Substantiation of using non-circular plain shaped surfaces to increase the compactness of power plants is presented. The criterion of thermohydraulic compactness is justified, which takes into account the layout and arrangement of heat transfer elements and their thermohydraulic efficiency. To simulate heat movement and transfer processes in elements of power plants, the computational fluid dynamics method is used. Verification is carried out with available literature data, the discrepancy of results does not exceed 2.4 %. For single elliptical and plane-oval shapes, there is a local maximum of efficiency, achieved at the axis ratio of 2.5 for elliptical and 2.75 for plane-oval. Studies of the change of the heat transfer coefficient in the tube bank rows are carried out. For the elliptical tube bank, heat transfer is stabilized from the fifth row. Heat transfer surfaces of circular, elliptical and plane-oval tubes with different combinations of geometric characteristics are considered. It is found that on the basis of elliptical tubes it is possible to reduce the volume of the heat transfer surface and increase the compactness of the entire power plant by 18.3 % compared to circular tubes and 2.4 % compared to flat-oval ones. Dimensionless indices of mass, volume, functional efficiency and service life of the heat exchanger are substantiated, allowing them to be compared as part of various power plants. It is advisable to continue studies of the features of heat transfer processes in close, with the relative longitudinal and transverse pitch ratio less than 1.5, banks of elliptical tubes with an axis ratio of 2.5

Keywords: power plant, compactness, heat exchanger, thermohydraulic efficiency, shaped surface, RSM turbulence model

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JUSTIFICATION OF EFFICIENCY OF PLAIN SHAPED HEAT EXCHANGE SURFACES TO INCREASE THE COMPACTNESS OF POWER PLANTS

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1. Introduction

The efficiency of power plants is increased by improving heat conversion processes in their elements. An effective method is the recovery and utilization of exhaust gas heat from the main and auxiliary engines of the plants. At the same time, the implementation of these processes leads to an increase in weight and size indicators and a decrease in the compactness of power plants [1–3].

The basis of most modern heat transfer surfaces is circular tubes. A promising direction of increasing compactness due to the intensification of heat transfer processes is the use of "well" streamlined tubes – plane-oval, elliptical, drop-shaped, etc. [4] and optimization of their arrangement in the bank. The use of such surfaces would improve the thermohydraulic characteristics of tube banks, which, in turn, would reduce their weight and size parameters and increase compactness.

In this regard, studies aimed at substantiating the use of shaped heat exchange surfaces to increase the compactness of power plants can be considered relevant.

2. Literature review and problem statement

To assess the compactness of power plants as a whole, a number of relative indicators are used, from which the relative power of the plant can be distinguished [5]:

$$\bar{N}_V = \frac{N_e}{V_{eq}}, \tag{1}$$

or

$$\bar{N}_M = \frac{N_e}{M_{eq}}, \tag{2}$$

where N_e is the power of generating equipment, kW; V_{eq} is the volume occupied by all plant equipment, m^3; M_{eq} is the weight of equipment, kg.

The compactness of the surfaces of heat transfer elements, as a part of the entire plant, is estimated by the geometric compactness index [6]

$$K_{geom} = \frac{F}{V}, \tag{3}$$

where F and V are the area and volume of the lateral heat transfer surface, respectively.

The results of studies of plane-oval tubes are presented in [4, 7, 8].

In [4, 7], the results of experimental studies of the efficiency of staggered banks of plane-oval tubes during the cross-flow around in the range of numbers Re=2,000...30,000 are presented. The studies were carried out for four types of tubes with axis ratios of 2; 2.5; 3.4; 5.0. The transverse and longitudinal pitch ratio S_1/S_2 is taken as a parameter of tube arrangement in the bank, and the results for the range S_1/S_2=0.375...1.438 are presented. It is noted that the advantage of this shape is its manufacturability; however, there is no analysis of the investigated shapes in terms of compactness and efficiency of heat transfer in banks.

In [8], the results of experimental and numerical studies of the efficiency of plane-oval tubes during oil flow around are presented. Single tubes with an axis ratio of 1.92 during flow around them along the major and minor axes investigated. The results for calculating the average heat transfer and hydraulic resistance are given, however, their use is limited to only one geometric size and the lack of recommendations for the arrangement of such tubes in banks.

The works [9, 10] present the results of comparative experimental studies of circular and elliptical tubes in the range of numbers Re=6,000...11,000 with an internal flow. Comparison of the obtained experimental data with the results of theoretical calculations using the *Dittus-Boelter* equation is shown. The possibility of using the presented results is limited by the data for the internal flow.

