Analytical Study on the Thermal Performance of a Partially Wet Constructal T-shaped Fin

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Abstract. The present paper addresses the thermal analysis of a T-shaped fin under partially wet condition by adopting a cubic variation of the humidity ratio of saturated air with the corresponding fin surface temperature. The point separating the dry and wet parts may lie either in the flange or stem part of the fin and so, two different cases having different governing equations and boundary conditions are analyzed in this paper. Since the governing equations are highly non-linear, they are solved by using an analytical technique called the Differential Transform Method and subsequently, the dry fin length, temperature distribution and fin performances are evaluated and analyzed for a wide range of the various psychometric, geometric and thermo-physical parameters. Finally, it can be highlighted that relative humidity has a pronounced effect on the performance parameters when the fin surface is partially wet whereas this effect is marginally small for fully wet surface.

Keywords: Constructal; Simultaneous heat and mass transfer; Partially wet; Analytical; Differential Transform Method

INTRODUCTION

Extended surfaces are widely utilised in various heat exchangers such as evaporator coils of refrigeration and air-conditioning systems, in order to augment the heat transfer rate between the primary surface and the surrounding environment. In such applications, the refrigerant evaporates while flowing through the finned evaporator coil by absorbing heat from the ambient air. If the temperature of the entire fin surface is above the dew point temperature of the air being cooled, then the fin surface remains dry. But if the temperature on the fin surface falls below the dew point temperature, then moisture from the air condenses on the fin surface by releasing latent heat of condensation and hence, the fin surface remains covered with a thin film of water. The fin surface becomes fully wet when the temperature on the entire fin surface is below the dew point temperature of the air being cooled, whereas the fin surface becomes partially wet if the dew point temperature lies between the fin base and fin tip temperature.

Analytical study on wet fins were first carried out by Threlkeld [1] and McQuiston[2]. Annular fin under wet condition was studied by Elmahdy and Briggs [3] by considering a linear variation of the humidity ratio of saturated air with the corresponding fin surface temperature. Later on, this linear model was adopted by various other researchers [4, 5]. Analysis of a partially wet fin is different and complex than that of a fully wet fin. Salah El-Din [4] studied a partially wet fin assembly of straight fins. Fully and partially wet annular fins were analyzed numerically by Sharqawy et al. [5]. Adopting a cubic variation of humidity ratio of saturated air with the fin surface temperature, Naphon [6] analyzed an annular fin under dry, fully wet and partially wet conditions. This cubic relation was also employed in several research works [7, 8, 12, 13]. Kundu et al. [7] developed an analytical model for analyzing fully wet straight fins of triangular profile whereas Kundu and Lee [8] studied analytically fully wet straight fins of rectangular, triangular, convex and exponential profiles. The need for more efficient heat exchanging devices has led to the application of Constructal theory [9] in the design of fins. Thermal performances and optimum design parameter analysis of a constructal T-shaped fin were presented in a few research works [10, 11].

Recently, Hazarika et al. [12, 13] analyzed T-shaped fin under fully wet condition. But, as already mentioned earlier, partially wet surface may result depending upon the psychometric, geometric and thermo-physical parameters. T-shaped fin under partially wet condition has not been analyzed till date. The motive of the present work is to develop an analytical model for the temperature distribution and thermal performance of a T-shaped fin under partially wet condition by adopting a cubic variation of the humidity ratio of saturated air with the corresponding fin surface temperature. Since, the point separating the dry and wet parts may lie either in the flange or stem part, two different cases having different governing equations and boundary conditions are presented in this paper. An analytical technique called the Differential Transform Method (DTM) is adopted for determining the temperature distribution from the non-linear energy balance equations. Besides predicting the temperature distribution and performance parameters, the present model also predicts the wet and dry parts of the fin for a wide range of the various geometric, thermo-physical and psychometric parameters.

