HYDROCARBON PROCESSING®

Originally appeared in *Hydrocarbon Processing,* September 2023. Posted with permission.

REFINING TECHNOLOGY

Reliable vacuum distillation requires reliable ejector system performance

J. R. LINES, Graham Corp., Batavia, New York

A fuels-oriented crude oil refinery must have good performance from its vacuum distillation section to deliver targeted yield and throughput. Reliable ejector system performance is critical to achieving that objective. A number of articles address the details for how ejectors and their associated condensers operate.[1–4] This article develops insights into how certain design elements influence ejector system performance. These discoveries were drawn from real-world experiences with the specifications for and operation of ejector systems in crude oil refining service.

Many refiners achieve solid performance from their vacuum distillation units and supporting ejector systems. However, when performance falls short, it can be a costly and frustrating experience. This article will highlight important variables to consider and how they affect performance.

Motive steam pressure and temperature. Supply conditions for motive steam used by ejectors in any service are important variables to define. Ejectors utilize a fixed orifice diameter that establishes the amount of steam that will pass through the orifice, depending on steam operating pressure and temperature. The higher the pressure, the greater the mass flowrate through the orifice; conversely, the higher the temperature, the lower the mass flowrate passing through the orifice. Motive steam will always be more than two times the ejector suction pressure—therefore, the orifice serves as a critical flow orifice where mass flowrate will pass no faster than sonic velocity. The Heat Exchange Institute (HEI) standard for steam jet ejectors provides an equation (Eq. 1) for the mass flowrate of steam through a critical orifice:

$$\textit{Mass flowrate} = 892.4 \times C_d \times Dia^2 \times (\textit{motive pressure/motive specific volume})^{0.5} \qquad (1)$$

where,

Mass flowrate: lb/hr
Pressure: psia
Specific volume: ft^3/lb
Dia: Nozzle orifice diameter, in.
C_d: Orifice discharge coefficient (typically 0.92–0.98)

Ejector performance curves for different motive pressure and temperature depict how suction pressure and maximum discharge pressure of the ejector vary with suction loading. Suction loading is defined as either a steam or air equivalent load.

FIG. 1 shows an example where an ejector system is designed for crude vacuum distillation service with the conditions shown in **TABLE 1.**

To build an additional factor of safety, the design specification specifies that the ejector system design motive pressure should be 85 psig at 440°F, which is 30 psi below the normal supply pressure and 15 psi below the expected minimum supply pressure.

The system being designed has the arrangement—1/4 of the capacity trains—and applies the HEI standard for developing steam equivalent loading. Each of the first-stage ejectors are designed to handle 4,500 lb/hr of steam equivalent loading.

A performance map (**FIG. 1**) for the first-stage ejector and condenser represents system performance at three different motive steam pressures of 85 psig, 100 psig and 115 psig, and a supply pressure at 440°F.

As the performance map indicates, there is little variation in suction pressure for the three different motive steam supply pressures. Although it is unclear from the graph, there is a minor rise in suction pressure with 4,500 lb/hr steam equivalent load between 85 psig and 115 psig as the pressure increases from 20 mmHg (millimeters of mercury) to 21.2 mmHg.

The notable difference is in the discharge capability of the ejector. As motive supply pressure increases, the maximum discharge pressure that the ejector can achieve increases. At 85 psig motive pressure, the ejector can achieve 103 mmHg maximum discharge pressure at a design 4,500 lb/hr steam equivalent suction load. Alternatively, with 100 psig or 115 psig motive pressure, a 119 mmHg and 132 mmHg maximum discharge pressure can be achieved, respectively.

There is a tradeoff for the greater discharge capability at higher motive supply pressure. Since the orifice diameter is fixed—as the motive steam pressure increases, more mass flowrate passes through the orifice, which is the energy increase creating greater discharge pressure capability—this adds to the condensing duty of the condenser receiving the ejector discharge flow. Approximately 30% greater steam flowrate passes through the orifice at 115 psig vs. 85 psig, thus adding more condensing duty and temperature rise on the cooling water. Consequently, the condenser pressure must rise.

