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HEAT TRANSFER IN CONDENSATION ON A VERTICAL TUBE IN A GRANULAR BED

P. T. Petrik and A. R. Bogomolov

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A theoretical solution of the problem of heat transfer in vapor condensation on a surface in a granular bed was first published in [1-3], treating the process on flat surfaces.

We have previously [4] proposed a heat-transfer model (Fig. 1) for the film condensation of a saturated vapor on a vertical cylinder immersed in a granular bed, taking into account the nonlinearity of the resistance law associated with liquid filtration in the granular bed. Bearing in mind the results of our hydrodynamic studies of the flow of a liquid film along a vertical cylinder in a granular bed [5], where we demonstrated the existence of liquid crossflow from the cylinder into the interior of the layer, we write the equation of motion with allowance for inertial effects in the form

$$\mu u/k + \rho C u^2/k = \rho g(1 - \rho''/\rho), \tag{1}$$

where u is the velocity, m/sec, k is the permeability of the granular bed, m^2, μ is the dynamic viscosity coefficient, m^2/sec, ρ and ρ'' are the liquid and vapor densities, and C is the coefficient of inertia: $C = 0.55$ [6].

We solve Eq. (1) for the velocity:

$$u = kg(1 - \rho''/\rho)G/\nu, \quad G = [(4\mathrm{Ar}_i + 1)^{1/2} - 1]/(2\mathrm{Ar}_i). \tag{2}$$

Here ν is the kinematic viscosity coefficient, m^2/sec, and $\mathrm{Ar}_i = gk^{3/2}C(1 - \rho''/\rho)/\nu^2$ is a modified Archimedes number characterizing the degree of departure from Darcy's law. The dimensionless parameter $G \to 1$ in the limit $\mathrm{Ar}_i \to 0$, and $G \to 0$ in the limit $\mathrm{Ar}_i \to \infty$.

Assuming that the liquid and the porous medium are in local thermal equilibrium and that convective transfer can be disregarded, we write the energy equation in the form

$$\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = 0, \tag{3}$$

where r is the radial coordinate, m, and T is the temperature, K.

We adopt the following boundary conditions: the wall temperature is constant, and the boundary of the film exists at the saturation temperature. To account for the outflow of condensate, we assume that liquid is sucked from the surface of the film into the volume along the normal to the surface of the cylinder at a constant rate V. On this basis the mass balance equation for the condensate at the film boundary acquires the form

$$(\lambda/r_h)(\partial T/\partial r) = \rho u\, d(R + \delta)/dz + \rho V. \tag{4}$$

Here λ is the effective thermal conductivity, W/(m·K), r_h is the latent heat of condensation, J/kg, R is the radius of the tube, m, and δ is the thickness of the film, m.

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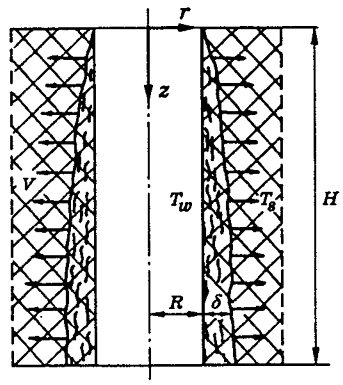

Fig. 1

Introducing the dimensionless variables

$$Z = za/(R^2u), \quad \eta = r/R, \quad \Theta = (T - T_w)/(T_s - T_w),$$
$$\Delta = (R + \delta)/R, \quad \psi = VR/a$$

(a is the effective thermal diffusivity, m^2/sec), we reduce Eqs. (3) and (4) to the form

$$\frac{\partial}{\partial\eta}\left(\eta\frac{\partial\Theta}{\partial\eta}\right) = 0. \tag{5}$$

