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Jim Lines, Graham Corp., USA, highlights the importance of ejector systems in urea plants.

Ejector systems are critical to the final concentration of a urea solution. Regardless of the end product, whether produced by granulation or prilling, ejector systems establish evaporator pressures that permit the removal of water to concentrate the urea solution at temperatures sufficiently low enough to minimise biuret formation. There are several process technologies for urea production. Saipem/Snamprogetti, Maire Tecnimont/Stamicarbon, Toyo Engineering Corp., Casale and NIIK offer the most frequently used. For each process technology, the ejector systems are critical to plant throughput and product quality. While critical to the success and profitability of a urea plant, ejector systems are viewed as not generally well understood. The

thermodynamics of ejector performance is not widely known and the vacuum condensers within an ejector system cannot be designed with conventional heat exchanger software.

This article provides a deeper review of ejectors and vacuum condensers in urea concentration processes so that specifiers, evaluators, purchasers and users of this critical process equipment understand the salient considerations necessary to provide reliable plant performance.

Ejector systems for urea concentration processes

The concentration section of a urea plant will have different ejector system requirements depending on the
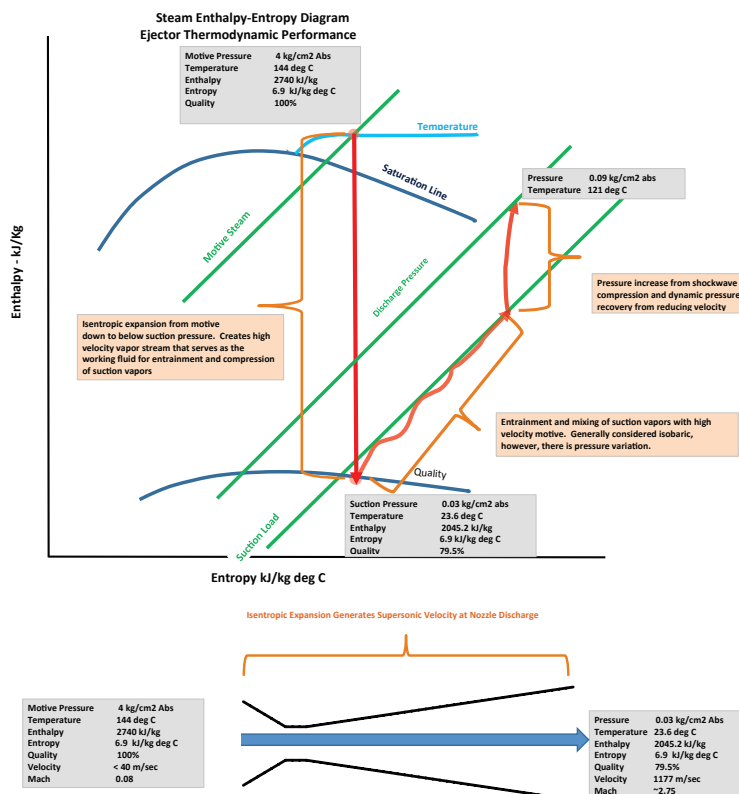
Figure 1. Ejector thermodynamics and expansion across motive steam nozzle.

type of final product. Urea produced via prilling or a Sandvik rotoform process requires urea melt with a water moisture content of less than 0.3 Wt%. A deeper vacuum (lower absolute pressure) is necessary to produce 99.7% urea melt. The initial concentration occurs at approximately 0.3 kg/cm² absolute pressure in the first stage evaporator. This concentration stage requires a vacuum precondenser followed by a single or two stage ejector system. The final concentration occurs in a second stage evaporator, operating at approximately 0.03 kg/cm² absolute. The final evaporator requires a three stage ejector system to achieve such a low operating pressure.

Granulation plants have just an initial concentration stage using a vacuum precondenser followed by a single or two stage ejector system. Granulation typically requires urea melt with water content of no greater than 4 – 5 Wt%.

