1. Introduction

Enhancing economic and ecological efficiency of thermal power plants (TPP) is very essential at present. Significant improvement of these characteristics is possible through a widespread application of steam-gas plants (SGP) of the gas-steam-turbine type (GST) [1]. However, weight, dimensions, and cost of one of the main elements of GST – boilers-utilizers, heat exchange surfaces of which are heated by relatively low-temperature exhaust gases of gas turbines, are high enough. That is why one of the important problems is decreasing metal intensity and cost of convective elements at the expense of increasing specific heat transfer surface.

2. Literature review and problem statement

A decrease in the metal intensity of boilers at present is achieved mainly by the development of specific surface area of heat exchange through tube finning on the side of flue gases. Transverse spiral-tape finning is most widely applied for heating surfaces of boilers and boilers-utilizers. Finning technology is employed in mass production. In addition, such finning provides sufficiently large development of heat transfer surface area – by 8...10 times or larger. Mass-dimensional, techno-economic and thermo-aerodynamic characteristics of spiral-tape finning are also quite high.

Plate transversal tube finning (also called “petal-type”, “N-type”) is becoming more and wider applied. However, according to results of studies [2, 3], thermo-aerodynamic and mass-dimensional characteristics of the “petal-type” finning only in certain cases can compete with spiral-tape finning.

Transverse finning weakens heat exchange relative to bare-tube heat exchangers. A positive effect is achieved only at the expense of increasing specific heat transfer surface. That is why an important problem is intensification of heat exchange in transversally-finned heating surfaces. In this regard, punched spiral-tape tube finning (also called “segmented”, “cut”, and “serrated”), the fragments of which are shown in Fig. 1, seems promising.

There are several publications on the research into heat transfer and aerodynamic resistance of banks of tubes with punched spiral finning, including [4–8] in staggered banks, and [9–11] in in-line banks. Evaluation of heat exchange intensification of fin punching is different in various papers. According to research [4, 6, 8] and [9, 10], an average heat exchange intensification in staggered banks is 25...38 %, in in-line banks, it is 15...27 % relative to the continuous spiral finning. Aerodynamic resistance of staggered banks increases by 20...40 %, of the in-line banks – by 18...36 %.

Availability of engineering methods of calculation of heat transfer and pressure loss in tube banks with the specified finning is of great importance for practical application. Information on calculation methods is limited. The first calculation method is presented in [12] and then with slight modifications by the American Corporation ESCOA...
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("Extended Surface Corporation of America") [13]. The method allows calculating heat exchange and aerodynamics of staggered and in-line tube banks with continuous and punched finning. The method does not take into account the influence of geometric characteristics of punched part of fins on the heat exchange.

Another method for the calculation of heat transfer of staggered and in-line tube banks with continuous spiral finning is presented in [14]. The method was used for calculation of heat exchange in banks of tubes with punched spiral finning [4, 10], and then in [8, 11]. The influence of geometry of the punched part was not taken into consideration in this method. At the same time, comparison of results of the experiment [8, 11] with the result of calculation by generalizations of [4, 10] for the conditions of the experiment [8, 11] showed a significant impact of this factor on heat exchange. In generalization of [8], the results of numerical research [15, 16] were used in order to consider geometrical characteristics of the punched part of fins.

It should be noted that results of calculation of heat transfer of in-line tube banks by generalization of [10] for the conditions of experiment in [11] were not consistent with the results of this experiment and generalization [11]. The character of influence of finning degree of heat transfer in various zones of a change relative to longitudinal tube pitch is not consistent.

Experimental studies of heat exchange of tubes with punched finning at additional deformation of the punched part of fins [17, 18, 19] have been recently carried out. Paper [17] presents results of research into tube banks with punched finning at petals rotation at the angle of 30° relatively the direction of incoming flow. There was an increase in heat transfer by 44 % at an increase in pressure loss by 16...40 % relative to continuous spiral finning. A somewhat different assessment of effect of petals turning was received in numerical study [18]: a positive effect is achieved at rotation angles of less than 15°. At the angles exceeding 15°, energy consumption for overcoming aerodynamic resistance exceeds amount of transferred heat. In present paper, heat exchange intensification by punching of continuous fins is estimated at 12.3 % at the same height of the fins. Article [19] explores heat exchange of a bank of tubes with punched fins with five shapes of petals. Heat exchange intensification by 176...21.5 % was obtained due to petals deformation. Pressure loss was not determined. Complexity of manufacturing of finning was not assessed either.

