6. V. P. Isachenko, Heat Exchange during Condensation [in Russian], Énergiya, Moscow (1977).
7. S. S. Kutateladze, Fundamentals of Heat Exchange Theory [in Russian], Atomizdat, Moscow (1979).
8. N. I. Mirmov and I. G. Belyakova, "Some results of a study of thermosiphons with binary heat-exchange agents," Inzh.-Fiz. Zh., 38, No. 5, 934 (1980).
9. E. M. Novokhatskii and A. N. Gorovoi, "Internal thermal resistance of a thermosiphon," Izv. Vyssh. Uchebn. Zaved., Énerget., No. 5, 87-92 (1978).
10. S. Bretshnaider, Properties of Gases and Liquids [in Russian], Khimiya, Leningrad (1966).

LOCAL HEAT EXCHANGE IN THE FILM CONDENSATION OF A STATIONARY VAPOR ON A VERTICAL SURFACE

G. I. Gimbutis

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A method is described for calculating the distributions of the heat-transfer coefficient and heat flux over the height of a condensation surface under all regimes of condensate film flow. Data on these distributions are also presented.

In designing condensers and calculating vapor condensation, one is usually dealing with mean values of the heat-transfer coefficient and heat flux over the entire height of the condensation surface. However, in certain cases the designer may be able to use such information as the distribution of local heat flux over the height of the condenser. These data could be valuable, for example, in organizing efficient delivery of the vapor to the condensation surface.

In the film condensation of vapor, local heat flux can be expressed as follows:

$$q_c = (T_s - T_f)\Big/\left(\frac{1}{\alpha} + R_c + \frac{1}{\alpha_1}\right). \tag{1}$$

The change in R_c and $1/\alpha_1$ over the height of the condensation surface can often be ignored in a first approximation. Thus, the change in q_c depends primarily on the change in T_f and α. For T_f, we can write

$$T_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^{x_s} \varepsilon_q dx_s, \tag{2}$$

if the heat-removing medium and condensation film constitute a forward flow. On the other hand,

$$T_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_1^{x_s} \varepsilon_q dx_s, \tag{3}$$

if there is a counter flow.

Since $\int_0^1 T_f dx_s = \overline{T}_f$, then Eqs. (2) and (3) respectively yield

$$\overline{T}_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^1 dx_s \int_0^{x_s} \varepsilon_q dx_s, \tag{4}$$

or

$$\overline{T}_f = T_{f_1} + (T_{f_2} - T_{f_1}) \int_0^1 dx_s \int_1^{x_s} \varepsilon_q dx_s. \tag{5}$$

According to [1], the heat-transfer coefficient for a nonisothermal surface should be averaged in conformity with the law

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$$\bar{\alpha} = \frac{\bar{q}_c}{T_s - \bar{T}_c} = \frac{\int_0^1 q_c dx_s}{\int_0^1 (T_s - T_c)\, dx_s}. \tag{6}$$

Then the mean heat flux

$$\bar{q}_c = (T_s - \bar{T}_f) \Big/ \left(\frac{1}{\bar{\alpha}} + R_c + \frac{1}{\alpha_1} \right). \tag{7}$$

Simultaneous solution of Eqs. (1), (2), (4), and (7) leads to the following relations between ϵ_q and ϵ_α for forward flow:

$$\varepsilon_q = \frac{\left(1 - \theta_1 \int_0^{x_s} \varepsilon_q dx_s\right)(1 + A)}{\left(1 - \theta_1 \int_0^1 dx_s \int_0^{x_s} \varepsilon_q dx_s\right)\left(\frac{1}{\varepsilon_\alpha} + A\right)}. \tag{8}$$

Accordingly, for counter flow

$$\varepsilon_q = \frac{\left(1 - \theta_1 \int_1^{x_s} \varepsilon_q dx_s\right)(1 + A)}{\left(1 - \theta_1 \int_0^1 dx_s \int_1^{x_s} \varepsilon_q dx_s\right)\left(\frac{1}{\varepsilon_\alpha} + A\right)}. \tag{9}$$

An additional relation between ϵ_q and ϵ_α can be found from the equation of the increment in condensate flow rate

$$\frac{d\Gamma}{dx} = \frac{q_c}{r}, \tag{10}$$

which can be reduced to the form

$$\frac{d\,\mathrm{Re}}{dx_s} = \mathrm{Re}_m \varepsilon_q, \tag{11}$$

if we consider that at $r \gg c(T_s - T_c)$ $\mathrm{Re}_m = 4\bar{q}_c H/(r\rho\nu)$.

