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VAPOR CONDENSATION ON AN INCLINED PLATE WITHIN A POROUS MEDIUM

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In chemical technology, thermal power generation, and other branches of technology the process of heating a surface by condensing vapor is widespread. The process of vapor condensation on smooth surfaces has been studied quite thoroughly [1-6]. Theoretical and experimental studies in this field have expanded the concepts of processes occurring in vapor condensation and permitted development of a technique for engineering calculations of condensation equipment.

Recently a number of technological applications have turned to condensation processes occurring under more complex conditions, for example, in narrow slits, or on surfaces located within porous media. This problem has not been investigated thoroughly. Although the first theoretical treatments have appeared [7, 8], experimental studies of vapor condensation on surfaces located in a porous medium are absent from the literature.

We will consider the problem of vapor condensation on an inclined plane surface located within a porous medium (Fig. 1). The vapor condenses on the outer surface of a moving condensate film.

We make the following basic assumptions: 1) Inertial forces developing in the film are small in comparison to viscous and gravitational forces; 2) there is no friction at the liquid-vapor phase boundary, and the temperature of the outer surface of the condensate film remains constant at the saturation temperature; 3) heat transfer is accomplished by thermal conductivity (effective) of the liquid across the film, while heat transport in the longitudinal direction may be neglected; 4) the physical properties of the liquid are temperature-independent.

To calculate velocity profiles in the liquid film moving along the inclined surface we will use Brinkman's filtration equation [9], which in essence is a simple superposition of Darcy's law and the equation of viscous flow in a porous medium. For planar flow in a porous medium under the action of gravity it has the form

$$\mu' d^2u/dy^2 - \mu' u/\Pi + (\rho' - \rho'')g_\varphi = 0, \tag{1}$$

with boundary conditions $y = 0$, $u = 0$; $y = \delta$, $du/dy = 0$, where u is the dimensionless velocity of the motion and y is the transverse coordinate; $g_\varphi = g\cos\varphi$; φ is the angle between the surface studied and the vertical; μ', ρ' are the liquid viscosity and density; ρ'' is the vapor density; Π is the permeability of the porous medium and δ is the film thickness. The solution of Eq. (1) is:

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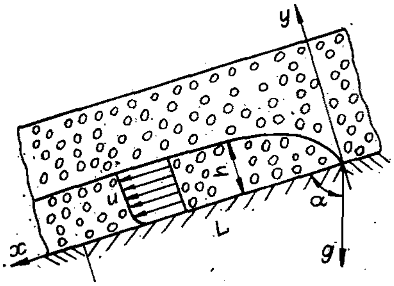

Fig. 1

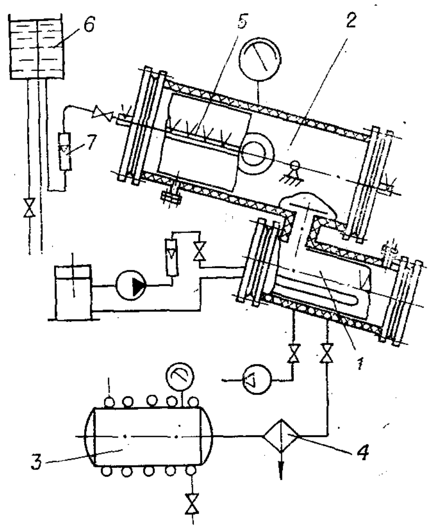

Fig. 2

$$u = \frac{\Pi}{\mu'}(\rho' - \rho'')\, g_\varphi \left(1 - e^{-y/\sqrt{\Pi}}\right).$$

