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# Water thickness effect on the fin efficiency and heat transfer for partially wet-surface heat exchanger

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**Abstract**

Heat and mass transfer, in this paper, is considered in one-row heat exchanger, that fins are hotter than air flow and water is added to fins. Related governing equations are derived by analyzing a two-dimension model in a unique cell of a heat exchange. These equations are numerically solved by finite difference method. Heat transfer and efficiency under partially wet surface are calculated by changes in thickness of water layer on the fins and wet percentage region of fin with constant airflow characteristics. In this study, Lewis Number as unity and water vapor saturation as parabolic are assumed. Obtained results show that increasing in thickness of fin leads to increasing thermal resistance; therefore, efficiency of fin decreases. But thickness of water layer dose not play a significant role in fin efficiency when water layer covering the surface of fins is thin or it covers a small region of fins because thermal resistance of water is not comparable with thermal conductivity of fin material. But where thickness of water layer is comparable with fin pitch or more surface of fins is wetted, fin efficiency and heat transfer change obviously because of increasing thermal resistance and changing in air flow velocity that cause more decreasing in efficiency of fins.

**Keywords:** *Heat exchanger; Partially wet-surface; Thickness of water layer*

## 1. Introduction

Wet fins can be categorized in two forms: cold fins and hot fins. When the cold fin temperature is less than dew point temperature of moist air, vapor in the moist air is condensed on the fin depending on heat exchanger situation part or full of the fin may be covered by water layer. But the case is different with hot fins. Water film on the fin is provided out of heat exchanger and as a result of evaporation on the fin, absolute humidity of airflow is increased which leads to significant decrease of temperature of fluid in tubes. That part or full of the hot fin to be wet does not depend on the situation of heat exchanger. Common instances of such application include evaporative coolers, evaporative condenser and cooling towers. In such devices, heat from the hot fluid is transferred to the atmosphere by direct or indirect interaction through both sensible heat exchange and evaporative latent heat exchange. The effect of water evaporation considerably improves the cooling performance. [1]

Since heat and mass transfer occur at the same time in an evaporative heat exchanger, the cooling process in an evaporative heat exchanger is more complicated in comparison with that of a sensible heat exchanger.

Numerous researches have been done on cold fin whether it is fully wet, partially wet or dry surface. Naphon [2] theoretically studied heat transfer

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characteristics of the annular under the mentioned conditions. In his study, effects of inlet condition of working fluids and the fin dimensions on the heat transfer characteristics and fin efficiency were considered and the obtained results were compared with those of other researchers. Kazeminejad [3] carried out analysis of a one-dimensional conduction heat transfer model and its efficiency under fully wet and dry surface conditions. Introducing a new method named "Finite Circular Fin Method", Wang and Pirompugd [4] investigated the performance of a plate fin-tube heat exchanger under fully and partially wet surface conditions. Assuming humidity ratio of the saturated air on the wet surface that varies linearly with the local fin temperature, Salahe Eldin [5] developed analytical solution for evaluating fin performance under fully and partially wet surface conditions.

Researches also include wet hot fin. Maclaine-Cross and Banker [6] examined the evaporative heat transfer between parallel plates through a 1-D approach. They proposed a simplified analysis model using the Lewis number as unity and the water-vapor-saturation line as linear. Using the same approach, Kettleborough and Hsieh [7] indicated that the wet bulb temperature is the most significant factor to the performance of an evaporative heat exchanger. Hsu et al. [8] developed the optimal design of an evaporative cooler based on a 1-D analysis. Chen et al. [9] suggested the performance evaluation method for the evaporative coolers in cylindrical or plate shapes. Stoitchkov and Dimitrov [10] worked on the evaporative cooling in a cross-flow plate heat exchanger with a more developed model than that of Maclaine-Cross and Banks [6]'s. The mentioned studies all were based on 1-D models. These models require depending on the experimental data or more advanced theoretical analysis concerning heat and mass transfer coefficients.

In the meantime, Tsay [11] studied the heat and mass transfer in a plate channel that were covered with thin water film based on a 2-D model and then calculated the heat and mass transfer coefficients. He also proved that the latent heat transfer functions more influentially on the heat transfer performance than the sensible heat transfer. Yan [12, 13] analyzed the evaporative cooling of a liquid film by a turbulent mixed convention through a 2-D analysis and examined the effect of the water supply temperature and mass flow rate. The obtained results of the 2-D analysis are definitely closer to what occurs in actual practice. porous fins may be more efficient than traditional fins duo to large effective heat transfer area, but in cooling system, condensation leads to wet fin surface and interrupts operation of heat exchanger. Recently several researchers have studied numerically and experimentally porous fins where they are fully wet [14-16]. Neither of researchers did consider the thickness of water layer and its effect on fin efficiency and heat transfer rate. Thus, in this study, 2-D analysis is used to find the effect of variations in thickness of water layer and percent of fin wetness region on efficiency of partially wet surface fin.

## 2. Modeling

In Fig. 1 a set of attached fins in one-row heat exchanger is displayed in which broken-lines show boundaries of a unified cell of the heat exchanger. To simplify the problem and derive the governing equations, following logical assumptions are supposed.

- Moist air flow is steady and its velocity is uniform.
- Variation of thermal properties and air flow pressure are negligible.
- The thermal contact resistance between thin layer of water with air and fin surface is negligible.
- The convective heat and mass transfer coefficient are constant.
- Water film is constantly replaced at its surface with water at the same temperature.
- Natural convection and radiation heat transfer are negligible.
- Heat transfer between tube surface and air flow is not included.
- Pressure drop is not important even where the velocity variations are significant.

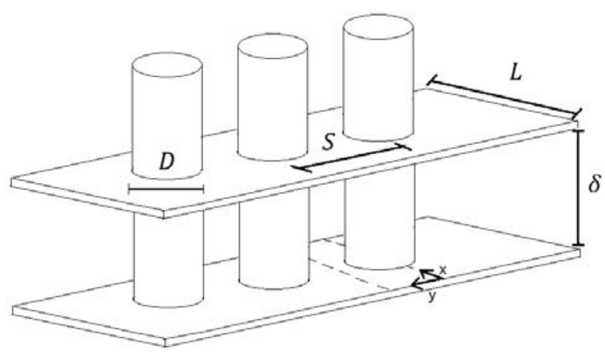


Fig.1. Schematic of fin-tube heat exchanger

Governing equations are obtained by energy balance of fin surface and energy and mass balance of moist air and water layer separately. Both fin surface and moist air flow include two governing equations because of partially wet surface fin, one for wet surface region and the other for dry surface region. But for water layer we have just one region.

