Experimental Investigation of Heat Transfer by PIN FIN

U S Gawai, Mathew V K, Murtuza S D
Asst.Prof (Mechanical Dept), JSPM’s ICOER, Pune

Abstract: - Heat transfer enhancement over surface results from the depression forming recesses rather than projections. Generically, such features are known as dimples, and may be formed in an infinite variation of geometries which results in various heat transfer and friction characteristics. Heat Transfer enhancement using dimples based on the principle of scrubbing action of cooling fluid taking place inside the dimple and phenomenon of intensifying the delay of flow separation over the surface. Spherical indentations or dimples have shown good heat transfer characteristics when used as surface roughness. The technology using dimples recently attracted interest due to the substantial heat transfer augmentations it induces, with pressure drop penalties smaller than with other types of heat augmentation. The proposed work is concerned with experimental set up for enhancement of the forced convection heat transfer over the dimpled surface and flow structure analysis within a dimple. The objective of the present work is to find out the heat transfer rate and air flow distribution on dimpled surfaces and all the results obtained will be compared with those from a flat surface.

Keywords: Dimple, Forced Convection, Heat transfer Enhancement, Pin fin.

I. INTRODUCTION
Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib tabulators, protrusions, pin fins, and dimples. These heat transfer enhancement techniques have practical. Application for internal cooling of turbine airfoils, combustion chamber liners and electronics cooling devices, biomedical devices and heat exchangers. The heat transfer can be increased by the following different Augmentation Techniques. They are broadly classified into three different categories:
(i) Passive Techniques
(ii) Active Techniques
(iii) Compound Techniques.

A. Passive Techniques
These techniques generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior (except for extended surfaces) which also leads to increase in the pressure drop. In case of extended surfaces, effective heat transfer area on the side of the extended surface is increased. Passive techniques hold the advantage over the active techniques as they do not require any direct input of external power. These techniques do not require any direct input of external power; rather they use it from the system itself which ultimately leads to an increase in fluid pressure drop. They generally use surface or geometrical modifications to the flow channel by incorporating inserts or additional devices. They promote higher heat transfer coefficients by disturbing or altering the existing flow behavior except for extended surfaces.

B. Active Techniques
These techniques are more complex from the use and design point of view as the method requires some external power input to cause the desired flow modification and improvement in the rate of heat transfer. It finds limited application because of the need of external power in many practical applications. In comparison to the passive techniques, these techniques have not shown much potential as it is difficult to provide external power input in many cases. In these cases, external power is used to facilitate the desired flow modification and the concomitant improvement in the rate of heat transfer.

C. Compound Techniques
A compound augmentation technique is the one where more than one of the above mentioned techniques is used in combination with the purpose of further improving the thermo-hydraulic performance of a heat exchanger. When any two or more of these techniques are employed simultaneously to obtain enhancement in heat transfer that is greater than that produced by either of them when used individually, is termed as compound enhancement. This technique involves complex design and hence has limited applications.

II. PRACTICES
Pin- Fins are used to increase the heat transfer rate from surface to the surrounding fluid when ‘h’ value is generally smaller on the surface. Familiar examples are the circumferential fins around the cylinder of motor cycle engine & pin fins attached to the condenser tubes at the back of domestic refrigerator. In present pin-fins are normally used in different shapes & sizes depending upon its applications as shown in following figures.

It is obvious that a fin surface sticks out from the primary heat transfer surface. The temperature difference with surrounding fluid will steadily diminish as one moves out along the fin. The design of the fins therefore required knowledge of the temperature distribution in the fin. The main objective of this experimental set up is to study temperature distribution in a simple pin fin.
III. EXPERIMENTAL SETUP

Fig. 3 shows a schematic diagram of the experimental system for the pressure loss and heat transfer measurements for the channels with pin fin-dimple. This experimental setup consists of a variable-speed blower, a settling chamber, a nozzle flow meter, a differential pressure transducer, a Lab View data acquisition system and a test section. The air is drawn into the wind tunnel by the blower, and the air mass flow rate is measured by the nozzle flow meter. The test fin has a length of 132 mm, which is made of 10 mm-thick. The clear Plexiglas channel wall can provide a good thermal insulation and a good optical access to the flow in the channel.

