International Communications in Heat and Mass Transfer xxx (2015) xxx–xxx

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ichmt

Optimal tip clearance in the laminar forced convection heat transfer of a finned plate in a square duct☆

Hae-Kyun Park, Bum-Jin Chung *

Department of Nuclear Engineering, Kyung Hee University, 1732 Deokyoung-daero, Giheung-gu, Yongin-si, Gyeonggi-do 446-701, Republic of Korea

ARTICLE INFO

Available online xxxx

Keywords:
Fin
Square duct
Tip clearance
Forced convection
Heat transfer
Analogy
Experiment
Numerical calculation

ABSTRACT

A joint experimental and numerical investigation of laminar forced convection heat transfer of a finned plate in a square duct is reported. The heat transfer rates were measured for fixed fin heights, fin thicknesses, and base-plate geometries, with systematically varied fin spacing, tip clearance, Prandtl number and Reynolds number. The optimal tip clearances were identified, which maximized the total heat transfer. The total heat transfer rate at the optimal tip clearance was maximum 33% higher than that at the largest tip clearance. It decreased as the Prandtl number increased and as the Reynolds number increased. The dependence of the heat transfer rate on the tip clearance weakened as the fin spacing increased.

1. Introduction

Fins are employed commonly to enhance the heat transfer of cooling systems [1–5]. They achieve this by increasing the total surface area available for heat transfer; however, if an excessively small fin spacing S is used, the heat transfer becomes limited by frictional losses [1]. Thus, an optimal fin spacing is expected, which will depend on the geometry and operating conditions of the heat exchanger; optimization of the fin spacing has been the focus of numerous studies [1,3,4].

Where a finned plate is located in a duct, optimization is also required for the tip clearance, i.e., the distance between the tips of the fins and the wall. A forced flow will partially bypass the fins where there is a gap between the fin tips and the wall. An excessively small tip clearance limits the bypass flow and reduces the heat transfer at the fin tips, because this reduces the flow at the fin tips; however, as the tip clearance increases, the bypass flow increases, limiting the total heat transfer of the finned plates. Therefore, there is the optimal tip clearance, which maximizes the overall heat transfer.

Here we investigated laminar forced convection heat transfer at a finned plate as a function of the tip clearance, the Prandtl number (Pr) and the Reynolds number (Re_{Dh}). We systematically varied the tip clearance, Pr and Re_{Dh}. The fin height, H was 5×10^{-3} m, the fin thickness, t was 3×10^{-3} m, the width of the base plate, W was 5×10^{-2} m and the length of the base plate, L was 5×10^{-2} m. Two different fin spacings, S of 2×10^{-3} m and 7×10^{-3} m were investigated, and the tip clearance was varied from 0 m to 1.5×10^{-2} m. The Pr ranged from 0.7 to 2014 and the Re_{Dh} from 500 to 1000. Experimental and numerical analyses were performed for $Pr = 2014$, with a range of moderate tip clearances to verify the simulated data. For very small tip clearances, which are difficult to achieve experimentally, only numerical simulations were carried out. Simulations were conducted using FLUENT 6.3.26. Mass transfer experiments were performed by exploiting the analogy of heat transfer using a cupric acid–copper sulfate (H_2SO_4–$CuSO_4$) electroplating system.

2. Background

2.1. General characteristics of finned plates

Heat transfer at a finned plate is influenced by H, S, t, L and W, as well as the flow conditions, as shown in Fig. 1. The total heat transfer increases with increasing H due to the increased area available for heat transfer. The fin spacing may either increase or decrease the heat transfer, as discussed above. The flow velocity also affects the heat transfer [2].

2.2. Fin spacing

Culham et al. [3] investigated the optimal design of a finned plate and calculated the heat transfer rate as a function of the fin spacing, with laminar forced convection and fixed values of W and t. As S increased, the total heat transfer area decreased, and the total heat transfer decreased.

☆ Communicated by W.J. Minkowycz.

* Corresponding author.
E-mail address: bjchung@khu.ac.kr (B.-J. Chung).

http://dx.doi.org/10.1016/j.icheatmasstransfer.2015.02.010
0735-1933/

Nomenclature

A	Flow area [m^2]
C	Molar concentration [mole/m^3]
D_h	Hydraulic diameter ($=4P/A$) [m]
D_m	Mass diffusivity [m^2/s]
F	Faraday constant, [96,485 C/mol]
g	Gravitational acceleration [9.8 m/s^2]
Gr_L	Grashof number ($g\beta\Delta TL^3/\nu^2$)
h	Heat transfer coefficient [W/m^2·K]
h_m	Mass transfer coefficient [m/s]
H	Fin height [m]
I_{lim}	Limiting current density [A/m^2]
k	Thermal conductivity [W/m·K]
L	Length of base plate [m]
n	Number of electrons in charge transfer reaction
Nu_{Dh}	Nusselt number (h_hD_h/k)
P	Wetted perimeter [m]
Pr	Prandtl number (ν/α)
Ra_L	Rayleigh number (Gr_LPr)
Re_{Dh}	Reynolds number (VD_h/ν)
S	Fin spacing [m]
Sc	Schmidt number (ν/D_m)
T	Temperature [K]
t	Fin thickness [m]
t_n	Transference number
U_x	Uncertainty of x
V	Velocity of flow [m/s]
W	Width of the base plate [m]

