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INFLUENCE OF THE ANGLE OF INCLINATION OF ROUND-FINNED TUBES IN A STAGGERED TUBE BUNDLE ON THE FREE CONVECTIVE HEAT EXCHANGE BETWEEN IT AND AN UNBOUNDED AIR SPACE

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Results of experimental investigations of the average heat transfer from five-row staggered bundles of tubes with knurled spiral fins, operating in the free air-convection regime, in the case where the angel of inclination of the longitudinal axis of the tubes changed from 0 to 60^o *are presented. The investigations were carried out by the method of complete heat simulation. The average heat transfer was measured in the direction of a free air flow in each row of a tube bundle. The experimental data were generalized by similarity equations for calculating the average heat transfer from the tube bundles and their individual rows in the range of change in the Rayleigh number characteristic of the operating conditions of industrial heat exchangers assembled from finned tubes of the type of the above-indicated tubes.*

Keywords: *round-finned tube, free air convection, reduced heat-transfer coefficient, similarity equation, transverse row, heat simulation.*

Introduction. In the last decade of the 20th Century and at the beginning of the 21st Century the free convective heat transfer by an external air flow around a body has found wide use in different technologies and industries as well as in power plants and heat-loaded electronic devices. In a yearly cycle of use of gas-liquid heat exchangers operating in the free-convection regime, heat is removed completely or partially for a time depending on the climatic conditions of the environment. The operation of such apparatus in the regime of free convective heat exchange makes it possible to decrease the power consumed by them, is useful in the environment protection, and, in a number of cases, serves to increase the operate reliability of power plants, in particular, nuclear plants in the case of their emergency cooling.

The free convective heat exchange is an integral part of the work of apparatus for recovering the heat exhausted by fans, of heat exchangers of air-conditioning systems of industrial and entertainment buildings, of heaters of the drying agent of moist materials, of air coolers of the oil of large electric power transformers, of heat-exchange sections of the apparatus for air cooling of natural gas in compressor plants [1] of gas mains, and of air-cooling apparatus for condensation and cooling of technological products of the petroleum and chemical industries [2]. It should be noted that approximately 10,000 air-cooling apparatus are used in the fuel-energy complex of Russia and Belarus and that the drivers of the fans of air-cooling apparatus used in the gas industry [3] call for more than 1.5 mln kW for their work. A number of investigations [1, 4] show that the power consumption of air-cooling apparatus can be decreased by changeover of their work into the regime of free air convection with the use of air cooled to a definite temperature. This regime makes it possible to decrease the electric energy consumed by an air-cooling apparatus in the yearly cycle of its work, depending on the climatic conditions of the locality and the thermophysical characteristics of the product cooled, to 45% without additional expenses. To develop a system for control of the switching-off of the fans of an air-cooling apparatus in accordance with the temperature of the ambient air, it is necessary to know the characteristics of the convective component Q_c of the heat flow in the apparatus. Such data are also necessary for development of recuperative heat exchangers operating in the regime of free convective heat transfer.

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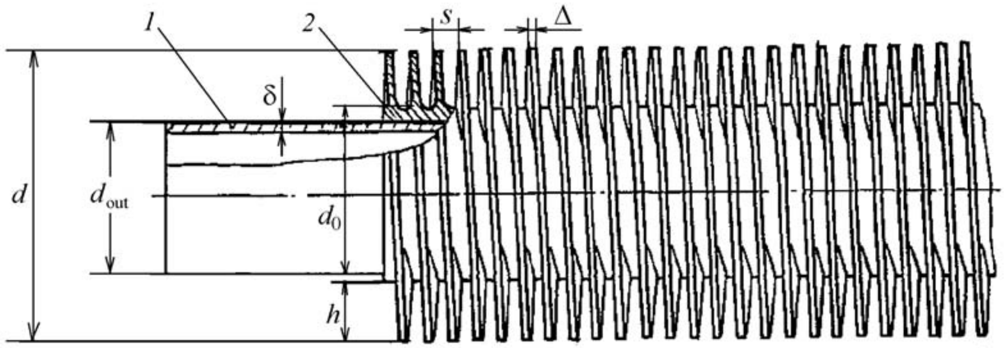

Fig. 1. Design of a bimetallic tube: 1, load-carrying tube; 2, finned shell of the tube with rolled spiral fins.

