SIZING OF VERTICAL VAPOR-IN-TUBE REFLUX CONDENSERS

By

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NOMENCLATURE

A	area, or empirical constant in Eqn. (3.7)
Ap	amplitude of interface disturbance
a	parameter defined by Eqn. (3.43)
B	empirical constant in Eqn. (3.7)
Bo	Bond number, defined by Eqn. (2.9)
C_f	interface enhancement factor
C_p	specific heat capacity, J/kg.K
c	condensation efficiency
c_i	imaginary part of wave celerity, m/s
d	tube diameter, m
E	entrainment mass flowrate, kg(liquid)/kg(total mixture)
E1, E2	empirical constants in Eqn. (2.7)
F	force, N
F_1, F_2	parameters defined by Eqns. (2.12) and (2.13) respectively
Fr	Froude number, defined by Eqn. (2.8)
f	friction factor
G	superficial vapor mass flowrate, kg/s.m^2
g	acceleration due to gravity, m/s^2
H	molar enthalpy, kJ/kmol
Δh_{lg}	latent heat, J/kg
J*	dimensionless parameter defined in Eqn. (2.1)
Ja	Jakob number, defined by Eqn. (2.17)

j	superficial velocity, m/s, or Colburn j-factor, or component index
K	Kutateladze number, defined by Eqn. (2.2)
k	wave number, m^{-1}
L	superficial liquid mass flowrate, $kg/s.m^2$, or tube length, m
$\dot{M}$	molar flowrate, kmol/s or kmol/hr
Mw	molecular weight, kg/kmol
$\dot{m}$	mass flowrate, kg/s
m, C	empirical constants
N_t	number of tubes
$\dot{n}$	condensation molar flux, $kmol/s.m^2$
P	pressure, N/m^2
ΔP	pressure difference, N/m^2
Q	heat transfer rate, W
q	heat flux, W/m^2
R	reflux ratio
R_f	fouling resistance, $m^2.K/W$
r	radius, m
S	circumference, m
T	temperature of hot process stream, K
ΔT_{lm}	logarithmic mean temperature difference, K
t	temperature of coolant stream, K
U	overall heat transfer coefficient, $W/m^2.K$
v	actual velocity, m/s
W	volumetric flowrate per unit width, $m^3/s.m$
x	quality, or mole fraction
y	mole fraction
Z	parameter defined by Eqn. (4.38)

z	mole fraction of the local condensing mixture, or coordinate

Greek letters

α	heat transfer coefficient, $W/m^2.K$
α_{oi}	heat transfer coefficient from the interface to the coolant, $W/m^2.K$
β	mass transfer coefficient, $kmol/s.m^2$
Γ	mass flowrate per unit width, kg/s.m
δ	diffusivity, m^2/s
∂	film thickness, m
ξ	mass transfer coefficient correction factor given by Eqn. (4.15)
ε	void fraction
η	dynamic viscosity, $N.s/m^2$
θ	tube end taper angle, radians
θ_s, θ_a	correction factors defined by Eqns. (3.42) and (3.44) respectively
κ	thermal diffusivity, $\kappa = \lambda/\rho C_p$, m^2/s
λ	thermal conductivity, W/m.K
ρ	density, kg/m^3
σ	surface tension, N/m
τ	shear stress, N/m^2
ϕ	dimensionless parameter in Eqn. (3.11), or parameter defined by Eqns.(4.16) and (4.26)
ϕ_{ltt}, x_{tt}	dimensionless parameters defined by Eqns.(3.47) and (3.48)

Subscripts

$\oplus$	reference area
1,2	component index

a	adiabatic
b	bulk
c	condensate
cool	coolant
cr	critical relative
crit	critical
eff	effective
f	friction
g	gas, or gravity
i	interface, or inside, or incremental element index
in	inlet stream
k	subscript g or l, or incremental element index
l	liquid
m	momentum, or incremental element index
o	outside
out	outlet stream
r	reduced
ref	reference
sv	sensible heat, vapor-side
t, T	total
TP	two-phase
w	wall
z	coordinate

Superscripts

-	mean

*	dimensionless
sat	saturated

CHAPTER I

INTRODUCTION

Condensers are heat-exchange equipment in which one or more of the condensable components of a vapor or vapor-gas mixture undergoes phase change into a liquid, due to heat exchange with a coolant fluid stream. They can be broadly classified, according to the contacting mechanism between the streams, into two types :

(a) Direct contact condensers, in which the liquid coolant stream is brought in direct contact with the vapor-gas mixture.

(b) Surface condensers, in which the coolant receives heat from the vapor-gas mixture across a wall, causing condensation to occur on the vapor-side wall surface. These condensers come in a variety of different configurations -- from the simple double pipe heat exchanger and the widely used shell and tube heat exchanger to the more recent plate type heat exchanger.

Reflux condensers belong to the class of surface condensers. Specifically, in a reflux condenser, the vapor-gas mixture flows upwards, with the condensate draining downward under the influence of gravity. Reflux condensers are widely used in the chemical process industries and the pharmaceutical industry for the control of chemical reactors, as internal condensers in distillation columns and also individually for the purpose of rough rectification. When used as partial condensers, they are also known as dephlegmators.

Configurations of Reflux Condensers

Reflux condensers are seen in a variety of configurations. Their most common configuration is the 1-1 type vertical shell and tube heat exchanger, shown in Fig. 1.1. In it, the vapor-gas mixture flows upwards inside the tubes and condenses on the side walls while the coolant stream flows in the shell. The condensed phase drains downward under

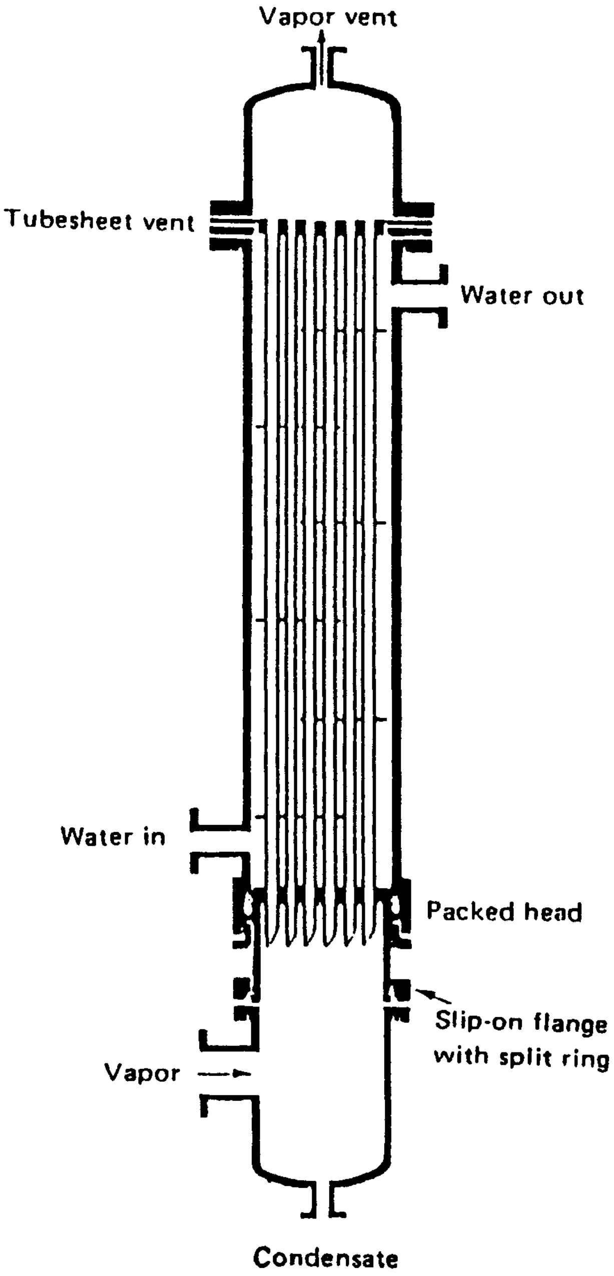


Fig. 1.1 A typical reflux condenser.

the influence of gravity, countercurrent to the rising vapor stream and the non-condensable gases leave the condenser through the vent nozzle at the top. In this configuration, they are mostly found mounted directly on (or sometimes even internal to) a reactor or a distillation column.

Two other configurations of reflux condensers are shown in Figs. 1.2 and 1.3. Fig. 1.2 shows a reflux air-cooled overhead condenser, in a V-shaped arrangement. It is custom designed for use as an integrated portion of a distillation column. Its V-shaped configuration permits a large induced draft fan on top. The tubes are arranged symmetrically around the column in four sets of inclined bundles. Dehne (1969) discusses its use as a simple condenser arrangement over a distillation column.

Fig. 1.3 shows a horizontal shell side reflux condenser. In the paper by Steinmeyer and Mueller (1974), Bell discusses such a condenser, which is used for partially removing a condensable vapor or vapor mixture from a non-condensable gas. The condensable vapor mixture is composed of two fractions - a heavy tarry material which is deposited on the lower tubes and a light component which drains downward and removes the tarry material by solution.

Though all the configurations discussed above show reflux condensers of the shell and tube type, it should be noted that they can be of other types also, e.g., a simple double pipe exchanger or a plate fin heat exchanger.

Applications of Reflux Condensers

A reflux condenser mounted on a reactor with a boiling solvent returns the condensate at a temperature close to that of the inlet vapor stream vaporized from the reaction mixture. The returned condensate serves as extra material added to the reactor at the reaction temperature, which is capable of absorbing the heat of reaction and vaporizing, only to be condensed and returned by the reflux condenser. This operation in conjunction

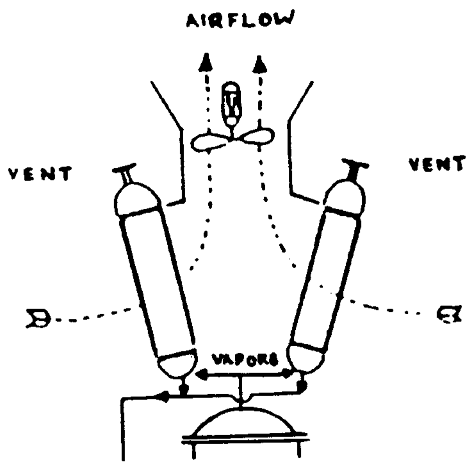


Fig. 1.2 Reflux air-cooled overhead condenser
(Dehne, 1969)

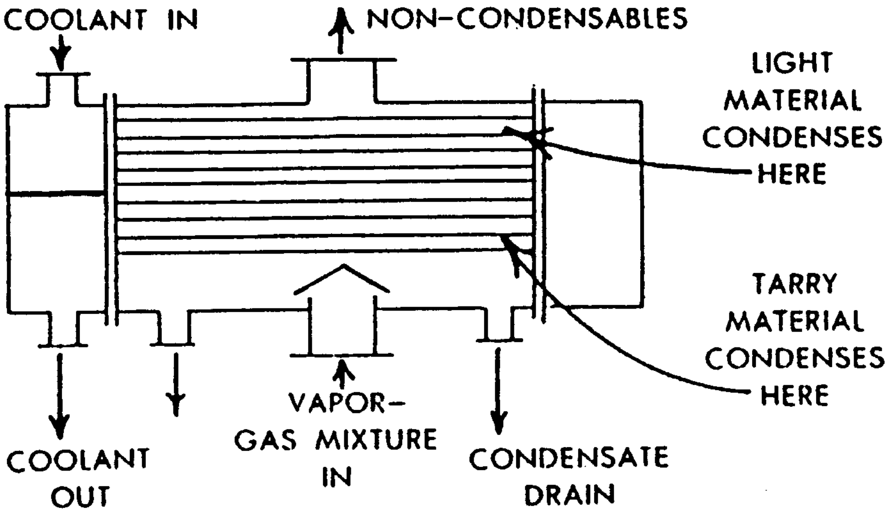


Fig. 1.3 Shellside partial reflux condenser
(Steinmeyer and Mueller, 1974)

with insignificant pressure losses helps to stabilize the operating temperatures and pressures of the reactor.

Mounted on a distillation column, a reflux condenser condenses wholly or partially the vapor mixture leaving the column, returning it as reflux to the column. In this process, it obtains additional rectification of the vapor-mixture for partial condensation. Because of the way it operates, it helps maintain a smooth and stable operation of the column.

In addition to the above uses, reflux condensers are also used as separation equipment in the hydrocarbon processing industry. The horizontal shell side reflux condenser of Fig.1.3 is such a condenser. The process of dephlegmation or partial reflux condensation (Chiu (1990)) is considered today a novel separation technique that offers good capabilities for separating gas mixtures. It combines mass transfer and heat exchange to achieve the desired separation. Air Products and Chemicals Inc. (Bernhard et al. (1988), Bernhard et al. (1986)) have been using the dephlegmator on a commercial scale in a wide variety of cryogenic gas separation applications, including selective removal of methane in the purification of H2 - CO synthesis gas and recovery of ethylene and other valuable hydrocarbons from FCC (fluid catalytic cracking), oil gasifier and dehydrogenation gas sources. They find the dephlegmator processes to be economical, reliable, efficient and also easy to operate and control.

Operational Characteristics and Advantages of Reflux Condensers

The basic traits of a reflux condenser operation are :

(a) The condensate stream is returned at a temperature near that of the inlet vapor stream, which is the hottest temperature of the system.

(b) The countercurrent nature of the vapor-condensate flow places an upper limit on the operating vapor velocity that can be used for a smooth reflux condenser operation. Thus, the operating vapor velocity is low.

(c) The pressure loss in the system is very small due to the low operating vapor velocity.

Traits (a) and (c) offer the following advantages for a reflux condenser compared to conventional condensers in which the vapor-condensate flow is cocurrent :

(i) The reflux condenser provides excellent thermal and mechanical stability of the system with which it operates, with relatively few or no controls.

(ii) Returning the condensate stream at a warmer temperature facilitates removal of smaller amounts of low boilers and also minimizes the quantity of dissolved light hydrocarbons and inerts. Thus, the reflux condenser provides enhanced separation capability compared to conventional condensers.

Trait (b) limits the vapor-handling capacity of a reflux condenser.

Reflux condensers are mostly used mounted on a reactor or a fractionation equipment. In this mode, they offer specific merits over conventional ground-mounted condensers, which include :

(i) Elimination of the reflux pump and the related pumping costs

(ii) Reduction of the requirements of piping and the attendant joints between the pipes - This reduces or eliminates the leakage problems and also minimizes pressure loss.

(iii) Saving of ground space and thus real estate cost

However, they are also limited by the following :

(i) Requirement of extra support structure, if units are large

(ii) Possible higher maintenance costs, due to their location

(iii) Higher installation costs, except for small units which can be prefabricated in one piece

Design Problems of Reflux Condensers

Despite the wide use of reflux condensers in the chemical process industries, they are poorly understood from a theoretical point of view, compared to conventional condensers.

Consequently, the design methods for these condensers are also poorly developed. Currently, the main problems associated with the design of reflux condensers are :

(a) Uncertainty in the correct prediction of the flooding point.

Flooding is one of the major disadvantages of reflux condensers. The flooding point denotes the upper physical limit of a steady countercurrent two-phase flow operation. Many experimental and analytical studies have been made of the flooding phenomena in vertical tubes. Both adiabatic and condensing cases have been studied. The result is a wide variety of correlations which can be used to predict the flooding velocity in countercurrent two-phase flow. The problem is however in the large disagreement between the correlations, as noted by several investigators, in predicting the flooding point for similar operating parameters. The reasons for this include (a) differences in the criteria used to define the flooding point, and (b) differences in the test-section entrance and exit geometries employed. As a result, no single correlation can be clearly identified which can predict the flooding point accurately for a wide range of the operating parameters.

(b) Poor understanding of the fluid mechanics, heat transfer and mass transfer aspects of the reflux condensation process.

Several studies have been made on the fluid mechanics of vertical countercurrent two-phase flow. Correlations have been developed to predict the key parameters of any gas-liquid flow, viz. pressure drop, mean film thickness and interfacial shear stress. Also, as mentioned earlier, correlations have been developed to describe the flooding point. However, the knowledge of the film flow hydrodynamics of a steady reflux condensation process is still limited.

The heat transfer aspects of the reflux condensation process are also not well understood. A survey of the literature shows that no study has yet been made specifically to evolve an empirical method of evaluating the condensing heat transfer coefficient in vertical countercurrent vapor-liquid flow.

Knowledge of the mass transfer aspects of the reflux condensation process is almost non-existent.

This thesis study is an attempt to address the above problems to the extent possible and then devise an approximate generalized design procedure for reflux condensers. It is limited to the most common configuration of reflux condensers, i. e. vertical vapor-in-tube reflux condensers. In Chapter II, the different flooding correlations and the comparative studies carried out on them are reviewed and a suitable strategy of predicting the flooding vapor velocity for any reflux condenser design problem as accurately as possible is evolved. In Chapter III, a survey is carried out of the different correlations available in the literature to predict the fluid mechanics and the heat transfer aspects of the reflux condensation process. Then, a suitable method of estimating the local heat transfer coefficient in the vapor core and in the condensate film is developed. In Chapter IV, the different design methods currently in use for pure vapor and multicomponent condensers are reviewed and their application to the design of reflux condensers is discussed. Then, the approximate design procedure developed specifically for multicomponent reflux condensers similar to the Silver-Bell-Ghaly method is presented. In Chapter V, the complete design procedure for a vertical vapor-in-tube reflux condenser is summarized. Also, some important mechanical design features are discussed. Finally, in Chapter VI, conclusions of this work and recommendations for future work are presented.

CHAPTER II

FLOODING PHENOMENON IN REFLUX CONDENSERS

Introduction

One of the major disadvantages of reflux condensers is their capacity limitation due to flooding. Flooding occurs when the inlet vapor velocity to the condenser is sufficient to reduce or even prevent the liquid from draining from the bottom of the condenser.

A number of analytical and experimental studies have been made of the flooding phenomenon in vertical tubes. The result is a wide variety of correlations which can be used to predict the flooding velocity in countercurrent gas-liquid flow. Correlations to deal with both adiabatic and condensing cases have been developed. Some of these correlations work very well for a limited range of fluid properties, equipment configurations and operating conditions. The problem is however in the ability of these correlations to predict consistently well for a wide range of the above parameters. Reviews of the flooding literature published by Deakin (1977) and Bankoff and Lee (1983) describe the limitations of the correlations. They also show wide disagreement in the relative predictive performance of the correlations for similar operating parameters. The main reasons for the latter include :

(a) Differences in the criteria used to define the flooding point.

(b) Differences in the test-section entrance and exit geometries employed.

It is evident that there exists no correlation today which can describe the phenomenon of flooding in vertical tubes completely and accurately, or even predict sufficiently well the conditions under which flooding will occur.

The objective of this chapter is to study the different flooding correlations and the comparative studies carried out on them to devise a suitable strategy to predict the flooding velocity for any reflux condenser design problem as accurately as possible.

Mechanism of Flooding

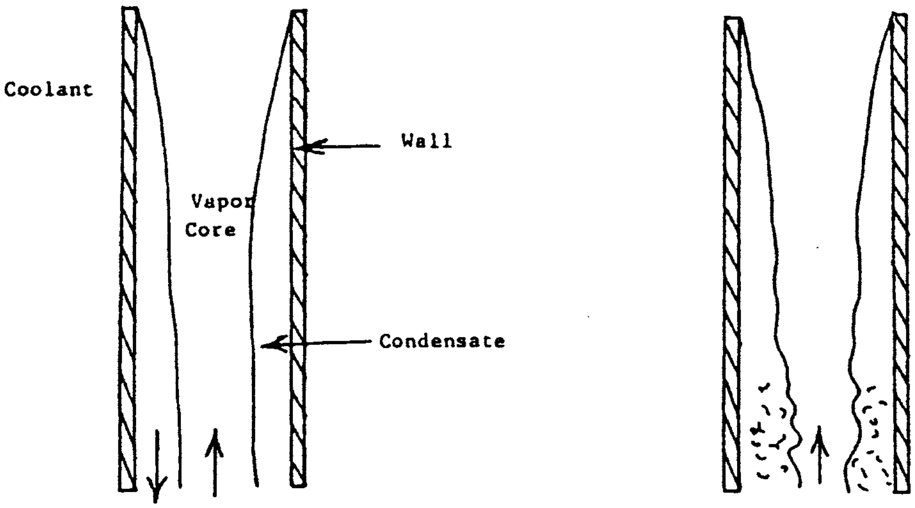
A simple description of the mechanism of flooding, as taken from Bankoff and Lee (1983), is as follows :

Vertical countercurrent two-phase flow is opposed by interfacial friction between the two phases. As the relative countercurrent mean velocity of the phases increases, the interfacial friction also increases monotonically. Hence, for a given geometry and liquid-gas pair, there is a maximum relative velocity that can be sustained in countercurrent flow. This point which describes the physical operating limit of countercurrent two-phase flow is known as the onset of flooding. Further increases in gas/vapor or liquid input rates result in only partial delivery of the liquid out of the bottom. Eventually, if the gas/vapor velocity becomes sufficiently high, none of the liquid is delivered at the bottom, and fully cocurrent upward flow is established.

The term "flooding" has been used by different investigators to describe various aspects of this transition from countercurrent flow to cocurrent flow. A more detailed look at the different aspects of the transition is provided by the description of the flooding mechanism in a reflux condenser, as suggested by Deakin (1977) from visual observation. See Fig. 2.1

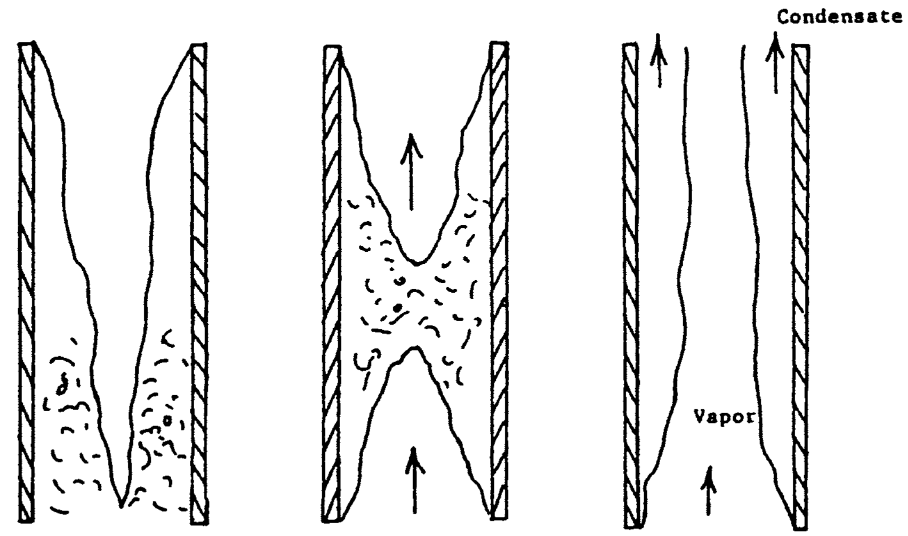
(1) At low vapor velocities a smooth falling film is observed.

(2) On increasing the vapor velocity small disturbance waves appear on the film, which are particularly marked at the vapor inlet.



a. Smooth downward flow of condensate. (Total reflux.)

b. Appearance of turbulence near entrance.



c. Bridging of waves across tube.

d. Vapor pushing out condensate. (No reflux.)

e. Upward cocurrent annular flow. (Climbing film.)

Fig. 2.1 Mechanism of flooding (Subramanyam, 1983)

(3) A further increase in velocity causes the waves at the vapor inlet to bridge across the tube and an intermittent churn flow is established; however the reflux rate is still constant.

