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CONDENSATION INSIDE NEAR HORIZONTAL TUBES IN CO-CURRENT AND COUNTER-CURRENT FLOW

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Abstract—Vapor condensation rate inside horizontal conduits was studied for co-current and counter-current flow of steam and the accumulated condensate at the bottom of the tube. When compared with the co-current flow case, the interfacial shear increases the axial pressure drop in the counter-current case and decreases the effective transfer surface. The effect of changing the inclination angle of the tube from 1° to 2° from the horizontal on the heat transfer rate is relatively unimportant. For a given tube length, co-current flow is the recommended mode of operation when a higher condensate production rate is desired at a given temperature driving force.

NOMENCLATURE

- a, geometrical parameter of accumulated condensate layer;
- A_v, vapor flow cross-section;
- b, distance from center $(R - H_l)$;
- C, constant, equation (3);
- C_1, constant, equation (13);
- D_H, hydraulic diameter;
- f, friction factor, equation (9);
- g, gravitational acceleration;
- H_l, liquid height of condensate layer;
- H_T, heat transfer coefficient;
- k_l, liquid conductivity;
- L, tube length;
- P_l, pressure in liquid layer;
- P_v, perimeter of vapor flow area;
- $\mathbf{q}_\phi$, local heat flux;
- Q_c, integrated peripheral heat flux;
- Q_l, liquid flow rate of condensate;
- Q_v, vapor flow rate;
- Q_{v0}, initial vapor flow rate;
- r, ratio of second derivatives, equation (2);
- R, tube radius;
- Re_v, Reynolds number, vapor;
- T, vapor temperature in tube;
- ΔT, temperature driving force;
- u, liquid velocity;
- u_v, vapor velocity;
- y_0, coordinate on solid–liquid boundary;
- x, horizontal coordinate;
- x_0, coordinate on solid–liquid boundary;
- y, vertical coordinate;
- z, axial coordinate.

Greek symbols

- α, a/R;
- β, b/R;
- λ, latent heat of condensation;
- θ, tube inclination, from horizontal;
- μ_l, viscosity, liquid;
- ρ_l, density, liquid;
- ρ_v, density, vapor;
- τ_i, interfacial shear stress;
- ϕ, peripheral angle;
- ϕ_m, peripheral angle of condensate layer.

1. INTRODUCTION

VAPOUR condensation inside horizontal conduits is commonly encountered in horizontal type refrigeration systems, thermal water desalination schemes and various types of air-cooled or radiation-cooled condensers. The high inlet velocity of the condensing vapor decreases as the condensation proceeds on the inside walls along the tube. The dynamics of the vapor–liquid interaction results in two-phase flows and different flow patterns are possible [1], ranging from stratified to annular flow regimes. The condensation mechanism in the stratified flow regime is predominantly affected by the viscous inertial body forces [2, 3] while in the annular flow regime the condensation process is controlled by the dynamics of the vapor and the adjacent condensate layers [4–6].

Previous studies of the various characteristic parameters of horizontal condensers have shown experimentally that the condensation heat transfer coefficient increases with steam flow rates and the tube's length to diameter ratio [7–10]. A presentation of the condensation rates at various steam flow rates is given by Rohsenow [11]. The effects of the accumulated condensate, at the bottom of the slightly inclined tube, on the heat transfer rate were presented by Chato [12] and Chaddock [13].

Condensation inside a horizontal evaporator–condenser has been studied by the authors [14, 15], mainly with relation to water desalination schemes whereby the evaporation of thin brine films, formed by the brine cascading on the outside of the horizontal tubes in the tube bundle is sustained by steam condensation inside the tube. The accumulated condensate can be drained either downstream or upstream of the vapor flow direction. This results in the tube being either positively (co-current) or negatively (counter-current) inclined (Fig. 1). In the range of flow rates considered in our previous studies ($Re_v < 35000$) the shear stress on the condensing film was assumed to be negligible. However, at higher steam flow rates this assumption of zero interfacial shear is not valid, although a stratified vapor–liquid flow still prevails. Whereas the effect of the interfacial shear stress on the stratified flow regime has been studied with respect to pressure drop and gas hold-up [16–20], the heat transfer and phase change aspects associated with the shear stress in stratified vapor–condensate flow have not been fully reported.