It is reasonable to assume the overall heat transfer rate will be constant regardless of motive supply pressure; therefore, in the duty equation $Q = U \times A \times MTD$, UA is a constant. Rearranging, $Q/MTD = UA$ and if UA is a constant while the duty increases due to greater motive steam, the MTD must increase to balance the equation. The only way MTD increases is if the pressure rises and increases the temperature at which the steam will begin to condense in the condenser. As can be seen above (**FIG. 1**), the

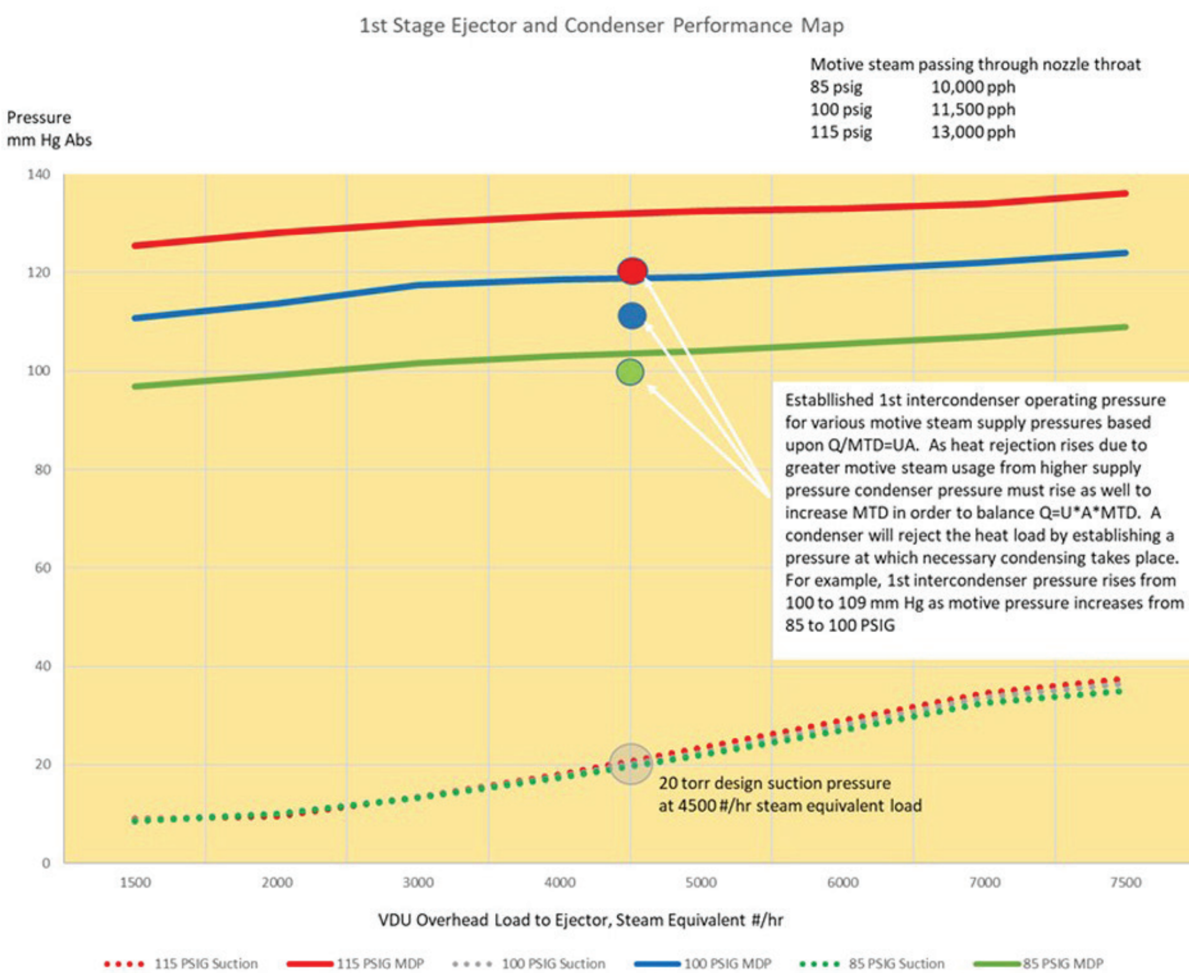

FIG. 1. A performance map for the first-stage ejector and condenser represents system performance at three different motive steam pressures.

