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Effects of psychrometric properties on fin performances of minimum envelope shape of wet fins

Balaram Kundu [a,*], Kwan-Soo Lee [b,*]

[a] Department of Mechanical Engineering, Jadavpur University, Kolkata 700 032, India
[b] School of Mechanical Engineering, Hanyang University, 222 Wangsimni-ro, Seongdong-Gu, Seoul 133-791, Republic of Korea

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ABSTRACT

A method based on the variational principle is used to determine a minimum shape of wet fins for effective transfer of heat in fins. In many air-cooling applications, fin surface becomes wet as the surface is lower than the dewpoint temperature of the surrounding air. On the wet surface, both the sensible and latent heat transfer take place due to an existence of respective driving forces. The fin temperature is also dependent upon the latent heat released on its surface from the condensed moisture. The humidity ratio of air adjacent to the fin surface for the latent heat transfer is a psychrometric function with temperature. Due to this, the analysis of wet fins is always difficult as the governing fin equation is complex in nature. A simple model based on a linearity function between humidity ratio and temperature has been an alternative approximate model. The present analysis is attempted to estimate an envelope shape of wet fins by minimising fin volume for a constraint heat transfer rate for an actual variation of humidity ratio with temperature. The effect of psychrometric properties of air on the minimum shape of wet fins has been studied in the present work. The fin performance for the least envelope shape of wet fins is proposed with the variation of thermo-psychrometric parameters. A comparison of results for the variation of linear and nonlinear humidity ratio with temperature on the analysis of the minimum envelope shape has been systematically investigated. This study is also highlighted errors in connection with the linear model to determine the least fin shape for efficient heat transfer. This analysis may be extremely important in those augmentation heat transfer apparatuses where a gain in weight creates always an excessive overburden. Due to this, the selection of the function of humidity ratio as a fin surface temperature may be extremely important for calculating minimum weight of a fin for the same heat transfer duty.

1. Introduction

A reduction of fin material is of an extreme importance where the system weight is a design constraint in case of the mobile application. In refrigeration, air conditioning, and aerospace systems, fins are attached to the primary surface to enhance the rate of heat transfer between the surrounding air and the refrigerant flowing through the evaporator tubes. Here, the fin surface is usually wet due to condensation of moisture by evolving latent heat.

It is a fact that the fin with variable thickness has ability to transfer effective heat and it may converge or diverge depending upon the fin surface conditions [1]. Schmidt [2] heuristically demonstrated that the optimum fin has a constant temperature gradient. Duffin [3] proved the Schmidt's statement mathematically for assuming constant thermophysical prosperities. The effect of arc length on the fin optimisation for a dry surface fin has been studied by Hanin and Campo [4] and they obtained a circular profile of the optimum fin instead of the parabolic one. Kundu and Das [5] investigated the volumetric heat generation on the optimum profile of longitudinal, spine, and annular fins. With the non-uniform heat transfer coefficient as a function of a spatial coordinate, Natarajan and Shenoy [6] calculated the optimum shapes of convective pin fins. Fabbri [7] employed genetic algorithm to determine the optimum profile shape of fins. All the above research works [2–7] focused on determination of the optimum profile shape under dry surface conditions for calculating a maximum heat transfer per unit volume.

Recently several design conditions have been taken to improve the heat transfer rate per unit volume for a common geometric fin. Among these, the porous and moving conditions are favourable design criteria investigated by many researchers. Bhanja et al. [8] found an enhancement of heat transfer from a continuously moving porous rectangular fin. Using Differential Transformation

* Corresponding authors.
E-mail addresses: bkundu@mech.net.in (B. Kundu), ksleehy@hanyang.ac.in (K.-S. Lee).

