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The Effect of Decreasing Sea Surface Temperature on the Design of Once-Through Surface Condensers in the CFPP

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ABSTRACT: Applying the once-through type generator cooling system by decreasing sea surface temperature value in the water area is a promising alternative. The application of the system will impact saving coal consumption due to reduced power consumption since it is no longer using the cooling tower in the once-through cooling system. In this condition, it is necessary to pay attention to the cooling water outlet, not more than 40 ℃. A surface condenser in this cooling system needs to plan the design dimensions and optimum performance. The Surface condenser function in the Coal Fired Power Plant (CFPP) system is to convert steam from the exhaust turbine into condensate water. Here, the condensate for the feedwater boiler will be recycled. This surface condenser design method uses heat transfer analysis with a weighted approach to the concept of heat transfer to determine the overall heat transfer coefficient of the surface condenser. The design result surface condenser has a heat transfer coefficient value of 3770.19 W/m2 ℃ and an effectiveness value of 0.5976 at a design vacuum pressure of 93.3 bar. The design results also indicate that the pressure drop value on the waterside is 0.3943801 bar (5.72 psi) and on the steam side is 0.129069 bar (1.872 psi). The simulation of the CFPP system using the Gate-cycle shows that implementing a once-through cooling system at the CFPP can save coal consumption by 21,224.48 tons per year, assuming there is no significant change in seawater temperature the plant (≤ 32 ℃).

KEYWORDS: boiler, coal, cooling tower, steam, surface condenser, water

I. INTRODUCTION

A surface condenser is an equipment that functions to change fluid from vapor to liquid. The working principle of the surface condenser is to flow steam into a room containing pipes (tubes). Steam flows outside the pipes (shell side) while water as coolant flows in the pipes (tube side). The surface condenser is the leading equipment in circulating cooling water in a steam power plant. According to sea surface temperature data, the seawater temperature in the water can reach 31.6 ˚C. This condition can occur because of the positive and negative correlation between rainfall and sea surface temperature (SST).

Changes in the SPL value in a certain period is to implement a once-through cooling system with a surface condenser design for better effectiveness [1]. Utilization of the once-through type generator cooling system can help to reduce the power consumption of the power plant system because the power used for cooling towers can be eliminated. Implementing a one-time cooling system is also a problem as the closed-cycle type's available data show the average value of the output water temperature in the past year was 43˚C. This condition is not possible because the output water's highest temperature must be less than 40˚C if a once-through type generator cooling system is used. Obtaining SPL data should not be more than 32˚C if this system is going to be implemented to reduce the output temperature value to less than 40˚C.

Researchers have made many efforts to improve the performance of surface condensers based on design aspects. The influence of condenser subcooling on the performance of the vapor compression system is Pottker and Hrnjak [2]. The study revealed that a 1°C change in cooling water temperature leads to a deviation of condenser pressure by 0.59 kPa, which corresponds to a variation in cycle heat rate of 0.36% and unit generation of 33000 kW [3]. This paper explains the role the condenser plays and describes some of the damage mechanisms that affect the entire condenser [4]. Surface condenser performance and quality of condensate water are essential [5]. Increasing the average condenser heat transfer coefficient will improve the power plant's thermal efficiency [6].

II. RESEARCH METHOD

The method used to design the surface condenser in this study is heat transfer analysis. This analysis is to determine the dimensions and optimum performance of the surface condenser. The design stages collect existing data as follows.

1. Data on SST (sea surface temperature) in water areas.
2. Data on the characteristics of the 660 MW CFPP, used as a reference for making simulations using the Gate- cycle software.
3. Operating data for the existing surface condenser.
4. Dimensions of the existing surface condenser.

Parameter

The several surface a condenser designs for the CFPP 660 MW once-through cooling system namely using software, manually to three-dimensional modelling. The method of using software simulates the CFPP system in the Gate-cycle software application. The Gate-cycle application can show the general design parameters of each principal component of the CFPP. Besides, Gate- cycle can also show the performance value of the CFPP system, such as net cycle power, net cycle LHV (Lower Heating Value) efficiency, and net cycle LHV heat rate. The initial stage simulation is an CFPP with a net capacity of 660 MW, LHV efficiency of 42-44%, and a closed- cycle generator cooling type. The second stage simulation changes the generator cooling type to once-through by removing the cooling tower and adding a cooling water source with characteristics that resemble seawater in coastal areas having a salinity value of 34 mg/g.

