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EXPERIMENTAL INVESTIGATION OF EVAPORATIVE PULSE-SPRAY IMPINGEMENT COOLING

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This paper describes an experimental facility and methods for measuring the aerodynamic and thermal parameters in the interaction between a pulse spray and a vertical surface. The spray-forming unit contains 16 atomizers with opening time regulation (from 1 to 10 msec) and a frequency of up to 50 Hz. The experiments were performed in the regime of evaporation of the liquid precipitated on the surface in the form of separate drops, rivulets, and a continuously flowing sheet. It has been shown that depending on the time parameters of the pulse spray, the integral heat transfer can effectively be controlled over a wide range.

Keywords: *heat transfer, pulse spray, heat transfer intensification, automatic control, impinging two-phase jets.*

Introduction. Cooling of heat-loaded surfaces by impinging gas-droplet streams is widely used in technology due to the high intensity of the heat transfer processes. However, the interaction of droplets with the surface, especially under heating conditions, is characterized by a large number of various factors that are difficult to take into account and hardly subject to theoretical analysis. Therefore, in investigating the heat transfer processes under spray cooling, of paramount importance are experiments.

Depending on the temperature of the surface being cooled by gas-droplet streams, there exist a diversity of regimes characterized by different heat transfer mechanisms. We can distinguish regimes of film and nucleate boiling, transient regimes, and one-phase cooling regimes. The overwhelming majority of works concerning cooling by gas-droplet streams are devoted to regimes with boiling [1–9], which is explained by the wide use of droplet cooling in metallurgy and nuclear power engineering. Investigations of one-phase cooling are comparatively not many, whereas in the last few years, because of the intensive development of computer engineering and the increasing power of heat released during the operation of processors, the need for new cooling systems has arisen. One promising way is the creation of a flowing sheet by sprinkling the surface with a gas-droplet stream.

Among the works on this topic, we can highlight [9], in which a detailed analysis of the factors influencing the heat transfer under spray cooling of heat-loaded surfaces was carried out and the efficiency of the regime where on the surface a liquid film of minimum thickness is created and continuously and uniformly replenished with drops precipitating throughout its flow area was shown. Similar investigations were also carried out in Russia. In [10], the heat transfer in spraying, by a droplet jet, a heated surface with a temperature below the boiling point was investigated. It was shown that the specific heat transfer in such cooling is higher than that using one-phase liquid flows. In [11], the heat transfer under interaction of individual liquid drops with a heated steel surface without boiling was investigated. Experimental results, as well as results of computational modeling, were presented. It was found that at large Weber numbers $\mathrm{We} \gg \mathrm{Re}^{0.5}$ the cooling efficiency weakly depends on the velocity and size of drops. A relation to the Prandtl number is mainly observed.

A wealth of theoretical and experimental data on the heat transfer in sprinkling, by gas-droplet jets, a surface with a temperature below the boiling temperature of liquid are given in [12]. The heat transfer on a plate sprinkled by a gas-droplet flow formed by an industrial atomizer was studied experimentally in [13]. Experimental data were compared with the developed theoretical models. However, discrepancies between the experimental results and the laws predicted by models were revealed. In a number of technological devices, effective cooling of extended surfaces is realized by multijet sprays. In so doing, the problem of choosing optimal regimes of heat transfer becomes more com-

