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Research Article

Experimental and Numerical Investigation of Forced Convection Heat Transfer in Heat Sink with Rectangular Plates on Vertical Base

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Abstract

It is essential to provide enough cooling in electronic components to avoid overheating and to increase the performance of the system. Improvement of the overall heat transfer performance, augmentations via fin patterns and various orientations are often employed. Among the fin configurations being implemented practically, rectangular fin is regarded as one of the most effective heat transfer augmentation methods for periodic vortex shedding and providing boundary layer restarting. The main objective of this experimental study is to evaluate and compare the rate of forced convection heat transfer enhancement of rectangular fin with fins at different angles (0–90°)with respect to different range of Re and heat fluxes. The variables for this study of forced convection are orientations of fins with various angles, heat fluxes and Re. The heat transfer coefficient increases with the rise of air velocity for all the configurations. The increase in the heat transfer coefficient was achieved at the angle of orientation 30° for inline and 0° for staggered arrangement. 0° fin staggered arrangement shows about 17% enhanced heat transfer coefficient as compared to 30° inline fin arrangement. For rectangular plate fin heat sinks it has been observed that there is boundary layer growth in forced convection, which decreases the heat transfer rate. To overcome this problem, some rectangular fins are attached to the plate fin. By doing so an average heat transfer coefficient of the rectangular plate fin heat sink was increased under the condition of varying fin angle and its arrangement, and the profit factor of the former is about 18% higher as compared to the other arrangements. Up till now all the researchers have analyzed the PFHS and rectangular PFHS for horizontal orientation but this project aims to find the heat transfer enhancement of PFHS for its vertical orientation fig.(1)(2) with rectangular plates at different range of heat fluxes, different fin angles, fin arrangements and Reynolds's numbers (Re).

Keywords: *Forced convection, fin angles, Re, Plate fin heat sink (PFHS), Rectangular Plate fin heat sink (RPFHS)*

1. Introduction

The enhancement of heat transfer is vital and active field of engineering research. Improvement in the effectiveness of heat sinks through suitable heat transfer augmentation technique result in considerable advantages and savings of costs. By using rectangular plate fins attached to the plate fin heat sinks on vertical base, considerable enhancements are demonstrated in the present work. Planted fins acts as a turbulence generator by disturbing the laminar boundary layer growth in the plate fin passage. Turbulence allows the mixing of the fluid, due to which rate of heat transfer increases.

Avram Bar-Cohen and Warren M. Rohsenow conducted experiment about the heat transfer performance of rectangular fin arrays. In their experiments, the preceding development of analytic relations for the optimum spacing between printed circuit boards or cards, modeled as thick isothermal or iso-flux surfaces, reveals significant departures from values associated with negligibly thick elements. Massimiliano Rizzi, Marco Canino, Kunzhong Hu proposed experimental investigation of fin heat sink effectiveness; this work describes heat sink effectiveness. The heat sink made up of Al and consists of a staggered fin array. The three heat sinks had constant fin height and fin diameter, but variable was the pitch.

The cooling was provided by air and the Reynolds number ranging from 400 to 17000. The shroud of channel touching the fin tips, to eliminate the flow bypass.N. Sahiti, F. Durst and A. Dewan worked to improve the performance of heat exchangers by implementing pin fins. Kai-ShingYang, Wei-Hsin Chu, Ing-Yong Chen, Chi-Chuan Wang performed a study of fin with circular, elliptic, and square cross-section and heat sink. Twelve fins with heat sink tested with inline and staggered arrangements. The fin density effect on the performance of heat transfer is examined. An appreciable influence showed by the circular pin fin

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density for inline arrangement whereas no such effect of fin density is observed for square fin cross section. An unique deflection of the flow pattern is observed with the inline circular fin. For all the three configurations with the staggered arrangement, the heat transfer coefficient increases with the rise of fin density.

