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Parametric Analysis Of Surface Condenser For 120 MW Thermal Power Plant

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Abstract

The conventional steam power plant working under the Rankine Cycle and the steam condenser as a heat sink and the steam boiler as a heat source have the same importance for the power plant operating process. Energy efficiency of the coal fired power plant strongly depends on its turbine-condenser system operation mode. Operating the condenser at optimum circulation water flow rate is essentially important to ensure maximum efficiency and minimum operating cost of the plant. To control the condenser variables like cooling water flow rate, condenser pressure, condenser temperature having vital importance on entire plant performance. For the given thermal power plant configuration, cooling water temperature or/and flow rate change generate alterations in the condenser pressure. Those changes have great influence on the energy efficiency of the plant. This project focuses 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. Reference plant GSECL-Sikka is working under turbine-follow mode with an open cycle cooling system having capacity of 120 MW situated at Sikka village near Jamnagar district under state of Gujarat, India. The study revealed that when plant runs at full load, the parameters like condenser pressure, heat rate, fuel consumption and cycle efficiency affected by cooling water flow rate. i.e. operating the condenser at optimum cooling water mass flow rate 13860 m^3/h instead of current flow rate 8750 m^3/h and plant input decreases as 10764288 Rs/year.

Key words: Cooling water flow rate, condenser pressure, condenser temperature, heat rate, cycle efficiency.

1. Introduction

The need for electrical energy will certainly continue to grow, and it has become imperative to lower the cost of electricity and enhance the operational economy of the turbine unit. Heat losses from the steam power plant cycle are mostly due to heat rejection through the condenser. Operating condenser at optimum variables is essentially important to ensure maximum efficiency and minimum input of the plant. The condenser operating conditions are of the great influence on the maximum generated power and the heat rate value. In the same time, the operating conditions of the cooling water system determine the operating conditions of the condenser. For cooling its condenser, steam power plants use basically two types of cooling systems: open-cycle and closed cycle [1,8]. Open-cycle or once-through cooling systems withdraw large amounts of circulating water directly from and discharge directly to streams, lakes or reservoirs through submerged diffuser structures or surface outfalls [2,6]. An open-cycle system depends on the adequate cool ambient water to support the generation at full capacity. A closed-cycle cooling system transfers waste heat from circulating water to air drawn through cooling towers. Conventional wet cooling towers depend on evaporating heat exchange and require a continuous source of fresh water to replace evaporation losses [3,7]. The ability of cooling towers to provide cold water to steam condensers of a thermoelectric unit decreases with increasing air temperatures and, in case of wet cooling tower, increasing humidity [14].

2. Literature Survey

A.N. Anozie& O. J. Odejobi [1] in 2011 had worked upon surface condenser cooling water flow rate and by optimizing it, decreased surface area of condenser for same heat duty. The computer program codes had been developed for simulation of power plant by varying cooling water mass flow rate. Decreased flow by optimization, he proved that condensate sub-cooling can be prevented and the enthalpy of feed water can be increased. Higher enthalpy FW means less fuel consumption in steam generator and finally fuel cost reduces [12]. Mirjana, S. lakovic.et al. [2] focused 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. As cooling water flow decreases the vacuum inside the condenser increases and heat rate reduces. This increased condenser pressure will decrease the power output of LP turbine which is not advisable. From mathematical model and analysis author proved that energy efficiency of condenser reduces when cooling water inlet temperature increases. Nirmalkumar P. Bhatt et al. [3] worked on surface condenser design based on cost optimization using genetic algorithm. For

Optimization, 3 different case study had been taken and reduces heat transfer area for same heat duty. Author had compared all three approaches for how tube velocity affects overall heat transfer coefficient for same diameter cooling water pipe of condenser. E. Nebot Et al. [4] worked on experimental work and model analysis for fouling deposition on tubes of condenser using sea water in thermal power plants. From experiment authors had proved that saline water fouling is directly proportional to water flow velocity. With higher possible velocity of water, fouling should be kept minimize. When fouling decreased the thermal resistance will also reduced and it will increase the heat transfer rate [11]. M.E. El-Dahshan et al. [5] analysed condenser of titanium tubes of desalination plant in UAE. Titanium condenser tubes exhibit outstanding resistance towards general pitting and crevice and stress corrosion cracking under a wide range of operating conditions. They also withstand fresh, brackish and seawater flow velocities up to 20m/s. Titanium exhibits these properties due to the presence on its surface of a thin, transparent, pore-free oxide film, which isolates the active metal from its surroundings [9].

