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# Investigation of thermal performance in natural convection from rectangular staggered interrupted fins

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## ABSTRACT

**It is investigated that special surface geometry or special fin pattern may enhance thermal performance as well as heat transfer coefficients considerably. In the work herein, steady-state external natural convection heat transfer from vertically-mounted rectangular Staggered interrupted finned heatsinks is investigated, experimentally and numerically. They are compared with continuous and inline interrupted fin arrays, also validating the existing results for continuous and inline interrupted fins. A systematic experimental and numerical, study is conducted on the effect of the fin array and Staggered interrupted fins array. FLUENT software is used in order to develop a three-dimensional numerical model for investigation of staggered interrupted fin effects. To perform an experimental study and to verify the analytical and numerical results which show that vertical rectangular staggered interrupted fins enhances the thermal performance of fins and reduces the weight of the fin arrays, which in turn, can lead to lower manufacturing costs.**

*Keywords*— **Efficient cooling techniques, Heat transfer argumentation, natural convection, staggered fin array.**

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## I. INTRODUCTION

The design of efficient cooling strategies is essential for reliable performance of high power density electronics. Now a day, the thermal losses of power electronic devices are increasing. Simultaneously, their sizes are decreasing. Consequently, heatsinks have to dissipate higher heat transfer rate in every new design. Therefore, finding efficient cooling solutions to meet these challenges is of paramount importance and has direct impacts on the performance and reliability of electronic and power electronic devices The importance of heat transfer augmentations has gained greater significance in areas as follows, fuel elements of nuclear power plants also in central processing units, bio medical devices, power electronics, gas turbine internal air foil cooling, ,microelectronic cooling and macro and micro scale heat exchangers. A tremendous amount of effort has been performed to develop new methods to increase heat transfer from fined surface to the surrounding flowing fluid such as air. Rib tabulators, an array of pin fins, and dimples have been employed for this purpose. The maximum temperature of the component is one of the main factors that control the reliability electronic products. Thermal management has always been one of the main issues in the electronics industry, power electronic system and its importance will grow in upcoming decades. The use of heat sinks is the most general application for thermal management in electronic devices packaging. Heat sink performance can be evaluated by several factors: material, flatness of contact surfaces, surface area, configuration, and fan requirements. Although there are a few investigations for the use of dimples under

laminar airflow conditions, there exist no experimental data with respect to the use of different dimple shapes for heat sink applications.[1]

### *1.1 Thermal Boundary Layer*

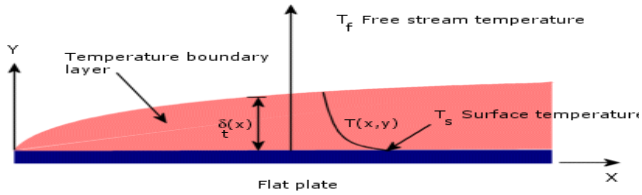


Fig.1.1 Thermal Boundary Layer

As shown in Fig 1.1, Heat will flow between a wall and the fluid adjacent to it when a temperature gradient is established between the wall and the fluid. Near the wall the fluid velocity increases from zero at the wall to the bulk velocity, sometimes not too distant from the wall relative to the radius of curvature. Likewise, the temperature changes from that at the wall to that in the free stream. The result is that the fluid temperature adjacent to the wall is assumed to be equal to the surface temperature of the wall at the interface and is equal to the bulk fluid temperature at some point in the fluid. The distance over which the temperature change occurs is called the thermal boundary layer. A momentum boundary layer also is present if the fluid is flowing past the wall. The momentum (hydrodynamic) boundary layer and the thermal boundary layer can affect each other. The distances over which the velocity changes from zero to the free stream velocity and the temperature changes from the wall temperature to free stream temperature are often different.

The main focus of this study is on natural convection heat transfer from an interruption, vertical and rectangular fins. However, a more general overview on these literatures in the area of natural heat transfer from fins is provided in this section. A number of varieties of theoretical expressions, graphical correlations also empirical equations have been developed to represent the coefficients for natural convection heat transfer from vertical plates and the vertical channels. These studies are mostly focused on geometrical parameters of the heatsinks as well as fins, such as fin spacing, fin height, fin length, as well as, fin directions also. Following study shows a summary of the literature review as it pertains to analytical, numerical or experimental type of work.