The boundary layer in the flow entering the test channel is tripped by the rough junction interface between the test channel and the contraction section and by the roughened surface at the entrance of the test channel. This experimental configuration has low inlet turbulence intensity and approximately a uniform inlet velocity field, which is typical of most pin fin channel heat transfer studies.

IV. PROPOSED WORK

The experiment is conducted to investigate the effect of dimple on the pressure loss and heat transfer characteristics in pin-fin dimple channel, where dimples are located on the pin-fins. A aluminum fin of rectangular cross section with various dimple depth is fitted in a long rectangular duct. The other end of the duct is connected to the delivery side of the blower and the air flows past the fin perpendicular to its axis. One end of the fin project outside the duct & is heated by a heater. Temperatures at five points along the length of the fin are measured by Chromel alumen thermocouples connected along the length of fin. The air flow rate is measured by orifice meter fitted on the delivery side of blower.

The experiment conduct for the pin-fin dimple with various dimple depths have been obtained and compared with each other for Reynolds number also analyses the further improvement in convective heat transfer performance.

V. CALCULATIONS

A] For Brass

1) Reynolds Number (Re) :-

$$Re = \frac{(v_{in}d_{in})}{\nu}$$
Re=139.95
2) Nusselt Number (Nu):
   \[ \text{Nu}=0.615 \times \text{Re}^{0.466} \]
   \[ \text{Nu}=6.1503 \]
3) Coefficient of Heat Transfer (h):
   \[ \text{h}=\frac{\text{Nu} \times k_{\text{air}}}{d_{\text{fin}}} \]
   \[ \text{h}=14.7706 \text{ w/m}^2\text{k} \]
4) Fin Parameter (m):
   \[ m=\left(\frac{h\times l}{(k \times A)}\right)^{1/2} \]
   \[ m=6.4763 \]
5) Fin Efficiency:
   \[ \eta=\frac{\tanh (ml)}{ml} \]
   \[ \eta=65\% \]
   
B) For Aluminum
1) Reynolds Number (Re):
   \[ \text{Re}=\left(\frac{\text{v} \times m_{\text{f}} \times L_{c}}{\nu}\right) \]
   \[ \text{Re}=2315.3413 \]
2) Nusselt Number (Nu):
   \[ \text{Nu}=0.615 \times \text{Re}^{0.466} \]
   \[ \text{Nu}=22.73 \]
3) Coefficient of Heat Transfer (h):
   \[ \text{h}=\frac{\text{Nu} \times k_{\text{air}}}{L_{c}} \]
   \[ \text{h}=119.3836 \text{ w/m}^2\text{k} \]
4) Fin Parameter (m):
   \[ m=\left(\frac{h\times l}{(k \times A)}\right)^{1/2} \]
   \[ m=5.733 \]
5) Fin Efficiency:
   \[ \eta=\frac{\tanh (ml)}{ml} \]
   \[ \eta=96\% \]

Comparison between Aluminum & Brass:

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<th>Brass</th>
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<tr>
<td>5.</td>
<td>Efficiency</td>
<td>96%</td>
<td>65.37%</td>
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VI. THERMAL CHARACTERISTIC CURVES

![Fig. 6. Reynolds Number Vs Friction Factor](image)

![Fig. 7 Reynolds number Vs Nusselt number](image)

VII. CONCLUSION

In the present work aluminum & brass plate were used as a test surface. Variation of Nusselt Number with Reynolds Number is investigated, with various parameter combinations. The experimental results gives heat transfer coefficient & efficiency of aluminum fin is greater than brass fin.

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REFERENCES