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Analysis of heat transfer in a partially wet radial fin assembly during dehumidification

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Abstract

This paper presents the analysis of heat transfer in a partially wet annular fin assembly during the process of dehumidification. In past studies, both fully dry and fully wet fins have been analyzed. New analytical formulation leading to a closed-form solution has been developed for a partially wet fin, which is most common in dehumidifier coil operation during air conditioning. The parameters that influenced the heat transfer rate in the finned tube structure are ratio of fin and wall thermal conductivities, ratio of fin thickness to fin pitch, ratio of wall thickness to fin pitch, ratio of fin length to fin pitch, cold fluid Biot number, ambient Biot number, the relative humidity and dry bulb temperature of the incoming air, and the cold fluid temperature inside the coil. Calculations were carried out to study the performance of the heat exchanger. The computed results included the temperature distribution in the wall and the fin and the fin efficiency. © 1999 Published by Elsevier Science Inc. All rights reserved.

Keywords: Partially wet fin; Dehumidification; Condensation

Notation

Bi	Biot number, h_2t/k_f
Bi_1	cold fluid Biot number, h_1p/k_w
Bi_2	ambient Biot number, h_2p/k_w
c_{pa}	specific heat of dry air (J/kg °C)
h_1	cold fluid heat transfer coefficient (W/m^2 °C)
h_2	ambient heat transfer coefficient (W/m^2 °C)
h_d	mass transfer coefficient (kg/m^{2}s)
h_{fg}	latent heat of condensation (J/kg)
k	thermal conductivity (W/m °C)
K	thermal conductivity ratio (k_f/k_w)
Le	Lewis number
p	half fin pitch (m)
P	aspect ratio (t/p)
q_{fin}	total heat transfer rate (W)
q_{max}	maximum (ideal) heat transfer rate (W)
r	radial coordinate (m)
r_i	internal radius of the tube (m)
r_o	external radius of the tube (m)
R	dimensionless radius within the fin (r/p)
R_1	dimensionless radius within the tube wall (r/p)
R_b	ratio of sensible to total heat transfer rate at base temperature
R_Θ	ratio of sensible to total heat transfer rate at temperature Θ
RH	relative humidity
t	half fin thickness ($=\delta/2$) (m)
T	temperature (°C)
T_1	cold fluid temperature (°C)
T_2	ambient dry bulb temperature (°C)
w	humidity ratio
W	dimensionless wall thickness $(r_o - r_i)/p$

Greek

δ	fin thickness (m)
ε	fin effectiveness
ζ	radial location separating the dry and wet surfaces (m)
η	fin efficiency
Θ	dimensionless temperature $(T - T_2)/(T_1 - T_2)$

Subscripts

1	cold fluid side
2	air side
dry	dry region
e	outer edge of the fin
f	fin
w	wall
wet	wet region
ζ	wet–dry interface

1. Introduction

Finned tube heat exchangers are widely used in air conditioning. An individual finned tube geometry is a reasonable

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representation of these heat exchangers. In a dehumidifier unit, the incoming air stream is cooled and dehumidified by circulating refrigerant through a coil. The refrigerant evaporates within the coil and removes heat from the air stream. The effectiveness of the heat exchanger primarily depends on the efficiency of the fin attached to the outer surface of the coil. As the moist air approaches the coil, the cooling takes place by the removal of sensible heat followed by condensation of water vapor contained within the air. The condensation process involves heat transfer with phase change and the cooling takes place by the removal of sensible as well as latent heat. The ratio of sensible to total heat is an important quantity that controls the heat transfer during a dehumidification process. This quantity is frequently used in sizing cooling coils for air conditioning units.

The topic of this paper is related to the analysis of fins used in cooling coil (dehumidifier) of an air conditioner. Research on these fins is based mainly on experimentation. Most air conditioning cooling coils have the coil surface temperature below the dew point temperature of the air being cooled. Therefore, simultaneous heat and mass transfer occur. Moisture condensation on the fin surface affects the overall fin efficiency. A special situation is encountered when the fin base temperature is lower but the fin tip temperature is higher than the dew point temperature of the surrounding air. This condition results in a partially wet fin. When the fin is partially wet its efficiency depends significantly on the relative humidity. The experimental data for the overall performance of dry and fully wet cooling coil with dehumidification have been reported by various investigators (Kays and London, 1964; Wang et al., 1997). However, only few theoretical works have been reported on condensation assuming fully wet fins or fin assemblies (Webb, 1994). Kazeminejad (1995) presented an analysis of rectangular one-dimensional fin assembly heat transfer with dehumidification under fully wet condition and incorporating the ratio of sensible to total heat transfer. In dehumidifier coils, however, annular fins are more common than rectangular fins. Rosario and Rahman (1998) presented a one-dimensional radial fin assembly model with condensation. Their findings indicated that the heat transfer rate increased with increment in both dry bulb temperature and relative humidity of the air.