### 2.1. . *Fin*

There are two equations that interpret balance of energy through fin material in two directions.

$$\frac{\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}+\frac{2k_w}{k\gamma}\frac{\partial t_w}{\partial z}=0 \qquad wet\ region \tag{1}$$

$$\frac{0\partial^2 t}{\partial x^2}+\frac{\partial^2 t}{\partial y^2}+\frac{2\dot{q}}{k\gamma}=0 \qquad dry\ region \tag{2}$$

Where:

$$\dot{q}=h(t_\infty - t) \tag{3}$$

In the above equation, $t_\infty$ show temperature of moist air.

Since the Lewis number in the moist air is close to 1 in general, the following relation can be applied. [17]

$$Le^{2/3}=\frac{h}{h_d CP_a}\approx 1 \tag{4}$$

### 2.2. *Air Stream*

Since the thickness of water layer on the fin is thin, water and fin temperatures are equal, but thickness of water layer is not ignored. Hence, energy equation will be couple for air flow.

$$\frac{\partial(t_\infty)}{\partial x}=-\frac{2h}{G_a CP_a \delta}(t_\infty - t_w) \quad wet\ region \tag{5}$$

$$\frac{\partial(t_\infty)}{\partial x}=-\frac{2h}{G_a CP_a \delta}(t_\infty - t) \quad dry\ region \tag{6}$$

The case in mass balance is the same because mass transfer is zero in dry region of fin and valuable in wet region.

$$\frac{\partial w_\infty}{\partial x}=-\frac{2h}{G_a CP_a \delta}(w_\infty - w_w) \quad wet\ region \tag{7}$$

$$\frac{\partial w_\infty}{\partial x}=0 \qquad dry\ region \tag{8}$$

To combine Eqs. (5) and (6), ratio of absolute humidity air variation to temperature variation can be obtained for wet and dry region separately.

$$c=\frac{(w_\infty - w_w)}{(t_\infty - t_w)} \qquad wet\ region \tag{9}$$

$$c=0 \qquad dry\ region \tag{10}$$

### 2.3. *Water layer*

Water layer plays a connective role that transfers heat between fin material and air steam; therefor, there is an equation for energy balance. Hint mass balance equation is satisfied automatically.

$$\frac{\partial^2 t_w}{\partial x^2}+\frac{\partial^2 t_w}{\partial y^2}+\frac{\mathrm{h}}{k_w\gamma_w}(t_\infty - t_w)\left(1+\frac{i_{fg}c}{CP_a}\right)+\frac{1}{\gamma_w}\frac{\partial t_w}{\partial z}=0 \tag{11}$$

For more simplifying the equation (11), temperature distribution of water layer in z direction is assumed linier. This assumption is logical because changing in thermal conductivity of water is unimportant.

$$\frac{\partial t_w}{\partial z}=\frac{t_w - t}{\gamma_w} \tag{12}$$

The new energy equation is written.

$$\frac{\partial^2 t_w}{\partial x^2}+\frac{\partial^2 t_w}{\partial y^2}+\frac{\mathrm{h}}{k_w\gamma_w}(t_\infty - t_w)\left(1+\frac{i_{fg}c}{CP_a}\right)+\frac{1}{\gamma_w}\left(\frac{t_w - t}{\gamma_w}\right)=0 \tag{13}$$

McQuiston [18, 19] assumed the value of c parameter for fins under fully wet- surface condition constant. But in partially wet-surface, the value of c parameter should be calculated locally by using equation (9) that needs direct relation between saturate absolute humidity and temperature. Coney [20] suggested the following parabolic equation:

$$w_w = 1.2075 - 0.9125 * 10^{-2} t_w + 0.1726 * 10^{-4} {t_w}^2 \tag{14}$$

The above equation can provide the local value of c parameter directly.

Now the boundary conditions are:

In connection of tube and fin

$$t=t_b \tag{15}$$

Since the fin is thin, no heat transfer occurs in head and tail of fin.

$$\frac{\partial t}{\partial x}=0 \qquad at\ x=0, x=L \tag{16}$$

The symmetry shown in Fig. 1 causes adiabatic in boundary, y=s/2

$$\frac{\partial t}{\partial y}=0 \qquad at\ y=0, y=S/2 \tag{17}$$

Boundary conditions for water layer are the same as fin's.

In connection of water layer and tube:

$$t_w = t_b \tag{18}$$

Adiabatic boundaries for water layer are assumed.

$$\frac{\partial t}{\partial x} = 0 \qquad at\ the\ boundryes \tag{19}$$

$$\frac{\partial t}{\partial y} = 0 \qquad at\ the\ boundryes \tag{20}$$

Temperature and absolute humidity of inlet air are known.

$$t_\infty = t_{\infty i} \qquad at\ x = 0 \tag{21}$$

$$w_\infty = w_{\infty i} \qquad at\ x = 0 \tag{22}$$

## 3. Non-dimensioning

Using the following non-dimensional variables, the governing equations can be rendered dimensionless:

$$X = \frac{x}{L} \tag{23}$$

$$Y = \frac{y}{L} \tag{24}$$

$$T = \frac{t - t_b}{t_{\infty i} - t_b} \tag{25}$$

$$T_w = \frac{t_w - t_b}{t_{\infty i} - t_b} \tag{26}$$

$$T_\infty = \frac{t_\infty - t_b}{t_{\infty i} - t_b} \tag{27}$$

$$W_w = \frac{w_w - w_b}{w_{\infty i} - w_b} \tag{28}$$

$$W_\infty = \frac{w_\infty - w_b}{w_{\infty i} - w_b} \tag{29}$$

$$C = \frac{i_{fg} c}{CP_a} \tag{30}$$

The governing equation can be non-dimensioning using dimensionless variables.

