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EXPERIMENTAL ANALYSIS OF NATURAL CONVECTION HEAT TRANSFER OF HEATED TRIANGULAR FINNED BASE PLATE WITHIN A NARROW AIR FILLED RECTANGULAR ENCLOSURE

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A B S T R A C T

An experimental study of natural convection heat transfer within a rectangular enclosure from the heated triangular fin array has been performed. The present study reported that increasing Ra ($365214 \leq Ra \leq 683826$) and fin height the Nu increases continuously. While Nu increases initially up to an extreme value, then tends to decrease by decreasing in fin spacing. The fin effectiveness increases with increase in fin height and it increases initially up to a maximum value, then tends to decrease with a decrease in fin spacing. The correlations are developed for Nu and verified with experimental data.

INTRODUCTION

Natural convection phenomenon from the finned surface in narrow enclosures has attracted considerable interest of investigators due to its eminent involvement in various environmental problems and engineering applications. The heat transport phenomena in a fluid where the fluid motion derives from the interaction of a variation in density caused by temperature difference with a gravitational field. The common areas in several engineering and environmental as, ranging from nuclear reactors to domestic heating, cooling of electronics components and computer. It may be summarized as in solar flat plate collectors, cooling of microelectronics circuits and computing devices, fire control, metallurgical and automotive industries, etc. In the present scenario, ever-increasing demand for compact systems leads to sophisticated packing density causing privilege heat generation. It grades to enhance in temperature, which is generating serious overheating problems. The use of extended surfaces is the most prominent way to enhance the heat transfer rate as it is more appropriate as well as cost-effective. Finned surfaces in Natural convection heat transfer are commonly used in practice to enhance heat transfer and they often augment the rate of heat transfer from a surface several folds. The closely packed thin metal sheets attached to the heat source to increase the effective surface area for convection and thus the rate of convection heat transfer from the source is many times than the without finned source. A number of studies have been done numerically and experimentally by considering various parametric ranges, enclosure designs, fin and base plate materials, working fluids and thermal boundary condition. Oz top *et al.* [2] carried out an experimental and computational study to analyze the natural convection in an inclined non rectangular enclosure for a wide range of influencing parameters. It was concluded that for trapezoidal enclosure in the horizontal position, the increase in the Rayleigh number leads the scenes to the temperature and velocity profiles, while the viscous effects due to the presence of fluid walls were neglected as per conclusion of [3]. A numerical investigation of 2D air filled rectangular enclosures heated from below and cooled from above had given an idea about flow pattern transitions [4-6]. An abrupt decrease in the Nusselt number was concluded for the flow transition from one cell to two cells steady state. Gelfgat [7] carried out a complete 2D and 3D parametric numerical study of Rayleigh-Bénard instability in a rectangular enclosure. An experimental and numerical study of free convective heat transfer for square enclosure characterized by a discrete heater located on the lower wall and cooled from the lateral wall having air as a working fluid, was carried out by Krane and Jessee [8]. It was perceived that for $Ra<10^4$, the conductive heat transfer was prevailing while the convective heat transfer began for $Ra>10^4$ and

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mushroom profile of the isotherms came into the picture at Ra>10^5. The system performance was considerably affected by enclosure orientation [9-11] and this effect could not be neglected. Altac and Kurtul [12] concluded the heat transfer enhancement with increasing enclosure tilt angle up to 22.5° and then it started decreasing steadily for all Rayleigh number values. Varol *et al*. [13] observed the heat transfer to be attenuated and less for plate positioned at 45° than at 135°. An experimental investigation of natural convection heat transfer in horizontal and vertical closed, narrow enclosures with heated rectangular finned base plate was done by Nada [1]. He concluded that the heat transfer augmented to increase in fin height and Rayleigh number. The heat transfer was noticed to increase with increase in fin density up to a certain limit, due to increase heat transfer surface area; afterwards it was tending to decrease due to decrease in flow intensity. The heat transfer rate for horizontal arrangement was noticed more pronounced as compare to vertical orientation. Dialameh *et al.* [14] carried out a numerical study to determine the effect of fin length over heat transfer coefficient for rectangular thick fin arrays having a short height (b≤50mm). A numerical study was done by Zerrin Bocu *et al*. [15] of Laminar natural convection heat transfer and air filled in 3D rectangular enclosures with pin arrays attached to hot wall.

