STUDY OF THE MECHANISM OF FILM CONDENSATION OF VAPOR UNDER THE INFLUENCE OF SURFACE FORCES

V. G. Rifert, P. A. Barabash,
A. I. Sardak, and A. N. Tobilevich

UDC 536.24:536.423

Measurements are made of local heat-transfer coefficients in the condensation of vapor with allowance for the effect of surface forces on hydrodynamics.

When a vapor condenses on a surface shaped in a particular manner (a fin), curvature of the condensate film results in a "Laplacian" pressure gradient dP/dx. Under the influence of this gradient, the film flows over the shortest possible path to the base of the fin. Despite the large number of theoretical and experimental studies of heat exchange on such surfaces — a detailed survey of which is offered in [1] — there are still no universal theoretical relations for heat transfer or substantiated recommendations on selecting a method and geometric parameters for shaping the surface, and there is substantial disagreement between the empirical and theoretical findings [1-4].

All of the theoretical studies adopt the following scheme of condensate flow on the shaped surface. The condensate flows toward the base of the fin (along the x axis) under the influence of a surface force and next to the base of the fin parallel to it under the influence of gravity. In solving the problem in the first zone of the pressure gradient dP/dx, the studies [2, 4-6, etc.] replaced the mean value $\Delta P/\Delta l$ between the high point (projection) and low point (depression) on the fin, where l is the height of the fin or half the spacing between the wire fins. Such a simplification makes it possible to obtain a finite relation for calculating the amount of vapor which condenses over the entire height of the fin. The thickness of the film over the fin height or between wires here changes roughly as $x^{1/4}$, i.e., as for the case of vapor condensation on a vertical wall in the gravity flow of a film. Only in the studies [3, 7-9] was dP/dx written exactly:

$$\frac{dP}{dx} = -\frac{d}{dx}\left\{\frac{\sigma d^2\delta/dx^2}{[1+(d\delta/dx)^2]^{3/2}}\right\}, \tag{1}$$

here (with T_W = const) we obtain the following nonlinear fourth-order equation for the condensate film:

$$\delta_x \frac{d}{dx}\left\{\delta_x^3 \frac{d}{dx}\left[\frac{d^2\delta_x/dx^2}{[1+(d\delta/dx)^2]^{3/2}}\right]\right\} = A, \tag{2}$$

where $A = 3\nu\lambda\Delta T/r\sigma$, with two boundary conditions at x = 0 ($d\delta/dx = 0$ and $d^3\delta/dx^3 = 0$) and two at the point of transition of the film into streams (the condition of smooth joining of the film and equality of the radius of curvature to its value R_p in the stream, which can always be calculated with known geometric parameters of the fin and a known condensate flow rate).

Equation (2) was solved in [7-9] by the Runge—Kutta method. Meanwhile, none of the studies indicated how the authors reduced the boundary-value problem to a Cauchy problem so as to make use of the above method. The solutions obtained in [7, 8] produced the profile of the film over the height of sinusoidal and triangular fins, which are characterized by thinning of the film next to the meniscus in the stream.

None of the well-known works experimentally verified theoretical solutions obtained for the section of condensate flow under the influence of dP/dx, and such verification is nearly impossible for small fins obtained by deformation of a wall. In the best case, measurements were made of the wall temperature at the base of fins or the mean temperature T_W for the entire surface and, accordingly, the mean heat-transfer coefficient $\bar{\alpha}_p$.

Kiev Polytechnic Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 49, No. 2, pp. 189-194, August, 1985. Original article submitted June 27, 1984.

