Performance Analysis of Surface Condenser under Various Operating Parameters

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ABSTRACT
The thermal power plants are used to generate power. The thermal power plants are designed based on required conditions (like a good quality of steam, pressure and temperature of steam etc.), but actually inlet conditions are not as per the designed conditions. In practical situations, when power plants are installed there are lots of constraints. This tends to reduce or increase output power and heat rate of thermal power plants. Due to these conditions, the designed power and heat rate are never achieved. Variations in the power outputs from plant are always a matter of disputes. So the parameters for power and heat rate are generated for different conditions of condenser pressure, flow rate of water through the condenser, Temperature difference. On the basis of site measurement and design data collection performance of the Condenser unit can be evaluated. These evaluations indicate that if operating conditions vary, then power output and heat rate also vary. This paper deals with the factors or parameters which reduced the efficiency of the condenser.

***Keywords* -** Flow rate of water, Power Output, Heat Performance Rate, Performance analysis

I. INTRODUCTION
The condenser is a heat transfer device or unit used to condense a substance from its gaseous to its liquid state, typically by cooling it. In doing so, the latent heat is given up by the substance, and will transfer to the condenser coolant. Use of cooling water or surrounding air as the coolant is common in many condensers. The main use of a condenser is to receive exhausted steam from a steam engine or turbine and condense the steam. The benefit being that the energy which would be exhausted to the atmosphere is utilized .A steam condenser generally condenses the steam to a pressure significantly below atmospheric. This allows the turbine or engine to do more work. The condenser also converts the discharge steam back to feed water which is returned to the steam generator or boiler. In the condenser the latent heat of condensation is conducted to the cooling medium flowing through the cooling tubes. [1]

In practical situations, when power plants are installed there are lots of constraints. This tends to reduce or increase output power and heat rate of thermal power plants. Due to these conditions, the designed power and heat rate are never achieved. [2-5]

The percentage ratio of the exergy destruction to the total exergy destruction was found to be maximum in the boiler system 86.27% and then condenser and stack gas 13.73%. In addition, the calculated thermal efficiency was 38.39 % while the exergy efficiency of the power cycle was 45.85%. [6]

II. DESCRIPTION
Basically, a condenser is a device where steam condenses and latent heat of evaporation released by the steam is absorbed by cooling water. Thermodynamically, it serves the following purposes with reference to the P-v diagram shown in Figure 1. Firstly, it maintains a very low back pressure on the exhaust side of the turbine. As a result, the steam expands to a greater extent and consequently results in an increase in available heat energy. The shaded area shown in the P-v diagram exhibits the increase in the work obtained by fitting a condenser unit to a non-condensing unit for the same available steam properties. In the P-v diagram,

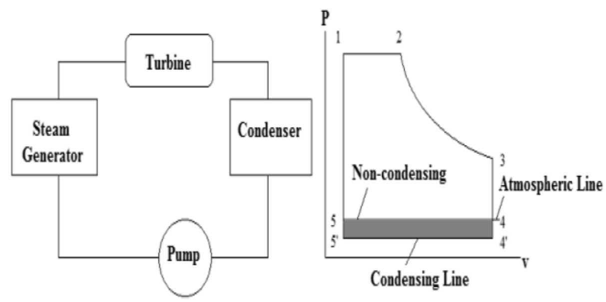
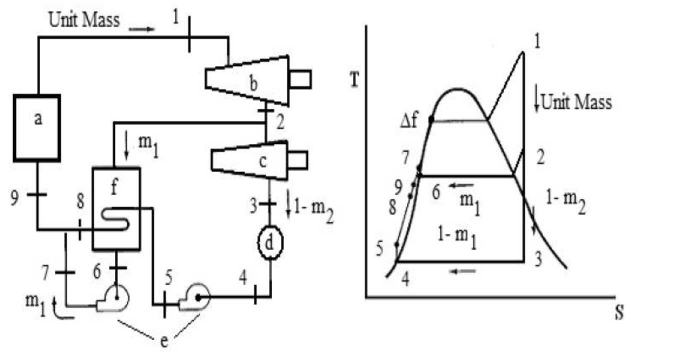


