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Heat transfer for film condensation of vapour

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Nomenclature

a - thermal diffusivity, m²/s; c_p - specific heat at constant pressure, J/(kg·K); D - tube outside diameter, m; g - acceleration of gravity, m/s²; h - condensation surface height, m; k - phase transformation number $r/c_p(T_{sat}-T_w)$; L - wetted surface or heat exchange length, m; Nu_M - modified Nusselt number $(\alpha/\lambda)/(\nu^2/g)^{1/3}$; $\overline{Nu}_M$ - mean modified Nusselt number $(\overline{\alpha}/\lambda)/(\nu^2/g)^{1/3}$; Pr - Prandtl number ν/a; q - heat flux density, W/m²; $\overline{q}$ - mean heat flux density, W/m²; r - latent heat of vaporization, J/kg; Re - Reynolds number of liquid film $4\Gamma/(\rho\nu)$; Re_m - Reynolds number maximum for condensate film at lower part of condensation surface $4\Gamma_m/(\rho\nu)$; T - temperature, K; $\overline{T}$ - mean temperature, K; v^* - dynamic velocity $(\tau_w/g)^{1/2}$; w - film velocity, m/s; x - longitudinal coordinate, m; y - transversal coordinate, m; Z - temperature gradient evaluating parameter; α - local heat transfer coefficient, W/(m²·K); $\overline{\alpha}$ - mean heat transfer coefficient, W/(m²·K); Γ - wetting density, $kg/(m\times s)$; Γ_m - wetting density maximum, $kg/(m\times s)$; δ - condensate film thickness, m; ε_q - ratio of heat flux densities, $q_w/\overline{q}_w$; ε_α - ratio of heat transfer coefficients $\alpha/\overline{\alpha}$; φ - dimensionless film velocity w/v^*; λ - thermal conductivity, W/(m·K); ν - kinematic viscosity, m²/s; ϑ - temperature field; $\overline{\vartheta}$ - mean temperature field; ρ - liquid density, kg/m³; τ - shear stress, Pa.
Subscripts: f - film flow; h - height; s - film surface; sat - saturated condition; w - wetted surface.

1. Introduction

The performance of thermal equipment frequently is affected by condensation, which may form on surfaces the condensate film. The variation of such film parameters has substantial influence on equipment performance being a function of the additional heat associated with latent heat when condensation occurs. Condensing heat transfer takes place in various industrial applications. It is an important part of refrigeration and cooling systems [1-3].

Film condensation of vapour flowing inside a vertical tube and between parallel plates was studied in [4]. A methodology was presented to determine numerically the heat transfer coefficients, the film thickness and the pressure drop. The analysis was based on the resolution of the full coupled boundary layer equations of the liquid and vapour phases and does not neglect inertia and convection terms in the governing equations. Turbulence in the vapour and condensate film was taken into account using mixing length turbulence models. The calculated results for the condensation of steam in a 24 mm diameter tube were compared with experimental values. The mean heat transfer coefficients for the condensation of vapour are also presented.

Study [5] develops a numerical model of laminar film condensation from a downward-flowing steam-air mixture onto a horizontal circular tube. The significant nonsimilarity of the coupled two-phase flow laminar film condensation is such that the boundary layer governing conservations of momentum, species and energy in the mixture and liquid phases are solved by finite-volume methods. Numerical analysis of both the local condensate film thickness and heat transfer characteristics elucidated the simultaneous effects of inlet-to-wall temperature difference and inlet air concentration, the Reynolds number of the mixture. It was shown that the local Nusselt number and liquid film thickness increase as both the noncondensable air mass fraction and the tube temperature decreases.

The purpose of study [6] was to find a convenient and practical procedure for calculating heat transfer of laminar film condensation on a vertical fluted tube. The condensate film on the tube surface along the axial direction was divided into two portions: the initial portion and the developing portion. The developing portion was analyzed in details. The film thickness equation of condensate film over the crest and the momentum equation of condensate film in the trough were established respectively after some simplifications and coupled with two-dimensional thermal conduction equation. The relationship between the heat transfer rate and the length of the flute was obtained through solving the equations numerically.