Ejector systems are critical to both plant output quality and capacity because when an ejector system underperforms, the moisture content increases, which negatively affects the urea product quality and the plant capacity. The ejector system design for urea processes must address numerous challenges in order to achieve reliable performance. These challenges include:

- Vapour-liquid equilibrium involving chemical reactions.
- Reactants that combine exothermically.
- Solids formation in booster ejector and vacuum condensers.
- Lack of reliable software for vacuum condenser design.
- Strict ammonia emission restrictions from the vacuum system.

Steam jet ejectors

Steam jet ejectors fall into two classifications:
- Non-critical: where the discharge pressure is less than approximately 1.8 times the suction pressure.

- Critical: where the discharge pressure is more than approximately 1.8 times the suction pressure.

'Critical' is a term that refers to the presence of a shockwave in the diffuser throat that serves to boost the pressure, increasing the discharge pressure above that of the suction pressure. Another aspect of critical ejectors is once the pressure ratio exceeds 1.8:1, the flow is choked, and, depending on how great this pressure ratio becomes, the flow will pass through the diffuser throat no faster than the speed of sound. This establishes the mass flowrate that can pass through the ejector for a given suction pressure. The same relation holds for the steam nozzle throat. Within a urea plant concentration section, the ejectors will be critical as the compression ratio is typically greater than 1.8:1 for each ejector stage.

A brief overview of compressible flow theory is important. This is a complicated fluid flow and thermodynamics discussion that is best kept in simple terms (Table 1).

The Mach number is the velocity relative to sonic velocity (speed of sound). Mach = 1 when velocity is equal to sonic velocity. Sonic velocity varies for specific gases or combinations of gases and with temperature. Higher molecular weight gases will have lower sonic velocities and hotter gases will have greater sonic velocities.

Moreover, there is a critical pressure ratio above which the flow is choked and will not pass through a given cross sectional flow area any faster than sonic velocity. This critical pressure ratio varies based on the type of gas and its properties. The generalised formula for critical pressure ratio is shown in Equation 1, where:

- P motive is the pressure of motive steam.
- P suction is the suction pressure to ejector.
- P discharge is the ejector discharge pressure.
- γ is the ratio of fluid specific heats.

$$\frac{P\,motive}{P\,suction} = \left(\frac{\gamma+1}{2}\right)^{\gamma/_{\gamma-1}} \text{ or } \frac{P\,discharge}{P\,suction} = \left(\frac{\gamma+1}{2}\right)^{\gamma/_{\gamma-1}}$$

**Equation 1.** Critical pressure ratio formulae.

An ejector leverages the behaviour of compressible fluids to first develop supersonic velocity by expanding high-pressure motive across a converging-diverging nozzle down to a pressure that is below the suction pressure of the ejector. Ratio of the motive pressure to suction pressure is always many times greater than the critical pressure ratio resulting in sonic velocity at the throat of the converging-diverging nozzle and supersonic flow in the diverging section of the nozzle. Flow through the throat of the converging-diverging nozzle is sonic and the corresponding mass flowrate that will pass through the nozzle is approximated by Equation 2a or 2b, where:

- M is the pound per hour.
- Cd is the nozzle discharge coefficient, dimensionless.
- Throat is the nozzle throat diameter (in.).
- γ is the ratio of specific heats, dimensionless.
- P is the motive pressure, PSIA (Lb/in² absolute).
- T is the motive temperature (degrees Rankine).