Fin equations:

$$\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} + \acute{M} \frac{\partial T_w}{\partial Z} = 0 \quad wet\ region \tag{31}$$

$$\frac{\partial^2 T}{\partial X^2} + \frac{\partial^2 T}{\partial Y^2} + M(T_\infty - T) = 0 \quad dry\ region \tag{32}$$

Air stream energy balance and mass balance equations:

$$\frac{\partial T_\infty}{\partial X} = -\acute{N}(T_\infty - T_w) \quad wet\ region \tag{33}$$

$$\frac{\partial T_\infty}{\partial X} = -N(T_\infty - T) \quad dry\ region \tag{34}$$

$$\frac{\partial W_\infty}{\partial X} = -\acute{N}(W_\infty - W_w) \qquad wet\ region \tag{35}$$

$$\frac{\partial W_\infty}{\partial X} = 0 \qquad dry\ region \tag{36}$$

And energy equation for water layer:

$$\frac{\partial^2 T_w}{\partial X^2} + \frac{\partial^2 T_w}{\partial Y^2} + \frac{\mathrm{h}L^2}{k_w \gamma_w}(T_\infty - T_w)(1 + C) + \frac{\mathrm{L}}{\gamma}\frac{\partial T_w}{\partial Z} = 0 \tag{37}$$

Where:

$$M = \frac{2hL^2}{k\gamma} \tag{38}$$

$$\acute{M} = \frac{2Lk_w}{k\gamma} \tag{39}$$

$$N = \frac{2hL}{G_a CP_a \delta} \tag{40}$$

$$\acute{N} = \frac{2hL}{G_a CP_a \acute{\delta}} \tag{41}$$

$$\acute{\delta} = \delta - \gamma_w \tag{42}$$

$$C = R\frac{W_\infty - W}{T_\infty - T} \tag{43}$$

$$R = \frac{i_{fg}(w_{\infty i} - w_\infty)}{CP_a(t_{\infty i} - t_b)} \tag{44}$$

$$W = AT + BT^2 \tag{45}$$

$$A = \frac{(t_{\infty i} - t_w)}{(w_{\infty i} - w_w)}(-0.9125 * 10^{-2} + 0.3452 * 10^{-4} * t_w) \tag{46}$$

$$B = \frac{(t_{\infty i} - t_w)^2}{(w_{\infty i} - w_w)}(0.1726 * 10^{-4}) \tag{47}$$

Now, the boundary conditions are developed as:

$$T = 0 \qquad at\ the\ connection\ with\ tube \tag{48}$$

$$\frac{\partial T}{\partial X} = 0 \qquad at\ x = 0, x = 1 \tag{49}$$

$$\frac{\partial T}{\partial Y} = 0 \qquad at\ Y = 0, Y = S/2L \tag{50}$$

$$T_w = 0 \quad at\ the\ connection\ with\ tube \tag{51}$$

$$\frac{\partial T_w}{\partial X} = 0 \quad at\ boundrys \tag{52}$$

$$\frac{\partial T_w}{\partial Y} = 0 \quad at\ boundrys \tag{53}$$

$$T_\infty = 1 \quad at\ X = 0 \tag{54}$$

$$W_\infty = 1 \quad at\ X = 0 \tag{55}$$

### 3.1. . *Efficiency*

Deriving governing equation for partially wet fin and moist air flow and solving these equations numerically, outlet air temperature and absolute humidity and distribution of fin temperature are found. Fin efficiency is calculated as ratio between actual heat transfer and ideal heat transfer that it means fin temperature is the same as tube temperature.

$$\eta = \frac{Q_{actual}}{Q_{ideal}} \tag{56}$$

Rosman [18] estimated fin efficiency as the following straightforward equation:

$$\eta = \frac{T_{\infty,e}}{T^{i}_{\infty,e}} \tag{57}$$

$T_{\infty,e}$ is dimensionless temperature of outlet air and $T^{i}_{\infty,e}$ is ideal dimensionless temperature of outlet air.

## 4. Numerical Solution

Eqs. (31-37) are complex and they include second order interdependence differential equations. So, it is difficult to solve them analytically, therefore, F.D numerical method is used. Considering nonrectangular physical domain, firstly grid should be generated by using elliptic partial differential equations. To solve equations, they are transferred from physical domain to computational domain.

Fig. 2 shows the obtained grid system by solving a system of elliptic partial differential equations and wet area that is wet 20% of fin surface near the tube. As shown in this figure, the grids distance is equal neither in x direction nor in y direction. So, for solving the governing equations numerically, governing equations have to be transferred to computational domain.

### 4.1. *Governing equations solving steps*

- Initialize water layer and fin temperature
- Solve Eqs. (31) and (32)
- Solve Eqs. (33) and (34)
- Solve Eqs. (35) and (36)
- Calculate C parameter locally
- Solve Eq. (37)
- Do the convergence test
- Repeat the first stage with new fin temperature

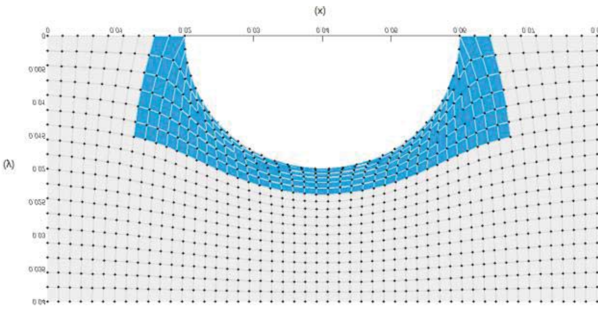


Fig.2. The obtained grid system and wet area (20% of fin surface)

Important properties of fin dimensions and air stream are shown in table 1.

Table 1. Important properties of fin dimensions and air stream

| property | value |
|---|---|
| Fin thickness | 0.0005(m) |
| Fin length | 0.08(m) |
| Fin Width | 0.04(m) |
| Tube diameter | 0.04(m) |
| Fin pitch | 0.005(m) |
| Thermal conductivity | $(\frac{W}{mK})221$ |
| Tube temperature | (K)330 |
| Inlet air velocity | (m/s)2 |
| Inlet air temperature | (K)310 |
| Relative humidity | 42% |

## 5. Validation

Effect of inlet air velocity as an important parameter has been investigated where the wet area percentage varies 5 to 100 and inlet air velocity up to 5 (m/s). Erens and Dreyer [21] have shown the effect of inlet air velocity on fin efficiency that fin geometry is similar to present study and fully wet. Low velocity, low effect on efficiency. At velocity more than 0.7(m/s), decreasing trend of efficiency is almost linear. Fig .3 illustrates the effect of inlet air velocity on fully wet fin efficiency.

