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Condensation inside tubes: pressure drop in straight horizontal tubes

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CONDENSATION INSIDE TUBES: pressure drop in straight horizontal tubes

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CONDENSATION INSIDE TUBES: pressure drop in straight horizontal tubes

1. NOTATION AND UNITS

		Units	
		SI	British
A	defined by Equation (4.14)	–	–
A', B	defined by Equations (4.10) and (4.11)	–	–
C	correlation constant in Equation (3.1)	–	–
Co	mean value of condensation number, from Equation (4.5)	–	–
CFR	correction factor for roughness, from Equation (3.1)	–	–
D	internal diameter of tube	m	ft
E	pressure drop correction factor (see Section 4.6)	–	–
Fr	Froude number, from Equation (4.6)	–	–
f	Fanning friction factor, from Equations (4.9) to (4.11)	–	–
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)
h	specific enthalpy	J/kg	Btu/lb
L	total cooled length of tube	m	ft
L_1	cooled length of tube before condensing begins	m	ft
L_2	cooled length of tube over which condensing occurs	m	ft
L_d	downstream uncooled length of tube	m	ft
L_u	upstream uncooled length of tube	m	ft
$\dot{M}$	total mass flow rate inside tubes	kg/s	lb/h
$\dot{m}$	total mass flux, see Equation (4.1)	kg/s m ²	lb/h ft ²
N_T	number of tubes in parallel	–	–
p	static absolute pressure	N/m ² (Pa)	lbf/ft ²
$\dot{Q}$	rate of heat transfer	W	Btu/h
$\dot{q}$	mean heat flux, see Equation (4.2)	W/m ²	Btu/h ft ²
R	enhancement in two-phase pressure drop, see Equation (4.15)	–	–

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Re_g	Reynolds number of vapour at quality of unity, from Equation (4.4)	—	—
Re_l	Reynolds number of liquid at quality of zero, from Equation (4.3)	—	—
s	standard deviation (see Section 5.4)	—	—
T	temperature	K	°F
We	Weber number, from Equation (4.7)	—	—
x_g	quality (dryness fraction), see Equation (4.8)	—	—
z	distance along the tube from inlet	m	ft
Δp	pressure drop	N/m ²	lbf/ft ²
Δh_v	latent heat of vaporisation	J/kg	Btu/lb
ε	surface roughness	m	ft
η	dynamic viscosity of fluid	kg/s m	lb/ft h
μ	errors in prediction of pressure drop in condensing flow using correlations for adiabatic flow, see Equation (5.1)	—	—
μ'	error in prediction of pressure drop in condensing flow after applying the correction factor for condensing flow defined in Section 4.6	—	—
ρ	density of fluid	kg/m ³	lb/ft ³
σ	surface tension	N/m	lbf/ft

Extra subscripts

<i>expt</i>	experimentally measured value
<i>fr</i>	frictional component
<i>g</i>	vapour
<i>in</i>	at inlet
<i>l</i>	liquid
<i>m</i>	mean
<i>out</i>	at outlet
<i>pred</i>	predicted value
<i>sat</i>	under saturation conditions
<i>T</i>	total
<i>t</i>	in tube
1	cooling single-phase vapour
2	cooling two-phase mixture

2. PURPOSE AND SCOPE OF THIS ITEM

This Data Item is one of a group on the condensation of vapours inside straight tubes of circular cross-section. The first Data Item, ESDU 89038³, deals with reflux condensation in vertical tubes, where the vapour flows up the tubes and the condensate drains down the tubes. The second Data Item, ESDU 90024⁴, gives a procedure for estimating pressure drop with vertical downflow. This, the third Item, deals similarly with flow along horizontal tubes. ESDU 91024⁵ deals with the estimation of the heat transfer coefficient of the film of condensate in the downflow condensation of a single-component vapour.

Whilst the scope of this Data Item is the pressure drop inside the tubes of a condenser, in practice the dominant pressure drop is often that at inlet to the tubes; this is dealt with in Section 6.1.

The prediction of pressure drop in a condenser is required for the design of downstream plant. It is also required so that the saturation temperature can be determined at the outlet and in some cases at several points along the tube. An excessive pressure drop may lead to a considerable reduction in temperature, resulting in a “temperature pinch” and fall in heat flux at some point along the condenser.

A data bank on pressure drop in condensing flow through straight horizontal tubes has been collected from the literature by staff at Harwell Laboratory for ESDU, as described in Section 5, and is used to develop the method of prediction presented here. The procedure is the same as that described in Section 2 of ESDU 90024⁴ for vertical downflow.

The total pressure drop can be estimated by following the numbered steps in Section 4. This is illustrated by the example in Section 7.

Section 5 discusses the data bank and compares the measured and predicted values of the pressure drop. An assessment of the accuracy of the predictions is made.

Section 6 discusses the prediction of the overall pressure drop in an industrial horizontal tubeside condenser.