A two-dimensional, steady state model of convective film-wise condensation of a vapour and noncondensable gas mixture flowing downward inside a vertical tube was developed in paper [7]. The noncondensable effect on the condensation has taken into account through boundary layer analysis of species concentration and energy balance. Numerical predictions were obtained for the condensation heat transfer coefficient of turbulent vapor flow associated with laminar condensate. The predictions were compared with the experimental data in the literature to assess the model.

Paper [8] presents an investigation into turbulent film condensation on a sphere with variable wall temperature. Under the wide range of vapour velocity, the wall temperature and the local film shear stress were considered. The result shows that under the high velocity vapour, the increase of the temperature amplitude will bring out a larger Nusselt number, and the increase is about 2.7-5.6%.

Besides, under the effect of the local film shear stress, the mean Nusselt number will decrease about 0.65-0.8%. Furthermore, the paper then discusses the influence of shears and temperature amplitudes on the local dimensionless film thickness and heat transfer characteristics.

The laminar film-wise condensation heat transfer coefficient on the horizontal tubes of copper and stainless steel was researched in [9]. The tests were conducted at saturation temperatures of 20 and 30°C, and liquid wall subcoolings from 0.4 to 2.1°C. The measured condensation heat transfer coefficients were significantly lower than the predicted data by the Nusselt analysis when the ratio of the condensate liquid film thickness to the surface roughness, δ / R_{p-v}, was relatively low. When the condensate liquid film was very thin, tube material affected the condensation heat transfer coefficient in the film-wise condensation.

2. Film condensation of vapour analysis

We consider laminar film condensation process. As shown in Fig. 1, there may be several complicating features associated with film condensation. The film originates at the top of surface and flows downward under the influence of gravity. The thickness δ increases with increasing h because of continuous condensation at the liquid-vapour interface, which is at T_{sat}. There is then heat transfer from this interface through the film to the surface, which is maintained at $T_w < T_{sat}$.

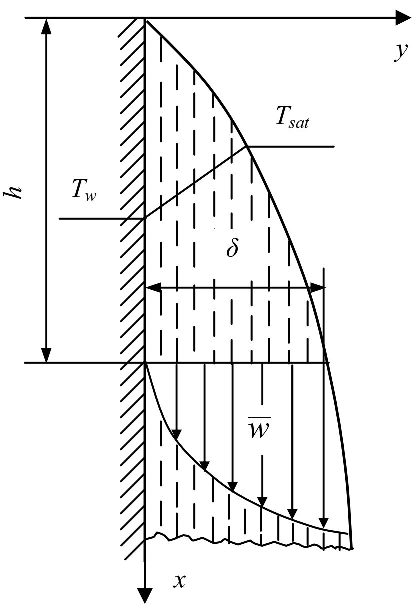

Fig. 1 Film condensation on a vertical surface

Heat transfer with film condensation resembles the heat transfer for film-surface evaporation at saturation temperature T_{sat}. This similarity therein lies that in both cases the heat flux density does not vary along the wetted surface. It must be said that heat flux densities on the condensate film surface and on the condensate surface (i.e. wall) are different. The heat flux density on the film surface is

$$q_s = r\frac{d\Gamma}{dx} \tag{1}$$

and on the condensate surface (wall), respectively

$$q_w = r\left(1+\frac{\overline{\vartheta}}{k}\right)\frac{d\Gamma}{dx} \tag{2}$$

where $\overline{\vartheta} = \left(T_{sat} - T_f\right)/\left(T_{sat} - T_w\right)$ and $k = r/c_p\left(T_{sat} - T_w\right)$.

However, a variable $\overline{\vartheta}/k$ usually is less than 1 and can be neglected. Then, as it appears from Eqs. (1) and (2), one can accept that the heat flux density does not vary across the condensate film. Usually, it is considered that at $Pr > 1$ and $k > 5$ variation of the heat flux density can be neglected.

If limit oneself within the interval of $1 \le Pr \le 10$, what often takes place in practice, then at $k = 1$ presumption that the heat flux density is constant leads to the error of heat transfer calculation less than 10% and at $k = 2$ less than 5%, respectively. These data attributed to the laminar flow of film condensate, however one can suppose that such conditions could also be take place at other film flow regimes. In practice, very rare occurrences when the value $k < 2$. Therefore, for the analysis of heat transfer in condensate film is fully acceptable the presumption of constant heat flux density.

