



Falling film evaporation in vertical tubes

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FALLING FILM EVAPORATION IN VERTICAL TUBES

CONTENTS

	Page
1. NOTATION AND UNITS	1
2. INTRODUCTION	4
3. PURPOSE AND SCOPE	5
4. LEADING CHARACTERISTICS OF FALLING FILM EVAPORATORS	6
4.1 Tube Dimensions	6
4.2 Thermal Characteristics	6
5. ANNULAR FILM COEFFICIENT	6
5.1 Flow Regimes in Vertical Tube-side Falling Films	6
5.2 Film Coefficients	7
5.2.1 Average coefficient in sub-cooled zone (tube entry)	7
5.2.2 Local film coefficient in evaporation zones	8
5.2.2.1 Evaporation from a laminar film, $Re_{f,x} < 30$	8
5.2.2.2 Evaporation from a gravity controlled wavy-laminar film	8
5.2.2.3 Evaporation from a gravity controlled turbulent film	8
5.2.2.4 Combination of wavy-laminar and turbulent film expressions	9
5.2.3 Evaporation from a film with vapour shear	10
5.3 Evaporation in the Presence of a Non-condensable Gas	10
5.4 Evaluation of Film Transport Properties	11
5.5 Onset of Nucleate Boiling	11
6. DATA BANK	12
7. DESIGN DETAILS	13
7.1 Liquid Distribution in Inlet Channel	13
7.2 Minimum Tube Wetting Rate	15
8. INFORMATION AND DATA REQUIREMENTS	16
9. CALCULATION PROCEDURE	17
9.1 Calculation of Film Coefficient	17
9.2 Calculation of Tube Length	18
9.3 Check for Onset Of Nucleate Boiling In Evaporation Zone	19

10.	WORKED EXAMPLE	20
11.	REFERENCES AND DERIVATION	25
	FIGURE	27
	APPENDIX A VAPOUR SHEAR ENHANCEMENT	28
A1.	Discussion	28
	APPENDIX B EVAPORATION FROM GRAVITY-CONTROLLED TURBULENT FILMS AT HIGH PRANDTL NUMBERS	29
B1.	Discussion	29
	APPENDIX C LIQUID FILM BREAKDOWN	31
C1.	Discussion	31

FALLING FILM EVAPORATION IN VERTICAL TUBES

1. NOTATION AND UNITS

		Units	
		SI	British
A	heat transfer area of tube internal surface, πDL	m^2	ft^2
C_0, C	constants in Equations (5.2) and (5.3) respectively	–	–
c_p	specific heat capacity	J/kg K	$\text{Btu/lb } ^\circ\text{F}$
D	internal diameter of tube	m	ft
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_{fg}	specific enthalpy of vaporisation (latent heat)	J/kg	Btu/lb
K_{min}, K_T	constants in Equations (C1.1) and (C1.2)	–	–
L	length of zone	m	ft
$\dot{M}$	mass flow rate per tube	kg/s	lb/h
m, n	exponents in Equations (5.2) and (5.3) respectively	–	–
Nu_f	modified film Nusselt number = $\frac{\alpha}{\lambda_f} \left[\frac{\eta_f^2}{\rho_f^2 g} \right]^{1/3}$	–	–
Pr_f	film Prandtl number = $\frac{\eta_f c_{pf}}{\lambda_f}$	–	–
$\dot{Q}$	rate of heat transfer	W	Btu/h
$\dot{q}$	heat flux	W/m^2	Btu/h ft^2
Re_f	film Reynolds number = $\frac{4\Gamma}{\eta_f} = \frac{4\dot{M}_f}{\pi D \eta_f}$	–	–
S	solubility, see Section 8	kg/kg	lb/lb
T	temperature	$\text{K or } ^\circ\text{C}$	$^{\circ}\text{R or } ^\circ\text{F}$
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 heating medium, 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 = $\dot{M}_g/(\dot{M}_g + \dot{M}_f)$	–	–