Wet fin efficiency has been studied by some researchers. Threlkeld (1970) proposed a rectangular fin model assuming that the fin was covered with a uniform condensate film. He developed an analytical expression for the overall fin efficiency by using the enthalpy difference as the driving force for the combined heat and mass transfer process. He assumed a linear relationship between the ambient air temperature and the corresponding saturated air temperature. His model showed that the wet fin efficiency was only slightly affected by the air relative humidity. ARI Standard 410-81 (1972) used an approach similar to Threlkeld (1970), but neglecting the presence of the water film on the fin surface. McQuiston (1975) developed an expression for wet fin efficiency for the case of a plane fin. Coney et al. (1989) presented a numerical solution for condensation over a rectangular fin taking into account the thermal resistance of the condensate film and using a second-degree polynomial to relate the humidity ratio with dry bulb temperature. Their results showed that there is negligible effect of condensate thermal resistance on the fin temperature distribution. Chen (1991) presented a two-dimensional model for fin efficiency with combined heat and mass transfer between water-wetted fin surface and moving moist air stream. The solution was obtained by finite-difference for rectangular fins. Srinivasan and Shah (1997) presented a summary of previous studies on condensation over rectangular fins.

Elmahdy and Biggs (1983) obtained the overall fin efficiency of a circular fin by taking into consideration the temperature distribution over the fin surface. Their work treated heat transfer and mass transfer separately by considering their respective driving force and then assumed a linear relationship between the humidity ratio of the saturated air on the fin surface and its temperature. Their numerical results indicate that the fin efficiency strongly depends on the relative humidity. McQuiston and Parker (1994) presented an analysis of circular fins using an approximation proposed by Schmidt (1949). Their model assumed a linear relationship between the humidity ratio and the dry bulb temperature. Hong and Webb (1996) derived an analytical formulation of fin efficiency of fully wet surface for circular fins. Their formulation was based on the exact solution of the governing differential equation after incorporating a linear relationship between the humidity ratio and the dry bulb temperature (McQuiston, 1975; McQuiston and Parker, 1994). Wang et al. (1997) derived a fully wet fin efficiency for circular fins using the formulation given by Threlkeld (1970). They obtained an analytical expression for the fully wet fin efficiency by utilizing the enthalpy difference as the driving force for the combined heat and mass transfer process. Wu and Bong (1994) studied fin efficiency under partially wet condition for plane rectangular fins. They showed that only when the fin is partially wet the overall fin efficiency became significantly dependent on the air relative humidity.

The present investigation has the following three objectives:

1. To develop a closed form analytical solution for heat transfer in a radial fin assembly under partially wet condition.
2. To study the effects of some important parameters like relative humidity and dry bulb temperature of the surrounding air and cold fluid temperature on the overall fin efficiency.
3. To study the enhancement of heat transfer in the fin assembly due to condensation.

2. Mathematical model

A radial fin assembly of uniform cross section and pitch under partially wet condition as shown in Fig. 1 is used for this investigation This assembly is found in many practical applications (Kern and Kraus, 1972; Kraus, 1982). A partially wet fin is encountered when the fin base temperature is lower but

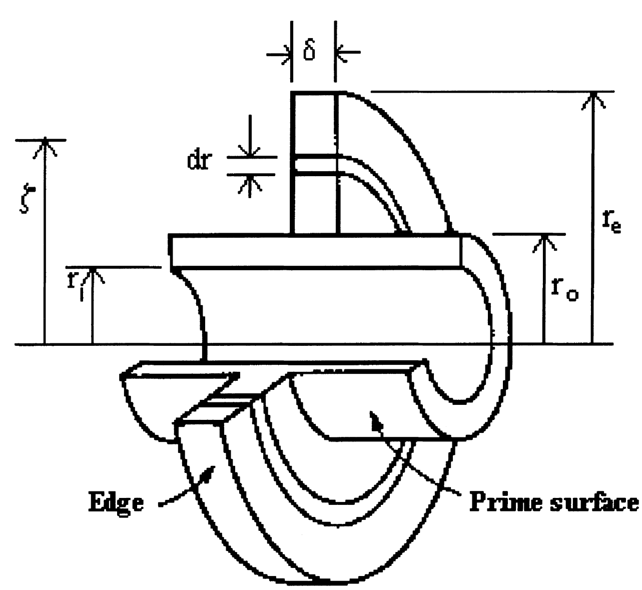

Fig. 1. Radial fin of uniform thickness.

the fin tip temperature is higher than the dew point temperature of the surrounding air. On the fin surface there is a radial location, $r = \zeta$, where the radial fin temperature equals the air dew point temperature, i.e., $T_{\text{fin}} = T_{\text{dew}}$. The fin is then separated into two radial regions: a wet region ($r_o \leqslant r \leqslant \zeta$), with the fin surface temperature lower than the air dew point temperature, and a dry region ($\zeta \leqslant r \leqslant r_e$) with fin surface temperature higher that air dew point temperature.

