Approximate estimation of the thermal resistance of the terms in the process of heat transfer through the finned wall

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Abstract. The paper deals with issues related to determining the value of the thermal resistance of the fin-wall contact in heat exchangers. The description of the process of heat transfer between heat carriers and the main factors and the key parameters which determine the thermal resistance between the rib and the wall is presented. The experimental stand for thermal-hydraulic studies of the tube bundle model was developed. A technique for estimating the thermal resistance of a rib-wall contact on the basis of experimental studies is proposed. An estimation of possible sources of errors that should be taken into account when performing calculations is carried out.

1. Introduction
When calculating heat exchangers with finned tubes of different geometry, the question arises about the correct consideration of the thermal resistance of the wall-fin assembly. There are two aspects here:

- a significant temperature gradient in the wall-fin contact area;
- quality of fin-wall thermal contact.

Figure 1 qualitatively illustrates the distribution of temperatures associated with thermal resistance:
I – convective heat transfer from the heat-transfer medium I to the fins and wall;
II – thermal resistance of fins;
III – thermal resistance of contact;
IV – thermal resistance of the wall;
V – convective heat transfer from the wall to the heat-transfer medium 2.
The components of processes I, II and V are studied in detail and repeatedly published, for example [1÷5].

The thermophysical properties that determine the thermal resistance of the wall IV are also usually known, but taking into account its non-isothermal viscosity makes an uncertainty which is difficult to estimate at the engineering level.

The greatest problems arise in connection with the region of the wall-fin contact (number III on figure 1). In [6], the effect of various parameters on the thermal resistance of fin-wall contact is examined more qualitatively than quantitatively. The authors [6] report the following key parameters:

- surface quality;
- thermal conductivity of contact materials;
- mechanical properties of materials;
- embodiment of the thermal contact area;
- specific pressure (stress state) of contacting materials.

To this list two factors can be added:

- possibility of corrosion at the point of contact;
- fin welding conditions in case of welded junction.

At moderate values of finning factors $\varphi_f$ and low-intensity heat transfer from the heat-transfer medium 1, the thermal contact resistance, usually, does not play a special role, and the error in its determination has little effect on the calculation results of the heat transfer surface.

However, in the case of large values of $\varphi_f$ or intense heat removal from the heat-transfer fluid 1, this resistance significantly affects the choice of the surface area values. This applies to condensation of steam from the steam-air (gas-vapor) mixture in boiler heat utilizers (welded fins $\varphi_f = 8 \div 10$, $\alpha_1 = 100 \div 150$ W / m²K), in air condensers and dry fan cooling towers (bimetallic pipes, fins $\varphi_f = 17 \div 20$, $\alpha_1 = 60 \div 80$ W / m²K).

The proposed procedure for estimating the thermal resistance is oriented to these devices and allows one to improve the accuracy of calculations on the one hand, and on the other hand to make a choice between different connection types based on quantitative rather than qualitative information.
2. Materials and methods

The methodological basis for the experimental evaluation of the thermal resistance is the well-studied regularities connecting the values of the dimensionless numbers Nu (or St) to the flow conditions of media characterized by Re numbers in the region of fully developed turbulent flow. In the case of flow over fins, it is expedient to additionally take into account the Bi number, which characterizes the conjugate process of convective heat transfer and thermal conductivity.

Thus, in accordance with the formula for the flow in the channel [3]

\[ \text{Nu} \sim \text{Re}^{0.8}\text{Pr}^{0.43} \]  

(1)

for flow through finned tubes bundle [1, 2]

\[ \text{Nu} \sim \text{Re}^{0.65}\text{Pr}^{0.33} \]  

(2)

To conduct research on the tube bundle, a tube bundle model was made from its tubes and tested on a special test bench, the scheme of which is shown in figure 2.

![Figure 2. Stand for investigation of heat transfer processes in a bundle of finned tubes; t, p – temperature and static pressure measurement; air, w – indices for air and water.](image)

Stand measuring system provides correct determination of the heat transfer coefficient \( k \) over a wide range of flow rates of the heating and heated mediums.

In the classical case, the total thermal resistance

\[ R = \frac{1}{k} = R_1 + R_{\text{wall}} + R_2 \]  

(3)

The thermal resistance of convective heat transfer from the side of the finned wall in accordance with (2)

\[ R_1 = R_{10} \left( \frac{\text{Re}_1}{\text{Re}_{10}} \right)^{0.65} \left( \frac{\text{Pr}_1}{\text{Pr}_{10}} \right)^{0.33} \]  

(4)

Where \( R_1, R_{10} \) – thermal resistances of convective heat transfer from the side of the finned wall in the calculated (0) and current (1) modes.

If air used as a coolant (1), the ratio \( \text{Pr}_1/\text{Pr}_{10} \) can be neglected.