The next step is to define the surface condenser design parameters. This parameter is obtained by doing a simulation with Gate-cycle; however, it is necessary to adjust several parameters with the following considerations:

a. The determination of the vacuum pressure value is based on the actual operating conditions of the ESPP from the results of previous studies [3], [7]-[11]. The amount of the vacuum pressure value of 0.0933 bar-A allows the LMTD (Logarithmic Mean Temperature Difference) to be higher, reducing the need for heat transfer area on the surface condenser.

b.The addition of the value of the design steam inlet temperature is 1 ℃ showing that the surface condenser's design considers the steam condition, which is still in a high thermal load when it enters the surface condenser. This condition often occurs in CFPP. Although the steam temperature difference is minimal, it ranges from 0.5-1 ℃ of its saturation state. It is necessary to design a steam inlet temperature more significant than 1 ℃ to reduce the high thermal load of steam by the tubes in the zone of these impact tubes.

c. The difference between the condensation temperature and the design output temperature is designed to be 0.5 ° C; this temperature is called condensate depression. This zone aims to reduce the effect of cavitation if the condensate enters the condensate pump, which is ultimately for the feed water boiler.

The main parameter data presentation of surface condenser design is as in Table.

Table 1. Main Parameters of CFPP 660 MW Surface Condenser Design

Shell side (Steam water)		
Parameter	Value	Unit
Inlet temperature	45.5	°C
Temperature condensation	44.5	°C
Outlet temperature	44	°C
Pressure vacuum	0.0933	barA
Mass flowrate	349.21	kg/sec
Tube side (Sea Water)		
Parameter	Value	Unit
Inlet temperature	31	°C
Outlet temperature	39	°C
Velocity inside tube	2.3 7.546	m/s ft/s
Salinity	34	%o
Other Parameters		
Cleanliness Factor	90	%
Tube gauge method	25	BWG
Tube OD	0.875	Inch
Tube Pitch	1.3×OD	-
Normal Tubes Thickness	0.508	mm
Tube material	Titanium Gr.2	ASTM B338GR2
Number of passes	2	-

The parameter data in Table 1 is the basis for manual and software surface condenser designs. The design of a three-dimensional surface condenser uses the steam program application. In addition to designing the surface condenser with simulations, it is also a manual calculation method to determine the detailed stages of creating a surface condenser. It is necessary to pay attention to several things in the calculation stage of this surface condenser design, namely:

a. Surface condenser design approach uses three zones, namely the desuperheating, condensing, and subcooling zones [12]. Explanation of additional zones on the surface condenser is in the book Process Heat Transfer [13], which discusses the desuperheating-condenser and condenser-sub-cooler sections. Three zones on the two-pass surface condenser will result in temperature and phase changes for the steam side (hot fluid) and an increase in temperature for the cold fluid side.

b.The three-zone analysis allows the calculation of the LMTD and the heat transfer coefficient for each zone. In designing the Surface Condenser, it is necessary to calculate the heat balance to the pressure drop. The results of these calculations must meet a pressure drop value of less than or equal to 0.137895 bar (2 psi) for steam fluid and 0.689476 bar (10 psi) for seawater fluid. Determination of the total impurity factor value for the surface condenser is 0.02641 m2˚C/kW [14], further testing the effectiveness value [15].

Manual Design Stage

The way to do the condenser design is by doing the following calculation steps:

Energy Balance Calculation

$$\sum q = \dot{m}h\,(\Delta h) \quad (1)$$

Where, q is total heat (each zone) (W), Δh is enthalpy difference (kJ/kg) and $\dot{m}_h$ is mass rate of hot fluid (kg /s).

LMTD calculations

$$\Delta T_{LM} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}} \quad (2)$$

$$\Delta T_{weighted} = \frac{Q_{total}}{\frac{\sum q}{\Delta T_{LMTD}}} \quad (3)$$

Where, Th, i is the temperature of the incoming hot fluid (°C), Th, o is the temperature of the hot fluid out (°C); Tc, i is the temperature of the cold liquid in (°C) and Tc, o is cold out fluid temperature (°C)