Aanalytical work on vertical channels was carried out by Elenbaas [2]. He investigated analytically and experimentally the isothermal finned heatsinks. And also developed an expression for Nusselt number and the where s is the spacing between two fins are as follow,

$$Nu_s = \frac{1}{24} Ra_s \left[1 - \exp\left(\frac{-35}{Ra_s}\right)\right]^{\frac{3}{4}}$$

The BarCohen and Rohsenow [3] were made for The analytical study resulted in general relations for natural convective heat transfer from vertical rectangular fins. Bar-also performed an analytical study to investigate the natural convective heat transfer from two parallel plates. They are developed relationship in between Nusselt number and the Rayleigh number Ra number for both, isothermal and isoflux plate cases and reported a correlation for the optimum fin spacing showing following equation.

$$Nu_s = \frac{hs}{k} = \left[\frac{576}{\left(\frac{Ra_s s}{L}\right)^2} + \frac{2.873}{\left(\frac{Ra_s s}{L}\right)^{0.5}}\right]^{-0.5}$$

Where$Ra_s$ is the Rayleigh number, which is mainly based on fin spacing between two adjacent fins, s, L is fins length also h is the convective heat transfer coefficient, and k is the fluid thermal conductivity, respectively. These aforementioned coefficients are mostly shown in Fig. 1.2.


Figure 1.2: Heatsink with continuous rectangular fins

For the three dimensional bodies [4] and [5], the Culham et al. [6] correlated the Nusselt number with characteristic length scale based on the squared root of the wetted area. The wetted area of a fin is that surface that is exposed to the air flow. Another numerical approach was to investigate the developing flow in the channel as well as the heat transfer between symmetrically heated, isothermal plates by Bodoia and Osterle [7] . They made aimed to predict the channel length required to attain a fully developed flow as a function of the channel width as well as wall temperature. A finite element method used by, Ofi and Hetherington [8] were to study the natural convective heat transfer from open vertical channels with uniform wall temperature. They also observed that fluid velocity may be vertical alone. Culham et al. [6] also used META, a numerical code, to simulate the natural convective heat transfer from a vertical fin array, also compared their results to their experimental data and the three dimensional models of [4] and [4] as mentioned in the previous section of review.

Several researchers were to be studied the natural convection heat transfer in vertical channels experimentally. Fujii [9] studied the heat transfer from inclined interrupted fin channels systems. In this study, it was examined that the thermal boundary layers were interrupted by the fins, a correlation developed, as shown following Eq., was fitted to be an experimental results. . Figure 1.3 shows the Fujii was used by these fin geometry.

$$Nu_s = \frac{1}{24} Ra_s . \left(\frac{s}{l}\right) \left\{1 - e^{-12.5 \Big/ \left[Ra_s \cdot \left(\frac{s}{l}\right)\right]^{\frac{3}{4}}}\right\}$$

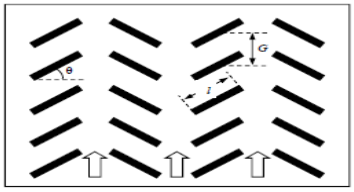


Figure 1.3:Fujii's [9] considered geometry for inclined interrupted fins

Naturalheat transfer coefficients for four different fin arrays also with three different base plates were calculated by Starner and McManus [10]. Flow patterns for each of these

cases were observed by using new smoke filaments. Parameters which were varied in their study were the fin spacing as well as the height, respectively. Welling and Wooldridge [11], who was investigated large arrays with comparable fin heights, confirmed the findings of [10] for the vertically based fin array orientation of the geometry. The natural convection as well as radiation heat transfer from twelve large vertically based fin arrays was investigated by the Chaddock [12].

