



ESDU 85041

Issued December 1985

With Amendment A

January 1988

Boiling inside tubes: general aspects and saturated wet-wall heat transfer

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BOILING INSIDE TUBES: GENERAL ASPECTS AND SATURATED WET-WALL HEAT TRANSFER

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BOILING INSIDE TUBES: general aspects and saturated wet-wall heat transfer

1. NOTATION AND UNITS

		Units	
		SI	British
A	internal surface area	m^2	ft^2
Bo	boiling number, defined in Equation (6.14)	–	–
C_{dr}	constant dependent on liquid/vapour density ratio, defined in Equation (6.8)	–	–
C_{pr}	constant dependent on reduced pressure defined in Equation (6.7)	–	–
c_p	specific heat capacity of fluid at constant pressure	J/kg K	$\text{Btu/lb } ^\circ\text{F}$
D	internal diameter of tube	m	ft
E	enhancement of coefficient due to boiling	–	–
g_n	gravitational acceleration	m/s^2 (9.81)	ft/h^2 (4.17×10^8)
h	enthalpy	J/kg	Btu/lb
J	mechanical equivalent of heat	– (1.0)	ft lbf/Btu (778)
$\tilde{M}$	relative molecular mass (molecular weight)	kg/kmole	lb/lbmole
$\dot{M}$	mass flow rate	kg/s	lb/h
$\dot{m}$	mass flux, from Equation (6.1)	kg/s m^2	lb/h ft^2
Pr	Prandtl number, from Equation (6.3)	–	–
p_c	critical pressure	N/m^2 (Pa)	lbf/ft^2
p_r	reduced pressure = p_s/p_c	–	–
p_s	saturation pressure	N/m^2 (Pa)	lbf/ft^2
$\dot{Q}$	rate of heat transfer	W	Btu/h
$\dot{q}$	heat flux = $\dot{Q}/A$	W/m^2	Btu/h ft^2

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r	radius of a nucleation site	m	ft
Re_{l0}	Reynolds number, at zero quality, from Equation (6.2)	–	–
T	temperature	K	°F
U	overall heat transfer coefficient, referred to internal surface area	W/m ² K	Btu/h ft ² °F
U'	combined heat transfer coefficient of heating fluid, wall and any fouling, referred to internal surface area	W/m ² K	Btu/h ft ² °F
x_g	quality (mass flow rate of vapour/total mass flow rate)	–	–
x_h	fraction of heat transfer rate, from Equation (7.1)	–	–
z	distance from tube inlet	m	ft
α_b	heat transfer coefficient for boiling liquid	W/m ² K	Btu/h ft ² °F
α_{Cb}	convective component of boiling coefficient	W/m ² K	Btu/h ft ² °F
α_g	heat transfer coefficient in superheating	W/m ² K	Btu/h ft ² °F
α_{l0}	heat transfer coefficient in preheating	W/m ² K	Btu/h ft ² °F
α_{nb}	nucleate component of boiling coefficient	W/m ² K	Btu/h ft ² °F
α_C	heat transfer coefficient of fluid to be vaporised	W/m ² K	Btu/h ft ² °F
Δh_v	latent heat of vaporisation	J/kg	Btu/lb
ΔT	overall temperature difference ($T_H - T_C$)	K	°F
η	dynamic viscosity of fluid	kg/s m	lb/h ft
θ	temperature difference between inside of wall and bulk of liquid to be vaporised ($T_w - T_C$)	K	°F
λ	thermal conductivity of fluid	W/m K	Btu/h ft °F
ρ	density of liquid	kg/m ³	lb/ft ³
σ	surface tension	N/m	lbf/ft

Extra subscripts

C	liquid to be vaporised
g	vapour
H	heating fluid

<i>in</i>	at inlet to tubes
<i>l</i>	liquid
<i>nb</i>	nucleate boiling
<i>ONB</i>	onset of nucleate boiling
<i>out</i>	at outlet from tubes
<i>s</i>	saturation
<i>w</i>	inside surface of wall (and any dirt)
<i>0</i>	at zero quality

2. PURPOSE AND SCOPE OF THIS DATA ITEM

Vaporisers are widely used in the process industries, operating over a wide range of pressures with many different fluids. They may be divided into two main categories.