It follows from the presented review that thermo-aerodynamic efficiency of banks of tubes with punched spiral finning was not studied enough. Even less explored are the banks of tubes with deformed elements of punched finning – petals (segments). Therefore, a very important issue of heat exchange intensification in finned convective heating surfaces of boilers is subject to additional research. In particular, it is required to determine conditions for effective practical application of specified heating surfaces.

3. The aim and objectives of the study

The goal of present research is to obtain new data to determine rational thermo- aerodynamic characteristics of banks of tubes with punched spiral-tape finning.

To accomplish the set goals, the following tasks had to be solved:

- to conduct calculation studies of thermo-aerodynamic efficiency of staggered and in-line banks of tubes with specified finning type;
- to generalize experimental results;
- to carry out experimental research into heat exchange and aerodynamic resistance of in-line banks of tubes with punched spiral finning based on results of experimental studies.

4. Experimental study of heat exchange and aerodynamic resistance of in-line banks of tubes with punched finning

4.1. Technique for studying the average-surface heat exchange and aerodynamic resistance

4.1.1. Technique of research into heat exchange

The study was conducted at the laboratory experimental setup, described in [6, 8, 11]. Heat exchange was explored with the method of complete thermal modeling under stationary conditions at forced convection and electric heating of all tubes in the bank. We determined average-surface heat transfer of deep tube rows in the banks when blown over with air. Thermal-physical parameters of air were determined at average air temperature in the row of tubes-calorimeters mounting. The study of aerodynamic resistance was performed under isothermal conditions at temperature of 20...22 °C.

Average-surface heat transfer coefficient of convection of tube-calorimeters was derived from formula:

$$\bar{\alpha} = \frac{Q}{H (t_i - t_s)} \text{ W/m}^2\text{°C},$$

where $Q$ is the heat release of the calorimeter, W; $H$ is the total surface area of tubes-calorimeters, m$^2$; $t_s$ is the averaged surface temperature of calorimeter (measured at rotation of a tube-calorimeter around its axis from 0° to 180° in direction of incoming flow with pitch of 30°), °C; $t_i$ is the average air temperature in the row of calorimeter mounting, °C.

The averaged temperature of the calorimeter surface was derived from formula:

$$t_s = \frac{\sum t_i H_i}{\sum H_i} \text{ °C},$$

where $t_i$ is the surface temperature of calorimeter section $H_i$, on which the $i$-th thermocouple was mounted.

Average air temperature in the row of calorimeter mounting was obtained from ratio:

$$t_s = t_s' + \frac{\sum Q' + 0.5Q}{GC_p} \text{ °C},$$
where $t'_e$ is the air temperature at the inlet to the studied bank, °C; $Q_e$ is the heat release of the rows, where the calorimeter is mounted, W; $\sum Q'_c$ is the total heat release of rows, preceding to the row of calorimeter mounting, W; $G$ is the air consumption, kg/s; $C_p$ is the specific mass thermal capacity of the air, kJ/kg°C.

Nusselt criterion is determined from formula:

$$Nu = \alpha \cdot d / \lambda_{st},$$

where $\lambda_{st}$ is the coefficient of thermal conductivity of the air, W/m°C.

Resistance of tube banks was determined by measured difference of static pressures before and after the bank. Pressure loss in the banks was determined by the difference of the measured pressure drops and resistance of the free channel, including resistance to friction and local resistance of mounting elements and spacing of tube boards. Resistance of the free channel was measured at the same air velocities, at which static drops of pressure on the banks were measured.

$$\Delta P = \Delta P_{st} - \Delta P_f.$$

The Euler numbers, referring to one transverse row of tubes, were calculated from obtained values $\Delta P_f$:

$$Eu = \frac{\Delta P_f}{Z_p U^2},$$

where $Z_p$ is the number of transverse rows of tubes in banks; $U$ is the air velocity in minimal cross section of one transverse row, m/s.

