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Troubleshooting vacuum operation of an inter-after condenser unit in an ethylene plant

A system of compressors powered by surface condensing steam turbines is inherent in the operation of a typical ethane cracker unit. These turbines run by extracting work from high-pressure steam, while a surface condenser condenses the turbine's exhaust to both maximize compressor horsepower and recover valuable condensate. In the surface condenser, a vacuum is created by the condensing steam. This vacuum is maintained by exhausting non-condensable load from the surface condenser via steam ejectors and inter-after condensing units. Non-condensable gases, which must be purged from the system, can originate from a number of sources: carbon dioxide (CO_2) entrained in the steam and air leaking through shaft seals into the low-pressure area of the surface condenser are two examples.

The technology for exhausting non-condensable gases to sustain vacuum has been in use for more than a century. While the systems tend to have a simple layout and are not overly complicated in terms of hardware, troubleshooting the loss of vacuum or underperformance of these units is not straightforward. A systematic approach is required to identify and rectify any issues that contribute to deteriorated performance. A process system operating with unstable and/or low vacuum directly affects turbine performance, a turbine's steam consumption and overall compressor efficiency.

In an ethane cracking unit, a troubleshooting study was undertaken to investigate an inter-after condensing unit and vacuum instability in the surface condenser. The troubleshooting study, summarized here, consisted of field observations, equipment review, trial runs and data collection. The goal of the study was to:

- Evaluate the unit's present operating conditions
- Understand its operating constraints
- Implement correction to sustain stable vacuum operation.

While ethylene plants are typically not known for the wide utilization and operation of their vacuum systems, troubleshooting and lessons learned from steam ejectors and condensing units span numerous process industries.

System overview. In the subject unit, a compressor turbine isentropically expanded high-pressure steam to a significantly lower pressure level. The expanded steam enters through the top of the condenser, as shown in **FIG. 1.** Once it enters the shell, steam spreads horizontally along the length of the shell while moving downward over the tube bundle. In this surface condenser, steam condenses on the shell side of the exchanger while heating the circulating cooling water inside the tubes. The condensed vapor falls to the bottom by gravity into the hotwell to be pumped out as valuable condensate. The surface condenser's vacuum created by the condensing steam is limited by the vapor pressure of the water at the cooling water inlet temperature, plus some approach temperature. Therefore, the cooling water temperature determines the best possible operating pressure—that is, the minimum achievable vacuum pressure—of the condenser.

For example, with a cooling water inlet temperature of 90°F, a condensing temperature of 100°F–110°F after approach is possible in the unit. This would correspond to a minimum possible operating pressure of 66 mmHg (absolute) based on water vapor pressure. Cooling water flow also affects the LMTD of the surface condenser. A lower cooling water flowrate will result in decreased LMTD across the exchanger. The unit's surface condenser is a fixed tube sheet split exchanger with cooling water on the tube side (**FIG. 1**). The shell is split into two independent compartments that permit the isolation of one side and periodic cleaning of cooling water tubes. The arrangement allows for continuous operation of the surface condenser, albeit at reduced capacity, during maintenance cleaning times.

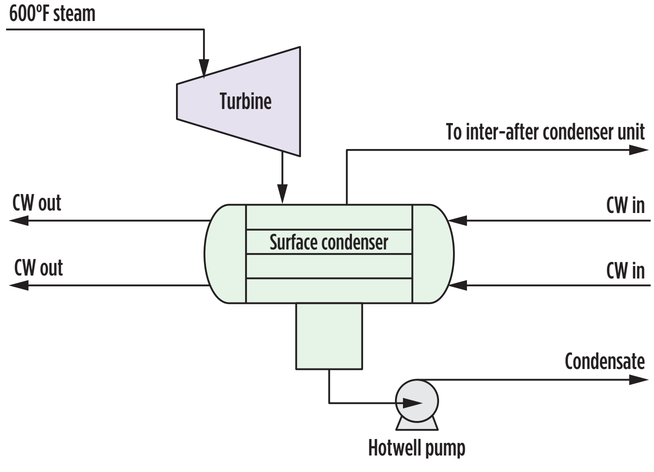

FIG. 1. Surface condenser flow diagram.