Figure 1: Key components of a thermal power plant working on a Rankine Cycle [7]

line 4-5 is non-condensing line when the condenser unit is not applied and line 4’-5’ is a condensing line when the condenser is used. Secondly, the exhaust steam condensate is free from impurities. Thermal efficiency of a condensing unit is higher than that of a non-condensing unit for the same available steam properties. In a reciprocating steam engine, the condenser pressure can be reduced to about 12 to 15 cm. of Hg. The thermodynamic analysis of condensate application is discussed in a thermal power plant using regenerative Rankine cycle with a closed feed water heater and pumped condensate as shown in the configuration of Figure 2. Condensate is pumped from the condenser through the Feed Water Heater (FWH) directly to the steam generator and to the turbine along the path 4-5-8-9-1. Ideally, P5 = P1 assuming no pressure drop occurs in the feed water heater and steam generator. As the operating pressure of the condenser is low due to an increased vacuum, the enthalpy drop of the expanding steam in the turbine will increase. This increases the amount of available work from the turbine. The low condenser operating pressure enables higher turbine output, an increase in plant efficiency and reduced steam flow for a given plant output. It is, therefore, advantageous to operate the condenser at the lowest possible pressure (highest vacuum).[10-12] The condenser provides a closed space into which the steam enters from the turbine and is forced to give up its latent heat of vaporization to the cooling water. It becomes a necessary component of the steam cycle as it converts the used steam into water for boiler feed water and reduces the operational cost of the plant. Also, efficiency of the cycle increases as it operates with the largest possible delta-T and delta-P between the source (boiler) and the heat sink (condenser). As the steam condenses, the saturated liquid continues to transfer heat to the cooling water as it falls to the bottom of the condenser, or hot-well. This is called sub-cooling, which is desirable up to a certain extent. The difference between the saturation temperature for the existing condenser vacuum and the temperature of the condensate is termed condensate depression. [13, 14]

This is expressed as a number of degrees condensate depression or degrees sub-cooled. However, the pump is designed according to the available net-positive-suction-head (NPSH) which is given as: NPSH is = Static head + surface pressure head – the vapour pressure of product – the friction losses in the piping, valves and fittings. There are two primary types of condensers that can be used in a power plant: 1. direct contact or jet condenser 2. surface condenser 3. Direct Dry Air cooled Condenser. Direct contact condensers condense the turbine exhaust steam by mixing it directly with cooling water. The older type Barometric and Jet-Type condensers operate on similar principles. The direct dry Air-cooled Condenser is beyond the scope of this paper.



(a)- Steam Generator, (b)- High Pressure Turbine, (c)- Low Pressure Turbine, (d)- Condenser (e)- Pump, (f)- Closed FWH

Figure 2: Regenerative Rankine Cycle feed-water-heater and pumped condensate [9]

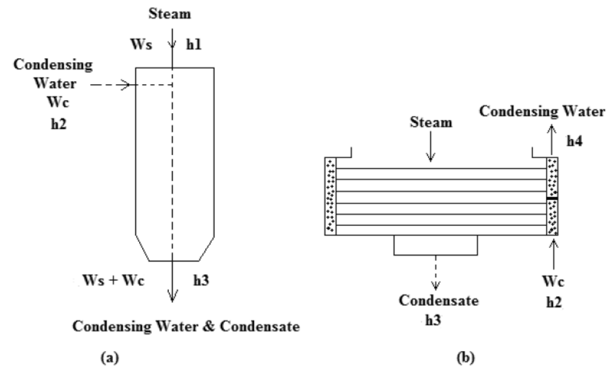


Figure 3: Energy exchange in a condenser

In a jet condenser, steam escapes with cooling water and this mixture inhabits recovery of condensate to be reused as boiler feed water. In this case, the cooling water should be fresh and free from harmful impurities. However, with moderate size turbine units the jet condensers can be used if enough supply of good quality cooling water is available. Steam surface condensers are the most commonly used condensers in modern power plants. The exhaust steam from the turbine flows in the shell (under vacuum) of the condenser, while the circulating water flows in the tubes. The source of the circulating water can be a river, lake, pond, ocean or cooling tower. Energy exchange in a condenser is analyzed by using the following steady state equations (see Figure 3):

Ws.h1 + Wc.h2 = (Ws + Wc) h3 …………… (1)
(Direct / Jet condenser)