2. Experimental setup and procedure

2.1 Experimental setup

The experimental setup showed in fig [3] mainly consists of flow control valve, u-tube manometer, 2 h.p. blower, honey comb mesh for streamed flow over fin pattern, duct, fin pattern, heater assembly and control panel. The fins are made from Al having conductivity of 202 W/mk. The rectangular fins are cut from 4mm Al fin and the fin base is cut from 6m Al fin. The rectangular fins are then mounted on the rectangular plate passage with the help of screw and the contact surface is covered with heat compound solution to reduce contact resistance. Holes drilled in the base portion of the fin array where three cartridge heaters were placed, connected to the mains via a dimmer stat. A calibrated wattmeter measures the heater input. A bakelite (K=0.233 W/mk) sheet which acts as an insulator were placed to cover the back of the heater which facilitate the heat to travel only in the fin array direction. Measurement of the temperature at various locations on the fin array and the surroundings are taken by thermocouple (copper-constantan).

Fig.1 Vertical Plate Fin Heat Sinks (PFHS)

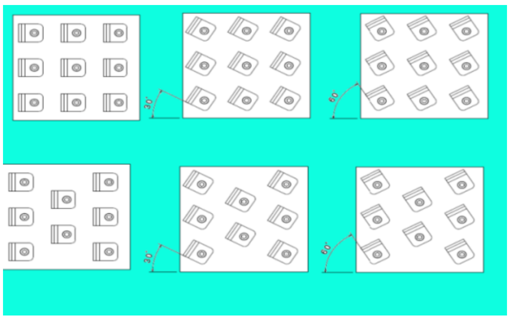

Fig.2 Vertical Plate Fin Heat Sinks (PFHS) with fin arrangements

2.2 Experimental procedure

Rectangular plate with fin array was tested for different Reynolds numbers for turbulent flow through non circular duct ranging from 4000 to 18,000 at different heater input starting from 50W, 80W, 100W and 125W and fin angle varying from 0°-60° in step of 30°. To find out amount of heat loss through fin array due to convection the heat transfer coefficient is calculated.

Nomenclature

Q : Heat input, W
L_c : Characteristic length, m
ρ : air density, kg/m^3
V : air velocity, m/s
μ : Dynamic viscosity, Pa-s
H : Fin height, m
L : Fin Length, m
T : Temperature, °C
H : coefficient of heat transfer, W/m^2.K
D_h : Hydraulic diameter of rectangular duct
h_{av} : Average coefficient of heat transfer, W/m^2.K
Re : Reynolds number = $\rho \cdot V \cdot D_h/\mu$
Nu : Nusselt number = $h_{av} \cdot lc/K$
K : Thermal Conductivity of air, W/m.K
Γ : Diffusion coefficient of air
α : fin angle
m : mass flow rate, kg/s
C_D :Coefficient of discharge for orifice,
a_0 : Cross section area of orifice (m^2),
d/D, diameter of pipe/ diameter of orifice,
g : Acceleration due to gravity (m/s^2),
H_a : Height of air column (m).
T_{in} : Air temp at the inlet of the duct (^{o}C),
T_{out} : Air temp at the exit of the duct (^{o}C),
C_p :Specific heat of air (KJ/kg K),
T_{bm} : Bulk mean temperature of air in the duct = (Tai + Tae / 2) (^{0}C).

Abbreviations

PFHS : Plate Fin Heat sinks
RPFHS : Rectangular Plate Fin Heat Sinks

Table 1 - Instrument Specification

Sr. No.	Name of Parts	Specification	Quantity
1	Blower	1 hp, 6.35 amp, 0.75 KW, 230 V, 1 Phase, 2800 rpm.	1
2	Thermocouple	T-Type	11
3	Honey Comb Mesh	200mm×100mm×100mm	1
4	Flow Control Valve	Butterfly Type	1
5	U-Tube Manometer	Water as a manometric fluid	1
6	Bakelite Sheet	As Insulation(K=0.232)	1
7	Sandwich Heater	Std. (238mm×168mm)	2 set
8	Heater input Controller and Data Logger	Sharp Tech	1
9	Duct	100mm×100mm×880mm	1

