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MODELLING OF A DELUGEABLE FLAT BARE TUBE BUNDLE FOR AN AIR-COOLED STEAM CONDENSER

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Abstract:-

In this paper, a one dimensional model was developed for the analytical evaluation of the thermal performance of a delugeable flat tube bundle to be incorporated in the second stage of an induced draft hybrid (dry/wet) dephlegmator (HDWD) of a direct air-cooled steam condenser (ACSC). A one-dimensional model is analysed by using three methods of analysis which are: Poppe, Merkel, and heat and mass transfer analogy. The model's accuracy was validated through a comparison of solutions obtained from the above-mentioned methods of analysis. Satisfactory correlations between the results were reached. However, heat transfer rate attained by Poppe method is higher by 2.89% and 9.87% than that obtained by Merkel, and heat and mass analogy methods, respectively. The difference in air-side pressure drop obtained by all the methods found to be insignificant. Furthermore, the best configuration of the flat tube bundle for the second stage of induced draft HDWD was identified through the comparison of its performance to the round tube bundle. The performance of the round tube bundle is found to be around 2 and 1.5 times of that of flat tube bundle, when both bundles operate in wet and dry operating modes respectively.

Keywords: - *Flat, Bare, Delugeable, Dephlegmator, Heat and mass transfer, wet, dry, Onedimensional*

1. INTRODUCTION

Recent increase in water tariffs coupled with shortage of water in the arid regions, led to the widespread application of air-cooled systems to reject heat into the environment in power plants incorporating steam turbines. However, these cooling systems experience performance penalties during the hot periods, which results in reduction of the output power of the steam turbine. This paper aims to evaluate the thermal performance of a delugeable flat bare tube bundle to be incorporated in the second stage of an induced draft HDWD of a direct ACSC. The HDWD is to replace a dry convectional dephlegmator in each street of an ACSC. The dephlegmator expels the non-condensable gases, stabilises and accelerates the steam flow in the primary condenser units by drawing out and condensing the excess steam. Therefore, it is worthwhile to enhance the dephlegmator performance, since this will directly lead to the overall performance improvement of the ACSC.

HDWD was proposed as an enhancement technique in an attempt to improve the performance and availability of the ACSC during the high ambient temperature, as well as to level the power production rate by lowering fluctuations caused by the ambient condition's changes, and minimise the water usage. The HDWD is found to be cost effective and uses around 20% less water than for the pre-cooling technology (Heyns & Kröger, 2012).

The HDWD can be forced or induced draft as shown in Figure 1. The HDWD has two stages connected in series and combined in one condenser unit. The first stage has inclined finned tubes and second stage has horizontal smooth galvanised steel tubes. The second stage operates in the dry mode as an ACC during cold or off peak periods, and in the wet mode as evaporatively cooled condenser during hot and peak periods. Deluge water sprayed on the surface of the second stage tubes during the wet operating mode is collected in collecting troughs under the tube bundle, while the water droplets blown up by the air are trapped by the drift eliminator above the tube bundle. In order to mitigate the corrosion and fouling risk during the wet operating mode, the smooth bare tubes are used in the second stage.

