Improved Steam Condenser Gas Removal System

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ABSTRACT

The Power Industry has traditionally focused on base load conditions. In the past, as utilities were operated at or near base load, part load operation was neglected. Most modern day plants, however, can expect frequent operation under part load conditions.

These events have led to a new awareness of the need for adequate venting at off design conditions. EPRI (Electric Power Research Institute) studies have identified the inter-relationship between adequate venting and good condenser performance. This paper will describe the heat transfer effects and interrelationship between venting systems and steam surface condensers, the characteristics of most common venting systems, and a recent adaptation, Hybrid Venting, which combines a steam ejector with a condensate or water cooled intercondenser backed by a liquid ring vacuum pump.

Hybrid systems offer several advantages over existing systems, which are detailed herein. Retrofit of existing liquid ring vacuum pump systems is also discussed.

EFFECT OF NON-CONDENSIBLES ON CONDENSER PERFORMANCE

The importance of proper condenser venting on steam surface condenser performance is not fully appreciated in the power industry. The effect at times is quite profound.

Heat transfer engineers have long been aware of the fact that as vapor is condensed in the presence of non-condensible gases, the condensed liquid film is no longer the primary resistance to heat transfer. The heat transfer surface becomes surrounded by a gas film which offers a substantially higher resistance to heat transfer. Moreover, the resistance to heat transfer increases in portions of the condenser as the gas vapor mixture is subcooled, and the non-condensible to vapor ratio increases. This happens in all power plant steam surface condensers operating under a vacuum. Whereas the entering steam may contain 1 lb (kg) of air in 10,000-30,000 lbs (kgs) of steam, the leaving mixture (based upon the required HEI minimum of 7.5 deg F (4.2 deg C) subcooling below saturation) is approximately 1 lb (kg) air and 2.2 lbs (kgs) of water vapor … approximately 30% noncondensibles.

Thus the heat transfer coefficient in the portion of the tube bundle adjacent to the air outlet will be substantially lower than the rate experienced in the main condenser bundle.

Fortunately, the percentage of non-condensibles present at the inlet is quite small, and the heat removed in the air cooling section will also be quite small. Still, the heat transfer surface required for this portion of the surface condenser can be as high as 8%.

The internal design of steam condenser tube bundles differ substantially with the manufacturer. Some use baffles, some shrouds for the air cooling section … all generally extract the non-condensibles in the portion of the condenser cooled by the coldest water to take advantage of the greatest temperature difference, subcooling the leaving vapors and reducing the vapor load to the venting system.

In reality, non-condensible concentration and gas/vapor subcooling starts to take place immediately as the steam penetrates the bundle. The effect, however, is negligible until non-condensible percentages rise and begin to affect the heat transfer coefficient. This effect becomes more pronounced as these gases approach the air removal duct, reaching a maximum at the outlet as described above. We can see, therefore, that the “air cooling section” really has no boundary, but increases or decreases, depending upon air leakage and venting system capacity.

If there is a severe imbalance between the venting system removal capability and the air leaking into the system, then the “air cooling” section will expand. If the temperature rise in each tube or group of tubes surrounding the air removal suction is measured, it will indicate little or no increase, while the tubes on the bundle periphery will show a substantial increase. This effect has been referred to in the literature as a gas “bubble” and is often first diagnosed as apparent tube side fouling. The explanation is actually that a larger portion of the condenser must now be devoted to “air cooling,” less surface is available for condensing, and condenser pressure rises. Evaluation of the condenser performance appears to indicate tubes are fouled, but cleaning the water side does not help.

The solution to this problem lies primarily in correcting the venting system deficiency described above. This may be due to the venting system characteristic, seal water temperature of liquid ring vacuum pumps for instance, excess air leakage, capacity drop off as condenser pressure decreases, or any combination of the above.

Improper tube bundle design causing excessive gas/vapor pressure drops can aggravate the situation. The use of heavy bundle densities, gas baffles, vapor by-passing, small vent lines, all increase the potential for this problem.