It is the purpose of the present work to evaluate the transport characteristics along a typical tube in a horizontal condenser. The interaction between the vapor and the condensate layers flowing along the tube is studied for co-current and counter-current flows.

2. THE PHYSICAL MODEL

Consider vapor flowing in a nearly horizontal tube, which is placed in a positive or a negative inclination, θ, from the horizontal plane. The vapor enters the tube at the $z = 0$, and flows along the tube in the z-direction. The initial high velocity decreases as the vapor moves and condenses along the tube. At a given set of operating conditions, an initial vapor feed Q_{v0} yields a practically zero velocity, corresponding to a practically complete condensation at the outlet, at $z = L$. The condensate drains down on the walls to the bottom of the tube and accumulates downstream. Whereas a high vapor flow rate interacts with an extremely thin condensate layer in the entry region of the tube, a relatively low velocity steam interacts at the outlet end with a thick condensate layer. Excluding extremely high vapor flow rates, a stratified vapor–liquid flow is assumed to exist along the whole tube. Moreover, the free liquid interface at any cross-section along the tube is assumed to be smooth and horizontal. The validity of the assumed horizontal interface is reasonable for the practical tube (38 mm I.D.) dealt with here since the effects of the preferential wetting and the liquid meniscus at the tube wall are negligible.

Assuming the condensate layer to be in laminar flow, the equation of motion is given by

$$\mu_l\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) = \frac{\partial P_l}{\partial z} - \rho_l g \sin\theta \tag{1}$$

where P_l is the local pressure in the liquid layer. The velocity distribution is obtained by integrating equation (1). Inasmuch as the integration is carried out in a stepwise fashion in the z-direction, the pressure drop term is introduced as a constant which depends on the local steam velocity. However, due to the presence of the second derivative with respect to the lateral coordinate x, a numerical integration method is required. A further simplification of equation (1) is achieved by introducing the ratio of the two second derivatives

$$r = \frac{\partial^2 u/\partial x^2}{\partial^2 u/\partial y^2}. \tag{2}$$

Introducing equation (2) into equation (1) yields

$$\frac{d^2 u}{dy^2} = \frac{1}{\mu_l(1+r)}\left(\frac{dP_l}{dz} - \rho_l g \sin\theta\right) \equiv C. \tag{3}$$

The ratio r in equation (3) can be taken either as a function of the interfacial position or, as in Buffham's analysis [17], assumed to be a constant. For relatively thin condensate films dealt with here, where $\partial^2 u/\partial y^2 \gg \partial^2 u/\partial x^2$ and $r \ll 1$ the solution for the velocity profile is insensitive to the value of r. Consequently, the ratio r may be ignored. Assuming no slip condition at the tube wall and a velocity gradient of τ_i/μ_l at the gas–liquid interface, equation (3) yields

$$U = \frac{C}{2}(y^2 - y_0^2) - \frac{\tau_i}{\mu_l}(y - y_0) \tag{4}$$

where τ_i is the interfacial shear stress in the flow direction at $y = 0$. τ_i is related to the axial pressure gradient by

$$\tau_i = \pm\frac{D_H}{4}\frac{dP_l}{dz} \qquad \begin{matrix} + \text{ co-current} \\ - \text{ counter-current} \end{matrix} \tag{5}$$

where D_H is the hydraulic diameter for the vapor flow and is given by

$$D_H = 4\frac{A_v}{P_v}. \tag{6}$$

The vapor cross-section A_v and the corresponding vapor-phase perimeter P_v are related to the peripheral angle ϕ_m of the condensate layer by

$$A_v = \frac{R^2}{2}\left\{\pi - \left[\frac{\pi}{180}2\phi_m - \sin(2\phi_m)\right]\right\}, \tag{7a}$$

$$P_v = \frac{R}{2}(\pi - \phi_m) + 2R\sin(\phi_m). \tag{7b}$$

The pressure-gradient is calculated locally based on the local vapor velocity

$$\frac{dP_l}{dz} = \frac{2 f u_v^2 \rho_v}{D_H} \tag{8}$$

where the vapor phase friction factor is given by the Blasius equation

$$f = 0.079\, Re_v^{-0.25}. \tag{9}$$

The liquid flow rate is obtained by integrating the liquid velocity field over the liquid cross-sectional area. Thus

$$Q_l = 2\int_0^a \int_0^{y_0} u \, \mathrm{d}y \, \mathrm{d}x \tag{10}$$

where a, shown in Fig. 1, is half the liquid chord length subtended by the angle ϕ_m.

