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Experimental And Numerical Study On Forced Convection For Plained And Dimpled Surface In A Long Triangular Fin Bar

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Abstract: - basically, we know that fin is extended surface use to improve the heat transfer rate. It is use as a secondary surface to remove heat from the primary surface. In this paper, we detailed study how can we improve or augment the heat transfer rate of fin so we are comparing two different fin surfaces but having same dimensions (L, B, H etc).

Introduction: - The removal of excessive heat from system components and I C engines is essential to avoid the damaging effects of burning or overheating. Therefore, enhancement of heat transfer is an important subject of thermal engineering. The heat transfer from surface may in general be enhanced by increasing the heat transfer coefficient between a surface and its surrounding or by increasing heat transfer area of the surface or by both.

Extended surfaces that are well known as a fin are commonly used to enhance heat transfer in many applications. Therefore, various types of fins like rectangular plate fins, square pin-fins and circular pin-fins are commonly used for both natural and forced convection heat transfers.

The conversion, utilization, and recovery of energy in every industrial, commercial, and domestic application involve a heat transfer process. Some common examples are coming from domestic application to industrial ones. Improved heat exchange, over and above that in the usual or standard practice, can significantly improve the thermal efficiency in such applications as well as the economics of their design and operation.

Key words: - forced convection, fins (extended surfaces), heat transfer etc.

Nomenclature

CD = Coefficient of discharge for orifice
TFO = Fluid temperature at the exit of the duct (°C)
β = d/D, diameter of pipe/ diameter of orifice
TFI=Fluid temperature at the inlet of the duct (°C)
Cp = Specific heat of air
Tpm = the average temperature of the test surface
T_{Fm} = the average temperature of air in the duct = TFO + TFI /2
As = projected surface area of test surface
ao = Cross sectional area of orifice
g = Acceleration due to gravity
Ha = Height of air column
m = Mass flow rate of air
Q = Convective heat transfer to air
h = heat transfer coefficient

Experimental procedure: - Bergles [1, 2] classified the mechanisms of enhancing heat transfer as active or passive methods. Those which require external power to maintain the enhancement mechanism are named active methods. On the other hand, the passive enhancement methods are those which do not require external power to sustain the enhancement characteristics.

Examples of passive enhancing methods are:

(a) Treated surfaces, (b) rough surfaces, (c) extended surfaces, (d) displaced enhancement devices, (e) swirl flow devices, (f) coiled tubes, (g) surface tension devices, (h) additives for fluids, and many others.

Heat transfer inside flow passages can be enhanced by using passive surface modifications such as rib turbulators, 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. Recently, dimples have drawn more attention because of the significant enhancement in heat transfer with a lower penalty in the increased pressure drop.

These studies employ flows over flat walls with regular arrays of spherical pits [3], flows in annular passages with a staggered array of concave dimples on the interior cylindrical surface [3], flows in single hemispherical cavities [4, 5], flows in diffuser and convergent channels each with a single hemispherical cavity [6], and flows in a narrow channel with spherically shaped dimples placed in relative positions on two opposite surfaces [7]. Heat transfer augmentations as high as 150 percent, compared to smooth surfaces are reported sometimes with appreciable pressure losses [3]. Other recent data show that the enhancement of the overall heat transfer rate is about 2.5 times smooth surface values over a range of Reynolds numbers and pressure losses are about half the values produced by conventional rib turbulators.

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Fig1:-experiment set up

Fig2:- plain fin surface (triangular long fin bar)

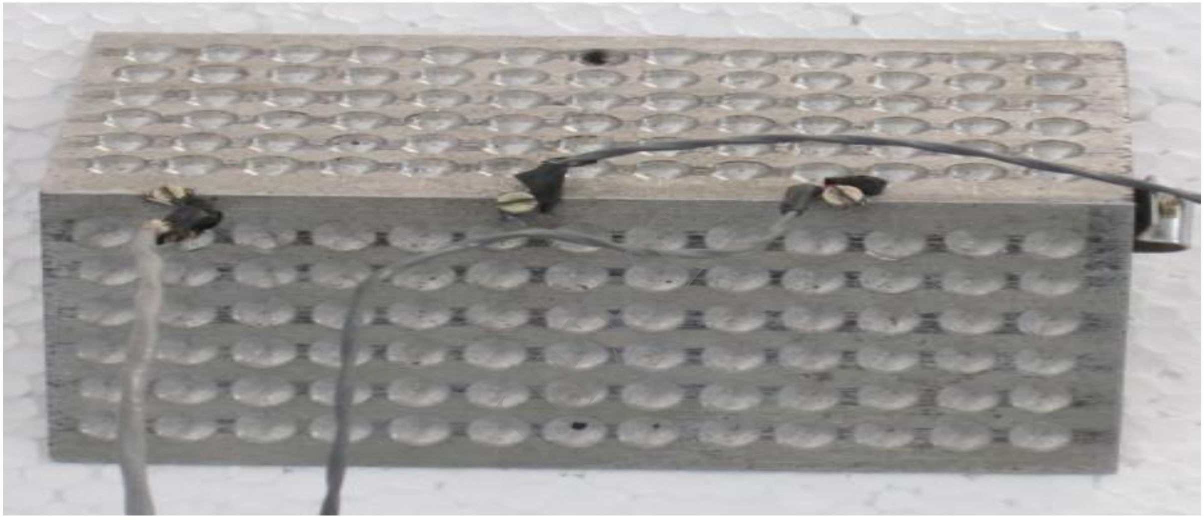

Fig3:- dimpled fin surface (triangular long fin bar)

