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A Feasibility Study on the Use of Cooling Towers in Thermal Power Plants Located in Tropical Region

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Abstract— In this work, the feasibility of using cooling towers has been studied theoretically for thermal power plants located in tropical region. Since cooling tower is directly coupled with the condenser of a power plant, the impact of selection of cooling tower on the design of condenser has been assessed. Electrical output of the plant is taken $1200MW_e$ and thermal efficiency is taken nearly 60% after superheating, thus around $800MW_t$ thermal load has been assumed on the condenser. The tertiary coolant water inlet temperature is assumed $28^0\mathrm{C}$ which is the normal reservoir temperature in tropical regions. Condensers with both single and multiple shell tanks have been studied. Design feasibility was evaluated based on two parameters; tube-side flow velocity and length to shell diameter ratio. From the results, it may be observed that there is no preferable design option for condensers with single shell tank and 2 shell tank, both for with and without cooling tower. For condensers with 3 and 4 shell tanks, multiple preferable design options were obtained. The number of design options is lower for condenser with cooling tower compared to the ones without cooling tower. Also, condenser size is found to be much larger for condensers with cooling towers than without cooling towers. As a result, use of cooling tower is not recommended.

Index Terms— Cooling Tower, Thermal Power Plant, Condenser, Vacuum Condenser, Tropical Region, Air Leakage, Heat Transfer Coefficient

1 Introduction

A condenser is a device that used in a thermal power plant to reject heat energy that is not utilizable. This energy is rejected in a low temperature reservoir (sink), which are usually large body of water such as ocean, lakes and rivers. As the thermal efficiency of a power plant depends on the sink temperature, it also largely depends on the weather conditions. For example, in tropical region where the atmospheric temperature during summer within the range of $35^0\mathrm{C}$-$45^0\mathrm{C}$ with relative humidity of 50-80%, the overall plant efficiency may reduce from 37.44% to 33.65% for 4kPa to 15kPa condenser pressure depending on the tertiary coolant water temperature [1]. The variation of inlet coolant temperature has a significant effect on the back pressure and thermodynamic efficiency of the condenser of a thermal power plant [2]. Not only condenser efficiency but also its power capacity is largely affected by ambient temperature [3]. Specifically, for a pressurized water reactor nuclear power plant with increasing of $1^0\mathrm{C}$ of coolant inlet temperature may reduce power output up to 0.45% [4]. Therefore, during the design of a power plant condenser, all the possible operating conditions including environmental conditions should be kept in mind. Otherwise, it may not only reduce the power plant efficiency but also result in failure to remove unused heat energy, which may lead to catastrophic event such as Fukushima disaster [5].

Since the thermal efficiency of a nuclear power plant is much lower compared to a fossil fuel power plant [6], it requires more heat rejection and thus a comparatively larger condenser. Also, some power plant required single shell tank while some others required multiple shell tanks [7]. Determining the optimum condenser's tube diameter is also important. Using a diameter that is too large increases the heat transfer area but leads to over dimensioning and higher cost. On the other hand, if its diameter is too small, water may flow faster through the tubes, which results in larger flow resistance and more pumping power [8].

Since most of the condensers used in power plant are vacuum condenser, there is always a possibility of air leakage. This may reduce condensation heat transfer coefficient and so the overall heat transfer coefficient [9], [10]. If a cooling tower is used alongside, the design of a condenser will be more complicated as it works on certain temperature gradients, mostly dependent on the atmospheric conditions [11], [12], [13]. Cooling tower performance also depends on cross-wind conditions [14], [15], [16]. Its shape and size also varies with ambient conditions. For example, at strong ambient winds, low height to diameter ratio of tower is recommended for better thermo-flow performances [17]. Cooling tower evaporation rate also varies with its height [18], which is also dependent on ambient conditions.

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From the above literature study, it is evident that the selection of cooling tower has association with the designing process of the condenser of a thermal power plant, the later one being one of the key factors on the overall performance of the plant itself. In this theoretical study, the feasibility of using cooling tower in a thermal power plant located in tropical region has been investigated on the basis of the design constraints for the condenser of a thermal power plant located in tropical region.