condenser operating pressure rises from 100 mmHg with 85 psig motive to 109 mmHg with 100 psig motive, and to 123.5 mmHg with 115 psig motive, with MTD correspondingly increasing from 24.2°F to 27°F to 30.8°F.

An important assessment becomes the impact of increasing temperature rise due to the greater heat rejection caused by the mass flowrate increase from higher motive pressure. Many refinery vacuum distillation ejector systems have the cooling water circuit flow in series from the first intercondenser to the second intercondenser and after condenser. Considering **FIG. 1,** the outlet temperature of the cooling water flowing from the first intercondenser to the inlet of the second intercondenser is 102°F when the motive is at 85 psig and 105.3°F when 115 psig is the steam pressure. The performance of the second intercondenser requires careful scrutiny to ensure it will not impede the performance or cause a break in the ejector system performance. Here, too, higher motive pressure increases the discharge capability of the second-stage ejector; however, the effect of higher condenser pressure increasing MTD is not as pronounced with greater operating pressure.

Arrangement. The orientation of the first-stage ejector with respect to the vacuum column, as well as the piping layout from an ejector to its condenser, must be considered. This is an overlooked aspect of ejector system integration into a vacuum distillation unit and often settled after a vendor has been selected. This requires discussion before vendor selection to:

- Align vendor bids so that each vendor is considering hydraulic losses similarly
- Calculate utility consumption with reasonable accuracy before selecting a vendor
- Minimize post-award change orders and design iteration.

The most common approach is to mount the first-stage ejectors vertically close to the column top, minimizing hydraulic losses and utilizing the ejector length to serve as piping between the tower top and structure level where the first intercondensers will be located. There are approaches where the first-stage ejectors are mounted horizontally on the structure next to the vacuum column with piping from the column overhead to the structure. A basic best practice (with respect to the ejector system cost and energy consumption) is to minimize hydraulic losses between the vacuum column and first-stage ejector—it is less impactful on the ejector system cost and overall energy consumption to incur piping hydraulic losses after the first-stage ejector discharge. A 2-mmHg piping hydraulic loss preceding the first-stage ejector is more impactful than a 5-torr piping loss between the first-stage ejector discharge and the first intercondenser inlet.

Various arrangements for the vacuum column to the first-stage ejector and from the first-stage ejector to the first intercondenser. The final layout is difficult to determine before detailed engineering and ejector system vendor selection. The actual arrangement, forces and moments on equipment from thermal growth and static loads can influence the final pip-

TABLE 1. Crude vacuum distillation service with conditions

Service	Measurement	
Ejector system suction pressure	20 torr	
Ejector system suction temperature	150°F	
Ejector system discharge pressure	1 psig	
Vacuum column overhead steam loading	13,500 lb/hr	MW = 18
Vacuum column overhead hydrocarbon vapor loading	4,050 lb/hr	MW = 153.7
Vacuum column overhead noncondensible gas loading	2,700 lb/hr	MW = 33
Cooling water supply temperature	90°F	
Maximum cooling water return temperature	110°F	
Normal motive steam pressure at 440°F	115 psig	
Minimum motive steam pressure at 440°F	100 psig	
Ejector system arranged in four(4) parallel 25% capacity trains		

ing between the ejector and condenser. This will affect hydraulic losses that must be accounted for and, therefore, final motive steam consumption. For example, expansion loops may be used to address stresses from thermal growth, as **FIG. 2** illustrates.