Ho and Chang [16] performed a two dimensional numerical analysis for heat transfer in finned square enclosure with water as a working fluid. Oztop and Abu-Nada [17] studied the natural convective heat transfer numerically for square enclosure filled with Nano fluids (Al_2O_3, TiO_2) and heated from the left vertical wall. The heat transfer was increased almost monotonically with volume fraction (φ) increase in all Rayleigh numbers and Nano fluids. An experimental analysis regarding heat transfer through triangular fin arrays at constant heat flux was conducted by Al-Jamal and Khashashneh [18]. They described the Nusselt number as a function of the maximum Reynolds number (Remax) at Prandtl number (PR) = 0.7. The natural convection heat transfer analysis from the arrays of isothermal triangular fins on air for vertical and horizontal orientations for an ample range of Ra by Karagiozis *et al.* [20] experimentally. A conjugate heat transfer problem of a second grade viscoelastic fluid through a triangular fin was studied by Hsu *et al*. [21] numerically for forced convection. A numerical investigation to study the control of mixed convection in lid driven enclosures using conductive triangular fins had been done by Sun *et al.* [19].

From the above studies, it has been concluded that a wide range of experimental as well as numerical investigation is concerned fluid flow and heat transfer characteristics in a fluid layer within the enclosures with heated bare base plate as well as with finned base plate. Above studies were carried out with different enclosure geometries, working fluids for wide ranges of influencing parameters. Though in case of finned base plates, the maximum work had been done for rectangular fins. Very few literatures are available regarding heat transfer from triangular fin geometry, although triangular fin provides maximum heat transfer per unit weight. In fact, there is no literature is available for heat transfer from triangular fin array within an enclosure. These facts forced the present investigation to be conducted to study the heat transfer from triangular fin array within air filled enclosure by experimentally, in order to analyze the effects of several persuading parameters on system performance. A heating element was placed in the bottom and at the top a cooling tank was facilitated with circulating cold water at a constant mass flow rate ($\dot{m}$ = 0.002381kg/s) by using a control device. The study was done at wide ranges of depending parameters like fin height (L), fin spacing (S) and Rayleigh number to analyze the effect on the Nu and the fin effectiveness to compare the heat transfer from bare plate.

Nomenclature

A: Effective surface area available for heat transfer [m^2]
a: Length of the enclosure [m]
b: Enclosure width [m]
g: Acceleration due to gravity [ms^{-2}]
h: Average convection heat transfer coefficient [W/m^2K]
H: Height of the enclosure [m]
K: Thermal conductivity [W/mK]
L: Fin height [m]
Nu: Average Nusselt number [-]
P_r: Prandtl number [-]
Q: Heat transfer rate [W]
Ra: Rayleigh number [-]
R: Resistance of the heat source [V^2/W]
S: Fin spacing [m]
T: Temperature [K]
T′: Plate thickness [m]
Ta: Ambient temperature [K]
T_c: Average temperature of the enclosure top surface [K]
T_h: Average temperature of the heated finned plate [K]
T_f: Film temperature [K]
V: Voltage supplied [V]
x: Thickness [m]

Greek letters

ε: Emissivity [-]
β: Coefficient of volume expansion [K-1]
ν: Kinematic viscosity of air fluid [m^2/s]
σ: Stefan Boltzmann constant [W/m^2K^4]

Subscript

a: Air
c: Convection
e: Conduction
r: Radition
w: Glass wall

Experimental setup

The schematic diagrams of the experimental test-rig are shown in the Fig. 1 (I, II). There are three main sections of the experimental setup

(1) Heating section, (2) Test section, (3) Cooling section

The heating arrangement is comprised of a nickel-chrome wired panel heater with dimensions 320 × 200 mm, mounted on a 3mm thick asbestos plate and insulated with mica sheets from all the sides. It was put within a rectangular wooden block and insulated with 50 cm thick glass wool layer from all the sides to minimize the heat loss. Pure aluminium was used as the material for base plate (T'plate = 3 mm) as well as fins due to its high thermal conductivity, low emissivity (ε = 0.052 $W/m^2\text{-}K^4$), structural strength and durability.

The test section included the triangular fin array within a narrow air filled rectangular enclosure with internal dimensions a × b × H = 320 x 200 x 50 mm. The fins are connected from the base plate with the help of aluminium screw. The effect of contact resistance between the fins and the base plate has been considered in the uncertainty analysis. The tested fin array combinations are given in table 1. The fin width is kept equal to the cavity lateral side (200 mm) for maximum heat transfer enhancement as per the conclusions of literatures [22, 23].

