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Convection Heat Transfer within Enclosures – An Analysis Review

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Abstract - *A very notable growth in electronics and computing technologies has been noticed for the last two decades and it is still continuing exponentially. This growth of these technologies has been reported to cause the enhanced rate of heat generation and thermal management problems. The parameters like fin length, fin spacing, Rayleigh number, type of working fluid used and the enclosure orientation are the most influencing parameters for convective heat transfer from the enclosures. This paper collects and analyzes the bodies of works written about natural convection phenomenon within the enclosure to study the effects of such parameters.*

Key Words: **Natural Convection, Thermal management, Parametric Design**

1. INTRODUCTION

Natural convection in the narrow enclosures has a wide area of its engineering applications in the present scenario. During past two decades, it has been turned out to be one of the major areas of research and still being motivated by the modern compact trend causing the thermal management problems to occur. Attachment of fins enhances the convective heat transfer due to increased effective surface area. Numbers of exhaustive experimental and numerical analyses are available regarding the study of free convective heat transfer for different influencing parameters like shape of enclosure, working fluids, boundary conditions, fin materials at numerous geometrical parameters like fin length, fin spacing, fin thickness, fin height at different enclosure orientations.

For significant convective heat transfer between the plates, the temperature difference imposed between the hot and cold plates must exceed a critical value and the Rayleigh number at this instant is known as critical Rayleigh number. Bénard cells come in to the picture above critical Rayleigh number [1]. Bénard cells are the cellular flow consisting of counter rotating two dimensional cells. They occur in plane horizontal layer of fluid heated from below and Buoyancy (gravity) is responsible for these convection cells. The important fact to be taken under consideration during design of setup is that the enclosure height must be less than the enclosure dimensions for proper circulation of Bénard cells. An extensive review on heat transfer in nanofluids was carried out by Das et al. [2] and the effects of volume fraction and temperature variations on thermal conductivity ratio were summarized for different nanofluids. The microscalic heat transfer was reviewed adequately by [3, 4].

Mukutmoni and Yang [5] observed the bifurcation sequence steady state periodic to quasi-periodic steady state with increase in Rayleigh number and reported the maximum oscillation amplitudes at the top corners along the vertical plane of symmetry perpendicular to the roll axes. The maximum oscillations for quasi-periodic regime were reported along the mid-plane causing the roll separation. The flow field was transferred from steady state to periodic at higher Rayleigh number. Literatures [6-8] concluded the flow pattern transitions consisting of a three step sequence - one cell steady to two cells steady followed by two cells oscillatory and one to three cells oscillatory flow patterns - through a numerical investigation on an air filled two dimensional rectangular enclosures heated from below and cooled from above. An abrupt reduction in the Nusselt number was concluded for the flow transition from one cell to two cells steady state. Although, the maxima of Nusselt number curve was reported before the first transition from one cell to two cells steady solution because of the progressive distortion of the one cell pattern leading to the reduction in heat transfer rate. Also, the heat transfer reduction was observed during the flow transition from one cell to three cells oscillatory solution. Ramesh and Venkateshan [9] reported the highest and least local heat transfer rates along the hot wall at the bottom and top of the wall respectively due to the growth of boundary layer along the hot wall of enclosure.

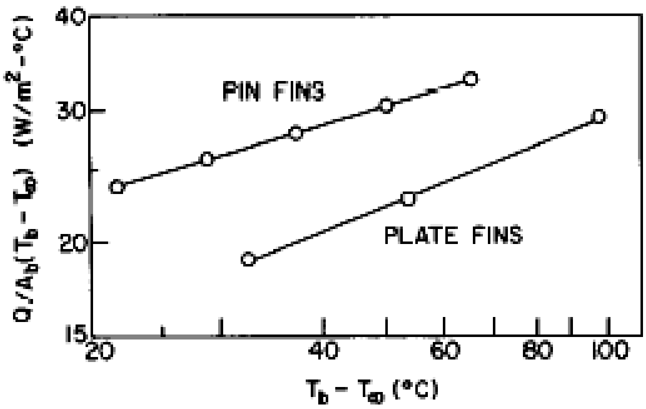

Chart-1: Comparison of convective heat transfer from pin fins and plate fins mounted on vertical base plates [17]

