Подані результати експериментального дослідження теплообміну та аеродинамічного опору коридорних пучків труб з просіченим спіральним оребренням. Запропоновані емпіричні рівняння для інженерних розрахунків тепловіддачі та аеродинамічного опору коридорних пучків труб. Подані також результати розрахункового дослідження тепло-аеродинамічної ефективності чотирьох типів поверхонь нагріву котла-утилізатора. Встановлено перевагу коридорних пучків труб з просіченим спіральним оребренням з цього параметру

Ключові слова: інтенсифікація теплообміну, аеродинамічний опір, просічене спіральне оребрення

Представлены результаты экспериментального исследования теплообмена и аэродинамического сопротивления коридорных пучков труб с просеченным спиральным оребрением. Предложены эмпирические уравнения для инженерных расчетов теплоотдачи и аэродинамического сопротивления этих пучков труб. Представлены также результаты расчетного исследования тепло-аэродинамической эффективности четырех типов поверхностей нагрева котла-утилизатора. Установлено преимущество коридорных пучков труб с просеченным оребрением по этому параметру

Ключевые слова: интенсификация теплообмена, аэродинамическое сопротивление, просеченное спиральное оребрение

UDC 536.24:533.6.011
DOI: 10.15587/1729-4061.2017.108587

INVESTIGATION OF THERMO-AERODYNAMIC CHARACTERISTICS OF BANKS OF TUBES WITH PUNCHED SPIRAL FINNING

I. Galushchak
PhD, Senior Lecturer
Department of Theoretical, Experimental and General Physics
National Technical University "Kharkiv Polytechnic Institute"
Kyrpychova str., 2, Kharkiv, Ukraine, 61002
E-mail: irene.gal17@gmail.com

S. Gorbatenko
Chief Executive Officer
LTD Teckhno Logicheskie Energosisteme
Vezelska str., 109, Belgorod, Russia, 308015
E-mail: gsv267@gmal.com

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 in boilers of traditional steam-turbine plants and boilers-utilizers of GST without decreasing their thermo-aerodynamic efficiency.

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

("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.

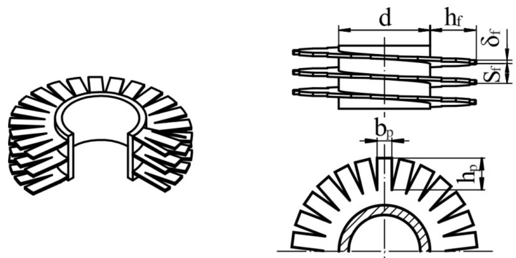


Fig. 1. Fragments of a tube with punched finning

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 17.6...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 carry out experimental research into heat exchange and aerodynamic resistance of in-line banks of tubes with specified finning type;

– to generalize experimental results;

– to conduct calculation studies of thermo-aerodynamic efficiency of staggered and 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:

$$\bar{\alpha} = \frac{Q_c}{H_c\left(\bar{t}_s - \bar{t}_a\right)}, \text{ W/m}^2\cdot{}^\circ\text{C}, \quad (1)$$

where Q_c is the heat release of the calorimeter, W; H_c is the total surface area of tubes-calorimeters, m^2; $\bar{t}_s$ 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; $\bar{t}_a$ is the average air temperature in the row of calorimeter mounting, °C.

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

$$\bar{t}_s = \frac{\sum t_i H_i}{\sum H_i}, \text{ °C}, \quad (2)$$

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:

$$\bar{t}_a = t'_a + \frac{\sum Q'_{rc} + 0{,}5 Q_{rc}}{GC_p}, \text{ °C}, \quad (3)$$

where t'_a is the air temperature at the inlet to the studied bank, °C; Q_{rc} is the heat release of the rows, where the calorimeter is mounted, W; $\sum Q'_{rc}$ 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:

$$Nu = \bar{\alpha} \cdot d / \lambda_a, \quad (4)$$

where λ_a 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.

$$\Delta P_b = \Delta P_{st} - \Delta P_{fc}. \quad (5)$$

The Euler numbers, referring to one transverse row of tubes, were calculated from obtained values ΔP_b:

$$Eu_0 = \frac{\Delta P_b}{Z_2 \rho U^2}, \quad (6)$$

where Z_2 is the number of transverse rows of tubes in banks; ρ is the air density at temperature before the bank, kg/m^3; 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 pneumo-metric method using a three-channel pneumo-probe. Differential pressure on the pneumo-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 D50141 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 bands 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 (ψ): S_f =5 mm (ψ=9.012) – series 1, S_f=6 mm (ψ=7.677) – series 2 and S_f =8 mm (ψ=6,010) – series 3. The other geometrical dimensions of tubes were the same: outer diameter of the tube d=28 m, fin height h_f =14.5 mm, petal height h_p =9.5 mm, petal width b_p =4.0 mm, fin thickness δ_f=1.0 mm. (Designations are shown in Fig. 1). Relative petal height h_p/h_f≈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 σ_2=S_2/d. Geometrical characteristics of some studied banks are shown in Table 1.

