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HEAT TRANSFER AND HYDRAULIC RESISTANCE IN CHANNEL COOLING SYSTEMS WITH A DISCONTINUOUS WALL

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The authors give results of measuring the hydraulic resistance, heat transfer, and temperature fields at the numbers Re = 70–3·10⁴ and Pr = 5.5–8 for cooling systems with rectangular channels with different kinds of discontinuity of their walls involving additional channels that act as intensifiers of heat transfer. The regions of Re numbers in which this intensification is energy-profitable are revealed.

Heat transfer in flow cooling systems of channel type with a small hydraulic diameter $d_h \le 1$ mm cannot be intensified efficiently in the region of developed turbulent flow as applied to flat-tube compact heat exchangers, since because of limitations of the pressure difference on the cooling system of a structure the working region of Reynolds numbers is in the range Re = 500–5000, which basically corresponds to laminar and transient regimes of liquid flow. Therefore the most acceptable measures to improve heat transfer can be methods that are based on organization and destruction of the flow structure. The use of contracted channels produces flow with unformed boundary layers [1]. The introduction of inhomogeneities into the flow, by acting on the shape of the channels, destroys the boundary layers, in which a major portion of the thermal resistance to heat transfer is concentrated. The action of pressure nonuniformities (contractor-diffuser-type cross section of the channel) is an efficient method of acting on the boundary layer [2, 3]. The introduction of inhomogeneities into the flow in the form of joint actions of a cross flow, discontinuities of the heat-exchange surface, rotations of the flow and its separation due to a contractor-diffuser flow led to an increase of 1.8–2 times in heat transfer with a simultaneous increase of 2.5–3 times in resistance.

The methods for intensifying heat transfer can be different depending on the objectives. With a limitation on the power of liquid pumping, energy-profitable intensification is the most appropriate ($\eta = \overline{\mathrm{Nu}}/\overline{\xi} \ge 1$). Therefore, under conditions of simple and technologically developed methods of formation of heat-exchange surfaces with small hydraulic diameters (for example, using milling or electric- erosion machining) and specific forms of their thermal loading (for example, a unilateral form in laser mirrors and other thermal-protection devices), realization of an energy-profitable method for intensifying heat transfer is, rather, an optimization problem.

Earlier we investigated a wafer cooling system [4, 5] that was a structure produced by forming, in a material, banks of channels of the same height that intersect at an angle ($45^o \le \varphi \le 90^o$). We studied the case where the steps of the formed fins in the transverse S_1 and longitudinal S_2 directions were approximately the same. The linear dimensions of a fin and a channel were also the same so that the fins either had the shape of a square ($\varphi = 90^o$) or were diamond-shaped ($\varphi < 90^o$) (in plan). The hydrodynamics of liquid flow in these systems differs fundamentally from the hydrodynamics of flow in rectangular channels [4]. In wafer systems when the hydraulic diameters of the channels are the same as in a channel system, heat transfer can be increased by a factor of 2-3 with a simultaneous resistance increase of 10–18 times [4, 5].

