ABSTRACT

Existing steam condenser venting equipment is often undersized, improperly installed or required to operate beyond its capabilities. The recent trend to part load operation, caused by market conditions, has accentuated the problem.

Inadequate venting equipment retains air in the condenser, which may raise the condenser pressure and will increase the absorption of oxygen in the condensate.

Conditions which contribute to inadequate venting are described and the characteristics of the two most common venting devices, steam ejectors and liquid ring vacuum pumps, are discussed.

A hybrid system which will permit condenser operation at low absolute pressure and will reduce dissolved oxygen in the condensate is described.

INTRODUCTION

The addition of large, more efficient Nuclear and Fossil plants, combined with reduced power consumption, cogeneration, and a host of other factors, increasingly imposes cycling or two-shift operation of existing units originally designed for base load operation.

This daily cycling, nightly and weekend shutdown, often requires part load steam condenser operation with consequent low operating pressures.

This article deals with one aspect of this problem: the need for venting equipment which is adequate under all conditions of operation.

We have found that existing venting equipment is often undersized, improperly installed, and required to operate beyond its capabilities.

The dynamic relationship between steam condensers and the associated venting equipment is sometimes ignored and may not always be fully understood.

It is worth noting that anytime the back pressure is higher than necessary because air is retained in the condenser, condensate oxygen absorption will increase, and the turbo-generator heat rate may be increased (References 1 and 2). This condition has been referred to as one where the condenser itself does not control the back pressure. The point is illustrated in Figure #1 - "The Effect of Air Leakage on Condenser Pressure", and is referenced in EPRI NP-3020 (1, 10).

The figure graphically demonstrates the dynamic balance which exists between a steam condenser and its venting equipment.

Harrington states (10): "The important considerations from a dissolved oxygen standpoint are as follows:

- When the air pump controls the back pressure, dissolved oxygen levels can be expected to rise.
- Excessive air inleakage can occur without affecting the operating vacuum, but the dissolved oxygen levels will rise."

We shall see that excessive water vapor carryover from the condenser air cooling section, as well as poor installation practices, contribute to venting equipment overloading.
We will also see that dissolved oxygen levels can be controlled by limiting the partial pressure of the air in the exit vent mixture. This is simply an application of “Henry's Law,” which indicates that oxygen will dissolve in water in proportion to its partial pressure and inversely to Henry's constant (which varies with temperature).

By sizing the venting equipment to remove large ratios of water vapor with each pound of air, we reduce the air partial pressure and, hence, the oxygen partial pressure over the condensing liquid and maintain an atmosphere within the condenser which limits the amount of oxygen that can be dissolved.

A thorough discussion of Henry's Law consideration is given in reference (5).

The importance of maintaining low dissolved oxygen levels in the condensate has been thoroughly discussed in several recent EPRI Reports (1, 3, 4, 5, 6, 7).

These reports identify dissolved oxygen as a major contributor to the corrosion process, especially in PWR nuclear plants and cycling fossil plants. Dissolved oxygen levels of 10 ppb or less are recommended. For modern plants, both fossil and PWR nuclear, goals of 5 to 7 ppb are usually cited.

If we wish to limit dissolved oxygen in the condenser, we must:

1. Keep air out of the condenser.
2. Remove air at a rate greater than the rate of ingress at all condenser loadings and at all operating pressures.
3. Properly design the condenser to avoid air pockets and insure good vapor distribution and deaeration.
4. Maintain an atmosphere within the condenser in which the O₂ partial pressure is low enough to satisfy Henry's Law (5).
5. Provide means to reheat condensate as it drops into the hotwell.

Reheating in deaerating hotwells at low load conditions (always a problem) is discussed in reference (6) and a device effectively utilized in several foreign plants is described.

Short of preventing all leakage, however, unless air is removed at a rate at least equal to its entry, and unless the O₂ partial pressure is held to the limits calculated by Henry's Law, dissolved oxygen levels will rise. Only adequate venting equipment can cope with this requirement.

Most venting equipment is sized in accordance with HEI Standards. These standards set design pressure at 1" Hg Abs or the condenser design pressure, whichever is lower.

HEI goes on to caution “Final Selection should consider compatible compatible operation of the condenser and its venting equipment over the full range of anticipated condenser operating pressure. In addition, the physical location of the equipment should be considered where the design suction pressure is selected.”