The following assumptions were made to complete the development of the theoretical model:

1. The thermal conductivities of the fin and the wall are constant.
2. The contact resistance between the tube and the fin is insignificant.
3. The convective heat transfer coefficients inside the tube and for heat transfer to the ambient are constant.
4. Temperature varies only in the radial direction.
5. Heat flows in the radial direction within the tube even in the presence of fins.
6. Condensation occurs when the surrounding air dew point temperature is reached.
7. Droplets can drain off the fin surface under the influence of the gravitational force.

Under these assumptions, the governing differential equations for the conservation of energy can be expressed in dimensionless form as the following.

For the fin, wet region ($r_o \leqslant r \leqslant \zeta$):

$$\frac{d^2\Theta_{\text{fwet}}}{dR^2} + \frac{1}{R}\frac{d\Theta_{\text{fwet}}}{dR} - \frac{Bi\Theta_{\text{fwet}}}{R_\theta P^2} = 0. \tag{1}$$

For the fin, dry region ($\zeta \leqslant r \leqslant r_e$):

$$\frac{d^2\Theta_{\text{fdry}}}{dR^2} + \frac{1}{R}\frac{d\Theta_{\text{fdry}}}{dR} - \frac{Bi\Theta_{\text{fdry}}}{P^2} = 0. \tag{2}$$

For the wall:

$$\frac{d^2\Theta_{\text{w}}}{dR_1^2} + \frac{1}{R_1}\frac{d\Theta_{\text{w}}}{dR_1} = 0. \tag{3}$$

The boundary conditions are the following.

At $R_1 = r_i/p$ (internal radius of tube):

$$\frac{d\Theta_{\text{w}}}{dR_1} = -Bi_1\,(1 - \Theta_{\text{w}}). \tag{4}$$

At $R_1 = R$ (wall–fin interface):

$$\Theta_{\text{w}} = \Theta_{\text{fwet}} \tag{5}$$

and

$$\frac{d\Theta_{\text{w}}}{dR_1} = -Bi_2\Theta_{\text{w}}\,\frac{(1-P)}{R_b} + (1 - \Theta_{\text{w}}) + KP\frac{d\Theta_{\text{fwet}}}{dR}. \tag{6}$$

At $R = \zeta/p$ (fin wet–dry interface):

$$\Theta_{\text{fwet}} = \Theta_{\text{fdry}} \tag{7}$$

and

$$\frac{d\Theta_{\text{fwet}}}{dR} = \frac{d\Theta_{\text{fdry}}}{dR}. \tag{8}$$

At $R = r_e/p$ (fin tip):

$$\frac{d\Theta_{\text{fdry}}}{dR} = -Bi\frac{\Theta_{\text{fdry}}}{P}. \tag{9}$$

The ratio of sensible to total heat transfer is expressed as

$$R_\Theta = \left[1 + h_d(w_2 - w)\frac{h_{\text{fg}}}{(T_2 - T_1)h_2\Theta}\right]^{-1}. \tag{10}$$

The heat and mass transfer coefficients can be related by using the Lewis relationship (McQuiston and Parker, 1994):

$$Le^{2/3} = \frac{h_2}{c_{\text{pa}}\,h_d}. \tag{11}$$

The Lewis number for moist air is approximately equal to 1.0. Therefore, for most air-conditioning applications:

$$h_d = \frac{h_2}{c_{\text{pa}}}. \tag{12}$$

Then Eq. (10) can be rewritten as

$$R_\Theta = \left(1 + \frac{h_{\text{fg}}(w_2 - w)}{c_{\text{pa}}(T_2 - T_1)\Theta}\right)^{-1}. \tag{13}$$

The overall fin efficiency, η, is defined as the ratio of the actual total heat transfer rate to the maximum total heat transfer rate,

$$\eta = \frac{q_{\text{fin}}}{q_{\text{max}}}. \tag{14}$$

In the case of partially wet radial fin assembly two regions have to be analyzed. In the wet region, the fin performance is a combination of heat and mass transfer. The actual total heat transfer, q_{fin}, must include both the sensible heat transfer and the latent heat transfer originated by mass transfer (condensation). The sensible heat transfer is due to convection from the air to the fin because of the temperature difference between the air and the fin, and the latent heat transfer is caused by the humidity ratio difference between the air and the fin surface. The maximum heat transfer rate, q_{max}, corresponds to an ideal fin whose surface temperature equals the temperature at the fin base under wet conditions.