PHYSICAL MODEL AND MATHEMATICAL FORMULATIONS

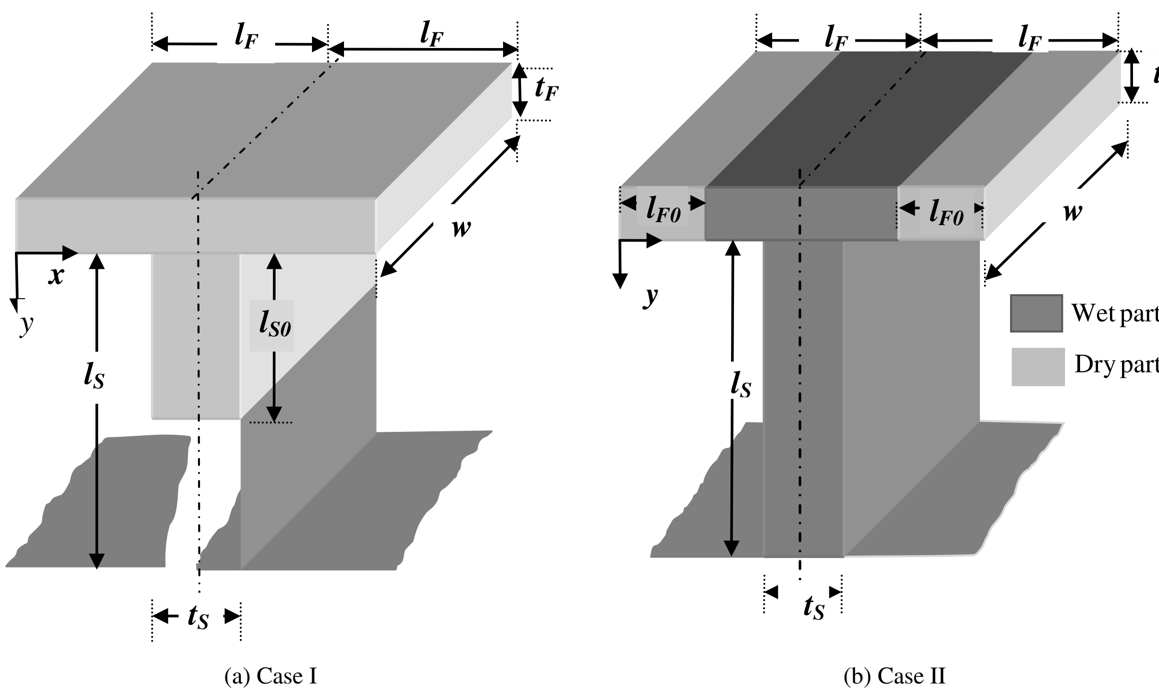


FIGURE 1: T-shaped fin (All dimensions are in m)

Figure 1shows a T-shaped fin having constant rectangular cross-section in the flange and stem part. The fin is attached to a surface at constant temperature T_B (^{0}C) and extends into moist air at constant temperature T_A (^{0}C). Partially wet surface is considered and the thermal resistance offered by the condensate layer is neglected. Fin thickness is taken to be very small as compared to the other dimensions and adiabatic fin tip is assumed. Radiation heat transfer is neglected and there are no bond resistances and no heat generation within the fin. Specific heat of ambient air ($C_{p,a}$, Jkg^{-1}K^{-1}), mass transfer coefficient (h_m, kg m^{-2} s^{-1}), latent heat of condensation (h_{fg} , Jkg^{-1}), thermal conductivity of fin material (k,Wm^{-1} K^{-1}) and convective heat transfer coefficient (h, Wm^{-2} K^{-1}) are taken as constants. Since, the point separating the dry and wet portion may lie either in the stem or the flange, two cases may arise :(i) Case I: The flange is completely dry and a part of the stem (up to l_{S0}) is dry and the rest portion of the stem (l_S- l_{S0}) is wet, as shown in Fig. 1(a) ; (ii) Case II: The entire stem is wet and a part of the flange (up to l_{F0}) is dry and the rest portion of the flange (l_F- l_{F0}) is wet, as shown in Fig. 1(b)

The 1-D steady state governing equations for the flange and the stem part are

$$\begin{bmatrix} 2h(T_F-T_A) \\ 2h(T_S-T_A) \\ 2\{h(T_S-T_A)+h_m h_{fg}(\omega_S-\omega_A)\} \end{bmatrix} = \begin{bmatrix} d(kt_F\, dT_F/dx)/dx \\ d(kt_S\, dT_S/dy)/dy \\ d(kt_S\, dT_S/dy)/dy \end{bmatrix} \begin{pmatrix} 0\le x\le l_F \\ 0\le y\le l_{S0} \\ l_{S0}\le y\le l_S \end{pmatrix} \begin{matrix} \text{dry domain for flange} \\ \text{dry domain for stem} \\ \text{wet domain for stem} \end{matrix} \quad (\text{for Case I}) \tag{1}$$

$$\begin{bmatrix} 2h(T_F-T_A) \\ 2\{h(T_F-T_A)+h_m h_{fg}(\omega_F-\omega_A)\} \\ 2\{h(T_S-T_A)+h_m h_{fg}(\omega_S-\omega_A)\} \end{bmatrix} = \begin{bmatrix} d(kt_F\, dT_F/dx)/dx \\ d(kt_F\, dT_F/dx)/dx \\ d(kt_S\, dT_S/dy)/dy \end{bmatrix} \begin{pmatrix} 0\le x\le l_{F0} \\ l_{F0}\le x\le l_F \\ 0\le y\le l_S \end{pmatrix} \begin{matrix} \text{dry domain for flange} \\ \text{wet domain for flange} \\ \text{wet domain for stem} \end{matrix} \quad (\text{for Case II}) \tag{2}$$

