Calculation and experimental analysis of heat transfer of assemblies of ribbed heat exchange elements

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Abstract. The results of thermal-hydraulic and strength tests of finned tube assemblies welded from KP 20 elements of four sizes are presented. The elements are representing by steel plate 0.4 mm in thickness, covered with a copper film 0.025 mm in thickness. One or two conical necks 17 mm long are extended in the plates. A set of placed in each other cones form a welded finned tube 23/20 mm with a toothed inner surface. As a result of the tests: high strength characteristics of these assemblies were demonstrated (internal pressure of burst under normal conditions is 40 MPa, resistance to sudden changes in temperature and freezing of water in “pipes”); a high average coefficient of thermal conductivity in the thickness of the ribs was confirmed at the level of 75 W/(m×K); the increase in the intensity of the heat transfer process is 2.15 times with the help of technological protrusions on the inner surface of the “pipe” to a turbulent single-phase flow in comparison with the calculated values for “smooth” pipes with a moderate increase in hydraulic resistance; the method of heat-hydraulic calculation of heat exchangers consisting of such “pipes” is proposed. The method is based on the ratios set out in the regulatory document “RD 24.035.05-89”. Thermal and hydraulic calculation of NPP heat exchanging equipment”, with amendments considering the high degree of finning of the tested “pipes” and the asymmetry of the edges of the ribs relative to the axis of the “pipes” bearing pressure, as well as the change in the value of the correction for the smallness of the tube bundle — we also identified areas of effective use of assemblies tested sizes of elements KP 20.

1. Introduction

In surface heat exchangers, with a significant difference in the heat transfer coefficients (α) of the heat exchanging media, fins are used on the side of the medium having a lower α value. Usually, easily accomplished external ribbing of tubular elements is used with the direction of the ribs along the flow of the medium. The greater the ratio of α values for heat exchanging media, the higher the degree of ribbing is required while maintaining a high coefficient of heat transfer efficiency along the height of the rib. An increase in the thermal conductivity of the material of the ribs, an increase in their thickness and a decrease in height have a positive effect on heat transfer.
Currently, the industry produces several variants of finned tubes. They contain a pipe carrying internal pressure and ribs connected to the outer surface of the pipe by different methods with different values of thermal contact resistance. There is a struggle to reduce this resistance.

For example, an original technology for manufacturing ribbed assemblies from KP 20 elements is known [1]. These elements represent a steel (St08KP) plate with a thickness of 0.4 mm, covered with a copper film with a thickness of 0.025 mm. One or two 17 mm long tapered necks are drawn in the plates by stamping. A set of nested cones after a series of technological operations forms a welded finned tube 23/20 mm in diameter with a toothed inner surface. A description of the design is given in [2]. Hydraulic rupture tests of three two-pipe assemblies, each containing 109 rectangular KP 20 elements (Figure 1), led to rupture of the assemblies at pressures of 39.4 MPa, 40.3 MPa and 40.1 MPa. These figures are equivalent to those for "monolithic" steel pipes of the same diameter and wall thickness.

![Diagram of the KP 20 two-pipe element, standard size 1.](image)

The considered finned assemblies have a number of advantages over the known prototypes:

— lack of contact thermal resistance between the ribs and the pressure-bearing pipe;
— technological protrusions on the inner surface of the "pipe" intensify heat transfer to this surface;
— the creation of a surface layer of copper on the ribs increases their average thermal conductivity in thickness and corrosion resistance.

To determine the resistance of the adhesion of elements to abrupt changes in water temperature, thermal cycling tests were carried out on the assembly described above: 301 change cycles over a period of 20–30 s in the temperature of the water flow supplied to the sample by 140–150°C at the upper range of these temperatures 170–180°C FROM. Each cycle was accompanied by a difference in the temperature of the water at the outlet from the upper and lower "pipes" of the assembly up to 50°C and a difference in the temperature of water and metal of the fin up to 120°C. The assembly retained its shape, dimensions and tightness both during testing and during subsequent pressure testing at a pressure of 10 MPa. Also, a cycle of 41 freezing and thawing of water was carried out in an assembly of 44 single-tube ring elements of KP 20. The assembly also retained its tightness during subsequent pressure testing at a pressure of 10 MPa.