(4) Eventually, a vapor velocity is reached which is sufficient to eject liquid from the top of the tube; this is accompanied by a dramatic rise in the pressure drop across the tube.

(5) If the vapor flow is increased further, climbing film annular flow is eventually established.

The relationship between pressure loss across a vertical tube and superficial vapor mass flux at inlet, observed by Deakin (1977), is shown in Fig. 2.2. This graph shows the wide transition region (A-B) between a fully countercurrent flow and a fully cocurrent flow. The pressure fluctuations in this region, particularly near the maximum, are worth noting. Associated with this transition region are a number of phenomena that have been used by different investigators to define their flooding point. Some of the definitions are listed below (Howell, 1987) :

(1) Onset of liquid entrainment.

(2) Sudden rise in liquid entrainment rate.

(3) Onset of liquid bridging.

(4) Sudden rise in the pressure drop across the tube.

(5) Flow pattern observations : As evident from the graph, in the transition region, the liquid in the tube has an unsteady chaotic flow pattern. Definitions of the flooding point based on visual observations of the flow pattern include (Howell, 1987) :

(a) " ..... the point where the liquid film becomes chaotic....."; (Wallis, 1961).

(b) " ..... where the film is disrupted....."; (Hewitt and Wallis, 1963).

(c) " ..... the hydrodynamic state of the system loses stability......"; (Alekseev, 1972).

(d) " .... the liquid film loses stability, ceasing to exist as such...."; (Imura, et al., 1977).

(e) " ..... the appearance of large disturbance waves at the gas-liquid interface....."; (Bankoff and Lee, 1983).

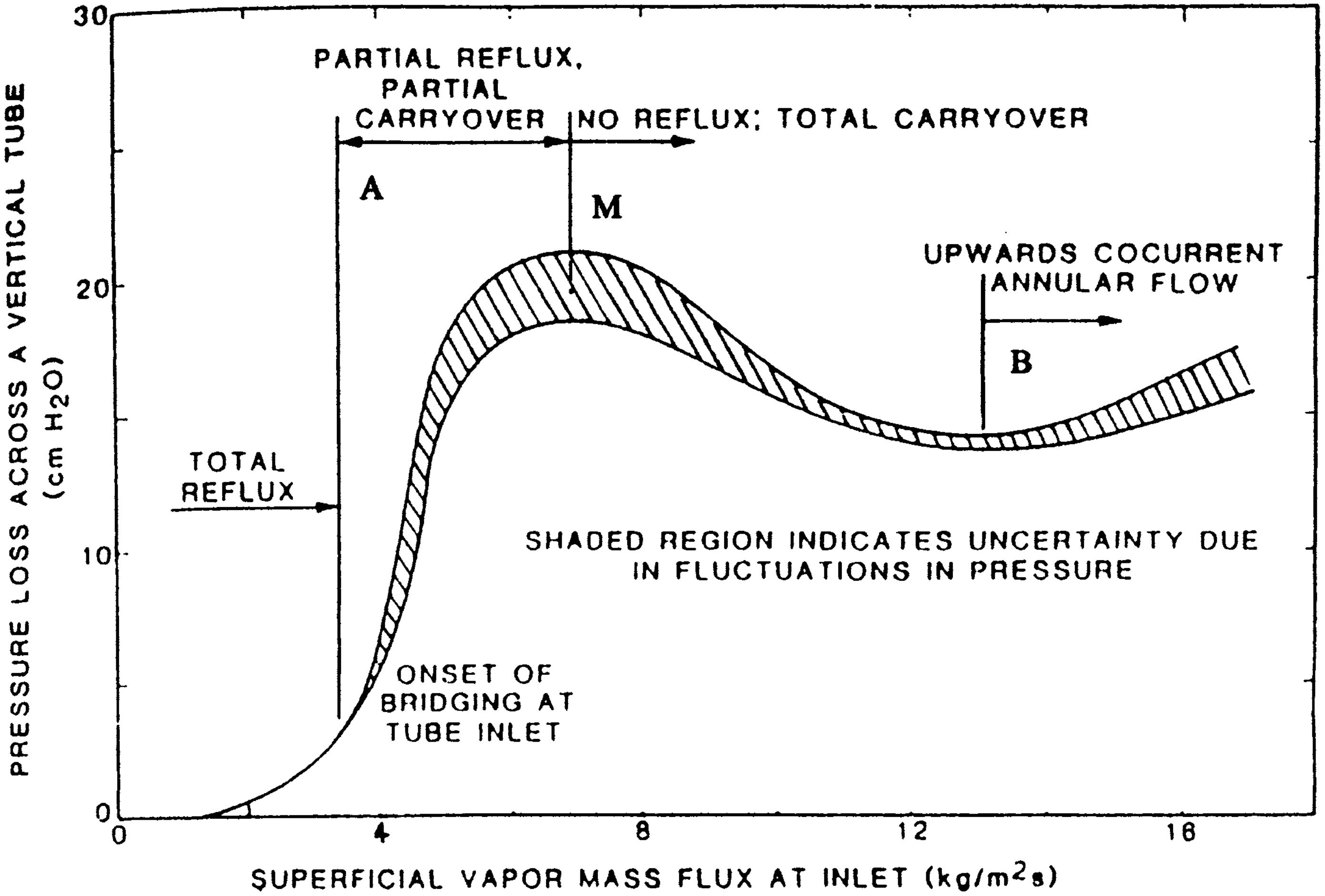


Fig 2.2 Typical variation of pressure loss with superficial inlet vapor mass flux in a reflux condensation process (Deakin, 1977)

(6) Film reversal : The point at which any of the liquid is in upward motion, even if the overall liquid flow is downward.

(7) Climbing film annular flow.

With such varied definitions of flooding, it is not surprising to see the large scatter in the data obtained from 22 different investigators, as reported by McQuillan and Whalley (1985). But, it is important to note here that besides differences in the definitions of the flooding point, there are several other factors which contribute as much or even to a greater degree to the scatter in the total flooding data. These include differences in the test-section geometrical parameters, particularly the entrance and exit geometries, and differences in the fluid properties.

Classification of the Flooding Correlations

In a vertical vapor-in-tube reflux condenser, the highest gas and liquid rates of flow occur at the bottom of the tubes. Hence, flooding begins at this location. This is not necessarily true for adiabatic countercurrent flow systems. Likewise, other differences can be noted between countercurrent condensing and adiabatic flow systems. But it is not clear whether these differences have any significant effect on the mechanism of flooding. Or, in other words, the effect of condensation on the mechanism of flooding in a vertical countercurrent two-phase flow system is yet to be determined.

Secondly, the correlations developed specifically for reflux condensers are very few in number compared to the extent of the total flooding literature. Many factors found to have a significant effect on the flooding mechanism in adiabatic systems have not been properly investigated with respect to condensing systems.

Keeping in mind the above two reasons, it is advisable to examine the entire flooding literature, i.e., flooding correlations on both reflux condensers and adiabatic systems, before evolving a strategy to determine the flooding gas velocity for any design situation.

Since this study deals with reflux condensers, it is helpful to classify the flooding correlations as :

(1) Flooding Correlations for Adiabatic Systems.

(2) Flooding Correlations for Reflux Condensers.

Flooding Correlations for Adiabatic Systems

Analytical Correlations

A number of analytical models have been developed to predict the onset of countercurrent flooding in vertical single tubes. The models differ widely in their description of the onset of flooding and the additional assumptions employed. Bankoff and Lee (1983) have classified the models into four main categories, based on their flooding point definitions :

(1) Stationary theory of a traveling wave : This theory considers flooding to be the result of interfacial instability between two superposed fluids flowing at different velocities. Models have been developed on this theory, after employing various assumptions such as potential flow (Imura, et al., 1977; Tien, et al., 1980), viscous laminar flow (Cetinbaduklar and Jameson, 1969) and finite amplitude surface waves (Zvirin, et al., 1979).

(2) Envelope theories : This class of models defines the flooding condition to be the limit of stable operating conditions as either the liquid or gas flowrate is increased. The latter definition suggests an envelope theory based on the steady hydrodynamic equations, the envelope being some limiting curve in the (J_g^*, J_f^*) plane that separates the operating region from the unattainable region for countercurrent flow. The envelope can be obtained by differentiating a one-parameter family of curves in the (j_g, j_f) plane, obtained by manipulation from the one-dimensional continuity and momentum equations, with respect

to the parameter. The parameter is usually the void fraction or the mean film thickness. Models based on this approach include the separated-cylinders model (Wallis, 1969), the drift-flux model (Wallis, 1969) and the separated-flow model (Bharathan et al., 1979; Dobran, 1981).

(3) Static equilibrium theories : This class is based on static equilibrium between gravity, shear stress and the aerodynamic pressure force exerted by the upward gas flow on the liquid, as shown by some form of a stationary liquid-gas interface. The flooding definitions in this are related to the postulated interfacial shapes. The models developed in this category include the stationary-wave model (Shearer and Davidson, 1965), the hanging film model (Wallis and Kuo, 1976) and the roll-wave model (Richter, 1981).

A summary of all the analytical models for vertical countercurrent flooding is given in Appendix A.

Empirical Correlations

A large number of flooding correlations based on experimental studies have been developed over the past 20 years. A majority of these correlations can be classified into two broad categories depending on the dimensionless parameter they are based on. The two dimensionless parameters are :

(a) The Wallis parameter.

(b) The Kutateladze number.

The Wallis parameter, introduced by Wallis (1961), is defined as :

$$J_k^* = j_k\left[\frac{\rho_k}{gd(\rho_l-\rho_g)}\right]^{1/2} \tag{2.1}$$

It represents the ratio of inertial force to hydrostatic force.

The Kutateladze number is defined as :

$$K_k = \overline{v}_k \left[\frac{\rho_k^{\,2}}{g\sigma(\rho_l - \rho_g)} \right]^{1/4} \tag{2.2}$$

The correlations belonging to the first category include Wallis (1961), Wallis (1962), Hewitt and Wallis (1963), Clift, et al.(1966), Hewitt (1977), and Dukler and Smith (1979). All of these are based on an equation of the form :

$$J_g^{*1/2} + mJ_f^{*1/2} = C \tag{2.3}$$

where m and C are experimentally determined constants which depend on fluid properties, particularly viscosity and the tube-end geometries. The value of m is usually found to lie between 0.8 and 1.0. The value of C varies from 0.7 to 1.0 depending on the tube-end geometries. Correlations of this type are among the most widely used flooding correlations for adiabatic systems.

Correlations based on the Kutateladze number include Tobilevich, et al.(1968), Pushkina and Sorokin (1969), Alekseev, et al. (1972), and Chung, et al. (1980). Tobilevich, et al. used the number in their study of flow regimes in evaporating equipment. In 1969, Pushkina and Sorokin used an expression similar to Eqn. 2.3, viz.

$$K_g^{1/2} + mK_f^{1/2} = C \tag{2.4}$$

to correlate their data. They found m = 0 and C = 1.79 for the critical gas velocity. Chung, et al. (1980) also used Eqn. 2.4 and found empirically that m = 0.65 to 0.80, with C being a function of the dimensionless diameter d*, which is defined as :

$$d^* = d\left[\frac{g(\rho_l - \rho_g)}{\sigma} \right]^{1/2} \tag{2.5}$$

Note the similarity between Eqn. 2.3 and Eqn. 2.4, in view of the following relationship between the two parameters :

$$K_k = d^{*1/2} J_k^* \tag{2.6}$$

Alekseev, et al. (1972) correlated their data on flooding in regular packings for the water-vapor - water system and also data on several other systems obtained from different investigators, with an equation of the form :

$$K_{g,crit} = C Fr^{E1} Bo^{E2} \tag{2.7}$$

for the critical gas-phase Kutateladze number. In this equation C, E1 and E2 are empirical constants. Alekseev found C = 0.2576, E1 = -0.22 and E2 = 0.26. Fr is the Froude number, defined as :

$$Fr = \frac{W_l g^{0.25} \left(\rho_l - \rho_g\right)^{0.75}}{\sigma^{0.75}} \tag{2.8}$$

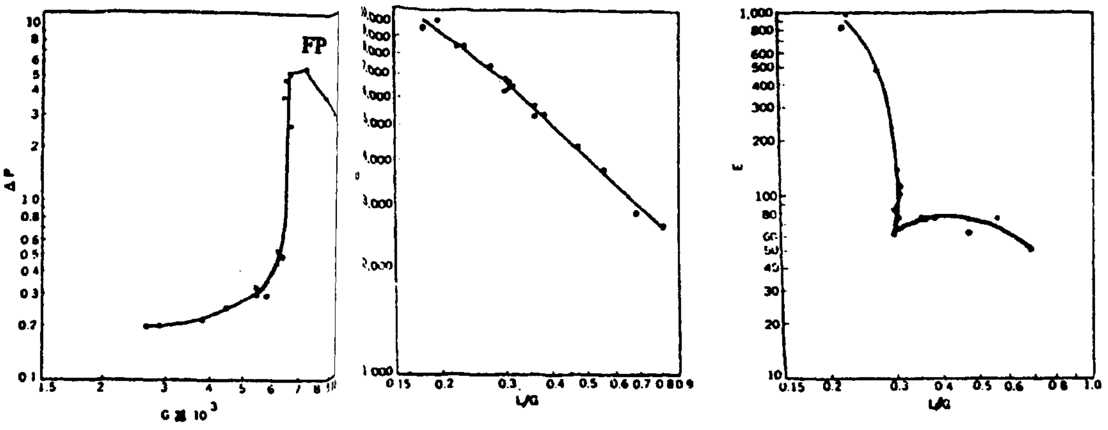
Bo is the Bond number, defined as :

$$Bo = \frac{d^2 \left(\rho_l - \rho_g\right) g}{\sigma} \tag{2.9}$$

Besides the conventional Wallis and Kutateladze numbers, correlations have been developed based on Reynolds number, Froude number and Weber number also. For a summary of all empirical flooding correlations, including those on reflux condensers, see Appendix B.

Flooding Correlations for Reflux Condensers

One of the first papers on flooding in reflux condensers is that of English, et al. (1963). These workers studied the flooding phenomenon associated with pure component condensation of four fluids, viz., water, carbon tetrachloride, n-propyl alcohol and n-heptane. Experiments were performed in single vertical 0.75 in. OD stainless steel tubes with ends cut at 0, 30, 60 and 75 degrees to the horizontal, at the vapor inlet. The data gathered were plotted in three ways : ΔP vs. G, G vs. L/G and entrainment mass flowrate, E vs. L/G. A sample graph for each of these types is shown in Fig. 2.3. A distinct break was noted in each graph, associated with the flooding point. English, et al. chose to define their flooding point as the second break point on the pressure drop vs. G curve (equivalent



(a) (b) (c)

Fig. 2.3(a) Pressure drop (in. of water) vs. superficial gas mass flowrate (lb/hr.ft^2)
Fig. 2.3(b) Gas flowrate (lb/hr.ft^2) vs. liquid-to-gas ratio
Fig. 2.3(c) Entrainment mass flowrate (lb/hr.ft^2) vs. liquid-to-gas ratio
(n-propyl alcohol with 75^{o} diagonally cut tube end, English et al., 1963)

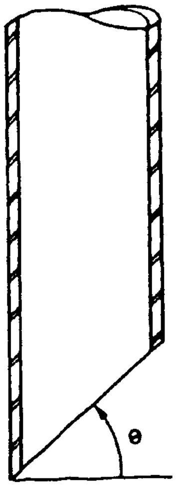


Fig. 2.4 Angle of taper at tube end

to point M of Fig. 2.2). They correlated their data with the following equation (in SI Units), to within ±10% accuracy.

$$j_g = 0.286\left\{\frac{d^{0.322}\rho_l^{0.419}\sigma^{0.097}}{\rho_g^{0.462}\eta_l^{0.150}j_f^{0.075}}\right\} \tag{2.10}$$

A taper on the tube end as in Fig. 2.4 permitted higher vapor flowrates before flooding. The increases noted in the study were 5, 25, and 54% for taper angle values of 30, 60 and 75 degrees respectively. The latter information is an important contribution of this study.

The paper by Diehl and Koppany (1969) is the next significant contribution to flooding in reflux condensers. They developed a simple correlation, based on their data on single component condensation and two-component wetted wall systems and also data from several other investigators (See Diehl and Koppany - Table 1). A broad range of physical properties, tube sizes and operating conditions was covered. The correlation is dimensional and can be written as follows :

$$j_g = F_1F_2\left(\frac{\sigma}{\rho_g}\right)^{½} \tag{2.11}$$

where $$F_1 = \left(\frac{d}{\sigma/80}\right)^{0.4} \qquad \text{if } F_1 < 1 \tag{2.12}$$

$$F_1 = 1 \qquad \text{otherwise}$$

and $$F_2 = \left(\frac{\rho_g j_g}{\rho_l j_l}\right)^{0.25} \tag{2.13}$$

As evident from the correlation, two physical properties, σ and ρ_g, and two design factors were noted to affect the flooding vapor velocity. The design factors include the effect of tube inside diameter when smaller than a certain critical diameter and the effect of the ratio of liquid to gas flowrates.

The unique feature of this correlation is the critical diameter effect. From their data on reflux conditions and the data of Verschoor (1938) on adiabatic conditions, the authors found that there exists a certain critical diameter above which the flooding vapor velocities are independent of the tube diameter. They developed a simple dimensional relationship to estimate the critical diameter as :

$$d_{crit} = \sigma/80 \tag{2.14}$$

where σ is in dynes/cm and d_{crit} is in inches.

This correlation is widely accepted today because it is fairly simple and is backed by a substantial database.

Besides the above two correlations for reflux condensers, a number of correlations have been developed for flooding with vapor condensation by a simple analytical extension of correlations of the type of Eqns. 2.3 and 2.4. The latter correlations are different in the sense that they are based on experiments in which both the liquid and its vapor are injected into the system. So, the phenomenon occurring in the above systems is not strictly reflux condensation, i.e. the liquid flowing in the system is not a result of vapor condensation alone. The effect of vapor condensation has been incorporated in these correlations by accounting for the reduction in the vapor flowrate due to condensation. It has been done assuming that the latent heat due to condensation is balanced by the sensible heat needed to raise the exit temperature of the liquid to saturation temperature. The dimensionless vapor inlet flow rate in Eqns. 2.3 and 2.4 has been replaced by the effective dimensionless vapor inlet flow rate. For example, Block and Crowley (1975) gave the following correlation :

$$\left(J^*_{g,eff}\right)^{1/2} + m\left(J^*_l\right)^{1/2} = C \tag{2.15}$$

in which $J^*_{g,eff}$, the effective dimensionless vapor inlet flow rate is given as :

$$J^*_{g,eff} = J^*_g - c\ Ja\ J^*_{l,in} \tag{2.16}$$

where Ja is the Jakob number defined as

$$Ja = \left(\frac{\rho_l}{\rho_g}\right)^{1/2} \frac{C_{pl}\left(T^{sat} - T_l\right)}{\Delta h_{lg}} \tag{2.17}$$

Tien (1977) suggested a similar correlation based on the Kutateladze number as :

$$\left(K_g - c\ Ja\ K_l\right)^{1/2} + mK_l^{1/2} = C \tag{2.18}$$

The correlation of Block and Crowley (1975) has been tested against steam-water data while Eqn. 2.18 has not been examined against any flooding data.

Particular attention has been focused on the evaluation of the coefficients m and C. Attempts in this direction have been that of Rothe and Crowley (1978) and Cudnik, et al.(1978). Wallis, et al.(1980) carried out experiments on the flooding characteristics of countercurrent steam-water annular flow in a vertical tube for various water temperatures. It was found that the flooding point, defined as the boundary on which water penetration is limited by the upward steam flow, can be described by Eqn. 2.3 with $C = 0.69 - 0.8$, depending on the tube-end condition, with $m \approx 1$ usually.

Comparison and Discussion

A number of articles have been published in recent years studying the different correlations, comparing them against each other and against experimental data. These include Deakin (1977), Bankoff and Lee (1983) and McQuillan and Whalley (1985). These studies have identified many factors which affect the flooding phenomenon. They are :

(a) Tube end conditions.

(b) Tube diameter.

(c) Tube length.

(d) Fluid properties, specifically liquid viscosity and surface tension.

(e) Phase change at the two-phase boundary. (e.g. condensation)

(f) Tube inclination.

(g) Relative values of liquid and gas mass flow rates, or in other terms, the liquid film thickness relative to the tube diameter.

While the effects of some of these factors like tube diameter are clear and well accepted by many investigators, the effects of other factors, viz. tube-end conditions, tube length, etc., are still far from understood.

A flooding correlation, whether developed analytically or empirically, is characterized by the flooding definition it is based on, the tube-end conditions it is related to, the influencing factors it has taken into account with their range considered and the way the influence of the various factors has been accounted for. Differences in the above aspects bring about differences in the flooding correlations. The following is a discussion of the above characteristics of flooding correlations.

Flooding-point Definition

As pointed out earlier, several definitions have been employed to describe the flooding point. It is difficult to ascertain whether these definitions are equivalent to each other. But, it should be noted that most of the commonly accepted and popular correlations are based on definitions which are quite closely related to each other. The common definitions employed include appearance of large disturbance waves at the two-phase interface, liquid bridging, sudden rise in the pressure drop and sudden rise in liquid entrainment. It is easy to see how these definitions could be describing the same or almost the same flooding point. So, as far as the widely used correlations are concerned, it can be taken that the differences in the definition do not affect the flooding velocity significantly.

