

EXXON EXXON ENGINEERING	HEAT EXCHANGE EQUIPMENT CALCULATION PROCEDURE, CONDENSATION PROPRIETARY INFORMATION - For Authorized Company Use Only		DESIGN PRACTICES	
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Revision Memo

12/99 Minor revisions and updated information throughout the section.

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SCOPE

This section presents design considerations and the recommended manual design procedure for shell and tube condensers.

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1. Gloyer, W., *Industrial and Engr. Chemistry*, 42, 7, 1361, July, 1950.
2. Devore, A., *Petroleum Refiner*, 38, 6, 205, June, 1959.
3. Akers, Deans, Crosser, *Condensing Heat Transfer Within Horizontal Tubes*, Presented at the Second National Heat Transfer Conference, AIChE-ASME, Chicago, Illinois, August 18 - 21, 1958.
4. Diehl, J. E. and Uhrh, C. H., *Petroleum Refiner*, 36, 10, 124, October 1959.
5. Chenoweth, J. M. and Martin, M. W., *Petroleum Refiner*, 34, 151, October 1955.
6. Heat Exchange Institute (HEI) Standards for Steam Surface Condensers.
7. Perry's Chemical Engineers' Handbook, 6th Ed.

GENERAL

The same basic approach that is used to design liquid-liquid exchangers is also used to design condensers. This approach uses a trial-and-error calculation to balance an assumed exchanger size against a calculated size, while staying within the prescribed limits of pressure drop. The basic difference lies in the so-called "zoning" of the condenser.

▶ Zoning is necessary because the concept of LMTD (based on the entire heat exchanger) is invalid whenever the heat transfer mechanisms vary from one region to another within the exchanger. The LMTD concept assumes that the heat transfer coefficients and the specific heats remain constant for both fluids. This is not the case for condensers. However, we may break down the exchanger into a number of zones and the LMTD concept can be applied separately in each of these zones. These zones are selected in a manner that the heat transfer coefficients and the specific heats (represented by the slope of heat release (T-Q) curve may be assumed to be constant. Therefore, all phase-change points, such as dew points (both hydrocarbon and steam) and bubble point define a start of a new zone. Additional zones may be required if the slope of the T-Q curve varies appreciably.

It should be noted here that this discussion is primarily concerned with the more difficult case of condensing a wide-cut hydrocarbon in the presence of steam. For narrow cuts [15 to 20°F (8 to 11°C)], zoning is usually unnecessary and the following method simplifies considerably.

▶ HTRI has developed updated empirical correlations and procedures for computer analysis, which are explained in their design manual and reports. For design and rating of condensers, the CST-3 Computer Program is recommended. This program may be used stand-alone, or through the heat exchanger network analysis program HEXTRAN.

THEORY AND DESIGN CONSIDERATIONS

ZONES

In setting up the zones, one assumes that the heat transfer coefficient is constant within the zone and that heat release is directly proportional to temperature change within the zone.

One can see from this that the larger the number of condenser zones, the more accurate would be the condenser design. Unfortunately, the calculation of each zone is so time consuming that, for hand calculation, it is impractical to set up more than two or three zones. As shown in **Figure 1**, the usual zones are:

1. Vapor cooling.
2. Vapor cooling + hydrocarbon condensation and subcooling.
3. Vapor cooling + water condensation and subcooling + hydrocarbon condensation and subcooling.

Figure 1 is somewhat simplified, in that the coolant is shown for a one-tube-pass unit. In this typical sequence, the hydrocarbon dew point is above the steam dew point. If the steam dew point should occur first, Zone 2 is essentially eliminated.

When better accuracy is desirable, Zone 3 is sometimes divided into two "sub-zones" of roughly equal heat duty.

In the absence of desuperheating, steam condensation or any other sharp break in the heat release curve, one condenser zone is adequate.

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THEORY AND DESIGN CONSIDERATIONS (Cont)

THE BASIC STEPS

The following steps are basic to the design of a unit for condensing wide-cut hydrocarbons in the presence of steam.

1. Determine the hydrocarbon dew point.
2. Determine the steam dew point.
3. Determine the heat released in each zone.
4. Determine the simple LMTD for each zone, then Δt_{ew} (weighted effective log mean temperature difference, °F or °C).
5. Estimate an overall coefficient (i.e., total area) for the condenser; determine mechanical features.
6. Calculate area required for each zone; sum the area.
7. Adjust mechanical features of the unit until the assumed total area is equal to the sum of the areas calculated for each zone.

These basic steps will next be discussed generally.

Dew Points; The Flash Curve - For wide-cut condensation, the hydrocarbon dew point is easily found by trial-and-error reductions in temperature to find the point where hydrocarbon vapor pressure equals its known partial pressure. (For the usual case of a fractionator overhead, the hydrocarbon is already at its dew point.)

The steam dew point is more difficult, since the total number of moles of vapor changes between the hydrocarbon dew point and the steam dew point. This change in composition must be determined before the steam dew point can be calculated.

It is usual practice to assume that as condensation proceeds, the vapor and liquid maintain an equilibrium composition. Hence, the composition and thermal properties of the system are obtained from equilibrium flash calculations. These calculations predict a "flash curve," which is a plot of temperature vs. "percent off" (percent not condensed). The design manual method for estimating the flash curve is a simplification of the Exxon Blue Book method.

The entire flash curve is not constructed. Instead, several points along the volumetric flash curve are calculated. These are plotted and connected with straight lines. The molal flash curve lies slightly below the volumetric curve, and the weight flash curve lies above the volumetric curve. For the sake of simplification, this separation of the curves is assumed proportional to the width of certain boiling ranges in the cut. The three curves are plotted on the same graph and used to predict composition and temperature of the vapor and liquid phases as condensation proceeds. An example is shown in **Figure 2**.

Since these curves represent conditions at one atmosphere partial pressure, an alignment chart is included to convert to the pressure of the condensing system (**Figure 3**).

The steam dew point is then found by a trial-and-error reduction in the system temperature to find the temperature, where water vapor pressure equals its calculated partial pressure. The method is described in detail in **Table 1**.

Heat Release in Each Zone - The following individual heat duties are calculated in each zone in which they occur. All duties occur simultaneously in the third zone only.

1. Cool entering liquid.
2. Cool vapor and gas that do not condense.
3. Cool condensing HC vapor.
4. Cool hydrocarbon condensate.
5. Cool condensing steam.
6. Cool steam condensate.
7. Remove latent heat of hydrocarbon.
8. Remove latent heat of steam.

To calculate Duties 3, 4, 5, and 6, one assumes that 50% (or the average quantity) of the material is cooled through 100% of the temperature change in the zone.

Although enthalpy charts can be used for these calculations, the use of average specific heats and average latent heats is adequate for most cases. Note that when enthalpy charts are used, Items 7 and 8 must be calculated at the average temperature (rather than inlet to outlet conditions) to avoid duplicating the sensible-heat portion of the duty.

Temperature Driving Force - After the zone heat duties have been calculated, the weighted average LMTD (Δt_{ew}) can be calculated. This is done by calculating the LMTD for each zone and then "weighting" the LMTD according to the amount of heat transferred in the zone. The conventional equation is as follows:

$$\Delta t_{ew} = \frac{Q}{\frac{q_{dh}}{\Delta t_{dh}} + \frac{q_{ds}}{\Delta t_{ds}} + \frac{q_{sc}}{\Delta t_{sc}}} \quad \text{Eq. (1)}$$



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THEORY AND DESIGN CONSIDERATIONS (Cont)

For explanation of symbols, see **NOMENCLATURE**.

Each zone Δt is corrected for noncountercurrent flow by multiplying by F_n from **Figure 2** of **Section IX-D**. F_n is calculated from the zone terminal temperatures.¹ This use of F_n is not theoretically sound but is justified, as a matter of convention, to make sure that enough shell passes are specified for the design temperature cross.

The use of this equation for weighting Δt 's is strictly a matter of industrial convention. It gives the true value of Δt_{ew} only when the coefficient is constant over the entire condenser; and this condition, of course, almost never prevails. But since the equation has been widely adopted, the use of other methods for weighting Δt 's might be misleading to exchanger suppliers.

In reality, Δt_{ew} does not enter into the condenser design and its value is of no real importance. The design is based on a summation of zone areas which are calculated from the individual zone Δt 's. The overall Δt_{ew} is estimated only so that an overall average U_o can be reported.

Heat Transfer Coefficients, Shell-Side Condensation - From an analytical viewpoint, a condenser is an extremely complex piece of equipment. The approach used in this manual is sometimes called the "Gloyer" (**Reference 1**) analysis, named after the first person to describe it in the literature. This method is not a theoretical analysis, but an empirical attempt to consider the various phenomena which occur simultaneously in a condenser. This approach considers the following mechanisms:

1. Vapor cooled by forced convection.
2. Liquid cooled by "dripping" from tube to tube.
3. Liquid cooled by forced convection along the bottom of the shell.
4. Condensation.

