



Reflux condensation in vertical tubes

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Item was monitored and guided by the following Working Party on behalf of the Heat Transfer Steering Group

Mr J.M. McNaught	— National Engineering Laboratory
Mr M.J.C. Moore	— John Brown Engineers and Constructors Ltd
Mr D.S. Robertson	— B.P. Research Centre
Mr R.E. Sacks	— Air Products Ltd.

The Heat Transfer Steering Group first met in 1966 and now has the following membership:

Chairman	
Mr D. Butterworth	— HTFS, Atomic Energy Authority, Harwell
Vice-Chairman	
Dr R. Cheesewright	— Brunel University
Members	
Mr R.J. Allam	— Air Products Ltd
Dr T.R. Bott	— University of Birmingham
Prof. P.J. Heggs	— University of Bradford
Mr T.J. Loraine	— Vital Technologies Ltd
Dr A. C. Pauls*	— Monsanto Company, USA
Mr A.C. Rapier	— Atomic Energy Authority, Windscale
Prof. R.H. Sabersky*	— California Institute of Technology, USA
Mr E.A.D. Saunders	— Independent
Mr R.A. Smith	— Independent
Dr J. Taborek*	— Independent
Mr M.A. Taylor	— Independent
Mr D.A. Winkworth	— Foster Wheeler Energy Ltd.

* Corresponding Member

The technical work involved in the assessment of the available information and the construction and subsequent development of the Data Item was undertaken by

Mr G.H. Walter	— Head of Heat Transfer, Internal Flow and Physical Data.
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REFLUX CONDENSATION IN VERTICAL TUBES

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REFLUX CONDENSATION IN VERTICAL TUBES

1. NOTATION AND UNITS

		Units	
		SI	British
A	heat transfer surface area	m^2	ft^2
Bo	Bond number* = $D^2(\rho_l - \rho_g)g/(g_c \sigma)$	—	—
c_f	Fanning friction factor = $\Delta p_f \left[4 \left(\frac{L}{D} \right) \frac{\rho V^2}{2g_c} \right]^{-1}$	—	—
c_p	specific heat capacity at constant pressure	J/kg K	Btu/lb °F
D	internal diameter of tube	m	ft
f_w	tube-wall friction factor in Equation (C2.1)	—	—
Fr	Froude number = $V/(g/D)^{1/2}$	—	—
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/lb h^2 (4.17×10^8)
h	specific enthalpy	J/kg	Btu/lb
K	static pressure drop coefficient	—	—
K_g	gas phase Kutateladze number = $V_g \rho_g^{1/2} [g g_c \sigma (\rho_l - \rho_g)]^{-1/4}$	—	—
L	tube length	m	ft
$\dot{M}$	mass flowrate per tube	kg/s	lb/h
$\dot{M}_{tot}$	total mass flow rate for all tubes	kg/s	lb/h
$\dot{m}_c$	condensation mass flux towards vapour-liquid interface	kg/s m^2	lb/h ft^2
N	number of tubes in a bundle	—	—
n	number of flooding data points	—	—
N_F	dimensionless number in Equation (B1.1) = $V_l D g^{1/4} (\rho_l - \rho_g)^{3/4} / [4 (g_c \sigma)^{3/4}]$	—	—
Nu	condensate film Nusselt number = $\frac{\alpha_c}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l (\rho_l - \rho_g) g} \right]^{1/3}$	—	—