Unfortunately, these requirements are often neglected.

The HEI instruction for establishing the venting equipment suction temperature is “The saturation temperature of the gas vapor mixture shall be considered as the steam temperature corresponding to the design pressure of the venting equipment less the greater of the following:

- either 25% of the difference between the steam temperature and the design inlet circulating water temperature or 7.5 F.

Common practice has been to simply design the venting equipment to vent the HEI tabulated air and water vapor quantities at 1" Hg Abs. and a suction temperature 7.5 F. below saturation or -71.5 F.; the air vapor mixture will be approximately 2.2 lbs. of water vapor per pound of dry air under these conditions.

Since most plants have been designed with base load in mind, the need to operate at pressures below 1" Hg Abs. has seldom been considered.

The curves of Figure #2 are a portion of the capacity curves for a condenser designed to operate at 3.6" Hg Abs. at full load with 97 F. cooling water. As can be seen at 30% load and 55 F. water, condenser vacuum is 0.6" Hg Abs.; with colder water, even lower pressures are attained. Even at full load, the condenser pressure is less than 1" Hg Abs. with 45 F. inlet cooling water.

Reference to Figure #3 indicates ejector capacity would be approximately 60% of design at 0.6" Hg Abs., and from Figure #4... liquid ring vacuum pump capacity is also reduced.

It is apparent that failure to anticipate part load and/or very low inlet cooling water temperature operating conditions result in undersizing of venting equipment.

When the power train is operated at reduced load and/or with very low inlet temperature cooling water, additional portions of the system normally operating above atmospheric pressure must operate under vacuum, thus increasing the areas exposed to air inleakage.
References (5) and (6) suggest that air leakage doubles. Reference (1) discusses three power plants operating at full and part load conditions, illustrating diagrammatically and specifically listing those additional portions of the power plant exposed to vacuum as loading is reduced. New leak paths occur as plants are cycled. Thus, at part load, not only is venting equipment inadequate because of loss of capacity at lower absolute pressures, but it is also inadequate because air inleakage increases.

For a complete description of this phenomenon, it is suggested that the original reference be read in its entirety.

The same report also points out that in order to prevent oxygen absorption in the condensate, it is necessary to minimize the partial pressures of the O₂ within the condenser. This is accomplished by designing the air removal equipment to remove a large amount of water vapor with the air.

Calculation of the required water vapor to dry air ratios produces the tabulation given in Table #1, which is simply a rephrasing of data from the original report. Reference (5), Table #6, and Appendix C thereof, explain the derivation.

If we consider the condenser of Figure #2 operating at 40% capacity with 50 F. condenser cooling water, the absolute pressure in the condenser will be 0.6” Hg Abs.

From the tabulation (Table #1), we see that if we wish to limit dissolved oxygen in the condensate to 7 parts per billion, the water vapor to dry air ratio must be 17.35 lbs. water vapor/lb. of dry air.

We have seen that present practice, however, is to size venting equipment to handle only approximately 2.2 lbs. water vapor/lb. dry air at 1” Hg Abs. (7.5 F. subcooling). This is because the table is based on the partial pressure of the air in the steam air mixture in contact with the condensate in the condenser. At full load, condensers and their integral air cooling sections are generally designed to increase this air partial pressure (assuming air flow per HEI condenser standards) to a value which results in the ratio of 2.2 lbs. of water vapor/lb. dry air at the pump suction. The ability of a given condenser to increase air partial pressure in the mixture at the pump suction is rapidly reduced as the heat rejection to the condenser is reduced.
The following discussion is based on the limiting condition where the air fraction at the pump is the same as that in the condenser. This is conservative and not quite applicable at the higher part loads, although indicative of the potential situation. It is, however, a reasonable approximation of the conditions at the very low loads associated with cycling and two-shift operations.

For further discussion of the air cooling section loss of effectiveness at low loads, reference should be made to S.W. Shor et al, Bechtel Group, Inc., EPRI NP-2294 (5).

At the above part load condition, to limit the dissolved oxygen to 7 ppb, we must remove 7.9 times (17.35 lbs. of water vapor/lb. of dry air versus 2.2 lbs. of water vapor/lb. of dry air), the amount of water vapor for which the equipment was originally designed.

