## Introduction

Whether for lube oil, fuel oil, or general fractionation, vacuum columns utilize ejector systems to maintain design vacuum levels within the column. Noncondensibles, cracked gases, hydrocarbon vapors and steam are removed from the column by the ejector system. Extraction of these fluids from the column is key to a proper vacuum level within the column and consequently, design charge rates and specification quality product are achieved.

Refiners do have lengthy operating experience with ejector systems. Ejector systems have been the mainstay for refinery vacuum distillation. Whether a crude vacuum tower operates as a ‘wet’, ‘damp’ or ‘dry’ tower, an ejector system is the vacuum producer. Different tower operating pressures and overhead load characteristics of wet, damp or dry operation affect only the configuration of an ejector system but the basic operating principle remains unchanged.

Even with lengthy operating experience, refiners view ejector systems with hesitation and uncertainty. This uncertainty results from an incomplete understanding of the basic operating principles of ejectors themselves and their interdependency with any vacuum condenser it supports or to which it discharges. There is only limited information in technical journals or books addressing operating principles of ejector systems. On a positive note, ejector systems are quite reliable and performance shortcomings are not a common problem. However, when operating problems do occur, they appear as a dramatic change in performance rather than a gradual loss of performance. Vacuum tower crisis is always critical and an immediate remedy is necessary. The purpose of this article is to offer a concise and complete overview of

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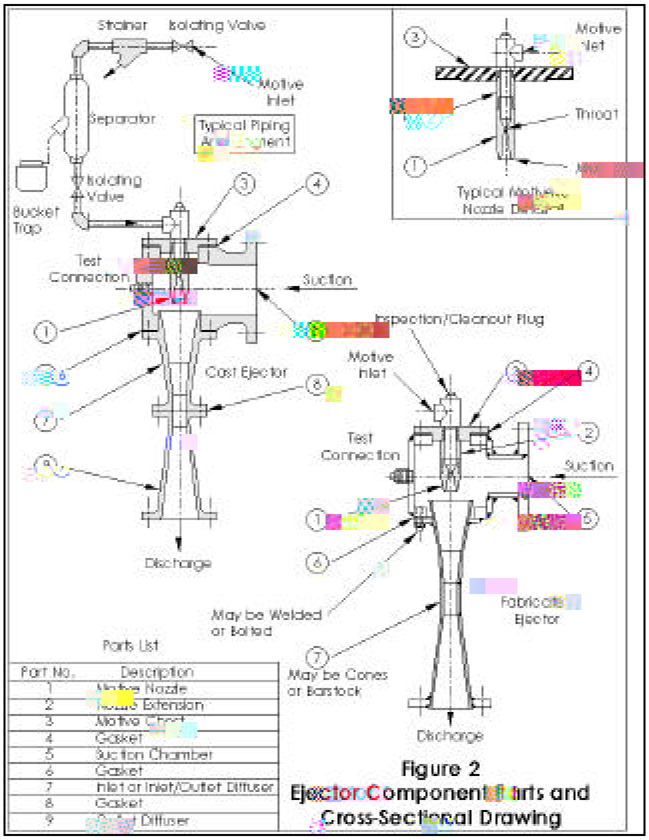


**Figure 2**
**Ejector Component Parts and Cross-Sectional Drawing**

## Operating principle

The basic operating principle of an ejector is to convert pressure energy of high pressure motive steam into velocity.

## Motive steam

Minimum motive steam pressure is important and is also shown on a performance curve. The manufacturer has designed the system to maintain stable operation with steam pressures at or above a minimum steam pressure. If motive steam supply pressure falls below design, then a motive nozzle will pass less steam. When this happens, the ejector is not provided with sufficient energy to compress the suction fluid to the design discharge pressure. The same problem occurs when the supply motive steam temperature rises above its design value, resulting in increased specific volume, and consequently, less steam passes through the motive nozzle.

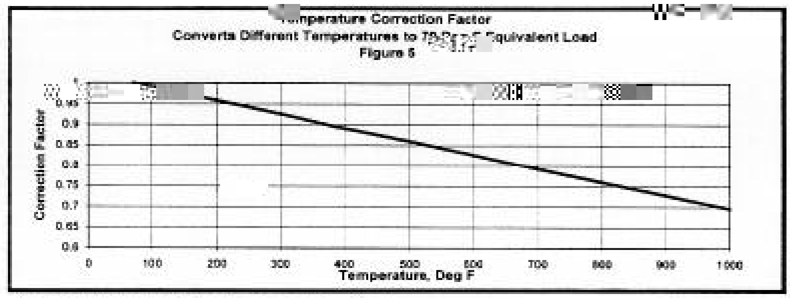


Figure 5. Temperature correction factor.