 0022-0841/85/4902-0886$09.50 © 1986 Plenum Publishing Corporation

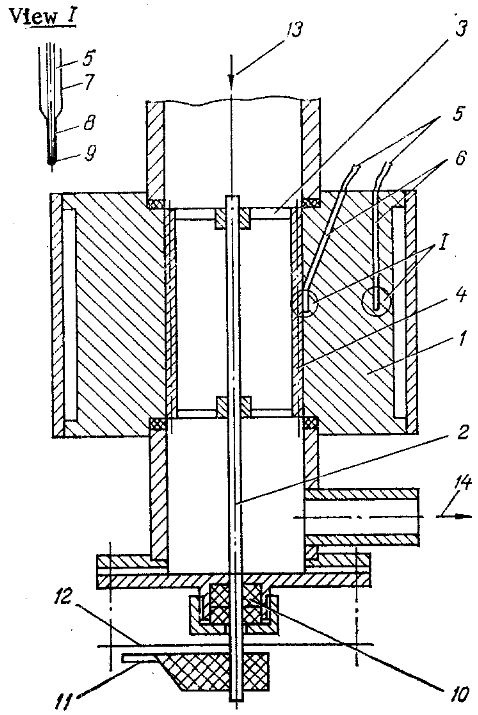


Fig. 1

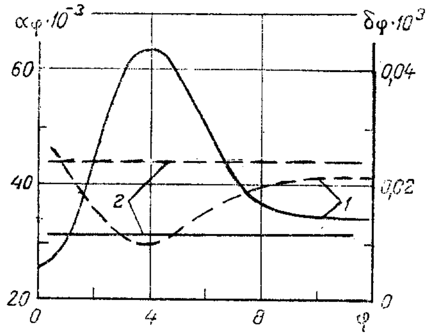


Fig. 2

Fig. 1. Experimental section.

Fig. 2. Change in local δ_φ and α_φ with w_V = 1.9 m/sec and q_φ = 375 kW/m^2; in the presence of fins; 2) without fins; solid lines) $\alpha_\varphi(\varphi)$; dashed lines) $\delta_\varphi(\varphi)$.

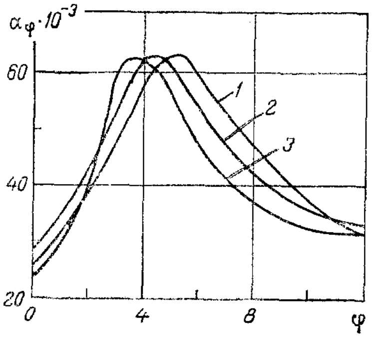
To create and intensify heat transfer during vapor condensation, the studies [6, 10] used a fin in the form of wires fastened along the generatrix for the vertical surface and in a spiral for the horizontal tube. An intensification of heat transfer here compared to the smooth tube was achieved with a distance between the wires s = 10-15 mm.

The use of such "fins" makes it possible to determine local heat-transfer coefficients with respect to x [11]. To do this, we prepared a test section (Fig. 1) in the form of a thick-walled tube made of brass and steel 1Kh18N10T with an inside diameter d = 30 mm and length L = 0.08 m. The outside diameter of the brass tube was 83 mm, while that of the steel 1Kh18N10T tube was d = 70 mm.

The section was positioned coaxially with a vertical tube L = 1.3 m. Inside the test section 1 we installed a frame consisting of a shaft 2 and spokes 3. Wires 1.7 mm in diameter were fastened to the ends of the spokes. The frame was installed in the tube so that the wires touched the walls of the tube 1 over its entire length. Copper—constantan thermocouples 5 with leads of diameter d = 0.1 mm were placed on one radius in the middle of the generatrix of the inside and outside walls of section 1. The thermocouples 5 were placed in a capillary tube 7. The thick end of the capillary tube 8, 0.4 mm in diameter, was welded to the thermocouple 5, forming the hot junction 9. The thus-prepared probes were introduced through channels 6 into the body of the working section 1. The probe on the inside surface of the tube was installed in a groove 10 mm long along the generatrix. The groove was then filled in with molten tin and the surface was ground.

The shafts of the frame were led outside the test section through the seal 10. A handle with a pointer 11 was attached to the outside of the seal, making it possible by means of scale 12 to determine the position of the frame with wires relative to the thermocouple to within 0.5°. Rotation of the frame made it possible to measure wall temperature at any distance from a wire (fin). Coolant water was pumped into the annular gap over the outside surface of the section.