Ws (hi – h3) = Wc (h4 – h2)………………… (2)
(Surface condenser)

Also, the exhaust steam enthalpy can be found from the turbine conditions line corrected for exhaust loss or by the energy relations as given below:

h1 = hi – W – Q – ∑ mh / 1- ∑ m … (3)

Where,
h1 = Exhaust-steam enthalpy, kJ/kg
hi = Prime-mover inlet steam enthalpy, kJ/kg

h2 = Inlet cooling water enthalpy, kJ/kg
h3 = Steam-condensate enthalpy, kJ/kg
h4 = Exit cooling water enthalpy, kJ/kg
Ws = Exhaust steam flow rate, kg per hrs.
Wc = Cooling water flow rate, kg per hrs.
W = Work output to turbine blades, kJ/kg per kg steam
Q = Radiation or other heat loss, kJ/kg per kg inlet steam
$\sum$ mh = Enthalpy of turbine extraction steam, kJ/kg per kg inlet steam
$\sum$ m = Total extracted steam, kg per kg entering steam In a surface condenser 'Terminal Temperature Difference' (TTD) is given as:

TTD = (Steam temperature) – (Cooling water exit temperature)

This is usually 2K or more. A low cooling water temperature rise helps to keep condensing steam pressure at low.

2.1 ELEMENTS OF SURFACE CONDENSER

The basic components of a surface condenser are shown in Figure 3. The heat transfer mechanism is the condensation of saturated steam outside the tubes and the heating of the circulating water inside the tubes. Thus, for a given circulating water flow rate, the water inlet temperature to the condenser determines the operating pressure of the condenser. As this temperature is decreased, the condenser pressure will also decrease. As described above, this decrease in the pressure will increase the plant output and efficiency. Steam condensation enables a vacuum and non-condensable gases will migrate towards the condenser. The non-condensable gases consist of mostly air that has leaked into the cycle from components that are operating below atmospheric pressure. These gases are also formed by the decomposition of water into oxygen and hydrogen. These gases must be vented from the condenser for the following reasons:

(a) The gases will increase the operating pressure of the condenser. This rise in pressure will decrease the turbine output and efficiency.
(b) The gases will blanket the outer surface of the tubes. This will severely decrease the heat transfer rates of the steam to the circulating water, and pressure in the condenser will increase.
(c) The corrosiveness of the condensate in the condenser increases as the oxygen content increases. Thus, these gases must be removed in order to enhance the life of components.

Table No. 1- Comparison between Designing & Operating parameters of Condenser

Parameters	Units	Design (MCR)	Operating
Unit Load	MW	120	75.94
CW inlet temperature	^oC	25.9	31.0
CW outlet temperature	^oC	40.0	42.0
CW temperature	^oC	14.1	11.0
Steam saturation temperature	^oC	42.6	44.7
Barometric pressure	mmHg	725	725
Vacuum	mmHg	661.5	654
Back pressure	mbar	84.6	94.6
Air suction depression temp.	^oC	38.4	46
CW circulated	kg/hr	33,230	35,300
Condensate collected	Kg/hr	1150	1120
Air suction depression temp	^oC	4.2	-1.3
ITD	^oC	16.7	13.7
TTD	^oC	2.6	2.7
LMTD	^oC	7.58	6.78
Surface area	m^2	20,000	20,000
No. of condenser tubes	Nos.	18,938	18,938
No. of plugged tubes	Nos.	Nil	Nil