An ejector may operate unstably if it is not supplied with sufficient energy to allow compression to its design discharge pressure. Unstable ejector operation is characterized by dramatic fluctuations in operating pressure. If the actual motive steam pressure is below design or its temperature above design, then, within limits, an ejector nozzle can be rebored to a larger diameter. The larger nozzle diameter allows more steam to flow through and expand across the nozzle. This increases the energy available for compression. If motive steam supply pressure is more than 20 - 30% above design, then too much steam expands across the nozzle. This tends to choke the diffuser. When this occurs, less suction load is handled by the ejector and suction pressure tends to rise. If an increase in suction pressure is not desired, then ejector nozzles must be replaced with ones having smaller throat diameters or the steam pressure corrected.

Steam quality is another important performance variable. Wet steam may be damaging to an ejector system. Moisture droplets in motive steam lines are accelerated to high velocities and become very erosive. Moisture in motive steam is noticeable when inspecting ejector nozzles. Rapidly accelerated moisture droplets erode nozzle internals. They etch a striated pattern on the nozzle diverging section and may actually wear out the nozzle mouth. Also, the inlet diffuser tapers and throat will have signs of erosion. On larger ejectors, the exhaust elbow at the ejector discharge can erode completely through. Severe tube impingement in the intercondenser can also occur but this is dependent upon ejector orientation. To solve wet steam problems, all lines up to the ejector should be well insulated. Also, a steam separator with a trap should be installed immediately before an ejector motive steam inlet connection. In some cases, a steam superheater may be required. Wet steam can also cause performance problems. When water droplets pass through an ejector nozzle, they decrease the energy available for compression. Furthermore, water droplets may vaporize within an ejector as temperature increases. Vaporized water droplets act as an additional load that the motive steam must entrain and compress. The effect is a decrease in load handling ability. With extremely wet steam, the ejector may even become unstable.

## Maximum discharge pressure

The maximum discharge pressure (MDP), also shown on the performance curve, is the highest discharge pressure that an ejector has the ability to achieve with the given amount of motive steam passing through the steam nozzle. If the discharge pressure exceeds the MDP, the ejector will become unstable and break operation. When this occurs, a dramatic increase in suction pressure is common. As an example, when a system designed to produce 15 mm HgA pressure breaks operation, suction pressure sharply increases to 30 - 50 mm HgA. This often causes a tower upset. Therefore, it is of paramount importance to make sure ejectors do not exceed their MDP.

Since increasing the discharge pressure above the MDP causes a loss of performance, it seems logical that lowering the discharge pressure below the MDP should have the opposite affect. This, however is not the case. Ejectors with a compression ratio, discharge pressure divided by suction pressure, higher than 2:l are called critical ejectors. Performance of a critical ejector will not improve if its discharge pressure is reduced. This is primarily due to the presence of the shock wave in the ejector diffuser throat.

## Condensers

### Component parts

Condensers are manufactured in three basic configurations: fixed tubesheet, U-tube or floating head bundle. Thermodynamically, these units perform identically. They differ only in ease of maintenance and capital cost. The fixed tubesheet unit, typically TEMA, AEM, BEM, AXM or BXM styles, has a bundle that is not removable from the shell. This unit is generally the least expensive to build. The major disadvantage of this type of unit is that the shellside of the condenser is not accessible for normal cleaning methods. The U-tube exchanger, TEMA, AEU or BEU, is the next most economical type of construction for a removable bundle. Since the bundle is completely removable from the shell, it allows thorough cleaning of the shellside as well as the tubeside. The major drawback to the U-tube unit is that the U-bend section of the tube can make

Table 1. Example of how various molecular weight gases are converted to the appropriate vapour equivalent

| Component | Flow rate | MW correction factor | Temp. correction factor | Equiv. flow |
|---|---|---|---|---|
| Water vapour | 100 pph | 1.0 | 0.96 | 104.2 pph |
| Air | 20 pph | 1.25 | 0.96 | 16.7 pph |
| Hydrocarbon | 50 pph | 1.86 | 0.96 | 28 pph |
| | | | Total equiv. flow* | 148.9 pph |

$$\text{* water vapour equivalent flow rate} = \frac{\text{(component flow rate)}}{\text{(MW correction factor)(temp correction factor)}}$$

