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A Modified McQuiston model for evaluating efficiency of wet fin considering effect of condensate film moving on fin surface

Xiangguo Xu, Liang Xia, Mingyin Chan, Shiming Deng *

Department of Building Services Engineering, The Hong Kong Polytechnic University, Hung Hom, Kowloon, Hong Kong SAR, PR China

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Abstract

A modified McQuiston model for evaluating the wet fin efficiency of cooling and dehumidification coils has been developed by modifying the existing popular McQuiston model and is reported in this paper. The condensate film moving on fin surfaces and its impacts on heat transfer have been taken into account in deriving the governing equation for fin temperature, and consequently, the enthalpy change of the moving condensate film has been included in the governing equation for fin temperature. The modified McQuiston model was validated by comparing its predictions with those using the McQuiston model under the same operating and boundary conditions. It is expected that the modified McQuiston model developed can be applied not only to the commonly seen air cooling and dehumidification but also to certain special operations such as industrial steam condensation.

Keywords: Wet fin efficiency; Moving condensate film; Total condensing rate; Governing equation

1. Introduction

Finned heat exchangers are used extensively to enhance heat and mass transfer. Among finned heat exchangers, an air cooling and dehumidifying coil is an important component in building heating, ventilation and air conditioning (HVAC) systems. Normally, simultaneous air cooling and dehumidification take place in an air cooling coil [1] and condensate moves downwards on the outer surface of the coil as a film when the coil surface temperature is below the dew point of the air passing through the coil and water vapor condenses on the fin surfaces. The fin efficiency is a key parameter for investigating the performance of a cooling coil. It is well known that a number of factors, such as fin geometry and material, air and coolant temperatures, flow rates, etc. affect fin efficiencies, as well as heat and mass transfer.

Over the past few decades, there have been studies on the efficiency of fins used in air cooling and dehumidification coils. Threlkeld [2] developed a model to investigate the fin efficiency of an air cooling coil by assuming a linear relationship between the air temperature and the corresponding saturated air enthalpy. His results suggested that the wet fin efficiency was independent of the relative humidity of the air upstream of the coil.

Nadar [3] performed a study on the efficiency of a wet vertical longitudinal fin attached to a pipe. By neglecting the contribution to the film condensation due to convective heat transfer, the governing equations for film thickness and fin temperature were obtained. Burmeister [4] provided a theoretical solution to the governing equations developed by Nadar. It was shown that both fin efficiency and the thickness of the condensate film were influenced by two dimensionless fin parameters, F_1 and F_2, defined as

$$F_1 = \frac{\rho_l(\rho_l - \rho_u) g h_{f,g} H^3}{k_l \mu_l (T_b - T_b)} \quad (1)$$

$$F_2 = \frac{t k_s}{2 H k_l} \quad (2)$$

These parameters were not related to the relative humidity of the moist air upstream of the fin, and neither was the wet

* Corresponding author. Tel.: +852 27665859; fax: +852 27657198.
E-mail address: besmd@polyu.edu.hk (S. Deng).