In each zone of the condenser, a heat transfer coefficient "h" is calculated for each of the above mechanisms. The heat transfer coefficient for the zone is then evaluated by weighting and averaging the individual coefficients:

$$h_{\text{zone}} = \frac{Q_{\text{zone}}}{\frac{q_1}{h_1} + \dots + \frac{q_4}{h_4}} \quad \text{Eq. (2)}$$

(Subscripts refer to above mechanisms.)

The overall zone coefficient is then calculated from the usual resistance equation:

$$\frac{1}{U_{\text{zone}}} = \frac{1}{h_{\text{zone}}} + r_o + r_w + r_{io} + R_{io} \quad \text{Eq. (3)}$$

and the area from

$$A_{\text{zone}} = \frac{Q_{\text{zone}}}{U_{\text{zone}} \Delta t_{\text{zone}}} \quad \text{Eq. (4)}$$

The total area of the condenser is simply the sum of the zone areas.

To calculate the coefficients for vapor cooling and "bottom flow" liquid cooling, one uses the average vapor and liquid quantities in the zone. These quantities are determined with the aid of **Figure 4**. This chart corrects for the variations in ΔT (between tube-side and shell side) with tube length. The usual relations for single-phase heat transfer are then used to calculate coefficients.

The coefficient for "drip cooling" is taken as 1.5 times the condensing coefficient. It is arbitrarily assumed that half of the liquid cooling duty is by "drip cooling" and half by "bottom flow" (for zones with all vapor entering).

To calculate the condensing coefficient, the following modifications of the Nusselt equation recommended by Devore should be used (**Reference 2**).

¹ For hydrofiner effluent-hydrofiner feed exchangers and other similar services where there is a "pinch" in the T-Q curves of the two streams, each shell of the final arrangement should be checked graphically for a temperature cross (i.e., plot shell-side and tube-side temperatures vs. duty curves on a single graph and compare inlet vs. outlet temperatures for each shell). If a temperature cross occurs, either the shell areas or the number of shells should be adjusted to remove the cross.

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THEORY AND DESIGN CONSIDERATIONS (Cont)

For Vertical Tubes

$$h_{\lambda} = \left(\frac{1.04 \times 10^4}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Customary}) \quad \text{Eq. (5)}$$

or

$$h_{\lambda} = \left(\frac{254}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Metric}) \quad \text{Eq. (5)M}$$

where:

$$\Gamma = \frac{12 W_c}{\pi N_t d_o} \quad (\text{Customary}) \quad \text{Eq. (6)}$$

or

$$\Gamma = \frac{1000 W_c}{\pi N_t d_o} \quad (\text{Metric}) \quad \text{Eq. (6)M}$$

This term is usually called "tube loading." The equation is restricted to:

$$\frac{\Gamma}{\mu_f} \leq 1090 \quad (\text{Customary})$$

or

$$\frac{\Gamma}{\mu_f} \leq 450 \quad (\text{Metric})$$

For Horizontal Tubes

$$h'_{\lambda} = \left(\frac{8.33 \times 10^3}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Customary}) \quad \text{Eq. (7)}$$

or

$$h'_{\lambda} = \left(\frac{204}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Metric}) \quad \text{Eq. (7)M}$$

where:

$$\Gamma = \frac{W_c}{L_c n_s} \quad \text{Eq. (8)}$$

The term n_s is the number of "condensate streams" in the bundle.

$$\begin{aligned} n_s &= 1.29 N_t^{0.480} \text{ for square tube layout, } 90^\circ. \\ n_s &= 2.08 N_t^{0.495} \text{ for rotated triangular layout, } 60^\circ. \\ n_s &= 1.02 N_t^{0.519} \text{ for triangular layout, } 30^\circ. \\ n_s &= 1.37 N_t^{0.518} \text{ for rotated square layout, } 45^\circ. \end{aligned}$$



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THEORY AND DESIGN CONSIDERATIONS (Cont)

The term L_c is the length of tube on which condensation occurs. If a condenser consists of one zone, L_c equals the tube length. Otherwise L_c is the fraction of the total tube length included in the zone in question. (If there are two shells in series, note that one zone might include 0.5L, the next zone 1.5L.)

The paper by Devore suggests that the value of h'_λ be increased by a multiplier which depends essentially on the number of tubes in the bundle. For a typical bundle, the value of this so-called "turbulence factor" is 1.5 to 2.5. This has not been included in this procedure. Instead, a modification is recommended which will relate the condenser design to vapor mass velocity. (An effect of mass velocity has been recognized for some time.)

A great deal of Exxon data on wide-cut condensation show that the overall outside coefficient varies linearly with G^x , where x is between 0.7 and 0.9. A similar effect probably holds for narrow cuts as well. Based on this, the condensing coefficient for horizontal bundles should be adjusted as follows:

$$h_\lambda = h'_\lambda \left(\frac{G_v}{5} \right)^{0.70} \quad (\text{Customary}) \quad \text{Eq. (9)}$$

or

$$h_\lambda = h'_\lambda \left(\frac{G_v}{24.4} \right)^{0.70} \quad (\text{Metric}) \quad \text{Eq. (9)M}$$

and

$$h_\lambda \text{ max} = 2.0 h'_\lambda \quad \text{Eq. (10)}$$

Where G_v is the average vapor mass velocity in the zone. Note that this mass velocity correction is not recommended for pure-component condensation.

In zones where both steam and hydrocarbon are condensing, the hydrocarbon wets the tubes preferentially. This forces the steam to condense on the hydrocarbon condensate film and the condensing coefficient for steam equals the condensing coefficient for the hydrocarbon.

Empirical Equation for Shell-Side Condensation - Not Recommended for Definitive Designs - Exxon has analyzed a considerable amount of data on wide-cut condensation. These analyses have resulted in an equation which fits the original data with an average accuracy of 10%. So far, however, the equation has not been tested on enough additional data to substantiate its use for definitive designs. The equation can be used for estimating coefficients and for checking condenser performance under various operating conditions. Because of its simplicity, it is hoped that an equation of this type will eventually be backed up by enough data to warrant use in design work.

The equation for the shell-side condensing coefficient, which includes the shell-side fouling factor, is as follows:

$$\ln(h_{oc}) = 0.82 \ln(G) + f(\alpha) - 0.002P_i \quad (\text{Customary}) \quad \text{Eq. (11)}$$

or

$$\ln(h_{oc}) = 0.82 \ln(G) + f(\alpha) - 0.00029P_i + 0.436 \quad (\text{Metric}) \quad \text{Eq. (11)M}$$

The function $f(\alpha)$ is shown in **Figure 5**. This curve shows that the shell-side coefficient first decreases sharply with increasing noncondensables, then slowly increases. "G" is inlet mass velocity and "P_i" is inlet pressure (absolute). The data analyzed include condensers in the following services:

1. Fluid coking - scrubber overhead.
2. Cat cracking - fractionator overhead.
3. Fluid hydroforming - scrubber overhead.
4. Pipestill overhead.
5. Prefractionator overhead.

The total range of the variables is as follows:

$$\begin{array}{llll} 1.47 < G < 20.9 \text{ lb/sec-ft}^2 & (\text{Customary}) & \text{or} & 7.18 < G < 102 \text{ kg/s-m}^2 & (\text{Metric}) \\ 0.008 < \alpha < 0.896 & & & & \\ 19.2 < P_i < 205 \text{ psia} & (\text{Customary}) & \text{or} & 132 < P_i < 1410 \text{ kPa abs} & (\text{Metric}) \end{array}$$

The equation includes no data on narrow-cut condensation and will give answers much too low for such cases.

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THEORY AND DESIGN CONSIDERATIONS (Cont)

Heat Transfer Coefficient: Condensation on the Tube Side - Condensation inside the tubes is seldom used in industry. Hence, little is known about the heat transfer coefficients in such service. In general, the coefficients are lower than for shell-side condensation.

If the condenser is vertical, Nusselt's analysis is undoubtedly as valid inside the tubes as on the outside. But in the usual case, such as an air-fin condenser, the unit would be horizontal. This tends to fill part of the tube cross-section with condensate, blanking some of the condensing area. This mechanism is completely outside of Nusselt's work.

The following empirical method has been proposed by Akers, Deans and Crosser (**Reference 3**). The method is based on data for condensing propane and Freon-12 in horizontal tubes. The authors correlated their data within about 20% by using an "equivalent" liquid mass velocity and the usual single-phase heat transfer relationships. Neither the Nusselt equation nor the Carpenter-Colburn equation would correlate the data.