In this particular instance, to address thermal growth-related stresses on the system, expansion loops were included in the final piping. This increased the hydraulic loss between the first-stage ejector discharge and the inlet to the first intercondenser from 1 mmHg to 2.5 mmHg. That may seem insignificant, but it is critical to consider with respect to first-stage ejector discharge capability. Using the example above, if the first-stage ejector cannot compress the operating pressure of the first intercondenser because piping losses were 2.5 mmHg rather than 1 mmHg, the overall system performance will fail. Consequently, the vacuum column overhead pressure of 20 mmHg will increase significantly, perhaps to > 30 mmHg, and be unstable. This will result in a tremendous loss in yield for the refiner.

It is critical for the engineering, procurement and construction (EPC), licensor and ejector system vendor to collaborate early in the process about layout, options and CAPEX/OPEX trade-off. When this critical conversation occurs after an ejector system order has been placed, it can result in surprises, design iterations, cost creep and schedule delays.

Recycle control. Vacuum column throughput will vary. Refiners often wish to control the tower top pressure when running at reduced throughput by recycling the load within an ejector system back to the tower top. This is very common. It is important to consider a proper location for where to pull the recycle back into the tower top.

The best practice is to have a recycle load where the mole fraction of non-condensible components is below the mole fraction of the non-condensible components for the design overhead load. In many designs, this is located only at the first-stage ejector discharge.

From a cost and operational standpoint, it is ideal to pull the recycle from the location of the highest operating pressure: the discharge of the last-stage ejector. To eliminate any issues with recycle control, map out the mole fraction of non-condensible gases throughout the ejector system. Select the location where the mole fraction of non-condensible gases in the recycle stream will be below that of the overhead load from the vacuum. If a recycle has higher mole fraction non-condensible gases, then the load entering the first-stage ejector will eventually have an elevated non-condensible gas level that negatively affects condenser performance. As cited in the previous example, the pressure may rise, from 20 mmHg to 30 mmHg (or higher) and the system will be unstable, ultimately leading to a system break.

FIG. 2. Vertically mounted first-stage ejectors and piping with expansion loops to address piping stresses that added hydraulic loss between first-stage ejectors and six intercondensers.

Mole fraction non-condensibles for different locations within the ejector system for the above example are listed in **TABLE 2.**

Overhead loading. The vacuum column overhead load will consist of steam, hydrocarbon vapors and various non-condensible gases. The amount of steam is tied to the steam used by the fired heater to affect

TABLE 2. Mole fraction non-condensibles for different locations within the ejector system

	Column overhead	First-stage ejector discharge	First intercondensers discharge	Second-stage ejectors discharge	Last ejectors stage discharge
Steam	13,500	53,500	2,065	10,250	11,000
Hydrocarbon vapor	4,050	4,050	3,885	3,885	2,457
Non-condensible gases	2,700	2,700	2,700	2,700	2,700
Mole fraction non-condensibles	9.50%	2.60%	36.20%	12%	11.20%
Remark		Suitable for recycle	Unsuitable for recycle	Unsuitable for recycle	Unsuitable for recycle

residence time as well as the amount of stripping steam important for distillation. Nonetheless, the steam in the overhead to the ejector system is predicted reliably. More challenging is the composition of the hydrocarbon loading and the amount of non-condensible gases. Both can vary due to crude slate, tower top temperature, cutpoint, fired heater performance, operating pressure and other variables.

Two key influencers for satisfactory performance of the ejector system are:

1. A reliable estimate of the amount of fired heater-cracked gases and other non-condensibles
2. The overhead hydrocarbon loading, and whether that load contains liquid droplets due to the absence of mist elimination at the top of the tower.

Excessive non-condensible loading will cause a break in ejector system performance where the vacuum column pressure increases 10 torr–15 torr, for example. This break is caused by the excessive non-condensible loading decreasing the amount of vapors condensed in a condenser that then must be handled at the downstream ejector. The preceding ejector cannot compress to a pressure needed for the downstream ejector to operate at the required elevated loading. Unwanted performance breaks and pressure in the column increase.

