



ESDU 84023

Issued September 1984

With Amendment A

August 1985

Shell-and-tube exchangers: pressure drop and heat transfer in shellside downflow condensation

ESDU 84023 A 1985

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SHELL-AND-TUBE HEAT EXCHANGERS: PRESSURE DROP AND HEAT TRANSFER IN SHELLSIDE DOWNFLOW CONDENSATION

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SHELL-AND-TUBE HEAT EXCHANGERS: Pressure drop and heat transfer in shellside downflow condensation

1. NOTATION AND UNITS

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

		<i>Units</i>	
		<i>SI</i>	<i>British</i>
a	parameter in Table 8.1 for Equation (5.10)	—	—
A	total external surface area for heat transfer	m^2	ft^2
c_p	specific heat capacity of fluid at constant pressure	J/kg K	$\text{Btu/lb } ^\circ\text{F}$
D	external diameter of tubes	m	ft
D_n	internal diameter of nozzle	m	ft
E	thermal effectiveness, from Equation (8.4)	—	—
g_c	constant in 2nd law of motion*	— (1.0)	$\text{lb ft/h}^2\text{lb f}$ (4.17×10^8)
g_n	gravitational acceleration	m/s^2 (9.81)	ft/h^2 (4.17×10^8)
H	depth of tube bundle	m	ft
H_n	average space between end of inlet nozzle and nearest obstruction	m	ft
J	mechanical equivalent of heat	— (1.0)	ft lbf/Btu (778)
k	number showing position of a layer of tubes, counted from top of bundle	—	—
K	pressure loss coefficient, see Equation (4.1)	—	—
L	heated length of tubes	m	ft
$\dot{M}_c$	mass flow rate of condensate per tube	kg/s	lb/h
$\dot{M}_n$	mass flow rate of condensate or vapour per nozzle	kg/s	lb/h
m	parameter from Table 8.1 for Equation (5.10)	—	—
$\dot{m}$	total mass flux between tubes in the centre of the shell, based on S_m	kg/s m^2	lb/h ft^2
n	negative exponent of Reynolds number in equation for pressure loss coefficient	—	—

For footnote see end of Notation section

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n_L	number of layers	—	—
n_t	number of tubes	—	—
P_t	pitch of tubes	m	ft
P_x	pitch of layers	m	ft
p	static pressure	N/m ² (Pa)	lbf/ft ²
$\dot{Q}$	rate of heat transfer	W	Btu/h
Re_s	Reynolds number, based on D and S_s	—	—
Re_m	Reynolds number, based on D and S_m	—	—
S_s	mean crossflow area in shell without tubes	m ²	ft ²
S_m	minimum area for crossflow between tubes in centre of shell	m ²	ft ²
S_{ne}	effective escape area under inlet nozzle	m ²	ft ²
T	temperature	K	°F
x_g	quality = $\frac{\text{mass flow rate of vapour}}{\text{total mass flow rate}}$	—	—
U	overall heat transfer coefficient, referred to the external surface area (A)	W/m ² K	Btu/h ft ² °F
U'	combined heat transfer coefficient for the coolant, the wall and fouling	W/m ² K	Btu/h ft ² °F
V	mean empty shell velocity, based on S_s	m/s	ft/h
V_a	axial flow velocity along top of condenser at the nozzle	m/s	ft/h
V_l^*	dimensionless flow rate of liquid from Equation (4.12)	—	—
z	distance measured down from top of bundle	m	ft
α	heat transfer coefficient	W/m ² K	Btu/h ft ² °F
α_c	heat transfer coefficient of film of condensate	W/m ² K	Btu/h ft ² °F
α_{gr}	gravity-controlled component of α_c	W/m ² K	Btu/h ft ² °F
α_{Nu}	heat transfer coefficient according to Nusselt, Equation (5.1)	W/m ² K	Btu/h ft ² °F
α_{sh}	shear-controlled component of α_c	W/m ³ K	Btu/h ft ² °F
Γ	condensate rate per unit length of tube ($\dot{M}_c/L$)	—	—
δ	thickness of film of condensate	m	ft
Δh_v	latent heat of vaporisation	J/kg	Btu/lb

