Research of horizontal single-row bundles of ribbon finned tubes at free convection

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Abstract. The intensity of free convection research leads to a wider understanding of the mechanisms and possibilities of using free-convection currents for the heat discharge or heat removal in many industries, in energy processes, in a large number of devices or systems. Free convection significantly affects the maximum values of heat fluxes, and studying them becomes quite important for problems in which other methods of heat transfer are not applicable or uneconomical. The phenomena of free-convective heat transfer are quite complex, processes of conjugate heat transfer, for example, radiation and convection can be observed. Theoretical methods of description and calculation are used for bodies of simple shape: ball, plate, cylinder. The methods of theoretical analysis for heat transfer of bodies of complex shape are too voluminous and cumbersome. The intensification of free-convective heat transfer is possible through the use of various types of fins. To calculate the natural convective heat transfer of finned surfaces, the criterial dependencies of an empirical nature are mainly used. The article presents the results of a study of free-convective heat transfer of single-row horizontal bundles of industrial bimetallic pipes with tape finning, the finning factor of 16.8. The position of the pipes in the bundle is characterized by the by transverse steps standardized for heat exchangers for air cooling $S_1 = 58; 61; 64; 70; 76; 86; 100$ mm, to which the relative steps corresponded $\sigma_1 = S_1 / d = 1.208; 1.271; 1.333; 1.458; 1.583; 1.792; 2.083$. An equation for calculating the free-convection heat transfer, taking into account the influence of the relative step in the range of Rayleigh number from 30,000 to 350,000 with an average deviation of $\pm 6.6\%$, has been obtained.

1. Introduction

Free convection is of great importance in many industries due to the problems of heat removal in devices or systems. The safety of the equipment at a time when the usual cooling options are problematic to use and the heat, released by the unit, is removed by free convection depends on understanding the process. This applies to most electronic devices and systems, as well as power plants, such as air-cooling units, dry cooling towers and others.

The number of publications on free convection in the world is increasing, which leads to a broader understanding of both the mechanisms and the possibilities of using free convective flows.

Intensification of free convective heat transfer is achieved by the development of the heat exchange surface, in particular by finning on the air side. Spiral-tape, longitudinal, knurled, mounted, soldered and welded, wire finning types are used in the industry.

Natural convective heat transfer is a complex process that takes place on the finned heat exchange surfaces. Theoretical methods of description and calculation are used in elementary cases. The heat
transfer of bodies of complex shape is usually considered in three-dimensional space. The processes of mixed heat exchange by radiation and convection are often observed on finned surfaces. The methods of theoretical analysis in this case are too voluminous, complex and to some extent impracticable at the current development stage of mathematical modeling tools. For this reason, traditional criterial dependences of empirical origin should be used to calculate the natural convective heat transfer of finned surfaces [1-7].

2. Research plant and experiments
The scheme of the experimental unit for the study of heat transfer by the natural convection of air in bundles of finned tubes is given in [8, 9].

In the experiments, we applied industrial bimetallic finned tubes with spiral wound fins, manufactured by CJSC Oktyabrskhimnash and used in heat exchangers, tube bundles of which can vary from 1 to 6-8 rows. The study of a new pipe size begins with a single-row bundle. The diameter of the carrier tube \( d_t = 25 \) mm, wall thickness \( \delta = 2 \) mm, the geometrical finning parameters: the outer diameter of the fin \( d = 48 \) mm; diameter at the base of the fin \( d_0 = d - 2h = 25.8 \) mm; fin height \( h = 11.1 \) mm; fin step \( s = 3.125 \) mm; average fin thickness \( \Delta = 0.35 \) mm; and tube finning coefficient \( \varphi = 11.2 \). The total length of the bimetallic tube, including the end sections, was 440 mm, and the heat-giving length of the finned part was 400 mm. The remaining tubes in the test bundles were made from the same batch of finned tubes and had the same construction as the calorimeter. This is important because of the use of industrial tubes. The tube heaters were constructed for their electrical resistances to be practically equal, and it was assumed that all the tubes in the bundle give the same amount of heat.

The construction of the experimental finned tube calorimeter, on which the experimental measurements were carried out, is presented in figure 1.

![Figure 1. Finned tube.](image)

The research on the free-convective heat transfer to air on bundles of finned tubes was carried out under conditions of complete thermal modeling, which implies heating of all tubes. Measurements were made on a calorimeter installed in the center of the bundle row.

The time during which the installation reached the stationary thermal regime was, as a rule, 1 ... 1.5 hours after the last adjustment of the electrical power. The mode was considered steady-state with thermocouple readings unchanged for 10 minutes. To register the average temperature at the base of the edges of the tube calorimeter, fourteen chromel-copel thermocouples were laid along the tube generator with a shift of 30° relative to each other. Prior to the installation, thermocouples were necessarily tipped with an accuracy of 0.1°C. The experiment was carried out with an increase in
power supplied to the heat exchange tubes from one experimental point to another. The maximum electrical power supplied to the tubes was limited by the tube wall temperature of about 200°C.