Good condenser design permits steam and air to readily penetrate the tube bundle with minimum pressure drop. Attempts to increase non-condensible velocities to improve heat transfer in the air cooling section or devices to control the vent rate in a particular section of the condenser, (proportional venting) impose an artificial pressure drop, thus reducing venting capacity. This can only be accommodated by larger venting equipment or better bundle design.

It is important to understand the phenomenon described above. Although the explanation is obvious, power plant engineers and heat transfer engineers tend to overlook the very common problem of venting system inadequacy and treat the symptoms, a low heat transfer rate apparently due to fouling, rather than the real cause ... inadequate venting.

VENTING EQUIPMENT

To undertake a complete evaluation of venting equipment, it is important that we consider the operation of the entire vacuum system. Although the phrase vacuum system is commonly used to describe the venting equipment, such terminology is misleading.

The function of the venting equipment is not to produce the vacuum, but rather to maintain the vacuum achieved by the steam surface condenser.

The steam surface condenser accepts a large volume of steam from the turbine exhaust. As the steam condenses on the cold condenser tubes the volume "collapses." It is this change of volume which results in the creation of the vacuum. Generally speaking, the greater the vacuum (lower condensing temperature), the more work will be done by the turbine, resulting in increased plant power output.

It is the function of the venting equipment to remove any non-condensible gas in-leakage which may enter the system through gaskets, packing, loose connections, or other sources. If the venting equipment is properly sized, the condenser vacuum will be set by cooling water temperature and heat transfer rate

If however, the venting equipment is not adequate, the condenser pressure (and turbine back pressure) will rise. The efficiency of the turbine and the amount of power generated will be reduced. In this situation, the venting equipment is limiting the condenser vacuum.

Whenever venting equipment is inadequate, it will reduce the plant's available generating capacity. Piskorowski et al. (1987) estimated that excess fuel cost of $600,000 per year occurred due to inadequate venting.

The Heat Exchange Institute Inc., 1984, Standards for Steam Surface Condensers, recommends that venting equipment for a Power Plant Turbine be designed for 1.0 in. HgA (3.39 kPa), or the condenser design pressure, whichever is lower.

Table 5 of the HEI Standards tabulate Venting Equipment Capacities for various condenser sizes, based on steam flow, number of condenser shells, and number of exhaust openings. Paragraph 4.1 (c) states, "It is recommended that the capacity of venting equipment be not less than the values shown in Table 5 to insure adequate removal capacity under commercial operating conditions."

Figure #1 (Harrington et al. 1971) demonstrates visually that if the venting system - regardless of type - is unable for any reason to remove the noncondensibles at the pressure which could be achieved by the condenser, condenser back pressure will rise and is controlled by the capacity of the venting equipment. Harrington also states, "When the air pump controls the back pressure, dissolved oxygen levels can be expected to rise."

