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Natural Convection Heat Transfer inside a Narrow Triangular Enclosure with Rectangular Staggered Finned Base Plate: An Empirical Correlation

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ABSTRACT: An experimental study of steady buoyancy-driven convective heat transfer inside a narrow triangular enclosure between hot base plate and inclined cold wall while others wall being isothermal has been performed. Aim of this study is to analyze free convection heat transfer in an air filled triangular cavity with rectangular staggered finned base plate. Experiments have been carried out in two orientations i.e. horizontal and vertical. In total 104 experiments were run to observe the effect of fin spacing, fin height, fin thickness and Rayleigh number on the Nusselt number. The work has been carried out in a triangular cavity of aspect ratio 0.3175. The Rayleigh number and Prandtl Number for different experiments were in the range 1779341 ≤ Ra ≤ 5559546 and 0.713-0.698 respectively. It has been revealed that Nusselt number is strong function of fin spacing and Rayleigh number while fin distance has moderate effect. The effect of several factors has been investigated individually and empirical correlations have been developed for Nusselt number in both orientations.

Keywords: Triangular enclosure; interrupted rectangular fins and natural convection.

INTRODUCTION: The need of effective cooling in modern devices has emphasized us to use effective cooling method and the most proposed way to increase heat transfer is either attaching the extended surfaces (fins) or employing some forced method to remove heat. Although, it has been found that attachment of fins to hot surfaces does not always help in enhancement of heat transfer rate. So, it becomes necessary to optimize the control parameters to maximize the heat transfer rate. The applications of natural convection inside triangular enclosure are in solar water heaters, nuclear reactor cooling, electronics devices, energy storage, cryogenics, building insulation, attic shape houses, geophysical systems and in passive cooling systems. In natural convection heat transfer convection cells are formed theses are known as Benard cells. The formation of Benard cells takes place at critical Rayleigh number when the flow characteristic of confined fluid takes place between laminar and turbulent flow.

Many researchers have conducted numerical as well as experimental investigation of natural convection inside the triangular enclosures. A comprehensive review has been represented by O.M. Kamiyo et.al [1] on natural convection inside the triangular enclosure. They suggested that Natural convection in triangular enclosures is an important problem. It displays well the generic attributes of this class of convection, with its dependence on enclosure geometry, orientation and thermal boundary conditions. It is particularly rich in its variety of flow regimes and thermal fields as well as having significant practical application. They also discussed the effect of pitch angle on the heat transfer inside the enclosure. E. Fuad Kent et.al. [11] conducted a study on laminar natural convection in right triangular enclosures and they found effects of flow patterns on the heat transfer. E.M. SPARROW et.al. [7] found out effects of orientation on the radiation/convection from pin fins in an open base plate array. Xu Xu et.al. [5] carried out a numerical analysis of laminar natural convective heat transfer around a horizontal cylinder inside a concentric air-filled triangular enclosure. They gave effects of Rayleigh number and aspect ratio on the Nusselt number. The buoyancy effect was modelled by applying the Boussinesq approximation of density to the momentum equation and the governing equations were iteratively solved using the control volume approach. K. Hooman [2] conducted a theoretical Analysis of Natural Convection in an enclosure filled with disconnected conducting square solid blocks to find out effects of number of blocks and porosity on the Nusselt number. He also gave Nu-Ra correlations by least square method and calculated heat transfer by assuming the blocks as an electrical resistant. A. Bairi [3] also gave a correlation for design of industrial equipment. He conducted experimental and numerical study inside a tilted rectangular cavity also investigated effects of tilt angle. G. Nardini and M. Paroncini [10] conducted experiment for natural convection in a square cavity with discrete sources. Abhishek Jain [4] used cognitive computing to extend the range of Grashof number in numerical simulation of natural convection. Ram Satish Kaluri et.al. [6] analysed Bejan's heat lines analyses of natural convection heat transfer in a right triangular enclosure and found out effects of aspect ratio and thermal boundary conditions. A scaling analysis was performed for transient boundary layer established adjacent to an inclined flat plate following a ramp cooling boundary condition by Suvash C. Saha et.al. [8]. S. Kenjereš et.al. [9] studied Rayleigh-Benard convection numerically and experimentally. Work on the interrupted fins [12-14] was carried out. Mehran Ahmadi et.al. [13] studied numerically and experimentally natural convection from rectangular interrupted fins similar to an heat sink having rectangular fins. He suggested that the interruptions will increase the heat transfer rate by resetting/interrupting the thermal and hydrodynamic boundary layers.