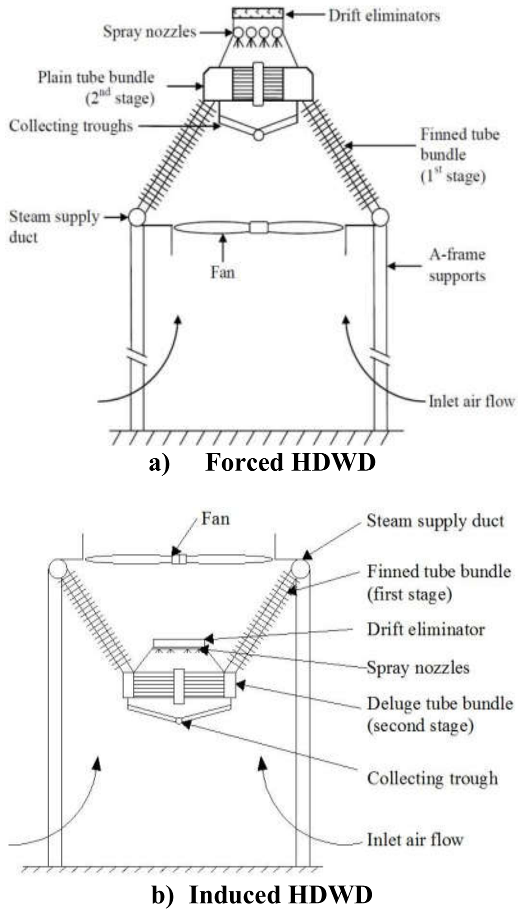


Figure 1: Schematic diagram of the HDWD

Analysis of the thermal performance characteristics of the evaporatively cooled heat exchangers has been a subject of several works. Some of the previous studies were performed by Parker & Treybal, (1961) and Mizushina, et al., (1967) who conducted the analytical and numerical analysis respectively, of the evaporative cooler. Authors applied the assumptions presented in Merkel F. (1926) which are: the Lewis factor equal to one, the evaporation of the deluge water was neglected and the enthalpy of the saturated air was considered being a linear function of the temperature. The deluge water temperature was taken to be constant. Authors such as Niitsu, et al., (1969), Nakayama, et al., (1988), and Hasan & Siren , (2003) conducted a comparative analysis between the round plain and finned tube bundle of an evaporative heat exchanger. The plain tube bundle showed higher mass transfer coefficient than finned. However, due to the large surface area of the finned tube bundle, the heat transfer rate for the finned tube bundle found to be higher.

The performance of the finned round tube bundle of an evaporative heat exchanger have been evaluated by several researchers such as Yang & Clark , (1975) and Leidenfrost & Korenic, (1979), while Zhang, et al., (2014) conducted an experimental study on the louver fin flat tube heat exchanger. The thermal performance characteristic of the oval tube bundle, operating under wet operating conditions, has been investigated in some studies such as in Hasan & Siren , (2004) and Zheng , et al., (2012).

There are numerous recently studies performed in an attempt to find a best configuration of the HDWD second stage's tube bundle. Heyns & Kröger, (2012), and Owen, (2013) investigated the performance of a delugeable bundle of 38 mm and 19 mm diameter round tubes for the HDWD, respectively. When the HDWD operates in wet operating mode, Owen, (2013) reported that the HDWD performance is two to three times of the convectional dephlegmator.

Reuter & Anderson, (2016) evaluated the performance characteristics of a round bare tube air-cooled heat exchanger bundle for the second stage of a HDWD during wet and dry operation. The authors determined the thermal performance, and heat and mass transfer coefficients from the experimental data, as well as from an Effectiveness-NTU model, whereby the Merkel method of analysis was employed during wet operation.

As stated above, in most of the studies round tube bundles were considered. The literatures lack information on specific performance investigation of the bare flat tube bundle operating under deluging conditions as well as the comparison of its performance to other bundle of different tubes geometries operating under similar conditions. Therefore, this paper evaluates the thermal performance of the bare flat tube bundle for an induced draft HDWD of an ACSC.

2. Materials and methods

2.1. One dimensional model

The thermal performance of a delugeable flat tube bundle is evaluated by means of one dimensional model. A schematic of two tubes in a delugeable flat tube bundle for the ACSC is shown in Figure 2.