Quéré [10] gave the ideas regarding the onset of unsteadiness to the disorder in flow with in enclosures. The natural convection heat transfer with in porous medium filled

enclosure was presented in the literatures [11-16]. A Comparative analysis for the pin-fin results to the results for plate fins was done by Sparrow and Vemuri [17] which lead to encourage the use of pin fins as reported in chart 1.
This paper presents an exhaustive review of both numerical and experimental studies carried out regarding natural convection phenomenon in an enclosure to study the effects of influencing parameters.

2. EFFECT OF VARIOUS INFLUENCING PARAMETERS VARIATION

2.1 Effect of Fin Height

Natural convection heat transfer within the enclosures as a function of fin height was observed by the literatures by Nada [1], Terekhov and Terekhov [18], Yazicioğlu and Yüncü [19]. The dependence of heat transfer on fin height has been illustrated in chart 2 as proposed by Dogan and Sivrioglu [20] which reflects net convective heat transfer augmentation with increase in fin length rather the rate of heat transfer augmentation was noticed to be decreased. This effect was attributed to the increased effective surface area for heat transfer causing heat transfer augmentation. Also, the hindrance to the fluid flow due to enhanced fin height caused the slower rate of heat transfer enhancement. The similar results were also concluded by the study conducted by Mobedi and Yüncü [21] for air filled horizontal base arrangement and this effect was attributed to the reduced amount of air entering from the middle part between the fins causing diminished heat transfer enhancement rate. These results were found in very excellent validation with Harahap and McManus [22].

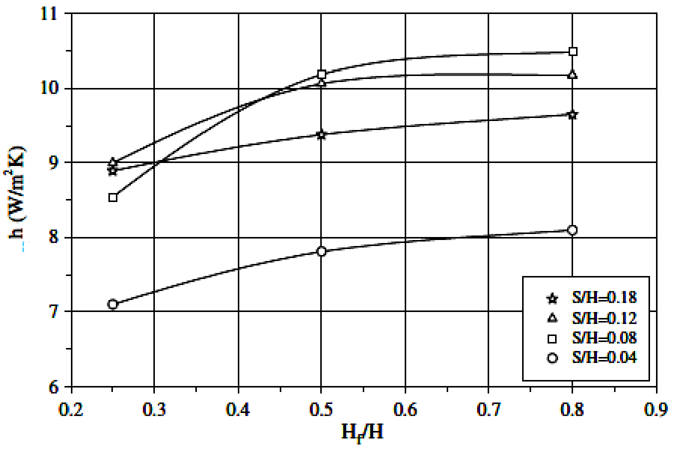

Chart-2: Variation of heat transfer coefficient as a function of fin height [20]

Dogan and Sivrioglu [20] carried out an experimental parametric study for an arrangement consisting of rectangular fin arrays heated from below in a horizontal channel and average heat transfer enhancement with fin height was concluded. This result was attributed to increased effective surface area as well as development of the buoyancy driven flows due to increased fin height. Fin height associated with the optimum fin spacing, for least material fins with constant heat transfer coefficient, was correlated by Bar-Cohen [23].

2.2 Effect of Fin Spacing Variation

The effect of the fin density variation over heat transfer was explained by number of experimental parametric studies as done by Nada [1], Sparrow and Vemuri [17], Dogan and Sivrioglu [20] for rectangular fin arrays heated from bottom side of the enclosure at broad ranges of influencing parameters. The heat transfer rate was noticed to increase up to an optimum value of fin spacing then it was decreased on further increase of fin spacing as illustrated in chart 3. This effect was attributed to the dominated increase in surface area effect to the increased hindrance effect to the flow up to an optimum value then the reversal was occurred.

