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# Analytical Sensitivity Analysis of the Performance characteristics of aDelugeableFlat Bare Tube Bundleto Operating Conditions and Geometric Parameters

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**ABSTRACT:** This paper analyses the sensitivity of the performance characteristics (heat transfer rate and air-side pressure drop) for both wet and dry operating modes of aDelugeableFlat Bare Tube Bundle (DFBTB) of an induced draft Hybrid (Dry/Wet) Dephlegmator (HDWD)'s second stage for a direct Air-Cooled Steam Condenser (ACSC) to the operating conditions (deluge water mass flow rate, frontal air velocity, steam and air operating conditions) and geometric parameters of the tube bundle (tube height, tube pitch, tube width). The validated one-dimensional analytical model developed based on heat and mass transfer analogy method of analysis in Angula[1] was employed in the study. The performance characteristics of the DFBTB were found to be less sensitive to the changes in the deluge water mass flow rate and air operating conditions ( $T_a$ and RH) duringwet operating mode compared to dry operating mode. The long tube height, large tube width, small tube pitch and high frontal air velocity were found to increase the tube bundle's heat transfer rate. However, this is associated with a high air-side pressure drop. The findings of this study were compared to the findings of similar studies performed with delugeable bundles of different tube geometries presented in literatures, and the good agreement wasattained.
**KEYWORDS:** Flat, Bare, Delugeable, Sensitivity, Heat and Mass transfer, Wet, Dry, One-dimensional

## I. INTRODUCTION

Due to water shortage and usage restrictions, the application of ACSCs to reject heat into the environment in power plantsincorporating steam turbines recently became wide. However, ACSCsexperience performance penalties during the hot periods, which results in reduction of the output power of the steam turbine. The HDWD is found to be the economical viable technique to improve the ACSC performance during the hot periods [2] and [3]. The HDWD is incorporated in each Street of the ACSC instead of conventional dephlegmator.The HDWD has first and second stages connected in series and in one condenser unit as shown in Figure 1. The first stage tube bundle geometry and operation aresame to that of conventional dephlegmator, while the second stage has horizontal smooth delugeable tube bundle which operates in the dry mode as an air-cooled condenser during cold or off-peak periods, and in the wet mode as evaporative-cooled condenser during hot and peak periods.The performance of the HDWD is mainly depends on the second stage delugeable tube bundle performance. Therefore, its performance sensitivity to operation conditions, as well as to its geometric parameters is significant.

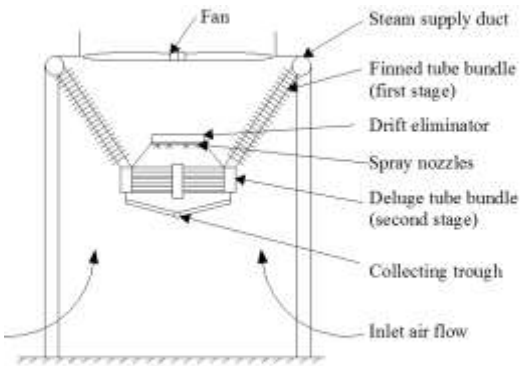



**Figure 1: Schematic diagram of the induced draft HDWD**[1],[4]

Several studies such as Heyns[5];Heyns & Kröger[2];Owen[3];Anderson[6]; Reuter and Andreson[7]; Graaff[8]and Angula[1] and[4]wereperformed to evaluate the HDWD performance through identification of a best configuration of the HDWD second stage's delugeable tube bundle that delivers good performance, as well as its performance sensitivity to some of operation conditions.However, in all above-mentioned studies, the delugeable round bare tube bundles were analyzed, except in Angula[1] and[4] where the DFBTB was evaluated, however the bundle's performance sensitivity analysis to operating conditions was not carried out.Therefore, in this paper, a DFBTB's performance sensitivity to operation conditions, as well as to its geometric parameters was performed.Heyns[5] and Heyns & Kröger[2]found the water film heat transfer coefficient as a function of the air mas velocity, deluge water mass velocity and deluge water temperature, while air-mass transfer coefficient andair-side pressure drop as a function of the air mass velocity and deluge water mass velocity.Both Reuter and Andreson[7] and Graaff[8] through the experimental data and model presented in Anderson[6] found that the air-side pressure drop increases as the air mass velocity increases, however it is insensitive to deluge water mass velocity change.

The performance sensitivity of an evaporative heat exchangers (condensers and coolers) with different tube geometries to operation conditions have been conducted in numerous studies such asYang and Clark[9]; Leidenfrost and Korenic[10]; Jahangeer et al.[11]; Hwang et al.[12] and Zhang et al.[13].Yang and Clark[9]in their experimental analysis of the thermal performance characteristics of three compact plate finned heat exchangers (plain-finned, louvered, and perforated fin heat exchangers), deluged with water and ethylene glycol mixture, found that as the air speed varied from 1.26 to 2.52 m/s, for the low Reynolds number of 500 and 1000, the air-side heat transfer coefficient increased from 40 to 45%, and there were no significant variations noted in the friction factors for all the heat exchangers. In Leidenfrost and Korenic[10]study, the performance of the finned counter flow evaporative condenser was investigated, and the condenser capacity was found increasing as air flow rate increases until the air flow starts to break the water film on the tubes surface.Jahangeer et al.[11]performed a numerical investigation of the plain round tube evaporative-cooled condenser performance for air-conditioning applications. It was found that as the water film thickness increases from 0.075 to 0.15 mm, the heat transfer coefficients decrease. While, as the air velocity increases from 1 to 4 m/s, the tube side and the overall heat transfer coefficient increase from $29000$ to $33000\ \mathrm{W/m^2K}$ and $330$ to $800\ \mathrm{W/m^2K}$, respectively.

In the performance evaluation of the compact round-tube louver-fin condenser, operating under wet and dry conditions at angle of $0°$ and $21°$ with the vertical conducted by Hwang et al.[12], at the air velocity of 1.4 m/s, the deluge water found to increase the heat exchanger capacity by 162% and 181%, and air-side pressure drop by 137% and 135% for the case of $0°$ and $21°$ angle with the vertical, respectively.In the experimental study of Zhang et al.[13]for the evaporative mist pre-cooling; deluging cooling and the combination of the two cooling systems on the louver fin flat tube heat exchanger, it was found that at the higher deluge water flow rate, the rate of water drainage increases faster than the rate of evaporation of water. Furthermore, they found that the heat exchanger capacity and air-side pressure drop increase with water mass flow rate and air velocity, respectively. As stated already, in most of the studies,the performance sensitivitiesof plain or finned round tube bundles or heat exchangers were considered. The literatures lack information on specific performance sensitivityanalysis of the DFBTB to the operating conditions and geometric parameters. Therefore, this paper addressed that.

## II. MATERIALS AND METHODS

The analytical sensitivity study of the heat transfer rate and air-side pressure drop of DFBTB to operating conditions (deluge water mass flow rate, frontal air velocity, steam and air operating conditions) and geometric parameters (tube height, tube pitch, tube width) was performed by means of a one-dimensional analytical model based on heat and mass transfer analogy method of analysis. This model is well presented in Angula [1] and its fully development will not be repeated in this work. In addition to that, the performance evaluation and configuration of DFBTB as well as its performance comparison to delugeable round bare tube bundle arealso presented in Angula[1] and [4].

**Model**

A schematic of two tubes in a DFBTB for the air-cooled condenser is shown in Figure 2. The governing equations were derived from the basic principle of conservation of mass and energy, and by using the approach presented in Dreyer[14] and Kröger[15]. The elementary control volume of the DFBTB as shown in Figure 3, was drawn at location shown in Figure 2.