Fig 3. Experimental setup

3. Data reduction

Mass flow rate of air across the orifice meter determined,

Using following relation

$$\dot{m} = C_D a_0 \times \frac{\rho_{air} \times \sqrt{2gH_a}}{\sqrt{1-\beta^4}} \quad (kg/s)\ldots\ldots\ldots\ldots \tag{1}$$

The heat gain by air is calculated from

$$Q = mC_p(T_{in} - T_{out}) \text{ (Watt)}\ldots\ldots\ldots\ldots\ldots\ldots \tag{2}$$

The heat transfer coefficient for test section

$$h = \frac{Q}{A_s \cdot (T_s - T_{bm})} \tag{3}$$

Nusselt number as,

$$Nu = \frac{hD_h}{K_{air}} \tag{4}$$

Input Values
1. Heat Flux (Adjusted from the Heater input controller by rotating the Knob to desired value of heat input)
Output measured Values.
2. Temperature at thermocouple Location (T_9 toT_{11})

Experimental Calculation to be followed

Rise in temperature of air = $\Delta T = (T_{in} - T_{out})$
Find the Properties of Air at T = $(T_9 + T_{10} + T_{11})/3$
$\dot{m}$ = Volume of Air × ρ (Density of air)
Volume of Air = $C_d \times A_{Orifice} \times \sqrt{2gH_{air}}$

$H_{air} = \left(\frac{\rho_{water} \times H_{water}}{\rho_{air}}\right)$

Q_{fin}=h× surface Area ×ΔT

$\Delta T = (T_s - T_f)$

$T_S = \left(\frac{T_1+T_2+T_3+T_4+T_5}{5}\right)$

$T_f = (T_9 + T_{10} + T_{11})/3$
Calculate h

4. Model Design

A three-dimensional rectangular duct of size 880.0×100.0×200.0 mm^3 is considered [6]. All the simulation is done in **CFD** (**C**omputational **F**luid **D**ynamics) FLUENT.

Table 2: Parameters of Numerical Study

Parameter	Dimensions
Area of base plate (mm^2)	148 * 148
Height of Fin (mm)	80
No of Plate Fin	9

The k–ε turbulence model is implemented. A three-dimensional, turbulent, incompressible, and steady flow with constant thermodynamic properties are assumed. Effects of Buoyancy and radiation heat transfer are negligible.

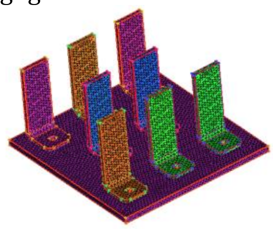

Fig.4a Meshed cut-plane view for air over fin array

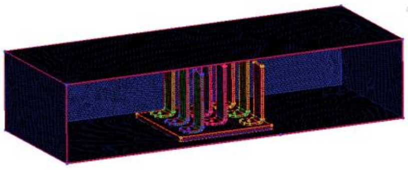

Fig.4b domain volume mesh

5. Results and observation

The six different set of Plate fin heat sink (PFHS) and rectangular plate fin heat sink (RPFHS) with different fin angle in this study discussed. The effects of the wind velocity and the types of RPFHS on the thermal performance are investigated. The parameters used in this study include V_{in} (V_{in} = 2.97, 3.45, 4.1, 4.4, 6.02 and 7.2 m/s), the fin angle varies from 0° to 60° in both inline and staggered arrangement The fin base plate area is heated with heating power ranging from 50W to 125W. The analysis is carried out by considering three different variables that are, Reyndols number, heat flux i.e applied to the base of fin array and different fin angle arrangements. All the six sets are analysed at different values of Reynolds number (ranging from 4000 to 18000) and different heat flux (50 W to 125 W) and the heat transfer coefficient is calculated. As it is seen from figures(Fig. 4a to 9b), by planting rectangular fins over plate fin heat sink the heat transfer rate increases as compared to the plate fin heat sink.