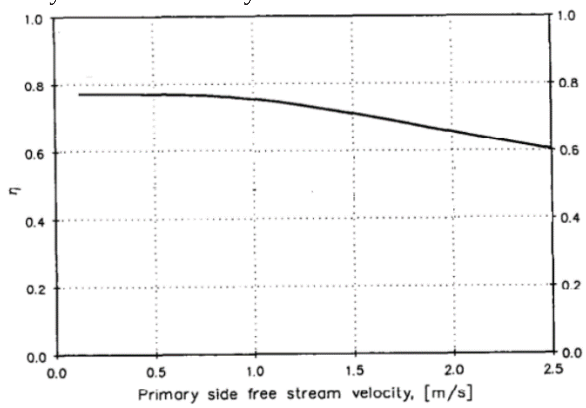


Fig.3. The fin efficiency for different inlet air velocity under fully wet fin condition [21]

To valid present work effect of inlet air velocity for several wetness percentages are calculated. Fig .4 shows efficiency of partially wet fin where the inlet air velocity varies from 0.5 (m/s) to 5(m/s). The significant result is the linear decreasing trend of efficiency as [21] has shown. More wetness percentage, more slop of linear decreasing trend of efficiency.

## 6. Result and Discussion

- *Thin water layer*

At first, calculating of efficiency of fin and heat transfer are concentrated with water layer that is thin. But the effect of this small thickness is not neglected. According to the method of solution, several thicknesses of water are determined and fin efficiency and total heat transfer (latent and sensible) under different percentage of wet area of fin are evaluated. Results show that the thickness of water layer plays insignificant roll on heat transfer and fin efficiency because this thermal resistance is not critical, thus unimportant, where the thickness of water layer is thin. This result expressed by another researcher, who also discussed cold fins under partially or full wet fins [22].

Difference in percent of wet area has a little effect on heat transfer and fin efficiency where the percentage is especially low.

Fig. 5 shows the fin efficiency when water layer is considered. Decreasing fin efficiency is obtained by increasing wet area percentage and thickness of water layer simultaneously, but with different reasons. When the percentage of wet area increases, the heat transfer increases, so gradient of fin temperature increases too. This causes decreasing of fin efficiency. On the other hand, increasing thickness of water area causes

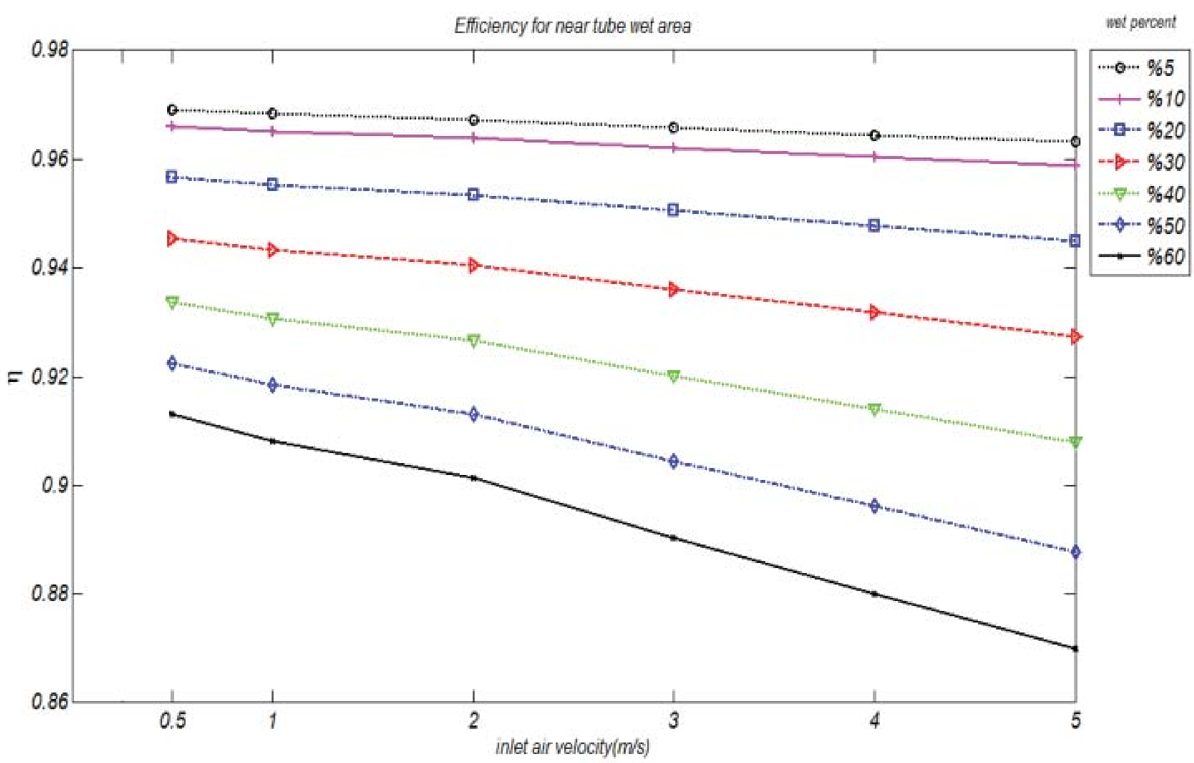


Fig.4. The fin efficiency for different inlet air velocity under partially wet fin condition

increasing in thermal resistance and decreasing heat transfer; hence, efficiency decreases.

Fig. 6 shows difference of fin efficiency between two situations. First, thickness of water layer is neglected and second, the thickness of water layer is played as an important parameter. Unimportance of water layer on fin efficiency is shown by Fig. 4 where the most difference of fin efficiency is about one percent. It is important to know that the percentage of wet area has more significant role in increasing thermal resistance, because in the fin with more wetness area thermal resistance increases, which causes more decreasing in fin efficiency.