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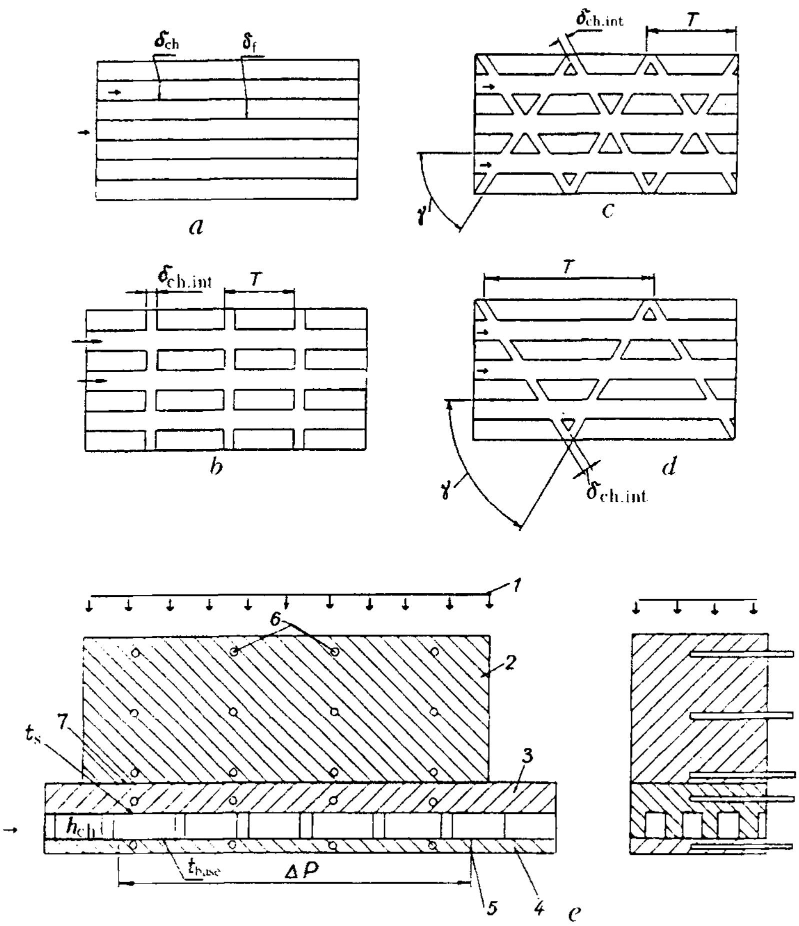

Fig. 1. View (plan) of systems and experimental scheme: a) channel system; b) channel system with transverse slots; c) channel system with tilted slots ("fishbone"); d) channel system with tilted slots ("zigzag"); e) scheme of an experimental portion [1) ohmic heater; 2) thermal wedge; 3) monolithic plate with a cooling system; 4) connected plate; 5) pressure tap; 6) thermocouples; 7) indium–gallium eutectic].

In this work, we propose a structure of a cooling system of a flat-tube heat exchanger that is different from a wafer cooling system and combines short channels with the production of pressure nonuniformities. Structurally, it is made by introducing a system of additional channels into a channel cooling system; the additional channels are arranged with a step T at angles $\pm\gamma$ to the main channels. The main channels acquire dissimilar lengths, and due to the dissimilar lengths of neighboring fins in the direction of the main stream a pressure gradient appears that produces a flow in the additional channels and facilitates efficient destruction of the boundary layer and earlier onset of the transition stage of the flow. We call this system a modified channel system. In it, heat transfer must be increased because of periodic efficient destruction of the temperature boundary layer so that heat transfer occurs on the initial thermal portion. Here, the term efficient means an action on the flow structure that would be accompanied by an insignificant increase in the hydraulic resistance. The periodicity of the action on the structure along the stream, according to recommendations of Kalinin et al. [6], must be approximately 10 hydraulic diameters of the channel.

Here we present results of an experimental investigation of the hydraulic resistance and heat transfer in several types of modified channel cooling systems at $Re = 70\text{–}3\cdot10^4$ and $Pr = 5.5\text{–}8$.

Thermohydraulic characteristics were studied on special models manufactured of copper (λ = 380 W/(m·K)) and consisting of two soldered plates. In one of the plates, a cooling system is formed (Fig. 1), the

TABLE 1. Characteristics of the Investigated Cooling Systems with a Discontinuous Wall

No. of model	Type of cooling system	δ_{ch}, h_{ch}, mm	d_h, mm	δ_f, mm	ε	Note
1	Channel with transverse slots (Fig. 1b)	1.0; 2.68	1.457	1.0	0.5	$\delta_{ch.int}$ = 1.1 mm; T = 19 mm
2	Channel with tilted slots ("fishbone," Fig. 1c)	1.0; 2.69	1.462	1.0	0.5	$\delta_{ch.int}$ = 2.0 mm; T = 36.5 mm; $\gamma = 38^o$
3	Channel (Fig. 1a)	1.0; 2.68	1.456	1.0	0.5	Indicator specimen
4	Channel with tilted slots ("zigzag," Fig. 1d)	1.0; 2.60	1.444	1.04	0.49	$\delta_{ch.int}$ = 1.1 mm; T = 26 mm; $\gamma = 66^o$
5	Channel with tilted slots ("fishbone," Fig. 1c)	1.0; 2.68	1.457	1.0	0.5	$\delta_{ch.int}$ = 1.0 mm; T =36 mm; $\gamma = 40^o$
6	Channel with tilted slots ("zigzag," Fig. 1d)	1.0; 3.0	2.4	2.0	0.5	$\delta_{ch.int}$ = 2.0 mm; T = 43 mm; $\gamma = 55^o$