The purpose of the thermohydraulic tests described below is to create a procedure for the thermohydraulic calculation of heat exchangers from KP 20. Particular attention is paid to the compliance of the test results with the calculations carried out according to the normative
recommendations [3–5] for heat transfer and hydraulic resistance of a single-phase flow (water, gas) as inside a "toothed tube", and outside with a transverse air flow around the finned tubes. The possibility of adjusting the regulatory recommendations was also considered.

The table shows the characteristics of the tested assemblies of KP 20 elements of four standard sizes.

### Table 1. Geometrical characteristics of assemblies from KP 20 elements

| Standard size | 1     | 2     | 3     | 4     |
|---------------|-------|-------|-------|-------|
| KP 20 element shape | rectangular | octagonal | round | rectangular |
| External dimensions of the element, mm | 130×90 | 128×56 | D56  | 130×90 |
| Number of "pipes" | 2      | 2      | 1      | 2      |
| Linear outer surface of heat transfer, m²/m | 3.55   | 1.69   | 0.71   | 6.38   |
| Linear weight, kg / m | 9.23   | 6.51   | 2.3    | 17.1   |
| Finning ratio | 24.57  | 11.7   | 9.83   | 44.14  |
| Finning step, mm | 6.5    | 6.5    | 6.5    | 3.5    |
| $w$, m/s | 0.46÷0.57 | 3.2÷19.1 | 5÷20  | 2.6÷5.8 |
| $Re_{vozd} = Re_{2} = \frac{wd_{th}}{\mu}$, $10^{-3}$ | 0.58÷1.63 | 4.4÷28  | 8.2÷33.3 | 3.5÷12.5 |
| $Re_{vozd} = Re_{2} = \frac{WL}{\mu}$, $10^{-3}$ | at $L=d_{th}$ | at $L=l_{0}$ | at $L=l_{0}$ | at $L=d_{th}$ |
| | 0.215÷0.76 | 8.7÷55.6 | 15.4÷43.7 | 0.84÷2.15 |

2. Installation for thermohydraulic tests

The stand (see Figure 2) contained a box 1, inside which was placed the investigated assembly 2 of KP 20 elements of the same standard size. There were no gaps between the walls of the box and the ribs. One assembly of two-pipe elements KP 20 or two assemblies with a clearance of 5 mm in height from one or two-pipe elements were placed in the box in height. The arrangement of the assemblies is corridor.

![Figure 2](image-url)

**Figure 2.** Stand layout: 1 - air box, 2 - tested assembly; 3 and 5 - flowmeter orifices; 4 - electric heater.
From below, cooling air was supplied to the box through a diffuser with an opening angle of 20°. From above, the box was connected to the atmosphere. The cooled medium was hot non-boiling water of high pressure ($P_1 \leq 1.8$ MPa, abs) supplied to the "pipes" of the assembly under study. Part of the experiments with assembling from elements of the first standard size was carried out with natural air draft with the air diffuser dismantled. The height of the box above the assembly was 0.25 m or 0.51 m. The direction of the water relative to the "teeth" was made "along the wool"; smooth narrowing of the section in the "teeth" and its sharp expansion when changing the inner diameter of the "pipe" in the range of 20/21 mm.

The flow rates of water $G_1$ and air $G_2$ were determined based on the measured pressure drops across diaphragms 5 and 3, as well as the measured pressures $P_1$ and $P_2$ and the temperatures of the medium $T_1$ and $T_G$ at these diaphragms. When the water temperature at the outlet of the assembly is below 70°C, the main measurement of its flow rate was carried out according to the rate of filling the measuring beaker with water.

For two-pipe assemblies, water was supplied in parallel to both pipes. The water temperature was measured at the outlet of the upper $T_{1u}$ and lower $T_{1n}$ pipes, and also its average value was measured after mixing the flows $T_{1a}$. For one-pipe assemblies, only the $T_{1a}$ value was measured. The distribution of air temperatures along the longitudinal axis of the duct at the entrance to the assembly $T_2$ and at the exit from it $T_{3a}$ was measured, respectively, using three and eight sensors. The air pressure drop $\Delta P_2$ was also measured when it passed through the assembly.