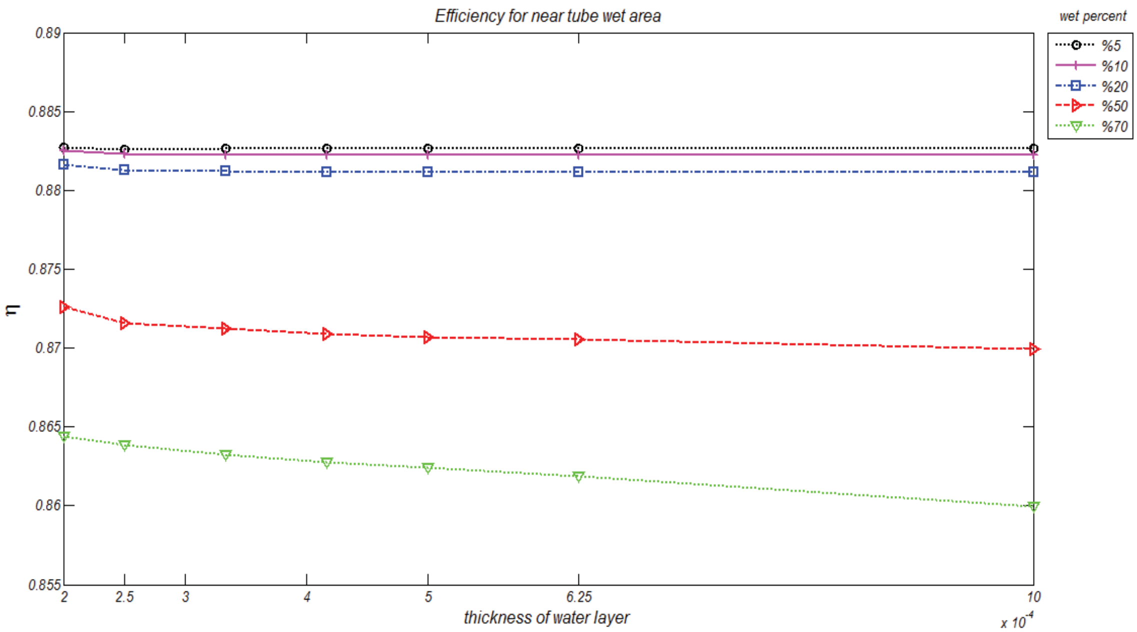


Fig.5. The fin efficiency for different thin water layer thickness and wet percentage (SI units)

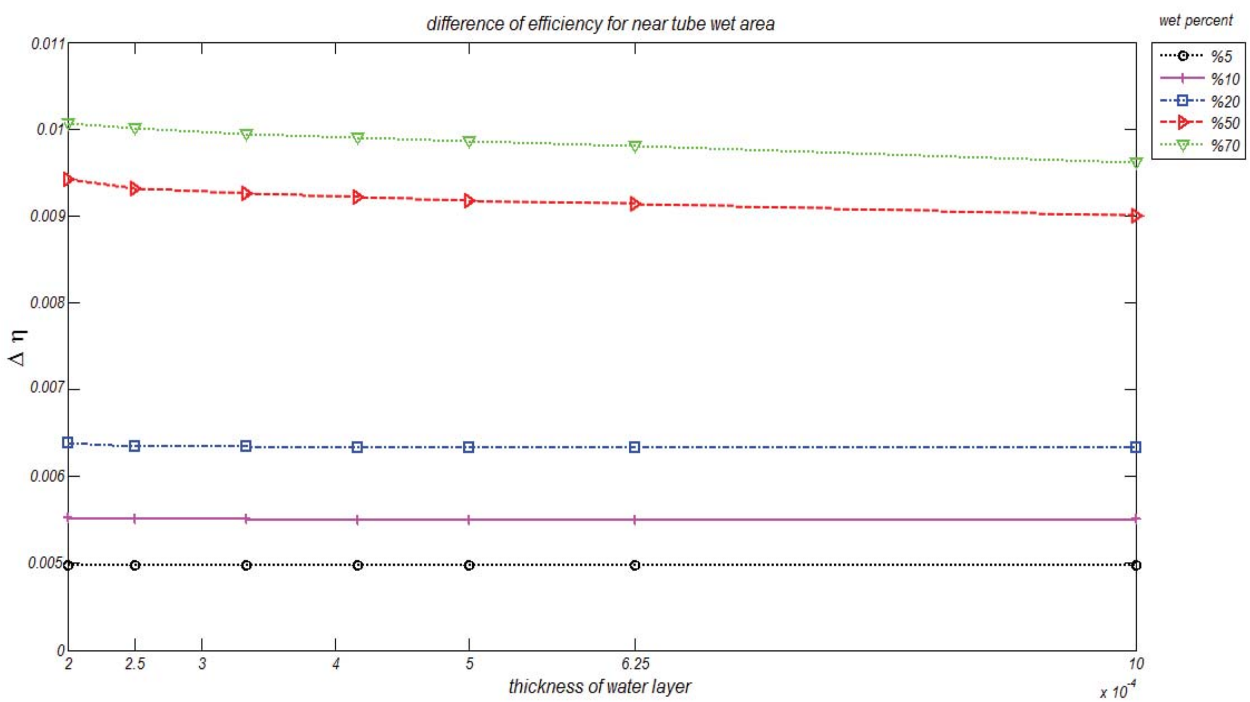


Fig.6. Difference of fin efficiency between two situations with thin water layer (SI units)

Heat transfer under the same situations is also calculated. Fig 6 shows variations of heat transfer in different percentage of wet area and thickness of water layer. It is obvious the heat transfer increases by increasing in percentage of wet area of fin, because the latent form of heat transfer becomes more as soon as the amount of wet region increases. The effect of water layer thickness is shown in Fig. 8, where the difference of heat transfer between two situations (first with thickness layer effect and second without thickness of water layer effect) is displayed. Importance of percentage of wet area on difference of heat transfer is clear because the thermal resistance is increased by increasing wet region; therefore, this difference is not major.