T_F and T_S are the local fin surface temperature (^{0}C) for flange and stem respectively. ω_F and ω_S are the humidity ratio (kg of water vapor per kg of dry air) of saturated air adjacent to the fin surface for flange and stem respectively. ω_A (kg of water vapor per kg of dry air) and *RH* are the humidity ratio and relative humidity of ambient air respectively. Chilton-Colburn analogy relates the h and h_m as $h=h_m C_{p,a} Le^{2/3}$ (*Le* is the Lewis number). The present analysis adopts the following cubic polynomial relation [6-8, 12, 13] between ω and T

$$\begin{bmatrix} \omega_F \\ \omega_S \end{bmatrix} = \begin{bmatrix} A_0+A_1T_F+A_2T_F^2+A_3T_F^3 \\ A_0+A_1T_S+A_2T_S^2+A_3T_S^3 \end{bmatrix} \text{where} \begin{bmatrix} A_0 & A_1 \\ A_2 & A_3 \end{bmatrix} = \begin{bmatrix} 3.7444\times10^{-3} & 0.3078\times10^{-3}(^0C^{-1}) \\ 0.46\times10^{-5}(^0C^{-2}) & 0.4\times10^{-6}(^0C^{-3}) \end{bmatrix} \tag{3}$$

The non-dimensional form of form of Eqns. (1) and (2) are

$$\begin{bmatrix} d^2\phi_F/dX^2 \\ d^2\phi_S/dY^2 \\ d^2\phi_S/dZ_S^2 \end{bmatrix} = \begin{bmatrix} H_F\phi_F \\ H_S\phi_S \\ C_{S0}+C_{S1}\phi_S+C_{S2}\phi_S^2+C_{S3}\phi_S^3 \end{bmatrix} \begin{pmatrix} 0\le X\le 1 \\ 0\le Y\le Y_0 \\ 0\le Z_S\le 1-Y_0 \end{pmatrix} \text{for Case I}; \begin{bmatrix} d^2\phi_F/dX^2 \\ d^2\phi_F/dZ_F^2 \\ d^2\phi_S/dY^2 \end{bmatrix} = \begin{bmatrix} H_F\phi_F \\ C_{F0}+C_{F1}\phi_F+C_{F2}\phi_F^2+C_{F3}\phi_F^3 \\ C_{S0}+C_{S1}\phi_S+C_{S2}\phi_S^2+C_{S3}\phi_S^3 \end{bmatrix} \begin{pmatrix} 0\le X\le X_0 \\ 0\le Z_F\le 1-X_0 \\ 0\le Y\le 1 \end{pmatrix} \text{for Case II} \tag{4}$$

$$[X;Y;X_0;Y_0;Z_F;Z_S;\phi_F;\phi_S] = [x/l_F;y/l_S;l_{F0}/l_F;l_{S0}/l_S;X-X_0;Y-Y_0;(T_F-T_A)/(T_B-T_A);(T_S-T_A)/T_B-T_A] \tag{5}$$

$$(Bi;\tau_L;\tau_T;\alpha;\alpha^*;H_S;H_F;\xi) = \left(ht_S/k;l_F/l_S;t_F/t_S;t_S/l_S;t_F/l_F;2Bi/\{\alpha^2(1+\upsilon)\};H_S\tau_L^2/\tau_T;h_{fg}/\left(C_{p,a}Le^{2/3}\right)\right) \tag{6}$$

$$\begin{pmatrix} C_{S0} & C_{S1} \\ C_{S2} & C_{S3} \end{pmatrix} = \{2Bi/\alpha^2(1+\upsilon)\} \begin{pmatrix} \{\xi(A_0+A_1T_A+A_2T_A^2+A_3T_A^3-\omega_A)\}/(T_B-T_A) & (1+\xi A_1+2\xi A_2T_A+3\xi A_3T_A^2) \\ (\xi A_2+3\xi A_3T_A)(T_B-T_A) & \xi A_3(T_B-T_A)^2 \end{pmatrix}; \begin{pmatrix} C_{F0} & C_{F1} \\ C_{F2} & C_{F3} \end{pmatrix} = \frac{\tau_L^2}{\tau_T} \begin{pmatrix} C_{S0} & C_{S1} \\ C_{S2} & C_{S3} \end{pmatrix} \tag{7}$$

X and Y are dimensionless distance for flange and stem part respectively. X_0 is the dry portion length for half of flange in dimensionless form. Y_0 is dry portion length for stem in dimensionless form. *Bi* is the Biot number. ξ is dehumidification parameter (^{0}C). ϕ_F and ϕ_S are dimensionless temperature for flange and stem part respectively.