Another study by Aihara [13, 14] investigated the natural convection as well as radiation heat transfer from an eleven large vertically based fin arrays. In the former work [13], He was to be conducted an experimental study of the heat transfer from the base plate in experiments. The effect of fin geometry and temperature on the average heat transfer coefficient has been studied and an empirical correlation was to be obtained. In their latter work [14], a series of experiments were focused on heat transfer from the fins reported. Based on their experimental data, they finding a correlated average Nusselt number.

Leung et al. [15, 16, 17, 18, and 19] and Van de Pol and Tierney [20] were investigated mostly focused on the effects of varying fin geometrical parameters, mainly, the array, and base plate orientation. They finding a relationship for the Nusselt number based on the fin spacing for different ranges of Rayleigh number Ra.

An important role plays in the heat transfer from fin arrays mainly in Radiation heat transfer. This has been shown by Edwards and Chaddock [21], for different value of Rayleigh number Ra as shown in following equation,

$$Nu_L = 0.59\, Ra_L^{\frac{1}{4}}$$

Chaddock [12], Sparrow and Acharya [22] a correlation developed, as shown following Eqation between Nusseltnumber and the Rayleigh number Ra, Saikhedkar and Sukhatme [23], Sparrow and Vemuri [24, 25], The common conclusion of the aforementioned studies was that the radiation heat transfer contributes between 25–40% to the total heat transfer from fin arrays in naturally cooled heatsinks.

$$Nu_s = 6.7\times10^{-4} Ra_s \left[1-\exp\left(\frac{7460}{Ra_s}\right)^{0.44}\right]^{1.7}$$

*1.2 Problem Statement:*

A schematic of the considered fin geometry with their salient geometric parameters is shown in Fig. 1.4. When theheatsink is heated, the buoyancy force causes the surrounding fluid to start moving therefore as a result thermal boundary layers start to develop at the bottom edges of the fins. The boundary layers mostly merge if the fins/channels are sufficiently long (continuous), creating a fully developed channel flow. Staggered Interrupted fins, therefore, disrupt the thermal boundary layer growth also maintaining a thermally developing flow regime, which can lead to a higher natural heat transfer coefficient.

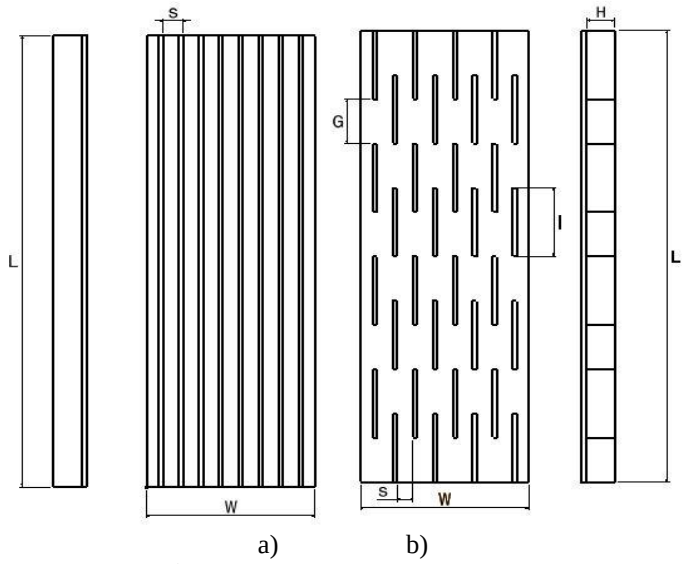


Fig. 1.4. Schematic of the considered heatsink geometry, a) continuous rectangular fin heatsink; b) Staggered interrupted rectangular fin heatsink

To investigate the effects of fin interruption in 3-D, we started by using the existing analytical models of to calculate the heat transfer rate. The idea is to be decoupling the effect of fin spacing from the staggered fin interruption. We also independently investigated the effect of staggered fin interruption experimentally.

## II. EXPERIMENTAL SETUP

The experimental study is to investigate the effects of fin interruption from the natural convection heat transfer of the considered rectangular vertical staggered fins. To enable this investigation, new custom-made test bed was designed. A number of heatsinks and single wall samples, with various geometrical parameters, were prepared. The first series of tests was designed to investigate the effect of staggered interruptions in 3-D and their comparison to the non-interrupted (continuous) 3-D channels (shown in fig 2.3 a forcontinuous and b for staggered interrupted model). The second series of tests were undertaken to validate the numerical data used for calculating the Nusselt number for the vertical fins.