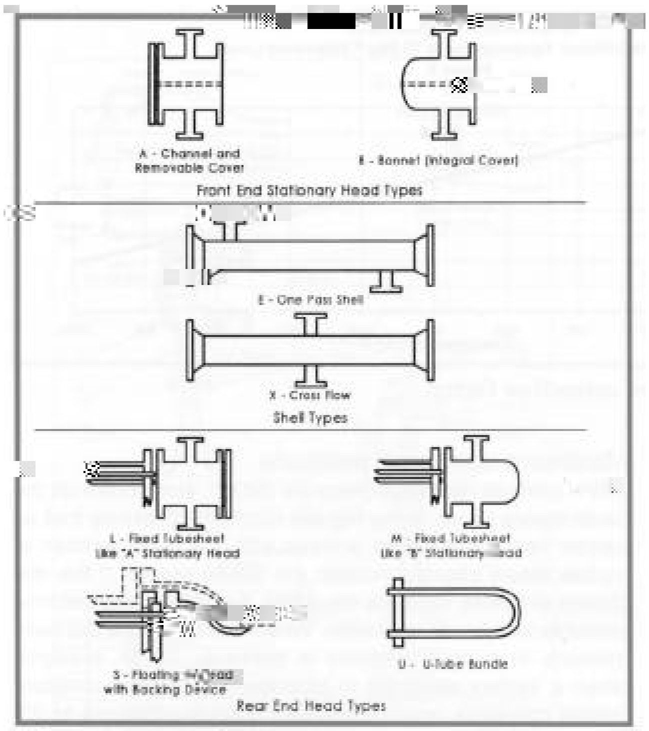


**Figure 6. TEMA designations.**

difficult cleaning of tube internal surfaces. Floating head the downstream ejectors. This makes the performance of ejectors very dependent on the upstream condensers.

The first intercondenser is the largest and most critical condenser from a design and operation standpoint. The pressure that the first intercondenser is designed to operate at is directly related to the maximum cooling water temperature for which the system is designed. The pressure inside the condenser must be high enough for condensation to occur. For instance, with 91 °F cooling water, an initial condensing temperature of approximately 115 °F is reasonable. This corresponds to a first stage intercondenser operating pressure of 76 mm Hg.

The equation for design of a vacuum condenser is the classic heat transfer relationship:

**Q = U x A x LMTD**

where:

**Q**= Amount of heat transfer required (btu/hr)

**U** = Overall heat transfer rate (btu/hr $ft^2$ °F)

**A=** Surface area of the condenser ($ft^2$)

**LMTD =** Log mean temperature difference (°F)

During the design phase, all of these variables are fixed. Q is fixed by the amount of steam being used by the upstream ejector and the amount of load coming over from the tower. The amount of steam that an ejector uses is directly related to the compression ratio. Therefore, a high design cooling water temperature results in a high minimum first intercondenser pressure which results in a high steam usage for the first stage ejector.

The heat transfer rate is a function of cooling water flow, process side condensing characteristics and tube material. Normally the heat transfer rate is determined for the tubeside and shellside separately and then combined into an overall heat transfer rate. The overall heat transfer rate is then used in the above equation to calculate the required surface area.

The surface area is set by the number of tubes in the condenser. The tubes in most crude vacuum system condensers are 3/4 in. diameter tubes and the surface area is calculated based on the external surface area of the tube.

The LMTD is a thermodynamic quantity that is used to calculate the amount of heat that is given up. The LMTD is set by the cooling water inlet temperature, cooling water temperature rise and the shellside inlet and outlet temperatures.

## Cooling water

When cooling water supply temperature rises above its design value, ejector system performance is penalized. A rise in cooling water inlet temperature decreases condenser available LMTD. When this occurs, the condenser will not condense enough and more vapors are carried out as saturated vapors with the noncondensible gases. As discussed in the preceding ejector section, this increased load to a downstream ejector cannot be handled by that ejector.

Similarly, if cooling water flow rate falls below design values, a greater temperature rise across the condenser occurs. Even if cooling water is at its design inlet temperature, a greater temperature rise reduces available LMTD. Condensation efficiency is reduced and additional load is passed on to a downstream ejector. Losses in cooling water flow occur over time as more process equipment is added to a cooling water loop or system pressure drop rises and reduces capacity of cooling water pumps. Furthermore, reduction in cooling water flow lowers the heat transfer rate.

Lower than design inlet cooling water temperature does not have a negative affect. Actually it often removes system performance problems. Typically summer months place the greatest strain on an ejector system. It is at this time that cooling water is warmest and demands on the cooling tower are the greatest. During winter months, the lower inlet cooling water temperature increases the safety margin for condenser operation as LMTD is greater than the design value.

## Fouling

Intercondensers and aftercondensers are subject to fouling like all other refinery heat exchangers. This may occur on the tubeside, shellside or both. Fouling deters heat transfer and, at some point, may compromise system performance.

Cooling tower water on the tubeside is prone to biological fouling or fouling due to corrosion products. Vacuum condensers are always designed to include a margin for fouling. Over time, Proprietary design procedures incorporate the following considerations:

- Condenser vapor inlet location and distribution area above the tube field so as to insure proper vapor entry to the shell and penetration into the tube field.
- Tube field layout and penetration areas to guarantee that flow distribution into the bundle is well maintained and pressure drop is held to a minimum.
- Noncondensible gas cooling section, where bulk condensate is separated from the vapor and final cooling to design saturation temperature is achieved.
- Bulk condensate and noncondensibles exit the shell at different locations and temperatures. In this way, noncondensibles and vapors are cooled below the condensate temperature to maximize condensation efficiency without contending with excessive condensate loading and associated thermal duty.
- Support plate spacing and bundle penetration areas to insure velocities are well below those necessary to establish vibration.
- Process vapors assessed to properly ascertain vapor/liquid equilibrium (VLE) conditions throughout the condensing regime.
- Condensing profile broken down into as many as fifty steps to properly determine the effective LMTD and VLE at each step.