3. Analysis of condenser variables

3.1 Process description

The case study plant GSECL-Sikka works with open cycle cooling water supply system for steam surface condenser and installed capacity is 120 MW. Sea water is taken from tunnel made at sea shore and it is collected in huge sump for constant water flow. Water is allowed to flow from tunnel to sump by gravity because sump level is quite low from sea level. Then pump sucks water from sump and sent it to common header which is connected with all pumps. There are three circulating water pumps used for cooling system. Two pumps for regular service and One extra pumps has been kept for standby. All the pumps are Vertical Turbine pumps. Each pump having installed capacity 14000 m^3/h. Entire plant runs based on coal used Rankine cycle having three steam expansion stage named HP, IP and LP turbine. One extraction from HP turbine outlet, two extraction from IP turbine and three extraction from LP turbine has been carried out for feed water heating into the FWH network. While designing the surface condenser all the steam extraction through the turbines cannot be considered because whenever any FWH is isolated or in maintenance, then bled steam of it will directly enter into the condenser. For over surface design generally 10% margin is preferred. i.e. main steam flow rate = 102 + (0.10 × 102) = 112 kg/s.

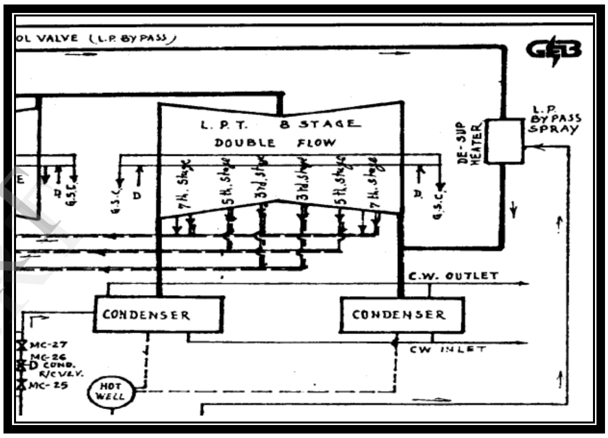

Fig.(1) LP turbine and Condenser arrangement of Rankine-steam cycle.

In GSECL, Sikka power station, two same size and same type shell and tube type surface condenser has been installed at the bottom of LP turbine to minimize the pressure drop inside the water tubes and ease of operation. Individual vacuum pump is provided to each condenser to maintain the proper vacuum inside the shell. As mentioned earlier that from cooling water system, discharge of both the pump is collected at common header and from header water is supplied to each condenser shell by same diameter and length pipe which is 8750 kg/sec for each condenser. As shown in fig. (1) due to double flow LP turbine inlet steam is splitted into two part and after work done on turbine blades, it will enter into the each condenser shell. Steam having mass flow 112 kg/s and at outlet the turbine 56 kg/sec will enter into the each condenser shell.

3.2. Analysis

In actual case cooling water mass flow can be found out by energy balance. i.e.

$$Q = m_{CW} C_p \, \Delta t$$

Here steam leaving the LP turbine is wet in case of isentropic expansion but in actual case it is slightly superheated or dry saturated. Here as per the plant working data the steam entering the condenser is considered as dry saturated having the value of dryness fraction is equal to one. By substituting the values in energy balance equation,

$$m_{CW} = 3854 kg/sec$$

Where, 0.85 is the cleanliness factor (heat transfer side)

ms = 56 kg/s for one condenser shell

hg = 2585 kJ/kg (At 0.1010 bar condenser design pressure from steam table)

Cp = 3.99 kJ/kgK for sea water

Δt = 8°C (design value of condenser working)

Obtaining mass flow rate is mCW = 3854 kg/sec~ 3.854 m3/sec ~13860 m3/hr

Current flow rate	Actual flow rate
8750 m^3/hr	13860 m^3/hr

By communicating the plant people about flow rate than it can be clear that the each circulating pump having 14000 m3/h installed discharge capacity but due to following problems discharge is less near about 37 % or 5110 m3/h compare with design value [13].

- Marine growth (Biological fouling) gradually increases inside the cooling water supply ducts due to sea water approximate 100 mm layer thickness. It creates resistance to flow and block the discharge passage.
- The casing, bowls and joints of circulating pumps which is vertical turbine type having leakage problems and air ingression, which affects the efficiency of pump.
- The impellers of pump got eroded due to cavitation problem. No doubt the impeller material had been selected to resist the salinity of sea water but after the long use it is eroded and facing corrosion problem.
- Lake of proper maintenance and lubrication.