The boundary conditions for Case I in dimensionless form are

$$\text{At } X=0,\ d\phi_F/dX=0 \tag{8a}$$

$$\phi_F|_{X=1} = \phi_S|_{Y=0};(2\tau_T/\tau_L)[d\phi_F/dX]_{X=1} = [d\phi_S/dY]_{Y=0};\phi_S|_{Y=Y_0} = \phi_D = \phi_S|_{Z_S=0};[d\phi_S/dY]_{Y=Y_0} = [d\phi_S/dZ_S]_{Z_S=0};\phi_S|_{Z_S=1-Y_0} = 1 \tag{8b}$$

The boundary conditions for Case II in dimensionless form are

$$\text{At } X=0,\ d\phi_F/dX=0 \tag{9a}$$

$$\phi_F|_{X=X_0} = \phi_D = \phi_F|_{Z_F=0};[d\phi_F/dX]_{X=X_0} = [d\phi_F/dZ_F]_{Z_F=0};(2\tau_T/\tau_L)[d\phi_F/dZ_F]_{Z_F=1-X_0} = [d\phi_S/dY]_{Y=0};\phi_F|_{Z_F=1-X_0} = \phi_S|_{Y=0};\phi_S|_{Y=1} = 1 \tag{9b}$$

Temperature Distribution

The present paper employs the Differential Transform Method [8, 14] for solving the highly non-linear governing equations. The differential transform of boundary condition (8a) and (9a) is obtained as $J_F(1)=0$. The differential transform of Eqn. (4) for Case I is

$$\begin{bmatrix}(i+1)(i+2)J_F(i+2)\\(i+1)(i+2)J_S(i+2)\\(i+1)(i+2)J_{Z_S}(i+2)\end{bmatrix}=\begin{bmatrix}H_FJ_F(i)\\H_SJ_S(i)\\C_{S0}\delta_{Z_S}(i)+C_{S1}J_{Z_S}(i)+C_{S2}\sum_{i_0=0}^{i}J_{Z_S}(i-i_0)J_{Z_S}(i_0)+C_{S3}\sum_{i_0=0}^{i}J_{Z_S}(i-i_0)\sum_{i_1=0}^{i_0}J_{Z_S}(i_0-i_1)J_{Z_S}(i_1)\end{bmatrix}\quad i=0,1,2,.. \tag{10}$$

For Case I, the following components are obtained for flange, dry part of stem and wet part of stem

$$\begin{bmatrix}J_F(2) & J_F(4) & J_F(6) & J_F(8) & ...\end{bmatrix}=\begin{bmatrix}H_FJ_F(0)/2 & H_FJ_F(2)/12 & H_FJ_F(4)/30 & H_FJ_F(6)/56 & ...\end{bmatrix} \tag{11}$$

$$\begin{bmatrix}J_S(2) & J_S(3) & J_S(4) & J_S(5) & J_6(6) & ...\end{bmatrix}=\begin{bmatrix}H_SJ_S(0)/2 & H_SJ_S(1)/6 & H_SJ_S(2)/12 & H_SJ_S(3)/20 & H_SJ_S(4)/30 & ...\end{bmatrix} \tag{12}$$

$$J_{Z_S}(2)=\left[C_{S0}+C_{S1}J_{Z_S}(0)+C_{S2}\{J_{Z_S}(0)\}^2+C_{S3}\{J_{Z_S}(0)\}^3\right]/2 \tag{13a}$$

$$J_{Z_S}(3)=\left[C_{S1}J_{Z_S}(1)+2C_{S2}J_{Z_S}(0)J_{Z_S}(1)+3C_{S3}J_{Z_S}(1)\{J_{Z_S}(0)\}^2\right]/6 \tag{13b}$$

$$J_{Z_S}(4)=\left[C_{S1}J_{Z_S}(2)+2C_{S2}J_{Z_S}(0)J_{Z_S}(2)+C_{S2}\{J_{Z_S}(1)\}^2+3C_{S3}J_{Z_S}(2)\{J_{Z_S}(0)\}^2+3C_{S3}J_{Z_S}(0)\{J_{Z_S}(1)\}^2\right]/12 \tag{13c}$$