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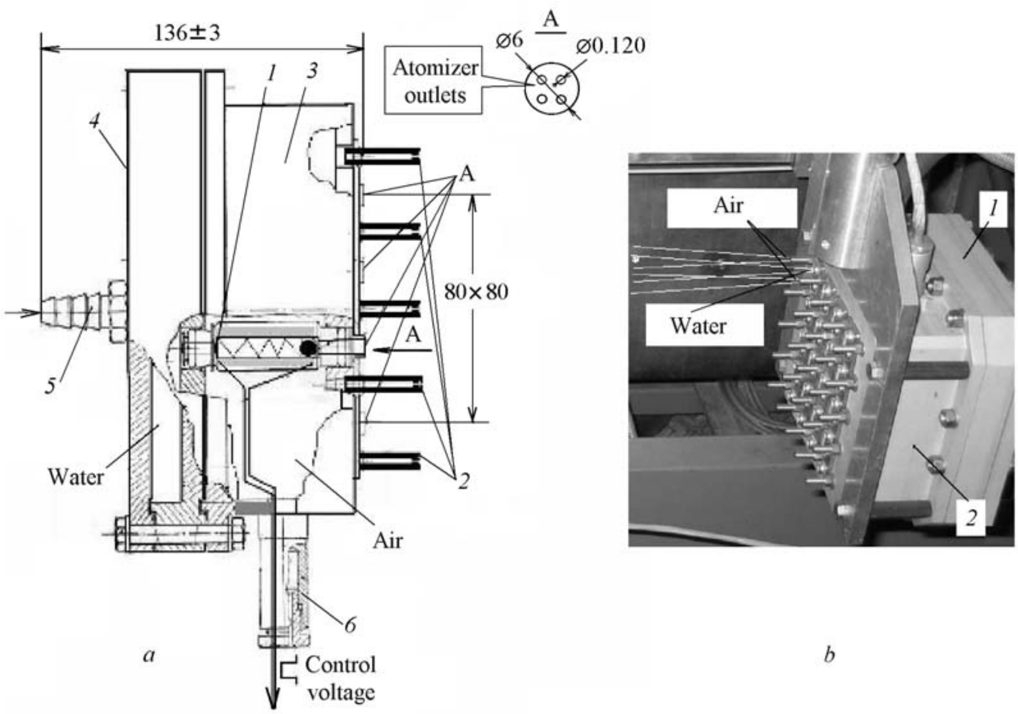


Fig. 1. Source of a multijet pulse spray. a) Scheme: 1) atomizer; 2) gas nozzle; 3) air chamber; 4) liquid chamber; 5) inlet hole of chamber 4; 6) lead-in of the atomizer control cable; b) external view: 1) and 2) liquid and air chambers.

plicated and multifactor [14]. It is known [15–17] that artificial disturbances introduced into the jet intensify the heat transfer processes. Therefore, nonstationary (pulse) sprinkling may lead to an additional increase in the efficiency of cooling heat-loaded surfaces. Selecting system parameters such as the spray velocity, the relation between the duration and frequency of pulses, etc., one can effectively control the heat transfer processes. In this paper, we present the results of investigations of the heat transfer in the process of sprinkling a heated surface by a pulse droplet jet with a temperature below the boiling point.

Experimental Facility and Measuring Technique. The experimental facility consists of the following basic elements: a programmed source of multijet spray (Fig. 1), a digital calorimeter with a heat exchanger (Fig. 2), and an automatic system for recording the gas-droplet flow parameters, the flow rate of the cooling liquid, the temperature, the pressure, the heat flow, and the thickness and velocity of the film on the heat exchanger surface.

The source of multijet pulse spray is designed in the form of a two-compartment block. Situated on the flat part of the injector are 16 jet liquid atomizers in the form of a 4×4 matrix. On the same surface around each liquid atomizer, 25 gas nozzles with an outlet diameter of 3.5 mm for creating a cocurrent air flow are positioned. The flow rate of the liquid through an atomizer and a gas nozzle is given by the inlet pressure of the liquid (P = 0.05–0.6 MPa). The liquid atomizer represents a sprayer consisting of four nozzles of diameter 125 μm turned on by one electromagnetic valve. The opening time of the atomizer was varied from 1 to 10 msec with an opening frequency of the valve from 1 to 50 Hz. The flow rate through the atomizer was regulated individually, which made it possible to form a gas-liquid flow with a given mass of the liquid phase and a given size of liquid droplets and turn off the flow for a given time for optical observations of the waves on the heat exchanger surface. Since the opening time of the electromagnetic valve is short (~0.1 msec), the flow rate of the liquid through the valve changed linearly depending on the turn-on duration (from 2 msec to continuous operation) and the pulse repetition frequency. Each atomizer was calibrated and from the data obtained the total rate of flow of the droplet phase was determined. The velocity distribution of the gas phase was measured with the use of a hot-wire anemometer in the absence of water supply through the atomizers.

