

Properly Employ Overhead Condensers for Vacuum Columns

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In a vacuum distillation system, the condenser is an important determinant of overall performance. Follow these guidelines for specifying, selecting and designing overhead vacuum condensers.

Vacuum distillation may be used to enhance the separation of hydrocarbons because of the improvement in relative volatility at the lower pressure. Vacuum may also be employed to break azeotropes by using two columns in series that operate at different pressures. And, it may be required to reduce the boiling point of the bottom composition so conventional heat-transfer fluids can be used for heating.

The improved performance of vacuum column operation must be weighed against the additional capital and operating costs of the vacuum system. Because of these high costs, vacuum operation is normally reserved for situations where the components are either very difficult to separate at higher pressures or where the products would be degraded at elevated temperatures.

Once a decision is made to operate under vacuum, it is generally advantageous to provide a condenser upstream of the vacuum source to reduce the size and cost of the vacuum system and recover process material. A vent condenser after the main condenser, possibly using a colder cooling medium, may be used to further reduce the load on the vacuum system.

The main process parameters involved in specifying vacuum systems are the absolute pressure and the amount of noncondensables. Other considerations in-



Typical overhead condenser installed on a vacuum distillation column.

clude the cooling medium and its temperature, the desirability of recovering process vapors, and the need to treat either the gaseous or liquid effluent stream, or both. In a batch system, the time necessary to pull the system down to the desired vacuum level is also relevant, and in fact might control the sizing of the vacuum system.

Pressure

Vacuum can be classified as high, medium or low, where “high vacuum” corresponds to the lowest absolute pressure: low vacuum \approx 760–100 torr; medium

Heat Transfer

vacuum \approx 100–10 torr; and high vacuum \approx 10–1 torr. The costs of producing vacuum increase exponentially with the level of vacuum.

Once the level of vacuum is established by evaluating the process requirements against cost and performance, the system can be specified. The first decision is how to allocate the pressure drop between the process user and the vacuum system and how to split this pressure drop between the piping and condenser. A rule-of-thumb is to allocate about 10% of the absolute pressure to the pressure drop in the piping and condenser. Because condensation increases with pressure, the pressure loss between the process equipment and condenser should be minimized. In the case of medium- and high-vacuum systems, the preferred arrangement is to integrally connect the process equipment and condenser. Eliminating the piping and mounting the condenser directly on the column minimizes the pressure drop between the column and condenser.

If the condenser is mounted away from the column, the piping should be generously sized, fittings minimized and valves avoided. Valves are expensive in large-bore piping and are sources of both in-leakage and pressure drop.

Noncondensables

Noncondensables can be introduced through in-leakage, they may be entrained or dissolved in the process fluid, or they may be produced in the distillation column by cracking of heavier components. During start-up, additional noncondensables in the form of contaminants, such as oil or grease, may be present in the system.

Two methods are available to estimate in-leakage. A crude technique developed by the Heat Exchange Institute (HEI) (1) provides an estimate of in-leakage as a function of absolute pressure and system volume for tight systems. Although quick and simple, there seems to be a general consensus that the results are overly conservative (2).

A somewhat more scientific basis is to count the fittings, valves, flanges and any other connections between the process and atmosphere, and estimate in-leakage based on either in-house or published data. In this approach, dynamic connections are assigned more weight than static connections.

The amount of noncondensables dissolved in the process stream or produced in the process must also be estimated and added to the noncondensables in-leakage estimate to determine a total noncondensable load. If a safety factor is desired, it is added to the total noncondensable load. Remember that the control system should be designed to accommodate the difference between the actual or expected noncondensables and the design allowance without introducing additional inerts.

The next step is to saturate the noncondensables with the process components and determine the total load to the condenser and the physical properties of the process stream.

It is obviously desirable to minimize in-leakage by eliminating unnecessary fittings and avoiding situations that permit the ingress of noncondensables. To the extent possible, the system should be welded and the distance from the process source through the condenser to the vacuum system minimized.

Temperature

A vent condenser will further reduce the load on the vacuum system, recover process material and reduce effluent-treatment requirements. It may be desirable to add refrigeration to maximize recovery.

At the same time, the design should be based on a reasonable coolant temperature and a realistic temperature approach. The cost of designing for the worst-case cooling water temperature and a close approach will be very high for a vacuum condenser.

Also, use caution when dealing with materials with high freezing points to avoid the possibility of freezing. A recirculating tempered-cooling loop may be provided to obtain a close approach while maintaining the process temperature above the freezing point.

