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Condensation inside tubes: condensate film coefficient for vertical downflow

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CONDENSATION INSIDE TUBES: condensate film coefficient for vertical downflow

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CONDENSATION INSIDE TUBES: condensate film coefficient for vertical downflow

1. NOTATION AND UNITS

		Units	
		SI	British
A	flow cross-sectional area	m^2	ft^2
c_p	specific heat capacity	J/kg K	$\text{Btu/lb } ^\circ\text{F}$
D	internal diameter of tube	m	ft
f_w	tube-wall friction factor in Equation (A1.1)	—	—
g	acceleration due to gravity	m/s^2 (9.81)	ft/h^2 (4.17×10^8)
g_c	constant of proportionality in Newton's 2nd law of motion	— (1.0)	lb ft/lbf h^2 (4.17×10^8)
j_g^*	gas phase dimensionless superficial velocity defined in Equation (5.1)	—	—
L	length of condensing zone	m	ft
$\dot{M}$	total mass flow rate per tube = $(\dot{M}_g + \dot{M}_l)$	kg/s	lb/h
Nu	modified condensate film Nusselt number = $\frac{a_c}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l(\rho_l - \rho_g)g} \right]^{1/3}$	—	—
n_1, n_2	exponents in Equations (B1.5) and (B1.6)	—	—
Pr	Prandtl number = $\eta c_p / \lambda$	—	—
$\dot{Q}$	cumulative heat load	W	Btu/h
$\dot{q}$	heat flux	W/m^2	$\text{Btu/ft}^2 \text{ h}$
Re_f	condensate film Reynolds number = $4\Gamma/\eta_l$	—	—
T	temperature	$\text{K or } ^\circ\text{C}$	$^\circ\text{F}$
T'	temperature at dry wall/wet wall transition	$\text{K or } ^\circ\text{C}$	$^\circ\text{F}$
U	overall heat transfer coefficient referred to inside diameter of tube	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h } ^\circ\text{F}$
U'	combined heat transfer coefficient for coolant, tube wall and any fouling referred to inside diameter of tube	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h } ^\circ\text{F}$
x_g	vapour mass fraction = $\dot{M}_g/(\dot{M}_g + \dot{M}_l)$	—	—
α	heat transfer coefficient	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h } ^\circ\text{F}$
Γ	condensate flow rate per wetted perimeter = $\dot{M}_l/\pi D$	kg/s m	lb/h ft

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δ	condensate film thickness	m	ft
δ^*	dimensionless condensate film thickness defined by Equation (A1.3)	–	–
η	dynamic viscosity	kg/s m (Pa s)	lb/h ft
Λ	dimensionless group defined in Equation (5.8)	–	–
μ	error in prediction of Nusselt number	–	–
λ	thermal conductivity	W/m K	Btu/h ft °F
ρ	fluid density	kg/m ³	lb/ft ³
τ_i	interfacial shear stress	N/m ² (Pa)	lb/ft ²
τ_i^*	dimensionless interfacial shear stress defined in Equation (B1.4)	–	–

Extra Subscripts

<i>c</i>	condensate film
<i>cs</i>	cold stream
<i>dw</i>	dry wall
<i>expt</i>	experimentally measured value
<i>g</i>	gas or vapour phase
<i>grav</i>	gravity controlled annular flow
<i>in</i>	at stream inlet
<i>init</i>	initial value
<i>L</i>	at point along tube where prescribed amount of condensation is complete
<i>l</i>	liquid phase
<i>lam</i>	laminar condensate film
<i>m</i>	mean value
<i>o</i>	at stream outlet
<i>pred</i>	predicted value
<i>sat</i>	at saturation
<i>sc</i>	in subcooling zone
<i>T</i>	refers to a condensate film as if total condensation of all vapour takes place
<i>turb</i>	turbulent condensate film
<i>w</i>	at vapour side surface of the tube wall or any dirt film present, as appropriate
<i>ww</i>	wet wall
<i>x</i>	at distance <i>x</i> from top of tube

2. INTRODUCTION

Condensation of a vapour will occur at a suitably cooled surface. If the surface is readily wetted the condensate will form as a film and drain away under gravity forces or under the effect of vapour shear at the vapour-liquid interface or some combination of both mechanisms. On non-wetting surfaces droplets will be formed which, with continuing condensation, will coalesce and eventually roll off the surface under the effect of gravity.