Table 1

Geometrical characteristics of studied tube banks

Number of location	σ_1	σ_2	Tube series					
			1		2		3	
			m	C_q	m	C_q	m	C_q
1	3.5	2.143	0.803	0.029	0.790	0.035	0.772	0.045
2	3.5	2.643	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	0.660	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	–	–	–	–	0.749	0.082

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

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

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_1/S_2) 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 $4.8\cdot10^3...4.5\cdot10^4$ by changing air consumption and equivalent diameter of banks.

Table 2

Geometric characteristics of in-line tube banks and empirical values of coefficients n and C_S

No. of bank	Tube series	S_1, mm	S_2, mm	S_1/S_2	H/F	d_e, mm	N	C_S
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	10.583	22.3	0.135	0.386
16	2	98	74	1.324	10.583	22.3	0.144	0.531
17	2	98	85	1.153	10.583	22.3	0.154	0.646
18	2	98	98	1.000	10.583	22.3	0.162	0.785
19	2	98	120	0.817	10.583	22.3	0.174	1.071
20	2	98	148	0.662	10.583	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_1/S_2 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

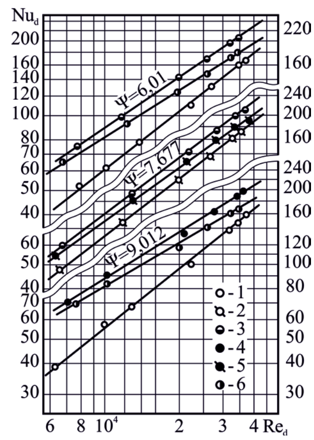
The main results of research are partially shown in Fig. 2. Fig. 2, *a* established dependence of the Nusselt criterion (Nu) on the Reynolds criterion (Re_d). For the examined banks, these dependences in logarithmic coordinates are linear with different slope angles of the straight lines, joining experimental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

$$\mathrm{Nu} = C_q \mathrm{Re}_d^m \tag{7}$$

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) %.

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*.



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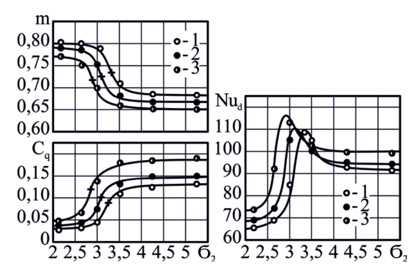


b *c*

Fig. 2. Results of research into heat exchange of the in-line banks of tubes with punched spiral-tape finning: *a* — dependences of the Nusselt numbers on the Reynolds numbers: 1—6 — numbers of tubes location in banks according to Table 1; *b* — dependences of coefficients m and C_q on relative longitudinal pitch (σ_2) and degree of finning of tubes (ψ); 1, 2, 3 are the tube rows; *c* — dependences of the Nusselt numbers on relative longitudinal pitch (σ_2) and degree of finning of tubes (ψ)

Results of experimental research into heat efficiency of finning are presented in Fig. 3 as a dependence of $E\psi_E$ on the dimensionless height of fin βh_f, where $\beta = \sqrt{2\alpha / \lambda_f \delta_f}$. Fig. 3 also shows results of a similar study for the staggered banks.

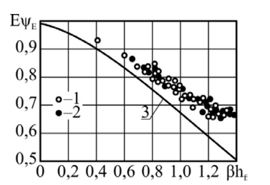


Fig. 3. Dependence of heat effectiveness coefficient of finned tube ($E\psi_E$) on the dimensionless height of fin (βh_f): 1 — staggered banks; 2 — in-line banks; 3 — calculation for continuous spiral finning by [14]