Fig 2.1 Actual Experimental Set-Up

The tested heatsinks are machined from aluminum alloy with a thermal conductivity of 130 W/m K and emissivity of 0.09 at 200C. A new custom-made testbed is designed for measuring natural convection heat transfer from heatsinks as shown in Fig. 2.1.The setup consists of an arrangement which starts with the dimmer stat. The duct which is made of plywood is the testing area and it enclosed the heated air. In the centre of the duct, an assembly of the test plate

followed by the plate heater has been arranged so as to facilitate proper heating of the heatsink, the plate heater is used to heat the MS plates to the desired temperatures as shown in Fig.2.2. It is based on Joule's effect. It consists of highly resistive coil of some alloys such as nichrome, tungsten etc. Of voltage 230volt &Power 440W.

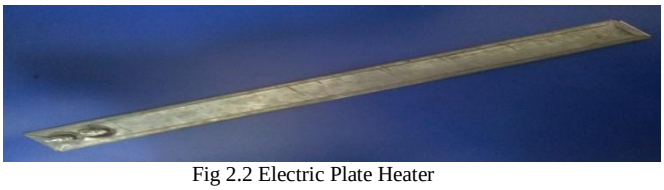

Fig 2.2 Electric Plate Heater

Six thermocouples type - K type (model no. TP-01A)have been connected to measure temperatures at various points in the plate, which are connected to a temperature indicator. A dimmer stat is connected to the plate heater so as to control the input power to the plate.

*2.2Experimental Procedures*

• The experimental set up is assembled and all the electrical connections are made.

• After checking all electrical connections power supply is switched on.

• The controller on the dimmerstat is operated to increase the voltage supplied to the plate heater from zero to a certain value so that the power input to the plate heater is set at 50 W.

•The console on the digital multimeter displays the power input to the heater. The temperature of the plate is continuously monitored until the plate reaches steady state.

• Temperatures of different thermocouples are continuously recorded at a regular interval of 10 min till the steady state is reached. After the steady state is reached, temperatures of different thermocouples are recorded from the temperature indicator display and power rating from multimeter is recorded. After this the power supplied to heater is changed to 60 W by using dimmerstat and whole procedure is repeated again.

## III.NUMERICAL ANALYSIS

Our literature survey indicates that there is a lack of in-depth understanding of the effects of various fin parameters, as shown in Figure 1.4, on the thermal performance of staggered interrupted fin arrays. This thorough understanding is a cornerstone of any comprehensive modeling and development program. As such, we will investigate natural convection heat transfer in the staggeredinterrupted fin array shown in Figs. 1.4. It should be noted that the targeted fin array has all the relevant and salient geometrical fin parameters and covers the targeted fins ranging from continuous straight rectangular fins to pin fins. In the following paragraphs, the physical effect of adding interruptions to the staggered fins on natural convective heat transfer is discussed. On the following subsections, the governing equations, numerical domain and the corresponding boundary conditions, the assumptions and the mesh independency are discussed. Some of the present numerical results are also presented in this chapter and compared against well-established analytical model available in the literature.

*3.1Governing equation and boundary conditions*

The present model consists of the external natural convection heat transfer from a single channel. The conservation of mass, momentum and energy in the fluid are based on assuming a fluid with constant properties and the Boussinesqapproximation for density temperature relation; summarize thegoverning equationsars as follow [26].