The heat flow from an air-cooling apparatus operating in the free-convection regime is determined as

$$Q = Q_c + Q_r . \tag{1}$$

By now reliable methods have been developed [5, 6] for calculating the radiative component of the heat transfer from a bundle of round-finned tubes exposed to an external air flow. However, the same cannot be said of the calculation of the convective component of the heat flow from such a bundle.

As follows from the reference book [7], the free convective heat transfer from bundles of round-finned tubes and even bundles of finned tubes [8] exposed to an external air flow was not investigated. We know that investigations in this direction were carried out only in [9] and in our works [10–13]; however, in these works the influence of the arrangement of the tubes in a tube bundle (the number of transverse rows, the orientation of the tubes in the space) on the intensity of the free convective heat transfer from it was not investigated to an extent required for the heat calculations.

The aim of the present work is to experimentally determine the average coefficients of convective heat transfer by a free air flow for each transverse row of tubes in a tube bundle and for the bundle as a whole for different angles of inclination of the tubes to the horizontal plane, to generalize the data obtain by similarity equations, and to determine correction coefficients estimating the influence of the arrangement of the tubes in a tube bundle on the heat transfer from it for the purpose of making up for the deficiency in this kind of investigations in the scientific and technical literature.

Object and Method of Investigation. We investigated staggered tube bundles consisting of five transverse rows of tubes. These bundles were assembled from commercial industrial bimetallic tubes (Fig. 1) with knurled spiral aluminum fins having the parameters $d \times d_0 \times h \times s \times \Delta$ = 55.6 × 26.5 × 14.55 × 2.91 × 0.75 mm. The load-carrying smooth tube with rolled fins was made from brass. The outer diameter of this tube d_{out} = 25 mm and the thickness of its wall δ = 2 mm. The finning coefficient $\varphi = F_t/F_0$ = 16.8. The length of the heat-eliminating part of the tubes in the tube bundles was equal to l = 300 mm. We used tubes of this standard size not accidentally because, first, such tubes are widely used in tube bundles of gas-liquid heat exchangers of different applications [14] and, second, we selected these sizes with regard for the parameters of the objects investigated in [10–13].

The tubes were placed at the corners of the equilateral triangle in the cascades of the bundles with a spacing $S_1 = S_2'$ = 64 mm. The spacing between the tubes in the longitudinal direction was S_2 = 55.4 mm. Each even transverse row contained six finned tubes heated by electric current, and each odd row contained five such tubes. Finned half-tubes were installed in the odd rows of tubes near the side walls of the bundles to provide the propagation of air through the cross sections of equal area along the height of a bundle.

The investigations were carried out by the method of complete heat simulation. A finned tube positioned at the center of each transverse row represented a calorimeter, with the use of which all the measurements were carried out. Thus, five calorimetric tubes were installed along the height of each bundle, which made it possible to simultaneously measure the initial parameters necessary for calculating the average heat transfer from each row of tubes and from the tube bundle as a whole in the steady-state heat state.

The reduced average coefficient of convective heat transfer from the ith transverse row of tubes of a tube bundle, operating in the regime of free air convection, is determined as

$$\alpha_{ci} = \frac{Q_{ci}}{F_i\,(t_{wi} - t_{em})}\,, \tag{2}$$

where $F_i = \pi d_0 \varphi l$ is the heat-eliminating area of the finning surface of the ith calorimeter. The reduced average coefficient of convective heat transfer from a tube bundle as a whole is equal to

$$\alpha_c = \frac{\sum_{i=1}^{i=5} Q_{ci}}{(t_w - t_{em}) \sum_{i=1}^{i=5} F_i}\,. \tag{3}$$

The average temperature of the calorimeter surface at the base of the fins is determined from the expression

$$t_w = \frac{\sum_{i=1}^{i=5} t_{wi}}{5}\,. \tag{4}$$

The adequacy of the values of α_{ci} and α_c depends on the reliability of calculation of the radiation component Q_{ri} of the heat flow from the calorimeter, which is determined mainly by the reliability of the reduced degree of blackness of the aluminum of the finned surface of the tube.