The heat transfer coefficient with dropwise condensation can be an order of magnitude greater than with film-wise condensation but the mode requires specially prepared surfaces. No established means are available for maintaining the surface quality over the sustained periods of operation required of industrial equipment. Consequently, dropwise condensation is not considered in this Data Item.

In a condensation process, resistance to heat transfer from the bulk vapour to the surface^{*} may be concentrated entirely in the condensate film or divided between the vapour-gas phase and the condensate film. When a single component vapour or an azeotropic mixture condenses, the resistance is confined to the condensate film, the heat transfer mechanism being that of conduction or convection through the film. There is an additional resistance in the vapour phase when a mixture of vapours, or vapours in the presence of non-condensing gases, are to be condensed. This arises because the heavy components (or 'low volatiles') must diffuse through the lighter components ('high volatiles') or non-condensing gases that accumulate adjacent to the condensate film-vapour interface. Consideration in this Data Item is limited to the condensate film.

3. PURPOSE AND SCOPE

This Data Item is concerned with the co-current downflow of vapour and condensate inside plain vertical tubes of circular cross-section. It presents the available data and provides correlations for predicting the condensate film coefficient. Separate correlations are presented for the local condensate film coefficient at a prescribed vapour mass fraction and the mean coefficient for the case where the saturation temperature is approximately constant over the entire condensing length.

This Item is not applicable to reflux condensation whereby vapour enters the tube at the bottom and flows upward whilst the condensate flows downward under gravity. ESDU 89038¹⁴ should be consulted for information on this process.

* In the absence of any dirt film the surface is that of the tube wall material adjacent to the liquid. In the presence of a dirt film the surface is that of the dirt film.

4. FACTORS INFLUENCING THE CHOICE OF IN-TUBE CONDENSATION

In the case of shell-and-tube heat exchangers, in-tube condensation is normally to be preferred in the following circumstances.

- (i) *When operating pressures are high.* High pressure on the shell side implies heavy, and consequently expensive, shells and scantlings.
- (ii) *When the condensing stream is corrosive.* Special, but costly, corrosion resistant materials usually make in-tube condensation less expensive; the shell and its internals can then still be in carbon steel or other relatively cheap material.
- (iii) *When the condensing stream is the more heavily fouling stream.* Unless chemical cleaning is specified, in-tube cleaning is easier and hence less expensive than cleaning the outside of a tube bundle.

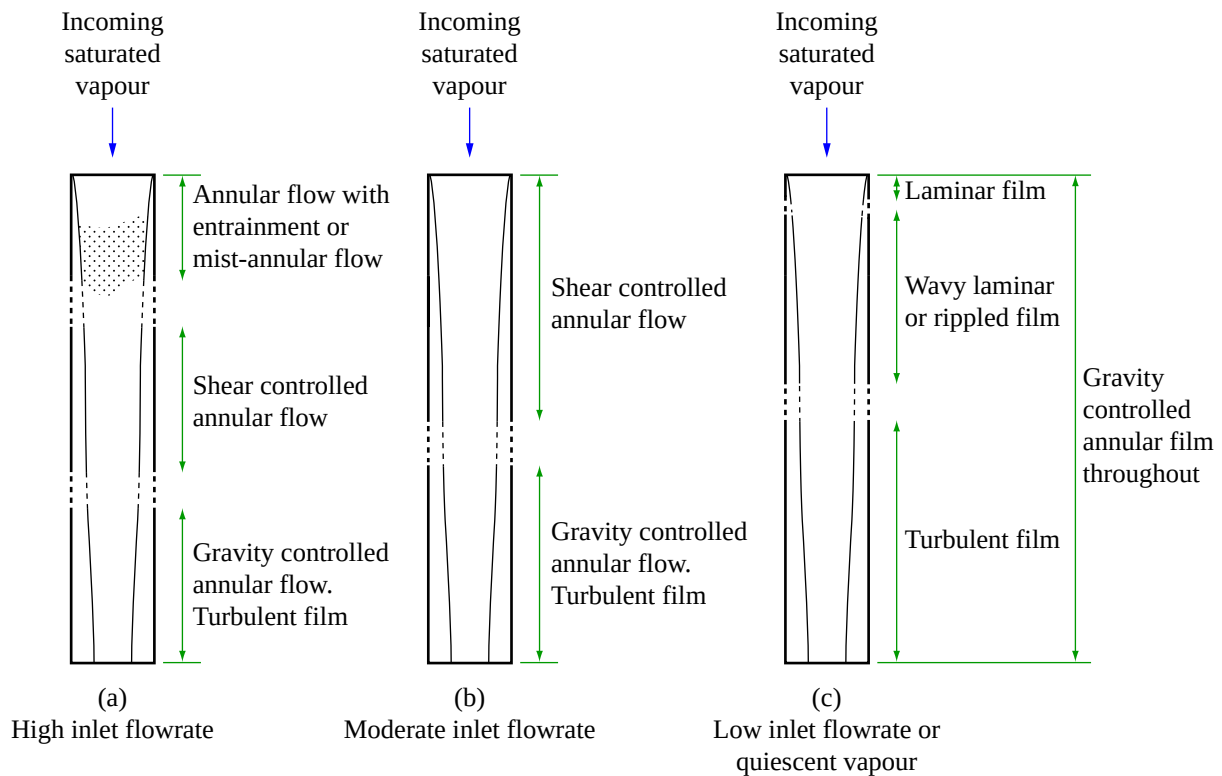
There are too many parameters in the thermal/hydraulic design of a condenser to give definitive guidelines as to whether vertical or horizontal orientation will result in a lower installed cost for a particular duty; however particular advantages of vertical tube orientation are:

