



Condensation of pure fluids in downflow on horizontal integral low-fin tube bundles

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CONDENSATION OF PURE FLUIDS IN DOWNFLOW ON HORIZONTAL INTEGRAL LOW-FIN TUBE BUNDLES

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CONDENSATION OF PURE FLUIDS IN DOWNFLOW ON HORIZONTAL INTEGRAL LOW-FIN TUBE BUNDLES

1. NOTATION AND UNITS

Two sets of self-consistent units are given – SI and British.

		SI	British
A, B	constants in Equations (A1.1) and (A1.2)	–	–
A_o	nominal area of bare tube based on D_o , $= \pi D_o L$	m^2	ft^2
A_{fo}	fin flank area of single fin	m^2	ft^2
A_i	tube inside area over length p	m^2	ft^2
A_R	ratio of tube external to internal area (see Appendix B)	–	–
A_r	nominal area of bare tube based on fin root diameter D_r , $= \pi D_r L$	m^2	ft^2
A_{ro}	inter-fin root area between two adjacent fins	m^2	ft^2
A_{to}	tip area of single fin	m^2	ft^2
B_t, B_f, B_{int}	constants in Equations (A2.7), (A2.8) and (A2.9) = 0.143	–	–
B_l	constant in Equation (A2.6) = 2.96	–	–
b	inundation exponent defined in Equation (4.6)	–	–
C	parameter defined in Equation (4.4)	–	–
c_p	specific heat capacity	J/kg K	Btu/lb °F
D_i	internal diameter of tube (see Sketch 1.1)	m	ft
D_o	diameter of tube over fins (see Sketch 1.1)	m	ft
D_r	diameter of tube at root of fins (see Sketch 1.1)	m	ft
f_f	fraction of unflooded part of fin flank blanked by retained condensate at fin root	–	–
f_{int}	fraction of unflooded part of interfin tube surface blanked by retained condensate at fin root	–	–
G	see Equations (A2.7), (A2.8) and (A2.9)	–	–
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 second law of motion	– (1.0)	lb ft/lbf h ² (4.17×10 ⁸)
h	fin height (see Sketch 1.1)	m	ft
h_{fg}	specific enthalpy of vaporisation (latent heat)	J/kg	Btu/lb
h_v	effective mean vertical fin height	m	ft
J	heat flux ratios defined in Equations (A2.7) to (A2.9)	–	–
L	incremental length of condenser tube	m	ft
$\dot{M}$	mass flow of vapour entering the condenser over an incremental length of condenser tube	kg/s	lb/h
$\dot{M}_c$	mass of vapour condensed in a bundle over an incremental length	kg/s	lb/h
$M_1, M_2 \dots$	mass of vapour condensed in the first (top), second and lower rows of tubes	kg/s	lb/h
m	parameter given by the Equation (A2.18)	–	–
N	number defining position of horizontal tube row, counted from top row where $N = 1$	–	–
Nu	condensate film Nusselt number = $\frac{\alpha_o D_o}{\lambda_l}$ (see Equation (4.1))		
n	number of tubes in vertical row	–	–
n_r	number of tubes in specified tube row	–	–
n_t	total number of tubes in bundle	–	–
p	fin pitch (see Sketch 1.1)	m	ft
p_{film}	pressure in condensate film	N/m ² (Pa)	lbf/ft ²
p_v	vapour pressure	N/m ² (Pa)	lbf/ft ²
Pr_l	liquid phase Prandtl number = $\frac{\eta_l c_{p,l}}{\lambda_l}$	–	–
$\dot{Q}$	rate of heat transfer	W	Btu/h
$\dot{q}$	heat flux	W/m ²	Btu/ft ² h
Re	Reynolds number defined in Equation (4.3)	–	–
Re_v	vapour Reynolds number	–	–
r_i	coolant fouling resistance	m ² K/W	ft ² h °F/Btu