TEMA data sheet

After the heat duty has been calculated, a condensing curve, as illustrated in Figure 1, is prepared. This curve will be used to calculate a weighted mean temperature difference to most accurately reflect the temperature difference throughout the heat exchanger.

A data sheet, such as provided in the Tubular Exchanger Manufacturers Association (TEMA) standards (3), is used to transmit pertinent fluid properties and mechanical preferences from the process engineer to the heat transfer specialist.

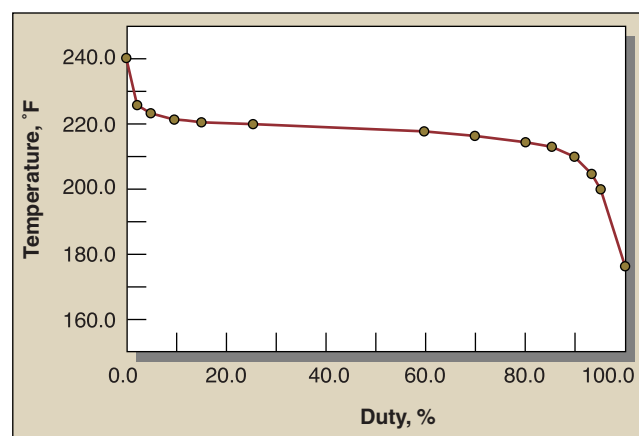


Figure 1. Typical heat release curve.

Relationship between condenser and column

Piping. The condensate piping must be properly designed to ensure good drainage at all times. This includes a correctly designed barometric seal with an ap-

appropriate allowance for the density of the condensate and any tendency to foam. For a fluid density of 50 lbs/ft³ and a tendency to foam, 45 ft is an appropriate height. The seal piping should be run vertically. If this is not possible, one or two 45-deg sections may be permitted, but this should be considered as a last-resort approach. Horizontal runs should never be used, as they might allow gas bubbles to form and agglomerate into vapor pockets, which would impair draining and might cause backup to the condenser. Remember also that condensate at sub-ambient temperature may warm up in the seal leg or hotwell and flash. To avoid this, it may be necessary to insulate the pipe.

In addition, the hotwell seal needs to have sufficient volume to fill the pipe to avoid breaking vacuum if the system is shut down. A 12-in. seal between the pipe outlet and hotwell overflow is suggested to accommodate minor level upsets.

Condensate collection and reflux distribution. Distribution of reflux is a critical factor in the operation of a vacuum column because of the liquid loading. It is also important to avoid entrainment of the returning reflux into the exiting vapor stream. The most common reflux arrangement is to remove part of the condensate as product (distillate) and return the remainder as reflux. Some columns return all of the liquid to the column and remove the product somewhat below the condenser from a specially designed drawoff stage. In either case, the condensate must first be collected in a central location for further distribution.

The most common way to collect condensate for a directly mounted condenser is to provide a welded annular trough around the inlet nozzle. Liquid can be withdrawn from this trough via either an overflow weir or a pipe to direct the flow to the column below or to a product line. Using an internal splitter is simple, but the reflux flowrate cannot be directly measured. At one time, it was common to withdraw the entire condensate and measure the reflux and product streams from the pumped discharge. With today's control systems, the reflux can be determined by performing a heat balance on the condenser using coolant temperatures and flows to calculate the total condensate and subtracting the distillate flow measurement, thereby permitting the use of the simpler internal splitter for reflux control.

Special consideration must be given to the reflux distributor in the column. As vacuum level increases, the column cross-section gets larger, but the reflux volume remains the same and may even decrease. In order to provide good distribution for wetting packing or ensuring uniform flow across trays, a specially designed distributor is normally used. These distributors become quite sophisticated if turndown is required, and it is most important that the distributor be level to ensure uniform flow across the column.

Selecting the design

The selection of the vacuum condenser type is mainly a function of the operating pressure and the allowable pressure drop. The size of the unit is then determined by the specified vapor mass flowrate and operating pressure.

Condensers operating at low vacuum (100–760 mm Hg) are the easiest to design; difficulty increases exponentially as the vacuum level increases. Experience shows that the most critical design factor for vacuum condensers is the proper allocation of pressure drop throughout the unit. Since the largest losses occur at the inlet nozzle before condensation starts, this region must be carefully analyzed. As the size of the condenser increases, the supporting structure becomes another important cost factor in optimization of the design.

