INFLUENCE OF NONIDEAL PLATE CONTACT ON HEAT TRANSFER IN COMPACT HEAT EXCHANGERS

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INTRODUCTION

Heat exchangers consisting of two or more plates, one of which has coolant channels, are widely used for the thermal protection of power-station components [1, 2]. The heat transfer in such systems is characterized by unidirectional heating, the presence of heat transfer through the solid phase, and inhomogeneity of the heat extraction along the channel perimeter. These characteristics were analyzed in detail in [3], where the results of numerical investigation of conjugate heat transfer in the given systems were confirmed experimentally ($Pr \approx 1$, $Re = 5\cdot 10^3$-$2.5\cdot 10^4$). It may be shown that, in determining the integral heat-transfer characteristics in most practical cases, it is sufficient to solve only the heat-conduction equations for the solid phase with a uniform distribution of heat extraction in the channels. Investigation ($Pr \sim 7$, $Re = 10^2$-$4\cdot 10^4$) [2] on the whole confirms this conclusion. At the same time, the error of this approach in determining the temperature of the rear (with respect to the heat flux) plate in turbulent flow or the temperature of the whole heat exchanger in the laminar region has been established.

In the analysis in [1-3], ideal contact of the ribs with the walls was assumed, and attempts to realize this condition were made in the corresponding experiments. However, in real heat exchangers, this may not be the case. Usually, soldering is used for reliable connection of the heat-exchanger plate (or plates) to ribs of the cooling system. The minimum thermal resistance of the soldered joint is determined by its thickness $h_{\ell i}$ and the thermal conductivity of the solder λ_{li}: $R_T = h_{\ell i}/\lambda_{\ell i}$. A typical theoretical value of R_T is $(0.3$-$10)\cdot 10^{-6}$ $m^2\cdot K/W$ with a solder thickness of the order of 0.1 mm. However, in practice, R_T may be significantly higher because of pores, solder flaws, oxide films, etc. The analysis of the influence of the thermal resistance between the plates and ribs on the heat transfer in [4-6] is inapplicable in this case.

The aim of the present work is to investigate the influence of nonideality of contact between plates and ribs of the heat exchanger on the integral heat-transfer characteristics.

An elementary symmetry cell of the heat exchanger in the cross section transverse to the coolant flow is shown in Fig. 1. The heat insulation of the cell at $y = 0$ and $y = (\delta_r + \delta_{ch})/2$ reflects the symmetry condition. The experimentally confirmed one-dimensional model of heat transfer [2] is extended here to the case of thermal resistance between the frontal plate and the ribs and also between the ribs and the rear plate. In the analytical solution of the one-dimensional heat-conduction equation, as in [2], constant heat extraction from the vertical (rib) and horizontal sections of the channel (α and α_0, respectively) is assumed. The temperature distribution is determined from the equation

$$\frac{d^2\vartheta}{dx^2} - m^2\vartheta = 0,$$

where $m^2 = 0$ when $0 < x < h_1$; $m^2 = 2\alpha/\lambda\delta_r$ when $h_1 < x < h_2$; $m^2 = 0$ when $h_2 < x < H$.

The boundary conditions are clear from Fig. 1. Suppose that the thermal resistance is concentrated in the contact zone of the rib with the frontal and rear plates (i.e., in the section from $y = 0$ to $y = \delta_r/2$; Fig. 1) in a layer of infinitely small thickness. When $x = h_1$, $x = h_2$, the distribution of the temperature and heat flux undergoes a discontinuity. The temperature differences at the thermal resistances R_{T1}, T_{T2} are determined from the expressions

