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Influence of location of submerged impact microjets on convective heat transfer during cooling of a heat-stressed surface by water and dielectric liquid

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Heat transfer during cooling of a heat-stressed surface in a small-sized channel by an annular and distributed array of flooded impact microjets of water and dielectric liquid Novec 7100 has been experimentally studied. The experimental setup is a copper block cooled by an array of microjets from nozzles with a diameter of 174, 327 μm and located at a distance of 1 mm from the surface. It has been established that the use of localized arrays of microjets makes it possible to obtain effective cooling of a heat-stressed surface with pumping energy comparable to energy costs for distributed microjets.

Keywords: convective heat transfer, impact microjet, water, dielectric liquid.

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The miniaturization of electronic equipment coupled with its increasing power levels necessitate the development of new methods for cooling of devices with highly localized regions of heat output. The heat output of modern computer microprocessors exceeds 100 W/cm^2, while certain lasers, microwave devices, and radars operate at the level of 1000 W/cm^2 [1]. Traditional air cooling techniques are inapplicable in such conditions and are substituted with active single-phase and two-phase cooling systems. The use of submerged impact microjets with nozzles positioned close to the cooled surface is one of the most efficient methods of single-phase cooling [2]. Systems of this kind are now used widely to cool computer microprocessors, radio transmitters, optical devices, and other equipment [3,4]. Swirling and excited jets are used to intensify the processes of heat transfer; nanoparticles may also be added to enhance the thermal conductivity of liquid [5,6]

A considerable number of papers focused on heat transfer in the process of impingement of submerged jets on a heated surface [7,8] have already been published, although studies into heat transfer in the case of impingement of distributed arrays of microjets are much more scarce [2,9,10]. Such jets provide an opportunity to intensify the process of heat removal considerably: heat fluxes up to 1100 W/cm^2 were achieved in [2] for a heater $80 \times 80\,\mu$m in size and a nozzle diameter of 112 μm at a temperature difference less than 30°C and a distance of 250 μm between nozzles and a heat-emitting surface.

The method of cooling of power electronic devices by distributed impact microjets has its drawbacks in practical applications. The maximum heat removal is observed in the region of jet impingement, while the process intensity for a wall jet is lower. This is the reason why high-density microjets are used to facilitate heat transfer; however, this entails an increase in energy spent on coolant pumping and makes the design of the liquid removal system more complex. At the same time, the heat transfer in a wall jet remains significant at a distance up to 15 jet diameters and may be used efficiently for removal of intense heat fluxes. This makes it possible to use localized microjet arrays to achieve high coefficients of heat transfer while minimizing the energy spent on coolant pumping and protecting the central part of the surface from jet impacts.

The aim of the present study is to examine experimentally the process of heat transfer in cooling of a heat-stressed surface by annular arrays of microjets and compare the obtained data on heat transfer and energy spent on coolant pumping with the corresponding data for a high-density distributed microjet array. Experiments were performed with the use of distilled water and dielectric liquid Novec 7100. At a temperature of 20°C, kinematic viscosity ν of Novec 7100 and water is 0.488 and 1.002 mm^2/s, while thermal conductivity λ of Novec 7100 and water is 0.07 and 0.6 W/(m $\cdot$ K), respectively.

The setup for experiments on microjet cooling was a closed circuit with a plunger pump, a pressure pulsation dampener, a filter, a turbine flow meter, a heat exchanger setting the temperature of cooling liquid, a working section, and a liquid cooler. An additional water circuit with a thermostat set the temperature of cooling liquid.

The heat-emitting element of the working section was fabricated from M1 copper. Six heating cartridges with a maximum power of 1200 W were mounted in the lower part of the copper block 8 cm in length and 4 cm in diameter. The upper part of this block had the form of a cylinder 1 cm in diameter and 1 cm in length with a 5-μm-thick electrodeposited nickel coating on its exterior surface. This cylinder was cooled by a microjet array (see Fig. 1).

