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Impingement cooling system for a large surface: multi-jet pulse spray*

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The paper presents the experimental study on the influence of parameters of a multi-jet impingement spray on the cooling efficiency for a large flat surface. The study is based on common principles for engineering systems with high-rate heat and mass transfer using the impinging jets; this enables the draining of hogh heat loads with a low flow rate of the coolant. These results, along with using the Nusselt and Reynolds criteria, give the approach for estimating the aggregated efficiency of heat transfer coefficient while cooling with a multi-jet impingement spray.

Keywords: cooling of large flat surfaces, impingement spray, coolant flow rate, air coflow, integral heat transfer coefficient.

Introduction

The purpose of this investigation is a study of opportunities for preliminary estimation of the integral efficiency and operability of impingement multi-jet spray system designed for cooling of a large surafce. The common approach to constructing the systmes for intense heat and mass transfer is based on single-jet impinging flows: this offers draining of high heat loads while using a moderate flow rate of liquid coolant [1, 2]. For example, the experiments [3, 4] found that the spray cooling ensures the same level of heat transfer as the jet cooling (but at low coolant flow rate). Depending on the tempertaure of the cooled surface, different regimes with different types of heat and mass transfer are observed. One can focus on film flow with boiling, transitional flow modes, and the regime with single-phase evaporative cooling [5 –7].

There exists a plenty of publications on cooling using gas-droplet flow for the situation of continuous supply of the coolant; they deal with cooling by thick films with boiling [3 –5]. The most part of experimetal and numerical studies in the field of gas-droplet cooling deal with sprays created by a single nozzle [5 – 9]. For the single-phase flwo modes, this type of cooling has been studied fundamentally [5, 6]. However, for the two-phase (gas-droplet) flows, the heat transfer problem for a set of jets impinging a cooled surafce was the object of study only in last years [7, 8]. As compared to the single-phase flow problem, this task is more complicated: then

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the system must ensure uniform irrigation of the cooled surface. This approach was developed by stages, starting from studying two parallel gas-droplet jets [9] and liquid jets [10].

The investigation in [11–15] demonstrated that the pulsed jet cooling offers a higher integral heat transfer coefficient than for the cooling with non-stop spray (at the same liquid flow rates). Designing of controllable and efficient cooling systems on the basis of pulse gas-droplet flow (spray) remains one of promising approahces for problem solving. Several investigations demonstrated the high efficiency of a pulsed multi-jet spray flow for the impingement cooling of large surfaces [11–15].

This short review of publications assumes that the two-phase flow cooling efficiency depends on a multitude of fluid dynamics parameters and the design of the "spray source – cooled surface" system. This complicates the estimation of the cooling system efficiency with a reasonable accuracy. Therefore, an attempt for generalization of experimental data obtained for a multi-jet impinging spray cooling system with the use of dimensionless similarity criteria for heat tarnsfer (i.e., Nusselt number and Reynolds number) remains a topical problem. This generalization would be valuable in calculating the surface-averaged heat transfer coefficient for different versions of the spray cooling system and search for optimal combinations of operation parameters.

1. Experimental setup and measurement technique

The study was performed on the experimental setup which is a prototype of a multi-jet pulsing cooling system (Fig.1). More details about this setup are available [13, 14]. The prototype consists of injector *1*, heat exchanger *2*, digital calorimeter *5*, and automatic data acquisition system *7*.

The coflow of air flow and coolant (water) flow was arranged through a special injector, which is a two-chamber spray generator (see Fig. 2). The generator front surface (a square with the size of $L = 0.14$ m) has chess-allocated 25 air nozzles with the diameter of 0.35 mm and 16 water supply nozzles with the diameter of 0.125 mm.

The water was fed through a set of nozzles in a pulse mode using a programmable valve: the valve can be opened with a certain frequency F during a certain time interval $\Delta\tau$. An accurately controlled amount of water (water pulse) is fed through each nozzle during the open interval (amount is proportional to the time interval $\Delta\tau$). The valve design allows one to assign the pulse repetition rate F from the following series: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 15, 20, 25, 30, 35, 40, 45, 50, 55, 60, 70, 80, 90, and 100 Hz. There are several possible durations of water pulses $\Delta\tau$: 2, 4, and 10 ms. The mass of water injected during a single pulse with duration $\Delta\tau$ was measured using a special calibration procedure [16].