To determine the local concentration of the liquid in the gas-droplet flow area depending on the operating conditions of the injector, a capacitive concentration meter was created and used. The recording principle is based on

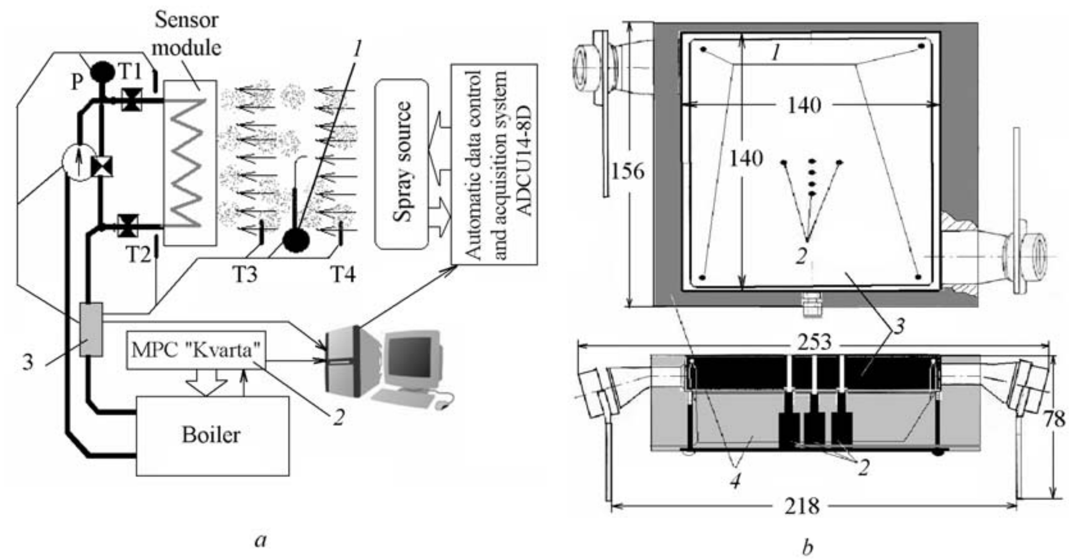


Fig. 2. Basic diagram of the experimental heat exchanger: a) diagram of the calorimeter [T1–T4, thermometers; P, manometer; 1) capacitive concentration meter; 2) microprocessor controller (MPC) — boiler control; 3) flow rate meter]; b) module of the heat exchanger with capacitive and thermal sensors [1) thermocouples; 2) capacitive sensors; 3) housing; 4) heat insulation].

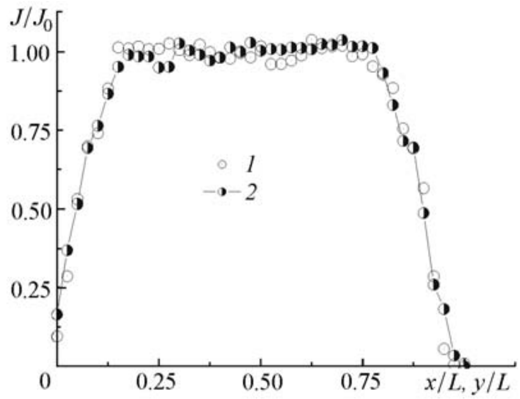
the change in the value of the probe capacity [18, 19]. Probe calibration was carried out with an air-water mixture with a given concentration.

The developed concentration recorder has made it possible to determine, in the process of experiments, the influence of the flow rate of the liquid and air on the moisture distribution in the gas-droplet stream. Scanning was carried out along two coordinates and at a different distance from the injector. Figure 3 shows the moisture distribution in the gas-droplet stream at a distance H = 50 mm from the heat exchanger in the mode of simultaneous turn-on of all valves. As is seen, a considerable part of the cross-section is occupied by a plateau with a practically constant content of droplets. And the concentration distribution therewith coincides in the orthogonal directions.

In the course of investigations not the whole of the stream formed falls on the surface of the heat exchanger. In this connection, we determined the influence of flow conditions on the distribution of moisture that settles on the heat exchanger surface. With the gravimetric method by precipitating droplets on filter paper, we determined the portion of the liquid falling on the working surface of the heat exchanger and on the surface surrounding it. Depending on the regime parameters, the quantity of settling liquid varied from 25 to 80%. In experimental data processing, we took into account the quantity of the liquid phase falling directly on the heat exchanger.

