STUDY OF RADIATIVE HEAT TRANSFER BETWEEN A SHELL AND A DISPERSED HEAT CARRIER (COOLANT)

Z. I. Geller and Yu. Z. Votlokhin UDC 536.248

The results of experiments aimed at determining the radiant component of heat transfer between a shell and a dispersed coolant are presented.

The heat transfer between the walls of a channel and dispersed material falling in a diathermal medium is determined by the convective heat transfer between the wall and the gas medium and between the gas and the moving particles, and also by the radiative heat transfer between the wall and the flow of coolant. The contact heat transfer between the particles and the wall may be neglected owing to the short period of contact, the small surfaces of contact, and the low specific pressure acting on the particle at the instant of contact.

Radiative heat transfer in dispersed media is described by the Stefan–Boltzmann law. Calculation of the radiative component is a complicated matter, as it involves the geometry of the channel, the shape of the particles, the particle distribution, the type of motion existing in the flow, the cross-sectional and longitudinal temperature distribution, the mutual screening effects, and the dissipation of thermal energy by the particles; for a given value of the effective area of the heat-transfer surface F_e, it amounts to a determination of the reduced emissivity of the system ε_{re}.

In order to calculate the integrated radiant heat transfer of the diffusely emitting and absorbing bodies making up the system (a cylindrical channel with a dispersed coolant moving within it) we assume that each elemental cylindrical volume of the channel having a height equal to the diameter of a particle contains exactly one particle; neglecting the screening of the particles in the vertical direction, we use the expressions [1]:

$$F_e = F_1 \tag{1}$$

and

$$\frac{1}{\varepsilon_{re}} = 1 + \varphi_{12}\left(\frac{1}{\varepsilon_1} - 1\right) + \varphi_{21}\left(\frac{1}{\varepsilon_2} - 1\right). \tag{2}$$

The irradiation coefficients are here related to the mutual radiation surface in the following way

$$\varphi_{12} = \frac{H_{12}}{F_1} = 1, \qquad \varphi_{21} = \frac{H_{21}}{F_2} = \frac{F_1}{F_2} \tag{3}$$

with

$$H_{12} = H_{21} = F_1.$$

In view of the screening effects, any increase in the number of particles per unit volume leads to a change in the mutual radiation surface, the value of which may be expressed in the form

$$H_{12} = H_{21} = \delta F_1. \tag{4}$$

The reduced emissivity of the system is then given by

$$\frac{1}{\varepsilon_{re}} = 1 + \delta\left(\frac{1}{\varepsilon_1} - 1\right) + \frac{\delta F_1}{F_2}\left(\frac{1}{\varepsilon_2} - 1\right) = 1 + \delta\left(\frac{1}{\varepsilon'_{re}} - 1\right), \tag{5}$$

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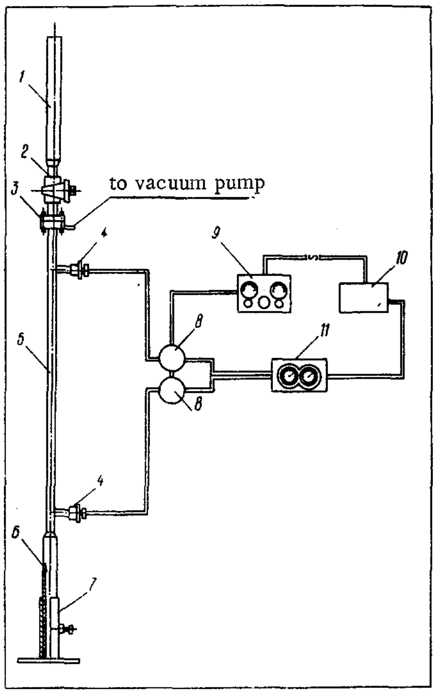

Fig. 1. Vacuum model for studying the motion of a falling layer of coolant: 1, 6) bunkers for coolant; 2) stop cock; 3) flanged joint for siting diaphragms; 4) acoustic sensors; 5) measuring section; 7) support; 8) amplifiers; 9) supply unit; 10) rectifier; 11) recording device.

where $1/\varepsilon'_{re} = (1/\varepsilon_1 + F_1/F_2)((1/\varepsilon) - 1)$ gives the reduced emissivity of the system without allowing for screening; δ is a parameter allowing for the change in the irradiation coefficients of the system associated with the screening, dissipation, and motion of the particles.