3.2.1. cooling water flow rate and condenser pressure. In energy balance formula only mass flow rate is variable and rest of all the parameter is constant. Due to open cycle cooling system water is taken from the sea ant its temperature is not in our hand. It changes as per the climate condition. The value of Δt is taken as 8°C in summer condition when the temperature of inlet water reaches at its maximum limit. Average temperature variation is shown below which is taken from plant efficiency department based on past 20 years data.

Season	Minimum (°C)	Maximum (°C)
Winter	18	22
Summer	28	33

For fixed value of Heat transfer as mentioned above in energy balance equation, in following interval, the mass flow rate value is changed and analysis can be carried out. To analyse the condenser variables, i.e. cooling water flow effect on various parameters, the flow value has been increased 500 m3/h up to optimum flow rate which is 13860 m3/h. By substituting above different flow rate value Δt can be obtained and from tout is found out.

$$\Delta t = t_{out} - t_{in}$$

For summer condition tin is considered as 33 °C which is maximum temperature in summer in plant region. As per the guidance of plant people of GSECL-Sikka. Surface condenser is indirect contact type heat exchanger in which 100 % heat transfer is not possible. Therefore Terminal Temperature Difference is added with outlet temperature which is 5 °C which is design value for surface condenser. After adding TTD with tout, at this temperature the corresponding saturation pressure is found from the steam table and this pressure become the condenser pressure. When condenser works at optimum cooling water flow, the vacuum pressure inside the condenser is 684 mm of hg which is taken from plant efficiency department. To find the intermediate values of pressure from steam linear interpolation is followed when needed. For above calculations and values has been loaded in MATLAB Program and results and graphs are obtained.

Table 1. Result table

Sr. No.	Cooling water flow rate (m^3/h)	Temp. Difference Δt (°C)	t_{out} + TTD (°C)	Condenser pressure (bar)
1	8360	13.28	51.28	0.1316
2	8860	12.69	50.53	0.1270
3	9360	11.86	49.86	0.1228
4	9860	11.26	49.26	0.1201
5	10360	10.72	48.72	0.1160
6	10860	10.22	48.22	0.1130
7	11360	9.81	47.78	0.1106
8	11860	9.36	47.36	0.1082
9	12360	9.01	47	0.1062
10	12860	8.65	46.63	0.1044
11	13360	8.32	46.32	0.1026
12	13860	8	46	0.1010

When cooling water flow rate decreases from its optimum value, then it leads to poor heat transfer due to temperature difference increases. Here for analysis purpose we kept constant heat transfer and inlet water temperature for summer season but variable is flow rate and corresponding cooling water outlet temperature. Based on this following graph shown in Fig. (2) has been plotted from MATLAB. Another plot has been taken from the calculations which is tabulated above and from this graph, Fig. (3) we can conclude that when cooling water flow rate increases, the condenser pressure decreases.