The boundary conditions are as follows:

$$\Theta = 0 \text{ at } \eta = 1; \quad \Theta = 1 \text{ at } \eta = \Delta.$$

The mass balance equation of the condensate has the form

$$(1/\mathrm{Ku})(\partial\Theta/\partial\eta) = d\Delta/d\eta + \psi, \tag{6}$$

where $\mathrm{Ku} = r_h/(c_p\Delta T)$ is the Kutateladze number, and c_p is the specific heat at constant pressure, J/(kg·K). Integrating (5) with allowance for the boundary conditions, we obtain the expression for the temperature profile

$$\Theta = \ln\eta/\ln\Delta,$$

which we substitute into (6) to obtain an equation for the thickness of the condensate film:

$$d\Delta/dZ = (\mathrm{Ku}\Delta\ln\Delta)^{-1} - \psi. \tag{7}$$

Film condensation on a vertical, isothermal surface immersed in a layer has been studied experimentally [7] with the working substance R-11. The experimental results have been compared with the theoretical conclusions in [3].

The experiments were carried out on a vertical plate of length 76 mm and width 57 mm. The height of the bed was 25 mm.

Comparing the experimental results with the theoretical dependence [3], the authors failed to obtain satisfactory agreement. In our opinion, the disparity between theory and experiment might be attributable to: 1) the impossibility of creating sufficiently thick films in the given experiments at the indicated condensate flow rates, so that one of the principal requirements set forth in the theoretical analysis of the problem could not be satisfied, i.e., the condition $\delta/d \gg 1$ was not observed (d is the diameter of an element of the granular bed, m); 2) the disregard for any possible transfer of liquid from the condensation surface into the interior of the layer in the experiments on a vertical surface in the presence of a granular bed; 3) the reasonable

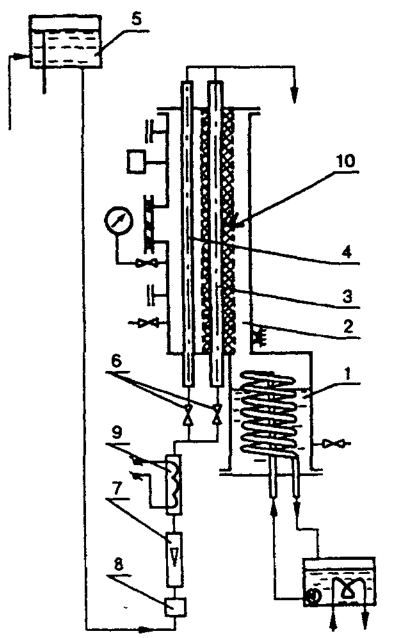

Fig. 2

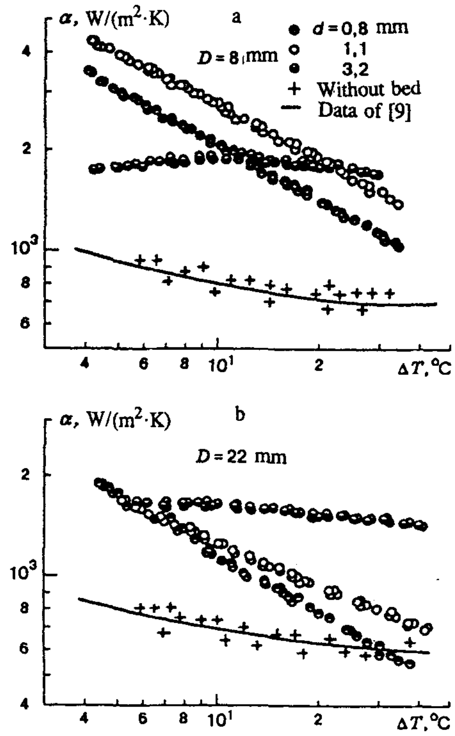

Fig. 3

assumption that the authors of the cited work did not adequately seal the test stand operating in vacuum during the experiments, so that atmospheric air could have penetrated the working volume of the condenser. This last assumption, in particular, is based on a comparison of the experimental data in the cited paper [7] with our own results [8] obtained in experiments on the condensation of R-12 vapor from vapor–air mixtures. Our comparison shows that the heat-transfer laws in [7] are similar to those obtained by us in condensation from mixtures.

