

## **Experimental Investigation of Natural Convection from Heated Triangular Fin Array within a Rectangular Enclosure**

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### **Abstract**

In this article an experimental investigation is made to predict the performance of heated triangular fin array within a vertically oriented and air filled rectangular enclosure. The experimental analysis is done to analyze the effects of several influencing parameters for their wide ranges; Rayleigh number  $295214 \leq Ra \leq 773410$ , fin spacing  $25 \text{ mm} \leq S \leq 100 \text{ mm}$  and fin height  $12.5 \text{ mm} \leq L \leq 37.5 \text{ mm}$  for constant heat flux boundary conditions at the heated and cooled walls of the enclosure. An empirical correlation is also developed relating Nusselt number to several influencing parameters.

**Keywords:** Thermal management; triangular fin array; rectangular enclosure; Rayleigh number.

### **1. Introduction**

In the present scenario, the general trend is to use compact systems especially in the electronic field. This ever-increasing demand for compact systems leads to higher packing density causing higher heat generation. It results to increase in temperature of the instruments beyond their permissible limits causing serious overheating problems. It affects the system performance and may lead to the equipment failure also. For the few decades the increasing numbers of these thermal management problems accentuate the need of effective cooling systems to enhance the system performance. Use of extended surfaces is the most prominent way to enhance the heat transfer rate as it is more convenient as well as economical. The weight and volume of the equipment are the important aspects of design in different engineering applications like electronic and computing industries; power plants or compact heat exchanger sectors.

The natural convection heat transfer associated with isothermal triangular fin array in the air was firstly reported by [A. Karagiozis,etal(1994)] for wide range of Rayleigh number. A numerical study regarding control of mixed convection in lid driven enclosures using conductive triangular fins was done by [Changzheng Sun, etal (2011)]. They concluded the triangular fins as an important control parameter for heat transfer, temperature distribution and flow field. Ref. [Changzheng Sun, etal (2004)] studied natural convection heat transfer within an air filled, two dimensional rectangular enclosures with adiabatic side walls and heated from below and cooled from above numerically for wide range of aspect ratio and Rayleigh number values. Ref. [M. Mobedi, H. Yuncu, etal (2003)] investigated numerically, the effects of different fin parameters for air filled horizontal base arrangement. They reported the reduced heat transfer enhancement with increase in fin height. This effect was attributed to the reduced amount of air entering from the space between the fins. Ref. [Charles D. Jone, etal s(1970)] observed fin weight as very crucial influencing parameter. It was reported that for given heat dissipation rate, light weight fins were reported to provide the maximum heat transfer rate within the optimized fin spacing range. Ref. [Yung-Shin Tseng, etal (2007)] suggested an optimum parametric design for the improvement of cooling of chip using Taguchi's philosophy. They reported the effort reduction of 50% approximately in performing the experiments and simulations by using such statistical design. Ref. [E. Bilgen, etal (2003)] reported the heat transfer suppression up to 38% by selecting appropriate fin parametric combinations.

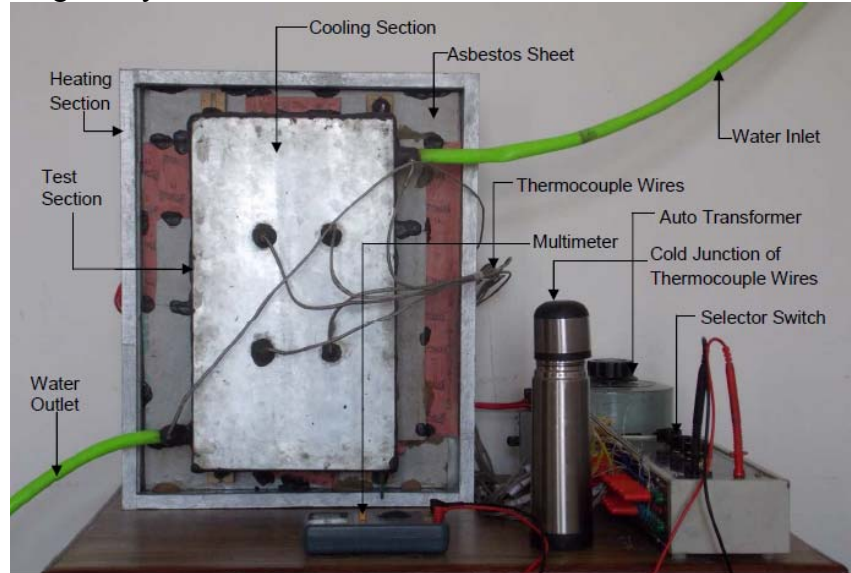
From the literature survey discussed above it can be clearly observed that a very little information is present regarding natural convection heat transfer is present, although numbers of literatures are available concerning heat transfer within the enclosures. To the authors' best knowledge; there is no literature available dealing with natural convection heat transfer from triangular fin array within an air filled rectangular enclosure. The present work deals with the natural convective heat transfer through triangular fin arrays within the rectangular enclosure.