Support Plate calculations

$$LT = (N_{sp} \times t_{sp}) + (2S_{sp2}) + ((N_{sp} - 1)_{p1}) \quad (4)$$

Where, S_{sp1} is the distance between the inner support plate(m), Ssp2: the distance between the outer support plate (m), tsp is the thickness of the support plate (m), LT is tube length (m), and N_{sp} is the number of support plates. Calculation of the net and gross heat transfer coefficient for each zone

$$U_c = \frac{h_o \, x \, h_{io}}{h_o + h_{io}} \quad (5)$$

$$Uc_{weighted} = \frac{\sum U_c . A_c}{\sum A_c} \quad (6)$$

(7) $$U_d = \frac{q}{A_d \, x \, \Delta T_{lm}} \quad (7)$$

(8) $$U_{d,weighted} = \frac{Q_{total}}{\sum \frac{q}{U_d}} \quad (8)$$

Uc, UC, weighted are the net heat transfer coefficient per zone and overall (W/m^2°C), respectively. Ac, Ad is the net and gross heat transfer surface area per site (m2). hio, ho is the inner and outer heat transfer coefficient (W/m^2 °C). UD, UD-weighted is the heat transfer coefficient in each zone and a whole (W/m^2 °C).

Calculation of fouling factors

$$R_{ft,calculated} = \frac{U_{c,weighted} - U_{d,weighted}}{U_{c,weighted} x U_{d,weighted}} \quad (9)$$

$$R_{TT} = L_T(R_T x R_1 x R_2) + \sum RE \quad (10)$$

Where RTT is total head loss (ft of water), LT is tube length (m), RT is tube loss (m of water/m of size), R1 is temperature factor correction, R2 is tube O.D. and gauge correction factor, and ∑RE is tube end losses (m of water).

Calculation of Pressure Drop in the tube

$$\Delta P_{tube} = H_{L\ total} \times \rho_f \quad (11)$$

$$H_{L\ total} = H_{L\ mayor} + H_{L\ minor} \quad (12)$$

$$H_{L\ major} = f \times \frac{L_T}{d_i} \times \frac{u_f^2}{2g} \quad (13)$$

$$H_{L\ minor} = k_{fric} \times \frac{u_{f\ moody}^2}{2g} \quad (14)$$

Where, $H_L$total is total head loss (m), H_L, major is head loss major (m), H_L, minor is minor head loss (m), fmoody is friction factor (searched by Moody's chart), kfric is loss coefficient, LT is tube length (m), di is the inner diameter of the tube (m), uf is cooling water fluid flow velocity (m /s), g is the velocity of gravity (m/s^2), ΔPtube is pressure drop on the tube (kg/m^2), and ρfis the density of cooling water fluid (kg/m^3)

Calculation of Pressure Drop in the shell

a. Equation for the pressure drop at the shell-side for the desuperheating zone and subcooling can be calculated by the following Equation:

$$\Delta P_{sheel-ds/sc} = N_L \times \left[\frac{\rho_f \times u_{max,o}^2}{2g}\right] \times f_{inc} \quad (15)$$

$$\Delta P_{shell-cond} = 4f_m \times N_L \times \rho_g \times \left[\frac{u_{max,o}^2}{2g}\right] \quad (16)$$

$$f_m = \frac{a}{R_{eo,max}^n} \quad (17)$$

Where, $\Delta P_{shell-\ desup/sc}$, $\Delta P_{shell\ -\ cond}$ is pressure drop on the shell-side of the desuperheating and subcooling zones, and the condensation zone (bar), finc, and fm are the friction factors of the Incopera diagram and the Blasius equation, respectively. The numbers of longitudinal tubes and maximum fluid flow velocity(m/s) in the tube bundle are NL and umax. ρf, ρg is the density of fluid and gas vapor (kg/m^3). Constanta a is the ratio of pitch outer diameter per tube, n is the Blasius exponent, and Reo, max is the maximum amount of Reynold steam.

Effectiveness Calculations

$$NTU = \frac{U_{total} . N_{tube\ total} . A_o}{C_{min}} \quad (18)$$

$$\varepsilon = 1 - \exp(-NTU) \quad (19)$$

Where NTU is the number of transfer units, C_{min} is fluid minimum heat capacity (kJ/kg°C), ε is effectiveness (%), and Ao is the outer surface area per 1 piece, (m^2). The value of electric energy production per year with a net power of 660 MW.