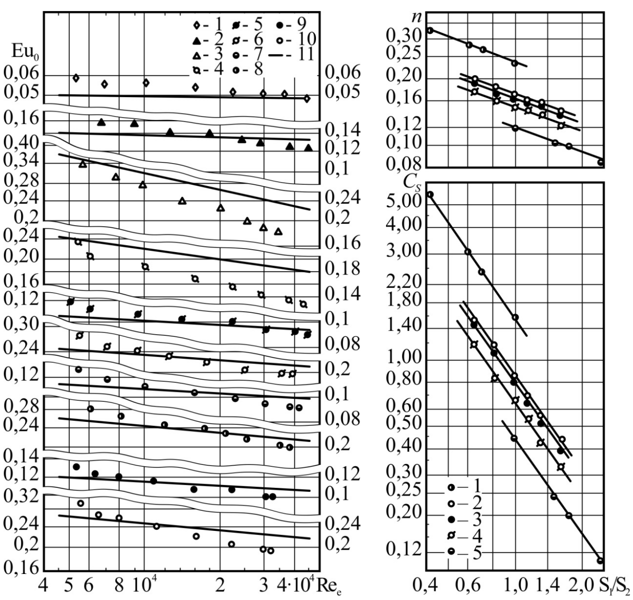
Thermal efficiency of the fin decreases as the height of a fin and convective heat transfer coefficient increase. Effectiveness of a fin is not dependent on the type of the tube bank layout whether it is staggered or in-line.

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 Eu_0 and Reynolds Re_e in logarithmic coordinates is linear. This gives grounds to use exponential equation to generalize results of the experiment:

$$Eu_0 = C_s \mathrm{Re}_e^{-n} \qquad (8)$$

with variable coefficients C_s and n, which depend on parameters, characterizing geometry of finned tubes and banks (H/F and S_1/S_2). Values of coefficients C_s and n 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 n and C_s for each bank were determined by the least squares method. RMS approximation error of experiment results by equation (8) was ±(1,3...2,9) %.



a *b*

Fig. 4. Results of research into aerodynamic resistance: *a* — dependences of specific values of Euler numbers (Eu_0) on Reynolds numbers (Re_e): 1—10 — numbers of tube banks according to Table 2, 11 — calculation by [9]; *b* — dependences of coefficients n and C_s 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=10.583; 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 Eu_0, 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 $\mathrm{Re}_e=10^4$.

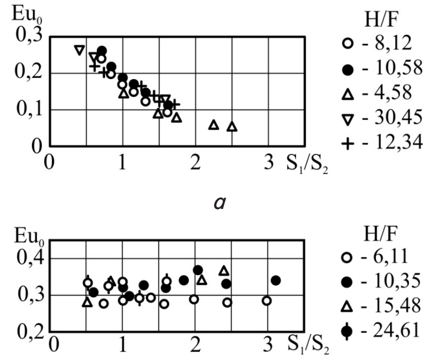


Fig. 5. Dependences of specific Euler numbers (Eu_0) on the ratios of pitches of tubes in banks (S_1/S_2) and reduced length of developed surface (H/F) for in-line and staggered banks of tubes with punched spiral finning: *a* — in-line banks; *b* — staggered banks

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_q (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:

$$m = b_1 th\{a_1[(\sigma_2)_0 - \sigma_2]\} + m_0. \qquad (9)$$

Change C_q – by function:

$$C_q = -b_2 th\{a_2[(\sigma_2)_0 - \sigma_2]\} + C_0. \qquad (10)$$

In equations (9) and (10), $(\sigma_2)_0$, m_0 and C_0 are the coordinates of points of tangensoid inflexion, determined in Fig. 2, *b*. Dependence $(\sigma_2)_0$ on the finning coefficient with error ±0.074 % was approximated by function [14]:

$$(\sigma_2)_0 = \frac{\psi}{7} + 2. \qquad (11)$$

Dependence m_0 on ψ was approximated with error ±0.188 % by formula:

$$m_0 = 0.654 + 0.0089\psi. \qquad (12)$$

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_0 in equation (10) are variable. Dependence $b_2=f(\psi)$ was approximated with average error ±2.3 % by equation:

$$b_2 = 0.321\psi^{-0{,}78}. \qquad (13)$$

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 ψ changes, and their ratio equals to 162. Taking into account presented results of the experiment, for calculation of coefficients m and C_q the following dependences are recommended:

$$m = 0,654 + 0,06th\left[2.5\left(\frac{\psi}{7} + 2 - \sigma_2\right)\right] + 0,0089\psi, \quad (14)$$

$$C_q = \left\{1,62 - th\left[2,5\left(\frac{\psi}{7} + 2 - \sigma_2\right)\right]\right\} \cdot 0,321\psi^{-0,78}. \quad (15)$$

Errors of calculation of m and C_q are determined by comparing the calculated and experimental values. Error of calculation m is ±1.036 %, error of calculation C_q is ±5.67 %.

