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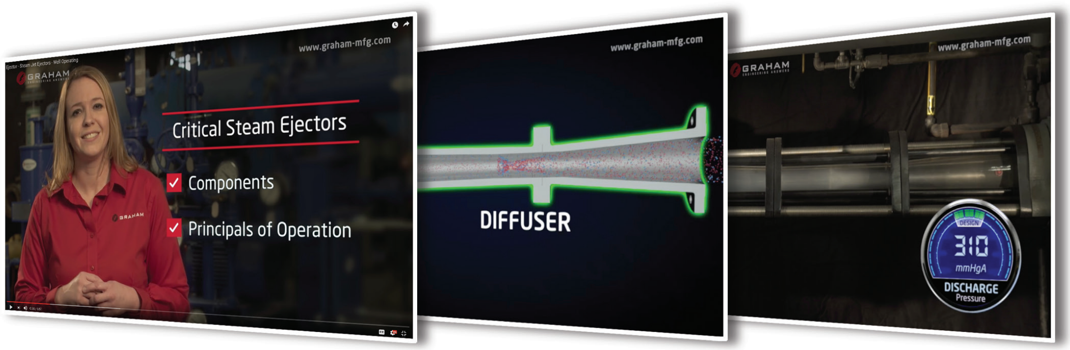

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Operating vacuum distillation ejector systems

Best practices and opportunities to deliver reliable ejector system performance and reduce performance risk

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Reliable ejector system performance is critical for every refiner. The performance of an ejector system correlates directly to vacuum gas oil yield and refinery profitability. Both charge rate and fractionation are impacted when distillation or fractionation operating pressure is not met. While they have been used widely in distillation service for decades, an understanding of best practices for specifying an ejector system and the important factors that affect ejector system performance are not always well known. This article provides a deeper review of ejector system performance, variables impacting performance, and best practices to specify an ejector system for vacuum distillation service.

Figure 1 An ejector system for a US Gulf Coast refiner: top left, first stage ejector; right, first stage condenser; bottom left, vacuum distillation column

Ejector system

An ejector system is a combination of ejectors and condensers arranged in series. The system produces and maintains sub-atmospheric pressure (a vacuum) within the distillation column to permit fractionation of crude oil into its various important components, such as light or heavy vacuum gas oils (LVGO and HVGO, respectively), and reduce the amount of lower valued residuum. The ejector system will continually extract from the distillation column cracked and inert gases along with associated saturated steam and hydrocarbon vapours. Failure to extract the gases and saturated vapours properly will result in an increase in distillation column operating pressure, thereby increasing residuum while lowering LVGO and HVGO yield. The ejector system extracts the gases at sub-atmospheric pressure and compresses them to a pressure typically above atmospheric pressure where they enter another refinery process for treating or repurposing of the gases.

An ejector

Ejectors are static equipment with no moving parts. The operating principle follows compressible flow theory. Medium or low pressure steam, typically less than 300 psig (43 kPag), is the energy source that performs the work and creates the vacuum. Steam is expanded isentropically across a converging-diverging nozzle where its pressure is reduced and converted to supersonic velocity. This pressure reduction and expansion to supersonic flow is what creates the vacuum. The low pressure region exiting the converging-diverging nozzle is lower than the distillation column pressure, thereby inducing flow from the column and pulling the cracked gases and inerts plus saturated vapours into the ejector. The vacuum column discharge is referred to as suction load or flow to the first stage ejector. The suction load is entrained by and mixes with the high velocity motive steam, and the combined flow remains supersonic. Again, compressible flow theory is applied where the supersonic mixture of load and motive passes through another converging-diverging conduit, referred to as a diffuser, where high velocity is converted back to pressure. A fundamental principle for compressible flow, which may be counter-intuitive, is that when flow is supersonic and the cross-sectional area of a flow path is progressively reduced, velocity actually decreases. The throat of the converging-diverging diffuser section of the ejector is where cross-sectional area is the smallest and a shock wave is established, which serves to boost pressure. **Figure 2** illustrates pressure and velocity profiles across an ejector with a clear step up in pressure at the throat where a shock wave is established.

An ejector, unlike a piston reducing volume to increase pressure, does not create a discharge pressure. Motive steam provides the energy necessary to compress and flow the mixture of motive and load to the operating pressure of a downstream condenser. If the pressure of the condenser is below the discharge capability of the ejector, the ejector will not cause the condenser to operate at a higher pressure. Conversely, if the operating pressure of a condenser downstream of an ejector is above the discharge capability of that ejector, referred to as a maximum discharge pressure (MDP), the performance of the ejector breaks down, the shock wave is lost, and typically suction pressure moves sharply higher. Suction pressure and therefore distillation column pressure may surge or become unstable once the shock wave is no longer present.