The cooling segment consisted of a cooling tank facilitated with circulating cold water at a constant mass flow rate ($\dot{m}$ = 0.002381kg/s) by using a flow rate control valve. The flow of water is maintained by a small pump (water cooler pump) to maintain the enclosure top surface at a constant low temperature as per literature [1]. It is considered as a natural convection heat transfer process because the heat transfer from the heated bottom surface to the top surface of the enclosure is done by a series of Benard cells of the air due to density difference caused by temperature difference. The enclosure height is kept small as comparable to the enclosure base dimensions in order to proper circulation of convection cells.

Procedure

Total eleven K- type Teflon coated chromel alumel thermocouples with an accuracy of 0.25˚C were used for measurement of temperature distribution at different points of the arrangement. Out of them six thermocouples were arranged symmetrically to measure the temperature of the hot base plate, four thermocouples were arranged to measure the temperature of cold surface and one was used to measure the temperature of the outside wall of the enclosure. The ambient temperature was measured with the help of mercury thermometer corresponding to different steady state reading of the thermocouples. The thermocouples were inserted and fixed through the holes drilled from the enclosure backside passing through the heater up to the finned base plate without disturbing the heat transfer within the enclosure. The thermocouple location has been shown in the Fig. 2 (a, b).

After mounting each fin array within the enclosure the heater was connected with a DC power supply through a variable transformer to control the input power at (10V, 20V, 30V, 40V) and hence the input heat flux and corresponding Rayleigh number were controlled. A digital multimeter with an accuracy of 0.5% for voltage as well as the resistance was used to record the output data. A constant mass flow rate of the water is allowed to circulate within the cooling tank using a flow control valve. The experiment was allowed to run about 4 hours so that the steady state condition (Temperature variations with respect to time, dT/dt was less than 1˚C/hour for all the thermocouples) was achieved. At this stage, the thermocouple readings were recorded by using a digital temperature indicator with the help of a selector switch. All the experiments were carried out in a windowless large room to avoid any interference of air currents and precautions were taken to maintain almost constant ambient temperature.

Data Reduction

The output data obtained are mainly the values of temperatures by means of thermocouples. The dimensionless Rayleigh number (Ra) is obtained using equation (1)

$$Ra=H^3g\beta(T_h-T_c)P_r/\upsilon^2 \qquad \text{………………..(1)}$$

Where T_h = (T1+T2+T3+T4+T5+T6)/6,

T_c = (T7+T8+T9+T10)/4

T1, T2, T3, …….,T10, are thermocouples reading.
All the fluid properties were determined by film temperature-

$T_f = (T_h+T_c)/2$

Using the energy balance, the net heat transfer rate (Q_{net}) within the enclosure is given by-

$$Q_{net}=Q_c+Q_r+Q_e \qquad \text{………………..(2)}$$

Using Ohm's law the input power (P_{in}) is given as-

$$P_{in} = V^2/R \qquad \text{………………..(3)}$$

The energy balance equation can be written as follows-

$$Q_{net}=V^2/R=Q_c+Q_r+Q_e \qquad \text{………………..(4)}$$

The conductive heat transfer rate (Q_e) at the enclosure side walls is determined as-

$$Q_e=-K_{w*}A_{w*}(\Delta T_w)/x_w=h_{a*}A_{w*}(T_w-T_a) \qquad \text{……………(5)}$$

Where h_a is the average heat transfer coefficient of the ambient air, ΔT_w = Temperature difference across the glass wall.

All the surfaces of the enclosure are assumed to be isothermal, gray and diffuse. The radiation heat loss Q_r from the enclosure internal hot surfaces (sides and bottom surfaces) is the net rate at which radiation is incident on the cold surface of the enclosure. Identifying the enclosure internal hot surfaces by the number i, the net rate at which radiation is incident on the enclosure cold surface is calculated as-

$$Q_r=\varepsilon_c A_c(\sigma T_h^4 - G_c) \quad \text{..................(6)}$$

Where the irradiation G_c is given by

$$G_c=\sum_{i=1}^{5}(Fc * Ji) \quad \text{...................... (7)}$$

Where (Fc * Ji) is the view factor between the enclosure cold surface and the i^{th} surface of the enclosure and Ji is the radiosity of that surface and is given by

$$Ji=\varepsilon_i \sigma T_i^4+(1-\varepsilon_i)* \sum_{j=1}^{5}(Fij * Jj) \quad \text{................(8)}$$

The view factors Fij between parallel and perpendicular surfaces were calculated based on graphs and expressions given in Incropera and De Witt [27] and Suryanarayana [28]. Equations (6-8) were solved together to find the radiation heat losses in terms of the surface temperatures of the enclosure inside walls. The average convective heat transfer coefficient between the hot and cold surfaces of the enclosure is determined as-

$$h=Q_c/A_c(T_h - T_c) \quad \text{................(9)}$$

The average Nusselt number for the heated bottom plate is calculated by using equation (10).