Most venting equipment designed for base load conditions is indeed inadequate for cycling and two-shift operation.

It is interesting to note that as condenser load increases and condenser pressure rises, venting equipment rapidly increases in capacity.

Refer to Figure #4, which shows the performance of a typical liquid ring vacuum pump for condenser exhauster service.

At 1" Hg Abs. and a 16 F. initial temperature difference between the condenser cooling water and the condensing temperature (at 1" Hg Abs. condensing temperature = 79.03 F. - for 16 ITD seal water must be 63 F.) holding capacity is 6.25 SCFM of free dry air and water vapor to saturate at 71.5 F. (7.5 F. sub-cooling).

At 2" Hg Abs., assuming the same 16 ITD capacity is 14 SCFM of free dry air and water vapor to saturate at 93.64 F. (2" Hg Abs. = 101.14 - 7.5 F. subcooling = 93.64 F.).

The potential air removal capacity increase is 14 SCFM / 6.25 SCFM or a factor of 2.24. There is, however, no reason to believe air leakage will increase as condenser loading increases and as condenser pressure rises from 1" Hg Abs. to 2" Hg Abs. Since the amount of air available to the pump does not increase, more pump capacity is available to remove water vapor, and the pump will simply "load up" on water vapor until a balance is reached.

A similar, though much less rapid, increase in capacity occurs with two stage condenser steam ejector systems. Figure #3 is a two stage condensing steam ejector air capacity curve. To save energy, two stage condensing steam ejector systems rely on condensation of most motive steam and load water vapor in an intercondenser between the first and second stage ejector. The second stage ejector operates at a much higher suction pressure and since most of the water vapor is condensed in the intercondenser, is essentially designed to handle only the design air leakage plus a small saturation load.

In general, however, any stage of an ejector system will double in capacity if the suction pressure doubles. The two stage ejector is limited by the second stage noncondensable (air) handling capacity; thus, its potential for air removal will not double, but if, as in the case discussed above, the air load remains the same and only the water vapor increases, the two stage condensing steam ejector would also approximately double in capacity at 2" Hg Abs. versus 1" Hg Abs. because the increased load is essentially water vapor.

Thus, as capacity approaches "Base Load", venting equipment capabilities are greatly improved. Air inleakage is reduced (more of power train is at positive pressure) and water vapor/air ratios are enhanced. Referring back to Figure #1, we can see that now the condenser is controlling, and the only power wasted is that required to run "excess" venting equipment, which now may appear to be more than adequate.

It may, indeed, be more than adequate from the standpoint that the venting equipment is no longer controlling the condenser pressure, but the "excess" capacity may be needed to limit oxygen absorption. Even if "not needed" to satisfy the design limit on dissolved oxygen, the excess capacity will inevitably result in lower dissolved oxygen levels.

Unfortunately, some system designers are misled by the characteristics of venting equipment operating at base load and by good housekeeping practices which limit air leakage. These designers have purchased venting equipment sized below HEI recommendations. When called upon to operate under cycling and/or at very low circulating water temperatures, such systems are totally
The American Society at Mechanical Engineers

The power expended by the "extra" venting equipment required to be sure that you can always track the condenser, is generally less than that gained from the improved heat rate achieved when the condenser operates at the lowest absolute pressure commensurate with cooling water and load conditions.

When the situation is encountered where turbine limitations prevent further heat rate improvement with lowered condenser pressure, it is still advantageous to have extra venting capacity in order to limit oxygen absorption.

Much has been written about the large increase in capacity experienced by liquid ring vacuum pumps if air leakage increases. This increase in the ability to handle greater air leakage, and water vapor carry-over, is an excellent characteristic of liquid ring vacuum pumps. It is, however, not reasonable to permit the condenser pressure to rise if air leakage increases, since reduced efficiency and oxygen damage will result. It makes more sense to install sufficient capacity to cope with reasonable temporary air leakage increases, to bring additional venting equipment on line as required, and to adopt maintenance practices which minimize inleakage.

Improper installation of venting equipment also limits capacity.

Any pressure drop between the air outlet of the condenser air cooling section and the suction connection of the venting equipment will reduce venting capacity.

Line sizes must be large enough to cope with all load and pressure conditions. Line length must be considered when locating venting equipment.