An ejector performance curve provides critical information about variables affecting performance. The two most important variables to understand and have correct for proper performance are: motive steam pressure and temperature; and the MDP an ejector is anticipated to operate against. Performance frustration and lost profit for a refiner stem most often from motive steam pressure falling below a minimum pressure or from discharge pressure in operation rising above MDP. In either of these two conditions, there is an abrupt negative change in performance, with distillation column operating pressure rising above its design operating pressure, and also pressure surging may occur. **Figure 3** shows a typical ejector performance curve. Notice that, for a given suction load, MDP capability increases with higher motive steam pressure. This particular ejector is designed for 7213 lb/h of water vapour equivalent load at 15 torr, discharging up to 104 torr when motive steam is at 220 psig. If motive steam pressure is 230, 240 or 250 psig, the MDP capability at 7213 lb/h of load is 109, 113 or 117 torr, respectively. Higher pressure motive will increase the motive mass flow rate along with the velocity exiting the converging-diverging nozzle and, therefore, energy from expansion increases, thus with higher motive pressure MDP capability is greater. A dashed line

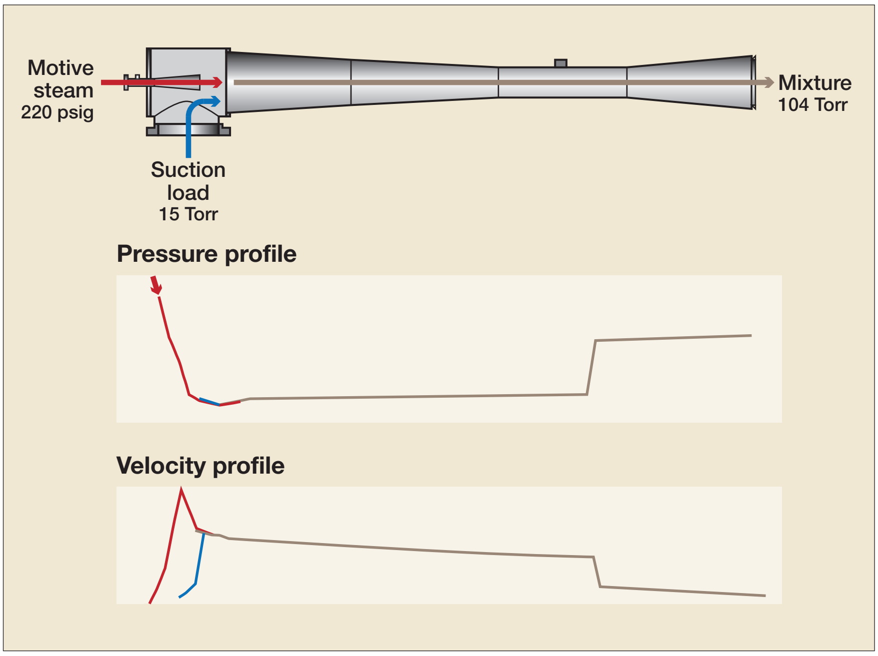

Figure 2 Pressure and velocity profiles within an ejector

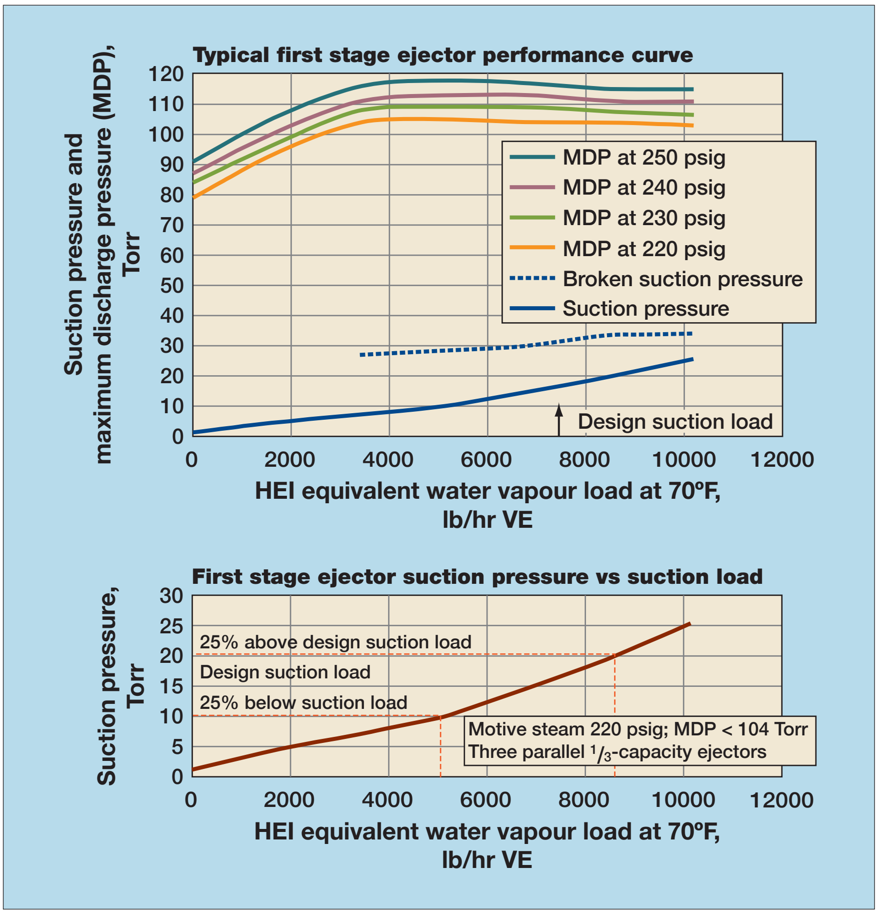

Figure 3 Typical ejector performance curve

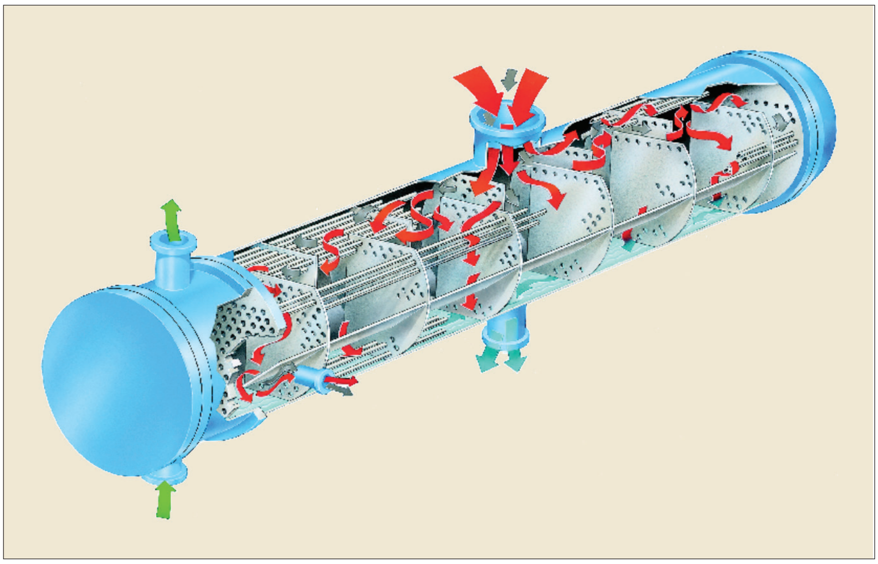

Figure 4 Cross-section of a TEMA "X" shell vacuum condenser with a longitudinal baffle for venting non-condensibles

shows an estimated suction pressure if the discharge pressure in operation exceeded MDP. There is essentially a doubling of the vacuum column discharge pressure, from 15 torr to 30 torr, should discharge pressure exceed MDP. That jump in pressure increases vacuum residuum, thereby reducing LVGO and HVGO cuts. The actual broken suction pressure will depend on discharge pressure. The higher the discharge pressure, the higher the broken suction pressure.