The following criteria must be factored into the selection of a shell-and-tube condenser:

Condensation inside vs. outside of tubes. Condensation inside tubes is not appropriate for vacuum overhead condensers because of the high tubeside pressure drop and the difficulty in piping and supporting a vertically mounted unit. Therefore, condensation on the outside of the tubes is the best choice.

Horizontal vs. vertical shell orientation. Vertical orientation for outside-tube condensation is justified for tube bundles mounted inside the column or possibly inside a receiving tank. However, it is more difficult to meet the low pressure-drop objectives when compared to horizontal designs because it is not possible to split the flow in a vertical arrangement. In addition, the use of a U-tube bundle may not be allowed, since the coolant is not drainable and a more costly floating head would have to be provided. The optimal design can be achieved in a horizontal shell.

Shellside flow arrangements. Dividing the flow crossing the tube bank can minimize pressure drop in a horizontal unit. Using divided shell flow and double segmental baffles can achieve this very effectively. Further decrease in pressure drop can be achieved by increasing the tube layout pitch, which will open up the cross-flow area. Typically, the vapor enters at the bottom in the center of the shell, the flow is divided inside the tubeless shell bottom, and noncondensables exit at the top in the center. The inlet nozzle is specially designed for collecting the condensate. The applied LMTD correction factor is the same as in a conventional divided-flow unit.

Mechanical design and materials of construction. The minimum piping pressure drop is achieved by eliminating the piping between the column and condenser. Mounting the condenser at the top of the column does present some mechanical and maintenance difficulties and may require a booster pump to supply coolant to tall columns, but there is a significant process advantage in minimizing pressure drop.

The unit must be self-supporting from the vapor inlet nozzle and designed to prevent in-leakage of the ambient air. This is important in order to avoid overloading the vac-

Nomenclature

A	= total required heat transfer surface, ft ²
D	= tube outer diameter, in.
g	= gravitational constant, 4.17×10^8 ft/h ²
h_c	= condensing heat-transfer coefficient, Btu/h-ft ² -°F
h_v	= convective heat-transfer coefficient of vapors, Btu/h-ft ² -°F
h_w	= tubeside heat-transfer coefficient of coolant, Btu/h-ft ² -°F
k	= thermal conductivity, Btu/h-ft-°F
L	= effective tube length, ft
N	= number of vertical tube rows
ΔP	= pressure drop, psi
ΔP_2	= two-phase flow pressure drop, psi
ΔP_L	= liquid-phase flow pressure drop, psi
Q	= heat duty, Btu/h
Q_v	= vapor cooling duty, Btu/h
R	= thermal resistance, h-ft ² -°F/Btu
Re	= Reynolds number
T	= temperature, °F
ΔT	= temperature difference, °F
U	= overall heat-transfer coefficient, Btu/h-ft ² -°F
W_c	= total mass flowrate of condensate, lb/h
x	= weight fraction of vapor (vapor quality)
X_{tt}	= Lockhart-Martinelli parameter for liquid and vapor in turbulent flow

Greek Letters

λ	= latent heat, Btu/lb
μ	= viscosity, lb/ft-h
ρ	= density, lb/ft ³
Φ_L^2	= Lockhart-Martinelli two-phase pressure drop multiplier

Subscripts

f	= fouling
i	= vapor/condensate interface
L	= liquid phase
m	= metal
n	= local value at n-th section along the condensing curve
o	= outer tube surface
v	= vapor phase

uum system and reducing the condenser performance. At the same time, maintenance requirements may dictate that the unit be removable. The best solution is to use a field-welded connection. This also assures minimum leakage.

If a welded design cannot be used, a mounting flange rated for 300 lb with spiral-wound gaskets would be the next-best choice. This arrangement can be quite expensive for large high-alloy-steel applications.

Leakage of coolant into the shell must be minimized. Using strength-welded tube-to-tubesheet joints per UW-20 of the ASME Pressure Vessel Code, section VIII, is strongly recommended.

The choice between a U-tube or fixed tubesheet tube bundle depends mainly on the thermal expansion stresses between the tubes and the shell. For a fixed tubesheet design, when the temperature difference between the tube wall and the shell at operating or any upset conditions results in excessively high stresses, a shell expansion joint is required. To avoid an expansion joint, a U-tube bundle may be selected. By using a non-removable channel with cover (TEMA N-U type), the shellside leakage would be minimized.

Trace impurities such as chlorides may concentrate at the phase change. If this is anticipated, corrosion-resistant materials of construction are required.

Entrainment of liquid. Entrainment should be minimized through proper baffling and deflector plates at the shell exit section of the heat exchanger. Excessive entrainment will reduce condenser capacity and prevent the optimum recovery of product.