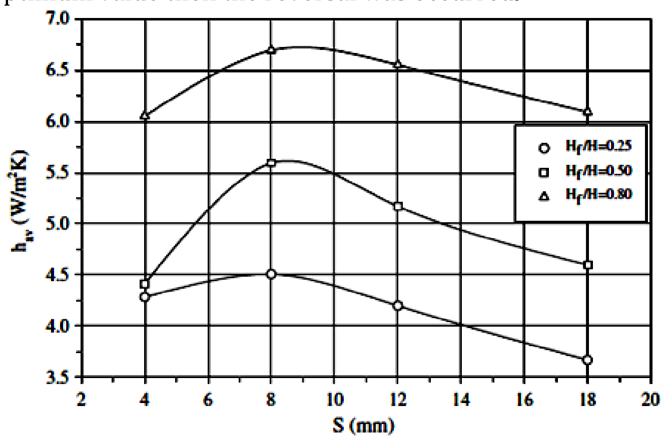

Chart-3: Convective heat transfer as a function of fin spacing [20]

It was also concluded that for a given temperature difference, the convective heat transfer rate from fin arrays takes on a maximum value as a function of fin spacing and fin height and an optimum value of fin spacing is always available for every fin height. Yazicioğlu and Yüncü [19] reported a range of optimum fin spacing depending upon Rayleigh number and fin height. Mobedi and Yüncü [21] reported the heat transfer enhancement with fin spacing due to retardation of the interference of boundary layer and fresh air was permitted to enter from the space between the fins. A two dimensional numerical analysis for heat transfer in finned square enclosure with water as a working fluid was performed by Ho and Chang [24]. The reduced heat transfer as compare to bare enclosure was reported due to weaker fluid flow due to increased fin density. Also, this reduction was observed to decrease with increase in Rayleigh number. The insensitiveness of fin spacing to the base to ambient temperature difference variations was discussed by Yüncü and Anbar [25]. He also reported that fin density is weakly sensitive to fin height and strongly sensitive to fin length. The optimum arrangement of rectangular fins on horizontal surfaces for free convective heat transfer using interferometric technique was found out by Jones and Smith [26]. A major advantage of using interferometric technique to study is that the heat transfer coefficient derived is for convection phenomenon only and they are independent of radiation. He reported fin weight as one of the important influencing parameter. He showed that the maximum heat transfer rate was observed for tall and light weight within the optimized spacing range. A correlation for optimum fin spacing was also proposed by Tamayol et al. [27] as-

$S_{opt}= 0.0231\ (T_w-T_a)^{-0.236}$

2.3 Effect of Rayleigh number variation

The heat transfer dependence on Rayleigh number was examined experimentally and analytically by different literatures as Nada [1], Hortmann and Scheuerer [8], Harahap and McManus [22], and the heat transfer enhancement with Rayleigh number increase was reported. Chart 4 reflects the Nusselt number variations with respect to Rayleigh number as reported by Mukutmoni and Yang [5]. It is quite clear from the graphs the rate of heat transfer augmentation was reduced with increase in Rayleigh number. A numerical analysis regarding triangular enclosures with protruding isothermal heater was done by Arnold et al. [28] and a constant Nusselt number for smaller value of Rayleigh number was observed which was increased with Rayleigh number. According to the computational analysis did by Srinivasan et al. [29], the increase in air velocity with increase in Rayleigh number, causing heat transfer enhancement from the cold wall. The increase in Nusselt number ratio (NNR) was noticed to be almost constant at $Ra=10^7$. As per conclusion by Wakitani [30] the optimum fin density value was observed virtually independent of the Rayleigh number for the horizontal fin array, although for the vertical fin arrays there was a probability for optimum fin spacing to occur at higher fin density with increase in Rayleigh number. The convergence of the plumes rising from each heated section was reported by Tric et al. [31] near the low surface at low Rayleigh numbers. However, at higher Rayleigh number, the plumes above each surface were observed interacting but not truly merged leading to the unsteady flow. An experimental and numerical analysis regarding the natural convection within an inclined non rectangular enclosure was carried out by Oztop et al. [32] for a wide range of influencing parameters. The increase in Rayleigh number leading the skewness to the temperature and velocity profiles for trapezoidal enclosure in the horizontal position was observed, while the viscous effects due to presence of fluid walls were neglected as per conclusion of [33, 34] regarding the absence of viscous effect for enclosure width to length ratio greater than 1.