Tube Entrance and Exit Conditions

Tube-end conditions include not only the geometries of the tube-ends but also the manner in which gas or liquid is introduced or removed from the tube. The different tube-end geometries used by investigators are shown in Fig. 2.5. The tube inlet geometries of English, et al. (1963) and Wallis (1961) are the ones most commonly found in reflux

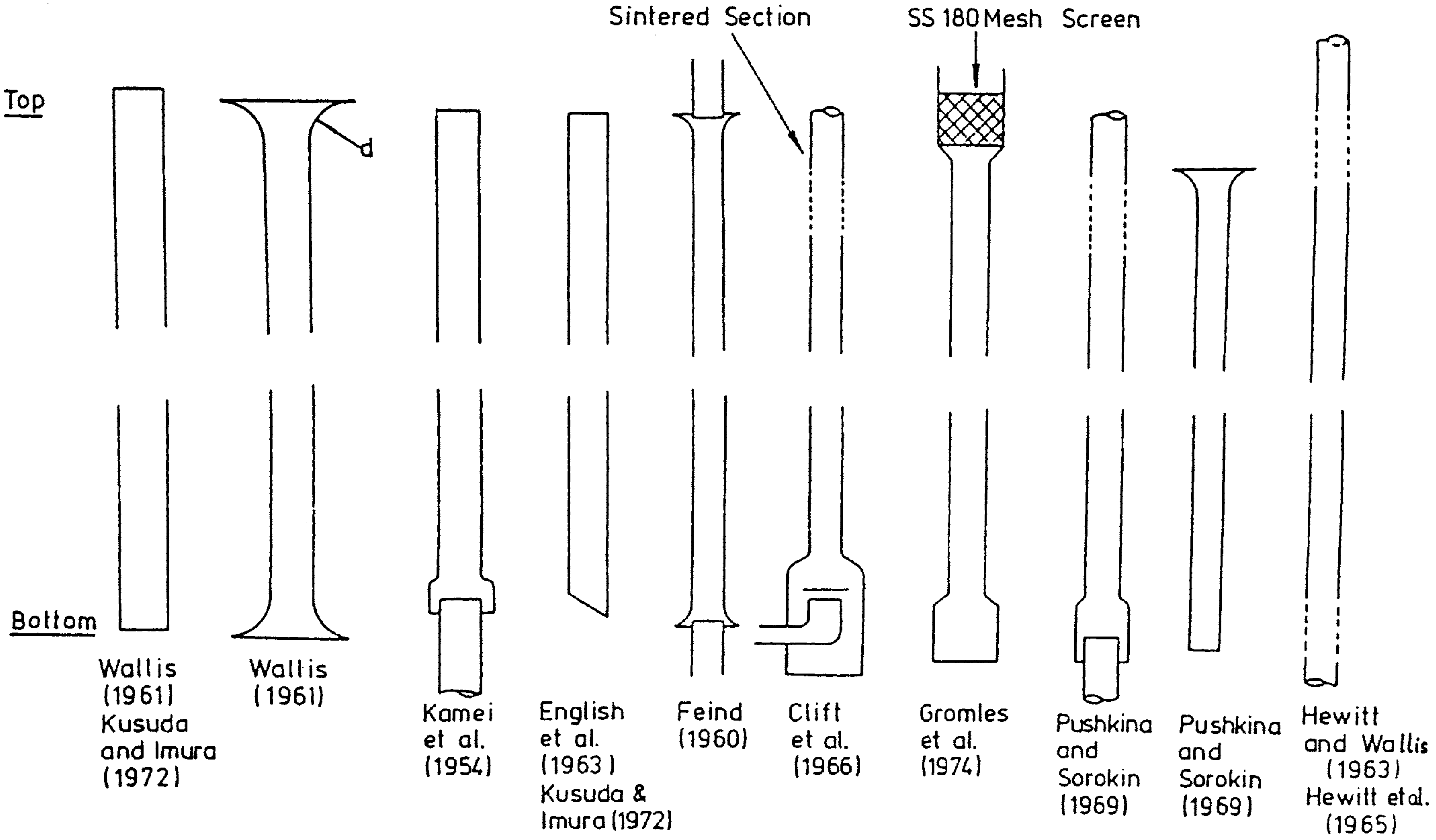


Fig. 2.5 Geometries of tube ends used in flooding studies (Imura et al. 1977)

condensers. It is evident from the figure that many different methods of gas and liquid injection and removal were used. Though investigators have ascertained that tube-end conditions have a considerable effect on the flooding velocities, their effect has not been properly incorporated in the correlations because no parameter which can uniquely describe the end conditions has been determined. Thus, most of the correlations account for the effect implicitly, making them specific for the end condition/(s) on which they are based, which in turn are not properly quantifiable.

In the correlations of the type of Eqns. 2.3 and 2.4, the constants m and C assume different values for different end conditions. However, it should be noted that these constants also account for variation in other influencing factors like fluid viscosity and tube diameter.

Other Factors Influencing the Flooding Phenomenon

(a) Tube diameter : The effect of tube diameter on flooding behavior has been found to differ for large and small tubes. It has been determined that the flooding gas velocity is independent of the tube diameter for dimensionless diameters d^* larger than approximately 40. Not all correlations account for this effect properly. All the Wallis type correlations account for tube diameter effect for $d^* < 40$. For $d^* > 40$, correlations based on the Kutateladze number work well. The correlation of Diehl and Koppany (1969) accounts for the effect for all d^*.

(b) Fluid properties, particularly liquid viscosity and surface tension : Liquid viscosity and surface tension have been noted to have opposite effects on the flooding velocity. An increase in liquid viscosity has been seen to decrease the flooding velocity. While some investigators have noted a significant effect, others like Hewitt (1977) believe that the effect is small. An increase in surface tension tends to increase the flooding velocity.

Though most researchers support this view, some like Suzuki and Ueda (1977) found no similar trend in their data.

(c) Phase change (condensation) : Though a condensing countercurrent flow system differs widely in operational characteristics from an adiabatic system, the effect of condensation on the flooding phenomenon is not clearly understood. It has been noted that condensing flow does not differ much from adiabatic flow (Bankoff and Lee, 1983) when flooding takes place at the bottom of the tube and the exit liquid is close to saturation temperature as in a reflux condenser.

The empirical correlations developed specifically for reflux condensers, e.g., English, et al.(1963) and Diehl and Koppany (1969), account for the effect of condensation implicitly. Other correlations for condensing cases have accounted for the effect by using an effective vapor flux to describe the reduction in the vapor flow up the tube.

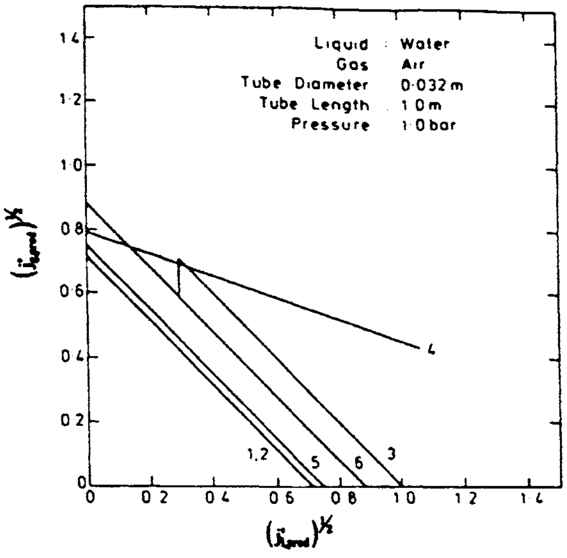
It is worthwhile to note here that it is extremely difficult to individually determine the effect of each factor on the flooding phenomenon as it is not always possible to vary one factor alone while keeping all other factors constant.

The next objective was to compare the various flooding correlations with each other to devise a suitable strategy for determining the flooding velocity accurately for a wide range of flow conditions. The criteria for the comparison were simplicity of use, versatility of application and accuracy of prediction. With the first criterion, some of the analytical correlations like Shearer and Davidson (1965) and Cetinbudaklar and Jameson (1969) were eliminated from consideration as they require the use of complicated solution techniques.

For comparing the remaining correlations with the latter two criteria, the comparison study of McQuillan and Whalley (1985) was primarily utilized. These workers compiled a data bank of 2762 experimental flooding data points from 24 different sources of data and used it to test the performance of 17 empirical and 5 theoretical flooding correlations. Their data bank is biased towards air-water flow (68% of the data) and against flow in large diameter tubes (78% of the data is for tubes of diameter less than 50mm.). When compiling the data bank, the authors were very careful in using the same conversion methods as the original work so as to bring all the data to the same form. The authors ignored differences in the flooding point definition and the tube-end conditions when comparing the correlations with the contents of the data bank as it was not possible to devise criteria by which the above factors could be suitably accounted for.

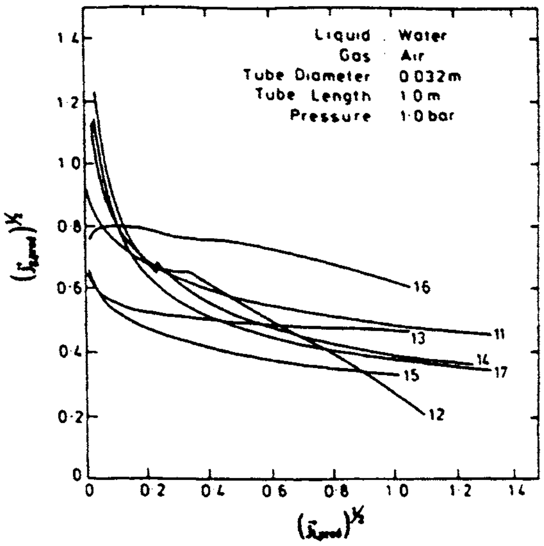
Figs. 2.6 - 2.9, as taken from McQuillan and Whalley (1985), show the flooding curves predicted by 22 different correlations for air-water flow in a 0.032 m diameter tube. The discrepancies between the correlations are considerable, even for correlations using a particular dimensionless group. A comparison among the correlations for reflux condensers alone, together with the generalized Wallis (1961) correlation, is shown in Figs. 2.10 and 2.11 (Deakin, 1977). The Andale correlation shown in these figures cannot be found in the open literature but it has been listed by English, et al. (1963), Deakin (1977) and Diehl and Koppany (1969) as a flooding correlation commonly used in industrial reflux condenser design. The poor agreement between the correlations is evident again in these figures. Since no fixed pattern can be observed in the figures to attribute to the discrepancies, it can probably be said that none of the 'quantifiable' factors like tube diameter, fluid properties or phase change are individually responsible for the differences in the graphs. A combined effect of the differences in accounting for these quantifiable factors might be significant but it is more likely that the random and unquantifiable factor of tube-end condition will be the main culprit responsible for the wide discrepancy.

The statistical quantities used by McQuillan and Whalley (1985) to evaluate the overall predictive ability of the correlations were number of data points that could be reasonably represented by the correlation, weighted percentage error (WE), weighted root mean



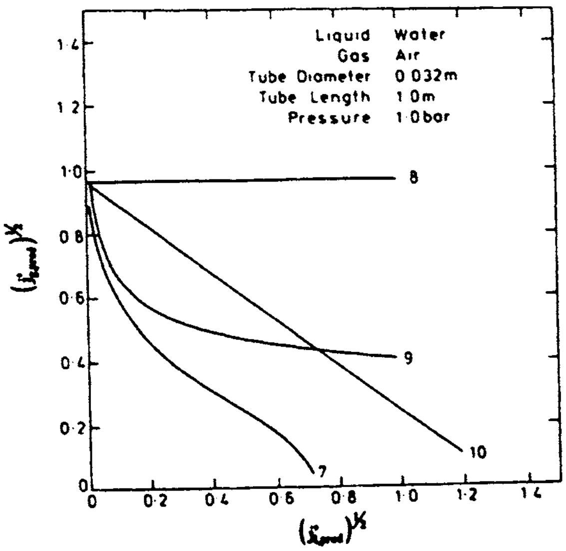
Correlation Number 1	Wallis (1961)
Correlation Number 2	Wallis (1962)
Correlation Number 3	Hewitt and Wallis (1963)
Correlation Number 4	Clift et al (1965)
Correlation Number 5	Hewitt (1977)
Correlation Number 6	Dukler and Smith (1979)

Fig. 2.6 Empirical flooding correlations using dimensionless superficial velocities



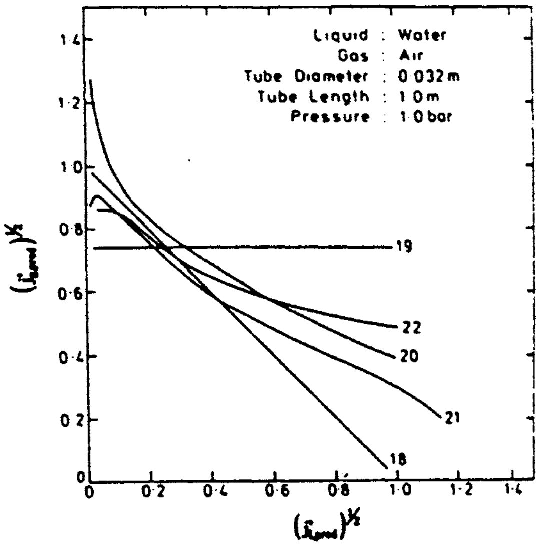
Correlation Number 11	Kamei et al (1954)
Correlation Number 12	Feind (1960)
Correlation Number 13	English et al (1963)
Correlation Number 14	Diehl and Koppany (1969)
Correlation Number 15	Grolmes et al (1974)
Correlation Number 16	Suzuki and Ueda (1977)
Correlation Number 17	Machej and Sokol (1979)

Fig. 2.8 Other empirical flooding correlations



Correlation Number 7	Tobilevich et al (1968)
Correlation Number 8	Pushkina and Sorokin (1969)
Correlation Number 9	Alekseev et al (1972)
Correlation Number 10	Tien et al (1979)

Fig. 2.7 Empirical flooding correlations using Kutateladze number



Correlation Number 18	Wallis (1969)
Correlation Number 19	Wallis and Kuo (1976)
Correlation Number 20	Imura et al (1977)
Correlation Number 21	Bharathan et al (1978)
Correlation Number 22	Richter (1981)

Fig. 2.9 Theoretical flooding correlations

(Figures 2.6 - 2.9, taken from McQuillan and Whalley, 1984)

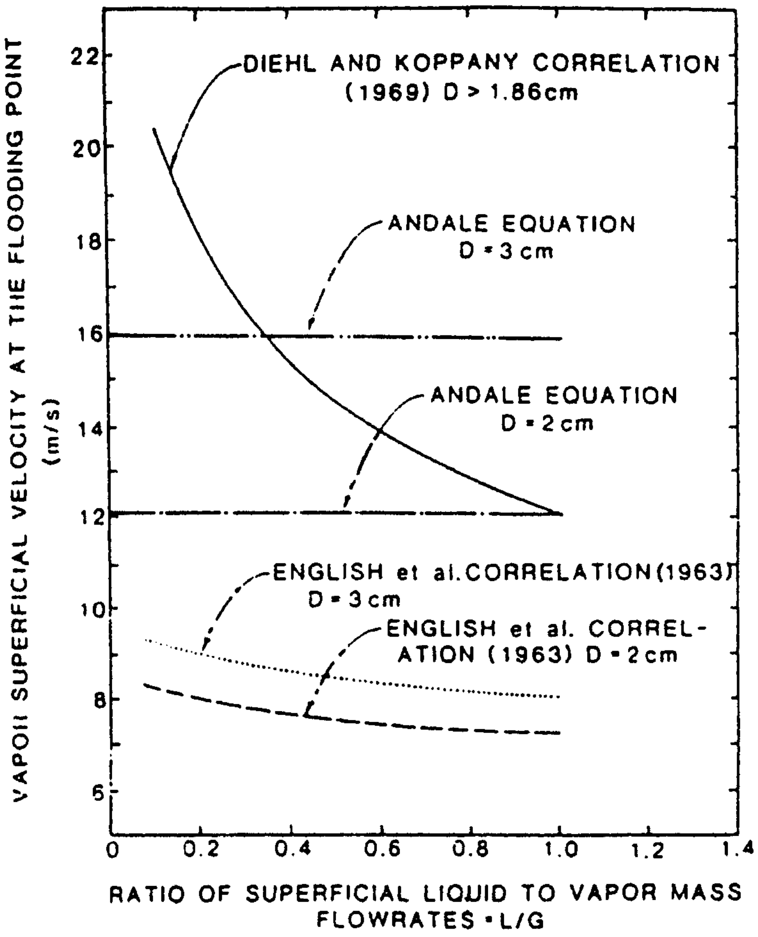


Fig. 2.10 A comparison of flooding correlations for reflux condensers (Condensing fluid = steam, P = 1 bar (abs), d_i = 10mm.) (Deakin, 1977)

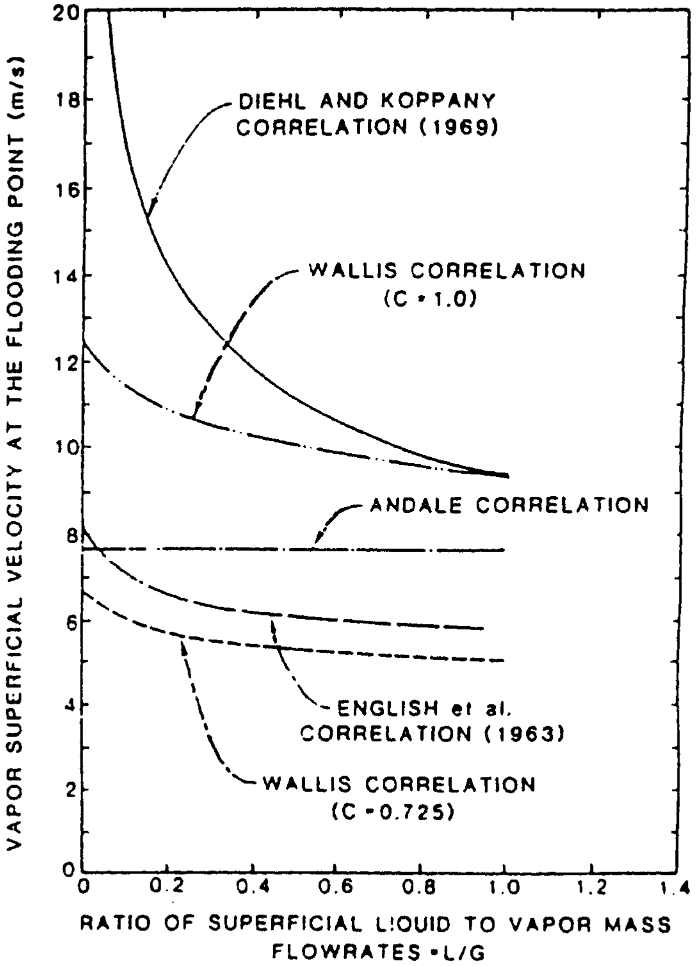


Fig. 2.11 Further comparisons of flooding correlations for reflux condensers (Condensing fluid = steam, P = 1 bar (abs)) (Deakin, 1977)

square error (WRMSE) and average deviation (AD). The weighted percentage error (WE) was defined as :

$$WE = \left[\frac{1}{n} \sum \left\{ \exp \left| \ln \left(\frac{j^{*}_{g,calc}}{j^{*}_{g,expt}} \right)^{1/2} \right| - 1.0 \right\} \right] * 100\% \qquad (2.19)$$

Similarly, the weighted root mean square error (WRMSE) and the average deviation (AD) were defined as :

$$WRMSE = \left[\frac{1}{n} \sum \left\{ \exp \left| \ln \left(\frac{j^{*}_{g,calc}}{j^{*}_{g,expt}} \right) \right| - 1.0 \right\} \right]^{1/2} * 100\% \qquad (2.20)$$

$$AD = \left[\frac{1}{n} \sum \left\{ \left(\frac{j^{*}_{g,calc}}{j^{*}_{g,expt}} \right)^{1/2} - 1.0 \right\} \right] * 100\% \qquad (2.21)$$

WRMSE should be low for low scatter in the data and AD should be ideally equal to zero.

The study found that empirical correlations were generally more successful than the theoretical correlations. Also, the correlations based on dimensionless superficial velocities were noticeably less successful than the other empirical correlations, particularly for high liquid velocities and for non air-water systems. The most successful correlation was found to be that of Alekseev, et al. (1972) which gave a weighted percentage error, WE = 28% with WRMSE = 46% and AD = 0.07%. English, et al. (1963) was the second best correlation which gave WE = 30% with WRMSE = 44% and AD = -0.07%. The correlation of Fiend, et al. (1960) was found to be equally accurate compared to the above correlations (WE = 28%, WRMSE = 45%, AD = 0.15%) but it could represent only 2619 data points out of a total of 2762 data points. A modified form of the correlation of Bharathan, et al. (1979) was noted as the most successful theoretical correlation. The Diehl and Koppany (1969) correlation was not as consistent and accurate in its predictions as any of the above correlations. The predictions of the correlations of Grolmes, et al. (1974) and Hewitt and Wallis (1963) were comparable to that of Diehl and Koppany (1969).

The authors found that Alekseev, et al. (1972) over-predicted the flooding flowrates for high liquid viscosities. They corrected this deficiency of the correlation by a small modification, giving the new correlation :

$$K_g = 0.286 Bo^{0.26} Fr^{-0.22} \left\{ 1 + \frac{\eta}{\eta_{water}} \right\}^{-0.18} \qquad (2.22)$$

The performance of this modified correlation was evaluated against the entire data bank. It was found to give the most successful prediction of the data with WE = 26%, WRMSE = 40% and AD = 0.0%.

Recommended Procedure for Predicting Flooding Velocity in a Reflux Condenser

The strategy for calculating the flooding vapor velocity for any reflux condenser design situation is as follows :

(a) First, compare the designed inlet geometry and the method of vapor injection to be used with the tube end conditions used in the literature. Pick out the correlation developed specifically for similar tube end conditions, irrespective of whether the correlation is for reflux condensation or adiabatic flow. Determine the flooding vapor velocity using this correlation.

(b) Use the modified Alekseev, et al. correlation, as proposed by McQuillan and Whalley (1985) to determine the flooding vapor velocity. This correlation was chosen for being the most successful correlation out of 23 flooding correlations, in predicting a wide range of flooding conditions, as determined by McQuillan and Whalley (1985).

(c) Use the English, et al. (1963) correlation to determine the flooding vapor velocity. This correlation was chosen as it was developed specifically for reflux condensers and also was found to be as accurate as the original Alekseev, et al. (1972) correlation in predicting a wide range of flooding conditions.

(d) Determine the flooding vapor velocity using the Diehl and Koppany (1969) correlation also, as it is the most widely used correlation today for reflux condenser problems.

(e) Compare the values obtained in parts (a), (b), (c) and (d) and take the lowest value as the flooding velocity.

CHAPTER III

FLUID MECHANICS AND HEAT TRANSFER

The objective of this chapter is two-fold :

(i) to survey the different correlations available in the literature to predict the fluid mechanics and the heat transfer aspects of the reflux condensation process.

(ii) to devise a suitable method of predicting the heat transfer coefficients in the vapor core and in the condensate film.

It is necessary to understand the fluid mechanics of any problem before its heat transfer aspects can be studied. Following is a discussion of the fluid mechanics of the reflux condensation process.

Fluid Mechanics

The hydrodynamics of vertical countercurrent two-phase flow was discussed to a certain extent in Chapter II. It was seen how the flow pattern changed as the gas/vapor flowrate was increased gradually till upward cocurrent annular flow was established. Refer to Fig. 2.1. Other observations made include :

(a) variations in the pressure drop across the tube, and

(b) variations in the liquid entrainment rate

However, the other key parameters of any gas-liquid flow, including mean film thickness and the interfacial shear stress, were not discussed.

The mean film thickness is an important parameter of countercurrent two-phase flow. The effect of a cocurrent or a countercurrent gas flow on the mean film thickness has been studied by several investigators including Hewitt and Wallis (1963), Collier and Hewitt (1964) and Hawley and Wallis (1982). Their results indicate that the mean film thickness in cocurrent two-phase flow tends to decrease as the gas flowrate is increased, owing to the interfacial shear, while it has an opposite effect in countercurrent two-phase flow.

Hawley and Wallis (1982) observed that if a steady countercurrent flow was maintained before the onset of flooding, the effect of interfacial shear on the mean film thickness was not significant. This was because in this region the interfacial shear was sufficiently small compared to the wall shear stress. Their data on the mean film thickness at low liquid Reynolds numbers and for low gas flows was seen to be well approximated by the traditional Nusselt (1916) equation :

$$\partial^* = 2.289\,\mathrm{Re}_l^{1/3} \tag{3.1}$$

where the dimensional mean film thickness and the liquid Reynolds number are defined as:

$$\partial^* = \partial\left(\frac{\rho_l(\rho_l - \rho_g)g\sin\theta}{\eta_l^2}\right)^{1/3} \tag{3.2}$$

$$\mathrm{Re}_l = 4\frac{\Gamma_l}{\eta_l} \tag{3.3}$$

See Fig. 3.1.