- (i) saving in plot area compared with a horizontally mounted unit, always provided that a separate structure is not required,
- (ii) the high measure of condensate subcooling attainable, if required, since the condensate to be subcooled normally flows as a falling film with an inherently high heat transfer coefficient.

Against these advantages must be set the fact that vertical condensers may be more difficult to clean than horizontally mounted units.

5. CONDENSATE FILM COEFFICIENT

5.1 Flow Regimes in Vertical Tube-side Downflow Condensation



Sketch 5.1 Flow patterns in vertical downflow condensation inside tubes of circular cross-section

The hydrodynamics of the condensate film must be appreciated and the flow patterns known in order to select the appropriate correlation for calculating the condensate film coefficient. In the case of vertical tube orientation the flow will normally be annular throughout the length of the tube. As condensation proceeds and the vapour velocity decreases, the ratio of shear forces to gravity forces acting on the condensate film will decrease. One of three possible flow sequences, illustrated in Sketch 5.1, may occur.

(a) High inlet flowrate

Initially the flow pattern is annular but high vapour velocities can give rise to entrainment. The continuous vapour phase then carries a significant proportion of the liquid phase in the form of entrained droplets. The annular film is consequently extremely thin and may even be unstable. Film coefficients are correspondingly high. Entrainment becomes progressively less and the film stabilises as condensation proceeds. The flow, and consequently the thickness of the condensate film, is then controlled by vapour shear at the vapour-liquid interface.

With further condensation vapour shear becomes progressively less and gravity more important as the controlling mechanism for condensate drainage.

Throughout the length of the tube the condensate film is turbulent.

(b) *Moderate inlet flowrate*

Flow patterns are similar to the high flowrate case except that any entrainment is confined to entry effects at the top of the tube.

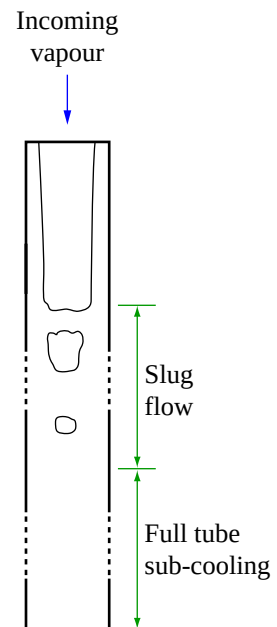
(c) *Low inlet flowrate or quiescent vapour*

Flow will be gravity controlled throughout in the absence of any significant vapour shear. Initially, over a short distance, the condensate film will be in laminar flow. As condensation proceeds the film, whilst remaining essentially laminar, becomes rippled. Eventually, and provided the tube is of sufficient length, the film becomes turbulent.

In laminar films the heat transfer mechanism is one of conduction through the film so, as the amount of condensate flowing down the tube increases and the film becomes thicker, the film coefficient decreases. In turbulent films the mechanism, outside the laminar sub-layer, is one of convection so the film coefficient will increase as condensation proceeds and the film becomes thicker.

