 3 \cdot 10^4$ was investigated. Experiments have shown that for an in-line arrangement, the level of heat transfer corresponds to a smooth tube bundle, however, the pressure loss was decreased by 25 %. With the staggered arrangement of tubes, the presence of indentations increases the heat transfer by 20 % and reduces the pressure loss by 35 %.

In [8], the results of the experimental study of heat transfer and hydraulic resistance of a four-row tube bundle of staggered arrangement with spherical surface indentations are presented. Deep (h/D=0.37) and shallow (h/D=0.085) spherical indentations were used. The Reynolds number was determined by the velocity of the flow entering the test section, its range was narrow enough, and was $10^4 ... 1.2 \cdot 10^4$. Studies have shown that the Reynolds analogy factor was 1.1...1.15 for deep indentations, and 1.3...1.5 for shallow ones.

The performed researches have shown the prospects of an external heat transfer augmentation at the cross flow of tube bundles by means of surface indentations. This method makes it possible to achieve an advanced growth of heat transfer in comparison to the growth of hydraulic losses. However, there are contradictions in the results of the works on tube bundles. Thus, in [7], a decrease in hydraulic losses was obtained by using tubes with indentations, and in [6] and [8] – moderate growth. Studies [8] were performed in a very narrow range of Reynolds numbers, which does not allow us to use these results for obtaining reliable dependences. For heat exchangers based on the cross-flow of tube bundles, the typical range of the Reynolds number is $10^3 ... 3 \cdot 10^4$. In this range, the results of the study of heat transfer and hydrodynamics of cross-flow of tube bundles with indentations are practically absent.

It is also of interest to study the effect of indentations of other shapes, different from spherical on heat transfer. For example, in [10] it is shown that for the flow in a flat channel, a higher level of heat transfer augmentation provides indentations of cylindrical shape.

3. The aim and objectives of the study

The aim of this paper is the experimental study of heat transfer and hydraulic resistance at cross-flow of the tube bundle of staggered arrangement with surface indentations of a truncated cone shape, in the Reynolds number range $3 \cdot 10^3 ... 2.5 \cdot 10^4$.

To achieve this aim, the following tasks will be solved:

– to obtain data on heat transfer, hydraulic resistance, heat transfer augmentation, hydraulic resistance increase and thermal hydraulic efficiency;

– to obtain correlations for calculating the heat transfer coefficient and the Euler number.

4. Materials and methods of research of heat transfer and hydraulic resistance at cross-flow of tubes

The experimental facility (Fig. 1) is an open-loop gas-dynamic circuit operated by the suction of atmospheric air by a radial-drive blower.

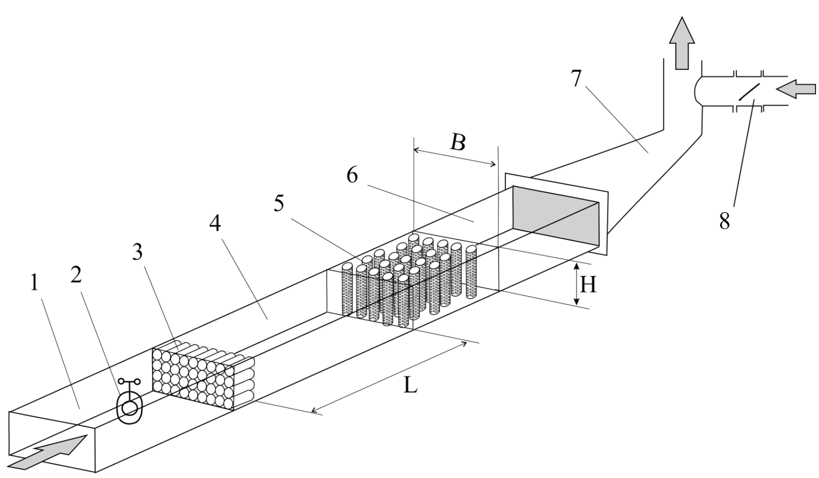


Fig. 1. Scheme of the experimental facility: 1 – input channel; 2 – flowmeter; 3 – honeycomb; 4 – pre-connected rectangular channel; 5 – test section; 6 –output rectangular channel; 7 – conical outlet nozzle; 8 – additional branch pipe with a valve

Air from the atmosphere entered the input rectangular channel 1, at the end of which a honeycomb was installed, which provides a uniform velocity field at the channel cross-section. In the input channel, there was a flowmeter (cup anemometer). The air then entered the pre-connected rectangular channel 4 with a height H=105 mm, a width B=200 mm and a length L=630 mm, and was fed into the test section 5. After that, the flow through the outlet rectangular channel 6 passed into the suction pipe of the fan through the conical nozzle 7. In the circuit, in-front of the fan, an additional branch pipe 8 with a valve is installed, for sucking air from the atmosphere, in order to enable the flow rate of the main flow to be controlled (towards its decrease). The walls of the channel, the test section and the outlet rectangular channel were made of plexiglas. The total length of the plexiglas part of the loop was 970 mm.

