Heat Transfer Enhancement by Air Jet Impingement on Heat Sink

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Abstract— The research is a fundamental investigation of heat transfer to an impinging air jet. Impinging jets are known method of achieving particularly high heat transfer coefficients and are therefore employed in many engineering applications. Impinging jets have attracted much research from the viewpoint of the fluid flow characteristics and their influence on heat transfer. The jet flow characteristics are highly complex and consequently the heat transfer from a surface subject to such a flow is highly variable. Numerous jet configurations have been studied and numerous experimental parameters exists that influence both the fluid flow and the heat transfer. The overall objective of the current research project is to conduct an investigation of the heat transfer mechanisms for an impinging air jet for heat sink with pin fin and dimpled surface.

The rapidly miniaturization of electronic components has resulted in need for thermal management in electronic products to increase their life. So thermoelectric cooling of equipments has been investigated and heat sink designs of electronic equipments using dimpled plate fin and pin fin has been compared with various parameters.

Keywords**:** Jet impingement, Pin-fin, dimple surface

I. INTRODUCTION

Jet impingement is a mature technique, which has been examined by many researchers as a method of heat transfer enhancement in a variety of applications [1, 5]. There have been a number of attempts to complement jet impingement [3, 4] with other enhancing techniques such as cross flow, ribs and tabulators. Attempts have been made to optimize each method in order to obtain effective heat transfer with low pressure loss. In order to augment the heat transfer, the boundary layer has to be thinned or be partially broken and restarted. Protruding ribs have been used widely to enhance the heat transfer [6]. However, the higher pressure loss, high maintenance and weight are the problems the application of ribs has. Dimpled surface has become into the consideration due to its potential in heat transfer augmentation, light weight, low pressure penalty and low.

To further increase the levels of heat transfer enhancement, the surfaces are modified to include pin fins [8] or cavities. In this study, the target surface has dimples machined on the surface and pin fns [9] are attached to it with certain pitch. It is expected that adding dimples to the target surface that has jets impinging on it will further enhance heat transfer. Surface dimples [5] are expected to affect the jet impingement structures and promote turbulent mixing and thus enhance heat transfer.

The effects of Reynolds number [2] and jet-to-plate spacing are investigated by many previous researchers that determine the turbulence intensity of the jets for fully developed turbulent jet.

In order to generate more coherent vortices and detach and restart the boundary layer more often, hence higher heat transfer, there have been numerous developments in surface protrusions, such as rib-roughness or tabulators, pin-banks or pin-fins [10].

An alternative method [11] of vortex generation is the use of concavity, which is defined as an indentation on the surface forming a recess rather than the protrusion. A dimple is one kind of concavity, and has a drag reduction characteristic in external flow over bodies such as is demonstrated by golf balls.

II. EXPERIMENTAL WORK

A schematic of the experimental apparatus is shown in Fig. 1. The air was supplied by a fan, and heated up by a 100W heater. The complete experiment has been performed on the setup as shown in the figure. The centrifugal type blower is used to supply ambient air at various velocities. The air from blower reaches the nozzle assembly [7] via the pipes. Air passes through an Orifice meter before reaching the nozzle assembly. U-tube water manometer has been connected across the orifice to measure the pressure drop in terms of mm of water column difference and thus maintain the desired velocities. The flow control knob helps varying the velocity. This way Reynolds number is varied from 6000 to 12000 by delivering air at corresponding velocities. The nozzle assembly consists of a 4 x 4 nozzle [3, 4] array providing multiple jets over the heat sink geometry.