- (1) Steam generators, to provide steam for heating or to drive turbines – either a fired boiler or a waste heat boiler.
- (2) Vaporisers for process fluids.

A short account of the applications of vaporisers in the process industries is given in Section 3. The large water-tube boilers in power stations are outside the scope of this Data Item.

A glossary of the specialised terms used in this Data Item is given in Section 9.

The objectives of this Data Item are as follows.

- (1) To describe the problems that may arise in the thermal rating of a vaporiser (Section 4).
- (2) To evaluate the correlations that have been published on wet-wall saturated boiling of pure liquids in straight tubes (Section 5).
- (3) To describe how such correlations may be applied to the thermal rating of a vaporiser.

In the first objective, a wide range of applications is covered. However the second and third objectives are limited by the range of the available experimental data. No quantitative information is given on the boiling of mixtures, on fouling, or on the criteria for the onset of dry-wall conditions.

This Item may be used over the complete range of saturated wet-wall conditions.

It is assumed that the liquid is in turbulent flow at zero quality. The procedure may be applied to flow up vertical tubes and to flow ($Re_{l0} > 4000$) along horizontal tubes at a velocity sufficient to avoid stratified flow. There is no consideration of the devices that may be inserted into tubes to enhance heat transfer or of the special surfaces that have been developed to enhance boiling.

It is important to appreciate that the problems in designing a steam generator differ considerably from those in designing a vaporiser for an organic liquid. In the former the rate of heat transfer is controlled by the thermal resistance of the gas that is heating the surface, provided that the wall is fully wetted by the boiling water. Hence there is no need to estimate the heat transfer coefficient of the boiling water accurately, but the important problem is the avoidance of dryout. By contrast the boiling coefficient is important in vaporising an organic liquid, using condensing steam or a hot liquid as the source of heat.

3. APPLICATIONS OF VAPORISERS IN THE PROCESS INDUSTRIES

The vaporisers used in the process industries can be classified as follows: -

- (1) **Steam generators** producing steam for (i) heating process fluids, (ii) space heating, (iii) driving a turbine connected to an electric generator, (iv) driving a turbine connected to a pump or compressor, or (v) use as a process fluid.

These steam generators are subdivided according to the source of heat into

- (A) fired boilers, where the source of heat is the hot flue gases produced in a furnace by the combustion of a fossil-fuel;
 - (B) waste heat boilers, where the source of heat is either the heat generated by an exothermic reaction carried out at a sufficiently high temperature, or the waste heat available in the flue gases from a high-temperature furnace.
- (2) **Reboilers** providing a source of heat at the base of a distillation column to vaporise some of the liquid that collects in the base of the column and return it as vapour or vapour-liquid mixture.
 - (3) **Evaporators** generating vapour for one of the following reasons:
 - (i) to revaporise a gas that has been liquefied for storage or for transportation,
 - (ii) to produce a pure vapour from a liquid that contains unwanted dissolved solids, *eg* desalination,
 - (iii) to concentrate a liquid containing dissolved solids,
 - (iv) to generate crystals in a liquid concentrated to saturation.
 - (4) **Chillers** in a refrigeration system cooling a process fluid to a temperature below ambient (by boiling the refrigerant).
 - (5) **Evaporators for heat pumps** extracting heat from the source.
 - (6) **Evaporators for heat transfer fluids** vaporising the high-temperature heat transfer fluid used to convey heat to a separate piece of equipment.