The test section (Fig. 2) consisted of a five-row bundle of tubes of a staggered arrangement. The diameter of all tubes d was 22 mm, its working length corresponded to the height of the channel H, and equals to 105 mm. The relative lateral and longitudinal steps of the tube arrangement in the bundle were respectively: S_1/d=1.7, S_2/d=1.2.

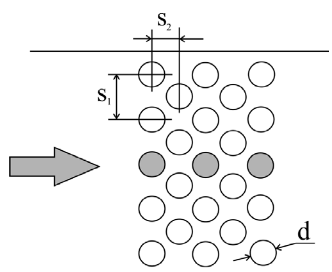


Fig. 2. Scheme of the test section for the study of cross-flow of a five-row tube bundle

The scheme of the conical surface indentation is shown in Fig. 3. The depth h was 1.3 mm, the diameters D and D_1 were 4 mm and 3 mm, the radius of rounding the edge r=0.25 mm. On the circumference of the tube, there were 15 rows of indentations in a staggered arrangement, total 318 indentations, the density γ of filling the surface was 55 %. The ratio of the actual tube surface accounting the relief to the surface of the smooth tube was 1.41.

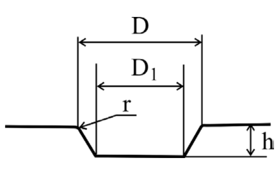


Fig. 3. Scheme of surface indentation

For comparison, experiments were conducted to measure the heat transfer coefficient and the hydraulic resistance of a smooth-tube bundle with a similar arrangement of tubes.

In heat transfer experiments, the average heat transfer coefficient over the surface of the tube was determined using a technique based on the use of a calorimeter containing melting ice.

The test tube was filled with distilled water and placed in a freezer to form ice. After the experiment, the amount of formed water was measured, and a posteriori, the average heat flux and heat transfer coefficient were determined. To check the mass balance, the amount of remaining ice was also measured. A detailed description of this technique is given in [11]. Measurements of the heat transfer coefficient were made for the central tubes of the 1st, 3rd and 5th rows (shaded in Fig. 1). The average on the bundle heat transfer coefficient was found by integrating the distribution of the heat transfer coefficient over the individual rows.

The air flow rate, air temperature, static and total pressure at the inlet to the test section, static pressure drop on the tube bundle, wall temperature of the tube were measured experimentally. The air temperature was determined by a standard mercury thermometer with a graduating mark of 0.1 °C. To determine the air flow rate, a cup anemometer was used, it was previously calibrated using a Pitot tube and integration of dynamic and static pressure across the channel cross-section. Micromanometer MKB-2500-0.02 was used in the measurements.

In the generalization of data to determine the thermophysical properties of air, the temperature of the incoming stream was used. The Reynolds number was determined from the diameter of the cylinder and the flow velocity at the minimum cross-section of the bundle. When determining the heat transfer coefficient, the area of a smooth cylinder was used, without taking into account the relief. The data on the hydraulic resistance were processed in the form of a dependence of the Euler number $Eu = \Delta p / \rho w_{\max}^2$ for the whole bundle divided by the number of rows, i. e. related to one row.

The air flow velocity when entering the test section varied in the range 1...9 m/s, its temperature was 15...20 °C, the Reynolds number Re_d varied from 3000 to 25000.

5. The results of experimental study of heat transfer and hydraulic losses at the cross flow of tubes

The data on the Nusselt number for individual rows and the averaged for the bundle for smooth pipes (a) and pipes with depressions (b) are shown in Fig 4.