In the literature we have not found any other experimental studies of heat transfer in vapor condensation on surfaces in a granular bed.

For the experimental investigation of heat transfer in condensation on a tube in a granular bed we used the test stand shown schematically in Fig. 2. The main body of the stand consisted of two offset, interconnected cylindrical vessels. The vessel 1 functioned as a boiler, and the vessel 2 as a condenser.

The condenser housed two test sections 3 and 4, the first serving as the working section, and the second as a control section.

The working section consisted of a tube encased in a steel mesh jacket. The space between the jacket and the tube was filled packed with a granular bed of steel pellets. Beds with pellets of diameter 0.8 mm, 1.1 mm, and 3.2 mm were used in the experiments. The temperature of the tube wall was determined from the averaged readings of 16 thermocouples caulked into the wall along a spiral curve.

Two different working sections were used in the experiments. The first had a length $H = 400$ mm and a diameter $D = 8$ mm; the corresponding dimensions of the second section were 770 mm and 22 mm. The control sections were tubes with dimensions matchings those of the working sections.

Heat was withdrawn from the sections by water admitted from the in-house water supply through the constant-level tank 5. Its flow was regulated and varied by means of the valve 6 and two rotameters: a turbine type 8 and a float type 7. The temperature of the coolant water was varied in the preheater 9. The frame of thermocouples 10 was set up to measure the temperature along the thickness of the bed in the working section.

The vapor temperature in the condenser was determined by measuring the saturation pressure and then calculating the temperature from the $P-T$ curve. All the tests were conducted at the saturation temperature $T_s = 60°C$.

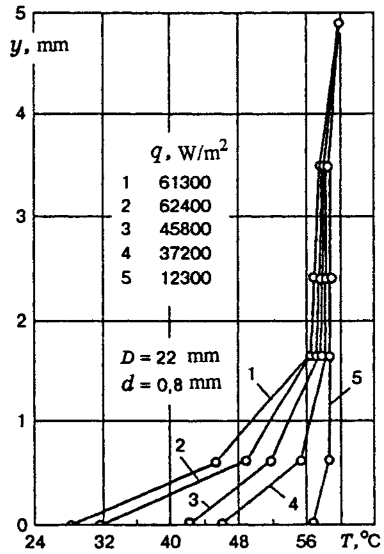

Fig. 4

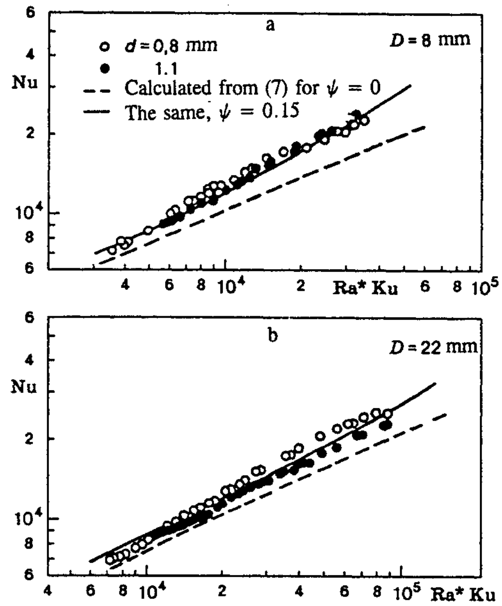

Fig. 5

The principal objective of the measurements was to determine the heat flux and the temperature drop, so that this information could be used to calculate the heat-transfer coefficient α.

The specific heat flux was estimated from the change of enthalpy of the coolant water. The validity of the measurements was checked prior to the main tests by conducting preliminary tests in the control sections. The results of these tests were compared with the data of other authors.