A similar break in performance arises when motive steam pressure is below 220 psig for example, while discharge pressure must be 104 torr.

In each case the break in performance is a result of insufficient energy available from the motive steam to perform the required compression. The shock wave breaks down, resulting in loss of compression across the ejector. Discharge pressure above MDP or motive pressure below design cause the shock wave to move out of the throat and into the converging section where it ultimately breaks down and compression is negatively impacted.

Vacuum system condensers

Condensers within an ejector system are positioned between ejector stages to condense steam and vapours in order to reduce energy requirements for the system. A vacuum condenser may also serve as a pre-condenser positioned between a vacuum column and an ejector system. By condensing steam and vapours it will reduce the loading to a downstream ejector, thereby lowering energy usage in the form of motive steam required by that ejector. A condenser within an ejector system is unlike a typical shell and tube heat exchanger, although it externally appears no different. It has similar construction features that follow Tubular Exchanger Manufacturer Association (TEMA) or American Petroleum Institute API 660 guidelines. However, the internal configuration is different due to operating under a vacuum, condensing vapours with non-condensibles present, handling non-ideally miscible condensates to ensure correct vapour-liquid equilibrium and to permit continual extracting of non-condensibles (see **Figure 4**). Distinct differences from conventional shell and tube heat exchangers are:

- Open areas above the tube bundle to permit flow distribution and reduce pressure loss
- Lack of conventional flow directing segmental or double segmental baffling in order to reduce pressure loss and appropriately manage vapour-liquid equilibrium
- Extracting non-condensible gases within a tube bundle, in most cases.

