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Improvement air condensers evaluation model

Svyatoslav Tsibulskiy[1,*], *Nikolay* Galashov[1], *Denis* Mel'nikov[1], *Alexandr* Kiselev[1], and *Al'bina* Bannova[1]

[1]National Research Tomsk Polytechnic University, 634050 Tomsk, Russia

Abstract. The results of analysis of the literature on the calculation of the heat transfer coefficient of an air condenser in the flow past a bundle of finned tubes by an air flow. The methods of calculation are disassembled, marked advantages and disadvantages of each. Calculations of the heat transfer coefficient for each method are given; the results compared with the experimental data.

1 Introduction

Air condensers (AC) are now widely used in a number of foreign TPPs with power units up to 1000 MW and at CCGTs. The main disadvantage of air condensers is a low heat transfer coefficient and large dimensions [1]. Therefore, it is advantageous to use them at TPPs, where turbines are installed with low steam flow rates to the condenser. This is primarily the turbines for utilization of combined-cycle gas turbines (CCGT), the steam consumption in the condenser is three times less than for conventional steam-turbine power units of the same unit capacity [2]. At the same time, TPP based on CCGT is becoming profitable to establish in the area of gas production and low temperatures of outside air.

One of the most serious problems of AC operation in winter during the condensation of water vapor is the condensate freezing in the tubes, which can lead to the destruction of tubes and the need to repair the condenser and stop the turbine. The solution to this problem is to use the AC in a CCGT with three cycles, where the low-boiling substances (LBS) [3] operates in the lower cycle, and its condensation can reliably pass at temperatures below -40 °C [4].

The problem of design and application of AC is the lack of reliable methods for their calculation. The most problematic issue is the cooling package finned tubes with air [5].

To create algorithm and program for calculating AC operating on the LBS, are made the analysis and a choice of the currently most acceptable methods for calculating a bundle of finned tubes when they are cooled by air.

2 Methods for calculating the heat transfer coefficient for a bundle of finned tubes

* Corresponding author: stzibulsky@tpu.ru

2.1 Method for calculating the heat transfer coefficient for a package of finned tubes of circular cross-section with a chess arrangement, A.I. Kirillov [6].

Method obtained based on experimental studies of heat treatment of the finned tubes.

The average convective heat transfer coefficient over the surface:

$$\alpha_k = Nu \cdot \lambda / d_H, \text{ W/m}^2\cdot\text{K}, \tag{1}$$

where λ – coefficient of thermal conductivity of air; the hydraulic beam diameter is defined as:

$$d_H = 2[t(s_1 - \delta) - 2\delta h)/2(h+t)], \text{ m}, \tag{2}$$

where t – step between fins on a tube, m; s_1, s_2 – The transverse and longitudinal pitch of the tubes in the tube bundle, m; h, δ – height and average thickness of the fin, m.

The Nusselt number is determined by the criterial equation:

$$Nu = 0.36 \cdot Re_n \cdot Pr^{0.33} C_Z \cdot C_S \cdot \psi^{-0.5}, \tag{3}$$

where:

$$n = 0.6 \cdot \psi^{0.07}; \tag{3a}$$

coefficient of finning:

$$\psi = [1 + 2 \cdot h(d_0 + h + \delta)]/(d_0 t); \tag{3b}$$

C_Z – correction factor for the number of rows of tubes in the bundle along the air flow; coefficient of the tube bundle form:

$$C_S = ((\sigma_1 - 1)/(\sigma'_2 - 1))^{0.1}; \tag{3c}$$

relative transverse and longitudinal steps of the tube bundle:

$$\sigma_1 = s_1/d_0, \ \sigma_2 = s_2/d_0; \tag{3d}$$

relative diagonal step:

$$\sigma'_2 = \sqrt{\sigma_1^2 / 4 + \sigma_2^2}; \tag{3f}$$

d_0 – outer diametr of the tube, m.

The Nusselt ratio is applicable for the range of the Renolds number $Re = 5 \cdot 10^3 \div 37 \cdot 10^4$ and the shape of the tube bundle $C_S = 0.46 \div 2.2$.

The heat transfer coefficient is defined as:

$$\alpha = \alpha_к \eta, \tag{4}$$

where:

$$\eta = 1 - 0,058 \cdot \beta \cdot h; \tag{4a}$$

$$\beta = \sqrt{2\alpha_к / (\lambda \cdot \delta)}. \tag{4b}$$

Range of change $\beta h = 0.1 \div 3.7$.

2.2 Method for calculating the heat transfer coefficient for a package of finned tubes of circular cross-section with a chess arrangement, A.N. Bessonny [7].

Reduced heat transfer coefficient:

$$\alpha = \alpha_k[(\psi - 1)E + 1]/\psi, \text{ W/m}^2\cdot\text{K}, \tag{5}$$

where E – fin efficiency factor; $\alpha_к$ is determined by the formula (1),
where:

$$Nu = St\cdot Pr^{2/3}\cdot Re\cdot Pr^{1/3}\cdot C_Z, \tag{6}$$

where:

$$St\cdot Pr^{2/3} = -7.5\cdot 10^{-7}Re + 0.0135; \tag{6a}$$

C_Z – coefficient taking into account the number of rows in the direction of air movement.

2.3 Method for calculating the heat transfer coefficient for a package of finned tubes of circular cross-section with a chess arrangement, E.N. Pis'menny [8].

The method presents a unified system of generalizing equations that allows one to find in the interval $Re=5\cdot 10^3\div 2\cdot 10^5$ the values of heat transfer coefficients for chess beams of cross-finned tubes with a finning coefficient $\psi=1.2\div 39.0$ and relative transverse and longitudinal pitch $\sigma_1=1.7\div 6.5$, $\sigma_2=1.3\div 9.5$, $(\sigma_1/\sigma_2 = 0.3\div 5.2)$.