The following basic physical magnitudes were measured: air consumption and its temperature before and after the studied tube banks, temperature of operating heat exchange surfaces of tubes-calorimeters, heat release of tubes-calorimeters, air pressure drop on tube banks. Air consumption was measured by the pneumatic method using a three-channel pneumatic probe. Differential pressure on the probe was measured with micromanometer MMN-300 of accuracy class of 0.6. Air temperature before and after tube banks was measured with a mercury thermometer of SP-25 type within measuring limits of 10...40 °C, as well as with thermometers of standard calibration XK. Temperature of operating surface of tubes-calorimeters was measured with thermocouples of standard XA calibration. Measurements were carried out at turning of the tube-calorimeter from 0° to 180° relative to direction of incident flow with pitch of 30°. Thermo-electromotive force of thermocouples was measured with a voltmeter V7-34A. The device error within the operating range, calculated from the passport formula, is (0.28...0.62) %. To determine heat release of tubes-calorimeters, we measured voltage and current power, supplying the heaters. The heaters were supplied with alternating current through AC voltage stabilizer. Voltage was measured with a voltmeter V7-38, amperage was measured with ammeter DS0141 of 0.2 accuracy class. Error of voltmeter V7-38, calculated from passport formula, was (0.081...0.091) %. Differential static air pressure on the banks was measured with micromanometer MMN-240 of 1.0 accuracy class. When processing empirical data, measurement errors were assessed. Analysis of measurement results at fitting of experimental setup showed that direct measurement error of most of the listed physical magnitudes was determined, basically, by systemic errors resulting from instrument errors. For this reason, measurements of specified physical magnitudes in the main experiments were not repeated. When measuring the surface temperature of the tube-calorimeter by thermocouples, systematic and random errors were close, that is why these measurements were repeated and random errors were determined. As a result of experiment processing, the following values of common measurement errors of basic parameters were established: Reynolds numbers (1.5...5.4) %, heat transfer coefficient (7.1...7.6) %, Nusselt number – (8.2...8.8) % and $Eu$ number – (5.4...15.3) %. For this part of the study, we manufactured and prepared new tubes-calorimeters with increased number of points of measuring of fin wall temperature and air temperature on the height of inter-fin channel. We also improved the program of computer processing of empirical data in terms of determining the surface area of heat exchange of finning.

The study used finned tubes with different pitches of fins ($S_f$) and, as a result, with different coefficients of finning ($\varphi$): $S_f = 5$ mm ($\varphi = 9.012$) – series 1; $S_f = 6$ mm ($\varphi = 7.677$) – series 2; and $S_f = 8$ mm ($\varphi = 6.010$) – series 3. The other geometrical dimensions of tubes were the same: outer diameter of the tube $d = 28$ m, fin height $h_1 = 14.5$ mm, petal height $h_2 = 9.5$ mm, petal width $b_1 = 4$ mm, fin thickness $\delta = 1.0$ mm. (Designations are shown in Fig. 1). Relative petal height $h_f/h_1 = 0.66$ is maximal permissible according to conditions of finning manufacturing. For a characteristic of the geometry of banks, we accepted relative longitudinal pitch of tubes $\sigma = S_f/d$. Geometrical characteristics of some studied banks are shown in Table 1.

| Geometrical characteristics of studied tube banks |
|---|---|---|---|---|
| Tube series | Number of location | $\sigma_1$ | $\sigma_2$ | $C_1$ | $C_2$ | $C_3$ |
|---|---|---|---|---|---|---|
| 1 | 3.5 | 2.143 | 0.803 | 0.029 | 0.790 | 0.035 | 0.772 | 0.045 |
| 2 | 3.5 | 2.043 | 0.801 | 0.031 | 0.784 | 0.040 | 0.752 | 0.071 |
| 3 | 3.5 | 3.036 | 0.788 | 0.044 | 0.755 | 0.075 | 0.700 | 0.135 |
| 4 | 3.5 | 3.500 | 0.710 | 0.110 | 0.680 | 0.130 | 0.669 | 0.179 |
| 5 | 3.5 | 4.286 | 0.685 | 0.128 | 0.671 | 0.150 | 0.653 | 0.186 |
| 6 | 3.5 | 5.286 | 0.683 | 0.130 | 0.670 | 0.150 | 0.653 | 0.189 |
| 7 | 2.5 | 2.500 | – | – | – | – | – | – |

The Reynolds number, related to the outer diameter of tubes, varied in experiences within $Re_f = (6.3...38.0) \cdot 10^3$ by changing air consumption through studied banks.

In the experimental study, we also determined reduced heat transfer coefficient ($\alpha$) and actual thermal efficiency of a fin taking into account uneven heat transfer on its heat exchange surface ($Ew_f$).

4.1.2. Technique for examining aerodynamic resistance

The study was conducted on the same experimental setup for the same types of tubes. We accepted reduced length of developed surface $H/F$ and ratio of transversal pitch to longitudinal pitch ($S_f/S_1$) as parameters that characterize the geometry of finned tubes and their location in banks. The values of these parameters for most of the studied banks are shown in Table 2.
The Reynolds number, related to equivalent diameter changed within $4.8 \times 10^2$...$4.5 \times 10^4$ by changing air consumption and equivalent diameter of banks.

It can be noted that the values of coefficients $m$ and $C_q$ practically coincide with the values, obtained in study [11]. Only in two experiments, the value of coefficient $m$ is different by 1.9...3.2 $\%$, and in three experiments, coefficient $C_q$ is different by 9...12 $\%$.

Dependences of the Nusselt criterion on the geometry of tubes and banks are shown in Fig. 2, c.