## **2. Experimental Test Rig and Data Reduction**

### **2.1. Experimental Test Rig**

The detailed pictorial view for experimental test rig has been illustrated in figure 1. It consists of mainly three sections- heating section, test section and cooling section. The heating arrangement contains a nickel chrome wired panel heater placed inside a wooden cover, insulated properly by a thick layer of glass wool and asbestos sheets to minimize the heat loss. The test section consists of a fin array of triangular geometry within an air filled rectangular enclosure ( $a \times b \times H=320 \text{ mm} \times 200 \text{ mm} \times 50 \text{ mm}$ ). Aluminum is used as a material for fins as well as heated base plate due to its high thermal conductivity (increasing function of temperature) and low emissivity values. The cooling arrangement is comprised of a cooling tank facilitated with cooling water flow at a constant mass flow rate. Thirteen K-type, Teflon coated thermocouples are used to measure the temperature distribution at different points within the enclosure. The input heat flux and hence the corresponding Rayleigh number, is controlled by

using an auto-transformer. The output data was recorded using digital multi-voltmeter after achieving steady state condition.



**Fig. 1:** Experimental Test Rig.

## 2.2. Data Reduction

The Rayleigh number is defined as

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

where, H is the enclosure height,  $T_h$  is the temperature of heated finned base plate,  $T_c$  is the temperature of enclosure top surface, g is the acceleration due to gravity, Pr is the Prandtl number and  $\nu$  is the kinematic viscosity of the working fluid (air). All the fluid properties were determined at film temperature, i.e.  $T_f = (T_h + T_c)/2$ . If V and R are the voltage applied and resistance of the heat source respectively, then using the energy balance, the net heat transfer ( $Q_{net}$ ) within the enclosure is given by-

$$Q_{net} = Q_{conduction} + Q_{convection} + Q_{radiation}$$

$$( Q_{net} = V^2 / R )$$

The conduction heat transfer ( $Q_{conduction}$ ) at the enclosure side walls is determined as-

$$Q_{conduction} = -k_{wall} A_{wall} \frac{\Delta T_{wall}}{x_{wall}}$$

where,  $K_{wall}$ ,  $A_{wall}$  and  $x_{wall}$  are the thermal conductivity, cross-sectional area and thickness of the glass walls of the enclosure.  $\Delta T_{wall}$  is the temperature difference between the inner and outer surfaces of glass walls of the enclosure. All the surfaces of

the enclosure are assumed to be isothermal, gray and diffuse and the net radiation heat transfer was calculated by the method of [8] using the equation-

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

where,  $F'$  is the view factor,  $A_r$  is the effective surface area available for radiation,  $\sigma$  is the stefan Boltzmann constant. The average convective heat transfer coefficient between the hot and cold surfaces of the enclosure was determined as-

$$h = \frac{Q_{convection}}{A(T_h - T_c)}$$

where,  $A$  is the effective surface area available for convective heat transfer. The average Nusselt number for the heated bottom plate was calculated using equation

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

where,  $K_a$  is the thermal conductivity of air determined at  $(T_h + T_c)/2$ .