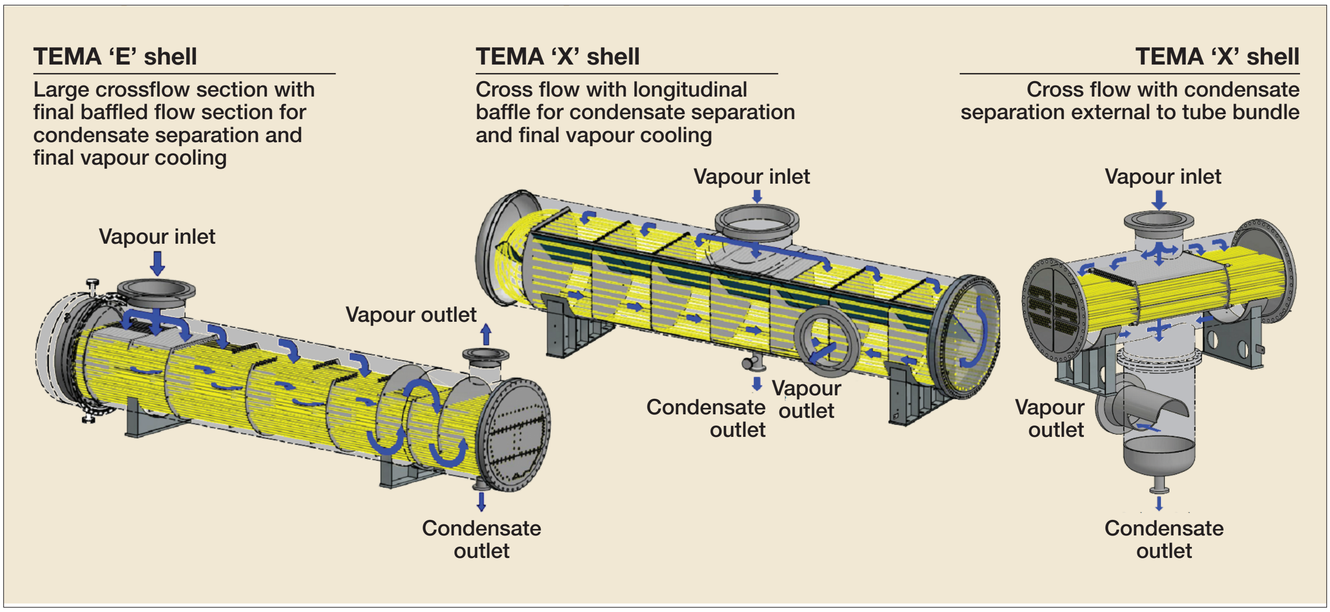

Figure 5 Three types of TEMA shell vacuum condensers

There are three typical configurations and the choice will depend upon the operating pressure, amount and type of condensable hydrocarbon vapours, and miscible condensate concerns related to vapour-liquid equilibrium. **Figure 5** shows the three types.

Vacuum column vapours are generally condensed shell side with condensing occurring on the outside diameters of the tubes. The shell side heat transfer coefficient is influenced by a) cracked gas, inerts and uncondensed vapour, b) the condensing coefficient for the steam and for the hydrocarbons, and c) the condensate film coefficient. A generalised resistance proration formula for the shell side heat transfer coefficient is:

$$hshellside = \left(\frac{1}{hgases\ and\ vapors} + \frac{1}{hcondensing} + \frac{1}{hcondensate\ film}\right)^{-1}$$

$h_{gases\ and\ vapours}$ will decrease from the top of the tube field to the bottom due to the increasing mole fraction of gases that are present as the vapours are condensed (increasing volume fraction of the gases).

$h_{condensate\ film}$ will decrease from the top of the tube field to the bottom due to the increasing thickness of condensate film. Moreover, hydrocarbon condensate forms a higher resistance to effective heat transfer than steam condensate. Hydrocarbon condensate has a much lower thermal conductivity, resulting in a lower ability to affect temperature change across the condensate film's thickness.

$H_{condensing}$ will vary based on whether steam or hydrocarbons are condensing at a given temperature or if both are condensing at that temperature.

The controlling coefficients are $h_{gases\ and\ vapours}$ and $h_{condensate\ film}$ with each varying throughout the heat exchanger tube bundle and becoming the lowest near the exit of a condenser due to the volume of gases being the highest and the condensate film thickness the greatest. **Figure 6** illustrates the temperature gradient for heat and mass transfer. Importantly, condensate film surface temperature must be at or below the local vapour dew point for condensation to occur. Temperature across the condensate film varies with condensate physical properties, where hydrocarbon condensate provides higher resistance to heat transfer than water, and a thicker condensate film results in greater resistance as well. **Figure 7** illustrates the challenge when a mixture of hydrocarbon vapours and steam must condense, and typically hydrocarbon vapours have a higher dew point than steam and will condense before steam. As **Figure 7** shows, hydrocarbon condensate film temperature must be below, in this example, 118°F (48°C) before steam will condense.

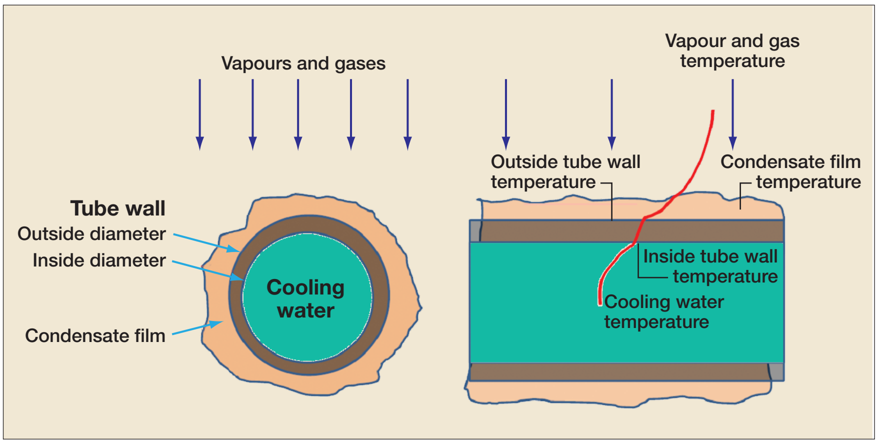

Figure 6 Temperature gradient hot side to cold side, across condensate film and tube wall

Specifying the distillation overhead loading to the ejector system

A third common performance issue for ejector systems in refinery vacuum distillation service is the actual compositional make-up of the loading to the ejector system exiting the vacuum column. The performance issue is often traced back to process simulation of the crude oil itself, the actual performance of the fired heaters, the performance of the atmospheric distillation column, or the vacuum column's performance. The vacuum