Table 2

| No. of Tube series | $S_y$ mm | $S_x$ mm | $S_y/S_x$ | $H/F$ | $d_c$ mm | $N'$ | $C_q$ |
|--------------------|----------|----------|-----------|--------|-----------|------|-------|
| 1                  | 3        | 148      | 60        | 2.467  | 4.578     | 49.9 | 0.085 | 0.122 |
| 2                  | 3        | 148      | 85        | 1.741  | 4.578     | 49.9 | 0.098 | 0.199 |
| 3                  | 3        | 148      | 98        | 1.510  | 4.578     | 49.9 | 0.102 | 0.243 |
| 4                  | 3        | 148      | 148       | 1.000  | 4.578     | 49.9 | 0.120 | 0.447 |
| 5                  | 1        | 60       | 148       | 0.405  | 30.446    | 7.7  | 0.333 | 5.650 |
| 6                  | 1        | 60       | 98        | 0.612  | 30.446    | 7.7  | 0.281 | 3.237 |
| 7                  | 1        | 60       | 85        | 0.706  | 30.446    | 7.7  | 0.269 | 2.512 |
| 8                  | 1        | 60       | 60        | 1.000  | 30.446    | 7.7  | 0.236 | 1.611 |
| 9                  | 3        | 98       | 60        | 1.633  | 8.117     | 28.7 | 0.123 | 0.329 |
| 10                 | 3        | 98       | 74        | 1.324  | 8.117     | 28.7 | 0.138 | 0.417 |
| 11                 | 3        | 98       | 85        | 1.153  | 8.117     | 28.7 | 0.141 | 0.550 |
| 12                 | 3        | 98       | 98        | 1.000  | 8.117     | 28.7 | 0.148 | 0.661 |
| 13                 | 3        | 98       | 120       | 0.817  | 8.117     | 28.7 | 0.158 | 0.885 |
| 14                 | 3        | 98       | 148       | 0.662  | 8.117     | 28.7 | 0.172 | 1.175 |
| 15                 | 2        | 98       | 60        | 1.633  | 10.383    | 22.3 | 0.135 | 0.386 |
| 16                 | 2        | 98       | 74        | 1.324  | 10.383    | 22.3 | 0.144 | 0.531 |
| 17                 | 2        | 98       | 85        | 1.153  | 10.383    | 22.3 | 0.154 | 0.646 |
| 18                 | 2        | 98       | 98        | 1.000  | 10.383    | 22.3 | 0.162 | 0.785 |
| 19                 | 2        | 98       | 120       | 0.817  | 10.383    | 22.3 | 0.174 | 1.071 |
| 20                 | 2        | 98       | 148       | 0.662  | 10.383    | 22.3 | 0.193 | 1.491 |
| 21                 | 1        | 98       | 60        | 1.633  | 12.343    | 18.9 | 0.143 | 0.439 |
| 22                 | 1        | 98       | 74        | 1.324  | 12.343    | 18.9 | 0.154 | 0.562 |
| 23                 | 1        | 98       | 85        | 1.153  | 12.343    | 18.9 | 0.158 | 0.692 |
| 24                 | 1        | 98       | 98        | 1.000  | 12.343    | 18.9 | 0.170 | 0.851 |
| 25                 | 1        | 98       | 120       | 0.817  | 12.343    | 18.9 | 0.184 | 1.175 |
| 26                 | 1        | 98       | 148       | 0.662  | 12.343    | 18.9 | 0.200 | 1.537 |

Table 2 shows that the experiments were carried out in fairly wide domains of variability that determine geometric parameters of tube banks. Specified ranges overlap considerably the limits of variability of $S_y/S_x$ and $H/F$, most commonly applied in practice.

5. Results of research into heat exchange and aerodynamic resistance

5.1. Results of research into heat exchange

The main results of research are partially shown in Fig. 2. Fig. 2, a established dependence of the Nusselt criterion (Nu) on the Reynolds criterion (Re). For the examined banks, these dependences in logarithmic coordinates are linear with different slope angles of the straight lines, joining experimental points. It allows approximating them by exponential equation:

$$\text{Nu} = C_q \text{Re}^m$$

with variable coefficients $C_q$ and $m$. The values of these coefficients for each straight line of the whole array of empirical data are determined by the least squares method and presented in Table 1 and in Fig 2, b. Approximation error was ±(0.21...1.07) $\%$.

Results of experimental research into heat efficiency of finning are presented in Fig. 3 as a dependence of $E_{w_f}$ on the dimensionless height of fin $b_f/\delta$, where $b_f = \sqrt{\alpha / \lambda \delta}$. Fig. 3 also shows results of a similar study for the staggered banks.