$$Nu=hH/K_a \quad \text{............(10)}$$

Uncertainty Analysis

The maximum possible care and precautions are normally taken during an experimental investigation to minimize the error. The maximum uncertainty of the measurements can be obtained using standard procedures as discussed in the literature [25]. The relation used for uncertainty measurement for the experimental investigation is mentioned in the equation (11).

$$\varepsilon_y=\sqrt{\sum_{i=1}^{n}\left(\left(\frac{\partial y}{\partial xi}\right)\varepsilon i\right)^2} \quad \text{............(11)}$$

where ε_y is the absolute uncertainty in the results for the measurement of $y(x_1, x_2, x_3 \ldots\ldots x_n)$, and εi is the uncertainty in the measurement of independent variable xi.

If $y = x_1^{a1}, x_2^{a2}, x_3^{a3} \ldots\ldots x_n^{an}$, Then the uncertainty is measured by using equation (12)

$$\varepsilon_y/y=\sqrt{\sum_{i=1}^{n}\left(\frac{ai}{xi}\varepsilon i\right)^2} \quad \text{...........(12)}$$

ε_y/y is known as relative uncertainty.

By using above relationship and uncertainty of calculated parameter from the table 2 the uncertainty associated with Nu is calculated and found that it is lie between 7-9% for all obserbations.

RESULTS AND DISCUSSION

The effects of different significant geometrical parameters such as Fin Height (12.5 mm ≤ L ≤ 37.5 mm), Fin Spacing (25 mm ≤ S ≤ 100 mm) over the Rayleigh number (365214 ≤ Ra ≤ 683826) on the steady state natural convection heat transfer and fin effectiveness for triangular fin arry has been anylised experimentaly. The Effect of above influencing parameters on the Nu and effectiveness is presented graphically.

Table 1 Tested Fin Array Combinations

Fin array combination	Input Voltage (V)	Fin height (L)(mm)	Fin spacing (S)(mm)	Number of fins (n)
1	10, 20, 30, 40	12.5	25	13
2	10, 20, 30, 40	12.5	50	7
3	10, 20, 30, 40	12.5	100	4
4	10, 20, 30, 40	25	25	13
5	10, 20, 30, 40	25	50	7
6	10, 20, 30, 40	25	100	4
7	10, 20, 30, 40	37.5	25	13
8	10, 20, 30, 40	37.5	50	7
9	10, 20, 30, 40	37.5	100	4

Table 2 Parametric Uncertainty Analysis

Calculated Parameter	Associated Uncertainty Range
Ra	4-7%
P_{in}	0.5-1.5%
Q_e	6-9%
Q_r	1-3%
Q_c	6-9%
h	6-9%

Heat Transfer

This section includes the effects of above influencing parameters on the natural convection heat transfer within air filled rectangular enclosure from the Triangular Fin arry has discussed.

Effect of Fin Height

The variation of the Nusselt number with fin height variation has been illustrated in the Fig. 3 (a-c). at different fin spacing and Rayleigh number values. From the figures it can be seen that for all the fin spacing and Rayleigh number the Nu increases continuously with an increase in fin height. It can be credited to the augmentation in the effective surface area available for heat transfer as well as increased in probability of the formation of separate Benard cells. However, at less fin spacing where very high fin density, increasing fin height increases the resistance of convection cell movement and this slightly reduces the heat transfer rate. This reduction of heat transfer can not overcome the increase of heat transfer due to the increase of surface area.

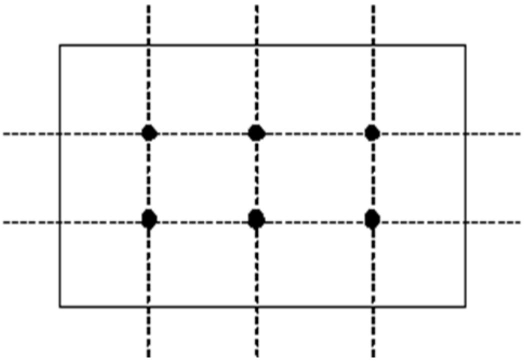


Fig 2 a

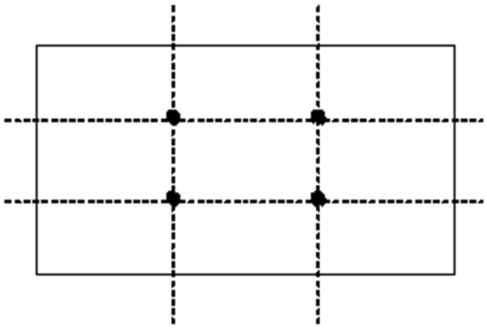


Fig 2 b

Fig. 2 a, b Thermocouples Location (a) Heated Base Plate; (b) Enclosure Top Surface.