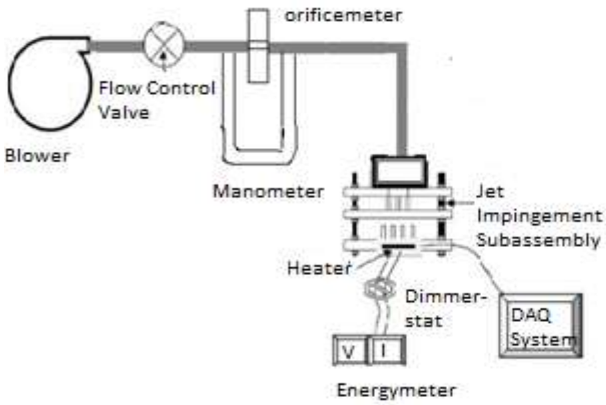

Fig.1 Experimental Setup

Heater is connected to the supply via a dimmer stat which helps maintain the heat input at 20W. The entire path from blower to nozzle assembly was carefully sealed to ensure no leakages occurred. T-type thermocouples have been used to detect temperatures at various points on the heat sink. These thermocouples are connected to the Digital temperature indicator having several channels to display temperatures at different points.

The heat sink position is varied with the help of suitable provisions to maintain Z/D ratio of 5. The heater is turned on and dimmer stat value is kept constant at 20W. It takes 30-45 minutes to attain steady state. The first iteration is performed at Reynolds number 6000. Air velocity required to maintain Re 6000 is calculated along with the corresponding water column difference in the U-tube manometer. The blower is turned on and flow is varied using flow control knob to match the water column difference in the manometer. Readings of temperatures at different points are noted at intervals of 5 min till 30 minutes after which it is observed to achieve steady states. The procedure is repeated for Re 6000, 8000, 10000 and 12000. The Z/D ratio [5] is changed to 5, 5.5 and 6 and readings for Re 6000, 8000, and 12000 are noted. This procedure is repeated for three heat sinks of pin fin and dimples designed.

Radiation heat transfer from bottom and side of heat sink with ambient is calculated based on asbestos temperature at the bottom and average temperature of heat sink.

2.1 Data Reduction

Reynolds number was calculated from the formula as

$R_e = (V_a * D_h)/\gamma_a$

Mass flow rate of air coming out from nozzle plate can be calculated from the formula as

$m_a = \rho_a * V_a * A_{pipe}$

& $m_a = C_d * (\pi/4) * d_0 * \rho_a * ((2 * g * h_a)/ (l-\beta^4))^{1/2}$

Height in the water column which is to be maintained for above Reynolds number can be calculated by

$h_w = h_a/\rho_w$

Bulk mean temperature can be calculated as

$T_m = Tav/4$

Heat consumed

$Q_s = h * A_s * (T_m - T_a)$, W

$Q_s = Q - (Q_r + Q_c)$, W

Radiation heat loss = $Q_r = A_1 * 5.67 * 0.6 * (((T_m+273)^4)-((T_a+273)^4)) * 10^{-8}$, W

Conduction heat loss = $Q_c = (K * A_2 * (T_5 - T_6)) / 0.005$, W

Heat transfer coefficient

$h = Q_s/[A_s * (T_m - T_a)$, W] [1]

Nusselt Number

$N_u = h * D_h/K_a$ [12]

2.2. Experimental Parameters

The following table gives one sample for readings of fin fin-fin and dimple surface parameters measured during experimentation out of six test pieces.

Table 1 Experimental procedure parameters

Time\Temp.	T1	T2	T3	T4	T5	T6
0	42	38	38	34	37	35
5	41	39	38	37	37	35
10	41	38	36	35	37	35
15	41	38	36	35	37	35

III. RESULT AND DISCUSSIONS

The following graph gives the detailed analysis of dimpled surface heat sink with various parameters like Nusselt number, Reynolds number and various jets to dimple plate spacing Z/D ratios from 5, 5.5 and 5.
Heat sink one has maximum enhancement at 8000 Re at Z/D =5.

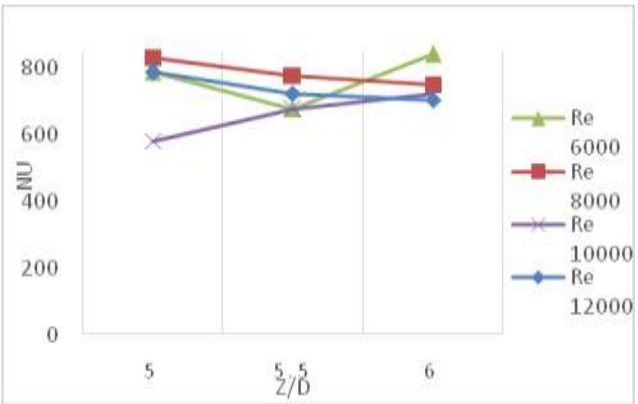

Fig 3.1 Heat Sink 1 (Dimple Diameter = 8mm (staggered)

Heat sink two with dimple 6mm and staggered arrangement of dimples shows maximum augmentation at Re =6000 and Z/D =5.5.