Often proprietary designs are compared to those determined by computer programs available from institutional organizations, research companies or software companies. These generic programs do not

## Process conditions

These are very important for reliable vacuum system operation. Process conditions used in the design stage are rarely experienced during operation. Vacuum system performance may be affected by the following process variables, which may act independently or concurrently:

- ***Noncondensible loading.*** Vacuum systems are susceptible to poor performance when noncondensible loading increases above design. Noncondensible loading to a vacuum system consists of air leaking into the system, lightened hydrocarbons, and cracked gases from the fired heater. The impact of higher than design noncondensible loading is severe. As non-condensing loading increases, the amount of saturated vapors discharging from the condenser increases. The ejector following a condenser may not handle increased loading at the condenser design operating pressure. The ejector before the condenser is not designed for a higher discharge pressure. This discontinuity in pressure causes the first ejector to break operation. When this occurs, the system will operate unstably and tower pressure may rapidly rise above design values.
- Noncondensible loadings must be accurately stated. If not, any vacuum system will suffer performance shortcomings. If noncondensible loadings are consistently above design, then new ejectors are required. New condensers may be required depending on severity.
- ***Condensible hydrocarbons.*** Tower overhead loading consists of steam, condensible hydrocarbons and noncondensibles. As different crude oils are processed or refinery operations change, the composition and amount of condensible hydrocarbons handled by the vacuum system vary. A situation may occur where the condensible hydrocarbon loadings are so different from design that condenser or ejector performance is adversely affected. This may occur in a couple of different ways. If the condensing profile is such that condensible hydrocarbons are not condensed as they were designed to, then the amount of vapor leaving the condenser increases. Ejectors may not tolerate this situation, resulting in unstable operation. Another possible effect of increased condensible hydrocarbon loading is an increased oil film on the tubes. This reduces the heat transfer coefficient. Again, it may result in increased vapor and gas discharge from the condenser. Unstable operation of the entire system may also result. To remedy performance shortcomings, new condensers or ejectors may be necessary.

***Tower overhead loading.*** In general, a vacuum system will track tower overhead loading as long as noncondensible loading does not increase above design. Tower top pressure follows the performance curve of the first-stage ejector. Figure 3 shows a typical performance curve. At light tower overhead loads, the vacuum system will pull tower top operating pressure down below design. This may adversely affect tower operating dynamics and pressure control may be necessary. Tower pressure control is possible with multiple element trains. At reduced overhead loading, one or more parallel elements may be shut off. This reduces handling capacity, permitting tower pressure to rise to a satisfactory level. If multiple trains are not used, recycle control is another possible solution. Here, the discharge of an ejector is recycled to the system suction. This acts as an artificial load, driving the suction pressure up. With a multiple-stage ejector system, design that lain, itl should sreases. Ej651  Tc 2.9cctiornh 0.1732 on. H
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exhausts to atmosphere, the sound pressure level should be checked for meeting OSHA standards, paragraph 1910.95 and Table G-12 and/or the local standards.

A thermostatic type condensate trap should be avoided since they have a tendency to cause a surge or loss of steam pressure when they initially open. This could cause the ejector to become unstable.

## Operation

*Start-up*

The ejector motive line should be disconnected as near as possible to the motive inlet and the lines blown clear. This is extremely important on new installations where weld slag and chips may be present and scale particles could exist. These particles could easily plug the motive nozzle throats. If a strainer, separator, and/or trap is present they should be inspected and cleaned after the lines are blown clear. The vapor outlet of the aftercondenser and condensate outlets should be open and free of obstructions and the cooling medium should be flowing to the condenser(s).

All suction and discharge isolating valves, if present, should be opened. If the unit has dual elements with condensers present, ensure the condenser is designed for both elements operating. If the condenser has been designed for one element operating, the suction and discharge valves should be opened to only one element (the other element being isolated).

The motive valve to the last ejector stage (‘Z’ stage) should then be fully opened. For optimum performance during an evacuation cycle the motive valves should always be opened starting with the ‘Z’ stage and proceeding to the ‘Y’, ‘X’, etc. stages. If a pressure gauge is present near the motive inlet, the reading should be taken to ensure the operating pressure is at or slightly above that for which the unit is designed. The motive pressure gauge should be protected with a pigtail to insure protection of the internal working parts of the gauge. The design operating pressure is stamped on the ejector nameplate.