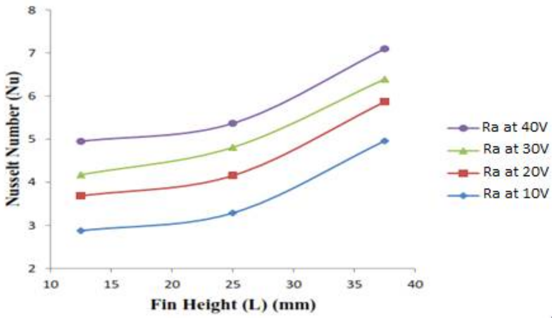


Fig 3 a

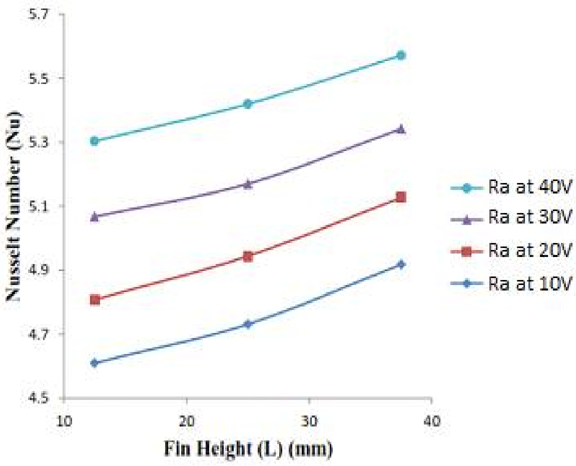


Fig 3 b

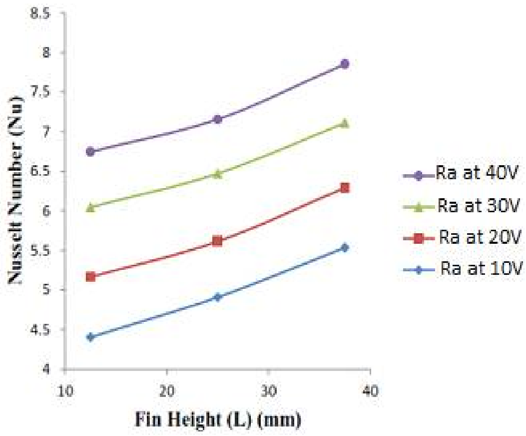


Fig 3 c

Fig 3 a-c Variation of Nusselt Number with Fin Height at different Rayleigh number and fin spacing- (a) S = 25 mm; (b) S = 50 mm; (c) S = 100 mm.

Effect of Fin Spacing

The dependence of Nu on fin spacing at the different Rayleigh number and fin height has been shown in Fig. 4 (a-c). It can be easily concluded from the curves that Nu increases initially with a decrease in fin spacing (increase in fin density) up to a maximum value then it starts decreasing on any further decrease in fin spacing. This behaviour may be credited for the two effects –firstly enhancement in the heat transfer due to the enhance in the effective surface area existing for heat transfer and secondly reduction is due to increase in interruption for the fluid flow within the enclosure. The increase of the fin density from the optimum value causes the increase in hindrance effect for the Benard cells due to boundary layer interactions. So, initially decreasing fin spacing results in improved heat transfer up to a certain value as the effect of surface area dominates the hindrance effect,