channels are produced by milling, and the burrs are removed. The plates are closed on the lateral surfaces by covers. The length, width, and height of the model were, respectively, 110, 30, and 11 mm. The geometry of the models and their characteristics are given in Table 1. The height of the additional (intermediate) channels is equal to the height of the main channels. Here the porosity was calculated as just the ratio of the free volume produced by the main channels to the entire volume of the cooling system. Each model is drained by 6–8 holes 0.8–1 mm in diameter for tapping the static pressure. The taps are located in both the main and auxiliary channels, which made it possible to determine pressure differences along and cross the channels. The model was glued into flanges and was fastened to supply and discharge collectors. The conditions at the inlet to the model corresponded to the conditions of discharge from a large volume and ensured a uniform velocity profile at the inlet. The procedure of a thermohydraulic experiment is described in [5, 6] in detail. The averaged hydraulic resistance was calculated from the measured pressure gradients Δp_i on the model lengths Δl_i and the flow rate G of the liquid, for which we used tap water at room temperature (t_{liq} = 10–25°C). In thermal experiments, we employed the thermal-wedge method. Heating was performed by an ohmic heater installed in a thermal wedge that was made in the form of an 80 × 50 × 30 mm copper prism (the wedge was arranged with its longest side along the model (Fig. 1)). Reliable thermal contact between the heater and the model was attained by appropriate treatment, fitting, and tightening of the surfaces and by placing an indium–gallium liquid-metal eutectic between them. To eliminate the effect of the thermal resistance of the soldered joint of the finned and unfinned plates on the integral heat-transfer characteristics [7], we performed the heating on the side opposite the soldering (on the side of the finned plate). The heat flux was measured in four cross sections along the heater length using twelve Chromel–Copel thermocouples and in four cross sections in the model. The temperature of the cooled surface t_s was determined by extrapolation of its measured distribution in the heater and in the model, and for monitoring the temperature of the model's rear surface t_{base}, four Chromel–Copel thermocouples were calked into it. The distribution of the heating of the water along the length of the heated portion of the model was taken to be linear. The structure of the thermal wedge ensured a uniform heat flux q in the measurement cross sections that could be varied in the range $q = (0.5–5)\cdot 10^5$ W/m^2. Under a change in the flow rate of the water we determined experimentally the reduced coefficient of heat transfer α_{red} ($\alpha_{red} = q/(t_s - t_{liq})$, t_{liq} is the mass-mean temperature of the liquid in the cross section) and the coefficient of "heat insulation" $R_{h.ins}$ of the lower plate ($R_{h.ins} = (t_{base} - t_{liq})/(t_s - t_{liq})$, t_{base} is the temperature of the lower, heat-insulated plate). The quantities α_{red} and $R_{h.ins}$ characterize the temperature field of the model.

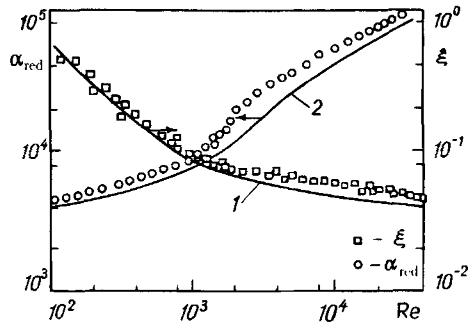

Fig. 2. Thermohydraulic characteristics of model 6 vs. Reynolds number: 1 and 2) hydraulic resistance and reduced heat transfer, respectively, of a cooling system without additional channels. α_{red}, W/(m^2·K).