For the calculation of vapour condensation, usually are used the mean heat transfer coefficients along the entire surface of condensation. Consequently, it is necessary to establish the relation between the local and the mean heat transfer.

According to [10], the heat transfer coefficient for nonisothermal surface can be averaged by the following regularity

$$\overline{\alpha} = \frac{\int_0^L \alpha \Delta T dx}{\int_0^L \Delta T dx} = \frac{\overline{q}_w}{\Delta T} \tag{3}$$

Therefore, in case of vapour condensation this regularity can be written as follows

$$\overline{\alpha} = \frac{\int_0^1 q_w dx_h}{\int_0^1 \left(T_{sat} - T_w\right) dx_h} = \frac{\overline{q}_w}{T_{sat} - T_w} \tag{4}$$

where $x_h = x/h$.

Then, taking into account that $q_w = \alpha\left(T_{sat} - T_w\right)$, we obtain that

$$\varepsilon_q = \varepsilon_\alpha \vartheta_w \tag{5}$$

where $\vartheta_w = \left(T_{sat} - T_w\right)/\left(T_{sat} - \overline{T}_w\right)$; $\varepsilon_\alpha = \alpha/\overline{\alpha}$ and $\varepsilon_q = q_w/\overline{q}_w$, respectively.

Let us assume that for the first approximation

$\left(1+\overline{\vartheta}/k\right)=const$. Also, we take into account that for $x=0$, $\Gamma=0$ and for $x=h$, $\Gamma=\Gamma_m$. Then, from Eq. (2) we can obtain the following expression

$$\int_0^{\Gamma_m} d\Gamma=\frac{1}{r\left(1+\overline{\vartheta}/k\right)}\int_0^h q_w dx \tag{6}$$

or

$$\Gamma_m=\frac{\overline{q}_w h}{r\left(1+\overline{\vartheta}/k\right)} \tag{7}$$

Taking into account the above-mentioned Eq. (2) can be written as follows

$$\frac{dRe}{dx_h}=Re_m\varepsilon_q \tag{8}$$

By neglecting a variation of physical parameters for film on the condensation surface, one can write as follows

$$Nu_M/\overline{Nu}_M=\alpha/\overline{\alpha} \tag{9}$$

Then, taking into account Eqs. (5) and (9), we can integrate Eq. (8) within the limits of $0\le x_h\le 1$ and $0\le Re\le Re_m$, and obtain the following expression

$$\overline{Nu}_M=\int_0^{Re_m}\frac{dRe}{Nu_M}=Re_m\int_0^1\vartheta_w dx_h \tag{10}$$

Considering that right integral is equal to 1, we can obtain the following general relation between the mean and the local heat transfer for film condensation of vapour on vertical surface

$$\overline{Nu}_M=\frac{Re_m}{\int_0^{Re_m}\frac{dRe}{Nu_M}} \tag{11}$$

It should be noted that this relation is also valid for inclined rectilinear surface of condensation.

If surface is curvilinear, then number Nu_M depends not only on the Re number (at given Pr number), but also on x_h. Usually this dependence can be expressed as the result of two functions, i.e.

$$Nu_M=f(Re)\varphi(x_h) \tag{12}$$

It follows from this that

$$\overline{Nu}_M=\frac{Re_m\int_0^1\varphi(x_h)\vartheta_w dx_h}{\int_0^{Re_m}\frac{dRe}{f(Re)}} \tag{13}$$

For example, at condensation of vapour on the surface of horizontal tube in laminar film flow can be used the following equation

$$Nu_M=1.1Re^{-1/3}\left(\sin\pi x_h\right)^{1/3} \tag{14}$$

where $x_h=x/0.5\pi D$.