The basic equations are as follows:

$$h_{\ell} = 5.03 \text{ Re}^{1/3} \text{ Pr}^{1/3} (12 k_f / d_i) \quad \text{Re} < 50,000 \quad (\text{Customary}) \quad \text{Eq. (12)}$$

or

$$h_{\ell} = 5.03 \text{ Re}^{1/3} \text{ Pr}^{1/3} (1000 k_f / d_i) \quad (\text{Metric}) \quad \text{Eq. (12)M}$$

$$h_{\ell} = 0.0265 \text{ Re}^{0.8} \text{ Pr}^{1/3} (12 k_f / d_i) \quad \text{Re} > 50,000 \quad (\text{Customary}) \quad \text{Eq. (13)}$$

or

$$h_{\ell} = 0.0265 \text{ Re}^{0.8} \text{ Pr}^{1/3} (1000 k_f / d_i) \quad (\text{Metric}) \quad \text{Eq. (13)M}$$

$$\text{Re} = \frac{G_e d_i}{29 \mu} \quad (\text{Customary}) \quad \text{Eq. (14)}$$

or

$$\text{Re} = \frac{G_e d_i}{1000 \mu} \quad (\text{Metric}) \quad \text{Eq. (14)M}$$

$$G_e = G'_\ell + G'_v \left(\frac{\rho_\ell}{\rho_v} \right)^{1/2} \quad \text{Eq. (15)}$$

G'_ℓ and G'_v are superficial average mass velocities of the liquid and vapor, respectively, based on the total flow area. The Nusselt, Prandtl and Reynolds numbers are in dimensionless forms. Physical properties used are those of the condensate film at average film temperature.

It was found that these equations correlated the data for vertical, inclined, and horizontal tubes.

Eqs. (12) through (15) are for the condensing coefficient only, and they would be applicable by themselves to narrow-cut condensation. When condensing a wide cut, the designer should follow the principles used on the shell side for wide cuts and calculate coefficients for gas and liquid cooling. For gas cooling, use the average vapor velocity in the zone to calculate a single-phase vapor heat transfer coefficient. The liquid cooling coefficient may be assumed to be equal to the condensing coefficient. There is, of course, no drip cooling and hence no required breakdown of the liquid cooling duty.

Pressure Drop - Pressure drop in condensers is a complex variation of two-phase flow that has not been successfully analyzed theoretically. The data are amenable to correlation, however, and such correlations are frequently reported in the literature.

The recommended procedure treats the two-phase system as a single phase, using the viscosity of the liquid and a density corresponding to the average density for both phases. The weighted average density is evaluated for each zone based on the inlet and outlet volume flow rates, or

$$\text{Average density} = \frac{2 \times \text{Mass rate}}{(\text{Vol. rate in}) + (\text{Vol. rate out})}$$

This calculation method is applicable for both tube-side and shell-side pressure drops using weighted average values of density and velocity. Both tube-side and shell-side pressure drops are evaluated by the "no change of phase" procedure as outlined in [Section IX-D](#).

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THEORY AND DESIGN CONSIDERATIONS (Cont)

The two-phase pressure drop calculated for each zone is multiplied by the ratio (Zone Area/Total Area) to estimate the zone pressure drop. The overall pressure drop is obtained by summing the zone pressure drops.

This procedure is believed to be conservative, i.e., actual pressure drops should be equal to or less than the calculated values. In cases where pressure drop is expensive or where a much lower than expected pressure drop would cause process difficulties, an independent check can be made by use of the Lockhart & Martinelli correlation (see Perry's Chemical Engineers' Handbook, 6th Ed., pages 5-40 to 5-42). This method is considered to be the best available generalized correlation for two-phase pressure drop in a system with heat (mass) transfer taking place. However, the accuracy of this correlation is only $\pm 50\%$. Thus, the actual pressure drop could be anywhere between 50% and 150% of the calculated value. Also, the HTRI CST-3 computer program is incorporated into HEXTRAN program and is available for problems requiring detailed analysis.

SPECIAL DESIGN CONSIDERATIONS FOR DRUMLESS CONDENSERS

When drumless condensers are designed, the following design criteria should be followed:

1. The condenser surface should be 110% of the surface required for condensing in order to account for surface normally covered by liquid.
2. The condenser should be equipped with a 2 in. (50 mm) vent located as close to the liquid outlet end as practicable.
3. A pot for liquid-vapor separation should be located on the condenser outlet. Pots up to 14 in. (356 mm) diameter should be sized for a liquid velocity of 1 ft (0.3 m) per second. The velocity in pots 16 in. (406 mm) and larger should be limited to 1.5 ft (0.5 m) per second. The pot should be 3 to 5 ft (0.9 to 1.5 m) long.
4. The condenser shell should be equipped with a gauge glass covering the entire shell diameter and pot.
5. The condenser should be elevated sufficiently to satisfy pump NPSH requirements with the condenser pot empty. In any event, the bottom of the condenser shell should be a minimum of 20 ft (6 m) above grade.
6. An anti-vortex baffle should be installed in the condenser pot.
7. The condenser should be located with its liquid outlet end on the pump side of the structure. All pump suction piping should be continuously sloped downwards towards the pump. Horizontal runs on the pump suction piping should be sloped at least 2 in. per 100 ft (50 mm per 30 m).
8. Pump suction lines up to and including 3 in. (76 mm) diameter should not have a liquid velocity in excess of 1.5 ft/sec (0.5 m/s). Four to 8 in. lines should be limited to 2.25 ft/sec (0.7 m/s). For 10 in. (254 mm) diameter and larger sizes, the liquid velocity should not exceed 3.5 ft/sec (1.1 m/s).
9. Pumps should be heavy-duty refinery type.
10. Pumps should be equipped with a recirculation line returning upstream of the condenser. The recirculation line should be equipped with a block valve and a restriction orifice sized for 25% of the normal pump capacity.

DETAILED DESIGN PROCEDURE

Table 2 lists the step-by-step procedure in the form of a calculation sheet. Each step is discussed below.

The condensation calculation procedure presented in this section is based on hydrocarbon (with or without small amounts of steam, less than 15% by weight) condensation. The procedure should not be used for the design of vacuum pipestill overhead condensers or steam surface condensers. A sizing procedure for vacuum pipestill overhead condensers is presented later in this section. Steam surface condensers should be sized using the procedures presented in the Heat Exchange Institute (HEI) Standards for Steam Surface Condensers.

1. Determine the inlet and outlet temperatures, t_1 , t_2 , T_1 and T_2 .
2. Determine the minimum number of shells in series; F_n (**Figure 2, Section IX-D**) based on exchanger inlet and outlet temperatures equal to or greater than 0.8.
3. Determine the overall duty Q and the weighted overall LMTD, Δt_{ew} .
 - a. If a T-Q curve (temperature vs. heat release) has already been prepared, determine Δt_{ew} as follows:
 - (1) From the curve, determine Q , T_{dh} and q_{dh} , T_{ds} and q_{ds} and q_{sc} .
 - (2) Calculate t_{dh} and t_{ds} .

$$t_{dh} = t_2 - (t_2 - t_1) \frac{q_{dh}}{Q} \quad \text{Eq. (16)}$$

$$t_{ds} = t_2 - (t_2 - t_1) \frac{q_{dh} + q_{ds}}{Q} \quad \text{Eq. (17)}$$



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(3) Calculate Δt_{dh} , Δt_{ds} and Δt_{sc} . These are the LMTDs in each of the three zones. Apply individual F_n 's (**Figure 2, Section IX-D**) to each zone, as calculated from zone terminal conditions.

(4) Calculate Δt_{ew}

$$\Delta t_{ew} = \frac{Q}{\frac{q_{dh}}{\Delta t_{dh}} + \frac{q_{ds}}{\Delta t_{ds}} + \frac{q_{sc}}{\Delta t_{sc}}} \quad \text{From Eq. (1)}$$

b. If a T-Q curve is not available, the method outlined in **Table 1** may be used.

4. Estimate the total area required for the condenser. See **Table 1, Section IX-C** for typical values of U_o .

$$A = Q / U_o \Delta t_{ew} \quad \text{Eq. (18)}$$

5. Calculate area per shell, number of shells and number of tubes according to the usual method.

6. Use the appropriate section of the manual to calculate the transfer coefficient and pressure drop for the noncondensing side.