TABLE 2. Generalization of Experimental Results on the Hydraulic Resistance and Heat Transfer of the Modified Channel Cooling Systems

No. of model	Hydraulic resistance			Heat transfer				
	Re	approximation		Re	α_{red} approximation		α_0 approximation	
		C_1	n_1		C_2	n_2	C_3	n_3
1	$(3–6.5)\cdot10^2$	82	–1	$(0.9–10)\cdot10^2$	3160	0.181	858	0.188
	$(5.6–15)\cdot10^2$	16.8	–0.75	$(8.0–37)\cdot10^2$	16.8	0.96	2.52	1.1
	$(1.5–20)\cdot10^3$	0.2	–0.132	$(3.7–16)\cdot10^3$	766	0.49	88	0.62
2	$(3–6.3)\cdot10^2$	101	–1	$(1–10)\cdot10^2$	861	0.46	214	0.49
	$(5.5–23)\cdot10^2$	3.3	–0.47	$(7.3–35)\cdot10^2$	960	0.45	198	0.51
	$(2.3–21)\cdot10^3$	0.33	–0.18	$(2.6–20)\cdot10^3$	873	0.47	116	0.57
3	$(1–10)\cdot10^2$	82.3	–1	$(1–20)\cdot10^2$	2280	0.26	602	0.275
	$(1–30)\cdot10^3$	0.37	–0.212	$(1.7–4.0)\cdot10^2$	0.942	1.3	0.085	1.46
				$(4–17)\cdot10^3$	289	0.614	20.7	0.8
4	$(0.8–2.25)\cdot10^2$	187.3	–1	$(1–10)\cdot10^2$	1832	0.31	502	0.32
	$(2.25–8)\cdot10^2$	36.3	–0.834	$(1–2.9)\cdot10^3$	54.4	0.82	9.4	0.9
	$(8–28)\cdot10^2$	1.74	–0.38	$(2.9–15)\cdot10^3$	590	0.523	73.2	0.65
	$(2.8–20)\cdot10^3$	0.16	–0.08					
5	$(0.75–4.6)\cdot10^2$	104	–1	$(1–9.5)\cdot10^2$	2567	0.263	680	0.276
	$(4–20)\cdot10^2$	7.8	–0.59	$(7.3–35)\cdot10^2$	38.2	0.892	5.7	0.99
	$(2–14)\cdot10^3$	0.26	–0.134	$(2.4–15)\cdot10^3$	1100	0.456	142	0.57
6	$(5–23)\cdot10^2$	0.223	–0.142	$(0.7–9.5)\cdot10^2$	1364	0.265	544.7	0.271
	$(2.3–36)\cdot10^2$	0.327	–0.189	$(9.4–30)\cdot10^2$	10.8	0.965	2.96	1.025
				$(3.0–26)\cdot10^3$	383.7	0.533	68.1	0.647

Simultaneously α_{red} is a characteristic of the effective heat transfer [4, 5] and, along with the surface heat transfer of the unfinned portion of the model (the averaged coefficient of heat transfer on this surface α_0), allows for the contribution of the prismatic fins (the averaged coefficient of heat transfer from the fins α) and the lower plate to it. In calculations of the thermally deformed state of a laser mirror (or other thermal-protection device) equipped with these cooling systems, the use of α_{red} and $R_{h.ins}$ unambiguously describes a one-dimensional temperature field along the height of the skeleton of the cooling system. This proposition is substantiated in detail in [5].