The work [11] presents the results of studying the banks of circular and elliptical, with the axis ratio of 2; 3; 4, tubes in the range of numbers Re=1,000...14,000. Expressions for calculating the thermohydraulic efficiency of single-row tube banks are presented. The paper does not specify how the given data can be extended for multi-row banks.

The paper [12] presents an analytical solution to the problem of heat transfer and resistance for an external flow around an elliptical cylinder with various axis combinations. Comparison of the values of heat transfer coefficients and hydraulic resistance in the range of numbers Re=100...100,000 with circular tubes is presented. The authors of the work limited themselves to studies of single shapes and did not indicate how to use the given data for tube banks.

To assess the thermohydraulic efficiency of heat transfer processes, the following are proposed:

– in [13], Reynolds analogy factor, defined as

$$FAR_\alpha = \frac{Nu/Nu_0}{f/f_0}, \tag{4}$$

where Nu is the Nusselt number, f is the drag coefficient. Index “0” defines the base channel, usually plain, to be compared;

– in [14], modified Reynolds analogy factor FAR_k, defined as

$$FAR_k = \frac{k/k_0}{\sum Eu/Eu_0}, \tag{5}$$

where k is the heat transfer coefficient, W/(m^2·K), Eu is the Euler number, index 0 corresponds to a cylindrical channel on both sides (as the most thermohydraulically studied). The disadvantage of indicators (3)–(5) is the possibility of only a separate assessment of either geometric compactness or thermohydraulic efficiency.

Thus, the available literature data do not allow determining changes in the compactness of power plants by improving heat transfer processes in their elements. The presented results are limited to studies of only the thermohydraulic efficiency of single and banks of plane-oval tubes and single and multi-row banks (transverse and longitudinal pitch ratio S_1/d=3.14 and S_2/d=1.57) of elliptical tubes. A joint analysis of the thermohydraulic efficiency and compactness of close (S_1/d and S_2/d<1.5) heat transfer surfaces due to the use of non-circular tubes and a corresponding change in the compactness of the power plant as a whole is not considered. This will allow solving the problem of combining increased thermohydraulic efficiency of heat transfer processes in the elements of power plants and their compactness.

3. The aim and objectives of the study

The aim of the study is to increase the compactness of power plants by improving the thermohydraulic efficiency of heat transfer processes in their elements.

To achieve the aim, the following objectives were set:

– to substantiate the change of compactness of power plants when changing the compactness of heat transfer elements;

– to substantiate a complex indicator of thermohydraulic compactness, taking into account the thermohydraulic efficiency and compactness indicator of the heat transfer surface, and, on its basis, to carry out a comparative assessment for single and banks of circular, elliptical and flat oval tubes;

– to determine the change of compactness of power plants depending on the performance of heat exchangers;

– to substantiate dimensionless indicators of compactness of heat exchangers as elements of power plants.

4. Materials and method of the study of thermohydraulic characteristics of plain shaped tubes

The method of the study is mathematical modeling of heat movement and transfer processes in elements of power plants. For this, the Computation Fluid Dynamics (*CFD*) method was used. The mathematical model of the processes was formed as a system of the following equations [15]:

– continuity equation

$$\frac{\partial\rho}{\partial t} + \nabla\left(\rho\vec{V}\right) = 0, \tag{6}$$

where ρ is the mass flow density, $\vec{V}$ is the vector of the local flow velocity;

– momentum conservation equation

$$\frac{\partial\left(\rho\vec{V}\right)}{\partial t} + \nabla\left(\rho\vec{V}\vec{V}\right) = \nabla p + \nabla\tau + \rho\vec{g} + \vec{F}, \tag{7}$$

where p is the static pressure; $\rho\vec{g}$ is the gravitational force acting per unit mass; $\vec{F}$ is the external forces acting on the flow; τ is the pressure tensor.

– energy conservation equation

$$\frac{\partial(\rho E)}{\partial t} + \nabla\left(\vec{V}(\rho E + p)\right) = \nabla\left(-\vec{J}_q + \left(\tau_{eff}\vec{V}\right)\right), \tag{8}$$

where $\vec{J}_q$ is the heat flux density, $E = h - \frac{p}{\rho} + \frac{V^2}{2}$ is the total energy of the working fluid, h is the enthalpy of the working fluid, the expression $\tau_{eff}\vec{V}$ determines viscous heating.