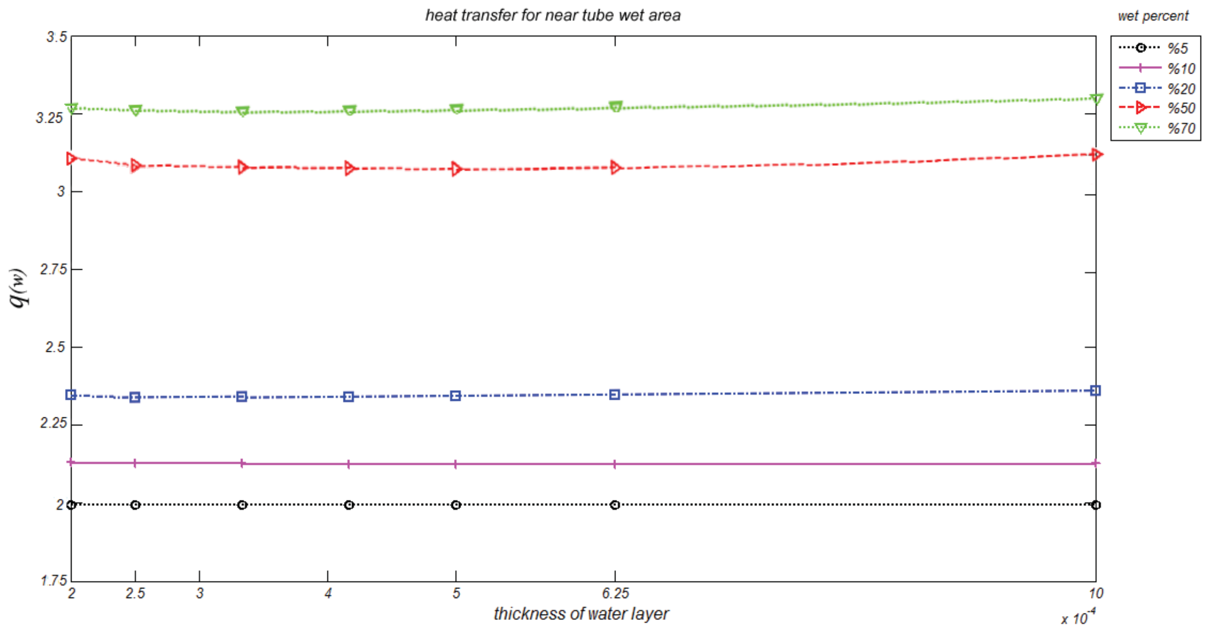


Fig.7. Heat transfer for different thin water layer thickness and wet percentage (SI units)

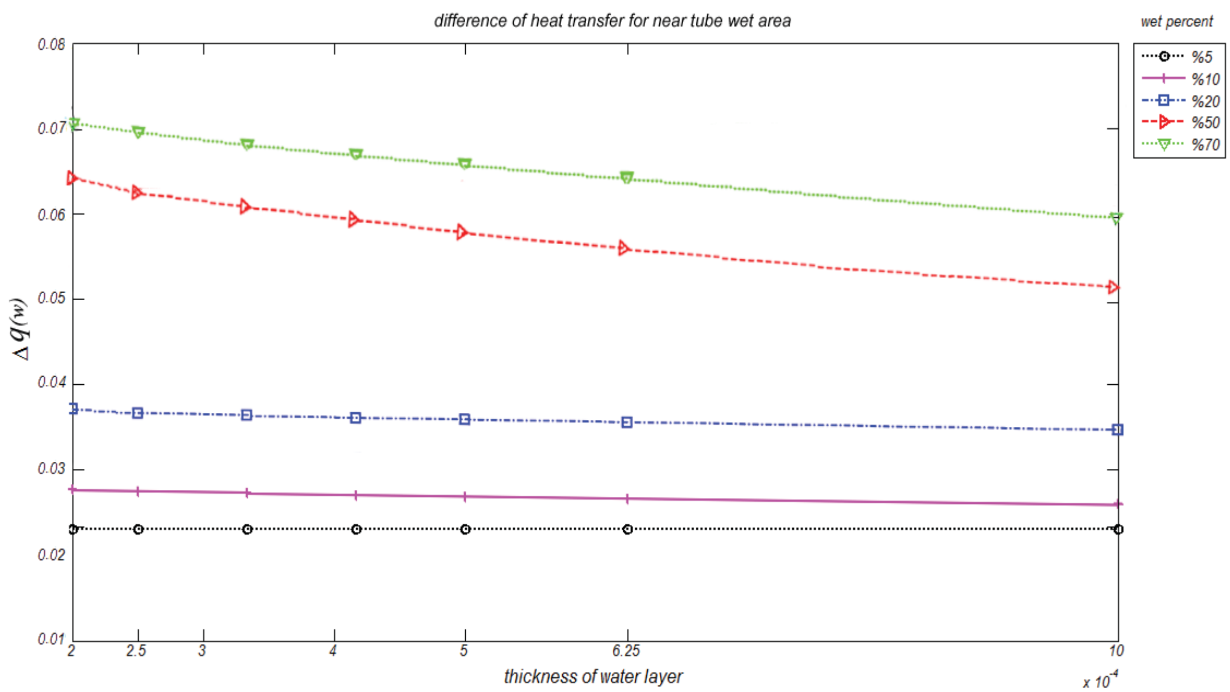


Fig.8. Difference of heat transfer between two situations with thin water layer (SI units)

- *Thick water layer*

In another state, water layer is thick and this thickness can change the velocity of air flow markedly. In this situation, the results are obtained considering frictionless. If the water layer thickness is comparable with fin pitch, significant variation in velocity of air flow occurs.

Efficiency of wet fin is shown in Fig 9. Reduction of fin efficiency through increasing percentage of wet area of fin and water layer thickness is marked. This reduction is really significant in comparison with Fig. 5, because increasing in velocity of air flow causes increasing in convective coefficients and more variations in gradient of fin temperature. Difference of fin efficiency between tow situations (first with thickness layer effect and second without thickness of water layer effect) is shown in Fig.10. It is obvious that the effect of water layer thickness and ignorance is really large because both thermal resistance and increasing in velocity cause more difference. It means that if the thickness of water layer is ignored the large error will generate, where the thickness of water layer is comparable with fin pith.