The pressures and pressure drops were measured by electronic pressure gauges and differential pressure gauges of the "Metran 100" type, having a class of 0.075 and 0.1, respectively. Taking into account the errors introduced by the elements included in the automatic data collection system, the rms relative error of these measurements is estimated at 0.245%.

The temperature of the media was measured with cable thermocouples of the KTMS KhK type, the hot junctions of which were introduced directly into the flow of the medium. Considering additional calibrations, the maximum absolute error in measuring the temperatures of the media, as well as the difference between their input and output values, was estimated at 0.5°C. The root-mean-square relative errors in determining the flow rates of water and air were 2.02% and 2.5%, respectively, and the thermal power of the assembly $N$, calculated from the balance ratio for both media, was 5.4%. The imbalance between the $N$ values for both media did not exceed 7%.

3. The results of thermohydraulic tests

The table shows the range of variation of air flow rates and Reynolds numbers $Re$. The relations for determining the characteristic size $l_0$ are given below.

To determine the degree of intensification of heat transfer from the water flow to the inner surface of the "pipe" with technological toothed protrusions, a series of experiments was carried out in the presence of boiling water at atmospheric pressure in the air box 1 and when pumping hot water $T_1 \leq 188$°C through the "pipes" at $P_1 \leq 1.8$ MPa. In a series of these experiments, the water temperature was at least 10°C below the saturation temperature.

The high intensity of heat transfer during boiling of water and low thermal resistance of the wall of the "pipe" made it possible with an error of no more than 11% to determine the coefficient of heat transfer to the inner surface of the "pipe" own. In the investigated range $Re = (7–33) \cdot 10^3$, it exceeded by $(2.15 \pm 0.1)$ times the value calculated from the known relation for a turbulent flow in a pipe $\alpha_{tr} = 0.023Re^{0.8} Pr^{0.4}/d$ [3–5] (see Figure 3). When compared with the available design recommendations for bumpy surfaces [5, 6], intensifiers in the form of transverse annular corrugations of a smooth configuration, located with a certain pitch, were the closest in design. The calculation according to the ratio proposed in [6] in relation to the geometry of the considered in-line channel gives a close value of the degree of intensification $\alpha_{tn} / \alpha_{tr}$=1.9.
Figure 3. Dependences $\alpha_{in} / \alpha_{tr} = f (Re)$ for heat transfer (♦) and $\lambda_{tr} = f (Re)$ for friction resistance (■) inside the "pipe" of KP 20. Lines: $\alpha_{in} / \alpha_{tr} = 2.15$ and $\lambda_{tr} = 0.0828$.

With the same direction of water flow, in separately conducted experiments with a single-pipe assembly, the value of the coefficient of hydraulic resistance by friction $\lambda_{tr}$ of the flow when passing through the "pipe" was determined. The results are also shown in Figure 3. They showed an early achievement of self-similarity (independence of the Re value) of $\lambda_{tr} = 0.0828 \pm 0.008$, which is only 2.11–2.84 times higher than the values calculated by the ratio $\lambda_{tr} = 0.11 \left( (\Delta / d) + (68 / Re) \right)^{0.25}$ [4] for technical pipes ($\Delta$ is the average absolute roughness of the technical pipe surface) For this copper-plated pipe, $\Delta = 10^{-2}$ mm was taken. Calculations according to the recommendations [6] for the above intensifier from transverse corrugations gave the ratio $\xi_{in}/\xi_{tr} = 2.51$. Consequently, internal asymmetric and one-sided abrupt changes in the "pipe" cross-section caused by the peculiarities of manufacturing assemblies with KP 20 elements are equivalent in the degree of heat transfer intensification and increase in hydraulic resistance to the special intensifying knurling proposed in [6] for compact heat exchangers.

The intensification of internal heat transfer is probably only relevant for recuperators of air before it is fed into the combustion chamber of gas turbines. For them, the intensification of internal heat transfer in assemblies of KP 20 elements reduces the energy consumption for pumping gas through the heat exchanger by a factor of 2.1–5.0 in comparison with the "smooth" pipe to obtain identical values of $\alpha_{in}$.