The system recording the influence of the spray parameters on the cooling efficiency is based on the digital calorimeter (see Fig. 2). It embodies the principle of recording, under dynamic conditions, the quantity of heat received by the heat-exchange module from the thermal energy source. Depending on the research problem, the heat exchanger, the heat transfer material, and the mode of operation of the heat or cold source are changed.

The measuring system of the device contains three basic apparatus-program modules: a temperature measuring unit, a module for measuring the flow rate, and a microprocessor module for data processing, control, storage, and transfer. The relative measurement error of the flow rate upon calibration of the device does not exceed 0.5% in the range of flow rates 200–3000 l/h and is less than 1% in the 100–190 l/h range of flow rates. The program-controlled microprocessor module maintains the execution of instructions on temperature measurement in five channels, pressure measurement in two channels, and flow rate measurement in three channels. Depending on the designed experiments, the calorimeter can come complete with an electric boiler ($dT/dt < 0.1$ deg/sec) or a cooler. This universal circuit design can be realized by the developed and manufactured prototype of an energy-efficient multiloop continuous-flow thermostat based on Peltier elements. It has been established experimentally that the thermostat control system permits maintaining the cold loop temperature with a static error of no more than 0.2^{o}C and a standard deviation of interme-



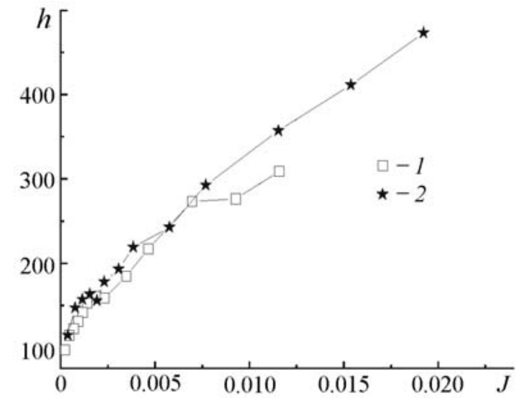


Fig. 3. Distribution of the droplet phase concentration in two orthogonal directions: 1) vertical direction; 2) horizontal direction.

Fig. 4. Distribution of the heat transfer coefficient in cooling the surface by a pulse droplet stream: 1) t = 2 msec; 2) 5. h, W/(m^2·K); J, kg/(m^2·sec).

diate temperature values of 0.1°C. As practice has shown, a total measurement error of the heat flow removed from the target, the heat exchanger, with account of a recording error of the water-air flow, as low as 1.5–3% can be attained, which makes it possible to determine with a fair accuracy the heat transfer parameters and the dependence of these parameters on the cooling flow geometry and conditions.

The heat-transfer surface of the calorimeter had sizes in plan of 150 × 150 mm and its total thickness was 50 mm. The heat exchanger was made of high-heat-conductivity copper, so that in the experiments the boundary condition T_w = const was realized and its validity was verified by thermocouple measurements of the heat-transfer surface at its different points. To intensify the heat removal inside the heat exchanger, a system of grooves increasing the surface area and promoting a decrease in the temperature gradient over the surface was provided. On the side and back surfaces the heat exchanger was covered with a heat-insulation layer of thickness up to 80 mm. In the course of experiments, rapid video filming of the spray formation was carried out, and the heat exchanger surface was photographed for qualitative determination of the character of the liquid film formed on the surface.

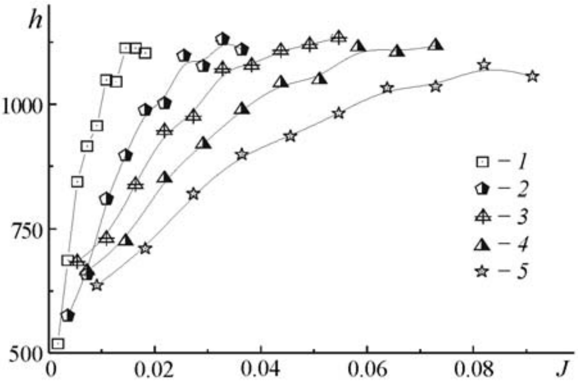
Experimental Results and Discussion. All experiments were carried out at room temperature of the water and air used to create a spray and at a constant temperature of the cooled surface equal to 70°C. The size of droplets at the outlet from the atomizer was 200–500 μm. In the first stage we performed experiments on the investigation of the heat transfer on the surface cooled by a droplet spray without additional air flow. The liquid was sprayed from a distance H = 530 mm from the heat exchanger surface, the frequency was varied from 1 to 50 Hz, and the pulse width took on values of 2–5 msec (Fig. 4).