The average convective heat transfer coefficient over the surface is determined by the formula (1). The hydraulic diameter of the bundle of tubes is determined by the formula (2).

The Nusselt number is determined by:

$$Nu=1.13Re^m Pr^{0.33} C_q C_Z, \tag{7}$$

where for the exponent:

$$m=0.7+0.08th(X)+0.005\psi; \tag{7a}$$

$$X=\sigma_1/\sigma_2-1.26/\psi-2; \tag{7b}$$

$$C_q=(1.36-thX)(1.1/(\psi+8)-0.014). \tag{7c}$$

Coefficient C_Z takes into account the effect on the heat transfer of the number of transverse rows of tubes in the tube bundle z_2.

For chess packages with $\sigma_1/\sigma_2 < 2$ and $z_2>8$ $C_Z=1$, while $z_2<8$:

$$C_Z=3.15z_2^{0.05}-2.5. \tag{8}$$

The reduced heat transfer coefficient is defined as:

$$\alpha=\alpha_k(E\psi_E F_f/F+F_t/F), \tag{9}$$

where surface area of tube without fins:

$$F_t=\pi\cdot d_0\cdot\psi\cdot L; \tag{9a}$$

surface area of the fins:

$$F_f=z_2 F_t B/s_1; \tag{9b}$$

total area of finned tube:

$$F=F_t+F_f; \tag{9c}$$

L – tube length; B – the width of the tube bundle; coefficient of theoretical efficiency of the tube fin:

$$E=th(\beta\cdot h'')/(\beta\cdot h''); \tag{9d}$$

where:

$$\beta=\sqrt{2\alpha_k/(\lambda\cdot\delta)}; \tag{9f}$$

$$h''=\left(h+\frac{\delta}{2}\right)\left[1+\left(0.191+0.054\frac{D}{d_0}\right)\ln\frac{D}{d_0}\right]; \tag{9g}$$

D – the outer diameter of the fins; correction for non-uniformity of heat transfer across the surface of the tube:

$$\Psi_E=1-0.016\left(D/d_0-1\right)\left[1+th(2\cdot\beta\cdot h-1)\right]\cdot \tag{9h}$$

The limits of applicability of equation $D/d_0=1.1\div4.0$; $\beta\cdot h=0.1\div4.0$.

2.4 Method for calculating the heat transfer coefficient for a package of finned tubes of circular cross-section with a chess arrangement, J. Moore [9].

Coefficient of heat transfer:

$$\alpha = 1000St\cdot Pr^{2/3}\cdot\rho\cdot c\cdot\omega/Pr^{2/3},\ W/(m^2K), \tag{10}$$

where ρ, c, ω – air density, heat capacity and velocity; Pr – Prandtl number;

$$St\cdot Pr^{2/3} = 0.134Re^{-0.319}(t/h)^{0.2}(t/\delta)^{0.1134}. \tag{10a}$$

Reynolds number:

$$Re = G\cdot d_H/(\mu\cdot A_{min}), \tag{11}$$

where G, μ – air flow rate and viscosity; A_{min} – minimum sectional area for the passage of air through the tube bundle; h, t, δ – height, pitch and thickness of the fin;
the hydraulic diameter and limits of application of equation:

$$d_H = 4h\cdot t/(h+t). \tag{12}$$

$$Re = 1100\div18000. \tag{13}$$

3 Experimental results

According to these methods, for the same initial data, heat transfer coefficient calculations for the AC section.

For the calculation, the following design parameters of the AC section were adopted:

1. Outer diameter of the support tube d_0 = 0.027 m.
2. The fin height h = 0.015 m.

3. The average thickness of the fins $\delta = 0.000735$ m.
4. Fin step $t = 0.0025$ m.
5. Transverse step $s_1 = 0.084$ m.
6. Longitudinal step $s_2 = 0.074$ m.
7. Number of longitudinal rows of tubes $z_1 = 4$ pcs.
8. Number of transverse rows of tubes $z_2 = 6$ pcs.
9. The width of the tube bundle $B = 0.945$ m.
10. Tube length $L = 12$ m.

The results of the calculation are given in the Table 1.

Table 1. The results of comparison of the methods for calculating the heat transfer coefficient for the air condenser section.

Autor	d_G, m	$St \cdot Pr^{2/3}$	Re	Nu	α, W/(m²K)
Kirillov [6]	0.0115	-	4929.7	38.9	80
Bessonny [7]	0.0115	0.0098	4929.7	40.9	64.2
Pis'menny [8]	0.0115	-	4929.7	27.8	51.9
Moore [9]	0.0115	0.0072	4805.3	-	67.6

The table shows that the highest heat transfer coefficient was obtained by calculating the formulas Kirillov, Moore and Bessonny, but the error in calculating them with respect to the experimental data is estimated to be above 20%. In this case, even when calculating using the Sleepless method, where the heat transfer coefficient is lower than that of Kirillov and Moore, the heat transfer coefficient is multiplied by 0.8.

When calculating the heat transfer coefficient using the Pis'menny formulas, the greatest number of geometrical and thermophysical parameters characterizing the heat exchange of the tube bundle with air is used. Taken into account the uneven heat transfer of the fin surface. The boundaries of the applicability of the formulas are clearly defined. The technique provides an error in the calculation of the heat transfer of finned tube packs below 20%.

As a result of the analysis performed in the construction of the VC model, the Pis'menny method was chosen for calculating the heat transfer coefficient, since they most accurately reflect the actual process of heat exchange of the finned tube bundle with cooling air.

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