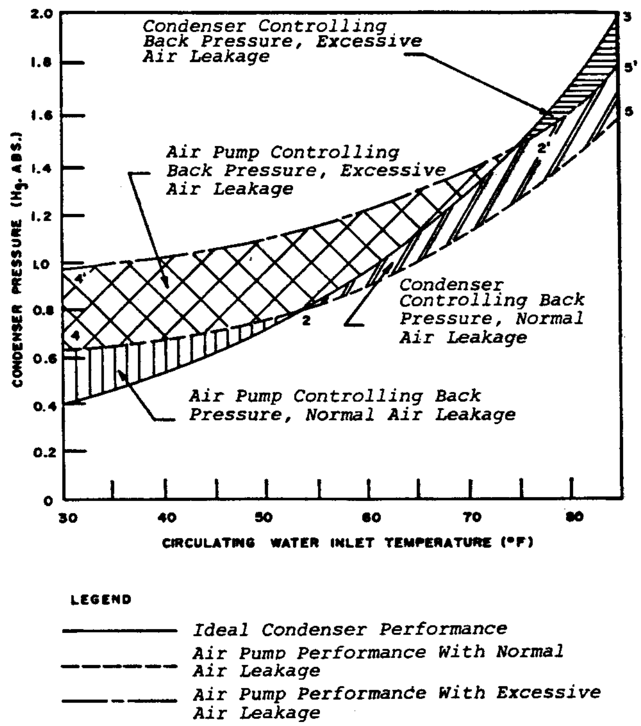

Fig. #1: Effect of Air Leakage on Condenser Pressure

The subject of enhanced venting/dissolved oxygen control has been covered by previous work, (Spencer and Impagliazzo 1984, and Athey and Spencer 1987), and will not be detailed herein. Suffice it to say, condensate dissolved oxygen can be controlled if the venting system is sized to remove sufficient water vapor such that the noncondensible partial pressure will be low enough to limit the amount of oxygen dissolved. (Henry's Law)

When power plants operate at base load, steam surface condensers typically operate at 2-3 in. HgA (6.8 - 10.2 kPa) or higher. At these pressures, venting equipment capacity will exceed the HEI tabulated values, which are set at 1 in. HgA (3.39 kPa). Note that the tabulated leakage rates specified by HEI do not take into consideration limitations of dissolved oxygen. HEl specifies that to achieve 7 ppb (.005 cc/L) of dissolved oxygen, leakage must be limited to 15%-25% of these values.

All venting systems gain in capacity to varying degrees as absolute pressure increases. However each of these systems lose capacity and behave quite differently when required to operate at lower absolute pressures. (See Table 2)

Recent trends to cycling, part loading, and two shifting, with concurrent cold condenser cooling water, allow the condenser to operate at lower absolute pressures. This can only be achieved if the venting equipment can accommodate the vacuum. (Of course if the turbine design limits the absolute pressure, no further gain in heat rate can be expected and it may not be desirable to lower condenser pressure below this point.) As the condenser operating pressure decreases, more of the heat cycle is exposed to vacuum and air in-leakage usually increases. Simultaneously all venting equipment capacity decreases, as the vacuum levels they are required to maintain increase.

Existing power plant condensers use a variety of venting devices, the most common of which are set forth below:

1) Two Stage Steam Ejectors

2) Two Stage Liquid Ring Vacuum Pumps (LRVP's)

3) Single Stage LRVP with Air Operated Ejector

4) Mechanical Blowers

5) Hybrid Steam Ejector/LRVP Systems

DESCRIPTION OF VENTING SYSTEMS

Two Stage Steam Ejectors

Air and water vapor are removed from the main steam condenser, enter the 1st stage ejector and are compressed to the interstage pressure by means of the high pressure motive steam. The load and motive steam are discharged to the inter condenser and a portion of the water vapor load and motive steam are condensed by means of cooling water or condensate from the main condenser. Non-condensibles and associated water vapor are removed from the inter condenser by the 2nd stage ejector, compressed to atmospheric pressure and are discharged through the after condenser. (Fig. #2)

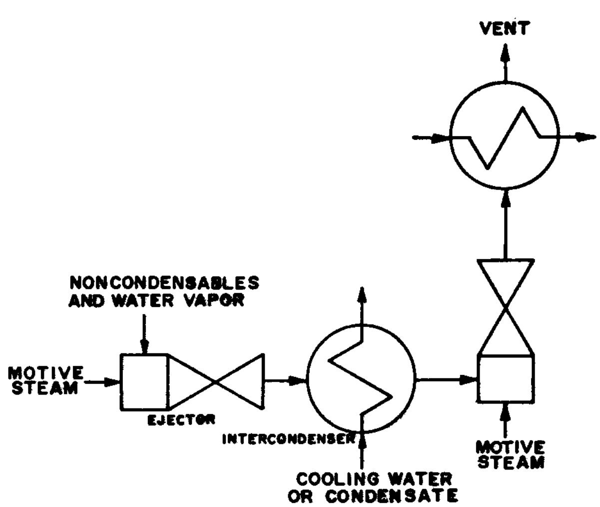

Fig. #2: Two Stage Ejector System

Two stage condensing ejector systems can be designed to operate at any condenser pressure and designs are not limited by the available cooling water temperature to the intercondenser (condensate cooled systems are common). These systems have no moving parts, are the most reliable, require the least maintenance of all venting systems, and are the least expensive in initial cost.