5.1 Thermal performance evaluation

Using the data obtained from the experiments, heat transfer and thermal performance characteristics of plate fin heat sinks are discussed below.

It is observed that as

i. Re increases Nu increases for all the heat inputs
ii. Heat trasnfer coefficient for various fin angle is different i.e for inline arrangement Nu increases first and then decreases and for staggerred arrangement Nu decreases as fin angle increases for all the heat inputs.

For the study convinience, following results are presented for Re = 6026 and Q = 50W for both inline and staggered arrangement. Though the relation between parameters remain almost same.

Pumping power P and profit factor J is calculated resp. by using the relation (13) and (14) resp.

$$P = (m/\rho)*\Delta P \qquad (13)$$

$$J = Q/P \qquad (14)$$

For the study convinience, following results are presented for Re = 6026 and Q = 50W for both inline and staggered arrangement. Though the relation between parameters remain almost same.

5.1.1 Fin angle (α) vs Nu

For inline arrangement, convective heat transfer coefficient increases with fin angle up to 30° and after 30°, it starts to decrease. The maximum heat transfer occurs at 30° for inline arrangement as shown in fig [5a]. in inline fin arrangement, the heat transfer coefficient is 34.86% more at 30° as compared to 0° whereas it found 20% as compared to 60°. In staggered arrangement heat transfer coefficient is maximum at 0°, after it starts decreasing as fin angle increases as shown ion fig [5b]. At 0° staggered arrangement, heat transfer is 4% greater as compared to 30° whereas it 34% as compared to 60°. Comparing both arrangements, it is found that at 0° staggered arrangement heat transfer is greater by 16.58% as compared to 30° inline fin arrangement.

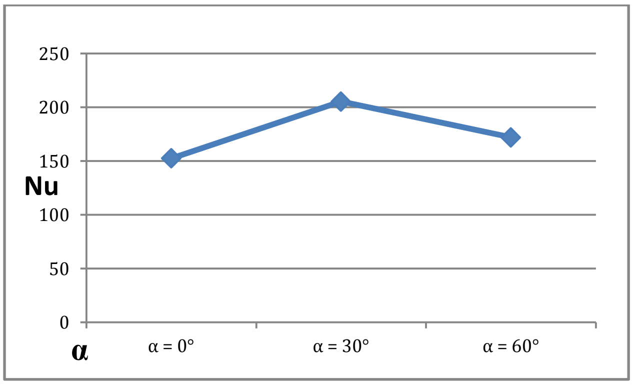

Fig. 5a Nu vs fin angles for inline arrangement at Q=50W and Re = 6026

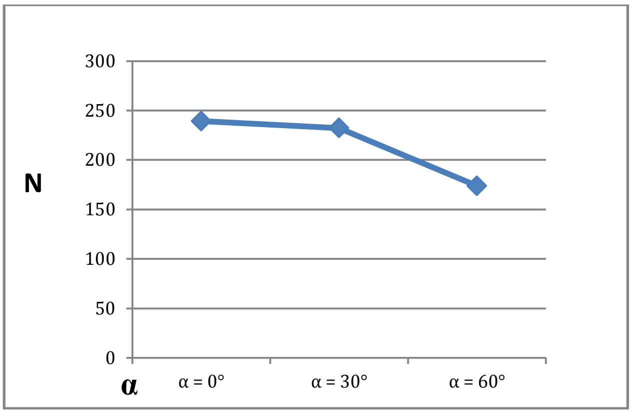

Fig 5b. Nu vs fin angles for staggered arrangement at Q=50W and Re = 6026

5.1.2 Re vs Nu

As Re increases Nu increases. For both 30° inline and 0° staggered fin arrangement, rate of heat transfer is greater at higher Re i.e. at 18000 fig [6a] and fig [6b] respectively. For 0° staggered fin arrangement, among Re range 4000 - 18000, the heat transfer rate is increased by 31.64%, 14.1069%, 26.662%, 23.0463%, 8.1974% and 2.3441%as compared to 30° inline fin arrangement fig [7a].