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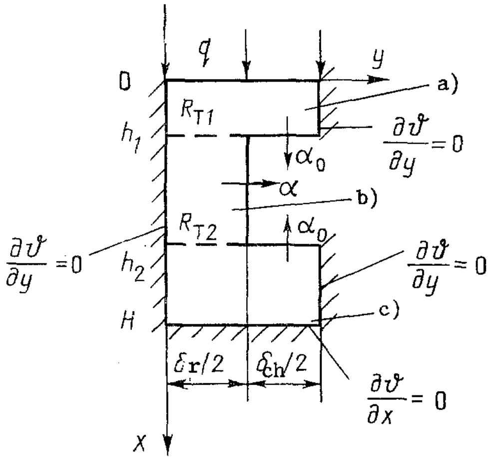

Fig. 1. Calculational cell of heat exchanger: a) frontal plate; b) rib; c) rear plate.

$$\vartheta(h_1^-)-\vartheta(h_1^+)=R_{T1}\left[-\lambda\frac{d\vartheta(h_1^+)}{dx}\right], \tag{1}$$

$$\vartheta(h_2^-)-\vartheta(h_2^+)=R_{T2}\left[-\lambda\frac{d\vartheta(h_2^-)}{dx}\right], \tag{2}$$

which are, in fact, matching conditions.

Under these assumptions

$$\Theta=\frac{\operatorname{ch}\psi_1}{1-\varepsilon+\varepsilon\alpha_0R_{T1}}\,\frac{\operatorname{ch}(mh+\psi_2)+\lambda mR_{T1}\operatorname{sh}(mh+\psi_2)}{\operatorname{sh}(mh+\psi_1+\psi_2)}+\frac{h_1-x}{m}, \tag{3}$$

$$0\leqslant x\leqslant h_1;$$

$$\Theta=\frac{\operatorname{ch}\psi_1}{1-\varepsilon+\varepsilon\alpha_0R_{T1}}\,\frac{\operatorname{ch}[m(h_2-x)+\psi_2]}{\operatorname{sh}(mh+\psi_1+\psi_2)},\ h_1^+\leqslant x\leqslant h_2^-; \tag{4}$$

$$\Theta=\frac{\operatorname{ch}\psi_1}{1-\varepsilon+\varepsilon\alpha_0R_{T1}}\,\frac{\operatorname{ch}\psi_2-R_{T2}\lambda m\operatorname{sh}\psi_2}{\operatorname{sh}(mh+\psi_1+\psi_2)},\ h_2^+\leqslant x\leqslant H, \tag{5}$$

where $\Theta = \vartheta\lambda m/q = (t - t_w)\lambda m/q$ is the dimensionless temperature; $\varepsilon = \delta_{ch}/(\delta_{ch} + \delta_r)$; $h = h_2 - h_1$; $m = \sqrt{2\alpha/(\lambda\delta_r)}$; $\operatorname{th}\psi_i = \varepsilon/(1 - \varepsilon + \varepsilon\alpha_0R_{Ti})\ (\alpha_0/\lambda m)$ $(i = 1, 2)$.

The reduced heat-transfer coefficient $\alpha_{re} = q/\vartheta(h_1^-)$ actually characterizing the temperature level in the frontal plate is

$$\alpha_{re}=\varepsilon\alpha_0+\frac{(1-\varepsilon)\lambda m\operatorname{th}(mh+\psi_2)}{1+\lambda m R_{T1}\operatorname{th}(mh+\psi_2)}. \tag{6}$$

The heat transfer is investigated experimentally by the method of [2]. The samples investigated consist of two 110 × 30 mm plates soldered together (overall thickness 5-8 mm), i.e., there is only one thermal resistance R_T in the samples. Rectangular channels are cut in one of the plates (the ribbed plate) for water cooling of the samples (Pr ~ 7). In contrast to [2], the heating is alternately from the ribbed and the unribbed plates. plates.

Experimental data on the reduced heat transfer as a function of the Reynolds number are given in Fig. 2 for various samples whose characteristics are given in Table 1. The Reynolds number is determined from the velocity of water in the channels, the hydraulic diameter of the channels d_h, and the mean water temperature over the sample length. For samples 1, 2 and 3, 4, two different highly conducting solders with a theoretical thermal resistance of no more than $0.5\cdot10^{-6}$ $m^2\cdot K/W$ are used.