For Case I, the final temperature distribution in the fin is obtained as

$$\left[\phi_F(X)\,;\phi_S(Y)\,;\phi_{Z_S}(Y)\right]=\left[\sum_{i=0}^{\infty}J_F(i)X^i\,;\sum_{i=0}^{\infty}J_S(i)Y^j\,;\sum_{i=0}^{\infty}J_{Z_S}(i)Z_S^{\,i}\right] \tag{14}$$

Thus, For Case I, the temperature distributions are functions of six unknowns $(J_F(0), J_S(0), J_S(1), J_{Z_S}(0), J_{Z_S}(1) \text{ and } Y_0)$, whose values are obtained by using the boundary conditions (8b) and by adopting the iterative Newton Raphson technique. The values of the unknowns are obtained after satisfying a convergence criterion of 10^{-6}.

In a similar manner, the final temperature distribution in the fin for Case II is obtained as

$$\phi_F(X)=J_F(0)+X^2\,H_FJ_F(0)/2+X^4H_FJ_F(2)/12+X^6H_FJ_F(4)/30+X^8H_FJ_F(6)/56+... \tag{15}$$

$$\begin{aligned}\phi_F(Z_F)=&J_{Z_F}(0)+J_{Z_F}(1)Z_F+\left[C_{F0}+C_{F1}J_{Z_F}(0)+C_{F2}\{J_{Z_F}(0)\}^2+C_{F3}\{J_{Z_F}(0)\}^3\right]Z_F^{\,2}/2+Z_F^{\,3}J_{Z_F}(1)\left[C_{F1}+2C_{F2}J_{Z_F}(0)+3C_{F3}\{J_{Z_F}(0)\}^2\right]/6\\&+Z_F^{\,4}\left[C_{F1}J_{Z_F}(2)+2C_{F2}J_{Z_F}(0)J_{Z_F}(2)+C_{F2}\{J_{Z_F}(1)\}^2+3C_{F3}J_{Z_F}(2)\{J_{Z_F}(0)\}^2+3C_{F3}J_{Z_F}(0)\{J_{Z_F}(1)\}^2\right]/12+...\end{aligned} \tag{16}$$

$$\begin{aligned}\phi_S(Y)=&J_S(0)+J_S(1)Y+\left[C_{S0}+C_{S1}J_S(0)+C_{S2}\{J_S(0)\}^2+C_{F3}\{J_S(0)\}^3\right]Y^2/2+Y^3J_S(1)\left[C_{S1}+2C_{S2}J_S(0)+3C_{S3}\{J_S(0)\}^2\right]/6\\&+Y^4\left[C_{S1}J_S(2)+2C_{S2}J_S(0)J_S(2)+C_{S2}\{J_S(1)\}^2+3C_{S3}J_S(2)\{J_S(0)\}^2+3C_{S3}J_S(0)\{J_S(1)\}^2\right]/12+...\end{aligned} \tag{17}$$

Actual Heat Transfer Rate and Fin Performance

For Case I, net actual heat transfer rate per unit width in dimensionless form are

$$
\begin{aligned}
Q_A &= q_A / k(T_A - T_B) \\
&= \left(4Bi\tau_L/\alpha\right)\int_0^1 \{\phi_F(X)\}dX + \left(2Bi/\alpha\right)\int_0^{Y_0}\{\phi_S(Y)\}dY \\
&+ \left[(2Bi)/\{\alpha(T_A-T_B)\}\right]\int_0^{1-Y_0}\left[(1+\xi A_1)(T_A-T_B)\phi_S(Z_S) + \xi(\omega_A - A_0 - A_1T_A) - \xi A_2\{T_A-(T_A-T_B)\phi_S(Z_S)\}^2 - \xi A_3\{T_A-(T_A-T_B)\phi_S(Z_S)\}^3\right]dZ_S
\end{aligned} \tag{18}
$$

q_A is the dimensional actual heat transfer rate through the fin per unit width (Wm^{-1}).