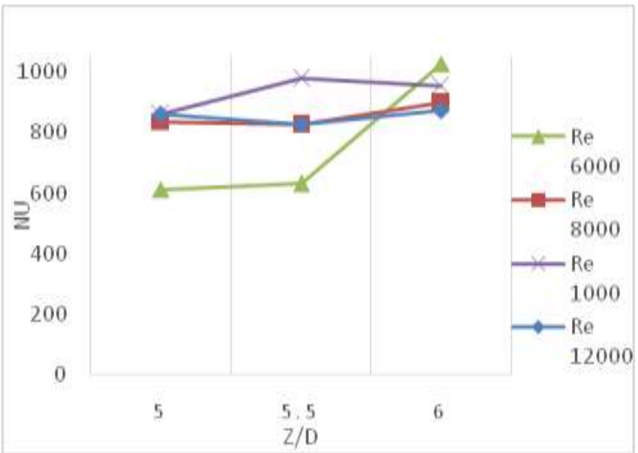

Fig 3.2 Heat Sink 2 Dimple Diameter =6 mm(staggered)

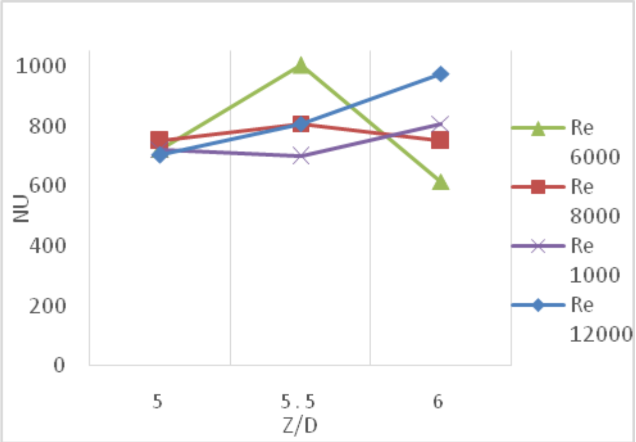

Fig 3.3 Heat Sink 3 Dimple Diameter =8 mm (inline)

The sample experimental results presented in the graphical form above shows that the dimple with 6mm spherical cavity shows maximum Nusselt number with staggered spacing. Also it shows that at Z/D = 5.5 and Reynolds number 6000 gives maximum heat transfer enhancement.

If we repeat the same experiment with same parameters with pin fin array of same sample piece with same Re = 6000, 8000, 10000 and 12000 with Z/D = 5, 5.5 and 6.

It has been found that heat transfer augmentation for maximum heat transfer is occurred with pin-fin array of 6mm diameter with 25 mm fin height.

The following graphical results show the comparisons between various pin-fin array arrangements similar to dimple surface.