*Shutdown*

There are two procedures to be considered when shutting down: method A is appropriate if it is desired to maintain the vacuum upstream of the first stage ejector (an isolating valve has to be present at suction) rather than allow pressure to rise to atmospheric pressure, in which case the valves should be closed in the following order:

- Close 1st stage suction valve.
- Close 1st stage motive inlet valve.
- Close 2nd stage suction valve.
- Close 1st stage discharge valve.
- Close second stage motive inlet valve.
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**Table 2. Case studies**

| | Description | Design | Case 1 | Case 2 | Case 3 | Case 4 |
|---|---|---|---|---|---|---|
| | | Ejector | | | | |
| P1 | 1st stg. suction press. (mmHgA) | 20 | 50 | 62 | 30 | 56 |
| P2 | 1st stg. disch. press. (mmHgA) | 83 | 96 | 102 | 80 | 108 |
| P3 | 2nd stg. suction press. (mmHgA) | 79 | 92 | 98 | 77 | 104 |
| P4 | 2nd stg. disch. press. (mmHgA) | 292 | 285 | 320 | 280 | 280 |
| P5 | 3rd stg. suction press. (mmHgA) | 277 | 275 | 305 | 271 | 272 |
| P6 | 3rd stg. disch. press. (psig) | 4.8 | 4.4 | 4.5 | 4.6 | 4.1 |
| | | | | | | |
| P7 | 1st stg. motive press. (psig) | 140 | 146 | 143 | 146 | 141 |
| P8 | 2nd stg. motive press. (psig) | 140 | 145 | 143 | 145 | 132 |
| P9 | 3rd stg. motive press. (psig) | 140 | 144 | 144 | 145 | 140 |
| | | Condensers | | | | |
| P10 | 1st I/C CW inlet press. (psig) | - | 66 | 60 | 63 | 51 |
| T1 | 1st I/C CW inlet temp. (˚F) | 91 | 82 | 76 | [illegible] | [illegible] |
| P11 | 1st I/C CW outlet press. (psig) | - | 61 | 55 | 55 | 46 |
| T2 | 1st I/C CW outlet temp. (˚F) | 112 | 98 | 97.5 | 102.5 | 109 |
| | 1st I/C CW ΔP (psi) | [illegible] | [illegible] | 5 | [illegible] | 5 |
| | 1st I/C CW ΔT (˚F) | 21 | 16 | 21.5 | 16.5 | 20 |
| P12 | 2nd I/C CW outlet press. (psig) | - | 56 | 49.5 | 47.5 | 41 |
| T3 | 2nd I/C CW outlet temp. (˚F) | 117.1 | 105 | 103 | 106 | 114 |
| | 2nd I/C CW ΔP (psi) | 5.1 | [illegible] | [illegible] | 7.5 | 5 |
| | 2nd I/C CW ΔT (˚F) | 5 | 7 | 5.5 | 3.5 | 5 |
| P13 | A/C CW outlet press. (psig) | - | 50 | 44.5 | 40.5 | 35.5 |
| T4 | A/C CW outlet temp. (˚F) | 122.6 | 110 | 108 | 110 | 120 |
| | A/C CW ΔP (psi) | [illegible] | 6 | 5 | 7 | 5.5 |
| | A/C CW ΔT (˚F) | 5 | 5 | 5 | 3.5 | 5 |

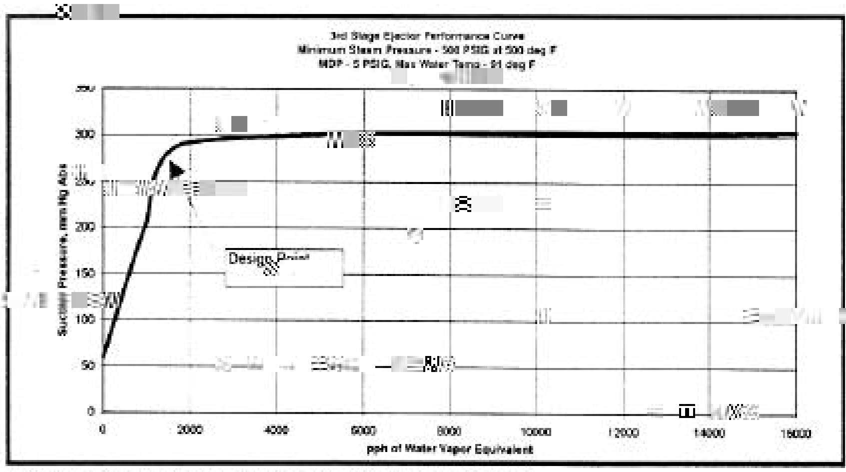


Figure 12C. 3rd stage ejector performance curve.

Measurement data can then be compared to the design data. This is done using the system performance curve and data sheets. It is often very helpful to be able to compare new data to baseline data taken when the system was operating correctly

Table 2 is a compilation of design and test data taken for the three stage crude system shown in Figure 11. The column marked 'Design' shows the design values for all the test points. The design suction, discharge and motive pressures, P1-9, are all taken from the system performance curve shown in Figure 12. The ejector discharge pressures are calculated from the curve assuming a maximum pressure drop of approximately 5% across each condenser. The design values for condenser inlet and outlet cooling water temperature and cooling water pressure drop, $\Delta p$, are obtained from the manufacturer's condenser data sheets. As shown, there are no design values given for the cooling water inlet and outlet pressures. For design and troubleshooting the only important number is the pressure loss across the condenser, not the actual pressure.