Marginally lower pressure in the surface condenser reduces steam consumption and improves the turbine's efficiency. The condensing steam creates a vacuum inside the surface condenser. As the system operates under vacuum, any non-condensable gases from steam or an external leak source will accumulate in the low-pressure area. These gases will quickly increase the operating pressure of the condenser, as the non-condensable gases that cannot be evacuated blank the tubes and reduce the capacity of the condenser. The gases must be vented from the condenser to help maintain vacuum. In the plant, the main devices used to vent the non-condensable gases are two-stage steam jet ejectors coupled with horizontally installed inter-after condensing units (**FIG. 2**).

The design of ejectors E-1, E-2 and E-3 in **FIG. 2** includes five main parts each: a motive steam nozzle, suction chamber, inlet diffuser, throat section and outlet diffuser.[1] Across the motive steam nozzle, 200 psig steam enthalpy is converted into kinetic velocity. Similar to the steam enthalpy conversion to velocity across the turbine's inlet nozzle, this is also an isentropic process. At the nozzle's discharge, the expanded steam creates a low pressure that entrains the process load into the high-velocity steam. Steam and non-condensable gases mix as they enter the inlet diffuser, where the velocity of the process flow decreases as it enters the diffuser throat. The throat section is the transition piece between the converging supersonic inlet diffuser and the diverging subsonic flow outlet diffuser. As the process flow moves through the throat, it transitions to a subsonic flow, creating a supersonic shockwave. In the outlet diffuser, the flow velocity is reduced further and, essentially, is converted back into pressure.

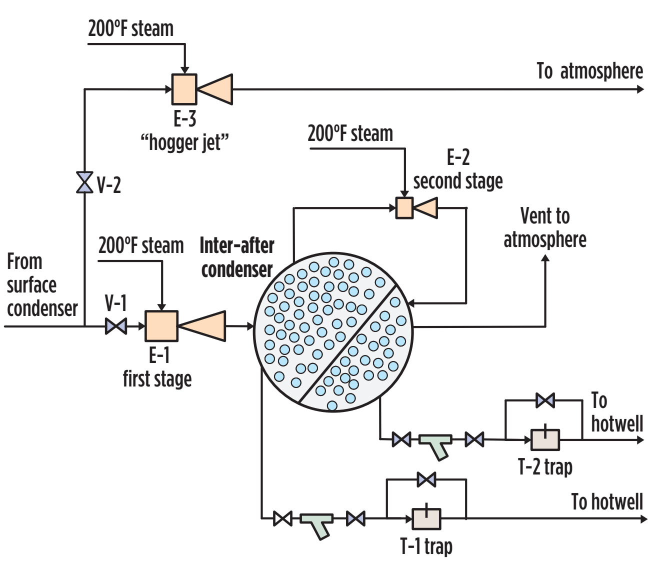

FIG. 2. Inter-after condenser unit flow diagram.

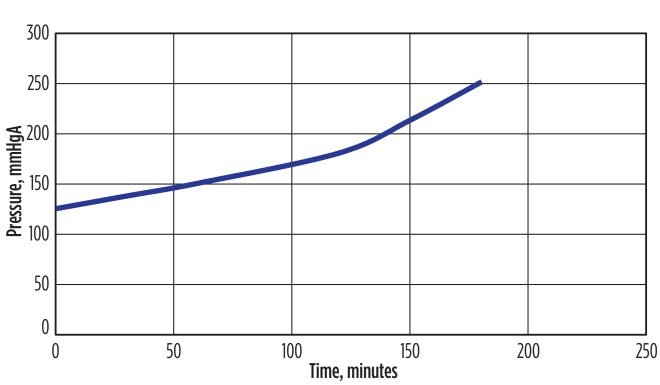

FIG. 3. Vacuum measured above the water level in the surface condenser's hotwell.

