Results of Thermal Tests of Recirculation Cooling Installations

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Abstract. The article presents the results from thermal tests of some recirculation installations for cooling air in nuclear power plant premises, including the volume under the containment. The cooling effect in such installations is produced by pumping water through its heat-transfer tubes. Air from the cooled room is blown by a fan through a bundle of transversely finned tubes and is removed to the same room after having been cooled. The finning of tubes used in the tested installations was made of Grade 08Kh18N10T and Grade 08Kh18N10 stainless steels or Grade AD1 aluminum. Steel fins were attached to the tube over their entire length by means of high-frequency welding. Aluminum fins were extruded on a lathe from the external tube sheath into which a steel tube had preliminarily been placed. Although the fin extrusion operation was accompanied by pressing the sheath inner part to the steel tube, tight contact between them over the entire surface was not fully achieved. In view of this, the air gap’s thermal resistance coefficient was introduced in calculating the heat transfer between the heat-transferring media. The air gap average thickness was determined from the test results taking into account the gap variation with temperature due to different linear expansion coefficients of steel and aluminum. These tests were mainly aimed at checking if the obtained thermal characteristics were consistent with the values calculated according to the standard recommendations with introduction, if necessary, of modifications to those recommendations.

1. Introduction
Nuclear power plant (NPP) premises are equipped with recirculation cooling installations (RCIs), also known as recirculation ventilation installations (RVIs). Air from the room being ventilated is pumped (by means of a fan) through the heat exchanger available in such an installation and returned to the same room. Such installations are of particular relevance for being placed in rarely attended premises whose air may contain harmful substances.

The NPO CKTI carried out thermalhydraulic tests of the Type ROU-6.3 recirculation cooling installation designed for the rated air flowrate \( G_{\text{air}} = 6300 \, \text{m}^3/\text{h} \) and intended to be used in the system for cooling air under the containment of the Rostov NPP units № 3, 4 [1]. The installation was manufactured by the Volgodonsk Works of Ventilation Articles (VZVI). Similar installations produced for the Rostov and Leningrad NPPs were tested. The latter installation used flashing freon for air cooling purposes; the freon itself was cooled by service cooling water circulating in the compressor-type chilling installation.
(air conditioner). The RUV-60 recirculation ventilation installation with $G_{\text{air}}=6\times10^4$ m$^3$/h for the Novovoronezh NPP was the largest tested installation.

The tested installations, which had the same staggered arrangement of finned tubes (referred to hereinafter as tubes) in the heat exchanger and the same annular shape of fins, differed in the used materials, geometrical characteristics, and manner in which the tube was engaged in contact with the fin: by welding a fin to the tube over the entire contact length by means of high-frequency welding or by crimping an aluminum tube around a steel tube with simultaneously extruding helical finning from the aluminum tube (Figure 1). The results from a numerical analysis of heat transfer intensity from the tube carried out according to the standard recommendations [2–4] have demonstrated excellent results. The aim of this study is to check the validity of the calculations against the results obtained from the tests of a few large heat-transfer installations carried out with the actually existing air velocities (Re).

2. Description of the tested installations, test facility, and measurement procedure

The tested installations comprised a heat exchanger and a centrifugal or an axial fan interconnected by an air duct (Figure 2). A louver-type moisture separator was placed inside the air duct for removing the moisture generated during steam condensation from humid air in the course of its cooling in the heat exchanger. The technical data of the finned tubes and tested ROU-6.3 and RUV-60 installations are given in [5]. The RUV-60 installation contained two heat exchangers connected in a series by air and having different numbers of identical tubes—540 pcs. in the large heat exchanger and 216 pcs. in the small heat exchanger—and with the corresponding number of rows along the bundle depth (10 and 4). Water was pumped through one heat exchanger during the tests. The tubes externally cooled by air were arranged horizontally in all installations. Water flowed sequentially through the row of tube passing over the height with the cross flow of heat-transferring media.