If not considered thoroughly, excessive hydrocarbon loading—primarily in the form of droplets that carryover to the first-stage ejector with the overhead load—can affect the maximum discharge capability of an ejector. The droplets are of a far higher density than the hydrocarbon vapors. When the droplets are impacted by the motive steam, energy is lost by the motive steam. This is much different than when the motive steam impacts hydrocarbon vapors with far lower density than droplets, and this lessens the discharge capability of that ejector. When this is known during the upfront design of the ejector system, this can be adequately addressed. If missed, the performance of the vacuum distillation column can be compromised.

The author's company has been involved in two successful revamp projects where liquid droplets in the overhead load were extensive and not considered. Initially, each of the refiners had unsatisfactory vacuum column performance in the summer months because the ejector system broke performance.

To remedy the poor performance, two retrofits were carried out in both cases. Mist elimination was added back into the tower top and the ejector motive nozzle was configured to produce greater discharge pressure without modifying motive steam consumption. The trade-off for the ejector was slightly diminished suction load handling capacity but far greater discharge capability. In both cases, after the retrofit the refiner was able to achieve the desired results even during the summer when cooling water is at its warmest.

Fouling. Fouling is a variable that is important to assess. Crude oils can have different fouling tendencies or foul very little, and the cooling water side can also present fouling issues. The condenser overall design heat transfer rates (U value) of a typical vacuum distillation ejector system are detailed in **TABLE 3.**

The rates can be different then the typical ranges shown in **TABLE 3** based upon the amount of non-condensible gases and hydrocarbon loading, whether vapor and/or liquid. Overall fouling factors for refinery service have ranged between 0.002 and 0.005, generally. The greatest CAPEX factor for a refinery vacuum distillation ejector system is the size of the first intercondensers. When considering the excess area 0.004 hr/ft^2/°F/Btu provides, it might be ideal to lower the overall fouling factor. In **TABLE 3,** for the first intercondenser, if 0.0025 overall fouling was applied rather than 0.004, the U value might range between 145 hr/ft^2/°F/Btu and 195 hr/ft^2/°F/Btu—as such, the condenser would be about 25% smaller in surface area and the overall system about 15% less expensive.

Establishing a prudent fouling factor is desirable considering the industry drive toward longer intervals between turnarounds or planned shutdowns. Process side fouling can be extensive, as shown in **FIG. 3.** Moreover, the cooling water side can present fouling challenges based of the sources for cooling water.

TABLE 3. The condenser overall design heat transfer rates (U value) of a typical vacuum distillation ejector system

	U value, Btu/hr/ft^2 °F	Excess area with 0.004 hr/ft^2/°F/Btu fouling
First intercondenser	120–150	90%–150%
Second intercondenser	100–130	65%–110%
Aftercondenser	80–110	50%–80%

A good understanding of the fouling tendency of both the process side (in particular, the hydrocarbons) and the cooling water to ensure an adequate fouling factor or excess surface area will permit an acceptable run time between planned maintenance intervals.

Cooling water supply temperature. In regions where summer temperatures may be extreme and certainly for older installations where the current hot summer seasons were not expected when ejector systems were designed, inlet cooling water temperature is a critical variable influencing reliable system performance. In the example used throughout this article, the cooling water inlet supply temperature for design purposes was set at 90°F. The data historian for cooling water temperature is noted in **FIG. 4,** which shows that 35 days in the summer months had an inlet water supply temperature above the design basis of 90°F.