Then, it follows that

$$\overline{Nu}_M=1.47Re^{-1/3}\int_0^1\vartheta_w\left(\sin\pi x_h\right)^{1/3}dx_h \tag{15}$$

Numerical integration of Eq. (11) leads to the results that are presented in Fig. 2. These theoretical data with sufficient accuracy can be approximated by the following semiempirical equation

$$\overline{Nu}_M=1.47Re_m^{-1/3}\begin{pmatrix}1+0.03Re_m^{0.2}+\\+0.00075Re_m^{0.8}Pr^{0.6}\end{pmatrix} \tag{16}$$

The comparison of this equation with theoretical and experimental data is presented in Fig. 3. As is well known, the expression $\overline{Nu}_M=f(Re_m,Pr)$ it is convenient

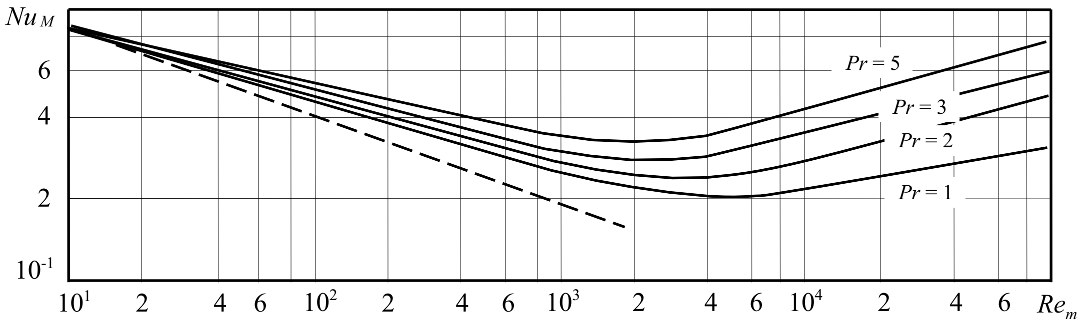

Fig. 2 Theoretical data of mean heat transfer for film condensation of vapour on vertical surface at various values Pr of the film: curves - theoretical calculation according to Eq. (11); dash line – Nusselt theory for laminar flow of condensate film; $\overline{Nu}_M=1.47Re_m^{-1/3}$

to use for practical calculations at given heat flux on the wall. However, frequently for calculation of condensation processes the temperature gradient proves to be a given variable. Then, it is convenient to apply the expression $\overline{Nu}_M = f(Z, Pr)$ or $Re_m = f(Z, Pr)$ [11], where

$$Z = \frac{\lambda h\left(T_{sat} - \overline{T}_w\right)g^{1/3}}{r\rho\nu^{5/3}} = \frac{Re_m}{4\overline{Nu}_M} \tag{17}$$

In this case theoretical data can be described by the following semiempirical equation

$$\overline{Nu}_M = 0.94Z^{-0.25}\left(1 + 0.04Z^{0.2} + 0.000045ZPr\right) \tag{18}$$

or

$$Re_m = 3.77Z^{0.75}\left(1 + 0.04Z^{0.2} + 0.000045ZPr\right) \tag{19}$$

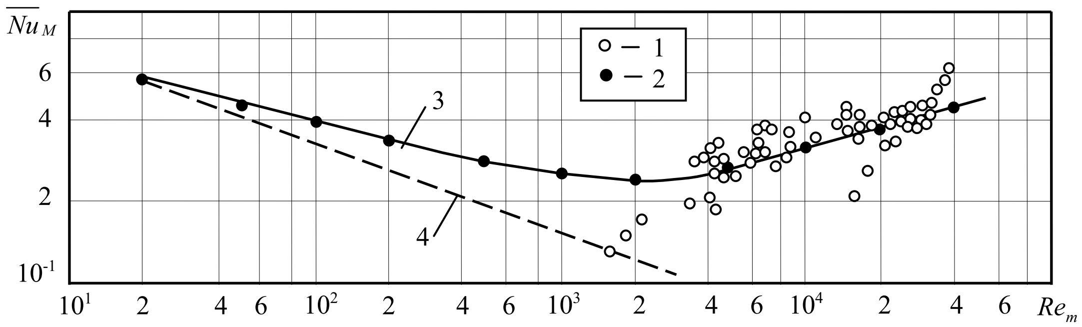

Fig. 3 Mean heat transfer for film condensation of vapour on vertical surface: *1* - experimental data of [12], $Pr \cong 5$; *2* - theory according to Fig. 2, $Pr = 5$; *3* - calculation according to Eq. (18), $Pr = 5$; *4* - Nusselt theory