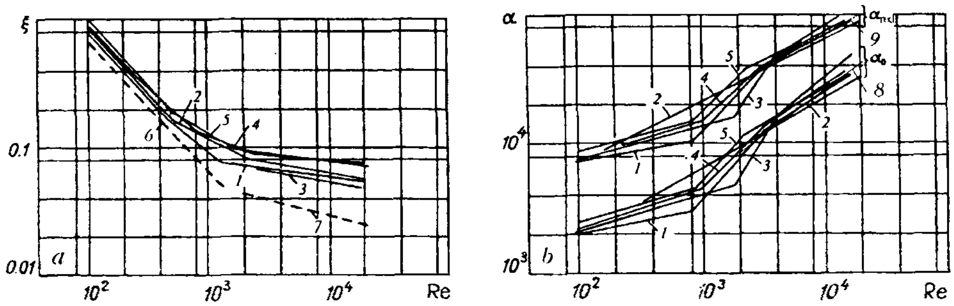

Fig. 3. Thermohydraulic characteristics vs. Reynolds number: a) coefficient of hydraulic resistance ξ (curves 1-5 correspond to the numbers of the models, curve 6 corresponds to $\xi = 66.6/\mathrm{Re}$, i.e., the hydraulic resistance of a prismatic channel in the laminar region of flow, curve 7 corresponds to $\xi = 0.316/\mathrm{Re}^{0.25}$, i.e., the hydraulic resistance of a prismatic channel in the turbulent region of flow); b) reduced coefficient of heat transfer α_{red}, W/(m^2·K), and the coefficient of heat transfer on the unfinned portion of the plate α, W/(m^2·K) (curves 1-5 correspond to the numbers of the models, curve 8 corresponds to the surface heat transfer in a prismatic channel in turbulent flow, curve 9 is the calculated reduced heat transfer of a prismatic channel obtained from curve 8).

In generalizing the results, we employed the Reynolds number Re, determined from the hydraulic diameter of the main channel d_h and the mass-mean velocity of the liquid in the channel w, and the coefficient of hydraulic resistance $\xi = \Delta p_i/[\rho w^2/2(d_h/\Delta l_i)]$. The maximum errors in determining the quantities did not exceed 10–12% for ξ (when $\mathrm{Re} > 500$), 3–6% for q, 10–15% for α_{red}, 4–7% for Re, and 5–25% for $R_{h.ins}$ depending on the flow regime (indirect values of measurements are determined with a confidence probability of 0.95).

Under the assumption of approximate equality of the averaged coefficients of surface heat transfer on a fin α and on unfinned surfaces α_0 in the channel system and based on experimental data on $\alpha_{red}(\mathrm{Re})$ and $R_{h.ins}(\mathrm{Re})$ and an analytical solution for a one-dimensional temperature field across the system's thickness [8] by the method presented in [5], we identified $\alpha_0 \approx \alpha$. The characteristic form of the obtained dependences is given in Fig. 2 for model 6. On this model in the beginning of the work the authors obtained higher α_{red} than in a similar channel cooling system [8] by a factor of 1.1–1.3 with a simultaneous increase of 10–15% in hydraulic losses thus revealing the efficiency of the method proposed for intensifying heat transfer. Models 1-5 were manufactured for further study of different versitions of introduction of "intensifying" additional channels and their dimensions and arrangement relative to the main channels. In this group, model 3 was an indicator specimen, in which there were no additional channels and which was a standard for revealing the intensifying effect in modified systems 1, 2, 4, and 5.

The obtained results on ξ, α_{red}, and α_0 for this group of models were generalized on segments of Re by power functions using the least-squares method and are given in Table 2 (without standard deviations or variances) and Fig. 3. Table 2 indicates the ranges of approximation of the coefficient of hydraulic resistance and the reduced and surface heat transfer, respectively, by the power functions $\xi = C_1\mathrm{Re}^{n_1}$, $\alpha_{red} = C_2\mathrm{Re}^{n_2}$, and $\alpha_0 = C_3\mathrm{Re}^{n_3}$ (where C_1, C_2, and C_3 are constants). The intervals of approximation over the Re number either go in succession or overlap each other somewhat.

In the purely laminar region of flow (which was located in the region $\mathrm{Re} < 250$ for all the models), the resistance is somewhat higher ($C_i = 80$–100) than the theoretical resistance $\xi = C/\mathrm{Re}$ ($C = 66.6$) for rectangular channels with a ratio of the sides $h_{ch}/\delta_{ch} = 2.7$, which is attributable to: a) the presence of the discontinuity in the channel along the length and the appearance of additional losses due to this; b) the previously established factor [9] of an increase in hydraulic resistance in the laminar region of flow in narrow channels with a small hydraulic diameter ($d_h \approx 1$ mm). The beginning of a deviation from the law $\xi \approx C_i/\mathrm{Re}$ differs in Re number for