In summary, the ejector uses kinetic energy in the form of a high-velocity stream to push back on the pressure at the outlet of the ejector. The outlet pressure that an ejector is subjected to is the result of the downstream equipment and is not set by the ejector. The ejector E-3 ("hogger jet") in **FIG. 2** is an auxiliary ejector used primarily to evacuate the condenser and turbine casing during startup. In the case of a performance issue with the main holding ejectors, the auxiliary ejector can be used as a temporary backup. Each component of the ejector is critical for proper vacuum operation, and seemingly negligible changes in geometry can have a significant impact on system performance.

The compressed steam and non-condensable gases from ejector E-1 enter the fixed-tube sheet inter-condenser, where a significant portion of that steam is condensed against cooling water flowing on the tube side. The non-condensable gases are further cooled, reducing the volume load to the second-stage ejector E-2. The condenser unit is separated by a welded partition that spans the entire condenser and creates two areas called—quite logically—the inter-condenser and the after-condenser. Ejector E-2 maintains the vacuum of the inter-condenser while exhausting non-condensables into the after-condenser. The ejector compression ratio is the ratio of absolute discharge pressure to suction pressure. Similar to the vacuum in the surface condenser, an ejector's individual compression ratio is mostly load related. The first-stage ejector E-1 pipes directly to the shell side of the surface condenser. If the air load is below design, the air removal system is in good condition, and steam and cooling water meet design conditions, the ejector E-1's compression ratio is set primarily by the surface condenser and the pressure at which the surface condenser wants to operate.

The after-condenser does not contribute to vacuum and operates at atmospheric pressure. It recovers additional steam condensate while exhausting non-condensable via atmospheric vent without the need for a silencer. Typically, condensate from the inter-condenser must be drained back to the hotwell, as the inter-condenser is always operating under vacuum. This may be accomplished by a condensate trap or, where elevation allows, by a highly reliable water loop seal. In the subject plant, condensate from the after-condenser was drained back to the hotwell through a float trap. The after-condenser can also be set up to drain condensate to an open drain without a trap.

Unit operation. Prior to the troubleshooting study, when ejector E-3 ("hogger jet") was in operation, vacuum in the hotwell would operate at 125 mmHg (absolute). As shown in **FIG. 3,** switching the unit to inter-after condenser operation would cause slow deterioration in vacuum.

System performance deterioration can occur in two areas within this unit: the surface condenser and the inter-after condenser unit, including the ejectors. The surface condenser unit operates with a cooling water inlet temperature of 87°F–91°F and a cooling water temperature difference of 10°F–17°F between the inlet and the outlet vs. the design temperature of 17.5°F. The measured hotwell temperature averages 120°F. The corresponding water vapor pressure to this temperature is an indication of the theoretical possible vacuum pressure inside the surface condenser, given that the non-condensable load is below design. These measured temperatures are well within design values of the surface condenser. However, if the cooling water temperature were to rise, the surface condenser's available LMTD would decrease. The system vacuum would be limited by the exhaust turbine temperature. The temperature of 120°F in the hotwell corresponds to water vapor pressure of 87.4 mmHgA. This temperature-pressure correspondence is close to the initial pressure of 90.9 mmHgA at the inlet to the inter-after condenser unit, measured with a pressure gauge. Rather than rely on installed distributed control system (DCS) instrumentation (pressure transmitter), it was important to perform a pressure survey between the hotwell and inlet to the inter-after condenser unit to ensure that no unnecessary significant pressure drop existed between the two units. The pressure survey between the surface condenser and the inter-after condenser unit revealed only a small pressure drop of about 7.6 mmHg, measured with the same pressure gauge.