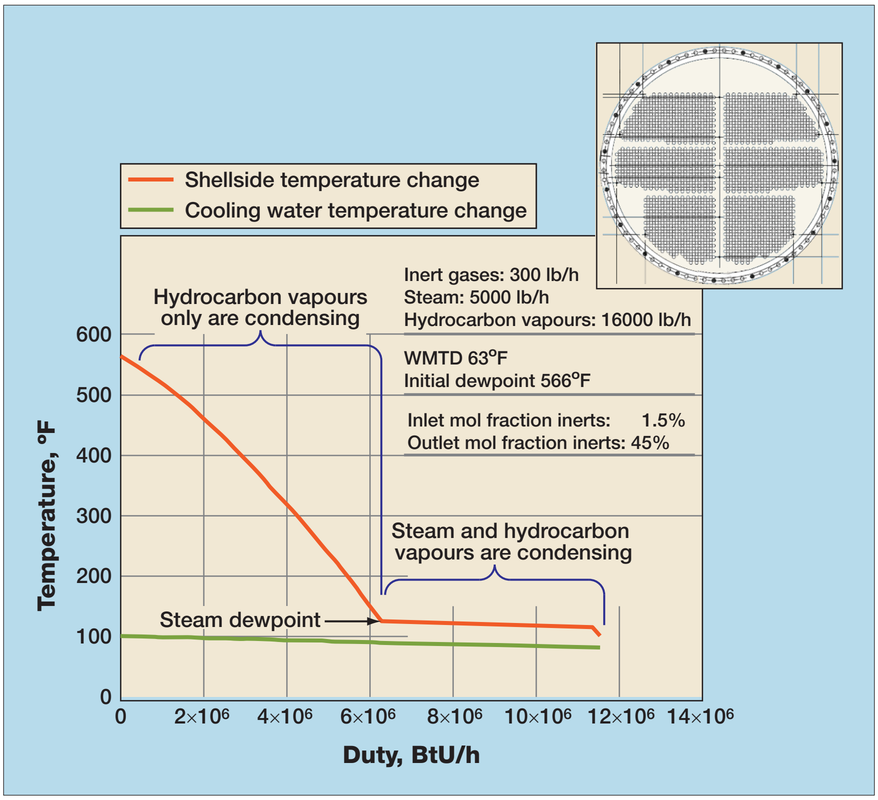

Figure 7 Vacuum column precondenser condensing curve and tube bundle

column overhead load to an ejector system is typically broken down as:

1. Steam used to maintain velocity in the fired heaters and for controlling partial pressure of hydrocarbons in the distillation column. This is generally predictable due to mass flow rate being set by the supply pressure and orifice diameters.

2. Cracked gases are generated in the fired heater. The amount of cracked gases will vary with the crude slate, the operating temperature of the fired heaters, and the amount of velocity steam. Typically, the higher the temperature, the greater the level of cracked gases. Also, the vacuum distillation process is at sub-atmospheric conditions, therefore ingress of air into the system must be considered and this is usually grouped with the cracked gases. Most often, C_6 hydrocarbons or lighter, where molecular weight is less than 90 lb/lb-mole, are grouped as cracked gases and considered non-condensible within the ejector system. To add safety, C_7 or C_8 hydrocarbons or lighter may be considered as non-condensible gases.

3. Condensible hydrocarbon vapours are generally C_7 and heavier hydrocarbons that, to varying degrees, will condense within the ejector system. Condensible hydrocarbons are developed using standard techniques that assess how much of the crude oil is vaporised at various temperatures. For example, 10% of the liquid volume is vaporised at 220°F (104°C) and by 250°F (120°C) 30% is vaporised. Hereto, crude slate affects how a crude oil is characterised. Light sweet, heavy sour, light tight shale and crude blends will all have unique characterisations. No two crude oils are alike. Moreover, understanding the method used to provide the distillation assay information is important: is it true boiling point, ASTM D-86, ASTM D-1160 or ASTM D-2887 information? Software or *API Technical Data Book* may be used for inter-conversion from one assay basis to another.

Table 1 shows an example of a typical compositional breakdown of vacuum distillation column overhead loading to an ejector system.

Vacuum distillation column overhead loading to an ejector system

Ejector suction pressure	15 torr	
Suction temperature	200°F	
Composition of suction load		
Component	#/hr	MW
Steam	12 200	18
Inerts (cracked gases)	1500	28
Hydrocarbon vapours	15 000	151.4
Total	**28 700**	**34.6**
HEI steam equivalent	21 640	
Load to each 1/3 first stage ejector	7213	
Hydrocarbon vapour normal boiling point breakdown		
Normal boiling point	#/hr	MW
150°F	750	100
220°F	750	110
280°F	3000	125
340°F	3000	150
400°F	3000	165
460°F	3000	190
550°F	1500	220

Table 1

Cautionary considerations related to condensable hydrocarbon loading

For expediency, process licensors may provide simply an average molecular weight for the condensible hydrocarbons along with normal boil point distribution. For example, from **Table 1**, the average molecular weight for the hydrocarbons is 151.4 lb/lb-mole while in actuality molecular weight varies with normal boiling point. The directional impact of this seemingly straightforward simplification is that more lower normal boiling point hydrocarbons are predicted to condense with a molecular weight of 151.4 versus, for example, 110 lb/lb-mole for a normal boiling point 220°F (104°C) pseudo-component. Consequently, in operation more hydrocarbon vapours exit a vacuum condenser than simulation would predict and potentially overload a downstream ejector. Best practice is to provide ASTM D-86 distillation assay information along with pseudo-component normal boiling points with corresponding molecular weights.