The inside surface of the section 1 was machined to a class 7 finish in accordance with All-Union State Standard (GOST) 2789-73, carefully degreased with alcohol (the wires were also degreased), and filled with condensate for 24 h before testing.



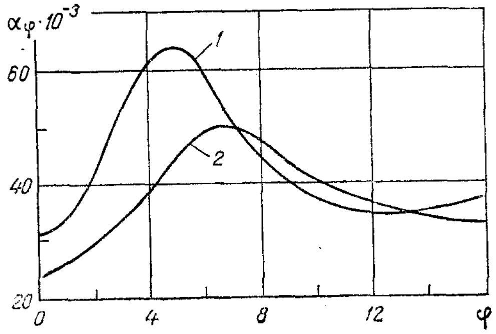


Fig. 3 Fig. 4

Fig. 3. Effect of the spacing s of the wires on the rate of heat transfer with $w_V = 1.9$; $q_\varphi = 375$ kW/m^2; 1) s = 5.85 mm; 2) 11.7; 3) 47.

Fig. 4. Effect of inclination of tube on $\alpha_\varphi = f(\varphi)$ with $w_V = 1.9$ m/sec and $q_\varphi = 375$ kW/m^2; 1) $\psi = 0$; 2) $\psi = 45°$.

The absence of drop condensation of vapor during the tests is evidence of the stable value of the thermocouple readings at different points of the tube perimeter during a significant period of time after the beginning of the steady-state regime.

The emf of the thermocouples, which were calibrated beforehand in a thermostat to within ±0.1°C, was measured with a Shch-68003 voltmeter.

To determine the local heat fluxes and heat-transfer coefficients, we numerically solved a heat-conduction equation written for an individual sector of the tube between a wire and the symmetry axis:

$$\frac{\partial^2 T}{\partial R^2} + \frac{1}{R}\frac{\partial T}{\partial R} + \frac{1}{R^2}\frac{\partial^2 T}{\partial \varphi^2} = 0 \tag{3}$$

with the boundary conditions: 1) $dT/d\varphi = 0$; on a radius passing through the wire and on the center between the wires; 2) empirically measured values of temperature were assigned on the contours of the surface.

The test data was analyzed on an ES computer. The analysis included a least-squares approximation of the empirical values of temperature using orthogonal Chebyshev polynomials, calculation of the temperature field by solving a system of n linear equations by the Gauss method, determination of the normal temperature gradient and local heat flux on the inside wall, and calculation of the local heat-transfer coefficients.

The experimental unit also included electric boilers to generate steam (13), a separator, an additional condenser for scavenge steam, condensate measuring vessels (14), a collection tank, and circulating pumps.

We condensed saturated water vapor at atmospheric pressure which was blown into the air for a certain period of time prior to the beginning of measurement. Vapor temperature was evaluated from the pressure measured by a standard manometer with graduations of 0.01 kg/mm^2 and was monitored with the thermocouple installed in the capillary tube in the vapor space in the measurement section.

This article presents results of tests involving vapor condensation inside section 1 (Fig. 1) in the case of absence of cooling of the tube ahead of 1. To eliminate the error possible for small values of q, the tests were conducted at the mean values $\bar{q}_\varphi \simeq 200$ and 400 kW/m^2 about the perimeter. The velocity of the steam at the temperature measurement site $w_V = 2$. The test section was positioned both strictly vertically and at a 45° angle to the horizontal.

For comparison with data on condensation in the absence of the influence of body forces, first we conducted tests without the frame with wires.