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Pr	Prandtl number = $\eta c_p / \lambda$	—	—
$\dot{Q}$	rate of heat transfer	W	Btu/h
Re_f	condensate film Reynolds number = $4(\Gamma/\eta_l)$	—	—
Re_g	vapour phase Reynolds number = $\rho_g V_g D / \eta_g$	—	—
r_f	fouling resistance	$m^2 K/W$	$ft^2 h F/Btu$
T	temperature	K or °C	°F
U	overall heat transfer coefficient	$W/m^2 K$	$Btu/ft^2 h F$
V	superficial velocity = $4\dot{M}/(\rho \pi D^2)$	m/s	ft/h
$V_{g,\alpha}$	actual vapour velocity	m/s	ft/h
$V_{g,crit}$	vapour superficial flooding velocity	m/s	ft/h
x_g	vapour mass fraction = $\dot{M}_g/(\dot{M}_g + \dot{M}_l)$	—	—
Z	parameter in Equations (8.1) and (8.2)	—	—
α	heat transfer coefficient	$W/m^2 K$	$Btu/ft^2 h F$
Γ	condensate flowrate per wetted perimeter = $\dot{M}_l/(\pi D)$	k/s m	lb/h ft
δ	condensate film thickness	m	ft
δ^*	dimensionless condensate film thickness defined by Equation (C2.4)	—	—
Δh_v	latent heat of vaporisation	J/kg	Btu/lb
Δp	pressure drop	N/m^2 (Pa)	lbf/ft^2
Δp_f	frictional pressure loss	N/m^2 (Pa)	lbf/ft^2
ε_{wrms}	weighted root mean square error	—	—
ε_{wm}	weighted mean error	—	—
ζ	deviation	—	—
η	dynamic viscosity	kg/s m	lb/ft h
η_w	dimensional constant in Equations (5.1a) and (5.1b) numerically equal to the viscosity of water at 20°C	kg/s m (0.001)	lb/ft h (2.42)
θ_m	logarithmic mean temperature difference	K	°F
λ	thermal conductivity	$W/m K$	$Btu/h ft F$
ξ	product $\dot{m}_c c_{pg} / \alpha_g$ (Equation (8.3))	—	—
ρ	fluid density	kg/m^3	lb/ft^3
σ	surface tension	N/m	lbf/ft

ϕ	angle of diagonal cut at tube bottom measured from horizontal	degrees	degrees
ϕ_g	two-phase multiplier: $\left[\frac{\text{two-phase frictional pressure gradient}}{\text{frictional pressure gradient assuming gas flowing alone}} \right]^{1/2}$	—	—
X	Martinelli parameter: $\left[\frac{\text{frictional pressure gradient assuming liquid flowing alone}}{\text{frictional pressure gradient assuming gas flowing alone}} \right]^{1/2}$	—	—

Extra subscripts

c	condensate film
eff	effective coefficient
g	gas or vapour phase
h	homogeneous
i	refers to internal surface area
in	at stream inlet
inc	increment
L	at bottom of a tube
l	liquid phase
lg	two-phase
m	mean value
o	refers to external surface area
out	at stream outlet
s	service or coolant stream
sat	at saturation
t	tube wall
tot	total flow or heat duty
x	at distance x from top of tube

* The Bond number is sometimes defined as $D[(\rho_l - \rho_g)g/(g_c \sigma)]^{1/2}$. Care must be exercised in transferring from one publication to another.

2. INTRODUCTION

Reflux condensation refers here to the condensation process in which vapour enters a vertical or inclined mounted condenser at the bottom of the heat transfer matrix and flows upward whilst the condensate formed flows downward under gravity. The vapour and condensate are thus in counter-current flow.

Occasionally reflux condensers are referred to as 'knock-back' condensers and are also sometimes called dephlegmators where the duty is partial condensation of a mixture: for example, overhead condensers on distillation columns.