7. Calculate U_{ox} , the overall coefficient neglecting the resistance on the condensing side.

$$\frac{1}{U_{ox}} = r_o + r_w + r_{io} + (R_o \text{ or } R_{io}) \quad \text{Eq. (19)}$$

8. For the condensing side, estimate the pressure drop so that the baffle pitch (or number of tube passes) can be established.

a. Calculate average mixture density ρ based on inlet and outlet flow volumes.

b. Assume a value of P_b (or N_p) and calculate ΔP . Use liquid viscosity at average condensing temperature with equations for no change of phase.

c. Adjust P_b or N_p to obtain desired approximate pressure drop. Note that this is an approximate pressure drop, used only in setting up P_b or N_p . A definitive ΔP will be calculated later.

9. Calculate A_{dh} , the area required for the desuperheating zone of the condenser.

a. Calculate the gas cooling coefficient (h_{gc}) using the usual method of no change of phase.

b. Calculate U_{dh} and A_{dh} .

$$U_{dh} = \frac{h_{gc} U_{ox}}{h_{gc} + U_{ox}} \quad \text{Eq. (20)}$$

$$A_{dh} = q_{dh} / U_{dh} \Delta t_{dh} \quad \text{Eq. (21)}$$

10. Calculate A_{ds} , the area required for the hydrocarbon condensation and steam desuperheating zone.

a. Estimate the condensate film temperature in the zone, assuming no temperature drop across the vapor film.

For Shell-Side Condensation

$$t_f = t_s - (1/2) (t_s - t_1) \left(1 - \frac{U_{zone}}{U_{ox}} \right) \quad \text{Eq. (22)}$$

For Tube-Side Condensation

$$t_f = t_t - (1/2) (t_t - t_s) \left(1 - \frac{U_{zone}}{U_{ox}} \right) \quad \text{Eq. (23)}$$

All temperatures refer to the zone only, note the overall condenser.

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- b. Calculate W_v , the average vapor flow rate in the zone. Use **Figure 4**.
c. Calculate W_ℓ , average liquid flow rate.

$$W_\ell = W - W_v \quad \text{Eq. (24)}$$

- d. Determine μ_f for the vapor; and μ_f and s_f for the liquid.
e. Calculate h'_{λ} the condensing coefficient.

(1) If on the tube side, calculate G_e and Re , obtain f , and calculate $h_{t\lambda}$.

$$G'_v = \frac{N_p W_v}{19.6 N_t d_i^2} \quad \text{(Customary)} \quad \text{Eq. (25)}$$

or

$$G'_v = 1.273 \times 10^6 \frac{N_p W_v}{N_t d_i^2} \quad \text{(Metric)} \quad \text{Eq. (25)M}$$

$$G'_\ell = \frac{N_p W_v}{19.6 N_t d_i^2} \quad \text{(Customary)} \quad \text{Eq. (26)}$$

or

$$G'_\ell = \frac{N_p W_v}{N_t d_i^2} \quad \text{(Metric)} \quad \text{Eq. (26)M}$$

$$G_e = G'_\ell + G'_v (\rho_\ell / \rho_v)^{1/2} \quad \text{From Eq. (15)}$$

$$Re = \frac{G_e d_i}{29 \mu} \quad \text{(Customary)} \quad \text{From Eq. (14)}$$

or

$$Re = \frac{G_e d_i}{1000 \mu} \quad \text{(Metric)} \quad \text{From Eq. (14)M}$$

Calculate $h_{t\lambda}$ from **Eq. (12)** or **(13)**.

(2) If on the shell side,

A. Vertical bundle

$$\Gamma = \frac{12 W_c}{\pi N_t d_o} \quad \text{(Customary)} \quad \text{From Eq. (6)}$$

or

$$\Gamma = \frac{1000 W_c}{\pi N_t d_o} \quad \text{(Metric)} \quad \text{From Eq. (6)M}$$

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THEORY AND DESIGN CONSIDERATIONS (Cont)

$$h_{\lambda} = \left(\frac{1.04 \times 10^4}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Customary}) \quad \text{From Eq. (5)}$$

or

$$h_{\lambda} = \left(\frac{254}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Metric}) \quad \text{From Eq. (5)M}$$

If Γ / μ_f is larger than 1,100 (or 450), the value of h_{λ} is quite conservative.

B. Horizontal bundle

Calculate n_s , the number of condensate streams in the bundle.

$n_s = 1.29 N_t^{0.480}$ for square tube layout, 90°

$n_s = 2.08 N_t^{0.495}$ for rotated triangular layout, 60°

$n_s = 1.02 N_t^{0.519}$ for triangular layout, 30°

$n_s = 1.37 N_t^{0.518}$ for rotated square layout, 45°

Calculate Γ , the tube loading. For this step A_{ds} must be assumed. The final calculated value of A_{ds} should be checked back for agreement (10 to 15%) with the assumed value.

$$\Gamma = \frac{W_c}{L_c n_s} \quad \text{From Eq. (8)}$$

$$L_c = \frac{A_{ds}}{A} (L - 0.5) N_s \quad (\text{Customary}) \quad \text{Eq. (27)}$$

or

$$L_c = \frac{A_{ds}}{A} (L - 0.152) N_s \quad (\text{Metric}) \quad \text{Eq. (27)M}$$

Calculate h'_{λ} , the condensing coefficient uncorrected for vapor velocity.

$$h'_{\lambda} = \left(\frac{8.33 \times 10^3}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Customary}) \quad \text{From Eq. (7)}$$

or

$$h'_{\lambda} = \left(\frac{204}{\Gamma^{1/3}} \right) k_f \left(\frac{s_f^2}{\mu_f} \right)^{1/3} \quad (\text{Metric}) \quad \text{From Eq. (7)M}$$

Calculate x_v , the fraction of shell free-flow area occupied by the vapor.

$$x_v = 1 - x_{\ell} \quad \text{Eq. (28)}$$

$$\frac{1}{x_{\ell}} = 1 + \left(\frac{W_v}{W_{\ell}} \right) \left(\frac{\mu_v}{\mu_{\ell}} \right)^{0.111} \left(\frac{\rho_{\ell}}{\rho_v} \right)^{0.555} \quad \text{Eq. (29)}$$



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Calculate average mass velocity of the vapor in the zone

$$G_v = \frac{W_v}{25 A_x x_v} \quad (\text{Customary}) \quad \text{Eq. (30)}$$

or

$$G_v = \frac{10^6 W_v}{A_x x_v} \quad (\text{Metric}) \quad \text{Eq. (30)M}$$

Calculate h_λ

$$h_\lambda = h'_\lambda \left(\frac{G_v}{5} \right)^{0.70} \quad (\text{Customary}) \quad \text{From Eq. (9)}$$

or

$$h_\lambda = h'_\lambda \left(\frac{G_v}{24.4} \right)^{0.70} \quad (\text{Metric}) \quad \text{From Eq. (9)M}$$

$$h_\lambda \text{ max} = 2.0 h'_\lambda \quad \text{From Eq. (10)}$$

f. Calculate h_{gc} , the vapor cooling coefficient.

(1) If on the shell side:

Use G_v , the average vapor mass velocity, to calculate the uncorrected vapor cooling coefficient h_{gc} . Use correlations for no change of phase. Use the same film temperature as calculated for the condensing coefficient.

Correct for the fact that heat flowing through the gas film must also overcome resistance of the condensate film.

$$\frac{1}{h_{gc}} = \frac{1}{h_\lambda} + \frac{C_1}{h'_{gc}} \quad \text{Eq. (31)}$$

where:

$$C_1 = 1 - \left(\frac{W_c}{W_{v1}} \left(\frac{T_1 + T_2}{2} - t_f \right) \right) \quad \text{Eq. (32)}$$

The factor "C₁" helps account for the additional gas cooling surface available from liquid droplets, ripples, etc. The temperatures T_1 and T_2 refer to the zone inlet and outlet temperatures.

The value of C_1 should fall in the range of 0.5 to 1.0. If the calculated value of C_1 is less than 0.5, set C_1 equal to 0.5; if the calculated value of C_1 is greater than 1.0, set C_1 equal to 1.0.

(2) If on the tube side:

Calculate G_v .

A. Calculate x_v [see Eq. (28)]

B. $G_v = G'_v / x_v$ Eq. (33)

Use G_v with the procedure for no change of phase and calculate h'_{gc} .

Calculate h_{gc} .

$$\frac{1}{h_{gc}} = \frac{1}{h_\lambda} + \frac{C_1}{h'_{gc}} \quad \text{Eq. (34)}$$



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THEORY AND DESIGN CONSIDERATIONS (Cont)

- g. Calculate h_{lc} , the liquid cooling coefficient. For vertical bundles, assume $h_{lc} = h_{\lambda}$.