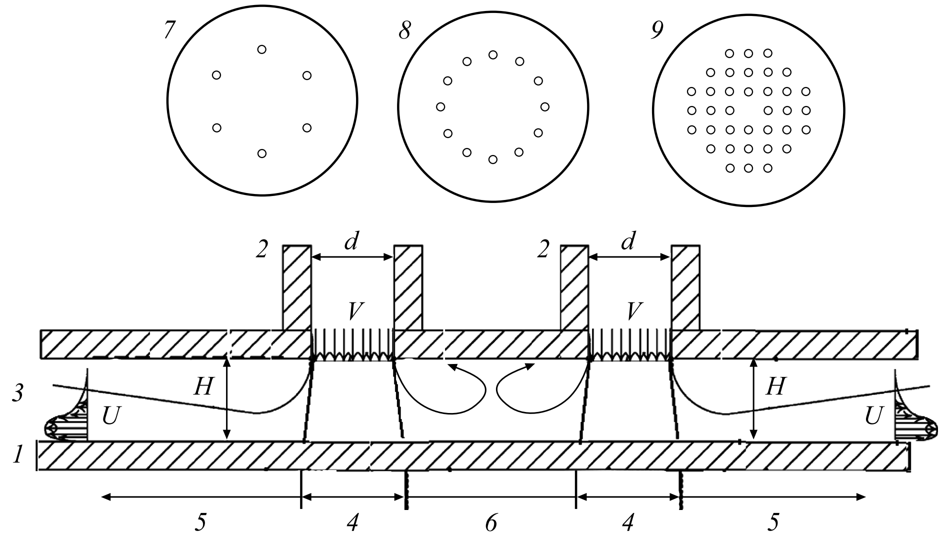

Figure 1. Submerged impact jet cooling system. *1* — Surface of the copper block, *2* — nozzle for microjet, *3* — channel for liquid removal, *4* — region of jet impingement on the surface, *5* — wall jet, *6* — region of interaction of wall jets, *7* — 6-jet array arrangement, *8* — 12-jet array arrangement, *9* — 36-jet array arrangement.

Four insulated thermocouples 0.5 mm in diameter were mounted along the axis of the small cylinder at distances of 1.02, 3.75, 5.94, and 8.72 mm from the exterior cooled surface to determine the heat flux and the heat-stressed surface temperature. These thermocouples were calibrated against a reference platinum thermometer. The side surface of the small cylinder was insulated with a fluoroplastic washer to reduce heat loss. The disbalance between the heat flux determined based on the power of heating cartridges (with heat loss taken into account) and the temperature gradient did not exceed 4%.

The micro-jet generator was a chamber with a replaceable brass head having orifices, determined the diameter of jets. Orifices were arranged as shown in Fig. 1 forming annular localized or distributed arrays of microjets in the channel 1 mm in height. The wall thickness of the brass head was 0.75 mm. Experiments were performed with the use of three types of heads shown in Fig. 1: six orifices of $327(\pm 3)\,\mu$m in diameter, arranged uniformly in a circle with a diameter of 5 mm (type 1); 12 orifices of $174(\pm 3)\,\mu$m in diameter arranged uniformly in a circle with a diameter of 5.5 mm (type 2); 36 uniformly distributed orifices of $174(\pm 3)\,\mu$m in diameter positioned within 1 mm from each other (type 3).

Prior to experiments, the working liquid was introduced into the working section and subjected to degasation with the use of a vacuum pump. The cooling liquid temperature, the cooled surface temperature, the cooling liquid flow rate, the pressure in the chamber upstream of the jet generator, and the pressure in the output chamber were measured in the course of experiments. The average velocity of liquid in jets varied from 2 to 22 m/s, the heat flux density varied up to 200 W/cm^2, and the initial liquid temperature varied from 20 to 40°C. The liquid flow rate was measured using a turbine flow meter with an uncertainty of 0.2 ml/s.

Figure 2, *a* shows the dependence of complex $\mathrm{Nu}_D/\mathrm{Pr}^{0.4}$ ($\mathrm{Nu}_D = hD/\lambda$) on jet Reynolds number $\mathrm{Re}_d = V_{jet}d/\nu$ in the case of microjet cooling with water. Here, h is the heat transfer coefficient, D is the heater diameter, V_{jet} is the jet velocity, d is the orifice diameter, and ν is the kinematic viscosity of liquid. The solid curve represents the results of calculation of heat transfer coefficient performed for a 36-microjet array (distributed over the heater surface) in accordance with the following formula [10]:

$$\frac{\mathrm{Nu}_D}{\mathrm{Pr}^{0.4}} = 0.509\mathrm{Re}_d^{0.5}\left(\frac{D}{d}\right)A_r + 0.0363\mathrm{Re}_{L^*}^{0.8}\left(\frac{D}{L^*}\right)(1 - A_r). \tag{1}$$