A common finding in operation is that the amount of condensible hydrocarbons exiting a vacuum column exceed the design basis. There are a number of possible causes for this:

- Atmospheric column over-flash
- Damaged stripping trays in atmospheric column
- Vacuum column top temperature
- LVGO vapour pressure
- Vacuum column stripping efficiency
- LVGO pumparound entrainment
- Varying crude slate
- Slop oil or recovered oil processing.

It is desirable to conduct a rigorous sensitivity analysis for 'what if' factors that could impact condensable hydrocarbon loading in operation, and then safely specify that loading for ejector system design. Conventional thinking is that excess hydrocarbon loading is unimportant or not materially impactful to ejector system operation. This notion stems from an ejector performance curve where, for example, if loading from **Table 1** was 30 000 lb/h of condensable hydrocarbons instead of the design 15000 lb/h, plant engineering would expect the first stage ejector to follow its performance curve. With 100% more condensible hydrocarbon loading, the Heat Exchange Institute (HEI) water vapour equivalent load is 29 300 lb/h or approximately 35% more than design 21 640 lb/h of HEI water vapour equivalent. Therefore plant engineering anticipates first ejector suction pressure to rise to 24 torr. Too often, 24 torr is not realised; however, the

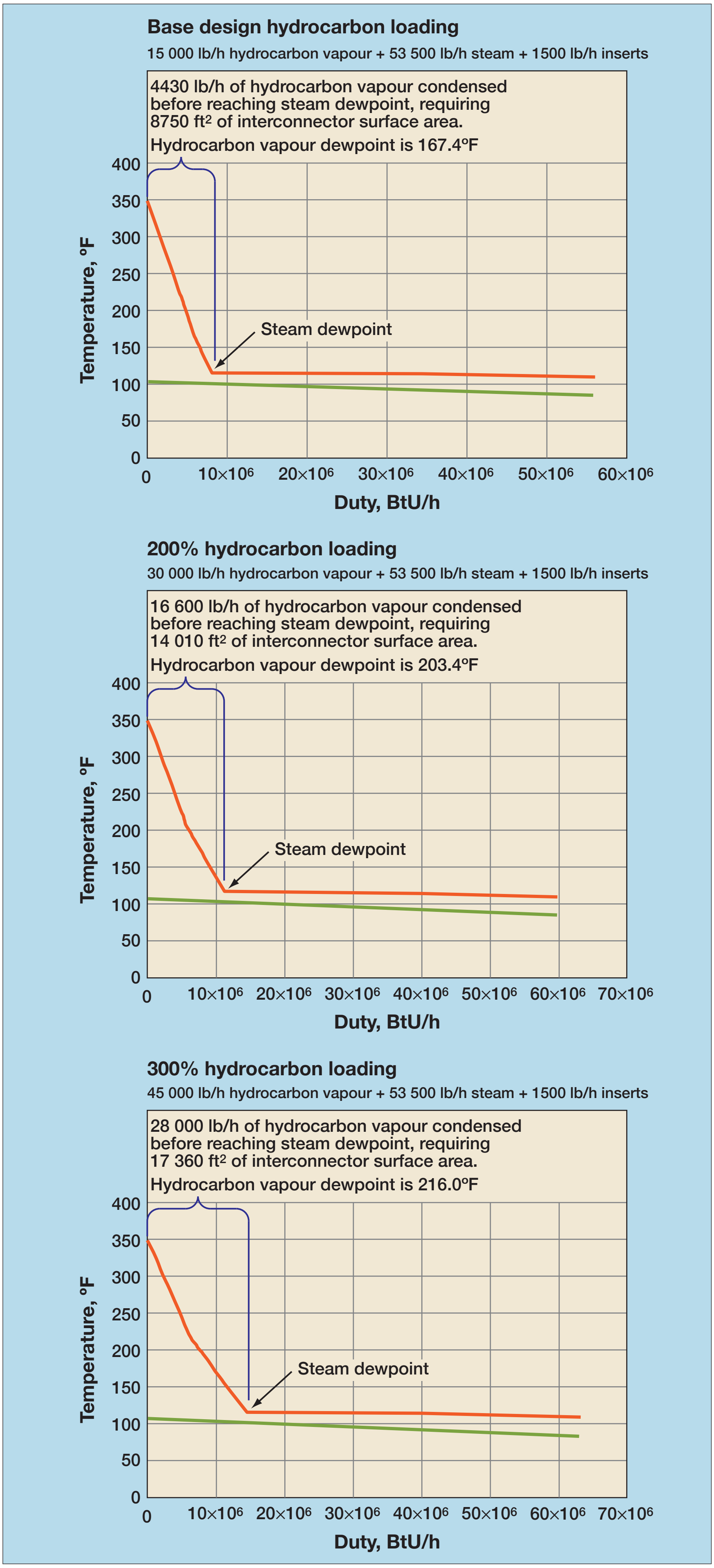

Figure 8 Effects of varying hydrocarbon loading

pressure rises to 30-40 torr. Why?