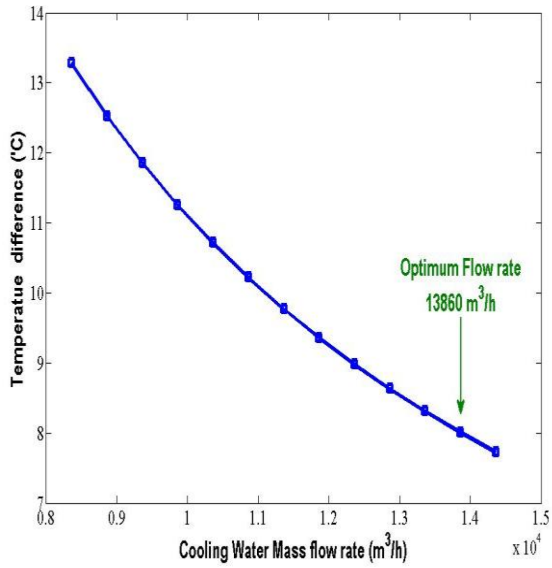

Fig. (2) Temp. Difference Δt v/s Cooling water inlet temperature

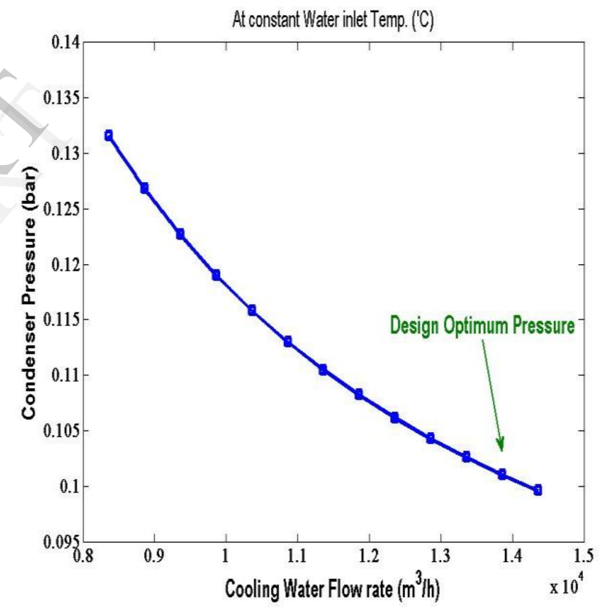

Fig. (3) condenser pressure v/s cooling water flow-rate

3.2.2. Condenser pressure and LP turbine power. GSECL, Sikka power plant having capacity of 120 MW and having 3 turbines for it, means combined output should be 120 MW and to analysed the effect of condenser pressure on power output, following calculations can be carried out:

$$\text{Turbine power} = P = m_s = (h_1 - h_2)$$

Where m_s = mass flow of steam entering into the steam turbine. There are 6 extraction is taken from the different turbines for FW heating purpose as mentioned earlier. Therefore from the working reading of the plant, these extractions should be deducted from the corresponding turbine inlet and power output of each turbine can be obtained. Enthalpy values at inlet and outlet of all the turbines are taken from the plant control room.

Mass flow of steam entering into the HP turbine = (main steam flow =112 kg/sec)

Mass flow of steam entering into the IP turbine = (HP turbine flow rate – no. of extraction taken from it)

Mass flow of steam entering into the LP turbine =(IP turbine flow rate – no. of extraction from it)

HP turbine power output = 38.321 MW.

IP turbine power output = 61.21 MW.

LP turbine power output = 27.14 MW.

Total plant capacity = $P_{HP} + P_{IP} + P_{LP}$ = 38.321 + 61.21 + 27.14 = 126.65 MW~120 MW because due to over surface design plant can produce 126.65 MW also for some hours.

Cooling water flow rate will affect the condenser pressure and condenser pressure will directly affect to the LP turbine power output. Hence the overall power output of plant is affected by condenser performance. For constant steam flow rate, when condenser pressure decreases the outlet enthalpy of LP turbine increases, i.e. enthalpy difference decreases and finally power output will go down for same fuel consumption. All the outlet enthalpy values can be taken from steam table and calculation has be carried out in MATLAB.

Table 2. Result table

Sr. No.	Condenser pressure (bar)	LP turbine Output (kW)	Difference (kW)
1	0.1010	27140	-
2	0.1026	27090	50
3	0.1044	27040	50
4	0.1063	26970	70
5	0.1080	26940	30
6	0.1108	26871	69
7	0.1130	26810	61
8	0.1160	26740	70
9	0.1201	26640	100
10	0.1226	26600	40
11	0.1279	26502	98