Table 1. Unique characteristics of compressible flow		
Flow regime	**Decreasing cross sectional area**	**Increasing cross sectional area**
Subsonic, velocity ‹ Mach 1.0	Velocity increases, pressure decreases and density decreases.	Velocity decreases, pressure increases and density increases.
Supersonic, velocity › Mach 1.0	Velocity decreases, pressure increases and density increases.	Velocity increases, pressure decreases and density decreases.
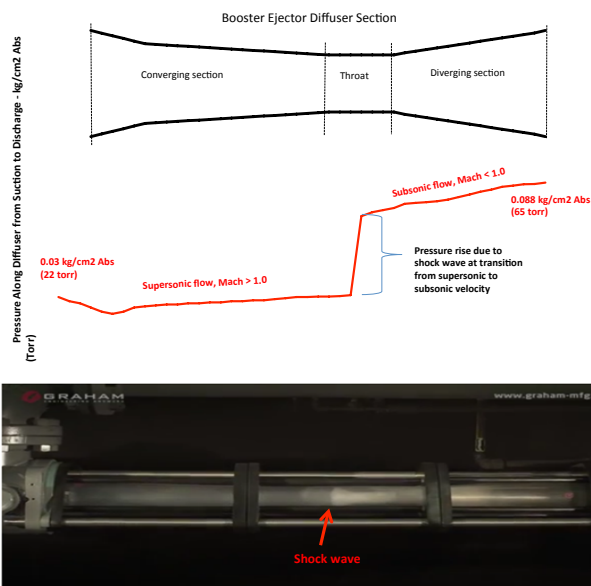
Figure 2. Pressure profile and shockwave that boosts pressure.
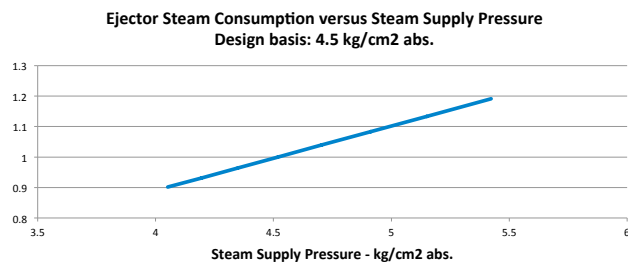
Figure 3. Typical steam supply pressure to ejectors and corresponding steam consumption.
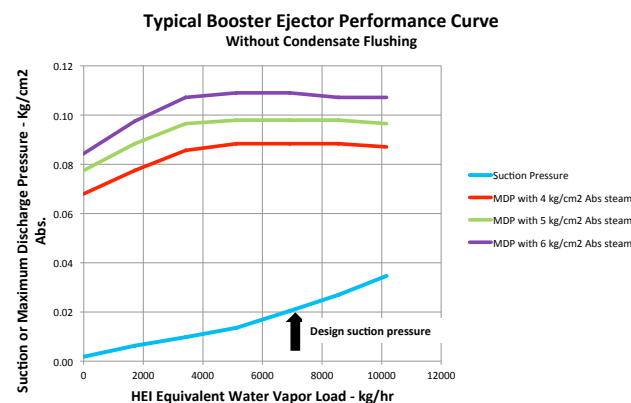
Figure 4. Typical booster ejector performance curve.

- MW is the molecular weight (Lbf/lbm-mole).
- ρ is the motive density (lb/ft^3).

$$M = 1336 * C_d * Throat^2 * (\gamma * P * \rho)^{1/2} * \left(\frac{2}{\gamma+1}\right)^{\left(\frac{1}{\gamma-1}+1/2\right)}$$

$$M = 409 * C_d * P * Throat^2 * \left(\frac{MW\gamma}{T}\right)^{1/2} * \left(\frac{2}{\gamma+1}\right)^{\left(\frac{\gamma}{\gamma-1}-1/2\right)}$$

Equation 2. Critical mass flowrate formulae where 2a (top) is when pressure and density are known and 2b (bottom) is when pressure and temperature are known.

Exchanger Institute formula for mass flowrate: M = 892.4 x Cd x Throat^2 x (P x ρ)$^{0.5}$.

By expanding to a pressure lower than the suction pressure, this will induce a flow of process vapours into the ejector where it is entrained by and mixes with the high-velocity motive steam. This mixture velocity is still supersonic. Figure 1 depicts thermodynamics of the ejector performance, including the isentropic expansion across a motive nozzle to develop supersonic flow, the quasi isobaric entrainment and mixing of suction vapours with motive steam and the compression across a shockwave and dynamic pressure recovery.