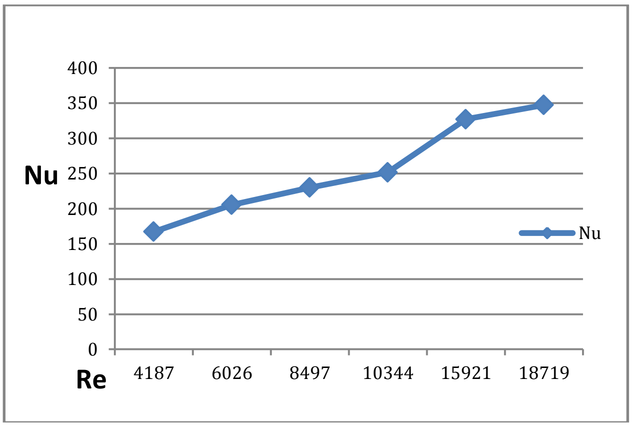

Fig 6a. Nu vs Re for inline arrangement α = 30° and Q=50W

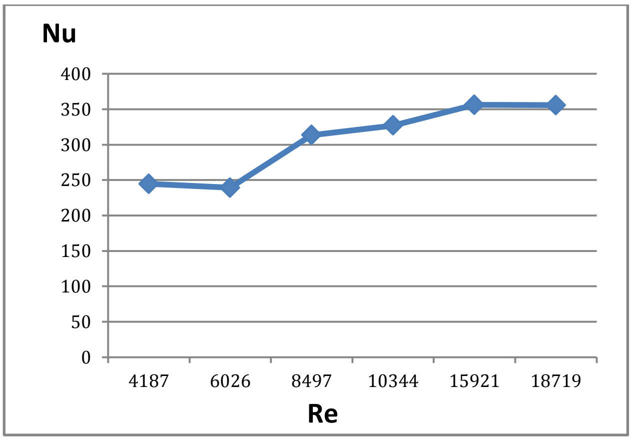

Fig 6b. Variation of Nu with Re for staggered arrangement

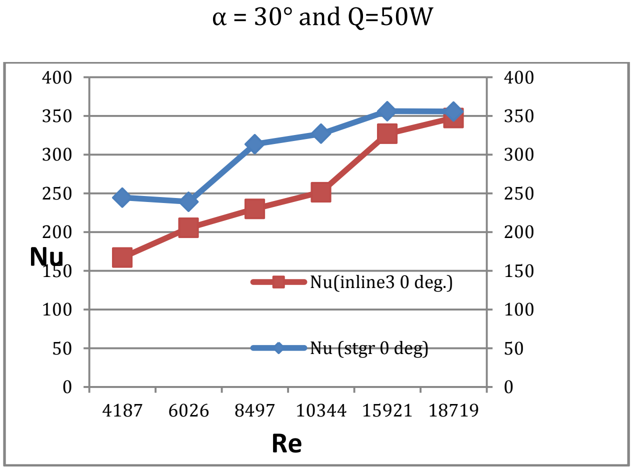

Fig 7a. Nu vs Re for both inline and staggered arrangement at Q = 50W

5.1.3 Pressure drop vs fin angle

In case of pressure drop, in both the arrangements it is minimum at 30°. In staggered arrangement at 30°, it is less by 3% as compared to inline arrangement. Fig. [7b and 8a]. Pressure drop increases as velocity increases fig.[8b and 9a]. Though it is found that heat transfer rate is maximum in 0° staggered fin arrangement but the pressure drop is 0.02% greater as compared to 30° inline fin arrangement.