![Figure 1](image1.png)  
**Figure 1.** Finned tube manufactured using the knurling method. (1) Tube, (2) air gap between the tube and sheath, (3) tube finning sheath, and (4) fin.

![Figure 2](image2.png)  
**Figure 2.** Test facility circuit arrangement: (1) fan, (2) heat exchanger, (3) pressurizer, (4) and (5) electric heaters, and (6) circulation pump driven by an adjustable speed motor.

In this article considers operation of the installations in the air-cooling mode without condensing the steam contained in the air. By changing the water temperature and flowrate at the inlet, as well as the air flowrate from one experiment to another, we succeeded in obtaining a certain amount of experimental data on the coefficients of heat transfer from the finned tube surface to air for comparison with the calculated results. The Type KSk3-6 air heater was in its delivery state just a heat exchanger, which was additionally fitted for the tests with a fan driven by an adjustable-speed electric motor. For testing the RUV-60 installation, a petal-type shutter was installed downstream of the axial fan outlet pipe, and flow limiting orifices were installed at the edge of the fan outlet pipe during some tests of the ROU-6.3 and ROU-35 installations for the possibility to adjust the air flowrate of the fan.

The water circuit instrumentation (see Figure 2) included means for measuring the pressure at the inlet $p_1$ and pressure drop in the heat exchanger $dP_1$ and the flowrate $G_1$ and temperature at the heat exchanger inlet $t_{1\text{in}}$ and at its outlet $t_{1\text{out}}$. The water flowrate was determined by measuring the pressure...
difference across the Venturi flow tube. The air circuit instrumentation included means for measuring underpressure downstream of the heat exchanger \( d_p^2 \) and the temperature at its inlet \( T_{in} \) and outlet \( T_{out} \).

All thermocouples used to measure water temperature were duplicated. Three to eight thermocouples were installed for measuring the field of air-flow temperatures over the air duct cross sections. A uniform field of temperatures at the fan outlet was obtained owing to air being agitated in the fan. Apart from subjecting all instruments to qualification at the Test-Spb Institution, the thermocouples passed additional calibrations before each series of tests, which included the following operations.

Owing to these measures, we succeeded in keeping the maximum absolute error (MAE) or the media temperature determination uncertainty according to [6] to be as low as 0.5°C. The same value of MAE was estimated for the measured changes in the temperature of each medium \( \Delta T \) and the average difference of temperatures between the heat-transferring media \( \Delta T_{\text{av}} \).

The rms error (RMSE) evaluated taking into account the errors introduced by the sensor reading transmission and automatic recording circuits did not exceed 0.212% for pressure measurements and 0.224% for pressure difference measurements. Computer-aided recording of instrument readings was performed at a 1 Hz frequency. The RMSE of determining the water flowrate \( \delta G_1 \) in the course of data processing did not exceed 5.5×10\(^{-3} \). During the tests, the combination of media flowrates and temperatures at the inlet was selected so that the change in the temperature of each medium and the average difference of temperatures between the media were no less than 10°C. As a result, the RMSE of determining the heat exchanger power \( \delta N \) did not exceed 0.093, and that of air flowrate \( \delta G_2 \) calculated based on the data from determining the heat exchanger power in its water circuit did not exceed 0.118. Our attempts to determine the air flowrate using the data from measurements of its velocity field in the fan outlet pipe, as well as attempts to set up the air flow stabilization section (during other tests) of the required length (more than 5 m) between the heat exchanger and fan in measuring the air velocity field by means of a heat-loss anemometer, gave essentially higher RMSE values as compared with that of the balance method used in the present study.

3. Results and Discussion
The further processing of the investigation data included calculations of the heat transfer coefficient \( k \) related to the total outer surface area \( F_s \) of the tested heat exchanger: \( k = N/(F_s \Delta T_{\text{av}}), \) where \( F_s = \pi d_{\text{sh}} L \) and \( L \) is the length of the tube working (finned) part.