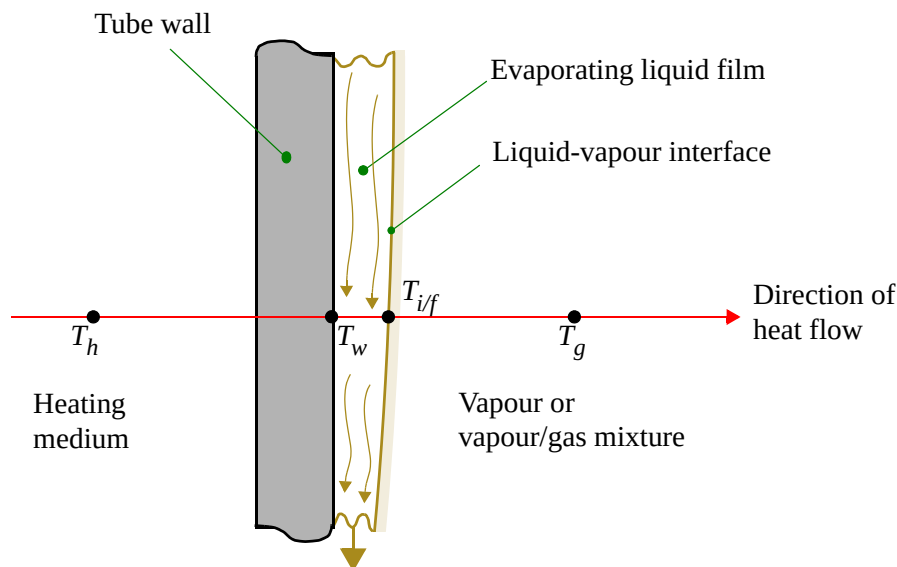
α	heat transfer coefficient	W/m ² K	Btu/ft ² h °F
Γ	liquid loading or liquid flow rate per wetted perimeter $= \dot{M}_f / \pi D$	kg/s m	lb/h ft
δ	liquid film thickness	m	ft
η	dynamic viscosity	kg/s m (Pa s)	lb/h ft
θ_{eff}	effective mean temperature difference	K	°F
Λ	dimensionless group defined in Equation (A1.4)	—	—
μ	relative error in prediction of Nusselt number defined in Equation (6.1)	—	—
λ	thermal conductivity	W/m K	Btu/h ft °F
ρ	fluid density	kg/m ³	lb/ft ³
σ	surface tension	N/m	lbf/ft
τ_i	interfacial shear stress	N/m ² (Pa)	lbf/ft ²
τ_i^*	dimensionless interfacial shear stress defined in Equation (A1.5)	—	—
χ	concentration, see Section 8	kg/kg	lb/lb

Extra Subscripts

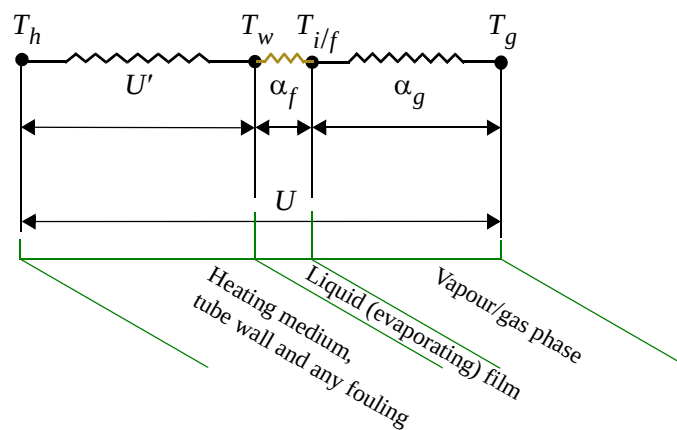
b	at boiling or saturation temperature
$evap$	in evaporating zone
exp	experimentally measured value
f	liquid film
g	gas or vapour phase
h	heating fluid
i/f	at liquid-vapour interface
in	at inlet
l	liquid
lm	logarithmic mean value
min	minimum value
$mixt$	mixture
$n-v$	non-vaporising component
onb	at onset of nucleate boiling

<i>pred</i>	predicted value
<i>sub</i>	in sub-cooled zone
<i>w</i>	refers to inside tube wall surface or any dirt film present, as appropriate
<i>x</i>	at distance <i>x</i> along tube from onset of evaporation
<i>Superscript</i>	
—	mean or average value

Notation for temperature distribution

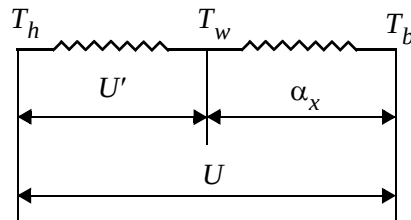


Corresponding thermal resistances and heat transfer coefficients are as follows.



A vapour/gas phase resistance is present when a two or more component mixture is being evaporated or when a liquid is being evaporated in the presence of a non-condensing gas.