Two stage ejector systems require a reliable steam source, generally 100-150 psig (690-1034 kPa) steam is used. Once equipment is built for a given motive steam pressure that pressure must be maintained or the ejector will become unstable and lose vacuum.

TWO STAGE LIQUID RING VACUUM PUMPS

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 the seal liquid, usually water. (Fig. #3) 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.

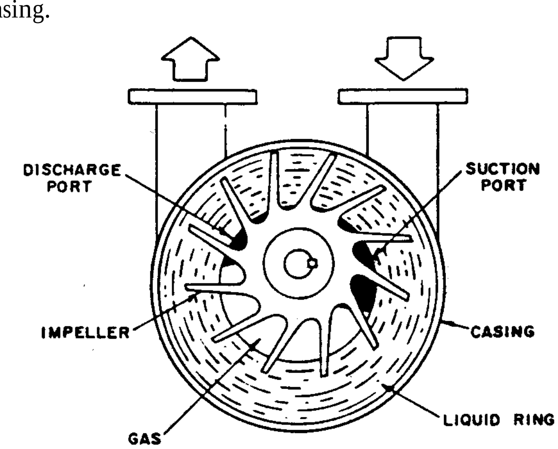

Fig. #3: Vacuum Pump Cross Section

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. This leaves the cell available to receive air or gas as it is presented to the inlet port on the next revolution. A small portion of the seal water is discharged with the vapor, and a constant supply of fresh seal must be maintained (30-50 gal/mm (6.8-11.4 m3/h)).

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 condenses water vapor entering with the gas. For condenser exhaust service, a closed loop (or total recirculation) seal system is commonly used. (See Fig. #4) The seal water temperature will be 3-5 deg F (1.7-2.8 deg C) warmer than the cooling water to the pump heat exchanger, which is normally taken from the same source as the condenser cooling water.

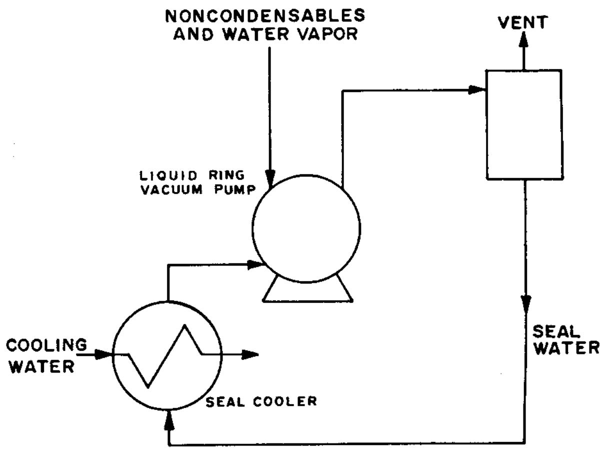

Fig. #4: Liquid Ring Vacuum Pump System

The vacuum attainable by a liquid ring vacuum pump is limited by the vapor pressure of the seal fluid. As the operating vacuum approaches the vapor pressure of the seal, more and more of the seal fluid will “flash” a into vapor. The capacity of the liquid ring vacuum pump is reduced as more of the impeller space is occupied by vapor from the seal fluid, leaving less space available to accept the incoming load. If allowed to continue, cavitation will occur inside the pump, resulting damage to internal surfaces, and preventing the pump from achieving greater vacuum levels.

To prevent cavitation, the operating vacuum of the liquid ring vacuum pump must be limited to approximately .25 in. Hg (.85 kpa) above the vapor pressure of the seal liquid.

Example, with 70 deg F (21.1 deg C) seal water, the vapor pressure is .74 in. Hg (2.5 kPa) thus the maximum vacuum obtainable is approximately .99 in. HgA (3.4 kpa), (.74 + .25 = .99 in.) regardless of pump size.