Here, $L^* = 0.5\left[\left(L_e/\sqrt{2} - \alpha d\right) + \left(L_e/2 - \alpha d\right)\right]$ is the distance from the jet boundary to the wall jet boundary, L_e is the distance between jets, $\alpha = 1.2$ is the jet diameter increases prior to impingement on the wall, $\mathrm{Re}_{L^*} = V_{jet}(1 + 2gH/V_{jet}^2)^{0.5}L^*/\nu$, H is the channel height, and A_r is the ratio of the total jet area to the heater area. A distributed array of only one fixed size (36 jets) was examined in our experiments. The results of calculations for distributed array revealed that the contribution of heat transfer from the region of jet impingement (first term) is significantly lower than the contribution of heat transfer from the wall jet (second term), since A_r assumes a value of 0.016. Experimental data lie slightly below the calculated values. It can be seen that the heat transfer at a certain fixed value of Re_d is maximal for the distributed microjet array and minimal in the case of orifices of $327\,\mu$m in diameter. At the same time, pressure drop ΔP and theoretical pumping power $Q_{pp} = \dot{m}\Delta P/\rho$ ($\dot{m}$ is the coolant mass flow rate) are also minimal in this case. The experimental data from Fig. 2, *a* were used to determine the dependence of the heat transfer coefficient on pumping power (Fig. 3, *a*). It can be seen that localized arrays with

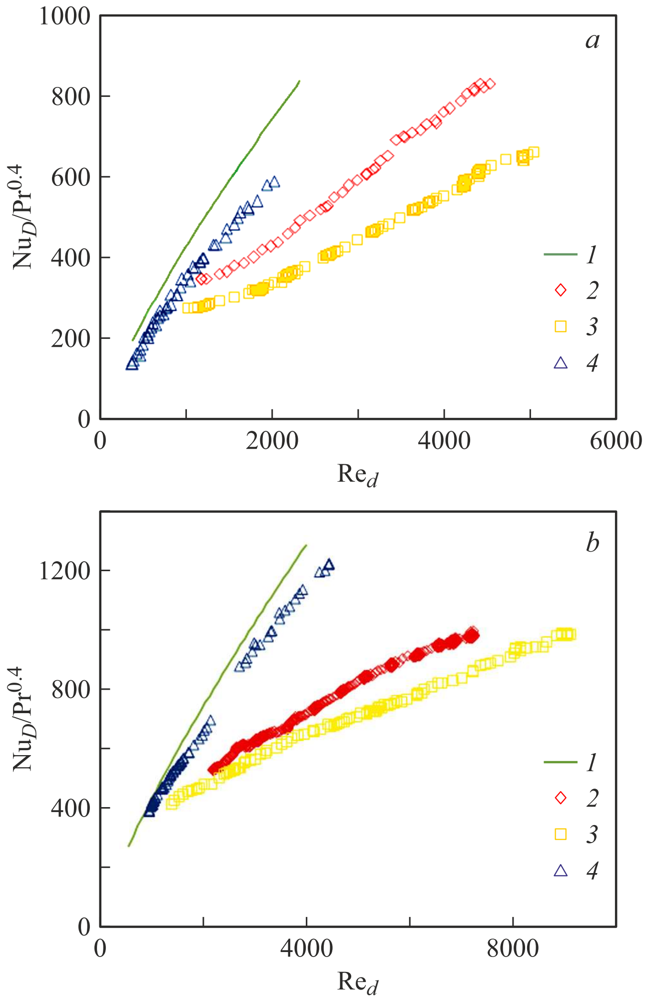

Figure 2. Dependence of the Nusselt number on the jet Reynolds number of for a jet in cooling with water (*a*) and with dielectric liquid (*b*). *1* — Calculation for 36 jets performed in accordance with formula (1), *2* — data for 12 jets of 174 μm in diameter, *3* — data for six jets of 327 μm in diameter, *4* — data for 36 jets of 174 μm in diameter.

a smaller number of water jets (*7* and *8* in Fig. 1) are more efficient than distributed jet arrays. In addition, these configurations protect the central part of the heat-stressed surface from dynamic jet impacts and achieve comparable heat transfer coefficient at lower coolant flow rates.