At 1" Hg Abs., a pound of water vapor occupies 652.3 ft.³ - at 0.6" Hg Abs., it occupies 1057.3 ft.³. For the same mass flow, this would increase the pressure drop by a factor greater than 2. We have seen, as explained above, that air leakage at low load may double and Wv/DA ratios for 7 ppb dissolved oxygen may increase by 7.9 times "normal" HEI ratios. (See Table #1 and text below.)

Line sizes must be increased to accommodate these conditions.

The most common type of improper installation of venting equipment and consequent limitation of capacity is associated with liquid ring vacuum pumps.

**PRINCIPLES OF LIQUID RING VACUUM PUMP OPERATION:**

The working parts of the liquid ring vacuum pump consist of a multi-bladed impeller mounted eccentrically in a round casing which is partly filled with liquid. The principle of operation is shown in the diagram Figure #5. As the impeller rotates, the liquid is thrown by centrifugal force to form a liquid ring which is concentric with the periphery of the casing.

Due to the eccentric position of the impeller relative to the casing and liquid ring, the spaces between the impeller blades fill with liquid during rotation and any air or gas trapped in the impeller space or cell is compressed and discharged from the casing through the outlet port leaving the cell available to receive air or gas as it is presented to the inlet port of the casing.

In addition to being the compressing medium, the liquid ring absorbs the heat generated by compression and friction, absorbs any liquid slugs or vapor entering with the gas stream, and also condenses water vapor entering with the gas.

Thus, in liquid ring pumps the seal liquid (water) is the compressant and the warmer incoming gas will approach the temperature of the seal fluid before the suction port closes and compression starts. The temperature of the seal fluid at this time is assumed to be the entering seal temperature plus the heat imparted to the seal fluid. This heat is the total of the pump horsepower plus any heat contained in the inlet air/vapor stream less the heat contained in the exit air vapor stream.

---

In addition to leakage (the only consideration for HEI tables), there are other sources of noncondensibles. In BWR units, hydrogen and oxygen gases are generated in the reactor from the disassociation of water. In some power plants, large quantities of steam are exported, thus requiring large quantities of water make-up. Depending upon degassing arrangements, condenser venting equipment may be called upon to remove all or part of these gases, as well as normal air leakage. During soot-blowing make-up rates increase, temporarily overloading make-up degasifiers with consequent increased call on condenser venting equipment.

During start-up, which may occur as many as 250 times a year in two-shift plants, the worst part load conditions exist.

Loading is very low, cooling water is available in excess, and may be cold. Under these conditions, venting equipment is required to operate at the lowest absolute pressures and greatest air leakage rates prevail. This too must be considered in venting equipment sizing.

---
Figure #4 is the curve for a liquid ring pump that would be selected in accordance with tabulated HEI Standards for a condenser designed to condense between 100,000 and 250,000 lbs./hr. of steam. HEI indicates venting capacity should be 7.5 SCFM of free dry air plus water vapor to saturate based upon subcooling 7.5°F below saturation temperature. Design vacuum is set by HEI at 1” Hg Abs. equivalent saturation temperature is 79.03°F. This capacity may also be expressed as 33.8 lbs./hr. of dry air and 74.4 lbs./hr. of water vapor.

The curve shown in Figure #4 is the typical curve supplied by liquid ring pump manufacturers for liquid ring pumps used as condenser exhausters. In most cases, pumps are selected for a 19°F ITD (Initial Temperature Difference between saturation temperature at the condenser pressure and the inlet cooling water to the condenser). These curves are based on a recirculated seal water system (discussed later) and contain an allowance for seal water temperatures 5°F higher than the cooling water entering the condenser. Thus, if you require, per HEI, a capacity at 1” Hg Abs of 7.5 SCFM of air plus water vapor to saturate with 7.5°F subcooling, you would require a 19°F ITD, i.e. cooling water must be 60°F = (79°F - 19°F) entering the steam condenser to achieve 1” Hg Abs. at the pump suction. If condenser cooling water is warmer, a larger pump is required to satisfy HEI requirements.

Many liquid ring vacuum pumps selected for this type of application are based upon a 19°F ITD. This is a reasonable design basis, since under base load conditions, condensers are designed for higher absolute pressures and higher water temperatures and only experience operating pressures of 1” Hg Abs. when cooling water reaches low temperatures as shown in Figure #2; at 100% load and 48°F cooling water by interpolation, condenser vacuum is 1” Hg Abs. Since 48°F is well below 60°F, pump capacity will be greater than HEI Standards.