What occurs in practice is that condensing efficiency in the first inter-condenser is reduced due to the greater hydrocarbon loading. There are two aspects to consider with added hydrocarbon loading:

1. How has it changed the dew point and thus the log mean temperature difference (LMTD)?

2. How will the greater hydrocarbon film thickness on the heat transfer tubes reduce heat transfer?

In most cases hydrocarbon vapours condense before steam reaches its dewpoint. The extent of hydrocarbon condensate cooling that must occur before the condensate film is below the steam dew point can materially alter condenser thermal capability. Often the effective overall heat transfer rate for the condenser drops measurably and as a consequence the operating pressure of the condenser rises in order to increase LMTD. The fundamental equation Q=U*A*LMTD is followed. Area (A) is fixed, Duty (Q) is known, and if overall heat transfer rate (U) is lowered due to excess hydrocarbon loading then LMTD must rise to balance the equation. To drive higher LMTD, operating pressure increases, which may result in the operating pressure exceeding the MDP capability of the ejector and, consequently, suction pressure breaks and is observed as a sharp rise above its predicted value.

The following evaluates a case where design basis was 15 000 lb/h of condensible hydrocarbon loading from the vacuum column; however, in the field, the loading was found to be two to three times more vapour based on oil measured from the condensate receiver. Moreover, the excessive hydrocarbon loading had higher percentages of higher molecular weight/higher normal boiling point hydrocarbons. See **Figure 8** for differences in the heat release curve, the amount of hydrocarbons that have condensed before steam reaches its dew point, and the additional inter-condenser surface area needed to address hydrocarbon condensing before steam begins to condense.

In this case, the base inter-condenser design was 26 240 ft^2 (2438 m^2). For two to three times the hydrocarbon vapour load, the required surface area is 31 500ft^2 (2926 m^2) to 34 850 ft^2 (3238 m^2). Put differently, area cannot be added to an installed condenser that was designed for 26 240 ft^2. Therefore, for 30 000 lb/h or 45 000 lb/h of hydrocarbon vapour loading the condenser is 20% or 33% under-surfaced. As a result, because surface area is now fixed, LMTD must rise to balance the fundamental equation Q = U*A*LMTD. To effect an increase in LMTD, condenser operating pressure, in this example, must rise 18 torr for the 45 000 lb/h case. At this required operating pressure, the first stage ejector MDP is surpassed by 18 torr and, therefore, the first stage ejector breaks performance. Consequently, the vacuum column pressure rises appreciably and potentially is unstable.

The process team wonders why the added hydrocarbon loading is affecting the system this way and why the first stage ejector is not simply tracking its performance curve. The root cause is the suppression of heat transfer in the first inter-condenser due to the excessive hydrocarbon loading that leads to a rise in its operating pressure. Once first inter-condenser operating pressure surpasses the MDP of the ejector that precedes it – in this example, MDP is 83 torr – vacuum column pressure abruptly rises higher.

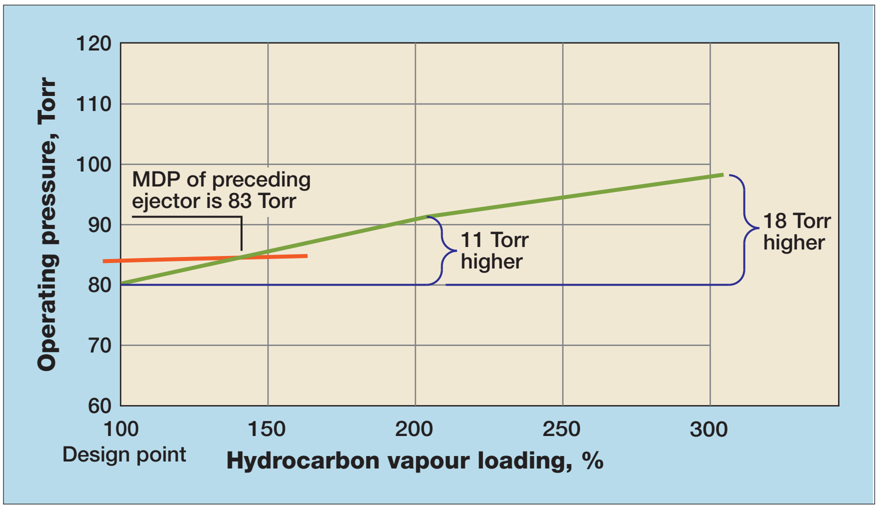

Figure 9 First intercondenser response to hydrocarbon loading

$$\text{mass flowrate of steam} = \frac{\text{lb} - \text{moles of inerts} * \text{partial pressure of steam} * 18}{\text{Condenser Pressure} - \text{summation of partial pressure of all condensible vapor}}$$

$$\text{mass flowrate of hydrocarbon}(i) = \frac{\text{lb} - \text{moles of inerts} * \text{partial pressure of hydrocarbon}(i) * \text{MW of hydrocarbon}(i)}{\text{Condenser Pressure} - \text{summation of partial pressure of all condensible vapor}}$$

Predicting and specifying design cracked gas load

Specifying conservatively the design cracked gas load is wise. Cracked gases are inerts within an ejector system and will not condense. At a given temperature and pressure within a condenser, steam and hydrocarbon vapours are directly correlated to the amount of inerts. The greater the level of inerts, the greater the amount of steam and hydrocarbon vapours that saturate the inerts and exit the condenser as vapours.