For Case II, net actual heat transfer rate per unit width in dimensionless form is

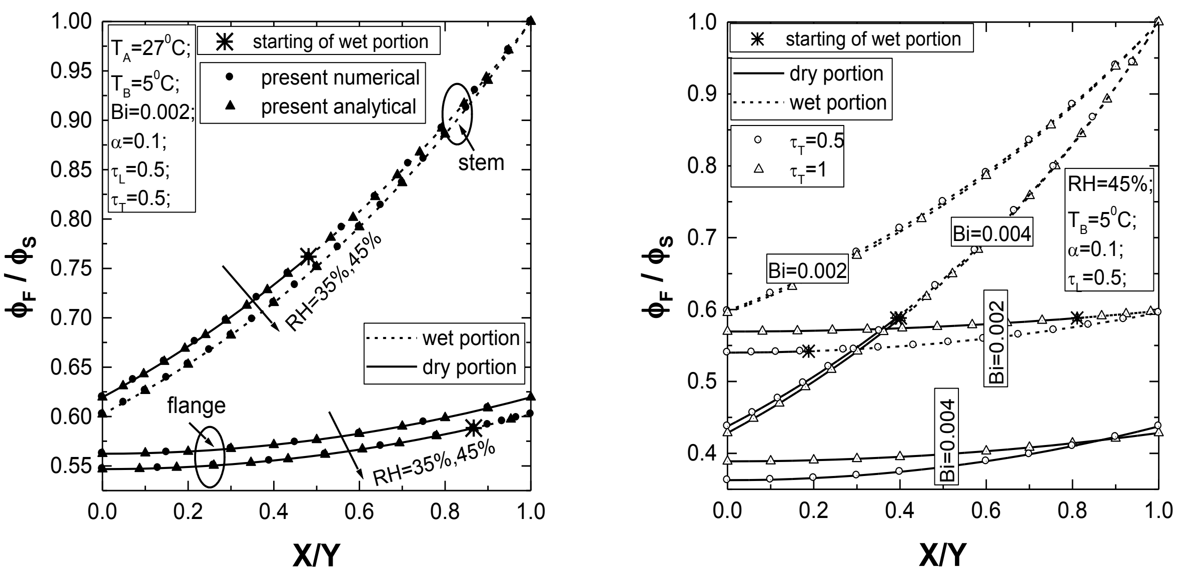
$$
\begin{aligned}
Q_A &= \frac{4Bi\tau_L}{\alpha}\int_0^{X_0}\{\phi_F(X)\}dX + \frac{4Bi\tau_L}{\alpha(T_A-T_B)}\int_0^{1-X_0}\left[(1+\xi A_1)(T_A-T_B)\phi_F(Z_F) + \xi(\omega_A - A_0 - A_1T_A) - \xi A_2\{T_A-(T_A-T_B)\phi_F(Z_F)\}^2 - \xi A_3\{T_A-(T_A-T_B)\phi_F(Z_F)\}^3\right]dZ_F \\
&+ \frac{2Bi}{\alpha(T_A-T_B)}\int_0^1\left[(1+\xi A_1)(T_A-T_B)\phi_S(Y) + \xi(\omega_A - A_0 - A_1T_A) - \xi A_2\{T_A-(T_A-T_B)\phi_S(Y)\}^2 - \xi A_3\{T_A-(T_A-T_B)\phi_S(Y)\}^3\right]dY
\end{aligned} \tag{19}
$$

Fin efficiency (η) and effectiveness (ε) are expressed as form, are

$$
[\eta;\varepsilon] = [Q_A/Q_I\,; Q_A/Q_{WF}] \text{ where } [Q_I; Q_{WF}] = \left[\left[2Bi(1+2\tau_L)\left\{1+\xi\left(\omega_A - A_0 - A_1T_B - A_2T_B^2 - A_3T_B^3\right)/(T_A-T_B)\right\}\right]/\alpha\,;\, Bi\left\{1+\xi\left(\omega_A - A_0 - A_1T_B - A_2T_B^2 - A_3T_B^3\right)/(T_A-T_B)\right\}\right] \tag{20}
$$

Q_I and Q_{WF} are ideal heat transfer rate and heat transfer rate through the same base area without fin per unit width in dimensionless form.