The following values were recorded in the stationary thermal mode:

a) summed electrical power for tube calorimeter;

b) air temperature in diagonally opposite corners of the chamber;

c) EMF of thermocouples installed at the base of the edges of the tube calorimeters.

The average reduced heat transfer coefficient by free convection from the calorimeter surface, $W / (m^2 \cdot K)$, was found by the formula:

$$\alpha_c = \frac{Q_c}{F(t_w - t_0)}$$

where $Q_c$ is the heat convective flux, $W$; $F$ is the heat transfer surface of finned tube, $m^2$, $F = \pi d_0 \phi l$; $t_w$ is the temperature of the tube at the base of the fin, determined by the arithmetic mean value of the EMF of 14 thermocouples laid at the base of the edges of the calorimeter, °C; $t_0$ is the ambient temperature, taken as the arithmetic mean temperature from thermometer readings located in diagonally opposite corners of the chamber to thermometers, °C.

The heat flux convection, $W$, was defined as

$$Q_c = W - Q_r - Q_l$$

where $W$ is the total thermal power supplied to the heater, $W$; $Q_r$ is the heat flux by radiation, $W$; and $Q_l$ is the heat loss from the ends of tubes, $W$.

The radiant component of the heat flux must be considered, since it can be up to 30% [6] of convective component. The calculation of the heat flux by radiation from the calorimeter $Q_r$ was performed according to [9, 10].

The results of the evaluation of end heat losses for a stabilized thermal regime are approximated by a linear equation:

$$Q_l = 0.053 \cdot t_w + 0.35.$$  

### 3. Results and discussion

A study of natural convective heat transfer of seven experimental models of single-row bundles collected from bimetallic finned tubes with a finning coefficient was carried out. The position of the tubes in the bundle is characterized by transverse steps $S_1 = 58; 61; 64; 70; 76; 86; 100$ mm, to which the relative steps corresponded $\sigma_1 = S_1 / d = 1.208; 1.271; 1.333; 1.458; 1.583; 1.792; 2.083$. The experimental results were processed and presented in the numbers of similarity of Nusselt and Rayleigh

$$Nu = \frac{\alpha_c d_0}{\lambda},$$

$$Ra = Gr \cdot Pr = \frac{g \beta d_0^3 (t_w - t_0)}{\nu a},$$

where $\lambda$ is the thermal conductivity coefficient, $\nu$ is the kinematic viscosity coefficient, $\alpha$ is the thermal diffusivity coefficient and $\beta$ is the volume expansion coefficient.

The diameter of the pipe at the base of the fins $d_0$ is taken as a characteristic size for the convenience of practical calculations. The characteristic temperature in the experimental data processing for the average heat transfer of the bundle is the wall temperature $t_w$ for $\nu$, $a$, $\lambda$ and the ambient temperature $t_0$ for $\beta$.

Experimental data for each series of experiments on convective heat transfer with an error of ± 5% were approximated by the equation of the form: $Nu = A \cdot Ra^\beta$. The results of the experiments are presented in figure 2.
As it can be seen from the figure, the dimensionless heat transfer coefficient is within the limits $\text{Nu} = 1.26…4.36$ if the Rayleigh number changes $\text{Ra} = (0.47…3.5) \cdot 10^5$.

When comparing the heat transfer of single-row horizontal bundles with different steps (figure), it is seen to gradually decrease with increasing breakdown step. For example, a bundle with a 58 mm step has a 31% higher heat transfer rate compared to a 100 mm bundle. This is due to the hydrodynamic features of the flow around tight bundles, in which more intense traction occurs due to better heating of the air in the space between the tubes.

The values of the coefficients $A$ and $n$ in the range of application of the Rayleigh number $\text{Ra} = (0.47…3.5) \cdot 10^5$ are given in the table 1.

When processing the experimental data, the heat transfer was estimated by radiation, which was 14…38% from the convective heat flux.

The generalization of the obtained data allowed obtaining a formula for single-row bundles with a finning coefficient in the whole range of steps used in industry in the form:

$$\text{Nu}=1.57 \cdot 10^{-2} \cdot \sigma^{0.53} \cdot \text{Ra}^{0.45},$$

which with a mean deviation of ±6.6% is valid for $\text{Ra}=(0.47…3.5) \cdot 10^5$.

### 4. Conclusions

As a result of the experiments and calculations, an equation for single-row horizontal bundles of bimetallic finned tubes with ribbon fins, produced by CJSC Oktyabrskhimmash, has been obtained.

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**Table 1.** $A$ and $n$ coefficients for single-row tube bundles $\varphi=11.2$

| $S_1$, mm | 58       | 61       | 64       | 70       | 76       | 86       | 100      |
|-----------|----------|----------|----------|----------|----------|----------|----------|
| $A$       | 0.0067   | 0.0061   | 0.009    | 0.0081   | 0.0092   | 0.0051   | 0.0147   |
| $n$       | 0.525    | 0.5306   | 0.4917   | 0.4912   | 0.4747   | 0.5178   | 0.4214   |

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**Figure 2.** Heat transfer of single-row horizontal bundles
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