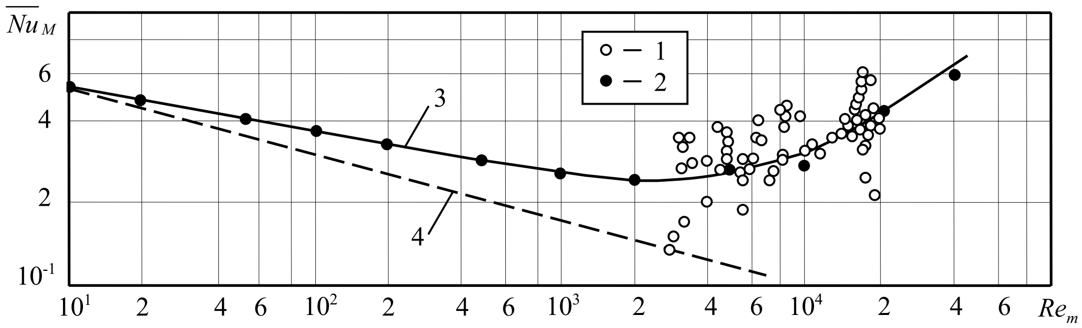

Fig. 4 Mean heat transfer for film condensation of vapour on vertical surface: *1* - experimental data of [12], $Pr \cong 5$; *2* - theory according to Fig. 2, $Pr = 5$; *3* - calculation according to Eq. (19), $Pr = 5$; *4* - Nusselt theory

Comparison of these equations with theoretical and experimental data is presented in Fig. 4.

3. Conclusions

Analysis of Eq. (11) showed that for rectilinear surface condensation the mean heat transfer coefficient does not depend on the nature of condensation temperature variation, if α it is averaged according to Eq. (4) with disregard of physical parameters variation for condensate. For the curvilinear surface (horizontal tube) this independence of α is absent and it is confirmed by Eq. (15).

It was observed that if to deny expressions in brackets of Eqs. (16) - (19), they become the theoretical equations resulted from Nusselt's theory for laminar flow of condensate film. Therefore, expressions in brackets can be considered as a multiplier, evaluating the increase of heat transfer for wavy and turbulent flow of condensate film comparing with laminar flow.

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ŠILUMOS ATIDAVIMAS, ESANT PLĖVELINEI GARŲ KONDENSACIJAI

R e z i u m ė

Straipsnyje analizuojamas šilumos atidavimas, kai šilumos srauto tankis plėvelės storyje nekinta. Praktiškai taip būna esant plėvelinei garų kondensacijai. Analizuojant plėvelinės garų kondensacijos procesą, įvertintas vidutinio šilumos atidavimo koeficiento ir šilumos srauto tankio pasiskirstymas kondensacijos paviršiaus aukštyje. Nustatyta, kad vidutinis šilumos atidavimo koeficientas nepriklauso nuo kondensacijos temperatūros kitimo pobūdžio. Pasiūlytos pusiau empirinės priklausomybės šilumos atidavimui apskaičiuoti, įvertinant temperatūros gradientą.

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HEAT TRANSFER FOR FILM CONDENSATION OF VAPOUR

S u m m a r y

Study provides an analysis of the heat transfer for constant heat flux densities across the film. A situation of this type is observed with film condensation. The analysis of film condensation includes the distribution of mean heat transfer coefficient and of heat flux density along the height of condensation surface. It has been determined that the mean heat transfer coefficient does not depend on the nature of condensation temperature variation. Semi-empirical relationships evaluating the temperature gradient are suggested.

С. Шинкунас, А. Кела

ТЕПЛООТДАЧА ПРИ ПЛЕНОЧНОЙ КОНДЕНСАЦИИ ПАРА

Р е з ю м е

В данной статье представлен теоретический анализ теплообмена при случае, когда плотность теплового потока одинакова по всей толщине пленки. Такой случай встречается на практике при пленочной конденсации пара. При анализе процесса пленочной конденсации пара теоретически рассматриваются такие его характеристики, как распределение среднего коэффициента теплоотдачи и плотности теплового потока по высоте поверхности конденсации. Установлено, что при конденсации средний коэффициент теплоотдачи не зависит от характера изменения её температуры. Предложены полуэмпирические уравнения для определения теплоотдачи учитывая температурный напор.

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