Dependences of Nusselt criterion of geometric characteristics of banks of finned tubes for deep rows, presented in Fig. 2, *c*, are of extreme character. Maximal heat exchange intensity is within a domain of variability of parameter σ_2=2.7...3.5. In the domain of variability σ_2=3.5...5.5, heat exchange intensity remains virtually unchanged and rather high. Higher values correspond to lower values of degree of finning ψ.

Extreme character of dependences, represented in Fig. 2, *c*, 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 (Re_d=(6.3...38)·10^3, h_f/d=0.4, h_p/h_f=0.66 and δ_f=1.0 mm), from equation (8) with determining of coefficients C_q and m from equations (14) and (15).

Petal width b_p=4 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_2>8, coefficient C_Z=1.0. For the banks Z_2=2...8, it is recommended to determine coefficient C_Z from formula

$$C_Z = 1,027 - \frac{0,264}{Z_2}, \quad (16)$$

derived from approximation of empirical values of C_Z by the least squares method. Approximation error is ±0.41 %.

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

$$C_h = 0,995\left(h_p / h_p^{max}\right)^{0,321}; \quad (17)$$

$$C_b = 0,925 - 0,125th\left(\frac{b_p}{4,0} - 1,4\right); \quad (18)$$

$$C_\delta = 0,94 + 0,057\left(\delta_f / \delta_{fb}\right). \quad (19)$$

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 ($Pr_g^{0,33}$), as it is accepted in [14] and other papers. Then equation (7) will be written as:

$$Nu_d = 1,13 C_z C_h C_b C_\delta\ _q Re_d^m Pr_g^{0,33}, \quad (20)$$

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 σ_2 and ψ change within the limits of experiment and at Re_d=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 turbulization, 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:

$$E\psi_E = 0,8 - 0,176 \cdot th\left[2\left(\beta h_f - 0,848\right)\right]. \quad (21)$$

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

$$n = 0,07\left(H / F\right)^{0.356}\left(S_1 / S_2\right)^{-0,381}, \quad (22)$$

$$C_s = 0,16\left(H / F\right)^{0.676}\left(S_1 / S_2\right)^{-1,44}. \quad (23)$$

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 equation:

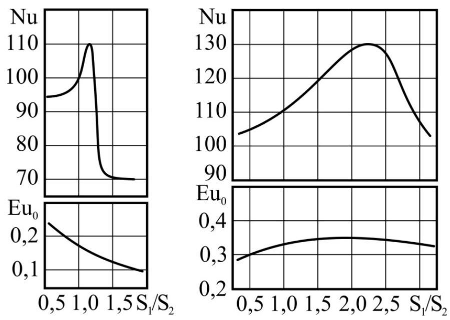
$$C_z = 0,97 + \frac{0,73}{Z_2^2}. \quad (24)$$

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 Eu_0 were compared with those, calculated by methods [9] and

[12] under identical conditions. In the first case, discrepancy was 2.3...10.6 %; in the second case, it was 10.2...23.4 %.

The results, presented here, of experimental research into heat transfer and aerodynamic resistance of in-line banks of tubes with punched spiral finning, as well of staggered banks [6, 8], allow establishing rational values of geometric characteristics of banks of finned tubes. For this purpose, Fig. 6 shows dependences $Nu=f(S_1/S_2)$ and $Eu_0=f_1(S_1/S_2)$ for the in-line and staggered pattern of banks. Nusselt numbers were determined at $Re_d=15\cdot10^3$ and $\psi=7.677$, Euler numbers were determined at $Re_e=10^4$ and H/F=10.358. Fig. 6 shows that the most advantageous ratio of heat transfer and resistance values for in-line banks is achieved in a domain of variability S_1/S_2=1.0...1.25; for staggered banks, it is achieved in a domain of variability S_1/S_2=2.0...2.5.



a *b*

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

Dependence 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:

$$E=\frac{Q[\mathrm{W}]}{\Delta P[\mathrm{Pa}]\cdot V\left[\mathrm{m}^3/\mathrm{s}\right]}. \qquad (25)$$

Comparison was performed only on the side of a gas coolant without taking into account efficiency of the fan. Results of calculations are presented in Fig. 7, which shows that in the experimental domain of variability S_1/S_2 of in-line banks (S_1/S_2=0.5...2.0) by thermo-aerodynamic efficiency, it is expedient to apply the in-line pattern of banks.