The flow across a motive steam converging-diverging nozzle, assuming no heat addition or removal and inviscid flow, follows the one-dimensional energy equation and the total (stagnation) enthalpy is constant throughout the nozzle. A simplification is that the velocity ahead of the nozzle is much less than the velocity exiting the nozzle, therefore, velocity of the motive for supply conditions can be neglected. Note in Figure 1 that the velocity ahead of the steam nozzle is less than 40 m/sec. and is 1177 m/sec. exiting the nozzle. The velocity at the discharge of the motive steam nozzle is proportional to the square root of the change in enthalpy for isentropic expansion.

It is now that sonic velocity comes into clearer view. Figure 1 shows that the velocity at the exit of the motive nozzle is approximately Mach 2.75 or 1177 m/sec. Consider a case where an ejector is using 3 kg/hr of motive steam for each kg/hr of steam (3:1 motive to load ratio). In such a case, the mixture of motive and load remains supersonic at Mach 2.06. As the mixture moves along a second converging-diverging conduit (the diffuser), a shockwave is established when the ratio of downstream pressure and the suction pressure exceeds the critical pressure ratio. This is where the importance of sonic velocity comes to light. An acoustic wave travelling in the opposite direction of the fluid flow propagates upstream at sonic velocity, which is slower than the supersonic velocity of the fluid. An acoustic wave, however, in subsonic flow permits the fluid flow field to adjust to contractions or obstructions in the flow path because the acoustic pressure wave informs the flow fluid of the pending obstruction or change in direction. When flowing at supersonic velocities, the flow fluid cannot adjust to such contractions or obstructions because the flow is travelling faster than the informative acoustic pressure wave, resulting in a shockwave that raises the pressure and reduces the volume-permitting flow to pass through the contraction at sonic velocity. Figure 2 illustrates the pressure profile across a converging-diverging diffuser with a pressure rise from a shockwave in the throat section. A digital image of an actual

shockwave inside a glass diffuser is shown directly below the pressure profile.

Steam supply conditions

An essential parameter for proper ejector performance is motive steam supply conditions. In urea plants, the steam provided as motive is available at low pressure and is saturated. The steam supply pressure will vary with plant operating rate. Proper ejector performance, for any ejector stage, is correlated directly to adequate motive steam supply condition. Best practice is to establish ejector performance based on lowest anticipated steam supply pressure. Figure 3 illustrates how the steam consumption for an ejector will vary based on the steam supply pressure. The lower the supply pressure, the less motive steam an ejector will consume and the less energy will be available for entrainment and compression.

Figure 3 illustrates how steam consumption by an ejector varies with the supply steam pressure. The steam pressure below the design basis for an ejector is harmful to the performance of that ejector and, therefore, harmful to the evaporator pressure and ultimately the urea moisture content. For example, if the plant design specifications detailed that the motive steam to the ejectors was 4.5 kg/cm² Abs, but in operation the actual supply conditions were 4.0 kg/cm² Abs, this would result in a 10% decrease in steam consumed by an ejector or 10% less working fluid available to perform the necessary entrainment and compression. An ejector will then break operation, resulting in suction pressure increasing dramatically. Also, the lower pressure steam in this operating range will have lower enthalpy, therefore, the velocity exiting the nozzle is lower, further lowering the energy available to perform the work within the ejector.

The ejector systems can become the limiting factor for facility throughput due to insufficient motive steam pressure available for the work the ejector system must perform.

Ejector performance curve

Each ejector has a unique performance curve that defines the suction pressure that the ejector will maintain as a function of the suction load when supplied with the design motive steam pressure and when the discharge pressure does not exceed its maximum capability. Figure 4 provides a typical booster ejector performance curve. It is important to note that the greater motive pressure does not generally affect suction capacity, however, it will improve discharge pressure capability. Importantly, an ejector does not create discharge pressure. It is the condenser downstream that establishes discharge pressure. For example, if an ejector can operate with 0.11 kg/cm² Abs discharge, the condenser that is downstream of the ejector establishes 0.09 kg/cm² Abs discharge pressure. The ejector discharges to the condenser pressure. It has the capability and the energy to produce greater compression, however, it will not do so.