The test results are shown in Figs. 2-4. The curvature of the film away from the wire is close to the curvature of the tube, while $dP/dx \ll \rho g$. The large curvature of the film next to the wires leads to the appearance of dP/dx, so that the film is thinner all over

$x(\varphi)$ compared to a smooth tube. The minimum temperature gradients $T_n - T_w$ are obtained at the point of transition of the film flow on the axis $x(\varphi)$ into stream flow. At this point, there is also a large temperature gradient near the wall—film boundary and, accordingly, maximum values of local heat-transfer coefficients. It should also be noted that a quite appreciable (particularly for the steel 1Kh18N10T tube) change in wall temperature on the inside surface at the point of the minimum ΔT was detected by the instrument with the smallest rotation of the frame — by $\Delta\varphi = \pm 0{,}5^\circ$ which corresponds to $\Delta x = 0.12$ mm.

If we assume that (due to the thinness of the film) $\alpha_\varphi = \lambda/\delta_\varphi$ on the entire segment between the wires, then the measured α_φ can be used to evaluate δ_φ. Figure 2 shows thus-calculated film profiles with respect to φ. It can be seen that over much of the surface the thickness of the film is considerably less than for similar conditions on the smooth tube, which is also due to significant intensification of heat transfer with respect to the mean values of $\bar{\alpha}_l$ on the section l by a factor of up to three.

Figure 3 shows the effect of the distance s between wires on α_φ. Even at the maximum (in the tests) value of s, heat transfer is intensified compared to the smooth tube. This is evidence of the fact that surface forces are exerting an effect. Meanwhile, the values of α_φ and δ_φ near the meniscus are similar for different values of s, while a sharp reduction in α_φ is seen on a small segment $\Delta x = 0.5\text{-}0.8$ mm. This means that the effect of the surface force predominates over the effect of gravity only near the meniscus. The force of gravity is important on the rest of the surface away from the wire. More condensate flows toward the fin (Fig. 4) and the width of the stream increases when the gravity component is smaller in magnitude, as occurs with an inclination of the surface.

Analysis of the data obtained on the local heat-transfer coefficients shows the need to allow for gravity and interphase friction in the solution of problems involving condensation of a vapor on a shaped surface. Making such an allowance will make it possible to obtain more accurate relations to calculate mean heat-transfer for the entire surface. The laws obtained for the process will also be valid for fins obtained by deformation of near its base.

NOTATION

s, spacing of fins; $l = s/2$; x, coordinate directed over the fin height; φ, angular coordinate; R, radial coordinate; δ, film thickness; ν, σ, λ, ρ, r, kinematic viscosity, surface tension, thermal conductivity, density, and heat of vaporization; ψ, angle of inclination of surface from the vertical; ΔT, temperature drop between vapor and wall; w_v, velocity of vapor at temperature measurement site; α_φ, local heat-transfer coefficient; $\bar{\alpha}_l$, mean heat-transfer coefficient on a section of length l; q_φ, heat flux on the inside surface of the tube averaged over φ.

LITERATURE CITED

1. V. G. Rifert, "Vapor condensation on a shaped surface," in: Energy and Mass Transfer Processes in Capillary-Porous Substances with Phase Transformations [in Russian], A. V. Lykov ITMO, Akad. Nauk Belorussian SSR (1982), pp. 149-170.
2. A. P. Solodov and V. P. Isachenko, "Study of heat-transfer in the condensation of vapor on fine-fiber tubes," in: Heat Exchange and Hydraulic Resistance [in Russian], Moscow Energy Institute (1966), pp. 85-98.
3. Markovich, Mikich, and Bergles, "Condensation on an inverted fibrous horizontal surface," Teploperedacha, 94, No. 3, 62-69 (1972).
4. V. A. Karkhu, N. V. Zozulya, V. A. Permyakov, et al., "Study of the effectiveness of using vertical finned tubes in low-pressure regenerative heaters," Trudy TsKTI, 140, 80-88 (1977).
5. N. V. Zozulya and V. A. Karkhu, "Analysis of the process of film condensation of a vapor on a fine-finned vertical surface," Zh. Prikl. Mekh. Tekh. Fiz., No. 3, 93-97 (1969).
6. V. G. Rifert and G. G. Leont'ev, "Analysis of heat transfer in the condensation of vapor on a vertical tube with wires to intensify the process," Teploénergetika, No. 4, 78-80 (1976).
7. Mori, Khidzikata, Kharasava, and Kakalyasi, "Optimization of the characteristics of condensers with external condensation surfaces," Teploperedacha, 103, No. 1, 116-124 (1981).