For the wet region ($r_o \leqslant r \leqslant \zeta$):

$$q_{\text{fin,wet}} = 2\pi k_f r_o \delta\left(\frac{dT}{dr}\right)_{r=r_o} - 2\pi k_f \zeta\delta\left(\frac{dT}{dr}\right)_{r=\zeta}, \tag{15}$$

$$q_{\text{max,wet}} = 2\pi\left(\zeta^2 - r_o^2\right)\left(h_2(T_2 - T_w) + h_d h_{\text{fg}}(w_2 - w_w)\right). \tag{16}$$

For the dry region ($\zeta \leqslant r \leqslant r_e$):

$$q_{\text{fin,dry}} = 2\pi k_f \zeta\delta\left(\frac{dT}{dr}\right)_{r=\zeta}, \tag{17}$$

$$q_{\text{max,dry}} = 2\pi\left(\left(r_e^2 - \zeta^2\right) + r_e\delta\right)\left(h_2(T_2 - T_\zeta)\right). \tag{18}$$

The maximum heat transfer rate includes both sensible and latent heat components because it is defined as the heat transfer rate that would exist if the dry region were at the fin base temperature, which in this case is the air dew point temperature.

Therefore, the overall wet fin performance can be expressed by the overall fin efficiency conserving the basic definition given by Eq. (14).

$$\eta_{\text{wet}} = \frac{q_{\text{fin,wet}}}{q_{\text{max,wet}}}. \tag{19}$$

After some substitutions and manipulations the wet fin efficiency can be expressed in dimensionless form as

$$\eta_{\text{wet}} = \frac{R_w 2P(-(d\theta/dR))_w - R_\zeta 2P(-(d\theta/dR))_{r=\zeta}}{\left(R_\zeta^2 - R_w^2\right)\left((Bi)/(PR_{\theta w})\right)}. \tag{20}$$

Similarly for the dry section:

$$\eta_{\text{dry}} = \frac{q_{\text{fin,dry}}}{q_{\text{max,dry}}}, \tag{21}$$

$$\eta_{\text{dry}} = \frac{2P^2 R_\zeta}{\left(R_e^2 - R_\zeta^2 + 2R_e R_\zeta\right)\theta_\zeta Bi}\left(-\frac{d\theta}{dR}\right)_{r=\zeta}. \tag{22}$$

The fin efficiency for the entire fin under partially wet condition is

$$\eta = \left(\frac{\zeta^2}{r_e^2}\right)\eta_{wet} + \left(1 - \frac{\zeta^2}{r_e^2}\right)\eta_{dry}. \tag{23}$$

This is the fin efficiency of a partially wet fin. The radial location $r = \zeta$ defines the separation between wet and dry regions. Once this location has been determined the fin efficiency can be calculated. The radial location $r = \zeta$ can be evaluated from continuity of heat flow at the separation between the dry and wet regions as presented by Eq. (8). This equation can be solved using an iterative procedure so that ζ can be found.

3. Results and discussion

The mathematical model developed in the last section was used to perform numerical simulation for conditions found in a typical air conditioning cooling coil under partially wet condition. Runge–Kutta method with shooting technique was used to integrate the differential equations. The following parameters were kept constant in all simulations:

$Bi_1 = 1.0,\ Bi_2 = 0.1,\ P = 0.25,\ W = 0.5.$

These values were calculated using heat transfer coefficients and geometric parameters that correspond to a typical direct expansion cooling coil used in air conditioning applications.

Fig. 2 presents the results of solving Eq. (8). The radial location ζ is important to determine the fin efficiency of a partially wet fin. This location separates the wet and dry parts of the fin. Fig. 2 shows the variation of ζ/r_e as a function of the surrounding air relative humidity. It may be noted that ζ/r_e increases rapidly with change in relative humidity when the fin is partially wet. When the dry bulb temperature is kept constant and the relative humidity is increased, the moisture content (humidity ratio) increases and results in higher dew point temperature. Therefore, a higher relative humidity results in condensation over a larger area of the fin.

