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Natural Convection Heat Transfer Inside an air Filled Narrow Rectangular Enclosure with Different Pin Fin Array Arrangement

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Abstract—*An experimental investigation on natural convective heat transfer inside a narrow rectangular enclosure has been performed. The bottom surface of enclosure is heated and the top surface is cooled while the rest of the surfaces are isothermal. The aim is to analyze free convection within an air filled enclosure having horizontal orientation. Experiments have been carried for two different cases: inline pin fin array and staggered pin fin array respectively. In total, 28 experiments were carried out to study the effect of fin spacing and Rayleigh number on Nusselt number. The work has been accomplished in a rectangular cavity of aspect ratio 0.15625. The Rayleigh number (Ra) and fin spacing (S) for experiments were in the range 278246 ≤ Ra ≤ 657361 and 25 ≤ S ≤ 100 respectively. It has been observed that Nusselt number is a strong function of Rayleigh number and fin spacing. Heat transfer is more in case of staggered pin fin arrangement as compared to inline pin fin arrangement. The effect of several parameters has been individually investigated and empirical correlations have been proposed for Nusselt number for both the cases.*

Keywords: *Natural convection, rectangular enclosure, pin fins.*

1. INTRODUCTION

Buoyancy driven natural convection heat transfer in cavities has become one of the significant areas of research due to its presence in various engineering applications such as nuclear reactor cooling, building insulation, automobiles and cooling of electronics. Due to overheating problems in most of the electronic devices, there is a need of efficient heat removal to maintain proper functioning of devices. The most common and potent technique of heat removal or enhancing heat transfer rate is to use fins. The extended surfaces i.e. fins increase the surface area available for heat transfer thereby increasing the heat transfer rate during operation. It is also economical and accessible. The enhancement of heat transfer depends on orientation of fins and its various geometric parameters. Numerous experimental and numerical investigation have been carried out regarding free convection in a rectangular cavity.

Bocu and Altac [1] carried out numerical investigation for laminar natural convection heat transfer in three dimensional rectangular enclosures with pin fins attached to hot surface. Nusselt number ratio (*NNR*) as a function of fin length, number of fins and aspect ratio (*AR*) was reported. Mobedi and Yuncu [2] carried out three dimensional numerical investigation of natural convection from short horizontal rectangular fin geometry on a horizontal base. The fin height and fin length were varied from 0.25 to 7 and 2 to 20-fin spacing respectively. Natural convection in horizontal and vertical closed narrow enclosures with heated rectangular finned base plate was reported by Nada [3] at wide range of Rayleigh number (*Ra*) for different fin length and fin spacing through experimental investigation. Dwivedi and Das [4] experimentally investigated the performance of triangular fins in a rectangular enclosure at wide range of Rayleigh number, fin spacing and fin height. A correlation was developed to analyze the performance of system under study. Silva and Gosselin [5] investigated laminar natural convection in a three dimensional cubical enclosure with rectangular fins attached to hot surface. They reported that fin aspect ratio did not have significant role for an enclosure with large volume fraction fin. Nogueira et al. [6] numerically investigated natural convection in a rectangular enclosure heated on one side and cooled on opposite side. Temperatures of hot and cold wall were kept constant. They reported the effect of Rayleigh number and aspect ratio on heat transfer and flow behavior in an enclosure. Nusselt number (*Nu*) was found to be strongly dependent on length to height ratio of enclosure and it increased with increase in length to height ratio of enclosure. Sparrow and Vemuri [7] found out effects of orientation on the radiation/convection from pin fins in an open base plate array. Numerical experiments have been carried out to investigate natural convection heat transfer and fluid flow characteristics by Arquis and Rady [8]. Al-Jamal and Khashashneh [9] performed an experimental analysis regarding heat transfer through triangular fin arrays at constant heat flux. They described Nusselt number as a function of maximum Reynolds number at Prandtl number 0.7. Corcione [10] carried out numerical investigation of laminar natural convection heat transfer in a rectangular cavity with different boundary conditions at the sidewalls. The cavity is heated from below and cooled from above. Comparison of heat transfer rates for various configurations was reported and results were expressed using correlation equations. Ganzarolli and Milanez [11] carried out numerical study of free convection in rectangular enclosure which is heated from bottom and sides are cooled symmetrically. Variation in Ra depending on height of enclosure was from 10^3 to 10^7 and AR from 1 to 9. Pr values were 0.7 and 7.0. Heat flux was uniform and wall temperature was also uniform at the boundaries. Results indicated that Pr had little influence on circulation of flow and heat transfer inside the enclosure.