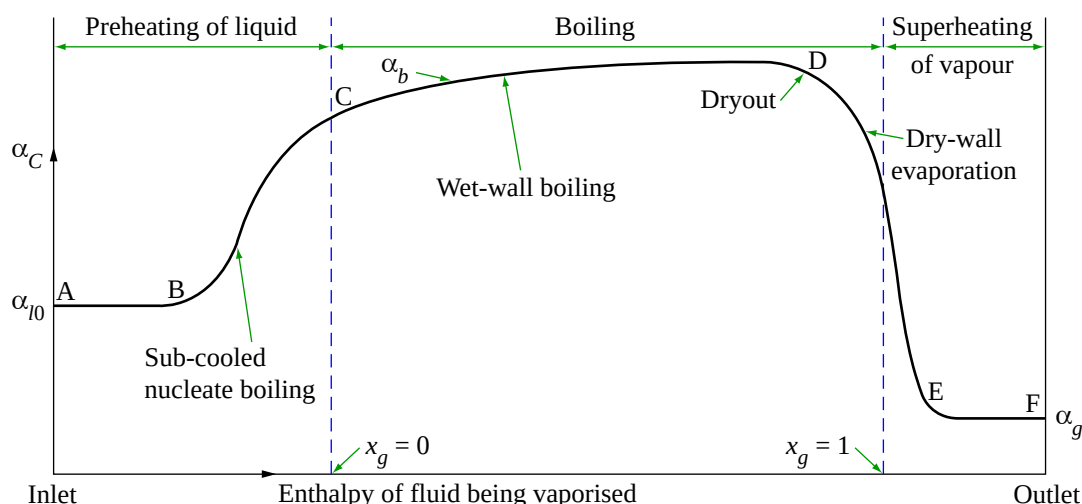
Engineers in the process industries may need to carry out thermal rating calculations for a vaporiser under one of the following circumstances:—

- (i) to choose between several competitive tenders for a new vaporiser and to check that the one chosen will be satisfactory in all respects for the duty specified,
- (ii) to design a new vaporiser,
- (iii) to rectify any faults that arise in the operation of a vaporiser,
- (iv) to predict how a vaporiser will perform under changed process conditions.

4. PROBLEMS IN THE THERMAL RATING OF A VAPORISER

When heating or cooling a fluid without phase change, the heat transfer coefficient does not usually change significantly from inlet to outlet of the heat exchanger.

With vaporisation there are very considerable changes. In a process vaporiser where the temperature difference between the hot and cold fluids is in the range of perhaps 5 to 50°C, the variation of the heat transfer coefficient for the fluid to be vaporised (α_c) is typically of the form shown in Sketch 4.1, where it is plotted against enthalpy for the full range of preheating, boiling and superheating.



Sketch 4.1 Changes in heat transfer coefficient in a complete vaporiser

From A to B there is single-phase heating of the liquid, so the heat transfer coefficient for the liquid (at zero quality, α_{l0}) can be calculated from the appropriate standard correlation. At C, the quality, $x_g = 0$.

The point B is known as “the onset of nucleate boiling”. This occurs when the wall is superheated a few degrees, sufficient for bubbles to begin to form at very small crevices or discontinuities in the surface, as described in Section 4.3. At the point C, the bulk of the liquid has been heated to its saturation temperature. Thus from B to C there is sub-cooled nucleate boiling and the heat transfer coefficient is enhanced. If the temperature difference is small, there is no subcooled nucleate boiling.