Design practice for reliable, problem-free performance requires the ejector design to be based upon the lowest motive steam pressure that is anticipated.
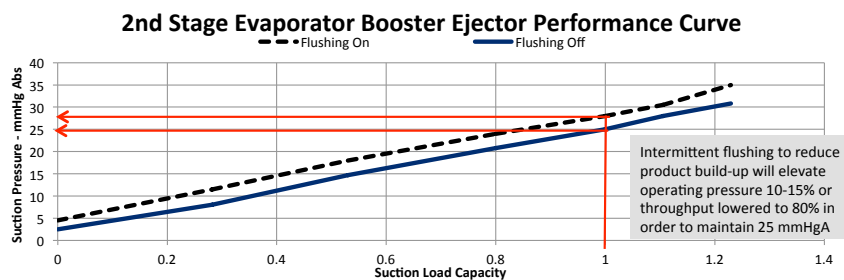

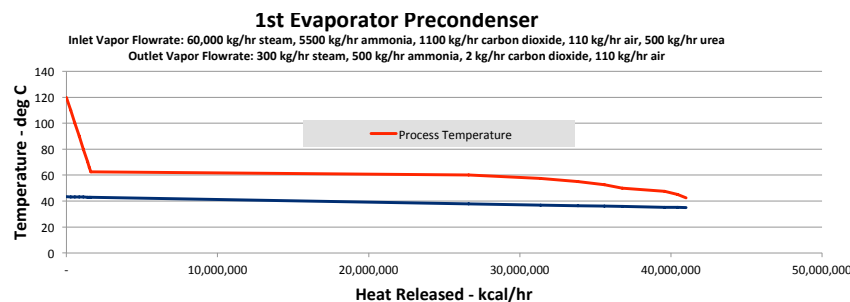**Figure 5.** Ejector with steam tracing and flushing connections.

Figure 6. First evaporator precondenser heat release curve.

## Variables that affect ejector performance

There are a few key variables that impact ejector performance. It is important to understand how these variables influence the formation and position of the shockwave. Videography illustrating shockwave position readily conveys how performance deteriorates when discharge pressure exceeds the maximum capability of an ejector; motive supply conditions result in less energy provided for compression; there is mechanical damage to or blockage within the ejector; or, the load conditions vary from design. Figure 8 notes varied shockwave formations. The direction of flow is from left to right.

If the motive supply pressure was 5.5 kg/cm² Abs, the shockwave would move to the right. In such a case, no performance would be lost. The only consideration is that more energy would be consumed by the ejector than would be necessary for the work.

Similar to the depictions above, the discharge pressure moves the shockwave upstream or downstream. High discharge that is above the MDP of an ejector forces the shockwave upstream out of the throat section, the shockwave dissipates where compression is lost and the ejector is in a broken condition where the evaporator pressure is elevated and unstable.