Several correlations have been developed to determine the interfacial shear stress in adiabatic/condensing countercurrent two-phase flow. In an adiabatic flow, the interfacial shear stress can be represented, similar to single-phase flow, by the interfacial friction factor defined by :

$$f_{i,a} = \frac{2\tau_{i,a}}{\rho_g\left(\bar{v}_g - v_i\right)^2} \tag{3.4}$$

where the interface velocity v_i is generally small compared to the mean gas velocity $\bar{v}_g$ and thus is often ignored. Wallis (1969) correlated the interfacial friction factor based on air-water data in wavy annular flow as :

$$f_{i,a} = 0.005\left(1 + 300\frac{\partial}{d}\right) \tag{3.5}$$

This can be approximated for thin films in circular pipes as :

$$f_{i,a} = 0.005\left(1 + 75(1-\varepsilon)\right) \tag{3.6}$$

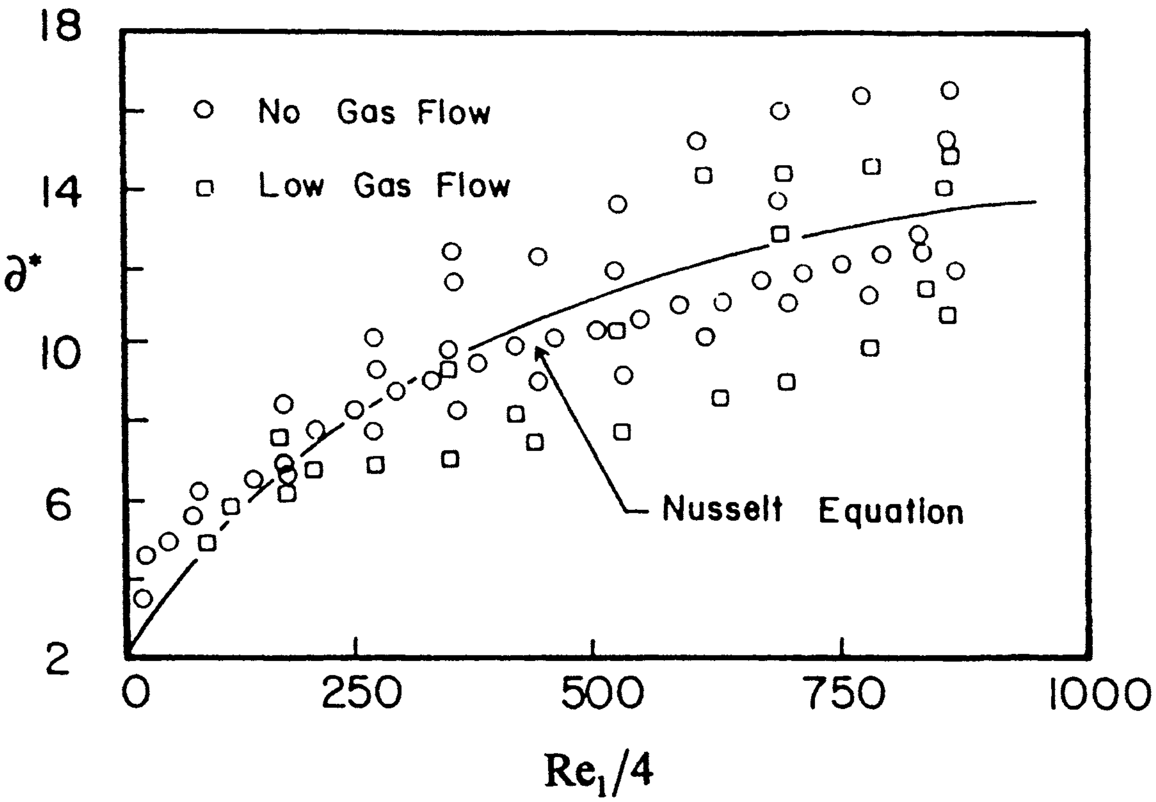


Fig. 3.1 Typical mean thickness data of a vertically falling film with low countercurrent gas flow (Hawley and Wallis, 1982)

An alternative empirical correlation was proposed by Bharathan et al. (1979) viz.

$$f_{i,a} = 0.005 + A\left(\frac{d^*}{2}\right)^B \left(1-\sqrt{\varepsilon}\right)^B \tag{3.7}$$

where A and B are constants given as :

$$\log A = -0.56 + \frac{9.07}{d^*} \tag{3.8}$$

$$B = 1.63 + \frac{4.74}{d^*} \tag{3.9}$$

The dimensionless diameter d^* is defined as :

$$d^* = d\left[\frac{g(\rho_l - \rho_g)}{\sigma}\right]^{1/2} \tag{3.10}$$

In condensing flow, the interfacial shear stress is altered due to the momentum transfer from the vapor to the liquid phase at the vapor-liquid interface. Silver and Wallis (1965) attempted to account for the effect of condensation by using the Reynolds flux concept in the vapor phase and derived the following equation :

$$\frac{f_i}{f_{i,a}} = \frac{\phi}{f_{i,a}} + \exp\left(-\frac{\phi}{2f_{i,a}}\right) \tag{3.11}$$

where ϕ and f_i are defined as :

$$\phi = \frac{2}{\rho_g \bar{v}_g}\left|\frac{d\Gamma_g}{dz}\right| \tag{3.12}$$

$$f_i = \frac{2\tau_i}{\rho_g \bar{v}_g^{\,2}} \tag{3.13}$$

Mickley et al.(1954) used the film theory to develop the equation :

$$\frac{f_i}{f_{i,a}} = \frac{\phi/f_{i,a}}{1-\exp(-\phi/f_{i,a})} \tag{3.14}$$

It is interesting to note that both Eqns. (3.13) and (3.14) indicate that f_i is approximately equal to ϕ when ϕ is large. Equating Eqns. (3.12) and (3.13) for large ϕ, the following expression is obtained for τ_i :

$$\tau_i = \bar{v}_g\left|\frac{d\Gamma_g}{dz}\right| \tag{3.15}$$

which means that the interfacial shear stress is equal to the product of the condensation rate and the mean vapor velocity when the condensation effect is dominant (large ϕ). Linehan (1968) proposed a linear equation in ϕ based on this result, postulating that the adiabatic shear stress is augmented by an amount exactly equal to the condensation rate times the mean vapor velocity :

$$\frac{f_i}{f_{i,a}} = 1 + \frac{\phi}{f_{i,a}} \tag{3.16}$$

and

$$\tau_i = f_{i,a}\frac{\rho_g \bar{v}_g^{\,2}}{2} + \bar{v}_g\left|\frac{d\Gamma_g}{dz}\right| \tag{3.17}$$

The above correlations for estimating the mean film thickness and the interfacial shear stress in a vertical countercurrent two-phase flow -- adiabatic or condensing -- situation will be used later in the chapter.

Heat Transfer

The heat transfer part of the reflux condensation problem consists of determining suitable methods of estimating the heat transfer coefficient at any point along the condenser tube both in the vapor core and in the condensate film. The latter case of estimating the condensing heat transfer coefficient is discussed first.

Condensate Film

A survey of the literature shows that no significant experimental effort has yet been made of measuring the condensing heat transfer coefficient in a reflux condensation situation and correlating the data with an equation. In this chapter, all the correlations available in the literature to predict film condensation on a vertical surface are studied. Then, based on them, a suitable strategy of predicting the local condensing heat transfer coefficient for a reflux condensation situation is developed.

The problem of film condensation on a vertical surface was first analyzed by Nusselt (1916a,b). The main assumptions of his model were :

(1) The condensate film is in creeping laminar flow

(2) Only gravity forces are acting on the film.

He gave the following expression for the local coefficient, α_z, at a distance z from the top of the plate or tube :

$$\alpha_z = \left[\frac{\lambda_l^3 \rho_l \left(\rho_l - \rho_g\right) g \Delta h_{lg}}{4 \eta_l \left(T_{sat} - T_w\right) z} \right]^{1/4} \tag{3.18}$$

In terms of the condensate Reynolds number, the local coefficient is given by :

$$\frac{\alpha_z}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l \left(\rho_l - \rho_g\right) g} \right]^{1/3} = 1.1 \mathrm{Re}_{c,z}{}^{-1/3} \tag{3.19}$$

where $\mathrm{Re}_{c,z}$ is given by Eqn. (3.3).

Kutateladze (1963) suggested a correction to the above equation to determine α_z more accurately in the laminar wavy region, i.e. for $\mathrm{Re}_{c,z} > 30$. The resulting equation is :

$$\frac{\alpha_z}{\lambda_l} \left[\frac{\eta_l^2}{\rho_l \left(\rho_l - \rho_g\right) g} \right]^{1/3} = 0.756 \mathrm{Re}_{c,z}{}^{-0.22} \tag{3.20}$$

This equation may be used till the onset of turbulence in the condensate film. For films falling under gravity, there is some controversy as to the Reynolds number at which turbulence begins. The general consensus is that turbulence starts at a Reynolds number of about 1600-1800.

Condensing heat transfer coefficients for turbulent falling films were first studied by Kirkbride (1934), who proposed an empirical correlation. Colburn (1934) contributed a more fundamental analysis of Kirkbride's data and presented his results in the form of a graph of the average coefficient against the condensate Reynolds number at the bottom of the tube. See Fig. 3.2. Labuntsov (1957) suggested the following semiempirical correlation for the local coefficient :

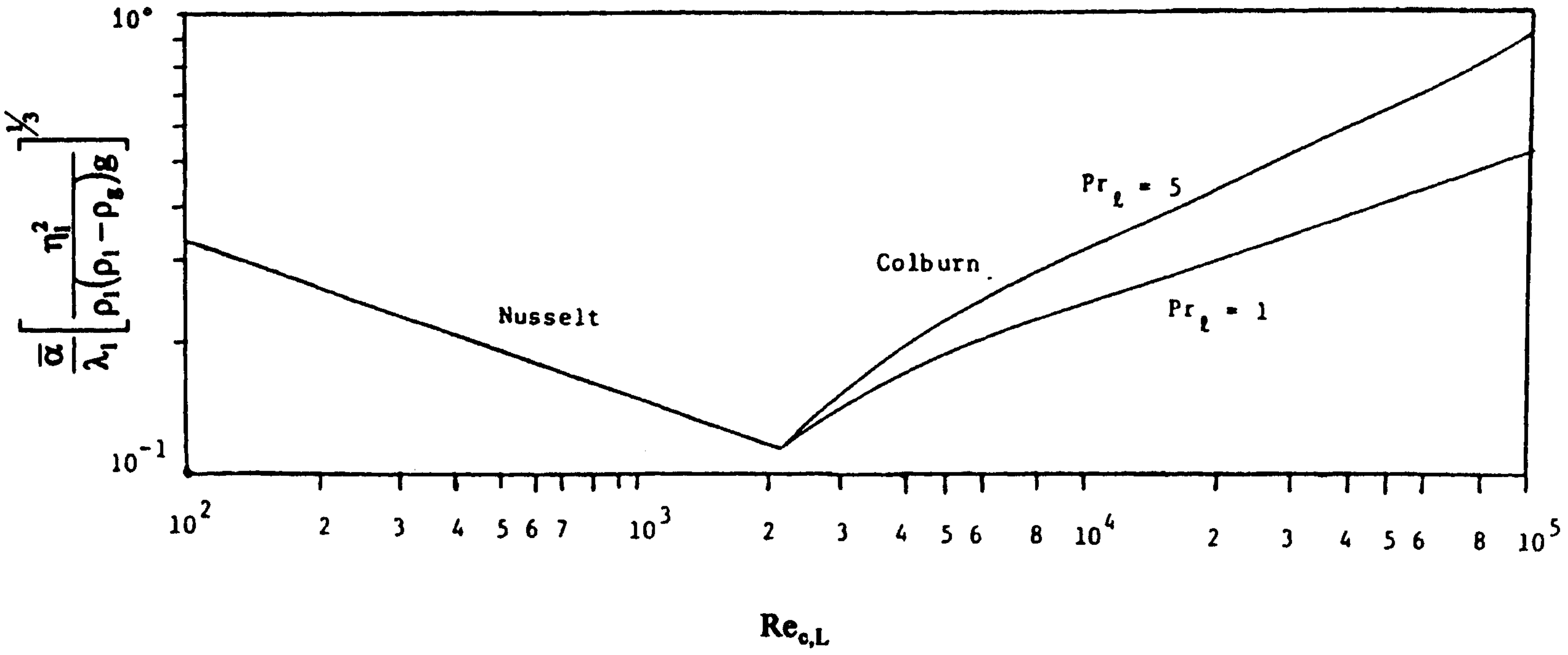


Fig. 3.2 Correlation for condensation on a vertical surface - No vapor shear

$$\frac{\alpha_z}{\lambda_l}\left[\frac{\eta_l^2}{\rho_l(\rho_l-\rho_g)g}\right]^{1/3} = 0.023\,\mathrm{Re}_{c,z}{}^{0.25}\,\mathrm{Pr}^{0.5} \tag{3.21}$$

The effect of interfacial shear on condensation was studied by several investigators. The earliest study was that of Nusselt (1916b) who derived a correlation for the heat transfer coefficient which applies only when the condensate is in laminar flow.

The next significant study was that of Carpenter and Colburn (1951). They obtained experimental data for local and average coefficients while condensing steam, methanol, ethanol, toluene and trichloroethylene inside a 0.459 in. ID, 8 ft. long vertical tube with inlet vapor velocities upto 500 ft/s downward. On the basis of their experimental data, they hypothesized that (a) due to vapor shear the condensate film becomes turbulent at much lower values of Reynolds number than in the absence of vapor shear, and (b) the major thermal resistance occurs in a laminar sublayer of the condensate film, whose thickness can be calculated from generalized velocity distributions developed for one-phase flow in pipes. By assuming a linear velocity profile in the laminar sublayer, they arrived at the following expression for the local heat transfer coefficient :

$$\alpha_z = 0.043\,\mathrm{Pr}_l^{1/2}\,\frac{\lambda_l \rho_l{}^{1/2} \tau_w{}^{1/2}}{\eta_l} \tag{3.22}$$

where τ_w is the shear stress in the laminar sublayer. This is the same as the wall shear stress due to the assumption of a linear velocity profile. The shear stress, τ_w was assumed to be affected by three factors -- gravity, momentum and friction -- and they were combined as follows :

$$\tau_w = \tau_g + \tau_m + \tau_f \tag{3.23}$$

The above relation between the forces for a cocurrent vapor-condensate flow situation is shown graphically in Fig. 3.3. τ_g, the gravity contribution was evaluated by an expression which required trial and error. τ_m was considered to be the momentum change in the condensing vapor and τ_f was based on the frictional pressure drop data of Carpenter

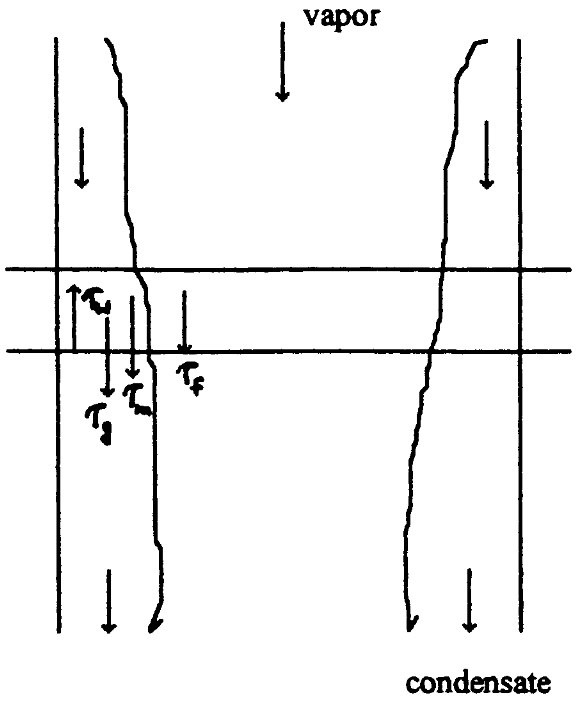


Fig. 3.3 Relation between τ_w, τ_g, τ_m and τ_f for cocurrent vapor-condensate flow

(1948). Reasonable agreement was found between the predictions of Eqns. (3.22) and (3.23) and experimental data.

Dukler (1960) studied the problem of fluid mechanics and heat transfer in thin liquid films in a detailed manner. He developed new equations for velocity distribution and film thickness, and from these, on the basis of the principles of the analogies, derived expressions for the local heat transfer coefficient. To determine the average coefficient, knowledge about the path of condensation along the tube is essential. Dukler devised a method of estimating the condensing path in a cocurrent downward flow condensation system with interfacial shear, for the special case of total condensation and a saturated inlet vapor. Based on this, he derived graphs of the average coefficient for several Prandtl numbers ranging from 0.1 to 1.0. Good agreement was found between his theory and the Carpenter (1948) data. The numerical work of Dukler was limited to the case of a vertical cocurrent downward flow falling-film system. It can be reworked for a countercurrent system, but the problem would be extremely involved mathematically and cannot be justified, considering the scope of this work.

Soliman et al. (1968) worked on the same lines as Carpenter and Colburn (1951) and presented a general heat transfer correlation for annular flow condensation. They derived new equations for the friction, momentum and gravity forces from a differential momentum balance analysis of an annular flow model and suggested ways of evaluating them. They also examined the relative magnitudes of contribution of the above forces to the wall shear stress, τ_w. Finally, a Carpenter-Colburn type correlation was suggested for the heat transfer coefficient viz. :

$$\alpha_z = 0.036\,\mathrm{Pr}_l^{0.65}\,\frac{\lambda_l \rho_l^{1/2} \tau_w^{1/2}}{\eta_l} \tag{3.24}$$

In this equation, the wall shear stress τ_w is to be evaluated from Eqn. (3.23) using the derived expressions for τ_g, τ_m and τ_f. This correlation was found to be in agreement with experimental data on both horizontal condensation and vertical downward condensation, over a range of vapor velocities from 20 ft/s to 1000 ft/s, a range of Prandtl numbers from 1 to 10, and a range of qualities from x=0.99 to x=0.03.

Other correlations developed for a vapor-shear dominated condensing situation include those by Traviss et al. (1971), and Boyko and Kruzhilin (1967).

All the correlations discussed above have been developed for a condensation process in which the vapor and the condensate flow together in the same direction. They may have to be modified before they can be applied to a reflux condensation problem. In this regard, the following observations on a reflux condensation flow situation would be very useful :

(1) Analogous to the analysis of Soliman et al. (1968) and Carpenter and Colburn (1951), the different forces acting on the condensate film of a reflux condensation process can be identified as shown in Fig. 3.4. They are related as :

$$\tau_w = \tau_g - (\tau_f + \tau_m) \qquad (3.25)$$

As evident, the forces of vapor shear and gravity act opposite to each other. They are both maximum at the bottom of the tube and they decrease continuously towards the top of the tube, where they are very small. The force due to momentum change, τ_m, does not vary much along the condenser tube. For a steady countercurrent two-phase flow to be maintained, the shear stress, τ_w should be positive, i.e., the force of gravity per unit area, τ_g, should be greater than the combined force per unit area due to vapor shear and momentum change, $(\tau_f + \tau_m)$, throughout the condenser tube. This means that τ_g should dominate through the entire condensation process. The condition at which $(\tau_f + \tau_m)$ becomes equal to τ_g, i.e. τ_w becomes zero, is equivalent to the onset of flooding.

(2) The gas flooding velocities observed in countercurrent annular two-phase flow for liquid to gas flowrate ratios (same as reflux ratio for a reflux condensation process) less than or equal to 1.0, range from close to zero to about 35 ft/sec. The higher flooding velocities are obtained for low values of L/G, i.e. low values of liquid flowrate at the bottom of the tube, which correspond to low rates of heat removal in

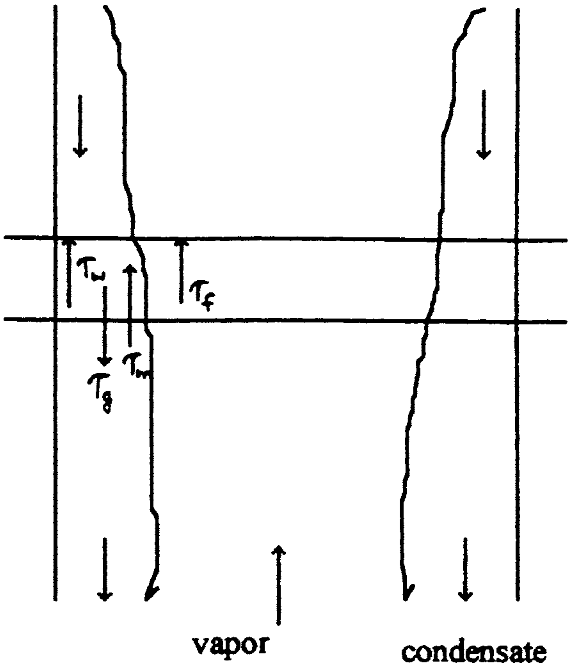


Fig. 3.4 Relation between τ_w, τ_g, τ_m and τ_f for reflux condensation

reflux condensation. Reflux condensers are normally operated with the vapor velocity well below the flooding velocity -- usually 0.6-0.7 times the flooding velocity -- so that a steady countercurrent operation can be maintained. The operating vapor velocities of reflux condensers [0 - 20 ft/sec] are usually small compared to the vapor velocities seen normally in vertical downward forced convection condensation processes. The data used by Soliman et al. (1968) to test the validity of their correlation covered a range of inlet vapor velocities from 20 - 1000 ft/sec.

The above points (1) and (2) suggest that vapor shear might be too low to affect the condensate film throughout the normal condensation process. However, since in a reflux condenser, the forces of vapor shear and gravity act opposite to each other and they are both maximum at the bottom of the tube, it is instructive to examine the relative magnitudes of the forces and then determine their relative importance accordingly. The values of the forces, τ_g, τ_f and τ_m can be estimated from expressions derived by a differential momentum balance analysis, similar to that of Soliman et al. (1968), of a countercurrent two-phase condensing flow situation. Following is a discussion of the derivation. The assumptions made include : (a) steady countercurrent flow, and (b) no radial pressure gradients.