The coefficient δ reflects the factors actually operating in the falling layer of dispersed coolant (the shape of the particles, their motion, the nonuniform particle distribution over the cross section, screening, and dissipation) and affecting the radiative heat transfer between the wall of the channel and the coolant. In every specific case it is therefore essential to make an experimental determination of the coefficient δ, the values of which may also be carried over to similar systems, subject to the satisfaction of certain boundary conditions [2].

We studied the motion of a falling layer of coolant in a cylindrical channel (passing through a diaphragm under vacuum conditions) in the model illustrated in Fig. 1. We determined the average velocity (average period spent by the particles in the channel) and the total spillage time of the coolant by means of a special set of equipment consisting of acoustic sensors (laryngophones with needles) introduced into the cavity of the channel, signal amplifiers, a supply unit, and recording devices [3] which enabled the time to be measured to an accuracy of 0.01 sec.

In all the experiments carried out in the vacuum models we used a fine fraction of fluvial quartz sand with an average equivalent diameter of $d_{eq} = 0.8$ mm. The rarefaction (absolute pressure less than 600 N/m^2) was maintained by means of a vacuum pump.

We measured the time spent by the particles in the measuring section as follows. After opening the tap, the coolant started spilling through with a mass velocity determined by the diaphragm currently installed. When the flow of coolant was interrupted (by closing the tap) and the impacts of the particles on the needle ceased, the first sensor started and the second sensor correspondingly stopped the counting device. The time required for the layer to pass through a distance of 1 m was measured by a pulse counter.

Figure 2 shows the dependence of the time τ spent by the particles in the measuring section on the mass velocity of the coolant G, both for a smooth channel (d = 25 mm) and also for the case in which the channel contained a spiral fitting (tightly pressed against its inner surface) with a pitch of 35 mm, this being made of 2.5 mm diameter wire. The average value of τ was determined to an error of no greater than ±(6-8%) from a series of several experiments.

The time spent by the particles in the smooth channel was found to increase for small values of G; this was attributed to the curvature of the particle trajectories arising from their collisions with the walls and with each other. As the mass velocity of the flow increases, so does the tightness of the particle flow; the trajectories straighten and τ diminishes.

The use of a spiral fitting under vacuum conditions leads to a sharp reduction in the mean flow velocity. The retarding action of the spiral on the falling flow of coolant is due to the development of rotational motion of the particles in the direction of the winding of the spiral, and also possibly to the reflection of the particles by collision with the latter.

We studied the heat transfer between the walls of the channel and the falling layer of coolant under vacuum conditions in a thermal model of the rotating type, illustrated schematically in Fig. 3. The model consists of two bunkers connected by a 32 × 3 mm tube and a heating furnace, all mounted on a framework which may be rotated through 180° and fixed vertically in a stand. Along the axis of the bunkers are special pockets for thermocouples with protective casings. The temperature of the inner surface of the channel was measured with thermocouples sited in longitudinal grooves milled in the body of the tube.

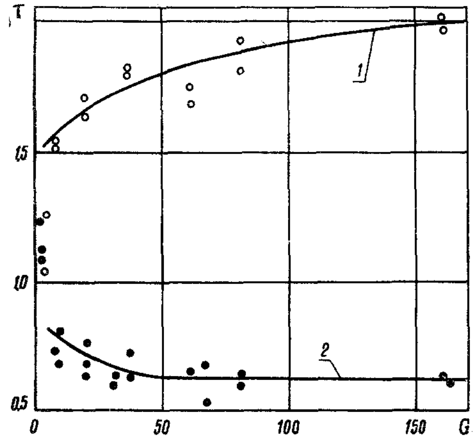

Fig. 2. Dependence of the time spent by the particles in the measuring channel (d = 25 mm and l = 1 m) on the mass velocity of the flow of coolant: 1) channel with spiral fitting; 2) smooth channel. τ, sec; G, g/sec.