Mass $$\frac{\partial p}{\partial t} + div(pu) = 0$$

x- Momentum

$$\frac{\partial (pu)}{\partial t} + div(puu) = -\frac{\partial p}{\partial x} + div(\mu gradu) + S_{mx}$$

Y- Momentum

$$\frac{\partial (pv)}{\partial t} + div(pvu) = -\frac{\partial p}{\partial y} + div(\mu gradv) + S_{my}$$

Y-Momentum

$$\frac{\partial (pw)}{\partial t} + div(pwu) = -\frac{\partial p}{\partial z} + div(\mu gradw) + S_{mz}$$

Internal energy

$$\frac{\partial (pi)}{\partial t} + div(piu) = -pdivu + div(kgradT) + \varphi + S_i$$

Equation of state $p = p(p,T)$ and $i = i(p,T)$

e.g. perfect gas

$p = pRT$ and $l = C_v T$

where y is the direction parallel to the gravitational acceleration and x is the direction normal to the gravitational acceleration, u is the flow velocity in x-direction and v is the flow velocity in y-direction, respectively.CFD Simulation Methodology used is that the finite volume based CFD solver ANSYS FLUENT was used for the simulations in this project work.The gravitational force, caused by the temperature difference between the staggeredFins' surface and the ambient conditions, acting in the domain were modeled using the Boussinesq model. The surrounding box/room surfaces were modeled as Adiabatic, Stationary, No-Slip Wall boundaries. The fluid can enter from the bottom surface and can leave the domain at the top surface. These were modeled using the Inlet / Outlet boundary conditions.In order to improve the accuracy of the simulation, Higher Order discretization schemes were used in the CFD simulations.

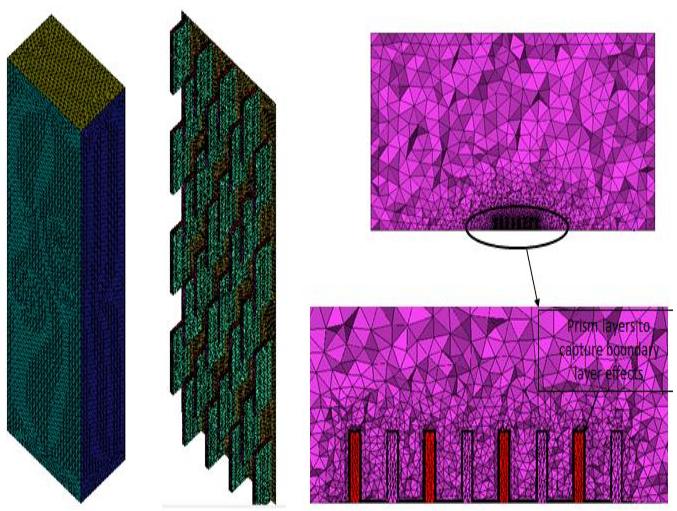

Fig 3.1 Staggered fin meshing

Boundary layer mesh was used for regions that are closer to the fin surface, in order to capture the flow behavior with a higher resolution. Figure3.1 shows a segment of the domain and the generated mesh for continuous and interrupted fins. The meshing type used for staggered fin is Tetra hedral with total mesh elements of 1,065, 996.which shown in fig.3.1.

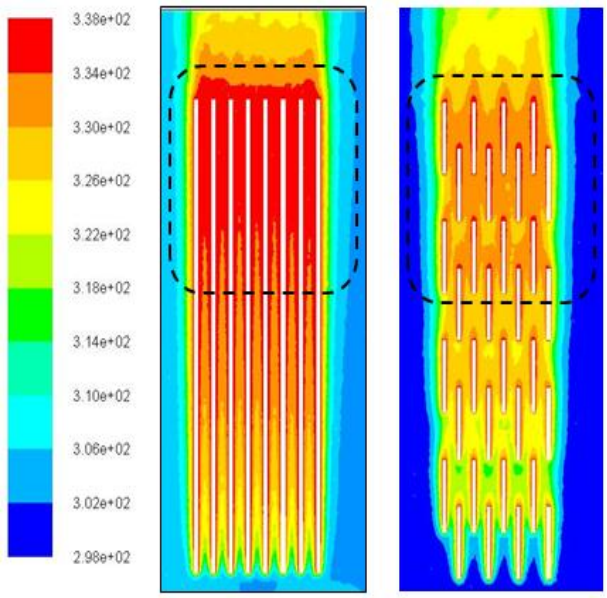


Fig.3.2 Temperature contours for considered fin geometry

The high temperature zones in the continuous fin arrangement&staggered fins are shown by dotted square region in figure no. 3.2.As can be seen in Fig.3.2 the high temperature zones in the continuous fin arrangement had been removed by having the staggered fin arrangement.