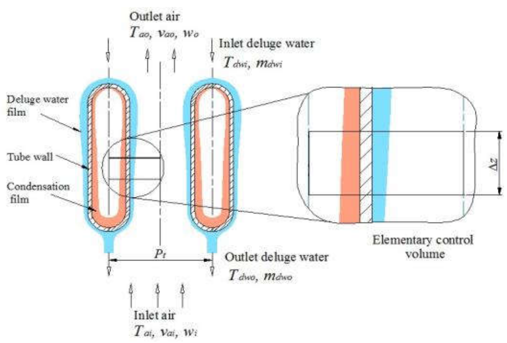


Figure 2: Schematic diagram of two adjacent tubes in a delugeable flat bare tube bundle

An elementary control volume is drawn from the centreline of the tube to the symmetry line between the two adjacent tubes as indicated in Figure 2. The elementary control volume of the delugeable horizontal bare flat tube bundle is shown in Figure 3. The counter-current flow configuration of air and water over the surface of the tubes is considered. The steam flows inside the tubes in the x-direction, and is condensed by the deluge water and cooling air on the exterior of the tubes. A condensation film is formed inside the tubes, and runs down the tube wall along the height due to gravity. The recirculating deluge water is distributed over the tube surface in the negative z-direction, while the inlet air is drawn upward from the bottom of the tubes in the $d$irection. The heat is transferred from the condensed steam through the condensation film, tube wall, deluge water film, and finally crosses the air-water interface to the air stream. Due to the direct contact of air and water at the air-water interface, the deluge water evaporates into the air stream.

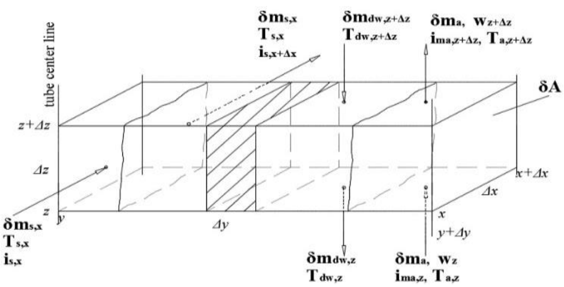


Figure 3: The elementary control volume of the delugeable horizontal bare flat tube bundle

2.2. Governing equations of a one-dimensional model

2.2.1. Tube bundle operated as an evaporative condenser

The governing equations are derived from the basic principle of conservation of mass and energy, and by using the approach presented in Kröger, (2004). The overall mass balance for the elementary control volume in Figure 3 is:

$$\delta m_s + \delta m_{dw,z+\Delta z} + \delta m_a(1 + w_z) = \delta m_s + \delta m_{dw,z} + \delta m_a(1 + w_{z+\Delta z}) \quad (1)$$

That can be simplified as

$$\Delta\delta m_{dw} = \delta m_a \Delta w \quad (2)$$

And the energy balance is:

$$\delta m_s i_{s,x} + \delta m_{dw,z+\Delta z} c_{pdw} T_{dw,z+\Delta z} + \delta m_a i_{ma,z} = \delta m_s i_{s,x+\Delta x} + \delta m_{dw,z} c_{pdw} T_{dw,z} + \delta m_a i_{ma,z+\Delta z} \quad (3)$$

Eq. (3) can be simplified as

$$\Delta T_{dw} = \frac{1}{\delta m_{dw,z} c_{pdw}} \left(\delta m_s \Delta i_s + \delta m_a \Delta i_{ma} - \Delta\delta m_{dw,z} c_{pdw} T_{dw,z}\right) \quad (4)$$

The evaporation rate of the deluge water into the non-saturated air stream is given by Dalton's evaporation law as

$$\Delta\delta m_{dw} = h_d(w_{sw} - w)\delta A_a \quad (5)$$

At the air-water interface, the heat transfer is due to the temperature and water vapour concentration difference between the saturated air at the interface and the air stream. Therefore, heat transfer across the air-water interface is expressed as

$$\delta m_a \Delta i_{ma} = h_a(T_{dw} - T_a)\delta A_a + h_d i_v (w_{sw} - w)\delta A_a \quad (6)$$