For footnote see end of Notation section

Δp	drop in static pressure	Pa	lbf/ft ²
ΔT	temperature difference ($T_h - T_c$)	K	°F
ΔT_s	drop in saturation temperature due to pressure drop, from Equation (6.6)	K	°F
η	dynamic viscosity of fluid*	kg/s m	lb/h ft
Λ	property index of fluid, defined by Equation (4.5)	—	—
λ	thermal conductivity of fluid	W/m K	Btu/h ft °F
ρ	density of fluid	kg/m ³	lb/ft ³
ϕ_c	condensation parameter, defined by Equation (5.6)		

Extra subscripts

a	axial flow along top of shell
C	coolant
e	elemental value
g	vapour flowing alone, without any liquid
$g0$	total flow rate as vapour
H	hot fluid
in	at inlet
l	liquid flowing alone, without any vapour
$l0$	total flow rate as liquid
m	mean
n	nozzle
out	at outlet
p	tubeside pass
s	saturation
sh	shear controlled
T	total
tp	two-phase

* Force = mass acceleration/ g_c . In SI units, the unit of force (N) is such that g_c is unity. In British units, using lb force and lb mass, g_c is numerically equal to the acceleration of a freely falling body

A dash (') after a symbol indicates an approximate value which may have to be re-calculated more accurately at a later stage in the calculation procedure.

2. PURPOSE AND SCOPE OF THIS DATA ITEM

This Data Item relates to the thermal design of tubular condensers in which the coolant flows through the tubes, which are horizontal, and a pure vapour flows across them in a downward direction. This type of single-pass shell is designated by TEMA³ as an “X” shell. The several possible arrangements are described in Section 3. The purpose of this Item is to provide methods of estimating the local values of the shellside pressure gradient and heat transfer coefficient at various values of the quality (ratio of mass flow rate of vapour to total mass flow rate). The former may then be combined with the pressure drop in the inlet nozzle to give the total pressure drop; the latter may be combined with the combined heat transfer coefficient for coolant, tube and fouling to give local values of the overall heat transfer coefficient. As the flow of vapour is essentially at right angles to the tubes, correlations for crossflow may be used. This information may be used to check that a condenser of given design will be adequate for a specified duty.

The basis for the calculation of pressure drop and heat transfer is described in Sections 4 to 6. The calculation procedure is summarised in Section 8. A worked example is given in Section 9.

The calculations are needed for one of the following four types of problem:

- (1) Checking – to decide whether a design that has been offered will be adequate for a specified duty – calculate the area required for the duty and compare it with the area that is available, determined from the drawing of the condenser.
- (2) Design – to design a condenser – assume a preliminary design and check it, as described in (1). Refine the design and repeat the calculations until a design is achieved that will meet the thermal requirements, the pressure drop limitations and give an economic design (this will be dealt with in a later Data Item).
- (3) Rating – to determine the outlet conditions from a condenser of known dimensions, given the flow rates and inlet temperatures of the fluids – guess the outlet quality, calculate the outlet temperature of the coolant and check, as in (1).
- (4) Testing – to evaluate the results of tests on an operating condenser – guess the sum of the internal and external fouling resistances and check, as in (1), that the condenser is adequate for the measured performance. If not, adjust the estimated fouling resistance until agreement is achieved.

The shellside downflow condenser dealt with in this Data Item is preferred when the volume flow rate of the hot vapour is many times greater than that of the coolant. It is also excellent when a very low pressure drop is essential. As the flow pattern is mainly downwards, any gases which substantially remain uncondensed are swept to the bottom of the condenser and may be vented to avoid the formation of stagnant pockets inside the condenser. Tube vibrations can easily be prevented by the use of many support plates, as described in Section 3.

The most common application is for condensing steam leaving a turbine at deep vacuum. They are also used in refrigeration and air-conditioning duties and for process duties over a wide range of conditions. For moderate pressures, other types of shell may be preferred, see TEMA³. Tubeside condensation is preferred when the vapour is at an appreciable pressure or when the condensate is corrosive.