r_o	condensate fouling resistance	$\text{m}^2 \text{ K/W}$	$\text{ft}^2 \text{ h}^\circ\text{F/Btu}$
s	interfin spacing = $p - t$ (see Sketch 1.1)	m	ft
T	temperature	K or $^\circ\text{C}$	$^\circ\text{F}$
T_r	tube wall temperature at fin root	K or $^\circ\text{C}$	$^\circ\text{F}$
T_s	saturation temperature of vapour	K or $^\circ\text{C}$	$^\circ\text{F}$
ΔT	vapour side temperature difference ($T_s - T_r$)	K	$^\circ\text{F}$
t	fin thickness (see Sketch 1.1)	m	ft
U	overall heat transfer coefficient	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
u_{max}	vapour velocity at minimum cross-sectional area based on D_o	m/s	ft/h
V	condensation volumetric flux	m/s	ft/h
x	distance radially outward along fin flank, with $x = 0$ at fin root	m	ft
x'	distance around fin perimeter	m	ft
x_g	linear dimension for gravity controlled condensate flow	m	ft
x_σ	linear dimension for surface tension controlled condensate flow	m	ft
α	condensation coefficient	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
α_i	coolant heat transfer coefficient	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
α_o	condensation coefficient based on A_o	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
α_r	condensation coefficient based on A_r	$\text{W/m}^2 \text{ K}$	$\text{Btu/ft}^2 \text{ h}^\circ\text{F}$
Γ	condensate flow rate per incremental tube length (see Section 5)	kg/m s	lb/ft h
δ	condensate film thickness	m	ft
$\varepsilon_{\Delta T}$	enhancement ratio	—	—
η	dynamic viscosity	kg/m s	lb/ft h
λ	thermal conductivity of condensate	W/m K	$\text{Btu/ft h}^\circ\text{F}$
λ_w	thermal conductivity of fin material	W/m K	$\text{Btu/ft h}^\circ\text{F}$
μ	relative error in prediction of condensate film coefficient	—	—
ρ	fluid density	kg/m^3	lb/ft^3
σ	surface tension	N/m	lbf/ft

Φ	function of ϕ given by Equation (A1.9)	—	—
ϕ	retention angle or position at which interfin space becomes full of retained condensate due to surface tension (see Sketch 3.1)	rad	rad
ψ	ratio of actual to theoretical outside surface area of tube	—	—

Additional subscripts

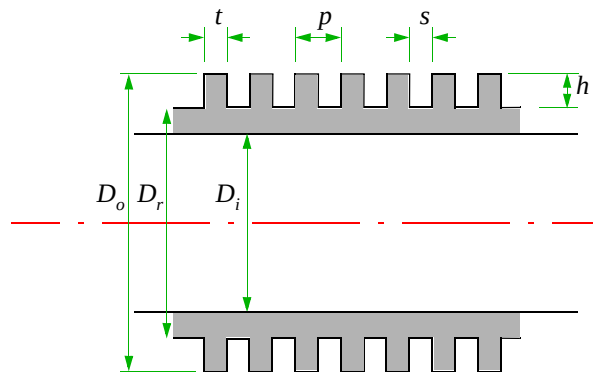
1	refers to top tube in a vertical row
<i>act</i>	actual value
<i>c</i>	refers to coolant fluid
<i>expt</i>	experimental or measured value
<i>f</i>	at fin flank
<i>film</i>	refers to condensate film
<i>flood</i>	in flooded region
<i>grav</i>	gravity component
<i>i</i>	inside tube
<i>int</i>	at interfin surface
<i>l</i>	liquid phase
<i>m</i>	at mean temperature of condensate film
<i>o</i>	outside tube
<i>pred</i>	predicted value
<i>r</i>	at fin root
<i>sh</i>	shear component
<i>sm</i>	smooth or plain tube
<i>t</i>	at fin tip
<i>th</i>	theoretical value
<i>v</i>	vapour phase

Abbreviations

<i>fpi</i>	fins per inch
<i>fpm</i>	fins per metre

Superscript

— mean or average value



Note 1 Tube wall and fin detail not to scale

Note 2 This sketch shows an idealised geometry. For purpose of analysis a rectangular fin cross-section is assumed. In practice the cross-section will be slightly trapezoidal with rounded corners resulting in the actual surface area per unit length of tube being less than the theoretical area. Correction for fin shape is presented in Section 8.3.