In spite of literature available for natural convection in enclosures, focus has been mostly on rectangular fins. Very few experimental studies are available regarding natural convection rectangular enclosures with pin fins. Hence the aim of the investigation is to have comparative analysis of natural convection inside narrow rectangular between two different arrangements i.e. staggered pin fin arrangement and inline pin fin arrangement. The staggered pin fin arrangement is closer to practical applications. Influence of various parameters such as fin spacing and Rayleigh number on heat transfer rate have been analyzed. Correlations have been developed for both the cases and predicted results are compared with experimental results to obtain the deviation associated with Nusselt number for both the cases.

2. EXPERIMENTAL SETUP AND PROCEDURE

The experimental setup is shown in Fig. 1. It consists of three sections: heating section, cooling section and test section. The heating section consists of a nickel chrome wired panel heater having dimensions of 320 mm × 200 mm, kept on an asbestos plate 3 mm thick and insulated from all the sides with mica sheet to maintain the constant heat flux arrangement. In order to minimize the heat loss from the enclosure, heating section is kept inside a rectangular wooden block and insulated from all the sides by 50 mm thick glass wool. The test segment consists of pin fin array (fin thickness = 10 mm, height = 25 mm). The internal dimensions of enclosure are 320 mm × 200 mm × 50 mm. Air is used as working fluid.

Window glass is used for enclosure side walls with thermal conductivity of 0.96 W/mK. The base plate with fins acts as the bottom surface of the enclosure and the bottom portion of the cooling tank acts as top surface of enclosure. The pin fin array is heated by the panel heater at the bottom of base plate and cooled simultaneously at the enclosure top surface by the cooling arrangement. The cooling section contains cold water circulating at a constant mass flow rate (m = 0.003204 kg/s) using a flow control valve in cooling tank. The base plates and fins are made up of pure aluminium. Fin arrays are shown in Fig. 2 and Fig. 3 respectively.

In order to control the power input, the panel heater is connected with a DC power supply using an auto-transformer to control the heat flux input and a Rayleigh number corresponding to it. Output data in terms of resistance and voltage is obtained with the help of digital multimeter having accuracy of ±0.4%. Eleven K-type Teflon coated chromyl alumel thermocouples are used for measuring temperature at various points with accuracy of ±0.25^{0}C. Six thermocouples are symmetrically placed to measure the base plate temperature and four thermocouples are also symmetrically placed for measuring the cold wall temperature. One thermocouple is utilized for measuring the outer temperature of enclosure side wall. The experiment for each configuration was carried out for about 3 hours so that the condition of steady state was achieved. When the steady state was reached, all the thermocouples readings were recorded with the help of a digital temperature indicator and a selector switch. To avoid disturbances caused by air currents, experiments were conducted in a large room without windows.

Table 1: Different cases for experiments

Sr. no.	Fin spacing (mm)	Number of fins	Type of fin array
1	25	52	inline
2	50	28	inline
3	100	16	inline
4	25	26	staggered
5	50	14	staggered
6	100	8	staggered
7	BARE PLATE	-	-

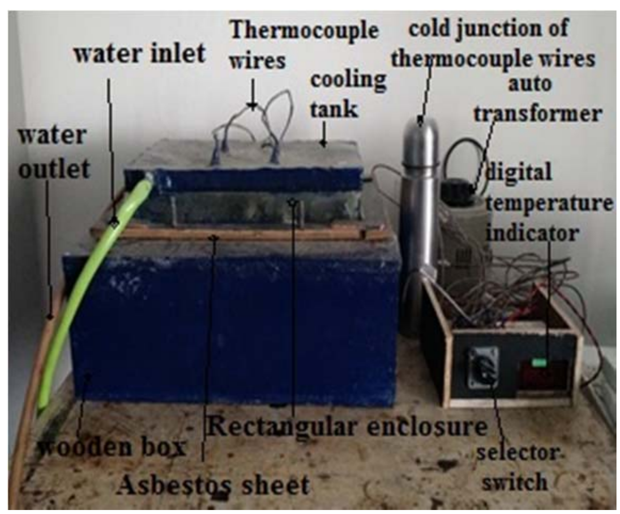

Fig. 1: Experimental setup.

Fig. 2: Staggered fin array of 50 mm spacing

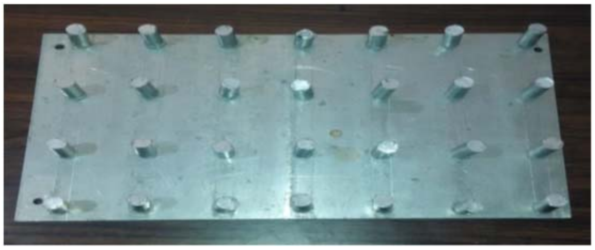

Fig. 3: Inline fin array of 50 mm spacing

3. CALCULATION PROCEDURE

Rayleigh number is obtained using relation

$$Ra = \frac{H^3 g\beta(T_h - T_c)Pr}{\upsilon^2} \tag{1}$$

All the fluid properties are calculated at film temperature $T_f = (T_h + T_c)/2$. The net heat transfer and conduction and radiation heat transfer [14] are calculated as,

$$Q_{net} = \frac{V^2}{R} = Q_{cond} + Q_{conv} + Q_{rad} \tag{2}$$

$$Q_{cond} = h_a A_w (T_w - T_a) \tag{3}$$

$$Q_{rad} = F' \varepsilon' A_r \sigma (T_h^4 - T_c^4) \tag{4}$$

Convection heat transfer is obtained using equation (2) and heat transfer coefficient was found using,

$$h = \frac{Q_{conv}}{A_c (T_h - T_c)} \tag{5}$$

Average Nusselt number was obtained as,

$$Nu = \frac{hH}{K_a} \tag{6}$$

The fin effectiveness was calculated as,

$$\mathcal{E} = \frac{Nu_{with\,fin}}{Nu_{bare\,plate}} \tag{7}$$

4. EMPIRICAL CORRELATIONS

From the analysis, it is found that Nu varies with Ra and dimensionless fin spacing (S/H). Two different correlations have been developed using the experimental data to represent Nu in the form of Ra and S/H.

4.1 For inline pin fin array:

$$Nu = 1.75 \times 10^9 \times \left(\frac{S}{H}\right)^{0.044} \times \exp\left(-0.2368\left(ln\left(\frac{S}{H}\right)\right)^2\right) \times (Ra)^{-3.2828} \times \exp(0.1362(ln(Ra))^2) \tag{8}$$

4.2 For staggered pin fin array:

$$Nu = 2.18 \times 10^9 \times \left(\frac{S}{H}\right)^{0.0399} \times \exp\left(-0.2207\left(ln\left(\frac{S}{H}\right)\right)^2\right) \times (Ra)^{-3.2912} \times \exp(0.1358(ln(Ra))^2) \tag{9}$$

Regression analysis of the experimental data was carried out to find the expression for Nu. The predicted values of Nu were calculated using the developed correlation and compared with the experimental results. Good agreement was obtained with maximum deviation of ±8% The deviation of Nusselt number was less in case of staggered arrangement as compared to inline arrangement.

5. RESULTS AND DISCUSSION

The experiments were carried out to analyze the effect of pin fin array on natural convective heat transfer in both the cases i.e. inline pin fin array and staggered pin fin array. The uncertainty involved in various parameters was found and depending upon the uncertainties in various parameters, the uncertainty in Nusselt number was obtained.

The Nusselt number (Nu) variation with fin spacing (S) is due to the dependence of heat transfer rate on the effects of following factors: First is increase in heat transfer rate due to increased effective surface area with increase in number of fins and second is reduction in heat transfer because of hindrance caused by the increase in number of fins to the flow of fluid inside the enclosure. Thus with decrease in fin spacing initially leads to an increased heat transfer rate due to dominance of surface area effect over the flow hindrance effect while with further reduction in fin spacing leads to reduced heat transfer rate since the flow hindrance effect now dominates the surface area effect.

The Nusselt number increases continuously with Rayleigh number for all configurations and at a given fin height as shown in Fig. 2. This result may be credited to the increment in the buoyancy force which increases flow driving force causing the augmented flow intensity due to enhanced Rayleigh number. The mixing within the air layer increases due to the increase of turbulence of vortices by increasing Ra and thereby resulting in enhanced heat transfer rate.

With decrease in fin spacing, fin effectiveness initially increases upto a certain fin spacing value, then it starts decreasing with further decrease in spacing between fins. This is because of Nusselt number variation with fin spacing value.

The fin effectiveness keeps on decreasing continuously with increase in Rayleigh number as shown in Fig. 3. This result is due to the fact that fin effectiveness is a relative measure of performance of system in presence of fins to the performance of system with bare base plate. Although, an increment in Rayleigh number leads to augmented heat transfer rate for both the configuration i.e. system with bare base plate as well as system with fins, heat transfer enhancement rate in case of fins is less with respect to that of bare plate. This is due to the formation of boundary layer in case of fins and results in reduction of fin effectiveness with increment in Rayleigh number.

The staggered pin fin array leads to larger convection cells resulting in increased heat transfer rate as compared to inline pin fin array for all tested configurations. For both the cases, with increase in fin density, convection cells become smaller and results in reduction in heat transfer rate.

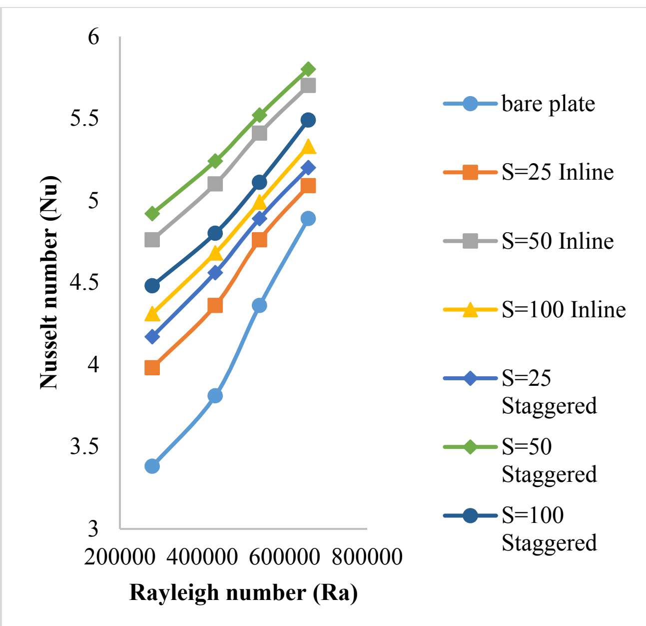

Fig. 4: Variation of Nu vs Ra.

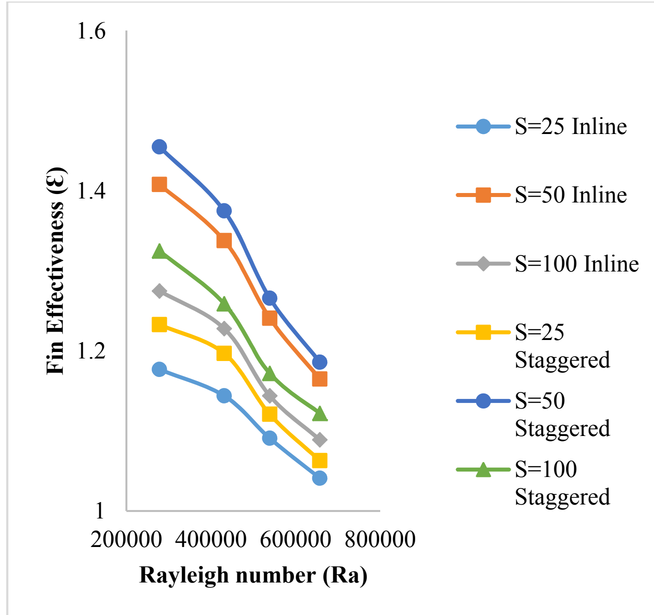

Fig. 5: Variation of Ɛ vs Ra.

6. CONCLUSION

The results show effects of parameters like fin spacing and Rayleigh number on heat transfer rate for a single aspect ratio enclosure, fin height and fin thickness. The following conclusions are obtained from the experimentation:

a) The Nusselt number increases with increase in Rayleigh number for all the cases.

b) Fin effectiveness decreases with increase in Rayleigh number for all tested configurations.

c) Fin effectiveness and Nusselt number increases with increase in fin spacing upto a certain fin spacing value then it starts decreasing with further increment in fin spacing.

d) Staggered pin fin array yields better heat transfer rates for all the cases as compared to inline pin fin array.

e) Deviation in values obtained from correlation was observed more for inline pin fin array as compared to staggered pin fin array.

7. NOMENCLATURE

AR	Aspect ratio
A_c	Effective surface area for convection (m^2)
A_r	Effective surface area for radiation (m^2)
A_w	Cross-sectional area of enclosure side walls (m^2)
F'	View factor
g	Acceleration due to gravity (m/s^2)
H	Enclosure height (mm)
h	Average heat transfer coefficient (W/m^2K)
h_a	Average heat transfer coefficient of air (W/m^2K)
K_a	Thermal conductivity of air (W/mK)
K_w	Glass walls thermal conductivity (W/mK)
L	Fin height (mm)
m	Mass flow rate (kg/s)
NNR	Nusselt number ratio
Nu	Nusselt number
P	Power
Pr	Prandtl number
Q_{net}	Net heat transfer (W)
Q_{cond}	Conduction heat transfer (W)
Q_{rad}	Radiation heat transfer (W)
Q_{conv}	Convection heat transfer (W)
R	Resistance of the panel heater (Ω)
Ra	Rayleigh number
S	Fin spacing (mm)
T_a	Ambient air temperature (^0C)
T_c	Temperature of cold surface (^0C)
T_f	Film temperature of working fluid (^0C)
T_h	Temperature of heated plate (^0C)
T_w	Surface temperature of enclosure wall (^0C)
V	Voltage applied (V)

7.1 Greek symbols

β	Coefficient of volume expansion (K^{-1})
ε'	Emissivity
υ	Kinematic viscosity (m^2/s)
σ	Stefan Boltzmann constant (W/m^2K^4)
$\mathcal{E}$	Fin effectiveness

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