International Journal of Heat and Mass Transfer 52 (2009) 2119–2131

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

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

Flow visualizations and heat transfer measurements for a rotating pin-fin heat sink with a circular impinging jet

Tzer-Ming Jeng [a,*], Sheng-Chung Tzeng [b], Hong-Ru Liao [b]

[a] *Department of Mechanical Engineering, Air Force Institute of Technology, GangShan 820, Taiwan, ROC*
[b] *Department of Mechanical Engineering, Chienkuo Technology University, ChanGhua 500, Taiwan, ROC*

ARTICLE INFO

Article history:
Received 30 May 2008
Received in revised form 20 October 2008
Available online 26 December 2008

Keywords:
Heat transfer
Jet impingement
Rotating heat sink
Smoke visualization

ABSTRACT

This work experimentally investigated the fluid flow and heat transfer behaviors of jet impingement onto the rotating heat sink. Air was used as impinging coolant, while the square heat sinks with uniformly in-line arranged 5×5 and 9×9 pin-fins were employed. The side length (L) of the heat sink equaled 60 mm and was fixed. Variable parameters were the relative length of the heat sink ($L/d = 2.222$ and 4.615), the relative distance of nozzle-to-fin tip ($C/d = 0$–11), the jet Reynolds number ($Re = 5019$–25,096) and the rotational Reynolds number ($Re_r = 0$–8114). Both flow characteristics of stationary and rotating systems were illustrated by the smoke visualization. Besides, the results of heat transfer indicate that, for a stationary system with a given air flow rate, there was a larger average Nusselt number (Nu_0) for the 9×9 pin-fin heat sink with $L/d = 4.615$ and $C/d = 11$. For a rotating system, a bigger Re_r meant a more obvious heat transfer enhancement (Nu_Ω/Nu_0) in the case of smaller Re, but Nu_Ω/Nu_0 decreased with increasing Re. In this work, Nu_Ω/Nu_0 in $L/d = 2.222$ is higher than in $L/d = 4.615$; among the systems in $L/d = 2.222$, bigger Nu_Ω/Nu_0 exists in the case of $C/d = 9$–11, but among the systems in $L/d = 4.615$, bigger Nu_Ω/Nu_0 exists in the case of $C/d = 1$–3. Finally, according to the base of $Nu_\Omega/Nu_0 \geqslant 1.1$, the criterion of the substantial rotation was suggested to be $Re_r/Re \geqslant 1.154$.

1. Introduction

Jet impingement can provide large localized cooling for many industrial applications, such as annealing metals, tempering glass, drying paper and textiles, and cooling of gas turbine blades and electronics equipment. Jambunathan et al. [1] reviewed the heat transfer data for single circular jet impingement and summarized four flow zones commonly appeared at such systems: (1) initial mixing zone, (2) established jet zone, (3) impingement zone, and (4) wall jet zone. In an impinging-cooling system, the fluid flow strongly influences the heat transfer. Consequently, the parameters that may affect heat transfer include: the jet Reynolds number (Re), the Prandtl number (Pr), the dimensionless distance of nozzle-to-plate (H/d), the dimensionless displacement from the stagnation point (x/d), the nozzle geometry, the flow confinement, the velocity profile at the nozzle exit, the turbulence intensity of jet flow, and etc.

Many studies had considered the heat transfer of the impinging jet onto a heated and stationary plate [1–6]. To further improve the cooling performance of jet impingement, the heat sink with extended surface is mounted onto the heated plate. Besides the increase of the heat dissipation area, the turbulence due to the separation and reattachment of fluid flow across fins also facilitates heat transfer between fluid and fins. The relevant researches are described below. Sathe et al. [7] presented the three-dimensional numerical simulations of the impinging jet onto the pin-fin heat sink. The heat sink used in their analysis, had a $59 \times 59 \times 2$ mm ceramic base, over which a total of 256 square pin-fins were arranged in a regular in-line pattern. Each pin-fin was 25 mm tall and had a square cross section of 2×2 mm. The cooling air entered the pin-fin heat sink through a nozzle situated 3 mm above the pin-fin tips at the center of the heat sink. Almost 64 pin-fins were directly impinged by the air jet in their tests. They reported that the part of the heated base directly below the nozzle was well cooled with the temperatures gradually increasing from the center towards the corner. Sparrow and his colleagues [8–10] experimentally and theoretically explored the heat transfer from the in-line cylindrical pin-fins with an oncoming longitudinal flow which turned 90 deg to exit the pin-fin array. The area of the fluid inlet was as large as the pin-fin array. The fin geometries (such as the relative fin height and the relative inter-fin spacing), Reynolds number, and the fluid inlet and exit geometries (such as with/without the inlet shroud and the extended base plate) were varied. They found that pin-fins situated adjacent to the edges of the array had higher heat transfer coefficient than those situated in the interior of the array. Additionally, they demonstrated that partial shrouding of the inlet could give rise to nearly uniform per-fin heat

* Corresponding author. Tel.: +886 7 6256040; fax: +886 4 7357193.
E-mail addresses: tm.jeng@msa.hinet.net, t_m_jeng@yahoo.com.tw (T.-M. Jeng).