Inserting equation (4) into equation (10) yields

$$Q_l = \int_0^a \left(-\frac{2}{3} C y_0^3 + \frac{\tau_i}{\mu_l} y_0^2 \right) \mathrm{d}x. \tag{11}$$

Note that $y_0 = (R^2 - x_0^2)^{1/2} - b$. Introducing $X = x/R \equiv \sin\phi$, equation (11) can be evaluated in terms of $\alpha(\equiv \sin\phi_m)$ and $\beta(\equiv \cos\phi_m)$. Stated differently, the liquid flow rate is related to ϕ_m, the angular parameter of the accumulated condensate. Integration of equation (11) thus yields

$$\frac{Q_l}{CR^4} = -\frac{1}{12}\left[2\alpha^3\beta - 15\alpha\beta + (3 + 12\beta^2)\arcsin\alpha + \frac{\tau_i}{\mu_l CR}(12\alpha - 4\alpha^3 - 12\beta\arcsin\alpha) \right] \tag{12}$$

where

$$\beta^2 = 1 - \alpha^2.$$

Equation (12) can be solved for α for any given value of the liquid flow rate. The latter is determined by the accumulated condensate layer. The accumulated condensate flow rate is evaluated from the condensation rate along the tube.

Previous studies by the authors of a typical evaporator–condenser tube [15, 16] showed that the thickness of the condensate film is of the order of 0.1 mm. Local velocity and temperature profiles have been derived [15, 16] for the evaporating and condensate films demonstrating that the condensate film can very reasonably be approximated by Nusselt's solution. Hence the local heat flux is given by

$$q_\phi = C_1 \left(\frac{\sin\phi}{\phi} \right)^{1/4}; \quad C_1 = \left(\frac{g\lambda\rho_l^2 k_l^3 \Delta T}{4\mu_l R} \right)^{1/4}. \tag{13}$$

Here ϕ is local peripheral angle (Fig. 1). The average film condensation rate over the arc 0–$(\pi - \phi_m)$ is obtained by integrating equation (13)

$$Q_c = \frac{2}{\rho_l\lambda(\pi - \phi_m)} \int_0^{\pi - \phi_m} \mathbf{q}_\phi \cdot \mathrm{d}\phi, \qquad = \frac{2C_1}{\rho_l\lambda(\pi - \phi_m)} \int_0^{\pi - \phi_m} \left(\frac{\sin\phi}{\phi} \right)^{1/4} \mathrm{d}\phi. \tag{14}$$

Equation (14), which gives the condensing film mass flow rate on the walls per unit length of tube, is now combined with equation (12), which gives the accumulated condensate flow rate. Thus, a mass balance in the axial direction reads

$$\pm \frac{Q_l|_{z+\Delta z} - Q_l|_z}{\Delta z} = Q_c, \qquad \begin{matrix} +, \text{ co-current}, \\ -, \text{ counter-current}. \end{matrix} \tag{15}$$

$$\mp \frac{CR^4}{12}\left[2\alpha^3\beta - 15\alpha\beta + (3 + 12\beta^2)\arcsin\alpha + \frac{\tau}{\mu_l CR}(12\alpha - 4\alpha^3 - 12\beta\arcsin\alpha) \right] = \left[\pm Q_l|_z + \frac{2C_1}{\rho_l\lambda(\pi - \phi_m)} \int_0^{\pi - \arcsin\alpha} \times \left(\frac{\sin\phi}{\phi} \right)^{1/4} \mathrm{d}\phi \right] \Delta z, \qquad \begin{matrix} +, \text{ co-current}, \\ -, \text{ counter-current}. \end{matrix} \tag{16}$$