It follows from Fig. 2 that, for samples 1-3 (Table 1), the discrepancy in α_{re} for heating from the ribbed and unribbed plates amounts to 25-45%. The difference in α_{re} for the same sample on heating from different sides has a unique interpretation: that the influence of the solder joint is felt on heating from the unribbed plate.

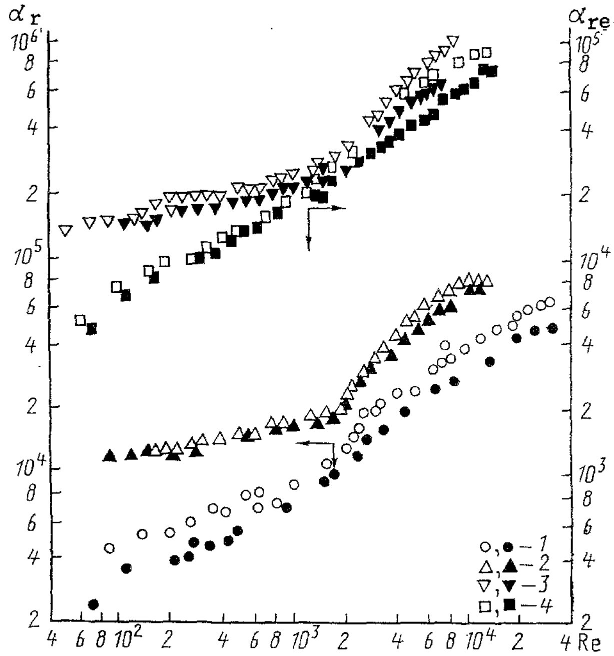

Fig. 2. Dependence of the reduced heat-transfer coefficient α_{re}, W/m²·K, on the Reynolds number: 1-4) experimental data for samples 1-4, respectively (Table 1); open symbols) heating from the ribbed plate; filled symbols) from the unribbed plate.

TABLE 1. Characteristics of Samples

Sample No.	λ, W/m·K	$h\cdot10^3$, m	$\delta_r\cdot10^3$, m	$\delta_{ch}\cdot10^3$, m	$d_r\cdot10^3$, m	ε
1	380	3,00	2,00	2,00	2,40	0,50
2	380	2,60	1,00	0,80	1,22	0,44
3	130	3,05	0,53	0,65	1,05	0,55
4	130	2,57	0,93	0,70	1,10	0,43

In the analytical model adopted, R_T may be determined from experimental data

$$R_T = \frac{\alpha_{re}(R_T = 0) - \alpha_{re}(R_T)}{\lambda m \operatorname{th}(mh + \varphi)\,[\alpha_{re}(R_T) - \varepsilon\alpha_0]}, \qquad (7)$$

where $\alpha_{re}(R_T = 0)$ is the reduced heat transfer in the heat exchanger with no thermal resistance; $\alpha_{re}(R_T)$ is the reduced heat transfer with heating from the unribbed plate; $\operatorname{th}\phi = \varepsilon\alpha_0/[(1-\varepsilon)\lambda m]$. With an error of 2-3%, it may be assumed that $\alpha_{re}(R_T = 0)$ in the present experiments is the reduced heat-transfer coefficient with heating from the ribbed plate. The values of R_T calculated in this way for samples 2-4 are shown in Fig. 3 as a function of the Reynolds number. With turbulent flow, as is evident from Fig. 3, R_T remains constant to an error of ±15%, which indicates, to a certain extent, the reliability of the chosen model. The increase in the error of R_T in the laminar region is associated, on the one hand, with the influence of unidirectional heating, when even a model without thermal resistance gives significant error [2], and, on the other, with the inaccuracy in calculating α_0 in the laminar region.

