EXPERIMENTAL STUDY ON PERFORMANCE OF STEAM CONDENSER IN 600MW SINGARENI THERMAL POWER PLANT

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ABSTRACT
In the present scenario, the rise in the demand for energy is increasing rapidly due to increase in fast growing world’s industrial sector. The conventional coal fired thermal power plant is one among the most effective and economical system developed for the conversion of heat energy into mechanical work. The thermal efficiency of power plant mainly depends on its turbine-condenser performance. The present work pertains to the study of performance analysis surface steam condenser operating at 600MW Singareni thermal power plant, India. The field data of the condenser were collected from the 600MW power plant operation record and the performance calculations were analysed at variable conditions. The efficiency of the unit has increased from 38.83% to 39.45% by reducing the condenser pressure to 65.21 mbar, which saves approximately 2.54 lakh rupees per day. The best possible back pressure that can be achieved in actual working conditions is evaluated using real-time operating parameters. Along with the fouling/air ingress analysis was also carried out which decreases the condenser efficiency, the reason might be due to the increase in condenser pressure slightly.

Key words: Air ingress, back pressure, condenser, fouling, Performance analysis, Thermal efficiency.

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1. INTRODUCTION
The concept of Exergy analysis is based on second law of thermodynamics which can be used as measure of both quality and quantity of energy. Yong Li and Lei Liu [1] performed exergy analysis on a 300MW coal based thermal plant and obtained irreversible losses distribution across different equipment of the plant at various part loads (100%, 75%, 50% and 30%) and
found out that maximum exergy loss is happening in the boiler, and in the turbine system maximum loss is in the condenser also, the coefficient of exergy loss is increasing with decrease in the plant load. Satish et al. So as maximum exergy loss occurs in boiler and condenser, if we can improve the efficiency of these two, the overall power plant efficiency can be improved considerable. The improvement of condenser efficiency is majorly depends on condenser parameters as Mayank R Pawar et al. [2] elaborates about the effect of cooling water flow rate, condensate temperature, cooling water inlet and exit temperature variations on the condenser pressure variation eventually and in return how the condenser pressure effects the net power output of the turbine and thermal power plant efficiency with the help of mini power plant experimental setup. And also discussed on how enthalpy and work vary according to condenser vacuum. Vikram Haldkara et al. [3] carried out the study on the influence of the cooling water temperature and flow rate on the condenser performance, and thus on the specific heat rate of the coal fired plant and its energy efficiency in Rayapati Power Generation Pvt. Ltd., Rajnandgaon having capacity of 7.5 MW. The study revealed that increasing of cooling water flow rate to maintain the same heat transfer rate at higher vacuum in the condenser, and thus to increase energy efficiency of the plant is a good way in optimizing plant operation. Abhay Kumar Sharma et al. [4] analysed for the parameters that are affecting the power plant efficiency negatively and concluded that causes which effecting the performance of condenser are cooling water inlet temperature, water flow rate and condenser pressure in energy efficiency of plant and also found out that the overall plant efficiency will reduce 2.7% because of these parametric deviation in the condenser. Along with condenser parameters the condenser defects such as air leakage and dirty condenser tubes results in poor condenser performance and Mohamed M. A. Ibrahim et al [5] focussed the study on influence of temperature as well as salinity on sea water thermo-physical properties causes the reduction in condenser overall heat transfer coefficient and effect on salinity on fouling factor variation at conceptual pressurized water reactor nuclear power plant (PWR NPP). It is concluded that the thermal efficiency of nuclear power plant decreases by approximately 0.2, respectively, for 5K and 10000 ppm increase in temperature and salinity of inlet seawater cooling inlet to the steam surface condenser. Karuppasamy et al. [6] carried out experiments on a hydro power plant and mentioned that the condenser performance is not up to the expectations due to several practical losses such as Low cleanliness factor, Low pumping function, exhaust capacity, Low heat transfer co-efficient, Air leakage of condenser are the reasons for the major losses in the efficiency of power plant and also proved that the condenser performance is the foremost important factor for achieving good efficiency of any power plant and depends on cooling tower water flow rate and inlet temperature of the power plant.