Sketch 1.1 Low-fin tube notation

2. INTRODUCTION

2.1 Background

Integral low-fin tubing has been widely used in condensation applications in refrigeration and air conditioning plant since the late 1940s. More recently low-fin tubing has been increasingly specified for condensers in the process industries generally.

Transverse circumferential fins of limited height (hence the term 'low-fin') are formed on a tube outside wall by a cold rolling process. Typical fin heights are from 0.75 mm to 3 mm (0.03 in to 0.12 in) on tubes of 12.7 mm to 38 mm ($\frac{1}{2}$ in to $1\frac{1}{2}$ in) nominal outside diameter with from 350 to 1580 fins per metre (9 to 40 fins per inch). Tubes are available in a range of ferrous and non-ferrous materials to suit process conditions.

The condensation rate is greater than that on plain tubes of the same nominal outside diameter (D_o) and length. This arises on account of (i) the increased surface area per unit length of tubing and (ii) surface tension effects. Use of low-fin tubing will therefore usually enable the size (tube length and/or bundle diameter) and hence weight of a condenser for a prescribed duty to be reduced compared with the equivalent plain tube unit.

Low-fin tubes are used to maximum advantage in the condensation of low surface tension fluids such as refrigerants and light hydrocarbons. With increasing surface tension the amount of retained condensate in the interfin zone (see Sketch 3.1) becomes progressively greater, thereby decreasing the surface available for heat transfer. The use of low-fin tubing with high surface tension fluids such as water is rarely justified.

However low-fin tubes are expensive compared with plain tubes of the same diameter so, for new plant, cost may well be the deciding factor on whether or not low-fin tubes are specified for a given duty. As a guide low-fin tubes will not normally be economically viable unless the outside bare or smooth tube heat transfer coefficient is less than a quarter of the inside coefficient. (In the case of very expensive materials such as hastelloy or titanium, the factor will be nearer one third.)

Replacement of plain tubes with low-fin tubes of the same overall diameter is often a cost-effective solution when plant up-rating is being considered. However, because for a given overall diameter (D_o) the tube internal diameter (D_i) will be smaller than that of the equivalent plain tube, the resultant tubeside pressure drop will be greater.