Subcooling of the condensate occurs as a result of the temperature gradient through the film. Further subcooling, if required, would normally occur by virtue of a falling film mechanism. Very occasionally the liquid loading (amount of condensate formed) is so high that slug flow occurs before condensation is complete and any subcooling required will then take place under full or flooded tube conditions, as illustrated in Sketch 5.2. In these circumstances both the condensate film coefficient and the subcooling coefficient will be considerably lower than if an annular film had been maintained. To ensure that a tube runs full of liquid, a loop seal or similar device should be installed in the condensate discharge line, thereby causing the bottom channel of the heat exchanger to be flooded.

Transitions between different flow regimes are never abrupt; they occur smoothly over a range of the controlling parameters and a finite length of the tube.



Sketch 5.2 Slug flow with full tube subcooling

5.2 Flow Regime Transitions

5.2.1 Mist-annular to annular transition

Although film coefficients are high, operation of industrial condensers well into the mist-annular regime can present two undesirable characteristics:

- (i) pressure drop may be unacceptably high,
- (ii) tube erosion may occur.

There are few data for vertical downflow that enable the transition to be specified with complete confidence under condensing as distinct from adiabatic conditions. Consequently, care must be exercised in using any procedure for ascertaining the position of the transition. However a reasonable assumption is that tube

orientation will not significantly affect flow patterns in shear controlled annular flows, where gravity effects are negligible, and so a simple criterion for flow inside horizontal tubes, proposed by Tandon *et al.*⁹, can be used for vertical flow. It is supported by condensation data, and suggests that entrainment is likely to be significant when

$$j_g^* = \frac{\dot{M}_g}{\frac{\pi}{4} D^2 [D \rho_g (\rho_l - \rho_g) g]^{1/2}} > 6 . \quad (5.1)$$

5.2.2 Annular to slug transition

Available data enabling the annular to slug transition to be specified under condensing conditions in vertical downflow are non-existent. Consequently, the basis of any prediction must be adiabatic gas-liquid data. These data are few and insufficient to warrant any re-correlation.

Barnea *et al.*¹⁰ have, on the basis of a semi-theoretical analysis, suggested that transition to slug flow will occur when

$$\frac{A_l}{A} = 0.35 . \quad (5.2)$$

Their data for air-water in 25 mm and 51 mm diameter tubes were in reasonable agreement with that criterion. The condensate film thickness, δ , must be ascertained in order to determine A_l , given by

$$A_l = \pi(D\delta - \delta^2), \quad (5.3)$$

$$\text{where} \quad \delta = 0.135 Re_f^{0.583} \left[\frac{\rho_l (\rho_l - \rho_g) g}{\eta_l^2} \right]^{-1/3} . \quad (5.4)$$

The derivation of Equation (5.4) is given in Appendix A.

5.3 Correlations for the Condensate Film Coefficient

The condensate film coefficient may be expressed either as the local (point) value, $\alpha_{c,x}$, or as the mean value, $\alpha_{c,m}$, over the total condensing length of the tube.

The recommended correlations have been identified by correlating data extracted from the literature. A description of the data banks and comparison of the predictions of the correlations with the data are given in Section 6.

5.3.1 Local condensate film coefficient

The condensate film in vertical downflow is subject to both shear forces and gravity forces, as described in Section 5.1 of this Data Item. The method presented here combines in one expression, in a form proposed by Chen *et al.*¹³, the theoretical analysis of Dukler² for the shear component with established forms of

correlation for gravity controlled wavy-laminar and gravity controlled turbulent film condensation. The correlation is

$$Nu_x = \frac{\alpha_{c,x}}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l(\rho_l - \rho_g)g} \right]^{1/3} = \left[\underbrace{\left(0.31 Re_{f,x}^{-1.32} + \frac{Re_{f,x}^{2.4} Pr_l^{3.9}}{2.37 \times 10^{14}} \right)^{1/3}}_{\text{Wavy-laminar gravity component}} + \underbrace{\frac{Pr_l^{1.5} \tau_i^*}{771.6}}_{\text{Shear flow component}} \right]^{1/2}, \quad (5.5)$$

Turbulent gravity component

where $Re_{f,x} = 4\Gamma/\eta_l = 4\dot{M}_l/\pi D\eta_l$, (5.6)

$$\tau_{i,x}^* = \Lambda \left(\frac{x_g}{1-x_g} \right)^{1.4} Re_{f,x}^{1.8}, \quad (5.7)$$

and $\Lambda = \frac{0.252 \eta_l^{1.177} \eta_g^{0.156}}{D^2 g^{2/3} \rho_l^{0.553} \rho_g^{0.78}}$, (5.8)

which is the dimensionless group presented by Chen *et al.*¹³

Typically, Λ lies in the range 5×10^{-6} to 2×10^{-4} .