2 Methodology

In order to identify the effect of selection of cooling tower on the design of the condenser, the design parameters with and without cooling tower must be compared. It has been assumed that the condenser coolant inlet temperature is around $28^0\mathrm{C}$, since the average reservoir water temperature in tropical region countries is nearly of this temperature during summer season. If cooling tower is used, the maximum possible temperature of coolant outlet from condenser should be $40^0\mathrm{C}$, since a $12^0\mathrm{C}$ ($\sim 30^0\mathrm{F}$) temperature gradient in a cooling tower is recommended [19]. However, for condensers with no cooling tower, no such constraint is applicable. As a result, three greater coolant outlet temperatures have been considered for no cooling tower scenario; $60^0\mathrm{C}$, $65^0\mathrm{C}$ and $70^0\mathrm{C}$.

Although there are multiple design options for a condenser for a thermal power plant, longitudinal, single pass condensers are the most commonly used ones [7]. A longitudinal condenser arrangement with four shell tanks is shown in Fig.1.

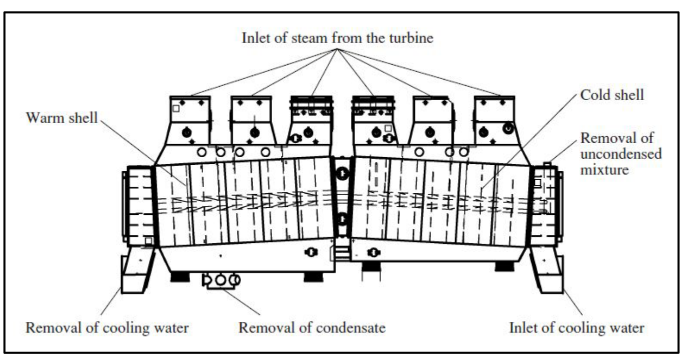

Fig. 1. A, longitudinal, single pass condenser with four shell tanks [7]

In this theoretical study, only longitudinal condenser design has been considered. The number of shell tanks is varied in the range 1-4 while the length of each shell tank is varied in the range 8-14m. The condenser tube diameter is taken 16mm with tube wall thickness of 0.5mm since these values are common for thermal power plants [7].
For designing a condenser, the mass flow rate on both shell and tube side are to be known. In order to calculate the mass flow rate of condensate in shell-side and coolant in tube-side, equations (1) and (2) may be used respectively.

$$\dot{m}_{shell} = \frac{\dot{Q}}{h_{fg}}$$

$$\dot{m}_{tube} = \frac{\dot{Q}}{C_p\left(T_{c,out} - T_{c,in}\right)}$$

The condenser coolant outlet temperature is usually very near to that of the tube wall temperature, which is again around $10^0\mathrm{C}$-$15^0\mathrm{C}$ less than condensate temperature. In this study, the tube wall temperature is taken $10^0\mathrm{C}$ less than condensate temperature and coolant outlet temperature is assumed to be approaching the wall temperature. Also, for the initial guess, overall heat transfer coefficient U is assumed to be 200 W/m.K
If U, $\dot{Q}$ and $LMTD$ are known, the surface area of the tubes may be determined from equation (3).

$$A_s = \frac{\dot{Q}}{U(LMTD)} \tag{3}$$

In this work, the electrical output of the plant is taken $1200MW_e$ and thermal efficiency is taken nearly 60% after superheating, thus around $800MW_t$ thermal load has been assumed on the condenser. The log-mean temperature difference ($LMTD$) may be calculated from equation (4).

$$\boldsymbol{LMTD} = \frac{(T_{c,o}) - (T_{c,i})}{\ln\frac{(T_{cond} - T_{c,o})}{(T_{cond} - T_{c,i})}} \tag{4}$$

Then based on the assumed tube diameter and tube length, the number of tubes may be calculated from equations (5).

$$N_t = \frac{A_s}{\pi . D_{tube} . L} \tag{5}$$

The tube bundle diameter may be calculated from equation (6),

$$D_b = D_{tube}\left(\frac{N_t}{K_1}\right)^{\frac{1}{n_1}} \tag{6}$$

Here single pass and square pitch arrangement are considered, thus K_1 will be 0.215 and n_1 will be 2.207 [20].Then the shell diameter may be calculated from equation (7),

$$D_s = D_b + BDC \tag{7}$$

Here, the bundle diameter clearance is a function of bundle diameter and may be obtained from suitable charts [20].
The velocity of coolant flow may be calculated from equation (8) and equation (9).

$$Gm = \frac{\dot{m}_c}{N_t} \tag{8}$$

(1)

$$v_t = \frac{Gm}{\rho_c} \tag{9}$$

The desirable range of tube-side flow velocity is 0.5-1.5 m/s to maintain optimum heat transfer coefficient as well as avoid over-turbulence.