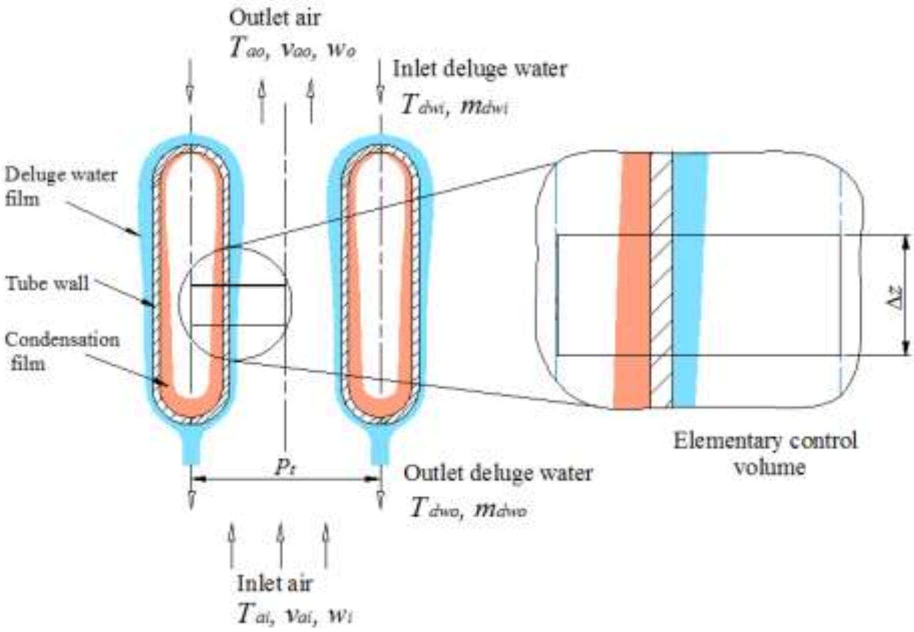



**Figure 2: Schematic diagram of two adjacent tubes in a DFBTB**[1]

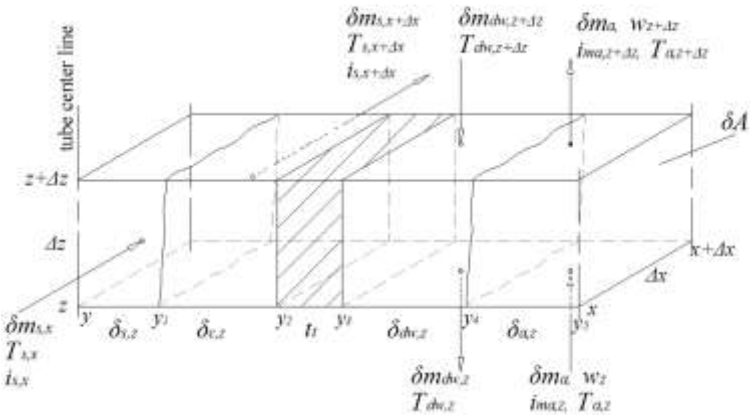



**Figure 3:The elementary control volume of the horizontal DFBTB**[1]

The differential equations describe the heat and mass transfer processes taking place in the elementary control volume are:

$$\Delta\delta m_{dw} = \delta m_a \Delta w \quad (1)$$

$$\Delta\delta m_{dw} = h_d (w_{sw} - w)\delta A_a \quad (2)$$

$$\delta Q_a = \delta Q_{ac} + \delta Q_{am} \quad (3)$$

$$\delta Q_{am} = \delta m_a \Delta w i_v = h_d i_v (w_{sw} - w)\delta A_a \quad (4)$$

$$\delta Q_{ac} = h_a (T_{dw} - T_a)\delta A_a \quad (5)$$

Therefore, Eq. (3) can be re-written as

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

Then,

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

The mass transfer convection coefficient, $h_d$ is determined from the analogy between heat and mass transfer at the air-water interface.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

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

where $U_a$ is the overall heat transfer coefficient.

2.1. Performance Sensitivity Analysis

During the performance sensitivity analysis of the DFBTB, the examined parameter was varied, while keeping the other parameters constant, and then the changes in heat transfer rate and air-side pressure drop were observed. The analysis was executed at two different standard operatingconditions: $T_s = 38$ ℃; $T_a = 32$ ℃; $RH = 38\%$ and$T_s = 45$ ℃; $T_a = 15$ ℃; $RH = 60$ %. The flow on the air-side was 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 was computed. The flow within the critical height was considered to be a developing flow, and therefore the heat transfer rate and air-side pressure drop within this region were determined by employing the external flow theories. The flow in the rest of the air flow channel ($\delta_a$) was considered to be a fully developed flow.

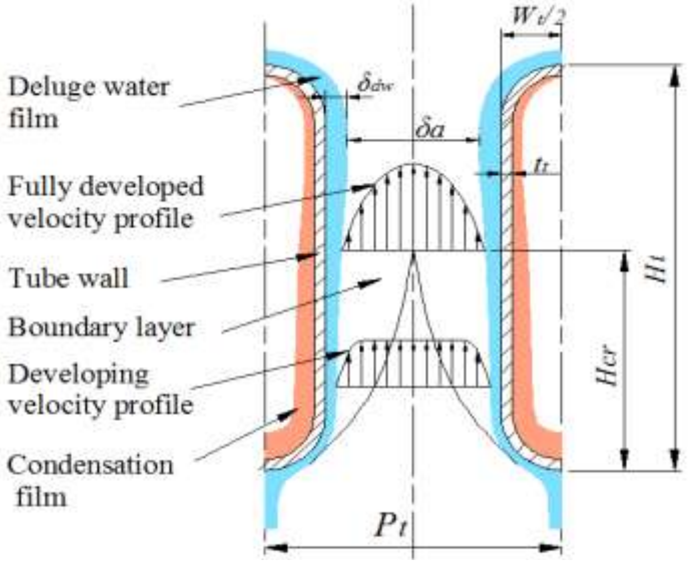


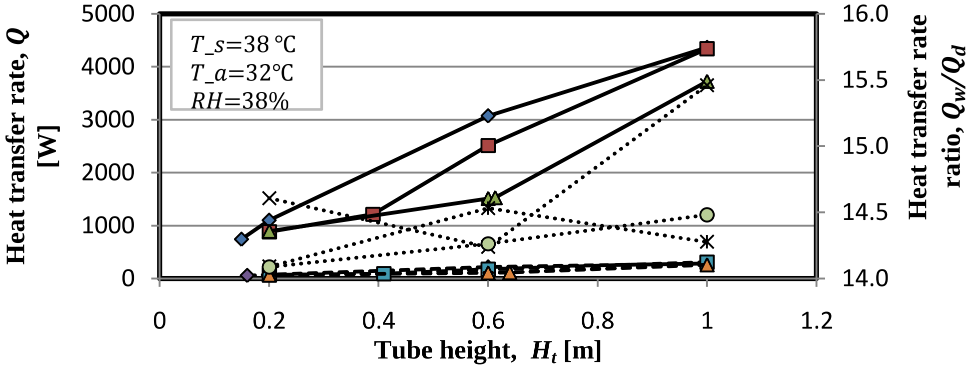

**Figure 4: Tube bundle section, illustrating the air-side flow between two adjacent tubes**[1]

To ensure that the obtained results are independent from the tube geometry, the air velocity between the tubes was kept constant for all cases in which DFBTB's performance sensitivity to air velocity was not investigated. Meanwhile, the deluge water mass flow rate was considered to be ten times the evaporation rate, and tube width was taken to be 20 mm for all cases in which the DFBTB's performance sensitivity to deluge water mass flow rate and tube width were not examined. Throughout the study, the tube wall thickness was kept constant at 1.5 mm. The analysiswas conducted for both wet and dry operating modes.