The heat exchanger *2* (see Fig. 1) is a squared plate made of high-heat-conductance copper with a polished working surface with the side length L=0.14 m and the thickness of 30 mm. The distance between the heat exchanger surface and the spray generator H equals 230 mm. The system for stabilization of the heat exchanger surface temperature T_w (water boiler) was kept at a steady value T_w=70 °C with the accuracy $dT_w = \pm 0.2$ °C. The temperature of air and liquid components of the injected air-droplet flow during experiments varied only slightly and for the liquid phase (distilled water) was ~13 °C and for air was ~22 °C.

The integral heat transfer coefficient is calculated by formula $\alpha = q/(T_w - T)$. Here q is the specific heat flux in W/m^2 recorded using the digital calorimeter *5* (see Fig. 1), T_w is the temperature of the working surface of the heat exchanger (70 °C), T is the coolant temperature (measured with thermometer T3). The technique for defining coefficient α in detail is described in [16].

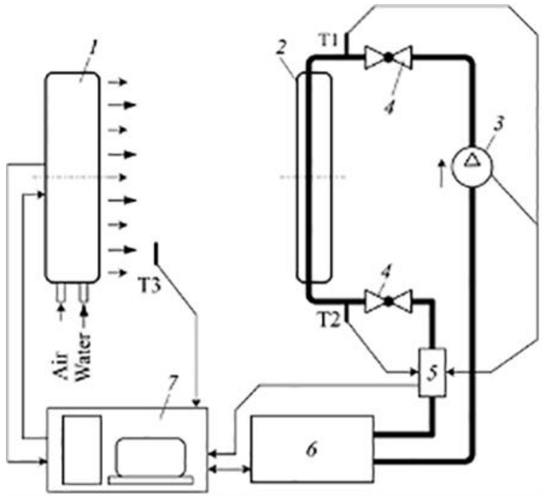

Fig. 1. Prototype of multi-jet pulsing cooling system.

1 — spray injector, *2*— heat exchanger, *3* — circulation pump,
4 — shutoff and control valves, *5*— digital calorimeter,
6 — water boiler for sustaining a steady temperature of the heat exchanger surface,
7— experimental data acquisition system,
T1, T2, T3 — thermometers for heat exchanger's inlet and outlet temperatures
and thermometer for water injector outlet.

The diagram in Fig. 2 shows three significant zones of multi-jet spray evolution: zone 1 from the nozzle outlet of the spray generator up to the flow cross section where the flow reacts to the presence of the heat exchanger; zone 2 covers the turn region of the air flow near the heat exchanger surface and the start of impinging the droplet flow on the cooled surface; and zone 3 is the region with a complicated interaction of the near-surface air flow and water droplets with the cooled surface.

Depending on the combination of numerous input parameters that influence the fluid dynamics and heat transfer (e.g., system geometry, thermophysical properties of coolant and air, operation of injectors), we predict the entire spectrum of heat and mass transfer for air-droplet flow with the cooled surface: from boiling and evaporation of droplets or liquid film patches up to convective heat transfer for a thick liquid film on a solid surface. Obviously, the parameters of a multi-jet flow in zone 1 (for the assigned geometry of the system) would define the further

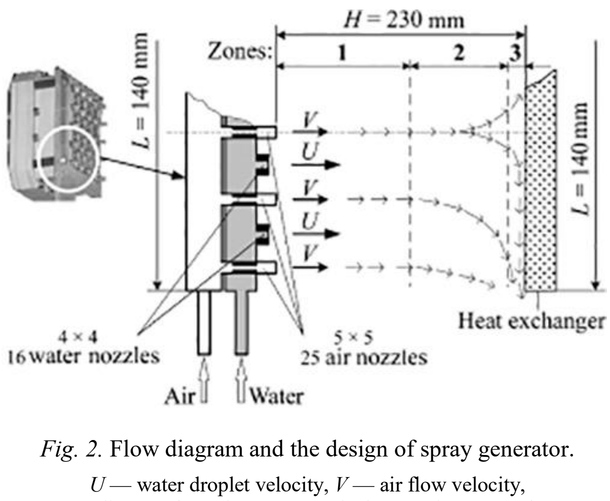

Fig. 2. Flow diagram and the design of spray generator.