An increased vapor flow and higher temperature into the inter-after condenser unit can indicate an air baffle leak in the surface condenser. To force steam and non-condensable vapors to go through the entire bundle, an air baffle is used in large surface condensers. The air baffle runs the entire length of the surface condenser and is welded to the shell to avoid steam bypassing the surface condenser and flowing directly into the inter-after condenser unit. Throughout the data collection period, an average temperature range of 103°F–115°F was recorded for vapor flowing into the air removal unit. A measured temperature of 115°F corresponded to a water vapor pressure of 76 mmHgA, which was well below the measured and deteriorating system vacuum (**FIG. 3**) when the surface condenser was operating with a full load.

The cooling water conditions operating below design suggests that load temperature from the surface condenser is not a limiting factor. Therefore, the factors that limit performance can either originate from high non-condensable loads (which can be checked at the inter-after condenser unit's vent, as discussed later) or from a bottleneck within the inter-after condenser unit itself. When running steam ejectors, it is important to look for surging or loud popping sounds coming from the steam ejectors. While many causes for ejector surging can exist, surging or popping sounds can be an indication of a vacuum break. Ejector surging can often lead to very hot vapor load temperatures flowing into the inter-after condenser, as steam will want to flow backwards—for example, from ejector E-1 toward the surface condenser—while the ejector recovers its shockwave. The resulting hot load temperature can lead to a misleading assessment of where troubleshooting attention should be focused.

To troubleshoot system performance, a rigorous raw data collection plan was developed. The following data collection points were identified and prepared prior to the initiation of the trial:

- Suction (inlet pipe or ejector suction chamber) and discharge pressure on each of the two ejectors, E-1 and E-2
- Pipe surface temperature on cooling water inlet and outlet
- Cooling water inlet and outlet pressure for the condenser
- Exhaust and suction temperature on the ejector jet
- Vapor flow from the after-condenser outlet atmospheric vent
- Motive steam pressure and temperature
- Pipe surface condensate drain temperatures.

Rather than using compound gauges to ensure both precision and accuracy, self-compensating, absolute-vacuum pressure gauges with isolation root valves and pigtails were installed at the two ejectors and leak-tested. It was also recommended to use a single electronic absolute pressure gauge for consistent readings throughout the trials.

Inter-after condenser no-load test. While the surface condenser was in operation running on the "hogger jet" ejector E-3, a no-load test was performed on the inter-after condenser unit, which was isolated by closing a load valve at the suction inlet to the first ejector followed by opening motive steam to both ejectors. Without load from the surface condenser, once steam was introduced the ejector exhausted from the first stage suction chamber to the minimum absolute pressure it was capable of producing. Typically, if the test showed that the ejector was operating at close to its design shutoff pressure before the introduction of load from the surface condenser, acceptable performance could be assumed along the full ejector's performance curve.

The system reached a vacuum of 99.5 mmHgA compared to the expected design no-load vacuum of approximately 10 mmHgA. While a small amount of condenser leakage through the isolation valve might be expected, this dramatic performance deterioration was not rational. The compression ratio of the primary ejector E-1 was 1.9:1, and the ratio of the secondary ejector E-2 was 4.0:1. This indicates that, in addition to the inter-after condenser unit being unable to sustain stable vacuum, it was also unable to operate at no-load design vacuum. Based on this no-load test compression ratio result, system troubleshooting could now be focused exclusively on the ejectors and inter-after condenser unit.

Steam supply. Both higher and lower steam supply pressures (vs. specified design) at 0%–20% will have detrimental effects on the stability of any or all ejectors. For example, motive steam pressures that exceed design can create choked flow and decrease the ejector's capacity. On the other hand, with lower-than-design steam pressure supply, the ejector will not receive sufficient energy to compress vent load from the surface condenser. During the trials, steam pressure was measured at 210 psig, approximately 10% above design. The pressure was deemed as acceptable and not detrimental to performance. Following the trial, the steam pressure to the inter-after condenser was lowered to design specification to save on steam consumption and condensate recovery.