Tube-side heat transfer coefficient may be calculated from equation (10) given by Eagle and Ferguson (1930) specifically for water [20].

$$h_{tube} = \frac{4200(1.35 + 0.02T_{mean})v_t^{\,0.8}}{d_i^{\,0.2}} \tag{10}$$

The shell-side heat transfer coefficient for condensation on a single horizontal tube surface may be calculated from equation (11) derived by Nusselt [21].

$$h_{cond} = 0.729\left\{\frac{g\rho_{h,l}(\rho_{h,l} - \rho_{h,v})h^*_{fg}(k_{h,l})^3}{\mu_{h,l}(T_h - T_w)d_0}\right\}^{0.25} \tag{11}$$

This equation does not provide accurate result for vacuum condenser due to air leakage and presence of non-condensable gas in steam. Experimental result shows that shell-side heat transfer coefficient approximately 10% of the calculated value. Thus the shell-side heat transfer coefficient may be calculated from equation (12),

$$h_{shell} = \frac{0.1(h_{cond})}{(N_{blank})^{0.25}} \tag{12}$$

The approximate average number of tubes in a condenser tube blank may be calculated from the equation (13),

$$N_{blank} = \sqrt{N_t} \tag{13}$$

Then the actual overall heat transfer coefficient may be calculated from equation (14),

$$\boldsymbol{U} = \left(\frac{\boldsymbol{1}}{\boldsymbol{h_s}} + \frac{\boldsymbol{1}}{\boldsymbol{h_t}}\right)^{-\boldsymbol{1}} \tag{14}$$

If the calculated overall heat transfer coefficient is close to the assumed one, then the other results (surface area of the tubes, number of tubes, condenser diameter) may be considered. Otherwise the calculated value may be used in equation (5) to calculate the surface area of the tubes and continued until the calculated value is close to the assumed value.

For condenser design, the preferable velocity of coolant flow is 0.5-1.5m/s. If any design condition results in velocity less than 0.5m/s, will not be acceptable since it should result in very low heat transfer coefficient at the tube side of the condenser. Similarly, velocity which is greater than 1.5m/s is also not suitable since it may result in over-turbulence. Also the preferable range of total tube length to shell diameter ratio is 5 to 10 [20]. Any design condition outside this range is dis

3 Results and Discussion

Table 1 shows the design parameters obtained for condensers with single shell tank, with and without cooling towers. From Table 1, it may be observed that no design option was obtained are acceptable for single shell tank, both with and without cooling tower. This is because all the possible designs have resulted in unacceptable flow velocities of inappropriate length to shell diameter ratio.

Table 2 shows the design parameters obtained for condensers with 2 shell tanks. From Table 2, it may be observed that for condenser with cooling tower (50^0C condensate temperature), no preferable design options are found. For 8m shell length, both tube side flow velocity and length to shell diameter ratio are below the preferable value. For 10-14m shell length its tube side flow velocities are within the preferable range (0.5m/s – 1.5m/s). However, the length to shell diameter ratios are much lower than the minimum acceptable value. Even for condensers without cooling tower, no preferable design options are found. For shell tank length up to 12m, both tube side flow velocities and length to shell diameter ratios are below the preferable value for all condensate temperatures. For 14m shell length and 70^0C condensate temperature, length to shell diameter ratio is below preferable value. On the other hand, for 14m shell length and 75^0C and 80^0C condensate temperature, length to shell diameter ratios are 5.117 and 5.331 respectively, which are within preferable range but tube side flow velocities are below the preferable value, thus can't be acceptable.

Table 3 shows the design parameters obtained for condensers with 3 shell tanks. From Table 3, it may be observed that for condenser with cooling tower (50^0C condensate temperature) and 8m and 10m shell length, tube side flow velocities are within the preferable range but length to shell diameter ratios are not. However, for 12m and 14m shell length, both tube side flow velocities and length to shell diameter ratios are within the preferable range. Therefore, these design options are acceptable. For condenser without cooling tower and 70^0C condensate temperature, tube side flow velocities and length to shell diameter ratios are both within the preferable range for shell length of 10-14m. Therefore, these design options are acceptable. Only for 8m shell length, both tube side flow velocity and length to shell diameter ratio are less than the preferable value. Similar results are observed for other condensate temperatures.