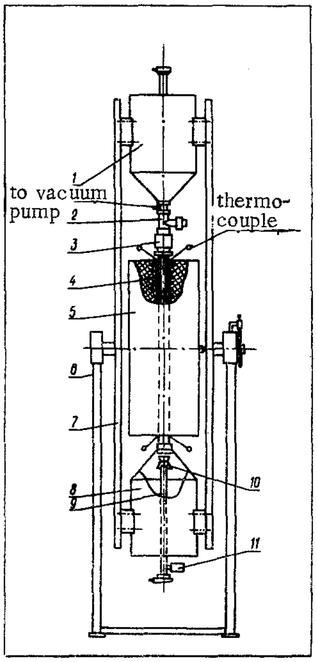

Fig. 3. Vacuum model for studying the heat transfer of a falling layer of coolant: 1), 8) bunkers for coolant; 2) chamber for diaphragm; 3) gland sealing; 4) heat-transfer section (d = 25 mm, l = 0.64 m); 5) heating surface; 7) rotating frame; 9) thermocouple pocket; 10) protective casing; 11) acoustic sensor.

In order to determine the radiant component of heat transfer we carried out at least two experiments at the same mass velocity. The first experiment was carried out at a mean wall temperature of the heat-transfer tube of 200-300°C and the second at 600-800°C. As the working temperature at the entrance into the heat-transfer section we took the temperature of the coolant measured in the upper bunker, the value of which differed little from the temperature of the surrounding air. As working temperature of the coolant at the exit from the heat-transfer section we took the temperature in the lower bunker at the end of the experiment with "fixation" of the layer [2].

Using this method the coefficient δ remains constant, while under vacuum conditions α_c remains also practically constant for pairs of experiments differing only in temperature level. This allows us to express the thermal balance for the first and second experiment in the form

$$Q_1 = \varepsilon_{re_1}\sigma_0\delta F_1 (T^4_{w_1} - T^4_{f1}) + \alpha_c \Delta t_1 F_2, \qquad (6)$$

$$Q_2 = \varepsilon_{re_2}\sigma_0\delta F_1 (T^4_{w_2} - T^4_{f2}) + \alpha_c \Delta t_2 F_2. \qquad (7)$$

Table 1 shows the initial data and the results of some of the experiments on heat transfer in a smooth channel and a channel containing a spiral fitting obtained by computer calculation based on Eqs. (5), (6), and (7).

TABLE 1. Initial Data and Results of Experiments on Heat Transfer between a Smooth Channel, a Channel with a Spiral Fitting, and a Falling Layer of Coolant (Individual pairs of experiments differing only in temperature conditions)

Quantities	Smooth channel				Channel with spiral fitting					
	experiments									
	1, a	1, b	2, a	2, b	1, a	1, b	2, a	2, b	3, a	3, b
G	57,5	57,5	85,5	85,5	19	19	58	58	32	32
t_{f_1}	25	34	34	27	26	26	27	38,5	24	26
t_{f_2}	30	50	29	43	37	84	33	56	32,5	56
t_f	27,5	42	26,5	35	31,5	55	30	47,2	28,2	41
t_w	279	576	233	503	281	700	316	574	274	545
q	4151	13750	6150	18800	3050	16880	5065	15410	4150	14250
α_s	16,5	25,7	29,7	40,2	12,2	26,2	17,7	29,2	16,9	28,3
F_1/F_2	1,30	1,30	1,89	1,89	1,08	1,08	3,71	3,71	1,95	1,95
ε_1	0,890	0,887	0,890	0,888	0,890	0,885	0,890	0,885	0,890	0,887
ε_2	0,658	0,683	0,643	0,675	0,653	0,695	0,620	0,682	0,653	0,679
φ_{12}	0,234	0,234	0,259	0,259	0,255	0,255	0,116	0,116	0,240	0,240
φ_{21}	0,304	0,304	0,490	0,490	0,275	0,275	0,430	0,430	0,470	0,470
ε_{re}	0,255	0,260	0,376	0,380	0,234	0,238	0,347	0,347	0,366	0,373
α_r	4,9	14,1	5,9	16,5	4,5	18,6	7,6	19,2	6,9	18,4
α_c	11,6	11,6	23,8	23,7	7,7	7,6	10,1	10,0	10,0	9,9