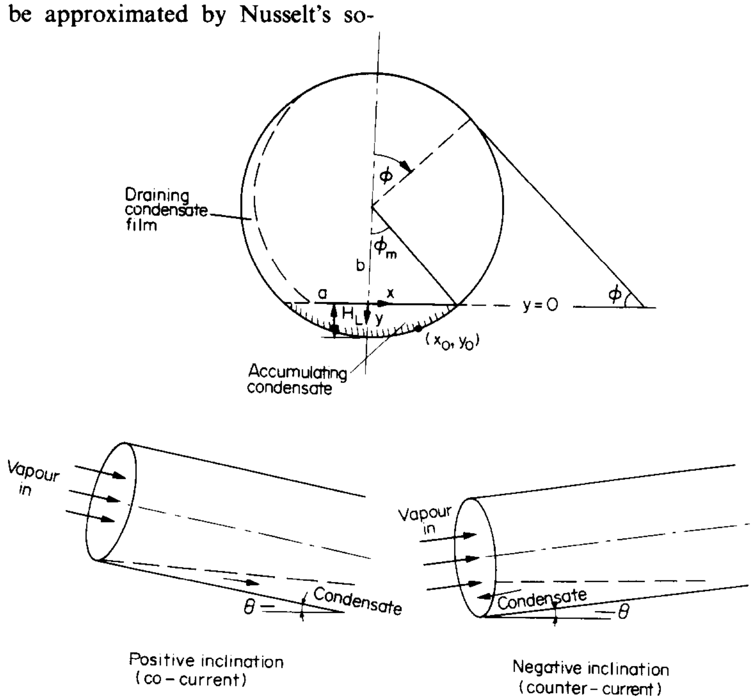

FIG. 1. The physical model and coordinates.

In the co-current flow case $Q_l = 0$ and the vapor flow rate is $Q_{v,0}$ at $z = 0$ (steam inlet). The calculation proceeds in a stepwise manner by first evaluating the parameter α for $z = \Delta z$ which, in turn is used to calculate the local liquid rate by equation (12). The local vapor flow rate is then corrected by accounting for the condensation rate. The new local values of Q_v, D_H, dP_l/dz and τ_i are then evaluated and the constant C [equation (3)] is also corrected to correspond to the local pressure gradient.

When taken with the lower sign equation (16) is used to evaluate the counter-current case whereby the steam is introduced at the lower edge of the positively inclined tube so that the accumulated condensate drains against the steam flow direction.

3. RESULTS AND DISCUSSION

The theoretical analysis presented here was applied to a typical horizontal evaporator–condenser tube in the horizontal tube evaporator–condenser desalination scheme. A tube length of 5 m, an inlet vapor flow rate of 7 kg m^{-1} and temperature range between 50 and 70°C were selected. These values are used as independent input data whereas the temperature gradient which yields a complete vapor condensation within the given tube length at the given temperature remains as a dependent parameter.

The effect of the inclination of the tube from the horizontal plane was studied in detail, and inclinations of 1° and 2° were compared for given operating conditions, in co-current and counter-current gas–liquid flow modes. Pressure drops, liquid height and heat transfer characteristics were evaluated for the various inclinations and modes of flow.

The axial pressure gradient is calculated from the instantaneous vapor flow rate and the available cross-section for vapor flow, both being, in turn, determined by the rate of condensation along the tube. Figure 2 presents the local pressure gradient at three temperature levels in co-current and counter-current flows. Note that for a given vapor mass flux the pressure drop

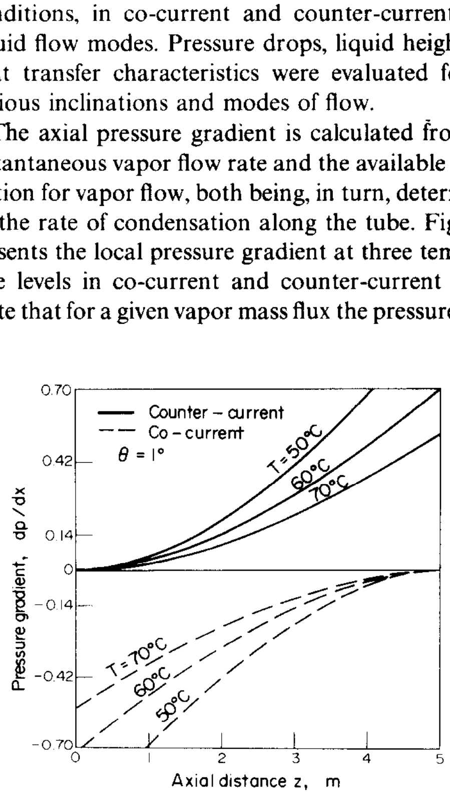

FIG. 2. Pressure gradient along the tube at various condensation temperatures in co-current and counter-current flows.