It is not unusual in the U.S. Gulf Coast, the Middle East or Southeast Asia for broken ejector system performance to occur in the hot summer months with cooling water is at its warmest and above the design basis. When this performance break happens,

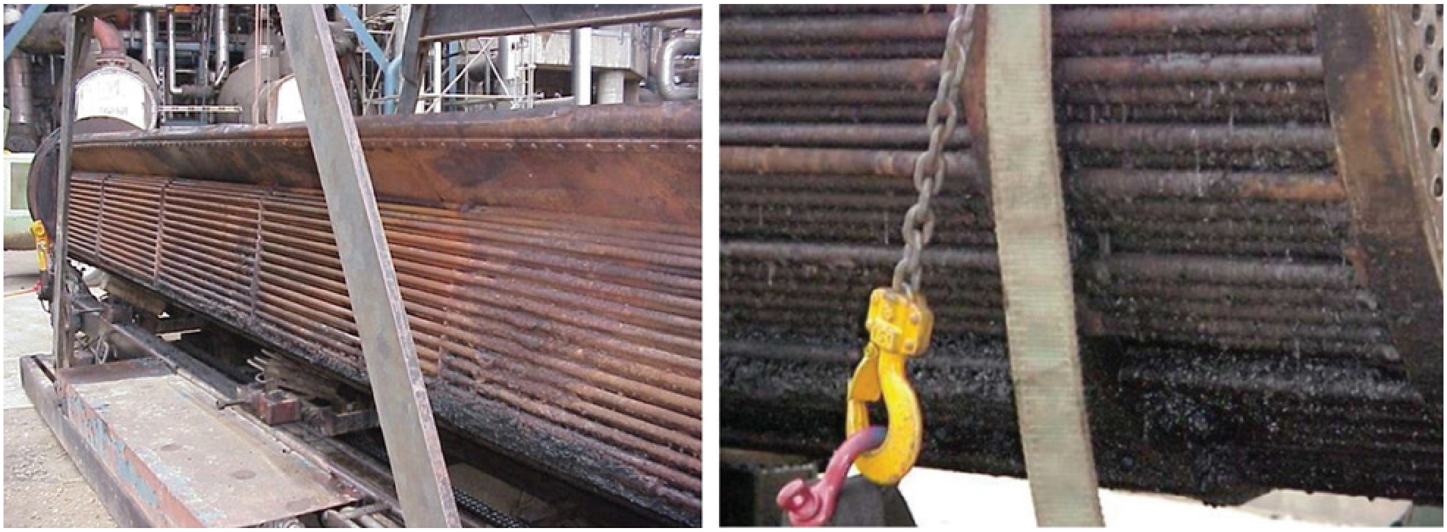

FIG. 3. Shell-side fouling from hydrocarbons in vacuum column overhead loading.