Freezing of condensing liquid on the tube wall. Some process liquids may have a freezing point close to the coolant temperature. The heat-transfer specialist will take this into consideration when designing the heat-transfer surface by checking the outer tube-wall-temperature, which must be kept at a safe minimum. This temperature can most effectively be controlled by increasing the coolant inlet temperature. The most critical part of the condenser is the bundle exit section, where the heat flux is at minimum and the heat-transfer coefficient is dominated by convective heat transfer.

Heat transfer calculations

The heat transfer mechanism in the condenser is partial condensation. Due to the presence of noncondensable gases in the multicomponent vapor mixture, the condensing vapors must be cooled along the condensing curve. The cooling process is a combination of heat and mass transfer that can be calculated by the rigorous method of Colburn and Hougen, which is described with practical examples by Hewitt, *et al.* (4). The complexity of mass transfer correlations and a lack of data on diffusivities for multicomponent mixtures of vapors led to the use of a simplified calculation method developed by Bell, *et al.* (5), which is the basis for the calculations presented here.

In this method, the incremental heat flux into the vapor/liquid interface must be equal to the heat flux from the condensing liquid film to the coolant:

$$dQ/dA = U(T_i - T_w) \quad (1)$$

$$dQ_v/dA = h_v(T_v - T_i) \quad (2)$$

Solving for T_i and introducing the simplifying parameter $Z = dQ_v/dQ$ yields:

$$\frac{dQ}{dA} = \frac{U(T_v - T_w)}{(1 + UZ/h_v)} \quad (3)$$

The overall heat-transfer resistance is:

$$\frac{1}{U} = \frac{1}{h_c} + R_{fo} + R_m + R_{fi} + \left(\frac{D_o}{D_i}\right) \frac{1}{h_w} \quad (4)$$

Integrating Eq. 3 reveals that the required heat-transfer surface area will be:

$$A = \int_0^Q \frac{(1 + UZ/h_v)}{U(T_v - T_w)} dQ \quad (5)$$

The shellside vapor-phase convective heat-transfer coefficient, h_v , is calculated using the Tinker type stream-analysis method, as described by Hewitt, *et al.* (4).

The condensing liquid-film heat-transfer coefficient is established by the correlations for single vapor condensation as given by McAdams (6) for two regions of flow. Gravity-controlled condensation occurs at $Re_L < 2,100$, and the heat-transfer coefficient in this region is:

$$h_c = 0.725 \left(\frac{k_L^3 \rho_L^2 \lambda g}{N \mu_L D \Delta T} \right)^{0.25} \quad (6)$$

For $Re_L > 2,100$, the heat-transfer coefficient becomes shear-controlled and can be established from the graphical interpretation given by McAdams (6), where the heat-transfer parameter is plotted as a function of Re_L :

$$h_c \left(\frac{\mu}{k_L^3 \rho_L^2 g} \right)^{0.33} = f(Re_L) \quad (7)$$

$$Re_L = \frac{4W_c}{NL\mu_L} \quad (8)$$

Pressure drop calculations

The pressure drop through the condenser is typically calculated from the inlet to the outlet nozzle of the shell, and it consists of following components:

$$\Delta P = \Delta P_{inlet} + \Delta P_{90\text{-deg-turn}} + \Delta P_{dome} + \Delta P_{entrance} + \Delta P_{2\Phi} + \Delta P_{outlet} \quad (9)$$

The single-phase shellside pressure drops are calculated by correlations as defined by Kern (7). The two-phase pressure drop across the bundle is established by applying the Lockhart-Martinelli method (8), where the pressure drop is calculated for a single phase (all liquid flow) and then corrected by a two-phase flow multiplier, shown graphically as a function of the Lockhart-Martinelli/Nelson parameter X_{tt} in the form:

$$\Phi_L^2 = f(X_{tt}) \quad (10)$$

The local Lockhart-Martinelli parameter is defined as:

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_L}{\rho_v} \right)^{0.5} \left(\frac{\mu_L}{\mu_v} \right)^{0.1} \quad (11)$$

The total two-phase pressure drop is then:

$$\Delta P_{2\Phi} = \sum (\Delta P_{L_n} \times \Phi_{L_n}^2) \quad (12)$$

Example: Application of computer programs

Vacuum condensers for one installation were sized by applying a combination of manual and computer calculations with full recognition of cost optimization.