The above system of equations is not closed. It is closed by adding semi-empirical dependences for the pressure tensor, heat flux, ideal gas equation, as well as differential equations of the turbulence model.

Newton's law. Using the assumption of neglecting bulk viscosity, the stress tensor can be represented as

$$\tau=\mu\left[\nabla\vec{V}+\left(\nabla\vec{V}\right)^{T}-\frac{2}{3}\nabla\left(\vec{V}I\right)\right], \tag{9}$$

where μ is the molecular viscosity coefficient, I is the unit vector.

Fourier's law. Heat flux is determined by the expression

$$\vec{J}_q=-\lambda_{eff}\nabla T, \tag{10}$$

where $\lambda_{eff}=\lambda+\lambda_t$ is the coefficient of effective thermal conductivity.

Mendeleev-Clapeyron's law. The basic ideal gas law, establishing the relationship between the main thermodynamic parameters, is as follows

$$p=R\cdot\rho\cdot T, \tag{11}$$

where R is the individual gas constant of the working fluid.

Selection of turbulence model for calculations of heat transfer processes in tube banks. The choice of turbulence model has a significant impact on the results of mathematical modeling of heat transfer processes in tube banks. A fairly large number of different hydrodynamic turbulence models are known: standard k-ε turbulence model, large eddy simulation model (*LES* model), Reynolds stress model (*RSM* model), and others. Heat transfer processes in tube banks are characterized by vortex flows in the inter-tube space, secondary and rotating flows in the channels, caused by the processes of flow and heat transfer of transfer media. Therefore, the use of the *RSM* model is justified in terms of computational resources, since simpler models cannot provide reliable results [15].

Solution of the presented mathematical model (6)–(11) was made using the control volume method [16].

Verification of the results was carried out by comparison with available literature data obtained experimentally [4]. The results of the comparison are shown in Fig. 1.

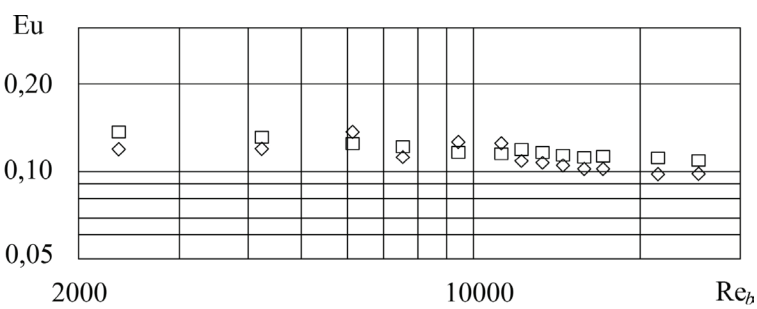

Fig. 1. Verification of the mathematical model: □ – data from [4]; ◊ – simulation results

The calculation was made for similar geometric parameters of the heat transfer surface in the range of Reynolds numbers 2,500...12,000. The discrepancy between the results did not exceed 2.4 %. This made it possible to use the accepted mathematical model (6)–(11), supplemented by the *RSM* turbulence model for further research.

5. Results of changing the compactness of power plants using shaped heat transfer surfaces

5. 1. Changing the compactness of power plants when changing the compactness of heat transfer elements

The volume occupied by all equipment of the plant in equation (1) can be represented as

$$V_{eq}=V_{en}+V_{he}+\sum V_{el}, \tag{12}$$

where V_{en} is the volume occupied by the main and auxiliary engines; V_{he} is the volume of the largest heat exchangers (waste heat recovery units, regenerators, etc.); ΣV_{el} is the total volume of the remaining elements of the plant.

The volume of the heat exchanger can be represented as

$$V=V_{ht}+V_{sh}+V_{\sum pip}, \tag{13}$$

where V_{ht} is the volume of the heat transfer surface, V_{sh} is the volume of the shell, $V_{\Sigma pip}$ is the total volume of piping elements (inlet and outlet branches, tube plates, partitions, etc.).

When upgrading heat exchange equipment, the change in its volume can be expressed as (index "0" corresponds to the basic heat exchanger)

$$\frac{V}{V_0}=\frac{V_{ht}\left(1+\bar{V}_{sh}+\bar{V}_{\sum pip}\right)}{V_0\left(1+\bar{V}_{sh0}+\bar{V}_{\sum pip0}\right)}. \tag{14}$$

Assuming that $\frac{V_{ht}\left(1+\bar{V}_{sh}+\bar{V}_{\sum pip}\right)}{V_0\left(1+\bar{V}_{sh0}+\bar{V}_{\sum pip0}\right)}\cong 1,$ the change in the volume of the heat exchanger and, accordingly, the power plant, can be estimated by

$$V=V_0\cdot\bar{V}_{ht}, \tag{15}$$

where V_0 is the volume of the base heat exchanger to be compared,

$\bar{V}_{ht}$ is the relative change in the heat exchanger volume when improving heat transfer processes.