In vapor condensation, it is usually the case that Re = 0 if $x_s = 0$. Then, having integrated (11), we obtain

$$\frac{\mathrm{Re}}{\mathrm{Re}_m} = \int_0^{x_s} \varepsilon_q dx_s. \tag{12}$$

If α is averaged according to (6), then $\int_0^1 \vartheta_c dx_s = 1$. Substituting this expression into (11) and integrating it over the entire ranges of x_s and Re – here considering $\varepsilon_q = \varepsilon_\alpha \vartheta_c$ – we obtain the following relation between $\overline{\mathrm{Nu}}_m$ and Nu_m:

$$\overline{\mathrm{Nu}}_m = \mathrm{Re}_m \left(\int_0^{\mathrm{Re}_m} \frac{d\,\mathrm{Re}}{\mathrm{Nu}_m} \right)^{-1}. \tag{13}$$

In vapor condensation in heat-exchange theory, it is assumed that the heat flux does not change across the film when $r[c(T_b - T_c)]^{-1} > 5$ and that the function $\mathrm{Nu}_m = f(\mathrm{Re})$ is independent of the character of nonisothermality of the condensation surface [1, 2]. In this case, as can be seen from Eq. (13), the function $\overline{\mathrm{Nu}}_m = f(\mathrm{Re}_m)$ is also independent of the character of nonisothermality of the condensation surface when the surface is vertical.

When the functions $\mathrm{Nu}_m = f(\mathrm{Re})$ and $\overline{\mathrm{Nu}}_m = f(\mathrm{Re}_m)$ are independent of the character of change in the temperature of the condensation surface, the ratio $\mathrm{Re}/\mathrm{Re}_m$ unambiguously determines the value of ϵ_α at given Re_m and Pr. Then, solution of systems (8) and (12) or (9) and (12) by the method of successive approximations allows us to find the relations $\epsilon_\alpha = f(x_s)$ and $\epsilon_q = f(x_s)$, i.e., the distribution of the heat-transfer coefficient and heat flux over the height of the condensation surface with given Re_m, Pr, A, and θ_1. For this, it is necessary only to have the specific relation $Nu_m = f(Re, Pr)$ for film condensation of vapor on a vertical surface.

Such a relation has yet to be established experimentally. We will therefore use the empirical relation $Nu_m = f(Re, Pr)$ presented in [3]. It was obtained with allowance for the transfer of momentum and heat caused by wave and turbulence processes occurring in a gravitational liquid film. The relation therefore embraces all regimes of flow of the film. This relation and the corresponding function $\overline{Nu}_m = f(Re_m, Pr)$ determined from Eq. (13) are shown in Fig. 1.

To facilitate calculations on the computer, the data in Fig. 1 can be adequately described by the semiempirical relations

$$Nu_m = 1.1\,Re^{-1/3}(1 + 0.02\,Re^{0.2} + 0.0009\,Re^{0.85}Pr^{0.65}) \tag{14}$$

and

$$\overline{Nu}_m = 1.47\,Re_m^{-1/3}(1 + 0.03\,Re_m^{0.2} + 0.00075\,Re_m^{0.8}Pr^{0.6}). \tag{15}$$

If the expressions in parentheses in Eqs. (14) and (15) are omitted, they become the theoretical equations which follow from the Nusselt theory for strictly laminar film flow. These expressions may therefore be regarded as a correction factor, allowing for the increase in heat transfer in wavy and turbulent film flows compared to strictly laminar flow.

Figure 2 shows values of ϵ_α and ϵ_q calculated by the above method for certain characteristic values of Re_m, Pr, A, and θ_1. The similar character of the relations $\epsilon_\alpha = f(x_s)$ and $\epsilon_q = f(x_s)$ is also seen at other Pr numbers.