The data on the blackness of the sheet aluminum given in thermal-engineering reference books cannot be used in this case since the structure of the surface of the metal, its roughness, and the oxidation state [14] change in the process of knurling of aluminum fins. Because of this, we have developed and realized [15] a procedure for experimental determining the reduced degree of blackness of the fins on a tube for our experimental conditions, which was used for calculating the value of Q_{ri}.

The experimental setup, the design of the calorimetric tubes, the transducers and measuring devices with which these tubes were equipped, and the order of conduction of experiments were described in detail in [16].

The experimental results were processed and represented by the similarity numbers $\mathrm{Nu}_i = \alpha_{ci} d_0/\lambda$, $\mathrm{Nu} = \alpha_c d_0/\lambda$, $\mathrm{Ra}_i = \mathrm{Gr}_i \mathrm{Pr} = \beta g d_0^3 (t_{wi} - t_{em})/(\nu a)$, $\mathrm{Ra} = \mathrm{GrPr} = \beta g d_0^3 (t_w - t_{em})/(\nu a)$, $\mathrm{Gr}_i = \beta g d_0^3 (t_{wi} - t_{em})/(\nu^2)$, and $\mathrm{Gr} = \beta g d_0^3 (t_w - t_{em})/\nu^2$. The physical parameters of the air λ, ν, a, and β were determined by the temperature of the embient air t_{em}. In the experiments the temperatures of the walls of the tubes and of the embient air changed within the ranges t_{wi} = 35–228^{o}C and t_{em} = 15–25^{o}C. The relative root-mean-square error in the experimental values of α_{ci}, α_c, Nu, and Ra did not exceed 5.0, 4.9, 5.4, and 2.9% respectively. We investigated five tube bundles I–V differing from each other by the angle of inclination γ of the longitudinal axis of the tubes to the horizontal plane; the investigations were carried out for the angles 0^{o}, 15^{o}, 30^{o}, 45^{o}, and 60^{o}.

Results of Experiments. Figure 2 shows average values of the convective heat transfer Nu_i from individual transverse rows of finned tubes in the staggered tube bundles I–V, obtained experimentally for angles of inclination of the tubes γ from 0^{o} (the horizontal position) to 60^{o}. When Ra_i, defining the velocity of the lifting air flow, increased, the heat transfer from each row of tubes increased with one and the same rate independently of the angle of inclination of the tubes. For example, when Ra_i increased from $0.3 \cdot 10^5$ to $3 \cdot 10^5$, the rate of heat transfer increased, on the average, by a factor of 1.7. Common to all the tube bundles is that the heat transfer from the rows of tubes decreases progressively from the first row to the last (five) one with increase in the angle of inclination of the tubes. The heat transfer from the first row of tubes depends most substantially on the angle of their inclination; it decreased by a factor of 1.7 when γ changed from 0 to 60^{o}. This parameter exerts a minimum influence on the heat transfer from the last (fifth) row of tubes; the rate of heat transfer from this row of tubes decreased by a factor of 1.25 with increase in the angle of inclination of the tubes. An increase in the angle of inclination of the tubes in the transverse row of