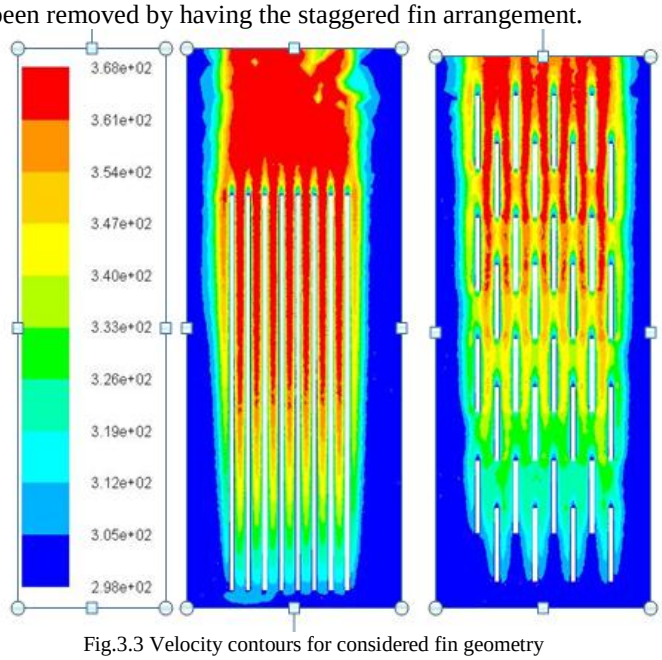


Fig.3.3 Velocity contours for considered fin geometry

.The velocity contours for the continuous fin arrangement&staggered fins is shown in figure no. 3.3.As can be seen in Fig.3.3 the accumulation of flow in the continuous fin arrangement has been removed by having the staggered fin arrangement. It was observed that the heat transfer rate of staggered interrupted fins is greater than continuous fins.

In order to improve the accuracy of the simulation, Higher Order discretization schemes were used in the CFD simulations, the set-up is shown in the image

## IV. RESULT & DISCUSSION

*4.1Continuous plate*

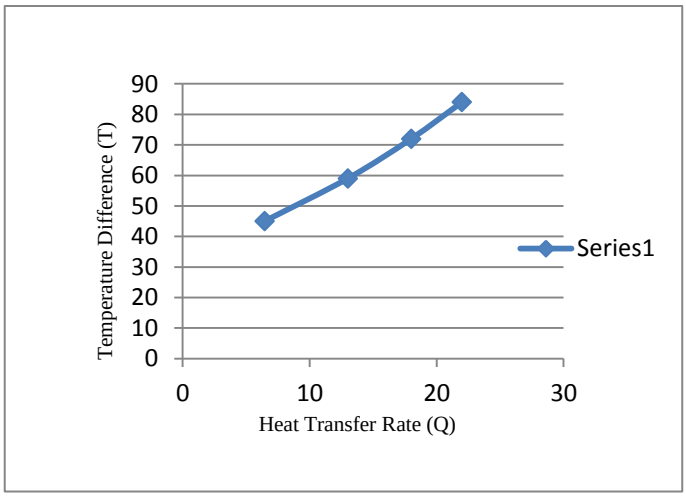


Fig. 4.1: Experimental data continuous fins(s = 9.5 mm, L =305 mm, H = 17 mm)

Fig.4.1 shows the variation of heat transfer rate with temperature difference over the plate to attaining the steady state condition. It can be seen that the heat transfer rate increases as the temperature difference increases over the surface of the plate increases.