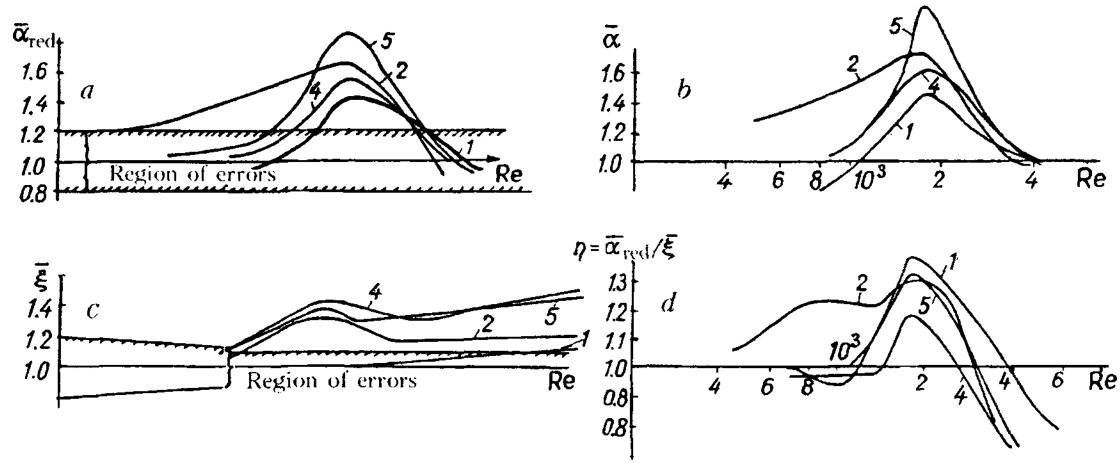

Fig. 4. Relative thermohydraulic characteristics of modified channel systems vs. Re number: a) relative reduced coefficient of heat transfer $\bar{\alpha}_{red} = \alpha_{red}/\alpha_{red.ind}$; b) relative surface coefficient of heat transfer $\bar{\alpha} = \alpha/\alpha_{ind}$; c) relative coefficient of hydraulic resistance $\bar{\xi} = \xi/\xi_{ind}$; d) relative energy efficiency $\eta = \bar{\alpha}_{red}/\bar{\xi}$.

different models. It is detected earliest in model 4 (Re = 250) and manifests itself at Re ≥ 700 for all the models. The range of the portion of transition flow differs in Reynolds number ($1400 \leq \mathrm{Re} \leq 2500$) but at Re = 2500 the resistance law $\xi = B_i \mathrm{Re}^{n_i}$ (where B_i is a constant, $n_i = -0.08$–0.212), which is characteristic of turbulent flow, is established. The level of resistance in this region for the indicator specimen is somewhat higher than the theoretical one (and is, apparently, associated with the roughness inherent in the milling and the solder in jointing and the deviation of the shape of the channel from rectangular). To reveal the effect of the additional channels on the resistance and the heat transfer, comparison is made in relation to the indicator specimen; in doing so, the nonidealities of the model fabrication technology are accounted for.