It is evident that the velocity of liquid motion varies only within a thin layer near the wall and remains constant over the main part of the film. We define the thickness of the wall layer with the condition that at its boundary the velocity differs by 1% from the velocity at the outer surface of the film:

$$\delta_r \approx 4{,}6\sqrt{\Pi}.$$

We will consider only the case of "thick" films, where the film thickness is much larger than the size of the elements forming the medium. For these conditions

$$\bar{u} = \frac{\Pi}{\mu'}(\rho' - \rho'')\, g_\varphi. \tag{2}$$

With the assumptions made the energy equation has the form

$$d^2t/dy^2 = 0,\ y = 0,\ t = t_c;\quad y = \delta,\ t = t_H.$$

The solution of this equation is:

$$(t - t_c)/(t_H - t_c) = \frac{y}{\delta}.$$

The heat liberation coefficient

$$\alpha = \frac{q}{t_H - t_c} = \frac{\lambda_{ef}(dt/dy)_{y=0}}{t_H - t_c} = \frac{\lambda_{ef}}{\delta}.$$

To determine the condensate film thickness we use the equation

$$\frac{\lambda_{ef}}{\delta}(t_H - t_c) = r\frac{dG}{dx}, \tag{3}$$

where r is the heat of condensation; $G = \rho\bar{u}\delta$ is the condensate mass flow per unit slot width; λ_{ef} is the effective thermal conductivity coefficient.

The effective thermal conductivity in a granular layer is usually described by expressions of the form

$$\lambda_{ef}/\lambda = A + B\,\mathrm{Re}\,\mathrm{Pr}.$$

For a granular layer consisting of thermally nonconductive elements, A and B are constant values. In the range of low Re we may neglect the second term.

Substituting Eq. (2) in Eq. (3) and integrating, we obtain

$$\delta = \sqrt{\frac{2\lambda_{ef}\Delta t\mu' x}{\rho'\Pi r(\rho' - \rho'')\, g_\varphi}},\quad \alpha = \sqrt{\frac{\rho'(\rho' - \rho'')\Pi r g_\varphi \lambda_{ef}}{2\mu'\Delta t x}}.$$

The mean heat liberation coefficient

$$\bar{\alpha} = \frac{1}{L}\int_0^L \alpha(x)\,dx = \sqrt{\frac{2\rho'(\rho'-\rho'')\Pi r g_{\varphi}\lambda_{ef}}{\mu'\Delta T L}}.$$

In dimensionless form

$$\overline{\mathrm{Nu}} = (2\mathrm{Ar}^*\mathrm{Pr}K)^{1/2},$$

where

$$\mathrm{Ar}^* = \frac{g_{\varphi}L\Pi}{\nu^2}\left(1-\frac{\rho'}{\rho''}\right),\quad \mathrm{Pr}_{ef} = \frac{\nu}{a_{ef}},\quad K = \frac{r}{c\Delta T},\quad \mathrm{Nu} = \frac{\alpha L}{\lambda_{ef}},$$

or

$$\overline{\mathrm{Nu}} = \left(2\frac{\Pi}{L^2}\,\mathrm{Ar}\,\mathrm{Pr}\,\mathrm{K}\right)^{1/2},\quad \mathrm{Ar} = \frac{g_{\varphi}L^3}{\nu^2}\left(1-\frac{\rho''}{\rho'}\right). \tag{4}$$

Equation (4) coincides with the result obtained in [8] for large values of K. We may rewrite Eq. (4) in the form

$$\mathrm{N}^* = 2\mathrm{Re}^{-1}, \tag{5}$$

where

$$\mathrm{N}^* = \mathrm{Nu}/\mathrm{Ar}^*;\ \mathrm{Re} = qL/r\mu.$$

A spherical piece of equipment, shown schematically in Fig. 2, was designed and constructed to perform experiments on condensation in porous media. The major components are the evaporator 1 and condenser 2. The device was designed for an operating pressure of 1.5 MPa. The system was mounted in a manner such that the evaporator-condenser portion could be rotated around the horizontal axis, with inclination varying over the range 0-90°. Khladon-12 was used as the coolant liquid. The main advantage of Khladon materials is the insignificant and easily determinable impurity level, which has practically no effect on the condensation process [6]. For the initial filling the working vessels were first evacuated thoroughly, then coolant from receiver 3 was fed through filter 4 into the evaporator and condenser until they were completely filled with liquid coolant, driving any residual air from the condenser. After the vessels are filled the liquid from the condenser is pumped back into the receiver, only the evaporator remaining filled.