Consider a differential element dz of the vapor core, as shown in Fig. 3.5. The momentum balance states that the summation of the forces acting on a control volume in the axial direction is equal to the difference between the rate of momentum leaving and entering the control volume, i.e.

$$\sum F_z = \sum (\dot{m}v)_{z,out} - \sum (\dot{m}v)_{z,in} \tag{3.26}$$

Applying this equation to the differential element of the vapor core gives :

$$-\frac{d}{dz}\left(PA_g\right) - \tau_{ig}S_g - \rho_g A_g g = \frac{d}{dz}\left(\bar{v}_g \dot{m}_g\right) - v_{gi}\frac{d\dot{m}_g}{dz} \tag{3.27}$$

Similarly, for the condensate layer shown in Fig. 3.5, the following equation arises :

$$-\frac{d}{dz}\left(PA_l\right) + \tau_{il}S_g + \tau_w S - \rho_l A_l g = \frac{d}{dz}\left(\bar{v}_l \dot{m}_l\right) - v_{li}\frac{d\dot{m}_l}{dz} \tag{3.28}$$

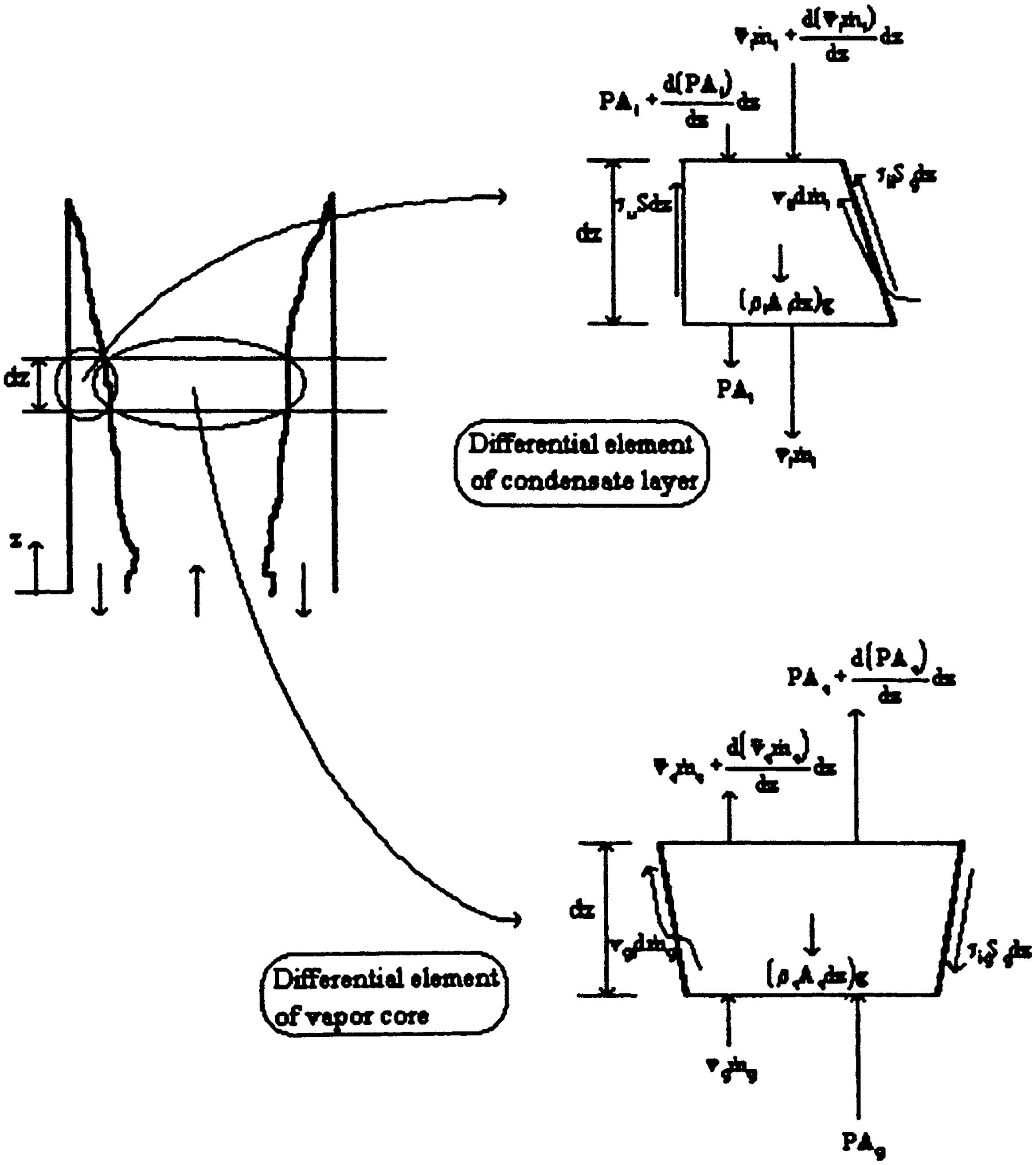


Fig. 3.5 Differential momentum balance analysis

If zero slip is assumed to exist between the vapor and the liquid at the vapor-liquid interface, then (i) $\tau_{ig} = \tau_{il} = \tau_i$, and (ii) $v_{gi} = v_{li}$. Further, since the vapor and the condensate flow in opposite directions, the velocities v_{gi} and v_{li} should be taken equal to zero for the zero-slip condition to be satisfied. If variations in A_l and A_g over the element dz are assumed to be negligible, then they can be taken out of the derivative in the first term of Eqns. (3.27) and (3.28). The resulting equations can, then be combined to eliminate the pressure gradient to give :

$$\tau_w = \left[(\rho_l - \rho_g)\frac{A_l g}{S}\right] - \left[\frac{\tau_i S_g}{S}\left(1 + \frac{A_l}{A_g}\right) + \left(\frac{A_l}{SA_g}\frac{d}{dz}(\dot{m}_g \bar{v}_g) - \frac{1}{S}\frac{d}{dz}(\dot{m}_l \bar{v}_l)\right)\right] \tag{3.29}$$

This equation is equivalent to Eqn. (3.25) such that :

$$\tau_g = (\rho_l - \rho_g)\frac{A_l g}{S} \tag{3.30}$$

$$\tau_f = \tau_i \frac{S_g}{S}\left(1 + \frac{A_l}{A_g}\right) \tag{3.31}$$

and

$$\tau_m = \left(\frac{A_l}{SA_g}\frac{d}{dz}(\dot{m}_g \bar{v}_g) - \frac{1}{S}\frac{d}{dz}(\dot{m}_l \bar{v}_l)\right) \tag{3.32}$$

τ_g represents the effect of the gravitational field, τ_f the effect of two-phase friction and τ_m the effect of momentum changes in the flow on the wall shear stress, τ_w. In terms of void fraction and tube diameter, the Eqns. (3.30), (3.31) and (3.32) can be written as :

$$\tau_g = (\rho_l - \rho_g)\frac{g(1-\varepsilon)d_i}{4} \tag{3.33}$$

$$\tau_f = \tau_i \frac{1}{\sqrt{\varepsilon}} \tag{3.34}$$

and

$$\tau_m = \left(\frac{1}{\pi d_i}\left(\frac{1-\varepsilon}{\varepsilon}\right)\frac{d}{dz}(\dot{m}_g \bar{v}_g) - \frac{1}{\pi d_i}\frac{d}{dz}(\dot{m}_l \bar{v}_l)\right) \tag{3.35}$$

Various correlations have been developed to predict local void fraction for cocurrent vapor-condensate annular flow. Evaluations of these correlations in the literature have found the correlation of Zivi (1964) to be the most successful (Soliman et al., 1968):

$$\varepsilon = \frac{1}{1 + \left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)^{2/3}} \tag{3.36}$$

where x is the local quality defined as :

$$x = \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_l} \tag{3.37}$$

This correlation can be assumed to hold good for countercurrent vapor-condensate annular flow.

It is worthwhile to note here that the maximum range of values of local quality, x for any reflux condensation problem is : $0.5 \leq x \leq 1$. The range of values of the void fraction, ε, is still less as $\rho_l \rangle \rho_g$. See Table 3.1.

From Eqns. (3.36), (3.37) and (3.33), the shear stress τ_g can be readily evaluated. For τ_f, the interfacial shear stress τ_i can be evaluated using the linear equation proposed by Linehan (1968), Eqn. (3.17) :

$$\tau_i = f_{ia} \frac{\rho_g \overline{v}_g^{\,2}}{2} + \overline{v}_g \left| \frac{d\Gamma_g}{dz} \right| \tag{3.17}$$

In this equation, the first term gives the adiabatic interfacial shear stress and the second term gives the effect of condensation mass flux on it. The latter cannot be evaluated as the condensation rate is not known a priori. For low condensation rates, the second term is small and can be neglected but for high condensation rates, it is dominant. If it is assumed that the first and the second term contribute equally to the interfacial shear stress (moderate condensation rates), then Eqn. (3.17) can be reduced to :

$$\tau_i = f_{i,a}\left(\rho_g \overline{v}_g^{\,2}\right) \tag{3.38}$$

This equation would underpredict τ_i when the condensation effect is dominant and it would overpredict for low condensation rates. These effects can be checked suitably, as explained later.

In Eqn. (3.35) for τ_m, the first term is always negative while the second term is positive and they tend to offset each others' effects. For countercurrent two-phase flow, the values of the above terms would be of the same order of magnitude. Soliman et al. (1968) found that for cocurrent two-phase flow, the term representing the addition of

Table 3.1 Values of void fraction, ε, as a function of local quality, x, for different $\frac{\rho_g}{\rho_l}$ values

$\frac{\rho_g}{\rho_l}=\frac{1}{1000}$		$\frac{\rho_g}{\rho_l}=\frac{1}{100}$		$\frac{\rho_g}{\rho_l}=\frac{1}{10}$	
x	ε	x	ε	x	ε
0.5	0.990	0.5	0.960	0.5	0.820
0.9	0.999	0.9	0.995	0.9	0.977

momentum to the liquid film by the condensing vapor was dominant for most of the condensing length. In countercurrent flow, this term would be close to zero as the interfacial velocities are very small (for zero slip at the interface, the interfacial velocities are equal to zero). Thus, for reflux condensation, the shear stress τ_m can be neglected. Eqn. (3.25) would then reduce to

$$\tau_w = \tau_g - \tau_f \tag{3.39}$$

where τ_g and τ_f can be estimated from Eqns. (3.36), (3.37), (3.33), (3.38) and (3.34). $f_{i,a}$ in Eqn. (3.38) can be found using the correlation of Wallis (1969) [Eqn. (3.6)] or Bharathan et al. (1979) [Eqns. (3.7) -- (3.10)].

The following strategy can be used to determine the local condensing heat transfer coefficient :

(1) Estimate τ_g and τ_f as described above.

(2) Using the criterion stated by Carpenter and Colburn (1951), if the value of τ_f is less than, say , half the value of τ_g, then consider the effect of vapor shear negligible.

Estimate the local condensing heat transfer coefficient using the appropriate gravity-controlled correlation, depending on the value of the local condensate Reynolds number, i.e., if $Re_{c,z} \leq 30$, use Eqn. (3.19); for $30<Re_{c,z}<1600$, use Eqn. (3.20) and for $Re_{c,z} \geq 1600$, use Eqn. (3.21) or Fig. 3.2.

(3) If vapor shear is significant, as determined by the above criterion, then estimate τ_w from Eqn. (3.39) and use the correlation suggested by Soliman et al. (1968), viz. Eqn. (3.24), to determine the local condensing heat transfer coefficient, assuming it holds good for reflux condensation. To ensure that the value obtained above for the coefficient is reasonable, compare it with the value obtained from the appropriate gravity-controlled correlation for the same conditions. If the former value is very high compared to the latter, which might arise if τ_f is significantly underpredicted (high condensation rate), then take the arithmetic average of the two values. Otherwise, choose the higher of the two values as the condensing heat transfer coefficient.

Vapor Core

The vapor-phase heat transfer coefficient is calculated in the literature as follows : First, the coefficient is calculated assuming single-phase vapor flow in a smooth conduit using the correlation of Chilton and Colburn (1934) viz.

$$\alpha_g = jC_{pg}\left(\frac{4\dot{m}_g}{\pi d_i^{\,2}}\right)(\mathrm{Pr})^{-2/3} \tag{3.40}$$

where the Colburn j-factor is given by :

$$j = 0.023\left(\frac{4\dot{m}_g}{\pi d_i \eta}\right)^{-0.2} \tag{3.41}$$

The single-phase coefficient is then corrected for :

(a) the effect of condensation, and

(b) the presence of the condensate layer.

Heat transfer in the vapor core is understood to occur by two parallel processes : (i) sensible heat transfer due to a temperature gradient, and (ii) diffusion of vapor molecules, because of a concentration gradient from the bulk vapor stream to the condensate surface, which carry sensible heat and condense at the interface giving up their latent heat.

The process of condensation affects the no-condensation heat transfer process in two ways:

(i) The condensation mass flux distorts the temperature profile in such a way that the sensible heat flux is reduced.

(ii) The condensing vapor molecules release sensible energy as they travel from the bulk stream which is at a higher temperature to the interface which is at a lower temperature.

Colburn and Drew (1937) and Ackermann (1937) derived a correction factor, θ_a to account for both effects simultaneously. It is given as :

$$\theta_a = \frac{a}{1-e^{-a}} \tag{3.42}$$

where the term a is defined as :

$$a = \frac{\dot{n}_t C_{pg}}{\alpha_g} \tag{3.43}$$

The expression for θ_a can be split into two terms to show the effects (i) and (ii) independently.

$$\theta_a = \frac{a}{e^a - 1} + a = \theta_s + a \tag{3.44}$$

The first term θ_s shows effect (i). Its value lies between zero and unity. It is to be used alone, when the temperature variation of the bulk gas-vapor mixture is to be determined. The value of θ_a is always greater than unity. It represents a correction to the heat transfer coefficient corresponding to total sensible heat flux from the vapor stream to the condensate surface.

It should be noted here that the term a in θ_a and θ_s depends on the condensation molar flux, which is not known a priori. Thus, the correction factors θ_a and θ_s to the single-phase heat transfer coefficient cannot be determined directly. They have to be obtained by iterative calculation together with the heat and the mass balance equations. Refer to Chapter IV.

The presence of the condensate layer is expected to increase the heat transfer rate at the interface due to increased turbulence and effective roughness. According to Sardesai et al. (1983), this enhancement may be accounted for by a multiplier C_f as the interface enhancement factor. The term C_f would have a minimum value of 1.0 corresponding to a smooth interface at very low velocity. For shear-controlled flow, it would be greater than unity. However, it would be difficult to quantify C_f theoretically and experimental data would be required to estimate it. It might be interesting to note here that for condensation inside a vertical tube with low vapor shear, the experimental studies of Owen et al. (1980) have shown the value of C_f to be close to unity.

Price and Bell (1974) accounted for the two-phase enhancement effect in a different way. They assumed the relationship between the heat transfer coefficient and the pressure gradient for two-phase flow to be analogous to that for single-phase flow such that :

$$\frac{\alpha_{TP}}{\alpha_g}=\left[\left(\frac{dp}{dz}\right)_{f,TP}\bigg/\left(\frac{dp}{dz}\right)_{f,g}\right]^{0.445} \tag{3.45}$$

In this equation, the frictional pressure gradient for two-phase flow was calculated using the Martinelli-Nelson correlation (1948) :

$$\left(\frac{dp}{dz}\right)_{f,TP}=\phi_{ltt}^{2}x_{tt}^{1.75}\left(\frac{dp}{dz}\right)_{f,g} \tag{3.46}$$

where $$x_{tt}=\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_l}\right)^{0.57}\left(\frac{\eta_l}{\eta_g}\right)^{0.11} \tag{3.47}$$

and $$\phi_{ltt}^{2}=\left[1+\frac{1}{x_{tt}}+\frac{3(1-P_r)}{\sqrt{x_{tt}}}\right] \tag{3.48}$$

In the present work, the two-phase enhancement effect is accounted for using the multiplier C_f suggested by Sardesai et al. (1983).

CHAPTER IV

DESIGN METHODS

The primary objective of this chapter is to develop an approximate design procedure for pure vapor and multicomponent total/partial reflux condensers. First, the literature is reviewed for the different design methods currently in use for condensers. Their application to the design of reflux condensers is discussed. Then, an approximate generalized design procedure is evolved for total/partial pure vapor and multicomponent reflux condensers.

The case of pure vapor condensation is discussed first.

Pure Vapors

Designing any heat exchange equipment for the condensation of pure vapors is relatively simple. For this case, there is no mass transfer resistance in the condensate film or in the vapor phase, and heat transfer resistance is concentrated in the condensate film. The design can be carried out in one step using :

$$Q_T = U_{\oplus} A_{\oplus} (\Delta T_{lm}) \tag{4.1}$$

if an average condensing heat transfer coefficient and thus an average overall heat transfer coefficient can be determined. The disadvantage of using an average condensing heat transfer coefficient is that the variation in the wall temperature and the physical properties along the condenser tube cannot be properly accounted for.

When the latter effects are important, the design can be carried out on a stepwise basis by evaluating the overall heat transfer coefficient, U, and the temperature difference between the condensing vapor stream and the coolant, (T_g-t) at several points along the condenser tube as a function of the heat transferred from the vapor entrance to that point,

Q. The required heat transfer surface can then be obtained by integrating the following equation numerically or graphically.

$$A = \int_0^{Q_T} \frac{dQ}{U(T_g - t)} \tag{4.2}$$

Eqn. (4.2) is the basic heat exchanger design equation. It should be noted here that U and A are interconnected as :

$$U_o A_o = U_i A_i = U_\oplus A_\oplus \tag{4.3}$$

where the subscripts refer to a particular reference area.

In any reflux condenser design problem, the quantities usually specified are the total condenser heat duty, Q_T, the flowrate and the complete thermodynamic state of the inlet vapor/vapor-gas mixture stream, the fractionation requirement specified as the desired flowrate or composition of one component in the outlet vapor stream and/or the flowrate and other conditions of the coolant. It should be noted here that it is not possible to specify fractionation requirements, heat requirements and coolant temperatures independently; only two of the three can be fixed independently. The condenser operating pressure can be assumed to be constant throughout as the pressure drop of the vapor stream in a steady countercurrent two-phase flow operation is usually negligible compared to the absolute pressure.

The overall heat balance for the condensation of a pure saturated vapor with a countercurrent coolant is (See Fig. 4.1) :

$$Q_T = \left(\dot{M}_{g,in} - \dot{M}_{g,out}\right)(Mw)\Delta h_{lg} = \dot{m}_{cool} C_{p,cool}\left(t_{out} - t_{in}\right) \tag{4.4}$$

where the latent heat of condensation, Δh_{lg} is a function of T^{sat}.

Given Q_T and the inlet vapor molar flowrate, the outlet molar flowrate of vapor can be determined and the mass flowrate of the coolant can be chosen such that the outlet coolant temperature is below its allowable upper limit fixed by engineering considerations.

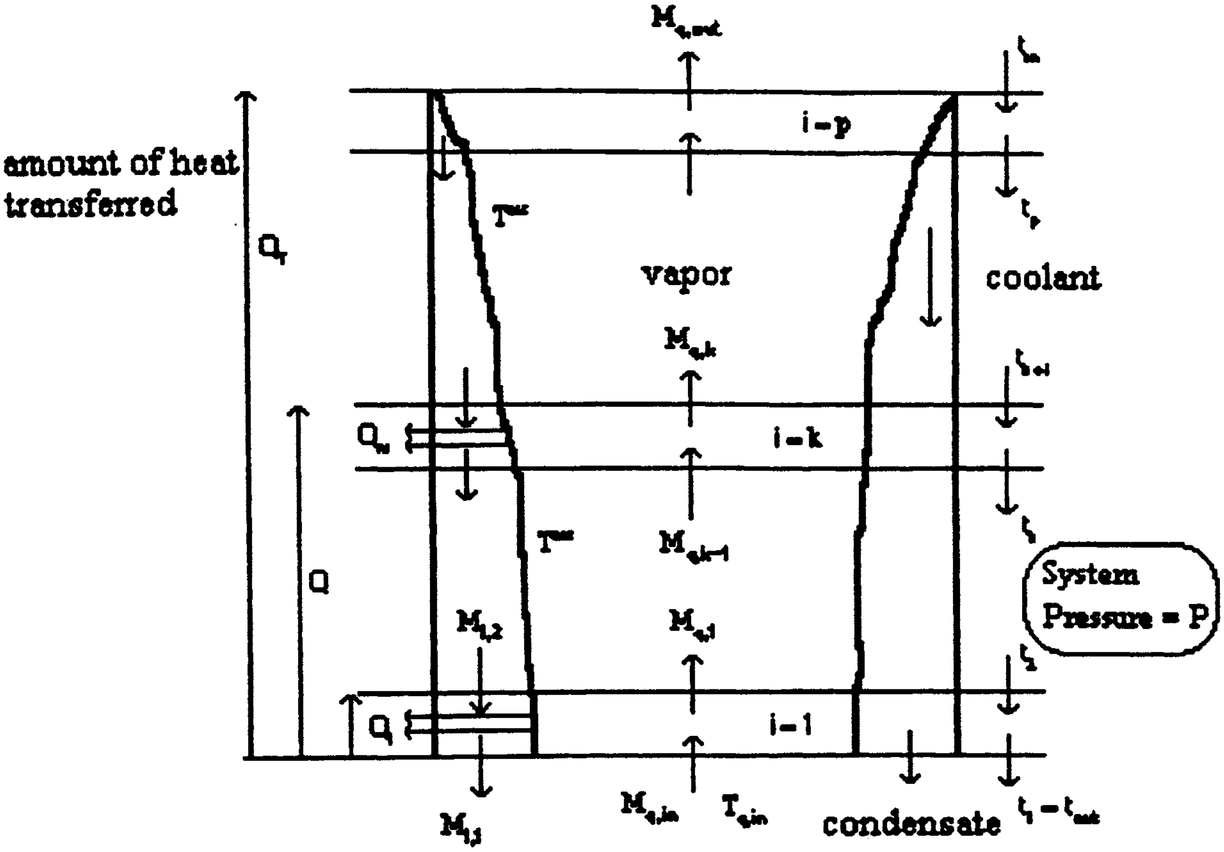


Fig. 4.1 Mass and energy balances for pure component condensation

The quantity $1/U(T_g\text{-}t)$ can be determined as a function of the heat transferred, Q, between the vapor entrance and the point where the bulk vapor and the coolant temperatures are T_g and t respectively, as follows :

First, small steps can be chosen in coolant temperature from t_{in} to t_{out} at, say, equal intervals. At each value of the coolant temperature, t, the quantities $(T_g\text{-}t)$ and Q can be readily determined as the vapor-side temperature, T_g is constant at T^{sat} for a saturated vapor and Q is given by :

$$Q = \dot{m}_{cool} C_{p,cool} \left(t_{out} - t\right) \tag{4.5}$$

The overall heat transfer coefficient, based on the outside area, U_O can be obtained from :

$$U_o = \frac{1}{\frac{1}{\alpha_i}\left(\frac{d_o}{d_i}\right) + R_{f,i}\left(\frac{d_o}{d_i}\right) + \frac{d_o \ln(d_o / d_i)}{2\lambda_w} + R_{f,o} + \frac{1}{\alpha_o}} \tag{4.6}$$

In this equation, the fouling resistances, $R_{f,i}$ and $R_{f,o}$ can be chosen by experience. The coolant-side heat transfer coefficient , which is also the shell-side coefficient, α_o can be estimated from known methods depending on its importance relative to the tube-side coefficient.

The only quantity to be determined is the vapor-side heat transfer coefficient, α_i. As stated by Colburn (1951), it is readily calculated from the kinetic theory of gases that under ordinary conditions, the temperature difference between the pure vapor and the condensate surface is negligible because of the high rate of molecular motion. In other words, the vapor-phase heat transfer resistance is negligible. Thus,

$$\frac{1}{\alpha_i} = \frac{1}{\alpha_c} \tag{4.7}$$

where α_c is the condensing heat transfer coefficient. α_c can be determined using the strategy described in Chapter III. The condensate Reynolds number at a point 'k' in the condenser (See Fig. 4.1) is given by :

$$Re_{c,k} = 4\frac{\left(\dot{M}_{g,k} - \dot{M}_{g,out}\right)(Mw)}{\pi d_i \eta_c} \tag{4.8}$$

$\dot{M}_{g,k}$, the vapor molar flowrate at point 'k' can be obtained using the incremental heat balance equation, viz.:

$$Q = \dot{m}_{cool} C_{p,cool} \left(t_{out} - t_k\right) = \left(\dot{M}_{g,in} - \dot{M}_{g,k}\right)(Mw)\Delta h_{lg} \qquad (4.9)$$

In this way, α_c, U and thus $U(T_g\text{-}t)$ can be determined at many points along the condenser tube as a function of Q. The heat transfer surface can then be evaluated by integrating Eqn. (4.2) numerically or graphically.