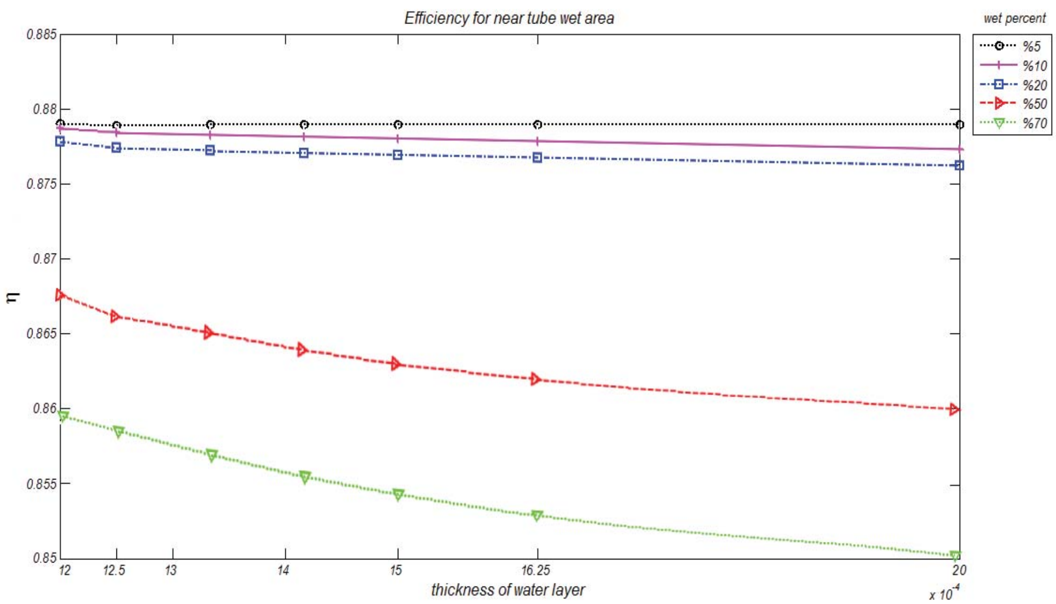


Fig.9. The fin efficiency for different thick water layer thickness and wet percentage (SI units)

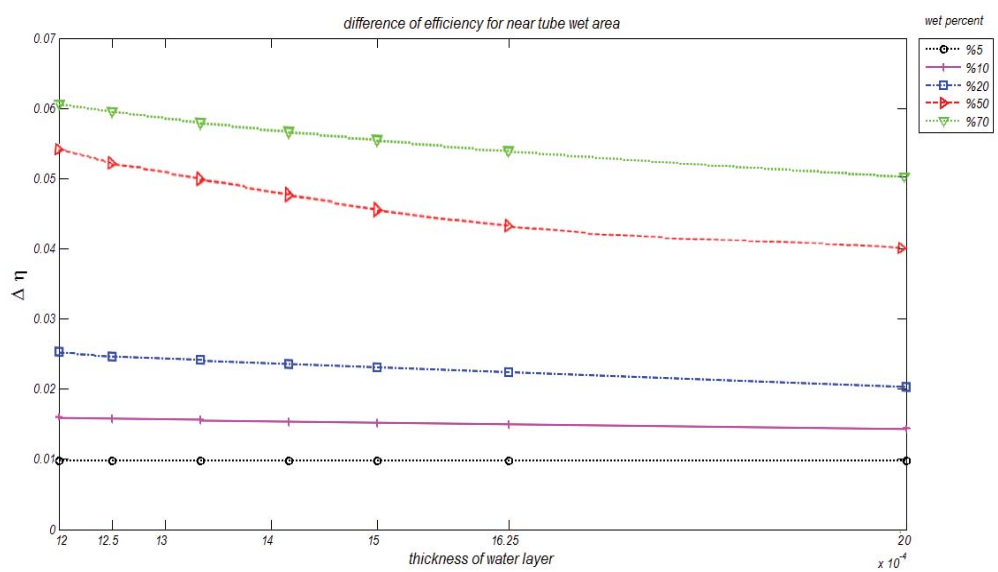


Fig.10. Difference of fin efficiency between two situations with thick water layer (SI units)

But heat transfer has a different story. Fig. 11 shows the heat transfer, where the water layer thickness is thick. Heat transfer increases when thickness of water layer becomes larger. It seems that thickness of water layer not only decreases heat transfer but also causes more heat transfer, because the increasing in velocity of air flow plays more important role than thermal resistance of thickness of water layer. Fig. 12 also shows the difference between tow ways of calculation (first with thickness layer effect and second without thickness of water layer effect). In this figure, the obtained difference of heat transfer is slight because the velocity is more important than thermal resistance.

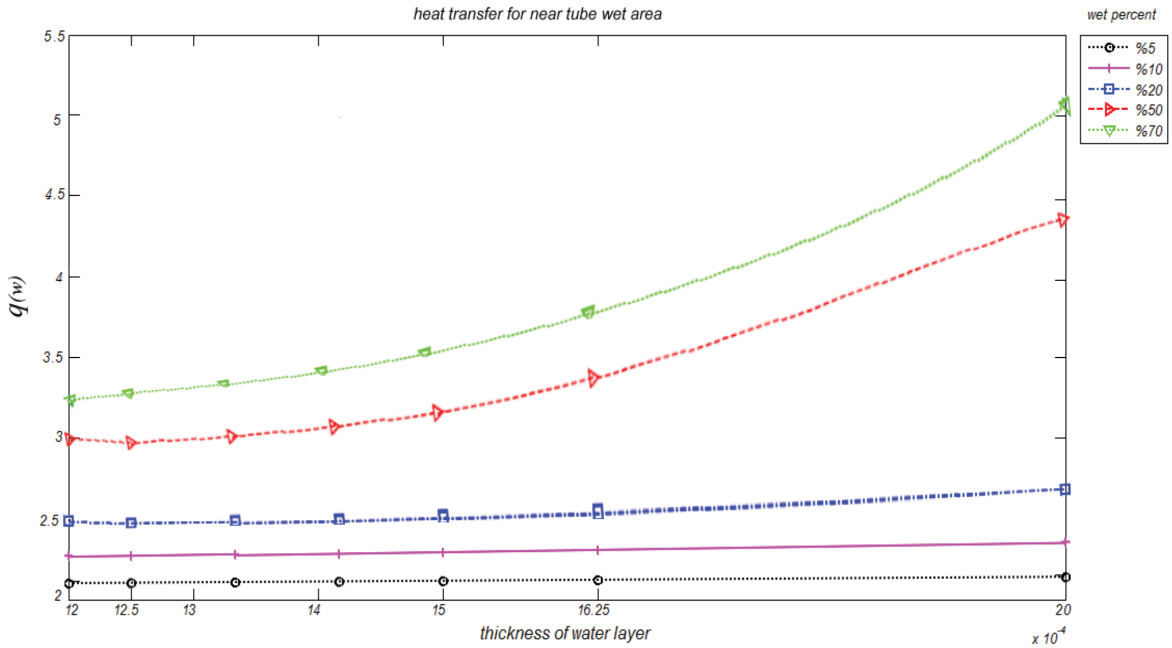


Fig.11. Heat transfer for different thick water layer thickness and wet percentage (SI units)