Through the Lewis factor which is $Le_f = h_a/c_{pma}h_d$, the Eq. (6) can be simplified as

$$\Delta i_{ma} = \frac{h_d \delta A_a}{\delta m_a}\left[Le_f(i_{masw} - i_{ma}) + (1 - Le_f) i_v (w_{sw} - w)\right] \quad (7)$$

In terms of the overall heat transfer coefficient, the heat transfer rate from the condensed steam to the deluge water film can be expressed as

$$\delta m_s \Delta i_s = U_a \delta A_a (T_s - T_{dws}) \quad (8)$$

Where U_a is the overall heat transfer coefficient, and it is determined as

$$U_a = \left[\frac{1}{h_c} + \frac{t_t}{k_t} + \frac{1}{h_{dw}}\right]^{-1} \quad (9)$$

For Merkel's simplified model (Merkel, 1926), the deluge water temperature is assumed to be constant, the deluge water evaporation is neglected, Lewis factor is equal to one ($Le_f = 1$), and outlet air is assumed to be saturated with water vapour. Therefore, the heat transfer from the condensed steam to the deluge water is equal to the heat transfer from the air-water interface to the air stream.

$$\delta m_a \Delta i_{ma} = U_a (T_s - T_{dws})\delta A_a \quad (10)$$

The deluge water's surface temperature can be determined from Eq. (10) as

$$T_{dws} = T_s - \frac{m_a(i_{mao} - i_{mai})}{U_a A_a} \quad (11)$$

For Poppe's formulation (Poppe & Rögener, 1984) , the heat transfer processes taking place within the one-dimensional control volume of a delugeable flat tube bundle are described by differential equations (2), (5), (7), (8) and (10).

2.2.2. Tube bundle operated as a dry air-cooled condenser

The tube bundle is operated as a dry air-cooled condenser during the cold and off-peak periods. The governing equation for a dry air-cooled condenser can be derived from Figure 3 and through the same procedures as in section 2.2.1. However, deluge water is neglected, and therefore $m_{dw} = 0\ kg/s$. The governing equation for a dry air-cooled condenser can be written as

$$m_a c_{pa}(T_{ao} - T_{ai}) = UA_a(T_s - T_a) \qquad (12)$$

Where U is overall heat transfer coefficient, and is determined as

$$U = \left[\frac{1}{h_c} + \frac{t_t}{k_t} + \frac{1}{h_a}\right]^{-1} \qquad (13)$$

2.3. Solution methods

The bundle's performance is investigated by employing three methods of analysis, which are: Poppe, Merkel and heat and mass transfer analogy. The major assumption made in all these methods is the constant temperature difference between the steam and deluge water or air. For both Poppe and Merkel analysis, the mass transfer coefficient at the air-water interface is attained from the relationship between heat and mass transfer coefficient at the air-water interface through a Lewis factor. The mass transfer coefficient is defined as

$$h_d = \frac{h_a}{c_{pma} Le_f} \qquad (14)$$

For Poppe approach, the Lewis factor is determined by employing Bosjnakovic equation (Bosjnakovic, 1965) as

$$Le_f = 0.866^{0.667}\left(\frac{w_{sw}+0.622}{w+0.622} - 1\right) / ln\left(\frac{w+0.622}{w+0.622}\right) \qquad (15)$$

For the heat and mass transfer analogy method, the mass transfer convection coefficient was determined from the analogy between heat and mass transfer at the air-water interface.

2.4. Validation of the model

In order to validate the accuracy of the model, three methods of analysis are employed and their results are compared. The model's validation was performed based on the following typical designing conditions of the delugeable tube bundle: steam temperature - $T_s = 45$ °C, atmospheric pressure - $p_a = 101325\ Pa$; ambient inlet air temperature - $T_a = 15$ °C; relative humidity - $\phi = 60$ %; air velocity - $v_a = 2\ m/s$; and deluge water mass flow rate - $m_{dw} = 10\ \Delta m_{dw}$, and the tube geometric data: tube outside height - $H_o = 0.6\ m$; tube outside width - $W_o = 0.02\ m$; tube wall thickness $t_t = 0.0015\ m$; and tube pitch - $P_t = 0.028\ m$.