Ajeet Singh Sikarwar et al. [7] mainly focuses the comparison of design and operating parameters of the surface condenser at ATPS. Worked on condenser performance deviation due to effect of air in leakage and tube fouling. Realised that the plant output is 75.24MW instead of 120MW due to these effects on surface condenser. Suggested symptoms and remedies for several causes for the faults and by rectifying these failures the plant efficiency and performance can be improved to a good extent. Swapnil M. Mule et al. [8] derived an equation for volume flow rate of vacuum pump by using Dalton’s law of partial pressures (using partial pressure of Air and steam) and also elucidated the impact of air ingress on condenser back pressure, heat transfer, shell side resistance to heat transfer, overall heat transfer coefficient by performing performance guarantee test in real time scenario. An empirical relation between amount of air ingress and condenser heat transfer coefficient is established.
Gaurav maiswal et al. [9] did a parametric study on the surface condenser operating at BHEL plant (525 MW) and plotted the variations in the power output to change in operating parameters of the condenser like CW Flow rate, temperature and also the effect of dirty tubes and air ingress on the condenser pressure and from the results it is concluded that condenser back pressure improves as CW inlet temperature drops and CW flow rate increases. Joseph W. Harpster [10] proposed three methods for quantifying the impact of two sources of performance degradation i.e., tube fouling and air ingress are outlined as well as how they can be converted to the equivalent economic loss. An equation for calculation of cost of fouling and air ingress is developed, so that the maintenance operations can be focussed on the one that is causing highest losses. Non-condensable gases (NCG) have a huge impact on the heat transfer in the steam side of the condenser; it increases the back pressure in the STC (steam turbine condenser). Dusan Strusnik et al. [11] worked on the impact of NCG on heat transfer and presented a simulation model which suggests the optimal location for placing of the extraction tubes of SEPS (Steam ejector pump system) which continuously removes the NCG form STC and also designed SEPS for an optimized Laval diameter of the nozzle which helps in utilizing minimum amount of motive steam for maintaining the optimum pressure in the STC.

The main objective of this research is to investigate the performance analysis steam condenser to maintain the best possible back pressure operating of condenser at 600MW Singareni thermal power plant, India. The results revealed in this paper will be really useful to the power plant by decreasing the back pressure that increases condenser efficiency and thereby increase in overall heat transfer coefficient further which increases the overall plant efficiency. From the insight gained from observing the trends of fig.3 and fig.4, we can come to a conclusion about the major factor that is responsible for increased condenser pressure and remedies related to the same can be practised to improve condenser efficiency considerably.

2. METHODOLOGY
A plants efficiency is optimum at design conditions but in practical conditions operating plant at these conditions is not possible all the time and hence the deviation in the efficiencies of the equipment and the plant. In this particular paper, data pertaining to a steam condenser of a real-time working thermal power plant is collected shift wise. Operating parameters like cooling water (hereafter referred as CW) inlet and outlet temperatures, flow-rates, saturated steam conditions at different part loads of the plant are collected and this data is used to calculate the theoretical pressures of the condenser and compare with the actual values of the condenser.

NOMENCLATURE
C.W :Cooling Water
\( t_1 \) :C.W temperature at inlet (in °C)
\( t_2 \) :C.W temperature at outlet (in °C)
\( t_3 \) :Saturated steam temperature (in °C)
\( t_4 \) :Temperature rise of cooling water (in °C)
\( t_m \) :Mean C.W temperature i.e. \( \frac{(t_1+t_2)}{2} \) (in °C)
\( d t_4 \) :C.W Temperature rise (design)
\( \Theta_1 \) :Initial temperature difference (I.T.D) i.e. \( (t_3-t_1) \) (in °C)
\( \Theta_2 \) :Terminal temperature difference (T.T.D) i.e. \( (t_3-t_2) \) (in °C)
\( m_w \) :Mass flow rate of C.W (in m3/hr.)
LMTD :Log Mean Temperature Difference
ΔT_C.W: C.W temperature difference i.e. (t1 – t2)

P_{sat}: Condenser/Back/Saturation Pressure (mBar)

T_{sat}: Corresponding Saturation temperature of P_{sat} (in °C)

TTD: Terminal Temperature Difference

Q: Quantity of Heat transferred

U: Overall Heat Transfer Coefficient (in W/m²°C)

C_p: Specific heat of water is 4.184 K J/Kg-°C

A: Area of condensation is 20149 (in m²)

F: Correction factor (F = 1 for condenser)

\eta_{turbine}: Turbine efficiency

\eta_{boiler}: The design boiler efficiency is 87.45%

HR: Heat Rate of Turbine/Unit

C.V: Calorific value of fuel is 3600 kcal/kg

mf: Mass flow rate of fuel

Logarithmic mean/average temperature difference (LMTD) is the reason for the happening of heat exchange in any heat exchanger, here condenser is also a special type of heat exchanger. If LMTD value is huge the amount of heat transferred to cold fluid is also more. The logarithmic mean/average temperature difference can be calculated using the mathematical formulation below equation (1).