The increase in waviness of the streamlines pattern at Ra>900 was concluded by Chu et al. [35] for an inclined trapezoidal enclosure filled with fluid saturated porous medium and initiation of a multi cellular structure in the middle portion of the enclosure was also observed. Nusselt number augmentation with increase in Rayleigh number up to a maximum value was observed at enclosure tilt angle (α=60°) and then it was reduced. An experimental and numerical analysis of natural 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 [36]. The conductive heat transfer was reported to prevail for $Ra<10^4$ while the convective heat transfer began for $Ra>10^4$ and the mushroom profile of the isotherms at $Ra>10^5$ was observed. The low Nusselt number values with increase in Rayleigh number were observed on the heat source which were increased at the boundaries reflecting the existence of high temperature gradients with intense recirculation of the fluid forming V shaped plot. The similar results were attained for the numerical study done by Khanafer et al. [37] for square enclosure having nanofluids as working fluid heated from the left vertical wall and the heat transfer augmentation with increase in Rayleigh number was reported because of increased flow strength.

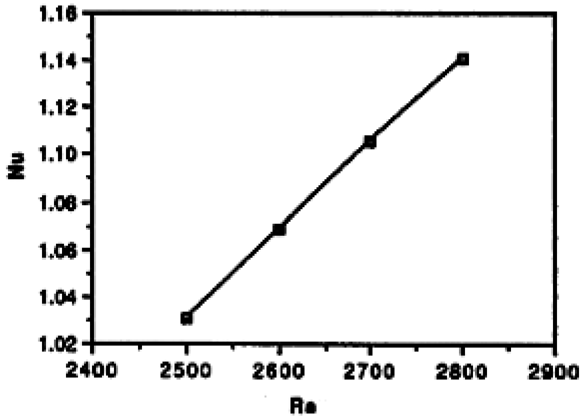

Chart-4: Average Nusselt number over a horizontal cross-section as a function of Rayleigh number [5]

As per analytical study did by Shohadaee [38] the symmetric flow field at low values of Rayleigh number for a square cavity diagonally divided by an inclined plate filled with porous medium was observed. A weak circulation with increase in Rayleigh number was initiated owing to hot vertical wall causing the increased interaction between hot and cold walls and increased heat transfer rate vice versa. Also, the formation of a bell-shaped trend along the inclined plate for local Nusselt number was examined which became wider at higher Rayleigh numbers indicating the reduced effects of the partitions. The effect of Rayleigh number variations on heat transfer in square and rectangular enclosures having air as a working fluid were reported by Shiralkar and Tien [39]. It was concluded that for the square enclosures the boundary layer formation at the top wall of the plate, initiating from the edges, was converged at the center of plate causing the formation of laminar vertical plumes attached to the plate which were noticed to be thick and having a wide base area on the isothermal plate therefore, the shifting of the location of the plume detached from the top wall towards the middle of the plate was reported with increase in Rayleigh number. The conduction regime associated with very weak flow was examined to be in heat transfer inside the base of detaching plumes causing a very low heat transfer rate at the top wall. The steady increase in heat transfer rate from the top and bottom walls was concluded. Also, the enhanced heat transfer rate was reported with increase in Rayleigh number for rectangular enclosure due to reduced thickness of thermal boundary layer. The Maximum local Nusselt number was reported in the vicinity of the top wall and was declined steadily along the cold wall towards the horizontal plate level. For rectangular enclosure the Nusselt number enhancement was reported due to temperature gradients exaggeration along the hot and cold walls. A correlation reflecting the dependence of Nusselt number upon Rayleigh number for all aspect ratios is as below as suggested by M. Ciofalo and T. G. Karayiannis [40]-

Nu=1 (for $Ra<10^3$)

Nu= f(A) $Ra^{1/4}$ (for $Ra>10^4$)