Typical applications of reflux condensation are in

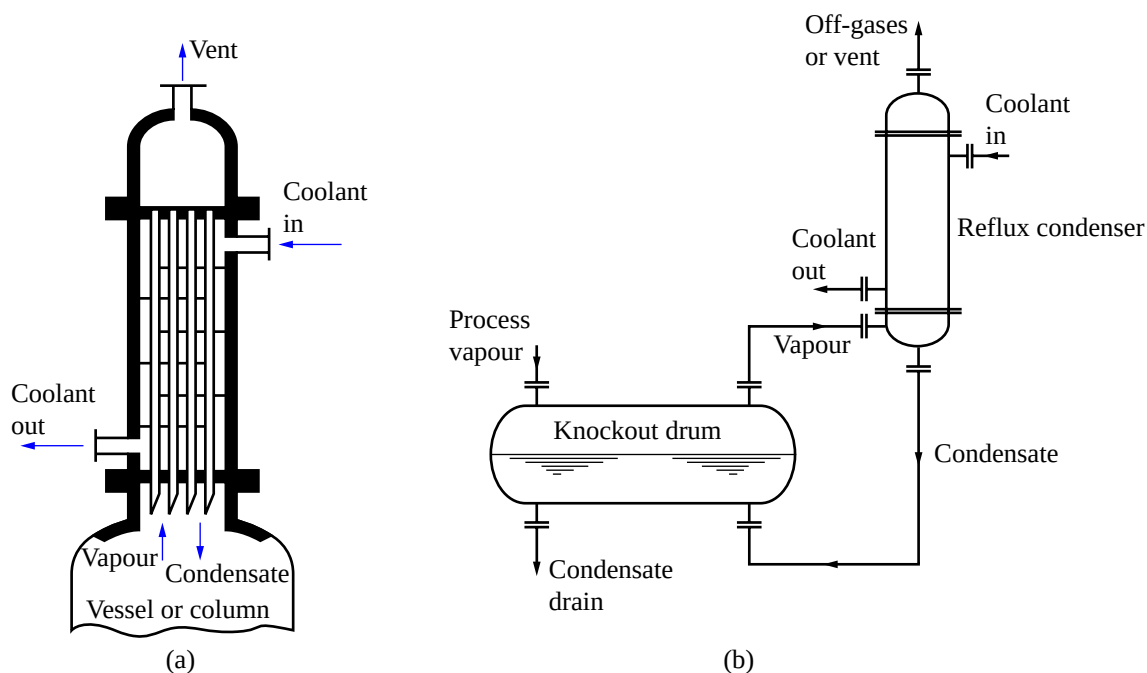
- (i) distillation column overhead condensers,
- (ii) vent condensers, such as on reactors or stirred vessels,
- (iii) the vent cooling section of air-cooled steam condensers,
- (iv) gravity assisted, wickless heat pipes.

The principle is also applicable to emergency core cooling systems in certain designs of pressurised water and boiling water nuclear reactors. This application is, however, specialised and outside the scope of this Data Item.

The most common heat transfer surface geometries along which reflux condensation may occur are:

- (i) inside the vertical or inclined tubes of a shell-and-tube or air-cooled heat exchanger,
- (ii) inside the vertical passages of a plate-fin exchanger,
- (iii) outside a helical coil bundle,
- (iv) outside a shell-and-tube bundle mounted within and at the top of a vessel or column,
- (v) through the vertically orientated passages of a spiral-plate heat exchanger.

Typical installation arrangements for vertically mounted shell-and-tube reflux condensers with condensation inside the tubes are shown, schematically, in Sketch [2.1](#)



Sketch 2.1 Reflux condenser installations:
(a) flange mounted on vessel or column
(b) set off associated vessel

The thermal-hydraulic performance of reflux condensers is limited by the phenomenon of flooding. Counter-current flow of condensate and vapour is opposed by interfacial friction between the phases. For any particular vapour-liquid combination and exchanger geometry there will be a maximum vapour velocity above which liquid will be entrained in the vapour phase or is otherwise carried upward rather than draining to the bottom. This velocity is the flooding velocity.

3. PURPOSE AND SCOPE

This Data Item is concerned with reflux condensation inside plain vertical tubes of circular cross-section. Its purpose is fourfold:

- (i) to provide the correlations for predicting the onset of flooding,
- (ii) to present the available data and formulate procedures for predicting the condensate film coefficient,
- (iii) to indicate how the pressure drop of the condensing vapour, in flowing through the condenser, may be estimated,
- (iv) to outline the combined heat and mass transfer effects in the reflux situation.

Data for other geometries are sparse. However, general principles applicable to vertical tubes are also relevant to other arrangements.