8. C. B. Panchal and K. I. Bell, "Analysis of Nusselt-type condensation on a vertical fluted surface," Numerical Heat Transfer, 3, No. 3, 357-371 (1980).
9. A. A. Babenko, M. M. Levitan, and D. I. Khrustalev, "Heat transfer in condensation on a surface with grooves," Inzh.-Fiz. Zh., 40, No. 6, 1022-1028 (1981).
10. V. G. Rifert, P. A. Barabash, and A. B. Golubev, "Rate of condensation of steam on horizontal tubes shaped with wire," Izv. Vyssh. Uchebn. Zaved., Énerg., No. 7, 106-110 (1980).
11. V. G. Rifert and A. N. Tobilevich, "Study of the mechanism of vapor condensation in a shaped tube," in: Physicochemical Processes in Power Plants [in Russian], A. V. Lykov ITMO, Akad. Nauk Belorussian SSR, Minsk (1983), pp. 144-148.

CONDENSATION COARSENING OF AEROSOL PARTICLES IN A COOLING VAPOR—GAS FLOW

E. V. Sugak and V. P. Isakov

UDC 621.175:541.182.2

A system of equations is obtained to describe the motion of a vapor—gas flow with aerosol particles in a channel of variable cross-sectional area in the presence of external heat transfer. The system is solved analytically for a quasiequilibrium process.

The principle of the condensation-associated coarsening of highly dispersed aerosol particles during the cooling of a vapor—gas flow is used to increase the efficiency of several existing wet-type dust catchers and separators [1-5] and to develop new designs [6-11]. However, broad use of the method is being held up by insufficient research on the process, which makes it difficult to develop simple, effective, and economical equipment for the fine cleaning of gases.

The goal of the present study is to obtain a model of the process of condensation coarsening of aerosol particles in a cooling vapor—gas flow with a variable cross section along its length.

The cooling of a vapor—gas mixture moving along a colder surface is accompanied by heat transfer through the gaseous boundary layer adjacent to it and subsequent condensation. If the mixture contains a disperse phase (solid or liquid aerosol particles), then condensation occurs not only on the surface of the channel but also on the particles. Also, in the event of significant supersaturation, nuclei are also formed — small drops of liquid (homogeneous condensation). Considering the small diameter of the nuclei (about 10^{-9} m [12]) and assuming that supersaturation is negligible (significant supersaturation being possible only in a pure gas or at very high cooling rates) and that the concentration of the disperse phase is relatively high, it can be reasoned that the content of liquid in nuclei is negligibly small and that the process of their formation need not be examined in the present case.

The relationship between the masses of condensate formed on the particles and on the channel surface undergoing cooling depends on the amount of supersaturation and the concentration of the disperse phase in the mixture. Amelin proved theoretically and it was confirmed empirically [12] that in the case of a large number of condensation nuclei (aerosol particles) in a flow, considerably more vapor condenses on them than on the walls. Thus, with a numerical concentration of particles of 10^8 m^{-3}, 99% of all condensate is formed on them [12]. Condensation on the walls may be substantial only in the case of their film spraying or when they have a developed surface.

Condensation growth of aerosol particles can be speed up by increasing the partial vapor pressure by expanding the vapor—gas flow in various types of diffusers as a result of a reduction in its velocity. Landau and Lifshitz showed [13] that vapor condensation is

Siberian Engineering Institute. Leningrad Engineering Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 49, No. 2, pp. 194-201, August, 1985. Original article submitted July 2, 1984.

 0022-0841/85/4902-0890$09.50 © 1986 Plenum Publishing Corporation