Production of electrical energy/year =
net power × power plant operating time/year (20)

Savings in Coal Energy Consumption
Coal energy consumption savings/year
=electrical energy production/year × heat rate generation (21)

Savings of Coal Amount/year

$$= \frac{\text{savings in coal energy consumption/year}}{\text{the calorific value of coal}} \quad (22)$$

III. RESULT AND DISCUSSION

***Simulation* Results of the 660 MW CFPP**

System Simulation of the ESPP system uses a gate cycle as in Table 2 and Figure 1.

Table 2. Performance of the 660 mw cfpp system with a gatecycle simulation

660 MW CFPP generator cooling system	*power* (MW)	*LHV* Efficiency, %	*LHV Heat rate* (kJ/kWh)
Closed - cycle	660.02	42.97	8,376.87
Once-through	665.58	43.34	8,306.59

The value of the nt cycle LHV efficiency for supercritical CFPP technology is in the range of 38.99 - 44.18% [16].

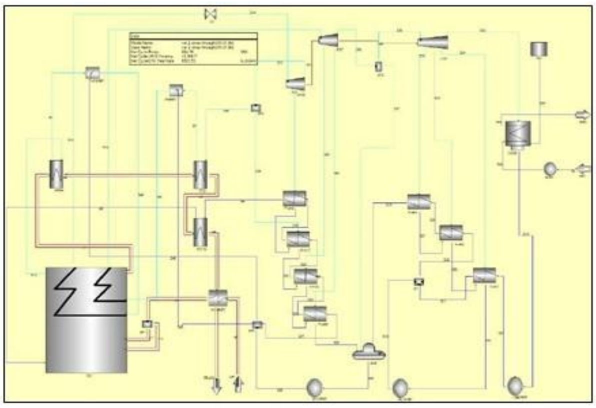

Fig. 1. Schematic diagram of a 660 MW CFPP Gate-cycle simulation with a once-through generator cooling system

Simulation Results of Surface Condenser Software

Figure 2 shows the schematic, performance, and design dimensions of the surface condenser.

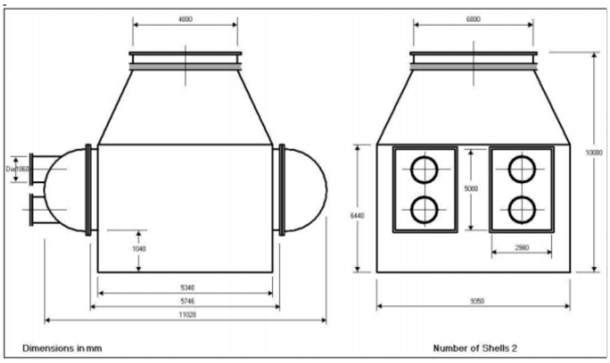

Fig. 2. Simulation results of the surface condenser program software (schematic view of the front and side)

Analysis of the Effect of the Implementation of the Once Through Surface Condenser Cooling System

The cooling system's design has a temperature rise (T_R) value of 8 °C so that the surface condenser in its optimum state can conduct steam condensation with the implication that there will be a change in the increase in seawater temperature at the outlet by 8 °C. Therefore, the maximum limit value of seawater's inlet temperature at once-through generator coolant is 32 °C. Adding T_R 8 °C, the amount ofsurface condenser outlet temperature becomes 40 °C. Based on the distribution of sea surface temperature data, the requirements for implementing a once-through generator cooling system can be applied throughout the month of the year, assuming there is no significant change in seawater temperature (≤ 32 °C). This condition is due to the influence of the surface condenser cooling water inlet- outlet points and ocean currents. In other words, the natural cooling process of outlet cooling water discharged into the sea runs optimally.

The application of the once-through generator cooling system will change the net power value of the CFPP system because it no longer uses power for the Cooling Tower. The cooling tower equipment replaced with cooling water sources directly from the sea possesses seawater characteristics adjusted by making corrections to seawater's salinity value to 34 mg/g. The amount of seawater salinity is significant because, with different salinity, seawater properties cost will also change [17].

CFPP with once-through generator cooling can produce higher performance than before. The amount of savings in coal energy consumption per year is 367,378,748,352 kJ. The heating value of LHV coal at the CFPP is 4134.23 kcal/kg (17,309.2 kJ/kg), so that the total coal savings is 2,1224.48 tons.