First, the data were processed for the 1st, 3rd and 5th rows as dependences of the Nusselt number on the Reynolds number, with the construction of the similarity equations in the form of power functions $Nu=a{\cdot}Re^n$. Data processing was carried out using the method of least squares. Then the mean integral values of these coefficients were found for the entire bundle. As a result of the generalization, the following dependences were obtained for the average heat transfer of the tube bundle:

Smooth-tube bundle:

$$\overline{Nu}_{fd} = 0{,}36 \cdot Re_{fd}^{0{,}6}. \quad (1)$$

Tubes with indentations:

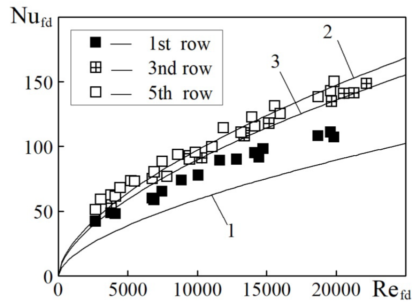
$$\overline{Nu}_{fd} = 0{,}41 \cdot Re_{fd}^{0{,}62}. \quad (2)$$

The root-mean-square deviations for the Nusselt numbers are 6 % for the maximum Reynolds number, and 13 % for the minimum.

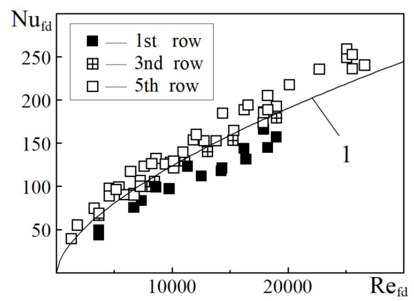
Fig. 5 shows the dependence of the Euler number for a smooth-tube bundle and a tube bundle with indentations.

For the reduced coefficient of resistance (per row) of the tube bundle with surface indentations, the following relationship is obtained:

$$Eu = (3.15 \pm 0.046) \cdot Re_{fd}^{-0.32 \pm 0.0017}. \quad (3)$$



a



b

Fig. 4. Heat transfer in tube bundles depending on the Reynolds number: *a* — the bundle of smooth tubes. Lines 1, 2 — deep and first rows, respectively [12]; Line 3 — the data of the present work for the entire bundle; *b* — a bundle of tubes with indentations. Line 1 — generalizing correlation for the bundle

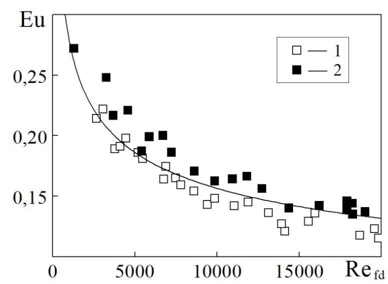
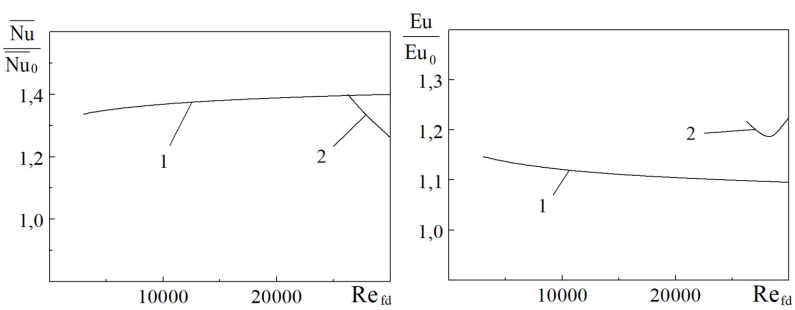


Fig. 5. The value of the average coefficient of resistance of the tube bundles, reduced to one row, depending on the Reynolds number: 1 — smooth-tube bundle; 2 — a bundle of tubes with surface indentations. The line is the dependence from [12]

As the analysis shows, in the considered range of the Reynolds number the factors of heat transfer augmentation and resistance growth vary slightly and have the following values: Nu/Nu_0=1.34...1.4; Eu/Eu_0=1.15...1.10 (Fig. 6).



a *b*

Fig. 6. Factors of heat transfer augmentation and hydraulic resistance growth: *a* — relative Nusselt number; *b* — relative Euler number. 1, 2 — respectively, the data of the present studies and the data of [8]

6. Discussion of the experimental results

The heat transfer data for a smooth tube bundle (Fig. 4, *a*) coincide with the dependence [12] for depth rows, but for the first row, the data of the present experiments are 20 % higher. Perhaps this is due to the turbulence of the flow after leaving the honeycomb. However, it can be noted that in determining the value of heat exchange intensification, this factor acts similarly for both variants studied, and does not affect the value of Nu/Nu_0.

In the dependence (2) of the Nusselt number on the Reynolds number for a variant of tubes with depressions, the exponent at the Reynolds number slightly increased, in comparison with the dependence (1) for a smooth tube bundle. This can be explained by the interaction of two types of vortex structures, which, on the one hand, are formed in the wake of the cylinders, and on the other, are generated by indentations. A similar phenomenon was analyzed in [8]. Interaction of vortices leads to additional turbulence of the flow and intensification of heat transfer.