Case studies 1 to 4 represent examples of different types

## Troubleshooting assistance tables

**Table 3. Ejector evaluation**

| Problem | Effect | Corrective action |
|---|---|---|
| Lower than design motive steam pressure. | Poor ejector performance. | Raise steam pressure or bore steam nozzles. |
| Higher than design motive steam pressure. | Reduced ejector capacity and wastes steam. | Reduce motive pressure or replace steam nozzles with new nozzles designed for a higher steam pressure. |
| Higher than design steam temperature (50 °F or more). | Poor ejector performance. | Raise steam pressure or bore steam nozzles. |
| Higher than design discharge pressure. | Poor ejector performance. | Look downstream for problems, for example:<br>◆ Condenser problem<br>◆ Downstream ejector problem<br>◆ Discharge piping restriction |
| Low ejector discharge temperature. Ejector discharge temperature should be superheated at least 50 °F above saturation. If not, the cause is wet motive steam. | Reduced ejector capacity or poor performance | Insulate steam lines.<br>Add moisture separator in motive steam line. |
| Higher than design suction pressure (assuming motive steam pressure and quality are normal and discharge pressure is equal or less than design.) | Greater than design load or mechanical problems with ejector. Either worn out internals or possible internal steam leak around steam nozzle threads. | Inspect internal dimensions and replace if necessary.<br>Tighten steam nozzle to steam chest if necessary or seal weld nozzle to steam chest. |

**Table 4. Condenser evaluation**

| Problem | Effect | Corrective action |
|---|---|---|
| High $\Delta P$ across shellside (As a rule of thumb, normally DP should be 5% of absolute design operating pressure or less) | Poor condenser performance:<br>◆ Shell side or tubeside fouling.<br>◆ Cooling water temperature higher than design.<br>◆ Low cooling water flowrate.<br>◆ Higher than design condensible hydrocarbon (approx. 20 - 30% above design). | Clean tubes.<br>Reduce temperature, increase cooling water flow.<br>Increase cooling water flow.<br>Reduce hydrocarbon load or larger condenser and downstream ejector required. |
| Higher than design tubeside $\Delta P$. | Poor condenser performance:<br>◆ Tubeside fouling.<br>◆ Higher than design cooling water flow. | Clean tubes.<br>Not a problem. |
| Higher than design tubeside $\Delta T$. | Poor condenser performance:<br>◆ Low cooling water flow.<br>◆ Higher than design duty. | Increase flowrate.<br>Increase cooling water flowrate or replace condenser. |
| High vapour outlet temperature. | Poor condenser performance. | Tube fouling.<br>Cooling water flowrate low or inlet temperature high.<br>Possible internal bypassing.<br>Check with manufacturer.<br>Downstream ejector not functioning and backstreaming. |

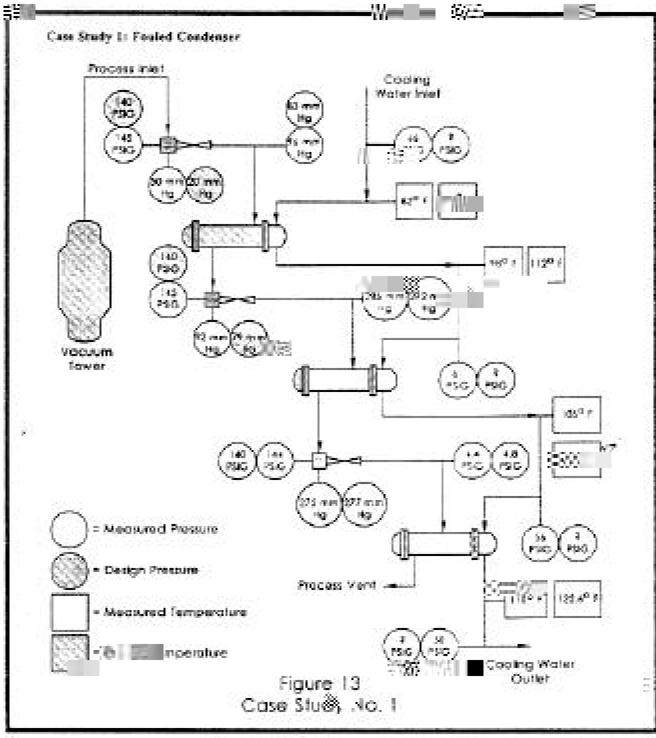


Figure 13. Case study 1: fouled condensor.