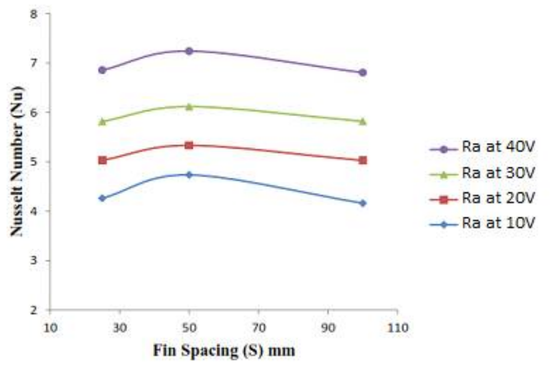


Fig 4 a

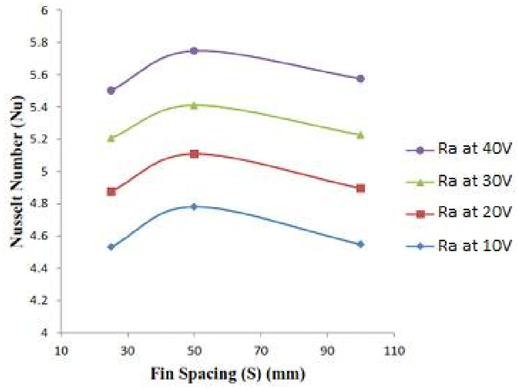


Fig 4 b

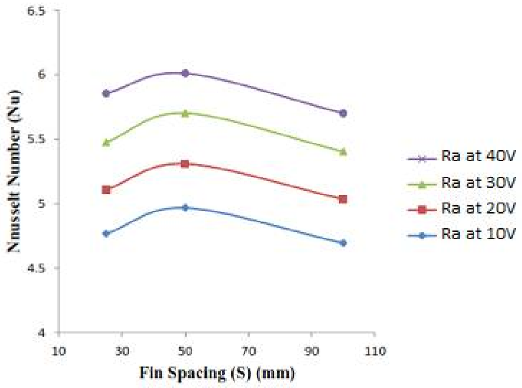


Fig 4 c

Fig 4 a-c Variation of Nusselt Number with Fin Spacing at different Rayleigh number and fin height - (a) L = 12.5 mm; (b) L = 25 mm; (c) L = 37.5mm.

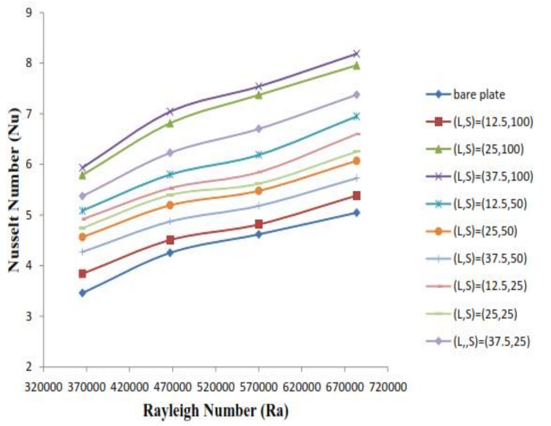
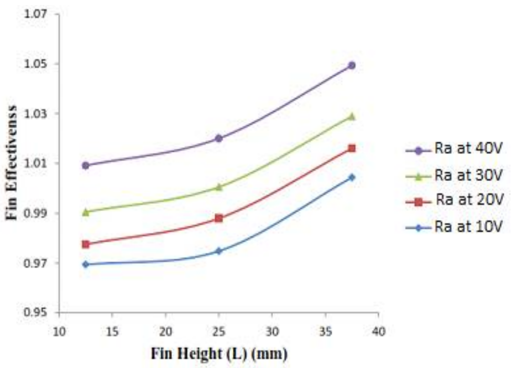


Fig. 5. Variation of Nusselt Number with Rayleigh Number at different combinations of fin height and spacing



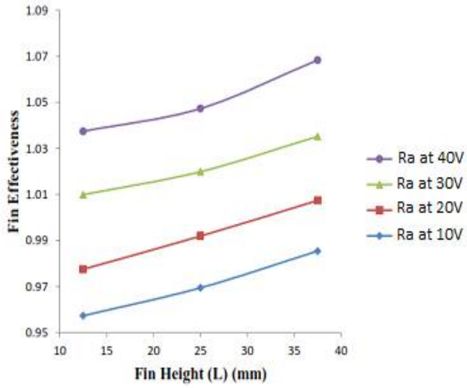


Fig 6 a **Fig 6 b**

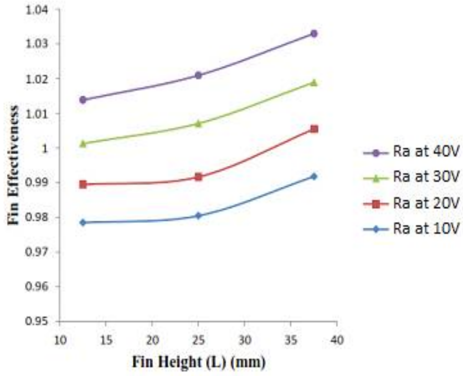


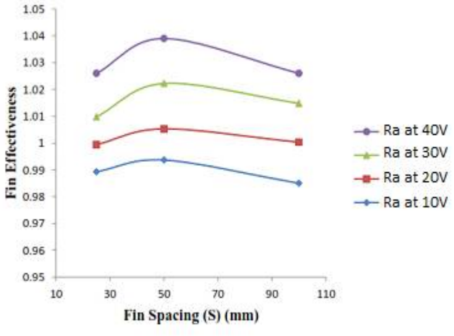
Fig 6 c

Fig 6 a-c Fin Effectiveness as a function of Fin Height at different Rayleigh number and fin spacing- (a) S = 25 mm; (b) S = 50 mm; (c) S = 100 mm.

while any additional decrease in fin spacing grades to reduce the heat transfer rate as hindrance effect overcomes the effect of surface area. The value of fin spacing which exhibits maximum heat transfer is known as optimum fin spacing (S_{opt}) and in this carrying out tests it was found to be at L = 37.5 mm.