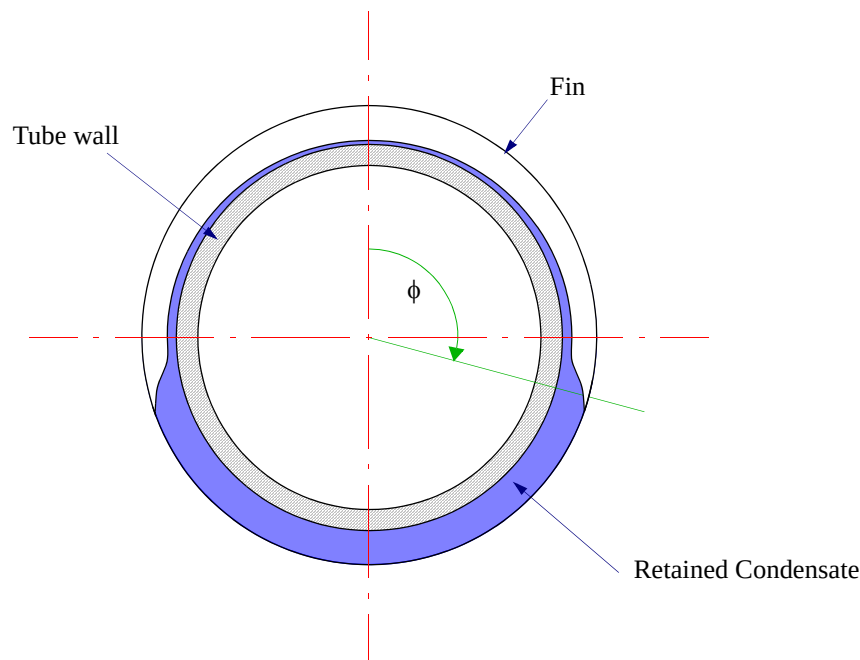
2.2 Purpose and Scope

This Data Item presents the available data and provides correlations for predicting the local condensate film coefficient for the downflow of a pure vapour or azeotropic mixture over horizontal low-fin tubes and tube bundles. It is applicable to those cases where the saturation temperature over the length of the condensing path remains approximately constant.

3. THE CONDENSATION PROCESS ON FIN TUBES

Enhancement of heat transfer on low-fin tubes is greater than might be expected solely on the basis of increased surface compared with plain tubes of the same diameter over the fins (D_o). This enhancement is due to surface tension effects.

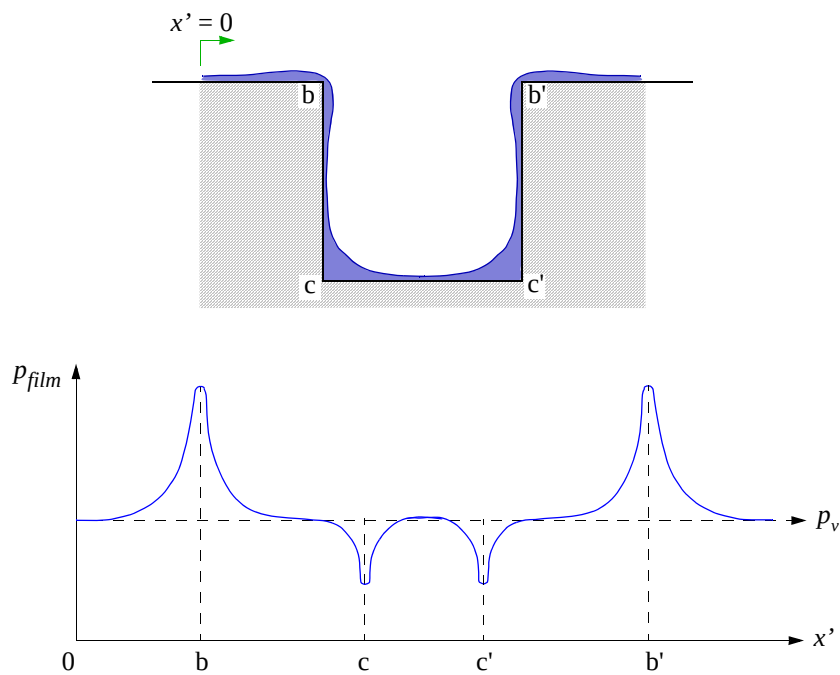
Condensate at the upper part of the tube is pulled towards the fin root by the combined effects of gravity and surface tension, whence it is drained circumferentially by gravity. At the bottom portion of the tube condensate is retained by capillary forces in the interfin zone, completely filling and effectively insulating that region (Sketch 3.1).



Sketch 3.1 Integral-fin tube with retained condensate

Abrupt changes of curvature of the condensate film/vapour interface occur near the fin tip and fin root. Convex curvature at the fin tip causes a pressure gradient in the film such that the flow on the fin flank is greater than if condensate drainage was due to gravity alone. The consequent thinning of the condensate film results in enhanced heat transfer. Conversely, concave curvature of the film at the fin root will cause condensate to be retained in this region in the form of a 'wedge'. Surface tension thus serves to thin the condensate film on the fin flanks, fin tips and in the interfin spaces while at the same time blanking a small area at the fin root corners.