The experimental results are presented in Fig. 4, which gives the dependence of the heat transfer coefficient on the flow rate of the liquid at various pulse widths. The heat transfer coefficient averaged over the heat exchanger surface is defined as follows:

$$h = Q_h / F_h \, (T_w - T_s) \, .$$

As is seen, for a pure liquid spray the pulse width has practically no effect on the heat transfer and the experimental data practically agree with each other. In the region of small flow rates of the liquid, the experimental curve shows a bend. Visual observations have shown that at these flow rates the rivulet flow of the liquid precipitated on the heated surface changes into a film flow.

The next set of experiments was carried out with the use of gas-droplet aerosol as a coolant. In the experiments, the pulse frequency was varied from 1 to 10 Hz, and the pulse width from 1 to 10 msec. The distance to the surface being cooled was 230 mm, and the surface temperature, as in the previous experiments, was 70°C. The average velocity of air in the two-phase coolant was 8 m/sec, and the ratio between the average-mass velocities of the liquid and gas phases was varied over the 0–0.01 range. Figure 5 shows the dependences of the heat transfer coefficients on the flow rate of the liquid phase of the coolant. Depending on the pulse width, the discrepancy between the



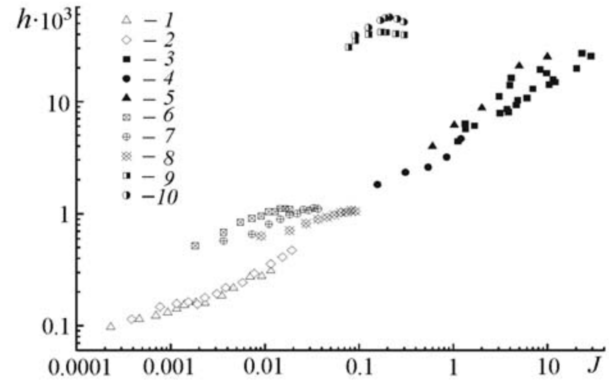


Fig. 5. Influence of the pulse width on the heat transfer coefficient in cooling the surface by a gas-droplet jet: 1) t = 2 msec; 2) 4; 3) 6; 4) 8; 5) 10. h, W/(cm^2·K); J, kg/(m^2·sec).

Fig. 6. Comparison between the obtained experimental data on the heat transfer coefficients on the surface in spray cooling and the data obtained by other authors. Droplet spray (our results): 1) 2 msec; 2) 5; 3) [10]; 4) [20]; 5) [21]; gas-droplet jet (our results, U_g = 8 m/sec): 6) 2 msec; 7) 4; 8) 10; experiments of [22]: 9) U_g = 82 m/sec); 10) 95. h, W/(m^2·K); J, kg/(m^2·sec).

data markedly increases, and the heat transfer maximum for shorter pulses is attained therewith at much lower rates of sprinkling. Consequently, the application of short pulses of supplying the liquid is more effective from the viewpoint of heat transfer intensification. In so doing, the value of the maximum proper is independent of the sprinkling rate.

Figure 6 presents the results of the experiments performed in comparison with the works of other authors on the investigation of the heat transfer in cooling heated surfaces by droplet and gas-droplet sprays. We chose from the other works those data that were obtained under conditions closest to our experiments.

As measurements have shown, in creating a droplet spray, during a pulse from the atomizer an ensemble of droplets of different sizes and velocities from 5 to 10 m/sec was formed. During the flight of droplets to the target, because of the difference in their sizes, their drag occurred differently and, therefore, the velocity difference between neighboring droplets increased. As a consequence, the spray became more homogeneous, especially with increasing distance to the target. This is confirmed by Fig. 6, from which it is seen that our data tend toward the data of the investigations of [10, 20, 21] in which the heat transfer in cooling the surface by droplet sprays was studied. In so doing, the sprinkling rate in the above works and in the present investigation differed by practically two orders of magnitude.