Simplified equations for the amount of vapour that saturates inerts gases are:

The partial pressure of steam is typically the saturation pressure corresponding to a given temperature because steam is immiscible in hydrocarbon condensate. The partial pressure of a hydrocarbon is the product of its mole fraction in the condensate multiplied by an activity coefficient multiplied by its saturation pressure corresponding to a given temperature. Hydrocarbon partial pressures are not straightforward because condensates that form follow non-ideal miscibility vapour-liquid equilibrium. Regardless of the complicated formula, the mass flow rate of vapour is directly proportional to the amount of inerts. If there is twice as much of the cracked gases, there will be twice as much vapour exiting the condenser and, therefore, twice the load for an ejector downstream.

Best practices for specifying ejector systems in crude oil vacuum fractionation service

1. Provide pseudo-component normal boiling point breakdown with individual molecular weight for each pseudo-component. If true boiling point, D-2887 or D-1160 assay information is available, to avoid uncertainty convert it to D-86 (normal boiling point) information

2. Run sensitivity analyses for atmospheric column overflash, vacuum column stripping efficiency and potential column top temperatures to understand the upper range for hydrocarbon vapour exiting the top of the vacuum column. Be conservative (overstate) regarding the mass flow rate.

3. Be careful to select conservatively the normal boiling boil distribution for the pseudo-components. A general guideline is that a greater weighting of lower normal boiling point pseudo-components results in less that will condense within the ejector system. A greater weighting of higher normal boiling point pseudo-components will result in more condensing of hydrocarbons in the first stage condenser. Understand the impact of greater hydrocarbon loading on suppressing the overall heat transfer performance. Consider field experience for how actual performance relates to a distillation column's simulated performance, in particular stripping efficiency, LVGO pumparound entrainment, and various 'what if' sensitivity analyses, to define range of performance outcomes.

4. Cracked gas and inerts should

be overstated from test data to account for actual fired heater performance. Make certain second and third stage ejectors are adequately sized to allow for errors in estimating the amount of non-condensibles. Overload of cracked gases presents problems for the second or third stage ejectors that manifest themselves as high and potentially unstable vacuum column pressure.

5. Steam loading to the ejector system is predictable based on supply pressure and the temperature of the steam and the orifice diameters that meter the steam to the vacuum column. There typically is little performance risk introduced by steam load estimates.

6. Motive steam supply conditions require thoughtful consideration. Ejectors are sensitive to steam pressure, especially when designed at the minimum supply pressure. Invariably over time, with added demands on the steam generating system, supply pressure to an ejector system will fluctuate downward. A safe practice that will use somewhat greater steam, however, and aid in performance reliability is to set motive pressure to establish a shock wave against the expected maximum discharge pressure at 90 to 95% of minimum supply pressure. This will provide operating flexibility and reliable performance that can be refined with a motive steam pressure reducing station that is typically in the steam supply system. This practice will eliminate frustration and costly profit shortfalls when vacuum column pressure increases due to insufficient motive steam pressure to an ejector system, resulting in broken ejector system performance where distillation column pressure increases dramatically.

7. Provide overlap between an ejector discharge and downstream condenser operating pressure. There are always hydraulic piping losses between ejector discharge and inlet to a downstream condenser, along with cooling water inlet temperature fluctuations and fouling within the condensers where overlap provides a margin of safety. It is always best practice to perform a hydraulic loss calculation once actual piping isometrics are complete. A good rule of thumb is to provide 10 to 15% overlap between an ejector MDP and the operating pressure of the downstream condenser. For example, if the operating pressure of a condenser is 100 torr or 250 torr, the preceding ejector should have an MDP >110-115 torr or 275-288 torr, respectively. Layout is not finalised until detailed engineering is completed and, to avoid time-consuming or frustrating iterations after an order, use the overlap concept to establish utility consumption and equipment sizes.

8. Cooling water inlet temperature should be considered the highest possible that the site will experience. Do not, for example, select a temperature that is satisfactory 95% of the time, say 85°F (29°C), when the plant water system can be as warm at 88°F (31°C). A few degrees error can result in several weeks of frustration in summer months when vacuum column pressure increases or becomes unstable due to broken ejector system performance where distillation column pressure increases dramatically.

Absolute best practice is to involve an ejector system supplier early in the specifying process to identify performance risks and methods to mitigate risk

9. Do not permit condenser designs where flow directing baffles are used, such as typical segmental or double segmental baffles, the entire length of the tubing. Hydrocarbon condensates are non-ideally miscible and require a configuration supporting integral condensation where vapours and condensate remain in contact throughout the majority of the condenser. Baffled units result in differential condensation that will lead to improper system performance due to greater percentages of the hydrocarbon load remaining in the vapour phase.

10. Ejector and condenser configuration should consider the second and third stage ejectors being at 150% capacity, for instance three 50% elements. This is so that if cracked gas estimation for design is too low, the system can accommodate up to 150% of design cracked gases and inerts. If actual cracked gas loading is below design, then it is possible to leave one of the elements idle so as not to waste energy. The condensers following the second and third stages should have 150% capacity to allow for all three ejector elements to be in operation. For the first stage ejectors and first stage condensers, consider multiple elements, such as three 40% trains or some other combination that provides operating flexibility.

11. Provide instrument connections at the suction and discharge of each ejector and at each connection for the condensers. This is important for field measurements. It is not uncommon for the control room DCS readings to be inaccurate, therefore field measurements can prove invaluable when performance issues arise. Having such connections available permits field measurements to be taken readily to aid in evaluating system performance.

12 Absolute best practice is to involve an ejector system supplier early in the specifying process to identify performance risks and methods to mitigate risk.

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