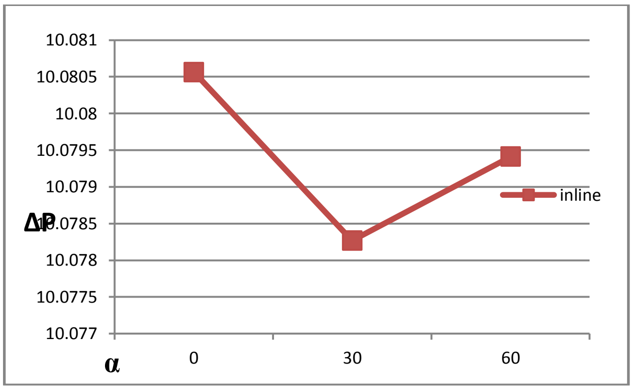

Fig 7b. pressure drop at various fin angle for inline arrangement at Re = 6026 and Q=50W

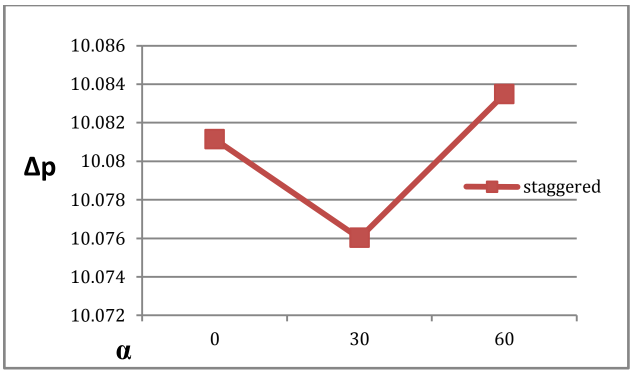

Fig 8a. pressure drop at various fin angle for staggered arrangement at Re = 6026 and Q=50W

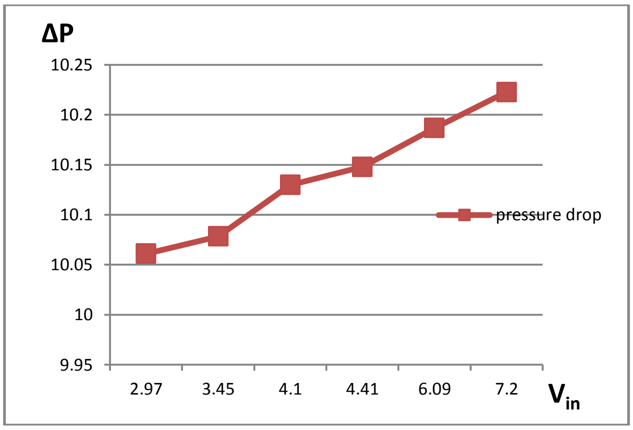

Fig 8b. pressure drop at various inlet velocity for inline arrangement at Re = 6026 and Q=50W

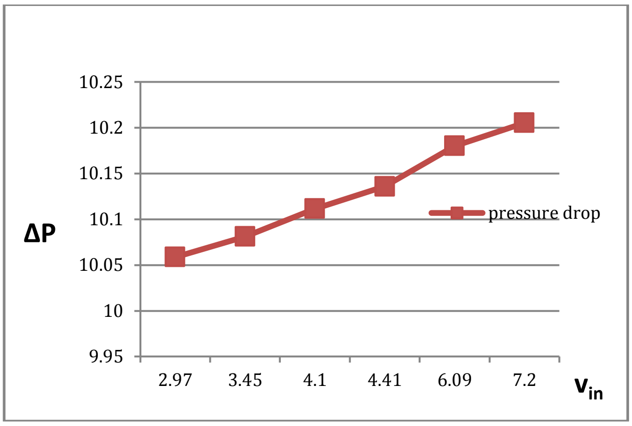

Fig 9a. pressure drop at various inlet velocity for staggered arrangement at Re = 6026 and Q=50W

5.1.4 Pressure drop vs Profit factor

Profit factor is calculated by using the equation (9) and (10). The graph pressure drop vs profit factor fig [9b and 10] implies that for both arrangements profit factor is maximum at 30°. Whereas it is greater by 19% in 30° inline fin arrangement as compared to 0° staggered arrangement.