RESULTS AND DISCUSSION

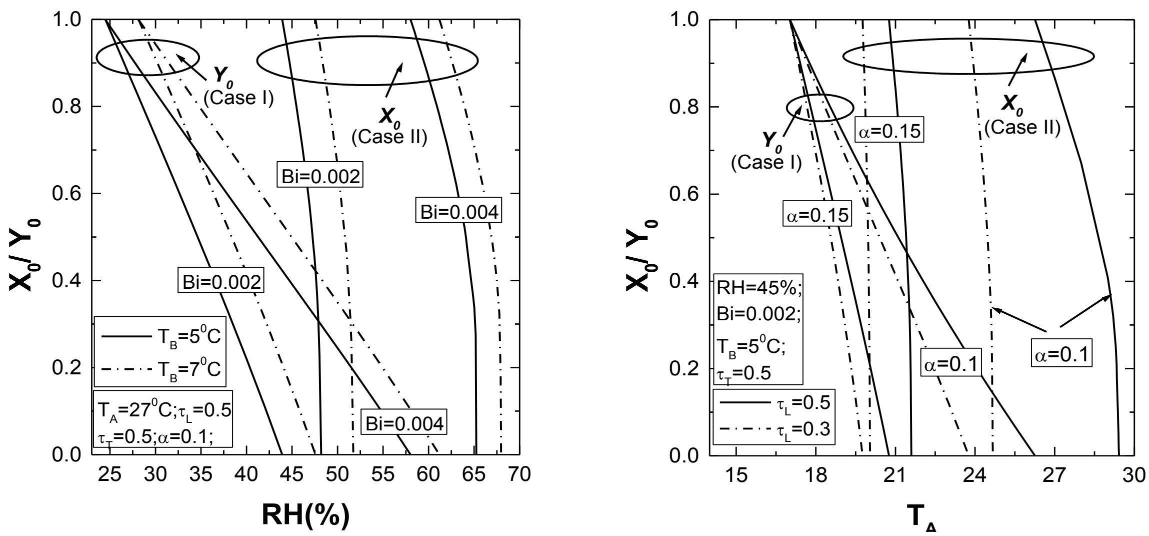


(a) (b)

FIGURE 2: (a) Comparison of dimensionless temperature distribution between the present analytical and numerical results ; (b) Effect of Bi and τ_T on the dimensionless temperature distribution

This section presents the effect of various parameters on the dry fin length, temperature distribution and fin performance. All the results are taken at constant ambient absolute pressure of 1.01325 bar. First, the present analytical model is first validated by a numerical model on the same problem. For the numerical results, Eqs.(4) are first discretised by Central difference scheme of fourth order accuracy and after that the difference equations are solved simultaneously by Gauss Seidel iterative procedure, satisfying the boundary conditions and a convergence criteria of 10^{-6}. As seen in Fig. 2(a), the temperature distributions from the present analytical model show a high accuracy with that obtained from the numerical model. As *RH* of the ambient air is increased, the moisture content of the air increases. This results in more amount of latent heat to be released due to increased moisture condensation, thereby increasing the fin surface temperature and thus decreasing the dimensionless temperature, as seen in Fig. 2(a). Increasing Biot number (*Bi)* increases the conduction resistance in the fin which increases the fin surface temperature, as seen in Fig. 2(b). The same figure shows that when the value of thickness ratio (τ_T) is increased, the dimensionless temperature decreases for the stem part but increases for major part of the flange and this is due to the increase in conduction resistance increases in the stem part but decrease in conduction resistance the flange part.