Table 4 shows the design parameters obtained for condensers with 4 shell tanks. From Table 4, it may be observed that for condenser with cooling tower (50^0C condensate temperature) and 8m shell length, tube side

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flow velocity is within the preferable range but length to shell diameter ratio is below the preferable value. For 10m shell length, both tube side flow velocity and length to shell diameter ratio are within the preferable range, thus this design option is acceptable. For 12m shell length, length to shell diameter ratio is within the preferable range but its tube side flow velocity is above the preferable value. For 14m shell length, both tube side flow velocity and length to shell diameter ratio are above the preferable value. For condenser without cooling towers, it may observe that for 8- 10m shell length, tube side flow velocities as well as length to shell diameter ratios are within the preferable range, thus these design options are acceptable. For 12-14mnshell length, tube side flow velocities are within the preferable range but their length to shell diameter ratios are above the preferable value.

TABLE 1
CALCULATED DESIGN PARAMETERS FOR SINGLE SHELL TANK

Condensate Temperature (^{0}C)		Shell length(m)	Single shell tank 8	10	12	14
With cooling tower	50^0C	Number of tubes	722753	543278	431838	356441
		Shell diameter(m)	14.522	12.761	11.50	10.544
		Tube-side hear transfer coefficient $(W/m^2.^0C)$	1255.469	1577.54	1895.576	2210.079
		Shell-side hear transfer coefficient $(W/m^2.^0C)$	213.764	221.529	201.513	233.513
		Overall hear transfer coefficient $(W/m^2.^0C)$	180.86	192.484	201.797	209.556
		Tube side flow velocity(m/s)	0.173	0.23	0.289	0.35
		Length to shell diameter ratio	0.551	0.784	1.043	1.328
Without cooling tower	70^0C	Number of tubes	464081	342625	269030	220088
		Shell diameter (m)	11.882	10.357	9.283	8.476
		Tube-side hear transfer coefficient $(W/m^2.^0C)$	899.697	1146.876	1391.652	1634.159
		Shell-side hear transfer coefficient $(W/m^2.^0C)$	248.833	258.452	266.384	273.154
		Overall hear transfer coefficient $(W/m^2.^0C)$	192.248	208.319	221.088	231.645
		Tube side flow velocity (m/s)	0.101	0.137	0.175	0.213
		Length to shell diameter ratio	0.673	0.966	1.293	1.652
	75^0C	Number of tubes	426530	313978	246042	200987
		Shell diameter (m)	11.437	9.956	8.915	8.135
		Tube-side hear transfer coefficient $(W/m^2.^0C)$	876.916	1120.467	1361.792	1600.972
		Shell-side hear transfer coefficient $(W/m^2.^0C)$	256.324	266.33	274.573	281.603
		Overall hear transfer coefficient $(W/m^2.^0C)$	195.086	212.015	225.463	236.576
		Tube side flow velocity (m/s)	0.095	0.129	0.165	0.202
		Length to shell diameter ratio	0.699	1.004	1.346	1.721
	80^0C	Number of tubes	391557	287838	225348	183957
		Shell diameter (m)	11.00	9.571	8.568	7.816
		Tube-side hear transfer coefficient $(W/m^2.^0C)$	867.614	1109.797	1349.833	1587.775
		Shell-side hear transfer coefficient $(W/m^2.^0C)$	263.87	274.218	282.737	290.001
		Overall hear transfer coefficient $(W/m^2.^0C)$	199.442	217.046	231.029	242.581

		Tube side flow velocity (m/s)	0.0915	0.124	0.159	0.195
		Length to shell diameter ratio	0.727	1.045	1.401	1.791