The data in Fig. 2 give us a picture of the distribution of the rate of vapor condensation over the height of a condensation surface, which might prove useful in organizing efficient delivery of vapor to the surface.

As already noted, there are no literature data on local heat transfer in the film condensation of vapor on a vertical surface. We can therefore only indirectly compare the above calculated results with experimental results.

The literature contains results from several experimental studies of heat transfer in the surface evaporation (without nucleate boiling) of a stabilized falling liquid film heated to the saturation temperature and descending over a vertical surface. In this case, as in the case of vapor condensation, heat transfer takes place at a nearly constant heat flux across the film. Thus, if we do not consider the effect of such factors as inertial forces in the film and changes in the physical properties of the liquid on heat transfer, then the theoretical relation $Nu_m = f(Re, Pr)$ is the same for both film condensation of a vapor and for the surface evaporation of a film heated to T_s. Thus, it is proper to compare the data in Fig. 1a with experimental data on heat transfer in the surface evaporation of a gravitational liquid film heated to T_s. Such a comparison is shown in Fig. 3 and, as can be seen, the agreement between the theory and the experiment, as well as Eq. (14), is quite satisfactory.

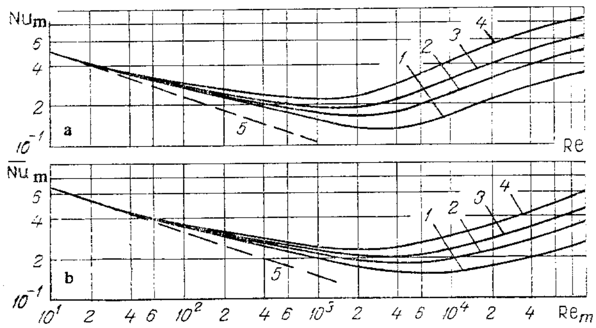

Fig. 1. Theoretical data on heat transfer at a constant heat flux across a gravitational film flowing down a vertical surface: a) local heat transfer; b) mean heat transfer in the film condensation of a stationary vapor; 1, 2, 3, 4) at Pr = 1, 2, 3, 5, respectively [3]; 5) Nusselt theory for strictly laminar film flow.

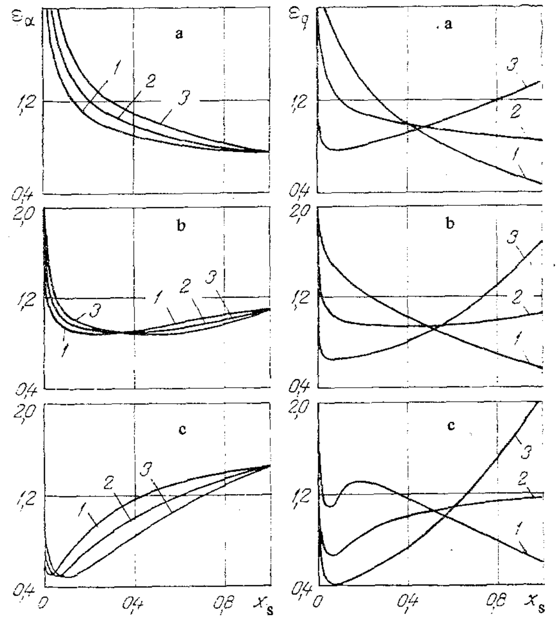

Fig. 2. Distribution of heat-transfer coefficient and heat flux over height of condensation surface at A = 1, Pr = 3; a) Re_m = 50; b) 5000; c) 50,000; 1) forward flow, θ_1 = 2/3; 2) θ_1 = 0; 3) counter flow, θ_1 = 2/3.