Radiation heat transfer plays a key role in heat transfer from fins some authors reported including E.M. Sparrow [7] that the radiation heat transfer contributes between 25 to 40% of total heat transfer from fin arrays in naturally cooled heat sinks. Large number of computational study has been carried out in triangular enclosures, there is no generalized correlation to predict heat transfer in triangular enclosures. To the author's knowledge no previous studies had been conducted on natural convection heat transfer inside a triangular enclosure with rectangular staggered fins. Therefore comparison of present data was not possible from previous data. Although some studies on triangular enclosures of different aspect ratios without fin and a single heater placed inside the cavity instead of fins are available.

In spite of number of existing literature on topic of natural convection inside the enclosures our review indicates that the focus has been mostly on the continuous fin and pin fin, no in depth study has been performed to investigate the natural convection heat transfer from staggered fins inside an enclosure which is more close to practical applications. As such in this paper we aim at investigating the effect of fin spacing, fin distance and Ra on natural convection inside a triangular enclosure. A proper selection of fin spacing, fin distance, Ra and orientation can lead to higher heat transfer. Thus, two different correlations have been developed and predicted results were compared with experimental one to find out the deviations associated with correlations.

MATERIAL NAD METHODS: The set up as shown in figures 1(a) & 1(b) consists of a triangular enclosure of window glass of 3.5 mm thickness of internal dimensions 315×200×100 mm^3 and enclosed by a sloping 1 mm thickness mild steel water tank of dimensions 347×200×20 mm^3. A frame made up of plywood having 12.7 mm thickness is filled with glass wool above which the triangular enclosure setup was mounted. The frame was designed such that it can be positioned in two orientations i.e. vertical and horizontal. Effective thickness of glass wool was kept 100 mm and on other sides it was 50 mm.

The power input wire and thermocouple wires adjusted in the frame. It was made air tight to prevent possible heat loss from the frame. A heater of nickel-chrome wire of 1mm diameter was made up of dimensions 315×200 mm^2 to produce heat for finned base plate. This heater was place on mica board followed by asbestos board, glass-wool, galvanized steel box respectively to prevent the possible heat loss to the other direction. The number of turns of wire was kept high to produce heat uniformly over base plate. The base plate thermocouples were adjusted in such a way that these should not disturb the convection cells.

Table 1: Shows total 13 sets.

S. No.	SPACING S (mm)	DISTANCE D (mm)	THICKNESS T (mm)	NUMBER OF FINS
1	25	10	1	44
2	25	17	1	40
3	25	24	1	36
4	37	10	1	18
5	37	17	1	15
6	37	24	1	13
7	37	10	3	18
8	37	17	3	15
9	37	24	3	13
10	65	10	1	6
11	65	17	1	5
12	65	24	1	5
13	BARE PLATE	…..	…..	0

A total of six thermocouples were installed in three rows each having two. Similarly, total four thermocouples were inserted in the water tank to take the temperature of cold plate. Each wall of enclosure was having one thermocouple to take the temperature of walls. Two thermocouples were placed nearby to the enclosure to take the ambient temperature.

The heater was connected to an AC auto transformer and a voltage supply of 10, 20, 30, and 40 volts was maintained and measured by a Digital Multi-meter of accuracy ±0.8%. All the thermocouple readings were recorded manually using a digital temperature indicator having accuracy of ±0.1°C. The base plate was 3 mm thick pure aluminum sheet and fins of 1mm and 3mm thick sheet. The pitch of fins was kept two times of spacing in longitudinal and equal to spacing lateral direction. Total 13 set shown in table.1 were run to find out effects of various parameters on heat transfer.

The fins on the base plate were physically attached with the help of bolts through the drilled hole in both base plate and fins. Tight physical contact was ensured in each set up. All the set were produced manually in the workshop.

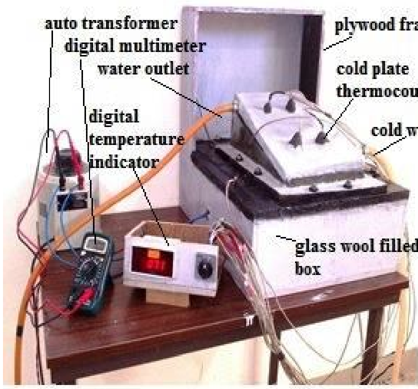

Figure 1(a): pictorial view of horizontal test bed.