TABLE 2
CALCULATED DESIGN PARAMETERS FOR 2 SHELL TANK

Condensate Temperature (^{0}C)		Shell length (m)	2 shell tank 8	10	12	14
With cooling tower	50^0C	Number of tubes	302296	230145	184569	153350
		Shell diameter (m)	9.786	8.65	7.827	7.198
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	2521.461	3136.156	3741.723	4339.615
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	238.372	246.637	253.536	259.477
		Overall hear transfer coefficient ($W/m^2.^0$ C)	216.204	227.188	236.073	243.542
		Tube side flow velocity (m/s)	0.413	0.542	0.676	0.813
		Length to shell diameter ratio	1.635	2.312	3.066	3.89
Without cooling tower	70^0C	Number of tubes	185389	139770	111343	92056
		Shell diameter (m)	7.843	6.902	6.227	5.714
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	1874.578	2349.845	2818.651	3281.921
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	279.076	289.105	297.44	304.597
		Overall hear transfer coefficient ($W/m^2.^0$ C)	240.626	255.33	267.1	276.908
		Tube side flow velocity (m/s)	0.253	0.336	0.422	0.51
		Length to shell diameter ratio	2.04	2.90	3.854	4.90
	75^0C	Number of tubes	169110	127292	101291	83678
		Shell diameter (m)	7.524	6.616	5.966	5.472
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	1838.156	2307.154	2769.889	3227.234
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	287.748	298.149	306.787	314.201
		Overall hear transfer coefficient ($W/m^2.^0$ C)	246.023	261.477	273.832	284.118
		Tube side flow velocity (m/s)	0.24	0.319	0.401	0.486
		Length to shell diameter ratio	2.127	3.023	4.022	5.117
	80^0C	Number of tubes	154701	116360	92544	76424
		Shell diameter (m)	7.226	6.352	5.727	5.252
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	1823.761	2290.446	2750.949	3206.123
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	296.348	307.089	316.006	323.658
		Overall hear transfer coefficient ($W/m^2.^0$ C)	252.399	268.453	281.281	291.954
		Tube side flow velocity (m/s)	0.232	0.308	0.387	0.469
		Length to shell diameter ratio	2.214	3.148	4.191	5.331

TABLE 3
CALCULATED DESIGN PARAMETERS FOR 3 SHELL TANK

Condensate Temperature (^{0}C)		Shell length (m)	3 shell tank 8	10	12	14
With cooling tower	50^0C	Number of tubes	184569	141194	113613	94630
		Shell diameter (m)	7.827	6.934	6.284	5.785
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	3741.723	4636.028	5516.405	6385.186
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	253.536	262.169	269.389	275.616
		Overall hear transfer coefficient ($W/m^2.^0$ C)	236.073	246.877	255.674	263.109
		Tube side flow velocity (m/s)	0.676	0.883	1.098	1.318
		Length to shell diameter ratio	3.066	4.327	5.729	7.26
Without cooling tower	70^0C	Number of tubes	111343	84589	67742	56226
		Shell diameter (m)	6.227	5.499	4.973	4.571
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	2818.651	3511.711	4194.561	4868.822
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	297.44	307.835	316.501	323.959
		Overall hear transfer coefficient ($W/m^2.^0$ C)	267.099	281.261	292.677	302.248
		Tube side flow velocity (m/s)	0.422	0.555	0.693	0.835
		Length to shell diameter ratio	3.854	5.456	7.239	9.188
	75^0C	Number of tubes	101291	76865	61507	51022
		Shell diameter (m)	5.966	5.266	4.761	4.375
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	2769.889	3454.106	4128.347	4794.178
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	306.787	317.554	326.526	334.244
		Overall hear transfer coefficient ($W/m^2.^0$ C)	273.832	288.68	300.633	310.643
		Tube side flow velocity (m/s)	0.401	0.529	0.661	0.797
		Length to shell diameter ratio	4.023	5.697	7.562	9.60
	80^0C	Number of tubes	92544	70191	56147	46562
		Shell diameter (m)	5.727	5.054	4.569	4.198
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	2750.949	3431.926	4103.022	4765.778
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	316.00	327.118	336.375	344.338
		Overall hear transfer coefficient ($W/m^2.^0$ C)	281.281	296.687	309.083	319.459
		Tube side flow velocity (m/s)	0.387	0.51	0.638	0.77
		Length to shell diameter ratio	4.191	5.936	7.880	10.005