The behavior of the dependence $\alpha_{red} = f(\mathrm{Re})$ (Fig. 3b) correlates well with the plot $\xi = f(\mathrm{Re})$ (Fig. 3a) in Re numbers. The strongest differences of heat transfer in different models are observed in the region Re = 700–3500, which corresponds to transition Re numbers and the initial region of turbulent flow. In the laminar region (Re < 250), the deviations do not exceed the errors; in the region $250 \leq \mathrm{Re} \leq 700$, the heat transfer is somewhat intensified. In the region Re ≥ 3500, the heat transfer of all the models is approximately the same and the intensifying effect degenerates. The relative (to the indicator model) increase in the reduced heat transfer and the resistance in the region $700 \leq \mathrm{Re} \leq 4000$ for modified channel systems 1, 2, 4, and 5 is given in Fig. 4a-c. The curves have an extremum $\bar{\xi} = 1.3$–1.45 and $\bar{\alpha}_{red} = 1.42$–$1.85$ (Re = 1500–1800) that is associated with the transition to turbulent flow in the indicator model and leads to an increase in the resistance and heat transfer in it. Singling out the surface heat transfer (with the condition $\alpha \approx \alpha_0$) enabled us to determine the relative increase in heat transfer for this region of Re (Fig. 4b), which attains a maximum $\bar{\alpha}$ = 1.4–2.0 at Re = 1700–1800. The strongest increase in $\bar{\alpha}_{red}$ and $\bar{\alpha}$ is noted in models 2 and 5 ($\bar{\alpha}_{red}$ = 1.65 and 1.85, $\bar{\alpha}$ = 1.72 and 2.02, respectively). As applied to the given problem the criterion of energy efficiency of the heat-exchange surface $\eta = (\bar{\alpha}_{red}/\bar{\xi})$ for [Re = idem] is determined as a characteristic of the efficiency of intensification of heat transfer in a flat-tube heat exchanger under conditions of a unilateral heat flux (Fig. 4d). In the determined range of Reynolds numbers ($1400 \leq \mathrm{Re} \leq 2600$), all the modified channel systems are energy-profitable. According to the maximum η_m, we arranged the models in order: 1, η_m = 1.38 (Re = 1800); 5, η_m = 1.32 (Re ~ 1700–1800); 2, η_m = 1.3 (Re ~ 1700–2000); 4, η_m = 1.18 (Re = 1800). The range of the region $\eta \geq 1$ differs for individual models but at Re > 4000 (in the region of developed turbulent flow) $\eta \leq 1$, i.e., the intensification is inefficient. But in this region, too, there is a positive aspect of intensification that is associated with a decrease of 1.5–2.0 times in the coefficient of heat insulation $R_{h.ins}$ in the modified systems as compared to the ordinary channel system, i.e., the heating of the main structure decreases substantially. To determine the efficiency of the proposed intensification method as applied to other problems of heat transfer, the criterion of energy efficiency must be represented in the form of the ratio $\eta_1 = \alpha/\xi$. The behavior of this criterion for modified systems is similar to that of the criterion η but the region of energy-profitable Re numbers expands somewhat and the maximum values of η_1 increase by 20–25%.

In the authors' opinion, the method proposed for intensifying heat transfer is promising since, with the labor content of the structure being increased by 8–10%, the method makes it possible to attain energy-profitable intensification (with a 30–65% excess of heat transfer over hydraulic resistance) in a wide range of flow velocities of the heat-transfer agent.

NOTATION

d_h, hydraulic diameter of the channel, $d_h = 4F_{ch}/\Pi_{ch}$; F_{ch} and Π_{ch}, area and perimeter of the channel; δ_{ch} and h_{ch}, width and height of the channel; $\delta_{ch.int}$, width of the intermediate channel; γ, intersection angle of the channels; δ_f, fin thickness; T, step of the channel in the transverse (S_1) and longitudinal (S_2) directions; ε, porosity, i.e., ratio of the void volume to the total volume of the unit cell of the cooling system; Δp_i, pressure difference for the distance Δl_i between the pressure taps; G, flow rate; w, mass-mean velocity of the flow; ξ, coefficient of hydraulic resistance; α_{red}, α, and α_0, coefficient of heat transfer (reduced, on a fin, on the unfinned portion of the plate); q, specific heat flux; t_s, t_{liq}, and t_{base}, temperature (of the surface, the liquid, and the base); ρ and μ_{liq}, density and dynamic viscosity of the liquid; λ and λ_{liq}, thermal conductivities of the material and the liquid; $Re = d_h w/\mu_{liq}$, Reynolds number; Pr, Prandtl number; $Nu = \alpha d_h/\lambda_{liq}$, Nusselt number; η and η_1, criterion of energy efficiency; $R_{h.ins}$, coefficient of heat insulation. Subscripts: liq, liquid; s, surface; red, reduced; h.ins, heat insulation; h, hydraulic; ch, channel; f, fin; ch.int, intermediate channel, i = 1-4, either the number of the cross section or the number of the coefficient (for C_i and n_i); base, base; m, maximum; ind, indicator specimen; red.ind, reduced for the indicator specimen.

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