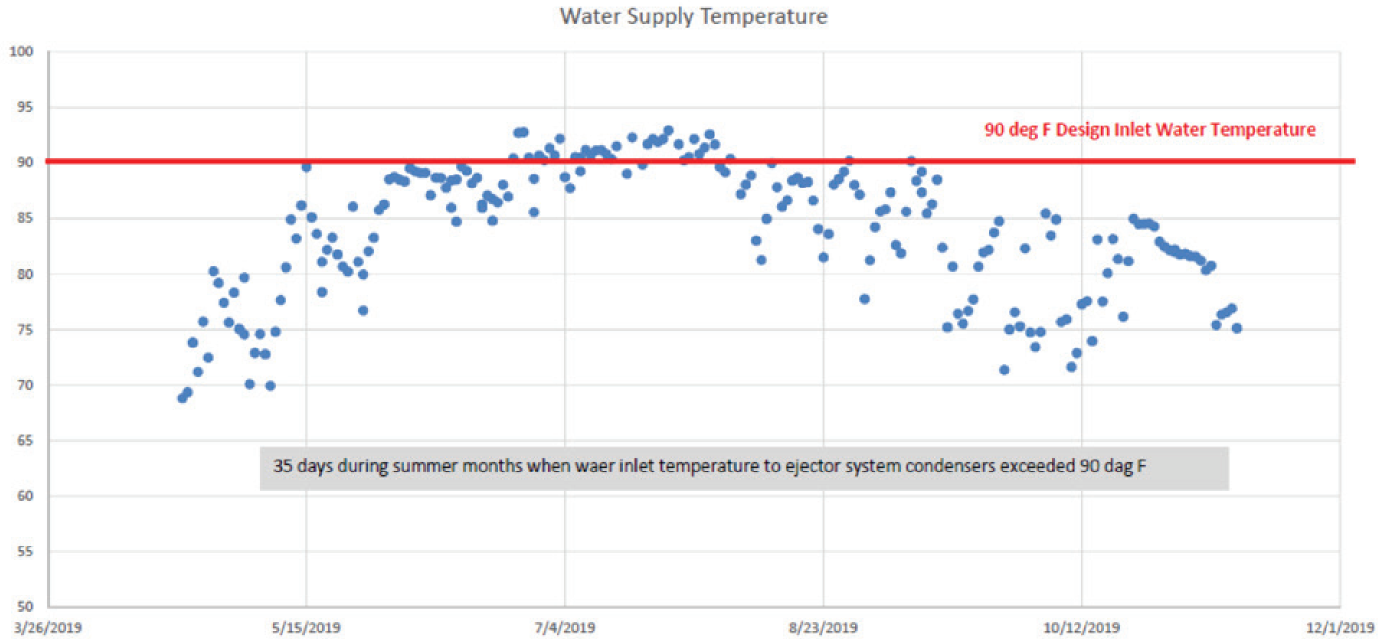

FIG. 4. The data historian for cooling water temperature.

the vacuum column pressure rises sharply: in the example presented here, the 20-mmHg design might jump to 30 mmHg. This results in significant yield and profit loss for the refiner.

Specifying a good design basis for cooling water inlet temperature is vital. The increase in summer temperatures, which are generally becoming hotter than historical norms, must be considered, as well as increased heat rejection on the cooling tower over time from pushing the refinery harder and adding future demand on the tower.

With rising inlet water temperature, the duty equation ($Q = U \times A \times MTD$) applies: UA is constant, and the heat rejection needed is constant; however, with elevated inlet water temperature, the MTD becomes lower. To compensate and ultimately reject the thermal duty, pressure rises in the condenser to establish a higher initial condensing temperature, thus increasing the MTD.

To mitigate this summer temperature challenge, the author's company was asked to develop an effective, low-cost solution. It is possible to replace the ejector motive steam nozzle and provide a new one that uses the same amount of motive steam but can produce a higher discharge pressure. The trade-off is a modest reduction in mass flowrate handling capacity. For example, it may be necessary to accept 1 mmHg–2 mmHg higher tower pressure, but pick up 4 mmHg–6 mmHg greater discharge capability, which often is adequate to overcome the summer temperature problem. There is no large CAPEX associated with this, as all major equipment and energy consumption remain unchanged. The only requirement is to replace the ejector motive steam nozzle with new designs capable of achieving greater discharge pressure.

Takeaways. Ejector system suppliers have extensive knowledge about real-world performance in the field; what can cause poor performance; how to design for minimal performance risk; how RFQ specifications are influencing OPEX, CAPEX and performance reliability; and what best practices ensure that refiners achieve their objectives. Involving an ejector system supplier early in the pre-FEED (front-end engineering design) and FEED phases is valuable. Once projects are in the EPC phase, it is difficult to make changes that could be deemed valuable to the end user. When an end user, licensor and ejector system supplier collaborate early in the scoping and specifying process, it leads to high satisfaction for the end user once the refinery is operating. When the parties do not thoroughly discuss the variables highlighted here and other performance influencers, it can lead to startup schedule delays, cost creep and under-performance of the refinery's operations. HP

LITERATURE CITED

1. Lines, J., "Coping with vacuum," *World Fertilizer,* May 2017.
2. Lines, J., "A mystery solved," *Hydrocarbon Engineering,* April 2017.
3. Lines, J., "Fouling in VDU ejector systems," *Petroleum Technology Quarterly,* January 2023.
4. Lines, J., "Operating vacuum distillation ejector systems," *Petroleum Technology Quarterly,* October 2016.

JIM LINES is an Engineer recently retired from Graham Corp., where he worked for 37 yr in various engineering and management positions. Lines has authored numerous articles related to heat transfer, vacuum process condensers and ejector systems..