When the vapour phase is an almost pure component, $T_{i/f} = T_b$, $T_g = T_b$ and the resistance diagram becomes

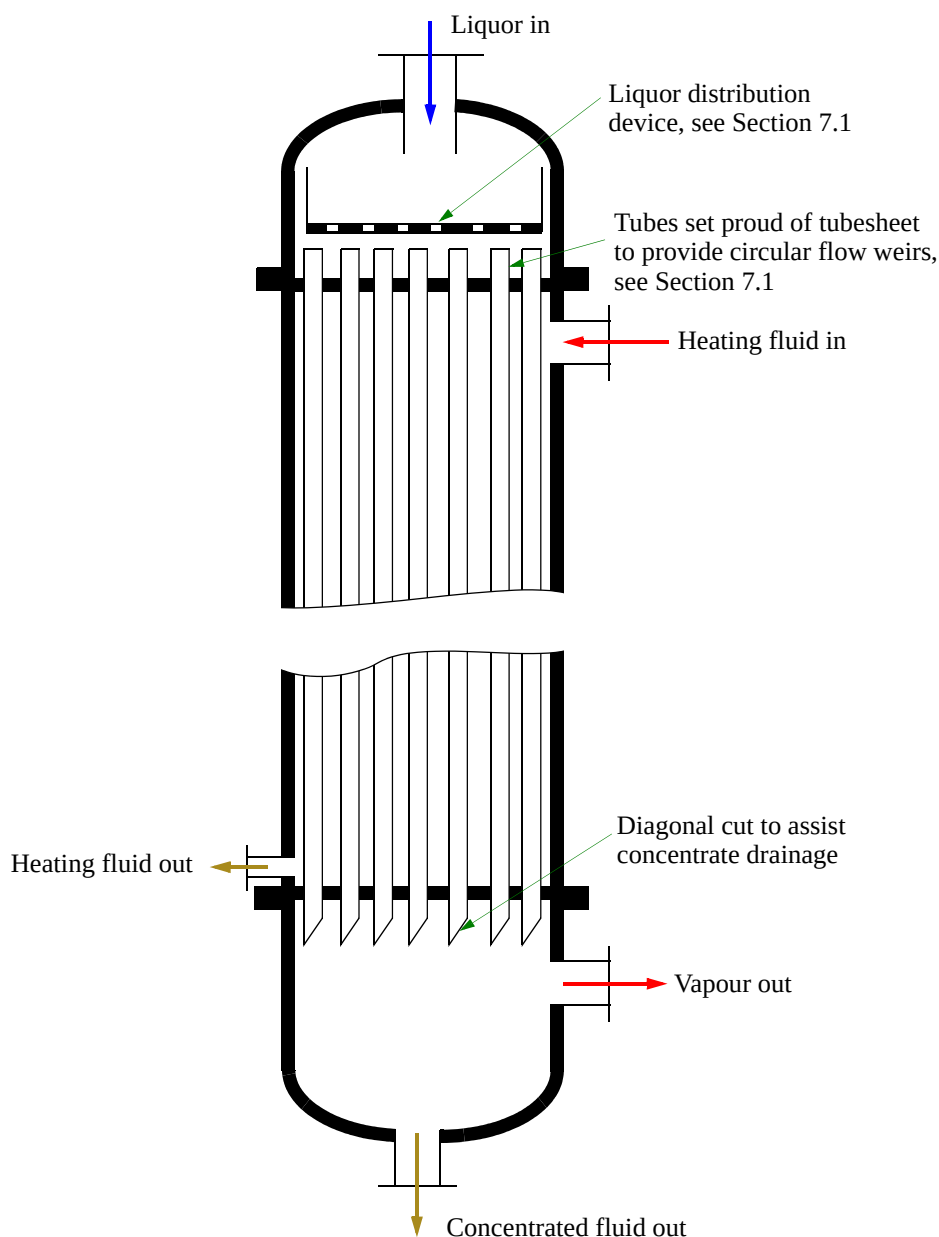


2. INTRODUCTION

The heat transfer mechanism for falling-film evaporation is one of conduction or convection through a liquid film from a heated surface to the vapour-liquid interface where evaporation takes place. Designs are such that no nucleation (nucleate boiling) occurs at the heated surface. Typically, liquor is fed to the tops of tubes arranged as a bundle and flows down the walls as a film under gravity. Vapour-liquid separation occurs usually at the bottom of the tubes, but some designs do have counter-current flow of liquid and vapour. A typical arrangement of a falling film evaporator is shown in Sketch 2.1.

Liquor may flow within or outside tubes, but structural and maintenance considerations, together with performance predictability and accepted practice, almost invariably dictate that evaporation occurs on the tube-side with the heating medium on the shell-side.