Thermal efficiency of the fin decreases as the height of a fin and convective heat transfer coefficient increase. Effectiveness of a fin is not dependent on the type of the tube bank layout whether it is staggered or in-line.
5.2. Results of research into aerodynamic resistance

The main research results are presented in Fig. 4. Experimental dependences of specific numbers of Euler (referred to one transverse row of tubes) on the Reynolds numbers, calculated by equivalent diameter, are presented in Fig. 4, a. For each bank, relationship of Euler numbers \( E_{u0} \) and Reynolds \( Re \) in logarithmic coordinates is linear. This gives grounds to use exponential equation to generalize results of the experiment:

\[
E_{u0} = C_0 \cdot Re_s^m
\]  

(8)

with variable coefficients \( C_0 \) and \( m \), which depend on parameters, characterizing geometry of finned tubes and banks \( (H/F \text{ and } S_1/S_2) \). Values of coefficients \( C_0 \) and \( m \) are shown in Table 2, and their dependence on \( S_1/S_2 \) and \( H/F \) is shown in Fig. 4, b. The values of coefficients \( m \) and \( C_0 \) for each bank were determined by the least squares method. RMS approximation error of experiment results by equation (8) was ±(1.3...2.9) %.

Fig. 4. Results of research into aerodynamic resistance: \( a \) — dependences of specific values of Euler numbers \( (E_{u0}) \) on Reynolds numbers \( (Re_s) \);  
\( 1 \rightarrow 10 \) — numbers of tube banks according to Table 2, 11 — calculation by [9]; \( b \) — dependences of coefficients \( n \) and \( C_0 \) on ratio of pitches of tubes in bank \( (S_1/S_2) \) and reduced length of developed surface \( (H/F) \):  
\[ 1 - H/F = 30.446; 2 - H/F = 12.343; 3 - H/F = 10.538; 4 - H/F = 8.117; 5 - H/F = 4.578 \]

Fig. 5, a shows dependence of specific Euler number on the geometric characteristics of tubes and their banks (for comparison, Fig. 5, b shows the same dependence for the staggered banks, constructed according to materials of research [6]).

As Fig. 5 shows, these dependences vary considerably both in values of \( E_{u0} \), and in the character of their change when changing parameter \( S_1/S_2 \). For both types of bank, specific values of Euler numbers were defined at \( Re_s = 10^4 \).

6. Discussion of results of research into heat exchange and aerodynamic resistance

Based on results of conducted experimental research into heat exchange, we accepted the equation as original. Character of changes of coefficients \( m \) and \( C_0 \) (Fig. 2, b) gives grounds to perform subsequent generalization of results of research into heat exchange according to procedure [14]. Then a change of \( m \) can be described by a function of the following form:

\[
m = b_1 \cdot th \left[ a_1 \left( \frac{\sigma_1}{\sigma_0} - \frac{\sigma_2}{\sigma_0} \right) \right] + m_0.
\]  

(9)

Change \( C_0 \) by function:

\[
C_0 = -b_2 \cdot th \left[ a_2 \left( \frac{\sigma_1}{\sigma_0} - \frac{\sigma_2}{\sigma_0} \right) \right] + C_{00}.
\]  

(10)

In equations (9) and (10), \( (\sigma_1) \), \( m_0 \) and \( C_{00} \) are the coordinates of points of tangents of inflexion, determined in Fig. 2, b. Dependence \( (\sigma_1) \) on the finning coefficient with error ±0.074 % was approximated by function [14]:

\[
(\sigma_1) = \frac{\psi}{\tau} + 2.
\]  

(11)

Dependence \( m_0 \) on \( \psi \) was approximated with error ±0.188 % by formula:

\[
m_0 = 0.654 + 0.0089 \psi.
\]  

(12)

Coefficients of equations (9), (10) \( b_1 = 0.06 \), \( a_1 - a_2 = 2.5 \) were determined as a result of processing empirical data. Coefficients \( b_2 \) and coordinate \( C_{00} \) in equation (10) are variable. Dependence \( b_2 = f(\psi) \) was approximated with average error ±2.3 % by equation:

\[
b_2 = 0.321 \psi^{0.78}.
\]  

(13)

Approximation of parameters \( m_0 \) and \( b_2 \) was performed by the least squares method.