At part load condenser operation, however, capacity falls off rapidly. The reasons for this capacity loss are inherent in the characteristics of liquid ring vacuum pumps (see below).

As indicated previously, the seal liquid in the pump absorbs the heat generated by compression and friction plus the heat produced by the condensation of water vapor entering with the air exhausted from the condenser.

The amount of seal water used by a given pump will vary with the pump size, number of stages, and the pump efficiency. Hence, the temperature rise of the seal within the pump will vary. The example, therefore, set forth below should be considered more qualitative than quantitative, but will illustrate the situation effectively.

In a recirculated seal system, the recirculated seal water is cooled by means of a heat exchanger, usually with the same water that cools the steam condenser. The recirculated water is, therefore, warmer than the entering condenser cooling water. For this example we will assume 5°F, although it may often exceed this approach.

For the pump considered:

\[
\text{BH P} = 42 \quad \text{Seal Flow} = 40 \text{ GPM} \\
\text{Seal Temperature entering the Pump} = 80°F \quad (75°F + 5°F)
\]

Heat absorbed is equal to:

\[
\text{Pump BH P} \times 2544 + \text{lbs./hr. water vapor condensed in the liquid ring} \times \text{the enthalpy of condensation} = 42 \times 2544 + 1002 (74.4-3.4) = 177,990 \text{ BTU/hr.} \quad \text{Eq. (3)}
\]

Seal water rise is:

\[
\frac{177,990 \text{ BTU/hr.}}{40 \text{ GPM} \times 8.3 \text{ lbs./gal.} \times 60 \text{ min./hr.}} = 8.9°F \quad \text{Eq.(4)}
\]

Thus, the effective temperature of the seal water in the pump is 13.9°F above the inlet temperature of the condenser cooling water. The lowest pressure which can be achieved in a liquid ring pump is slightly higher than the seal fluid equilibrium saturation pressure. For the purpose of this example, we will assume the sat-
Pump seal water rise will be constant and will not vary with steam condenser load. Cooling water rise in the steam condenser is not constant and will vary with load.

Thus, for an effective seal temperature of 88.9°F.

Pump operating pressure will be $1.373"$ Hg Abs. (equivalent pressure to 88.9°F.), plus $0.2"$ Hg Abs. = $1.373 + 0.2 = 1.573"$ Hg Abs. Eq. (5)

Pump seal water rise will be constant and will not vary.

Consider two possibilities:
1. Condenser selected for 20°F. cooling water rise.
2. Condenser selected for 30°F. cooling water rise.

At full load, lowest achievable condenser pressure is, for the case considered above, the saturation pressure equivalent to the leaving cooling water temperature $+5$ F. (minimum approach to cooling water exit temperature per HEI Standard) =

1. $75$ F. $+ 20$ F. $+ 5$ F. $= 100$ F. - equivalent to $1.933"$ Eq. (6)
2. $75$ F. $+ 30$ F. $+ 5$ F. $= 110$ F. - equivalent to $2.596"$ Eq. (7)

At 50% load, calculations yield:
1. $1.422"$ Hg Abs. (pump governs condenser pressure) Eq. (8)
2. $1.660"$ Hg Abs. Eq. (9)

At 30% load, calculations yield:
1. $1.253"$ Hg Abs. (pump governs condenser pressure) Eq. (10)
2. $1.378"$ Hg Abs. Eq. (11)

Thus, we see that at reduced condenser loads, the pump limits the vacuum that can be achieved by the condenser.

Referring back to Figure #1, we are in the area of the curve where the air pump is controlling back pressure. Subcooling will result, air will accumulate in the condenser until a balance is achieved and dissolved oxygen will increase.

Actually, the situation deteriorates more rapidly as we approach 30% load. As we have seen, air leakage increases, therefore, the pump is called upon to handle more air and more water vapor. Even worse, to maintain 7 ppb dissolved O₂, we should carry over (see Table #1) 17.35 lbs. Wv/lb². DA resulting in 15,180 BTU/lb. DA of additional heat absorbed by the same GPM of seal water.