Fig. 3 shows the dimensionless temperature distribution over the fin plotted against the dimensionless radius. The three regions can be distinguished from the slope of the curve: wall, wet fin, and dry fin. It may be noted that the temperature

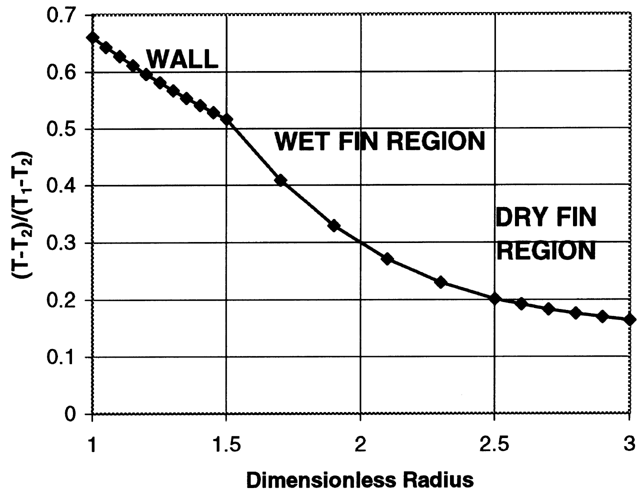

Fig. 3. Variation of dimensionless temperature of the assembly ($T_1 = 7°C$, $T_2 = 27°C$, RH = 30.5%, $K = 1$).

distribution is linear in the wall region. A larger slope in the wet region is indicative of a large rate of heat transfer due to condensation on the fin surface. A lower slope in the dry region is expected because the heat transfer is by pure convection without any phase change.

Fig. 4 presents dimensionless temperature Θ as a function of the dimensionless radius with variation in the surrounding temperature T_2. An increase in the air side temperature increases the heat transfer rate in all regions: wall, wet fin, and dry fin. It can be noticed that dimensionless temperature at all regions decreases with increase in T_2. A larger air dry bulb temperature translates to a larger moisture content (humidity ratio) when the relative humidity remains constant. It is important to note that any change in T_2 affects the radial location ζ separating the wet and dry surfaces because of changes in the air dew point temperature.

The dimensionless temperature as a function of the dimensionless radius for the variation in the cold fluid temperature T_1 is shown in Fig. 5. An increase in T_1 increases the fin temperature and therefore decreases the difference between the fin and its surrounding. T_1 also affects the radial location ζ

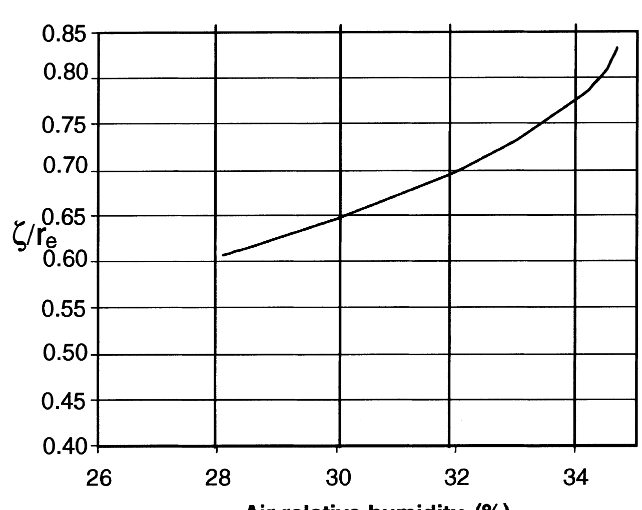

Fig. 2. Radial location separating the wet and dry regions as a function of the surrounding air relative humidity ($T_1 = 7°C$, $T_2 = 27°C$, $K = 1$).

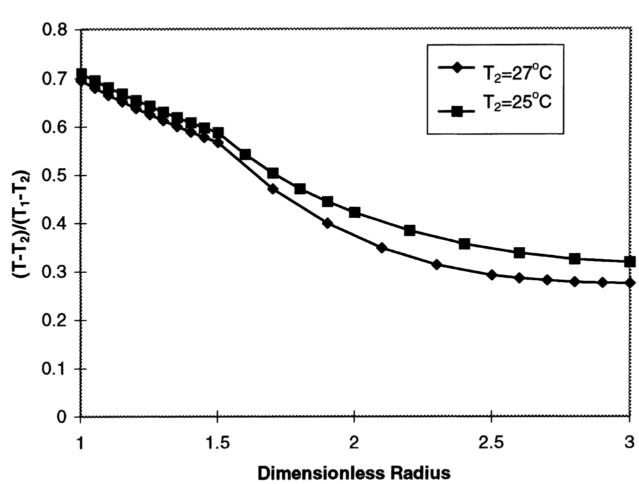

Fig. 4. Variation of dimensionless temperature distribution with variation in surrounding air dry bulb temperature ($T_1 = 7°C$, RH = 30.5%, $K = 1$).