0017-9310/$ - see front matter © 2008 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2008.10.028

Nomenclature

A	area of the heated surface (m^2)
C	nozzle-to-fin tip distance (m)
d^*	inner diameter of the vertical tube connecting the nozzle (m)
d	inner diameter of the nozzle (m)
D^*	outer diameter of the nozzle (m)
D	diameter of the circular and heated plate (m)
h	heat transfer coefficient (W/m^2 °C)
H	nozzle-to-plate distance (m)
H_f	height of the pin-fins (m)
I	input current (A)
k	conductivity (W/m °C)
L	side length of the square heat sink (m)
Nu	average Nusselt number, $Nu = \frac{hL}{k_f} = \frac{q_c L}{(\overline{T_w} - T_j)k_f}$
q_c	convective heat flux (W/m^2)
q_{Loss}	heat loss (W/m^2)
q_t	total input heat flux (W/m^2)
Re	jet Reynolds number, $Re = \frac{\rho_f U_j d}{\mu}$
Re_r	rotational Reynolds number, $Re_r = \frac{\rho_f \pi \Omega L^2}{120\mu}$
t	thickness of the spreader (m)
T	temperature (°C)
V	input voltage (V)
U_j	average fluid velocity at the nozzle exit (m/s)
w	width of the fins (m)

Greek symbols

μ	viscosity of fluid (kg/m/s)
ρ	density (kg/m^3)
θ	nondimensional temperature, $\theta = \frac{T - T_j}{\overline{T_w} - T_j}$
Ω	rotational rate of the heat sink (rpm)

Superscript

–	average

Subscripts

0	stationary state
a	ambient
f	fluid stream
j	jet nozzle
nc	nature convection
w	heated wall
Ω	rotating state

transfer coefficients throughout the array. Furthermore, modifications of the exit geometry affected only the less tightly packed arrays and then only at the outermost row of pin-fins. Hansen and Webb [11] experimentally investigated the heat transfer from finned heat sink with a normally impinging air jet. Four kinds of fin were employed, including the square pin-fin, the pyramidal fin, the concentric fin and the annular fin. As compared to the bare plate, enhancement of the overall heat transfer in their systems was demonstrated by a factor ranging from 1.5 to 4.5. Various nozzle diameters (d, less than the base diameter of the finned heat sink) and nozzle-to-plate distances (H) were used in their tests. They reported that the heat transfer typically decreased with increasing the nozzle diameters or increasing H/d. In addition to, the relationship between the average Nusselt number and the Reynolds number for $H/d = 5$ was proposed. Ledezma et al. [12] provided an experimental, numerical and theoretical study of the heat transfer from the in-line square pin-fins with an oncoming longitudinal flow. They recommended the correlations for the optimal inter-fin spacing to have the maximum thermal conductance in their systems with various fin heights, fin thicknesses, side sizes of the square base plate, Prandtl numbers and oncoming air velocities. Brignoni and Garimella [13] experimentally studied the heat transfer from the circular pin-fin heat sink with a confined air jet impingement. The heat sink had a 20 × 20 mm base, which was 2.4 mm thick. The fin was 0.9 mm in diameter and 16.4 mm in height. Various nozzle-to-fin tip distances, nozzle diameters and numbers of nozzle were used in their measurements. They found that the nozzle-to-fin tip distance had only modest effect on the heat transfer. Enhancement factors for the heat sink relative to a bare plate were in the range of 2.8–9.7, with the largest value being obtained for the largest single nozzle (12.7 mm diameter). Kondo et al. [14,15] provided a zonal model, based on a series of semi-empirical formula, to determine the thermal resistance and the pressure drop of the finned heat sinks with impingement cooling. Their heat sink configurations were the plate-fin heat sink and the circular pin-fin heat sink. They also performed some typical experiments and flow visualizations to validate the zonal model and used the zonal model to design the optimal finned heat sink. Certainly, due to the leakage flow from the gap between the nozzle and the finned heat sink, their zonal model will not be applicable for a gap more than around 5 mm. Maveety et al. [16–18] experimentally and numerically investigated the heat transfer from the in-line square pin-fin heat sink with a round air jet impingement. The heat sink had a 50.8 × 50.8 mm base. Two heat sinks with 13 × 13 and 7 × 7 pin-fin geometries were used. The diameter of the round nozzle was 6.4 mm. They reported that the optimal performance was achieved when the relative nozzle-to-fin tip distance, C/d, was from 8 to 12 at $Re = 4 \times 10^4$–5×10^4. Additionally, by following the Ledezma et al. [12] method, they showed that the performance of 7 × 7 pin-fin geometry was superior to that of 13 × 13 pin-fin geometry. They also illustrated the utility of numerical tests in the design and optimization of such systems. Besides, they demonstrated that the cooling performance gains could be obtained by inserting a deflector plate above the heat sink. El-sheikh and Garimella [19] experimentally investigated the heat transfer enhancement of air jet impingement by using pin-fin heat sinks. In their study, the heat transfer coefficient, for both pinned and unpinned heat sinks, is only modestly dependent on the nozzle-to-target plate spacing (H/d). They also found that the heat transfer coefficient increases as the nozzle diameter decreases at a fixed flow rate. Li and his colleagues [20,21] used infrared thermography to measure the thermal performance of heat sinks with confined impinging air jet. The ratio of heat sink length to nozzle diameter equaled 10 in their tests. They indicated that the thermal resistance decreases with increasing the fin width. Increasing the fin height also decreases the thermal resistance, but the effects are less prominent than those of the fin width. They also reported that an appropriate impinging distance with minimum thermal resistance can be found at a specific Reynolds number, and the optimal impinging distance increases as the Reynolds number increases. Issa and Ortega [22] experimentally measured the pressure drop and heat transfer of a square jet impinging onto the square pin-fin heat sink. Both the sizes of the nozzle and the heat sink base were 25 × 25 mm. The varied parameters were pin-fin height, pitch and thickness, and the nozzle-to-fin tip distance. They provided a systematically parametric study for such systems. Their conclusions in the aspect of the heat transfer indicated that the overall thermal resistance decreased with increasing pin density