*4.2 Interrupted plate*

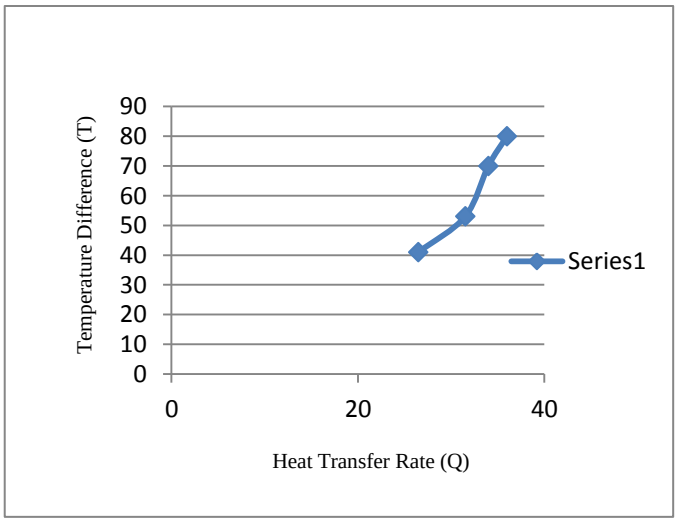


Fig.4.2experimental data for fins interrupted fins (n = 8, s = 9.5 mm,L = 305 mm, H =17 mm, l =37 mm, G =30 mm).

Fig. 4.2 shows the variation of heat transfer rate with temperature difference over the plate to attaining the steady state condition. It can be seen that as increase in temperature difference the heat transfer rate increase.

This variation can be observed due to the fact that because of the presence of interrupted over the plates, the turbulence created by the air over the plate increases thus increasing the heat transfer rate. Moreover, the area is contact is increased, which in turn increase the heat transfer rate.

*4.3Staggered plate*

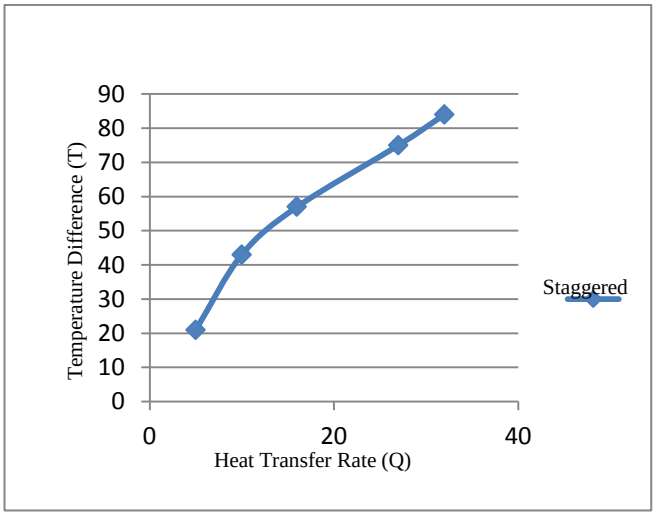


Fig. 4.3experimental data for staggered fins (G=30mm)

Fig. 4.3 shows the variation of heat transfer rate with temperature difference over the plate to attaining the steady state condition. It can be seen that as increase in temperature difference the heat transfer rate increase.

This variation can be observed due to the fact that because of the presence of interrupted over the plates, the turbulence created by the air over the plate increases thus increasing the heat transfer rate. It was observed that the heat transfer rate of staggered interrupted fins greater than continuous fins.

*4.4Comparison between continuous plate and staggered plate*

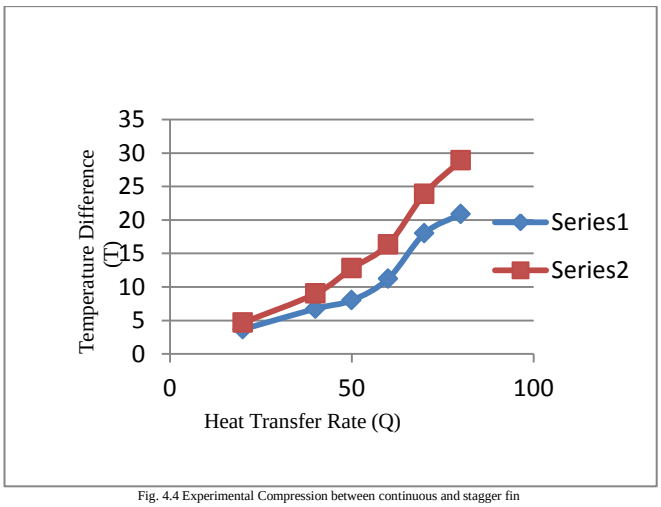


Fig. 4.4 Experimental Compression between continuous and stagger fin

Fig.4.4 series 1 & series 2 show the variation of heat transfer rate with temperature difference over the plate to attaining the steady state condition forcontinuous fins and staggered interrupted fins respectively. It can be seen that as increase in temperature difference the heat transfer rate increase.