Equation (5.7) gives a satisfactory fit to the available data (see Section 6.1). However, τ_i^* calculated from this equation attains a maximum value at $x_g = 0.81$ (see Section B.1). At higher values of x_g Equation (5.9) should be used.

$$\tau_{i,x>0.81}^* = 0.383 \Lambda \left(\frac{Re_{f,x}}{1-x_g} \right)^{1.8}. \quad (5.9)$$

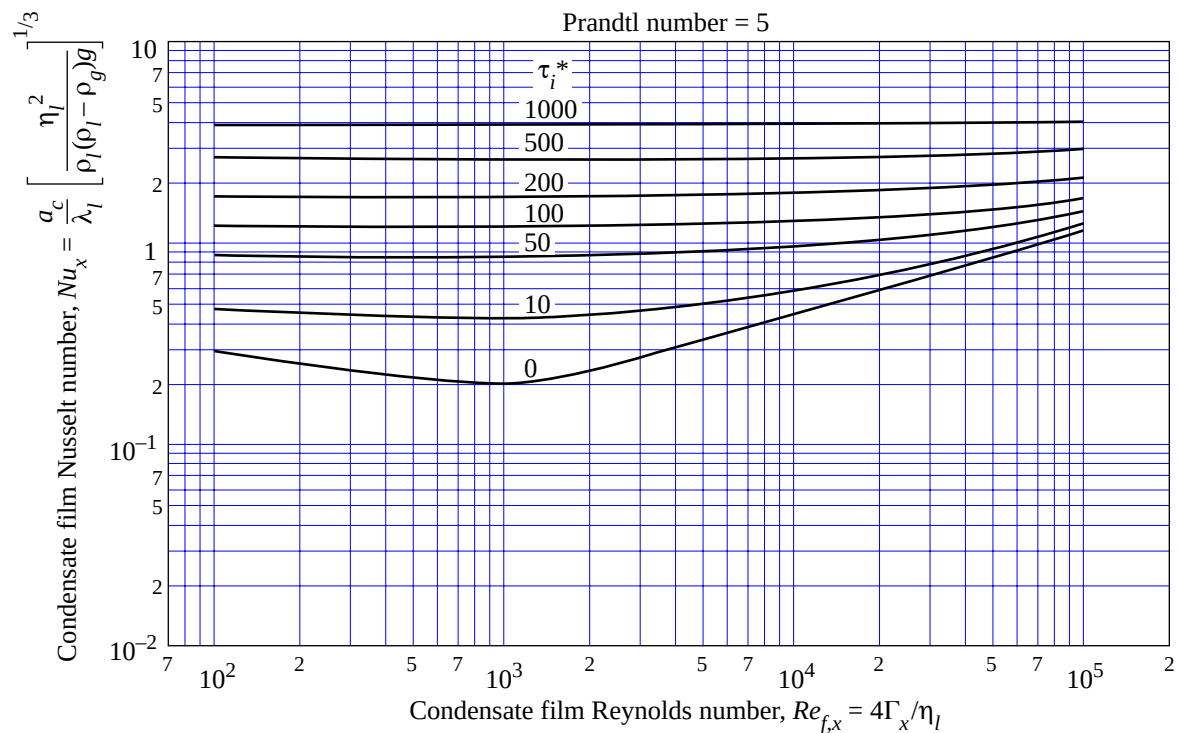
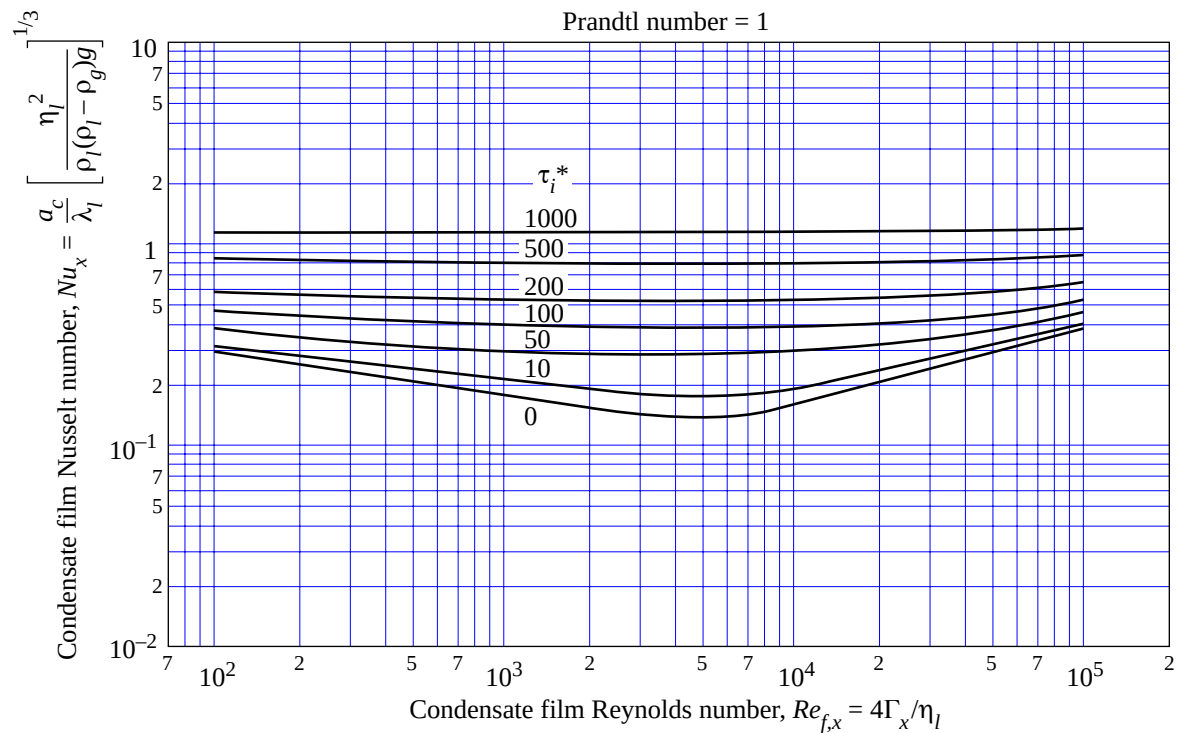
Equation (5.9) will lead to a small underprediction of the condensate film coefficient but at high values of x_g that coefficient is high and, when combined with the coolant, fouling and tube wall coefficients, the resultant reduction in overall coefficient, U , will be small – in most cases less than 5%.