The value of \( k \) can also be represented as follows

\[
k = 1/\rho \left\{ d_{\text{sh}}/(\alpha_i d_m) + d_{\text{sh}} \ln(d/d_m)/(2 \lambda_{\text{sh}}) + d_{\text{sh}} \ln((d+2 \lambda_{\text{sh}})/(2 \lambda_{\text{sh}})) + d_{\text{sh}} \ln([d_{\text{sh}}/(d+2 \lambda_{\text{sh}})]/(2 \lambda_{\text{sh}})) + 1/(\alpha_2 \rho) \right\}^{-1}, \tag{1}
\]

where \( \lambda_{\text{sh}} \) is the mean gap between the tube and sheath, \( m; \lambda_{\text{w}}, \lambda_\ell, \) and \( \lambda_{\text{g}} \) are the heat conductivities of the tube wall, fin, and air in the gap (the latter is determined at the air temperature in the gap), \( W/(m K); \alpha_i \) is the coefficient of heat transfer from water to the tube inner surface, \( W/(m^2 K); \) and \( \alpha_2 \) is the coefficient of heat transfer to air related to the total outer surface area of tubes, \( W/(m^2 K) \).

For the installations with steel finning, the fins in which were welded over the entire length to the tube outer surface, the third and fourth terms in the denominator of (1) are equal to zero, and the sheath outer diameter is replaced by the tube outer diameter. The same applies to the expression for calculating the surface area. Therefore, by using the \( \alpha_1 \) value calculated from the Krausold–Nusselt relationship [2–4] for turbulent flow of water inside the tubes, we determined the sought value of \( \alpha_2 \) from expression (1). The fraction of thermal resistance \( (\alpha_2)^{-1} \) in the performed tests was 44–88% of its total value \( k^{-1} \), and, with a small error of calculating the other terms in (1), \( \delta (\alpha_2) = 0.101–0.134 \).

The regulatory documents [2, 3] use the same expression for calculating the heat transfer coefficient related to the total outer surface of tubes

\[
\alpha_{\text{calc}} = (E \mu_0 F/F_s + F_i/F) \alpha, \tag{2}
\]

where \( E = f(\beta h_t) \) is the finning efficiency coefficient, the value of which for annular fins is calculated analytically using the Bessel functions; \( \mu_0 \) is the coefficient taking into account the effect of fins becoming wider toward their base; \( \psi = (1–0.058 \beta h_t) \) is the coefficient that takes into account heat transfer nonuniformity over the fin surface; \( \beta = (2 \alpha/(\lambda_i \delta_{\text{sh}})^{0.5}) \); \( h_t \) is the fin height, \( m; F_i \) and \( F_s \) are the fin and tube surface areas, \( m^2; \) and \( \alpha = N_u \lambda_f /h_0 \).
The difference between the methods described in [2] and [3] lies in the quantity taken as the characteristic size $l_o$: this is the tube outer diameter $d$ (or the sheath outer diameter $d_{sh}$ if there is a sheath) in [3], and this is a more intricate complex in [2] calculated from the formula

$$l_o=\left(1-F_d/F_s\right)d+\left(F_d/F_s\right)(0.25\pi(d^2-d^2)^{0.5}).$$

For the case of staggered tube bundles placed in a cross flow, the Nusselt criterion related to the total outer surface area is given, according to [2], by

$$Nu = 0.36RePr^{0.33} \varphi^{0.5} C_sC_n,$$

where $\varphi = 0.6\varphi^{0.07}$; $C_s$ is the correction factor for the number of tube rows $z$ in the bundle along the flow ($C_s = 1.0$ at $z>4$); and $C_n=\left[(\sigma_1-1)/(\sigma_2^*-1)\right]^{0.1}$ is the bundle form factor. Formula (4) is recommended to be used at $Re = w_2 l_0/v_2 = 5 \times 10^3-3.7 \times 10^3$, $\varphi=1.0-21.2$, $l_0=0.012-0.178$ m, and $(\sigma_1-1)/(\sigma_2^*-1)=0.46-2.2$, where $\sigma^2=(\sigma_1^2/4+\sigma_2^2)^{0.5}$; $w_2$ is the air velocity in the narrow section; $v_2$ is the kinematic viscosity coefficient; and $\sigma_1$ and $\sigma_2$ are the transverse and longitudinal relative pitches of tubes in the bundle.