For this validation, the flow on the air-side is considered based on Figure 4. The critical tube height (H_{cr}) at which two air-side boundary layers reach the symmetric line between the tubes as shown in Figure 4 is computed. The flow within the critical height is considered to be a developing flow, and therefore the heat transfer rate and air-side pressure drop within this region are determined by employing the external flow theories. The flow in the rest of the air flow channel (δ_a) is considered to be a fully developed flow.

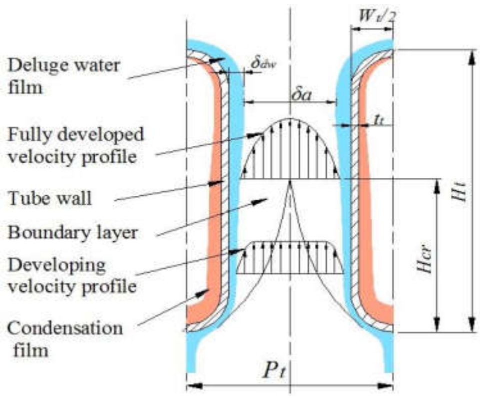


Figure 4: Tube bundle section, illustrating the air-side flow between two adjacent tubes The output data which obtained by different methods of analysis are shown in Table 1.

Table 1: Comparison of results obtained by different methods of analysis

Description	Symbol	Units	Method of Analysis		
			Poppe	Merkel	Heat and mass transfer analogy
Condensation heat transfer coefficient	h_c	W/m^2 K	11720.21	11831.16	12093.52
Deluge water mean temperature	T_{dwm}	°C	43.936	44.007	44.062
Deluge water evaporation rate	Δm_{dw}	kg/s	0.001585	0.001420	0.001416
Deluge water heat transfer coefficient	h_{dw}	W/m^2 K	4412.01	4579.32	4585.80
Outlet air temperature,	T_{ao}	°C	27.261	29.846	27.224
Outlet air humidity ratio	w_o	kg/kg	0.02948	0.02707	0.02714
Air-side heat transfer coefficient (critical region)	$h_{a,cr}$	W/m^2 K	19.453	19.499	19.425
Mass transfer coefficient (critical region)	$h_{d,cr}$	m/s	0.01991	0.01878	0.01781
Air-side heat transfer coefficient (fully developed region)	$h_{a,fd}$	W/m^2 K	40.013	39.995	39.746

Table 1: Comparison of results obtained by different methods of analysis, continued.....

Description	Symbol	Units	Method of Analysis		
			Poppe	Merkel	Heat and mass transfer analogy
Mass transfer coefficient (fully developed region)	$h_{d,fd}$	m/s	0.04095	0.03852	0.03577
Heat transfer rate	Q	W	4903.93	4767.16	4463.33
Air-side pressure drop	ΔP	p_a	30.915	31.001	30.555
Tube critical height	H_{cr}	m	0.30324	0.30266	0.30176

The heat transfer rate achieved by Poppe method is found to be higher than that obtained by Merkel, and heat and mass analogy methods by 2.89% and 9.87%, respectively. The discrepancies between solutions are mainly attributed to the assumptions of constant temperature difference between steam and deluge water or air made in the model. The difference in air-side pressure drop yielded from all methods is found to be insignificant. Therefore, the model is demonstrated to be valid for the analysis of the thermal performance of a delugeable flat bare tube air-cooled steam condenser bundle.