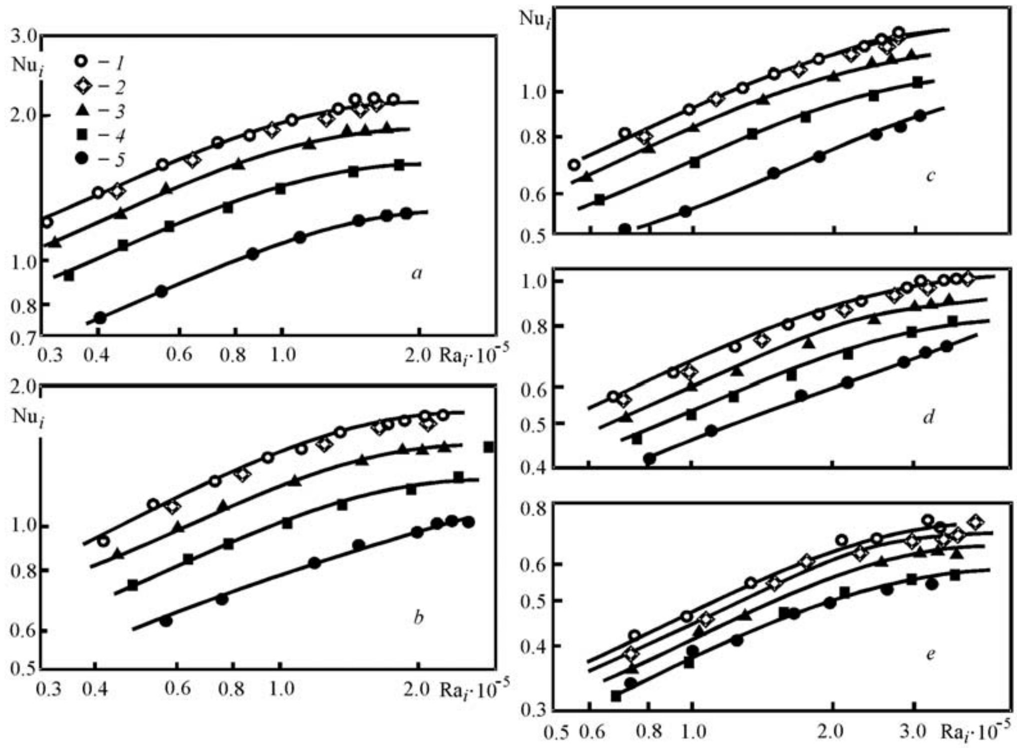

Fig. 2. Average convective heat transfer Nu$_i$ in the transverse rows of staggered tube bundles operating in the regime of free air convection: a–e) for rows 1–5 respectively; 1–5) for tube bundles I–V respectively; the curves) data calculated by Eq. (5).

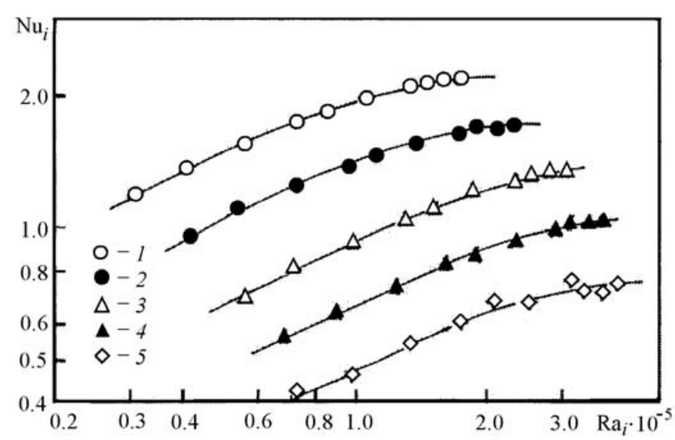

Fig. 3. Average convective heat transfer Nu$_i$ in the transverse rows of a horizontal staggered bundle of tubes: 1–5) for 1–5 rows respectively; curves) data calculated by Eq. (5).

a tube bundle γ from 0 to 15^{o} had no influence on the heat transfer from this row of tubes. The angle of inclination of the tubes in the fifth row analogously influenced the heat transfer from this row of tubes; however, this effect was observed at γ = 45 and 60^{o}.

The most favorable hydrodynamical conditions for the air flow in a bundle of tubes are realized in the case of a horizontal arrangement of tubes, where the rate of heat transfer is maximum. When γ increases in a bundle of inclined tubes, the faces of the fins disturb the lifting air flow and, coincidently, the aerodynamic drag of the tube bundle increases; in this case the velocity of the air in the interfin spaces decreases. The increase in the rate of the

TABLE 1. Values of Coefficients in Eq. (5) for Calculating the Average Convective Heat Transfer in the Transverse Rows of Staggered Inclined Tube Bundles