Note that the absolute thermal resistance of the given samples is $(2.5\text{-}7.0)\cdot10^{-6}$ m²·K/W, which is 14 times higher than the thermal resistance of solder material of the same thickness. Direct measurements of R_T on special cylindrical copper samples show that the thermal resistance of the joint produced with the same solder and by the same technology as in cooled samples 1 and 2 is $(3\text{-}5)\cdot10^{-6}$ m²·K/W. The practical agreement of the values of R_T obtained in direct and indirect measurements using the analytical model is further indication of the correctness of the model, taking account of the presence of thermal resistance.

The influence of thermal resistance on the reduced heat transfer is now analyzed. Sup-

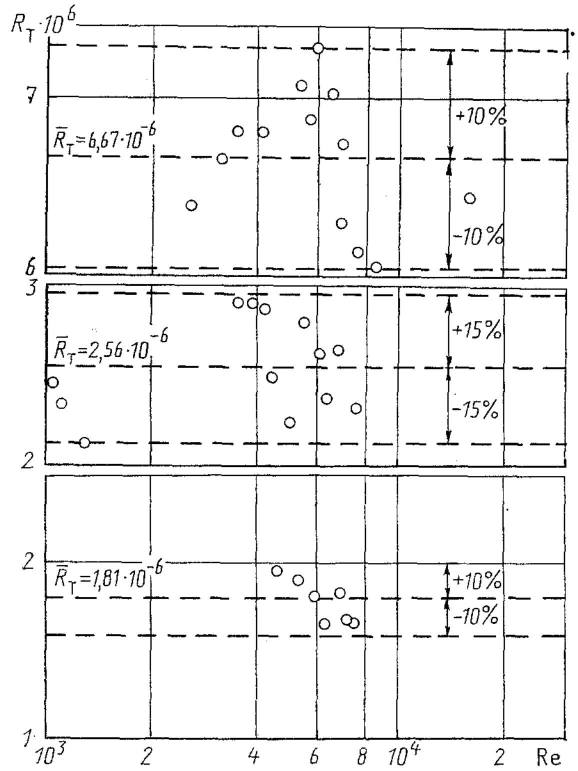

Fig. 3. Thermal resistance R_T ($m^2 \cdot K/W$) at various Reynolds numbers for samples 2-4 (reading downward).

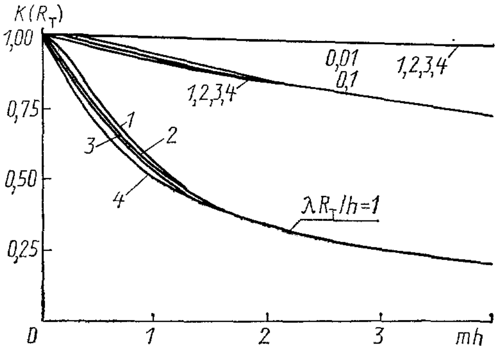

Fig. 4. Dependence of $K(R_T)$ on the dimensionless heat transfer mh with various values of $\lambda R_T/h$ and ϕ = 0.25 (1), 0.50 (2), 0.75 (3), and 1.0 (4).

pose that there is only one thermal resistance, and heating is from the unribbed plate, i.e., from the solder. In Fig. 4, $K(R_T)$ is shown as a function of the rate of heat extraction from the ribs mh and the dimensionless complex $\lambda R_T/h$ characterizing the ratio of thermal resistances of the solder joint and the rib. Here

$$K(R_T) = \frac{\alpha_{re}(R_T) - \varepsilon\alpha_0}{\alpha_{re}(R_T = 0) - \varepsilon\alpha_0} \quad (8)$$

and is numerically equal to the ratio of the heat extraction from the frontal plate through the ribs with the actual and zero thermal resistances. When $R_{T1} = R_T$, $R_{T2} = 0$, it follows readily from Eq. (6) that

$$K(R_T) = [1 + (\lambda R_T/h)(mh)\,\mathrm{th}\,(mh + \varphi)]^{-1}. \quad (9)$$