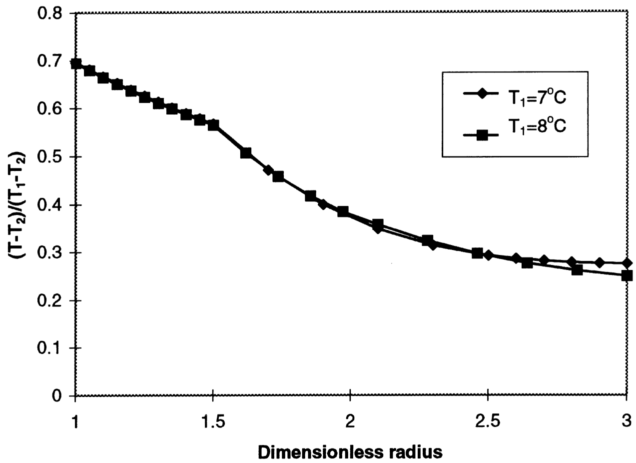

Fig. 5. Variation of dimensionless temperature distribution with variation in cold fluid temperature ($T_2 = 27°C$, RH = 30.5%, $K = 1$).

separating the wet and dry surfaces because of changes in the fin temperature.

Variations of $(T - T_2)/(T_1 - T_2)$ with the dimensionless radius R for changes in the relative humidity can be seen from Fig. 6. It can be noticed that dimensionless temperature decreases with increase in relative humidity in all three regions. A larger relative humidity at a given dry bulb temperature translates to a larger moisture content in the air. It is important to note that any change in air relative humidity also affects the radial location ζ separating the wet and dry surfaces because of changes in the air dew point temperature.

Fig. 7 plots the variation of fin efficiency η as a function of relative humidity of the ambient air. In this calculation, the ambient temperature was assumed to be 27°C and the fluid inside the coil was assumed to be at 7°C. It may be noted that the fin remains entirely dry at low relative humidity. Then there is only sensible heat transfer. As long as the air intake temperature remains constant, the fin efficiency also remains constant. As soon as the condensation begins, the fin efficiency decreases with relative humidity. A large decrease is seen in the partially wet region and somewhat gradual decrease when the entire fin participates in the condensation process. Fig. 7 also plots the wet fin efficiency reported by several previous studies.

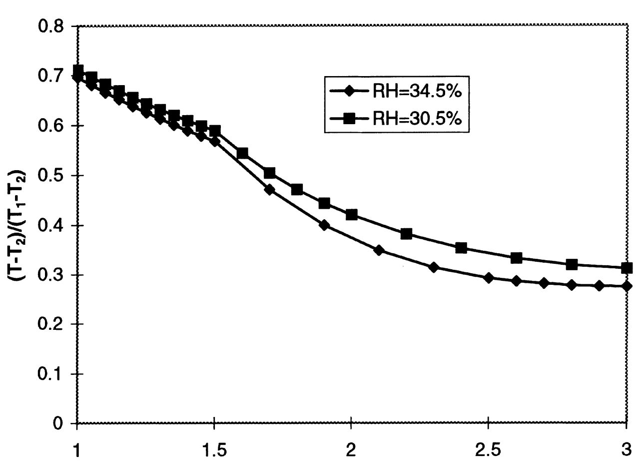

Fig. 6. Variation of dimensionless temperature with variation in relative humidity ($T_1 = 7°C$, $T_2 = 27°C$, $K = 1$).

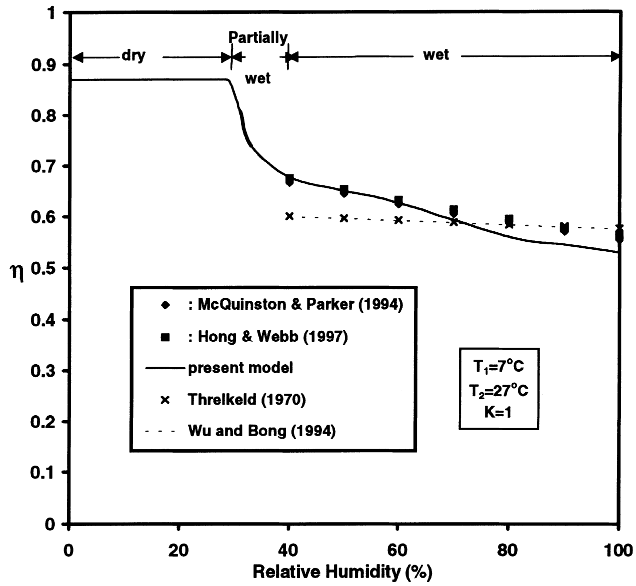

Fig. 7. Fin efficiency under partially wet condition.