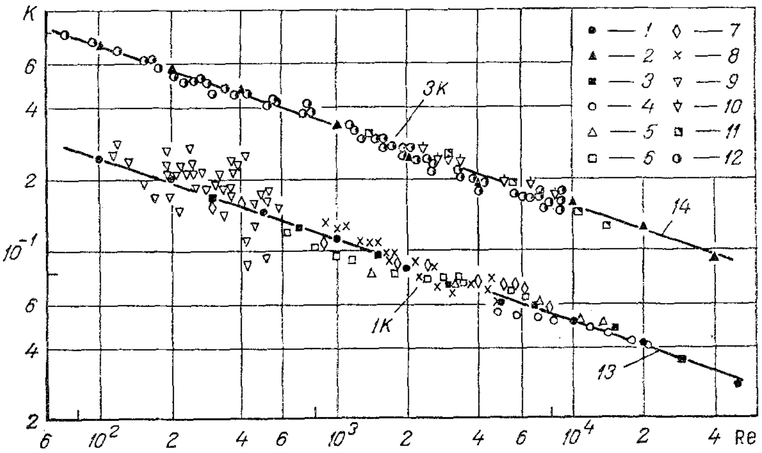

Fig. 3. Generalized data on heat transfer in the surface evaporation of a gravitational liquid film heated to T_s and flowing down a vertical surface: 1, 2, 3) theoretical data for Pr = 2, 3, 5, respectively; 4, 5, 6, 7) experimental data, water, Pr = 1.77; 2.91; 5; 5.7 [4]; 8) experimental data, water, Pr = 1.75 [5]; 9) experimental data, salt water, Pr = 1.6-2.2 [6]; 10) experimental data, water, Pr = 1.75 [7]; 11) experimental data, water, Pr = 1.75 [8]; 12) experimental data, Khladon 11, Pr = 4.16 [9]; 13 and 14) from Eq. (14); $K = Nu_m/(1 + 0.02\,Re^{0.2} + 0.0009\,Re^{0.85}Pr^{0.65})$.

The data in Fig. 1b can be compared with experimental data on the mean heat transfer in the film condensation of a stationary vapor on a vertical surface. Such a comparison is shown in Fig. 4 and illustrates that all of the values agree quite satisfactorily.

Thus, analysis of Figs. 3 and 4 leads us to the conclusion that the above-established relations $\epsilon_\alpha = f(x_s)$ and $\epsilon_q = f(x_s)$ (see Fig. 2) correctly reflect the distribution of the heat-transfer coefficient and heat flux over the height of a condensation surface in the film condensation of a pure stationary vapor.

It is also apparent from Figs. 3 and 4 that Eq. (14) can be used for engineering calculations of heat transfer in the