\[ LMTD = \frac{(\theta_1 - \theta_2)}{2.3log_{10}(\theta_1 / \theta_2)} \]  

There are mostly two results occurs by changing CW inlet temperature. The first and primary is to alteration of steam saturation temperature by same amount as the CW temperature change, assuming all the other factor are kept constant. This in return will change the corresponding back pressure. The other effect is caused because of the change in the heat transfer of CW water film in contact with the condenser tubes with the temperature of water. These two changes are in opposite directions. The magnitude of second effect is approximately equal to the fourth root of the mean CW temperature. The mathematical formula for the effect of variation of C.W inlet temperature on condenser back pressure is given by equation (2)

\[ \text{Actual LMTD} = (\text{Design LMTD}) \times \sqrt[4]{\frac{\text{actual tm}}{\text{operating tm}}} \]  

The major effect of a change of C.W flow is the change of the C.W temperature rise. Thus, if the flow rate is halved the temperature rise will be doubled, by keeping all other things constant. The other effect which also acts in the same direction as the first one is the one that results from the change in heat transfer rate, due to the change in thickness of C.W boundary film. It is approximately proportional to the square root of the flow rate. The mathematical formula for the effect of variation of C.W flow on condenser back pressure is given by equation (3) and operating C.W temperature rise is given by equation (4)

\[ \text{Actual LMTD} = (\text{Design LMTD}) \times \sqrt{\frac{\text{design C.W flow}}{\text{actual C.W flow}}} \]  

Operating C.W temperature rise, \( t_4 \) = \( dt_4 \times \left( \frac{\text{design C.W flow}}{\text{actual C.W flow}} \right)^{0.5} \text{°C} \)  

If the heat given to the surface condenser varies due to change in C.W temperature rise the value of logarithmic mean/average temperature difference (LMTD) also varies and the
mathematical formula for the Effect of variation heat given to the surface condenser on back pressure (part loads) is given by equations (5) and (6)

\[
\text{Operating C.W Temperature rise} t_4 = dt_4 \times \left( \frac{\text{Operating Plant Load}}{\text{Design Plant load}} \right)
\]  
(5)

\[
\text{Actual LMTD} = \text{Design LMTD} \times \left( \frac{\text{Operating Plant Load}}{\text{Design Plant load}} \right)^{\frac{2}{3}}
\]  
(6)

The performance of total power plant and turbine with relation to the condenser back pressure is calculated by using the below formulae. By using equation (7) the HR of turbine at any operating load can be computed. HR correction factor can be arrived from the standard BHEL graph with the help of corresponding condenser back pressure. The turbine HR for any back pressure can be computed with the help of equation (8)

\[
\text{Turbine heat rate (HR) of the power plant} = (m_{f_{C.W}}) \times \text{Kcal/KW} - \text{hr.}
\]  
(7)

New turbine HR = \text{Turbine HR (design)} \times \text{Kcal/KW} - \text{hr.}

(8)

Overall heat transfer co-efficient (U) also known as film co-efficient is utilised to calculate the amount of latent heat given to the cold fluid by hot fluid. The Overall Heat transfer coefficient is given by equation (9)

\[
Q = U \times A \times (LMTD \times F)
\]  
(9)

3. RESULTS AND DISCUSSION

3.1. Effect of condenser parameters on back pressure

The effect of variation of C.W inlet temperature on back pressure by keeping load and C.W flow constant at 600MW and 31650 m³/hr. is plotted in fig.1 and it is found that as cooling water inlet temperature increases, the back pressure in the condenser increases. [2] Actual values of back pressure are taken from real time data noted at various CW inlet temperature and theoretical values are calculated by using the equation (2).

The effect of variation of C.W flow rate on back pressure by keeping load and C.W inlet temperature constant at 600MW and 33°C respectively is plotted in fig.2 and it is found that as cooling water flow rate increases, the back pressure in the condenser decreases.[2] Theoretical back pressure values are calculated by using the equation (3).
The effect of variation of plant load on condenser back pressure is plotted for two different scenarios, first one is by keeping C.W inlet temperature and C.W flow rate at 33°C and 31650 m$^3$/hr. respectively as indicated by fig.3 and second one is plotted when the plant was shut down for maintenance purpose and restarted. Readings of back pressure are noted as load increased from 0 MW to 600 MW and C.W flow rate kept at 36000 m$^3$/hr. as indicated by fig.4. [9]