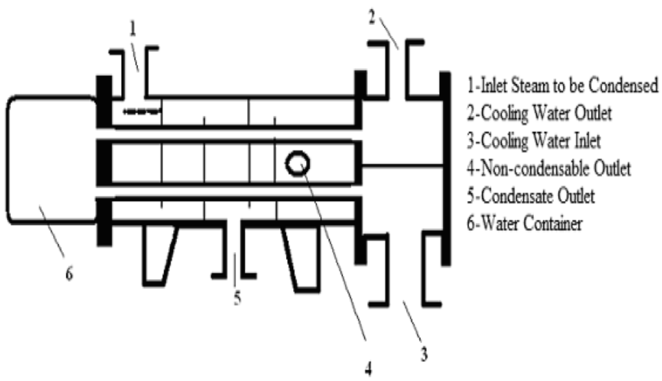


Fig. 4: Surface Condenser

2.2 AIR REMOVAL

The two main devices that are used to vent the no condensable gases are Steam Jet Air Ejectors and Liquid Ring Vacuum Pumps. Steam Jet Air Ejectors (SJAE) use high-pressure motive steam to evacuate the non-condensable from the condenser (Jet Pump). Liquid Ring Vacuum Pumps use liquid to compress the evacuated non-condensable gases and then these are discharged into the atmosphere. Condensers are equipped with an Air-Cooler section for the removal of non-condensable gases. The Air-Cooler section of the condenser consists of a number of tubes that are baffled to collect the no condensable. Cooling of the non-condensable gases reduces the volume and size of the air removal equipment. Air removal equipment must operate in two modes: hogging and holding. Prior to admitting exhaust steam to a condenser, all the non-condensable gases must be removed. In hogging mode, large volumes of air are quickly removed from the condenser in order to reduce the condenser pressure from atmospheric to a predetermined level. Once the desired pressure is achieved, the air removal system can be operated in the holding mode to remove all non-condensable gases. [7]

III. PERFORMANCE ANALYSIS OF CONDENSER

Performance Evaluation of Amarkantak Thermal Power Station by Performance Analysis of Steam Turbine Cycle,

3.1 COMPARISON OF DESIGN AND OPERATING PARAMETER OF CONDENSER

Insulation and steam drain systems based on present operating condition of plant and then compare it with design performance. The generating consists of two condenser units, each having same specification. The flow rate of water through the condenser, Temperature difference and pressure were measured. On the basis of site measurement and design data collection performance of the Condenser unit 1 can be evaluated. Data required for this analysis is shown in the table.

IV. CALCULATIONS

4.1 DEVIATION DUE TO INLET TEMPERATURE

Design cooling water inlet temperature (t_1) = 25.9 ^oC
Design cooling water outlet temperature (t_2) = 40.0 ^oC
Design average Temperature $(t_1+t_2)/2$ = 32.95 ^oC
Design Steam saturation Temperature (t_3) = 42.6 ^oC
Corresponding back pressure (corresponding to t_3) =84.6 mbar

Design, LMTD = $\{DT_1-DT_2) / \ln (DT_1/DT_2)\}$ = 7.62 ^oC
Where
$DT_1 = t_3 - t_1 =$ 16.7
$DT_2 = t_3 - t_2 =$ 2.6
Operating Cold Water Inlet Temp. (t_1) = 31.0 ^oC
Increase in Inlet Temperature from Design= 5.10 ^oC
New average Temperature = 32.95 + 5.10 ^oC = 38.05 ^oC
New saturation Temperature = 44.7 ^oC
New LMTD = $\{(\text{Design LMTD}) \times (\text{Design Avg. Temp.} / \text{New Avg. Temp.})^{1/4}\}$ = 7.34 ^oC
But New LMTD = $\{(DT_1-DT_2)/ \ln \{(t_3-t_2)\}$
From the above formula,
New Calculated Saturation Temperature (t_3) = 47.58 ^oC
Corresponding back pressure = 110 mbar
Due to the higher inlet temperature than the design value, it is found condenser is getting devit (loss) of, 25.4 mbar of vacuum.

4.2 DEVIATION DUE TO C.W. FLOW AND LOAD

Design CW Temp. Rise ($D\ t_4$) = $t_2 - t_1$ = 14.1 ^oC
Design Steam saturation Temp. (t_3) = 42.6 ^oC
Design LMTD = 7.58 ^oC
Design flow (Q1) = 33,230 m^3/hr
Present flow (Q2) = 35,300 m^3/hr
Design load (L_1) = 120 MW
Operating Load (L_2) = 75.94 MW