This differential varies somewhat according to pump model and the amount of heat added to the seal water by condensible load, however the importance of vacuum limitations set by seal water temperature cannot be overemphasized.

Even if the pump is oversized, it cannot obtain lower pressure levels than that permitted by the vapor pressure of the seal water. This limitation is a major disadvantage when the condenser is operated under part load conditions, as will be discussed later.

Liquid ring vacuum pumps are normally supplied completely packaged with controls for fully automatic operation. No separate hogging device for initial evacuation is required, as the pump gains capacity rapidly at higher suction pressures. Initial cost is high, but operating cost is normally lower than a two-stage ejector system.

SINGLE STAGE LRVP WITH AIR OPERATED EJECTOR

This device uses atmospheric air as the motive fluid for the ejector. The vacuum pump must handle the air leakage load as well as the substantial amount of additional air which enters as motive. This system is not very efficient, and since the advent of two stage liquid ring vacuum pumps, use has been limited. (Fig. #5).

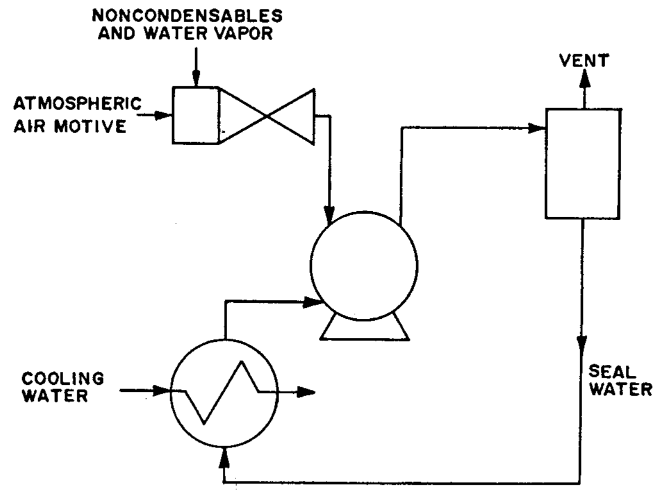

Fig. #5: Liquid Ring Vacuum Pump
With Air Operated Ejector

With an air ejector, the liquid ring vacuum pump operates at a higher interstage pressure than a simple liquid ring vacuum pump, thus these devices are less affected by the vapor pressure of the seal liquid. However, the large volume of air reaching the liquid ring vacuum pump requires large expensive machines. Energy requirements are high, maintenance is comparable to a standard liquid ring vacuum pump system.

COMPRESSORS AND MECHANICAL BLOWERS

The use of these devices for condenser exhaust service has decreased as maintenance requirements have caused operating problems with those systems. These devices are not limited by cooling water temperature. Energy use is lowest, however original cost and maintenance tend to be expensive.

HYBRID SYSTEM

This paper describes a Hybrid system which entails combining a first stage steam ejector discharging into a surface intercondenser, which in turn is coupled to a liquid ring vacuum pump. (Fig. #6)

The vacuum pump thus operates at a higher interstage pressure. Three distinct advantages are gained:

1) The pump suction pressure is now much higher than the vapor pressure of the seal water. Thus the seal temperature will no longer limit the suction pressure.
2) The major portion of the motive steam is condensed in the intercondenser, leaving only a small portion to enter the vacuum pump.
3) The volume of air and water vapor to be handled by the pump is reduced at the higher interstage pressure, resulting in a much smaller pump. (If a 100 NP (75 kW) pump is required at 1.0 in. HgA (3.39 kPa) 20 HP (15 kW) may be sufficient in a hybrid system.)

This hybrid system offers distinct advantages over the previously described systems, and can be readily designed to operate with existing liquid ring vacuum pump systems.