4. ADVANTAGES AND DISADVANTAGES OF REFLUX CONDENSATION

4.1 Advantages

- (i) Reflux condensers may be flange mounted on top of a column or reactor vessel (see Sketch 2.1) or installed within a column in which case piping and installation costs will be reduced in comparison with a conventional overhead condenser set off the column and in which vapour and liquid flow co-currently.
- (ii) Reflux condensers are self-venting. The light components and inert gases are swept to the top of the condenser by the vapour flow.
- (iii) On distillation column duties, reflux condensation is accompanied by additional vapour-liquid separation equivalent to several additional separation stages.
- (iv) In a reflux condenser the rising hot vapour will be cooled by the condensate flowing down the tube walls. The net effect of this heat transfer between vapour and condensate is to minimise cooling of the condensate. A 'hot' condensate return reduces the heat input requirement of the vessel or column and represents an energy saving.
- (v) A further consequence of hot condensate return is that condensation of unwanted light (low boiling point) components is prevented. They are discharged at the top of the condenser together with any non-condensing gases that may be present.
- (vi) With dirty vapours a reflux condenser will act as a scrubber.

4.2 Disadvantages

- (i) The onset of flooding limits the maximum design vapour velocity at tube inlet. More tubes are necessary to meet this condition than would be the case with co-current flow of vapour and condensate. The shell diameter will consequently be larger and the unit more expensive.

Furthermore, the predictive accuracy of the flooding correlations (see Sections 5.2 and 5.3) is such that a significant safety margin has to be applied to any calculated flooding velocity.
- (ii) The flooding limitation also implies that vertical, in-tube reflux condenser designs are not normally appropriate to situations where large amounts of non-condensing gases are present.
- (iii) The possibility exists of flow maldistribution between tubes or groups of tubes, with attendant parallel path instability, occasioned by the very low tube-side pressure drop inherent in in-tube reflux condensation.
- (iv) Reflux condensation in tubes requires a single tube-side pass and, in the case of a shell-and-tube exchanger, the insertion of expansion bellows or a packed head to relieve the resultant tube stresses may be required. The use of bellows or a packed head may not be acceptable to the equipment user.
- (v) On distillation duties, control of the liquid return rate of the column can only be accomplished by changing the column heat duty.

5. FLOODING

5.1 The Flooding Mechanism

The detailed mechanism of flooding is not fully understood. In the context of reflux condensers, understanding is further obscured by the following considerations.

- (i) Much of the experimental work on flooding has been undertaken on adiabatic systems with no mass transfer, typically the air-water combination.
- (ii) The basic experimental layout for the study of flooding phenomena has consisted of introducing the liquid phase to the top of the test section or tube through an upper plenum or porous injector and introducing the gas or vapour at the bottom of the test section, from a lower plenum. The liquid film, before the onset of flooding, is assumed to be of uniform thickness along the length of the tube. In a reflux condenser no significant amount of liquid enters at the top of the tube and the liquid (condensate) film increases in thickness from the top to the bottom of the tube.
- (iii) Momentum transfer from the vapour to the liquid phase will occur with phase change. Interfacial shear will be augmented as a result.

There is general acceptance that at low vapour or gas velocities the falling film is smooth or contains small ripples, depending on the film Reynolds number. With increasing vapour or gas velocities larger disturbance waves appear on the film. Mechanisms for the onset of flooding, proposed in the literature and based on adiabatic data, are summarised by Zabaras and Dukler²⁰ and include the following.

- (a) At flooding the waves become unstable, grow rapidly and either
 - (α) block the gas flow, causing the liquid to be forced up as intermittent slugs, or
 - (β) are broken up by the gas and flow upwards as entrained droplets.
- (b) At gas flowrates below flooding, the waves are seen to flow downward. As the flooding point is reached with increasing gas flowrate, the waves become stationary. A further increase in gas flowrate causes the waves to flow upward and flooding is imminent.

In their own work on 50.8 mm internal diameter tubes, Zabaras and Dukler observed that waves propagate only downward and were never of such an amplitude as to bridge the tube. In contrast to these observations Deakin *et al*²⁸, in a limited study specific to reflux condensation of steam at atmospheric pressure in a 19 mm diameter glass tube, noted that waves did bridge across the tubes.

5.2 Correlation of Flooding Data

A comprehensive data bank of 2762 experimental flooding data points was compiled from the literature by McQuillan and Whalley³². The range of their data was as follows.