It is evident from Fig. 4 that, when $\lambda R_T/h < 0.01$, the influence of thermal resistance on the heat transfer is practically nonexistent in the most common practical range of mh. Further increase in $\lambda R_T/h$ leads to sharp decrease in $K(R_T)$ at small mh and less sharp varia-

TABLE 2. $K(R_T)$ as a Function of mh, $\lambda R_T/h$, and ϕ ($R_{T1} = 0$, $R_{T2} = R_T$)

mh	$\lambda R_T/h$	φ: 0,2	0,4	0,6	0,8	1,0	1,2
0,2	0,01	1,000	1,000	0,999	0,999	0,999	0,999
	0,1	0,998	0,996	0,993	0,991	0,990	0,989
	1	0,982	0,958	0,938	0,922	0,911	0,903
0,4	0,01	1,000	0,999	0,999	0,999	0,999	0,999
	0,1	0,998	0,994	0,991	0,989	0,987	0,986
	1	0,980	0,950	0,924	0,905	0,891	0,881
0,6	0,01	1,000	0,999	0,999	0,999	0,999	0,999
	0,1	0,998	0,995	0,991	0,989	0,987	0,986
	1	0,982	0,953	0,929	0,911	0,898	0,889
0,8	0,01	1,000	1,000	0,999	0,999	0,999	0,999
	0,1	0,998	0 995	0,992	0,990	0,988	0,987
	1	0,984	0,960	0,940	0,925	0,914	0,907
1,0	0,01	1,000	1,000	0,999	0,999	0,999	0,999
	0,1	0,999	0,996	0,994	0,992	0,990	0,989
	1	0,987	0,968	0,952	0,940	0,932	0,926
1,5	0,01	1,000	1,000	1,000	1,000	0,999	0,999
	0,1	0,999	0,998	0,997	0,996	0,995	0,994
	1	0,994	0,984	0,976	0,971	0,967	0,964
2,0	0,01	1,000	1,000	1,000	1,000	1,000	1,000
	0,1	1,000	0,999	0,998	0,998	0,998	0,997
	1	0,997	0,993	0,990	0,987	0,985	0,984
2,5	0,01	1,000	1,000	1,000	1,000	1,000	1,000
	0,1	1,000	1,000	0,999	0,999	0,999	0,999
	1	0,999	0,997	0,996	0,995	0,994	0,994
3,0	0,01	1,000	1,000	1,000	1,000	1,000	1,000
	0,1	1,000	1,000	1,000	1,000	1,000	0,999
	1	0,999	0,999	0,998	0,998	0,998	0,997
4,0	0,01	1,000	1,000	1,000	1,000	1,000	1,000
	0,1	1,000	1,000	1,000	1,000	1,000	1,000
	1	1,000	1,000	1,000	1,000	1,000	1,000

tion in $K(R_T)$ when mh > 2. The heat transfer from the unribbed surface α_0 and the porosity ε appearing in ϕ have little influence on $K(R_T)$.

Thus, the thermal resistance has the greatest influence in heat exchangers made of heat-conducting materials with high ribs. In practice (mh < 4-5), the thermal resistance of the solder joint may be disregarded in heating from the unribbed plate if it is no more than 1% of the rib thermal resistance h/λ. For highly conducting materials (λ = 100-400 W/m·K) with a rib height $h = (1\text{-}4)\cdot10^{-3}$ m, the thermal resistance of the solder joint must be no more than $(0.25\text{-}4)\cdot10^{-7}$ $m^2\cdot K/W$, i.e., must be comparable with the thermal resistance of the solder material itself. This imposes increased requirements on the solder-joint quality, from the viewpoint of both strength and thermal properties.