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Nomenclatures

A	humidity ratio parameter for linear model, see Eq. (34)
B	a gradient of saturation curve with consideration of straight line on psychometric chart, K^{-1}, see Eq. (35)
C_j	coefficient of a polynomial function for humidity ratio as a function of temperature for $j = 0, 1, 2, 3$, see Eq. (2)
$c_{p,a}$	specific heat of dry air at constant pressure (J kg^{-1} K^{-1})
D_j	variables defined in Eq. (5) for $j = 0, 1, 2, 3$
E	a function defined in Eq. (A1)
E_j	variables used in Eq. (A28)
h	convective heat transfer coefficient (W m^{-2} K^{-1})
h_{fg}	latent heat of condensation (J kg^{-1})
k	thermal conductivity of fin material (W m^{-1} K^{-1})
l	fin length (m)
L	dimensionless fin length, hl/k
Le	Lewis number
p	ambient pressure (bar)
p_v	vapour pressure of moisture (bar)
q	actual heat transfer rate (W)
Q	dimensionless heat transfer rate, $q/2kw(T_a - T_b)$
q_i	ideal heat transfer rate (W)
Q_i	dimensionless ideal heat transfer rate, $q_i/2kw(T_a - T_b)$
q_w	heat transfer rate through the base area for no fin condition (W)
Q_w	dimensionless heat transfer rate without fin, $q_w/2kw(T_a - T_b)$
s	surface area per unit width (m)
S	dimensionless surface area per unit width, hs/k
T	local fin surface temperature (K)
T_a	ambient temperature (K)
T_b	fin base temperature (K)
T_L	tip temperature (K)
U	dimensionless fin volume per unit width, $h^2V/2wk^2$
V	fin volume (m^3)
w	fin width (m)
x, y	coordinates (m)
X, Y	dimensionless coordinates, $hx/k, hy/k$
y_b	semi-base thickness as shown in Fig. 1 (m)
Y_b	dimensionless semi-base thickness, hy_b/k
y_L	semi-tip thickness (m)
Y_L	dimensionless semi-tip thickness, hy_L/k
WLA	with length of arc
WOLA	without length of arc

Greek letters

α	spatial function defined in Eq. (A7)
ε	fin effectiveness
η	fin efficiency
λ	Lagrange multiplier
θ	dimensionless fin surface temperature, $(T_a - T)/(T_a - T_b)$
θ_L	dimensionless tip temperature, $(T_a - T_L)/(T_a - T_b)$
θ_p	dimensionless temperature parameter defined in Eq. (33)
ϕ	relative humidity
$\varphi(X)$	spatial function
ξ	Latent heat parameter, $h_{fg}/c_{p,a}Le^{2/3}$ (K)
ψ	temperature function, see Eq. (A25)
ω	humidity ratio for saturation air on the fin surface (kg of w.v. per kg of d.a.)
ω_a	humidity ratio for ambient air (kg of w.v. per kg of d.a.)

Method, Collocation Method and Least Square Method, Hatami et al. [9] determined the temperature distribution in porous fins with internal heat generation. Turkyilmazoglu [10] considered a stretching and shrinking mechanism mounted on the surface of a longitudinal rectangular fin for the heat transfer enhancement and efficiency. Homotopy perturbation method was employed by Cuce and Cuce [11] to investigate the thermal performance of straight porous fins whereas Moradi et al. [12] studied the effect of convection and radiation in the analysis of performances of porous triangular fins with temperature dependent thermal conductivity. Das and Ooi [13] analysed to predict the multiple combinations of parameters in a naturally convective porous fin for a given temperature distribution.

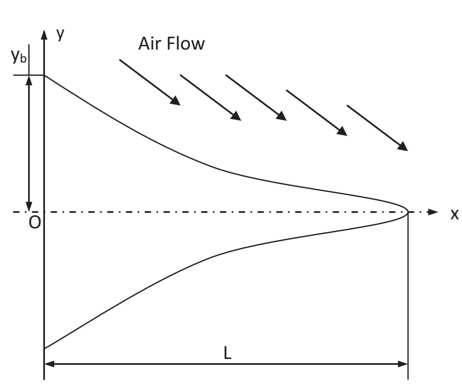

Fig. 1. Schematic diagram of an optimum fin.

An extensive literature of wet fins under dehumidification of air has been presented for a specified geometric fin [14–23]. However, for searching a critical design condition, thermal engineers always engage to determine a fin shape which transfer maximum heat transfer per unit volume. This design condition is always superior in comparison to the optimisation of the common geometric fins. With this design consideration for determination of the optimum profile shape, it may be extremely important where a system moves from one place to another due to extra cost requirement for any additional weight carrying. Therefore, the minimum profile shape fin may reduce the transportation cost in the application of automobiles and aircrafts for the attachment of the air conditioning system. The research on optimum profile fins under dehumidifying conditions was commenced firstly by Kundu [24]. He determined optimum profiles of three commonly used fins, namely, longitudinal, annular and spine, under wet conditions. In his work, he assumed the humidity ratio of saturated air at the fin surface to be a linear function with temperature.

It is important to note from the previous studies that a linear model is generally used between the mass driving force and the temperature difference between the surface and ambient which capitulates to a limited solution for a given inlet air temperature and fin base temperature. Physically, the relation between humidity ratio of a saturated air and dry bulb temperature is not a linear one. However, it always follows the psychrometric correlations. The comparison of these two models for the minimum shape of fins under wet condition has not yet been done. The study for fin performances of the minimum shape of wet fins with variation of psychrometric parameters for both the models may be extremely essential in those vehicle systems where the weight