Tube internal diameter	4 – 300 mm (78% of the data were for tubes less than 50 mm diameter).
Liquid density	741 – 1252 kg/m ³ .
Liquid viscosity	0.0001 – 0.098 kg/s m.
Surface tension	0.017 – 0.072 N/m.

All data were for adiabatic (no mass transfer) conditions.

Twenty-two flooding correlations were compared with the data. The criteria of comparison were the weighted percentage error, ε_{wm} , the r.m.s. weighted error, ε_{wrms} , and the deviation, ζ . These quantities are defined in Appendix A.

The most successful correlation was that of Alekseev *et al.*⁹ McQuillan and Whalley³² modified the correlation to improve accuracy. It is presented here (Equations (5.1a) and (5.1b)) in a rearranged form, convenient for reflux condenser design and analysis.

$$V_{g,crit} = 1.08D^{0.25} \left[\frac{\rho_l^{0.18} (\rho_l - \rho_g)^{0.28}}{\rho_g^{0.59}} \frac{\sigma^{0.13}}{\left(1 + \frac{\eta_l}{0.001}\right)^{0.15}} \right] \left(\frac{\dot{M}_l}{\dot{M}_g} \right)^{-0.18} \quad \text{with SI units}^* \quad (5.1a)$$

$$V_{g,crit} = 9400D^{0.25} \left[\frac{\rho_l^{0.18} (\rho_l - \rho_g)^{0.28}}{\rho_g^{0.59}} \frac{\sigma^{0.13}}{\left(1 + \frac{\eta_l}{2.42}\right)^{0.15}} \right] \left(\frac{\dot{M}_l}{\dot{M}_g} \right)^{-0.18} \quad \text{with British units.}^* \quad (5.1b)$$

Further information on the Alekseev *et al.* equation will be found in Appendix B.

The next most successful correlation was that of English *et al.*⁷, also presented in rearranged form.

$$V_{g,crit} = 0.31 \frac{D^{0.30}}{(\cos \phi)^{0.32}} \left[\frac{\rho_l^{0.46} \sigma^{0.09}}{\rho_g^{0.50} \eta_l^{0.14}} \right] \left(\frac{\dot{M}_l}{\dot{M}_g} \right)^{-0.070} \quad \text{with SI units,} \quad (5.2a)$$

$$V_{g,crit} = 8750 \frac{D^{0.30}}{(\cos \phi)^{0.32}} \left[\frac{\rho_l^{0.46} \sigma^{0.090}}{\rho_g^{0.50} \eta_l^{0.14}} \right] \left(\frac{\dot{M}_l}{\dot{M}_g} \right)^{-0.070} \quad \text{with British units.} \quad (5.2b)$$

For total condensation of vapour, where all the vapour is condensed and discharges from the bottom of the tube, $\dot{M}_l/\dot{M}_g = 1$. As the amount of inert gas and uncondensed vapour discharging at the top of the tubes increases, $\dot{M}_l/\dot{M}_g$ will correspondingly decrease.

The statistical analysis of the original form of these correlations, as presented by McQuillan and Whalley,³² is summarised in Table 5.1.

TABLE 5.1 Statistical analysis of correlations

	Alekseev <i>et al.</i> ⁹	Modified Alekseev <i>et al.</i> ¹⁹	English <i>et al.</i> ⁷
r.m.s. weighted error %	46	40	44
weighted error %	28	26	30
deviation	0.07	0.00	-0.07

* Indices have been rounded to two significant figures. See Appendix B.

The English *et al.*⁷ correlation, Equations (5.2a) and (5.2b), has been noted, partly because the weighted errors and deviation are close to those of Alekseev *et al.*, and partly because it was derived from condensing data. The limitations of the English *et al.* correlation should, however, be borne in mind. They are:

- (i) the correlation was based on a total of only 56 flooding determinations,
- (ii) the 56 data points were for four compounds: water, *n*-propyl alcohol, *n*-heptane and carbon tetrachloride, and four tube-end diagonal cuts,
- (iii) one tube diameter was used throughout,
- (iv) no raw data are presented.

The range of their physical property data was as follows.

Liquid density	610 – 1480 kg/m ³ .
Vapour density	0.6 – 5.5 kg/m ³ .
Liquid viscosity	0.00028 – 0.00049 kg/s m.
Surface tension	0.018 – 0.059 N/m.