Figure 3 shows the experimental data in coordinates $\alpha = f(\Delta T)$, where $\Delta T = T_s - T_w$ is the temperature drop, and T_w is the wall temperature, K. We see that the experimental data for pellet beds with $d = 0.8$ mm and 1.1 mm are almost parallel to one another, and the slope of the lines drawn through these data corresponds to a power exponent of $-1/2$ for ΔT, consistent with the dependence given in [1]. The experimental data for the bed of 3.2-mm pellets in both tubes shows that the heat-transfer coefficients in this case remain almost constant for all the given values of ΔT. Results obtained in vapor condensation on the control section (a smooth tube) at the same saturation temperature are shown in the graphs for comparison. Clearly, they are situated far below the data for the tube packed with the granular bed.

We have performed experiments to determine the characteristics of the behavior of the liquid film flowing along the surface of a vertical cylinder in a granular bed [5]; the results show that the flow of liquid along the surface in a granular bed under these conditions cannot be described by the customary filtration laws, because the vertical position of the working section is accompanied by liquid crossflow induced by surface forces. The singular profile of the temperature distributed measured along the thickness of the bed in the condensation experiments (Fig. 4) also indicates the presence of condensate crossflow. Clearly, the experimental results differ from their counterpart obtained in condensation on an inclined plate [1, 2], which exhibits two distinct zones with different temperature distribution laws as a result of the difference in the heat resistance near the condensation surface and in the main condensate flow. In the case of condensation on a tube a third zone appears, characterized by the fact that the temperature in it remains practically constant over a fairly extended length, but without reaching the saturation temperature. This type of temperature distribution can be attributed to the fact that the temperature measurements were performed under conditions such that the subcooled condensate leaving the film flowed through the granular bed in separate jets, forming a two-phase flow zone.

Experimental data for granular beds consisting of round glass beads with $d = 1.1$ mm and 0.8 mm are compared with the analytical relation (7) in Fig. 5 [$\mathrm{Nu} = \alpha H/\lambda$ is the Nusselt number, and $\mathrm{Ra} = gkH(1\rho''/\rho)G/(a\nu)$ is a modified Rayleigh number]. The ordinary differential equation (7) has been solved numerically by the Runge–Kutta method.

The dimensionless suction rate ψ is determined from a comparison with experimental data on the average heat transfer and is taken equal to 0.15. The values of the effective heat-transfer coefficients are taken from experiments with condensation

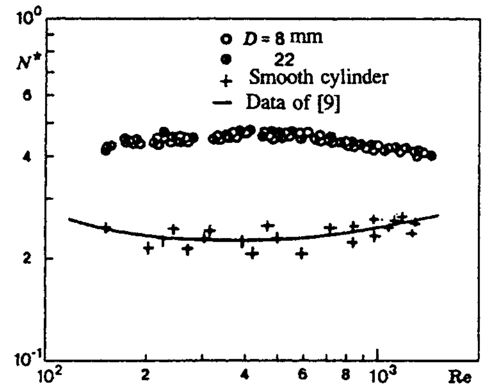

Fig. 6

on an inclined plate and can be calculated approximately from the well-known equation $\lambda = m\lambda_l + (1 - m)\lambda_T$. The permeability of the layers is calculated using the first term of the Ergun equation, according to which

$$k = m^3 d^2 / [A(1 - m)^2]$$

with our experimentally determined coefficient $A = 180$.

The porosity of the layers m is determined by a standard procedure; for granular beds with $d = 0.8$ mm and 1.1 mm we have $m = 0.358$ and 0.367, respectively. The graphs exhibit satisfactory agreement between the experimental and calculated results.

The experimental data for a granular bed consisting of round pellets with $d = 3.2$ mm cannot be described by Eq. (7), because in the experiments the film thickness was much smaller than the grain size. In Fig. 6 these results are compared with condensation data obtained on a smooth tube, where it is evident that a heat-transfer regime in which the modified Nusselt number $N^* = (\alpha/\lambda)(\nu^2/g)^{1/3}$ remains practically constant is observed over the entire range of the Reynolds number Re = $qH/(r_h\mu)$ (q is the heat flux density, W/m^2), and the experimental results lie far above the data for the surface without the granular bed.

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