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EFFECT OF THE NONISOTHERMICITY OF THE WALL ON HEAT TRANSFER IN JET WETTING

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A method for solving a conjugate periodic convective-conductive problem for a "heat-transfer agent—wall" developed earlier system is used for calculation of the effect of the wall parameters on heat transfer under conditions of jet wetting.

In the majority of cases, real processes of heat transfer are characterized by periodic velocity and temperature pulsations about the average values. These fluctuations can be smooth (wave flow of a liquid film, flow of a heat-transfer agent that pulsates in time along bodies) or relaxation (the slug flow regime of a two-phase flow, nucleate boiling and transient boiling of a liquid). The pulsations of the characteristics can be spatial and temporal in character, have the form of progressive and standing waves, and possess a cellular space-time structure. Temperature pulsations in the wall region of the flow that lead to disturbances of the temperature field of the wall are a consequence of thermohydraulic pulsations in the volume of the heat-transfer agent. Under conditions of thermal conjugation, both local and averaged differences of the "heat-transfer agent—wall" temperatures must depend on the thermophysical properties of the wall, its thickness and configuration, and the conditions of the external heat supply. In the final analysis, the heat-transfer coefficient defined as the quotient of division of the averaged density of the heat flux by the averaged temperature difference, too, will be a function of the thermal effect of the wall.

The problem of allowing for the thermal effect of the wall on the averaged heat transfer in heat-transfer processes with periodic intensity was stated in general form for the first time in [1, 2]. A new method for analysis of the conjugate convective-conductive problem for a "heat-transfer agent—wall" system was proposed.

According to [1, 2], the action of the heat-transfer agent on the wall is considered to be equivalent to prescribing on the heat exchange surface the "true" heat-transfer coefficient α that varies periodically both along the surface of the body and with time. The quantity α is represented as a superposition of the averaged $\langle\alpha\rangle$ and pulsational ψ components:

$$\alpha = \langle\alpha\rangle (1 + \psi) \tag{1}$$

which is the quotient of division of the local quantities:

$$\alpha \equiv q_s/\vartheta_s . \tag{2}$$

From solution of the heat-conduction equation, we determine the temperature field in the wall, by which we calculate the standard heat-transfer coefficient, i.e., the quotient of division of the averaged heat-flux density $\langle q_s\rangle$ by the averaged temperature difference $\langle\vartheta_s\rangle$:

$$\alpha_m = \langle q_s\rangle/\langle\vartheta_s\rangle . \tag{3}$$

The quantity α_m, which is the objective of a traditional heat-transfer experiment and is used in applied calculations, is called the "experimental" heat-transfer coefficient in [1-4]. The true averaged heat-transfer coefficient, by definition (2), will be equal to

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$$\langle\alpha\rangle = \langle q_s/\vartheta_s\rangle\,. \tag{4}$$

From the difference in the averaging procedures in (3) and (4), it follows that α_m and $\langle\alpha\rangle$, in the general case, will not be equal. A quantitative measure of their difference is defined by the "coefficient of the thermal effect of the wall"

$$\varepsilon = \alpha_m/\langle\alpha\rangle\,. \tag{5}$$

which, in the general case, will depend on the following parameters: the thermophysical properties of the heat-transfer body, its geometry (a single- or multilayer flat plate, a cylinder, a sphere) and characteristic dimension (thickness, radius), the space and time periods of the pulsations, the method of heat supply, and the amplitude and mode of pulsations of the true heat-transfer coefficient (smooth or relaxation functions ψ, a progressive or standing wave).

According to the basic idea of the method, a value of α whose averaged part $\langle\alpha\rangle$ is adopted from the corresponding section of the theory of convective heat transfer in a nonconjugate formulation and whose pulsational part ψ is prescribed by the mechanism of hydrodynamic pulsations in the heat-transfer agent corresponds to each process of heat transfer with periodic intensity. Then, for walls having various thermophysical properties, thicknessess, configurations, and methods of heat supply, the sought value of α_m is determined (as the product of solving the heat-conduction equation for the wall) by relation (3).

A method of analysis of the thermal effect of the wall on the heat transfer in the case of periodic intensity pulsations was developed in [3, 4]. Analytical solutions for three basic characteristic laws of pulsations of the true heat-transfer coefficient that correspond to harmonic, inverse harmonic, and step functions ψ were obtained. Detailed tables and nomograms for the central characteristic of the analysis – the coefficient of the thermal effect of the wall $\varepsilon = \alpha_m/\langle\alpha\rangle$ – were constructed.

These solutions have, however, a very cumbersome form: sums of infinite continued fractions and functional series. In [5], the conclusion was made that it is apparently impossible to obtain analytical solutions for functions ψ different from the three enumerated functions. At the same time, because of the multiparameter nature of the problem direct numerical modeling of the conjugate problem for processes of heat transfer with periodic intensity is inefficient. In [6, 7], a simple approximate method for calculating ε based on asymptotic expansions of a boundary condition of the third kind into power series in the parameters was proposed. The investigations of [8-14] were devoted to further development and universalization of the method for calculating the averaged heat-transfer coefficient. The problem of space-time periodicity of convective heat transfer on both sides of a heat-transfer wall is considered in [15, 16]. A relation for the averaged heat-transfer coefficient that incorporates as particular cases the corresponding problems of the experimental heat-transfer coefficients on each side of the wall is obtained, and thus a higher hierarchic level is reached by the method. In [17-22], relations obtained earlier for ε were adapted to applied problems of evaporation of thin liquid films, nucleate boiling and transient boiling, burnout, and the flow of a heat-transfer agent in assemblies of fuel elements. The set of results obtained was summed up in [23] in the form of an approximate theory of processes of heat transfer with periodic intensity.