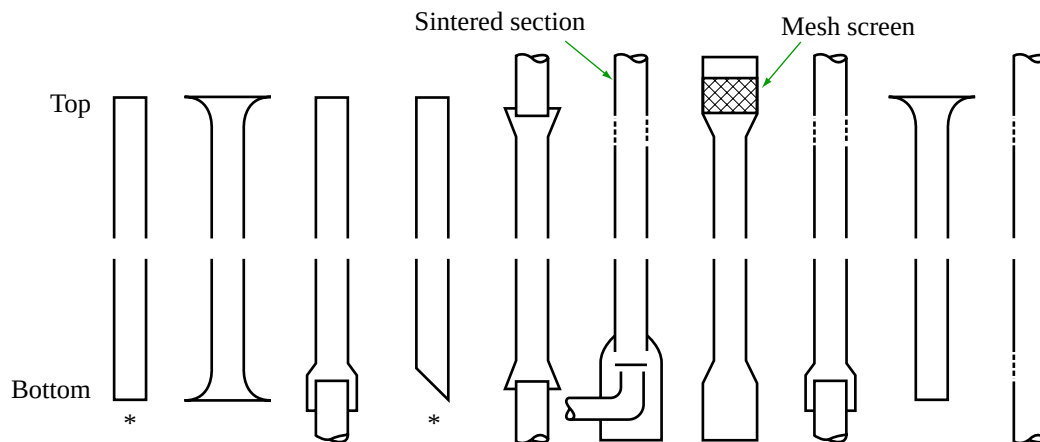
5.3 Data Scatter

The considerable data scatter implicit in the values of the weighted and r.m.s. weighted errors in Table 5.1 largely arises on account of two factors:

- (a) the wide variety of test section entry and exit geometries,
- (b) differences in the method of determination of the flooding point.

The gas flooding velocity is significantly increased by introducing the liquid phase as a 'smooth' film, for example over a radiused or tapered entry, compared with a square tube-end entry. This is attributed to the flow disturbances associated with a sharp entry increasing interfacial friction. In the case of reflux condensers there is seldom upper head introduction of liquid and the normal square cut tube-end at the top may not be expected to impair flooding performance significantly. Liquid exit tube-end style also affects flooding performance.

McQuillan and Whalley³² stated they were unable to devise criteria by which the various tube-end conditions might be accounted for. Some of the tube-end geometries they assessed are shown in Sketch 5.1.



*Either of these bottom tube-ends may be used in reflux condensers

Sketch 5.1 Some tube-end geometries used in flooding experiments

To date the flooding point has been determined by one of three methods:

- (i) noting the vapour velocity at which a large increase in entrainment occurs,
- (ii) noting the vapour velocity at which the pressure drop increases dramatically,
- (iii) visual observation.

These methods are not entirely consistent. It has been found⁸ that the flooding point determined by pressure drop can involve entrainment loads of 0.01 to 0.05 kg liquid per kg of vapour.

5.4 Comparison with Data Obtained under Reflux Condensing Conditions

Flooding data under reflux condensing conditions are limited. The available data^{23,27,28} are compared with the predictions of McQuillan and Whalley and English *et al.* in Figures 1a, 1b and 1c. None of the experimental equipment is fully typical of reflux condenser practice on account of the small diameter of the tubes used. Deakin *et al.*²⁸ used a 19 mm internal diameter tube whilst Verschoor²³ experimented with tubes of 3.5 mm to 9 mm internal diameter. Reflux condensers are usually designed with internal diameters in the range of 25 mm to 50 mm.

5.5 Improvements in Performance with Respect to Flooding

5.5.1 Tube-end diagonal cut

English *et al.*⁷ and Holmes⁴ have reported significant increases in the flooding velocity, $V_{g,crit}$ when the bottom tube-end is cut obliquely. Their data have been summarised by Butterworth³⁰ and are shown here in Table 5.2.