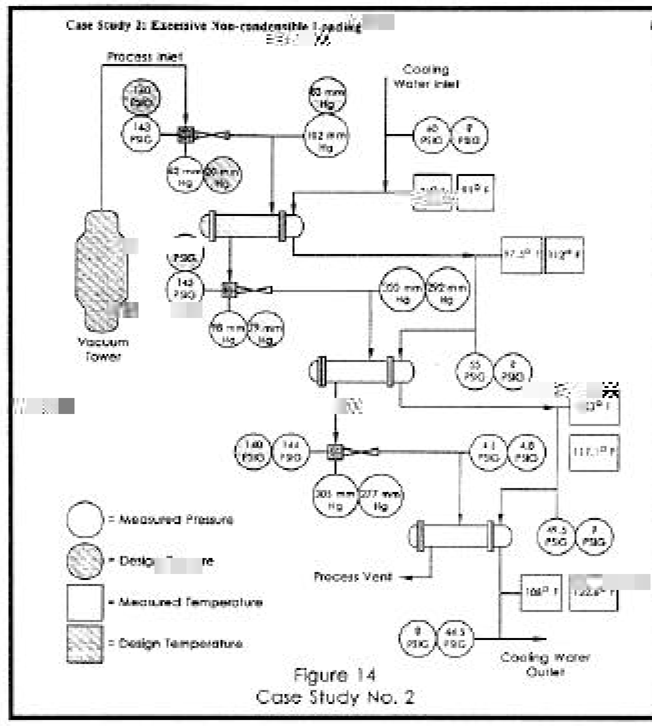


Figure 14. Case study 2: excessive non-condensible loading.

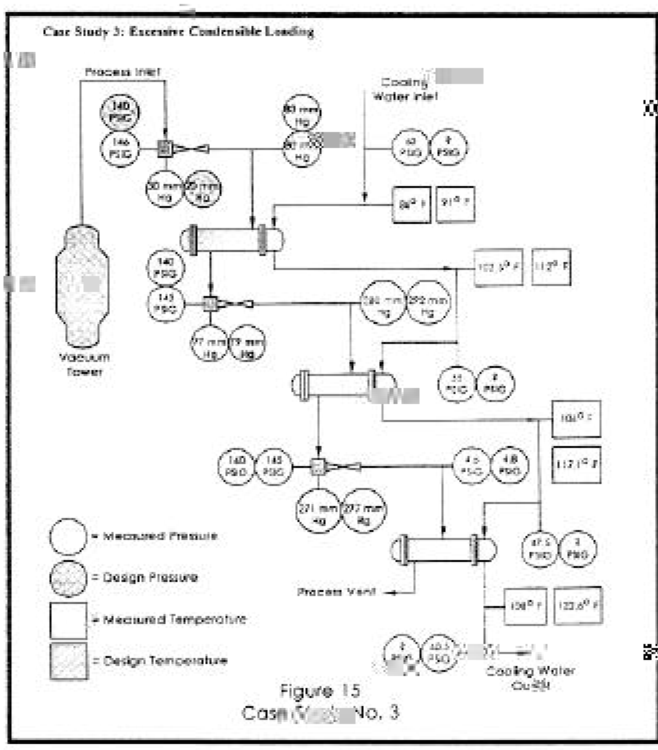


Figure 15. Case study 3: excessive condensible loading.

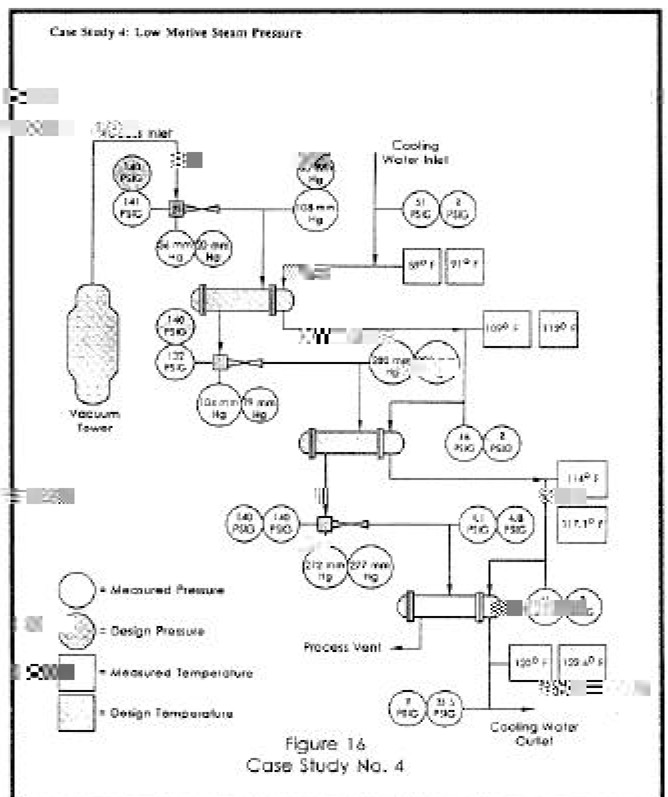


Figure 16. Case study 4: low motive steam pressure.

of common performance problems. In each case, a different problem was found with the equipment. After each case has been dicussed, there will be an additional section on how mechanical failures can also contribute to the symptons shown.

**Case study 1:**
fouled condenser

The most common performance problem with steam ejector systems is lower than design steam pressure. For this reason, motive steam pressure is always the first data steam pressure is always the first data that should be examined. In this case, the motive steam pressures at each ejector, P7-9, are all above design and should not pose any performance problems. Next, the ejector suction and discharge pressures are examined, starting with the third stage ejector. The process begins with the last stage because if that is not working, then the other stages will not work either.