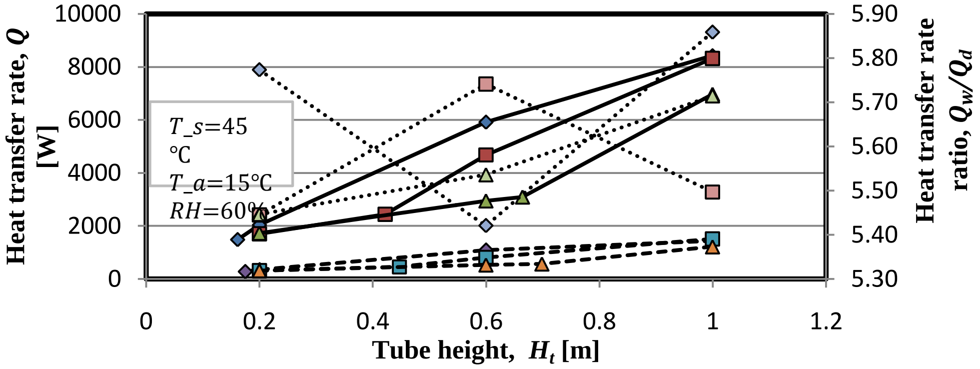
## III. RESULTS AND DISCUSSION

### 3.1. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Tube Height and Tube Pitch

To study the sensitivities of the heat transfer rate and air-side pressure drop to the tube height and tube pitch, the tube height was varied from 0.2 to 1 m, whereas tube pitches of 25, 28 and 30 mm were considered. The obtained results of the heat transfer rate ($Q_w$ and $Q_d$) and air-side pressure drop ($\Delta P_w$ and $\Delta P_d$) for wet and dry operating modesare plotted in Figure 5 to 7. From the yielded results, it was found that the heat transfer rate and air-side pressure drop are sensitive to both tube height and pitch, regardless of the operating conditions and modes. The heat transfer rate and air-side pressure drop increase as the tube height varies from 0.2 to 1 m. This is due to the fact that the tube height increases the heat transfer surface area and the length of the air flow channel between the tubes, which in turn directly impact on the rate of heat transfer and air-side pressure drop. The increase of the tube pitch widens the air flow channel, thins the boundary layer thickness, and lengthens the tube critical height. Consequently, the air-side heat transfer and friction coefficients drop. As a result, the heat transfer rate and air-side pressure drop decrease.




**a)** $T_s = 38$ °C, $T_a = 32$ °C and $RH = 38\%$




**b)** $T_s = 45$ °C, $T_a = 15$ °C and $RH = 60\%$

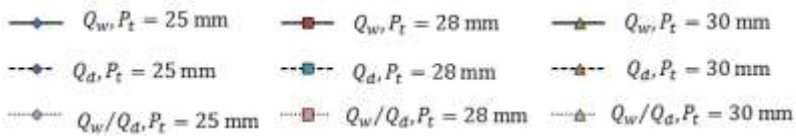



**Figure 5:Sensitivity of heat transfer rate to the tube height and pitch**

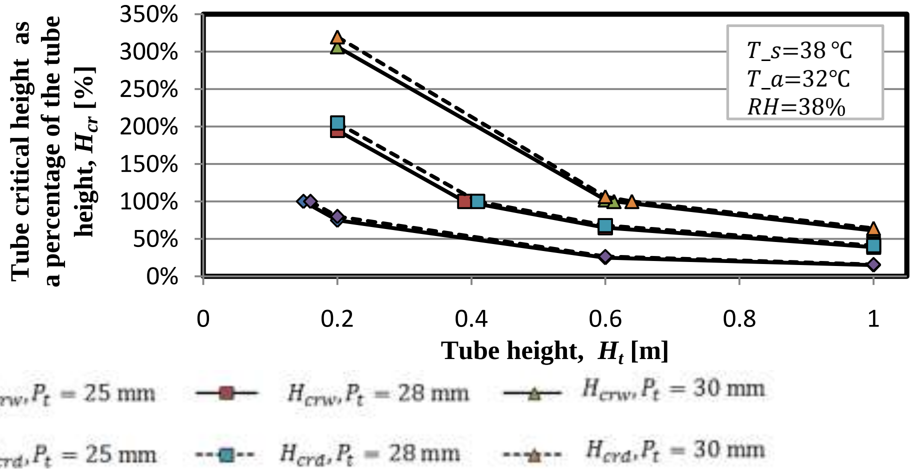
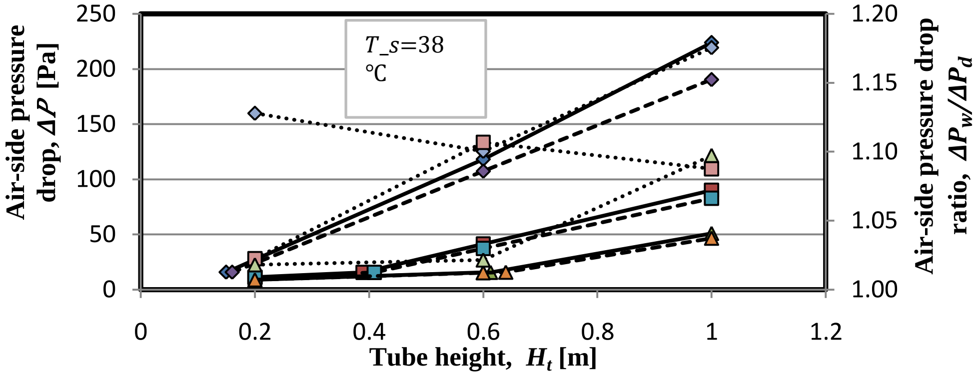
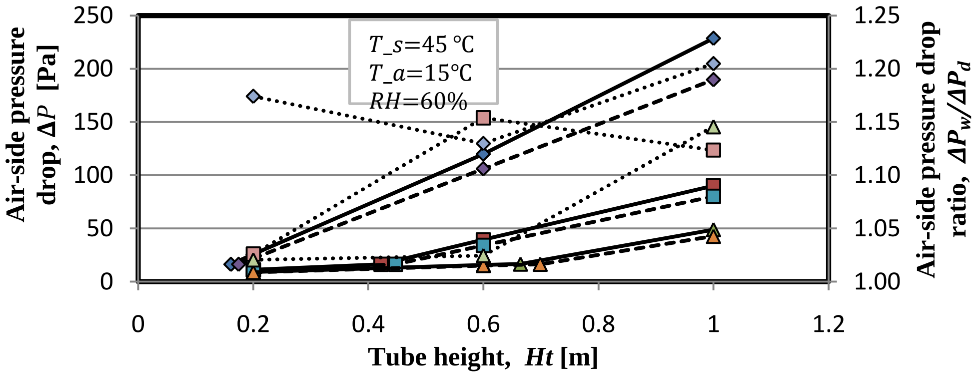