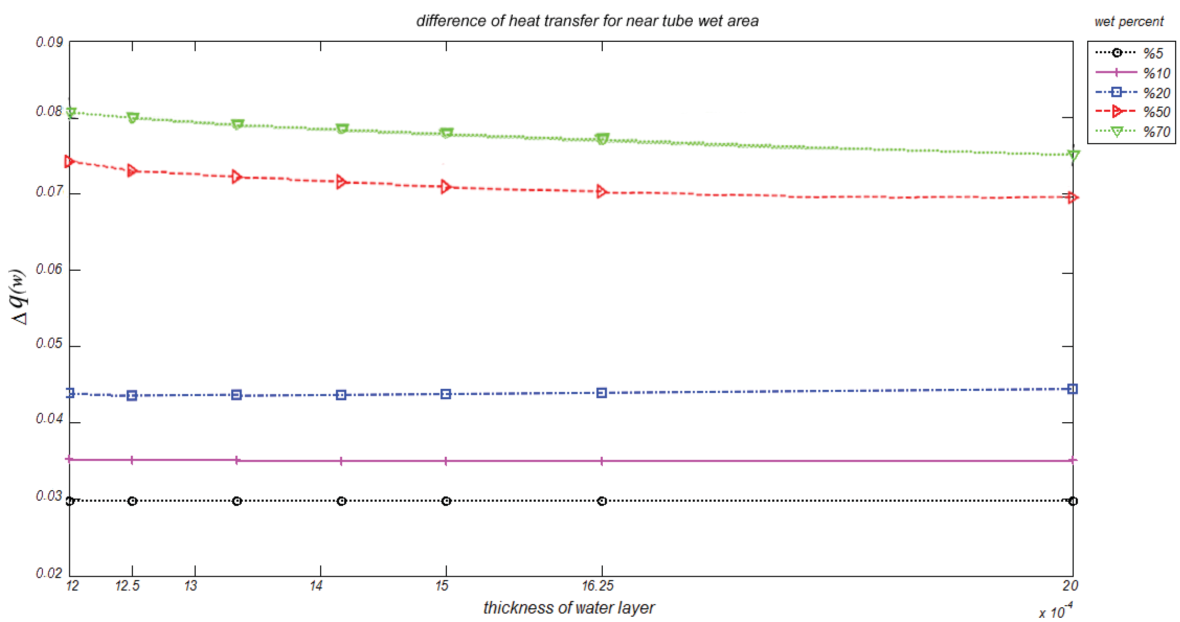


Fig.12. Difference of heat transfer between two situations whit thick water layer (SI units)

## 7. Conclusion

Thermal resistance of water layer under partially wet fin is calculated under tow situations: first thin water layer and second thick water layer. Efficiency is more dependent on wet area of fin than the thickness of water layer. It means that this thickness does not play a significant role to determine fin efficiency. This result is true for heat transfer too. But an important neglected parameter in this equations and results is pressure drop. Pressure drop cannot neglect where the thickness of water layer is thick. Therefore, this result certainly cannot be used in practical operations.

**Nomenclature**

| | |
|---|---|
| $A$ | *Parameter defined in Eq. (46)* |
| $B$ | *Parameter defined in Eq. (47)* |
| $C$ | *Parameter defined in Eq. (30)* |
| $c$ | *Parameter defined in Eq. (9)* |
| $C_p$ | *Specific heat of the air–water vapor mixture,* $(\frac{J}{kg.K})$ |
| $G_a$ | *Mass rate,* $(\frac{kg}{s.m^2})$ |
| $h$ | *Heat transfer coefficient,* $(\frac{W}{Km^2})$ |
| $h_d$ | *Mass transfer coefficient,* $(\frac{kg}{s.m^2})$ |
| $i_{fg}$ | *Evaporative latent heat,* $(\frac{j}{kg})$ |
| $k$ | *Fin thermal conductivity,* $(\frac{W}{Km})$ |
| $k_w$ | *Water thermal conductivity,* $(\frac{W}{Km})$ |
| $L$ | *Fin length,* $(m)$ |
| $M$ | *Parameter defined in Eq.(38)* |
| $M'$ | *Parameter defined in Eq.(39)* |
| $N$ | *Parameter defined in Eq. (40)* |
| $N'$ | *Parameter defined in Eq.(41)* |
| $\dot{q}$ | *heat flux,* $({W}/{m^2})$ |
| $\dot{q}_l$ | *Sensible heat flux,* $({W}/{m^2})$ |
| $\dot{q}_s$ | *Latent heat flux,* $({W}/{m^2})$ |
| $R$ | *Parameter defined in Eq.(44)* |
| $S$ | *Width fin,* $(m)$ |
| $T$ | *Dimensionless fin temperature* |
| $T_w$ | *Dimensionless water temperature* |
| $T_\infty$ | *Dimensionless air temperature* |
| $t$ | *Fin temperature,* $(K)$ |
| $t_b$ | *Tube temperature,* $(K)$ |
| $t_w$ | *Water temperature,* $(K)$ |
| $t_\infty$ | *Air temperature,* $(K)$ |

| | |
|---|---|
| $t_{\infty i}$ | *Inlet air temperature*, $(K)$ |
| $w_w$ | *Saturate absolute humidity at water temperature*, $(\frac{kg}{kg})$ |
| $w_\infty$ | *Air absolute humidity*, $(\frac{kg}{kg})$ |
| $w_{\infty i}$ | *Inlet air absolute humidity*, $(\frac{kg}{kg})$ |
| $\delta$ | *Fin pitch*, $(m)$ |
| $\eta$ | *Fin efficiency* |
| $\gamma$ | *Fin thickness*, $(m)$ |
| $\gamma_w$ | *Water thickness*, $(m)$ |

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