The overall pressure profile in the condensate film along a fin is illustrated in Sketch 3.2¹⁶.



Sketch 3.2 Typical pressure distribution in the condensate film in the unflooded region

4. CONDENSATE FILM COEFFICIENT

4.1 Flow Regimes on Low-fin Tubes

Condensate flow on low-fin tubing may be expected to be subject to both shear forces and gravity forces as well as surface tension generated forces. The magnitude of the shear and gravity components will be a function of vapour flow conditions, in particular of vapour velocity. The flow in a gravity controlled condensate film may be expected to be either laminar or turbulent, depending upon the magnitude of the condensate film Reynolds number.

4.2 Flow Regime Transitions

In laminar films in or on plain tubes the heat transfer mechanism is one of conduction through the film. For downflow over a vertical row of horizontal tubes condensate from one row will discharge onto the next, progressively thickening the film and decreasing the film coefficient. With further condensate formation, expressed as a film Reynolds number, $4\Gamma/\eta_l$, the film may be expected to become progressively unstable and the flow in the film will become turbulent. The heat transfer mechanism then becomes one of convection. The film coefficient will now increase as the film Reynolds number increases.

Chu and McNaught¹², however, in work on downflow over a low-fin tube bundle over a wide range of Reynolds numbers, observed that no point was reached where the condensate film coefficient tended to increase with Reynolds number, so characterising transition from a laminar condensate film to a turbulent film. Hence for downflow in tube bundles, typically TEMA X-shells, a laminar correlation for the condensate film coefficient should be used. This is consistent with the available data.

If the vapour velocity at the minimum cross-sectional flow area of a bundle is sufficiently high the flow and consequently the thickness of the condensate film will be in part controlled by vapour shear at the vapour-liquid interface and in part by gravity. As condensation proceeds down a bundle the vapour shear component becomes progressively less and the gravity component more important as the controlling mechanism. The two components may be combined by the relationship presented in Section 4.3.3.

4.3 Correlations for the Condensate Film Coefficient

The correlations presented in Sections 4.3.1 and 4.3.2 were developed for condensation on single tubes. Extension of these methods to downflow condensation on tube bundles is presented in Section 4.3.4.

An enhancement ratio, $\varepsilon_{\Delta T}$, is defined as

$$\varepsilon_{\Delta T} = \frac{\text{condensate film coefficient for finned tube}}{\text{condensate film coefficient for plain tube at same vapour side } \Delta T}$$

Both coefficients are based on the smooth tube area at the fin root diameter.

Account is taken of condensate retention as well as surface tension and gravity effects on the flow of the condensate film.

Expressions are obtained for the mean heat flux for the fin tip, fin flanks and interfin space which, when multiplied by the appropriate surface areas and added, give the mean heat flux for the tube.

Derivation of expressions for $\varepsilon_{\Delta T}$ is outlined in Appendix A. Two cases are presented as follows.

- (i) The surface temperature of the fin is assumed constant and equal to the fin root temperature; that is, there is no temperature gradient along the fin. This case may be used with small error when the tube material is of high thermal conductivity (copper for example) and when condensate thermal conductivity is low, such as for a refrigerant. See Appendix A2.1.
- (ii) For low thermal conductivity materials such as bronze or steel and high thermal conductivity condensates conduction along the fin with resultant reduction in ΔT should be taken into account in accordance with the procedure proposed by Briggs and Rose¹⁰. See Appendix A2.2.

When low thermal conductivity materials are to be used, the fins ideally should be short and thick. In this way the temperature gradient along the fin and the accompanying loss of performance will be minimized. Conversely, with high conductivity materials the fins should be longer and thinner allowing more fins per unit length of tube and hence enabling greater surface to be installed.