3. POSSIBLE ARRANGEMENTS OF CROSSFLOW SHELLSIDE CONDENSERS

As the shellside downflow condenser is the preferred type for handling large volumetric rates of flow of vapour, pressure drop being important, care is necessary in the design of the inlet nozzles to provide an adequate cross-sectional area for flow of the vapour from the inlet nozzle to the top of the first row of tubes. Four possible ways of achieving this are illustrated in Sketches 3.1 to 3.4; the details of these schemes are described below. The optimum number of tubeside passes for the coolant depends on the required volumetric rate of flow of the coolant and the size of condenser required. It is independent of which method is employed for introducing the vapour into the condenser. A single pass of coolant is shown in Sketch 3.1, two passes in Sketches 3.3 and 3.4, and four passes in Sketch 3.2. In each sketch, the stippled regions in cross-sections on the left-hand-side are the regions that contain the tubes.

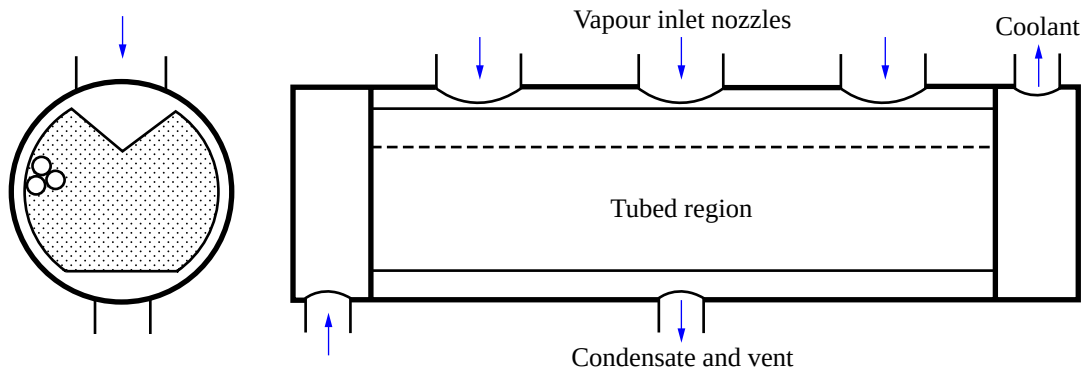
3.1 Design Details

Support plates, such as the typical plate shown in Sketch 3.2, are placed at regular intervals along the length of the shell to support the tubes and avoid vibrations. The support plate in the centre of the inlet nozzle(s) is especially important. Each tube is supported by every plate, and the pitch of the plates is sufficiently close to avoid tube vibrations, as described by Chenoweth⁵.

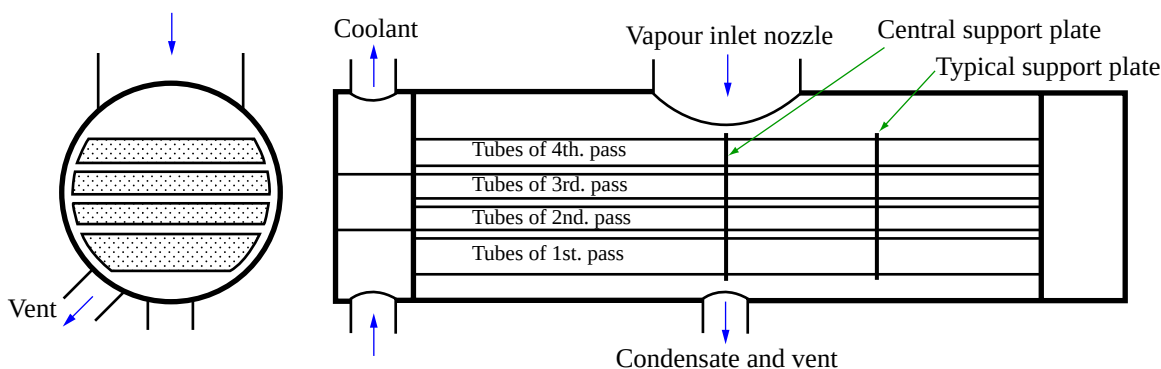
In each sketch the branch at the bottom of the shell drains the condensate from the shell (see Section 7); it can also be used to purge the shell of air during start-up and to purge any incondensable gases that may collect during operation. Alternatively, a separate branch may be supplied for the latter function placed near to the bottom but above the maximum level of condensate which may be estimated as described in Section 7. this method of venting is shown in Sketch 3.2.