The fin effectiveness ( $\varepsilon$ ) is defined as

$$\varepsilon_{fin} = \frac{Q_{finned\ surface}}{Q_{bare\ surface}}$$

### 3. Results and Discussions

#### 3.1. Effect of Rayleigh Number

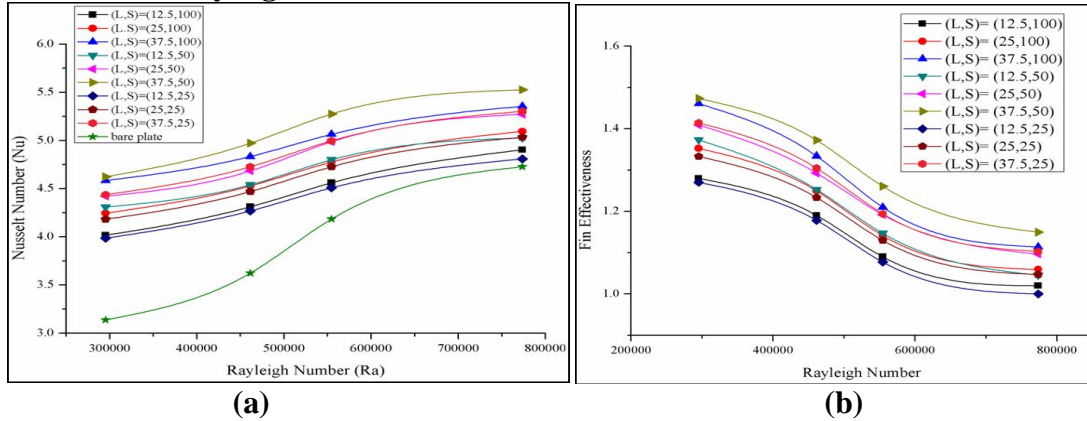


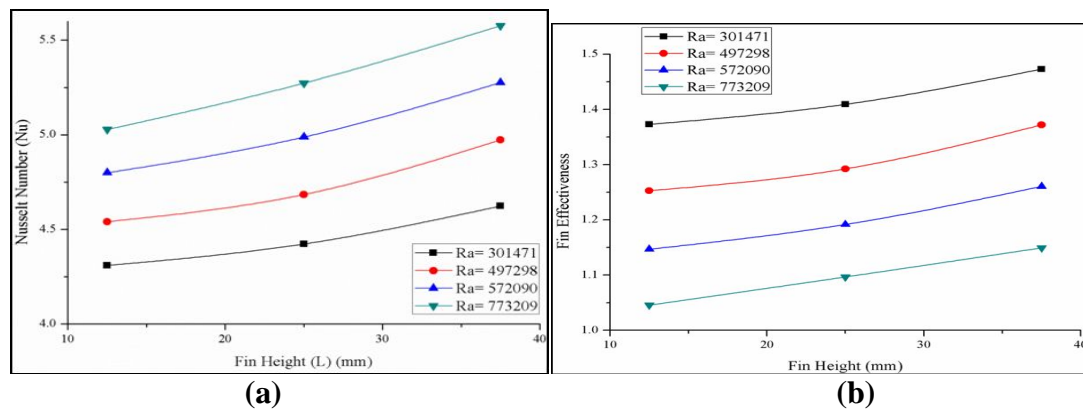
Fig. 2: Effect of Rayleigh Number Variation on System Performance (a) Nusselt Number; (b) Fin Effectiveness.

Figures 2(a) and 2(b) shows the variations of Nusselt number and fin effectiveness with Rayleigh number for all the tested fin array configurations. The continuous increase in Nusselt number with Rayleigh number is reported at all the fin height and

fin spacing combinations. This result may be attributed to the increased buoyancy force causing increased flow intensity due to improved Rayleigh number value. The figures show that for all tested fin array combinations and at any Rayleigh number within the range, the Nusselt number is observed higher than that investigated in bare heated base plate case which can be attributed to the increased effective surface area available for heat transfer due to addition of extended surfaces. The fin effectiveness keeps on decreasing continuously with increase in Rayleigh number. This result may be attributed to the fact that fin effectiveness is a relative measure of improved system performance because of addition of fins to the system performance with bare base plate. Increase in Rayleigh number results to increase in heat transfer for both cases, i.e. systems with bare base plate as well as with finned base plate, although, the rate of heat transfer augmentation is more for no fins condition due to low rates of buoyancy force increase and viscous force reduction (due to boundary layer formation), as compare to the systems with heated finned base plate. It sources the reduction of fin effectiveness with increase in  $Ra$ .

**3.2. Effect of Fin Height**

The variations of Nusselt number and fin effectiveness with fin height have been illustrated in figures 3 (a) and 3(b) respectively at a particular fin spacing value. As it can be seen from the graphs attained that both Nusselt number and fin effectiveness increases continuously with increase in fin height for all the fin spacing and Rayleigh number values. This result can be attributed to the increased effective surface area for heat transfer as well as increased probability of formation of separate convection cells due to increase in fin height causing enhanced Nusselt number. Increase in Nusselt number directly refers to improved convection heat transfer rate.

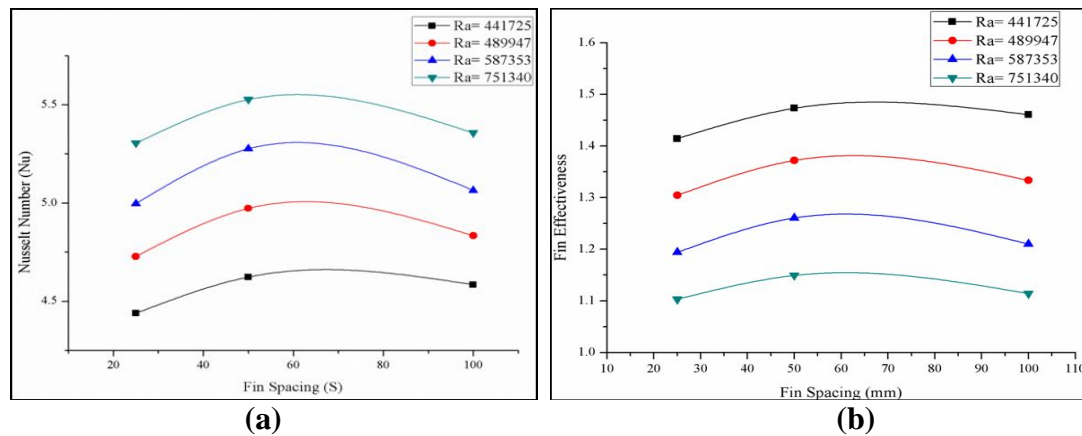


**Fig. 3:** Effect of Fin Height Variation on System Performance (S= 50 mm) (a) Nusselt Number (b) Fin Effectiveness.

**3.3. Effect of Fin Spacing**

Figures 4 (a-b) depicts the effect of fin spacing variations on system performance at a particular fin height. It can be easily concluded from the curves that both Nusselt

number and fin effectiveness increase initially with decrease in fin spacing i.e. increasing number of fins up to a maximum value then start decreasing on any further reduction in fin spacing. This behaviour may be attributed to the two effects of fin spacing - enhanced heat transfer due of increased effective surface area available for heat transfer and reduced heat transfer due to increased hindrance to the fluid flow within the enclosure. The increase of the fin density causes the increased hindrance effect due to boundary layer interactions. So, initially decreasing fin spacing results to enhanced heat transfer up to a certain value as surface area effect dominates the hindrance effect while any further decrease in fin spacing results to reduced heat transfer rate and the fin effectiveness vice versa as hindrance effect overcomes the surface area effect.



**Fig. 4:** Effect of Fin Spacing Variation on System Performance ( $L= 37.5$  mm) (a) Nusselt Number (b) Fin Effectiveness.

### 3.4. Empirical Correlation

It has been observed that Nusselt number for heated finned base plate within an enclosure is very strong function of various geometry and flow parameters as; fin height ( $L$ ), fin spacing ( $S$ ) and Rayleigh number ( $Ra$ ). In order to predict the performance of an air filled rectangular enclosure having triangular finned base plate, the statistical correlation for Nusselt number is developed for wide ranges of different operating and design parameters.

$$Nu = 648.72 \left(\frac{L}{H}\right)^{0.2091} \left(\frac{S}{H}\right)^{0.0106} (Ra)^{-0.9214} \exp\left(0.0705 \left(\ln\left(\frac{L}{H}\right)\right)^2 - 0.0949 \left(\ln\left(\frac{S}{H}\right)\right)^2 + 0.0224 \left(\ln(Ra)\right)^2\right)$$

In order to verify the empirical correlation the predicted results are compared with the experimental results and the maximum variation of 2.6% is observed. Thus, the developed Nusselt number correlation shows a very good agreement with experimental results in range of influencing parameters investigated in the present work.

#### **4. Conclusions**

Natural convection heat transfer from triangular fins within an air filled and vertically oriented rectangular enclosure has been studied experimentally. The system performance (Nusselt number and fin effectiveness) as the functions of fin height, fin spacing and Rayleigh number have been reported. An empirical correlation relating Nusselt number to the influencing parameters has also been suggested with maximum variation of  $\pm 2.6\%$ .

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