The condenser contains an active section, a diagram of which is shown in Fig. 3. A three-sided metal channel 8 is surrounded by a layer of thermal insulation 9. Plastic fins 1 with viewing windows 3 are installed on the sides of the channel. To measure the thermal flux liberated by coolant condensation the upper surface of the rectangular channel has a temperature measuring unit consisting of two resistance thermometers 6 separated by a plate 7. On top of this unit is a copper plate 4 with thermocouples 5. The space between plate 4 and grid 2 is filled by pouring in various materials. Water was circulated within the rectangular channel to absorb the heat liberated by coolant condensation. The cooling water was fed from a constant level vessel 6 (Fig. 2) through a flow meter 7 (Fig. 2). The outer surface of the evaporator and condenser were thermally insulated and provided with guard heaters. The temperature-measuring equipment was calibrated with a specially constructed device using the power level generated by an electric heater. When hot water from the thermostatic vessel was supplied to the evaporator heat exchange some fraction of the vapor entered the condenser and condensed in the operating section. The liquid coolant flowed along the inclined surface back into the evaporator. The guard heater maintained the temperature of the condenser body equal to (or a fraction of a degree higher than) the saturation temperature for a given pressure.

To measure the temperature distribution over the height of the granular layer a special movable bar carrying thermocouples was constructed and placed at various points in the layer. During the experiments the flow rates and temperatures of the heating and cooling water, and the temperature and pressure in the vapor space of the condenser were also recorded.

To verify proper operation experiments were performed involving condensation on the inclined plate in the absence of any packing material. The results of these experiments are shown in Fig. 4, where 1 is the Nusselt solution for a smooth film [1], written in the form [4]

$$\mathrm{N}^* = \overline{\mathrm{Nu}}\,\mathrm{Ar}^{-1/3} = 0.925\,\mathrm{Re}^{-1/3}, \tag{6}$$

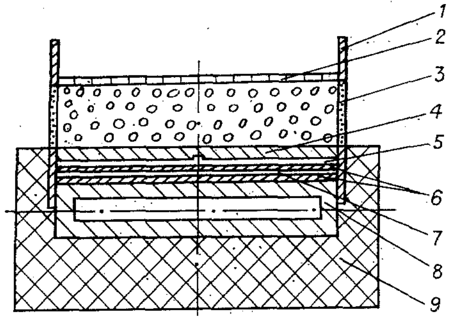

Fig. 3

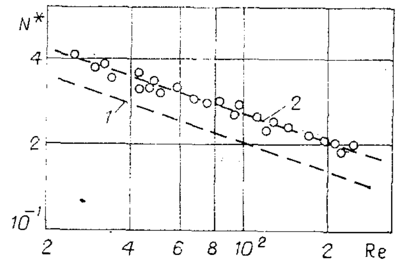

Fig. 4

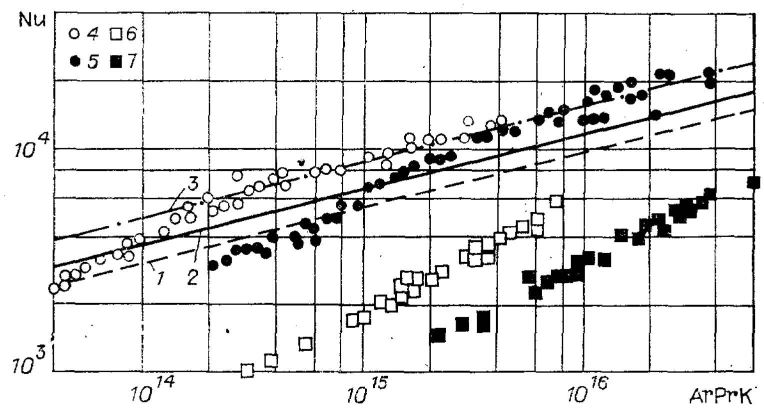

Fig. 5

which corresponds to $\overline{Nu} = 0.943(ArPrK)^{1/4}$; 2 is an expression considering the effect of wave formation on heat exchange [4],

$$N^* = 1.18\,Re^{-1/3} \tag{7}$$

and corresponding to the expression [2]

$$\overline{Nu} = 1.13\,(ArPr\,K)^{1/4}.$$

It is evident that the results obtained agree with Eq. (7).