Sketch 3.1 shows three separate vapour inlet nozzles with an untubed sector at the top of the shell. Sketch 3.2 shows a single large inlet nozzle with a large untubed segment at the top of the shell. Sketch 3.3 shows an arrangement in which a smaller shell is completely filled with tubes, the vapour being distributed along the top through a vapour inlet box (two inlet nozzles are shown here). In the arrangement shown in Sketch 3.4 a larger shell is used. This acts as the pressure vessel, but the flow of the vapour is controlled by a thin cylindrical shroud around the bundle. The tubes extend to the shroud, which has large gaps in its top to allow the vapour to reach the top tubes and small gaps at the bottom to drain the condensate.

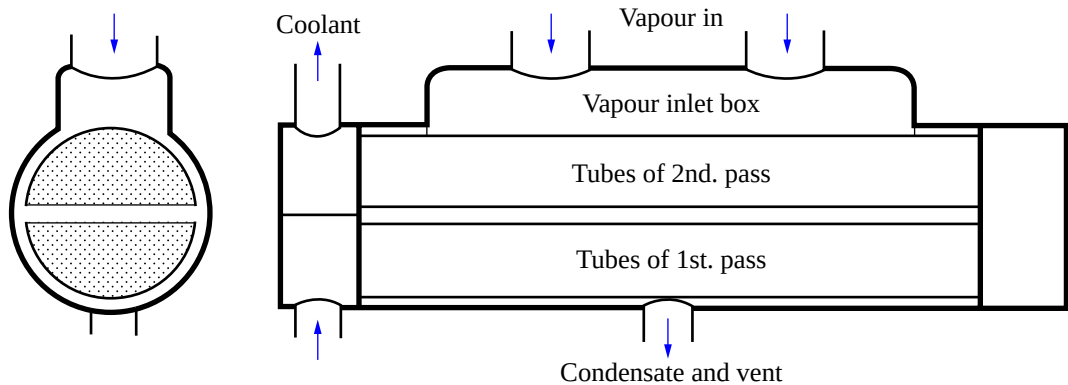
Other items are not shown in the sketches, but required by TEMA³, are sealing strips and impingement plates. Sealing strips are placed between the outermost tubes and the shell (or shroud) to reduce the amount of leakage between the bundle and the shell. This is not a serious problem when condensing a pure vapour; in this case the sealing strips may be omitted provided that the gap is small between the outermost tubes and the shell (or shroud). Impingement plates are placed opposite the inlet nozzles to protect the first and second layers from damage by erosion, but these increase the pressure drop and may introduce extra problems in the avoidance of the tube vibrations. Perforated plates are sometimes used to alleviate these problems.



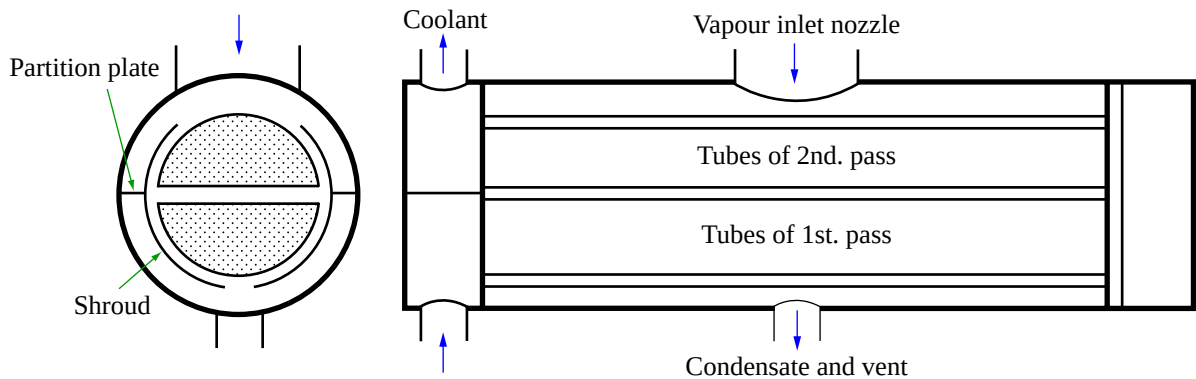
Sketch 3.1 Shellside downflow condenser with single-pass coolant