Let us consider what happens when power plants encounter low cooling water temperatures while operating at or near full load.

Consider our condenser, see Figure #2, now operating at 100% load and 40°F. cooling water. The condenser is capable of achieving $.85"$ Hg Abs.

On the basis of a recirculated seal water, temperature rise will be (assuming no increase in air leakage) the same 8.9°F.

Effective seal water temperature $= 8.9$ F. $+ 5$ F. $+ 40$ F. $= 53.9 = .418"$ Hg Abs.

To this we must add $.2"$ Hg Abs. (as above). Total pressure achievable in the pump is $.618"$ Hg Abs.

This is better than the $.85"$ Hg Abs. that the condenser can achieve and the liquid ring vacuum pump is adequate. The liquid ring pump was picked for 7.5 SCFM, 19 ITD. Referring to the liquid ring vacuum pump performance curve, Figure #4, capacity at $.85"$ Hg Abs. and 19 ITD is 7.0 SCFM, dry air plus water vapor to saturate - 7.50 F. subcooling. Actual ITD $= 74.13$ (temperature equivalent to saturation pressure at $.85"$) minus inlet cooling water temperature (40 F.) $= 34.13$ F. At 100% load and 40 F., the pump will be very adequate for the specified conditions. Should air inleakage increase or load conditions decrease, the situation will deteriorate.

Although the seal water temperature rise in the pumps may not limit condenser vacuum under “normal” HEI conditions, this rise does limit the capability of the venting system to remove additional water vapor so as to minimize dissolved oxygen in the condensate only when condenser cooling water temperature is high and high absolute pressures are maintained within the condenser is the liquid ring pump able to absorb “extra” water vapor, and thus assist in preventing oxygen absorption.

The situation described above is often aggravated by installation practices at many power plants.
1. Load is 17.25 lbs. Water vapor per lb. of dry air at 0.6" Hg Abs. 
   Motive is 21 lbs. of 150 lb. steam per lb. of dry air.

2. Load to new inter-condenser is 38.35 lbs. of water vapor per lb. 
   of dry air.

3. Air handling capacity of existing venting equipment, at new operating 
   pressure of 2" Hg Abs., is about 310% of design for liquid ring vacuum 
   pump systems and 130% of design for steam jet ejector systems.
Figure #6 illustrates three different liquid ring vacuum pump seal arrangements. Comparable achievable vacuums for the conditions considered above are (assuming 5 F. approach and cooling water temperature in each case) equal to the entering cooling water to the main condenser.

**Once Through System:**

1. \( 75 \text{ F.} + 8.9 \text{ F.} = 83.9 \text{ F.} = 1.17'' \text{ Hg Abs.} + .2'' \text{ Hg Abs.} = 1.37'' \text{ Hg Abs.} \)  
   \[ \text{Eq. (12)} \]

**Simple Recirculation System**

2. \( 75 \text{ F.} + 5 \text{ F.} + 8.9 \text{ F.} = 88.9 \text{ F.} = 1.37'' \text{ Hg Abs.} + .2'' \text{ Hg Abs.} = 1.57'' \text{ Hg Abs.} \)  
   \[ \text{Eq. (13)} \]

**CCW Recirculation System**

3. \( 75 \text{ F.} + 10 \text{ F.} + 8.9 \text{ F.} = 93.9 \text{ F.} = 1.60'' \text{ Hg Abs.} + .2'' \text{ Hg Abs.} = 1.80'' \text{ Hg Abs.} \)  
   \[ \text{Eq. (14)} \]

Few, if any, power plants use the once through system, due primarily to corrosion and contamination problems.

System (2) is a Simple Recirculation Seal System (most common), but seal water must be warmer than the cooling water, at least 5 F., and achievable liquid ring vacuum pump vacuum is reduced.

System (3) is a double indirect system. It has been a recent trend, particularly where salt or brackish water is used for the steam condenser, to centrally cool all service water, except that going to the steam condenser in a closed cooling water exchanger (CCW). This procedure minimizes fouling and simplifies auxiliary equipment heat exchanger metallurgy. Unfortunately, it imposes a double penalty on the vacuum pump, a condition seldom contemplated during air pump design stage.

We have seen that venting equipment can be inadequate for many reasons.

Proper equipment location and due consideration of pressure drop can alleviate some of these capacity losses.