Processing of empirical data showed that parameters \( C_0 \) and \( b_2 \) change similarly, when \( \psi \) changes, and their ratio
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equals to 162. Taking into account presented results of the experiment, for calculation of coefficients \( m \) and \( C_s \) the following dependences are recommended:

\[
m = 0.6544 + 0.0646 \left( 2.5 \left( \frac{\psi}{7} + 2 - \sigma_2 \right) \right) + 0.0089 \psi, \tag{14}\]

\[
C_s = \left( 1.62 - 0.7h \left( 2.5 \left( \frac{\psi}{7} + 2 - \sigma_2 \right) \right) \right) - 0.321 \psi^{-0.38}. \tag{15}\]

Errors of calculation of \( m \) and \( C_s \) are determined by comparing the calculated and experimental values. Error of calculation \( m \) is \( \pm 1.036 \% \), error of calculation \( C_s \) is \( \pm 5.67 \% \).

Dependences of Nusselt criterion of geometric characteristics of banks of finned tubes for deep rows, presented in Fig. 2, are of extreme character. Maximal heat exchange intensity is within a domain of variability \( \sigma_2 = 2.7…3.5 \). In the domain of variability \( \sigma_2 = 3.5…5.5 \), heat exchange intensity remains virtually unchanged and rather high. Higher values correspond to lower values of degree of finning \( \psi \).

Extreme character of dependences, represented in Fig. 2, is caused by specific features of flow hydrodynamics of finned tubes in banks [14].

Based on results of the study, it is proposed to perform calculation of heat transfer of deep rows of the in-line banks of tubes with punched spiral finning for conditions, accepted as basic (\( \text{Re}_{\text{c}} = (6.3…38) \cdot 10^4 \), \( h_o/d = 0.4 \), \( h_o/h_{th} = 0.66 \) and \( \delta = 1.0 \text{ mm} \)), from equation (8) with determining of coefficients \( C_s \) and \( m \) from equations (14) and (15).

Petal width \( h_p = 4 \text{ mm} \) was accepted as basic for the following reasons: its increase worsens heat transfer, while its decreasing reduces hardness of structures and increases tubes’ tendency to vibration.

Impact of the number of transverse rows of tubes in the bank on heat exchange was evaluated by correction factor \( C_z \). As a result of experimental study of banks with a few rows, it was found that for single-row banks and banks with the number of transverse rows \( Z > 8 \), coefficient \( C_z = 1.0 \). For the banks \( Z < 2…8 \), it is recommended to determine coefficient \( C_z \) from formula

\[
C_z = 1.027 - 0.264 \frac{Z}{Z^2}. \tag{16}\]

derived from approximation of empirical values of \( C_z \) by the least squares method. Approximation error is \( \pm 0.41 \% \).

When geometric dimensions of finning are different from those, accepted as basic ones, we should introduce to equation (7) correction coefficients \( C_s, C_a \) and \( C_b \) determined from results of numerical analysis [16] from equations:

\[
C_s = 0.995 \left( \frac{h_p}{h_{th}} \right)^{0.311}; \tag{17}\]

\[
C_b = 0.925 - 0.125 h_b \left( \frac{h_p}{4.0} - 1.4 \right); \tag{18}\]

\[
C_a = 0.94 + 0.057 \left( \frac{\delta_i}{\delta_p} \right). \tag{19}\]

If it is necessary to apply proposed formulas for heat transfer calculation when tube banks are blown over by other gases, in particular combustion products, we should introduce the Prandtl number of these gases in power of 0.33 (\( \text{Pr}^{0.33} \)), as it is accepted in [14] and other papers. Then equation (7) will be written as:

\[
\text{Nu}_o = 1.13 C_s C_a C_b \frac{\text{Re}_{\text{c}}}{\text{Pr}^{0.33}}, \tag{20}\]

where coefficient 1.13 was acquired through dividing unity by the Prandtl number of air in power 0.33. The value of Prandtl number of air was accepted at temperature of 30 °C.

Comparison of results of calculation of heat transfer with the results of the experiment showed that average relative calculation error is 5.68 %. Results of experimental research into heat exchange were also compared with results of calculation with the use of technique [14] for banks of tubes with continuous spiral finning with a view to establishing the values of heat exchange intensification by punching of fins. Heat exchange intensification under the same conditions in in-line banks is \( 17.1…32.8 \% \) when \( \sigma_2 \) and \( \psi \) change within the limits of experiment and at \( \text{Re}_{\text{c}} = \text{const} \). Heat exchange intensification occurs due to periodic renewal of hydrodynamic and thermal boundary layers and a decrease in their thickness on each petal. At stall from sharp petal edges, there occurs a flow turbulence, which also intensifies heat exchange. A certain increase in the thermal efficiency of fins contributes to an increase in heat transfer.

Results of experimental determining of thermal effectiveness of fins were approximated by dependence, based on function of hyperbolic tangent by the least squares method:

\[
\psi = 0.8 - 0.176 \cdot h_b \left( 2(bh_b - 0.848) \right). \tag{21}\]

Mean square error of approximation is 2.25 %.