5. 2. Complex indicator of thermohydraulic compactness of heat transfer surfaces

The amount of heat transferred by the heat transfer surface element of a power plant is determined based on the heat transfer equation

$$Q=\alpha\cdot F\cdot\Delta t.$$

Since the heat transfer coefficient α depends on a large number of factors – surface shape, thermophysical parameters of heat transfer media, etc., in order to increase the transferred heat flux Q, it is advisable to increase not only α, but the product α·F. From the provision of compactness, this product should tend to a maximum with a minimum volume, i.e.

$$\frac{\alpha\cdot F}{V}=\alpha\cdot\frac{F}{V}\rightarrow\max. \tag{16}$$

According to the relationship (2), the heat transfer efficiency is determined by the Reynolds analogy factor FAR_α, while for non-intensified surfaces this indicator is 1, and differs from 1 for intensified ones. Then equation (16) can be represented as a criterion of "thermohydraulic" compactness of heat exchange

$$K_{thc}^{\alpha} = FAR_{\alpha} \cdot K_{geom}, \tag{17}$$

or heat transfer

$$K_{thc}^{k} = FAR_{k} \cdot K_{geom}. \tag{18}$$

Features of the formation of compact heat transfer surfaces of circular, elliptical and plane-oval tubes are considered. The condition of constant area of heat transfer surface is accepted.

Single tubes. To assess the geometric and thermohydraulic compactness, the diameter of a circular tube d=0.022 m was taken. The geometric characteristics of plane-oval tubes were taken according to [4], and elliptical – calculated [17]. Three options for comparison are considered – with a constant equivalent diameter d_{eq}, circumference L and cross-sectional area S. When calculating elliptical tubes, the semi-axis ratio (a/b) in the range of 1.5...2.7 are accepted. Initial data for modeling the external flow around and internal flow are taken according to [18].

The results of the calculation are shown in Fig. 2.

In the calculations of $FAR_{\alpha 1}$ and FAR_k, the results of calculations for a circular tube were taken as a basic option.

Analysis of the results allows drawing the following conclusions. When comparing tubes with the same equivalent diameter, the best heat transfer efficiency in terms of the $FAR_{\alpha 1}$ index is exhibited by elliptical tubes in almost the entire range of semi-axis ratios. However, under heat transfer conditions, plane-oval tubes have the best indicators of thermohydraulic compactness (Fig. 2). Moreover, for elliptical tubes, a local maximum of the thermohydraulic compactness coefficient is observed in the region a/b=2.5, while for a plane-oval tube this value is a/b=2.75.

Similar results are observed when comparing tubes with the same circumference L and cross-sectional area S.

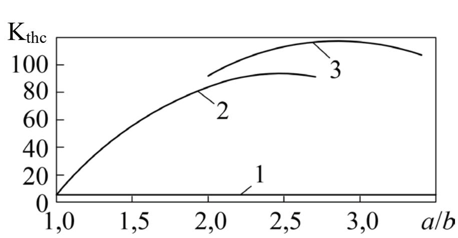

Fig. 2. Comparative characteristics of single shapes under heat transfer conditions with d_{eq}=const: 1 – circle, 2 – ellipse, 3 – plane-oval

Of great importance in calculating the external flow around surfaces is the correct choice of the determining size in terms of similarity numbers. According to [19], the determining factor during the external flow around bodies is the size, which boundary layer formation depends on. Therefore, when calculating the flow around non-circular bodies, it is advisable to take the size of the minor semi-axis b as the determining one. Then from equation (4), it can be found that the coefficient of heat transfer during the flow around non-circular bodies will be higher by

$$\alpha = \alpha_0 \cdot \frac{d_{eq}}{b}.$$

Tube banks. The study of the change in the heat transfer coefficient along the tube bank rows is shown in Fig. 3. For comparison, summarized data are given according to [20].

The literature data on the distribution of the heat transfer coefficient over the tube rows are rather ambiguous. If [20] provides data on heat transfer stabilization, starting from the third row, then, according to [21], stabilization is carried out from the 5...6 rows. This depends on the initial turbulence level of the first rows of tubes and is taken into account by the corresponding correction factors.