**Figure 6: Tube critical height as a percentage of tube height**




a) $T_s = 38$ ℃, $T_a = 32$ ℃ and $RH = 38\%$




b) $T_s = 45$ ℃, $T_a = 15$ ℃ and $RH = 60\%$

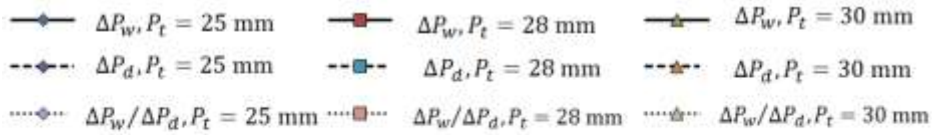
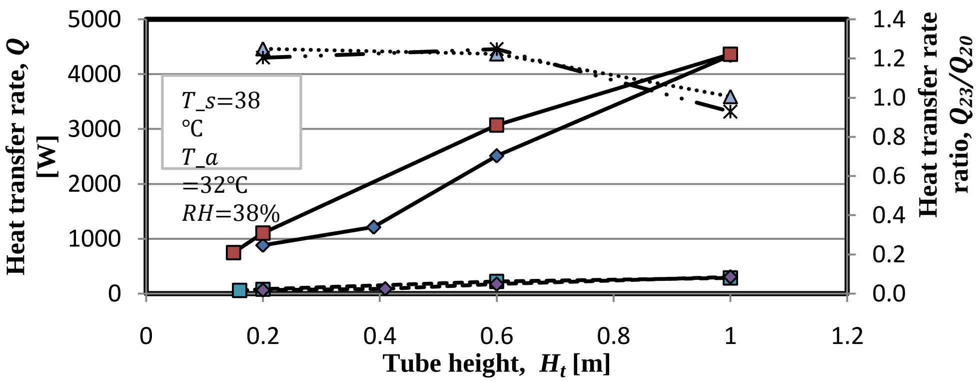


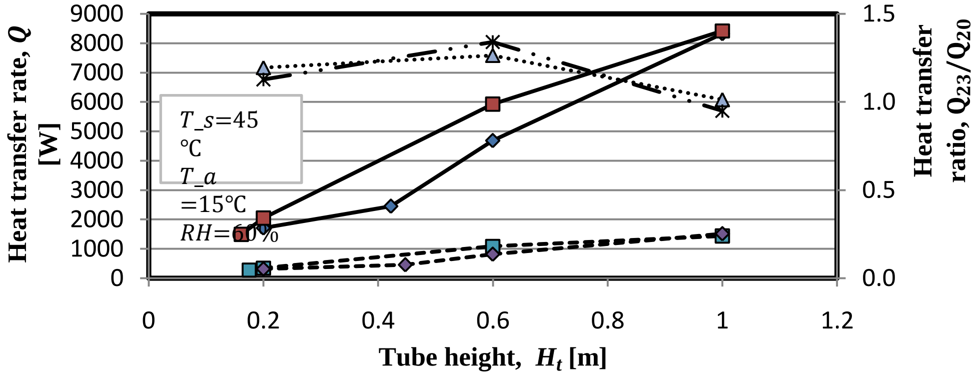

**Figure 7: Sensitivity of air-side pressure drop to tube height and pitch**

**3.2. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Tube Width**

To examine the sensitivities of the heat transfer rate and air-side pressure drop to tube width, the tube width of 20 and 23 mm were considered. The investigation was conducted for a fixed tube pitch of 28 mm, since this tube pitch provides the best heat transfer rate at a reasonable air-side pressure drop. For the same reason, the 28 mm tube pitch was also used in all the investigations presented in sections 3.3, 3.4, 3.5 and 3.6. As depicted in Figure 8 and 9, for both operating modes and conditions, it was observed that as the tube width varies from 20 to 23 mm, both heat transfer rate and air-side pressure drop increase. However, the increase of the air-side pressure drop was found to be higher than that of the heat transfer rate. This is due to the fact that the increase of the tube width reduces the air flow area between the tubes. This thickens the boundary layer thickness, which shortens the tube critical height. Consequently, the air-side heat transfer and friction coefficient increase. For both dry and wet operating modes, air-side pressure drop ratio ($\Delta P_{23}/\Delta P_{20}$) for tube width of 23 mm and 20 mm was found to be greater than two, while the heat transfer rate ratio ($Q_{23}/Q_{20}$) is less than two.



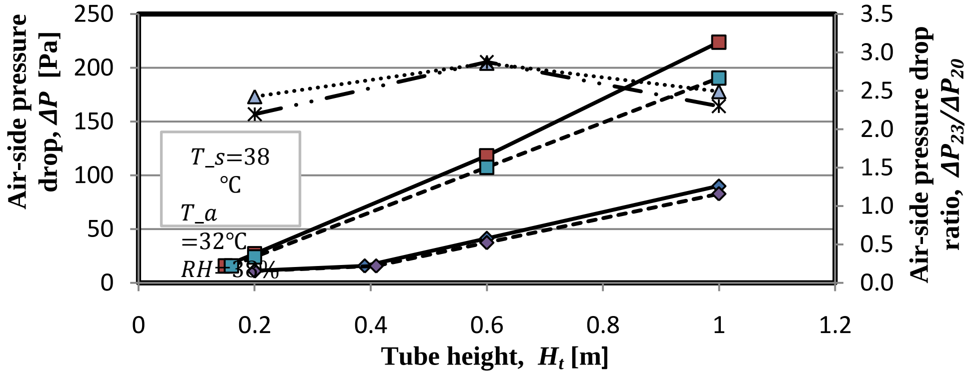

a) $T_s = 38$ ℃, $T_a = 32$ ℃ and $RH = 38\%$



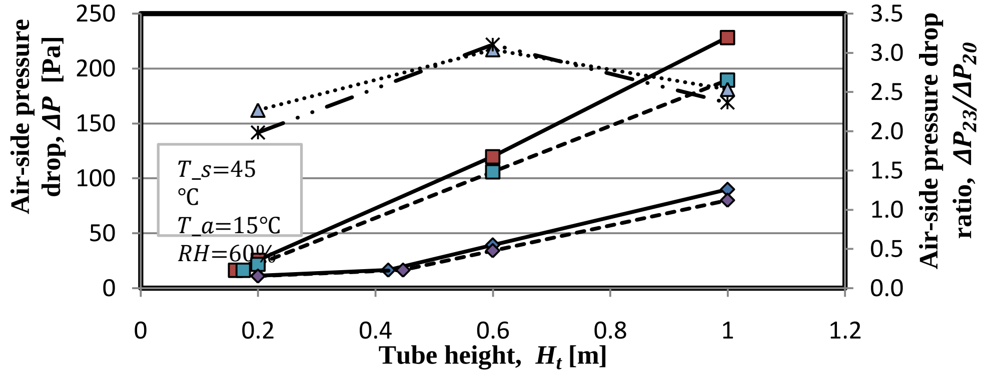

**b)** $\boldsymbol{T_s = 45}$ ℃, $\boldsymbol{T_a = 15}$ ℃ **and** $\boldsymbol{RH = 60\%}$

$Q_w, W_t = 20$ mm — $Q_w, W_t = 23$ mm — $Q_d, W_t = 23$ mm — $Q_d, W_t = 20$ mm — $Q_{w23}/Q_{w20}$ — $Q_{d23}/Q_{d20}$