Sketch 3.2 Shellside downflow condenser with four-pass coolant and side vent



Sketch 3.3 Shellside downflow condenser with vapour inlet box



Sketch 3.4 Shellside downflow condenser with shrouded bundle

3.2 Causes of Poor Performance

On occasions, carefully designed condensers have seriously failed to achieve the predicted performance. The main obstacles to the efficient operation of condensers handling pure vapour are discussed below.

- (i) Maldistribution. It is necessary to obtain a good distribution of the incoming vapour along the length of the shell, without introducing an excessive pressure drop, as discussed in Section 3.1. It is important to avoid stagnant zones because vapour thought to be pure may contain small traces of incondensables which would collect in a stagnant zone after a few days of operation, thus rendering that area of the condenser useless.
- (ii) Poor drainage. To avoid flooding of the condenser the outlet nozzle must be an adequate drain, as described in Section 7.
- (iii) Inadequate venting. It is essential to install a vent for purging the condenser on start-up and for removing any incondensables that may be inadvertently introduced.
- (iv) Fouling. It is very difficult to predict the amount of fouling⁴, although much research has been carried out on fouling by cooling water. Very little information is available on fouling by a condensing vapour, but this is seldom a source of trouble.
- (v) Frozen condensate. If the inlet temperature of the coolant is less than the freezing temperature of the condensate, it is necessary to check whether the temperature of the wall (including any fouling layer) could fall below the freezing temperature under any conditions of operation. If so, a layer of frozen condensate will form on the outside of the tubes to such an extent that the interface temperature equals the freezing temperature. This results in a loss in thermal performance and an increase in pressure drop.

4. PRESSURE DROP IN CROSS-FLOW CONDENSERS

Most of the pressure drop in a shellside downflow condenser occurs between the inlet nozzle and the top of the bundle of tubes. This reduces the saturation temperature within the condensing zone, so it should be estimated before attempting to estimate the rate of heat transfer. Too great a pressure drop significantly reduces the available temperature difference and also affects the equipment that follows the condenser; especially it affects the design of the vacuum equipment for a steam condenser.

This Section is concerned with the estimation of the total pressure drop in the condenser, from the inlet nozzle to the drain at the bottom of the shell. In the downward flow of vapour and condensate across a bundle of horizontal tubes, it may be assumed that the two phases are in separated flow and that gravity affects the flow of condensate but not the flow of vapour. There is a pressure drop due to the acceleration of the vapour as it enters the bundle and a pressure rise due to deceleration in the bundle as condensation proceeds. In the absence of experimental data on these items, they are neglected as it is expected that they approximately cancel each other. Thus the problem is reduced to the calculation of the pressure loss due to wall friction and form drag alone.

The first step in the estimation of the two-phase pressure loss in the bundle is to estimate the pressure gradient at various values of the quality first with the vapour flowing alone and then with the liquid flowing alone. These may then be combined to give the two-phase pressure gradient. It is assumed that the vapour is initially saturated or slightly superheated.