U — water droplet velocity, *V* — air flow velocity,
H — distance from the source to the heat exchanger surface.

character of flows and processes occurring in zones 2 and 3. The entire multitude of local flow parameters on the zones of drifting the air-droplet mixture, finally, defines the integral heat transfer coefficient for the cooled surface.

Taking those considerations, we can divide the process of studying the cooling system based on the multi-jet spray source into two main stages. The first stage is the experimental study of the dependence of the integral heat transfer coefficient on the conditions for the formation of a multi-jet air-droplet flow and its component properties and parameters. This stage is aimed at the generalization of experimental data on surface cooling with a multi-jet spray system using the dimensionless heat transfer criteria (Nusselt number and Reynolds number). If this generalization is successful, the empirical dependencies of dimensionless criteria can be a foundation for estimating engineering methods for calculating the surface-averaged heat transfer coefficient at different versions of the spray cooling system. This is beneficial in searching the optimal combination of the input parameters. The second stage of investigation is the experimental study of flow patterns for the second and, mainly, the third zone of flow evolution. The ultimate result of this investigation would help in explaining the results on the integral heat transfer coefficient (the first stage of study) and can be fruitful in developing the theoretical or simulation approaches for this type of complicated cooling systems.

2. Results and discussion

Here we consider the results from the first stage. We conducted two series of experiments on calculating the integral coefficient of heat transfer for the presented experimental setup.

For the first experimental series, the average velocity of air coflow V_a was evaluated as 1.9, 3.1, 4.4, 5.1, 5.8, and 5.9 m/s. The value of V_a velocity was defined by averaging of data from 16 points for anemometer measurement of air-droplet flow (at the distance of 150 mm from the heat exchanger surface). For every value of velocity V_a, four experiments were conducted on finding the heat transfer coefficient for those total flow rates of the coolant M (0.0027, 0.0054, 0.008, and 0.0135 kg/s) being injected through 16 nozzles of the cooling setup. The change in the water flow rate was ensured by changing the frequency of the pulses that open the controlled valves of the air-droplet source and were 1, 2, 3, and 5 Hz. The pulse duration was equal to 2 ms. The liquid/air mass ratio in the gas-droplet stream in this experimental series varied from 0.002 to 0.3.

Figure 3 presents the results of the first experimental series in the format of the integral heat transfer coefficient α vs., the average air coflow velocity V_a. This plotting indicates that the increase in the coolant flow rate M (at a steady air coflow velocity) enhances the integral heat transfer coefficient α. For example, the growth in M from 0.0027 to 0.0135 kg/s increases the coefficient α (at $V_a = 0$) by 3.4 times. Meanwhile, for the air velocity $V_a = 5.9$ m/s, this gain was 2.1 times. Figure 3 indicates that the gain in the air coflow velocity from 0 to 6 m/s (at the steady value of M) increases significantly the integral heat transfer coefficient α: at the mass

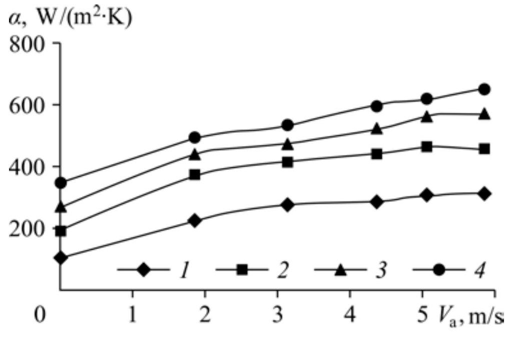

Fig. 3. Integral heat transfer coefficient α vs air coflow velocity V_a at different liquid mass flow rates M. M = 0.0027 kg/s (*1*), 0.0054 kg/s (*2*), 0.0081 kg/s (*3*), 0.0135 kg/s (*4*).