2.5. Performance analysis

The best configuration of a delugeable flat tube bundle was selected through a comparison of its performance to round tube bundle presented in Reuter & Anderson, (2016). The comparison is performed by investigating the effect of the tube pitch, tube height, and the steam flow area on the heat transfer rate ratio (Q_r/Q_f) and air-side pressure drop ratio ($\Delta P_r/\Delta P_f$) of the round and flat bundles. This study is conducted for the constant frontal area and fan power, which is equal to the value of the round tube bundle. Furthermore, the deluge water mass flow rate is taken to be seven and half times the evaporation rate ($m_{dw} = 7.5\ \Delta m_{dw}$), as also considered for the round tube bundle. The Merkel method of analysis is employed in evaluation of the performance of the considered tube bundle configuration, since it is the same method that has been used in Reuter & Anderson, (2016). The critical performance measures are identified to be heat transfer rate, airside pressure drop, and fan power. The best tube bundle configuration is found to be the one that provides a higher rate of heat transfer at a reasonable air-side pressure drop.

The round tube bundle considered by Reuter & Anderson, (2016), is designed in an attempt to enhance the performance of round tube bundles presented by Heyns & Kröger, (2012) and Owen, (2013). The bundle is a multi-row with 39 tubes per row and 25 rows. The round tube bundle consists of tubes of 15.8 and 19 mm inner and outer diameter respectively, which are in a triangular arrangement with the tube pitch of 38 mm. The above mentioned bundle is found to be 2.4 m wide, 1 m high and 2.5 m long. The performance data of this bundle are shown in Table 2.

Table 2: The performance data of a round tube bundle presented by Reuter & Anderson, (2016)

Description	Symbol	Units	Operating mode	
			wet	dry
Heat transfer rate	Q	MW	5.12	0.627
Air-side pressure drop	ΔP	P_a	22.70	22.50
Fan power	P_F	W	417.98	417.89

To examine the impact of tube pitch on the performance ratio of the bundles, four tube pitches of 23; 25; 30 and 25 mm are considered. For each tube pitch, the performance ratio was analysed for three different steam flow areas of $A_{sf1} = 0.368\ m^2$, $A_{sf2} = 2A_{sf1}$ and $A_{sf3} = 2A_{sf1}$; whereby the first steam flow area is equal to the round tube bundle's steam flow area. The tube width corresponding to each steam flow area was computed. This investigation is conducted for both wet and dry operating modes, and the tubes' height is kept constant at 1 m. The number of the tubes per row computed for each considered pitch is shown in table 3

Table 3: Number of tubes per row for the flat tube bundle configurations

Description	Symbol	Units				
Tube pitch	P_t	mm	23	25	30	35
Number of tubes per row	n_{tr}		125	115	96	82

The effect of the tube height on the performance ratio of the bundles was examined for three tube heights of 0.5; 1; and 1.2 m. The tube pitch of 30 mm was considered in this investigation, despite a low performance obtained at this pitch compared to the performance yielded at the tube pitch of 23 and 25 mm. The investigation of the tube height effects at small tube pitches is found to become impossible and unrealistic for short tubes as the steam flow area changes from A_{sf2} to A_{sf3}.

3. Results and Discussions

3.1. Influence of tube pitch on the performance ratio of the bundles

In Figure 5 and 6 the heat transfer rate ratios (Q_r/Q_f) and air-side pressure drop ratios ($\Delta P_r/\ \Delta P_f$) of the round and flat tube bundle are illustrated. From Figure 5, it can be seen clearly that, for a constant tube pitch, there is a slight variation in the obtained results for different tube widths. However, a large difference in the yielded results at various tube pitches is observed.

This shows that the tube pitch has a measurable effect on the bundle's performance than the tube width. Furthermore, it is found that for small tube pitches, the tubes with small width perform better than those with large widths. Additionally, it is also noted that at the large tube pitches, there is a slight increase in heat transfer rate for the wider tubes. This is due to the fact that for large tube pitches, the air-side pressure drop between the tubes is low. Therefore, since the fan power is constant, the air velocity between tubes is increased in order to achieve the targeted fan power. As shown in Figure 6, the air-side pressure drop ratio is found to increase as the tube pitch increases, and drop as the tube width increases.