The vapor which is handled by a reflux condenser is usually saturated. For a saturated vapor, the above design procedure can be used. If the vapor is superheated, the problem can be dealt with as follows. In a reflux condenser, because of the countercurrent nature of flow, the entering vapor is in contact with the draining condensate film. The condensate phase is at or below the saturation temperature of the component at the system pressure. Thus, the superheated vapor condenses right from the entrance and because of thermodynamic equilibrium at the vapor-liquid interface, the condensate surface is at the saturation temperature. The design can be carried out similar to that for saturated vapor, using the following equation :

$$A_o = \int_0^{Q_T} \frac{dQ}{\alpha_{oi}\left(T^{sat} - t\right)} \qquad (4.10)$$

where α_{oi} is the heat transfer coefficient from the interface to the coolant, based on the outside area. The overall heat balance equation can be written as :

$$Q_T = \left(\dot{M}_{g,in} - \dot{M}_{g,out}\right)(Mw)\Delta h_{lg} + \left(\dot{M}_g C_{pg}\right)_{in}(Mw)\left(T_{g,in} - T^{sat}\right) \qquad (4.11a)$$

$$Q_T = \left(\dot{m}C_p\right)_{cool}\left(t_{out} - t_{in}\right) \qquad (4.11b)$$

The molar flowrate of vapor at any point in the condenser tube can be determined from the appropriate incremental heat balance equation. It is assumed that the vapor and the liquid streams leaving the first incremental element are in equilibrium with each other at the vapor bulk temperature. In other words, all the desuperheating is assumed to occur in the first element. Thus, the incremental heat balance equation for any element can be written as :

$$Q_T = \left(\dot{M}_{g,in} - \dot{M}_{g,k}\right)(Mw)\Delta h_{lg} + \left(\dot{M}_g C_{pg}\right)_{in}(Mw)\left(T_{g,in} - T^{sat}\right) \tag{4.12a}$$

$$Q_T = \left(\dot{m}C_p\right)_{cool}\left(t_{out} - t_{k+1}\right) \tag{4.12b}$$

Once the integrand in Eqn. (4.10) is evaluated as a function of Q, the required heat transfer surface can be readily obtained by integrating Eqn. (4.10) numerically or graphically.

Vapor Mixtures

Condensation of vapor mixtures differs from pure vapor condensation in two ways :

(a) Mixture condensation is always non-isothermal. As the heavier components are preferentially condensed, the remaining mixture has a lower dew-point and the temperature of condensation decreases monotonically.

(b) Mass transfer resistances are introduced in addition to the heat transfer ones, in the vapor stream and in the condensate film.

A qualitative diagram of the mixture condensing process at a point some distance from the beginning of condensation is given in Fig. 4.2. The temperature profile in it indicates that there are sensible heat effects in both the liquid and the vapor phases and the concentration gradients indicate that there are diffusional resistances to the transfer of mass in each phase. The controlling resistance in the condensation of vapor mixtures is generally the vapor-phase resistance to heat and mass transfer.

In any condensation problem, the resistance to heat transfer in the condensate film is not in series with that in the vapor phase, i.e. the heat transfer resistance from the bulk vapor to the coolant cannot be determined by simply adding the resistances in the vapor phase and in the condensate film. This is because of the condensation occurring at the interface. Thus the overall heat transfer coefficient from the bulk vapor to the coolant is difficult to determine.

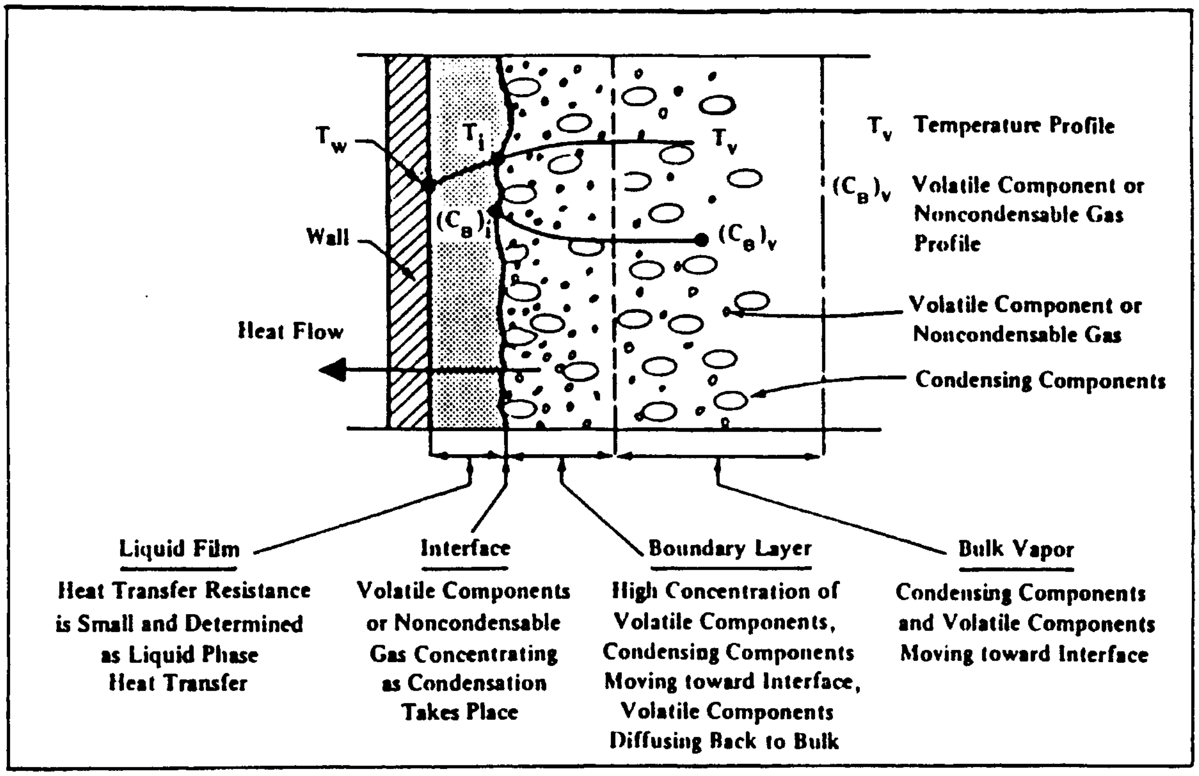


Fig. 4.2 Multicomponent condensation (Sardesai et al., 1983)

Models of varying complexity and accuracy are available in the literature to account for the different processes that occur in the condensation of vapor mixtures. Design methods based on these models are of two basic kinds, viz. :

(1) The approximate design methods (such as those of Kern (1950), Silver(1947), Ward (1960), Bell and Ghaly (1972) and Sardesai et al.(1983)), which are based on an equilibrium model.

(2) The more physically realistic differential methods, which are based on models such as film theory, penetration theory and boundary layer theory.

In this chapter, the different methods currently in use are reviewed and their application to the design of reflux condensers is discussed. The advantages and the pitfalls in the methods and the suitability of their application to different design problems is evaluated. Only those methods which are of immediate relevance to practicing design engineers in the chemical process industries are considered.

(1) Differential Methods

This class of methods includes those based on models such as film theory, penetration theory and boundary layer theory. The latter two models are more sophisticated than the film theory models and are relatively less investigated in the literature. They are limited to geometries like vertical tubes and describing the condensation of a vapor in the presence of a noncondensable gas or the condensation of a binary vapor mixture. Extensions of these models to three or more component systems are few in number. Also, these models have not been developed to a point where they could be used for the tube-side design of shell and tube heat exchangers. Thus, only methods based on film theory are discussed.

In the film theory methods, local heat and mass transfer rates are calculated and then integrated numerically along the length of the condenser using a set of one-dimensional differential material and energy balances. These methods demand detailed information about the fluid streams and their physical properties throughout the condensation process.

It would be convenient to study the film theory methods in a stepwise fashion starting from :

(a) the condensation of a pure vapor in the presence of a noncondensable gas,

(b) the condensation of a binary vapor mixture, and lastly

(c) the condensation of multicomponent (≥ three components) vapors or vapor-gas mixtures.

(a) Single condensable vapor with noncondensing gas : The earliest work on the design of heat exchangers for the condensation of pure vapors with noncondensing gases is that of Colburn and Hougen (1934). They formulated the equations and the design procedure for condensing a saturated vapor mixture in a fairly rigorous form. Subsequent studies on the subject include those of Colburn and Edison (1941), Smith (1942) and Colburn (1951). Some of them suggested simplifications to the trial and error calculations of the Colburn-Hougen procedure but they could not appreciably broaden the range of application.

With the presently available knowledge on the multicomponent condensation process, the design problem can be formulated, in fundamental terms, as follows :

The condensation molar flux is given by :

$$\dot{n}_1 = \beta_g \ln \frac{1-y_{1i}}{1-y_{1b}} \qquad (4.13)$$

This equation is obtained by assuming all the resistances to heat and mass transfer in the gas phase occur in a fictitious laminar layer of certain thickness adjacent to the condensate film. It can be expressed alternatively as :

$$\dot{n}_1 = \xi \beta_g \frac{y_{1b}-y_{1i}}{1-y_{1b}} \qquad (4.14)$$

where

$$\xi = \frac{\phi}{e^{\phi} - 1} \tag{4.15}$$

and

$$\phi = \frac{\dot{n}_1}{\beta_g} \tag{4.16}$$

Eqn. 4.14 is useful at low condensation rates, where ξ tends to unity. The mass transfer coefficient, β_g can be calculated conveniently using the Chilton-Colburn (1934) analogy between heat and mass transfer :

$$\beta_g = \frac{\alpha_g}{C_{pg}}\left(\frac{\delta_{12}}{\kappa_g}\right)^{2/3} = \frac{\alpha_g}{C_{pg}}\left(\frac{Pr}{Sc}\right)^{2/3} \tag{4.17}$$

The heat flux equation is given by :

$$\alpha_{oi}(T_i - t) = \alpha_g{}'(T_g - T_i) + \dot{n}_1 \Delta h_{lg} \tag{4.18}$$

The form of Eqn. (4.18) was derived by Colburn and Hougen (1934). This equation is obtained by assuming that there is subcooling in the condensate layer and that all the heat transferred from the bulk vapor stream to the vapor-liquid interface passes through the condensate film and the wall to the coolant. The rate of heat flow from the vapor stream to the condensate surface is understood to occur by two parallel processes : first., sensible heat transfer due to the temperature gradient, and second, diffusion of vapor molecules, because of a concentration gradient, from the main stream to the condensate surface, which carry sensible heat and condense at the interface giving up their latent heat. As noted above, the sensible heat transfer rate is enhanced by the mass transfer process. This is accounted in Eqn (4.18) by the term $\alpha_g{}'$, which is the gas phase heat transfer coefficient corrected for mass transfer effects as :

$$\alpha_g{}' = \alpha_g \frac{a}{1 - e^{-a}} \tag{4.19}$$

where

$$a = \frac{\dot{n}_1 C_{pl}}{\alpha_g} \tag{4.20}$$

This correction to α_g was derived by Ackermann (1937) and by Colburn and Drew (1937).

Eqn (4.18) is to be used to iteratively calculate the interfacial temperature, T_i, given the state of the gas phase (i.e. its bulk phase composition and temperature) and the coolant temperature, t. To solve this equation, it is necessary to assume that equilibrium conditions prevail at the interface. The change in the temperature and the composition of the bulk vapor phase along the condenser tube is determined by integrating the gas phase heat and mass balance equations along the vapor path. The heat balance equation is :

$$\frac{dT_g}{dA} = -\frac{\alpha_g^\bullet (T_g - T_i)}{\dot{M}_g C_{pg}} \tag{4.21}$$

where $\alpha_g^\bullet$ is defined as :

$$\alpha_g^\bullet = \alpha_g \theta_s = \alpha_g \frac{a}{e^a - 1} \tag{4.22}$$

It should be noted here that $\alpha_g^\bullet$ is different from α_g', defined by Eqn. (4.19). α_g' is the heat transfer coefficient that corresponds to the total sensible heat flux from the vapor stream to the interface while $\alpha_g^\bullet$ corresponds only to the sensible heat flux occurring due to the temperature gradient (i.e. the sensible heat flux due to the condensing vapor molecules is not included in this). $\alpha_g^\bullet$ is used in Eqn. (4.21) as it is known that the temperature change of the bulk gas-vapor mixture is only due to the heat flux arising from the temperature gradient.

The mass balance equation is :

$$\frac{dy_{1b}}{y_{1b} - 1} = -\frac{\dot{n}_1 dA}{\dot{M}_g} \tag{4.23}$$

where $\dot{M}_g$ is the total molar flowrate of the vapor stream.

For using the above model for design purposes, a trial and error procedure is usually necessary. First, a value is assumed for one of the design parameters. The heat and the mass balance equations, Eqns. (4.21) and (4.23), are integrated along the vapor-flow path. At each step of the integration, the conditions at the interface are obtained by iterative evaluation of the heat and the mass flux equations, Eqns. (4.14)and (4.18). At the end of the above process, a value is obtained for one of the specified variables. This value is compared with the specified value of that variable. The process is repeated for different values of the design parameter till a match is obtained between the calculated and the specified values of the given quantity.

(b) Binary vapor mixtures : The classic formulation of the local transport rate equations in binary vapor condensation is by Colburn and Drew (1937) .This model has been used by many investigators for the design of condensers (downdraft, horizontal and updraft or reflux condensers) for binary vapor mixtures. The work of Kent and Pigford (1956) is one of the first to use the Colburn-Drew model in a design mode They studied experimentally and theoretically the performance of reflux condensers in comparison to conventional equipment for adiabatic distillation plus total condensation. The design procedure they developed for reflux condensers was obtained solely from the point of view of mass transfer by an adaptation of the transfer-unit concept to partial condensation. It consists of equations relating the number of transfer units to liquid and gas -phase resistances, surface area and amount of condensation. It is based on the assumption that the heat transfer rate or the rate of condensation is constant throughout the condenser.

Like the work of Kent and Pigford (1956), many studies (Pressburg and Todd (1957) , Estrin et al. (1965)) experimentally confirmed the Colburn-Drew model but they did not extend it appreciably to a design mode. In 1974, Price and Bell presented a complete design procedure based on the Colburn-Drew model for binary vapor condensers. The mass transfer resistance in the liquid phase was neglected, i.e., a well-mixed condensate film was assumed. The procedure employed mass and energy balances in an incremental fashion to compute the change in the temperature and the composition of the bulk vapor stream, which are different from the differential equations of the Colburn-Drew model. The predictions of this design procedure were not compared with any experimental data;

they were compared with those of the approximate design method of Bell and Ghaly (1972). It was found to be less conservative (in the sense of area required to remove a certain amount of heat) compared to the latter method, as expected.

One of the first significant works to use the Colburn-Drew theory completely in a design mode is that of Cave et al. (1987). They developed a rigorous mathematical model specifically for the design of dephlegmators/reflux condensers. Mass transfer resistances in the liquid phase were included. Also, the changes in the temperature and the composition of the vapor phase were computed using differential material and energy balance equations similar to those proposed by Colburn and Drew (1937). The proposed model was validated by testing against five series of experimental runs carried out with two different binary systems. The agreement between the predicted and the measured values for all the variables - compositions, temperatures and heat transfer surface - was found to be satisfactory.

The complete binary vapor condensation design problem can be expressed as follows : The condensation molar flux and the heat flux expressions are similar to those in part (a). The total condensation molar flux is given by :

$$\dot{n}_T = \beta_g \ln \frac{z - y_{1i}}{z - y_{1b}} \tag{4.24}$$

The alternative expression for n_t is :

$$\dot{n}_T = \xi \beta_g \ln \frac{y_{1b} - y_{1i}}{z - y_{1b}} \tag{4.25}$$

where ξ is given by Eqn. (4.15) and ϕ is defined as :

$$\phi = \frac{\dot{n}_T}{\beta_g} \tag{4.26}$$

Unlike part (a), in binary vapor condensation, there is mass flux occurring in the condensate film also. This can be expressed, analogous to Eqn. (4.24), as :

$$\dot{n}_T = \beta_g \ln \frac{z - x_{1b}}{z - x_{1i}} \tag{4.27}$$

The heat flux equation is :

$$\alpha_{oi}(T_i - t) = \alpha_g'(T_g - T_i) + \dot{n}_T \Delta h_{lg} \tag{4.28}$$

The term, a, in the expression for α_g' (Eqn. (4.19)) is defined here as :

$$a = \frac{\dot{n}_T C_{p,cv}}{\alpha_g} \tag{4.29}$$

where $C_{p,cv}$ is the specific heat capacity of the condensing vapor mixture, obtained by :

$$C_{p,cv} = zC_{pg1} + (1-z)C_{pg2} \tag{4.30}$$

As in part (a), the conditions at the interface (T_i, y_i, x_i) are determined by using Eqns. (4.25), (4.27) and (4.28) and assuming that equilibrium conditions prevail at the interface, given the state of the gas phase and the coolant temperature.

The heat balance equation to be used to determine the change in vapor temperature along the condensation path is Eqn (4.21), with the term a in the expression for $\alpha_g^\bullet$ defined as in Eqn. (4.29). To evaluate the change in the bulk vapor phase composition, the mass balance equation for the vapor phase is to be used viz.

$$-\frac{dy_{1b}}{y_{1b} - z} = -\frac{\dot{n}_T dA}{\dot{M}_g} \tag{4.31}$$

For change in the bulk liquid phase composition, the mass balance equation is :

$$\frac{dx_{1b}}{x_{1b} - z} = -\frac{\dot{n}_T dA}{\dot{M}_l} \tag{4.32}$$

Eqns. (4.21), (4.31) and (4.32) have to be numerically integrated to determine values of T_g, y_{1b} and x_{1b} at any point along the condensation path.

Common simplifications made to the above problem are :

(i) The condensate phase is assumed well-mixed, i.e. there are no diffusional resistances to mass transfer in the liquid phase. In this case, x_{1i} can be assumed equal to x_{1b}. This situation is equivalent to one with very high liquid phase mass transfer coefficients. This assumption is usually made when the liquid phase mass transfer coefficients are difficult to determine due to lack of experimental data. (Butterworth, 1984; Colburn and Drew, 1937).

(ii) There is little mixing in the liquid phase so that the liquid composition near the interface is the same as the condensing fraction i.e. $x_i = z$. This situation is equivalent to one with very low liquid phase mass transfer coefficients. (Butterworth, 1984; Colburn and Drew, 1937).

(iii) Constant heat flux or condensation rate is assumed throughout the condensation path (Kent and Pigford, 1956; Honda et al., 1992).

The design procedure for binary vapor condensers based on this film model is similar to that discussed in part (a).

(c) Multicomponent mixtures : The Colburn-Drew equations were first extended to multicomponent vapor or vapor-gas mixtures by Toor (1964) and Stewart and Prober (1964). A corrected version of the multicomponent film model incorporating a generalized matrix method for the solution of the Stefan-Maxwell equations was presented by Krishna and Standart (1976) and Krishna et al. (1976). The above two models are commonly referred to as "interactive" models of mass transfer because they allow for diffusional interactions, where the diffusional behavior of a constituent depends on all independent concentration gradients. In the noninteractive film model, also known as the effective diffusivity model, an effective diffusion coefficient is defined for constituents with respect to the mixture as a whole. In this model, the mass transfer calculations are effectively based on the approach of Colburn and Hougen (1934).

The multicomponent condensation design problem, as described by the above models, will not be presented here as it is extremely complicated and beyond the scope of this work.

Krishna et al. (1976) compared the interactive models in their ability to predict the condensation of a five component hydrocarbon vapor mixture accompanied by hydrogen as the inert gas flowing down inside a vertical tube heat exchanger cocurrent to the condensate. They demonstrated that the deviations of the calculated separation for the simpler models compared to the matrix model was about 20%. Webb and Sardesai (1981) presented a comparative study of the interactive and the noninteractive models with experimental data for multicomponent condensation inside a vertical tube. They found that the Krishna-Standart method predicted the experimental condensation rates within an accuracy of ±10% They also found that the predictions of the Krishna-Standart method and Toor's linearized theory were within ±1% of each other for a range of experimental conditions. The difference between the Krishna-Standart method and the effective diffusivity method was found to be less than 10%.

Summarizing the papers, it can be said that the interactive models are more complex than the noninteractive model, that the interactive models are of roughly equal quality and that the matrix model of Krishna-Standart gives the most precise description of multicomponent mass transfer but it also requires the most computing time.

Rohm (1980) applied the matrix model of Krishna-Standart and developed a generalized macroscopic model to simulate the steady state behavior of a dephlegmator(reflux condenser) for multicomponent vapor systems. He used this model and the other known mass transfer models for calculating the dephlegmation of a ternary vapor mixture of methanol, ethanol and water with specific operating conditions. He found the differences in the calculated separation rate between the models to be insignificant with respect to practical condenser design.

The above described film theory methods give a precise description of the condensation process. However, they are very complicated and therefore time consuming to use. They demand detailed knowledge about the fluid streams and their physical properties. A particular problem is the lack of data on diffusion coefficients, which are required.

(2) Approximate Methods

These methods are useful when knowledge about the physical properties of the two phases, particularly diffusion coefficients, is not complete and when a preliminary (rough) design of the condenser is required. In these methods, the condensation process is assumed to follow the equilibrium condensation curve. This assumption eliminates the problem of calculating mass transfer coefficients and the interface conditions, simplifying the calculation procedure considerably as compared with the film theory methods.

Several authors have published methods of this type. The earliest among them is that of Silver (1947), who developed a general purpose method for the design of coal gas condensers. Subsequently, Ward (1960) and Bell and Ghaly (1972) described procedures similar to Silver's method. The general method of Silver-Ward-Bell-Ghaly can be described as follows.

The sensible heat flux, q_{sv} for cooling of the bulk gas-vapor mixture core is given by :

$$q_{sv} = \alpha_g (T_g - T_i) \tag{4.33}$$

The total heat flux, q_t is given by :

$$q_t = \alpha_{oi} (T_i - t) \tag{4.34}$$

This equation makes the minor assumption that all the heat released in cooling the condensate layer crosses all of that layer. Eliminating the interface temperature, T_i between Eqns. (4.33) and (4.34) gives :

$$q_t = \left(\frac{1}{\alpha_{oi}} + \left(\frac{q_{sv}}{q_t} \right) \frac{1}{\alpha_g} \right)^{-1} (T_g - t) \tag{4.35}$$

If U_o is defined as the effective overall heat transfer coefficient of the process from the bulk vapor to the coolant stream such that

$$q_t = U_o (T_g - t) \tag{4.36}$$

then, from Eqns. (4.35) and (4.36),

$$\frac{1}{U_o} = \left(\frac{1}{\alpha_{oi}} + \left(\frac{q_{sv}}{q_t} \right) \frac{1}{\alpha_g} \right) \tag{4.37}$$

Bell and Ghaly define the ratio $\frac{q_{sv}}{q_t}$ as a factor Z .

$$Z = \frac{q_{sv}}{q_t} \tag{4.38}$$

From the above equations, the final design equation can be written as :

$$A_o = \int_0^{Q_T} \frac{1 + \frac{Z\alpha_{oi}}{\alpha_g}}{\alpha_{oi}(T_g - t)} dQ \tag{4.39}$$

or,

$$A_o = \int_0^{Q_T} \frac{1}{U_o(T_g - t)} dQ \tag{4.40}$$

This reduces to the appropriate form for the extreme values of Z. Z = 0 corresponds to the case of a pure component condensing isothermally and Z = 1 corresponds to cooling of a gas without condensation.

As stated by Sardesai et al. (1982), Eqn. (4.39) is rigorous except for the minor assumption concerning the heat released in cooling the condensate layer. Loss of rigor occurs, when the following assumptions are made in order to simplify the evaluation of the terms in Eqn. (4.39) :

(a) The condensation process is assumed to follow the equilibrium condensation curve i.e. the liquid and the vapor phases are assumed to be in equilibrium at the vapor bulk temperature. This is the most important assumption as this is what makes the design methods approximate. Z and $(T_g - t)$ are conveniently evaluated using this assumption.

(b) α_g is calculated from a correlation for convective heat transfer to a dry wall assuming that the vapor-gas mixture flows alone. Any possible enhancement of the coefficient due to the presence of a rippled film rather than a dry wall is ignored.

(c) α_g is not corrected for mass transfer effects.