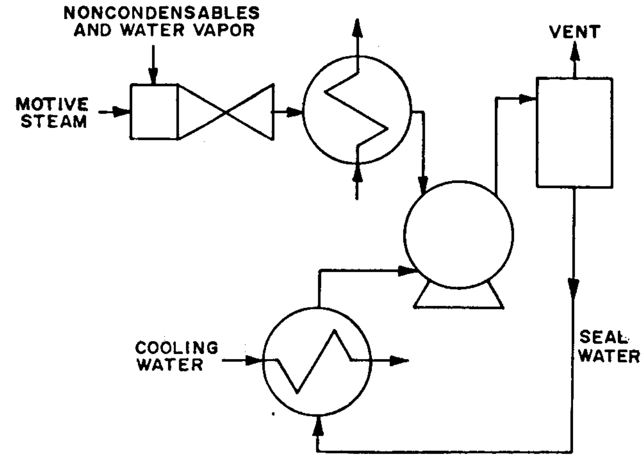

Fig. #6: Hybrid System

SIZE COMPARISON

Recommended capacities for venting equipment are given in Table 5 of the Heat Exchange Institute's "Standards for Steam Surface Condensers."

For comparison purposes, we will look at a two-stage steam ejector, liquid ring vacuum pump, and Hybrid Unit all sized per HEI design, as well as a Hybrid unit sized for double the HEI leakage.

For our example, let us consider a single shell condenser with one steam opening designed for 3,500,000 lb/hr (1,587,600 kg/h) steam at 3.0 in. HgA (10.2 kPa) with 90 deg F (32.2 deg C) cooling water.

The HEI tabulated value required is 20 Standard ft 3/min (34 m3/h) air plus the associated water vapor to saturate at 1.0 in. HgA (3.39 kPa) and 71.5 deg F (21.9 deg C). (90 lb/hr (40.8 kg/h) air plus 198 lb/hr (89.8 kg/h) water vapor).

Table #1 includes a comparison of venting equipment based on this example.

For comparison of utility requirements, Table 1 includes the total steam equivalent for each system. Although this provides a useful comparison, the important fact is that the total utility requirement for each of the venting systems is much less than one-tenth of one percent of the total plant steam.

PART LOAD OPERATION

Liquid ring vacuum pumps are sized based on the assumption that the initial temperature difference (lTD), for a given condenser, is a fixed number. (lTD is the difference between the saturation temperature at the condenser pressure, and the inlet cooling water temperature.) As the lTD increases, vacuum pump capacity will increase, (See Fig. #7).

In our example, with a condenser pressure of 3.0 in. HgA (10.2 kPa), the corresponding saturation temperature is 115 deg F, (46.1 deg C). With 90 deg F (32.2 deg C) cooling water, the lTD is thus 25 deg F (13.9 deg C) (115 -90 = 25 deg F).

Once we have calculated the ITD, the vacuum pump can be chosen from the pump performance curve, (Fig. #7). A large lTD will allow use of a small vacuum pump, a small lTD will require a larger model.

Table #1 SIZE COMPARISON OF VENTING EQUIPMENT

	LB/Hr Motive Steam @ 100 PSIG	BHP	Motor HP	* Total Steam Equiv Lb/Hr	** % of Total Plant Steam
2 Stg Ejector	1050	0	0	1050	.030%
LRVP	0	72	100	565	.016%
Hybrid System	550	15	20	668	.019%
Hybrid for 2 x HEI	1100	30	40	1336	.038%

* Based on an assumed plant steam rate of 10 lb/hr steam/kW and assumed motor efficiency of 95%.