It is found from fig.3 that as amount of plant load increases, the back pressure in the condenser decreases. Actual values of back pressure are taken from real time data noted at various plant loads and theoretical values are calculated by using equation (6). It is also observed the as plant load increases the gap between theoretical and actual decreases, this trend of theoretical and actual plots indicates the effect of air ingress, whose effect tends to decrease as plant load increases. [8][11] From the fig.4, it is found that plant load increases from 0 to 600 MW, the back pressure in the condenser decreases. Actual values of back pressure are taken from real time data noted at various part loads and theoretical values are calculated by using the equation (6). It is also observed that as plant load increases the gap between theoretical and actual increases, this pattern between theoretical and actual plots indicate the effect of dirty tubes, whose effect tends to increase plant load increases. [10]

3.2. Effect of condenser back pressure on turbine and overall plant efficiency
Using equation (8) and From fig.5, it is found that as condenser back pressure increases, the turbine efficiency as well as overall plant efficiency decreases. So we need to ensure that the back pressure of the condenser to be as minimum as possible in order to increases the turbine efficiency as well as overall plant efficiency.

Figure 1 Deviation of back pressure with C.W inlet temperature.
Experimental Study on Performance of Steam Condenser in 600MW Singareni Thermal Power Plant

**Figure 2** Deviation of condenser back pressure with C.W flow rate.

**Figure 3** Deviation of condenser back pressure with Operating Plant Load.
3.3. Effect on overall heat transfer coefficient (U)

Along with these the calculations are also made for overall heat transfer coefficient using the equation (9) by varying the plant load as well as C.W inlet temperature and C.W flow rate is kept constant at 31650 m³/hr. The results were plotted in fig.6 and it is found that as Plant load increases, the overall heat transfer coefficient also increases for all inlet temperatures. This indicates that we need to ensure the full load working of the plant in order to have higher heat transfer coefficient which directly effects overall plant efficiency. The calculations for

Figure 4 Variation of Actual & Theoretical back pressure along with Plant Load

Figure 5 Variation of Turbine & Plant efficiencies along with condenser back pressure for 600MW and 480MW
Overall heat transfer coefficient of condenser by varying the C.W flow rate and keeping C.W inlet temperature 33°C and plant load 600MW constant and results were plotted in fig.7, it is found that as CW flow rate increases, overall heat transfer coefficient also increases decreases. This indicates that we need to ensure high CW flow rate in order to have higher heat transfer coefficient which directly effects overall plant efficiency.

We need to operate the condenser at minimum possible back pressure (optimum); deviation from which has its effects on the condenser performance and operating condenser at very low pressures (less than optimum) also has its drawbacks.

4. MONETARY SHIFT CAUSED DUE TO VARIATION OF CONDENSER BACK PRESSURE

The tonnes of coal required by the plant running at 600 MW (full load condition) and cost of coal required per day of operation are calculated using equations (10) and (11) respectively and their fluctuation with condenser back pressure is plotted in fig.8 and it is found that as condenser back pressure increases, the amount of cost incurred for power production per day increases. This also indicates that we need to ensure the back pressure in condenser is minimum in order to produce power with minimum amount of expenditure.

![Effect of Operating Plant Load On Overall Heat Transfer Coefficient](image)

**Figure 6** Change of Overall heat transfer co-efficient along with different C.W inlet temperatures.

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Figure 7 Change of Overall heat transfer co-efficient along with C.W flow rate.

Figure 8 Monetary shift w.r.to condenser back pressure for 600MW and 480MW.

5. CONCLUSIONS
As plant load increases, overall plant efficiency has two effects that act in opposite direction. The primary one being increase of back pressure that decreases condenser efficiency and second one is increase in overall heat transfer coefficient which increases the overall plant efficiency. As flow rate increases overall plant efficiency has two effects that act in same direction. The primary one being decrease of back pressure that increases condenser efficiency and second one is increase in overall heat transfer coefficient which also increases the overall plant efficiency. Few causes for each high condenser back pressure defect and
remedies are if C.W flow being less, one of the C.W pumps is defective and remedy is locate and deal with/show up the defective pump, if Malfunctioning of C.W pump and condenser tubes chocked and remedy is clear the fouling in the surface condenser pipes, if Excessive air ingress and remedy is detect and connect the positions of air in leakage/air ingress by using the helium detection test and if Malfunctioning of air venting equipment and low gland seal steam pressure and remedy is deal with the air venting apparatus and exactly correct the seal steam supply pressure. The following are the negative impacts of having very low back pressure

- Increased CW pumping power
- Increased leaving losses
- Reduced condenser temperature
- Increased wetness of steam

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