<u>Procedure</u>:-

(1)Switch ON the mains system (2). Switch ON blower. (3) Adjust the flow by means of gate valve to some desired difference in the Manometer level (4) Switch ON heater
(5) Start the heating of the test section with the help of dimmerstat and adjust desired heat input with the help of Voltmeter and Ammeter. (6) Take readings of all the six thermocouples at an interval of 10 min until the steady state is reached. (7) Note down the heater input

<u>Data reduction</u>: - these are basic formulas have been used in experimental investigation. Those are as follows.
The mass flow rate of air is determined from the pressure drop across the orifice meter, using a following relation

$$m' = C_D \; a_o \times \frac{\rho_{air} \times \sqrt{2 g H_a}}{\sqrt{1-\beta^4}}$$

Where,
CD = Coefficient of discharge for orifice
ao = Cross sectional area of orifice
β = d/D, dimeter of pipe/ diameter of orifice
g = Acceleration due to gravity
Ha = Height of air column

The useful heat gain of the air is calculated as:

$$Q = \dot{m} C_p (T_{FO} - T_{FI})$$

Where,

TFO = Fluid temperature at the exit of the duct (°C)

TFI= Fluid temperature at the inlet of the duct (°C)
m = Mass flow rate of air
Cp = Specific heat of air
Q = Convective heat transfer to air

The heat transfer coefficient for the test section is:

$$h = \frac{Q}{A_S} X (T_{pm} - T_{Fm})$$

Where,
Tpm is the average temperature of the test surface
T_{Fm} is the average temperature of air in the duct = TFO + TFI /2
As is projected surface area of test surface
h is convective heat transfer coefficient
The Nusselt number as,

$$Nu = \frac{h\, x\, Dh}{K_{air}}$$

Where,
Nu is the average Nusselt number of the test surface
Dh is the hydraulic diameter of the rectangular duct
K air is the thermal conductivity of air

The friction factor was determined from measured vales of pressure drop across the test section using:

$$f = \Delta P\, D_h / 2\, \rho_{air}\, L\, V^2_{air}$$

Where,
f is friction factor of the test surface
ΔP is pressure drop across the test surface.
ρ air is the density of air
L is the length of the test surface
V air is the velocity of the air through rectangular duct.

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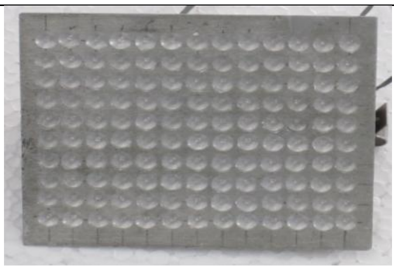
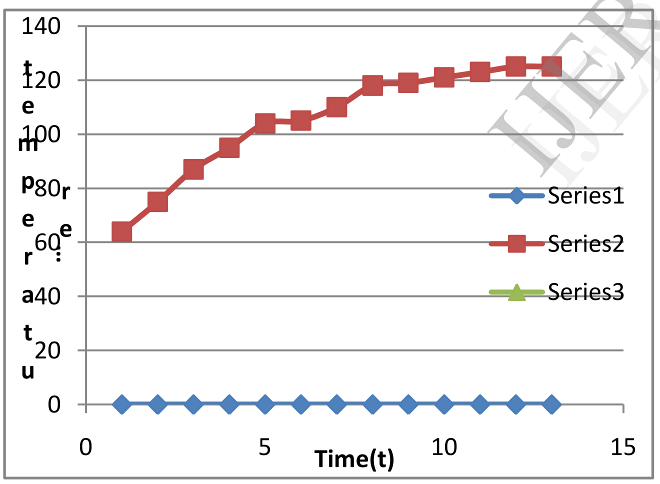
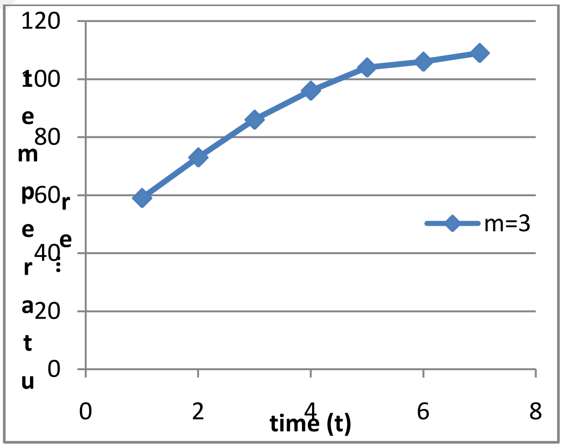
Result and discussion:-