In [3], another correlation was proposed:

$$Nu = 0.113RePr^{0.33} C_sC_n,$$

where $n = 0.7 + 0.08\xi + 0.005\varphi$; $\xi = \tanh(\lambda)$; for staggered tube bundles, the following holds:

$x = \sigma_1/\sigma_2 - 1.26/\varphi - 2$; $C_s=1.36-(\varphi+0.14)$; and $C_n=1.0$ at $z>7$.

The authors of [3] did not specify the application boundaries for expression (5). However, the authors of [7], the recommendations of which are similar to those used in [3], indicated them: $Re = w_2 d/v_2 = 5 \times 10^3-2 \times 10^3$, $\varphi=1.2-39$; $\sigma_1=1.7-6.5$; $\sigma_2=1.3-9.5$; and $\sigma_1/\sigma_2=0.3-5.2$. It should also be pointed out that, according to [7], the coefficient that considers the heat transfer nonuniformity over the fin surface is given by $\psi=1-0.016(d/d-1)[1+\tanh(2\beta h-1)]$.

Figure 3 shows a comparison between the experimental coefficients of heat transfer to air $\alpha_{exp} = \alpha_2$ for installations with welded steel finning and their calculated values $\alpha_{calc}$ evaluated according to the recommendations given in [2, 3].

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The relative deviations $\delta \alpha = |\alpha_{exp}/\alpha_{calc} - 1|$ obtained in the calculations carried out according to [2] are as follows for their maximal, algebraic mean, and rms values, respectively: $\delta \alpha_{max}=0.09$, $\delta \alpha_{a.m}=0.03$, and $\sigma=0.04$; in the calculations carried out according to [3], $\delta \alpha_{max}=0.11$, $\delta \alpha_{a.m}=0.04$, and $\sigma=0.04$. All the above-mentioned values of the quantities are within the $\alpha_{exp}$ determination error and within the permissible deviation from the calculation according to [2, 3].
The regulatory document [4], which is the last one in the chronological order, contains an outdated procedure for calculating the Nu criterion [8] without separating the contribution the heat transfer from the nonfinned part of the tube surface and that from the surface of fins introduce in the average value of \( \alpha \) with taking into account the heat conductivity of their material, whereas expression (2) does take this separation into account. The calculation procedures developed by foreign authors [9, 10] are also simplified in nature and are inferior to the procedures outlined in [2, 3].

Both installations the test results of which are analyzed in Fig. 3 have geometrical characteristics that are often used in real heat exchangers, including \( \varphi=7.0 \) and 7.5 as well as the air flow velocities \( w_{\text{air}}=5.6–13.7 \) m/s. In all likelihood, the researchers who developed formulas (3)–(5) used the combination of these parameters. The tested assemblies fitted with aluminum fins had a higher value of the finning ratio \( \varphi \), and, as the results of experiments reported in [11] have shown—which were carried out on small tube assemblies with welded fins at an increased value of \( \varphi \)—the value of the Re number exponent \( n \) in (5) is overestimated, and the ratio \( \alpha_{\text{exp}}/\alpha_{\text{calc}} \) decreases monotonically as Re increases. The exponent \( n \) in (4) under the same conditions has a smaller value and satisfactorily describes the function \( \alpha_{\text{exp}}=f(\text{Re}) \). Therefore, those who analyze the results of tests on installations with aluminum finning should give preference to the recommendations outlined in [2]. It should be pointed out as an argument in favor of this choice that both the recommendations also consider the case of a nonfinned tube (\( \varphi=1 \)), for which \( n=0.6 \) in the considered range of Re (at \( l_{\text{g}}=d \)) according to the recommendations of [2–4], which is not the case in [3]. It can also be conjectured that with \( \varphi \to \infty \) (a flat slit channel), \( n \to 0.8 \). These limiting conditions are fulfilled in [2] and give higher values of \( n \) according to [3].