The coefficient of resistance of a smooth tube bundle (Fig. 5) turned out to be smaller by 10 % than the dependence in [12]. Apparently, this is due to the influence of the external turbulence of the flow, which reduces the drag of the first row. This effect is a consequence of a decrease in the velocity gradient in the vicinity of the front critical point and a decrease in the rarefaction in the aft region of the cylinder [13].

A comparison with the data of [8] for deep indentations, recalculated by the value of the bulk velocity in the minimum cross section, shows that the data on the heat transfer augmentation correlate with each other. There is some difference in hydraulic resistance, but it should be taken into account that in the experiments [8] a smaller lateral step was used.

Thus, according to the data of the present experiments, the value of the Reynolds $\dfrac{Nu/Nu_0}{Eu/Eu_0}$ analogy factor is 1.17...1.27; in [8] it is 1.1...1.15 for deep depressions, and 1.3...1.5 for shallow ones.

Based on the results of the study of heat transfer and hydraulic resistance, an assessment of the reduction in the heat exchange surface was made when surface depressions were used for gas recuperation in turbine installations. Recuperation is used to heat incoming air with exhaust gases. In such devices, the gas and air rate differ slightly from each other. In this case, the number of transfer units is written as follows:

$$NTU = \frac{k \cdot S}{\dot{m} \cdot C_P}, \tag{4}$$

where NTU is the number of transfer units, which has a unique relationship with the efficiency of heat exchange for a given heat exchange scheme [14]; k is the heat transfer coefficient, W/(m·K); S – heat exchange surface, m²; $\dot{m}$ – mass rate, kg/s; C_P – heat capacity at constant pressure, J/(kg·K). Using this relationship, it is possible to determine the ratio of heat exchange surfaces with a constant gas flow rate and given heat exchange efficiency (or NTU):

$$\frac{S}{S_0} = \frac{k_0}{k}, \tag{5}$$

where k_0 and k are the values of the heat transfer coefficient for a smooth-tube bundle and for a bundle with intensifiers. For the variant of pipe making by punching out of thin sheet material, indentations are obtained on the outside of the pipes and protrusions on the inside (Fig. 7). According to [15], the level of heat exchange intensification in turbulent (Re_d=6·10^3...7·10^4) flow within a tube with spherical indentations is 1.4...1.6. In the present paper, for the external flow, the heat exchange intensification is 1.34...1.4. Thus, to estimate the heat exchange surface, it can be assumed that the values of the intensification factors for internal and external heat exchange are approximately equal.

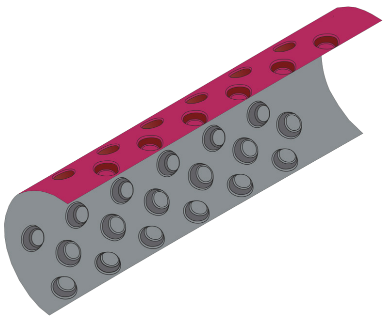


Fig. 7. Scheme of the wall of the tubular heat exchange surface

In addition, one can neglect the thermal resistance of the tube wall due to its smallness. Then the value of S/S_0, taking into account the results of the experiments performed in this paper, is 0.75...0.7. The actual reduction of the heat exchange area can be changed with a different level of intensification of internal heat exchange, in this case a more accurate calculation is required. In conclusion, it can be noted that the above estimation is suitable for any flow scheme of coolants, since it deals with the ratio of quantities, and not with their absolute values.

7. Conclusions

1. The data on the heat transfer and hydraulic resistance of a five-row tube bundle with surface indentations as dependences of the Nusselt and Euler numbers on the Reynolds number in the range 3000<Re<25000 are obtained. The heat transfer augmentation factor is 1.34...1.40 with the factor of increase in the hydrodynamic resistance of 1.10...1.15. The value of the Reynolds analogy factor is 1.17...1.27, which indicates an advanced increase in heat transfer compared to an increase in hydraulic losses.

2. The relations are obtained for the calculation of heat transfer and hydraulic resistance, in the form of dependences of the Nusselt number and the Euler number on the Reynolds number. It is shown that these dependences have a power-law form, and allow determining the coefficients of heat transfer and hydraulic resistance in the development of tube-type heat exchange equipment. Evaluation by means of an expression for the number of transfer units has shown that the use of tubes with surface depressions allows reducing the heat-exchange surface by 25...30 %.

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