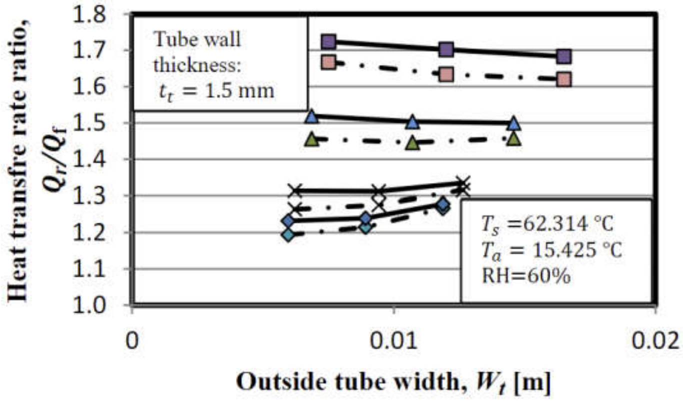


Figure 5: Effect of the tube pitch on the heat transfer rate ratio of round and flat tube bundle

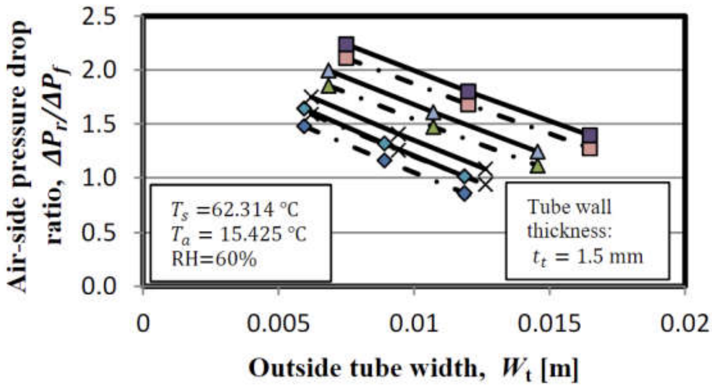
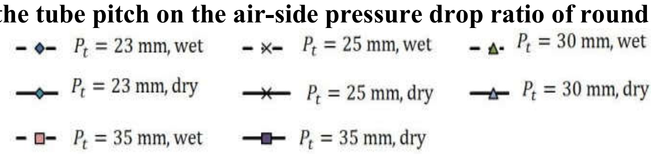


Figure 6: Effect of the tube pitch on the air-side pressure drop ratio of round and flat tube bundle



3.2. Influence of tube height on the performance ratio of the bundles

Figure 7 and 8 displays the obtained ratios of the heat transfer rate (Q_r/Q_f) and air-side pressure drop ($\Delta P_r/\Delta P_f$) of a round and flat tube bundle. The results show clearly that the short tube height has a measureable negative impact in the bundle performance especially for the large tube width. This is due to the fact that for the constant steam flow area, the tube becomes wider when its height decreases.

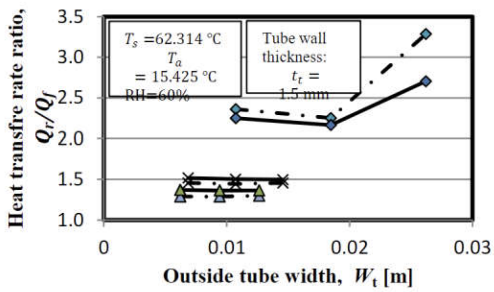


Figure 7: Effect of the tube height on the heat transfer rate ratio of round and flat tube bundle