Processing of results of experimental research into aerodynamic resistance of in-line banks of tubes with punched spiral finning showed possibility to generalize them based on similarity equation (8) with variable coefficients \( n \) and \( C \).

Dependences, presented in Fig. 4, in logarithmic coordinates by straight lines, were approximated by the least squares method by power equations:

\[
n = 0.07 \left( H / F \right)^{0.35} \left( S_1 / S_2 \right)^{0.381}, \tag{22}\]

\[
C = 0.16 \left( H / F \right)^{0.02} \left( S_1 / S_2 \right)^{1.44}. \tag{23}\]

Mean relative error of approximation by equation (22) is 0.75 %, by equation (23) – 3.0 %.

For engineering calculations of aerodynamic resistance of multi-row in-line banks of tubes with punched spiral finning, equations (7), (22), and (23) are proposed. For banks with a few rows, it is necessary to introduce correction coefficient \( C_z \) into equation (8). In order to determine it, results of the experiment were approximated by the least squares method by power equation:

\[
C_z = 0.97 + 0.73 \frac{Z}{Z^2}. \tag{24}\]

Mean square error of approximation is 0.83 %.

Comparison of results of experimental study with calculation according to (7), (22), and (23) showed that average discrepancy is 6.78 %. Absolute experimental values \( \text{Eu} \) were compared with those, calculated by methods [9] and
Thermo-aerodynamic characteristics of various types of heating surfaces of boiler-utilizer according to gas turbines

| Element of boiler | Q, MW | $\alpha_1$, W/m°C | $E_{t1}$, W/m²°C | $k$, W/m°C | h, m²°C | Z, | $\Delta P$, Pa | E |
|------------------|------|-----------------|-----------------|--------------|---------|---|----------------|---|
| **Staggered banks of tubes with finned surfaces** | | | | | | | | |
| HPSS             | 27.06 | 84.76 | 0.730 | 64.08 | 39.54 | 77.56 | 6 | 318.2 | 268.0 |
| HPE              | 68.60 | 79.20 | 0.753 | 61.50 | 54.31 | 232.68 | 18 | 802.0 | 269.5 |
| LPSS             | 1.51 | 56.50 | 0.810 | 46.75 | 16.68 | 258.5 | 2 | 80.9 | 59.0 |
| Shell            | 184.47 | – | – | – | – | 92445 | 72 | 2597.4 | 223.8 |
| **Staggered banks of tubes with punched finning** | | | | | | | | |
| HPSS             | 27.06 | 120.42 | 0.746 | 90.95 | 54.84 | 5587 | 5 | 329.5 | 258.8 |
| HPE              | 68.60 | 112.20 | 0.767 | 87.71 | 75.92 | 16702 | 14 | 771.1 | 280.5 |
| LPSS             | 1.51 | 80.23 | 0.819 | 67.21 | 23.12 | 2092 | 2 | 80.9 | 52.5 |
| Shell            | 184.47 | – | – | – | – | 68009 | 50 | 2610 | 222.8 |
| **In-line banks of tubes with punched finning** | | | | | | | | |
| HPSS             | 27.06 | 86.23 | 0.811 | 71.63 | 44.04 | 6957 | 6 | 231.1 | 369.0 |
| HPE              | 68.60 | 92.79 | 0.808 | 76.84 | 65.31 | 19417 | 17 | 550.0 | 393.2 |
| LPSS             | 1.51 | 61.69 | 0.865 | 54.25 | 21.06 | 2379 | 2 | 52.8 | 91.2 |
| Shell            | 184.47 | – | – | – | – | 80640 | 70 | 1823 | 319.0 |
| **Staggered bare-tube banks** | | | | | | | | |
| HPSS             | 27.06 | 102.20 | – | 108.50 | 87.25 | 3512 | 23 | 465.2 | 183.4 |
| HPE              | 68.69 | 98.69 | – | 104.27 | 88.68 | 14307 | 94 | 1551.7 | 139.4 |
| LPSS             | 1.51 | 95.26 | – | 97.01 | 72.21 | 694 | 5 | 60.2 | 79.3 |
| Shell            | 184.47 | – | – | – | – | 56562 | 373 | 1407 | 143.0 |

Basic results of calculations for three functional elements of a boiler – high pressure steam super-heater (HPSS), high-pressure evaporator (HPE) and low-pressure steam super-heater (LPSS) – and the shell as a whole are shown in Table 3.

Note: heat transfer coefficients of gases $a_1=a_1+a_2$ are shown for bare-tube banks in box $a_1$. 

oundation of Nusselt criterion on ratio of tubes’ pitches in bank $S_1/S_2$ for the in-line bank was acquired from transformation of curve $2$ from Fig. 2, $c$.