When vapour shear is negligible, Equation (5.5) simplifies to

$$Nu_x = \left(0.31 Re_{f,x}^{-1.32} + \frac{Re_{f,x}^{2.4} Pr_l^{3.9}}{2.37 \times 10^{14}} \right)^{1/6}. \quad (5.10)$$

The form of relationship between modified Nusselt number, Nu_x , and condensate film Reynolds number, $Re_{f,x}$, for a range of values of the interfacial shear stress, τ_i^* , and for two values of film Prandtl number, Pr_l , is illustrated in Sketch 5.3.

The derivation of Equation (5.5) is outlined in Appendix B.



Sketch 5.3 Relationship between modified Nusselt number and condensate film Reynolds number (Equation (5.5))

5.3.2 Mean condensate film coefficient

A mean coefficient, where there is no liquid flow at entry, may be defined by

$$\frac{Re_{f,L}}{Nu_m} = \int_0^{Re_{f,L}} \frac{d(Re_{f,x})}{Nu_x}. \quad (5.11)$$

The integration is strictly valid only where the saturation temperature and wall temperature are constant, that is for single component vapours condensing at constant pressure. In order that the integration of Equation (5.11) may be carried out, some approximations are made (see Appendix B). As a consequence, mean values of the condensate film coefficient given by Equation (5.12) must be regarded as no more than fair approximations, only suitable for preliminary calculations.