**Figure 8: Sensitivity of heat transfer rate to tube width**




**a)** $\boldsymbol{T_s = 38}$ ℃, $\boldsymbol{T_a = 32}$ ℃ **and** $\boldsymbol{RH = 38\%}$




**b)** $\boldsymbol{T_s = 45}$ ℃, $\boldsymbol{T_a = 15}$ ℃ **and** $\boldsymbol{RH = 60\%}$

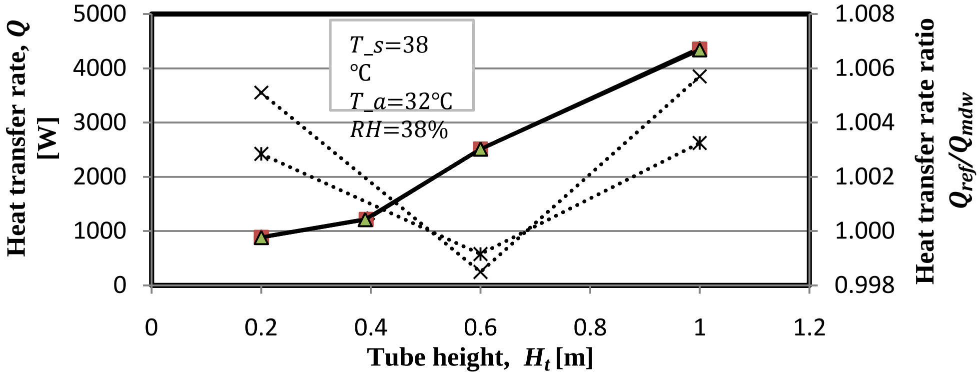
$\Delta P_w, W_t = 20$ mm — $\Delta P_w, W_t = 23$ mm — $\Delta P_d, W_t = 23$ mm — $\Delta P_d, W_t = 20$ mm — $\Delta P_{w23}/\Delta P_{w20}$ — $\Delta P_{d23}/\Delta P_{d20}$

**Figure 9: Sensitivity of air-side pressure drop to tube width**

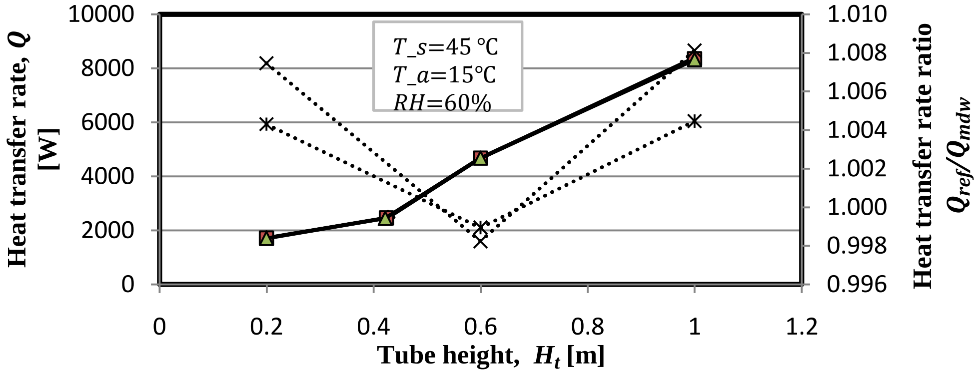
### 3.3. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Deluge Water Mass Flow Rate

The sensitivities of the heat transfer rate and air-side pressure drop to deluge water mass flow rate were examined at three deluge water mass flow rates:$m_{dw1} = 5\,\Delta m_{dw}$, $m_{dw2} = 7.5\,\Delta m_{dw}$ and$m_{dw3} = 10\,\Delta m_{dw}$, whereby $\Delta m_{dw}$ is the rate of evaporation. For the clear demonstration of sensitivity of the heat transfer rate and air-side pressure drop to deluge water mass flow rate, the heat transfer rate ($Q_{mdw\,1} = Q_{ref}$) and air-side pressure drop ($\Delta P_{mdw\,1} = \Delta P_{ref}$) achieved when the deluge water mass flow rate is equal to $m_{dw1}$ , were taken to be reference data.

As can be seen in Figure 10 and 11, the variations in the obtained results are small. This indicates that, the performance of the DFBTB is insensitive to the changes in deluge water mass flow rate. However, from a closer observation it was found that as the deluge water mass flow rate increases from $m_{dw1}$ to $m_{dw3}$there is a slight decrease in the rate of heat transfer and insubstantial increase in the air-side pressure drop. The increase of the deluge water mass flow rate thickens the deluge water film; as a result, the air flow area between tubes reduces. The thick deluge water film may build resistance to the heat transfer, which may result in the drop of the heat transfer rate. At the tube height of 0.6 m, the opposite changes in heat transfer rate were obtained. This is due to the effect of tube critical height which decreases as the deluge water mass flow rate increases. Jahangeer et al.[11] reported the similar findings as they found that, as the water film thickness increases from 0.075 to 0.15 mm, the heat transfer coefficients decrease. In addition to that, these findings also concur withYang and Clark[9]'s experimental data, that showed independent of air-side pressure drop from deluge water mass flow rate and an increase of heat transfer rate with deluge water mass flow rate up to a maximum rate, then no substantial changes were observed.




**a)** $\boldsymbol{T_s = 38}$ ℃, $\boldsymbol{T_a = 32}$ ℃ **and** $\boldsymbol{RH = 38\%}$




**b)** $\boldsymbol{T_s = 45}$ ℃, $\boldsymbol{T_a = 15}$ ℃ **and** $\boldsymbol{RH = 60\%}$

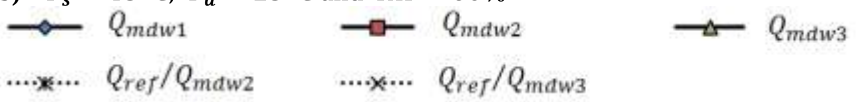
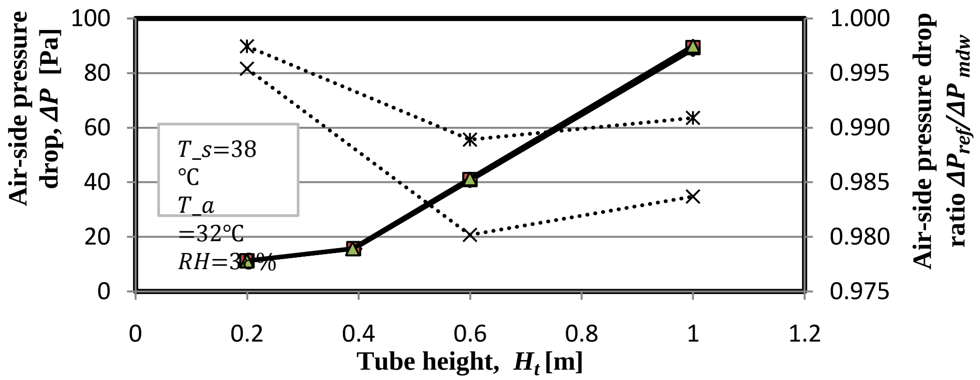
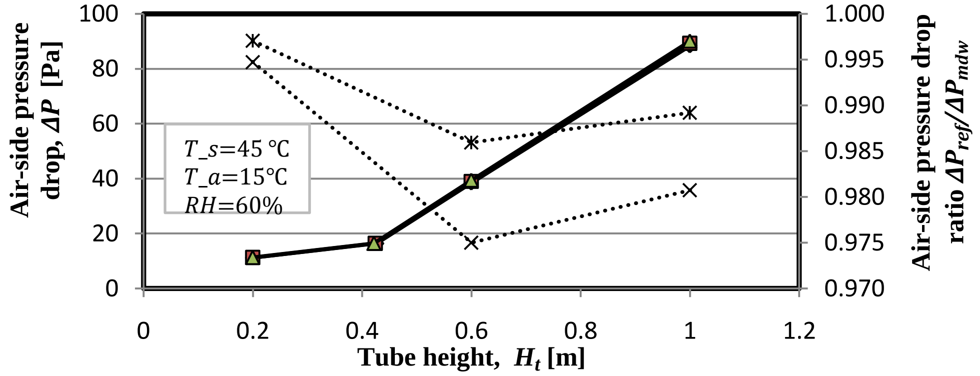