With evaporation inside tubes the liquid flows down the tubes as an annular film. Resistance to heat transfer in the case of evaporation of a single component liquid is concentrated entirely in the liquid film. There is an additional mass transfer or diffusional resistance in the liquid phase, resulting in reduced heat transfer, when evaporation from a liquid mixture takes place. One physical explanation is that preferential evaporation of the more volatile or light component(s) concentrates the less volatile or heavy component(s) at the interface, increasing the equilibrium temperature at the interface. The temperature driving force across the film, $(T_w - T_{i/f})$, and hence the effective evaporation rate is reduced. When the vapour phase is an almost pure component (such as water in a concentration process), the vapour phase mass transfer resistance will be negligible. When the vapour phase is also a mixture, a vapour phase diffusional resistance may be present and the effective evaporation rate will be reduced further.



Sketch 2.1 Sectional arrangement of falling film evaporator with co-current flow

3. PURPOSE AND SCOPE

This Data Item presents the available data and provides correlations for predicting the local film coefficient for flow inside plain vertical tubes of circular cross-section and hence the performance of falling film evaporators under steady state conditions. Data available for assessing liquid mixture effects are insufficient for developing reliable design procedures*. Consequently this aspect of falling film evaporation is not considered further in this Data Item.

* A method accommodating mass transfer effects in falling film evaporation, based on film theory, has been presented by Palen *et al.*¹³ The paper references most relevant work undertaken since 1980. It should be consulted for guidance where diffusional resistance in the liquid film is deemed to be significant.

4. LEADING CHARACTERISTICS OF FALLING FILM EVAPORATORS

4.1 Tube Dimensions

Length typically 6 m to 11 m (20 ft to 36 ft) but may be as short as 1.5 m to 3 m (5 ft to 10 ft), particularly in deep vacuum applications. Diameters are typically 25 mm to 64 mm (1 in to 2½ in).

4.2 Thermal Characteristics

- (i) Liquid retention times are short, implying liquid hold-up and overall inventory are low.
- (ii) Suppression of nucleate boiling implies low liquid film temperature differences, $(T_w - T_{i/f})$. As a rule the overall temperature difference $(T_h - T_g)$ will be less than 15 K (27°F) and frequently less than 10 K (18°F).
- (iii) Film heat transfer coefficients are high.
- (iv) Pressure drop through the tubes is often negligible.
- (v) Absence of nucleate boiling and low temperature differences reduce possible fouling tendencies.
- (vi) The flow pattern is that of a liquid annulus with a vapour core. Because a two-phase mixture is absent in the vapour core there will be no hydrostatic head effect*. Absence of an imposed hydrostatic head allows a falling film evaporator to be used at absolute pressures approaching the hydrostatic head of the film thickness.

The combination of short liquid retention times and the ability to operate at low overall temperature differences makes falling film evaporation particularly suitable for concentrating most heat sensitive liquids, whilst the absence of any significant hydrostatic head effect can make falling film evaporators a superior alternative to thermosyphon reboilers in vacuum service (also in terms of operational stability).

5. ANNULAR FILM COEFFICIENT

5.1 Flow Regimes in Vertical Tube-side Falling Films

Flow within an annular film when evaporation is taking place at the free surface may be either laminar or turbulent, depending upon the value of the liquid film Reynolds number, $Re_f = 4\Gamma/\eta_f$. At the lowest Reynolds numbers ($Re_f < 30$) the flow will be fully laminar. At higher values of Reynolds number the flow becomes firstly rippled or wavy whilst remaining essentially laminar. At even higher Reynolds numbers the flow becomes turbulent.

In laminar films, the heat transfer mechanism is one of conduction through the film so, as evaporation proceeds and the film becomes thinner, the film coefficient may be expected to increase. However, this effect may be partially or even wholly offset by the increase in viscosity as the liquor becomes progressively more concentrated. (In laminar flow the film thickness is proportional to the cube root of the viscosity.)

With turbulent films the mechanism, outside the laminar sub-layer, is one of convection and the film

* The hydrostatic head effect may best be understood by reference to a liquid to be evaporated and introduced at the bottom of a vertical tube. The boiling temperature will be higher at the bottom of the tube than towards the top on account of the static pressure associated with the head of liquid or two-phase mixture above. The mean overall temperature difference, which is the driving force for heat transfer, will be reduced. This effect is particularly significant where overall temperature differences are small.

coefficient will decrease as evaporation proceeds and the film becomes thinner.

If sufficient vapour is generated and the vapour velocity down the tubes is high enough, gravity flow will be enhanced by vapour shear at the vapour-liquid interface.