The mean coefficient* is given by

$$Nu_m = \frac{a_{c,m}}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l(\rho_l - \rho_g)g} \right]^{1/2} = \left(Re_{f,L}^{-0.44} + \frac{Re_{f,L}^{0.8} Pr_l^{1.3}}{1.718 \times 10^5} + \frac{Pr_l^{1.5} \tau_{i,m}^*}{771.6} \right)^{1/2}, \quad (5.12)$$

$$\text{where} \quad \tau_{i,m}^* = \Lambda Re_T^{1.8} \left(\frac{Re_{f,L}}{Re_{f,T}} \right)^{0.4} \left(1.25 + 0.39 \frac{Re_{f,L}}{Re_{f,T}} \right)^{-2}. \quad (5.13)$$

When vapour shear is negligible, Equation (5.12) simplifies to

$$Nu_m = \left(Re_{f,L}^{-0.44} + \frac{Re_{f,L}^{0.8} Pr_l^{1.3}}{1.718 \times 10^5} \right)^{1/2}. \quad (5.14)$$

The derivations of Equations (5.12) and (5.13) are outlined in Appendix B.

5.3.3 Evaluation of condensate film transport properties

The expressions for the condensate film coefficients given in Sections 5.3.1 and 5.3.2 are based on the assumption of constant fluid properties across the film and were derived at a mean film temperature given by

$$T_{c,m} = T_{sat} - 0.75(T_{sat} - T_w). \quad (5.15)$$

Evaluation of properties at this temperature is considered appropriate for laminar films. In the absence of data at higher condensate film Reynolds numbers, this expression should be used for wavy-laminar and turbulent film conditions. Gregorig *et al.*²⁶ noted that the use of this expression also reduced the random error of their data (for steam) in the turbulent film region.

* Strictly, an integrated mean value of the overall heat transfer coefficient, U , should be used in calculation procedures in preference to U obtained from the expression $1/U = 1/U' + 1/\alpha_{c,m}$. However it is usually more convenient in practice to use this expression.

In practice, variations in density, specific heat and usually thermal conductivity across the condensate film are small, particularly if the temperature difference, $(T_{sat} - T_w)$, across the film is small. These properties may be evaluated at the vapour saturation temperature without introducing significant error. If temperature differences are large, greater than about 20 K, the condensate viscosity, η_l , must be evaluated at the mean temperature, $T_{c,m}$.

Vapour phase properties should be evaluated at the vapour saturation temperature.

6. DATA BANKS

A data bank was prepared from information given in Derivations 16 to 23 for local coefficients and derivations 24 to 27 for mean coefficients. The scopes of the data banks are given in Tables 12.1 and 12.2 respectively.

The basis of comparison with predictions in all cases was the error, μ , defined as

$$\mu = \frac{Nu_{pred} - Nu_{expt}}{Nu_{expt}}. \quad (6.1)$$

6.1 Local Data with Vapour Shear

Data for horizontal tubes (Derivations 19, 20 and 21) were included in order to have sufficient data. This was justified on the basis that tube orientation is immaterial for shear controlled annular flows. No data for values of the j_g^* parameter less than 1.5 were included. This value is regarded as signalling the onset of the transition to stratified or wavy flow in the case of horizontal tube orientation.

The data of Bae *et al.*,^{19,20,21} Goodykoontz and Dorsch¹⁸ and Goodykoontz and Brown¹⁷ were obtained at a series of stations along an instrumented tube. Data from the point nearest to vapour entry were rejected in all cases in order to eliminate any tube entry effects such as excessive local entrainment. This consideration did not apply in the case of the Blangetti *et al.*²³ data that were obtained at a particular station well along the tube.

Every third data set of Goodykoontz and Brown¹⁷ was included, as were all the sets of Goodykoontz and Dorsch¹⁸ where the condensing fluid pressure drop over the length of the tube was less than 10 kPa (1½ lbf/in²).

The data of Carpenter¹⁶ extend the range of compounds from steam and halogenated hydrocarbons to other organic fluids. The scatter of these data is wide but, as with the data from the other sources, the majority of points lie within the ±30 % error band.

For all data sources the error, μ , was plotted against the following local quantities:

- (i) condensate film Reynolds number, $Re_{f,x}$,
- (ii) dimensionless interfacial shear stress, $\tau_{i,x}^*$,
- (iii) vapour mass fraction, x_g ,
- (iv) vapour phase Reynolds number, $Re_{g,x}$,
- (v) flow regime parameter, j_g^*