**Figure 10: Sensitivity of heat transfer rate to deluge water mass flow rate**




**a)** $\boldsymbol{T_s = 38}$ ℃, $\boldsymbol{T_a = 32}$ ℃ **and** $\boldsymbol{RH = 38\%}$




**b)** $T_s = 45$ ℃, $T_a = 15$ ℃ and $RH = 60\%$

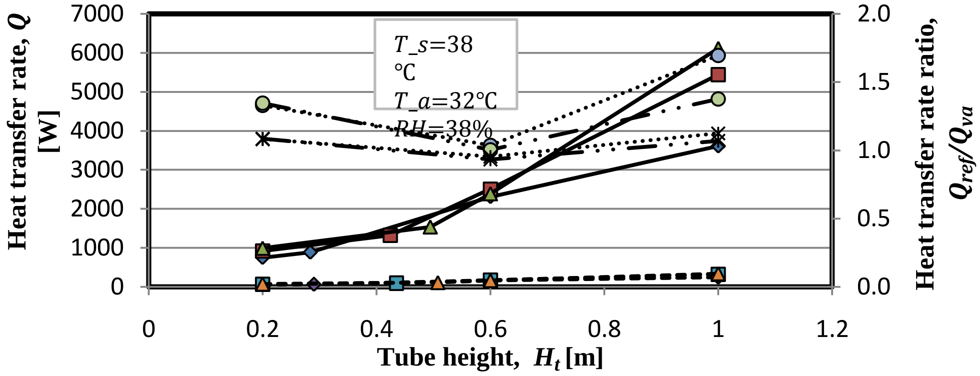
$\Delta P_{mdw1}$ $\Delta P_{mdw2}$ $\Delta P_{mdw3}$

$\Delta P_{ref}/\Delta P_{mdw2}$ $\Delta P_{ref}/\Delta P_{mdw3}$

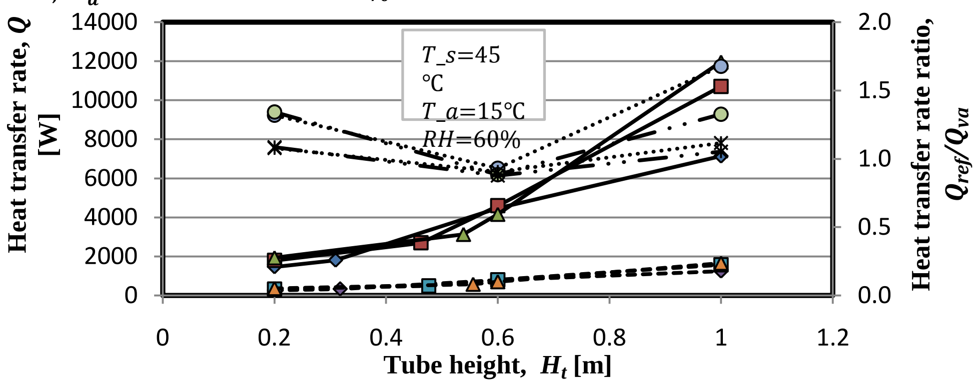
**Figure 11: Sensitivity of air-side pressure drop to deluge mass flow rate**

### 3.4. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Frontal Air Velocity

In order to study the sensitivities of heat transfer rate and air-side pressure drop to frontal air velocity, three frontal air velocities of $v_{a1} = 2\ m/s$, $v_{a2} = 3\ m/s$ and $v_{a3} = 3.5\ m/s$were considered. For the critical analysis, the heat transfer rate ( $Q_{va3} = Q_{ref}$) and air-side pressure drop ($\Delta P_{va3} = \Delta P_{ref}$) obtained at air velocity of $v_{a3} = 3.5\ m/s$were taken to be reference data. For all considered operating conditions and modes, the heat transfer rate and air-side pressure drop were found to increase as the air velocity varies from 2 to 3.5$m/s$as depicted in Figure 12 and 13. Furthermore, it was noted that, the changes in the air-side pressure drop is higher than in the heat transfer rate. This shows that, the high frontal air velocity increases the tube bundle's performance; which is associated with high air-side pressure drop.These results are in good agreement with findings ofYang and Clark[9]; Leidenfrost and Korenic[10]; Jahangeer et al.[11]; Hwang et al.[12] and Zhang et al.[13]as well with experimental results of Reuter and Andreson [7] which was found to concur with Nitsu et al. [16]'s data.For short tube heights, there is an overlap in the heat transfer rate and air-side pressure drop values, which were attained at different air velocities. This is due to the effect of the tube critical height, which increases with the frontal air velocity.




**a)** $T_s = 38$ ℃, $T_a = 32$ ℃ and $RH = 38\%$



**b)** $T_s = 45$ °C, $T_a = 15$ °C **and** $RH = 60\%$

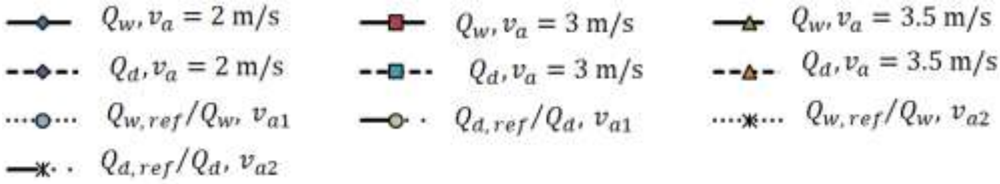
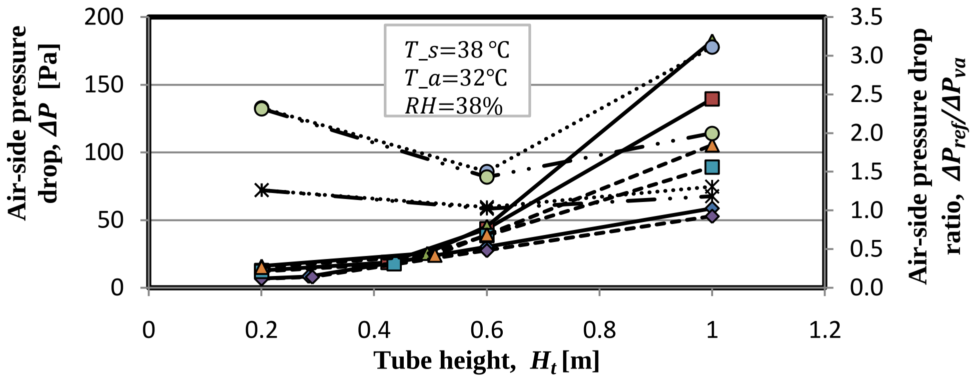
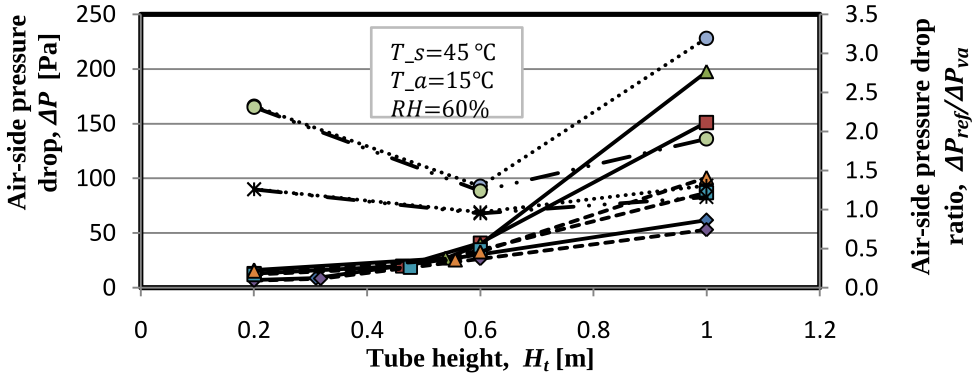