The flows of working media of power plants have relatively high velocities and temperatures; therefore, a high degree of turbulence was set in the simulation. This causes the difference in the values for the circular tube bank according to [20] and those calculated in the test simulation.

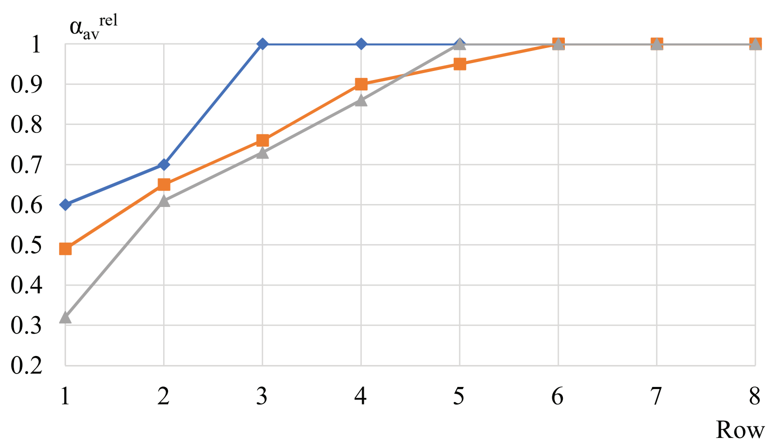

Fig. 3. Changes in the relative heat transfer coefficient for rows of staggered circular and elliptical tubes: ♦ – values for the circular tube bank [20]; ■ – values for the circular tube bank obtained from test simulation results; ▲ – values for the elliptical tube bank

The average heat transfer coefficient was calculated as [20]

$$\bar{\alpha} = \frac{\sum_{i=1}^{n} (\alpha_i \cdot F_i)}{\sum_{i=1}^{n} F_i},$$

where i is the number of rows; α is the heat transfer coefficient of the i-th row; F is the area of the i-th row tubes.

It was found that for the elliptical tube bank, heat transfer stabilization begins from the fifth row. Values of correction factors – for the first row – 0.32, for the second – 0.61, the third – 0.75, the fourth – 0.86. Starting from the fifth row, the heat transfer coefficient stabilized and equaled the average in volume.

5. 3. Changing the compactness of power plants

Assessment of the change of compactness of power plants depending on the efficiency of circular, elliptical and plane-oval tube banks is performed. The comparison was made for the following options: upgrading the existing heat transfer element and designing a new one.

For the case of upgrading, the following options are considered:

– with the same relative pitches S_1/d_{eq} and S_2/d_{eq} and the same lateral surface area;

– with the same number of heat transfer elements.

In the first case, the thermohydraulic compactness calculated by the FAR_k index was 0.057, 0.056 and 0.055 for circular, elliptical and plane-oval tubes, respectively. However, the relative change in the volume of the heat exchange surface, and, accordingly, the entire plant, according to the relationship (15), was 0.817 for the bank of elliptical tubes and 0.841 for plane-oval tubes. In the case of the same number of heat transfer elements, the thermohydraulic compactness calculated by the FAR_k index was 0.072, 0.151 and 0.112 for circular, elliptical and plane-oval tubes, respectively.

For the case of designing a new heat transfer element, an option with the same geometric compactness indicator is considered. The values of the thermohydraulic compactness indicator were $K_{thc}^{k}=0.077$, 0.143 and 0.116 for circular, elliptical and plane-oval tubes, respectively.

Thus, for given geometrical dimensions of the heat exchange surface, the heat transfer surface formed from elliptical tubes will have higher thermohydraulic compactness.

These applications of elliptical tubes as heat transfer surfaces will make it possible to increase the compactness of the entire power plant up to 18.3 % compared to circular tubes and up to 2.4 % compared to plane-oval ones.

5. 4. Dimensionless indicators of compactness of heat exchangers of power plants

The mass and volume indicators presented in equations (1) and (2) for estimating the compactness of power plants are dimensional. This makes it difficult to use them to determine the comparative efficiency of various power plants. In addition, the ambiguity of the effect of the input quantities on the final result makes it difficult to use them at various hierarchical levels of analysis.

Therefore, it is advisable to substantiate complex dimensionless indicators for analysis. For this, the following dimensional values are taken: mass (kg), length (m), temperature (deg), and dimensionless indicators for heat exchange equipment are the mass of the heat exchanger (HE) M_{HE}, its volume V_{HE}, functional efficiency η_{HE} and service life.