TABLE 4

CALCULATED DESIGN PARAMETERS FOR 4 SHELL TANK

Condensate Temperature (^{0}C)		Shell length (m)	4 shell tank			
			8	10	12	14
With cooling tower	50^{0}C	Number of tubes	130719	100259	80820	67407
		Shell diameter (m)	6.696	5.939	5.387	4.962
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	4930.902	6096.767	7244.056	8375.95
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	264.707	273.633	281.105	287.555
		Overall hear transfer coefficient ($W/m^2.^0$ C)	249.992	260.755	269.56	277.028
		Tube side flow velocity (m/s)	0.954	1.244	1.543	1.85
		Length to shell diameter ratio	4.779	6.736	8.911	11.285
Without cooling tower	70^{0}C	Number of tubes	78176	59633	47891	39832
		Shell diameter (m)	5.306	4.695	4.251	3.912
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	3740.371	4644.942	5535.689	6414.893
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	310.884	321.585	330.523	338.224
		Overall hear transfer coefficient ($W/m^2.^0$ C)	285.316	299.225	310.496	319.986
		Tube side flow velocity (m/s)	0.60	0.787	0.98	1.179
		Length to shell diameter ratio	6.031	8.52	11.29	14.316
	75^{0}C	Number of tubes	71016	54123	43439	36112
		Shell diameter (m)	5.081	4.493	4.068	3.742
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	3679.874	4573.091	5452.759	6321.096
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	320.711	331.788	341.035	349.00
		Overall hear transfer coefficient ($W/m^2.^0$ C)	292.927	307.483	319.262	329.170
		Tube side flow velocity (m/s)	0.572	0.751	0.936	1.125
		Length to shell diameter ratio	6.298	8.902	11.80	14.965
	80^{0}C	Number of tubes	64841	49397	39634	32943
		Shell diameter (m)	4.876	4.311	3.903	3.59
		Tube-side hear transfer coefficient ($W/m^2.^0$ C)	3656.636	4545.709	5421.339	6285.723
		Shell-side hear transfer coefficient ($W/m^2.^0$ C)	330.376	341.803	351.342	359.558
		Overall hear transfer coefficient ($W/m^2.^0$ C)	301.093	316.184	328.39	338.652
		Tube side flow velocity (m/s)	0.553	0.725	0.904	1.088
		Length to shell diameter ratio	6.563	9.278	12.299	15.60

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If the sizes of the condensers are compared, it may be observed that the condensers with cooling towers, greater shell lengths and number of shell tanks are required in order to satisfy the limiting conditions compared to the condensers without cooling towers. Therefore, using cooling tower should significantly increase the cost of condenser. Also, the installation and maintenance cost of cooling tower is very high. As a result, it may be opined that used of cooling tower is not feasible for thermal power plants located in tropical region.

4 CONCLUSION

In this work, the feasibility of using cooling tower in the tertiary coolant circuit of a thermal power plant in tropical region is studied from the condenser designing point of view. Theoretical analyses are conducted to find the design parameters of condensers with and without cooling towers. The number of shell tanks is varied in the range 1-4 while the length of each shell tank is varied in the range 8-14m. Condenser tube diameter is taken 16mm with tube wall thickness of 0.5mm. Condenser coolant inlet temperature is assumed to be around 280C, since it is the average reservoir water temperature in tropical region countries during summer season.

For condenser with cooling tower, temperature of coolant outlet from condenser is taken 400C. For condensers with no cooling tower, three coolant outlet temperatures have been considered; 600C, 650C and 700C. For condenser design, the preferable velocity of coolant flow is 0.5-1.5m/s while the preferable range of total tube length to shell diameter ratio is taken 5 to 10. Results indicate that only a few preferable design options are available for the condenser with cooling tower compared to the condenser without cooling tower. Also, the condensers with cooling towers require greater shell lengths and number of shell tanks compared to the condensers without cooling towers. As a result, using cooling tower should increase the cost of condenser significantly. Therefore, use of condenser in thermal power plants located in tropical region is not recommended in this study.

This study has only analyzed the design constraints of condenser associated with the selection of tertiary coolant circuit. However, the effect of using cooling tower on the overall efficiency of the plant is not considered in this study. Further study may be conducted to observe the influence of cooling tower on the efficiency of a thermal power plant located in tropical region.

Nomenclature

$\dot{Q}$	Condenser load, W
U	Overall heat transfer coefficient, W/m.K
A	Area, m2
$LMTD$	Log-mean temperature difference, 0C
h	Convective heat transfer coefficient, W/m.K
T	Temperature, 0C
L	Length, m
D	Diameter, m
C_p	Heat capacity, J/kg.K
ρ	Density, kg/m3
k	Thermal Conductivity, W/m2.K
μ	Dynamic viscosity, kg/m.s
$\dot{m}$	Mass flow rate, kg/s
v	Velocity, m/s
BDC	Bundle diameter clearance, m

Subscripts,

tube	Tube side data
shell	Shell side data
c	Coolant
s	Tube wall surface
i	Inlet
o	outlet
$inner$	Inner
$outer$	Outer
$mean$	Mean
$blank$	Tube blank
b	Bundle
l	Liquid phase
g	Gaseous phase
cond	Condensation
fg	Liquid-vapor interface

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