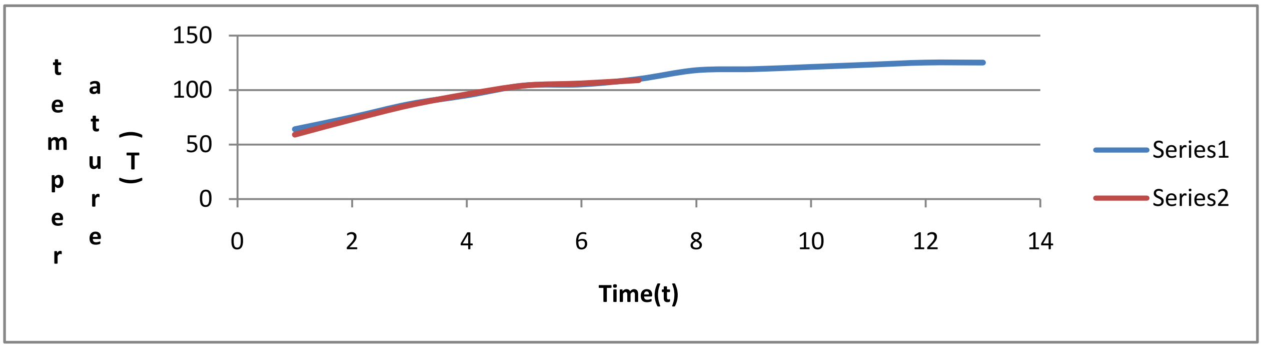
Fin surface	Heat transfer coefficient	Heat transfer rate
 Plain fin surface	22.4	90.74
 Dimpled fin surface	26.32	107.2396





Graph1:- for plain fin surface

Graph2:- for dimpled fin surface



Graph3:- combined graph for plain and dimpled fin surface

Conclusion:-

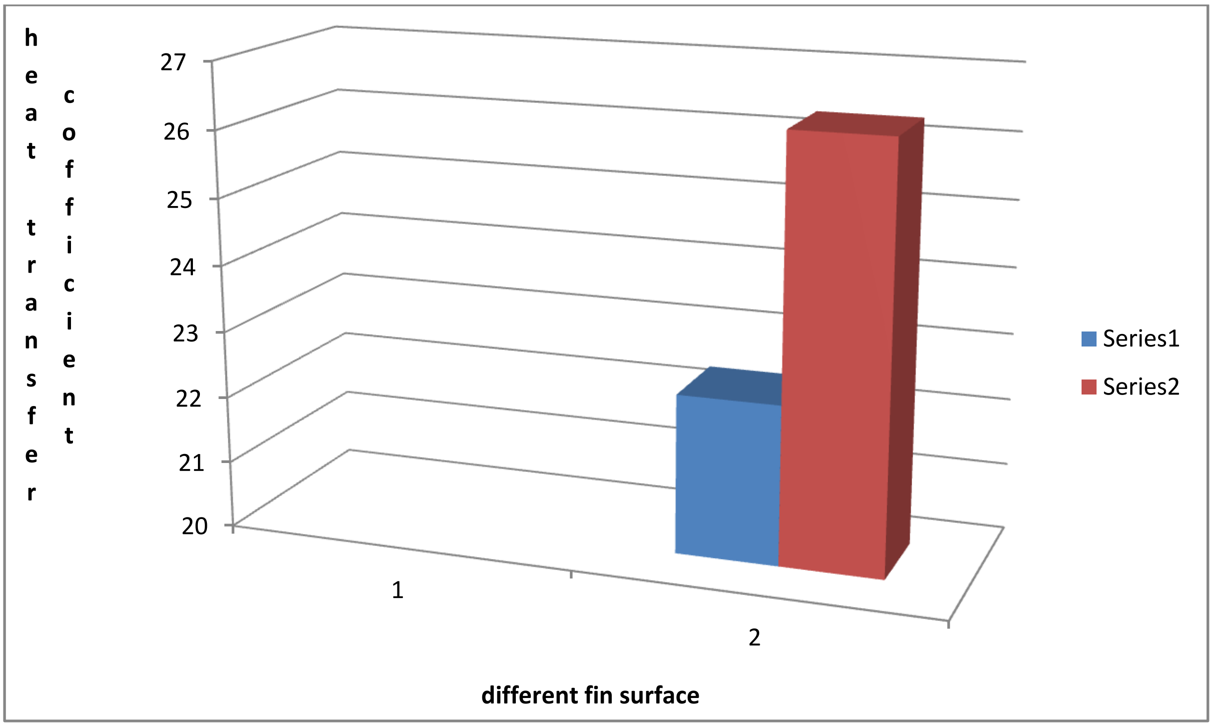


Fig4:-graph between different fin surface and heat transfer coefficient

Through all the experiment procedure we come to know that dimpled fin have more heat transfer rate and having more heat transfer coefficient than plain fin. So it is better to use dimpled surface than plain surface for more heat transfer rate and heat transfer coefficient. We come to know that a simple arrangement can increase more heat transfer rate than plain fin surface at low cost. this type of passive method is easily applicable and more effective.

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