In the main experiments with the cooling of assemblies by the air flow, 85–88% of the total thermal resistance between the heat exchanging media was occupied by the thermal resistance of heat transfer to air from the outer surface of the "pipe" and ribs at the minimum value of the temperature difference between the media $\Delta t_{av} = 121^\circ C$. Therefore, the root-mean-square relative error in determining the average experimental value of $\alpha$, referred to the total outer surface of pipes and fins $\alpha_{ex}$ by the "heat exchanger method", did not exceed 10%. The values $\alpha$ were compared with the values $\alpha_{up}$ calculated according to the recommendations [3, 4] for a rib of constant thickness along its height:

$$\alpha_{up} = ( F_{tr} / F_{ex} + E \varphi_{E} F_{p} F_{z} ) \alpha_{in}$$  \hspace{1cm} (1)

at $F_{tr} / F_{ex} = \left[ 1 - (\delta / S)_{p} \right] / \varphi = F_{tr,p} \left[ 1 - (\delta / S)_{p} \right]$, $F_{p} / F_{ex} = (1 - F_{tr} / F_{ex})$;

$E = \text{th}(m_{h} \rho p / m_{h} \rho)$ — the coefficient of efficiency of the rib, calculated under the assumption that the values $\alpha = \alpha_{in}$ remain constant for the entire surface of the rib; $m = [2 \alpha_{in} (\delta / \lambda_{p})]^{0.5}$;

$\varphi_{E} = 1 - 0.058 m_{h}$ — coefficient taking into account the uneven heat transfer along the surface of the rib;
where \( C_x \) is a coefficient that depends on the pitch of the pipes in the bundle and the ribbing coefficient; \( C_z \) - correction for the number of rows of pipes \( z \) along the gas flow. At \( z \geq z_{avt} \), \( z_{avt} = 8-10 \) [3] or \( z_{avt} = 4 \) [3], the correction is \( C_z = 1 \). At \( z < z_{avt} \), according to the recommendation [3], \( C_z \) grows with an increase in the number of pipe rows. The same character of \( C_z \) change is described in the recommendations [4, 5] for non-ribbed transversely flushed pipes. At the same time, for finned tubes in [4], the dependence \( C_z = f(z) \) has the opposite character. For most of the tested samples, \( z = 2 \), \( C_z = 0.76 \) and \( C_z = 1.1 \) according to the recommendations [3] and [4], respectively, i.e., there is a difference of 1.45 times, \( L \) - characteristic size, m. In [3] \( L = d_r \); in [4] \( L = l_0 = f(d_r, D_p; \varphi) \), where \( d_r \) and \( D_p \) are, respectively, the outer diameters of the pipe and rib, m; \( \varphi = F_{tr} / F_{tr,p} \) - ribbing coefficient - the ratio of the total outer surface of a finned and non-ribbed pipe. The same values of \( L \) are used when calculating \( \text{Re} = wL/v_a \), \( w \) is the air speed in the minimum flow area, m/s; \( \lambda_a \) and \( v_a \) - coefficients of thermal conductivity and kinematic viscosity of air at its average temperature in the gap, \( W/(m\cdot K) \) and \( m^2/s \) respectively.

\( P_t \) is the Prandtl number at the same temperature. Recommendations for choosing the values of the multiplier \( A \) are given in [2].

The value \( n = f(\varphi) \) for the tested standard size of KP 20 elements is, according to [3], \( n = 0.830 \); and according to [4] \( n = 0.763 \). In [3] there is no indication of the limits of application of relation (1). In [4], the limits are limited by the values \( \text{Re} = (1-37) \times 10^4 \); \( v_a = 1-18.5 \); \( l_0 = (0.027-0.178) \) m. This range does not include experiments with assembling the first and fourth standard sizes.

4. Discussion of test results

The values of \( \alpha_{crx} \) was compared with \( \alpha_{pr} \) calculated according to [3, 4]. The comparison also involved the results of comparative experiments on assemblies of pipes with annular ribs of several domestic manufacturers, carried out under the same conditions as experiments with assemblies from KP 20 elements of the third standard size: two pipes with annular ribs installed one above the other with a gap of 5 mm. The comparison shows the following:

1. Recommendations [3] more correctly reflect the influence of the number of pipe rows along the gas flow. So, the difference in the values \( \alpha_{crx} = f(\text{Re}) \) when testing one and two assemblies installed one above the other from elements of the second standard size (4 "pipes" installed one above the other) corresponded to the change in the value of \( C_z \) according to [3].