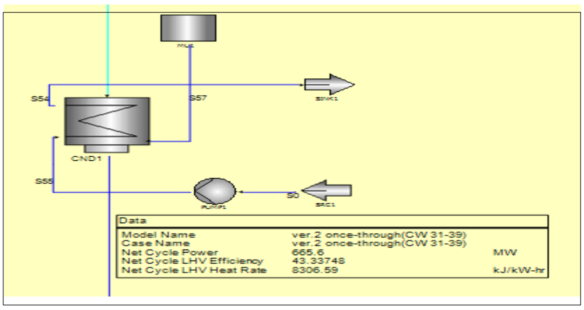

Fig. 3. Once-through generator cooling system at the 660 MWCFPP

Surface Condenser Design Results

1. The image sketch of the manual design

The image design on the surface condenser design uses the 2016 Solid-works application in Fig. 4 and Fig 5.

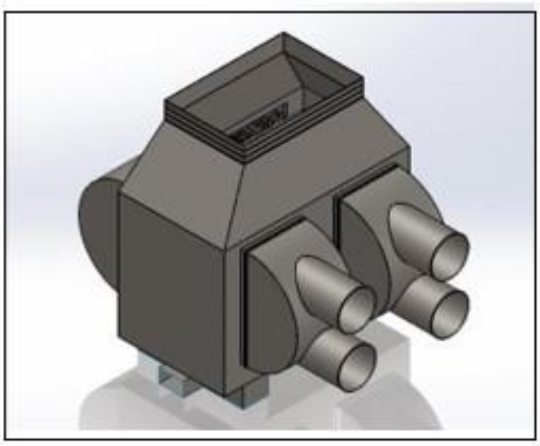

Fig. 4. Shows the results of the 3D surface condenser design

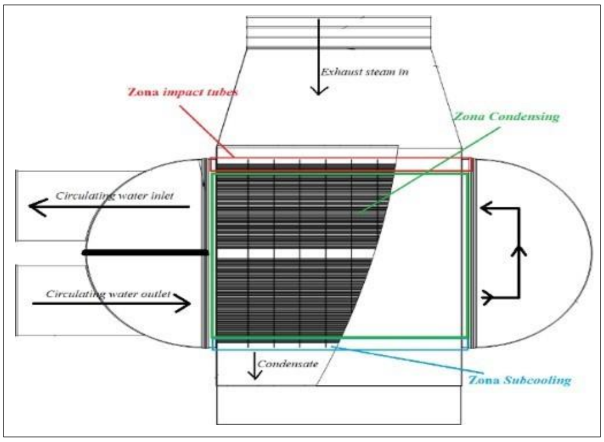

Fig. 5. Parts of the surface condenser

Caption:

No.	Component Name	material
1.	*Exhaust neckpieces*	ASTM A285 Gr-C
2.	*Exhaust neck expansion joint*	ASTM A285 Gr-C
3.	*Shell*	ASTM A285 Gr-C
4.	*Inlet-outlet water boxes*	ASTM A516-70
5.	*Circulating water inlet-outlet*	ASTM A516-70
6.	*Support plates*	ASTM B11 C60800 *(Aluminum Bronze)*
7.	*Hot well*	ASTM A285 Gr-C
8.	*Return water box*	ASTM A516-70
9.	*Water box pass partition*	ASTM A516-70
10.	*Tube sheets*	ASTM B265 Gr.2
11.	*Tubes*	ASTM B338 Gr.2

Material Analysis of Tube Surface Condenser

With or without inhibitors, the corrosion rate of Titanium material is 0, which is different from other materials that have a high corrosion rate. Carbon steel and Al-Bronze, for example, under normal conditions (without deaeration and inhibitor), the corrosion rate reaches 5 mpy and 1.035 mpy (millimeter per year), respectively. Titanium materials, which have a corrosion rate of <0.02 mpy, are grouped into categories with outstanding corrosion resistance levels. The higher the effect of corrosion on a material, the possibility of erosion of the material is relatively high [18]. So that the tube life becomes shorter because it is prone to leakage due to corrosion. Based on this analysis, use class 2 titanium for the surface condenser design.

Analysis of Surface Condenser Working Principles

The process that occurs in each surface condenser zone is as follows**:**

a. Impact tubes Zona

In this zone, the steam temperature decreases until it reaches its saturation temperature. The initial temperature of steam entering this zone reaches 45.5 °C; the design of the number of tubes of 1540 will reduce the steam temperature by 1 to 44.5 °C or, be precise at the steam saturation temperature value [19]. The cooling water fluid temperature on the tube side will increase from 35 °C (T_{box}) to 38.8 °C (T_4). The result of the energy balance analysis calculation is the temperature value. The number of tubes planned is 1540; this value appears after a specific calculation process with heat transfer analysis. Figure 6 shows an illustration of the direction of fluid flow and the condenser tube surface's proportions.