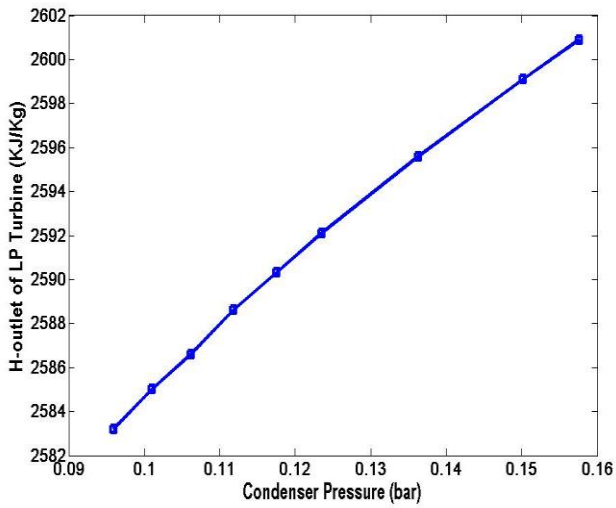

Fig. (4) Outlet enthalpy of LP turbine v/s Condenser pressure

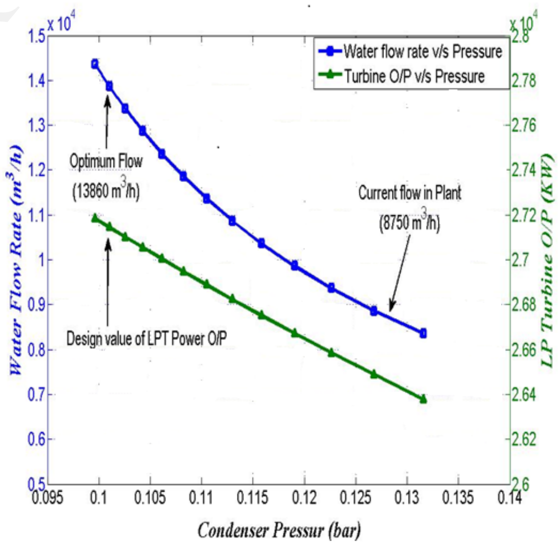

Fig. (5) Water Flow rate and LP Turbine Output v/s Condenser pressure

As mentioned earlier that when condenser pressure increases, the power output of LP turbine decreases and to obtain same overall generator output we have to keep steam load on condense and power output constant and against the outlet enthalpy of LP turbine, inlet enthalpy should be increased accordingly. From that we can obtain fuel consumption rate and specific Heat rate.

$$Power = m_s(h_{in} - h_{out}) \text{ (For LP turbine)}$$

By substituting design values,

$$Power = 81.95\ (2916.18 - 2585)$$

If h_{out} increases due to condense pressure rise, then enthalpy drop in LP turbine reduces and power output decreases. For same output following values of h_{in} can be found out from MATLAB programme.

$$\text{i.e. } 27140 = 81.95\ (h_{in} - 2585.60)$$

Assuming that enthalpy drop of HP and IP turbine remain same as per the design condition after deducting the losses in steam transmission lines.

Table 3. Result table

Sr.No.	Condenser Pressure (bar)	Inlet Enthalpy of LPT (kJ/kg)
1	0.1010	2916.18
2	0.1026	2916.77
3	0.1044	2917.35
4	0.1063	2918.18
5	0.1080	2918.53
6	0.1108	2919.46
7	0.1130	2920.14
8	0.1160	2921.027
9	0.1201	2922.26
10	0.1226	2922.75
11	0.1279	2924.28

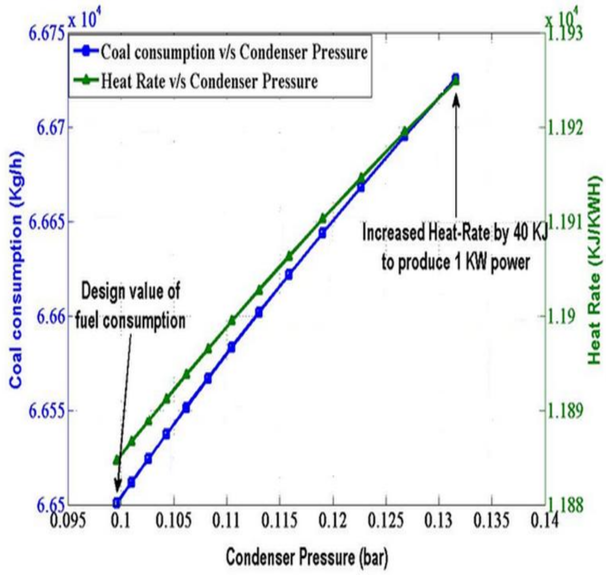

Fig. (6) Coal consumption and Heat rate v/s Condenser pressure

3.2.3. Heat rate and Cycle efficiency. Similarly to gain higher enthalpy at inlet fuel consumption increases and due to more coal consumption Heat rate of plant will increase which is inversely proportional to Cycle Efficiency. Assuming the linear relation between coal consumption and enthalpy rise for unit different values of fuel consumption can be obtained. From it plant Heat rate can also be found out [12]. i.e.

$$HR = \frac{B\ H_i}{\eta_{SB} P}$$

Where,

HR = heat rate (kJ/kWh)

B = fuel consumption (kg/h)

H_i = Gross calorific value of coal (kJ/kg)

η_{SB} = Boiler unit efficiency

P = Generator output (kW)