2. For all tested assemblies of KP 20 elements \( \varphi = 9.83-44.14 \) as well as for samples of finned tubes of a number of Russian firms \( \varphi = 4.87-8.10 \) with aluminum fins or stainless steel fins welded or crimped to the main pipe, there is a correspondence between the values \( \text{Ex} \) and \( \alpha_{pr} \) when calculating \( \alpha_m \) according to the recommendations of [4], but assuming the value \( C_z = 0.83-0.96 \), which is close to the recommended in [3] \( C_z = 0.9 \) for the tested two-row samples. Therefore, for all tested assemblies, it is recommended to calculate the value of \( \alpha_m \) according to relations (1) and (2), using the recommendations [4], but to calculate the coefficient considering the smallness of the beam, use the recommendations [3]. At the same time, the upper limit of the range of these using recommendations expands to \( \varphi = 44.14 \). For all compared experiments, the range of values \( \text{Re} = w/d_r \) / \( v_a = (15-60) \times 10^3 \) was considered. This corresponded to the air velocity for the assembly of KP 20 elements \( w = (5-20) \) m/s, which is typical for the operation of water economizers of boilers and air-cooled devices for various purposes.

3. The results of calculation and experiment agreed when the value \( \lambda_p = 75 \) W/(m*K) averaged over the rib thickness was taken, considering the high thermal conductivity of the copper-coated rib coating.

Based on the above research results, for assemblies of KP 20 elements of the second and third standard sizes, the maximum values of \( \alpha_m \) were determined, at which the “combined” efficiency factor
of the rib ($R=\Phi_0$) decreases to an acceptable value equal to 0.7. Accordingly, these values are $\alpha_\mathrm{n} = (33 \text{ and } 75) \text{ W/(m}^2\cdot\text{K})$. The first value is typical for the operation of air-cooled heat exchangers with natural draft with a pipe of considerable height - air-cooled systems of passive heat removal from the second circuit of the NPP-2006 through steam generators, dry cooling towers. The second value is typical for the operation of the cooling apparatus of the reactor room of the NPP combined with the fan. At $t_{\text{air}} = 100{\degree}\mathrm{C}$, the given values of $\alpha_\mathrm{n}$ correspond to air velocities of 7 m/s and 17.2 m/s.

Assemblies of the first and fourth standard size had the same dimensions of the KP 20 elements (see Figure 1) and differed only in the spacing of these elements, see the table. These assemblies had high finning ratios, exceeding the values for the range of recommendations [4]. In addition, the applied element KP 20 had a significant asymmetry in the location of the edges of the edges relative to the axis of the "pipes". At the same time, the above solution for the efficiency coefficient of the rib $E$ is correct only for an annular rib, and with a certain degree of approximation - for an octagonal element, i.e. second and third standard sizes of assemblies.

5. Numerical calculation of temperature distribution

For the KP 20 element shown in Figure 1, the problem of heat transfer along the rib from the outer surface of the "pipes" to the air flow washing over the rib was solved numerically using the ANSYS software package. In this case, the surface of the rib was divided into 40178 finite elements. Similarly, to the methods [3, 4], the value of $\alpha_\mathrm{n}$ unchanged over the entire surface of the rib was considered, and the value $\lambda_\phi = 75 \text{ W/(m}^2\cdot\text{K})$ was also kept. For the given values $\alpha_\mathrm{n}$ and $\Delta T = T_{\text{air}} - T_2$, the value of the heat flux along the fin from the outer surface of the pipes $Q_2$ and the value $E = Q_2 / Q_\infty$, where $Q_\infty = \alpha_\mathrm{n}F_\phi\Delta T$ is the heat flux along the edge at $\lambda_\phi = \infty$. In the range of values $\alpha_\mathrm{n} = 10 \text{ to } 100 \text{ W/(m}^2\cdot\text{K})$, the function $E=f(\alpha_\mathrm{n})$ is approximated by the relation $E = -0.1555\ln(\alpha_\mathrm{n}) + 0.9014$. Applying this ratio, on the basis of the experimental data $\alpha_\mathrm{ex}$, the corresponding values $\alpha_\mathrm{n} = \alpha_\mathrm{ex}/E$ were calculated.