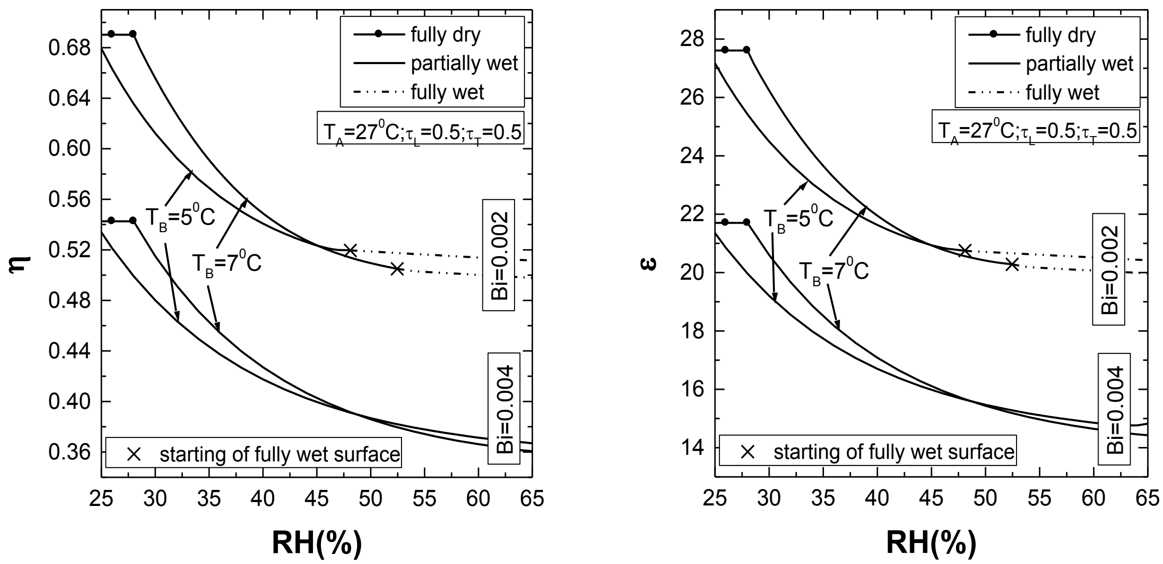
Figure 3 investigates the effect of various psychometric, geometric and thermo-physical parameters on the length of the dry portion in partially wet T-shaped fin. As seen in Fig. 3(a), at very low *RH*, both the stem and the flange are fully dry. But as *RH* is increased, moisture starts condensing on the fin surface, staring from the fin base i.e Y_0 decreases and at a particular *RH*, the entire stem part becomes fully wet (i.e Y_0 =0) whereas the flange part remains fully dry. With further increase in *RH*, moisture also starts condensing in the flange part i.e. X_0 decreases and at a particular relative humidity X_0 becomes equal to zero, which means that the entire fin surface becomes fully wet. Figure 3(a) depicts that at a particular *RH*, increasing *Bi* increases the length of the dry portion. This is due to the increase in conduction resistance in the fin, which increases the fin surface temperature. Also, the fin surface remains partially wet up to a higher value of *RH* with the increase in *Bi*. The same effect is noticed for increasing values of T_B and the reason is the increase in temperature in the entire fin with the increase in T_B. Fig. 3(b) shows that at a particular T_A, reducing the aspect ratio (α) of stem increases the length of dry portion and this is due to the increase in the conduction resistance in the fin. At lower α, fully wet condition is reached at higher value of T_A. Figure 3(b) also shows that increasing the length ratio increases the dry portion length.



(a) **(b)**

FIGURE 3: Dry length for a partially wet fin as a function of air relative humidity and ambient air temperature: (a) Effect of *Bi* and T_B; (c) Effect of α and τ_L

Figure 4 shows the influence of *RH*, *Bi* and T_B on the fin performances. Fin performances under dry condition are obtained by setting h_{fg}=0. Fin performance is independent of *RH* for fully dry surface, as seen in Fig. 4. For wet surface, increasing *RH* increases the moisture content of the air and this increases the latent heat flux, thereby enhancing the actual heat transfer rate, ideal heat transfer rate and heat transfer rate in un-finned condition. But increment of ideal heat transfer rate and heat transfer rate in un-finned condition are more compared to the actual heat transfer rate and hence both the performance parameters decrease with the increase in *RH*. This effect of *RH* on fin performance is pronounced for partially wet surface but very marginal for fully wet surface. The same figures show that lower values of *Bi* result in better fin performances and the reason is the decrease in conduction resistance in the fin with the decrease in *Bi*. Figure 4 also shows that for partially wet surface, increasing T_B enhances the fin efficiency and effectiveness up to a certain value of *RH* but beyond that value of *RH*, both the performance parameters declined with the increase in T_B. This is due to the decrease in the actual heat transfer rate, ideal heat transfer rate and heat transfer rate in un-finned condition with the increase in T_B. But in case of fully wet surface, the performance parameters showed a declining trend with the increase in T_B.



(a) **(b)**

FIGURE 4: Effect of *RH*, *Bi* and T_B, on fin performances: (a) Fin efficiency; (b) Fin effectiveness

CONCLUSIONS

Based on the above discussions, the following concluding remarks can be stated

- The most important parameters which govern whether a given T-shaped fin is fully or partially wet are the *RH* and T_A. The length of the dry portion decrease with the increase in *RH* or T_A
- The dry portion length increases with the increase in *Bi.* Also, at larger values of *Bi*, the fin surface remains partially wet up to a higher value of *RH*. The same effect is noticed for increasing values of T_B. The dry portion length increases with the decrease in α and also, at lower α fully wet condition is reached at higher values of ambient temperature. But a reverse effect is noticed for length ratio
- Fin performance declines with the increase in *Bi*. For partially wet surface, the performance parameters decrease sharply with the increase in *RH*. But for fully wet surface, the effect of *RH* on fin performance becomes marginally small. In case of partially wet surface, fin performances increase with the increase in T_B up to a certain value of *RH* but beyond that, it decreases. For fully wet surface, fin performances always decrease with the increase in T_B.

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