Number of a row	γ, deg	$A_i \cdot 10^3$	$B_i \cdot 10^{-5}$	n_i	Range of $\mathrm{Ra}_i \cdot 10^{-5}$
1	0; 15	10.40	2.9	0.46	0.3–2.0
	30	9.05	2.9	0.46	
	45	11.60	3.4	0.42	
	60	8.70	4.0	0.42	
2	0; 15	5.83	3.5	0.48	0.42–2.5
	30	5.00	3.5	0.48	
	45	8.00	4.7	0.42	
	60	17.2	∞	0.33	
3	0; 15	4.65	6.0	0.46	0.57–3.1
	30	4.15	6.0	0.46	
	45	4.50	6.8	0.44	
	60	5.65	∞	0.40	
4	0; 15	3.35	6.8	0.46	0.7–3.8
	30	3.00	6.8	0.46	
	45	3.70	8.7	0.43	
	60	5.70	∞	0.38	
5	0; 15	2.38	6.8	0.46	0.7–4.0
	30	2.27	6.8	0.46	
	45	2.10	6.8	0.46	
	60	1.90	6.4	0.46	

heat transfer, caused by the disturbing action of the inclined fins on the air flow, is smaller than the decrease in this rate caused by the decrease in the velocity of the air in the tube bundle. It has been established that the air flow in the interfin spaces of inclined-tube bundles is renewed with a low rate. Therefore, the heat exchange in the rows of inclined tubes in a tube bundle deteriorates as the angle between the tubes and the horizontal plane increases. The heat transfer from the transverse row of tubes remains unchanged when γ changes from 0 to 15° because the hydrodynamics of the flow changes insignificantly in this case.

Figure 3 illustrates the dynamics of change in the convective heat transfer from the horizontal tube bundle with $\gamma = 0^{\circ}$ in the direction from the first row of tubes to the last one. Measurements have shown that the average temperatures of the walls of the tubes in different rows of this bundle t_{wi} differ markedly under definite heat conditions. When a maximum electric power of 99 W was supplied to each tube of the indicated tube bundle at an air temperature of 24°C, the average temperature of the walls of the tubes at the base of the calorimeter fins changed in the direction of the air flow in the following way: it was equal to 119°C in the first row of tubes, 226°C in the fourth row of tubes, and 228°C in the fifth row of tubes. The dependences $\mathrm{Nu}_i = f(\mathrm{Ra}_i)$ are qualitatively similar for different rows of tubes; however, because of the different temperatures of the walls of the tubes in the different rows, the ranges of change in Ra_i are shifted to the right for each of the next transverse rows as compared to the first one. Analysis of Fig. 3 shows that, at Ra_i = const, the heat transfer from each next row is smaller by 35–40% than that from the previous row. A maximum heat transfer is characteristic of the first row of tubes in the indicated tube bundle. In this tube bundle, on the average, as compared to the first row of tubes with $\gamma = 0^{\circ}$, the heat transfer is smaller by a factor of 1.25 for the second row of tubes, by a factor of 2.15 for the third row, by a factor of 3 for the fourth row, and by a factor of 4.25 for the fifth row. The average convective heat transfer from a row of tubes depends mainly not on the velocity of the free air flow heated in the lower-lying rows but on its temperature.

The experimental values of Nu_i, presented in Figs. 2 and 3, are generalized with a deviation of $\pm 5\%$ from the approximation curve, by the similarity equation

$$\mathrm{Nu}_i = A_i \, \mathrm{Ra}_i^{n_i} \left[1 - \exp\left(-B_i/\mathrm{Ra}_i\right)\right] . \tag{5}$$

The values of the coefficients A_i, B_i, and n_i and the ranges of Ra_i in which Eq. (5) can be used are presented in Table 1. It should be noted that, at $\mathrm{Ra}_i < 1.5 \cdot 10^5$, the exponential term can be disregarded in the calculation of Nu_i by Eq. (5).