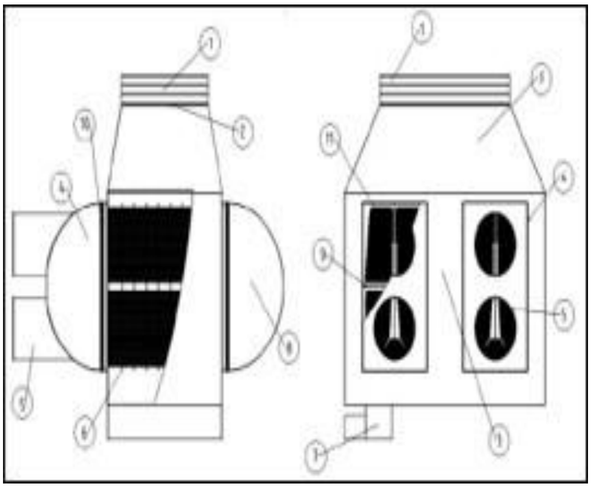

Fig. 6. Zones on the surface condenser

b. The condensing zone

Another function of the surface condenser is to create the lowest possible turbine backpressure or process, which re-operates the pressure while condensing the steam. When steam hits a relatively cold tube, it reduces. The effect of this condensation is the rapid change in state from gas to liquid.

This phase change results in a considerable reduction in the specific volume, and it is this volume reduction creates a vacuum in the condenser. When the condensation process is maintained, the vacuum condition will be maintained long as the condenser is kept free of air. The vacuum ventilation system supports the condenser vacuum by continuously expelling the air entering the system.

Non-condensable air in the design, generally due to leaks in pipes, around shaft seals, and valves, enters the condenser and mixes with the steam.

There will be no change in steam temperature at this stage, meaning that the steam temperature will remain constant at its saturation value, namely 44.5 ° C. The amount of cooling water required to take the heat load on the steam is large enough so that the steam will change phase to condensate. Meanwhile, the cooling water fluid temperature on the tube side will increase from 31℃ (T_{inlet}) to 39 ℃ (T_5). The calculation of the energy balance analysis produces the temperature value. So that, this value can calculate the LMTD for this zone, which is 8.9 ℃. The number of tubes planned is 57136; this value appears after the specific calculation process with heat transfer analysis. There is a significant difference in the number of tubes for the condensing zone with other zones. A simple explanation is that the amount of phase change enthalpy in steam is much higher than with steam without phase change. This condition resulted in a surge in the number of tubes. Many other things cause differences in these tubes, such as differences in fluid properties and different heat transfer parameter values. If the phase changes or remains, of course, will experience significant differences.

c. The subcooling zone

After the vapor condenses, the saturated liquid continues the heat transfer process against the cooling water flow. It then falls to the bottom of the surface condenser called the hot well. This zone is called subcooling. Some subcooling degrees can prevent the condensate pump's cavitation before feeding the condensate as a feedwater boiler. The design difference between the condensation temperature and the design output temperature is 0.5 °C. This difference makes the vapor that has changed its phase into condensate to be cooled back in the subcooling tube zone. The value of several degrees of subcooling is known as condensate depression. Excessive condensate depression will reduce plant operating efficiency because it has to reheat the condensate in the boiler. At this stage, the condensate temperature decreases from 44.5 °C to 44 ° C, and there is no phase change in the condensate. Meanwhile, the cooling water fluid temperature on the tube side will increase from 31 ℃ (Tinlet) to 35 ℃ (T1).

d. Analysis of Surface Condenser Performance

The explanation of the heat transfer coefficient analysis is as follows. Most surface condenser surfaces tend to acquire additional heat transfer resistance, which increases with time. This resistance may be in the form of a weak oxidation layer or other circumstances; it may be a thick deposit of scale, such as that produced from cooling ocean water. The fouling effect is considered by involving an additional thermal resistance of 0.027568 m2 ℃ / kW, which will be compensated by a different heat transfer surface area of 2377.26 m^2.