It is important to note that when conditions cause a shockwave in an ejector to collapse (in industry parlance this is referred to as a 'break' or that the 'ejector is broken'), the flow will backstream. Backstreaming occurs because the pressure downstream of a shockwave is greater than the upstream pressure. The shockwave serves as a barrier maintaining lower pressure upstream and greater pressure downstream of the shockwave. When a shockwave collapses, the high pressure rushes back towards the low upstream pressure in the process vessel. If uncorrected, surging can occur where the process pressure oscillates up and down as the shock reestablishes and breaks cyclically. Moreover, the system may simply operate in a broken condition where evaporator pressure is elevated above design.
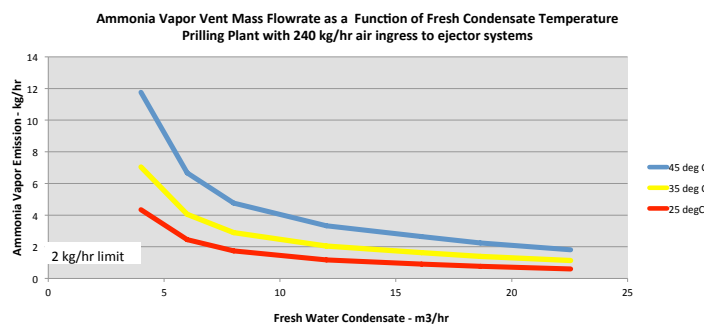
Figure 7. Ammonia vent flowrate versus fresh condensate flowrate and temperature.
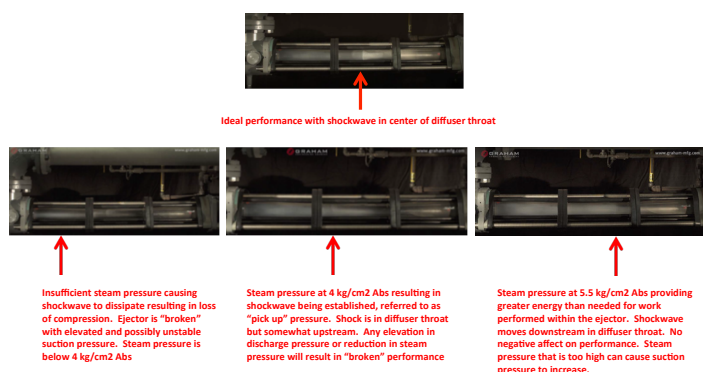
Figure 8. Shockwave formation and position versus steam pressure.

Booster ejector flushing

The evaporator overhead load contains urea mist along with water vapour, ammonia, carbon dioxide and air. Solids can form and build up within a booster ejector. This results in a blockage in the cross-sectional area and disruption of the flow within the booster ejector. Performance will degrade if not addressed, resulting in elevated or surging evaporator pressure when the ejector compression breaks down.

Booster ejectors are often equipped with steam tracing to maintain warm temperatures to reduce precipitation of solids. Also booster ejectors may have condensate flushing connections that introduce flushing liquid that is low concentration aqueous ammonia for the removal of product buildup within the ejector. Flushing operations are never continuous and should be done intermittently – daily or once per shift. Continuous or intermittent flushing will elevate the second stage evaporator pressure, due to flashing of the flushing solution and reduction of the vapour entrainment capability of the motive steam. Figure 5 provides a comparison of booster ejector performance without flushing and when flushing condensate is introduced. Condensate flushing increases operating pressure by approximately 10%. As noted previously, this higher operating pressure for extended durations will increase the urea melt moisture content to unacceptable levels. A general guideline for flushing the booster ejector is once per shift for 10 min.

Vacuum condensers

The condensers in the vacuum concentration section of a urea plant, within an ejector system, are particularly difficult to design. There is no commercial software available for modelling thermal and chemical reactions and hydraulic performance within these condensers. Only vacuum technology and ejector system

suppliers with proven experience should be considered for this service.

The process load to these condensers includes water vapour, ammonia, carbon dioxide, urea mist and air. Calculating the vapour-liquid equilibrium must consider chemical reactions that are exothermic and release considerable heat. The heat and material balance across these vacuum condensers can be markedly different from the actual performance if the vapour-liquid equilibrium and condenser design are not properly considered. Figure 6 provides a heat release curve for a typical precondenser receiving process load from a first stage evaporator.

Ammonia solubility in steam condensate is well documented, however, if a second chemical reaction of gaseous carbon dioxide and aqueous ammonia is not properly considered or the condenser is not configured correctly to permit chemical and phase equilibria, then the mass and volumetric flowrate exiting that condenser will be vastly greater. Referring to Figure 6, the gas/vapour exiting the condenser is 912 kg/hr and 1.5 m^3/sec for mass and volumetric flows, respectively. If the secondary reaction is ignored or the condenser internal configuration does not maintain vapours and condensate in intimate contact, then the gas/vapour exiting the condenser can be as high as 5600 kg/hr or 6.5 m^3/hr. If an ejector downstream of this condenser is not designed for a higher flowrate, the evaporator pressure will rise along with the urea moisture content.