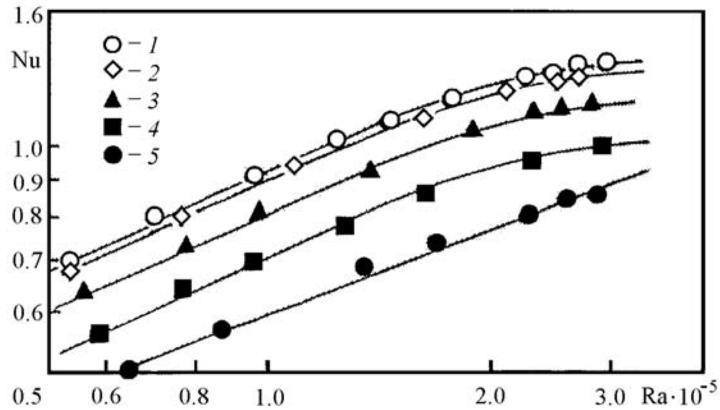

Fig. 4. Average convective heat transfer Nu of staggered five-row bundles operating in the regime of free air convection: 1–5) for tube bundles I–V respectively; curves) data calculated by Eq. (6).

TABLE 2. Values of Coefficients in Eq. (6) for Five-Row Tube Bundles

γ, deg	$A \cdot 10^3$	$B \cdot 10^{-5}$	n
0	5.82	5.8	0.44
15	5.68	5.8	0.44
30	5.15	5.8	0.44
45	4.48	6.0	0.44
60	10.60	∞	0.35

The experimental data obtained for the third and fourth rows of tubes with $\gamma = 60^{\circ}$ were generalized by purely exponential dependences obtained for $B_i \to \infty$.

The influence of the angle γ on the average convective heat transfer from the tube bundles I–V (Fig. 4) is qualitatively similar to the influence of this angle on the heat transfer from each row of tubes. An increase in the angle γ leads to a decrease in the rate of the heat transfer that, at $\gamma = 60^{\circ}$, comprises 67% of the heat transfer from the horizontal-tube bundle with $\gamma = 0^{\circ}$. The rates of heat transfer from the bundles of tubes with $\gamma = 0^{\circ}$ and $\gamma = 15^{\circ}$ differ by no more than 3%.

The average convective heat transfer from the five-row bundles of tubes I–V is generalized, with an error of ±5%, by the similarity equation

$$\mathrm{Nu} = A\,\mathrm{Ra}^n\,[1 - \exp(-B/\mathrm{Ra})]\,, \tag{6}$$

which is valid at $\mathrm{Ra} = (0.55\text{–}3.0)\cdot 10^5$. The values of the coefficients A, B, and n are presented in Table 2.

The data obtained for the average convective heat transfer from the tube bundles I–V allowed us to calculate the correction coefficient (Fig. 5), estimating the influence of the angle of inclination of the tubes in a tube bundle on the heat transfer from it, by the formula

$$C_\gamma = \mathrm{Nu}_\gamma / \mathrm{Nu}_\mathrm{I}\,, \tag{7}$$

where Nu_γ and Nu_I are the Nusselt numbers for the average free convective heat transfer from the bundles of tubes with $\gamma = 0^{\circ}$, 15°, 30°, 45°, and 60°, and the horizontal-tube bundle I with $\gamma = 0^{\circ}$, calculated by Eq. (6).

Consequently, in the case of free air motion around a staggered bundle of finned tubes, the average convective heat transfer from it at γ ranging from 0 to 60° and Ra changing in the range being investigated is calculated by the formula

$$\mathrm{Nu}_\gamma = C_\gamma \mathrm{Nu}_\mathrm{I}\,. \tag{8}$$

Earlier, the values of C_γ were determined for staggered bundles of such finned tubes with $\varphi = 16.8$ and a number of transverse rows $z = 2$, 3, 4 [3], and 6 [13]. Comparison of these values with the analogous values obtained

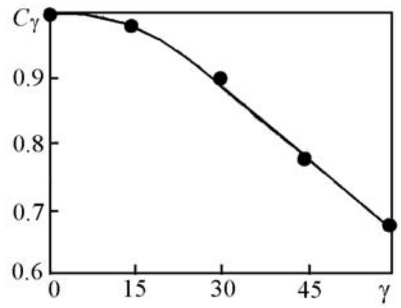

Fig. 5. Correction coefficient C_γ of the average heat transfer in the five-row tube bundles I–V: points) calculated values. γ, deg.

in the present work (Fig. 5) shows that the coefficient C_γ is practically independent of the number of transverse rows in a tube bundle. Therefore, in the case where the average heat transfer is calculated with an error not greater than ±7%, the data presented in Fig. 5 can be used for tube bundles with $z \le 8$.