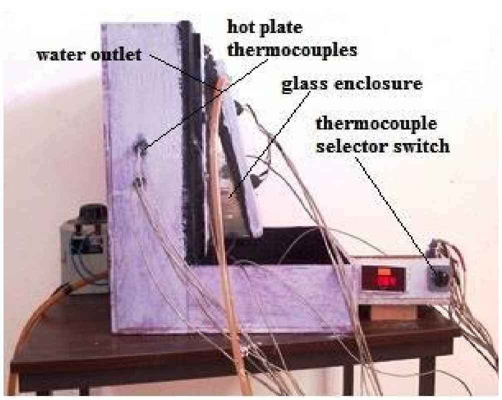

Figure 1(b): pictorial view of vertical test bed.

Formulation: Dimensionless Rayleigh number was found using relation:

$$Ra = \frac{g\beta\,[illegible]}{\alpha^2}\mathrm{Pr} \quad (1)$$

Heat transfer by natural convection was determined by subtracting radiation and conduction loss through the glass wall to the environment from heat generated by heater.

$$Q - \frac{V}{R} - Q_c + Q_r + Q_e \quad (2.a)$$

$$Q_c = Q - Q_r - Q_e \quad (2.b)$$

Heat transfer coefficient was found using the relation:

$$h = \frac{Q_c}{A(T\ [illegible]\ T)} \quad (3)$$

and this was used to find out the average Nusselt number using relation:

$$Nu = \frac{hL}{k} \quad (4)$$

with the help of above equations (1)-(4) average Nusselt number was found.

Empirical correlations: Two different correlations have been developed from the experimental data to give the $\overline{Nu}$ in the form of Ra, spacing S and distance D respectively for 1 mm thick fins. It has been found from the analysis that $\overline{Nu}$ varies with Ra, S and D. Hence, a correlation between dimensionless parameters has been developed for both orientations.

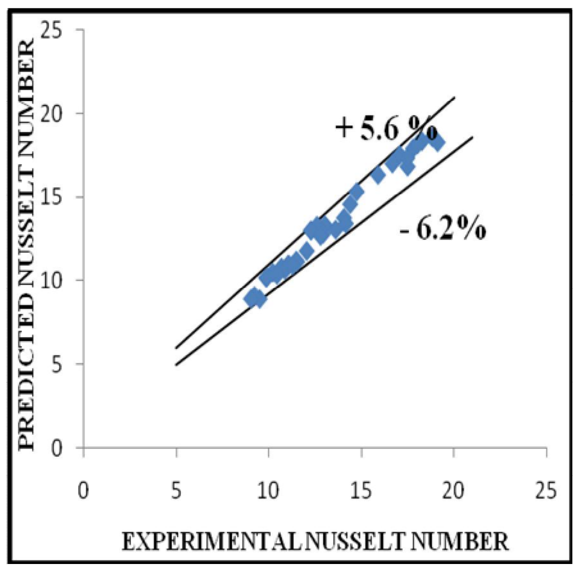

Figure 2(a): Plot of predicted nu vs experimental nu in vertical enclosure.

(a)Vertical orientation:

$$\overline{Nu} = 4.8728\ Ra^{0.1229}\left(\frac{-}{H}\right) \quad \left(\frac{-}{H}\right) \quad - 0.016839$$

(b) Horizontal orientation

$$\overline{Nu} = 0.6733\ Ra^{0.2341}\left(\frac{-}{H}\right) \quad \left(\frac{-}{H}\right) \quad - 0.1149$$

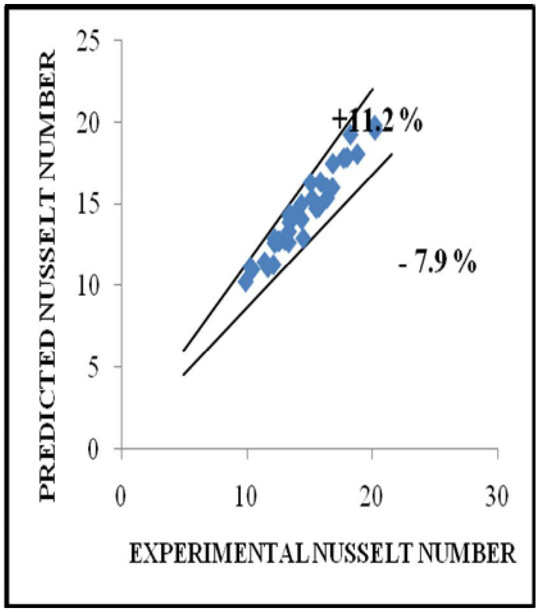

Figure 2(a): Plot of predicted nu vs experimental nu in horizontal enclosure.

The experimental data was fitted using regression analysis to find out an expression for $\overline{Nu}$. These are developed for the range of parameters 1779341 ≤ Ra ≤ 5559546, 0.25 ≤ S/H ≤ 0.65 and 0.1 ≤ D/H ≤ 0.25. The predicted values of Nusselt number were calculated using the developed correlation and were compared with the experimental result. The maximum percentage error was found 6.2% in vertical orientation and 11.18% in horizontal orientation. The predicted and experimental values were plotted in fig 2(a), 2(b) to show the distribution of deviations in the Nusselt number for both orientations.

RESULTS AND DISCUSSION: The experiments were performed to find out the effects of different fin arrays on the free convective heat transfer in both orientations i.e. horizontal and vertical. The uncertainty associated to all the parameters was found out and based on all parameters uncertainty, the uncertainty in Nusselt number was found for both the orientations i.e. vertical and horizontal orientation.

The variation of $\overline{Nu}$ with S can be attributed to the dependence of heat transfer rate in enclosed air layer upon the competition between the effects of following factors (1) decrease in heat transfer due to insertion of fins because maximum heat transfer occurs near to the intersection of base wall and inclined wall. The insertion of fins might disturb the convection cells (2) the increment or the decrement in the S or no. of fins causes the disturbance in the intensity of convection cells (3) increase or decrease in the number of convection cells. . Increasing the number of convection cells leads to higher heat transfer rate. With increment in spacing the resistant to flow of convection cells decreases resulting in higher heat transfer.

The staggered array of fins leads to smaller convections cells resulting in reduction of heat transfer. For both the orientations with increment in fin density the S decreases which causes smaller convection cells hence heat transfer rate decreases in the cavity. With increasing Ra the rate of heat transfer increases and variation of $\overline{Nu}$ with Ra is observed periodic in some cases. For vertical triangular enclosure without fins a flow field is characterized by a single large stable convection cell that rises along heated wall and descends along the cold wall and with insertion of fins that get disturbed resulting in the reduction of $\overline{Nu}$ or the heat transfer.

In horizontal orientation the difference is very less in $\overline{Nu}$ and in vertical orientation the variation is significant. It can be stated here that in vertical enclosure staggered fins are having significant effect on heat transfer because of stable convection cells. Due to more fin density the convection cells got disturbed contributing to reduction in heat transfer which we can't overcome by simply increasing surface area.

It can be observed in any case that the $\overline{Nu}$ increases with increase in Ra. This can be attributed to the increase of buoyancy force over the viscous force. Buoyancy force enhances the driving force of convection cells which increase the flow intensity of the convection cells which increases the heat transfer. Also increasing Ra increases the mixing within the air enclosed due to increased turbulence of vortices that result in higher heat transfer.

CONCLUSION: The results show effects of parameters like distance, spacing, Rayleigh number and thickness for a single aspect ratio enclosure. The uncertainty in all the parameters accounts in the measurement of the Nusselt number. Some important findings were pointed out during the experimentation-

a) The Nusselt number increases with increase in Rayleigh number. In some cases the variation was observed periodic.

b) The fins having thickness 1mm were more effective than the fins having thickness 3mm. So, thin fins are more suitable inside the enclosures.

c) The effect of fin distance was observed much in vertical orientation as compares to horizontal orientation.

d) Deviations in correlation of horizontal orientation were observed more in comparison to vertical orientation.

Nomenclature:

Ra	Rayleigh Number
$\overline{Nu}$	Average Nusselt Number
S	Fin Spacing
D	Fin distance from inclined wall
H	Enclosure Height
T_h	Hot plate temperature
T_c	Cold plate temperature
A	Area of base plate
h	Heat transfer coefficient
l	Characteristic length
k	Thermal conductivity of air
ν	Kinematic viscosity of air
β	Volume expansion coefficient
Gr	Grashof Number
Pr	Prandtl Number
T	Fin thickness
g	Acceleration due to gravity
Q	Heat Supplied by Heater
Q_c	Convection Heat Transfer
Q_r	Radiation Heat Transfer
Q_e	Conduction Heat Transfer
V	Voltage Supplied
R	Resistance

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