Greek symbols

α	Thermal diffusivity [m^2/s]
β	Volume expansion coefficient [1/K]
μ	Viscosity [kg/m·s]
ν	Kinematic viscosity [m^2/s]
ρ	Density [kg/m^3]

Seri Lee [1] analyzed heat transfer at a finned plate numerically and found an optimal number of fins. As S decreased from a relatively large spacing, the total heat transfer increased due to the increased area; however, an optimal spacing was found, and a further decrease in S resulted in a drop in the total heat transfer due to a decrease in the flow rate because of increased frictional losses.

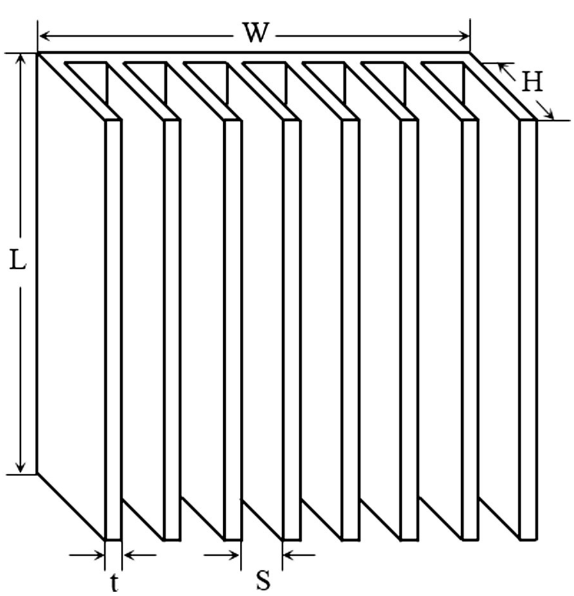

Fig. 1. Finned plate geometry.

Dogan et al. [4] performed numerical and experimental analyses of the convection heat transfer at finned plates with S/H ranging from 0.04 to 0.18, with the tip clearance normalized to H in the range of 0.20–0.75, and with Re_{Dh} ranging from 250 to 2300. When S was smaller than the optimal value, the boundary layers that developed at the fin surface overlapped, which obstructed the fluid flow.

2.3. Tip clearance

Dogan et al. [4] reported that the total heat transfer increased as the tip clearance decreased. In particular, the increase in the total heat transfer as the tip clearance decreased was larger when Re_{Dh} was larger. Kim et al. [5] and Min et al. [6] found an optimal tip clearance, whereby maximum forced convection heat transfer of the finned plate occurred using experimental and simulation analyses. The heat transfer increased as the area available for heat transfer increased and as the flow velocity near the tips of the fins increased gradually, whereas the heat transfer decreased with an increase in the bypass flow at the tip clearance region for wide tip clearances.

2.4. Prandtl number

Bejan and Scuibba [7] performed theoretical analyses of laminar forced convection heat transfer between parallel plates with Pr varying from 0.7 to 1000. They reported that the thickness of the boundary layer varied with Pr, which affected the optimal plate spacing at which maximum total heat transfer occurred.

3. Experiment

3.1. Methodology

A heat transfer problem can be solved using a mass transfer experiment based on the analogy between heat and mass transfer. This is because the mathematical model used to describe the two phenomena is the same [8]. Table 1 lists the corresponding governing parameters. Here, measurements were made using a sulfuric acid–copper sulfate (H_2SO_4–$CuSO_4$) electroplating system. The technique is attractive, as it provides a simpler, cheaper, and more accurate method by measuring the electric current, compared with the use of temperature [9,10]. This technique was developed by several research groups [11–14] and is a well-established experimental methodology [15–21]. The physical properties were calculated using Eqs. (1)–(8), which were reported by Fenech and Tobias and were found to be accurate to within ±0.5% at 20–22 °C [14].

$$\rho\left(\mathrm{kg/m^3}\right) = \left(0.9978 + 0.06406C_{H_2SO_4} - 0.00167C^2_{H_2SO_4} + 0.12755C_{CuSO_4} + 0.01820C^2_{CuSO_4}\right) \times 10^{-3} \tag{1}$$

$$\mu(cp) = 0.974 + 0.1235C_{H_2SO_4} + 0.0556C^2_{H_2SO_4} + 0.5344C_{CuSO_4} + 0.5356C^2_{CuSO_4} \tag{2}$$

Table 1
Dimensionless numbers for the analogous systems.

Heat transfer		Mass transfer	
Nusselt number	$\frac{h_h D}{k}$	Sherwood number	$\frac{h_m D}{k}$
Prandtl number	$\frac{\nu}{\alpha}$	Schmidt number	$\frac{\nu}{D_m}$
Rayleigh number	$\frac{g\beta\Delta T D^3}{\alpha\nu}$	Rayleigh number	$\frac{gD^3}{D_m\nu}\frac{\Delta\rho}{\rho}$