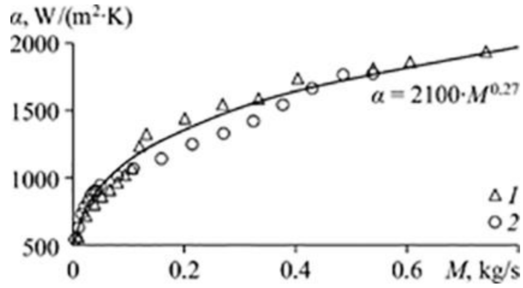

Fig. 4. Dependency of the integral heat transfer coefficient α on the flow rate of coolant at a constant air coflow velocity V_a = 5 m/s. *1*: 10 ms — 1÷55 Hz, *2*: 4 ms — 1÷100 Hz.

flow rate 0.0027 kg/s, this gain was 2.8 times, but at the mass flow rate of 0.0135 kg/s, this gain is only two times.

For the second experimental series, the air coflow velocity was constant and equal to V_a = 5 m/s. The variation of the water flow rate was provided by a change in the opening pulse signals for the programmable control valves. We present here two subseries of experiments — for two fixed duration of valve opening: 4 and 10 ms. For the short-pulse variant (4 ms), we conducted 20 tests with the following pulse repetition rates: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 15, 20, 30, 40, 50, 60, 70, 80, 90, and 100 Hz. These valve operation modes provided the variation of water mass flow rate from 0.0054 to 0.54 kg/s. The long-pulse series (10 ms) included 17 tests with the following pulse repetition rates: 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 15, 20, 25, 30, 40, 45, and 55 Hz. This provided the variation in the water mass flow rate from 0.013 to 0.74 kg/s. The water/air mass ratio in the gas-droplet flow for this test series varied from 0.0046 to 6.2.

Figure 4 presents the results of the second experimental series in the format of the integral heat transfer coefficient α vs. coolant mass flow rate M. These data indicate that the increment in the mass flow rate M (at a steady air coflow velocity V_a) increases the integral heat transfer coefficient α: the dependency $\alpha(M)$ is a power law with the exponent factor ~0.27. One can see also that the integral heat transfer coefficient α depends only on the liquid injection flow rate and is independent of the duration or pulse repetition rate for the valve opening at a fixed flow rate.

The experimental curves presented in Figs. 3 and 4 on flat surface cooling can be considered as surface impact from an “equivalent” impinging spray generator. Here we assume that the generator creates a uniform air-droplet flow with a typical size (heat exchanger side length L) and the average velocity V_a (air coflow velocity). We assume also that the water droplets in this flow are uniformly distributed over the flow cross section with the specific mass flow rate (through the cross section) equal to $m = M/S$, kg/(s·m^2), where M is the liquid mass flow rate from the injector, S is the flow transversal cross section (i.e., heat exchanger area). With this model of the cooling system, we can present the experimental data from Figs. 3 and 4 as a plotting in dimensionless parameters Nu = $f(\mathrm{Re}\cdot(m/m_s))$. Here $\mathrm{Re} = L\cdot V_a/\nu$ is the Reynolds number defined from the typical system size L, average air coflow velocity V_a, and the air kinematic viscosity ν, m/m_s is the specific dimensionless liquid flow rate through the transversal cross section S; $m_s = 1$ kg/(s·m^2) is the unit (equal to one) specific flow rate for liquid through the flow cross section S; $\mathrm{Nu} = \alpha L/\lambda$ is the Nusselt number defined from the integral heat transfer coefficient α, characteristic setup size L and water thermal conductivity λ at the outlet temperature of the water injector. The characteristic size of setup was the length of the squared-shaped heat exchanger L = 0.14 m, and the air velocity V_a is the experimental average air coflow velocity.