The staggered fin arrangement improves the heat transfer rate between 37 – 56 % as compared to the continuous fins. The results are shown in Fig.4.5.

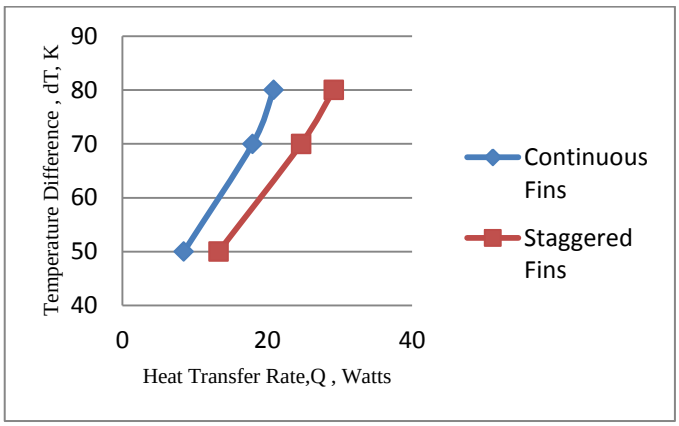


Fig.4.5 Numerical simulation validations of fins

As compare to Fig. 4.4 and 4.5 showing experimental & Numerical results respectively.

Increase in the surface area available for heat transfer and reduction in the hydrodynamic resistance for the fluid flow over the surface, resulting in less pressure drop.

The vortices formed inside the staggered results in thinning and to disturb the thermal boundary layer formed over the surface during coolant flow and serve ultimately to bring about enhancement of heat transfer between the fluid and its neighbouring surface at the price of less increase in pressure.

## V.CONCLUSION

The effects of interruptions on vertically-mounted rectangular staggered interrupted heatsinks were studied experimentallyand.numerically. Thestaggered interruptions will enhance the heat transfer rate by resetting/interrupting the thermal as wellas hydrodynamic boundary layers.The purpose of these interruptions is to reset the thermal boundary layer associated with the fin in order to decrease thermal Resistance. Our experimental and numerical results shows an increase in heat flux from the heatsink when staggered interruptions were added. The entire experiment on the apparatus was performed successfully and effectiveness of all plate for different sets of input parameters was evaluated. It is observed from the experimental results that the convective heat transfer rate is enhanced as staggered interruption change the flow. This can be attributed to the fact thereby increasing the heat transfer rate from the heated plate. As compared to the continuous flat plate, there is marginal yet significant rise in the convective heat transfer coefficient of heat transfer increase with change the dimension. As result in saving material cost and reduces weight of plate. The experimental and numerical results show a considerable increase in the heat flux from a heatsink compared to the equivalent continuous heat flux from heatsink. To support and validate the results obtained from the experiment, CFD simulation of the experiment was performed successfully and those results were compared with that obtained from the experiment.

## NOMENCLATURE

| | |
|---|---|
| A | surface area, [m2] |
| g | gravitational acceleration, [m/s2] |
| G | fin interruption length, [m] |
| h | convection heat transfer coefficient, [W/m2 K] |
| H | fin height, [m] |
| I | electrical current, [A] |
| k | thermal conductivity, [W/m K] |
| l f | in length, [m] |
| L | enclosure length, [m] |
| n | number of interruptions |
| N | number of fins per row |
| Nu | Nusselt number |
| P | pressure, [Pa] |
| Pinput | input power, [W] |
| Pr | Prandtl number |
| Q | heat transfer rate, [W] |
| Ra | Rayleigh number |
| s | fin spacing, [m] |
| t | fin thickness, [m] |
| T | temperature, [K] |
| u | flow velocity in x direction, [m/s] |
| v | flow velocity in y direction, [m/s] |
| V | electrical voltage, [V] |

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