2.4 Effect of Enclosure Orientation

Literatures [41-47] reported the enclosure orientation as one of the most important parameters influencing the natural convective heat transfer within the enclosure. Mukutmoni and Yang [5] compared the heat transfer rate for in-line and staggered arrangements for horizontal and vertical orientations. He concluded the more heat dissipation for in-line horizontal arrangement of the pins as compare to vertical arrangement while the more uniform distribution of heat dissipation was observed for staggered orientations. As per the conclusion drawn by the parametric experimental study done by Sparrow and Vemuri [17] the actual differences in the heat transfer performance of the three orientations (horizontal fins with vertical base plate, vertical fins with horizontal down facing base plate and vertical fins with horizontal up facing base plate) were reported to depend on both fin density and the Rayleigh number. Amongst the three orientations, the vertical up facing fin array yielded the highest heat transfer rates followed by the horizontal fin arrays and the vertical down facing fin arrays respectively. The optimal fin density and for practical values of the Rayleigh number, the heat transfer was reported about 15% smaller for horizontal arrangement as compare to vertical and up facing and the heat transfer for the vertical down facing array was noticed about 20% less than that for the vertical up facing array. According to the analysis done by Varol et al. [48] the heat transfer rate was observed more attenuated at plate positioned at 45° as compare to plates at 135°. The highest temperature gradients were reported in the vicinity of hot and cold walls at α=0° which was decreased with increase in inclination angle causing lowering of heat transfer through the walls. As per conclusion of the study carried out by Altac and Kurtul [49] the heat transfer enhancement was reported up to 22.5° after which a steady decrease was observed for all Rayleigh number values. Similar results were reported by Heindel et al. [50]. The dependence of Nusselt number over enclosure tilt angle has been illustrated in Chart 5 as reported by Baytas and Pop [53]. The maximum and minimum values of Nusselt numbers were observed at tilt angle values of 180° and 270° respectively as by literature of Lee [51]. The existence of a broad maxima within the tilt angle ranging in between 45° to 60° was suggested by Biondi and Ciofalo [52] through his experimental analysis while for tilt angle range of 75° to 90°, a slight change in the slope of Nusselt number curve was reported. Baytas and Pop [53] observed that the flow pattern was changed from one cell at α=165° and Ra=100 to a two cells at α= 15°, 135°, 165° and Ra= 500 and 900.

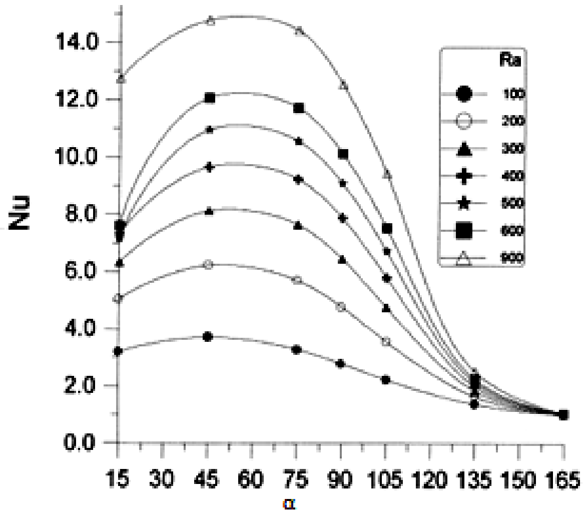

Chart-5: Effect of Tilt angle variation on Nusselt number [15]

Nada and Moawed [54] analyzed the heat transfer within the tilted rectangular enclosures heated from the bottom wall and with three different venting arrangements experimentally at constant heat flux to study the effect of different operating ratios at different tilt angles (α= 0°-180°). The decrease and increase in Nusselt number for top and side venting arrangements respectively was reported with increase in tilt angle up to a critical value. The heat transfer due to side venting arrangement was reported dominating the heat transfer due to top venting arrangement beyond this critical value.