For horizontal bundles:

- (1) If on the shell side

Calculate G_{ℓ}

$$G_{\ell} = \frac{W_{\ell}}{25 A_x x_{\ell}} \quad (\text{Customary}) \quad \text{Eq. (35)}$$

or

$$G_{\ell} = \frac{10^6 W_{\ell}}{A_x x_{\ell}} \quad (\text{Metric}) \quad \text{Eq. (35)M}$$

Using G_{ℓ} , with the procedure for no change of phase, calculate h'_{lc} , liquid cooling obtained by "bottom flow."

Calculate the liquid cooling obtained by "drip cooling."

$$h_{dc} = 1.5 h_{\lambda} \quad \text{Eq. (36)}$$

Calculate h_{lc}

For all vapor entering;

$$h_{lc} = \frac{2h_{dc} h'_{lc}}{h_{dc} + h'_{lc}} \quad \text{Eq. (37)}$$

For liquid + vapor to the zone;

$$h_{lc} = \frac{(W_{\ell} + W_c) h_{dc} h'_{lc}}{\frac{W_c}{2} h'_{lc} + \left(W_{\ell} + \frac{W_c}{2}\right) h_{dc}} \quad \text{Eq. (38)}$$

- (2) If on the tube side

$$h_{lc} = h_{\lambda} \quad \text{Eq. (39)}$$

- h. Calculate h_{ds} , the weighted zone coefficient on the condensing side.

$$h_{ds} = \frac{q_{ds}}{\frac{q_{\lambda}}{h_{\lambda}} + \frac{q_{gc}}{h_{gc}} + \frac{q_{lc}}{h_{lc}}} \quad \text{Eq. (40)}$$

- i. Calculate U_{ds} , the zone coefficient.

$$U_{ds} = \frac{h_{ds} U_{ox}}{h_{ds} + U_{ox}} \quad \text{Eq. (41)}$$

- j. Calculate A_{ds}

$$A_{ds} = \frac{q_{ds}}{U_{ds} \Delta t_{e(ds)}} \quad \text{Eq. (42)}$$

- k. Compare A_{ds} calculated here with A_{ds} assumed to calculate Γ . If necessary, repeat calculation to obtain agreement.

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THEORY AND DESIGN CONSIDERATIONS (Cont)

11. Calculate A_{ds} , the area required for the steam condensing zone. Calculations for this zone are performed in the same way as for the previous zone. Note the following:
 - a. The condensing coefficient for steam equals the hydrocarbon condensing coefficient.
 - b. The cooling coefficient for condensed steam equals the hydrocarbon liquid cooling coefficient..
12. Calculate the total condenser area.

$$A = A_{dh} + A_{ds} + A_{sc} \quad \text{Eq. (43)}$$

13. If the area calculated in the previous step equals the area estimated in Step (4), proceed to the next step.
If the calculated area is not equal to the estimated area, repeat the calculation with a better estimated area until agreement is reached.
14. Calculate the overall coefficient, U_o .

$$U_o = \frac{Q}{A \Delta t_{ew}} \quad \text{Eq. (44)}$$

15. Calculate the clean coefficient, U_c

$$\frac{1}{U_c} = \frac{1}{U_o} - r_{io} - r_o + 0.001 \quad \text{(Customary)} \quad \text{Eq. (45)}$$

or

$$\frac{1}{U_c} = \frac{1}{U_o} - r_{io} - r_o + 0.00018 \quad \text{(Metric)} \quad \text{Eq. (45)M}$$

16. Calculate pressure drop on the condensing side.
 - a. Refer to Step 8 and calculate two-phase pressure drops for the overall condenser at the average conditions of each zone.
 - b. Calculate zone pressure drops

$$\Delta P_{\text{zone}} = \Delta P_{\text{at zone conditions}} \left(\frac{A_{\text{zone}}}{A_{\text{total}}} \right) \quad \text{Eq. (46)}$$

- c. Calculate ΔP

$$\Delta P = \Delta P_{dh} + \Delta P_{ds} + \Delta P_{sc} \quad \text{Eq. (47)}$$

VACUUM PIPESTILL OVERHEAD CONDENSER DESIGN PROCEDURE

Since a vacuum pipestill overhead condenser heat duty primarily results from condensing steam, the detailed condenser calculation procedure presented in this section is not applicable for design of these units. However, the HEXTRAN Program with HTRI CST-3 program may be used. For preliminary hand calculations, when sizing a vacuum pipestill overhead condenser, the following calculation procedure is recommended.

1. Use an overall heat transfer coefficient of 130 Btu/hr-ft²-°F or 738 W/m²-K. (Values between 110 to 130 Btu/hr-ft²-°F (625 and 738 W/m²-K) have been used for past designs.)
2. Use the steam dew point temperature, rather than the hydrocarbon dew point temperature, as the condensing zone inlet temperature for calculating the effective temperature difference.
3. Design for 3 to 12 mm Hg (0.4 to 1.6 kPa) pressure drop. The pressure drop should be estimated based on 1/2 the calculated pressure drop using inlet vapor conditions. (Past designs frequently employed divided flow, TEMA Type J, shells with double segmental, modified disk and donut, baffles at or near maximum baffle spacing to obtain low pressure drop values.)
4. Estimate nozzle pressure drop based on three velocity heads for inlet and outlet nozzles.

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NOMENCLATURE

A	=	Total exchanger area, ft ² (m ²)
A _{dh}	=	Area of desuperheating zone, ft ² (m ²)
A _{ds}	=	Area of hydrocarbon condensation and steam desuperheating zone, ft ² (m ²)
A _{sc}	=	Area of steam condensation zone, ft ² (m ²)
A _s	=	Area/shell, ft ² (m ²)
A _x	=	Free flow area between shell baffles, in. ² (mm ²)
B ₁	=	Bundle factor for shell-side heat transfer, dimensionless
B ₂	=	Bundle factor for shell-side pressure drop, dimensionless
c	=	Specific heat at caloric temperature, Btu/lb-°F (kJ/kg-K)
c _f	=	Specific heat of the shell-side fluid at average film temperature, Btu/lb-°F (kJ/kg-K)
D	=	Shell I.D., in. (mm)
D _t	=	Diameter of tube bundle ("outer tube limit"), in. (mm)
d _i	=	Tube I.D., in. (mm)
d _o	=	Tube O.D., in. (mm)
f	=	Fanning friction factor, dimensionless
F _{bt}	=	Correction factor to account for baffle type, dimensionless
F _n	=	Correction factor for log mean temperature difference (due to partially concurrent flow), dimensionless
F _F	=	F shell correction factor, dimensionless
F _J	=	J shell correction factor, dimensionless
F _s	=	Shell-side pressure drop correction factor, dimensionless
F _t	=	Tube-side pressure drop correction factor, dimensionless
F _v	=	Correlation factor used to determine average vapor flow, dimensionless
G	=	Mass velocity, lb/sec-ft ² (kg/s-m ²)
G _e	=	Equivalent mass velocity of liquid and vapor in 2-phase flow (see Figure 4), lb/sec-ft ² (kg/s-m ²)
G _ℓ	=	Average mass velocity of liquid, lb/sec-ft ² (kg/s-m ²)
G' _ℓ	=	Superficial mass velocity of liquid, lb/sec-ft ² (kg/s-m ²)
G _v	=	Average mass velocity of vapor, lb/sec-ft ² (kg/s-m ²)
G' _v	=	Superficial mass velocity of vapor, lb/sec-ft ² (kg/s-m ²)
h _{dc}	=	Liquid cooling coefficient resulting from "dripping" in the condenser, Btu/hr-ft ² -°F (W/m ² -K)
h _{dh}	=	Film coefficient for condensing side of desuperheating zone, Btu/hr-ft ² -°F (W/m ² -K)
h _{ds}	=	Film coefficient for condensing side of hydrocarbon condensing zone, Btu/hr-ft ² -°F (W/m ² -K)
h _{sc}	=	Film coefficient for condensing side of steam condensing zone, Btu/hr-ft ² -°F (W/m ² -K)
h' _{gc}	=	Gas cooling coefficient uncorrected for condensate film resistance, Btu/hr-ft ² -°F (W/m ² -K)
h _{gc}	=	Gas cooling coefficient, Btu/hr-ft ² -°F (W/m ² -K)
h _{io}	=	Inside film coefficient corrected to outside area, Btu/hr-ft ² -°F (W/m ² -K)
h' _{ℓc}	=	Liquid cooling coefficient resulting from "bottom flow" in the condenser, Btu/hr-ft ² -°F (W/m ² -K)
h _{ℓc}	=	Weighted liquid cooling coefficient, Btu/hr-ft ² -°F (W/m ² -K), assuming half of liquid cooling is "drip" cooling and half is "bottom flow" cooling
h _o	=	Shell-side film coefficient no change of phase, Btu/hr-ft ² -°F (W/m ² -K)
h _{oc}	=	Shell-side condensing coefficient including fouling factor, Btu/hr-ft ² -°F (W/m ² -K)
h' _λ	=	Shell-side condensing film coefficient uncorrected for effect of vapor mass velocity, Btu/hr-ft ² -°F (W/m ² -K)
h _λ	=	Shell-side condensing film coefficient, Btu/hr-ft ² -°F (W/m ² -K)
h _{tλ}	=	Tube-side condensing film coefficient, Btu/hr-ft ² -°F (W/m ² -K)