Thermal resistance of convective heat transfer from the coolant flow in the channel, see (1)

\[ R_2 = R_{20} \left( \frac{\text{Re}_2}{\text{Re}_{20}} \right)^{0.8} \left( \frac{\text{Pr}_2}{\text{Pr}_{20}} \right)^{0.43} \]  

(5)
Then it is necessary to select the base point (index 0) with the given flow rate and parameters of both heat carriers, to determine the heat transfer coefficient $k_0$ and the total thermal resistance at this point

$$R_0 = \frac{1}{k_0} = R_{10} + R_{\text{wall}} + R_{20}$$  \hspace{1cm} (6)

This point corresponds to the base values $Re_{10}$, $Pr_{10}$, $Re_{20}$, $Pr_{20}$.

An experiment for wide range of fluid 1 flow rate is performed. The results are approximated using a smooth curve. Points on the curve with index 1 with known heat transfer coefficient $k_1$ and the numbers $Re_{11}$ and $Pr_{11}$ are selected.

$$R_1 = \frac{1}{k_1} = R_{10} \left( \frac{Re_{11}}{Re_{10}} \right)^{0.65} + R_{\text{wall}} + R_{20}$$  \hspace{1cm} (7)

Subtracting (7) from (6), we obtain

$$\Delta R = \left( \frac{1}{k_0} - \frac{1}{k_1} \right) = \left( 1 - \frac{Re_{11}}{Re_{10}} \right)^{0.65} R_{10}$$

whence

$$R_{10} = \frac{\Delta R}{\left( 1 - \frac{Re_{11}}{Re_{10}} \right)^{0.65}}$$  \hspace{1cm} (8)

$R_{10}$ can be compared to its estimated value determined by [1, 2], and if necessary the value of the exponents in (2) and derived formulas can be refined.

A similar procedure is carried out for coolant 2 with constant parameters of coolant 1, for which its thermal resistance $R_{10}$ is known. In this case, the experimental values of $k_2$, $Re_2$, $Pr_2$ are known

$$R_2 = R_{10} + R_{\text{wall}} + R_{20} \left( \frac{Re_2}{Re_{20}} \right)^{0.8} \left( \frac{Pr_2}{Pr_{20}} \right)^{0.43}$$  \hspace{1cm} (9)

Subtracting (9) from (6), after the transformation we obtain

$$Re = \frac{R_0 - R_2}{1 - \left( \frac{Re_2}{Re_{20}} \right)^{0.8} \left( \frac{Pr_2}{Pr_{20}} \right)^{0.43}}$$  \hspace{1cm} (10)

The value of the convective heat transfer coefficient is useful to compare with the calculated data [2, 4, 5].

The final step finds the value $R_{\text{wall}}$

$$R_{\text{wall}} = R_0 - (R_{10} + R_{20})$$  \hspace{1cm} (11)

It is important to note that this parameter takes into account the thermal conductivity of the fin and the non-isothermicity of the wall. On the one hand, this is important for engineering calculations, on the other—this somewhat limits the range of velocities on the side of both coolants, which is associated with a change in finning efficiency and heat exchange conditions on the channel walls.
An experimental study of a tube bundle assembled from finned tubes shown in figure 3 was carried out.

3. Results of studies
The results of the experiments are shown in figure 4.

Figure 3. The tube bundle for evaluating the thermal contact resistance of the fin-wall assembly. All dimensions are given in millimeters.

Figure 4. Dependences of heat transfer (1) and convective heat transfer coefficients (2) on the Re number: a) under variable air flow rate; b) under variable water flow rate.
4. Discussion and Conclusions
Since the method is of the nature of an approximate evaluation, it is important to determine the sources of errors and their effect on the results of calculations. The error may be contained in the following positions:

- the thermal resistance of an fin-wall contact is calculated as a small difference of large numbers, significant impact has the convective heat transfer coefficient calculation error on the side of the finned wall;
- an error in the determination of the exponents in formulas (1) and (2) is possible, in particular for Re<7000 for a flow in a finned tube; it is better to use the formula [5] here, in particular, taking into account the influence of the inlet section of the pipeline;
- it is also necessary to estimate the influence of the thermal conductivity of the fin and wall on the results of calculations;
- the error in the experiment will also affect the results through the error in calculating the total resistance, and in determining the Re number.

As a result of using the procedure described above, the thermal resistance of the contact zone was determined, which was \( R_c = 3.59 \times 10^{-4} \frac{m^2 \cdot K}{W} \). According to the classical formula for a cylindrical tube \( \varnothing 24.8 / 20.35 \text{ mm} \) for a thermal conductivity of the wall \( \lambda = 45 \frac{W}{m^2 \cdot K} \), then this value is \( 0.55 \times 10^{-4} \frac{m^2 \cdot K}{W} \), that is 7 times less.

In our case the maximal relative error of approximating \( k = k(Re) \) is equal less then 1%. So the relative error of \( R_{wall} \) will be approximately 30%.

Thus the fin-wall thermal resistance of experimental tube will be \( (3.59 \pm 1.08) \times 10^{-4} \frac{m^2 \cdot K}{W} \) with probability 95%.

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