IMPROVING CYCLE EFFICIENCY OF THERMAL POWER PLANTS BY ENHANCING VACUUM PUMP CAPACITY

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ABSTRACT: The paper provides an innovative approach towards improving the cycle efficiency, and hence the heat rate, of thermal power plants, particularly those located at higher temperature areas. Under such hot conditions, the vacuum pump capacity gets extremely affected, which is a major reason for vacuum deterioration.

Efficiency of a vacuum pump depends directly upon its air removal capacity, which in turn depends upon the temperature of the sealing water. Thus, manufacturers suggest maintaining the sealing water temperature at 15^0C, where the capacity of the vacuum pump would be the maximum achievable capacity. However, the temperature of 15^0C is challenging for the locations where the sealing water is cooled by water whose temperature is governed by the ambient conditions. Hence the desired capacity of the vacuum pumps is not achieved and other options compromising on efficiency have to be resorted.

The paper discusses this idea of improving the sealing water temperature of a typical rotary vane vacuum pump. Two methods have been proposed and validated by heat balance equations, namely (i) using chilled water generated in the AC plant as coolant for sealing water and (ii) using tubular evaporator heat exchanger. In both the methods, it has been found that vacuum pump flow increases by around 33% when sealing water temperature is 15^0C. The results formulated in the paper also consider the effect of sub-cooling in case the vacuum achieved falls below the design value. Moreover, the effects of reduction of seal water temperature on impeller life have also been elucidated in this paper.

Presently, during summers, the condenser vacuum is maintained around 92mmHg against the design vacuum of 76mm Hg. Thus, improvement of capacity of vacuum pumps can translate into tremendous profit to our organization, because each mm of Hg improvement, implies an annual financial benefit of Rs 29.87 lakhs.
KEYWORDS: Heat Rate, Vacuum pump, Cycle Efficiency

1. INTRODUCTION

1.1 EFFECT OF CONDENSER VACUUM ON CYCLE EFFICIENCY

Figure-1 shows a typical modfified Rankine Cycle which is the underlying cycle for all coal based thermal power plants. The line 7-8 represents the heat dissipated in the condenser, which depends upon the condenser vacuum.

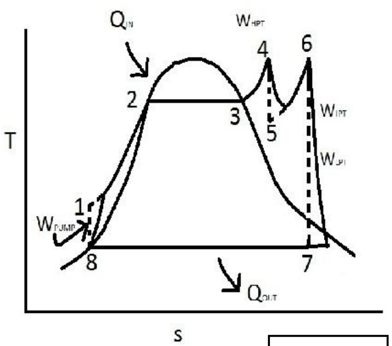

FIGURE-1

The cycle efficiency is defined as the work done by the cycle versus the heat input into the cycle. Thus as the area under the curve increases, the cycle efficiency shall increase. To obtain higher efficiency, we can lower the line 7-8, i.e., the condenser back pressure. However, design of the condenser entails a design vacuum to be maintained in the condenser. Operation of the plant above the design vacuum compromises the cycle efficiency, whereas too low vacuum below the design shall cause sub-cooling and increase in pump energy consumption, leading to inefficient operating conditions.

Thus to obtain the maximum point of the cycle efficiency, we have to obtain the optimum location of line 7-8, that is, maintain the vacuum under design conditions. This task is harder than it sounds as several factors affect the condenser vacuum, like cooling water inlet temperature, condensate temperature, flow rate of cooling water and steam, and amongst others the rate of air & condensible evacuation from the condenser.

The rate of evacuation from the condenser obviously affects the condenser vacuum. Hence, we propose to improve the rate of evacuation from condenser by improving the vacuum pump efficiency.

1.2 VACUUM PUMPS: EFFECT OF SEALING WATER TEMPERATURE ON CAPACITY

The liquid ring vacuum pump is a constant volume device, which means that at a given speed the volume of gas taken in will be a constant. If the gas steam coming to the vacuum pump contains a condensable vapour, and the vapour is condensed before it is drawn into the bucket of the liquid ring pump, the gas will occupy less space than it had before the gas was condensed. Since the pump draws in a constant volume, a measurement of the flow before condensation will be higher than the capacity of the vacuum pump as stated in the pump curves.

This theory is governed by the principle theory of Dalton's law of Partial Pressures. If a liquid is kept enclosed in an evacuated closed chamber some of the liquid will evaporate until equilibrium is reached between the gaseous phase (vapour) and the liquid phase. Since the chamber is closed the higher amount of gas will be evidenced by a higher pressure. This pressure is called the saturated vapour pressure which has a direct relation with temperature.

In FGUTPP, there are 6 rotary vane liquid ring vacuum pumps, 2 in each of Units 3, 4 and 5, where the sealing liquid is water. As outlined in the schematic Figure-2, the sealing water is routed through a separator tank via a recirculation pump into a cooler where the water is cooled by clarified cooling water and enters back into the pump as the working fluid. The vacuum pump cooler is a shell and tube heat exchanger where the enthalpy of the incoming sealing water is lost to cooling water circulating through the tubes. The specification of the existing cooler is attached in the Annexure-1.

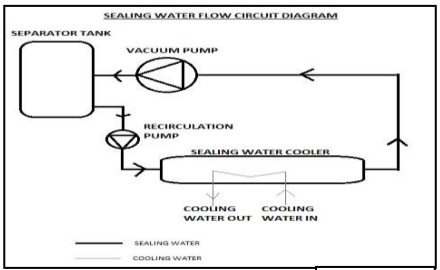

FIGURE-2

The liquid seal temperature is critical to the performance of the water ring pump. An increase of a few degrees can reduce the pump capacity by 30% or more, and can lower the maximum suction pressure the pump can deliver. Figure-3 is a typical capacity correction factor versus sealing water temperature cure supplied by the manufacturer. The graph shows the improvement in capacity over a range of sealing water temperatures. With a seal-water temperature of approx. 25°C, the maximum attainable end-vacuum is 55 bar absolute for a single-stage water ring vacuum pump. At that end-vacuum, the capacity correction factor is 0.58, which means that the pump capacity at that temperature and end-vacuum is only 58% of the published capacity of a given pump. As the seal water temperature decreases, the capacity correction improves. The capacity correction is 1.0 at 15^{0}C, implying that a vacuum pump operates at 100% of its rated capacity when the sealing water temperature is 15^{0}C.

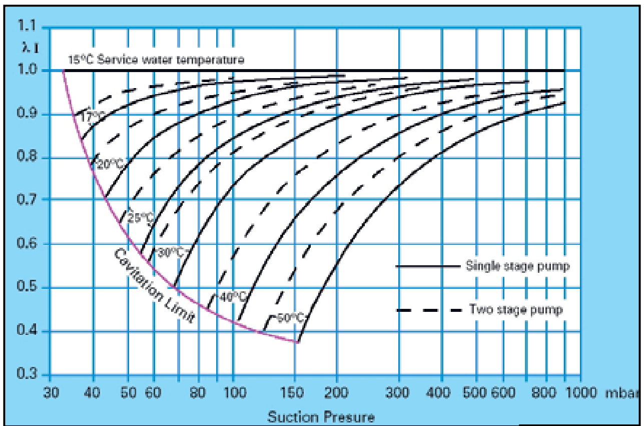

FIGURE-3

2. THEORITICAL ANALYSIS

The effect of saturated vapour on the dry air flow can be understood easily by understanding the thermodynamics behind the windfall of this power consumption. As the sealing water temperature reduces, the inlet air+vapour mixture temperature also reduces and the vapour condenses into water, thus increasing the efficiency of the vacuum pump by bridging the gap between the actual and rated capacity. The actual increase in air flow with the reduction in sealing water temperature to 15^0C was calculated based on the principle that the actual flow was a function of temperature factor and the condensing factor assuming that the rated airflow remains constant.

So, Actual Flow = Rated Flow X (Temperature Factor X Condensing Factor)

Table-1[1] gives a comparative statement about the deviation in actual capacity from the rated capacity for different sealing water temperatures and incoming air/water vapour mixture temperatures.

Sealing Water F°	Vacuum in Torr	Air/Water Vapor Mixture at F°					Sealing Water F°	Vacuum in Torr	Air/Water Vapor Mixture at F°				
		77	86	95	104	122			77	86	95	104	122
		Factor							Factor				
50	125	1.15	1.21	1.30	1.42	2.0	77	125	1.12	1.15	1.22	1.32	1.72
	90	1.21	1.31	1.47	1.70	–		90	1.18	1.23	1.35	1.52	--
	70	1.29	1.42	1.67	2.15	–		70	1.23	1.32	1.50	1.80	--
	50	1.48	1.71	2.28	–	–		50	1.38	1.59	1.95	--	--
	30	2.05	--	–	–	–							
60	125	1.18	1.23	1.30	1.48	2.0	86	125	1.11	1.15	1.20	1.31	1.68
	90	1.26	1.30	1.40	1.54	–		90	1.17	1.22	1.31	1.48	2.18
	70	1.32	1.41	1.56	1.90	–		70	1.21	1.32	1.49	1.75	--
	50	1.48	1.68	2.06	–	–		50	1.35	1.55	1.90	--	--
68	125	1.12	1.18	1.27	1.37	1.82	95	125	1.10	1.15	1.21	1.29	1.60
	90	1.19	1.27	1.39	1.57	–		90	1.15	1.21	1.31	1.45	2.05
	70	1.25	1.39	1.59	1.91	–		70	1.20	1.30	1.45	1.70	--
	50	1.42	1.65	2.10	–	–		50	1.33	1.50	1.80	--	--

TABLE-1

From the table, it can be seen that the condensing factor and temperature factor for sealing water temperature at 35^0C are 2.05 and 0.75 respectively. For the same vacuum pump working at the same operating conditions, for sealing water temperature of 15^0C, the condensing factor and temperature factor are 1.91 and 1.00 respectively. Thus, the actual capacity increases by 24.22% of the capacity when sealing water temperature was 35^0C.

So, it is possible to increase the capacity of the vacuum pump by 24.22% of the existing capacity, just by reducing the sealing water temperature to 15^0C.

This approach to improve condenser vacuum by enhancing the vacuum pump capacity is not unusual, as initiatives such as running both vacuum pumps to improve capacity and hence vacuum have already been taken. However, running both vacuum pumps in parallel does not often yield desired results as the demerits of auxiliary power consumption overcome the benefits of vacuum improvement.

Improving the condenser vacuum by reducing the temperature of working fluid of vacuum pump is the novelty of this paper. The authors have proposed two strategies to improve the sealing water temperature to 15^0C.

1) By using the existing standby AC compressor to generate chilled water as coolant for sealing water

2) By using a tubular evaporator heat exchanger.

Both the methods have been vividly detailed and backed by mathematical analysis. The equations used are the basic heat balance equations and the authors have tried their level best to adhere to the design specifications of the existing cooler in order to maintain equanimity of the theoretical analysis and very low or no extra expenditure.

Before embarking upon the different methods outlined under sections 2.1 and 2.2 to attain seal water temperature of 15^0C, the following specifications of the existing cooler are worthy to be noted:

- Working liquid flow : 23 m^3/hr
- Cooling liquid flow : 50 m^3/hr
- Working liquid temperature rise in vacuum pump : 8-10^0C

2.1 Strategy-1: Using Existing Standby AC Compressor

The strategy-1 uses one AC compressor system along with a chilled water pump in order to generate chilled water at 7^0C. This water is then fed to the existing cooler as the coolant for the incoming hot sealing water. The desired temperature drop in chilled water, i.e. Δt should be 5^0C. The advantage of using the existing heat exchanger is that the fouling factor, heat transfer coefficient and surface area of heat transfer remains the same, and so the temperature rise or ΔT in the sealing water will also remain 8-10^{0}C as per cooler specifications. The only unknown factor is the amount of chilled water required for the heat transfer, which can easily be obtained by heat balancing.

$Q_{\text{rejected by sealing water}} = Q_{\text{gained by chilled water}}$

$M . C . \Delta T = m . c . \Delta t,$ ...Equation-1

$\Rightarrow 23 . C . 10 = m . C . 5,$

As the specific heat capacity of water does not change appreciably in the above range of temperatures, m = 46kg/hr.The required mass flow of chilled water is the same as the flow of existing cooling water, which means that no extra piping modifications are required to be done. Moreover, 50kg/hr is also within the rated the flow of a chilled water pump which is 70kg/hr. Hence this design is technically feasible.

The practical feasibility of this strategy is also befitting considering that AC compressors have become redundant in most of the plants with the advent of new technology like VAMs or Screw Chillers; yet, routine overhauling of these standby compressors is carried out at extra cost. Thus, operation of these standby compressors to generate chilled water for vacuum pump cooler is practical.

2.1.1 Effect of Sealing Water Temperature on Vacuum Pump Impeller Life

The effect of sealing water temperature on the minimum absolute attainable suction pressure of the pump is shown in Figure-2, where the red line shows the demarcation below which cavitation is most likely to occur severely damaging the impeller.

To control or prevent cavitation, the sealing water must be able to support the vacuum level established in the condenser without boiling or vaporizing. The vaporization of the sealing water sets up the structures of cavitation, but the damage is caused when the vapour bubbles collapse, not when they form. A high velocity micro-jet of water tears away at the metallic surfaces of the pump internals when the collapse occurs [2].

It can be seen from Figure-2 that as the sealing water temperature increases, the minimum attainable suction pressure of the vacuum pump improves, thus reducing the chance of cavitation. A typical vacuum pump impeller costs approximately 11 lakhs [3].So, when the sealing water temperature is reduced to 15^{0}C, along with the benefit of vacuum improvement, there is the added advantage of impeller life improvement.

2.1.2 Cost Benefit Analysis of Strategy-1

In order to calculate the cost benefit of this strategy, the expenditure incurred on auxiliary power consumed due to operation of a standby AC Compressor and chilled water pump is tabulated against the benefits of vacuum improvement and impeller life improvement.

This paper emphasizes that since there is no direct relation between capacity of vacuum pump and condenser vacuum impovement, hence the exact vacuum improvement by capacity improvement of 24.22% is unknown. Thus, the authors have provided a break even point, meaning the exact improvement in vacuum which will return full return on investment.

<u>Expenditure to obtain cooling water at 15^{0}C</u>

Total Aux. power Consumption = 125 (compressor) + 25 (chiller water p/p) + 20 (CW p/p) HP [4] = 170HP = 126.67 kW.

No. of units consumption per day = 126.67kW x 24hr= 3072 kWH

Generation cost = Rs 3.50 per unit [5], thus, APC per day = 3072 x 3.50 = Rs 10,752

The above strategy has to be in service for only 9 months, i.e., excluding winters.

Yearly APC = Rs.10752 x 270days = Rs 29,03,040

Yearly maintenance cost of overhauling of compressors= Rs 25,000

Piping and misc cost for 25 years life = Rs 10,00,000 (approx)

Hence, the total yearly expenditure = Rs 29,03,040 + Rs 25000 + Rs 10,00,000 / 25 = **Rs 29.5 Lakhs**

Savings in terms of Vacuum and impeller life improvement

Assuming an improvement in vacuum of 'X' mm of Hg,

{(X x 8184.96 x 270) + 2.75} Lakhs =Rs 29.5 Lakhs, i.e., X = 1.21

This means only 1.21 mm HG reduction of condenser back pressure will bring break even. Operating above breakeven point, each mm HG reduction of condenser back pressure will give benefit of Rs 22.56 Lakhs per annum.

2.2 Strategy-2: Using Tubular Evaporator type Heat Exchanger

This strategy is another approach to achieve the desired sealing water temperature of 15^0C using the existing cooling medium, but the cooler set up requires design modifications. Although this strategy is just as effective as before, yet this setup may be recommended for the locations where standby AC compressor system is not available or where Strategy-1 cannot be implemented.

The tubular evaporator works on the basic principle that the boiling point of water reduces with reduction in pressure. The shell of this heat exchanger is maintained at a vacuum of 10mm of Hg where the boiling point of water is 11^0C. Thus, as soon as the cooling water is sprayed onto the cooler tube bundle, due to the low pressure, there is immediate boiling of water, and the latent heat is extracted from the cooler tubes, thereby cooling the sealing water that runs inside. The scheme of this strategy is shown in Figure-4.

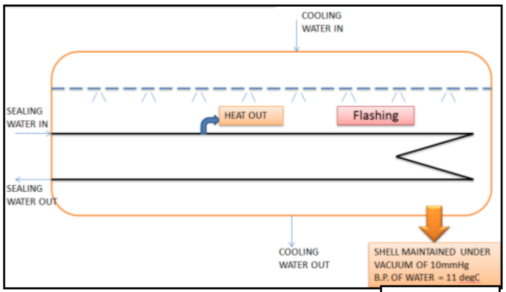

FIGURE-4

This type of heat exchanger is already in practice in the Vapour Absorption Machine, which is installed in every plant. Hence the strategy is practically feasible.

Once again, the calculations consider the desired inlet and outlet temperatures of sealing water as 25 and 15^0C respectively, and using heat balance equations, the heat exchanger is designed. However, to maintain close adherence to design specifications (Table-1), the heat balance shall include the same efficiency and fouling factor.

$Q_{\text{rejected by sealing water}} = Q_{\text{gained by cooling water}}$, or, $U . A . LMTD = m.c. \Delta T$ ...Equation-2

LMTD of the heat exchanger = $[(T_1 - T_0) - (T_2 - T_0)] / \log [(T_1 - T_0)/ (T_2 - T_0)]$

$$= [(25-11) – (15-11)] / \log [(25-11)/(15-11)] = 7.98^0C$$

1050 (J/s.m^2.k). A . 7.98 = 23000 (Kg/hr) . 4.18 (KJ/KgK) . 10

Thus, ideal surface area required for heat transfer, A = 31.87 m^2.

In an effort to maintain as close adherence to existing conditions as possible, the tube fouling factor has been considered to remain constant, i.e., 0.42 x 10^4 m^2 ^{0}C/kcal. Hence the final surface area required = 38.89 m^2

This area indicates the minimum surface area necessary for achieving sealing water temperature of 15^{0}C. Although the surface area is nearly double the surface area of the existing cooler tube bundle, the modification considered under this strategy is limited to a shell redesign only, as the required tube surface area can be easily obtained by parallel operation of both the tube bundles. Thus, this modification is very minimal and can easily be accomplished with existing resources. Figure-5 shows the modified scheme of sealing water after parallel arrangement of tubular evaporator type coolers.

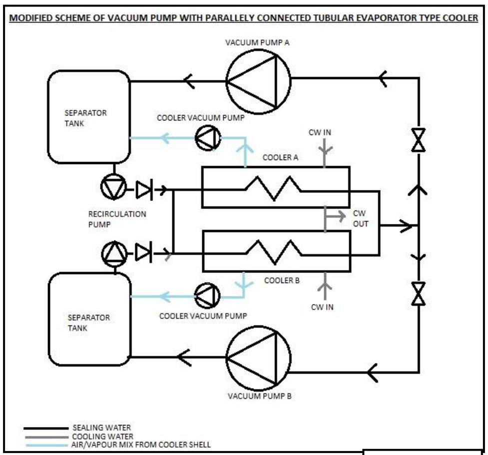

FIGURE-5

2.2.1 Cost Benefit Analysis of Strategy-2

The authors have tried to maximise the potential of available resources to minimise the extra expenditure incurred in this strategy. However, the avenues of expenditure and savings have been tabulated below.

Expenditure in terms of	Savings in terms of
Cooler shell procurement (Rs 27.10 Lakhs) [6]	Heat Rate (Rs 22.09 per mmHg improvement)
Vacuum Pump for shell (Rs 85,893) [7]	Impeller life (Rs 2.75Lakhs)
Maintenance and Installation (Rs 2 lakhs)	

If we assume the break even value of vacuum improvement to be 'X', and compute accordingly, then we can obtain that X=1.23. Thus after an improvement of 1.23mmHg in vacuum, not only will we get our return on investment, but we will also benefit with the improved heat rate.

3 RESULTS

The capacity of vacuum pump can be increased by 24.22% of its existing air removal capacity. This can be done simply by reducing the temperature of sealing liquid to 15^{0}C. The two strategies proposed to achieve this result are equally feasible. The end result of both the models proposed is the same: sealing water temperature at the vacuum pump cooler outlet is 15^{0}C. From the above calculations, it is evident that there exists a break-even point at 1.21 mm HG reduction of condenser back pressure only. Operating above breakeven point, each mm HG reduction of condenser back pressure will give benefit of Rs 22.56 Lakhs per annum and a 6mm HG reduction will give benefit of Rs 135.35 Lakhs per annum in operation of 9 months only. Improvement in exhaust vacuum by 10 mm Hg reduces the steam consumption in the turbine by about 1.1 %. Improvement in turbine efficiency varies significantly from 0.24 % to 0.4 % [8].

4 IMPLICATIONS

Apart from the commercial implications shown under section 2.1.2 and 2.2.1, the small improvement of heat rate (2.03kCal/kWh) for each mm of Hg translated into unthinkable inferences. As the exact drop in vacuum achieved cannot be theoretically calculated, hence the implications have been formed for every 1mm of Hg improvement in vacuum.

4.1 Implication on Coal Savings

Reduction of heat rate means lesser coal is required to generate one unit of electricity. Using the specific coal consumption of 0.66kg/kWh and average station heat rate of 2442kCal/kWh for May 2016 [5], we can achieve annual coal savings of 925.64 tonnes for each mm of Hg improvement in vacuum.

4.2 Environmental Implications

Reduction in coal burnt has a direct impact upon the flue gas emission. Stoichiometric equations for complete combustion prove that for every mole (12gms) of carbon burnt, one mole (44gms) of carbon dioxide is produced. Thus for every 925.64 tonnes of coal savings, around 1476 tonnes of CO_2 emissions can be reduced, and considering the Certified Emission Reduction unit rate of 7.91 euros, the emissions can translate into an annual saleable value of Rs 8.76 lakhs. On a greener note, reduction of 1467 tonnes of carbon dioxide emission is equivalent to planting about 4000 trees! As the coal burnt reduces, thus the amount of SOx, NOx and other particulate matter also reduces, thereby helping us to adhere to the rigorous environmental norms. Also, the ash content reduces, thus, saving APC as the loading of ash handling equipments also reduce.

5 CONCLUSION

Under the existing conditions, the vacuum pump is being inadvertently underestimated and made to run at only about 70% of its rated capacity due to high sealing water temperatures in geographically hotter areas. By reducing the sealing water temperature to 15^0C, capacity improvement of 24.22% in the vacuum pump can be achieved. This possibility opens a window of strategies for reduction of sealing water temperature, two of which have been elaborated in this paper.

The proposed modification of the process indicator has impact manifold, not only does it foretell a profit of around Rs 22.56 Lakhs per annum per mmHg reduction of condenser back pressure, but in the coming days of stricter environmental norms, it provides relief in terms of reduced ash generation, as well as reduced carbon dioxide and SO_X, NO_X emission.

GLOSSARY OF TERMS & SYMBOLS:

APC	Auxiliary Power Consumption
HR	Heat Rate
Q	Heat
M	Mass flow rate of sealing water
C	Specific Heat Capacity of water
m	Mass flow rate of chilled water
c	Specific Heat Capacity of chilled water
U	Universal Heat Transfer Coefficient
A	Surface Area of Tubes
LMTD	Log Mean Temperature Difference
T_1	Sealing water inlet temperature
T_2	Sealing water outlet temperature
T_0	Cooling water inlet temperature

ANNEXURE-1

HEAT EXCHANGER

Working liquid	Water
Quantity	22.84 M^3 / hr
Pressure reduction	0.32 bar
tF1 – tF2	3.4 K
Cooling liquid	Water
Quantity	49.9 M^3 /hr
Pressure reduction	0.14 bar
TF2 – tF2	1.6 K
Exchange surface	21.5 m^2
Fouling factor	0.42 x 10^4 M^2 °C/ kcal

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FIGURE-5	Modified Sealing water flow with parallel Coolers
TABLE-1	Temperature and condensing factors for varying seal water temperature

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