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k	=	Thermal conductivity at caloric temperature, Btu/hr-ft-°F (W/m-K)
k_f	=	Thermal conductivity of fluid at average film temperature, Btu/hr-ft-°F (W/m-K)
k_w	=	Thermal conductivity of the tube metal at average tube temperature, Btu/hr-ft-°F (W/m-K)
ℓ	=	Tube thickness, ft (m)
L	=	Tube length, ft (m)
L_c	=	Length of tube exposed to condensation, ft (m)
LMTD	=	Log mean temperature difference, °F (K)
MW	=	Molecular weight, lb/mol (kg/kmol)
N_b	=	Number of shell baffles
N_p	=	Number of tube passes per shell
N_s	=	Number of shells in series
N_t	=	Number of tubes in the bundle
N_{tc}	=	Number of tubes across the center line of the bundle
Nu	=	Nusselt number, hd/k , dimensionless
n_s	=	Number of condensate streams in the shell
P_b	=	Baffle pitch, in. (mm)
P_i	=	Inlet pressure, psia
Pr	=	Prandtl number, c_p/k , dimensionless
P_t	=	Tube pitch, in. (mm)
ΔP_{dh}	=	Pressure drop in desuperheating zone, psi (kPa)
ΔP_{ds}	=	Pressure drop in hydrocarbon condensing zone, psi (kPa)
ΔP_{sc}	=	Pressure drop in steam condensing zone, psi (kPa)
ΔP_t	=	Total tube-side pressure drop, psi (kPa)
ΔP_{tf}	=	Tube pressure drop due to friction, psi/tube pass (kPa/tube pass)
ΔP_{tr}	=	Tube pressure drop due to turns, psi/tube pass (kPa/tube pass)
ΔP_{sf}	=	Shell-side pressure drop due to friction, psi/shell (kPa/shell)
ΔP_{sr}	=	Shell-side pressure drop due to turns, psi/shell (kPa/shell)
ΔP_s	=	Total shell-side pressure drop, psi (kPa)
Q	=	Total heat duty, Btu/hr (W)
q_{dh}	=	Heat duty in desuperheating zone, Btu/hr (W)
q_{ds}	=	Heat duty in hydrocarbon condensing zone, Btu/hr (W)
q_{sc}	=	Heat duty in steam condensing zone, Btu/hr (W)
q_{gc}	=	Heat duty from gas cooling in a zone, Btu/hr (W)
q_{lc}	=	Heat duty from liquid cooling in a zone, Btu/hr (W)
q_λ	=	Heat duty from condensation in a zone, Btu/hr (W)
R_c	=	Total resistance (clean) to heat transfer, hr-ft ² -°F/Btu (m ² -K/W)
R_{io}	=	Inside film resistance to heat transfer corrected to outside area, hr-ft ² -°F/Btu (m ² -K/W)
R_o	=	Outside film resistance to heat transfer, hr-ft ² -°F/Btu (m ² -K/W)
R_t	=	Total resistance (dirty) to heat transfer, hr-ft ² -°F/Btu (m ² -K/W)
r_i	=	Tube-side fouling factor referred to tube inside surface area, hr-ft ² -°F/Btu (m ² -K/W)
r_{io}	=	Tube-side fouling factor referred to tube outside surface area, hr-ft ² -°F/Btu (m ² -K/W)
r_o	=	Outside fouling factor, hr-ft ² -°F/Btu (m ² -K/W)
r_w	=	Resistance of tube wall metal at average wall temperature, hr-ft ² -°F/Btu (m ² -K/W)
Re	=	Reynolds number
S	=	Free width between baffles, in. (m)



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NOMENCLATURE (Cont)

S_f	=	Specific gravity of the condensate film
T_{dh}	=	Hydrocarbon dewpoint, °F (°C)
T_{ds}	=	Steam dewpoint, °F (°C)
t_{dh}	=	Temperature of the fluid being heated, after it gains enough heat to have cooled the condensing stream from T_{dh} to T_2 , °F (°C)
t_{ds}	=	Temperature of the fluid being heated, after it gains enough heat to have cooled the condensing stream from T_{ds} to T_2 , °F (°C)
T_m	=	Tube sheet design temperature, °F (°C)
T_1	=	Inlet temperature of fluid being cooled, °F (°C)
T_2	=	Outlet temperature of fluid being cooled, °F (°C)
t_1	=	Inlet temperature of fluid being heated, °F (°C)
t_2	=	Outlet temperature of fluid being heated, °F (°C)
t_f	=	Average shell-side film temperature, °F (°C)
t_s	=	Caloric temperature of the shell fluid, °F (°C)
t_t	=	Caloric temperature of the tube fluid, °F (°C)
t_w	=	Average tube wall temperature, °F (°C)
Δt_{dh}	=	Log mean temperature difference in desuperheating zone, °F (°C)
Δt_{ds}	=	Log mean temperature difference in hydrocarbon condensing zone, °F (°C)
Δt_{sc}	=	Log mean temperature difference in steam condensing zone, °F (°C)
Δt_e	=	Log mean temperature difference corrected for nonideal countercurrent flow (Effective temperature difference), °F (°C)
Δt_{ew}	=	Weighted effective log mean difference, °F (°C)
Δt_m	=	Log mean temperature difference for true countercurrent flow, °F (°C)
U_c	=	Overall clean coefficient of heat transfer, Btu/hr-ft ² -°F (W/m ² -K)
U_{dh}	=	Overall heat transfer coefficient in desuperheating zone, Btu/hr-ft ² -°F (W/m ² -K)
U_{ds}	=	Overall heat transfer coefficient in hydrocarbon condensing zone, Btu/hr-ft ² -°F (W/m ² -K)
U_{sc}	=	Overall heat transfer coefficient in steam condensing zone, Btu/hr-ft ² -°F (W/m ² -K)
U_o	=	Overall duty coefficient of heat transfer, Btu/hr-ft ² -°F (W/m ² -K)
U_{ox}	=	Overall condenser heat transfer coefficient, neglecting the resistance of condensate film, Btu/hr-ft ² -°F (W/m ² -K)
u	=	Compressibility factor of the vapor
V	=	Velocity, ft/sec (m/s)
$V_{\ell 1}$	=	Volume liquid entering a zone, ft ³ /hr (m ³ /s)
$V_{\ell 2}$	=	Outlet liquid volume, ft ³ /hr (m ³ /s)
V_v	=	Volume vapor entering a zone, ft ³ /hr (m ³ /s)
V_{v1}	=	Inlet vapor volume, ft ³ /hr (m ³ /s)
V_{v2}	=	Outlet vapor volume, ft ³ /hr (m ³ /s)
W	=	Mass flow rate, lb/hr (kg/s)
W_c	=	Quantity condensed in a zone (or overall), lb/hr (kg/s)
$W_{\ell 1}$	=	Inlet liquid rate, lb/hr (kg/s)
$W_{\ell 2}$	=	Outlet liquid rate, lb/hr (kg/s)
W_{ℓ}	=	Average liquid flow rate, lb/hr (kg/s)
W_v	=	Average vapor flow rate, lb/hr (kg/s)
W_{v1}	=	Inlet vapor flow rate, lb/hr (kg/s)
W_{v2}	=	Outlet vapor flow rate, lb/hr (kg/s)



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x_v	=	Fraction of cross-sectional flow area occupied by vapor
x_ℓ	=	Fraction of cross-sectional flow area occupied by liquid
Y_{th}	=	Tube-side heat transfer correlation factor
μ	=	Viscosity at caloric temperature, cP (Pa•s)
μ_f	=	Viscosity of fluid at average film temperature, cP (Pa•s)
μ_ℓ	=	Average liquid viscosity, cP (Pa•s)
μ_v	=	Average vapor viscosity, cP (Pa•s)
μ_w	=	Viscosity of the tube-side fluid at tube wall temperature, cP (Pa•s)
α	=	Ratio vapor rate to inlet vapor rate, shell-side condensation
β	=	In a condenser: Weight fraction condensed
Γ	=	Tube loading, lb/hr-ft (kg/s-m)
$\Delta 1,$ $\Delta 2,$ $\Delta 3$	=	Correlation factors used in calculating equilibrium flash curve
λ	=	Average latent heat of vaporization, Btu/lb (kJ/kg)
ρ	=	Fluid density (No change of phase), lb/ft ³ (kg/m ³)
ρ_ℓ	=	Average liquid density, lb/ft ³ (kg/m ³)
$\rho_{\ell 1}$	=	Inlet density of liquid, lb/ft ³ (kg/m ³)
$\rho_{\ell 2}$	=	Outlet density of liquid, lb/ft ³ (kg/m ³)
ρ_v	=	Average vapor density, lb/ft ³ (kg/m ³)
$\rho_{v 1}$	=	Inlet density of vapor, lb/ft ³ (kg/m ³)
$\rho_{v 2}$	=	Outlet density of vapor, lb/ft ³ (kg/m ³)