TABLE 5.2 Average increase in flooding velocity with diagonal cut tube-ends

Diagonal cut of tube-end, ϕ , measured from the horizontal	Tube internal diameter, mm			
	14	16	25	50
	Flooding velocity multiplier			
0° (cut square)	1.0	1.0	1.0	1.0
30°		1.05		
60°	1.5	1.3	1.2	1.0
75°		1.5		

The factors presented in Table 5.2 are based on limited data derived from single tube tests and considerable care should be exercised in any application and extrapolation.

5.5.2 Tube inclination

Hewitt¹⁴, experimenting with various air-liquid combinations, noted that it was possible to increase flooding velocities significantly by inclining tubes away from the vertical. The flooding velocity increased but then decreased as the angle of inclination to the vertical changed from 0° to 80°. The optimum angle was of the order of 20°, somewhat dependent upon tube diameter.

There are, as yet, no reports of inclined shell-and-tube bundles being installed on plant.

5.6 Recommended Correlation for Prediction of Flooding

Re-correlation of data is not practicable because an insufficient quantity is available under reflux condensing conditions. In particular, the range of tube diameters is limited.

The extent and range of the data, the magnitude of the weighted errors and deviation and comparison of predictions with such data as are available all dictate that the Alekseev *et al.* equation as modified by McQuillan and Whalley³² is the appropriate one to use. This is Equation (5.1a) or (5.1b) of Section 5.2. Additionally, a safety factor needs to be applied to the calculated flooding velocity for two reasons, as follows.

- (i) To accommodate the extent of the data scatter.
- (ii) All data are for tests on single tubes. In a multi-tube heat exchanger some maldistribution of the flow is almost certain, so for some tubes the onset of flooding may occur sooner than in other tubes. In addition to undesirable carry-over, such local flooding may lead to parallel path instabilities across the exchanger that, in turn, could upset column control.

A factor of 0.7 or less should prove adequate.

It is recommended that, wherever practicable, the tube ends are cut obliquely at about 70° to the horizontal, allowing an increased flooding velocity. Alternatively, the cut may be regarded as an increased safety factor in accordance with Table 5.2.

6. CONDENSATE FILM COEFFICIENT

Condensate film coefficients, in gravity controlled as distinct from shear controlled flows, are conveniently presented as plots of condensate film Nusselt number as a function of condensate film Reynolds and Prandtl numbers.

The condensate film coefficient may be either the local value, $\alpha_{c,x}$, or the mean value, $\alpha_{c,m}$, integrated over the total length of the tube.

The mean coefficient may be derived from the local value, provided the tube wall temperature is sensibly constant along its length, from the expression

$$\frac{Re_{f,L}}{\alpha_{c,m}} = \int_0^{Re_{f,L}} \frac{d(Re_{f,x})}{\alpha_x} \quad (6.1)$$

where $Re_{f,L}$ is the Reynolds number at the bottom of the tube and $Re_{f,x}$ is the local Reynolds number.

The available data for reflux condensation, Derivations 26, 31 and 33, are presented in Figures 2 and 3. The full lines represent 'well tried' correlations for gravity controlled co-current flow of vapour and condensate.

For $Re \leq 30$ the condensate film is assumed to be laminar,
 $30 < Re < 1600$ the film is assumed to be rippled-laminar,
 $Re \geq 1600$ the film is assumed to be turbulent.

The correlations are, respectively, those of Nusselt²², Nusselt as modified by Kutateladze²⁵ and Labuntsov²⁴.

The range of the data is summarised in Table 6.1.

TABLE 6.1 Range of data for the condensate film coefficient

Data Source →	Onda <i>et al.</i> ²⁶	Gross and Hahne ³³	Chen <i>et al.</i> ³¹
Fluid	Methanol	R13 B 1	Methanol, water
Equipment	Reflux tube	Two-phase closed thermosyphon	Two-phase closed thermosyphon
Operating Pressure	Near atmospheric	Reduced pressures from 0.5 to 0.8	Near atmospheric
Tube internal diameter, mm	21	40	14.2
Length of condensing section, m	0.31, 0.81, 1.32, 11.81	0.77	0.94
Re_f	30 – 370	580 – 9260	20 – 220
Coefficients	Mean	Mean	Local