The present work is devoted to the use of the method developed in [1-23] for allowing for the effect of the parameters of the heat-transfer body on heat transfer for the particular case of spatial nonisothermicity.

Results of an experimental determination of the heat-transfer coefficients in jet flow of air around spheres with internal heat release are given in [24]. The specimen under study was a hollow sphere with a heater on its interior surface. Specimens manufactured from different materials (brass, steel, stainless steel, and fluoroplastic) were studied. The ratio of the outside diameter of a spherical shell to its inside diameter did not change and was 1.4 for all the specimens.

The dependences obtained in [24] for the Nusselt number are approximated by the following expression:

$$\mathrm{Nu_m} = 5.8\,(\lambda_w/\lambda)^{-0.38}\,\mathrm{Re}^{0.28 + 0.1\log(\lambda_w/\lambda)}\,. \tag{6}$$

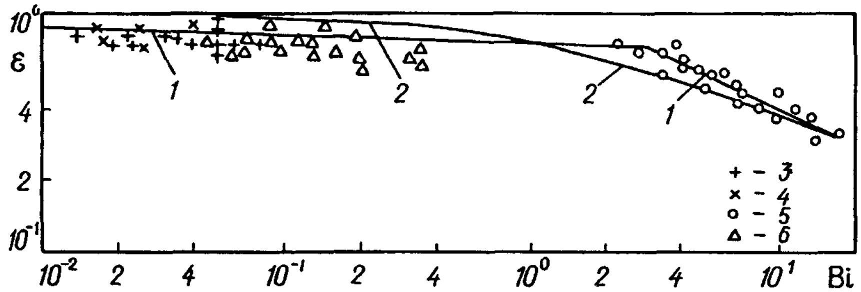

Fig. 1. Coefficient of the thermal effect of a wall as a function of the Biot number: 1) according to [25], 2) by (12) of the present work, 3-6) data of [24] [material of the wall: 3) brass, 4) steel, 5) fluoroplastic, 6) stainless steel].

The experimental results of [24] were analyzed qualitatively in [25], the chain of reasoning of which is very similar to the idea of the method developed in [1, 2].

For a quantitative description of the effect of thermal conjugation found experimentally in [24], the author of [25] employed the following empirical method. The Biot number for a spherical wall is introduced:

$$\mathrm{Bi} = \frac{\langle \alpha \rangle \Delta}{\lambda_w (D_1/D_0)} . \tag{7}$$

For material with an infinitely high thermal conductivity ($\mathrm{Bi} \to 0$), the surface of the sphere is considered to be isothermal and the effect of thermal conjugation is absent. The coefficient of the thermal effect of the wall as a function of the Biot number is sought:

$$\varepsilon = f(\mathrm{Bi}) . \tag{8}$$

Based on extrapolation of experimental data of [24] to the asymptotic case $(\lambda_w/\lambda) \to \infty$, we found the following dependence of the true Nusselt number averaged over the surface on the Reynolds number:

$$\mathrm{Nu} = 0.44\,\mathrm{Re}^{0.6} . \tag{9}$$

Finally, all the experimental points of [24] were approximated in [25] using (9) by the following empirical dependences:

a) for $0.01 \le \mathrm{Bi} \le 3$

$$\varepsilon = 0.75\,\mathrm{Bi}^{-0.063} ; \tag{10}$$

b) for $3 \le \mathrm{Bi} \le 20$

$$\varepsilon = 1.15\,\mathrm{Bi}^{-0.45} . \tag{11}$$

We use results of [10-13] to generalize the experimental data of [24]. In the case under consideration, the spatial nonisothermicity can be approximated by a two-dimensional harmonic function of the true heat-transfer coefficient "expanded" on the exterior surface of the sphere. For this problem, we can obtain a simple approximate relation for the coefficient of the thermal effect of the wall as a function of the Biot number:

$$\varepsilon \approx (1 + 0.15\,\mathrm{Bi})^{-1} , \tag{12}$$

which describes satisfactorily the experimental data of [24] (see Fig. 1).

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NOTATION

α, true heat-transfer coefficient; $\langle\alpha\rangle$, true averaged heat-transfer coefficient; ψ, pulsational component of the true heat-transfer coefficient; α_m, experimental heat-transfer coefficient; $\varepsilon = \langle\alpha\rangle/\alpha_m$, coefficient of the thermal effect of the wall; q_s, density of the heat flux through the heat-exchange surface; ϑ_s, difference of the "heat-exchange surface–heat-transfer agent at infinity" temperatures; D_0 and D_1, outside and inside diameters of the spherical shell; $\Delta = D_0 - D_1$, thickness of the spherical wall; λ_w, thermal conductivity of the wall; λ, thermal conductivity of the liquid; $Nu = \langle\alpha\rangle D_0/\lambda$, Nusselt number constructed from the averaged heat-transfer coefficient; $Nu_m = \alpha_m D_0/\lambda$, Nusselt number constructed from the experimental heat-transfer coefficient; Re, Reynolds number.

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