Mass index. The mass of the heat exchanger can be calculated as

$$M_{HE}=\rho_{eff}\cdot V, \tag{17}$$

where $\rho_{eff}=\rho_{mat}\cdot(1-\Pi)+\rho_{air}\cdot\Pi$; ρ_{mat} is the density of heat transfer surface material, kg/m^3; ρ_{air} is the air density, kg/m^3; $\Pi=\frac{V-V_{mat}}{V}=1-\frac{V_{mat}}{V}$; $V_{mat}=F\cdot\delta_w$ is the volume of the heat transfer surface of the heat exchanger, m^3; $V=\frac{F}{K_{geom}}$ is the total volume of the heat exchanger, m^3; F is the heat transfer surface area, m^2; K_{geom} is the geometric compactness factor, 1/m; δ_w is the wall thickness of the heat transfer surface of the heat exchanger, m.

Heat transfer surface area is calculated as

$$F=\frac{Q}{k\Delta t},$$

where Q is the amount of heat transferred by the heat exchanger, W; k is the heat transfer coefficient, W/(m^2·K); Δt is the average logarithmic temperature difference between heat transfer media in the heat exchanger, °C.

Given the above, equation (17) is converted to

$$M=\frac{Q}{k\cdot\Delta t}\left\{\rho_{mat}\cdot\delta_w-\rho_{surf}\left[\frac{1}{K_{geom}}-\delta_w\right]\right\}.$$

Thus, the mass of the heat exchanger can be represented as a function of the following parameters

$$M=M\left(Q,k,\Delta t,\rho_E,\delta_w,\rho_{surf},K_{geom},\eta_{HE}\right).$$

If we represent k and Δt as

$$k=\frac{1}{\frac{1}{\alpha_1}+\frac{\delta_w}{\lambda_w}+\frac{1}{\alpha_2}},$$

where α_1 and α_2 are the heat transfer coefficients of hot and cold transfer media, respectively, W/(m^2·K); λ_w is the heat conductivity coefficient of heat transfer surface material, W/(m·K);

$$\Delta t=\frac{\Delta T_b-\Delta T_m}{\ln\frac{\Delta T_b}{\Delta T_m}}=\frac{(T_h-T_c)_b-(T_h-T_c)_m}{\ln\frac{(T_h-T_c)_b}{(T_h-T_c)_m}},$$

then the mass of the heat exchanger can be represented by the following function

$$M=M\left(Q,\ \alpha_1,\ \alpha_2,\delta_w,\ \lambda_w,t_0,\ \rho_{mat},\ \rho_{surf},K_{com},\ \eta_{HE}\right). \tag{18}$$

For further conversion of equations (17) and (18), the dimensional method was used [19].

The primary values are: mass – [M], kg; length – [L], m; time – [T], s; temperature – [θ], °C.

Performing dimensional transformations of quantities and transforming variables using a new dimension scale, we obtained

$$M\cdot[\mathrm{M}];\ Q\cdot\left[\frac{M\cdot L^2}{T^3}\right];\ \alpha_1\cdot\left[\frac{M}{T^3\cdot\theta}\right];\ \alpha_2\cdot\left[\frac{M}{T^3\cdot\theta}\right];$$

$$\Delta_w\cdot[L];\ \lambda\cdot\left[\frac{M\cdot L}{T^3\cdot\theta}\right];\ t_0\cdot[\theta];\rho_m\cdot\left[\frac{M}{L^3}\right]\rho_a\cdot\left[\frac{M}{L^3}\right];\ K_{geom}\cdot\left[\frac{1}{L}\right].$$

To form a functional relationship, the following single-valuedness conditions are assigned

$$Q_n\cdot\left[\frac{M\cdot L^2}{T^3}\right]=1;\ t_0\cdot[\theta]=1;\ \delta_w\cdot[L]=1;\rho_m\cdot\left[\frac{M}{L^3}\right]=1,$$

where Q_n is the nominal value of the heat exchanger power, W.