The requirements on the thermal resistance of the solder joint may be significantly reduced by changing the heat-exchanger construction so that heating is from the ribbed plate ($R_{T1} = 0$, $R_{T2} = R_T$). Table 2 gives the results of calculating $K(R_T)$ as a function of mh, $\lambda R_T/h$, and ϕ. In this case, taking account of Eq. (6), $K(R_T)$ may be calculated from the formula

$$K(R_T) = \frac{\operatorname{th}(mh + \psi_2)}{\operatorname{th}(mh + \varphi)}.$$

It follows from Table 2 that, in the characteristic range of the determining parameters for most heat exchangers, heating from the ribbed plate permits practical elimination of the influence of the solder-joint thermal resistance. It is interesting to note that, with increase in cooling intensity mh, $K(R_T)$ initially decreases, since the heat extraction from the unribbed part of the rear plate is reduced. With further increase in mh, the temperature difference over the height of the rib increases, and the boundary condition at the lower end of the rib approaches the heat-insulation condition, in which the presence of thermal resistance may be disregarded.

CONCLUSION

The experimental data obtained here and the corresponding analysis indicate the need to take account of thermal resistance in the contact zone of plates with the cooling system of the heat exchanger. This applies, above all, to heat exchangers of highly conducting material, in which the heating is from the unribbed plate, or multilayer heat exchangers. In these cases, stricter requirements must be imposed to ensure high-quality thermal contact of the plates with the cooling system.

NOTATION

a, thermal diffusivity; d_h, hydraulic diameter of channel; h_1, h_2, characteristic dimensions of heat-exchanger cell (Fig. 1); H, Heat-exchanger thickness; $K(R_T)$, coefficient defined in Eq. (8); $m = \sqrt{2\alpha/(\lambda\delta_r)}$; q, heat-flux density at heat-exchanger surface; R_{T1}, R_{T2}, thermal resistances at $x = h_1$, $x = h_2$; u, flow rate of water in channel; x, coordinate over the heat-exchanger thickness; α_0, heat-transfer coefficient from unribbed surface of channel; α, heat-transfer coefficient from ribbed surface of channel (from rib); $\alpha_{re} = q/\vartheta(h_1^-)$, reduced heat-transfer coefficient; δ_{ch}, δ_r, channel and rib width; $\varepsilon = \delta_{ch}/(\delta_{ch} + \delta_r)$; λ_{re}, λ, thermal conductivity of solder and heat-exchanger materials; $\vartheta = (t - t_w)$, excess temperature of heat exchanger with respect to water; $\Theta = \vartheta\lambda m/q$, dimensionless temperature; $\psi_i = \operatorname{arcth}(\varepsilon\alpha_0/\{[(1 - \varepsilon) + \varepsilon\alpha_0 R_{Ti}]\lambda m\})$, dimensionless complex ($i = 1, 2$); $Pr = \nu/a$, Prandtl number; $Re = ud_h/\nu$, Reynolds number.

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INFLUENCE OF NONAXISYMMETRIC TEMPERATURE PROFILE IN LAYER OF DIFFERENT-TEMPERATURE COMPONENTS ON RADIANT HEAT TRANSFER

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On the basis of analysis of the radiant heat transfer in a layer of different-temperature components with a nonaxisymmetric temperature profile, the influence of the content of low-temperature components, the optical thickness of the layer, and the temperature profile on the resulting radiation flux is investigated. Practical recommendations for the design of steam-boiler furnaces are given.

The increasing utilization of the considerable reserves of young lignite, which contains more ash and moisture and tends to produce more slag, has a lower heat of combustion, is dangerously prone to explosion in the finely disperse state in air, and increases the ecological and size demands on steam boilers, has intensified the development and industrial introduction of new ignition methods and new furnaces, including low-temperature vortical (LTV) furnaces, furnaces with a circulating fluidized bed (CFB), flame-layer (FL) furnaces, etc. As shown in [1], the characteristics of ignition in such furnaces lead to the existence

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