4.3 FOR TEMPERATURE DIFFERENCE USE DT_1

For temperatures use t_1,
New C.W. Temp. Rise (DT_4) = $(Q_1/Q_2) \times (L_2/L_1) \times D\ t_4$ = 8.35 ^oC
New LMTD = Design LMTD x $(Q_1/Q_2)^{1/2}$ X (L_2/L_1) = 4.63 ^oC
New C.W. inlet Temp. (t_1) = Design inlet Temp. = 25.9 ^oC
New C.W. outlet Temp (t_2) = (t_1+DT_4) = 34.25 ^oC
But new LMTD = $\{(DT_4)/ \ln (T_3-T_1)/(T_3-T_2)\}$
From the above formula, new calculated saturated temperature (T_3) = 35.90 ^oC
Corresponding back Pressure = 60.0 mbar
Due to the higher flow and lower load than the design value, it is found; condenser is getting credit (Gain) of, 24.6 mbar of vacuum
Net credit to the condenser due to inlet temperature, flow and load = (24.6-25.4) = - 0.8 mbar
Net debit to the condenser due to inlet temperature, flow and load = 0.8 mbar

4.4 DEVIATION DUE TO AIR INGRESS/DIRTY TUBE

Air suction depression temperature is found 1.3 ^oC, which is even, less than the design value of 4.2 ^oC. So it can be concluded that there is negligible

amount of air ingress inside the condenser tube or in ejector itself.
Net debit to the condenser as calculated above = 0.8 mbar Vacuum that should be obtained at the condenser is nearly same as that Design value. (Design value 84.6 mbar and calculated 85.4 mbar)
But actual measured value is = 94.6 mbar
So deviation due to dirty/fouled tube = 9.2 mbar
So, Total Deviation by these parameters (Loss) = 9.2 + 0.8 + 25.4= 35.4 mbar

4.5 FAULT TRACING INSTRUCTIONS

Þ = Reference power output,
Δh/Δt = Rate of vacuum drop in the condenser
t1 = C.W. Inlet temperature,
tg = (t4-t2), TTD steam/CW
t2 = C.W. Outlet temperature,
tg1 = (t3-t2), TTD condensate/CW
ΔÞ = C.W. Pressure drop across the condenser tube,
t = (t2-t1), CW temperature rise
t3 = Hot well condensate temperature,
Δh = Vacuum drop in mm Hg.
t4 = Exhaust steam temperature
Δt = Time in minutes.

Table No. 2 – Fault Diagnosis Instructions

l. No.	Fault	Symptoms	Cause	Remedy
1.	Low vacuum	i) t=high ii)t=higher corresponding to turbine loading and excessive ΔÞ iii) tg & tgl excessive	C.W. flow being less a) One of the CW pumps defective b) Malfunctioning of CW pump, Condenser tubes choked a) Excessive air ingress b) Malfunctioning of air venting equipment. c) Gland seal steam pressure low.	Attend the defective pump -Do- Clean the tubes Locate & plug the points of air ingress. Attend the air-venting equipment. Correct seal steam supply pressure.
2.	Rapid fall in condenser vacuum	iv)tg=high, tgl=normal and flooding of condenser	a) Fault in hot well level b) Condensate pump defective	Attend the fault Attend the defective pump Attend the fault
3.	Leak test reveals high Dh/Dt	i) C.W. supply, turbine gland seal system, air venting & condensate transfer system normal i) Vacuum pumps not in operation ii) Turbine gland seal steam low pressure iii) Leakage of air from vacuum breaker valve	Fault in low vac. trip system a) Severe air ingress b) Vacuum breaker valve open c)Rupturing diaphragm damaged d) Condenser drain valve open e) Local level gauge damaged f) Leakage in piping of vacuum system Air leakage in turbine. Turbine gland seal system defective a) Leakage from valve seat b) Improper seal water supply to the valve gland	Locate & plug the points of air ingress Close the valve Replace the diaphragm Close the valve Repair/replace level gauge. Locate & plug the leakage Point Locate & plug the leakage points Rectify the gland seal system defect Attend valve seat Improve sealing water supply

V. CONCLUSION

From all the analysis of ATPS, this paper realized that the power plant has proposed on 120MW but they could get worked on 75.24 MW. This paper evaluated all the aspects of condenser which affecting the performance of power plant. This paper worked on three causes which affecting the performance of condenser are deviation due to inlet temperature of cold water is 25.4mbar,deviation due to cold water flow and load 0.8mbar, deviation due to air ingress/dirty tube, so total deviation of pressure in the condenser is 35.4mbar. Eventually, this paper find that the total efficiency of a power plant will reduces to 0.4% by all these deviation in the condenser and by overcome these three reasons, the performance of power plant can be rises with a good level.

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