**Figure 12: Sensitivity of heat transfer rate to frontal air velocity**




**a)** $T_s = 38$ °C, $T_a = 32$ °C **and** $RH = 38\%$




**b)** $T_s = 45$ °C, $T_a = 15$ °C **and** $RH = 60\%$

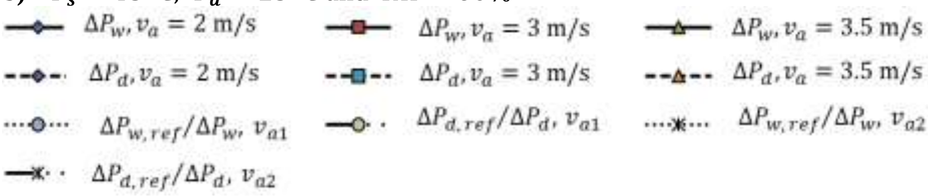



**Figure 13: Sensitivity of air-side pressure drop to frontal air velocity**

### 3.5. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Steam Operating Conditions

The sensitivities of the heat transfer rate and air-side pressure drop to steam operating conditions were investigated for two steam temperatures: $T_s = 45$ °C and $T_s = 60$ °C at an air temperature of $T_a = 15$ °C. Figure14 and 15 show that the heat transfer rate is more sensitive to steam temperature than the air-side pressure drop, and this is more noticeable in wet operating modes than in dry operating modes. When steam temperature changes from 45 to 60 °C, the heat transfer rate increases by 2 times and by 1.5 times for wet and dry operating mode respectively. The sensitivity of air-side pressure drop to steam temperature was found to be inconsequential. However, the air-side pressure drop is slightly higher at high steam temperature than at low steam temperature.

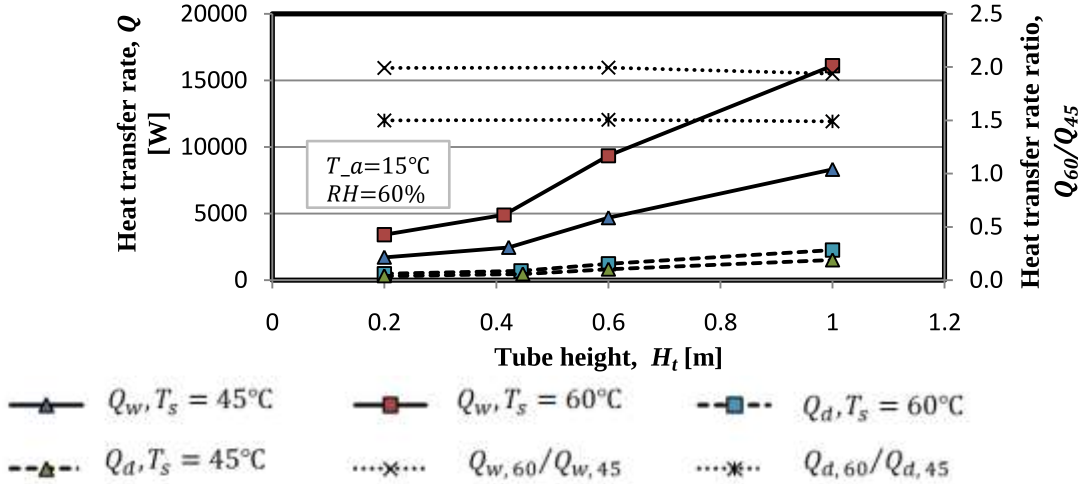



**Figure 14: Sensitivity of heat transfer rate to steam temperature**

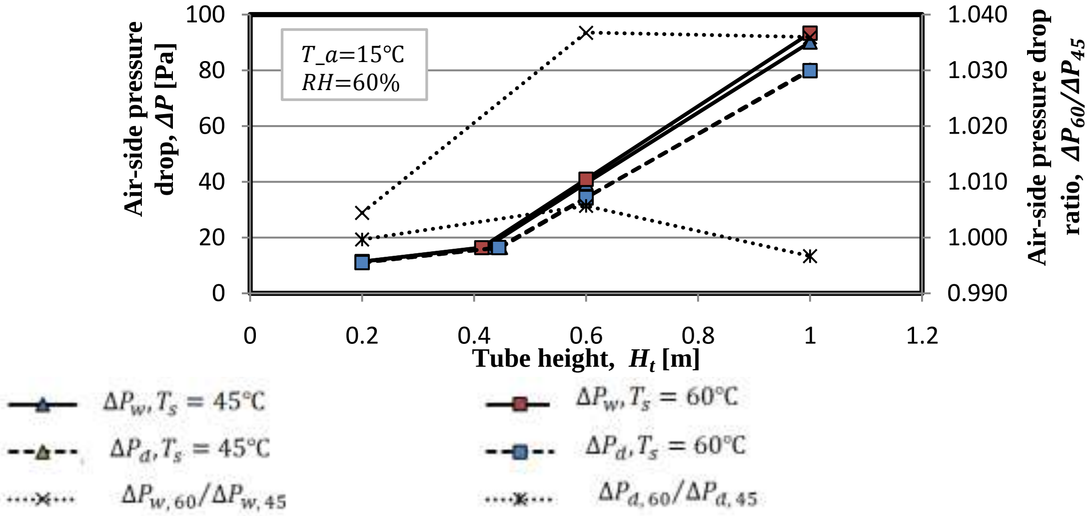
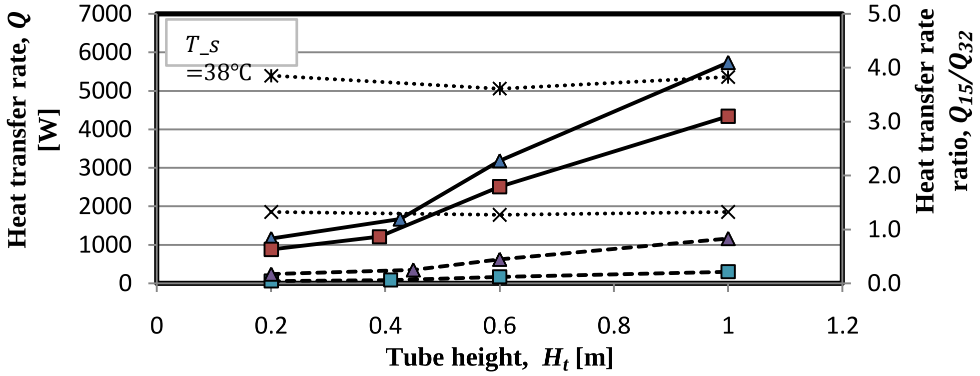