Figure 2, *b* shows the dependence of complex $Nu_D/Pr^{0.4}$ on jet Reynolds number for dielectric liquid Novec 7100. The kinematic viscosity of Novec 7100 is significantly lower than the one of water. This causes an increased Re value, and the transition to turbulent flow in jets occurs earlier. The values of complex $Nu/Pr^{0.4}$ for Novec 7100 are close to the corresponding values for water. As in the case with water, the results of calculations performed in accordance with (1), which are represented by the solid curve, lie slightly above the experimental data for distributed jets. The heat transfer at a certain fixed value of Re_d is maximal for the distributed microjet array and minimal in the case of orifices with 327 μm in diameter. Figure 3, *b* shows the dependence of the heat transfer coefficient on pumping power for the experimental data from Fig. 2, *b*. Compared to the distributed microjet array, the localized array with a jet diameter of 327 μm (*7* in Fig. 1) allows one to reduce considerably the pumping power if complex $h/Pr^{0.4}$ assumes a value lower than $4\,kW/(m^2 \cdot K)$. At higher values of the heat transfer coefficient, the efficiency of the distributed microjet array increases slightly, although it still remains comparable to the efficiency of localized microjet arrays. This may be associated with a reduced pressure drop at the jet generator for a less viscous liquid (Novec 7100).

The obtained data reveal that, owing to the dominant influence of wall jets on heat transfer in the examined conditions, localized arrays of submerged impact microjets provide efficient cooling of a heat-stressed surface with the pumping power being comparable to the pumping power in the case of distributed microjets. A simplified design

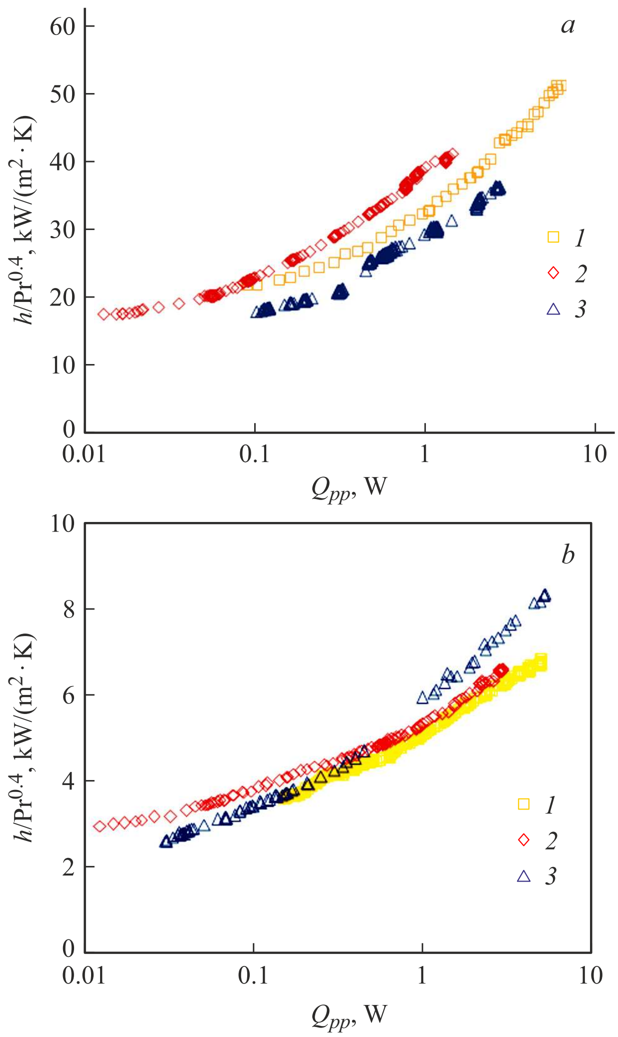

Figure 3. Dependence of heat transfer coefficient vs pumping energy during cooling: with water (*a*) and with dielectric liquid (*b*). *1* — Data for 12 jets of 174 μm in diameter, *2* — data for six jets of 327 μm in diameter, *3* — data for 36 jets of 174 μm in diameter.

of the cooling system and a reduced coolant flow rate are indicative of bright prospects for application of localized arrays of submerged impact microjets in cooling of power electronic devices.

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Conflict of interest

The authors declare that they have no conflict of interest.

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