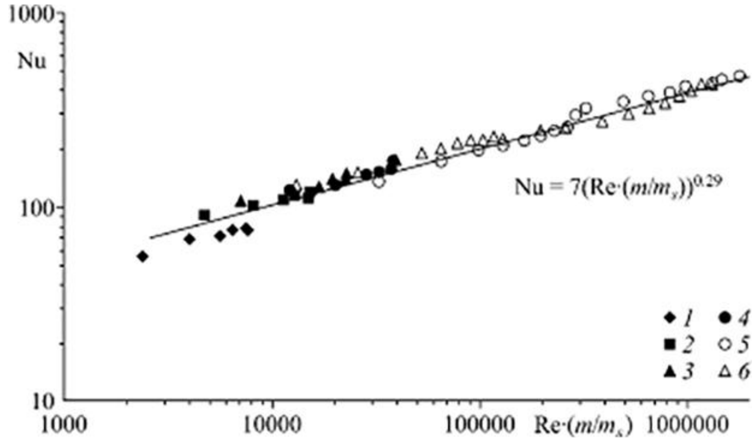

Fig. 5. Dependency of Nusselt number on Reynolds number
and on dimensionless coolant flow rate.
M = 0.0027 kg/s (*1*), 0.0054 kg/s (*2*), 0.0081 kg/s (*3*),
0.0135 kg/s (*4*),0.0135÷0.74 kg/s (*5*), 0.0054÷0.54 kg/s (*6*).

Figure 5 presents the same results of all experiments as a plotting $\mathrm{Nu} = f(\mathrm{Re}\cdot(m/m_s))$. This curve confirms that the offered model of equivalent cooling system enables the generalization of obtained experimental data in the form of a dependency between dimensionless values with three parameters: Nu, Re, and m/m_s. This empirical dependency can be approximated by the following power law:

$$\mathrm{Nu} = 7[\mathrm{Re}\cdot(m/m_s)]^{0.29}.$$

This approximation enables calculating the parameters of a multi-jet impingement spray with alternative geometries and flow rates.

This approach for evaluating efficiency that type of cooling systems can be validated by experimenting with similar models for a wide range of geometry characteristics and flow rates.

Conclusion

Experiments were conducted for measuring the integral heat transfer of a flat surface with cooling by a multi-jet impingement system. The experiments were conducted for variation in the air coflow velocity in the range 0 – 5.9 m/s and the interval of coolant mass flow rate 0.0027– 0.74 kg/s. The liquid was injected into the air flow in a pulse mode. The duration of one liquid pulse was chosen from values 2, 4, and 10 ms, while the pulse repetition rate varied from 1 to 100 Hz. It was found that for these experimental conditions, the integral heat transfer coefficient α depends only on the mass flow rate of injected coolant and is independent of pulse characteristics produced by the controllable valves. An equivalent model for multi-jet impingement spray system for large surface cooling was developed. This allows us to generalize the experimental data in a form of approximate dimensionless function $\mathrm{Nu} = 7[\mathrm{Re}\cdot(m/m_s)]^{0.29}$. This approximation law can be applied for calculating the parameters of a cooling multi-jet impingement spray system with other parameters.

Nomenclature

U — water droplet velocity, m/s,
V_a — air coflow velocity, m/s,
L —length of the heat exchanger side, m,
H — distance between the source and heat exchanger, m,
$\Delta\tau$ —pulse length, ms,
F — valve operation frequency, Hz,
S —flow cross section, m^2,
T_w — temperature of heat exchanger surface, °C,
T — coolant temperature, °C,
q — specific heat flux, W/m^2,
M — liquid mass flow rate, kg/s,
m_s — normalized (equal to one) specific liquid flow rate through the flow cross section S, kg/(m^{2}s),
Re — Reynolds number,
Nu — Nusselt number,
α — integral heat transfer coefficient, W/(m^2·K),
ν — air kinematic viscosity, m^2/s,
λ — water thermal conductivity, W/(m·K).

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