By substituting the values,

$$HR = \frac{66500\ \times 18421.92}{0.859\ \times 120000} = 11884.53\ kJ/kWh$$

Here variable is only B = Fuel consumption, rest all are fixed and by putting different values of B, we can obtain values for HR. From the above calculation we can plot the following graph for as a analysis result. Finally cycle efficiency is affected from the condenser variables which is:

$$\eta_{cycle} = \frac{H_{bo} - H_c}{H_{bo} - H_{bi}}$$

Where,

H_{bo} = enthalpy of outlet steam from boiler (kJ/kg)

H_{bi} = enthalpy of feed water into the boiler (kJ/kg)

H_c = enthalpy of steam into the condenser (kJ/kg)

By substituting the values,

$$\eta_{cycle} = \frac{3557.4 - 2585}{3557.4 - 1008.4} \times 100 = 38.15\ \%$$

Table 4. Result table

Sr. No.	Coal Consumption (kg/hr)	Heat Rate (kJ/kWh)	Cycle Efficiency (%)
1	66500	11884	38.12
2	66513	11887	38.10
3	66527	11889	38.05
4	66546	11893	38.03
5	66553	11894	38.00
6	66574	11897	37.80
7	66590	11901	37.78
8	66610	11904	37.75
9	66639	11909	37.74
10	66659	11911	37.73
11	66692	11919	37.70

To find the efficiency variable is only Hc, because it changes as per the condenser pressure changes. Rest of all is kept fixed for analysis purpose. Based on the above equation design value of efficiency is found out and then different values can be obtained by varying condenser enthalpy. Form MATLAB programme the following values are tabulated and graphs are plotted.

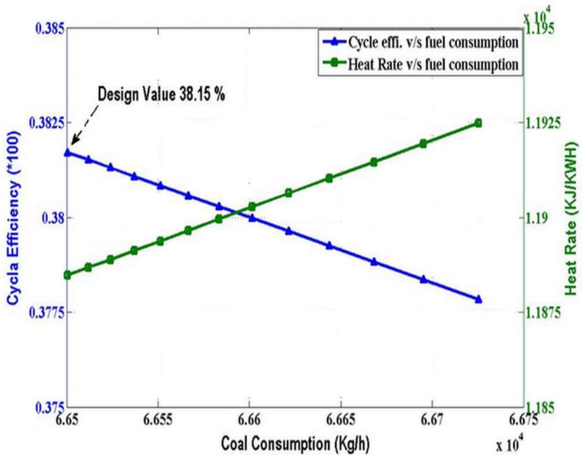

Fig. (7) Cycle Efficiency and Heat rate v/s Coal consumption

From the graph shown in fig. (6) we can observe that when condenser pressure increases, coal consumption also increases and to produce same power more heat energy have to supplied into the Boiler which increases Heat rate. Heat rate is the inverse of efficiency and input increases, the overall cycle efficiency decreases which is shown in fig. (7).

4. Effect on plant economy

The entire above analysis is done based on one condenser shell and taking the splitted steam flow inside the condenser shell due to twin shell condenser arrangement in case study power plant. Same effects can also be considered for second condenser shell because both the shell having same size and construction as well as operational conditions.

Increased coal consumption = 66692 – 66500 = 192 kg/h
Coal price rate = Rs. 3200/tone = Rs. 3.2/kg
Therefore,

Increased input cost = Coal Consumption × Price

= 192 × 3.2

= 614.4 Rs/hr = 14745.6 Rs/day

= 5382144 Rs./year/each condenser shell.

So, Overall input cost increased is

(2 × 5382144) = 10764288 Rs./year.

5. Conclusion

Steam power plant strongly depends on its cold end operating conditions, where the condenser is the key of the heat exchange system. In this paper, influence of cooling water flow rate is considered for power plant "GSECL-Sikka" with once-through cooling system based on the thermodynamic model. A numerical and repetitive calculation was done by using the Matlab programming platform. It is essential to operate the condenser variable at their optimum level for better performance of entire power plant and it is clear that:

- Lower flow rate leads to increase the condenser pressure than design value & poor heat transfer will occur, while higher the flow rate leads the condensate under cooling which is also not desirable because it decreases the cycle efficiency.
- Lower the condenser pressure tends to decrease power output of LP turbine.
- Lower the LP turbine output, higher the Heat rate and higher heat rate means lower the plant cycle efficiency.

6. Acknowledgement

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