Performing the following transformations

$$[L]=\frac{1}{\delta_w};\ [M]=\frac{L^3}{\rho_m}=\frac{1}{\rho_m\cdot\delta_w^3};\ [\theta]=\frac{1}{t_0};$$

$$\left[T^3\right]=Q_n\left[M\cdot L^2\right]=Q_n\frac{1}{\rho_m\cdot\delta_w^3}\frac{1}{\delta_w^2}=\frac{Q_n}{\rho_m\cdot\delta_w^5},$$

and substituting the obtained expressions into equation (18), one can finally obtain in a dimensionless form

$$\frac{M}{\rho_m \cdot \delta_w^3} = f\left\{\begin{matrix} \frac{Q}{Q_n}; \alpha_1 \frac{\delta_w^2 \cdot t_0}{Q_n};\ \alpha_2 \frac{\delta_w^2 \cdot t_0}{Q_n}; \\ \lambda \frac{\delta_w \cdot t_0}{Q_n}; \frac{\rho_a}{\rho_m}; K_{geom} \cdot \delta_w; \eta_{HE} \end{matrix}\right\}. \quad (19)$$

Processing of open Internet data [22–25] on heat exchanger designs made it possible to obtain the following relationship for the dimensionless index

$$\overline{m} = \frac{M}{\rho_m \cdot \delta_w^3} = \left(\alpha_1 \frac{\delta_w^2 \cdot t_0}{Q_n}\right)^{-0.198} \times$$
$$\times \left(\alpha_2 \frac{\delta_w^2 \cdot t_0}{Q_n}\right)^{-0.369} \cdot \left(\lambda \frac{\delta_w \cdot t_0}{Q_n}\right)^{-0.156} \cdot \left(\eta_{HE}\right)^{-0.0243}. \quad (20)$$

Volume index. The volume of the heat exchanger can be defined as

$$V_{HE} = \frac{F}{k \cdot \Delta t \cdot K_{geom}}.$$

Similar to equations (17) and (18), the volume of the heat exchanger can be represented as a function of the following parameters

$$V_{HE}=V\,(Q, \alpha_1, \alpha_2, \delta_w, \lambda_w, t_c, t_h, K_{com}, \eta_{HE}), \quad (21)$$

where t_c and t_h are the temperatures of the cold and hot transfer medium, respectively, °C.

Performing transformations similar to the above, the relationship for calculating the volume of the heat exchanger can be represented as

$$\frac{V_{HE}}{\delta_w^3} = f\left\{\begin{matrix} \frac{Q}{Q_n}; \alpha_1 \frac{\delta_w^2 \cdot t_0}{Q_n};\ \alpha_2 \frac{\delta_w^2 \cdot t_0}{Q_n}; \\ \lambda \frac{\delta_w \cdot t_0}{Q_n};\ K_{com} \cdot \delta_w; \eta_{HE} \end{matrix}\right\}. \quad (22)$$

Functional index. The functional efficiency of the heat exchanger η_{TA} is determined as

$$\eta_{HE} = \frac{t_c^{out} - t_c^{in}}{t_h^{out} - t_c^{in}} = \frac{\delta t_c}{\delta t_{\max}},$$

where t_c^{in}, t_c^{out} and t_h^{out} are the temperatures of the cold transfer medium at the inlet and outlet and the hot transfer medium at the outlet of the heat exchanger, °C.

According to [20, 21]

$$\frac{t_h^{out} - t_c^{out}}{t_h^{in} - t_c^{in}} = e^{-mkF},$$

where $m = \left(\frac{1}{C_h} + \frac{1}{C_c}\right) = \left(\frac{1}{G_h \cdot c_p^h} + \frac{1}{G_c \cdot c_p^c}\right)$ in the case of direct flow and $m = \left(\frac{1}{C_h} - \frac{1}{C_c}\right) = \left(\frac{1}{G_h \cdot c_p^h} - \frac{1}{G_c \cdot c_p^c}\right)$ countercurrent flow of heat transfer media, and G_h and G_c are the mass flow rates of hot and cold transfer media, kg/s; c_p^h and c_p^c are the specific mass heat capacities of hot and cold transfer media, J/(kg·deg).

After performing the corresponding transformations, the expression for the functional efficiency indicator is obtained

$$\eta_{HE} = \frac{\delta t_c}{\delta t_{\max}} = \frac{\left[1 - e^{-mkF}\right]}{\left[1 + \frac{C_c}{C_h}\right]}. \quad (23)$$

Service life index. The service life of the heat exchanger T_{HE} can be represented as

$$T_{HE} \Rightarrow \min\ \{T_1,\ T_2,\ \ldots, T_i\},$$

where T_1, T_2, ... T_i, i=1...n are the service lives of individual heat exchanger units.