The calculation was carried out by the finite element method in a three-dimensional spatial setting. The design scheme is a section of four plates connected to a monolithic tube with ribbing.

6. Discussion of the results of thermal calculations

In figures 4, 5 show examples of the results of the calculations performed in the form of temperature fields.

![Figure 4. Temperature distribution over the surface of the plates at $\alpha_\mathrm{n} = 21 \text{ W/m}^2 \cdot \degree\mathrm{C}$, $t_{\text{air}} = 0\degree\mathrm{C}$.](image)
An analysis of the calculation results showed that in cases of modes with $\alpha_n < 50 \text{ W/m}^2\text{°C}$ in the range $\Delta T < 130 \text{°C}$, an uneven distribution of temperatures along the edges of the plate is characteristic. In this case, the formation of zones of a significant decrease in temperature takes place. For such modes, it is recommended to change the shape of the element, namely, the manufacture of an oval-shaped plate or in the form of an eight.

In cases of modes with $\alpha_n > 100 \text{ W/m}^2\text{°C}$ in the range of $\Delta T > 130 \text{°C}$, a decrease in the surface with a temperature higher than the air temperature by more than 10% of $\Delta T$ is characteristic, to a diameter of 50 mm around the axis of the tubes. For such modes, it is recommended to use heat exchangers with a round element shape.

7. Analysis of hydraulic resistance

The experiments also determined the drag coefficients related to one row of the considered finned "tubes" with an external air flow around the assemblies. In this case, the root-mean-square relative determination error reached 30%. This was mainly due to the high relative errors in the measurement of small pressure drops $\Delta P_2$ on the tested small-row assemblies at moderate flow rates. The determination was carried out both with forced pumping of air through the assembly and with its natural draft (the second only with the assembly of the first standard size). In this case, it was determined at what value of $\zeta$ there is a correspondence between the driving head and the aerodynamic resistance of the assembly.

For a two-row assembly with the third (round) standard size of KP elements 20 $\zeta = 0.8$ at the value $\zeta = 0.185$ calculated according to [5]. A significant role in the discrepancy is played by the correctness of the specification of the coefficient $C_z$, which considers the small row of the beam. Coefficient according to the recommendation [5] at $z = (1; 2$ and 3), non-monotonically changes several times, amounting, respectively, at $\text{Re} = 10^4$ $C_z = (1.8; 0.4$ and $0.8$).

For the assembly with the first standard size of KP 20 elements, the correspondence between the experimental and calculated according to [7] values of $\Delta P_2$ took place when $\Delta P_2$ was considered as the sum: resistance of a two-row in-line bundle of smooth pipes at $\zeta = 0.18$ (flow velocity in the minimum section of the bundle); frictional resistance in the intercostal gap (flow velocity in the channel section,
uncorroded by pipes) at $\lambda_{tr} = A/Re$ (at $A = 89$ - the calculated value [7] for laminar flow with the existing ratio of the width and depth of the slot).

8. Conclusion

1. Thermal hydraulic and strength tests of finned assemblies welded from KP 20 elements of four standard sizes were carried out.
2. The possibility of their use in real gas-water heat exchangers at a pressure of the medium inside the "pipes" up to 5 MPa and a temperature of the metal up to 160°C with the additional presence during operation of sharp thermal changes of the medium and its freezing inside the "pipes" has been demonstrated.
3. For the external and internal pipe flow of a single-phase medium, ratios for heat transfer and hydraulic resistance of the considered media flows with the integration of these ratios into the general procedure for thermohydraulic calculation of heat exchangers presented in RD 24.035.05-89 are proposed. Thermal and hydraulic calculation of NPP heat exchange equipment.
4. Calculation of heat transfer showed that in cases of modes with $\alpha_n < 50 \text{ W/m}^2\text{°C}$ in the range $\Delta T < 130\text{°C}$ it is advisable to manufacture an oval or eight-shaped plate.

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