Effect of Rayleigh Number

The Nusselt number variation in Rayleigh number has been illustrated in the Fig. 5 at different combinations of fin length and spacing values. It can be observed from the curves that the Nusselt number increases continuously with Rayleigh number at all the fin height and fin spacing



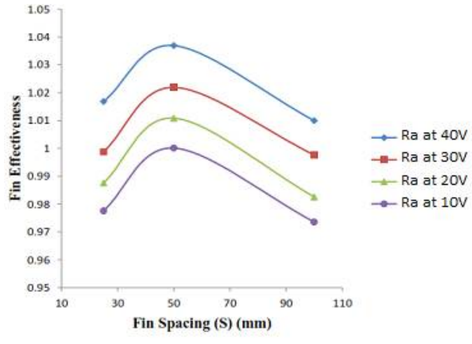


Fig 7 a **Fig 7 b**

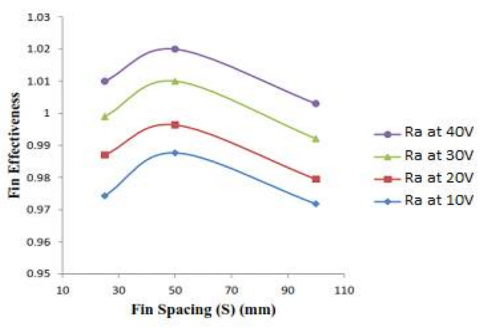


Fig 7 c

Fig 7 a-c Fin Effectiveness as a function of Fin Spacing at different Rayleigh number and fin height - (a) L = 12.5 mm; (b) L = 25 mm; (c) L = 37.5 mm.

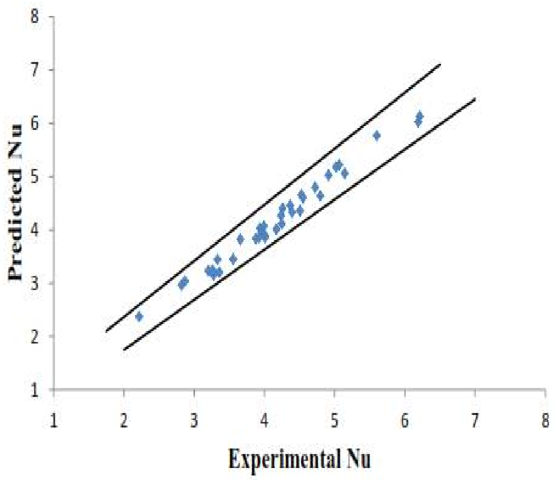


Fig. 8 Plot of Predicted values vs. Experimental values of Nu

combinations. This result may be attributed to the increase in buoyancy force caused by enhancing flow intensity due to improved Rayleigh number value. The figure shows that for all experienced fin array combinations and at all Ra within the range, the Nusselt number is observed higher than that investigated in bare heated base plate, which can be accredited to the increase in the effective surface area available for heat transfer due to addition of extended surfaces.

Fin Effectivenss

The finned surface effectiveness is most important parameter to be considered in analyzing the design and performance of a finned surface. The use of extended surfaces cannot be recommended unless the heat transfer enhancement justifies the added cost and complexity associated with the fins. The performance of the fins is described on the basis of fin effectiveness [26]. The fin effectiveness (ε_{fin}) has been determined by using equation (13) as per literature survey [1].

$$\varepsilon_{fin}=\frac{(\mathrm{Nu})\text{with fin}}{(\mathrm{Nu})\text{without fin}} \qquad \text{........................(13)}$$

if $\varepsilon_{fin} = 1$, It indicates that the fin addition does not affect the heat transfer rate. If $\varepsilon_{fin} <1$, It indicates that the fin acts as insulation and if $\varepsilon_{fin} >1$, then the attachment of fins results in enhanced heat transfer from the surface. This section covers the effects of several influencing parameters like fin height (L), fin spacing (S), as well as the Rayleigh number on the effectiveness of triangular fin, within air filled rectangular enclosure.

