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Numerical Investigation of Natural Convection Heat Transfer from Horizontal Pin-Fin with Annular Fins

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Abstract—*Natural convection heat transfer from an annular fins attached to a horizontal pin-fin has studied and compared with conventional pin-fin of same material, numerically by varying the Rayleigh number (Ra) in laminar flow conditions. The computations were carried out by varying fin spacing to fin thickness ratio (S/t) in the range of 1.6, 2.4, 3.5, and 5.3 respectively. In the present study, numerical simulations were conducted based on Navier-Stokes equation along with the energy equation on horizontal pin-fin with annular fins of optimal thickness using the algebraic multi-grid solver of FLUENT 15. Optimization study of the conjugate heat transfer characteristics has carried out to find the best S/t ratio for maximum heat transfer in laminar flow. The effect of parameters like Ra, Nu, efficiency and effectiveness are analyzed.*

Keywords—*Natural Convection, Pin Fin, Heat Transfer, Optimization*

I. INTRODUCTION

[1]Andrew T. Morrison optimized the fin geometry of rectangular cross section at a constant fin spacing under natural convection conditions in steady state [2] Carried out an analysis of heat transfer by theoretical modelling on a heat sink.[3] Abdul Aziz focused on thermal performance of an annular fin with uniform thickness losing heat by convection [4] Raj Bahadur optimized PPS polymer pin fin heat sink on basis of optimum array, material, height and optimum center-to-center distance for better thermal performance. [5] Elsha deals with the heat transfer performance of solid and hollow/perforated pin fin heat sinks relying on natural convection furthermore the heat transfer experiments of solid pin fin array are also conducted for the sake of performance comparison and the importance of orientation of the heat. [6] Studied various flow models in natural convection heat transfer and fluid flow characteristics of vertical annular elliptical finned tube heat exchangers for various fin spacings, and compared fin efficiency of the annular circular and elliptical fins. [7]They carried an analysis in the direction of heat augmentation capacity of rectangular heat fin array to dissipate heat at faster rate. They have conducted a numerical and experimental investigation on plane fin by using Active and passive heat transfer techniques that are commonly employed for heat transfer augmentation in fluids and found that the disturbance in flow helps to accelerate the rate of heat transfer. [8] Compared analytically the thermal performances of optimized plate-fin and pin-fin heat sinks under fixed volume condition. A new correlation of the heat transfer coefficient for pin-fin heat sinks was proposed and validated experimentally for the optimization.[9] Chi-Yuan Lai theoretically studied the thermal performance of annular fin heat sink by considering the dimensionless parameter and heat transfer coefficient ratios, to find the optimum outer radius and number of annular fins.[10]They, conducted an experimental investigation to quantify and compare the natural convection heat transfer enhancement of perforated fin array with various fin spacing, perforation angle, perforation diameter, pitch of perforation and heater inputs. They also establishes optimized fin setup for various parameters of fin geometry and its effect on heat transfer results.

[11],[12]. Optimized the conjugate heat transfer characteristics by considering the fin spacing and fin-to-tube diameter ratio as basic parameters for maximum heat transfer in a heat exchanger along vertical and horizontal respectively. In this paper an annular fin was designed by considering fin material as a constraint. The computations were carried out by varying fin spacing to fin thickness ratio (S/t) in the range of 1.6, 2.4, 3.5, and 5.3 respectively.

Nomenclature			
Ra	Rayleigh Number	h	Heat transfer co-efficient (W/m^2K)
g	Acceleration due to gravity (m/s^2)	A_s	Total surface area of plain fin (m^2)
β	Coefficient of thermal expansion	N	Number of fins in an array
T_w	Wall Temperature (^0C)	A_{fin}	Annular pin fin area (m^2)
T_∞	Ambient Temperature (^0C)	A_b	Bare tube area (m)
d	Diameter of Annular pin fin in (mm)	Nu	Nusselt number
D_m	Diameter of plain fin (mm)	k	Thermal conductivity (W/mK)
Q	Rate of Heat transfer (W)	η_{fin}	Efficiency of fin (%)
Q_{actual}	Actual heat transfer (W)	ν	Kinematic viscosity (m^2/s)
Q_{max}	Maximum heat transfer (W)	α	Thermal diffusivity (m^2/s)
S	Space between fin in (mm)	t	Thickness (mm)

Table 1:

II. MATHEMATICAL MODELLING

The governing differential equation for steady, laminar, constant property, three dimensional flow with incompressible ideal gas assumption is given by

A. *Continuity equation:*

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \quad (1)$$

B. Momentum Equation

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\left(\nabla^2 u\right) \quad (2)$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\nabla^2 v\right)-\rho g \quad (3)$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\nabla^2 w\right) \quad (4)$$

C. Energy Equation

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}=\alpha\left(\nabla^2 v\right) \quad (5)$$

At the pressure and temperature of the surroundings

$$P=P_{atm} \quad , T=T_\infty$$

Energy equation for conduction in solid

$$\nabla^2 T=0$$

III. Important Parameters

The surrounding fluid is air with incompressible ideal gas assumption with the fluid properties measured at film temperature $T_{film}=\left(T_w+T_\infty\right)/2$ where T_∞ the ambient temperature. The results will henceforth be represented using the following non-dimensional parameters.

Rayleigh number based on the tube diameter can be calculated

From the following relation

$$\mathrm{Ra}=\frac{g\beta\left(T_w-T_\infty\right)D_m{}^3}{\nu\alpha} \quad (6)$$

Where D is the diameter of the Plain pin fin and d is the diameter of annular pin fin.

As per Newton's law of cooling, the total convective

Heat transfer from the fins and the tube can be written as:

$$Q=hA_s\left(T_w-T_\infty\right) \quad (7)$$

Where the average heat transfer coefficient *(h)* is based on the total surface area (A_s). This includes the fin area (A_{fin}) and the bare tube area (A_b). If there are N fins in the array, total surface area can be given by:

$$A=NA_{fin}+A_b \quad (8)$$

The total heat transfer rate, Q is obtained from the numerical computations. The average heat transfer coefficient, h can be calculated using Eq. (9).

$$h=Q/A_s\left(T_w-T_\infty\right) \quad (9)$$

The experimental Nusselt number can be calculated as

$$Nu=\frac{QD_m}{A_s\left(T_w-T_\infty\right)k} \quad (10)$$

Where $D_m = (Di+Do)/2$

The theoretical Nusselt number can be calculated as:

$$Nu=0.53\left(Gr\,\mathrm{Pr}^{1/4}\right) \quad (11)$$

The fin efficiency (η_{fin}) of a fin can be defined as

$$\eta_{fin}=\frac{Q_{actual}}{Q_{\max}} \quad (12)$$

$$\eta_{fin}=\frac{Q}{A_s h\left(T_b-T_\infty\right)} \quad (13)$$

IV. Experimental Set Up

Fig. 1: Experimental Set Up

A Pin fin apparatus is as shown in Fig 1. All experimentations were conducted on natural convection considerations, the models were placed at which the heating element was located in a duct as shown, and Digital wattmeter has been provided to measure power input to the heater. Heat input can be vary by regulator.6 k-type Thermocouples are provided to measure the average surface temperature of the models. A multichannel temperature indicator has been provided to monitor different temperature points.

A pin-fin of diameter 15mm and length 193mm was modified to annular pin-fin of same weight having 12mm, 18mm as inner and outer diameter respectively. Ten annular projections were provided by keeping the length as constant.

Space (mm)	Thickness (mm)	S/t
10	6	1.6
12	5	2.4
14	4	3.5
16	3	5.3

Table 1: Variation of Space with Respect to Thickness of Annular Pin-Fin

V. Experimental Procedure

- Switch on the mains and console after ensuring the given model has fitted in the duct.
- Open the windows provided on the top and bottom of the duct for conducting experiment in Natural convection.
- Switch on the heater and adjust the power input to approx. 50 Watts.
- After conducting experiment in natural convection mode, Increase the power supplied to the heater as to maintain the same temperature before starting the blower.
- After attaining steady state condition, note down the temperature readings.

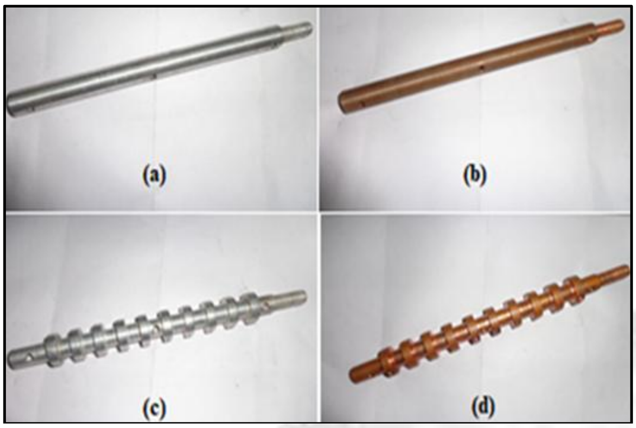


Fig.2: (A) Aluminium Pin-Fin, (B) Copper Pin-Fin(C) Aluminium Annular Pin-Fin, (D) Copper Annular Pin-Fin

VI. RESULTS& DISCUSSION

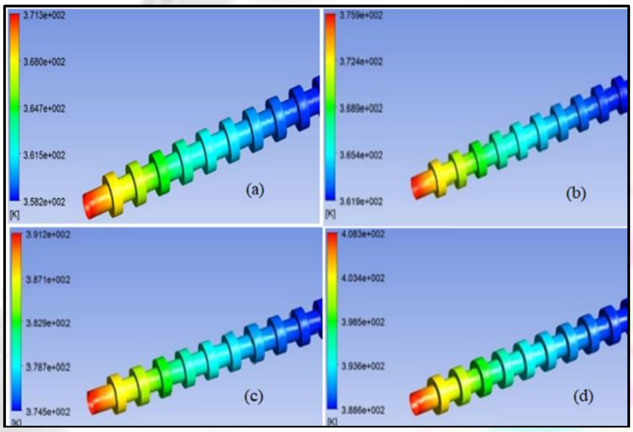


Fig.3: Temperature Distribution along the Length of the Aluminium Annular Pin-Fin at Different Heat Inputs(A) 20W, (B) 25W, (C) 30W, (D) 35W

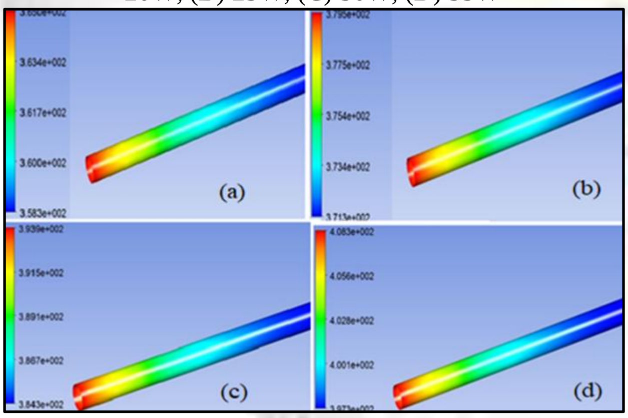


Fig.4: Temperature Distribution along the Length of The Aluminium Pin-Fin At Different Heat Inputs At (A) 20W, (B) 25W, (C) 30W, (D) 35W

Fig. 3 and Fig. 4 depict the noticeable distributing patterns of temperature along the annular and pin-fins at four different heat inputs. It was observed from the contours, the temperature distribution is more uniform in annular pin-fin when compare with pin-fin. The result shows at 1.6 space to thickness ratio annular pin-fin and pin-fin. Even though copper has more thermal conductivity than aluminium, the less density, better thermal conductivity and abundance of aluminum attracts the engineers to design variety of heat sinks economically. Hence the simulation shown in Fig.3 and Fig.4 represents the temperature distribution along the length of Aluminium annular pin fin and Aluminium pin fin.

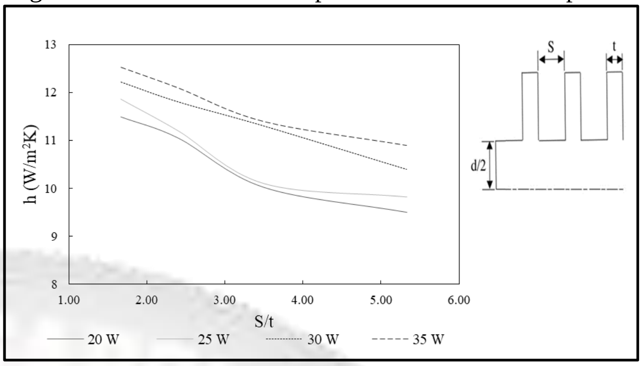
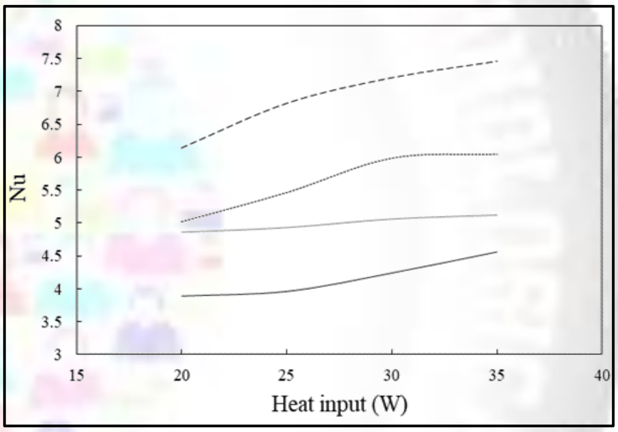
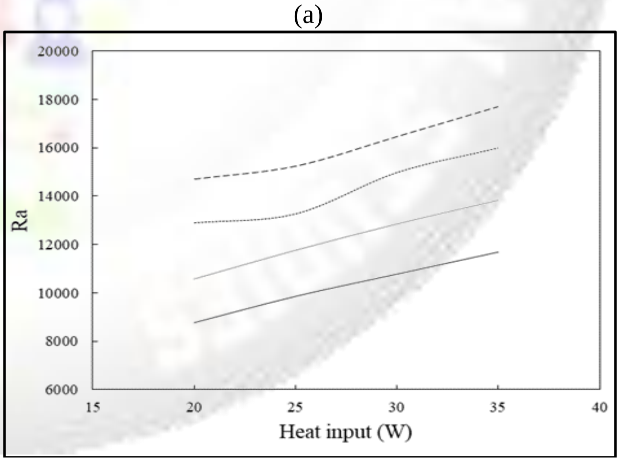


Fig. 5: Average Heat Transfer Co-Efficient as a Function of S/t Ratios at Different Heat Inputs for Aluminium Annular Pin-Fin

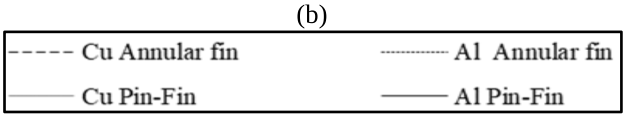
Average heat transfer co-efficient as a function of S/t ratios at different heat inputs were plotted for Aluminium annular pin- fin as shown in Fig.5. It was observed that the maximum heat transfer co-efficient occurs at 1.6(S/t) space to thickness ratio for various heat inputs. The similar trend was observed in heat transfer co-efficient with increasing S/t ratio.

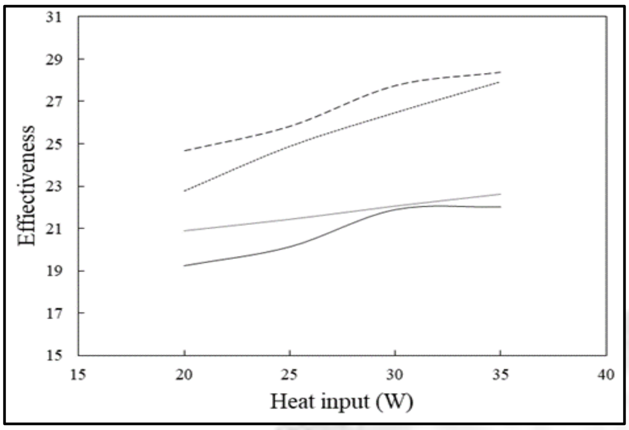


(a)

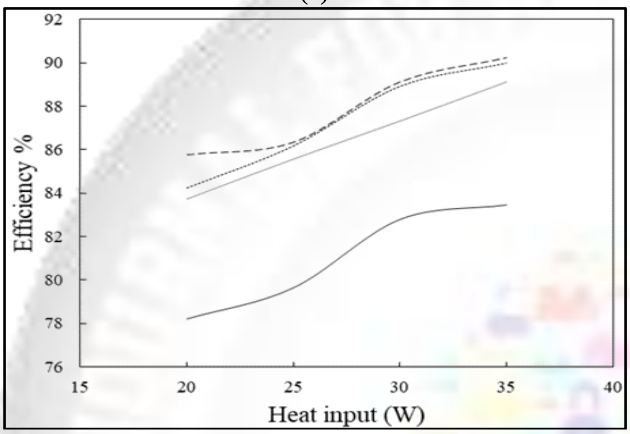


(b)





(c)



(d)

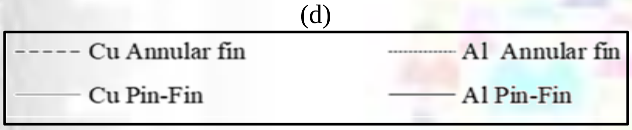
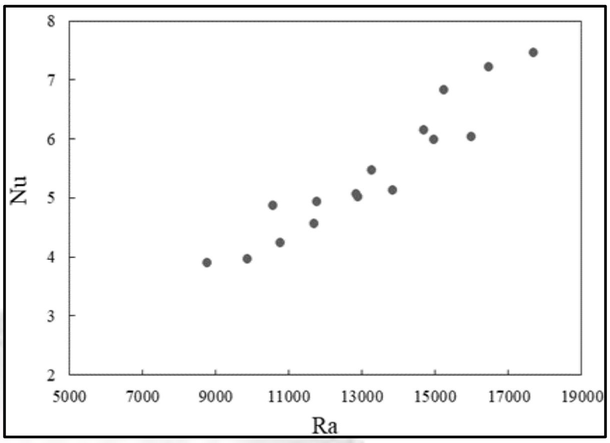
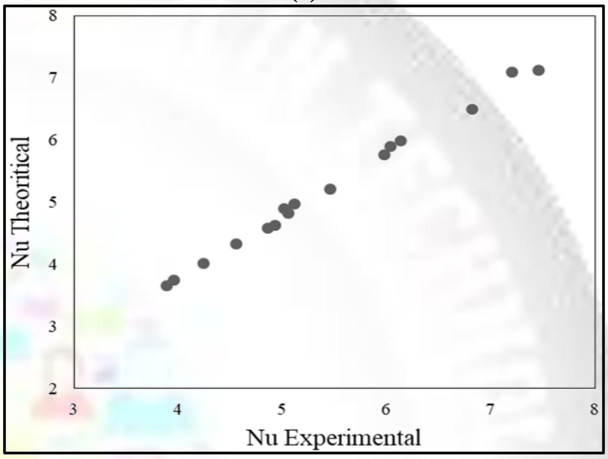


Fig. 6: Variation of (a) Nusselt Number, (b) Rayleigh number, (c) Effectiveness and (d) Efficiency as a Function of different Heat Inputs

Variation of Nusselt number, Rayleigh number, Effectiveness and Efficiency as a function of different heat inputs were plotted for pin fin and annular pin- fin of copper and aluminium material as shown in Fig.6. Since copper has better thermal conductivity the copper annular pin fin shows a significant thermal performance when compare with other pin-fins. As the Nusselt number rises as a function of heat input, at 1.6 S/t ratio itwas observed the similar trend in Rayleigh number, Effectiveness and Efficiency. Though Copper has more thermal conductivity. The epic characteristics like, less density, cost, better thermal conductivity and more abundance of Aluminium, attracts the engineers to design variety of heat sinks. Fig 7(a), Fig 7(b) shows the linear and scattered distribution of Nu as a function of Rayleigh number and the distribution of Nu theoretical as a function of experimental.



(a)



(b)

Fig.7: Variationof (a) Rayleigh Number Vs Nusselt Number, (b) Nu Experimental Vs. Nu Theoretical

VII. CONCLUSIONS

Heat transfer enhancement as a primary motive, different computations were carried out by varying fin spacing to fin thickness ratio (S/t) in the range of 1.6, 2.4, 3.5, and 5.3 respectively, on annular pin-fin. While comparing annular pin-fin at 1.6 S/t ratio with conventional pin-fin, it shows better thermal performance and also found good agreement with the experimental results under Natural convection conditions. So the study concludes that the increase in space between the annulars, decreases the thickness of the exposed projections, which in turn leads to increase the internal resistance of the fin. The present paper constrained to 10 annular projections for the study and suggests adding material based on constructal (optimal) way shows better thermal performance for further studies.

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