The fin efficiency calculated by using the method of McQuiston and Parker (1994) and Hong and Webb (1996) show good agreement with the present model in the wet fin region. The formulations by Threlkeld (1970) and Wu and Bong (1994) however show much smaller change in fin efficiency with change in ambient air relative humidity. Both of these studies considered rectangular fin geometry instead of circular fin, and therefore, the difference in the nature of efficiency variation is not surprising. Due to larger area of a circular fin for any given fin length (radius), the effects of moisture condensation is also larger.

Fig. 8 shows variation of fin efficiency η with air relative humidity for changes in the surrounding dry bulb temperature T_2. A decrease in the air side temperature T_2 increases the fin efficiency in both partially wet and fully wet regions. It can be noticed that the relative humidity where the fin temperature is exactly equal to the air dew point varies with the surrounding temperature T_2.

The variation of fin efficiency η as a function of air relative humidity with the variation in the cold fluid temperature T_1 is

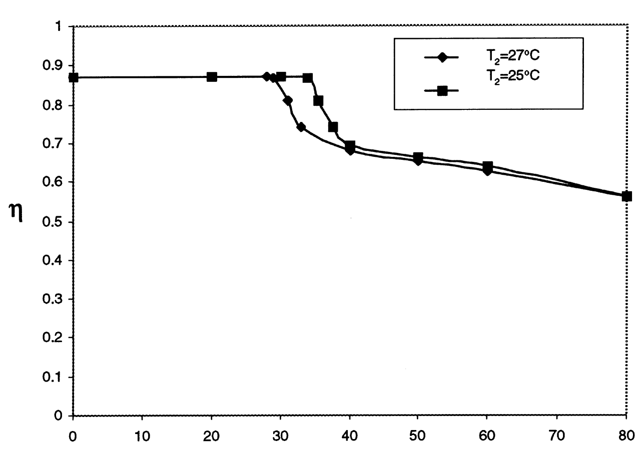

Fig. 8. Variation of fin efficiency with variation in surrounding air dry bulb temperature ($T_1 = 7°C$, $K = 1$).

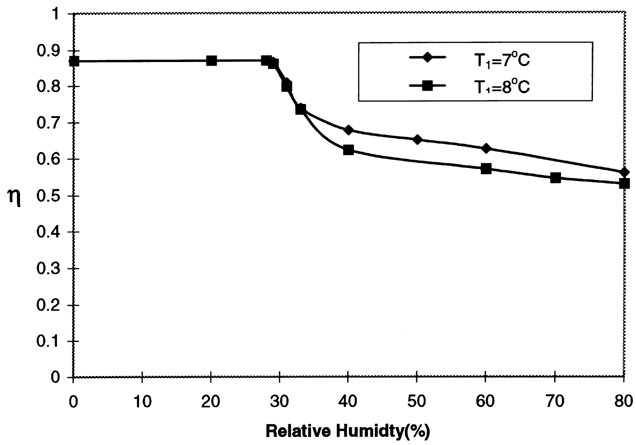

Fig. 9. Variation of fin efficiency with variation in cold fluid temperature ($T_2 = 27°C$, $K = 1$).

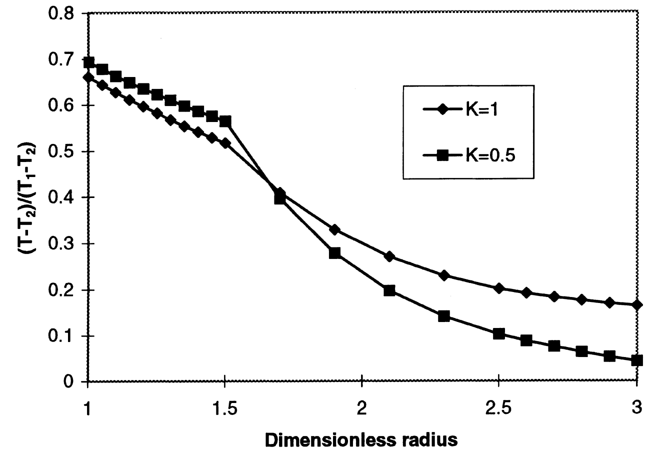

Fig. 11. Variation of dimensionless temperature distribution for different thermal conductivity ratios ($T_1 = 7°C$, $T_2 = 27°C$, RH = 30.5%).

presented in Fig. 9. Any increase in the cold fluid temperature T_1 decreases fin efficiency η especially in the fully wet region. In the partially wet region this influence is not very large. It may be noted that the extent of the partially wet zone changes significantly with change in the cold fluid temperature.