The latter two assumptions can be suitably relaxed without making the design method very complicated. McNaught (1979) has suggested a method for correcting α_g for the effect of simultaneous mass transfer from film theory. According to film theory, the sensible heat flux for cooling of the bulk gas-vapor mixture is given by :

$$q_{sv} = \alpha_g^{\bullet}\left(T_g - T_i\right) \tag{4.41}$$

where $\alpha_g^{\bullet}$ is defined as in Eqn. (4.22). This correction to α_g accounts for the effect of condensation molar flux on the heat transfer rate. If this equation is used instead of Eqn. (4.33), the following design equation is obtained :

$$A_o = \int_0^{Q_T} \frac{1 + \dfrac{Z\alpha_{oi}}{\alpha_g \theta_s}}{\alpha_{oi}\left(T_g - t\right)} dQ \tag{4.42}$$

Since θ_s is always less than 1, the modification always gives a lower overall heat transfer coefficient.

The difficulty in applying Eqn. (4.42) compared to Eqn. (4.39) is the evaluation of the term a in the expression for θ_s, which depends on the condensation molar flux, $\dot{n}_T$. Since $\dot{n}_T$ is not known a priori, an iterative calculation of the correction together with that of the local heat transfer rates is essential. McNaught (1979) has expressed a in terms of the interfacial temperature, T_i, and has suggested a method for the iterative calculation.

McNaught has also presented numerical comparisons of the Silver-Bell-Ghaly method, the Colburn-Hougen method and his modified method for two vapor-noncondensing gas examples. His main observations include :

(i) The film theory correction term is particularly significant where the mass flux is comparatively high and the gas phase is tending to control the heat transfer rate. Where larger fouling factors are applied, the differences between the methods are less marked.

(ii) The original Silver-Bell-Ghaly method can, in some circumstances, underestimate the required heat transfer surface compared to film theory. This is in contrast with the studies of Price and Bell (1974) and Krishna et al. (1976) which observed consistent conservative prediction by the approximate method. The suggested modification was found to correct this underestimation.

Sardesai et al.(1983) presented a complete design method incorporating analytical corrections to account for both two-phase flow effects and mass transfer effects on α_g. In their paper, they have termed the Silver-Bell-Ghaly method as a resistance proration method as it involves proration of the condensate and the gas phase resistance to obtain the overall resistance to heat transfer. The enhancement in α_g due to the presence of the condensate has been accounted for by introducing a multiplier, C_f as the interface enhancement factor. The term C_f has a minimum value of 1.0 corresponding to a smooth interface at very low velocity. In shear-controlled flow regime, it is likely to be greater than unity. However, it is difficult to theoretically quantify the value of C_f, and hence, experimental data are required to correlate it. For multicomponent condensation with low vapor shear inside a vertical tube, C_f has been found to be close to unity by the experimental studies of Owen et al. (1980).

Condensation mass flux has been shown by Sardesai et al. (1983) to distort the temperature profile in such a way that the sensible heat flux is reduced. (See Fig. 4.3). This effect has been accounted for in a way similar to McNaught (1979). Thus, the final equation for q_{sv} is :

$$q_{sv} = \alpha_g \theta_s C_f \left(T_g - T_i\right) \tag{4.43}$$

The design equation obtained with this is :

$$A_o = \int_0^{Q_T} \frac{1 + \dfrac{Z\alpha_{oi}}{\alpha_g \theta_s C_f}}{\alpha_{oi}\left(T_g - t\right)} dQ \tag{4.44}$$

As noted earlier, an iterative calculation is again required to determine the correction factor θ_s, together with the local heat transfer rates. Sardesai et al. (1983) have suggested a self-converging procedure which involves converging on θ_s. They have tested the performances of the original resistance proration method and the modified method against

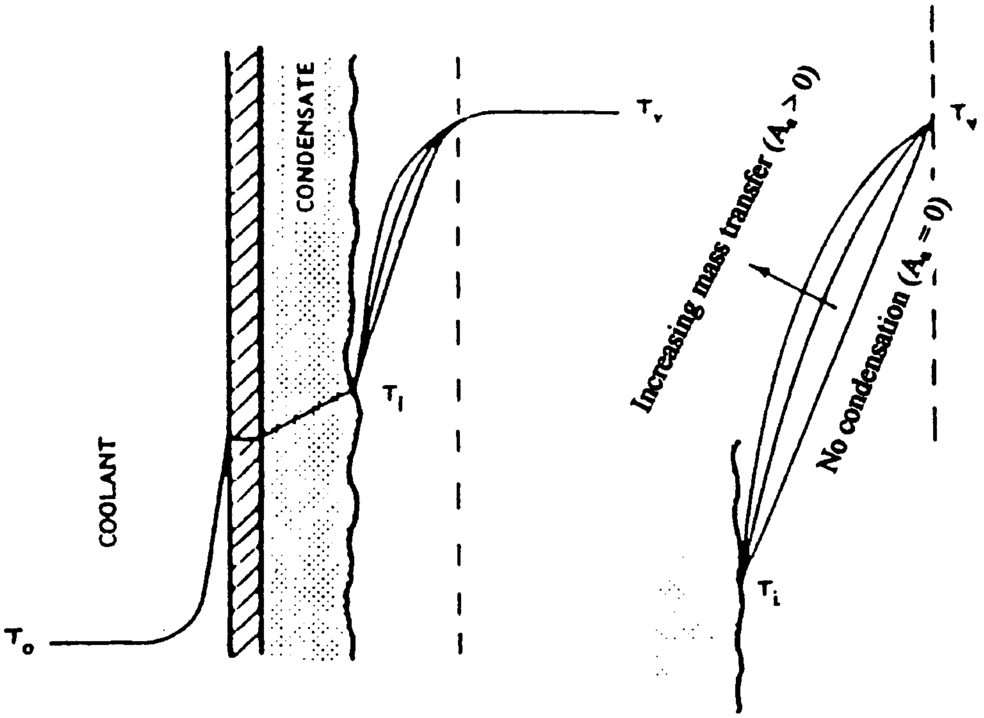


Fig. 4.3 Effect of mass transfer on temperature profile
(Sardesai et al., 1983)

experimental data for binary and multicomponent condensation of gas-vapor mixtures in downflow inside a vertical tube. The modified method was found to predict all data within $\pm 30\%$. Other observations include :

(i) The original resistance proration method overpredicts the overall heat transfer coefficient, U_o increasingly as θ_s decreases. This trend has been eliminated by the modified method.

(ii) When θ_s is close to unity, the difference in the predictions of U_o between the original and the modified method is not significant.

To use the design equation (Eqn. (4.44)) to determine the required heat transfer surface, the stepwise design procedure suggested by Bell and Ghaly (1972) and the iterative calculation method suggested by Sardesai et al. (1983) to evaluate the correction factor θ_s can be combined to give the following design procedure :

(1) Calculate the condensing curves :

(a) Total condensing stream enthalpy versus the local vapor stream temperature, T_g, assuming thermodynamic equilibrium between the phases.

(b) Vapor molar flowrate, $\dot{M}_g$ versus T_g.

(c) Total heat removed from vapor from inlet to given point, Q versus T_g.

(d) Coolant temperature, t versus T_g.

(2) Calculate as a function of Q, $dQ_{sv} = -\dot{M}_g dH_g$, or in finite increments $\Delta Q_{sv} = -\left(\dot{M}_g\right)_{avg} \Delta H_g$, where ΔQ_{sv} is the vapor sensible heat removed in the increment, $\left(\dot{M}_g\right)_{avg}$ is the mean vapor molar flowrate in the increment and ΔH_g is the decrease in the vapor enthalpy per unit mole in the increment.

(3) Calculate Z as a function of Q.

(4) Calculate α_c and α_g as functions of $\dot{M}_g$ and $\dot{M}_l$ respectively, and then as functions of Q. Calculate α_{oi} as a function of Q.

(5) Assume C_f equal to unity, unless it can be estimated from available experimental data.

(6) Assume $\theta_s = 1$.

(7) Calculate $\dfrac{1+\dfrac{Z\alpha_{oi}}{\alpha_g\theta_s}}{\alpha_{oi}\left(T_g-t\right)}$, which is the same as $\dfrac{1}{U_o\left(T_g-t\right)}$, as a function of Q.

(8) Calculate $-\dfrac{dx}{dA}=\left[\dfrac{U_o\left(T_g-t\right)-\left(\dot{m}_gC_{pg}+\dot{m}_lC_{pl}\right)\dfrac{dT_g}{dA}}{\dot{m}_T\Delta h_{lg}}\right]$ as a function of Q. The value of $\dfrac{dT_g}{dA}$ can be obtained from the condensation curve as follows :

$$\frac{dT_g}{dA}=\left(\frac{dQ}{dA}\right)\left(\frac{dT_g}{dQ}\right)=q_t\left(\frac{dT_g}{dQ}\right)=U_o\left(T_g-t\right)\left(\frac{dT_g}{dQ}\right)$$

(9) Calculate $a=\dfrac{\dot{m}_T\left(-\dfrac{dx}{dA}\right)C_{pg}}{\alpha_g}$ and θ_s from Eqn. (4.22) as a function of Q.

(10)For each value of Q, compare the present value of θ_s and the earlier value of θ_s. If θ_s is not converged for a certain value of Q, repeat steps (7) to (9) for that Q, with the last calculated value of θ_s.

(11)Once θ_s is converged at all values of Q, calculate the heat transfer area by integrating numerically or graphically the final values of the function of step (7) from Q = 0 to Q = Q_T.

Equilibrium condensation curves, required in step (1) of the above procedure, are of two types : integral and differential. In calculating the integral curve, it is assumed that condensate and vapor keep together as they flow through the condenser and that they are intimately mixed. In the differential type, it is assumed that the condensate, once formed, is separated from the vapor although it continues to flow parallel to it and, it is assumed, brought to the same temperature. There is no clear evidence as to which type of curve should be used in a given circumstance. In practice, most condensers, whether shell-side or tube-side, are designed assuming integral condensation. Butterworth (1984) clearly discusses the calculation of the integral and the differential condensation curves. However, his calculation method is limited to condensers in which the vapor and the condensate flow together in the same direction.

For reflux condensers, the equilibrium condensation curves cannot be calculated like those discussed by Butterworth (1984). This is because, the vapor and the condensate flow in opposite directions in a reflux condenser. The following general design procedure has been developed specifically for reflux condensers.

The assumptions made are similar to those of the Silver-Bell-Ghaly type of method. They include :

(1) The condensation process is assumed to follow the equilibrium condensation curve, i.e., at any point in the condensation path, the liquid and the vapor phases are assumed to be in equilibrium with each other at the vapor bulk temperature and the system pressure.

(2) Condensation is assumed to occur at constant pressure.

The overall component and total mole balances for n components, can be written as :

TOTAL : $$\dot{M}_{g,in} = \dot{M}_{g,out} + \dot{M}_{l,out} \tag{4.45}$$

COMPONENT :

j = 1 $$\dot{M}_{g,in} y_{1,in} = \dot{M}_{g,out} y_{1,out} + \dot{M}_{l,out} x_{1,out} \tag{4.46a}$$

..

j = k $$\dot{M}_{g,in} y_{k,in} = \dot{M}_{g,out} y_{k,out} + \dot{M}_{l,out} x_{k,out} \tag{4.46b}$$

..

j = n-1 $$\dot{M}_{g,in} y_{n-1,in} = \dot{M}_{g,out} y_{n-1,out} + \dot{M}_{l,out} x_{n-1,out} \tag{4.46c}$$

The component mole balance for the nth component, i.e. j = n, can be obtained by subtracting the sum of all the component mole balance equations for j = 1 to n-1 from the total mole balance equation.

The overall energy balance equation can be written as :

$$Q_T = \dot{M}_{g,in} H_{g,in} - \dot{M}_{g,out} H_{g,out} - \dot{M}_{l,out} H_{l,out} \tag{4.47}$$

where $H_{g,in}, H_{g,out}$ and $H_{l,out}$ are mixture enthalpies which are functions of the temperature , the pressure and the composition of the mixture. The latter can be obtained from thermodynamic tables directly or from component enthalpy values assuming a certain mixing rule. They can also be estimated by equation-of-state evaluations.

As mentioned earlier, the quantities usually known in a reflux condenser design problem are:

(1) The flowrate and the complete thermodynamic state of the inlet vapor stream, i.e., $\dot{M}_{g,in}, T_{g,in}, y_{j,in}(j=1,..n)$.

(2) The condenser operating pressure, P, assumed constant.

(3) The fractionation or heat removal requirement specified as :

(i) Reflux ratio, $R = \frac{\dot{M}_{l,out}}{\dot{M}_{g,in}}$, or

(ii) The composition in the outlet vapor stream of the most volatile component or the component of interest, or

(iii) The total condenser heat duty, Q_T; and

(4) The inlet temperature of the coolant stream, t_{in}.

First, the terminal conditions of the condenser can be determined from the overall mass and energy balance equations. With available knowledge about $y_{j,in}(j=1,..n)$ and P, and the assumption of equilibrium between the phases, the composition of the outlet condensate stream i.e. $x_{j,out}(j=1,..n)$ and the saturation temperature of the inlet vapor stream can be determined from equilibrium dew-point calculations.

When (1), (2) and (3(i) or 3(ii)) are specified, the calculations are simple. The n mass balance equations can be solved for $\dot{M}_{g,out}$ and the outlet vapor stream compositions, $y_{j,out}(j=1,..n)$ [if 3(i) is known] or the unknown outlet vapor compositions and $\dot{M}_{l,out}$ [if 3(ii) is known]. To determine Q_T from Eqn. (4.47), the temperature of the outlet vapor stream should be known. This can be obtained from dew-point calculations at the top of the condenser, given P and $y_{j,out}(j=1,..n)$, known from mass balance calculations.

When Q_T and (1) and (2) are specified, the mass balance and the energy balance equations should be solved simultaneously to determine the unknown quantities. This would require an iterative procedure, considering the nature of Eqn. (4.47) viz. :

(a) Assume a certain reflux ratio, R

(b) Solve the mass balance equations to determine $\dot{M}_{g,out}$ and $y_{j,out}(j=1,..n)$.

(c) Determine $T_{g,out}$ from equilibrium calculations using $y_{j,out}(j=1,..n)$ and P.

(d) Solve the energy balance equations to determine $\dot{M}_{l,out}$ and thus, the reflux ratio, R.

(e) Check if R(calculated) = R(assumed). If no, go to (a) and continue calculations until the above condition is satisfied.

The total heat transfer rate, Q_T, is related to the coolant conditions as :

$$Q_T = \dot{m}_{cool} C_{p,cool}(t_{out} - t_{in}) \tag{4.48}$$

Using this equation, the mass flowrate of the coolant can be chosen such that t_{out} is less than its allowable upper limit fixed by engineering considerations.

The next step is to carry out incremental mass and energy balances with equilibrium dew-point calculations through the condenser from the vapor entrance to the top. The increment size is fixed as small steps in the coolant temperature, t from t_{out} to t_{in}. The mass balance equations for the first increment i = 1 can be written as (See Fig. 4.4) :

TOTAL :

$$\dot{M}_{g,0} + \dot{M}_{l,2} = \dot{M}_{g,1} + \dot{M}_{l,1} \tag{4.49}$$

COMPONENT :

$j = 1$

$$\dot{M}_{g,0} y_{1,0} + \dot{M}_{l,2} x_{1,2} = \dot{M}_{g,1} y_{1,1} + \dot{M}_{l,1} x_{1,1} \tag{4.50a}$$

..

$j = k$

$$\dot{M}_{g,0} y_{k,0} + \dot{M}_{l,2} x_{k,2} = \dot{M}_{g,1} y_{k,1} + \dot{M}_{l,1} x_{k,1} \tag{4.50b}$$

..

$j = n-1$

$$\dot{M}_{g,0} y_{n-1,0} + \dot{M}_{l,2} x_{n-1,2} = \dot{M}_{g,1} y_{n-1,1} + \dot{M}_{l,1} x_{n-1,1} \tag{4.50c}$$

In the above equations, the known quantities are $\dot{M}_{g,0}(=\dot{M}_{g,in})$, $y_{j,0}(j=1,..n)$, $x_{j,1}(j=1,..n)$[from equilibrium calculations] and $\dot{M}_{l,1}(=\dot{M}_{l,out})$[from overall mass and

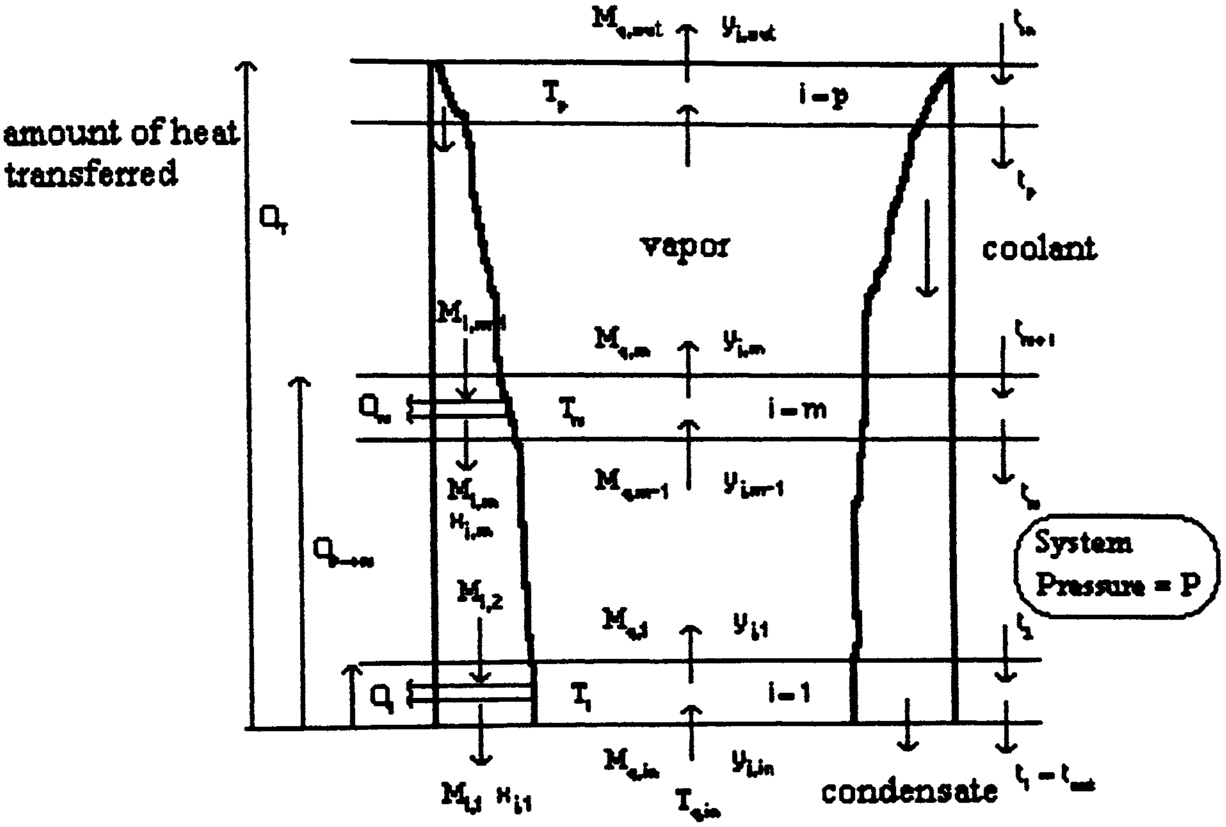


Fig. 4.4 Mass and energy balances for multicomponent reflux condensation

energy balance equations]. For each increment, it is assumed that the condensate and the vapor leaving the increment are in equilibrium with each other at the temperature of the increment. Thus, for the first increment, $y_{j,1}(j=1,..n)$ should be in equilibrium with $x_{j,1}(j=1,..n)$ at T_1 and P. It has also been assumed before that $x_{j,1}(j=1,..n)$ are in equilibrium with $y_{j,0}(j=1,..n)$ at T_1 and P. This means that $y_{j,1}(j=1,..n)$ are the same as $y_{j,0}(j=1,..n)$, i.e. there is no change in the composition of the vapor stream in the first increment. In other words, the composition of the condensing mass at i = 1 is the same as the composition of the vapor stream.

If the given value of $T_{g,in}$ is greater than T_1 (the saturation temperature), the inlet vapor stream is superheated; if $T_{g,in}$ is equal to T_1, the inlet vapor stream is at its dew-point.

The incremental energy balance for i =1 can be written as :

$$Q_1 = \dot{M}_{g,0}H_{g,0} + \dot{M}_{l,2}H_{l,2} - \dot{M}_{g,1}H_{g,1} - \dot{M}_{l,1}H_{l,1} \tag{4.51a}$$

In terms of coolant conditions (assuming a countercurrent coolant stream),

$$Q_1 = \dot{m}_{cool}C_{p,cool}(t_1 - t_2) \tag{4.51b}$$

Q_1, the heat transferred in the increment i = 1, can be obtained from Eqn.(51b). In Eqn. (51a), the term $H_{l,2}$ depends on $x_{j,2}(j=1,..n)$, T_2 and P.

Also, by assumption, it is known that $x_{j,2}(j=1,..n)$ are in equilibrium with $y_{j,2}(j=1,..n)$ at T_2 and P, i.e.

$$x_{j,2}(j=1,..n) = f\{y_{j,2}(j=1,..n),\ T_2 \text{ and } P\} \tag{4.52}$$

Thus, for i = 1, there are (n+2) unknowns and (n+2) equations relating them. Simultaneous solution of the equations is required to evaluate the unknowns. The form of Eqn. (4.52) suggests that an iterative calculation method is essential. The following iterative scheme can be used :

(a) Guess a value for $\dot{M}_{l,2}$.

(b) Evaluate $\dot{M}_{g,1}$ from Eqn.(4.49).

(c) Evaluate $x_{j,2}(j=1,..n)$ from Eqns.(4.50a,..b,..c)

(d) Evaluate T_2 from equilibrium dew-point calculations using $x_{j,2}(j = 1,..n)$ and P.

(e) Calculate $\dot{M}_{l,2}$ from Eqn. (4.51a). Check if this value is equal to the assumed value. If no, go to step (a) and rework the above steps. The above calculation is carried out until the assumed and the calculated values of $\dot{M}_{l,2}$ match.

Similarly, for i = m, the incremental mole and energy balance equations with the equilibrium equation can be written as :

Mole balances :

TOTAL :
$$\dot{M}_{g,m-1} + \dot{M}_{l,m+1} = \dot{M}_{g,m} + \dot{M}_{l,m} \tag{4.53}$$

COMPONENT :

j = 1
$$\dot{M}_{g,m-1} y_{1,m-1} + \dot{M}_{l,m+1} x_{1,m+1} = \dot{M}_{g,m} y_{1,m} + \dot{M}_{l,m} x_{1,m} \tag{4.54a}$$

..

j = k
$$\dot{M}_{g,m-1} y_{k,m-1} + \dot{M}_{l,m+1} x_{k,m+1} = \dot{M}_{g,m} y_{k,m} + \dot{M}_{l,m} x_{k,m} \tag{4.54b}$$

..

j = n-1
$$\dot{M}_{g,m-1} y_{n-1,m-1} + \dot{M}_{l,m+1} x_{n-1,m+1} = \dot{M}_{g,m} y_{n-1,m} + \dot{M}_{l,m} x_{n-1,m} \tag{4.54c}$$

Energy balance :

$$Q_m = \dot{M}_{g,m-1} H_{g,m-1} + \dot{M}_{l,m+1} H_{l,m+1} - \dot{M}_{g,m} H_{g,m} - \dot{M}_{l,m} H_{l,m} \tag{4.55a}$$

$$Q_m = \dot{m}_{cool} C_{p,cool} \left(t_m - t_{m+1}\right) \tag{4.55b}$$

Equilibrium condition :

$$x_{j,m+1}(j = 1,..n) = f\{y_{j,m+1}(j = 1,..n),\ T_{m+1} \text{ and } P\} \tag{4.56}$$

In Eqns. (4.54a,..b,..c), $y_{j,m}(j = 1,..n)$ can be obtained from equilibrium dew-point calculations using $x_{j,m}(j = 1,..n)$ and P. The (n+2) unknowns include $\dot{M}_{l,m+1}, \dot{M}_{g,m}$, T_{m+1} and $x_{j,m+1}(j = 1,..n-1)$. They can be evaluated by an iterative scheme as described above for i = 1.