Without taking into account heat exchanger repairs, the service life at preliminary design stages can be estimated as equal to the service life of the power plant, i.e.

$$T_{HE}=T_{pp}.$$

At the engineering and detailed design stages in accordance with [26–28], the service life of an individual heat exchanger unit can be represented as

$$T_i=T\ (p,\ t,\ \delta_w,\ K_{geom},\ \sigma \cdot 10^T/t,\ n_i,\ \varphi_i,\ Y_i),$$

where p and t are the pressure, Pa and temperature, °C, of the heat transfer medium, respectively; δ_w is the wall thickness of the heat exchanger, m; K_{geom} is the geometric compactness factor, 1/m; $\sigma \cdot 10^T/t$ is the conditional limit of long-term strength, MPa; n_i is the safety factor; φ_i is the estimated strength factor of the part; Y_i is the shape factor.

Using the above dimensional method, the expression for determining the service life is presented as the following relationship

$$\frac{T}{T_p} = f\left\{\frac{p}{p_e};\ \frac{t}{t_e};\ K_{geom} \cdot \delta_w; \frac{\sigma \cdot 10^T / t}{p}; n_i; \phi_i; Y_i\right\}, \quad (24)$$

where T_e is the estimated service life of the heat exchanger, Pa; p_e is the estimated pressure of the transfer medium, Pa; t_e is the estimated temperature of the heat transfer medium, °C.

6. Discussion of the results of the study of using shaped surfaces to increase the compactness of power plants

A significant reserve for increasing the efficiency of power plants is the recovery and utilization of exhaust gas heat. An increase in the efficiency of these processes leads to an increase in weight and size and a decrease in the compactness of plants, which limits the possibilities for their implementation. In addition, the limited initial data on the composition of power plants makes it difficult to assess the change of their compactness when improving heat transfer processes.

Representation of the volume of the power plant as the sum of its elements in the form of relationships (14) and (15) allows evaluating the change of compactness depending on the change in the characteristics of the main heat exchanger element – heat transfer surface. Based on these data, it is possible to estimate the change in the compactness of the entire plant based on relationships (1) and (2).

Improving the efficiency of heat conversion processes and compactness of power plants is possible through the use of shaped heat transfer surfaces. The proposed indicator of thermohydraulic compactness – equation (17) – allows a comprehensive assessment of heat transfer efficiency depending on the layout of surfaces. The results on the basis of this indicator presented in Fig. 2 allow substantiating the geometric dimensions of the elements of the heat transfer surfaces to achieve maximum efficiency of tube banks.

The results of changing the compactness of the power plant should be refined for specific conditions. They will depend on the purpose of the plant, layout, efficiency requirements and production capabilities.

The proposed complex of dimensionless indicators (19), (22)–(24) will allow substantiating the most efficient heat exchangers of the power plant to achieve maximum compactness. The limitation of the presented results is that only one dimensionless relationship is available for a comparative assessment of the mass of heat exchangers of various power plants. The absence of other dependencies is objectively explained by the lack of initial data in open sources to obtain them. If data are available, mathematical processing is not difficult.

The mathematical apparatus used made it possible to determine the geometric parameters of banks of plain shaped tubes. Therefore, the development of the presented work consists in experimental studies of non-circular heat transfer surfaces with a transverse and longitudinal pitch ratio of less than 1.5.

The results show the efficiency of using non-circular tubes to increase the compactness of power plants. For stationary power plants, if the compactness requirements are not decisive for them, the use of plane-oval tubes can be recommended, since they are more manufacturable. For portable plants, for which compactness is important, elliptical tubes are preferable.

In general, the results obtained will make it possible to solve the problem of combining increased thermohydraulic efficiency of heat transfer processes in the elements of power plants and their compactness.

7. Conclusions

1. The change of compactness of power plants is proposed to be evaluated by changing the volume of heat transfer elements while improving heat transfer processes.

2. The criterion of thermohydraulic compactness of the surface is substantiated, taking into account both the layout and arrangement of heat transfer elements and their thermohydraulic efficiency. On its basis, it was found that for single elliptical and plane-oval shapes there is a local maximum of this indicator, which is achieved at the axis ratio of 2.5 for elliptical and 2.75 for plane-oval. At the same time, single plane-oval tubes have a higher value of compactness index. Elliptical tube banks have higher thermohydraulic compactness.

3. The change of compactness of power plants depending on the parameters of heat exchangers was estimated. It was found that when using elliptical tubes as heat transfer surfaces, it is possible to increase the compactness of the entire power plant up to 18.3 % compared to circular tubes and up to 2.4 % compared to plane-oval ones.

4. Dimensionless indicators of mass, volume, functional efficiency and service life of the heat exchanger were substantiated, allowing them to be compared as part of various power plants.

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