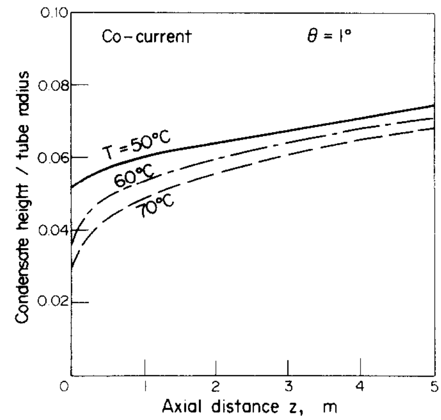

FIG. 3. Accumulated condensate height at the bottom of the tube at various condensation temperatures for co-current flow.

is proportional to f/ρ_v. Thus, increasing the temperature level in co-current flow yields an increase in the pressure gradient due to the decrease in the density of the vapor. Also, the effect of the gas shear on the accumulated condensate layer is to decrease the condensate height H_l which results in a smaller friction factor. On the other hand, the effect of the interfacial shear in the counter-current flow case leads to a thicker accumulated condensate layer, resulting with a higher friction factor and a larger pressure drop. Obviously,

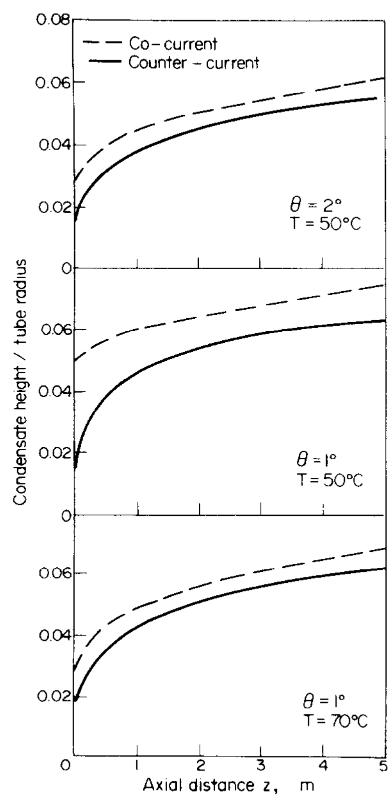

FIG. 4. Comparison of the condensate height in co-current and counter-current flows at various condensation temperatures and different tube inclinations.

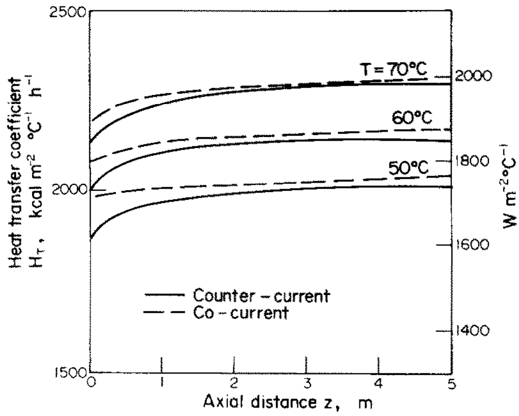

FIG. 5. Heat transfer coefficient at various condensation temperatures in co-current and counter-current flows.

the effects in both cases are more pronounced at the entry region of the vapor where the vapor flow rate is the highest. Based on the same reasoning, the absolute value of the pressure at a certain temperature level is higher for the counter-current flow. It is to be noted that only a very slight effect on the pressure drop has been realized when the tube inclination angle was increased to 2°.

Figure 3 represents the effects of the temperature level on the height of the accumulated condensate layer at the bottom of the tube. As is indicated by the curves in this figure, an increase in the temperature level yields, due to higher interfacial shear (see Fig. 2a), a thinner condensate layer. The effect of temperature level is reduced, as the inclination of the tube is increased. Again, this is due to the milder effect of the interfacial shear on the thinner film associated with the higher inclination angle of the tube. Similar trends are obtained for counter-current flow.