For steam quality, a source of dry steam close to saturation is preferable for the ejectors. Moisture in the steam will erode the internals as well as reduce suction load capacity for the ejector. To ensure dry steam, the insulated inlet line in the subject unit was routed from the top of steam supply header. If such a configuration is not available, or if moisture is known to be present in the steam, a separator and floater trap can be used to improve the steam's quality. Measuring the temperature on the discharge of the ejector or inspecting the internals can determine if wet steam is present. The temperature of the vapor at the discharge of the ejector will be much cooler if a high degree of liquid is present.[2] The steam outlet temperature was measured on the "hogger jet" by inserting a thermometer into the exhaust stream. The steam temperature measured at the outlet of the "hogger jet" was 255°F–260°F, an expected temperature range for the exhaust steam to the atmosphere. Erosion of nozzle internals and diffusers, as mentioned, is evidence of wet steam presence and requires internal inspection of the ejector, to be performed at a later date.

Condensate recovery. It is important to ensure an unobstructed pathway for condensate drainage. If a strainer, separator and/or trap are present, they should be inspected and cleaned after the lines are blown clear. During initial startup, it is easy for a strainer to become plugged from deposits or rust within a stagnated unit. The subject unit contained two traps: one on the drain from the inter-condenser, and one on the after-condenser. Low flow of condensate to the traps due to a plugged 1-in. strainer or undersized traps will cause an eventual backup of water into the condenser.

A flooded condenser will lower the available surface area for additional steam condensation from the ejector and cause unstable or deteriorated vacuum. Free water exhausting from the vent is an obvious sign that condensate has backed up and that the after-condenser has flooded. A flooded condenser will produce subcooled condensate. During the trials, the traps were opened and inspected for any mechanical issues that might inhibit function, as well. The traps on the unit were floater/thermostatic traps. The thermostatic part of a floater trap is a vent spring valve that, based on measured temperature, will vent air and non-condensable gases. The thermostatic vent in a normal floater trap application is designed to open at a few degrees below saturation. The thermostatic element contracts, pulling the valve head off the valve seat. The trap opens and discharges air and condensate, which is undesirable, particularly if the trap is connected to the hotwell and the condensate is subcooled. These traps are typically used to discharge condensate in modulating conditions, such as traditional heat exchangers, air-handling coils and typical steam header stations. In a vacuum system, it is recommended to use a simple floater trap (no thermostatic element) or a loop seal. As shown in **FIG. 4,** the thermostatic valve part of the condensate trap was removed and blocked with an NPT plug during the troubleshooting trials.

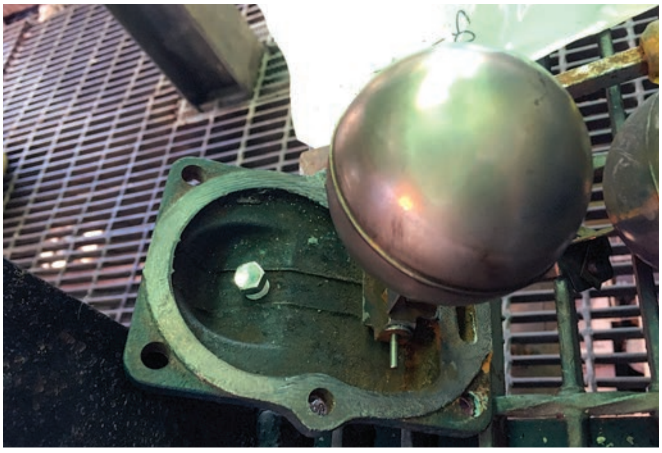

FIG. 4. Floater trap with thermostatic vent plugged.

It is also recommended that the condensate drain piping from the inter-condenser not tie in to the after-condenser piping at any point before the hotwell. This avoids condensate backup problems in the drain lines, i.e., backing condensate up from one condenser into the other as they operate at different pressures. The difference in pressure may also allow air load from the after-condenser to recycle back into the inter-condenser. In the subject unit, both lines drained condensate independently.