The first series of experiments was performed using glass spheres 3.2 mm in diameter as a packing material. The condensate film thickness was varied from $3\cdot10^{-3}$ to $7\cdot10^{-3}$ mm, i.e., was much less than the glass sphere diameter. The results of these experiments are denoted by the digits 4 and 5 of Fig. 5. It is evident that the character of the dependence of Nu on ArPrK just as in Nusselt theory, but line 3, which averages the experimental data, is located approximately 30% higher than line 2, which corresponds to Eq. (7) [the number 1 denotes the curve obtained with Eq. (6)]. This can apparently be explained by the fact that due to increase in film thickness near the surface of the spheres (surface wettability) the film thickness decreases in the space between the spheres. Such intensification of heat exchange in condensation has also been found, for example, in [11].

Turning to Fig. 5, we note that with increase in liquid film thickness (decrease in the complex ArPrK) the experimental data deviate from the dependence Nu $\sim$ $(ArPrK)^{1/4}$ and begin to correspond to Nu $\sim$ $(ArPrK)^{1/2}$ (the braking effect of the spheres begins to be effective).

Thus, with increase in film thickness the character of the flow therein changes. The flow in a thin film is determined by the force of gravity and the friction force on the plate, and with increase in film thickness the mean velocity increases (the Nusselt solution). For a relatively thick film friction on the sphere surface produces a significant effect, and under such conditions the mean velocity in the film is independent of thickness (film motion in a granular medium).

Figure 5 also shows experimental results for coolant vapor condensation on sections of various lengths, with river sand used as the packing material, with a mean equivalent particle diameter of 0.8 mm (particle size varied from 0.6 to 1.0 mm). The results of these experiments are denoted by the digits 6 and 7. The properties of the granular medium (permeability, porosity, effective thermal conductivity coefficient) depend significantly on the

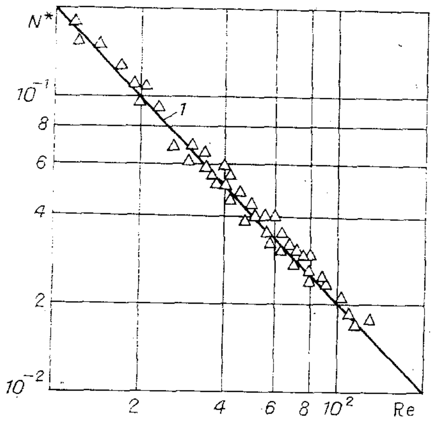

Fig. 6

concrete form of the elements and their packing, the latter being determined by the conditions under which the material is poured into the measurement volume. To determine the permeability (and porosity) of the sand a special working section was constructed which corresponded in form and size to the sand-filled channel of the main condensation experiments, the only difference being that the grid was replaced by a liquid-impermeable wall. The auxiliary and main channels were filled under identical conditions. Permeability and porosity of the pour were determined by the conventional method of [10].

Figure 6 shows the experimental results on heat exchange in vapor condensation on a plane inclined surface immersed in sand ($d \sim 0.8$ mm) in the form of the function $Nu/Ar^* \sim Re$. The effective thermal conductivity was calculated from readings of thermocouples located in the sand at various distances from the wall. As was noted above, with film flow of a liquid in a granular medium, the flow velocity is determined only by the permeability of the medium and the angle of inclination of the experimental surface, being independent of film thickness. As a result, Re, as determined from the liquid velocity in the pore space, the element diameter, and the liquid viscosity, varied over a relative narrow range ($Re_d \sim 3$-20) under our experimental conditions (for sand with $d \sim 0.8$ mm). Under these conditions the temperature change across the film thickness was linear and independent of Re_d.

The results obtained with regard to effective thermal conductivity agree qualitatively with data presented in [10] for the same Re range. The experimental data on heat exchange (Fig. 6) agree satisfactorily with the theoretical Eq. (5).

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