It is interesting to compare thermo-aerodynamic efficiency of staggered and in-line banks of tubes at all other conditions being equal. Here, we performed this comparison by the evaluation of Kirpichov criterion, establishing the ratio between the heat amount, transmitted by a heat exchanger and the amount of energy it took to pump coolants through it:

$$E = \frac{Q [W]}{\Delta P [Pa]} \frac{[W]}{m^3/s}.$$ (25)

Comparison was performed only on the side of a gas coolant without taking into account efficiency of the fan. Results of calculations are represented in Fig. 7, which shows that in the experimental domain of variability $S_1/S_2$, the ratio of heat transfer and resistance values for in-line banks is achieved in a domain of variability $S_2/S_1=1.0...2.0$; for staggered banks, it is achieved in a domain of variability $S_1/S_2=2.0..2.5$.

**Fig. 6.** Dependences of heat transfer and specific aerodynamic resistance of in-line and staggered banks on parameter $S_1/S_2$: $a$ – in-line banks; $b$ – staggered banks

**Fig. 7.** Values of Kirpichov criterion for the in-line and staggered tube banks: $1$ – in-line banks; $2$ – staggered banks
Table 3 shows that in-line tube banks with punched finning have the highest thermo-aerodynamic efficiency. Results of the calculation study also showed that the higher thermo-aerodynamic efficiency of heating surfaces, the more intensive heat transfer inside tubes, and the higher the number of transverse rows of tubes $Z$, in the bank.

7. Conclusions

As a result of conducted study, we obtained new data on thermo-aerodynamic characteristics of transverse-streamlined tube banks with punched spiral finning. 1. Experimental study of heat exchange and aerodynamic resistance of in-line tube banks with punched spiral finning at the maximum permissible height of fin punching was conducted. It was found that heat exchange intensification relative to tube banks with continuous finning under conditions of the experiment is 17.1...32.8 %. Specific aerodynamic resistance increases by 18...40 %. The nature and degree of influence of geometric characteristics of finning and tube banks on heat exchange and aerodynamic resistance were determined. 2. Results of experimental study were generalized. A set of equations for engineering calculations of heat transfer and aerodynamic resistance of in-line tube banks with punched spiral finning was proposed. They include original power criterial equations with variable coefficients, establishing relationship between Nusselt and Euler criteria with Reynolds criterion. To calculate variable coefficients in heat exchange equation, equations based on hyperbolic tangent function are recommended. Parameters, characterizing the geometry of a finned tube – coefficient of finning and geometry of tube banks – relative longitudinal pitch of tubes, were accepted as input variables. We recommend power equations to calculate variable coefficients during determining aerodynamic resistance. Ratio of transverse pitch of tubes in a bank and longitudinal pitch of reduced length of developed surface are accepted as input variable parameters. Relative calculation error of heat exchange is 3.68 %, of specific aerodynamic resistance – 6.78 %.

3. We performed calculation study of thermo-aerodynamic efficiency of in-line and staggered tube banks with punched spiral finning using the results of experimental study. They assessed thermo-aerodynamic efficiency of heating surfaces with punched spiral finning of one shell of boiler-utilizer of power unit PGU-345. It was found that a Kirpichov criterion for the in-line pattern of tube banks is 312.0, for staggered pattern, it is 222.8.

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Protection of electrical plants of 6–750 kV from overvoltage has an important role in the operation of electric power facilities. Currently, the main way of protection of electrical equipment from overvoltage in electric networks of 6–750 kV is the application of nonlinear surge arresters (SA). That is why correct selection of SA in the course of designing is essential. Nowadays, majority of designing organizations select SA practically without taking into account the forms and duration of overvoltage action that can arise in the network, for which this selection is performed. This approach may lead to a damage of SA during operation due to the influence of overvoltage with large values of stored energy. Today, selection and application of SA are regulated by the following documents:

1. In Ukraine:
   - SOU-N II 40.12-00100227-47 "Non-linear overvoltage limiters of 110–750 kV voltage. Guidance on selection and application";
   - SOU-N MEV 40.10010227-67:2012 "Non-linear overvoltage limiters of 6–35 kV voltage. Guidance on selection and application in switchgear"

2. In Russia:
   - "Guidelines on application of overvoltage limiters in electric networks of 110–750 kV";
   - "Guidelines on application of non-linear overvoltage limiters in electric networks of 6–35 kV".

3. International standards:
   - IEC 60099-5 Suppressors for overvoltage protection. Part 5. Recommendations on selection and application.

4. Developments of companies – manufacturers of SA: