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Experimental Study of Concentric Staggered Annular Fin with Radial Outlet and Staggered Rectangular Fin with Lateral Outlet

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Abstract

In this paper, an experimental study of a concentric staggered annular fin with radial outlets and a staggered rectangular fin with lateral outlets for steady-state natural convection, and forced convection is presented. The objective of this work is to investigate the effect of shape, profile and arrangement of the fins in heat transfer with minimum blockage ratio, stagnation, and pressure drop. In this regard, the two different shapes of fins are designed with the same material, identical extended surface area, width, and thickness; and studied experimentally with identical thermal load and boundary conditions, as well as identical flow characteristics. In case of forced convection, the orientation of the base-plate of the fins is kept perpendicular to the direction of flow to achieve velocity vectors parallel to the fin surfaces. The overall heat transfer coefficient, fin efficiency, effectiveness, and temperature distribution of both fins are compared for different base-plate's temperature of the fins and different free stream air velocity; and also for natural convection. It is found that the shape of the fins has a significant effect on heat transfer, especially in the case of forced convection. Though both fins are performed almost the same in natural convection, the concentric staggered annular fin with radial outlets performed better in forced convection. It is concluded that the concentric staggered annular fin with radial outlets can be used in the practical field instead of the conventional rectangular fin, particularly for forced convection.

Graphical Abstract

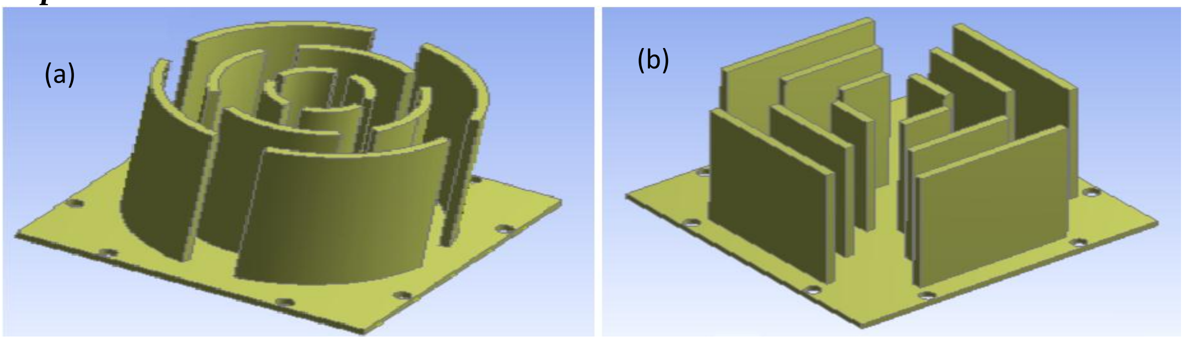

***Figure 1:** (a) Concentric Staggered Annular Fin, (b) Staggered Rectangular Fin*

***Keywords:** Annular fin, effectiveness of fin, effect of shape of fin, extended surfaces, fin efficiency, rectangular fin*

INTRODUCTION

Fins are extended metal surfaces used to remove heat from the primary surface of a hot body in an effective and efficient manner to maintain a limited temperature in the hot body [1]. It facilitates to convert the optimum amount of thermal energy into mechanical or electrical energy and vice versa without sacrificing the performance, life span of the structures

[2], and dissipate rest amount of heat to the ambient. There are numerous practical application of fins such as cooling of electric chips and devices [1], automobile and aircraft engines [3, 4], heat rejection devices for space vehicle [1, 5], air-cooled engine, radiator, gas turbine blade [5], transformer [6] and air-conditioning systems, [7, 8] etc.

The evolution of the thin extended surface technology is first started in 1922 by Harper and Brown with mathematical analysis; these extended thin surfaces are known as cooling fin or fin [3]. Since then, there are different types of fins developed by numerous researchers with supporting mathematical and numerical analysis such as longitudinal fin of rectangular, trapezoidal and parabolic profile; cylindrical tube with longitudinal fins, radial fin of rectangular and trapezoidal profile; cylindrical spine, truncated conical spine, and truncated concave parabolic spine [3], etc. Still, the fin is an important topic among researchers to develop the efficient way of heat dissipation from a hot body by designing different profile and shape of thin extended surfaces and without using any moving parts [9]. Due to the advancement of material science, manufacturing technology, growing of light weight and micro size object concept, managing and using of thermal energy, as well as, dissipation of thermal energy are the most challenging issue for designing of densely packed and high-heat-flux dissipating microelectronic devices; which are widely used in the aerospace and other industries [10]. Recent advancements in micro-fabrication techniques in electronic and other fields, the researchers are becoming more interested to work on designing and evaluation of the performance of fin to ensure efficient dissipation of heat from a hot body to the surrounding [10]. The miniaturization of electronics devices required a higher rate of heat dissipation per unit volume to obtain the reliability of the electronic components at the desired level [5]. The power of computer microprocessors is becoming double every eighteen months while keeping their volume constant [4]. Due to the rapid increase of performance of the electronic and other equipment, it is required to have an enhancement in thermal energy management to avoid overheating, malfunctions and degradations [4]; which leads the researcher to look for further enhancement of cooling by fins [3]. Fin also provides some additional benefits to the system such as heat transfer by conduction, as well as, enhance structural strength [4, 9].

The cooling performance of fin depends on numerous factors such as number of fins or fin density, shape and arrangement e.g. inline, radial, staggered [1]; geometry of cross-section e.g. circular, square, elliptical, NACA, lancet etc.; fin's diameter, thickness, length, height and pitch [2, 3]. Fluid flow characteristics also influence on heat transfer through fin such as acceleration, cross flow, vortex and wake in the flow; blockage area ratio and cross flow blowing ratio; disturbance of flow by fin or position of stagnations [9]; properties of fins material, spacing of fins and fluid [10]; type of fluid flow e.g. forced or natural convection; nature of fluid flow e.g. laminar, turbulent, vortex flow, and also location of fan or blower [5].

There are numerous works [1-19] reported on the enhancement of heat transfer through fins by using pin fins, perforated plate fins, a jet impingement in cross-flow; varying geometric shape, size, arrangement, and orientation. Nguyen H. and A. Aziz [1] worked on rectangular, trapezoidal, triangular and concave parabolic shapes of convecting-radiating fins by finite difference method for one-dimensional steady-state condition and neglecting heat transfer by radiation. From their study, it is found that the efficiency and heat transfer through the fins of other profile shape is within 11% of commonly used rectangular shape. Hong, S. K., D. H. Rhee and H. H. Cho [2] worked on the effect of arrangement and shape of the fin on heat transfer for impingement or effusion cooling with cross flow. They found that 40-45% enhancement of heat and mass transfer coefficient in case of a rectangular fin with high blowing ratio. Kraus, A. D. [3] worked on reviewing fin literature from 1922 to 1987 about configuration, optimization, arrangement, numerical analysis and mathematical techniques of heat transfer through fin. Starner, K. and H. McManus [9] worked on heat transfer from the rectangular-fin arrays for free convection. They found that fin spacing, height, arrangement, and orientation effect on heat transfer from the rectangular-fin arrays. İzci, T., M. Koz, and A. Koşar [10] worked on the effect of the shape of the fin on the thermal-hydraulic performance of micro-pin fin heat sinks. It is found that the rectangular shape pin fin has the highest pressure drop and friction factor, and cone-shaped micro pin-fin has the highest thermal performance over pin-fin. Reddy, R. S. k., et al., [5] worked on the effect of pin-shaped fin for forced convection heat transfer by using ANSYS. It is found that the perforated pin fin has higher heat transfer performance. Lorenzini, G. and S. Moretti [4] worked on the numerical analysis of heat transfer from a Y-shaped fin. A/K Abu-Hijleh, B [6] investigated numerically on cross-flow forced convection heat transfer from a horizontal cylinder with an equally spaced radial fin. Kim, Y. H., et al. [7] worked on the effect of the alignment of the fin and tube on the heat transfer performance of a fin-tube heat exchangers with large fin pitch. Li, P., and K. Y. Kim [11] worked on optimization of staggered elliptical pin-fin by using Reynolds-averaged Navier-Stokes analysis, and found that shorter fin with a longer pitch performed better; and also elliptic pin-fin perform better than the circular pin-fin. Dewan, A., et al. [12] investigated on the effect of spacing and material of circular pin fins; and found that higher thermal conductivity yields better heat transfer without an increase of pressure drop and higher spacing of fin results in lower heat transfer. Oswal, S., H. Jagtap, and A. Mane [13] worked on the factors which are affecting on the thermal performance of fins numerically. Asadi, M., and R.H. [8] numerically investigated on constant cross-section area fin for an interaction of radiation with convection. Arul, N., et al. [14] worked on the experimental and computational analysis of various types of fins. Eldhose, A., B. Paul, and T.J. Sebastian [15] investigated pin-fin numerically by varying shape and materials. Though numerous works are reported on fins, in our best knowledge, no work is being reported on the evaluation of performance of heat transfer by Concentric Staggered Annular and Staggered Rectangular Fins in case of steady-state natural convection, and forced convection flow to a direction parallel to the fin surfaces.

The objective of this work is to investigate the effect of the shape of fins on heat transfer. In this regard, a Concentric Staggered Annular Fin with radial outlets and a Staggered Rectangular Fin with lateral outlets are designed with an identical base-plate, fin length, thickness, material, and surface area. The fins are studied experimentally with the same thermal load and boundary condition, as well as flow parameters in the case of steady-state natural and forced convection. To maintain identical thermal load and boundary condition, a heating element of 40 Watts capacity with fin holder is designed, fabricated and attached at the back side of the base-plate of the fins. The heating element is insulated around three sides to ensure the heat flow in the direction parallel to the fin surfaces. In both cases, the distributions of temperature along the length of the fins are found experimentally and compared with the theoretical results. In this work, the respective steps of designing, modeling, and fabrication of fins; as well as the experimental procedure, results, and analysis are described in the subsequence paragraphs.

METHODOLOGY

Governing Equation

The following non-dimensional equations and non-dimensional numbers are used to determine the convective heat transfer coefficient for natural and forced convection [16]. In this work, heat loss due to radiation from the fin is neglected.

Continuity equation,

$$\nabla^* \cdot v^* = 0 \quad (1)$$

Momentum equation,

$$\left(\frac{\partial v^*}{\partial t^*} + v^* \cdot \nabla^* \theta\right) = \frac{1}{R_e} \nabla^* P^* + G_r\,\theta\,\vec{e_z} + \frac{1}{R_e}\,\nabla^{*2} v^* \quad (2)$$

Energy equation,

$$\frac{\partial v^*}{\partial t^*} + v^* \cdot \nabla^* \theta = \frac{1}{R_e} \cdot \frac{1}{P_r}\,\nabla^{*2}\theta \quad (3)$$

where non-dimensional parameter,

$$v^* = \frac{v}{V}\,;\; t^* = \frac{t}{L/V}\,;\; T^* = \frac{T - T_\infty}{T_0 - T_\infty} = \theta$$

$$\nabla^* = L\nabla\,;\; P^* = \frac{P}{\mu\,V/L}\,;\; \nabla^{*2} = L^2 \cdot \nabla^2$$

Grashof number,

$$G_r = \frac{g\,\beta\,(T_0 - T_\infty) \cdot L^3}{\nu^2}$$

Prandtl number, $P_r = \frac{\mu\,C_p}{k}$

Reynolds number, $R_e = \frac{\rho V L}{\mu}$

Rayleigh number,

$$R_a = P_r \cdot R_e = \frac{g\,\beta\,(T_0 - T_\infty) \cdot L^3}{\nu \cdot \alpha}$$

Thermal diffusivity, $\alpha = \frac{k}{\rho\,C_P}$

Nusselt Number, $N_u = \frac{h_c \cdot L}{k}$

For horizontal plate and laminar flow [17]

$$10^4 < R_a < 10^8; \quad N_u = 0.56 \cdot (G_r \cdot P_r)^{0.25} \quad (4)$$

For horizontal plate and turbulent flow

$$10^8 < R_a < 10^{12}; \quad N_u = 0.13 \cdot (G_r \cdot P_r)^{0.25} \quad (5)$$

Condition of infinitely long fin [18]

$$L_{Infinite} \geq 2.65\sqrt{\frac{k\,A}{h\,P}}$$

In case of heat transfer through fin tip by convection, the temperature distribution and heat transfer through fin is given by the following two equations [19],

$$\frac{T(x) - T_\infty}{T_0 - T_\infty} = \frac{cosh[m(L-x)] + \left(\frac{h}{m\,k}\right) sinh[m(L-x)]}{coshmL + \left(\frac{m}{h\,k}\right) sinh(mL)} \quad (6)$$

$$\dot{Q}_{Convection} = \sqrt{hpkA}(T_0 - T_\infty)\,\frac{\sinh mL + \left(\frac{h}{mk}\right)\cosh ml}{\cosh ml + \left(\frac{h}{mk}\right)\sinh mL} \quad (7)$$

Fin efficiency [19],

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin,max}} = \frac{\tanh mL}{mL} \quad (8)$$

$$\dot{Q}_{fin,max} = h\,A_{fin}\,(T_0 - T_\infty) \quad (9)$$

Fin effectiveness [19],

$$\epsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no\,fin}} = \frac{\dot{Q}_{fin}}{h\,A_b(T_0 - T_\infty)} \quad (10)$$

The convective heat transfer coefficient for forced convection is found by using Logarithmic Mean Temperature Difference (LMTD) [19],

$$T_{lm} = \frac{T_{out} - T_{in}}{ln\left(\frac{T_0 - T_{in}}{T_0 - T_{out}}\right)} \; and \; h_c = \frac{\dot{Q}_{fin}}{A_{fin} \times T_{lm}} \quad (11)$$

The amount of heat released and absorbed by the air in the case of forced convection is evaluated by,

$$\dot{Q}_{fin} = \dot{m}_{air} C_p \Delta T = \dot{m}_{air} C_p\,(T_{out} - T_{in}) \quad (12)$$

$$\dot{m}_{air} = A_d\,V_{air}\,\rho_{air} \quad (13)$$

where, A_d = Cross-sectional area of the test section of the wind tunnel, T_{out} and T_{in} is the temperature of the air at the outlet and inlet of the working section of the wind tunnel.

Design and Fabrication of Fins

A Staggered Rectangular Fin with lateral outlets and a Concentric Staggered Annular Fin with radial outlets are designed and fabricated with an identical size of base-plate of 106 mm x 106 mm size, fin length of 70 mm, thickness 3 mm, surface area, and fin spacing of 15 mm. During design, the peripheral length of each segment of cylindrical fins is kept equal to the respective length of each segment of Staggered Rectangular Fins; and the width of the lateral outlet of rectangular fins is also kept equal to the respective width of the radial outlet of annular fins. The segments of fins are welded to the base-plate. Both fins and base-plates are made of 3 mm aluminum plate. The 3D design tool SolidWorks 2015 is used to design the fins.

Annular Fin with Radial Outlet

The design, geometry, and fabrication of Concentric Staggered Annular Fin with radial outlets are shown in Fig. 2 to Fig. 3.

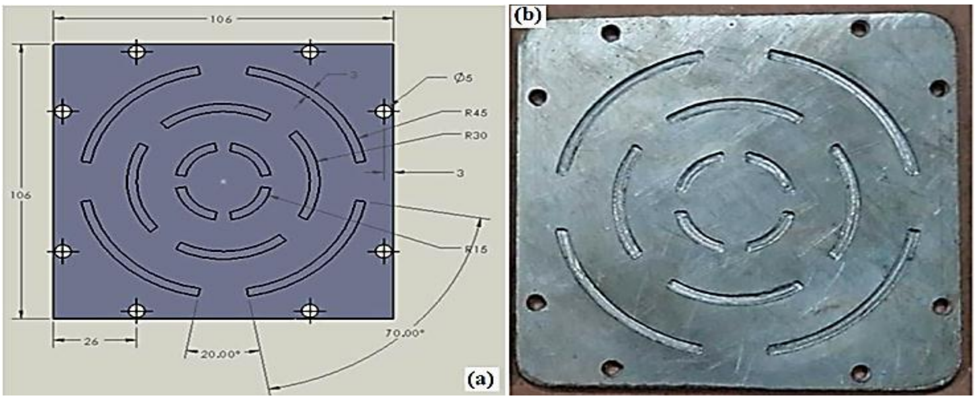

Figure 2: *(a) 3D Modeling of Base-Plate of Concentric Staggered Annular Fin (Dimension in mm), (b) Fabricated Base-Plate*

The angular or radial gap between the cylindrical segments of annular fin plates is kept 20^0 and diameter of outer, middle and inner fins is 90 mm, 60 mm and 30 mm respectively. The cylindrical segments of the Concentric Staggered Annular Fin are fabricated with aluminum by sand casting with appropriate allowances. The final radii and thickness of the cylindrical fin segments are obtained by machining in the milling machine. To measure the temperature at multiple locations of the fins, an indentation is made along the length of each cylindrical fin segment at a distance 57 mm, 42 mm, 27 mm and 13 mm from the base-plate. The cylindrical segments are joined to the base-plate by aluminum arc welding. The fabricated cylindrical segments of outer, middle and inner fin plate of Concentric Staggered Annular Fin are shown in Fig. 3.

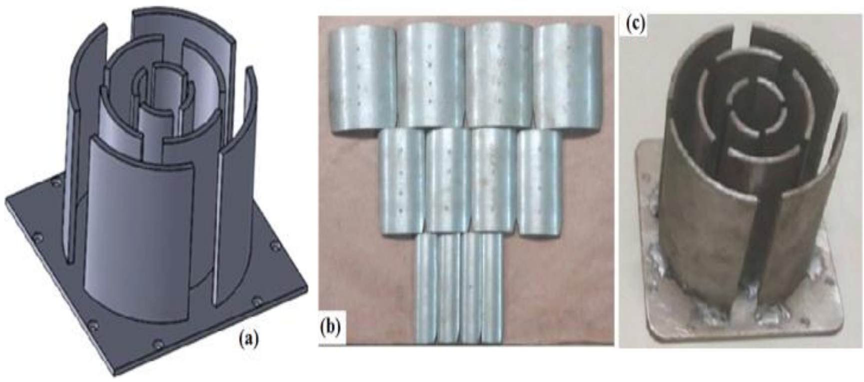

Figure 3: *(a) 3D modeling of Concentric Staggered Annular Fin, (b) 70 mm long Cylindrical Segments of Outer, Middle and Inner Fin, and (c) Fabricated Concentric Staggered Annular Fin with Base-Plate.*

Staggered Rectangular Fin

The design, geometry, and fabrication of Staggered Rectangular Fin with lateral outlets are shown in Fig. 4 to Fig. 5. The surface area, thickness, width and spacing of the fin segments are kept the same as of Concentric Staggered Annular Fin. The width of the outer, middle and inner rectangular segments of fins are 53.14 mm, 34.82 mm and 16.50 mm respectively; and the thickness of the respective rectangular fin segments is 3 mm. The rectangular fin segments are fabricated from the 3 mm thick aluminum plate and welded to the 3 mm thick aluminum base-plate by aluminum arc welding.

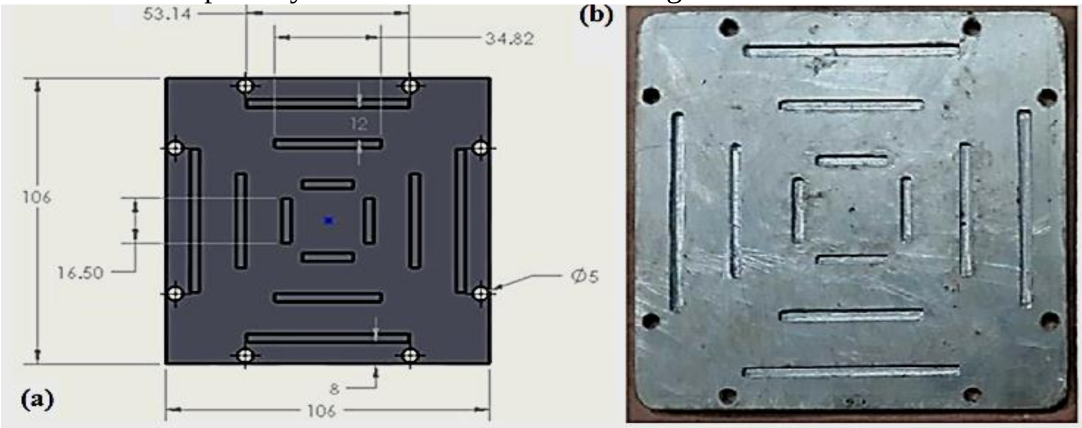

Figure 4: *(a) 3D Modeling of Base-Plate of Staggered Rectangular Fin (Dimension in mm), (b) Fabricated Base-Plate*

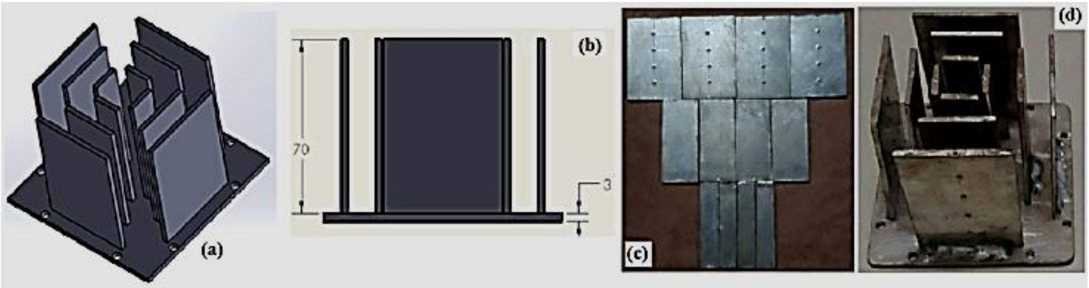

Figure 5: *(a) 3D Model of Staggered Rectangular Fin, (b) Front View Staggered Rectangular Fin (Dimension in mm), (c) 70 mm long Rectangular Segments of Outer, Middle and Inner Fin, (d) Staggered Rectangular Fin with Base-Plate*

Heating Element

The heating element consists of a rectangular steel plate, an aluminum casing, a flat electric heater, and an AC voltage controller. The aluminum casing and the four outer sides of the fin base-plate are insulated with Glass Wool to ensure heat flow in a direction perpendicular to the base-plate of the fin only and minimize the loss of thermal energy from the heating element. Due to the Glass Wool insulations, the base-plate of the fins is heated only by absorbing thermal energy from the heater.

The thickness of the rectangular steel plate is 10 mm, size 200 mm x 125 mm with a 107 mm x 107 mm square slot at the center. The 3D model and fabricated steel plate for supporting the fin are shown in Fig. 6.

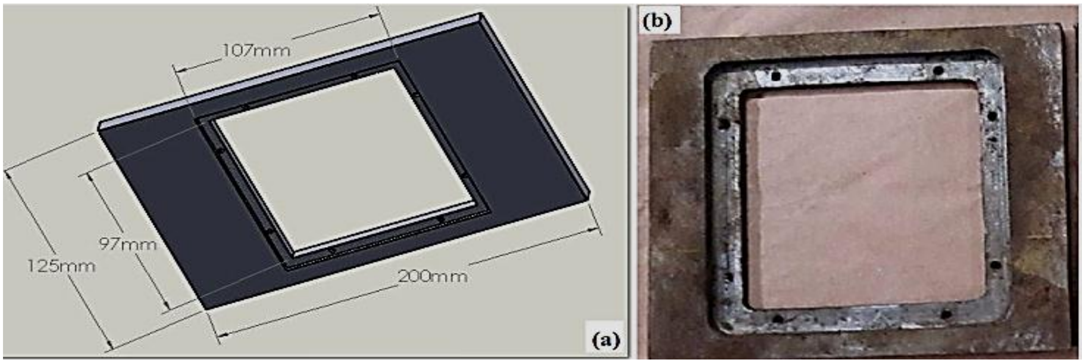

Figure 6: *(a) 3D Modeling of 10 mm Thick Supporting Steel Plate, (b) Fabricated Supporting Steel Plate of Fin.*

The Casing is fabricated from 0.5 mm thick aluminum sheet; and the length, width, and depth of the Casing are 200 mm, 126 mm and 35 mm respectively. The 3D model and fabricated Casing, and 40 Watts Flat Electric Heater are shown in Fig. 7.

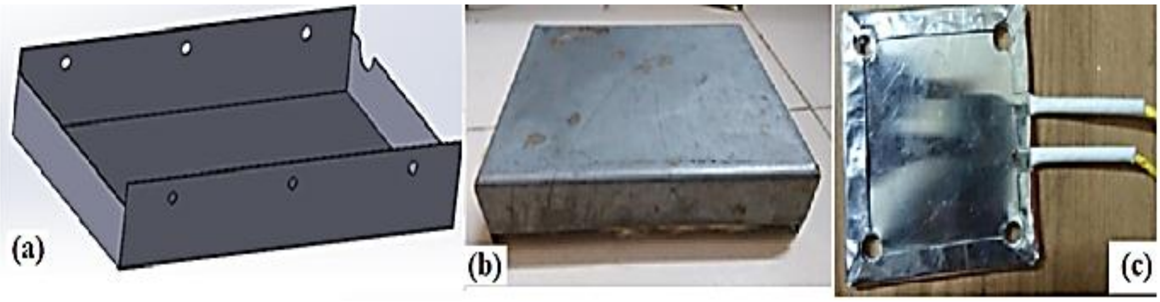

Figure 7: *(a) 3D Model of Heater Casing; (b) Aluminum Heater Casing; and (c) 40 Watts Flat Electric Heater*

The different level of constant temperature at the fin base-plate is maintained by regulating an AC Voltage Controller, which is attached to the heater. Pen type digital thermometers are used to measure the temperature at different locations of the fins. Anemometer is used to measure the velocity of the air. The equipment used in the experiment is shown in Fig. 8.

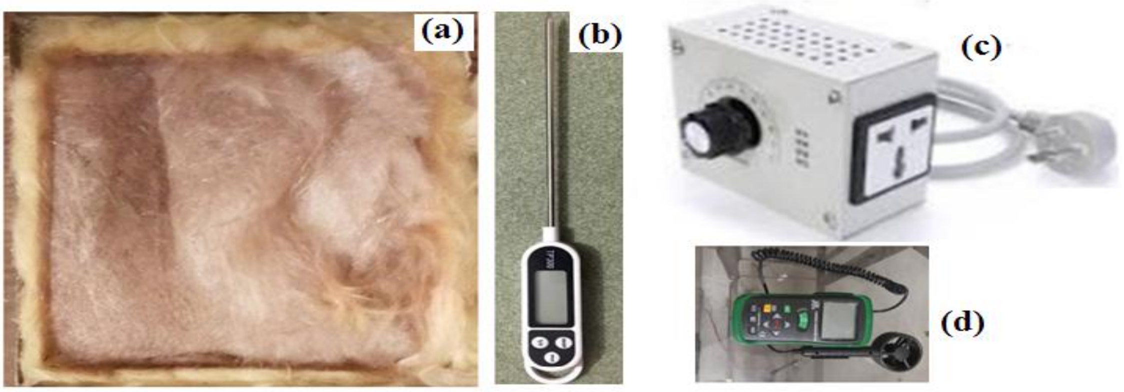

Figure 8: *(a) Locally Available Glass Wool, (b) Pen Type Digital Thermometer, and (c) Anemometer, (d) AC Voltage Controller*

The final assembly of Staggered Rectangular Fin and Concentric Staggered Annular Fin with Electric Flat Heater is shown in Fig. 9.

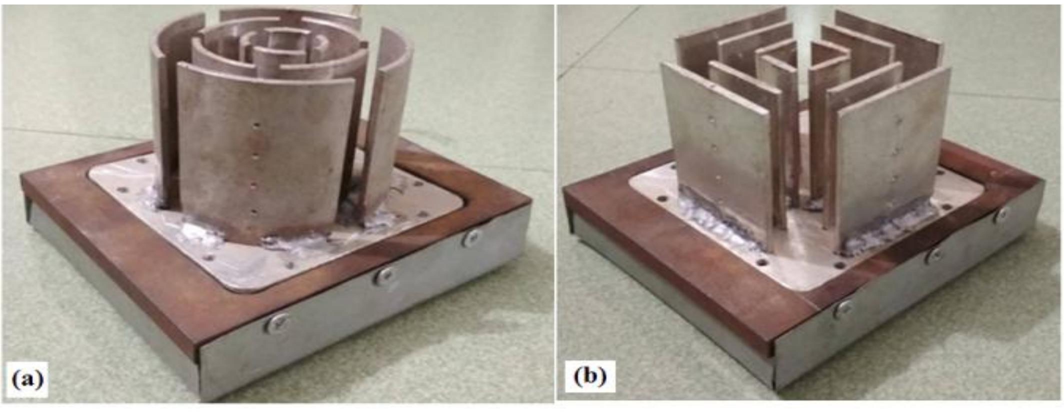

Figure 9: *(a) Concentric Staggered Annular Fin; (b) Staggered Rectangular Fin*

Experimental Procedure
Natural Convection
For natural convection, the fins are placed on a wooden stool in such a way that the segments of the fin surfaces remained horizontal as shown in Fig. 10. The base-plate of the fins is heated to the temperature of 60°C, 70°C, 80°C, and 90°C respectively by regulating the AC voltage controller. After obtaining the steady-state condition, the temperature at the ends of the fins and specific locations of the fins from the base-plate is recorded. The heat transfer coefficient is determined by using Grashof number, Rayleigh number, Nusselts number, the governing equation (4) and (5). The theoretical distribution of temperature is obtained by using the governing equation (6).

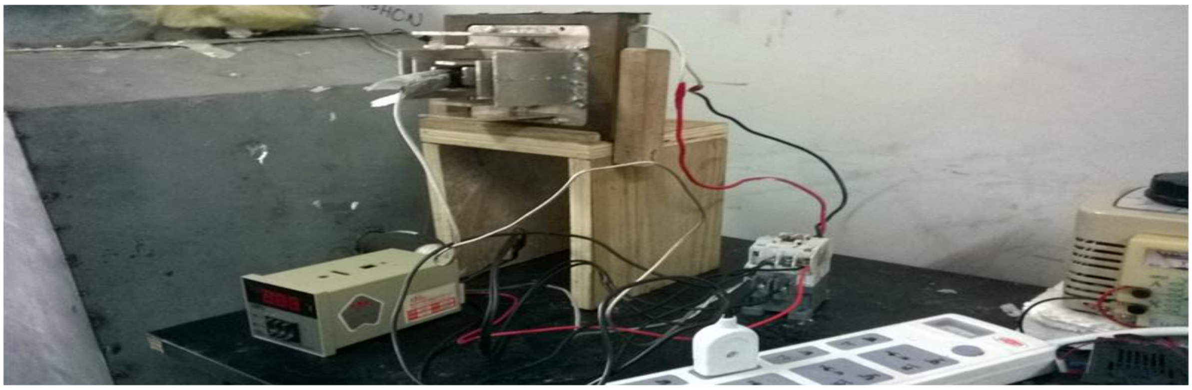

Figure 10: *Experimental setup for natural convection.*

Force Convection
Fins are placed in the wind tunnel to make sure that the velocity vector of air is parallel to the fin surfaces as shown in Fig. 11. The temperature of the fins at a distance 13 mm, 27 mm, 42 mm, 57 mm, and at the tips are recorded for the base-plate temperature of 60 ^{0}C, 70 ^{0}C, 80 ^{0}C, and 90 ^{0}C; and also for 1.5 m/s and 2.5 m/s air velocity. The temperature of the air at the inlet and outlet of the test section of the wind tunnel are also recorded. The cross section of the test area of the wind tunnel is 16 inch by 16 inch. The amount of heat transfer from the fin is determined from the governing equations (11), (12) and (13).

Figure 11: *Setup of fin in subsonic wind tunnel for force convection.*

EXPERIMENTAL RESULTS
Natural Convection
The theoretical and experimental temperature distributions of both fins for steady-state natural convection and different temperatures of base-plate are shown in Fig. 12 to Fig. 15.

Forced Convection
The theoretical and experimental temperature distributions of both fins for steady-state force convection of 1.50 m/s and 2.50 m/s and different temperatures of the base-plate are shown in Fig. 16 to Fig. 23.

Natural Convection

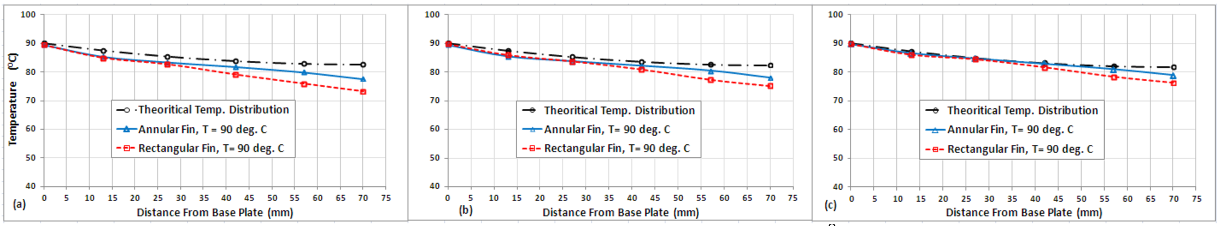

Figure 12: *Natural Convection and Base-Plate Temperature 90°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

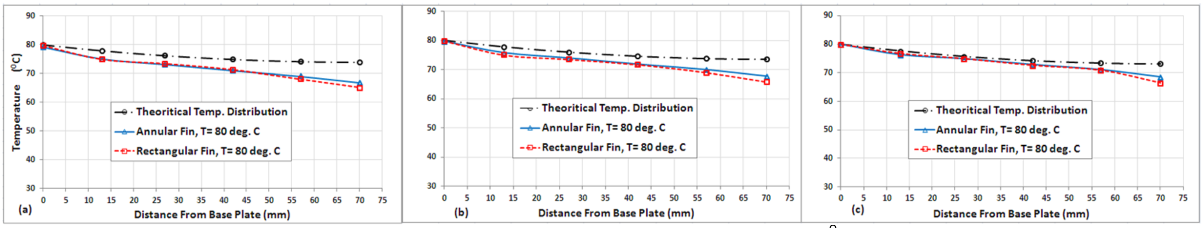

Figure 13: *Natural Convection and Base-Plate Temperature 80°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

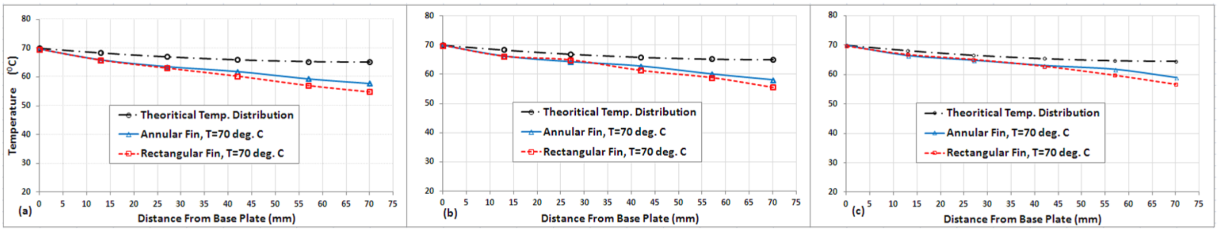

Figure 14: *Natural Convection and Base-Plate Temperature 70°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

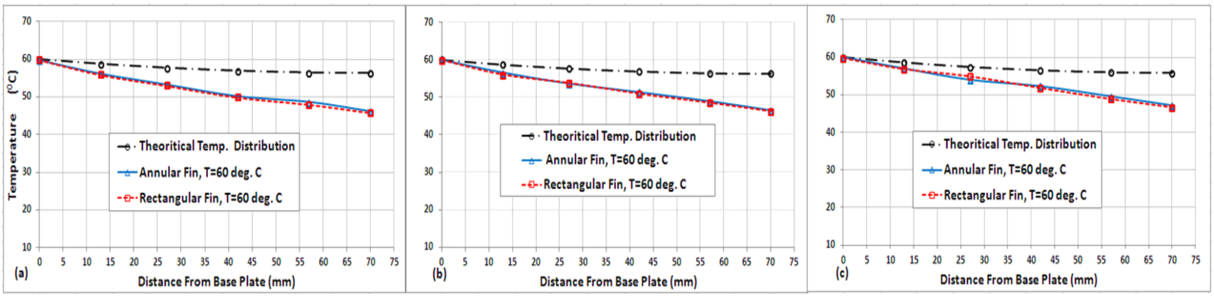

Figure 15: *Natural Convection and Base-Plate Temperature 60°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

Forced Convection

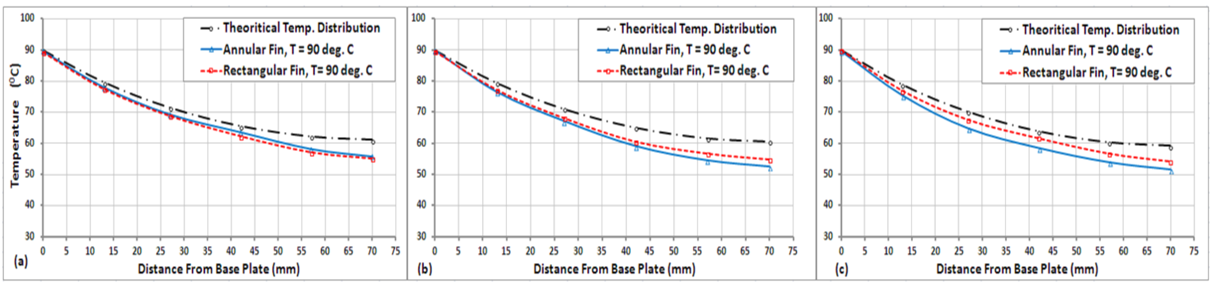

Figure 16: *Forced Convection, V = 1.5 m/sec and Base-Plate Temperature 90°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

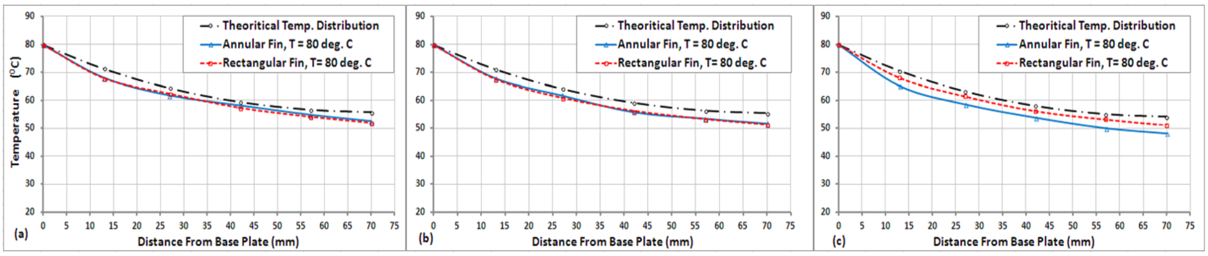

Figure 17: *Forced Convection, V = 1.5 m/sec and Base-Plate Temperature 80°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

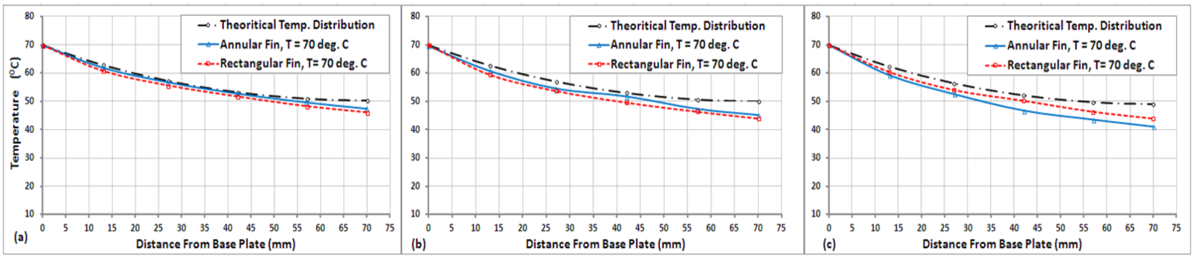

Figure 18: *Forced Convection, V = 1.5 m/sec and Base-Plate Temperature 70°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

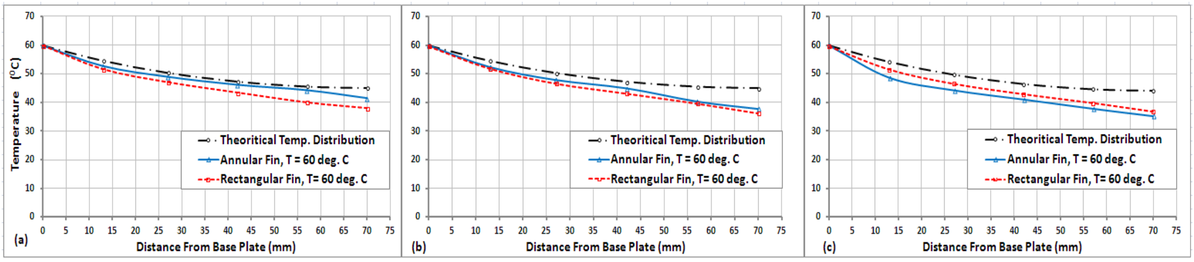

Figure 19: *Forced Convection, V = 1.5 m/sec and Base-Plate Temperature 60°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

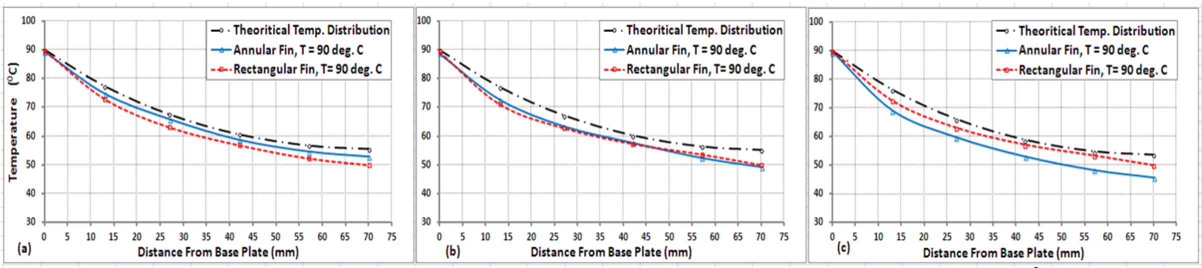

Figure 20: *Forced Convection, V = 2.5 m/sec and Base-Plate Temperature 90°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

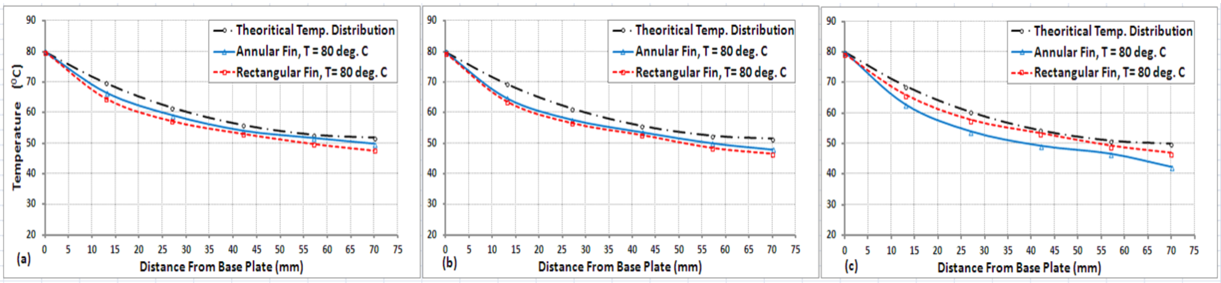

Figure 21: *Forced Convection, V = 2.5 m/sec and Base-Plate Temperature 80°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

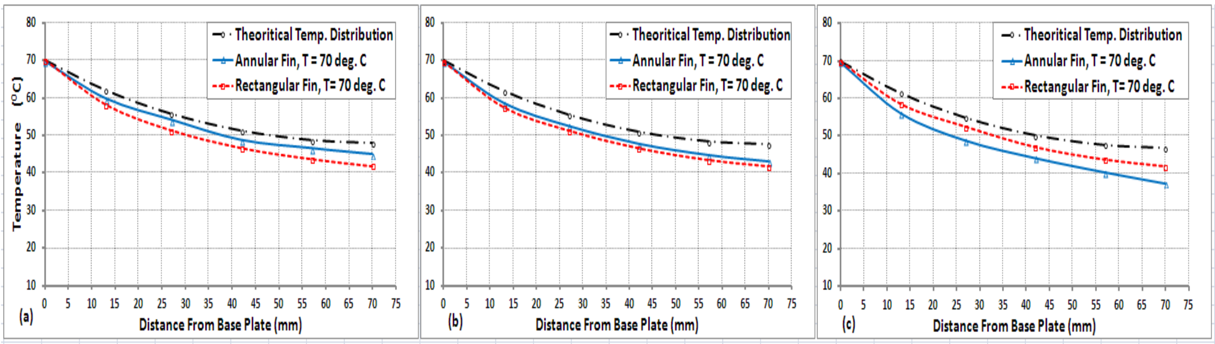

Figure 22: *Forced Convection, V = 2.5 m/sec and Base-Plate Temperature 70°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

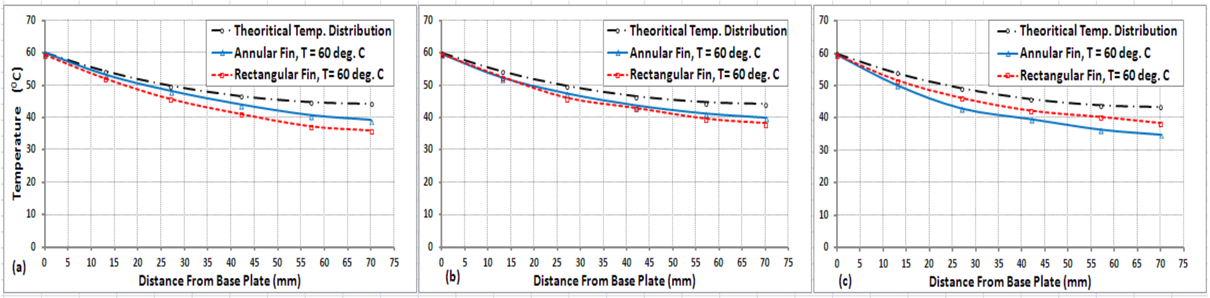

Figure 23: *Forced Convection, V = 2.5 m/sec and Base-Plate Temperature 60°C: (a) Outer Fin, (b) Middle Fin, (c) Inner Fin.*

DISCUSSION

Natural Convection

In the case of natural convection, the temperature distributions of outer and middle fins of the Concentric Staggered Annular Fin are found higher than the Staggered Rectangular Fin for the base-plate temperature of 90°C, 80°C and 70°C (Fig. 12 to Fig. 14). However, the temperature distributions of the inner fins are almost the same in both fins (Fig. 12(c), 13(c) and 14(c)). For 60°C base-plate temperature, the temperature distributions in both fins are almost identical. For an ideal or highest performance fin, the temperature distribution is equal or close to the base-plate temperature. In that respect for the higher base-plate temperature, the Concentric Staggered Annular Fin is performed better than the Staggered Rectangular Fin. The fin segments of the Staggered Rectangular Fin have flat surfaces which facilitate to move the air quickly and easily after absorbing heat from adjacent of the flat surfaces. In the case of Concentric Staggered Annular Fin due to the curved surfaces, air is not able to move quickly and faster as of flat surfaces and getting more time to stick with the fin surfaces, which facilitates to hold a relatively higher temperature of the fin surfaces. However, at a relatively low temperature 60 ^{0}C of the base-plate, both fins perform almost the same. The heat transfer coefficient and Nusselt number are almost the same for both fins and shown in Fig. 25(a).

Forced Convection

In the case of forced convection, the Concentric Staggered Annular Fin remains at a higher temperature relative to the Staggered Rectangular Fin except the innermost fins (Fig. 16 to Fig. 23). For free stream air velocity of 2.5 m/s, the Concentric Staggered Annular Fin is maintained a relatively higher temperature than that of 1.5 m/s free stream air velocity. An ideal fin remains at a temperature equal or close to the base-plate temperature performed highest. In that respect, the Concentric Staggered Annular Fin performed better than the Staggered Rectangular Fin. The convective heat transfer coefficient, efficiency, and effectiveness of both fins are shown in Fig. 24 and Fig. 25.