4.1 Single-phase Pressure Gradient in Flow Across Tube Bundles

The pressure gradient in single-phase flow may be estimated by the method of ESDU 79034¹³. The pressure gradient is given by

$$\frac{dp}{dz} = -K \frac{\rho V^2}{2D} \quad (4.1)$$

the pressure loss coefficient, K , is the number of velocity heads lost in a depth of bundle equal to the external diameter (D) of the tubes. It can be determined from Figure 1, 2 or 3, taken from ESDU 79034¹³, according to the pattern of the tubes; it depends on the pitch-to-diameter ratio, P_t/D , and the Reynolds number based on the external diameter of the tubes and the mean empty shell velocity, given by

$$Re_s = \rho V D / \eta \quad (4.2)$$

Thus at any value of the quality (x_g = mass flow rate of vapour/mass flow rate of vapour plus condensate) it is possible to determine V , Re_s , K and $(-dp/dz)$ for either phase flowing alone, without the other.

4.2 Two-phase Correction Factors in Flow Across Tube Bundles

If it is assumed that the two phases are flowing separately, it can be shown that the two-phase pressure gradient is given by

$$\left(-\frac{dp}{dz} \right)_{tp} = \left[\left(-\frac{dp}{dz} \right)_g^{1/(2-n)} + \left(-\frac{dp}{dz} \right)_l^{1/(2-n)} \right]^{2-n} \quad (4.3)$$

where subscripts g and l denote the values of the pressure gradient if the vapour and liquid were flowing

alone, without the other. The exponent n is the negative slope of the curve of K plotted against Re_e on logarithmic co-ordinates, as in Figures 1 to 3. In laminar flow, n has a value of 1; when $Re \geq 100$, n is in the range 0.1 to 0.3.

Barbe *et al.*⁹ show excellent agreement with the above theory in experiments with air and water flowing down over a bank of coiled tubes at atmospheric pressure and rates from 35 to 208 kg/s m². The ratio $(-dp/dz)_g/(-dp/dz)_l$ ranged from 3 to 8000.

It is convenient for calculating the pressure drop in a condenser to express the local two-phase pressure gradient as a factor of the pressure gradient that would exist if the flow were all vapour. Thus Equation (4.3) becomes

$$\left(-\frac{dp}{dz}\right)_{tp} = \left(-\frac{dp}{dz}\right)_{g0} \left[x_g + (1 - x_g) \Lambda^{1/(2-n)} \right]^{(2-n)} \quad (4.4)$$

where subscript $g0$ denotes that the pressure gradient has been calculated on the assumption that all the flow is vapour, *i.e.* it is the pressure gradient at the beginning of the condensing zone, x_g is the local value of the quality and Λ is the property index, which is defined as the ratio of the pressure gradient with all liquid flow to that with all vapour flow. It is given by

$$\Lambda = \frac{\rho_g}{\rho_l} \left(\frac{\eta_l}{\eta_g} \right)^n \quad (4.5)$$

Figure 4 shows the ratio $(-dp/dz)_{tp}/(-dp/dz)_{g0}$ plotted against x_g for different values of Λ , taking $n = 0.2$.

4.3 Estimation of Total Pressure Drop in Condensing Zone

It is possible at this stage to make an approximate estimation of the pressure drop across the bundle, by assuming that x_g , the quality, varies linearly with z , the distance down from the top of the bundle. The first step is to calculate Δp_{g0} the pressure drop for flow as 100% vapour, *i.e.* with no coolant in the tubes, as described in Section 4.1. The approximate pressure drop is obtained by multiplying this by the correction read from Figure 5 for the value of Λ calculated from Equation (4.5). A more accurate calculation may be made after completing the heat transfer calculations and estimating a relationship between x_g and z . For chosen values of x_g it is possible to calculate the pressure gradient from Equation (4.4) and the distance (z) at every chosen values of x_g . The pressure drop is obtained by integrating the pressure gradient:

$$\Delta p_{tp} = \int_0^H - \left[\frac{dp}{dz} \right]_{tp} dz \quad (4.6)$$

where H is the total depth of the bundle. Equation (4.6) may be solved graphically or numerically. The total pressure drop is then given by

$$\Delta p_T = \Delta p_{n\ in} + \Delta p_a + \Delta p_{tp} \quad (4.7)$$

where $\Delta p_{n\ in}$ is the pressure drop in the inlet nozzle, and Δp_a is the pressure drop in axial flow along the top of the shell, calculated as described in the next Section.