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TABLE 1
T-Q CURVE CALCULATION PROCEDURE

I. CALCULATION PROCEDURE

- From the predicted ASTM distillation of the condensing hydrocarbon, determine t_{10} , t_{30} , t_{50} , t_{70} , and t_{90} . Then calculate $\Delta 1$ from:

$$\Delta 1 = \frac{2}{3} (t_{70} - t_{10})$$

- From **Figure 6**, determine $\Delta 2$ and $\Delta 3$.
- Calculate the 10, 30, 50, 70, and 90 Vol. % points on the atmospheric flash curve.

- $t_{10f} = t_{10} + (\Delta 1 - \Delta 3) - \Delta 2$
- $t_{30f} = t_{10} + \frac{2}{3} (t_{30} - t_{10}) + (\Delta 1 - \Delta 3) - \left(\frac{\Delta 1}{3} + \frac{\Delta 2}{2} \right)$
- $t_{50f} = t_{10} + 0.48 (t_{50} - t_{10} - \Delta 1) + (\Delta 1 - \Delta 3)$
- $t_{70f} = t_{10f} + \frac{3}{2} \Delta 2$
- $t_{90f} = t_{10} + 0.23 (t_{90} - t_{10} - 2\Delta 1) + (\Delta 1 - \Delta 3) + \Delta 2$

- Estimate the temperatures at 10, 30, 50, 70, and 90 wt% flashed and at the same mol% flashed.

$$\begin{aligned} t_{30w} &= t_{30f} + 0.10 (t_{50f} - t_{10f}) & t_{10w} &= t_{10f} + 0.05 (t_{50f} - t_{10f}) \\ t_{50w} &= t_{50f} + 0.10 (t_{70f} - t_{30f}) & t_{90w} &= t_{90f} + 0.05 (t_{90f} - t_{50f}) \\ t_{70w} &= t_{70f} + 0.10 (t_{90f} - t_{50f}) \\ t_{30m} &= t_{30f} - 0.04 (t_{50f} - t_{10f}) & t_{10m} &= t_{10f} - 0.02 (t_{50f} - t_{10f}) \\ t_{50m} &= t_{50f} - 0.04 (t_{70f} - t_{30f}) & t_{90m} &= t_{90f} - 0.02 (t_{90f} - t_{50f}) \\ t_{70m} &= t_{70f} - 0.04 (t_{90f} - t_{50f}) \end{aligned}$$

- Prepare a graph (see **Figure 2**) with percent off on the X-axis and temperature on the Y-axis.
 - Plot the five points calculated for the volume flash curve. Connect the points as follows:
 - A straight line from t_{30f} through t_{10f} , intersecting the 0% axis.
 - A straight line from t_{70f} through t_{90f} , intersecting the 100% axis.
 - A straight line between t_{50f} and t_{30f} .
 - A straight line between t_{50f} and t_{70f} .
 - Plot the five weight percent flashed points and connect each adjacent point with a straight line. Connect t_{10w} with the 0 vol. % intercept, and t_{90w} with the 100 vol. % intercept. Repeat this process with the five mol percent flashed points.
- Estimate T_{dh} , the dew point of the hydrocarbon. (For the overhead of the fractionation tower, $T_1 = T_{dh}$.)
 - On the flash curve prepared above, the *atmospheric-pressure* dew point is the point at 100% NOT condensed.
 - Calculate the partial pressure of the condensable hydrocarbon and adjust the atmospheric-pressure dew point to this partial pressure, using the alignment chart, **Figure 3**.

$$\text{HC Partial Press.} = \frac{\text{mol of Condensable HC}}{\text{Total mol of vapor}} \times P_1$$

- Estimate T_{ds} , the dew point of steam in the system. This is the temperature at which the vapor pressure of water is equal to the partial pressure of the water in the system.
 - This calculation is done in tabular form, as shown below.
 - The calculations for the table should be carried out at increasing values of weight percent condensed until the first point is found where the vapor pressure of water @ system temperature is less than the calculated partial pressure of water in the system.

TABLE 1 (Cont)
T-Q CURVE CALCULATION PROCEDURE

- c. Prepare a graph with temperature on Y-axis and pressure of the X-axis. From the table just completed, plot the last 3 or 4 steam partial pressure vs. system temperature points. On the same graph, plot the last 3 or 4 water-vapor-pressure vs. system temperature points. The intersection of the two curves gives the approximate steam dew point of the system (see **Figure 2**).

Wt. % Hydrocarbon Condensed ⁽¹⁾						
Vapor Temp. @ Atm. Press. ⁽²⁾						
Mol % HC not Condensed ⁽²⁾						
Mol of HC not Condensed						
Mol of Non-Condensables ("NC")						
Mol of Steam						
Total Mol of Vapor						
Total Pressure, psia or kPa ⁽³⁾						
Partial Pressure of Steam, psia or kPa						
Partial Press. of Condensables, psia or kPa						
System Temperature, °F or °C ⁽⁴⁾						
Vapor Press. of H ₂ O at Sys. Temp. ⁽⁵⁾ , psia or kPa						

Notes:

- (1) Arbitrary values.
- (2) Read from flash curves.
- (3) Assume percent pressure drop = mol percent condensed.
- (4) Vapor temperature at 1 atm. adjusted to the partial pressure of condensables by means of the alignment chart, **Figure 3**.
- (5) From steam tables or from alignment chart **Figure 3**. (Use the point marked H₂O as the "normal boiling point".)

8. Determine the amount and type of heat duty in each zone. Evaluate specific heat (c) and latent heat of vaporization (λ) at the arithmetic average zone temperature.

- a. Sensible heat transferred from material which passes through the zone without changing phase.

$$q = W c \text{ (Temp Change)}$$

- b. Sensible heat transferred from vapor or steam which condenses in the zone.

- c. Sensible heat transferred from hydrocarbon or water which has condensed in the zone.

$$q = \frac{W_c}{2} c \text{ (Temp Change)} \quad (\text{c of vapor})$$

$$q = \frac{W_c}{2} c \text{ (Temp Change)} \quad (\text{c of liquid})$$

- d. Latent heat transferred from condensing hydrocarbon or steam.

$$Q = \lambda W_c$$

Note that Item (8a) may include as many as 4 calculations: steam; non-condensables; vapor which does not condense; and any entering liquid.

Item (8b) may include two calculations: vapor and steam.

Items (8c) and (8d) may each include two calculations: hydrocarbon and water.

The above calculations, performed for each zone, will give q_{dh} , q_{ds} , and q_{sc} . These values permit the calculation of the LMTD's and overall temperature difference (Δt_m) – also referred to as the Mean Temperature Difference (MTD).

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TABLE 2
CALCULATION PROCEDURE FOR CONDENSER DESIGN

Exchanger Number E - _____
 Service _____

Refinery _____
 Calc By/Date _____

I. ZONE DUTIES AND TEMPERATURES*

A. Vapor Desuperheating Zone (If Present); 1st Zone

	CUSTOMARY UNITS	METRIC UNITS
q_{dh}	= _____ Btu/hr	_____ W
Fluid Being Condensed (Shell/Tube)		
T_1	= _____ °F	_____ °C
T_{dh}	= _____ °F	_____ °C
Fluid Being Heated (Shell/Tube)		
t_2	= _____ °F	_____ °C
t_{dh}	= _____ °F	_____ °C
$R = \frac{T_1 - T_{dh}}{t_2 - t_{dh}} =$ _____	= _____	
$j = \frac{t_2 - t_{dh}}{T_1 - T_{dh}} =$ _____	= _____	
$\Delta t_m = \frac{(T_1 - t_2) - (T_{dh} - t_{dh})}{\ln \left(\frac{T_1 - t_2}{T_{dh} - t_{dh}} \right)}$	= _____	_____
	= _____ °F	_____ °C
From charts given in Section IX-D		
$F_n =$ _____ for this zone**		
$\Delta t_{e(dh)} = \Delta t_m F_n$	= _____ °F	_____ °C
For shell-side condensation:		
$t_s = 0.4 (T_1 - T_{dh}) + T_{dh}$	= _____ °F	_____ °C
$t_t = 0.4 (t_2 - t_{dh}) + t_{dh}$	= _____ °F	_____ °C
For tube-side condensation:		
$t_t = 0.4 (T_1 - T_{dh}) + T_{dh}$	= _____ °F	_____ °C
$t_s = 0.4 (t_2 - t_{dh}) + t_{dh}$	= _____ °F	_____ °C
$\frac{q_{dh}}{\Delta t_e}$	= _____ Btu/hr-°F	_____ W/K

* Use LMTD correction factor for each zone, based on zone temperatures.