Elimination of double indirect cooling systems, if it is not possible to make proper allowance for them in the system design, can help liquid ring vacuum pumps.

None of these steps will solve the inadequacy of some existing venting systems at part load condenser operation.

To solve this problem, EPRI NP-2294 (Reference 5) recommends the installation of a refrigeration system between the main steam condenser and the existing vacuum system (Figure #7).

This arrangement goes a long way toward alleviating the problem.

The refrigerated condenser, between the existing steam condenser and the existing venting equipment, acts to reduce the water vapor imposed by Henry's Law considerations and impaired air cooling section capability, so that this can be reduced to approximately 2.2 lbs. Wv/lb. DA, . . . the same ratio as under "normal" 1'' Hg Abs., H E I conditions for which most venting equipment is designed by appropriate pre-cooling.

This arrangement, however, does not address all of the problems we have discussed above. We have seen that at low load:

1. Air leakage increases.
2. Condenser pressure drops below 1'' Hg Abs.
   a. Liquid ring vacuum pumps lose capacity,
      1) due to high seal temperature
      2) due to reduced capacity at low pressure.
   b. Ejectors lose capacity due to reduced capability at low absolute pressure
   c. Pressure drop between condenser and venting equipment increases by a factor of at least 2.

Thus, the existing venting equipment may still be inadequate. A much simpler arrangement, which treats the entire problem, is shown in Figure #8.

A new booster and intercondenser are interposed between the existing condenser and its venting equipment.

The booster(s) is sized to:

1. Operate at minimum anticipated condenser pressure at no load/low load conditions.
2. Remove large water vapor/dry air ratios required to limit dissolved oxygen to 7 ppb, based on air leakage paragraph 3.
3. Remove double the tabulated H E I air leakage.
4. Compensate for increased pressure drop between condenser and venting equipment created by larger flow rate and volume.

5. Compress to an intercondenser pressure which permits existing venting equipment to:
   a. Handle double air leakage
   b. Operate with higher seal temperatures (liquid ring vacuum pump).
   c. Recover a portion of the heat normally rejected to condenser cooling water.

Let's look at some figures:

At .6" Hg Abs. - To attain 7 ppb dissolved O₂, each pound of air must contain 17.35 lb. Wv (Table #1). The refrigeration system reduces this to 2.2 lbs. Wv/lb. DA, but vacuum is still at .6" Hg Abs. and as seen in Figure #8, existing ejector capacity is reduced to 55% of that at 1" Hg Abs.

As we have seen, seal water rise in the liquid ring vacuum pump exceeds that in the steam condenser and at part load, the existing liquid ring vacuum pump could not attain 0.6" Hg Abs.

The booster arrangement, Figure #8, illustrates that by compressing with the steam ejector booster to a new higher interstage pressure, the capacity of the existing venting equipment increases. An existing liquid ring vacuum pump will handle 310% of the air load it could handle at 1" Hg Abs. The existing ejectors will only handle a 130% increase in air load (plus, in both cases, additional water vapor). Existing ejector systems are usually twin element, two stage, each element good for 100% capacity, therefore, to handle double the air load at part load conditions, both existing ejector elements would have to be turned on.

Since we have increased the operating pressure of the existing venting equipment, obviously we can now tolerate a higher seal water temperature (saturation temperature is 101.14 F. versus 79 F.) in the liquid ring pump.

Heat rejected to the new intercondenser, cooled by condensate from the main steam condenser, can be returned to the main steam condenser at very low loads to help deaerate the condensed liquid; at high loads, it can be fed forward to the boiler feed system, recovering a portion of the energy required to operate the steam ejector booster, as well as the heat contained in the substantial saturation component of the air vapor mixture.

The booster system of Figure #8 was subsequently suggested as an alternate to the refrigeration system at the Fossil Plant Cycling Workshop. (8)

A combined hybrid liquid ring vacuum pump/steam ejector system is shown in Figure #9. This unit is completely automated and incorporates the best abilities of both ejectors and liquid ring vacuum pumps. When load conditions warrant, the booster ejector can be shut off and only the liquid ring vacuum pump is run.

Systems of this type can track the condenser at all loads, at all anticipated pressures, while maintaining Wv/DA ratios required to limit dissolved O₂ to 7 ppb.