** % of total plant steam (3,500,000 lb/hr).

SI Conversions: 1 psi = 6.895 kPa
1 HP = .746 kW
1 lb/hr = .4536 kg/h

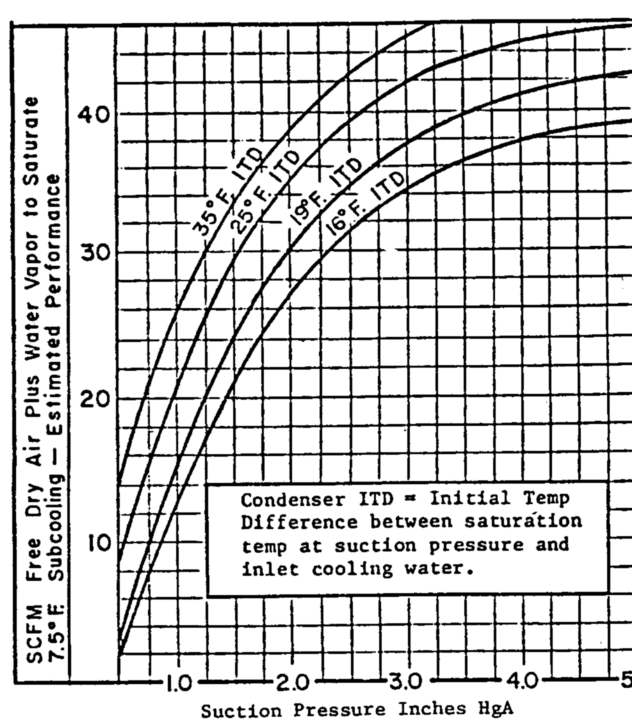

Fig. #7: Vacuum Pump Performance Curve

Although present HEI Standards do not discuss the sizing of liquid ring vacuum pumps, this method, based on the condenser lTD, is used by all liquid ring vacuum pump manufacturers for condenser exhaust service.

Liquid ring vacuum pump selection is based on the condenser ITD, which is assumed constant. This assumption of a fixed lTD is acceptable when the condenser is operated under full steam load conditions. Hybrid systems, however, are designed for the maximum anticipated cooling water temperature.

But what happens when this same condenser is operated under part load?

Table 2 lists predicted condenser pressures and venting equipment capabilities for various steam loads and cooling water temperatures for the condenser in our previous example. As can be seen in Table 2, the condenser ITD decreases rapidly under part load conditions.

The reduced ITD causes a severe reduction in the vacuum pump capacity, especially when the condenser is operated at 50% load or below.

Under full load conditions, the condenser in our example is capable of operation at 1.97 in. HgA (6.67 kPa), with 75 deg F (23.9 deg C) water, condenser ITD is 25.6 deg F, (14.2 deg C) and vacuum pump capacity is well above design. But at 50% load with 75 deg F (23.9 deg C) cooling water, from Table #2, the same condenser can operate at 1.33 in. HgA (4.5 kPa). The ITD is now 12.9 deg F (7.2 deg C) and vacuum pump capacity falls to 75% of design. The hybrid system is not dependent on the ITD. Capacity is 100% for either condition.

If air leakage is at design, the vacuum pump will operate at approximately 1.75 in. HgA (5.9 kPa) (Fig. #7). The vacuum pump will limit the condenser operating pressure and the condenser will operate at the elevated pressure. The difference (1.75 - 1.33 = .42 in. Hg or 1.42 kPa) may require $70,000 extra fuel cost if the condition occurs for 3 months, (prorating the example given by Piskorowski et al. (1987)).

This operational limitation, under part load conditions, is a major disadvantage of a liquid ring vacuum pump for condenser exhaust service.

Many power plant operators are not aware that the venting equipment may be limiting the condenser under part load conditions. Only reference to the condenser performance curve will reveal the fact that the operating pressure is above the pressure the condenser is capable of robbing the plant of available energy.

SUMMARY

The rational for minimizing venting equipment size is to save parasitic power consumption and initial capital cost. However, the penalty for inadequate venting can result in much greater power loss in the plant heat rate, and higher condensate dissolved oxygen levels.

Hybrid systems designed to HEI Standards, reduce initial cost, increase useful energy, and respond well to lower condenser pressure operation.

It is relatively easy to retrofit a steam ejector and intercondenser to an existing vacuum pump installation. Such retrofits can double or even quadruple existing venting capacity while lowering the suction pressure that can be achieved. Since the existing liquid ring vacuum pump capacity will increase substantially at the now higher interstage pressure, some existing equipment can be relegated to stand-by or hogging service. Retrofit systems permit the first stage ejector to be shut down when the condenser operates at higher absolute pressures, or when reduced air leakage permits.