3. PUBLISHED CORRELATIONS FOR PRESSURE DROP IN ADIABATIC TWO-PHASE FLOW AND THEIR APPLICATION TO CONDENSING FLOW

From a balance of force and momentum it can be shown that the pressure gradient in two-phase flow is the sum of three components, as follows:

- (1) frictional component, see Section 3.1 of ESDU 90024⁴,
- (2) gravitational component, not relevant with horizontal flow,
- (3) momentum-change component (sometimes called “accelerational”) see Section 3.3 of ESDU 90024⁴.

The total pressure drop over a straight length of tube is determined by integrating the sum of these components of the pressure gradient with respect to the distance along the tube.

During condensation the values of the components change considerably with decreasing quality (defined in Section 4.2). It was decided to analyse the data for two-phase pressure drop with condensing flow by comparing the measured values with values predicted by using the best available correlations for adiabatic flow. Some improvements have been made to the correlation for the frictional component, as described below. From the analysis it has been possible to provide a system of correction factors that can be applied to the total pressure drop predicted as described now for adiabatic flow. The methods used and the reasons for choosing them have been described in Sections 3.1 and 3.3 of ESDU 90024⁴.

The evidence on the effects of **roughness** on two-phase pressure drop is sparse, but it has been shown that an increase in the roughness of the surface of the tube gives a smaller contribution to the two-phase than to the single-phase pressure drop. This is because, in two-phase flow, the roughness of the vapour-liquid interface tends to have a greater influence than the roughness of the wall (except for extremely rough tubes). Chisholm¹ has suggested that this can be allowed for by applying a correction factor, *CFR*, to the two-phase term in the equation used for estimating the frictional component of the pressure drop.

This is given by

$$CFR = \frac{1 + 10^{-C\varepsilon/D}}{2} \quad (3.1)$$

where *C* is a correlating constant, ε is the surface roughness and *D* the internal diameter of the tube. Thus for a smooth pipe, $\varepsilon = 0$ and *CFR* = 1.0. As the roughness increases, *CFR* tends to 0.5.

The policy adopted in the Data Item on pressure drop in boiling flow²⁰ was to use the simplified version of Friedel’s correlation² for the frictional component and to multiply the second term, which is zero for single-phase flow, by *CFR* obtained from Equation (3.1). It was found that the closest agreement between experimental and predicted pressure drop was obtained with *C* = 220. In the absence of any data on the effect of roughness on pressure drop in condensing flow, the same value of *C* is assumed in this Data Item and is incorporated in the last term in Equation (4.15).

4. PROCEDURE FOR PREDICTING PRESSURE DROP IN CONDENSING FLOW

This section describes how it is possible to predict the pressure drop through a cooled horizontal tube with condensing flow using the correlations recommended in the previous section. Section 4.6 shows how the accuracy of the calculation can be improved by applying a correction factor based on the analysis of the data bank described in Section 5.

4.1 Data Required and Preliminary Calculations

The following information is required, using a self-consistent system of units, as in Section 1:

specific enthalpy at inlet and outlet, h_{in} and h_{out} , or quality, see Equation (4.8), $x_{g,in}$ and $x_{g,out}$,

internal diameter, D , surface roughness, ε , and number of tubes in parallel, N_T , (some values of ε can be found in Derivations 7 and 14),

upstream uncooled length, L_u , cooled length, L , and downstream uncooled length, L_d ,

mass flow rate of condensing fluid, $\dot{M}$,

rate of heat transfer, $\dot{Q}$,

pressure and temperature at inlet, p_{in} and T_{in} , or at outlet p_{out} and T_{out} ,

physical properties of the condensing fluid:

latent heat of vaporisation, Δh_v ,

density of liquid and vapour, ρ_l and ρ_g ,

dynamic viscosity of liquid and vapour, η_l and η_g ,

surface tension of liquid against its vapour, σ ,

relationship between saturation temperature, T_{sat} , and pressure, p ,

relationship between ρ_g and T_{sat} .

Other properties may be assumed to be independent of T_{sat} .

At this stage, the following preliminary calculations should be carried out.

- (1) Determine the total mass flux from

$$\dot{m} = \frac{4\dot{M}}{\pi D^2 N_T} \quad (4.1)$$

- (2) Calculate the mean heat flux in the cooled length from

$$\dot{q} = \frac{\dot{Q}}{\pi D L N_T}. \quad (4.2)$$

- (3) Calculate the Reynolds number when all the flow is liquid from

$$Re_l = \frac{\dot{m} D}{\eta_l}. \quad (4.3)$$

Also calculate the Reynolds number when all the flow is vapour from

$$Re_g = \frac{\dot{m} D}{\eta_g}. \quad (4.4)$$

- (4) Calculate the mean value of the condensation number from

$$Co = \frac{\dot{q}}{m \Delta h_v}. \quad (4.5)$$

- (5) Determine the density ratio, ρ_l/ρ_g , under conditions at inlet or outlet, whichever is given. The correlations developed here have been checked against data with density ratios in the range 13 to 2100. As discussed in Section 5.4, the error did not show any tendency to vary with density ratio, so it is reasonable to suppose that the procedure may be used outside this range.