It is interesting to compare the height of the condensate layer for the two modes of flows (Fig. 4). Two interesting points are demonstrated. The first is that the difference between the two modes of flow increase as the temperature levels decrease. The second is that as the tube inclination increases these differences become more moderate. The two phenomena are understandable in view of the associated effects on the pressure gradient, already discussed with reference to Fig. 2.

Figure 5 represents the peripherally averaged heat transfer coefficient along the tube for various temperature levels, for co- and counter-current flows. As is expected, higher temperature levels yield higher transfer coefficients. Also, in view of the higher level of the accumulated condensate associated with co-current flows, the corresponding heat transfer coefficient based on vapour phase flow area are higher than those realised in counter-current flow.

4. CONCLUSIONS

The interactions between the condensing vapor and the accumulated condensate layer flowing along a nearly horizontal tube were compared for co-current and counter-current vapor–liquid flows. The effects of tube inclination and temperature level of condensation were evaluated.

In general, the axial pressure drop increases as the condensation temperature level increases in the co-current flow case and decreases as the condensation temperature increases in the counter-current flow case. However, in both flow cases, the height of the condensate layer increases at the higher temperature levels and is consistently higher for co-current flows. This leads to apparently higher heat transfer coefficients for the co-current film (since the transfer area for film condensation is smaller). Actually the nominal heat transfer coefficients, based on tube diameter, are practically identical. However, for a given tube length and a desired condensate production, a 10–13% larger temperature driving force is required for the counter-current case as compared with the co-current flow case.

It is also to be noted that the differences between co-current and counter-current flows become less significant as the inclination angle of the tube increases and the effect of gravity forces predominate.

Thus, the co-current vapor and condensate flow in the near horizontal tube is the preferred mode of operating the evaporator–condenser unit.

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CONDENSATION A L'INTERIEUR D'UN TUBE PRESQUE HORIZONTAL AVEC ECOULEMENT A CO-OU CONTRE-COURANT

Résumé—On étudie le débit de condensation de vapeur à l'intérieur de conduits horizontaux pour un écoulement à co-ou contre-courant de vapeur d'eau et de condensat accumulé au fond du tube. Comparé au cas du co-courant, le cisaillement interfacial accroît la chute de pression axiale dans le cas du contre-courant et diminue le transfert effectif à la surface. Le fait de changer l'inclinaison du tube de 1° à 2° à partir de l'horizontale est relativement sans importance sur le transfert. Pour une longueur de tube donnée, l'écoulement à co-courant est le mode recommandé quand on désire un débit plus élevé de condensat pour un écart de température donné.

KONDENSATION IN NAHEZU HORIZONTALEN ROHREN BEI GLEICH- UND GEGENSINNIGER STRÖMUNG

Zusammenfassung—Die Kondensation in horizontalen Rohren wurde für gleich- und gegensinnige Strömung des Dampfes und des unten im Rohr angesammelten Kondensats untersucht. Im Vergleich zur gleichsinnigen Strömung erhöht im Fall der Gegenströmung die Schubspannung an der Grenzfläche den Druckabfall in axialer Richtung und vermindert die wirksame Übertragungsfläche. Eine Änderung des Rohr-Neigungswinkels von 1° auf 2° gegenüber der Waagerechten beeinflußt den Wärmeübergang nur unwesentlich. Bei gegebener Rohrlänge und treibender Temperatur-Differenz wird die Betriebsweise mit gleichsinniger Strömung empfohlen, um eine größere Kondensationsrate zu erreichen.

КОНДЕНСАЦИЯ ВНУТРИ ПОЧТИ ГОРИЗОНТАЛЬНЫХ ТРУБ ПРИ СПУТНЫХ И ПРОТИВОТОЧНЫХ ТЕЧЕНИЯХ

Аннотация — Проведено исследование скорости конденсации пара внутри горизонтальных труб при спутном и противоточном течении пара и конденсата, образующегося на дне трубы. По сравнению со спутным течением сдвиг на поверхности раздела фаз при противотоке увеличивает аксиальный перепад давления и уменьшает эффективную поверхность переноса. Изменение угла наклона трубы от 1 до 2° от горизонтального положения почти не оказывает влияния на интенсивность теплопереноса. Для труб исследуемой длины рекомендуется спутный режим течения, если желательно получить более высокую скорость конденсации при заданном температурном градиенте.