When anticipated condenser operating conditions indicate that operating pressures will be lower then normal for any of the reasons cited above, hybrid systems can be designed at the outset to accommodate these problems.

The authors find it difficult to understand the obsessive desire to "undersize" venting equipment. The energy consumption of all venting systems is minuscule when compared to the total quantity of energy involved in the equipment serviced by these venting systems.

Certainly the improvement in heat transfer rate in the air cooling section and within the condenser, producing even a slight improvement in vacuum, will more than compensate for the minute increase in the venting equipment energy requirements.

Inadequate venting can cause power loss and an increase in condensate dissolved oxygen. Conversely, "over venting" can produce power gain and limit condensate dissolved oxygen.

Hybrid systems are more versatile than conventional venting devices, and can be readily retrofitted to existing liquid ring vacuum pump installations to remedy problems and enhance plant performance.

Table #2 PREDICTED CONDENSER PRESSURES AND VENTING EQUIPMENT CAPACITIES

STEAM LOAD (%)	COOLING WATER TEMP (°F)	EST COND. PRESS (In HgA)	COND ITD (°F)	2xHEI HYBRID (%)	HEI HYBRID (%)	HEI LRVP (%)
100	90	3.00	25	280	140	210
	75	1.97	25.6	200	100	175
	60	1.30	27.2	200	100	140
	45	.90	30.8	180	90	110
75	90	2.51	18.8	200	100	175
	75	1.62	19.2	200	100	125
	60	1.05	20.5	200	100	80
	45	.69	23	140	70	60
50	90	2.09	12.6	200	100	120
	75	1.33	12.9	200	100	75
	60	.84	13.8	170	85	40
	45	.53	15.5	80	40	0
40	90	1.93	9.9	200	100	95
	75	1.22	10.2	200	100	55
	60	.76	10.8	150	75	0
	45	.47	12	50	25	0
30	90	1.79	7.5	200	100	75
	75	1.13	7.8	200	100	0
	60	.69	8	140	70	0
	45	.42	9	20	10	0

Condenser design is 3,500,000 lb/hr steam at 3.0 In. HgA with 90 deg F cooling water temperature. HEI Capacity required is 20 SCFM (Standard ft3/min.).

ITD = initial temperature difference between the saturation temperature at the condenser pressure, and the inlet cooling water temperature.

SI Conversions:
deg C = (deg F-32)/1.8
1 in. Hg = 3.386 kPa
1 lb/hr = .4536 kg/h
1 ft3/min = 1.699 m3/h

Table #3 COMPARISON OF VENTING EQUIPMENT

STEAM EJECTORS

Advantages
- low capital cost
- low maintenance, most reliable
- will not limit condenser pressure at part load conditions to 1.0 in. HgA (3.39 kPa) at design air leakage
- can maintain condenser pressure below 1.0 in. HgA (3.39 kPa) if air leakage is less than design

Disadvantages
- not usually automated, although available at increased cost
- Separate hogging jet required
- Condensate drain required
- Overload characteristics for NC's not as great as LRVP's

LIQUID RING VACUUM PUMPS

Advantages
- Compact, Floor Mounted
- Fully Automatic
- No separate condensate drain needed
- Capacity increases more rapidly at increased pressure
- No Separate hogger required

Disadvantages
- high capital cost
- limits condenser pressure at part load conditions especially under cold condenser cooling water conditions

HYBRID UNITS

Advantages
- compact, floor mounted
- fully automatic
- Capital cost of automated package equal to or significantly lower than LRVP
- will not limit condenser pressure at part load conditions to 1.0 in. HgA (3.39 kPa) at design air leakage
- can maintain condenser pressure below 1.0 in. HgA (3.39 kPa) if air leakage is less than design
- can be designed for increased capacity at the minimum anticipated condenser pressure at no load/low load cold water conditions, thus limiting the dissolved oxygen content of the condensate throughout the operating range and at the same time accommodate increased leakage as more of the heat cycle is subject to vacuum.

Disadvantages
- separate hogging ejector may be required.
- somewhat less responsive than LRVP when air leakage is greater than HEI design