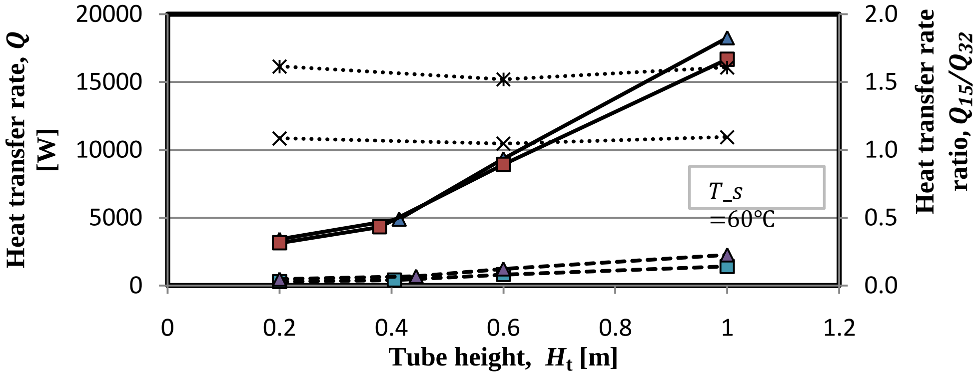

**Figure 15: Sensitivity of air-side pressure drop to steam temperature**

**3.6. Sensitivity of the Heat Transfer Rate and Air-side Pressure Drop to Air Operating Conditions**

The sensitivities of the heat transfer rate and air-side pressure drop to air operating conditions wereexamined for two air operating conditions:$T_a = 15$ °C, RH = 60 %; and $T_a = 32$ °C, RH = 38%; and at two different steam temperatures $T_s = 38$ °C and $T_s = 60$ °C. From the ratios of the heat transfer rate ($Q_{15}/Q_{32}$) and air-side pressure drop ($\Delta P_{15}/\Delta P_{32}$), which wereobtained at different air operating conditions as depicted in Figure 16 and 17, the heat transfer rate was found to be more sensitive to the operating conditions of air than the air-side pressure drop. This is highly noticeable at low steam temperatures in dry operating modes than in wet operating modes. As the air temperature changes from 15 to 32 °C, for dry operating mode, the heat transfer rate drops by 4 times and by 1.5 times for the operation at steam temperature of$T_s = 38$ °C and $T_s = 60$ °C respectively. While for wet operating modes, the heat transfer rate drops by around 1.2 times or less for both steam temperatures. Similar results were reported by Owen[3], where turbine back pressure was found to increase as the ambient air temperature increases, which resulted in decreasing of temperature difference between steam and ambient air ($\Delta T = T_s - T_a$), then as result, the heat transfer rate dropped. From this analysis it is clear that the performance of the delugeable tube bundle is independent from the ambient air conditions. Regardless of the steam temperature, the effect of air operating conditions on the air-side pressure drop was found to be insignificant.




**a)** $\mathbf{T_s = 38}$ ℃




**b)** $\mathbf{T_s = 60}$ ℃

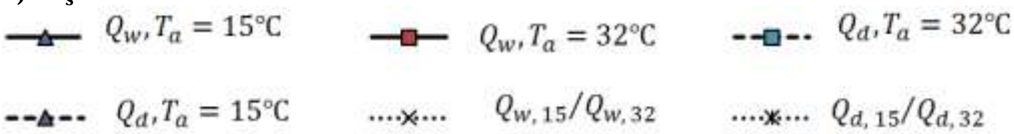
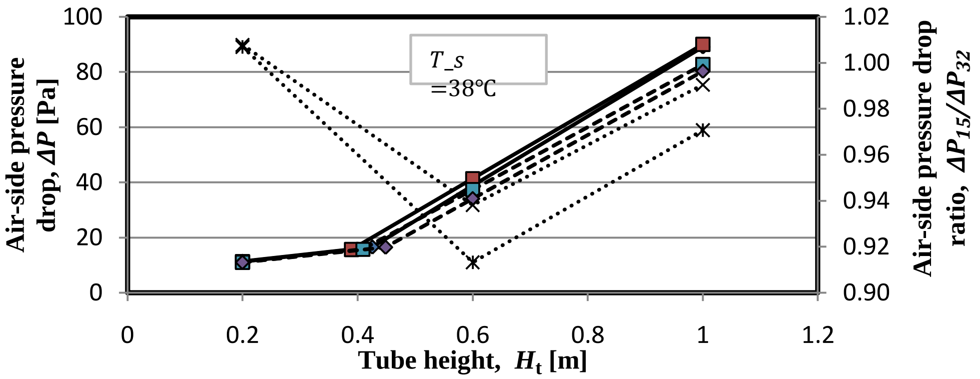
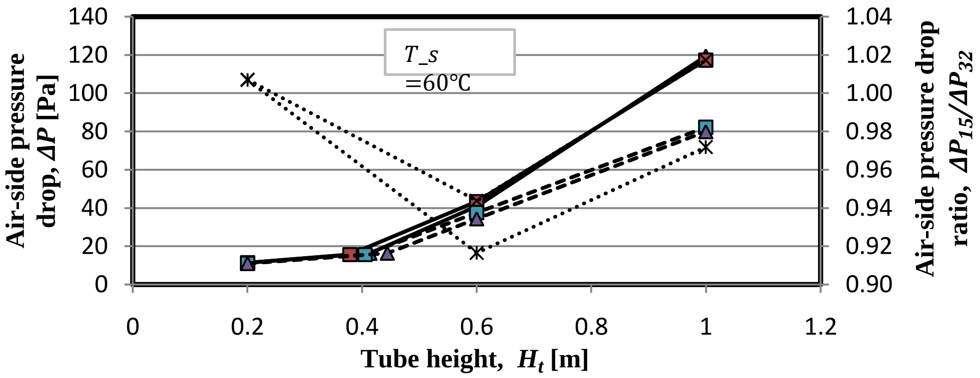



**Figure 16:Sensitivity of heat transfer rate to air operating conditions**




**a)** $\mathbf{T_s = 38}$ ℃




**b)** $\mathbf{T_s = 60}$ ℃

$\Delta P_w, T_a = 15°C$ — $\Delta P_w, T_a = 32°C$ — $\Delta P_d, T_a = 32°C$

$\Delta P_d, T_a = 15°C$ — $\Delta P_{w,15}/\Delta P_{w,32}$ — $\Delta P_{d,15}/\Delta P_{d,32}$

**Figure 17:Sensitivity of air-side pressure drop to air operating conditions**

## IV. CONCLUSION

From the analysis of the performance sensitivity of the DFBTB to operating conditions and its geometric parameters, it was found that the long tube height, large tube width, small tube pitch and high frontal air velocity increase the tube bundle's performance. However, this performance is associated with high air-side pressure drop. Therefore, a careful selection of these parameters is required during the designing and configuration of such tube bundle. The impact of deluge water mass flow rate on the bundle performance is found to be slight; this shows that the tube bundle's performance is insensitive to the variations of deluge water mass flow rate.The heat transfer rate sensitivity to steam temperature is high for wet operating modes than dry modes, and while, the air-side pressure drop sensitivity to steam temperature is insignificant. Therefore, operating the deluged tube bundle at high steam temperature is preferable. The influence of air operating conditions at high steam temperature, and for wet operating modes is found to be low, which shows that the performance of a deluged flat tube bundle depends less on air ambient conditions.

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**NOMENCLATURE**

| | | | | |
|---|---|---|---|---|
| A | Area | $m^2$ | Subscripts | |
| $c_p$ | Specific heat at constant pressure | $[J/kg\ K]$ | a | Air |
| | | | s | Steam |
| | | | w | wet |
| | | | d | dry |
| H | Height | $[m]$ | t | tube |
| h | Heat transfer coefficient | $[W/m\ K]$ | va | Air velocity |
| | | | ac | Convection heat transfer |
| | | | am | Mass transfer |
| | | | cr | Critical |
| $h_d$ | Mass transfer | $[kg/m^2\ s]$ | dw | deluge water |
| | | | dwm | Deluge water mean |

| | | | | |
|---|---|---|---|---|
| | coefficient | | dws | Deluge water surface |
| | | | sw | Saturated water |
| i | Enthalpy | [J/kg] | $y_1, y_2, y_3, y_4, y_5$ | Location or position |
| m | Mass flow rate | [kg/s] | | |
| P | Pitch | [m] | | |
| Q | Heat transfer rate | [W] | | |
| RH | Relative humidity | [%] | | |
| t | Thickness | [m] | | |
| T | Temperature | [℃] | | |
| U | Overall heat transfer coefficient | $[W/m^2K]$ | | |
| W | Width | [m] | | |
| w | Humidity ratio | [kg /kg ] | | |
| x, y, z | Co-ordinate or distance | [m] | | |