Since air leakage will vary from plant to plant, and from time to time within a plant, and will vary with condenser load, it is not possible to optimize the equipment size actually required. It is suggested that multiple part size parallel steam ejector boosters be installed, each with isolating suction valves, so that they may be turned on or off with load changes. A single condenser designed to accommodate the total steam ejector booster effluent, and cooled by full condenser condensate flow should be installed.

Installation of a hybrid system designed to accommodate these greatly increased loads will be useless unless condenser air removal piping is modified to accommodate the vastly increased volume. It may be necessary to internally modify condenser air cooling section vent lines, and, of course, in new units the air cooling section itself may need some modification.

Inadequate venting contributes to balance of plant deterioration. Referring back to Figure #1: at any time that the venting equipment limits condenser vacuum, air will accumulate, air partial pressure will increase, and condensate O₂ content will rise, leading to additional use of hydrazine for scavenging.

The consequences are varied - hydrazine tends to break down to form ammonia, which concentrates in the air cooling section of condensers and causes corrosive failure of copper alloy tubing. Often this results in cooling water inleakage with consequent corrosion of downstream components, including denting in PWR steam generators.

In the absence of copper alloys, the effect of high oxygen content in the condensate may be reduced. Other causes of cooling water leakage are also present and can introduce contaminants, which in combination with oxygen, can be damaging.

Hydrazine effectiveness in oxygen scavenging is a function of time and temperature. As a result, hydrazine scavenging is not generally capable by preventing oxygen attack in components between the condensate and feed pumps.

In power plants equipped with deaerating heaters, equipment downstream of this heater is protected against O₂, and because at elevated temperatures hydrazine scavenging is effective. Maintaining low O₂ content in condenser condensate will protect equipment in the low temperature end.

A thorough discussion of dissolved oxygen problems and the desirability of its elimination is contained in references (5) and (1).

The use of a steam ejector booster/condenser combination will improve the capabilities of all existing systems and enable them to cope with two-shift and cycling conditions. For systems using ejectors, it will be necessary, if you need to double air capacity, to utilize the standby ejector system. For systems combining steam
ejector boosters with vacuum pumps, the ability to operate liquid ring vacuum pumps at the new higher interstage pressure will solve the problem.

When venting equipment is adequate for all load conditions, an atmosphere is maintained within the condenser which limits oxygen absorption. The condenser, at all times, operates at the lowest possible absolute pressure.

Existing venting equipment, particularly under part load conditions, is often inadequate. Modification can be readily made to enhance capacity. Precaution should be taken not to limit capacity by faulty installation. Equipment must be designed with sufficient capacity at the lowest operating pressure in order to assure low dissolved oxygen and improved heat rate. Simply turning on additional ejectors or liquid ring vacuum pumps without the other necessary modifications described above, cannot be relied upon to solve the problem.

REFERENCES


**SI UNITS CONVERSION CHART**

<table>
<thead>
<tr>
<th>From</th>
<th>To</th>
<th>Action</th>
</tr>
</thead>
<tbody>
<tr>
<td>F.</td>
<td>C.</td>
<td>Subtract 32, Divide by 1.8 (F-32)/1.8</td>
</tr>
<tr>
<td>in.abs.</td>
<td>mm</td>
<td>Multiply by 25.4</td>
</tr>
<tr>
<td>lb. H₂O/lb.</td>
<td>air kg H₂O/kg air</td>
<td>No conversion necessary</td>
</tr>
<tr>
<td>F. ITD</td>
<td>C. ITD₃</td>
<td>Divide by 1.8</td>
</tr>
<tr>
<td>SCFM</td>
<td>Std. M³/h</td>
<td>Multiply by 1.695</td>
</tr>
<tr>
<td>lb.</td>
<td>kb</td>
<td>Divide by 2.20462</td>
</tr>
<tr>
<td>HP</td>
<td>kw</td>
<td>Multiply by .74571</td>
</tr>
<tr>
<td>GPM</td>
<td>M³/h</td>
<td>Multiply by .227</td>
</tr>
<tr>
<td>BTU/hr.</td>
<td>W</td>
<td>Multiply by .293</td>
</tr>
<tr>
<td>BTU/lb.</td>
<td>kj/kg</td>
<td>Multiply by 2.326</td>
</tr>
</tbody>
</table>