** Use LMTD corrections factor, based on terminal temperatures to evaluate minimum number of shells in series. F_n must be equal to or greater than 0.8.

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TABLE 2 (Cont)
CALCULATION PROCEDURE FOR CONDENSER DESIGN

B. Hydrocarbon Condensing Steam Desuperheating Zone (2nd Zone)

	CUSTOMARY UNITS	METRIC UNITS
q_{ds}	= _____ Btu/hr	_____ W
Fluid Being Condensed (Shell/Tube)		
T_{dh}	= _____ °F	_____ °C
T_{ds}	= _____ °F	_____ °C
t_{dh}	= _____ °F	_____ °C
t_{ds}	= _____ °F	_____ °C
$R = \frac{T_{dh} - T_{ds}}{t_{dh} - t_{ds}} =$ _____	= _____	_____
$j = \frac{t_{dh} - t_{ds}}{T_{dh} - T_{ds}} =$ _____	= _____	_____
$\Delta t_m = \frac{(T_{dh} - t_{dh}) - (T_{ds} - t_{ds})}{\ln \left(\frac{T_{dh} - t_{dh}}{T_{ds} - t_{ds}} \right)}$	= _____ °F	_____ °C
From charts given in Section IX-D , $F_n =$ _____ for _____ shells		
$\Delta t_{e(ds)} = \Delta t_m F_n$	= _____ °F	_____ °C
For shell-side condensation: $t_s = 0.4 (T_{dh} - T_{ds}) + T_{ds}$ $t_t = 0.4 (t_{dh} - t_{ds}) + t_{ds}$	= _____ °F = _____ °F	_____ °C = _____ °C
For tube-side condensation: $t_t = 0.4 (T_{dh} - T_{ds}) + T_{ds}$ $t_s = 0.4 (t_{dh} - t_{ds}) + t_{ds}$	= _____ °F = _____ °F	_____ °C = _____ °C
$\frac{q_{ds}}{\Delta T_e}$	= _____ Btu/hr-°F	_____ W/K

C. Steam Condensing Zone (3rd Zone)

q_{sc}	= _____ Btu/hr	_____ W
Fluid Being Condensed (Shell/Tube)		
T_{ds}	= _____ °F	_____ °C
T_2	= _____ °F	_____ °C
t_{ds}	= _____ °F	_____ °C
t_1	= _____ °F	_____ °C
$R = \frac{T_{ds} - T_2}{t_{ds} - t_1} =$ _____	= _____	_____
$j = \frac{t_{ds} - t_1}{T_{ds} - T_2} =$ _____	= _____	_____

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TABLE 2 (Cont)
CALCULATION PROCEDURE FOR CONDENSER DESIGN

	CUSTOMARY UNITS	METRIC UNITS
$\Delta t_m = \frac{(T_{ds} - t_{ds}) - (T_2 - t_1)}{\ln \left(\frac{T_{ds} - t_{ds}}{T_2 - t_1} \right)}$	= _____ °F	_____ °C
From charts given in Section IX-D , $F_n =$ _____ for _____ shells		
$\Delta t_{e(sc)} = \Delta t_m F_n$	= _____ °F	_____ °C
For shell-side condensation:		
$t_s = 0.4 (T_{ds} - T_2) + T_2$	= _____ °F	_____ °C
$t_t = 0.4 (t_{ds} - t_1) + t_1$	= _____ °F	_____ °C
For tube-side condensation:		
$t_t = 0.4 (T_{ds} - T_2) + T_2$	= _____ °F	_____ °C
$t_s = 0.4 (t_{ds} - t_1) + t_1$	= _____ °F	_____ °C
$\frac{q_{sc}}{\Delta t_e}$	= _____ Btu/hr-°F	_____ WK

D. Weighted Temperature Difference

$\Delta t_{ew} = \frac{Q}{\frac{q_{dh}}{\Delta t_{e(dh)}} + \frac{q_{ds}}{\Delta t_{e(ds)}} + \frac{q_{sc}}{\Delta t_{e(sc)}}}$	= _____	_____
where: $Q = q_{dh} + q_{ds} + q_{sc}$	= _____ °F	_____ °C

II. SHELL-SIDE CONDENSATION

A. Overall Tube-Side Calculation (No Change of Phase)

1. Assumed value of U_o	= _____ Btu/hr-ft ² -°F	_____ W/m ² -K
2. $A = Q / U_o \Delta t_{ew}$	= _____ ft ²	_____ m ²
3. Tube metal: _____		
Thermal conductivity, k_w	= _____ Btu/hr-ft-°F	_____ W/m-K
Tube outer diameter, d_o	= _____ in.	_____ mm
Wall thickness, ℓ	= _____ ft	_____ m
Tube insides diameter, d_i	= _____ in.	_____ mm
Tube length, L	= _____ ft	_____ m
4. Tube pitch (P_t) and layout (Note 1)*	= _____ in.	_____ mm
5. $A_s = A / N_s$ (Note 2)*	= _____ ft ²	_____ m ²
6. $N_t = 3.82 A_s / (L - 0.5) d_o$	= _____	_____
$N_t = 318 A_s / (L - 0.152) d_o$ (Metric)		

* General notes contained on last page of calculation form.

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TABLE 2 (Cont)
CALCULATION PROCEDURE FOR CONDENSER DESIGN

	CUSTOMARY UNITS	METRIC UNITS
7. Assumed N_p	= _____	_____
8. t_s overall = $0.4 (T_1 - T_2) + T_2$	= _____ °F	_____ °C
9. t_t overall = $0.5 (t_1 + t_2)$	= _____ °F	_____ °C
10. Tube-side properties at t_t		
a. Density, ρ	= _____ lb/ft ³	_____ kg/m ³
b. Viscosity, μ	= _____ cP	_____ Pa•s
c. Specific Heat, c	= _____ Btu/lb-°F	_____ kJ/kg-K
d. Thermal Conductivity, k	= _____ Btu/hr-ft-°F	_____ W/m-K
11. Tube-side flow rate, $W = Q / c (t_2 - t_1)$	= _____ lb/hr	_____ kg/s
12. $V = \frac{N_p W}{(19.6) \rho N_t d_i^2}$	= _____ ft/sec	
$V = \frac{(1.273 \times 10^6) N_p W}{\rho N_t d_i^2}$ (Metric)		_____ m/s
13. If tube-side fluid is water		
$h_{io} = \frac{368}{d_o} (V d_i)^{0.7} \left(\frac{t_t}{100} \right)^{0.26}$	= _____ Btu/hr-ft-°F	
$h_{io} = \frac{1.27 \times 10^4}{d_o} (V d_i)^{0.7} \left(\frac{1.8 t_t + 32}{100} \right)^{0.26}$ (Metric)		_____ W/m ² -K
$\Delta P = 0.020 F_t N_s N_p \left(V^2 + 0.158 L \frac{(V)^{1.73}}{(d_i)^{1.27}} \right)$	= _____ psi	
$\Delta P = 1.48 F_t N_s N_p \left(V^2 + 22.9 L \frac{(V)^{1.73}}{(d_i)^{1.27}} \right)$ (Metric)		_____ kPa
14. For fluids other than water, evaluate overall tube-side heat transfer coefficient and pressure drop by the procedure outlined in Section IX-D , tube-side iteration.		
h_{io}	= _____ Btu/hr-ft-°F	_____ W/m ² -K
ΔP_t	= _____ psi	_____ kPa
15. $R_{io} = 1 / h_{io}$	= _____ hr-ft ² -°F/Btu	_____ m ² -K/W
16. Tube-side fouling factor, r_i	= _____ hr-ft ² -°F/Btu	_____ m ² -K/W
17. $r_{io} = (d_o / d_i) r_i$	= _____ hr-ft ² -°F/Btu	_____ m ² -K/W
18. Shell-side fouling factor, r_o	= _____ hr-ft ² -°F/Btu	_____ m ² -K/W