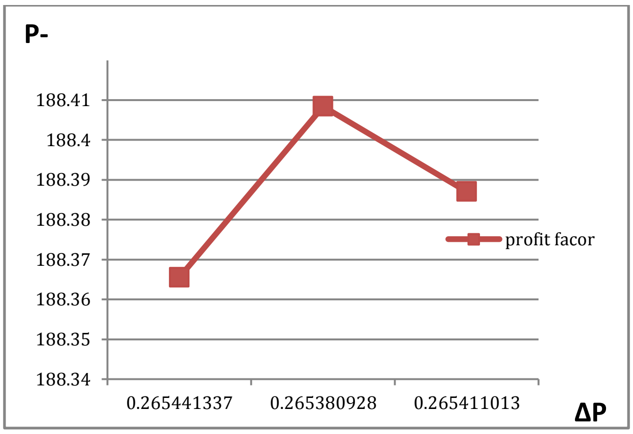

Fig 9b. Profit facor vs pressure drop

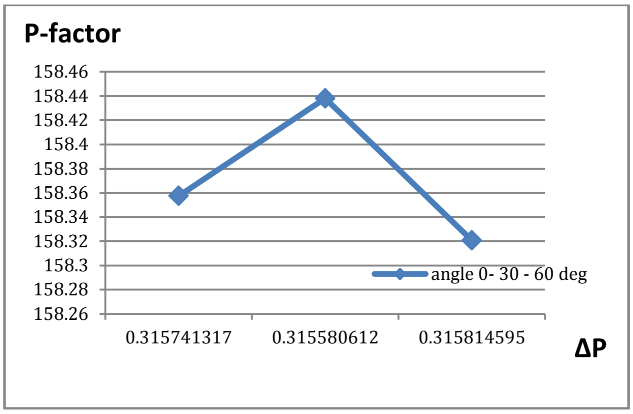

Fig 10. Profit facor vs pressure drop

Conclusions

In the present paper the effect of insertion of solid rectangular fins in the passage of plate fin heat sink in inline and staggered manner with varying fin angle manner on turbulent forced convective heat transfer in a 3D channel been experimentally and computationally examined. The study shows that rectangular plate fins considerably enhanced the heat transfer for the range of Reynolds number considered in the present study. The use of rectangular plate fins although increase the weight and the cost but the enhancement in heat transfer is considerably enough to compensate the cost of manufacturing. Therefore it can be concluded that the use of rectangular plate fin heat sink is beneficial in the fins of air cooled engine as well as in many other industrial applications wherever there is a necessity of faster heat removal.

The experiment is carried out with the supply heat from 50W, 80W, 100W and 125W.The Reynolds No. from 4000 - 18000 in the step size of 2000. The fin angles are varied from 0°, 30° and 60° for inline and staggered arrangement.

It can be concluded that,

1) Fin angle and arrangement plays an vital role in increasing the coefficient of heat transfer. Selecting larger fin angle with any arrangement is not always going to increase heat transfer rate.
2) Comparing both arrangements, it is found that at 0° staggered arrangement heat transfer is greater by 16.58% as compared to 30° inline fin arrangement.
3) As Re increases Nu increases. For 0° staggered fin arrangement, among Re range 4000 - 18000, the heat transfer rate is increased by 31.64%, 14.1069%, 26.662%, 23.0463%, 8.1974% and 2.3441%as compared to 30° inline fin arrangement
4) In case of pressure drop, 0° staggered fin arrangement the pressure drop is 0.02% greater as compared to 30° inline fin arrangement.
5) Profit factor is greater by 19% in 30° inline fin arrangement as compared to 0° staggered arrangement

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