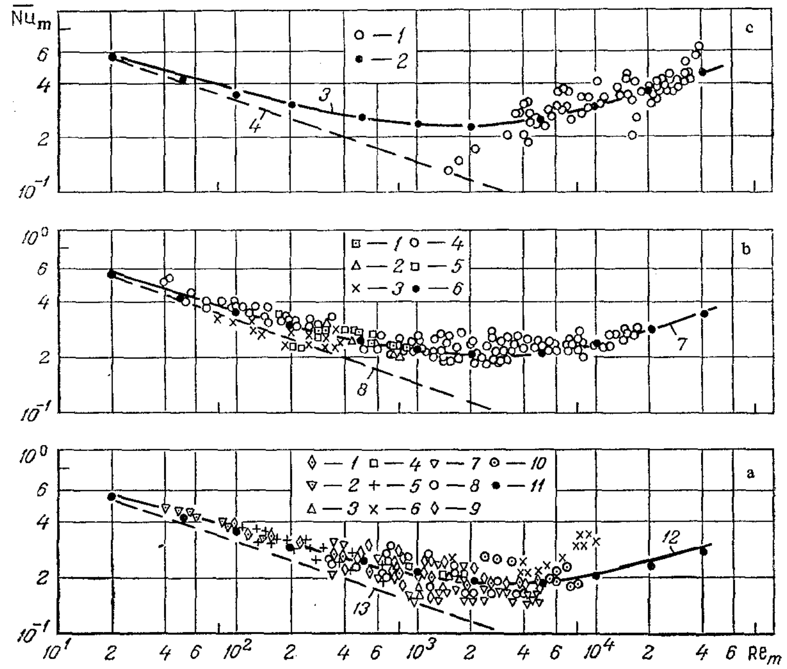

Fig. 4. Mean heat transfer in the film condensation of a pure stationary vapor on a vertical surface: a) Pr ≃ 2; 1) data of Kutateladze; 2) Shekriladze–Ratiani; 3) Lozhkin–Kanaev; 4) Zozulya; 5) Burov; 6) Gudemchuk; 7) Strobe; 8) Gebbard; 9) Maisenburg; 10) Jacob–Érk–Éck (data on stream condensation taken from [1, 10, 11, 12]); 11) theoretical calculation with Eq. (13) at Pr = 2; 12) calculation with Eq. (15) at Pr = 2; 13) Nusselt theory with strictly laminar film flow; b) Pr ≃ 3; data of Zozulya, steam [11]; 2) Kozitskii, Khladon 12 vapor [12]; 13) Mazyukevich, Khladon 12 vapor [12]; 4) Gogonin–Dorokhov–Sosunov, Khladon 21 vapor [12]; 5) Kozitskii, Khladon 22 vapor [12]; 6) theoretical calculation with Eq. (13) at Pr = 3; 7) calculation with Eq. (15) at Pr = 3; 8) Nusselt theory; c) Pr ≃ 5; 1) results of Badger, diphenyl vapor [13]; 2) theoretical calculation with Eq. (13) at Pr = 5; 3) calculation with Eq. (15) at Pr = 5; 4) Nusselt theory.

surface evaporation of a gravitational liquid film heated to the saturation temperature and flowing over a vertical surface, while Eq. (15) can serve the same purpose for the film condensation of a pure stationary vapor on a vertical surface.

NOTATION

$Nu_m = \frac{\alpha}{\lambda}\left(\frac{\nu^2}{g}\frac{\rho}{\rho-\rho_0}\right)^{1/3}$; $\overline{Nu}_m = \frac{\bar{\alpha}}{\lambda}\left(\frac{\nu^2}{g}\frac{\rho}{\rho-\rho_0}\right)^{1/3}$; $Re = 4\Gamma/(\rho\nu)$; $Re_m = 4\Gamma_m/(\rho\nu)$; $Pr = c\rho\nu/\lambda$; $Z = \frac{\lambda H(T_s - \bar{T}_c)}{r\rho\nu^{5/3}}\left(g\frac{\rho-\rho_0}{\rho}\right)^{1/3}$; $A = \left(R_c + \frac{1}{\alpha_1}\right)\bar{\alpha}$; $x_s = x/H$; $\varepsilon_q = q_c/\bar{q}_c$; $\varepsilon_\alpha = \alpha/\bar{\alpha} = Nu_m/\overline{Nu}_m$; $\vartheta_c = (T_s - T_c)/(T_s - \bar{T}_c)$; $\theta_1 = (T_{f_2} - T_{f_1})/(T_s - T_{f_1})$; α, $\bar{\alpha}$, local and mean heat-transfer coefficients for heat transfer from the condensation direction; α_1, coefficient for heat transfer from the direction of the heat-removing medium; R_c, heat resistance of the wall; λ, thermal conductivity of the liquid; ν, kinematic viscosity of the liquid; c, specific heat of the liquid; ρ, density; ρ_0, density of the vapor (gas); r, heat of phase transformation; x, coordinate along the condensate flow; H, height of condensation surface; q_c, $\bar{q}_c$, local and mean heat flux on wall; Γ, Γ_m, local and maximum density of condensate spray; T_s, saturation temperature; T_c, $\bar{T}_c$, local and mean temperature of condensation surface; T_f, T_{f_1}, T_{f_2}, $\bar{T}_f$, local, initial, final, and mean temperature of heat-removing medium; g, gravitational constant.

LITERATURE CITED

1. V. P. Isachenko, Heat Exchange in Condensation [in Russian], Énergiya, Moscow (1977).
2. S. S. Kutateladze, Principles of the Theory of Heat Exchange [in Russian], Atomizdat, Moscow (1979).
3. G. I. Gimbutis, Heat Exchange in the Film Condensation of Vapor and Surface Evaporation of a Gravitational Film of Liquid Heated to the Saturation Temperature, Deposited at VINITI (All-Union Scientific-Research Institute of Technical Information), No. 2287 (1980).