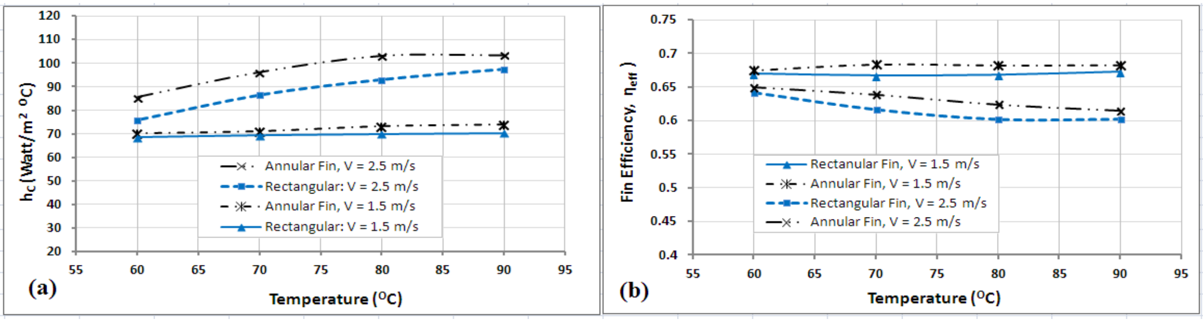

Figure 24: *(a) Convective Heat Transfer Coefficient, h_c for Force Convection; and (b) Fin Efficiency for Force Convection.*

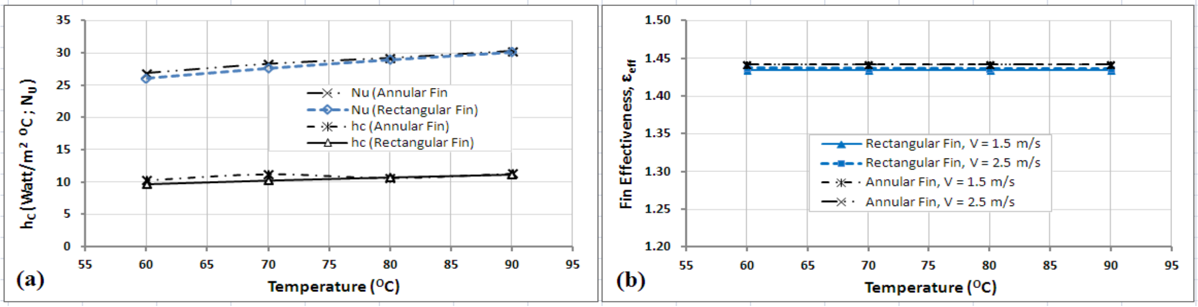

Figure 25: *(a) Convective Heat Transfer Coefficient, h_c for Natural Convection; and (b) Fin Effectiveness.*

In the case of forced convection, the convective heat transfer coefficient, fin efficiency is found higher for Concentric Staggered Annular Fin than the Staggered Rectangular Fin (Fig 24). The convective heat transfer coefficient and efficiency of Concentric Staggered Annular Fin are increased with increasing of free stream air velocity and up to a fin base-plate temperature of 80°C, and then it becomes constant. For both free stream velocity of 1.5 m/s and 2.5 m/s, the Concentric Staggered Annular Fin is performed better than the Staggered Rectangular Fin in term of heat transfer and fin efficiency (Fig. 24). The fin effectiveness of Concentric Staggered Annular Fin is also found slightly higher than the Staggered Rectangular Fin at free stream velocity of 2.5 m/s, which is shown in Fig 25 (b).

Due to some unavoidable reasons and difficulties, the temperature distributions of the fins for all base-plate temperatures are varied from the theoretical distribution. First, the fin segments are welded to the fin base-plate instead of making the whole fins the from a single piece solid material, as a result, an air gap remained between the fin segments and the base-plate; which couldn't be eliminated totally. Second, the heater is fabricated by using locally available components/materials and not able to maintain the constant presetting temperature perfectly during operation. As a result, the base-plate temperature of the fin has fluctuated a little bit from the presetting value. Third, 3.0 mm thick aluminum plate is used to fabricating the fin, base-plate, etc. However, the actual alloy compositions of the 3.0 mm aluminum plate are unknown, and an average value of thermal conductivity of aluminum is used in the analysis which might be varied from the actual value. Fourth, the wind tunnel used in the experiment is not able to maintain a

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constant air velocity perfectly. As a result, the free stream air velocity has fluctuated a little bit from the presetting value. However, the results of the experimental study presented in this paper could be improved further by minimizing those difficulties.

CONCLUSION

There is numerous application of fin such as cooling of electronic equipment, engines, hot spots of space vehicles such as an airplane, missiles, rockets, space shuttles, etc. Due to the advancement of material science, a growing concept of using micro-scale objects, light-weight structure, and space vehicle with higher Mach number, the mechanisms of dissipation of the heat is becoming most challenging. The following assumptions are made from this work.

- The design and analysis of the Concentric Staggered Annular Fins with radial outlets presented in this paper could be used to dissipating of heat by incorporating/adopting the free stream air velocity parallel to the fin surfaces instead of flowing parallel to the fin-base plate, which could be implemented in high-speed space vehicles.
- The performance of the Concentric Staggered Annular and Staggered Rectangular Fin is evaluated for forced convection of 1.5 m/s and 2.5 m/s and also for natural convection. It is found that in the case of forced convection, the Concentric Staggered Annular Fin performed better than the Staggered Rectangular Fin. The Concentric Staggered Annular Fin will be performed better at the higher free stream air velocity.
- The performance of the Concentric Staggered Annular and Staggered Rectangular Fins could be examined further for higher free stream air velocity i.e. Mach number >1 with considering the compressibility of the fluid.
- The result presented in this paper could be improved by minimizing the difficulties which are faced during the experimental study.

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MOMENCLATURE

T_∞	Free stream Temperature of air
T_0	Temperature of the base-plate of fin
T_{out}	Temperature of air at outlet of the wind tunnel
T_{in}	Temperature of air at inlet of the wind tunnel
h_c	Convective heat transfer coefficient
Q	Heat Transfer from the fin
m	Mass flow rate
L	Length of fin
R_e	Reynolds number, $R_e = \frac{V\rho L}{\mu}$
C_p	Specific heat of air at constant pressure
K	Thermal conductivity of fin material
N_u	Nusselt Number, $N_u = \frac{h\,L}{k}$
P_r	Prandtl number, $P_r = \frac{\mu\,C_p}{K}$
R_a	Rayleigh number, $R_a = \frac{g\,\beta\,(T_2-T_0)\,L^3}{\nu\,\alpha}$
G_r	Grashof number, $G_r = \frac{g\,\beta\,(T_0-T_\infty)\cdot L^3}{\nu^2}$
μ	Absolute viscosity of air
ρ	Density of air
ν	Kinematic viscosity of air $\left(\nu = \frac{\mu}{\rho}\right)$
η_{eff}	Efficiency of the fin
ϵ_{eff}	Effectiveness of the fin
β	Thermal expansion coefficient
α	Thermal diffusivity

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