In this way, calculations can be carried out until the last incremental element i = p. The values obtained from the evaluations at i = p for $\dot{M}_{g,out}(=\dot{M}_{g,p})$ and $y_{j,p}(j = 1,..n)$ should match those obtained from overall balance equations. If a match is not found, either the increment size is too large and needs to be reduced or the accuracy of the calculations needs to be checked.

Once the above incremental calculations are completed, the term to be integrated in the design equation can be obtained as a function of $Q_{0\to m}$, the heat transferred from the vapor entrance till the end of the element i = m, as follows:

For $i = m$, $t = t_{m+1}$, $T_g = T_{g,m}$

$$Q_{0\to m} = \dot{m}_{cool}C_{p,cool}\left(t_{out} - t_{m+1}\right) \tag{4.57}$$

$$\frac{1}{\alpha_{oi,m}} = \frac{1}{\alpha_{cool,m}} + (\text{wall} + \text{fouling resistances}) + \frac{1}{\alpha_{c,m}}\left(\frac{d_o}{d_i}\right) \tag{4.58}$$

$\alpha_{c,m}$ is obtained from the strategy described in Chapter III, depending on the value of the condensate Reynolds number, $Re_{c,m}$.

$$Re_{c,m} = 4\frac{\Gamma_m}{\eta_c} = 4\frac{\dot{M}_{l,m}(Mw)}{\pi d_i \eta_c} \tag{4.59}$$

$\alpha_{g,m}$ is obtained from the correlation by Chilton and Colburn (1934) [Refer to Chapter III] using $\left(\frac{\dot{M}_{g,m-1} + \dot{M}_{g,m}}{2}\right)$. Z_m is calculated as follows :

$$Z_m = \frac{q_{sv}}{q_t} = \frac{\left(\frac{\dot{M}_{g,m-1} + \dot{M}_{g,m}}{2}\right)\left(H_{g,m-1} - H_{g,m}\right)}{\dot{m}_{cool}C_{p,cool}\left(t_m - t_{m+1}\right)} \tag{4.60}$$

C_f is assumed to be unity unless related experimental data are available to estimate it. This is justified as it has been shown by Owen et al.(1980) that C_f is close to unity for condensation inside a vertical tube with low vapor shear.

To determine $\theta_{s,m}$, the following procedure needs to be followed :

(a) Assume $\theta_{s,m} = 1$

b) Calculate $\left(\dfrac{1+\dfrac{Z\alpha_{oi}}{\alpha_g\theta_s}}{\alpha_{oi}\left(T_g - t\right)}\right)_{i=m}$, which is the same as $\left(\dfrac{1}{U_o\left(T_g - t\right)}\right)_{i=m}$

(c) Calculate the condensation molar flux, $\dot{n}_{T,m}$ from :

$$\dot{n}_{T,m} = \frac{\dot{M}_{g,m-1} - \dot{M}_{g,m}}{\Delta A_o} \tag{4.61}$$

where ΔA_o, the area of the increment can be determined as :

$$\Delta A_o = \frac{Q_m}{q_{t,m}} = \frac{\dot{m}_{cool} C_{p,cool}\left(t_m - t_{m+1}\right)}{U_o\left(T_{g,m} - t_{m+1}\right)} \tag{4.62}$$

(d) Calculate $\theta_{s,m}$ from $\theta_{s,m} = \left(\dfrac{a}{e^a - 1}\right)_{i=m}$ (Eqn. (4.22)), where a_m is given by $a_m = \dfrac{\dot{n}_{T,m} C_{pg}}{\alpha_{g,m}}$ (Eqn. (4.23)).

(e) If $\theta_{s,m}$ is not converged, go to step (b) with the last calculated value of $\theta_{s,m}$.

Thus, the value of $\left(\dfrac{1+\dfrac{Z\alpha_{oi}}{\alpha_g\theta_s}}{\alpha_{oi}\left(T_g - t\right)}\right)_{i=m}$ with the converged value of $\theta_{s,m}$ is obtained as a function of $Q_{0\rightarrow m}$.

In the above manner, calculations are carried out for all the increments (i = 1 to p) to obtain $\dfrac{1}{q_{t,i}}$ as a function of $Q_{0\rightarrow i}$. The required heat transfer surface can then be determined by integrating $\dfrac{1}{q_t}$ numerically or graphically with respect to Q.

The above described design procedure is for the case of partial condensation, where $\dot{M}_{g,out} \neq 0$. The problem of total condensation has to be dealt with differently.

For $\dot{M}_{g,out} = 0$, the overall total and component mole balance equations, Eqns.(45) and (46a,b,c) reduce to :

TOTAL : $$\dot{M}_{g,in} = \dot{M}_{l,out} \tag{4.63}$$

COMPONENT :

j=1,n-1 $$\dot{M}_{g,in} y_{j,in} = \dot{M}_{l,out} x_{j,out} \tag{4.64}$$

This means that the molar flowrate and the composition of the vapor-gas mixture and the condensate are identical to each other at the vapor entrance. The reflux ratio, R, defined as $R = \frac{\dot{M}_{l,out}}{\dot{M}_{g,in}}$ is always equal to unity.

The overall energy balance equation, Eqn. (4.47) reduces to :

$$Q_T = \dot{M}_{g,in} H_{g,in} - \dot{M}_{l,out} H_{l,out} \tag{4.65}$$

To determine Q_T using this equation, the temperature of the outlet condensate stream should be known. Assuming the condensate stream is saturated, this can be obtained from equilibrium calculations using $x_{j,out}(j = 1,..n)$ and P. Once Q_T is determined, the mass flowrate and the outlet temperature of the coolant stream can be fixed suitably such that Eqn. (4.48) is satisfied.

The incremental mole balance equations, Eqns. (4.49), (4.50a,b,c), (4.53) and (4.54a,b,c) reduce to :

TOTAL :

$$\dot{M}_{g,i} = \dot{M}_{l,i+1} \tag{4.66}$$

COMPONENT :

j=1,n-1

$$y_{j,i} = x_{j,i+1} \tag{4.67}$$

for all i, i = 1, p. This means that at every point of the condensation path, the molar flowrate and the composition of the vapor stream and the condensate are identical to each other.

Unlike the case of partial condensation, for total condensation, the increments are not fixed a priori as steps in the coolant temperature. Assuming that the liquid and the vapor phases leaving an increment are in equilibrium with each other, the increment calculations are carried out as follows. For i = 1, from Eqn. (4.67), it is known that :

$$x_{j,1}(j = 1,..n) = y_{j,0}(j = 1,..n) \tag{4.68}$$

$y_{j,1}(j = 1,..n)$and T_1 are determined from equilibrium calculations using $x_{j,1}(j = 1,..n)$ and P. Also, $x_{j,2}(j = 1,..n)$ is obtained using :

$$x_{j,2}(j = 1,..n) = y_{j,1}(j = 1,..n) \tag{4.69}$$

Similarly, for $i = m$, $x_{j,m}(j = 1,..n)$ [same as $y_{j,m-1}(j = 1,..n)$] are known. $y_{j,m}(j = 1,..n)$ [same as $x_{j,m+1}(j = 1,..n)$] and T_m are determined from equilibrium calculations using $x_{j,m}(j = 1,..n)$ and P.

The above calculations are repeated until, for an increment $i = p$, $y_{j,p}(j = 1,..n)$ is obtained very close to unity. This is the last increment and the number of increments thus is equal to p. The range of the coolant stream temperature, t from t_{in} to t_{out} is now divided into p equal parts, designating the size of each increment. For any increment $i = m$, the only unknown quantity is the molar flowrate of the condensate stream entering the increment, $\dot{M}_{l,m+1}$ [which is the same as the molar flowrate of the vapor stream leaving the increment, $\dot{M}_{v,m}$]. This is obtained using the incremental energy balance equation, Eqn. (4.55a) rewritten as :

$$Q_m = \dot{M}_{g,m-1}\left(H_{g,m-1} - H_{l,m}\right) - \dot{M}_{g,m}\left(H_{g,m} - H_{l,m+1}\right) \tag{4.70}$$

where Q_m is obtained from Eqn. (4.55b). The above calculation is carried out for all increments, $i = 1$ to p. For $i = p$, $\dot{M}_{l,p+1}$ should be obtained as zero.

The remaining calculations to determine the integrand $\dfrac{1}{q_{t,i}}$ in Eqn. (4.44) as a function of $Q_{0\rightarrow i}$ and then the required heat transfer surface by numerical or graphical integration are the same as for partial condensation.

CHAPTER V

PROCESS DESIGN AND MECHANICAL DESIGN

Process Design

The overall design procedure for a vertical vapor-in-tube reflux condenser is as follows

1. Select a tube diameter, d_i. Select a suitable tube inlet geometry and a convenient method of vapor injection and condensate removal. See Fig. 2.5, Chapter II. Calculate the flooding vapor velocity using the strategy described in Chapter II. This flooding velocity is for a case in which the vapor flow is distributed evenly between the tubes of a tube bundle. To correct for maldistribution effects, the design flooding velocity can be reasonably assumed to be 70% of the above value. The operating vapor velocity is then chosen as 0.7 times the corrected flooding velocity.
2. From the total vapor flow and the operating vapor velocity, determine the number of tubes required using :

$$\left(\dot{m}_g\right)_T = \rho_g v_g \left[\left(\frac{\pi d_i^2}{4}\right) N_t\right] \tag{5.1}$$

Select a suitable tube pitch and layout and determine the shell size required to hold N_t tubes. Use a full tube-count if more tubes can be accommodated. Note that there can only be one tube-side pass in a vertical vapor-in-tube reflux condenser.

If the reflux condenser is to be mounted directly on top of a reactor/distillation column, it is advisable to compare the shell diameter obtained with the diameter of the reactor/distillation column. Comparable shell diameters can be obtained by adjusting the tube diameter and the tube end conditions suitably.

3. Carry out the appropriate design calculations described in Chapter IV depending on - (a) whether the condensation is of a pure vapor or a multicomponent mixture, and (b) whether the condensation is total or partial.

   Though the above design procedures refer specifically to condensation in a single tube, they can be applied directly to a complete condenser tube bundle, as always, by assuming that all the tubes in the bundle behave identically. In the above evaluations, care should be taken however in calculating the tube-side heat transfer coefficients, which should be based on flowrates occurring in a single tube.

   The shell-side heat transfer coefficient can be calculated using the Delaware method for shell-side design or other known methods.

4. Calculate the required tube length from the heat transfer surface obtained above using :

$$A_o = (\pi d_o L) N_t \qquad (5.2)$$

5. Repeat the above procedure for different values of the design parameters such as tube diameter and layout, tube inlet geometry, coolant stream conditions, etc., to arrive at a 'good' design.

Mechanical Design

Some of the important mechanical design considerations for a vertical vapor-in-tube reflux condenser are :

1. Higher operating vapor velocities and thus a higher capacity utilization of the reflux condenser can be sought by :

(a) Use of large diameter tubes.

(b) Use of a suitable tube inlet geometry and a convenient method of vapor injection and condensate removal that can allow a smooth and steady countercurrent flow of vapor and condensate at the tube entrance for high relative velocities of flow. The different tube end conditions studied in the literature are shown in Fig 2.5, Chapter II. It is worthwhile to note here the relative simplicity of the angled tube-end design and its proven advantages over a square tube-end design, as shown by English et al. (1963).

However, care should be taken to avoid flooding and entrainment in the connected piping.

2. Venting for in-tube condensation is simple as the flow path is fixed and the point of accumulation of the non-condensables is clear. For a vertical vapor-in-tube reflux condenser, the vent nozzle should be placed at the top of the upper header. The shell-side should be vented through the upper tubesheet.
3. The differential expansions of the shell and the tubes of a heat exchanger due to large temperature differences are accounted for by suitable mechanical design features that avoid the build-up of thermal stress. For a discussion on the different construction features used to deal with the thermal stress problem, refer to Bell (1993). For a vertical vapor-in-tube reflux condenser, a fixed tubesheet with a shell-side expansion joint or a floating head design could be used depending on the magnitude of the thermal stress problem. Fixed tubesheet designs should work most of the time because of the relatively small temperature differences involved in a reflux condenser. Among floating head designs, the 'outside packed lantern-ring' arrangement could be preferred considering its advantages for a single tube-side pass construction.

CHAPTER VI

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. There is wide disagreement between the different analytical and experimental correlations available in the literature for predicting the flooding velocity in vertical tubes. The principal reasons for the disagreement are - (a) differences in the criteria used to define the flooding point, and (b) differences in the inlet tube-end conditions, whose effect on the flooding phenomenon is difficult to account for. The effect of condensation on the flooding phenomenon is also not clearly understood. Keeping the above points in mind, a suitable method has been developed for predicting the flooding vapor velocity for any reflux condenser design problem as accurately as possible.
2. The film flow hydrodynamics and the heat transfer aspects of a reflux condensation process are not well understood. For determining the condensing heat transfer coefficient, the available correlations for a cocurrent vapor-condensate flow situation with zero vapor shear should work well enough, as vapor shear is expected to be negligible throughout a normal reflux condensation process. However, a method has been devised to estimate the relative magnitudes of the forces of vapor shear and gravity on the condensate film. Based on this, a general strategy for predicting the local condensing heat transfer coefficient has been developed.
3. The film theory methods of condenser design are complicated to use. The accuracy of their predictions is greatly dependent on the accuracy of the input quantities and the relevant calculations. For a preliminary condenser design, the approximate methods should be preferred. In this work, an approximate generalized design procedure has been developed specifically for multicomponent total/partial reflux condensers, using

the Silver-Bell-Ghaly design equation modified by McNaught (1979) and Sardesai et al. (1983) [Eqn. (4.44)].

Recommendations

1. (a) An experimental study needs to be carried out to determine the effect of different tube end conditions on the flooding velocity in a vertical countercurrent adiabatic or condensing two-phase flow system. A means of uniquely characterizing the tube end conditions needs to be determined.
   (b) The relation between the different flooding point definitions used in the literature has to be established. Further studies on flooding should employ a common definition for the flooding point. Onset of liquid entrainment would be a good definition for the flooding point as it is the most restrictive and is likely to be the most important to a condenser designer who cannot allow condensate loss by entrainment.
   (c) The effect of condensation on the flooding phenomenon has to be investigated.
2. A complete experimental and analytical study has to be carried out to develop a reliable method of predicting the condensing heat transfer coefficient in a reflux condensation flow situation for a wide range of flow conditions.
3. Sample reflux condenser design problems have to be worked out using the design procedure developed in this work to confirm its correctness and offer modifications to it to improve it further.

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APPENDIXES

APPENDIX A

SUMMARY OF ANALYTICAL MODELS FOR VERTICAL COUNTERCURRENT FLOODING

Model	Basic Equation(s)	Empirical Equation(s)	Fluids Compared	Flooding Equation
Potential flow Imura et al.(1977)	Laplace Bernoulli	Mean film thickness Critical wavelength	Air-water Air-ethyl alcohol Air-n-heptane	$\bar{v}_g + \bar{v}_l = \frac{\sigma}{\rho_g}\left(k - \frac{1}{r - \partial}\right)^{1/2}$
Viscous flow Cetinbudaklar and Jameson (1969)	Orr-Sommerfeld	None	Air-water Air-glycerol	$\bar{v}_g = \bar{v}_g(\text{Re},\ kc_i,\ \text{fluid properties})$ $\bar{v}_{g,\text{cnt}} = \min(\bar{v}_g),\ \text{when } kc_i > 0$
Finite-amplitude wave Zvirin et al. (1979)	Momentum	Interfacial shear	Air-water Air-glycerol Air-silicone oil	$\frac{J_g^*}{\varepsilon} + \left(\frac{\rho_g}{\rho_l}\right)^{1/2} \frac{J_l^*}{1-\varepsilon} = \bar{v}_{cr}\left(\frac{\rho_g}{(\rho_l - \rho_g)gd}\right)^{1/2}$ $\frac{2f_i\left(J_g^*\right)^2}{\varepsilon^{2.5}} + \frac{2f_w\left(J_l^*\right)^2}{(1-\varepsilon)^2} = 1 - \varepsilon$
Separate cylinders Wallis (1969)	Continuity Momentum	None	Air-water	$\left(J_g^*\right)^{2/n+1} + \left(J_l^*\right)^{2/n+1} = 1,$ where n is a constant

APPENDIX A

(CONTINUED)

Separated flow Bharathan et al. (1979)	Momentum	Interfacial shear	Air-water	$F(\varepsilon, J_g^*, J_l^*) = \frac{2f_i (J_g^*)^2}{\varepsilon^{2.5}} - \frac{2f_w (J_l^*)^2}{(1-\varepsilon)^2} - (1-\varepsilon) =$ $G(\varepsilon, J_g^*, J_l^*) = \frac{\partial F}{\partial \varepsilon} = 0$
Stationary wave Shearer and Davidson (1965)	Wave profile	Pressure distribution	Air-water	$Ap = Ap(\bar{v}_g, \bar{v}_l, \text{ fluid properties})$ $\frac{d(Ap)}{d\bar{v}_g} \to \infty$
Hanging film Wallis and Kuo (1976)	Bernoulli	None	Air-water	$K_{g,crit} = 1.87$
Roll wave Richter (1981)	Momentum	Mean film thickness Interfacial shear	Air-water	$\frac{f_w}{4} Bo (J_g^*)^4 \left(Bo (J_g^* J_l^*)^2 + 4 \right) + 150 f_w (J_g^*)^2 = 1$

(Adapted from Bankoff and Lee, 1983)

APPENDIX B

SUMMARY OF EMPIRICAL CORRELATIONS FOR VERTICAL COUNTERCURRENT FLOODING

Authors	Tube size, m	Test fluids	Experimental Range	Flooding Equation
Kamei et al. (1954)	d = 0.019, 0.0316, 0.0418 L = 2.5	Air-water Air-millet jelly Air-soap solution	$50 < Re_l < 700$	$\frac{\rho_g j_g}{\rho_l j_l} = \frac{36.2}{Re_l^{1.225}} \left(\frac{\rho_g}{\rho_l}\right)^{0.13} \left(\frac{\eta_g}{\eta_l}\right)^{0.71} \left(\frac{d^3 \rho_l^2 g}{\eta_l^2}\right)^{0.231} Bo^{0.23}$
Feind (1960)	d = 0.02, 0.05 L = 0.96, 2.5	Air-water Air-diethylene glycol solution	$0.15 < Re_l < 3000$	$m \frac{Re_g}{Re_l^n} \left(\frac{\rho_l}{\rho_g}\right)^{0.4} \left(\frac{\eta_g}{\eta_l}\right)^{0.75} + 1.4 * 10^4 = 1300 \left(\frac{r}{\partial}\right)^{1.25}$ $m = 58.2$, $n = 0.33$ if $Re_l \le 400$ $m = 157.7$, $n = 0.5$ otherwise
Wallis (1961)	d = 0.0127, 0.019, 0.0254, 0.051 L = 1.22	Air-water		$\left(J_g^*\right)^{1/2} + \left(J_l^*\right)^{1/2} = C$ $C = 0.725 - 0.875$
Hewitt and Wallis (1963)	d = 0.0318	Air-water	$J_l^* < 0.30$	$\left(J_g^*\right)^{1/2} + \left(J_l^*\right)^{1/2} = 1$
English et al. (1963)	d = 0.019	Water Carbon tetrachloride n-propyl alcohol n-heptane	$0.1 \le \frac{L}{G} \le 1.0$	$j_g = 0.286 \left(\frac{d^{0.322} \rho_l^{0.419} \sigma^{0.097}}{\rho_g^{0.462} \eta_l^{0.150} j_l^{0.075}}\right)$

APPENDIX B

(CONTINUED)

Clift et al. (1966)	$d = 0.0318$ $L = 1.83$	Air-water Air-glycerol	$0.01 < J_l^* < 0.09$	$\left(J_g^*\right)^{1/2} + 0.34\left(J_l^*\right)^{1/2} = 0.79$ $K_{g,crit} = 2.24$
Tobilevich et al. (1968)	$d = 0.0327, 0.0525$ $L = 3.0$	Air-water Air-sugar solution Steam-water	$Re_l < 1800$	$Fr = a \exp\left\{ b\left(\frac{\rho_l}{\rho_g}\right)^{-0.2} K_g \right\}$ $a = 0.129$, $b = -14.14$ if $Fr > 0.012$ $a = 0.065$, $b = -10.12$ otherwise
Pushkina and Sorokin (1969)	$d = 0.31, 0.031, 0.012$ $L = 2.5$	Air-water	$0.005 < J_l^* < 0.064$	$K_{g,crit} = 3.2$
Diehl and Koppany (1969)	$d = 0.0266, 0.0525, 0.0158$	Hydrogen-diesel oil Air-water Steam-water		$j_g = F_1 F_2 \left(\frac{\sigma}{\rho_g}\right)^{0.5}$ $F_1 = \left(\frac{d}{\sigma/80}\right)^{0.4}$ if $F_1 < 1$ $F_1 = 1$ otherwise $F_2 = \left(\frac{\rho_g j_g}{\rho_l j_l}\right)^{0.25}$
Alekseev et al. (1972)		Steam-water	$J_l^* \leq 1.0$	$K_{g,crit} = 0.2576 Fr^{-0.22} Bo^{0.26}$

APPENDIX B

(CONTINUED)

Grolmes et al. (1974)	d = 0.004-0.025 L = 0.04-1.30	Nitrogen-water	$0.4 < Re_l < 140$	$\bar{v}_g = 1.15\left(\frac{\rho_l}{\rho_g}\frac{g\partial}{0.006+200\partial^2(\eta_{ref}/\eta)^{0.44}}\right)^{0.5}$
Hewitt (1977)	d = 0.0127, 0.0318 L = 1.22, 2.44	Air-water Air-glycerol Air-silicone oil	$0.01 < J_l^* < 0.275$	$\left(J_g^*\right)^{1/2} + \left(J_l^*\right)^{1/2} = C$ C depends on fluid property
Suzuki and Ueda (1977)	d = 0.029, 0.01, 0.018 L = 0.5 - 2.0	Air-water Air-glycerol		$\left(\frac{\rho_g(\bar{v}_g + \bar{v}_l)^2}{\rho_l g\partial}\right)^2 = a\log X + b$ $X = \left(0.63 Re_l^{-1/3} Bo^{1/4}\left(\frac{\eta_l}{\eta_g}\right)^{2/3}\right)$ a and b are tube length dependent
Dukler and Smith (1979)	d = 0.051 L = 3.94	Air-water	$77.5 < Re_l < 822.8$	$\left(J_g^*\right)^{1/2} + \left(J_l^*\right)^{1/2} = 0.88$
Chung et al. (1980)	d = 0.0159, 0.0318, 0.046, 0.070 L = 0.914	Air-water Air-silicone oil Air-white oil	$0.128 < K_l < 2.265$	$K_g^{1/2} + mK_l^{1/2} = C_1 \tanh\left(C_2 d^{*1/4}\right)$ m = 0.65 - 0.80 $C_1 = 1.79 - 2.1$, $C_2 = 0.8 - 0.9$

(Adapted from Bankoff and Lee, 1983)

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