The recommendations mentioned above were used for calculating the values of \( \alpha_{1} \) and \( \alpha_{2} \), and the results from testing the ROU-35 installation (an installation with knurled aluminum finning that does not have positively tight contact between the aluminum sheath inner surface and the steel tube outer surface) were generalized. It was found from that analysis that the third term in (1) accounts for 4–16% of the total thermal resistance to heat transfer between the heat-transferring media. Its fraction increases with increasing the calculated air temperature in the considered gap \( t_{\text{g}} \), which was attributed to the difference between the linear expansion coefficients of Grade AD1 aluminum \( \alpha_{\text{al}} \) and Grade 08Kh18N10T steel \( \alpha_{\text{st}} \). For the considered temperature range of \( t_{\text{g}} \), we have \( \alpha_{\text{al}}=16.1 \times 10^{-6} \text{ K}^{-1} \) and \( \alpha_{\text{st}}=23.5 \times 10^{-6} \text{ K}^{-1} \) [12, 13].

With the aluminum tube wall heat conductivity \( \lambda_{\text{al}}=200 \text{ W/(m K)} \) adopted according to [12], the solution of (1) with respect to the average air gap enabled us to obtain the dependences \( \delta_{\text{g}}=f(t_{\text{g}}) \) (Fig. 4): experimental (1) and calculated (2) derived from the equation

\[
\delta_{\text{g}}=\delta_{0}+(\alpha_{\text{al}}-\alpha_{\text{st}})t_{\text{g}}d/2, \tag{6}
\]

where \( \delta_{0} \) is the conditional average gap, \( \mu \text{m} \) at 0°C, which was determined proceeding from the condition of minimizing the average difference between the values of \( \delta_{\text{g}} \) obtained from the experiment and calculated from (6) (for the considered tests, \( \delta_{0}=8.5 \mu \text{m} \)).

![Figure 4. Dependence of \( \delta_{\text{g}} \) vs. \( t_{\text{g}} \) for the ROU-35 installation for \( \delta_{0} = 8.5 \mu \text{m} \).](image-url)
Expression (6) is simplified in nature and does not take into account the difference between the tube wall and finning sheath temperatures nor does it take into account elastic deformation of mating materials when they come into force contact with each other on a certain part of the surface.

The results obtained from the tests of tubes with aluminum finning we carried out at JSC Kalorifernyi Zavod (Air Heater Works) in the city of Kostroma in different years have shown that the technologies for manufacturing them have been improved, as a result of which smaller values of $\delta_{\text{g,in}}$ have been obtained. Thus, the previous tests of small tube assemblies manufactured at the Works in 2000 gave $\delta_0=22.5$ μm, whereas the gap measured in the tubes of the Type KSk3-6 air heater manufactured in 2014 was found to be $\delta_0=4–8$ μm at $t_g=53–116^\circ\text{C}$. This testifies that the roughness protrusions on the contacting surfaces experience elastic deformation and that its value decreases (without loss of contact between the mating surfaces) as $t_g$ increases.

4. Conclusions
(1) The results obtained from the tests of real recirculation air cooling installations for NPPs have corroborated the validity of the recommendations set out in the RD (Guiding Document) 24.035.05-89 for calculating heat transfer at the external finned surface of heat-transfer tubes.
(2) Some mistakes in RD 24.035.05-89 were corrected and its application area was extended.
(3) The thermal resistance of the air gap between the tube and finning must be taken into account if positive contact between their surfaces (like that secured by welding, soldering, etc.) cannot be guaranteed. The procedure proposed for calculating the air gap thermal resistance considers the difference between the linear expansion coefficients of the tube and finning materials.

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