Comparison Pin-fin Heat sinks & Z/D ratios

For Z/D = 5
For Z/D = 5.5
For Z/D = 6

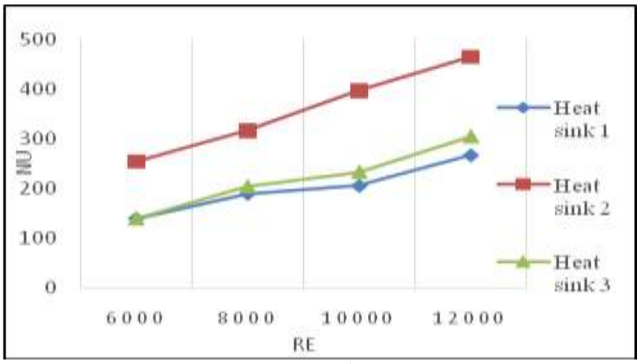

Fig 3.4 N_u vs. R_e for Z/D = 5

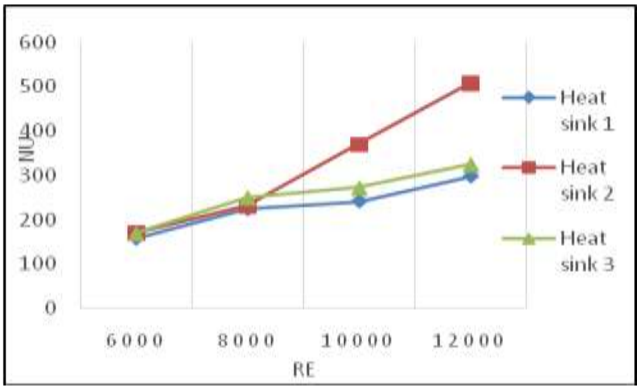

Fig. 3.5 N_u vs. R_e for Z/D = 5.5

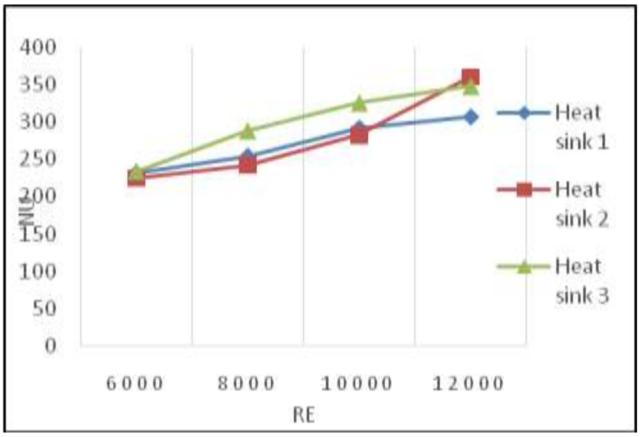

Fig. 3.6 N_u vs. R_e for Z/D = 6

It can be observed from graph that pin fin array of 6mm diameter heat sink with staggered array shows better results for heat transfer augmentation for all Z/D ratios.

IV. CONCLUSIONS

This experiment was performed for investigating effects of different cavities on heat transfer with jet impingement on dimples as well as pin-fin array. Main conclusions are summarized as

i) Use of dimples and pin-fin array increase heat transfer compared to flat plate
ii) Heat transfer enhancement increases with increase in Reynolds number.
iii) Nusselt number is maximum for dimpled surface compared with pin-fin array.
iv) It was found to be Nu higher for heat sink with dimple of 6mm staggered spacing with Z/D =5.5.
v) Distance between nozzle plates and test plate affects heat transfer augmentation.
vi) For Pin-fins Z/D =5.5 showed maximum heat transfer for all heat sinks and for dimples surface Z/D =5.5 for 6mm dimple surface.
vii) If Z/D ratio is reduced then more turbulence will occur and less heat transfer. So it shows maximum heat transfer will occur at optimum Z/D ratio.

NOMENCLATURE

A_{plate}	Area of Test plate (Ai plate), m^2
D	Nozzle diameter, m
Z	Nozzle to target surface spacing, m
C_d	Discharge Coefficient of air
C_p	Specific heat of air at constant pressure, J/KgK
d_o	Diameter of Orifice, m
g	Acceleration due to gravity, m/s^2
h	Convective heat transfer Coefficient of air, W/m^2K
Ka	Thermal Conductivity of air, W/m K
h_{air}	Height of air column, m
h_w	Height of water column, m
M_a	Mass flow rate of air, kg/s
Q_c	Convective heat transfer to air, W
T_m	Mean Temperature, °C
T_{12}	Ambient temperature

DIMENSIONLESS PARAMETER

N_u	Nusselt Number
R_e	Reynolds Number
P_r	Prandlt Number

SYMBOLS

ρ_a	Air Density, kg/m^3
ρ_w	Water Density, kg/m^3
μ	Coefficient of Viscosity, $N\text{-}s/m^2$

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