It is difficult to define a simple criterion for when fin efficiency becomes important. Temperature drop along the fin is governed by the magnitudes of the fin flank and fin tip condensation film coefficients, α_f and α_t , given by Equation (A2.15) and (A2.16) respectively. As these coefficients are not known initially, they cannot be used to ascertain whether or not the modified expressions for conduction along the fins should be used.

As a rough guide, conduction along the fin may be significant when the heat transfer coefficient (based on the plain tube area at the fin root diameter), calculated on the assumption of uniform surface temperature conductivity of the fin, is greater than about 20 000 W/m² K and the thermal conductivity of the fin material is less than about 100 W/m K. These values are based on extensive calculations¹⁷ using both models for a wide range of fluids, fin materials and tube geometries, and are supported by the available experimental data for tubes of low thermal conductivity.

4.3.1 Gravity controlled flow

The gravity controlled condensate film coefficient, α_{grav} , is predicted by the method proposed by Rose^{11, 16}.

4.3.2 Shear controlled flow

The vapour shear controlled component is predicted by the method proposed by Cavallini *et al.*¹³ and, where present, is combined with the gravity component in accordance with Equation (4.5).

The shear-controlled film coefficient, α_{sh} , is obtained from

$$\alpha_{o, sh} = \frac{\lambda_l}{D_o} Nu_{o, sh} \quad (4.1)$$

where $Nu_{o, sh} = C Re^{0.8} Pr_l^{0.33}$, (4.2)

$$Re = \frac{\rho_v u_{max} D_o}{\eta_l} \left(\frac{\rho_l}{\rho_v} \right)^{0.5}, \quad (4.3)$$

$$C = 0.03 + 0.166 \frac{t}{p} + 0.07 \frac{h}{p}. \quad (4.4)$$

4.3.3 Combination of gravity and shear components

The gravity and shear components are combined by the relationship

$$\alpha_o = (\alpha_{o, sh}^2 + \alpha_{o, grav}^2)^{1/2}. \quad (4.5)$$

For values of the gas phase Reynolds number, $(\rho_v u_{max} D_o / \eta_v)$, less than about 70 000, the shear component will be small and may usually be neglected for hand calculations.

4.3.4 Application to tube bundles

For a vertical row of horizontal tubes a proportion of condensate from one row of tubes will discharge on to the next in line, thereby increasing the condensate film thickness on the lower tubes and decreasing the film coefficient. This process is referred to as inundation.

A substantiated mechanistic model of the inundation process is not available as the process is complex and subject to a number of unsystematic effects, as listed here.

- (i) Condensate does not usually fall as a continuous sheet, uniform along the length of the tubes.
- (ii) Splashing with considerable droplet formation may occur.
- (iii) Some liquid may flow down the gaps between tube rows.
- (iv) Tubes may be slightly bowed resulting, in the case of plain tubes, in some liquid flow along the tubes. Note that fins reduce axial spreading.

Data for the mean heat transfer coefficient, $\bar{\alpha}_o$, for a bank of horizontal tubes have been correlated by expressions of the form

$$\bar{\alpha} = \alpha_1 n^{-b}, \quad (4.6)$$

where α_1 is the heat transfer coefficient for the top tube in the row.

The coefficient for the N^{th} tube from the top will be

$$\alpha_N = \alpha_1 \left\{ N^{(1-b)} - (N-1)^{(1-b)} \right\}. \quad (4.7)$$

Some values of the exponent, b , for low-fin tubes have been reported in the literature and obtained under gravity controlled laminar or wavy/laminar film conditions as follows.*

b	Fluids	Reference
0.06	R-12, acetone, steam, <i>n</i> -butane	$\left. \begin{array}{c} 1 \\ \text{reported in } 5 \\ 4 \end{array} \right\}$
0.045	steam	
0.15	R-11	
0.0	R-11	

* A value of $b = 0.167$ is frequently used with plain tubes.