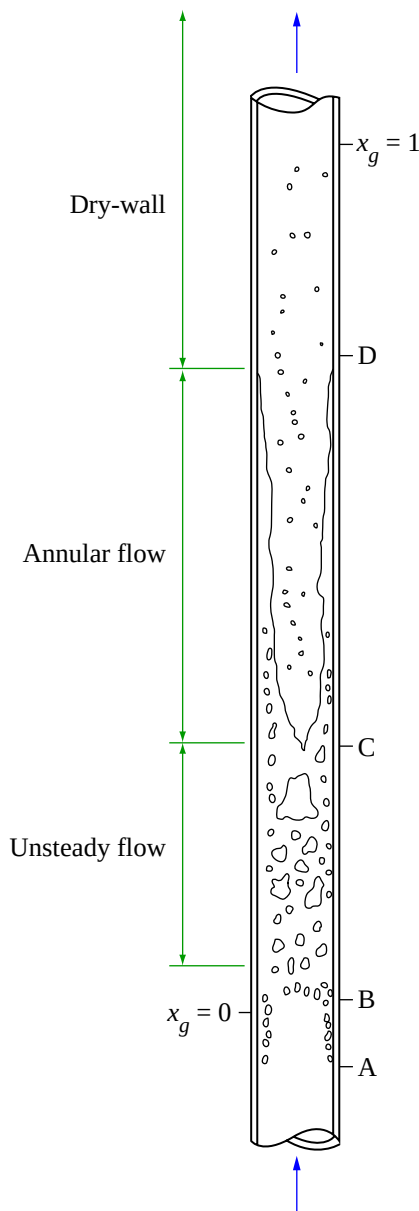
From C to D there is wet-wall boiling, at the saturation temperature, and the considerable enhancement in the heat transfer coefficient is discussed in Section 4.4. If there is no subcooled nucleate boiling, a liquid may become superheated before boiling begins. Most industrial liquids contain sufficient nuclei, in the form of suspended solids and dissolved gases, to prevent excessive superheating. However there is a possibility that bulk superheating may occur; this is dangerous in that it leads to very unsteady flow because of the intermittent production of vapour (see Section 4.8.2). It can be avoided if it is possible to ensure that the heat flux at zero quality is sufficient to give subcooled nucleate boiling, as described in Section 4.3. An extreme case of bulk superheating of liquid is sometimes seen in the vaporisation of liquid alkali metals.

At D dryout occurs, due to incomplete wetting of the heated surface. This is discussed in Section 4.4. The region D to E is the “dry-wall” region (see Section 4.6). At E the heat transfer coefficient has fallen to α_g , the value for single-phase superheating of the vapour, calculated from the appropriate standard correlation (as used for liquid heating). The fact that E is slightly beyond the position where the equilibrium quality, $x_g = 1$, indicates that superheating of the vapour begins before all the liquid has evaporated, so that heat is being transferred from the wall to the dry vapour near to the wall and thence to the drops flowing in the core of the vapour.

In a gas-heated steam generator, the increase in α_C at the point B in Sketch 4.1 is steeper, and the fall at point D is extremely steep, to less than the value of α_g .

4.1 Importance of Flow Patterns in a Vaporiser

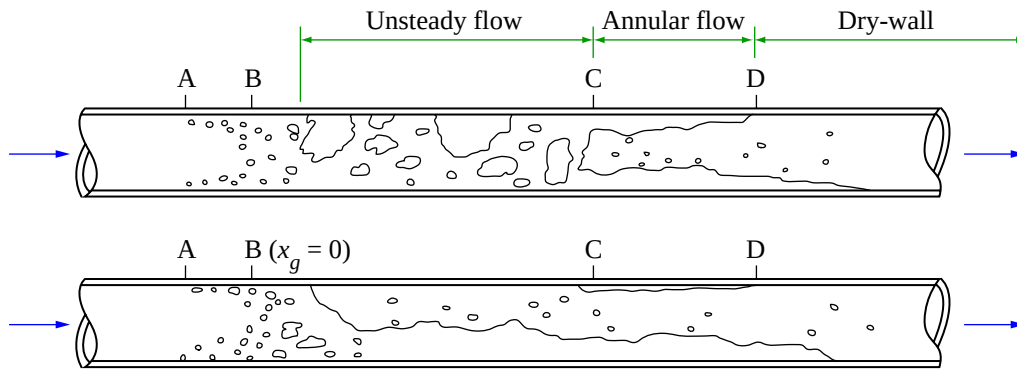
To understand the problems in the design of a vaporiser it is necessary to realise that the pattern of two-phase flow changes considerably as the quality changes from zero to unity. A typical flow pattern for a vertical tube is illustrated in Sketch 4.2. Gravity influences boiling in a horizontal tube, as shown in Sketch 4.3: the lower sketch is for a low mass flux (only slightly greater than the minimum defined at the end of Section 4.4) while the upper sketch is for a mass flux several times greater.