In dense-packed staggered tube bundles with $58 \le S_1 \le 64$ the average temperatures of the walls of the tubes are very different even under one and the same heat conditions. Such difference between the temperatures of the tubes in a tube bundle makes the determination of the redistribution of the radiative heat flows over the rows of the bundle and the calculation of the real rates of the convective heat flows Q_c from individual rows very difficult. However, the sum of the convective heat flows from the tubes-calorimeters of a tube bundle is reliably determined because the error in calculating the radiative heat flow for one row of tubes is compensated by the error with an opposite sign for the other row. Since, under definite heat conditions, the values of Ra_i are very different, the average value of Nu_{cal} for a tube bundle, calculated by the dependences $\mathrm{Nu}_i = f(\mathrm{Ra}_i)$ obtained for individual rows as the arithmetic mean $\mathrm{Nu}_{cal} = \Sigma\,\mathrm{Nu}_i/z$, is larger than the value of Nu calculated by dependences (6) for the whole bundle. In dense-packed staggered bundles of finned tubes with $\varphi = 16.8$ the difference between the values of Nu_{cal} and Nu reaches 25% at $z = 4$, 20% at $z = 5$, and 13–15% at $z = 6$. Equations of the form $\mathrm{Nu}_{cal} = f(\mathrm{Ra}_i)$ are used for construction of theoretical models of the free convection heat transfer by air in bundles of finned tubes. Consequently, the installation of tubes-calorimeters in each transverse row of a tube bundle is a necessary condition for obtaining reliable experimental data on the average free convective heat transfer generalized by an equation of the form of (6) necessary for engineering calculations of heat exchangers.

Conclusions. Similarity equations have been dirived on the basis of the data of experimental investigations on the average free convective heat transfer from individual transverse rows of round-finned tubes in a tube bundle and from the whole tube bundle found in an unbounded air space. In the simulation of the average convective heat transfer from a dense-packed multirow staggered bundle of tubes it is necessary to introduce tubes-calorimeters in each transverse row since the determination of the average heat transfer from the tube bundle by the measured values of the heat transfer Nu_i from individual rows of tubes leads to its overestimation, as compared to the average value of Nu measured at one moment at Ra = const, by 13–25% for different numbers of transverse rows. An increase in the angle of inclination of the tubes in a staggered tube bundle from 0 to 60° leads to a decrease in the rate of the free convective heat transfer by 67% because of the deterioration of the hydrodynamics of the air flow in the interfin spaces of the tubes.

NOTATION

A, B, coefficients; a, thermal diffusivity of the air, $\mathrm{m^2/sec}$; C, correction coefficient; d, outer diameter of a fin, mm; d_0, diameter of the fin at its base, mm; d_{out}, outer diameter of the load-carrying tube, mm; F, heat-eliminating area of the finning surface, $\mathrm{m^2}$; F_0, heat-eliminating area of the bare tube of diameter d_0, $\mathrm{m^2}$; Gr, Grashof number; g, free fall acceleration, $\mathrm{m^2/sec}$; h, height of a fin; i, number of a transverse row in a tube bundle; l, length of the finned part of a tube, mm; Nu, Nusselt number; n, exponent; Pr, Prandtl number; Q, heat flow, W; Ra, Rayleigh number; S_1, transverse spacing between the tubes, mm; S_2, diagonal spacing between the tubes, mm; s, pitch of a fin, mm; t, temperature, $^\circ$C; z, number of transverse rows; α, heat-transfer coefficient, $\mathrm{W/(m^2 \cdot K)}$; β, coefficient of volume expansion of air, K^{-}; γ, angle of inclination of tubes, deg; Δ, average thickness of a fin, mm; δ, thickness of the wall of the load-carrying tube, mm; λ, heat conductivity coefficient of air, W/(m·K); ν, kinematic-viscosity coefficient of air, m^2/sec; φ, finning coefficient of a tube. Subscripts: c, convection; r, radiation; out, outer; em, embient air; cal, calculated; w, wall; t, tube.

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