As with the booster ejector, the primary vacuum condensers can have solid product formation on the tubes that acts as fouling, thereby lowering the condensing efficiency. The primary condensers should have flushing nozzles that operate intermittently, perhaps once per shift for 10 – 15 min., to remove any solids that form.

Ammonia emission from ejector system

A design variable that continues to receive careful scrutiny is the mass flowrate of the ammonia vapour vented from the ejector system. A critical design variable is limiting the ammonia emission in accordance with international emission standards. As plant capacity moves progresses towards mega urea plants of 5000 tpd or greater capacity, this design variable must be addressed differently. Single train capacity has increased from 1200 tpd to approaching 4000 tpd, with next generation plants targeting 5000 tpd, and maintaining ammonia vent levels within international standard cannot be accomplished without additional measures.

Larger plant capacities translate into facilities with greater vessel volumes, larger diameter flanges and greater potential for air ingress. Ammonia vent mass flowrate is directly proportional to air mass flowrate. 200 kg/hr air ingress handled by an ejector system will have twice the ammonia vent level as a system with 100 kg/hr air ingress, assuming all else is equal. There are four variables that influence the mass flowrate of ammonia vented from the ejector system scrubber:

- Mass flowrate of air.
- Amount of fresh condensate available to absorb ammonia within the vent scrubber.
- Temperature of the fresh condensate.
- Operating pressure of the scrubber.

Process licensors specify the amount of air ingress, therefore, it is unlikely that a design basis would be permitted with lower air ingress. The operating pressure of the scrubber is slightly above ambient pressure as the last ejector stage will compress to 1.05 – 1.15 kg/cm^2 Abs. A higher pressure will drive more ammonia into the solution, however, the energy consumption by the last stage ejector will limit this. If the amount of fresh condensate used to absorb the ammonia is flexible, then one can design a scrubber based on unrestricted condensate in order to reach the emission levels. However, processes do not generally have an abundance of fresh condensate. Typically, 5 – 10 m^3/hr of fresh condensate is available for scrubber absorption liquid.

The final option is chilling the fresh condensate. Colder fresh condensate will permit greater absorption of ammonia into the solution thereby lowering the vent stream to 2 kg/hr or less, for example. Typically, fresh condensate is available at 45°C for scrubbing ammonia from the vent stream.

Most facilities must keep fresh condensate usage to under 10 000 kg/hr (10 m^3/hr). In order to meet the ammonia emission restrictions, the condensate must be cooled. Prilling plant booster ejectors have a suction pressure of nominally 0.03 kg/cm^2 Abs. By introducing the 45°C fresh condensate into a flash vessel that is connected to the booster ejector suction, the condensate can flash cool to the saturation temperature, corresponding to 0.03 kg/cm^2 absolute, which is approximately 24°C. The flash steam load to the booster ejector is minimal in comparison to the process load, thus system utilities (steam consumption) are not materially altered with this concept. 24°C fresh condensate can absorb more ammonia gases into solution than 45°C fresh condensate. Figure 7 indicates a large amount of fresh condensate is required (20000 kg/hr or 20 m^3/hr) when the temperature is 45°C. Fresh condensate requirements are lowered to 6000 – 7000 kg/hr (6 – 7 m^3/hr) when fresh condensate is flash cooled to 24°C before entering the vent scrubber. There is added cost for a flash vessel, associated piping, controls and low NPSH pump, however, this provides an elegant means of achieving ammonia vent limit of 2 kg/hr, for example, notwithstanding plant capacity or unduly altering ejector system energy consumption.

Summary

Urea plant ejector systems operate in a demanding service and the required design cannot be met with commercially available software. Proprietary know-how and proven experience are required by an ejector system supplier to ensure trouble-free performance. The ejectors themselves, while static equipment, are not well understood nor are the variables that affect performance. An ejector system supplier should always be involved early during the plant concept evaluation to make certain essential project success variables are vetted. The ejector systems can limit a plant's throughput and profitability, therefore, engaging with an ejector system supplier early in the projection development stage to ensure profit objectives are attained. **WF**