The liquid is initially sub-cooled and heated in single-phase flow. Nucleation begins at point A, bulk boiling at B. At very low qualities the vapour is present as bubbles, known as “bubble flow”. In horizontal tubes there is a bigger concentration of bubbles nearer to the top of the tube. When the volume rate of flow bubbles is about a third of the total volume rate of flow, the bubbles coalesce, and flow becomes unsteady. In vertical tubes, “churn flow” occurs. At low mass flux in a horizontal tube, “separated flow” occurs, the liquid flowing along the bottom of the tube and the vapour along the top, with waves forming at the interface. At high mass flux in a horizontal tube there is a region of “slug flow”. When the volume rate of flow of vapour considerably exceeds that of the liquid, the flow becomes “annular”, as at C in the sketches. Most of the liquid flows along the wall in the form of an annulus, whilst the vapour plus some of the liquid, in the form of entrained drops, flows at considerably higher velocity in the centre of the tube. In a horizontal tube the annulus is thinner at the top than at the bottom of the tube.

At the point marked D in the sketches the wall becomes dry, because the supply of liquid to the wall by impingement of drops is less than the rate of evaporation. The quality at which the tube becomes dry depends on the mass flux, the heat flux, the relative densities of liquid and vapour, and the diameter of the tube, as discussed in Section 4.5, which also deals with the consequences. With a vertical tube, the transition to dry-wall conditions is abrupt: with a horizontal tube, first the top of the tube is wetted intermittently; then a dry patch forms on the top of the tube; finally the dry area spreads to the bottom of the tube.

Sketch 4.2 Flow pattern in boiling flow up a vertical tube

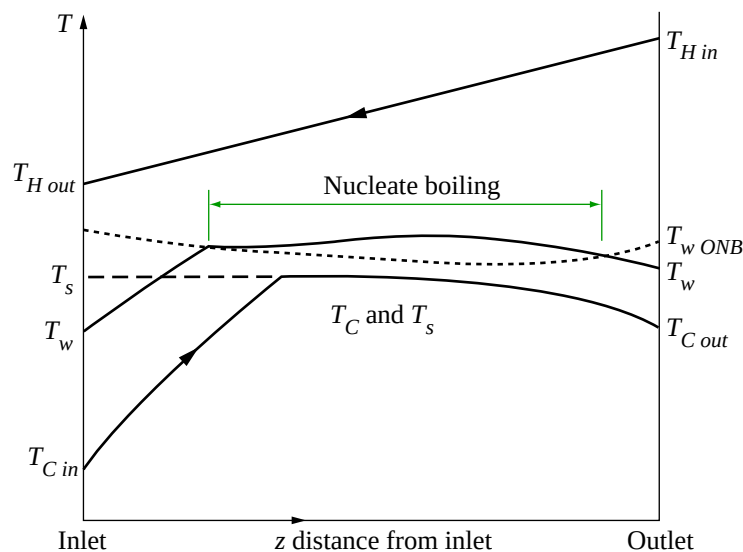


Sketch 4.3 Flow patterns in boiling flow along a horizontal tube

4.2 Estimation of Local Heat Flux

This Data Item is concerned with the estimation of heat flux at various locations along a vaporiser with wet-wall vaporisation inside tubes. It is assumed that the liquid to be vaporised enters the tubes at a temperature slightly below the saturation temperature and leaves at a quality that is sufficiently low to avoid dryout.