Effect of Fin Height

The dependency of fin effectiveness on fin height has been shown in the Fig. 6 (a-c). at different Rayleigh number and fin spacing. It can be seen from the graphs that the value of fin effectiveness is near about 1. The effectiveness increases with continuous increase in fin height for wide ranges of fin spacing and Rayleigh number values. This result can be attributed to the continuous increase in heat transfer rate due to the addition of fins (increased surface area) as discussed in section 6.1.1.

Effect of Fin Spacing

The variation of fin effectiveness with fin spacing has been illustrated in Fig. 7 (a-c). at various fin height and Rayleigh number values. It can be observed from the curves that initially fin effectiveness increases with decrease in fin spacing up to a maximum value then tends to decrease. This effect indicates the variation in the system performance (heat transfer enhancement) due to movement of convection cells or formation of split Benard cells between each two adjacent fins and latter increased in resistance for the convection cell movement with a decrease in fin spacing.

Development and Verification of Correlations for Nusselt Number

In order to predict the performance of the triangular finned base plate within an air filled rectangular enclosure, statistical correlations for Nusselt number are developed at wide ranges of different operating and design parameters such as; fin height (L), fin spacing (S) and Rayleigh number (Ra) is given as-

$$\mathrm{Nu}=f(L/H, S/H, Ra) \qquad \text{………………(14)}$$

In order to determine the functional relationship between the Nusselt number (Nu) and dimensionless fin height (L/H), fin spacing (S/H) and Rayleigh number (Ra) the experimental outcomes, i.e. 'Nu values' are plotted against the above dimensionless parametere in the range of ($1\leq L/H\leq 2.225$), ($0.449 \leq S/H\leq 2.225$) and ($296559\leq Ra\leq 660003$) values using power law model of curve fitting by regression analysis.

The resulting expression is written as-

$$\mathrm{Nu}= C_o e^{-0.0434(\ln(Ra))^{\wedge}2} \mathrm{x}(Ra)^{0.512} \mathrm{x} e^{0.692\left(\frac{L}{H}\right)} \mathrm{x} e^{0.084\left(\ln\left(\frac{S}{H}\right)\right)^{\wedge}2} \mathrm{x}\left(\frac{S}{H}\right)^{0.3036} \qquad \text{………………(15)}$$

It is observed that the predicted values of Nu by using the correlations given in equation 15. is lie -within the range of ±4% of experimentally observed data values. Thus, the developed Nusselt number correlation shows very good agreement with experimental results investigated for the above range of influencing parameters.

Comparision with the literature

The variation of Nu with Ra has been found similar to the experimental study of natural convection from triangular fin array done by Gaurav Kumar, Kamal Raj Sharma *et al.* [29]. It is found that variations of Nu with Ra and fin spacing, some similar to the numerical study of natural convection heat transfer in a finned horizontal fluid layer by E. Arquis and M. Rady [30].

CONCLUSIONS

An experimental investigation regarding natural convection heat transfer from the heated triangular fin array within a narrow air filled enclosure is performed. The present study has been done to analyze the effects of several inducing factors on system performance. The study is performed at given ranges of influencing parameters like fin height, fin spacing, and Rayleigh number.

On the basis of this investigation, the major concluding remarks can be highlighted as follows

1. The continuous increase in the Nusselt number with fin height and Rayleigh number (increase in the effective surface area for heat transfer) is observed while with a decrease in fin spacing, the average Nusselt number increases (increase in more surface area for heat transfer) firstly up to a maximum value than tends to decrease due to hindrance effect of the fluid flow.
2. The effectiveness of the finned surface is a very strong function of fin spacing, fin height and Rayleigh number. The fin effectiveness increases with increase in fin height. While with a decrease in fin spacing, the fin effectiveness increases initially up to a maximum value then tend to decrease.
3. The optimum fin spacing (S_{opt}) is determined as the most important design parameter for influencing the system performance. The influencing parametric combination Ra = 683826, L = 37.5 mm and S = 50 mm have been determined as the optimum factor of combination for the present study.
4. The new empirical correlations are proposed for the prediction of the Nusselt number of the above tested ranges of influencing parameters and given as-

$$Nu = Co e^{-0.0434(\ln(Ra))^2} x (Ra)^{0.512} x e^{0.692(\frac{L}{H})} x e^{0.084(\ln(\frac{S}{H}))^2} x (\frac{S}{H})^{0.3036}$$

The above proposed correlations are successfully verified with experimental data. The ±4% inaccuracy between the experimental and predicted numerical results have been observed.

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