Ejector inspection. The primary and secondary ejector steam chests and outlet diffusers were disconnected and inspected. The secondary ejector diffuser was not smooth and showed signs of some erosion, corrosion and fouling. If the motive nozzle and diffuser throat have increased by about 7% (area) or 3% (diameter) vs. the initial installation, replacement of the ejector is recommended. When inspecting the steam chest, a typical mechanical issue is a loose motive steam nozzle. During inspection, a slightly loose motive steam nozzle was found on the second-stage ejector E-2. A loose nozzle will easily leak steam across its threads and bypass discharge of the nozzle. The steam that bypasses the nozzles becomes a supplemental and unnecessary load in the air chamber of the ejector and decreases operational vacuum and stability. The nozzle should be inspected for signs of wear and corrosion. To fix a loose nozzle, the threads of the existing steam nozzle can be temporarily wrapped with a sealing device like Teflon tape for the duration of trials. A long-term fix is to seal weld the nozzle to the spacer of the motive chest, or to replace the motive chest entirely.

Unit leaks. Vacuum systems are vulnerable to vacuum pressure deterioration when the non-condensable load to a vacuum system increases above its specified design. In addition to expected non-condensable loads coming from condensing steam or condensate traps that load the surface condenser through thermostatic vents, two possible areas were also identified where the extraneous introduction of non-condensables might occur. The first was ambient air, as both the exhaust of the steam turbine and the condenser operate under significant vacuum.

The potential for air leakage into the system is always prevalent, and this leakage can occur through flanges, threaded piping, valve pickings or from expanding seal/joint failure on the turbine itself. One method of testing for external leaks is with ordinary shaving cream. When applied to all potential leak piping and joints, shaving cream will be sucked into any unsealed openings. The unit was also checked for cooling water tube leaks. A cooling water tube leak into the shell of the inter-after condenser unit can overwhelm traps, back-flood the condenser and plug condensate strainers over time. Channel head covers at each side of the condenser unit were removed, and steam was introduced to the shell side to check for tube leaks. A secondary option can be to perform a hydrotest.

Another leak source can occur from the process itself. When the inter-after condenser is operating, the ejector E-3 "hogger jet" is offline and is isolated by valve V-2 in **FIG. 2.** If the isolation valve leaks, it could easily suck in air from outside and overload the secondary ejector. The unit also had bypass valves around the floater traps, which were also identified as possible leak sources. A vacuum deterioration occurred when block valves on the condensate drain were opened to the hotwell prior to introducing motive steam to the inter-after condenser. In response, the bypass valves around the traps were removed and blinded during the troubleshooting. Condensate trap bypass valves can only be opened temporarily to drain the condensate while the trap is being serviced. Without the trap in service, the surface condenser pressure will be affected.

Ejectors can also get overloaded from an internal partition failure.[3,4] The inter-condenser is separated from the after-condenser by a partition plate welded to the shell. The secondary ejector exhausts steam into the after-condenser, which operates at higher pressure (atmospheric) than the inter-condenser, which operates under vacuum. A partition failure resulting in a leak would expose the higher pressure after-condenser to the inter-condenser operating under vacuum, leading to operational upset on the first ejector. To offline test for any internal partition leak, the system was set up in the following way:

- Outlet of the second stage ejector was routed to the atmosphere
- Inlet to the after-condenser and its atmospheric vent were blinded and blocked off
- Both shell sides of the condenser were drained and condensate lines were isolated
- A vacuum pressure gauge was connected to the after-condenser
- The unit was isolated from the surface condenser. The "hogger jet" E-3 remained online to sustain the vacuum in the surface condenser required for plant normal operations.