Heat transfer and aerodynamic resistance were defined at the same air velocity.

Thermo-aerodynamic efficiency of heating surfaces of double-shell boiler-utilizer for PGU-345 was determined. The shells were independently connected to two gas turbines. Calculations were performed for one shell. The boiler was made in a horizontal layout with vertical tube banks of heating surfaces. In the shell, heating surfaces of nine functional elements of the boiler were located in sequence.

The options of execution of heating surfaces were explored: staggered banks with continuous spiral finning, staggered banks with punched spiral finning, in-line banks with punched finning and staggered bare-tube banks. Banks of tubes with punched spiral finning were calculated according to procedure, described in this article and in [6, 8], with continuous finning according to [14], bare-tube – by standard methods of thermal and aerodynamic calculation of boiler units. Values of aerodynamic resistance, listed in Table 3 and accepted in calculation of a Kirpichov criterion, were calculated considering operational correction k_c=1.1 in accordance with regulations.

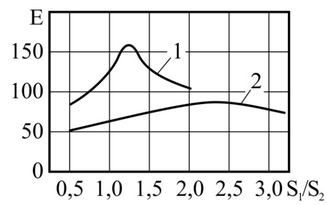


Fig. 7. Values of Kirpichov criterion for the in-line and staggered tube banks: 1 – in-line banks; 2 – staggered banks

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.

Table 3

Thermo-aerodynamic characteristics of various types of heating surfaces of boiler-utilizer according to gas turbines

Element of boiler	Compared parameters, heating surface type								
	Q, MW	$\bar{\alpha}$, W/m°C	$E\psi_E$	α_{1r}, W/m²°C	k, W/m²°C	H, m²	Z_2	ΔP, Pa	E
Staggered banks of tubes with continuous finning									
HPSS	27.06	84.76	0.730	64.08	39.54	7756	6	318.2	268.0
HPE	68.60	79.20	0.753	61.50	54.31	23268	18	802.0	269.5
LPSS	1.51	56.50	0.810	46.75	16.68	2585	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	771.7	280.5
LPSS	1.51	80.23	0.819	67.21	23.12	2092	2	90.9	52.5
Shell	184.47	–	–	–	–	68009	59	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	52.28	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	68.60	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	4067.5	143.0

Note: heat transfer coefficients from gases $\alpha_1=\alpha+\alpha_R$ are shown for bare-tube banks in box α_{1r}

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_2 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 5.68 %, 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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UDC 621.314
DOI: 10.15587/1729-4061.2017.108567

INFLUENCE OF ENERGY CHARACTERISTICS OF SURGE ARRESTERS ON THEIR SELECTION

S. Shevchenko
Doctor of Technical Sciences, Professor
Department of electricity transmission
National Technical University
"Kharkiv Polytechnic Institute"
Kyrpychova str., 2, Kharkiv, Ukraine, 61002
E-mail: syurik42@rambler.ru

S. Khlomko
Director
Merejaenergobud, Ltd.
Zroshuval'na str., 15, Kyiv, Ukraine, 02099
E-mail: s.khlomko@gmail.com

O. Berchuk
Project Manager
HITEK, Ltd
Vidradnyi ave., 95e, Kyiv, Ukraine, 03061
E-mail: beerch1977@gmail.com

Виконані розрахунки теплових режимів обмежувача перенапруг нелінійного. Встановлено, що при виборі енергетичних характеристик обмежувача перенапруг нелінійного необхідно користуватися його вольт-амперною характеристикою. Використання емпіричних формул з нормативних документів не завжди виправдано. Встановлено, що для отримання коректних значень енергії, яку поглинає обмежувач перенапруг нелінійний, необхідно використовувати воль-амперну характеристику
Ключові слова: обмежувач перенапруг нелінійний, вольт-амперна характеристика, теплові режими обмежувача перенапруг, енергія перенапруги

Выполнены расчеты тепловых режимов ограничителя перенапряжений нелинейного. Установлено, что при выборе энергетических характеристик ограничителя перенапряжений нелинейного необходимо пользоваться его вольт-амперной характеристикой. Использование эмпирических формул из нормативных документов не всегда оправдано. Установлено, что для получения корректных расчетных значений энергии, которую поглощает ограничитель перенапряжений нелинейный, необходимо использовать вольт-амперную характеристику
Ключевые слова: ограничитель перенапряжений нелинейный, вольт-амперная характеристика, тепловые режимы ограничителя перенапряжений, энергия перенапряжений

1. Introduction

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 40.100100227-67:2012 "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: