Condenser Vacuum Improvement and Related Parametric Analysis: A Case Study on A 120MW Thermal Power Plant

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Abstract— In the present global scenario, a huge emphasis is laid on efficiency and emissions and with respect to a thermal power plant the condenser is one important auxiliary equipment whose vacuum is one of the critical parameters which is often overlooked and has a vast impact on the plant performance even with a seemingly inconsiderable deviation from the optimum value. The demands placed on utility generating units are very significant and therefore at the Tata Power thermal power plant the purpose of this study was to identify all the factors which affect the condenser performance. With reference to the ideal Carnot heat engine, a lower sink temperature provides for a better work output and therefore the cooling tower performance also plays an integral role and is thus also included in the ambit of this study. The improvements were precisely quantified which occurred after the scheduled maintenance outage. The result of this study provides for methods to achieve better performance and sustainability through certain operational methods and retrofitting, while simultaneously meeting all the demands imposed.

Key words: Vacuum, Performance, Carnot heat engine, Cooling tower, Outage, Sustainability, Retrofitting

I. Introduction

In every power utility unit the condenser vacuum enhancement has been a continuous endeavor since this factor greatly affects the plant output in terms of the heat rate required for a certain Megawatt of power generation and the associated emissions. The design and operation of steam turbines must be such that the exhaust temperature is kept at as low a temperature as possible, so that a maximum Enthalpy drop occurs during the conversion of heat into work [1].The Fig. 1 shows the increase in heat interaction on the T-S (Temperature-Specific Entropy.) diagram, as depicted by the hatched area which is achieved on increasing the operating condenser vacuum from a pressure of (P.) to (Pi). The back pressure of the condenser determines the amount of latent heat that has to be removed by the condenser for the vapor to condense thus making it evident that more the heat that is removed through efficient heat transfer will lead to a lower back pressure. Improving the vacuum will undoubtedly generate a better payoff as the efficiency of the turbine would increase as rightly justified by the Carnot Heat Engine, which would facilitate for a higher temperature drop. Such an improvement will also reflect in the cost analysis of the power plant. Putman [2] outlines the various degrading factors to which the steam condenser is prone to, such as fouling of condenser tubes, water leakage from the tubes to the shell side and air ingress. The condensation of vapors flowing over a cold surface is initiated and sustained when the temperature of the surface is maintained below the dew point temperature of the vapor. The pioneering work in the field of condensation is due to Nusselt [3] who predicted the heat transfer coefficient of stationary pure vapor in film condensation on a vertical plate. Mc Adams [4] also presented the heat transfer coefficient for horizontal and vertical tubes which however are of little use for estimating the condensation rate under a vacuum of 1 to 1 ½ in. Hg abs.

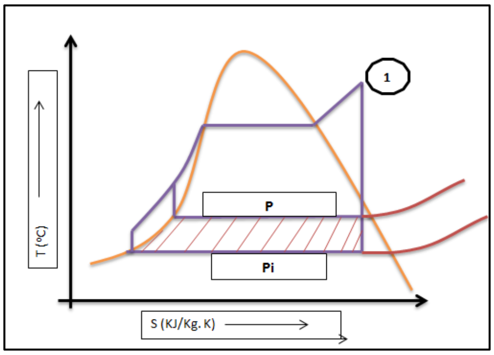

Fig. 1: Effect of Improving the Condenser Vacuum

The importance of proper condenser venting is not fully appreciated in the power industry; however its effect is quite profound. In the presence of non-condensable gases, the primary liquid film is no longer the only resistance to heat transfer. Othmer [5] has also shown that when as little as 1% of air by volume is mixed with the inlet steam, the heat transfer coefficient falls from 2000 to 1100 units with a temperature difference of 20°F between the two fluids. The cooling tower is an important aspect of the closed loop water cooled condenser as it rejects the low grade heat to the atmosphere. The cooling tower in its different sizes is the cheapest way to cool large quantities of water [6]. The most generally accepted concept for cooling tower performance was developed by Merkel in 1925 [7] and the same is used in this study. In the thermal power plant the scheduled outage for maintenance purposes is bound to generate improvement, but simultaneously a thorough understanding of the various performance parameters is of prime importance so that the result is driven towards the best attainable values. Thus the behavior of every related parameter with respect to the condenser vacuum must be constantly monitored in order to be a step ahead obtaining quick and efficient outcomes.

II. Condenser Vacuum and its Importance

In a power utility unit a lot of attention is focused upon the design of the exhaust end of the machine. The main control parameter is the back pressure measured at the turbine exhaust, since any deviation in this value from the optimum directly affects the heat rate of the machine. In a condenser, the latent heat of condensation is rejected at constant temperature and pressure to the cooling medium and therefore there is a huge decrease in specific volume as the vapor is converted into a saturated liquid. This decrease in specific volume is the main reason as to why a vacuum is developed in the condenser. To illustrate the importance of this factor, consider the Fig. 2 which is a plot of pressure (absolute.) variation with specific volume.

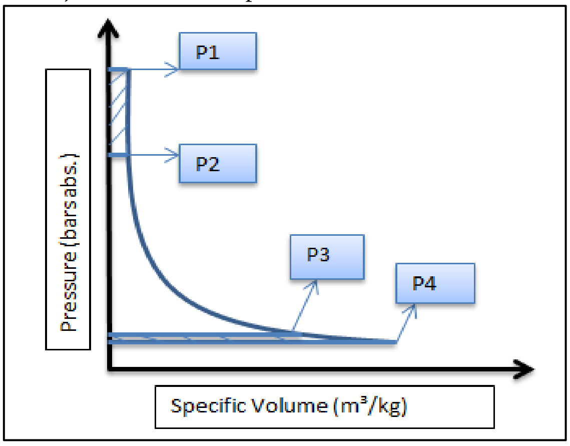

Fig. 2: Pressure v/s Specific Volume for Dry Saturated Steam

Here the pressure at the inlet of the turbine is (P1.) and respective specific volume. As the steam expands through the turbine while generating work, the pressure decreases until it reaches a value of (P3.) and a higher specific volume. The resulting work would be the equivalent to the area under the P-v curve. Now if the final pressure was reduced by a small amount from (P3.) to (P4.), the steam would expand further thus causing an additional amount of work under the same input conditions, which is depicted as the hatched area at the lower end of the graph. A comparable effect can be obtained by increasing the inlet pressure of the steam to a great extent, up to pressure (P1.) which is nothing but an inefficient method and therefore a lot of attention is concentrated on the condenser back pressure. Improving the vacuum leads to a lesser specific steam consumption (ssc.) since more work output is obtained with a certain fixed quantity of steam. Fig.3 shows the decrease in (ssc.) at full load after the plant shutdown which caused a vacuum improvement on 5th April 2015.

III. FACTORS THAT INFLUENCE CONDENSER VACUUM

A. Vacuum Loss Due To High Cooling Water Temperature:

Considering the same flow rate, a higher cooling water (CW.) inlet temperature will lead to a higher outlet temperature and thus for the same Terminal Temperature Difference (TTD.) will lead to the saturation temperature to be higher thus causing for a

Higher vacuum pressure at this saturation temperature. Fig. 4 shows the initial working conditions which provide a condenser vacuum (P*.) as represented by the dashed line and the solid line represents the conditions when the inlet cooling water temperature is higher leading to a vacuum pressure of (P**).

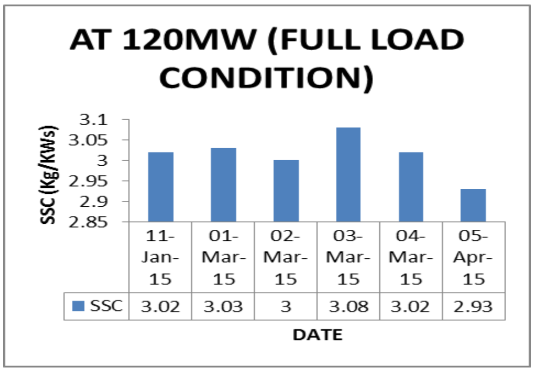

Fig. 3: Specific Steam Consumption V/S Time.

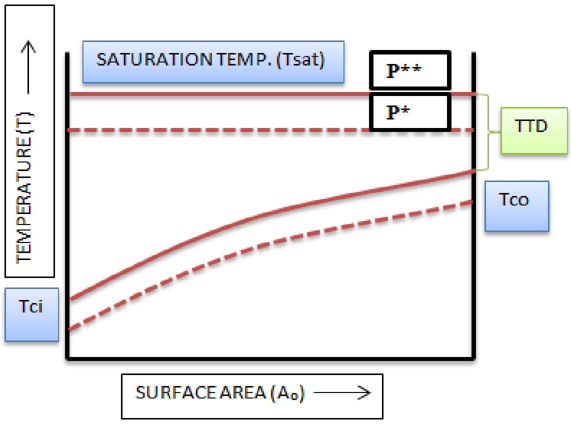

Fig. 4: Change in Vacuum Pressure with Change in Inlet CW Water Temperature.

With respect to the design conditions it was found that for a 3° higher CW temperature, the loss in back pressure is 1.324 KPa. The above factor can also be improved by improving the performance of the cooling tower used in design. During the maintenance process in the plant, the areas where rectification was incorporated with respect to the cooling tower were, the nozzle positioning, cleaning and replacement of fills and drift eliminators, cleaning and maintenance of the induced draught fan and cleaning of the cooling tower structure.

Such a loss can be minimized by admitting an abnormal amount of cooling water which will cause a lesser temperature rise but this improvement can be justified only when the turbine output improvement due to vacuum increase is larger than the power required for circulating an increased quantity of water. The Fig. 5 shows the daily record of variation of condenser vacuum with change in cooling water temperature thus proving that water at a lower temperature provides better vacuum.

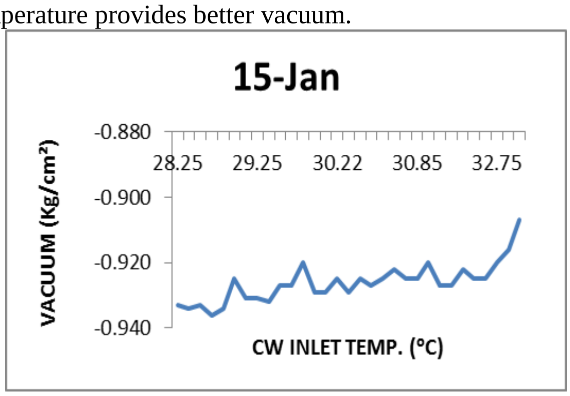

Fig. 5: Vacuum Variation with Inlet Cooling Water Temperature.

B. *Loss Due To Incorrect Cooling Water Quantity:*

This kind of a loss is can usually be controlled by regulating the cooling water outlet valve, however if this does not cause a considerable change then fouling of condenser tubes might be fouled. Here is was found that for a 3° higher temperature rise caused by reducing the flow rate, then the loss in back pressure is 2.319 KPa from design, keeping the other factors at design. Both the factors mentioned above can be controlled effectively for the given heat load (Q).

$Q = m \cdot Cp \cdot \Delta T$ ……………………..….. (1.1)

Where; m= mass flow rate (kg/s); Cp= Specific heat capacity of water; ΔT= temperature rise.

C. *Loss Due To Air Ingress:*

Since the condenser operates at vacuum conditions it is obvious that air in-leakage. Air ingress is a highly detrimental factor to condenser vacuum and as well to the condensate formed due to high concentration levels of oxygen. The thermal conductivity of air is very low and thus poses a high resistance to heat transfer which ultimately affects the condenser back pressure. By mathematically equating the heat rate for across air and copper with similar conditions, it can be seen that the thermal resistivity of a 0.254 millimeters thick air film is approximately the same as offered by a slab of copper with 3.78 meters thickness. Thus during the scheduled outage, using the helium test the areas of air ingress must be detected and then sealed. In the study of the present power plant the use of two, two- stage, liquid ring pumps were used which is usually an inefficient process. Therefore the efficient method would be to use a hybrid system which comprises of a single, two- stage liquid ring vacuum pump and a Steam Jet Air Ejector (SJAE), which can be introduced easily through retrofitting.

D. *Loss Due To Dirty Tubes:*

Such a loss cannot be controlled much but it can however be kept under check by cleaning the condenser when offline. In the maintenance operation at the plant, an acid cleaning was done using sulphonic acid (RSO_3H). Dirty tubes and air ingress is evident when the TTD is much higher than the design value. A high TTD means a less Condenser Efficiency (ɳ). Here is was found that for a 3° higher TTD due to air ingress/ dirty tubes, the loss in back pressure was found to be equal to 2.133KPa.

ɳ= (Actual temperature rise)/ (Max. temperature rise)

With reference to Fig. 3,

ɳ= (Tco – Tci)/ (Tsat – Tci)…….………………………. (1.2)

Where; Tco= cooling water outlet temperature; Tci= cooling water inlet temperature; Tsat= saturation temperature.

By simple mathematics; ɳ= (Range)/ (Range+ TTD).

As seen in Fig. 6 from the daily record of parameters in the power plant, a low TTD is obtained at higher loads. Thus when the plant operates at full load it is most efficient. Fig. 7 shows the difference in the design and actual pressure drop. A maximum pressure drop difference must always be strived for through regular maintenance of the condenser.

Pressure drop

(ΔP) total = ΔPf +ΔPr …………………………………. (1.3)

Pressure drop due to friction;

ΔPf (PSI units) = (0.00234 x V^1.84 x L)/ (di ^ 1.16) …………………..……... (1.4)

Where; V= velocity of water in tubes (fps.); L= Length of tubes (ft.); di= Inner tube diameter (inches).

Pressure drop due to return losses; ΔPr (PSI) = (1.2 x V²)/ (2 x g’) for a single shell pass.

Where gravity g’ (ft. /s²) = 32.3.

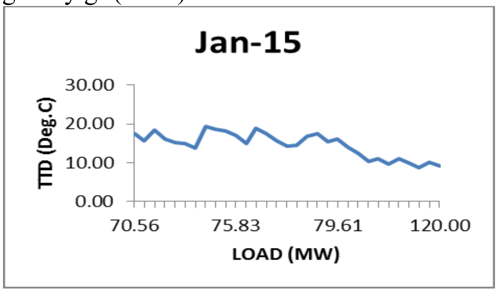

Fig. 6: Terminal Temperature Difference (TTD) Variation with Load.

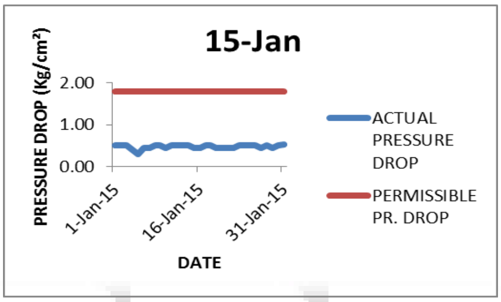

Fig. 7: Pressure Drop Variation with Time

IV. DESIGN DATA AND PARAMETRIC CONSIDERATIONS

Table 1; lists the design parameters of the condenser and cooling tower that were considered in this study.

CONDENSER	COOLING TOWER
Steam Quantity – 258 TPH	Water flow – 18,000 m³/hr/Tower
Pressure – 10.477 KPa (absolute)	Number of cells - 6
Hood Temp.- 46.6 °C	Hot water temperature - 43°C
Cooling water quantity - 16,000 m³/hr	Cold water temperature - 33°C
Velocity of water in tubes – 1.96 m/sec	Wet bulb temperature – 28.8°C
Cooling surface – 7774m²	Range - 10°C
Outer Tube diameter – 22mm	Approach – 4.2°C
Inner tube diameter – 18 BWG	Evaporation losses – 289.8 TPH
Effective tube length – 7400mm	Drift losses – 1.8 TPH
Number of tubes – 14,000	Max. fan discharge – 17,89,369.2 m³/hr
Cleanliness factor - 85%	Ground Area – 100.8meters x 16.8 meters
Tube material – 90/10 Copper – Nickle	Fill height – 1.5 meters

Table 1: Lists the Design Parameters of the Condenser and Cooling Tower That Were Considered In This Study

A. *CONDENSER:*

Log Mean Temperature Difference (LMTD) = $(\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$…………… (1.5)

Where; ΔT_1 = (Saturation temperature – Cooling water inlet temperature) ; ΔT_2 = (Saturation temperature – Cooling water outlet temperature).

Heat load

(Q) = U x A x (LMTD)………………… (1.6)

Where; U = Overall heat transfer coefficient; A = Cooling surface area.

Vacuum pressure is calculated at the saturation temperature of the steam (hood temperature).

B. *Cooling Tower:*

Effectiveness = $(Th_1 - Tc_1) / ([Th_1 - Tc_1] + [Tc_1 - Tw_1])$……………. (1.7)

Where; Th_1= Hot water temperature at cooling tower inlet; Tc_1= Cold water temperature at cooling tower outlet; Tw_1= Wet bulb temperature.

I.e. Effectiveness = (Range)/ (Range + Approach).

(L/G) ratio Where; L = water flow rate/ ft² of ground area; G= air flow rate/ ft² of ground area.

V. COMPARISON OF QUANTIFIED PARAMETERS

A. *Condenser:*

The Table 2, displays all the quantified parameters before and after the Maintenance shutdown, along with the design values all at 120MW (full load).

Parameter	Design	Before Shutdown	After shutdown
(Q) in [KW]	1,68,630.48	1,95,284.38	1,50,560.8
TTD in [°C]	3.523	9.37	6.34
Temp. Rise of CW [°C]	9.077	9.4	6.91
LMTD [°C]	7.12	13.530	9.3743
U [W/m²K]	3,046.57	1856.629	2065.99
η [in %]	72.03	50.07	52.15
Absolute Pressure (Vacuum)[KPa]	10.477	11.9219	11.2493
Loss in back Pressure [KPa]		3.15 At (η=72.03%)	0.3631 At (η= 72.03%)

Table 2: Displays All the Quantified Parameters Before and After the Maintenance

B. *Cooling Tower:*

The improvement in the cooling tower is difficult to compare in actual conditions since the number of fans operating are reduced when the desired temperature drop of water is attained. The four major factors which affect the size of the cooling tower are, the heat load, the approach, the range and the wet bulb temperature. The tower size varies directly with the heat load while the other three parameters are kept constant, while Approach, Range and the Wet bulb temperature individually vary inversely with the tower size while the other three factors are kept constant and only one factor is varied at a time. The Table 3 shows the different parameters that were measured with respect to the cooling tower at full load. The Number of Diffusion units (NDU)/ Tower difficulty, at design was found to be equal to 1.50 using the Merkel Equation. Fig. 8 shows the variation of (NDU) and Packing function with respect to the (L/G) ratio, the intersection of both the lines are solved at the point of intersection to obtain the required (L/G) ratio. Using the packing function and NDU curves at different (L/G) ratios it was found that the required (L/G) ratio was lesser than the design which implied that the cooling tower in operation was undersized. Table 4 shows these calculated values.

Parameter	Design	Before Shutdown	After shutdown
Effectiveness (%)	70.42	55.32 (6 fans running)	50 (4 fans running)

Table 3:

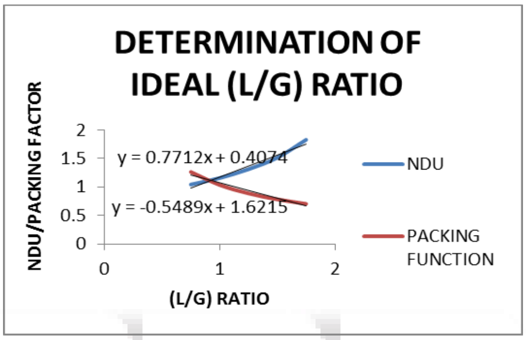

Fig. 8: (NDU) and Packing Function Versus (L/G) Ratio

Parameter	Design	Required
(L/G) ratio	1.4932	0.9197
Max. Fan discharge (m³/hr.)	17,89,369.2	29, 05,213.16 (at constant water flow rate).

Table 4:

VI. CONCLUSION

In this study, the importance of vacuum improvement has been thoroughly understood. A daily record was maintained of all the primary factors involved in the condenser and cooling tower performance. The various quantified critical parameter values before shutdown were compared with the values obtained after shutdown. The shutdown carried out for maintenance brought about an improvement of the condenser vacuum. The overall heat transfer coefficient after shutdown was improved by 11.27%.The TTD decreased after the condenser cleaning process carried out during the maintenance process. A considerable amount of decrease was observed in the specific fuel consumption which justifies the improvement induced after the condenser was overhauled. A suggestion was prosed for further improving the performance by adjusting (reducing) the cooling water flow rate so that the condenser operates at an efficiency of 68.50 in comparison to the 52.15% operating efficiency. Also the cooling tower induced draught fans are undersized and an increase in fan discharge is required, which is about 62.3%, which will then generate an (NDU) of 1.1166 in comparison to 1.50 which is the design NDU.

VII. ACKNOWLEDGMENT

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