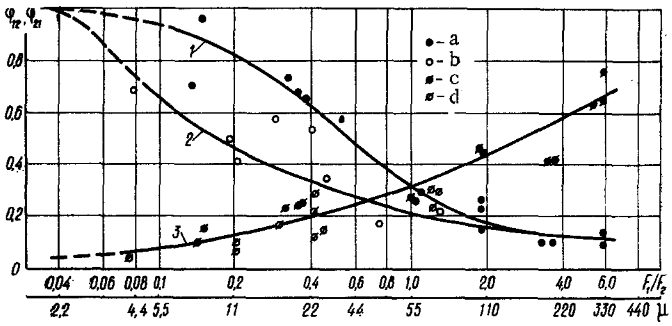

Fig. 4. Dependence of the irradiation coefficients φ_{12}, φ_{21} on the parameter F_1/F_2: 1) φ_{12} for experiments with a spiral fitting (a); 2) φ_{12} for experiments with a smooth channel (b); 3) average φ_{21} for a channel with a spiral fitting (c) and a smooth channel (d). μ, kg/m^3.

Figure 4 shows the dependence of φ_{12} and φ_{21} on μ and F_1/F_2. We see from the graph that on increasing F_1/F_2 (and hence also the screening of the particles) the irradiation coefficient φ_{12} falls both for the smooth channel and also for the channel with a spiral fitting. Over the whole range of variation of μ the value of φ_{12} for the channel with the spiral fitting is greater than for the smooth channel. This is due to the more uniform distribution of the particles in the channel with the spiral fitting, and also to their greater mobility resulting from the twisting effect and the impacts of the particles on the spiral, which increases their "visibility."

The coefficient φ_{21} rises with increasing F_1/F_2. The influence of the spiral fitting is only slight, since in the expression $\varphi_{21} = \delta\ F_1/F_2$ the decisive factor is F_1/F_2.

Analysis of the relation $\varepsilon_{re} = f(F_1/F_2)$ plotted from the experimental results obtained at high temperatures ($t_w > 500$°C) showed that ε_{re} increased with increasing surface area of the particles in the heat-transfer zone, and in the limit showed a value corresponding to the ε_{re} for a compact layer.

In analyzing the experimental data we used the emissivity of quartz given in [4] and that of 1Kh18 steel subjected to sand blasting given in [5] to represent the temperature dependence of the emissivity of the coolant ε_1 and that of the inner surface of the channel ε_2.

NOTATION

G	is the mass velocity of the coolant, g/sec;
T_f, t_f, t_{f1}, t_{f2}	are the average temperatures of the coolant and temperature of the coolant at the entrance to and exit from the heat-transfer section respectively;
T_w, t_w	is the average temperature of the heat-transfer section;
ΔT	is the logarithmic mean temperature difference;
q	is the thermal flux, W/m^2;
α_s	is the total heat transfer coefficient, $W/m^2 \cdot deg$;
α_r, α_c	are the radiative and convective heat-transfer coefficients respectively, $W/m^2 \cdot deg$;
F_1	is the total surface area of coolant particles, m^2;
F_2	is the internal surface area of heat-transfer section, m^2;
F_1/F_2	is the parameter;
σ_0	is the emissivity of an absolute black body, $W/m^2 \cdot {}^\circ K^4$;
ε_{re}	is the reduced emissivity of the system;
ε_1, ε_2	are the emissivities of the coolant and channel walls respectively;
φ_{12}, φ_{21}	are the irradiation coefficients;
H	is the mutual radiation surface;
μ	is the concentration of coolant.

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