- (6) Determine the viscosity ratio, η_l/η_g . The correlations have been checked here against data with viscosity ratios in the range 9 to 31. Friedel's method of predicting the frictional component is recommended for values of the viscosity ratio up to 1000.

- (7) Calculate the Froude number for all liquid flow from

$$Fr = \frac{(\dot{m}/\rho_l)^2}{g D}. \quad (4.6)$$

(In many other sources, the Froude number is defined as the square root of that given by Equation (4.6).)

- (8) Calculate the Weber number for all liquid flow from

$$We = \frac{\dot{m}^2 D}{\rho_l \sigma g_c}. \quad (4.7)$$

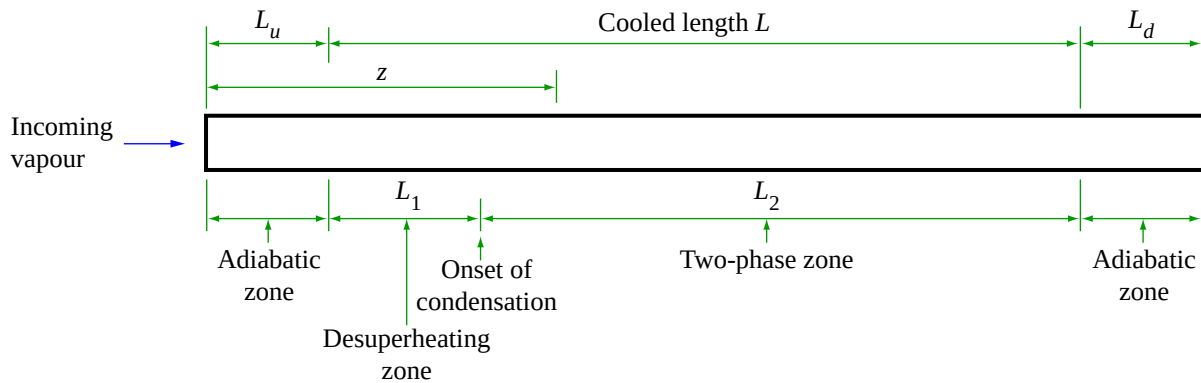
4.2 Relationship between Quality and Length

It is necessary that heat transfer calculations have already been carried out to check that the required rate of heat transfer, $\dot{Q}$, can be achieved in the cooled length, L , that is available. The procedures given in ESDU 91024⁵ for heat transfer calculations with downflow condensation should also be applicable to horizontal tubes in the annular flow region. The heat transfer calculations will have been done by dividing the cooled length into various sections defined by the quality at the inlet to that section. (Quality is defined as the ratio of the mass flow rate of vapour to the total mass flow rate.) As the vapour and liquid are assumed

to be in equilibrium, the quality at any point is the same as the thermodynamic quality, given by

$$x_g = \frac{h - h_l}{\Delta h_v} \quad (4.8)$$

where h is the local value of the enthalpy, determined from heat transfer calculations, and h_l is the enthalpy of saturated liquid. With an initially superheated vapour, the quality at inlet is greater than one.



Sketch 4.1 Definition of lengths

To be conservative it is assumed that condensation does not start until the bulk temperature of the vapour has been reduced to the dew point and that the pressure drop in the desuperheating zone can be estimated by the normal procedures for single-phase flow. Thus it is necessary for the heat transfer calculations to include an estimate of L_1 , the length of the tube required for single-phase desuperheating, and $L_2 (= L - L_1)$, the length of tube required for two-phase condensing. Sketch 4.1 shows the different lengths. If there is a two-phase mixture at inlet, $L_1 = 0$ and $L_2 = L$.

- (9) For the condensing zone, choose several values of the quality between those at inlet and outlet. Carry out heat transfer calculations to determine the length of tube required for each decrement in quality. It is likely that sections used for the thermal calculations may also be used for calculating the pressure drop. The number of steps needed for both calculations depends on the accuracy required and the amount by which the quality changes from inlet to outlet. For hand calculations, it may be adequate if the incremental change in quality along each section is in the region of 5 to 10%.

Check that the total length available does not differ significantly from the sum of the estimated lengths required for heat transfer in the single-phase and two-phase regions. If it does, revise the input data accordingly. Thus, if the length is not adequate, either the length provided must be increased or the temperature of the cooling fluid must be reduced.

- (10) The correlations given here have been checked in regions of annular/dispersed flow, where the vapour velocity is high and some of the liquid is dispersed in the vapour. They will not apply to the end of a tube where the quality is approaching zero, nor will they apply at an exceptionally low mass flux. Consult Table 8.2 to check that the prevailing conditions do not differ greatly from those for which the correlations have been checked. Towards the end of a tube, where quality is much reduced from that at inlet, the negative pressure gradient is comparatively low, and it may possibly rise to a slight positive value; thus errors in its estimation will not introduce any error in the estimation of the total pressure drop in the tube.