The mechanisms of heat transfer through an annular film are similar, whether the process is one of condensation of a vapour or evaporation of a liquid in the absence of nucleation at the heated surface. Correlations for the prediction of local liquid film coefficients may therefore be expected to be of similar form in both cases. This has been assumed in much of the literature. Here, available data **specific to evaporation** are presented.

Transitions between different flow regimes are not abrupt; they occur smoothly over ranges of the controlling parameters and a finite length of the tube.

ESDU 91024¹² may be consulted for further information on the hydromechanics of annular films in vertical tubes.

5.2 Film Coefficients

If a sub-cooled liquid zone exists the average or mean coefficient in that zone may be estimated from the expressions presented in Section 5.2.1. In the evaporative zone variations in the liquid physical properties as evaporation proceeds are such that local or point values of the film coefficient should be used (see Section 5.2.2). The total heat exchanger area will be

$$A = \sum_0^{\dot{Q}} \frac{\Delta \dot{Q}}{U(T_h - T_g)}. \quad (5.1)$$

5.2.1 Average coefficient in sub-cooled zone (tube entry)

At tube entry the liquid may be sub-cooled. In these circumstances all the heat transferred from the tube wall goes to raise the bulk temperature of the liquid to its saturation or equilibrium temperature. No heat is transmitted across the free surface.

The appropriate expression depends on whether flow within the film is laminar, transitional or turbulent as defined by the value of the film Reynolds number, $4\Gamma/\eta_f$. The expressions are those of Wilke¹, derived from data over the range $5 \geq Pr_f \leq 210$, and as presented by Gani⁷ and Seban⁵.

$$\frac{\bar{\alpha} \delta}{\lambda_f} = C_0 \left(\frac{4\Gamma}{\eta_f} \right)^m Pr_f^{0.344}, \quad (5.2)$$

where δ is the film thickness given by

$$\frac{\rho_f \delta (g \delta)^{0.5}}{\eta_f} = C \left(\frac{4\Gamma}{\eta_f} \right)^n. \quad (5.3)$$

TABLE 5.1 Constants for Equations (5.2) and (5.3)

Flow	Laminar $4\Gamma/\eta_f \leq 1600$	Transitional $1600 < 4\Gamma/\eta_f \leq 3200$	Turbulent $4\Gamma/\eta_f > 3200$
C_0	0.029	0.212×10^{-3}	0.181×10^{-2}
m	0.533	1.2	0.933
C	0.866	0.073^*	0.137
n	0.5	0.84^*	0.75

* Two sets of values for C and n (0.095, 0.80) and (0.051, 0.87) giving values of $\rho_f \delta (g\delta)^{0.5} / \eta_f$ within 10% of one another over a wide range of film Reynolds numbers are presented by Seban⁵. The values in Table 5.1 are means of these values.

5.2.2 Local film coefficient in evaporation zones

For evaporation at the free surface to take place the whole of the heat taken up from the tube wall is transmitted across the film, implying a temperature gradient. Therefore, before evaporation can take place, all the heat transferred from the tube wall must go into superheating the film to the extent that the liquid at the liquid-vapour interface is at its equilibrium (saturation) temperature. Data are few for heat transfer in this thermally-developing region and no substantiated calculation methods are available. However, this zone is usually short compared with the overall lengths of tube required in most falling film evaporator applications and may be accommodated by an additional length of tubing.

5.2.2.1 Evaporation from a laminar film, $Re_{f,x} < 30$

The Nusselt solution for heat transfer through a laminar gravity controlled film is

$$Nu_x = \frac{\alpha_x}{\lambda_f} \left[\frac{\eta_f^2}{\rho_f^2 g} \right]^{1/3} = 1.10 Re_{f,x}^{-1/4}. \quad (5.4)$$

5.2.2.2 Evaporation from a gravity controlled wavy-laminar film

$$Nu_x = 0.82 Re_{f,x}^{-0.22}. \quad (5.5)$$

This is Equation (5.4) multiplied by an enhancement factor suggested by Chun and Seban⁴.

5.2.2.3 Evaporation from a gravity controlled turbulent film *

$$Nu_x = 0.0097 Re_{f,x}^{0.29} Pr_f^{0.63}. \quad (5.6)$$

* The correlation of Chun and Seban⁴, $Nu_x = 0.0038 Re_{f,x}^{0.4} Pr_f^{0.65}$, is frequently cited. Åsblad and Berntsson¹⁷ point out the experimental procedure resulted in a systematic variation of maximum Reynolds number with evaporation temperature and thus with Prandtl number, and questioned the suitability of extrapolation beyond the following experimental limits.

Prandtl number	5.7	2.9	1.8
Maximum Reynolds number	8000	14 500	21 000