Sketch 4.4 illustrates the relationship between the various temperatures plotted against z , the distance from inlet. The dashed line is the saturation temperature of the liquid; this falls towards the outlet, owing to pressure drop, which may be high in the boiling region (see Section 4.9). The upper line is the temperature of the heating fluid. This is drawn for counter-current flow of the two streams. Occasionally co-current flow is used. Often heat is supplied by a condensing stream, so T_H is approximately constant. Nucleate boiling takes place over the length marked "nucleate boiling", showing that, in this case, nucleate boiling is suppressed at high quality. It is required to estimate the heat flux at various locations defined by the enthalpy of the boiling liquid. It is then possible to determine the length of tube needed to transfer sufficient heat to take the boiling liquid from one location to the next. The range in Sketch 4.4 corresponds to the range of enthalpy from point A in Sketch 4.1 to a point between C and D.



Sketch 4.4 Temperature/length diagram for a wet-wall vaporiser with a nucleate boiling zone in a horizontal tube

The local heat flux is given by

$$\begin{aligned}\dot{q} &= \alpha_C(T_w - T_C) \\ &= U'(T_H - T_w) \\ &= U(T_H - T_C)\end{aligned}\quad (4.1)$$

where T_H is the temperature of the heating fluid, T_w that of the wall and T_C that of the liquid being vaporised. U' is the combined heat transfer coefficient of the heating fluid, the wall and any fouling. U is the overall heat transfer coefficient, given by

$$U = [(1/U') + (1/\alpha_C)]^{-1}. \quad (4.2)$$

The estimation of the extent of the nucleate boiling zone is dealt with in the next section. Section 4.4 deals with the estimation of the boiling heat transfer coefficient and Section 4.5 discusses dryout.

4.3 The Conditions for Nucleate Boiling

Although this Data Item is not concerned with the estimation of the enhancement in heat transfer coefficient due to nucleation in the preheating regions, it is advisable to determine whether there will be any nucleation in this zone, because it may considerably enhance the boiling coefficient at low quality, and because nucleate boiling eliminates the danger of gross superheating of the liquid and the consequent very erratic flow, known as “bumping” or “chugging”. To avoid this danger, the onset of nucleate boiling should occur before the bulk temperature of the liquid reaches saturation temperature. The temperature of the wall must exceed the saturation temperature by an amount that depends on the flux ($\dot{q}$), the physical properties of the liquid and its vapour, and the nature of the surface. Accounts of how nucleate boiling begins have appeared in many textbooks, eg Chapters 4 and 5 of Reference 42.

The correlations recommended for the estimation of the temperature of the wall at the onset of nucleate boiling have been discussed by Davis and Anderson¹³. One that is frequently used is

$$T_{w\text{ ONB}} = T_s + \left(\frac{8 \sigma \dot{q} T_s}{\lambda_l J \Delta h_v \rho_g} \right)^{1/2}, \quad (4.3)$$

where J is the mechanical equivalent of heat, Δh_v is the latent heat of vaporisation, λ_l is the thermal conductivity of the liquid, ρ_g is the density of the vapour and σ is the inter-facial surface tension between the liquid and vapour.

Equation (4.3) is likely to underpredict $T_{w\text{ ONB}}$ because it is based on the assumption that the surface in contact with the liquid to be vaporised contains a large size range of cavities that will act as sites for nucleation. In practice the largest of these sites tend to be flooded with liquid, so that they are not effective, especially if the liquid has good wetting properties. The smaller sites require a greater superheat before they become active. Davis and Anderson¹³ show that if it is assumed that sites of radius r will initiate nucleation, the required wall temperature is given by an equation which has been rearranged in the form

$$T_{w\text{ ONB}} = T_s + \frac{2 \sigma T_s \left(\frac{1}{\rho_g} - \frac{1}{\rho_l} \right)}{J \Delta h_v r \left(1 - \frac{r \alpha_{l0}}{\lambda_l} \right)}. \quad (4.4)$$