When motive steam was introduced to the second ejector only, the isolated and blinded after-condenser remained at atmospheric pressure while the inter-condenser held at a stable vacuum. Had the internal partition been compromised, the after-condenser pressure would fall to vacuum. Since the pressure in the after-condenser remained near-atmospheric, the test showed that the internal division plate had not failed.

To qualitatively determine excessive total leakage throughout the system, the non-condensable vapor flow should be measured. After the modifications mentioned above were completed, a quantitative test was used to actually measure the flow of saturated vapor and non-condensables exiting the unit at the after-condenser atmospheric exhaust vent. Shown in **FIG. 5,** a 4-gal grocery trash bag was attached to the exhaust vent with duct tape and rubber bands. A stopwatch was utilized to measure the fill time of the bag, giving an estimate of total volumetric vapor flow through the 1.5-in. vent.

The flow was calculated to be about 15 sf^3/hr at 95°F. Based on the measured temperature, the vapor was about 5.5% (vol) saturated water. The flowrate for the non-condensable gases was calculated to be about 1 lb/hr, well below the maximum load design value of 33 lb/hr for the inter-after condenser. For future periodic performance checks, it was recommended to install a permanent air leakage meter if one is not already part of the unit.

Unit fouling. Within the subject plant, surface, inter- and after-condensers were subject to the same fouling mechanisms, as are all other cooling water exchangers. This fouling can occur in any portion of the condenser. Fouling on the tube side (cooling water) or the shell side—depending on the unit's steam quality, or magnitude of a condition known as vapor binding of the tubes[5]—will lower heat transfer, condense steam at a degraded pressure and affect vacuum performance. If severely fouled, the condenser will be unable to condense motive steam at the design operating pressure.

The surface condenser was taken offline (one cooling water side partition at a time) and the tubes were cleaned. Once both shell sides were fully cleaned, the temperature inside the hotwell decreased by an average of 8°F. Given design flexibility in any plant, the authors recommend for routine maintenance procedures to include periodic cleaning of condenser bundles. Prior to the initiation of the cleaning step, it is also worthwhile to inspect for tube leaks in the surface condenser. If one side of the condenser is isolated, monitor for water conductivity change with a meter in the condensate from the hotwell. If a conductivity meter is unavailable, test for hardness with a pronated dye solution such as the Eriochrome Black T solution, which changes color in an alkaline environment from blue (no hardness indication) to purple (hardness present) when complexed with magnesium or calcium. A large conductivity or hardness color change would indicate that one side of the condenser is leaking more cooling water into the steam condensate system than the other. If a large tube leak is detected, caution must be exercised when opening the shell side to plug the tubes, as air will be sucked into the surface condenser.

Inter-after condenser follow-up analysis. The inter-after condenser unit was placed back into operation to record final data after the trials and modifications above. The unit achieved

FIG. 5. Trash bag test on the after-condenser vent to measure the total non-condensable load from the surface condenser and inter-after condenser units.

stable vacuum operation of 61 mmHgA–89 mmHgA upstream of the primary ejector E-1 under full plant load operation vs. the initial pressure of 125 mmHgA (prior to vacuum deterioration, as in **FIG. 3**) at the beginning of the study. The suction pressure to the secondary ejector E-2 decreased by a significant 114 mmHg. With the thermostatic vent on the floater traps plugged, bypass around traps to the hotwell isolated, and other leak checks completed, no high, non-condensable vapor flow coming from the after-condenser vent was evident.

Takeaway. Ejectors and inter-after condenser units constitute a critical part of stable turbine and surface condenser operation. Consistent and sustainable vacuum is crucial for turbines and the compressors they power. Without these seemingly simple ejector/condenser units, stability is jeopardized and turbine steam efficiency is compromised. The inherent simplicity of operation should not be taken for granted when multiple bottlenecks may be acting simultaneously to the detriment of any system performance. Answers to operational problems exist in the field, and will always become more visible with a systematic approach and a good field testing plan. By establishing frequent performance checks, the avoidance of future process problems and the minimization of operating costs are achievable. HP

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