By monitoring the parameters of the drop in tube side pressure, vacuum degradation, and terminal temperature difference (TTD) can detect impurities on the condenser surface. Of course, along with the increase in fouling, there will be a high deviation in the parameter values previously mentioned. This decrease in surface condenser performance will affect the rate of heat transfer between steam and cooling water. Another implication is that it can increase the temperature rise value so that the surface condenser output temperature will increase, which is more than 40. The cleaning system commonly used on large scale surface condensers is the ball cleaning system.

The use of more than one zone on the surface condenser allows the calculation of each zone's individual U values . In each surface condenser zone, there is a separate and significant net coefficient over each surface. By weighing the U of each heat load and the individual design coefficient, this allows a reasonably good estimate to determine a single coefficient. Heat load and unique heat transfer coefficient values for each zone affect the overall single heat transfer coefficient, which is 3,770.19 W/m^2 ℃.

Analysis to get the heat transfer coefficient value is obtained from complex geometric relationships such as surface shape, dimensions, surface roughness ¸ fluid characteristics, and boundary conditions. The correlation represents a dimensionless number with several parameters. So, varying the numbers will explain the actual model of the Equation.

Tube configuration is related to the arrangement of tube arrangement, the number of longitudinal tubes, transverse tubes, and the size of the tubes pitch. The role of tube configuration is significant in analyzing the heat transfer coefficient because it can lead to the following factors.

i. The number of longitudinal tubes can change the correlation and correction factors of the Nusselt Equation.
ii. The more longitudinal tubes, the lower the heat transfer coefficient value because it increases the effect of condensate inundation.
iii. The more longitudinal tubes, the higher the steam pressure drop value at the shell side.
iv. The use of a staggered tube arrangement generally results in an increase in heat transfer amount compared to an aligned tube arrangement.
v. Increasing the pitch tube distance will decrease the steam velocity and Reynold number so that the amount of pressure drop on the steam subsides. Meanwhile, reducing the pitch distance will reduce the surface condenser size and weaken the tube sheet and cause fluid expansion problems in the tube to increase the pressure drop on the steam side. The right pitch distance is necessary so that the dimensions of the surface condenser are not too big. Typically, the pitch ratio value is 1.3 or more to avoid this effect.

d. Comparison of Surface Condenser Design Results

Table 3 shows the manual surface condenser design parameters and software design. Comparison between the results of manual surface condenser design and software, it can be concluded that there is no significant difference. The average deviation of the design results is 1.9066 %.

Table 3. Comparison of the parameter values of the surface condenser design manual and software

Parameter	Design Manual	Design Software	Unit
vacuum pressure	93.3	93.3	
condensation temperature	44.5	44.48	°C
enthalpy steam	2,461.7	2,451.8	kJ/kg
specific volume of steam	15.7	14.9	m^3/kg
cooling water temperature in / out	31 to 39	31 to 39	°C
cooling water flow rate	87,839.2	86,147.6	m^3/h
cooling water type	*seawater*	*seawater*	-
heat duties	795,970.4	790,121.8	kW
LMTD	8.87	8.88	°C
heat transfer coefficient design	3,770.19	3,761.14	W/m^2°C
displacement surface area hot	23,682.7	23,676	m2
number of tubes	59,068	59,006	pcs
pitch tubes	28.89	28.9	mm
tubes length	5,74	5,747	mm
cooling water pressure drop	0.394	0.397	bar

IV CONCLUSSION

The conclusions from the surface condenser design results in the CFPP 660 MW one-time generator cooling system are as follows.

1.The difference between the results of manual and software surface condenser design is not significant. In general, the dimensional deviation of the heat transfer surface area is 0.028%.

2. Surface condenser performance parameters calculated by manual design show a higher value than the condition of the existing surface condenser, namely as follows.

a.The design heat transfer coefficient is 3770.19 (W/m2°C) compared to the existing only 3754.93 (W/m2 °C).

b.The design pressure drop is 5.72 psi, lower than the existing 7.7088 psi.

c. The surface condenser effectiveness in the design is higher, namely, 0.5976 compared to the existing condition, which is only 0.5899.

3. Meanwhile, the effect of implementing the 660 MWCFPP once-through coolant on the system is as follows.

a.The net power of the CFPP has increased from 660.02 MW to 665.58 MW.

b.LHV net cycle efficiency increased from 42.97% to 43.34%.

c. The CFPP heat rate decreased from 8,376.87 kJ/kWh to 8,306.5987 kJ/kWh.

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