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Nomenclature

A	surface area of fin, m^2
A_c	cross sectional area of fin, m^2
C	ratio defined as $(w - w_a)/(T - T_a)$ by Eq. (8), kg/(kg dry air °C)
C_{pa}	specific heat of moist air, kJ/(kg °C)
C_{pw}	specific heat of water, kJ/(kg °C)
F_1	dimensionless fin parameter, $[\rho_l\ (\rho_l - \rho_v)\ g\ h_{f,g} H^3]/[k_l\ \mu_l\ (T_a - T_b)]$
F_2	dimensionless fin parameter, $(t\ k_s)/(2H\ k_l)$
g	gravitational acceleration, m/s^2
H	fin height, m
h	convective heat transfer coefficient, $W/(m^2\ °C)$
h_m	convective mass transfer coefficient, $kg/(m^2\ °C)$
$h_{f,g}$	latent heat of vaporization of water, W/kg
k_l	thermal conductivity of water, W/(m °C)
k_s	thermal conductivity of fin, W/(m °C)
L	fin width, m
Le	Lewis number
$m_{co}(x)$	total condensing rate, kg/s, defined by Eq. (7)
P	fin perimeter, m
RH	relative humidity of bulk air
T	fin temperature, °C
T_a	dry bulb temperature of bulk air, °C
$T_{a,w}$	wet bulb temperature of bulk air, °C
T_b	fin base temperature, °C
t	fin thickness, mm
w	moisture content of air near fin surface, kg/kg dry air
w_a	moisture content of bulk air, kg/kg dry air
x, y	spatial coordinate, m
Greek letters	
θ	relative temperature, $T - T_a$, °C
θ_b	relative fin base temperature, $T_b - T_a$, °C
μ_l	dynamic viscosity of saturated water, pa s
ρ_l	density of saturated water, kg/m^3
ρ_v	density of saturated vapor, kg/m^3
η	fin efficiency
η_{Mc}	fin efficiency obtained by using McQuiston model

fin efficiency. McQuiston [5] studied the efficiency of a wet straight fin by adding the effect of condensation heat to the governing equation for fin temperature developed by Schmidt [6], and thus, a new governing equation for fin temperature considering combined heat and mass transfer was derived as follows:

$$\frac{d^2\theta}{dx^2} = \frac{hP}{k_s A_c}\left(1 + \frac{C h_{f,g}}{C_{pa}}\right)\theta \tag{3}$$

Eq. (3) has been well known as the McQuiston model for evaluating wet fin efficiency. In deriving the McQuiston model, it was assumed that the driving force for the mass transfer, i.e. the difference between the moisture content of bulk air and that of air near the fin surface, was linearly related to the difference between the temperature of bulk air and that of air near the fin surface. The McQuiston model suggested that the wet fin efficiency strongly depended on the moisture content of the air upstream of a cooling coil and was lower than the dry fin efficiency. The McQuiston model has been well adopted in evaluating the wet fin efficiency of air cooling and dehumidifying coils ever since it was established.

Using the McQuiston model, Wu and Bong [7] analyzed the efficiency of a partially wet straight fin by treating separately the driving forces for heat transfer and mass transfer. It was suggested that the efficiency of a partially wet fin depended significantly on the relative humidity of the air upstream of the cooling coil.

The application of the McQuiston model has been extended to other types of fins. For example, McQuiston and Parker [8] analyzed the wet fin efficiency of circular fins. The analytical solution for a circular fin was presented by Kern and Kraus [9]. Hong and Webb [10] developed a quantitative evaluation method to simplify wet circular fin efficiency calculations. Elmahdy and Biggs [11] derived an algorithm to determine the efficiency of a circular fin. It was indicated that the relative humidity of the air upstream of the fin would significantly impact the wet fin efficiency. Kundu [12] applied the McQuiston model to study the efficiency of a fully wet straight tapered fin.

As can be seen from the previously reported studies, there existed two different views, derived by Threlkeld and McQuiston, as to whether the wet fin efficiency was influenced by air moisture content or relative humidity. However, among all the previous studies on wet fin efficiency, the effect of the moving condensate film on heat transfer in a fin-film system has never been taken into account. Xu et al. [13] recently suggested through an experimental study that the condensate retained on a fin surface should not have been simply regarded as a still layer but a layer moving downwards, as the condensate, in fact, kept draining off the fin surface. With regard to the wet fin efficiency, since the temperature of the condensate flow changed in the direction of motion because the fin temperature changed, the motion of the condensate layer would, consequently, change the energy balance between the moving condensate film and the fin surfaces. It followed that in evaluating the efficiency of wet fins, the effect of condensate moving downwards on a fin surface on the heat transfer in a fin-film system must be considered.

This paper reports on the development of a new model, which can be regarded as a modified McQuiston model, for