4. K. R. Chun and R. A. Seban, "Heat transfer to evaporating liquid films," J. Heat Transfer, 93, No. 4, 391-396 (1971).
5. T. Fujita and T. Ueda, "Heat transfer to falling liquid films and film breakdown," Int. J. Heat Mass Transfer, 21, 109-118 (1978).
6. W. Unterberg and D. K. Edwards, "Evaporation from falling saline water films in laminar transitional flow," AIChE J., 11, No. 6, 1073-1080 (1965).
7. V. G. Rifert, "Heat transfer in the vaporization of a liquid film flowing over a shaped vertical surface," Inzh.-Fiz. Zh., 39, No. 5, 833-837 (1980).
8. Yu. M. Tananaiko, "Study of heat transfer during boiling in flowing films," in: Heat and Mass Transfer [in Russian], Vol. 2, Nauka i Tekhnika, Minsk (1968), pp. 173-179.
9. H. Struve, "Der Wärmeubergang and einem verdampfenden Rieselfilm," VDI-Forschungsh., Düsseldorf, No. 534 (1969).
10. D. A. Labuntsov, "Heat transfer in the film condensation of pure vapors on a vertical surface and horizontal tubes," Teploénergetika, No. 7, 72-80 (1957).
11. N. V. Zozulya, "Study of heat transfer in vapor condensation on vertical tubes," in: Heat Transfer and Thermal Modeling [in Russian], Izd. Akad. Nauk SSSR, Moscow (1959), pp. 287-297.
12. I. I. Goginin, A. R. Dorokhov, and V. I. Sosunov, "Heat exchange in the film condensation of a stationary vapor on a vertical surface," in: Heat Transfer in Boiling and Condensation [in Russian], Inst. Teplofiz. Sib. Otd. Akad. Nauk SSSR, Novosibirsk (1978), pp. 60-76.
13. G. Greber, S. Érk, and U. Grigul', Principles of Heat Exchange Science [Russian translation], IL, Moscow (1958).

APPLICABILITY OF BOUNDARY-LAYER THEORY TO CALCULATION OF HEAT TRANSFER UNDER SEPARATION CONDITIONS

Yu. F. Gortyshov and I. M. Varfolomeev **UDC 536.25**

The conditions are examined under which methods and relations developed for attachment flow are applicable to regions of separation flow.

Recently, the number of experimental and theoretical studies has been increasing where the authors examine the possibility of calculating the processes during separation by the method based on the theory of a boundary layer in nonseparation streamlining, this method having been rather thoroughly tested for a wide range of applications. The problem has not only great practical but also fundamental theoretical importance, inasmuch as studied pertaining to it will reveal differences between the physics of separation flow and the physics of attachment flow. No single view on this matter has yet been developed, apparently because of the multitude of modes and forms of separation flow. There is no doubt that the possibility of applying the relations varied for attachment flow to conditions of separation flow must be examined individually in each specific case.

An important aspect of the problem is determining whether there exists an analogy between friction and heat transfer in separation flow. For determining the friction in this study the authors used a known indirect method [1, 2]. In accordance with that method, the frictional velocity (or dynamic velocity) was selected so as to ensure the required direction of the measured velocity near the wall. The method was used here for determining the friction in two-dimensional grooves streamlined by a compressible gas. The study covered a wide range of parameter variation: of the relative groove depth $\bar{H} = H/L$ from 0 to 1.0 and the Reynolds number $N_{Re} = u_e x/\nu$ in the stream core from $2.5 \cdot 10^5$ to $3.5 \cdot 10^6$, with the Mach number N_{Ma} equal to 3.5, 4.0, and 4.5 successively.

On the graph in Fig. 1 are indicated the readings of velocity obtained with a total-head Pitot tube near the wall at four sections along the x-coordinate in grooves of various depths. The frictional velocity had been selected so as to make the experimental points fit on the Karman curve for a buffer layer with $u^* = -3.05 + 5$ (ln y^*) within the $5 \leqslant y^* \leqslant 30$ range, with $u^* = u/u_\tau$ and $y^* = yu_\tau/N$.

"A. N. Tupolev" Kazan Institute of Aviation. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 43, No. 3, pp. 397-401, September, 1982. Original article submitted March 17, 1981.

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