Here, the third stage discharge pressure, P6, and third stage suction pressure, P5, are both below design. Thus, the third stage ejector is operating correctly and its load must be within design limits. Since the third stage ejector load is within design limits, the second intercondenser must be working properly. Next, the second stage ejector discharge pressure, P4, is examined. It is also below design, indicating an acceptable shellside $\Delta P$ of 3.5%. Remember, pressure drop across a vacuum condenser should be less than 5% of its operating pressure.

Moving to the second stage ejector suction, P3, the system's problems begin to show up. P3 is 13 mm Hg higher than design. It is not possible for the first stage ejector to compress its load to 96 mm Hg Abs, 13 mm Hg greater than the 83 mm Hg Abs design, and still maintain a suction pressure of 20 mm Hg Abs. The higher than design first stage discharge pressure is causing the first stage ejector to break operation. The logical cause of the high second stage ejector suction pressure is a fouled first intercondenser. To confirm this, the cooling water data is examined.

The cooling water pressure drop on all three condensers is normal, indicating cooling water flow rate is approximately at design. The cooling water temperature rise is low across the first intercondenser and high across the second intercondenser. The low temperature change on the first intercondenser indicates that the cooling water is not absorbing as much heat as it should and therefore, must be fouled. As previously discussed, a fouled condenser allows greater vapor carry over to the downstream ejector. This accounts for the high second stage ejector suction pressure and high second intercondenser cooling water temperature rise.

### Case study 2:
excessive noncondensible loading

Following the same thought process as case study 1, motive steam pressure is not a problem. The third stage ejector discharge pressure is also under design. It is noted that the third stage ejector suction pressure is higher than design, measured at 305 mm Hg Abs versus a design of 277 mm Hg Abs. This appears to affect first and second stage ejector performance.

Possible causes of an elevated suction pressure are cooling water flow rate below design, cooling water inlet or outlet temperature greater than design, condenser fouling or higher than design loading to the ejector. Reviewing cooling water data suggests no abnormalities, i.e. pressure drop across each condenser seems acceptable and cooling water temperatures are below design values. With cooling water pressure drop and temperature rise at each condenser close to design values, fouling may be ruled out. The remaining possible cause is an increased load to the ejector.

Common performance problems arise when noncondensible gas loading exceeds the design value.

Higher non-condensible loading results in increased loading to downstream ejectors. This is due to a higher mass flow rate of noncondensibles plus their associated vapors of saturation.

The elevated pressure at the third stage ejector suction causes the second stage to break operation. Again, this is because the second stage ejector is unable to compress its load to a pressure greater than 292 mm Hg Abs. Therefore, there is an increase in the suction pressure of the second stage as it breaks operation. This, in turn, forces the first stage to break operation and the suction pressure to the system increases from 20 mm Hg Abs to 62 mm Hg Abs.

### Case study 3:
excessive condensible loading

This case is characterized by a modest loss in lower top pressure. Once again, the steam pressure to each ejector is satisfactorily above design. The third stage ejector suction and discharge pressures are below design. The second stage ejector suction and discharge pressures are also below normal, as is the first stage ejector discharge pressure. The only pressure that is abnormal is the first stage ejector suction pressure.

The cooling water data indicates all three condensers have higher than design cooling water pressure drops and lower than design temperature rises. This indicates that: the high cooling water pressure drop is an indication of either fouling or high cooling water flow rate. The low $\Delta T$ indicates that either the condensers are fouled or that there is a high cooling water flow rate. The previous analysis of the suction pressures of the second and third stage ejectors show no signs of fouling, i.e. elevated suction pressures. The conclusion must be that there is a higher than design cooling water flow rate to the condensers. Higher cooling water flowrate does not affect ejector system performance. The elevated first stage suction pressure and tower top pressure must be the result of a high condensible load causing the ejector to run out further out on its curve.

### Case study 4:
low motive steam pressure

Using the same method as previous case studies provides a quick answer to this performance problem. The steam pressure on the second stage ejector is below design. As discussed earlier, this will cause the second stage to break operation. When this second stage ejector breaks operation, its suction pressure rises above the maximum discharge pressure of the first stage ejector. This results in broken operation for the first stage ejector and increased tower top pressure. This situation will correct itself if the second stage ejector steam pressure is increased.

### Mechanical problems

Now that examples of how process conditions, fouling and utilities will affect system performance have been seen, it needs to be understood what affect mechanical problems will have on a system. A common mechanical problem is a loose steam nozzle. When a steam nozzle becomes loose it begins to leak steam across the threads. The leaking steam then becomes load to the ejector. If the loose nozzle occurs in the first stage ejector the affect will be an overloaded first stage ejector. If the leak occurs in the

second or third stage ejector, the data will look similar to a fouled condenser.
Inspection of ejector internals should be done periodically.