In creating gas-droplet aerosol, droplets got into the cocurrent air flow and their velocity tended to the airflow velocity. Thus, the concurrent flow created a tendency towards grouping of droplets. Moreover, the air flow intensifies the evaporative processes on the surface and turbulizes the film formed, which leads to an increase in the heat transfer coefficients. In Fig. 6, the data of experiments corresponding to the gas-droplet aerosol lie higher than the data on the cooling by a droplet stream. For comparison, Figure 6 presents the data of [22], where the heat transfer in cooling the surface by a gas-droplet jet was investigated. They lie much higher than our experimental curves, which is explained by the higher velocity of the cocurrent air flow. In our experiments, the air flow velocity did not exceed 10 m/sec, and in the experiments of [22] it was equal to 80–90 m/sec. In Fig. 6, a discrepancy between the results of [22] depending on the air flow velocity is seen. In our experiments, there is also a discrepancy between the values of the heat transfer coefficients due to both the influence of the cocurrent flow and the nonstationarity of the liquid phase. This is shown convincingly by the data of Fig. 5.

But in general, the mechanism of heat transfer intensification in a pulse spray and in the presence of a cocurrent air flow remains to be elucidated. The complex and multifactor character of the process makes it impossible to do this, and more detailed experimental studies are needed.

CONCLUSIONS

1. With the use of an automated experimental facility designed for investigating the heat transfer processes in cooling surfaces by a pulse gas-droplet stream, we have obtained initial data on the flow structure and the average heat transfer. The duration of the liquid supply pulse was varied over the 2–10-msec range with a pulse repetition frequency of up to 50 Hz. The spraying unit of the facility permits forming a pulse spray with an independent variation of the pulse frequency and width.

2. The test experiments have shown that a total measurement error of the heat flow removed from the target with account for the recording error of the water-air flow as low as 1.5–3% can be attained, which makes it possible to determine with a fair accuracy the heat transfer parameters and their dependence on the cooling flow geometry and conditions.

3. For a pure liquid spray, a change in the pulse width with no change in the sprinkling rate has no appreciable effect on the heat transfer intensity. With the use of short pulses for opening the liquid atomizers to form a gas-droplet spray with a cocurrent air flow the heat transfer maximum is attained at much smaller values of the sprinkling rate. In so doing, the value of the maximum itself remains practically unaltered with varying pulse width.

4. Qualitative agreement with the literature data on the heat transfer in the interaction of droplet and gas-droplet sprays with vertical surfaces has been obtained. It has been shown that a cocurrent air supply leads to a significant intensification of the heat transfer between the spray and a vertical obstacle.

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NOTATION

d_p, droplet diameter, m; F_h, surface area of the heat exchanger, m^2; h, heat transfer coefficient, W/(m^2·K); H, distance between the heat transfer surface and the atomizer, m; J, density of sprinkling the heat exchange surface with atomized liquid, kg/(m^2·sec); J_0, density of sprinkling the heat exchange surface on the two-phase jet axis, kg/(m^2·sec); L, vertical and horizontal dimensions of the heat exchanger, m; P, pressure, Pa; Q_h, thermal energy delivered to the heat exchanger, W; $\mathrm{Re} = \rho_p U_p d_p/\mu_p$, Reynolds number of the droplet; t, time, sec; T, temperature, ^{o}C; T_s, temperature of the liquid supplied to the atomizer, ^{o}C; T_w, temperature of the heat exchange surface, ^{o}C; U_g, gas velocity, m/sec; U_p, velocity of droplets, m/sec; $\mathrm{We} = \rho_p U_p d_p/\sigma$, Weber number; x, horizontal coordinate, m; y, vertical coordinate, m; ρ_p, density of the droplet, kg/m^3; μ_p, dynamic viscosity coefficient of the droplet, N·sec/m^2; σ, surface tension coefficient of the liquid, N/m. Subscripts: 0, parameter on the two-phase jet axis; g, gas; p, droplet; s, atomizer; h, w, heat exchanger, heat transfer surface.

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