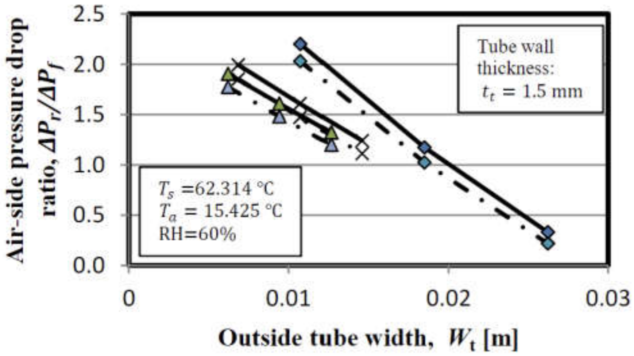
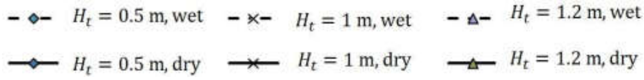
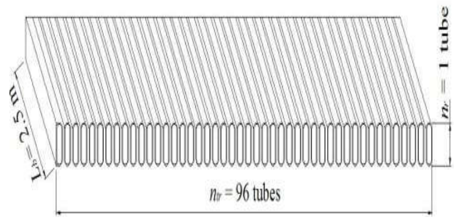


Figure 8: Effect of the tube height on the air-side pressure drop ratio of round and flat tube bundle

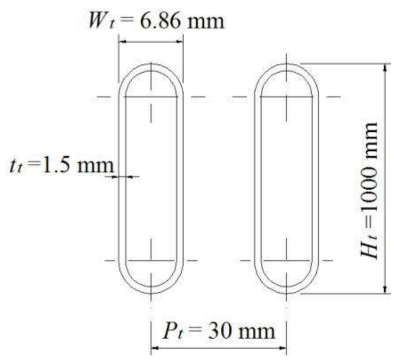


3.3. Tube bundle selection

The best flat tube bundle configuration is selected based on the dimensions and steam flow area of the round tube bundle. From the data presented in section 4.1 and 4.2, the small tube pitches deliver high performance than large pitches. However, as mentioned early their performance analysis becomes unrealistic and impossible at some points. Therefore, the chosen bundle consists of tubes of 1 m long, inside and outside width of 3.86 and 6.86 mm respectively, and arranged with a tube pitch of 30 mm. The tube layout and dimensions of the flat tube bundle's configuration are demonstrated in Figure 9. As it is already illustrated in Figures 5 to 8, round tube bundle performs better than the flat tube bundle. In comparison with the selected flat tube bundle configuration, the performance of the round tube bundle is found to be around 2 times, and 1.5 times, when both bundles operate as an evaporative and dry air-cooled condenser respectively.



a) Tube layout



b) Tube dimensions

Figure 9: Tube bundle layout and dimensions

4. Conclusions

In this paper the one-dimensional model employed in the analysis of the thermal performance of the deluged flat tube bundle of a second stage of an induced draft HDWD of an ACSC was developed. This model was used to identify the best flat tube bundle configuration, through parametric study whereby the tube pitch, steam flow area, and tube height were varied, and their effects on the heat transfer rate and pressure drop were observed. One-dimensional model is presented by a set of governing differential equations. For the validation of the model, the governing equations were solved analytically by means of Poppe, Merkel and heat and mass analogy methods of analysis and their results were compared.

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Nomenclature

A	Area	m^2	Subscripts	
c_p	Specific heat at constant pressure	[J/kg K]	a	Air
d	Diameter of duct	[m]	cr	Critical
H	Height	[m]	c	Condensate
h	Heat transfer coefficient	[W/m K]	dw	deluge water
h_d	Mass transfer coefficient	[$kg/m^2 s$]	dwm	Deluge water mean
i	Enthalpy	[J/kg]	dws	Deluge water surface
k	Thermal conductivity	[W/ mK]	f	Flat
L	Length	[m]	o	outlet
m	Mass flow rate	[kg/s]	sf	Steam flow
P	Pitch	[m]	sw	Saturated water
Q	Heat transfer rate	[W]	t	Tube
RH	Relative humidity	[%]	tr	Tube row
t	Thickness	[m]		
T	Temperature	[°C]		
U	Overall heat transfer coefficient	[W $/m^2K$]		
W	Width	[m]		
w	Humidity ratio	[kg / kg]		