Even though a larger efficiency is seen when the fin is dry, the rate of heat transfer increases by a very significant amount as the condensation takes place on the fin surface. This is demonstrated in Fig. 10 where the fin effectiveness, ε, is plotted as a function of relative humidity for $T_1 = 7°C$ and $T_2 = 27°C$. The fin effectiveness is defined as the ratio of actual total heat transfer rate over the heat transfer rate under fully dry condition. It measures the augmentation of heat transfer rate due to condensation on the fin surface. It can be noticed that the rate of heat transfer increases by about 40% when the entire fin participates in the phase change process.

The results presented above assumed that the fin and the tube are made of the same material, i.e., $K = 1$. In many dehumidification processes, fins are made of lightweight materials such as aluminum, whereas the tube is made of copper or steel. Variations of $(T - T_2)/(T_1 - T_2)$ with the dimensionless radius R for changes in thermal conductivity ratio can be seen in Fig. 11. It can be observed that dimensionless temperature decreases with decrease in thermal conductivity ratio in the fin region. A lower fin thermal conductivity results in larger temperature gradient along the fin. This indicates a smaller heat transfer rate and therefore larger dimensionless temperature in the tube. It is important to note that any change in thermal conductivity ratio also affects the radial location ζ separating the wet and dry surfaces.

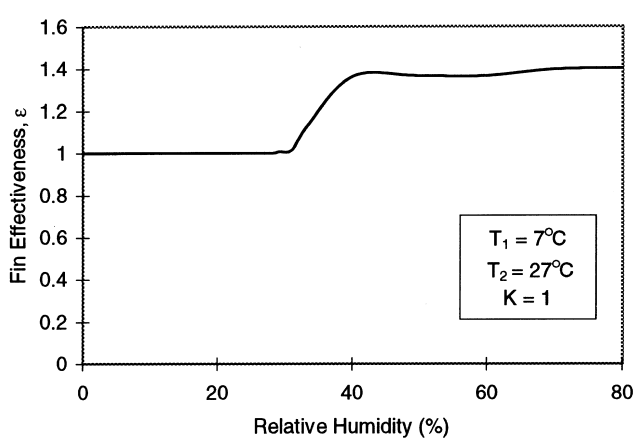

Fig. 10. Variation of fin effectiveness with relative humidity.

The variation of fin efficiency η as a function of thermal conductivity ratio is shown in Fig. 12. Any decrease in thermal conductivity ratio decreases the fin efficiency η in all three regions. It may be also noted that the fin remains partially wet over a larger range of relative humidity when the thermal conductivity is lowered. A decrease of fin efficiency with decrease of thermal conductivity ratio is quite expected because the fin maintains a larger temperature gradient along its length. A higher thermal conductivity will result in more uniform temperature along the fin contributing to a higher efficiency and larger rate of heat transfer.

4. Conclusions

A one-dimensional analytical model representing the partially wet condition has been developed. The findings indicates that the overall fin efficiency depends on the condition of the fin surface, whether it is dry, partially wet, or fully wet. The

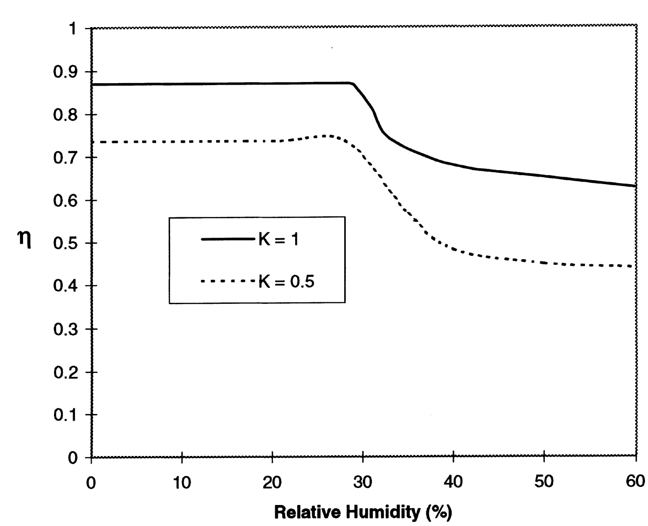

Fig. 12. Variation of fin efficiency for different thermal conductivity ratios ($T_1 = 7°C$, $T_2 = 27°C$).

dimensionless temperature, Θ, decreased with temperature and relative humidity of the surrounding air. Under partially wet condition, the fin efficiency changes rapidly with air relative humidity. The range of relative humidity over which the partially wet condition exists depends on intake air temperature, refrigerant temperature inside the coil, as well as the thermal conductivity ratio of fin and tube materials. The enhancement of heat transfer due to condensation was found to be quite significant. The complete understanding of heat transfer phenomenon covering dry, partially wet, and fully wet conditions is believed to be very useful for an efficient design of dehumidification processes.

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