2.5 Dependence on Volume Fraction of Nanofluids

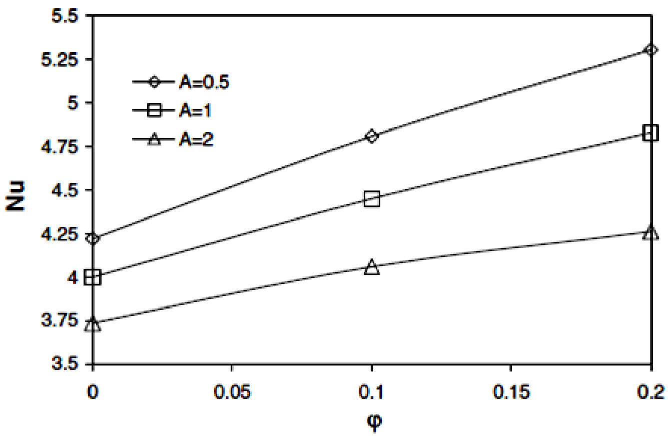

Chart-6: Variation of mean Nusselt number with respect to volume fraction for Cu-water nanofluid at different aspect ratio [55]

A numerical analysis regarding the free convective heat transfer within the nanofluid filled and left vertical wall heated square enclosure was carried out by Oztop and Abu-Nada [55]. The Nusselt number variation with respect to volume fraction has been drawn in chart 6. The study was carried out for three different nanoparticles: Cu, Al_2O_3 and TiO_2. The monotonic enhancement in heat transfer with volume fraction (φ) was concluded for all Rayleigh numbers and nanofluids. The increasing order of heat transfer was reported as pure fluid < TiO_2 < Al_2O_3 < Cu nanoparticles. It was also examined that at higher Rayleigh number values the mean Nusselt number difference was increased with increase in volume fraction of nanoparticles and this result was attributed to the dominated convective heat transfer. Also, the heat transfer enhancement with increased volume fraction of nanoparticles causing increased thermal conductivity was noticed. These results were also supported by the literature [56]. Literatures [57,58] also suggested the similar results. The increase in optimum fraction value (φ_{opt}) with decrease in the nanoparticle size was concluded by Corcione [59]. This enhancement was reported more remarkably with increase in both the nanofluid temperature and the slenderness of the enclosure. With increase in volume fraction of nanoparticles was examined with increase in F(Ra) while at smaller volume fractions F(Pr) was examined almost unchanged. Increased F(Pr) with volume fraction was also predicted at higher values of volume fractions. The increase in heat transfer enhancement was reported with decrease in volume fraction up to a certain value (φ_{opt}) then it was decreased on further decrease in volume fraction value. This effect was attributed to the dispersion of large amount of solid nanoparticles into the base liquid at initial stage causing increase in heat enhancement because of increased thermal conductivity of the nanofluid. On further decrease in volume fraction the decreased heat transfer enhancement rate was attributed to the excessive growth of the nanofluid effective viscosity. The Nusselt number variation with nanoparticle diameter has been drawn in chart 7.

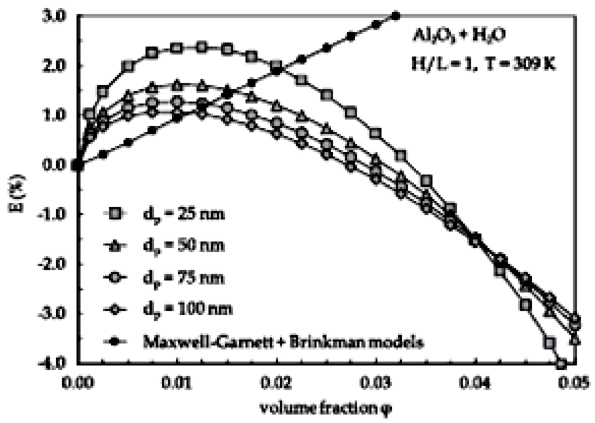

Chart-7: Distribution of heat transfer enhancement for a square cavity at different nanoparticle diameters [59]

3. SUMMARY AND CONCLUSIONS

There is an ever growing need to increase the heat flux removal level for thermal management of electronic devices or chips. This review paper provides a better understanding of various influencing parameters and their effects on heat transfer rate.

Some of the major conclusions are enlisted as below:

(1) Convective heat transfer rate increases with increase in fin height.
(2) Convective heat transfer rate initially increases with increase in fin spacing up to a maximum value after which it starts decreasing if fin spacing is increased further.
(3) Convective heat transfer augmentation occurs as Rayleigh number increases.
(4) Increase in volume fraction enhances the heat transfer rate.

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