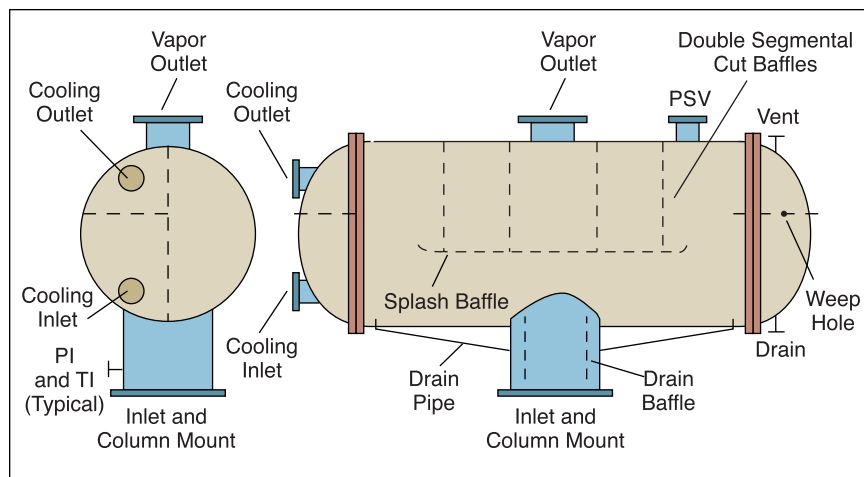
The TEMA (3) designation of the condenser was type BJM, meaning divided-flow, fixed tubesheets, and bonnet heads with the condensing-vapor inlet nozzles located on the top of the shell. This case was a special design in which the flow arrangement was reversed and one vapor inlet nozzle was located at the bottom to also catch returning condensate. The full flow was returned to the column as reflux via drain holes at the shell's bottom and a half-pipe drain. The detailed design is shown in the setting plan in Figure 2.

For the thermal design, the HTRI condenser computer program was used. This program follows, in principle, the calculation methods outlined here, but with more exact and improved correlations. Since the program is not completely adaptable to the special features of this design, some modifications were made to satisfy the standard input:

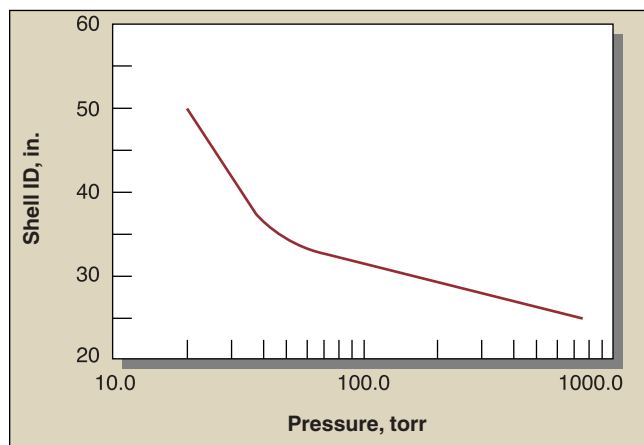
1. The HTRI designation of BJ21M was specified, denoting two inlet nozzles and one outlet nozzle. Therefore, the single inlet had to be split into two nozzles with an equivalent mass-flow velocity. The effect on heat transfer of the upward flow at the inlet section with the liquid flowing down was neglected and checked for flooding only.

2. The shell size was specified as the equivalent diameter of the real shellside flow area, which corresponds to the full shell area minus the segmental-inlet-flow area.

The pressure drop determined by the computer program was modified by manual calculations to account for all contributing components, in accordance with Eq. 9. The shell size was also established by manual calculations so that the diameter would agree with the bundle tube count and more accurately simulate the shellside flow areas.



■ Figure 2. Setting plan for condenser.



■ Figure 3. Condenser shell diameter dependence on operating pressure.

Pressure drop distribution and condenser size

The frictional losses in the dome area and the outlet nozzle losses of the noncondensable vapors are negligible. The distribution of the total pressure drop among the components presented in Eq. 9 can be expressed as a percentage of the total pressure losses for a typical design lumped into three groups, as shown in the table.

It can be seen that the inlet losses represent an appreciable portion of the pressure loss. Because these pressure losses are mainly a function of shell diameter, they directly affect the size and cost of the condenser. Figure 3 illustrates the dependence of the shell diameter on the operating pressure using the duty and flow

from the design example and allocating 10% of the operating pressure to the condenser pressure drop. CEP

Table. Pressure drop distribution in a typical condenser.

Inlet nozzle + impingement on the dome plate35%
90-deg turn of the flow into the bundle25%
Tube bundle in two-phase flow40%

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