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Enclosure Phenomena in Confined Natural Convection

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ABSTRACT: *Through proper experimentation, the role of an external enclosure on confined natural convective heat transfer on a square flat plate is explored. The effect and the extent of effect of different external enclosure on heat transfer rates is investigated. The phenomenon is articulated in terms of deviations in convective heat transfer coefficient. The role of controlling parameters viz., plate orientation, surface roughness, enclosure distance, type of enclosure and enclosures in distinct configurations is probed and optimized for wide-ranging applications. To simplify the heated surface orientation and related heat transfer analysis, a novel zonal system with respect to the surface orientation is proposed. Results indicate that enclosures significantly affect the transportation of heat from source under varying conditions. Flow behavior with the respective variations is understood to play formidable role in energy transference. Smoother surfaces are useful in conservation of heat and with increasing surface orientation becomes more effective in transfer of heat.*

Keywords: *Natural convection, enclosure, heat sink, heat transfer coefficient.*

I. INTRODUCTION

Heat transfer is one of the fundamental sciences of practical and functional importance. It is necessary to understand the transfer and conservation of heat energy for different working systems under diverse conditions.

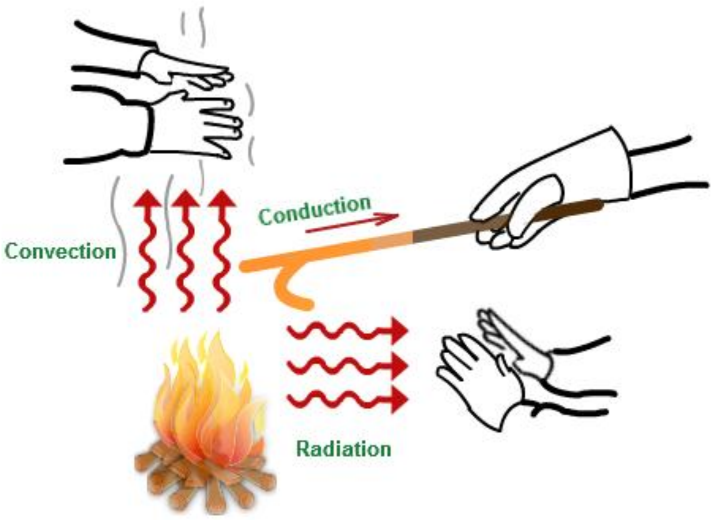


Figure 1: Schematic of modes of heat transfer.

Respective modes of heat transfer work as a cumulative sum and the dominant one redefines the governing principles of any working system (figure 1). The heat transfer theory points to predict the energy transfer that takes during transfer of heat. Of respective modes, convective heat transfer refers to the subjective heat transportation between a hot body and the surrounding fluid. This transference of heat is studied in two domains viz., natural convection and forced convection (please see figure 2). The natural mode refers to the fluid motion by buoyant forces arising due to density gradients as a result of temperature gradients. Whereas, forced convection marks the boosted fluid drive as upgradation of natural convection for enhanced heat transfer.

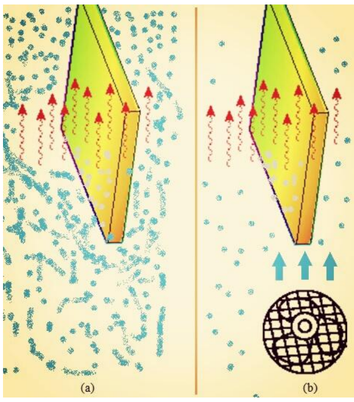


Figure 2. Schematic of (a) Natural convection (b) forced convection.

Natural convection is prominent in nature with applications ranging from the need of cooling to heating under different conditions. The heat transfer for most of cases is studied by investigating the heat transfer coefficient. Some of the prominent applications includes heat exchangers, power plants, reactor cores cooling, turbine blade cooling, automobile engines, cooling of electronic chips and transistors, high voltage electric transformers and many household applications. An interesting aspect in these applications is convective heat transfer in the presence of an enclosure. The enclosures are likely to act as an external heat influence viz., a heat sink by taking a part of heat transferred and are likely to affect the heat transfer with due interactions. With natural convection, the presence of an enclosure partially obstructs and redirecting the hot flow to affect the primary heat transfer (please see figure 3).
Confined natural convection concerns with the secondary fluid generation and related influence on transfer of heat. The phenomenon is widely encountered in wide range of applications.

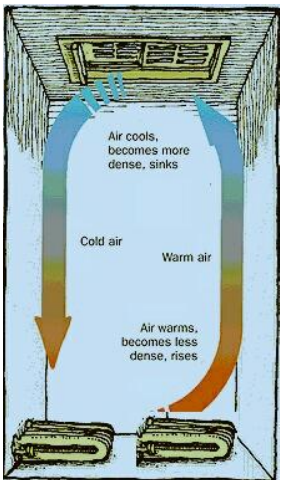


Figure 3. Schematic of the flow behavior in presence of an enclosure in natural convection

Though, most of the convective heat transfer problems deals with the confined natural convection influenced by the presence of an external enclosure and is *an issue yet to be comprehensively addressed*. The intricacy of the problem deals with uneven heat and mass interactions and thus had prevented a comprehensive understanding.

Following the classical work of Ostrich [1] and Kierkus [2] over laminar free convection heat transfer on plates, appreciable research efforts have contributed to the understanding of the confined convective heat transfer. The contributions are reported in several reviews like [3] - [11]. The works provide an excellent review on the developments up to the end of the century. Raos [12] carried out investigation on the laminar natural convection phenomena in enclosed spaces. A 2-D rectangular object with differentially heated sides and adiabatic horizontal walls was selected as real physical enclosure. Results of the study presented good base for definition of the object parameters in engineering practice containing natural convection phenomena. Bazylak et al., [13] presented computational analysis of the heat transfer due to an array of distributed heat sources on the bottom wall of a horizontal enclosure. The heat sources were modeled as flush-mounted sources. Optimum heat transfer rates and the onset of thermal instability triggering various regimes was found to be governed by the length and spacing of the sources and the width-to-height aspect ratio of the enclosure. Spacing equal to that of the source length was noted to provide effective convective heat transfer.

Mariani and Coelho [14] carried out a numerical study to investigate steady heat transfer and flow phenomena of natural convection of air in enclosures with varying aspect ratios and a local heat source on the bottom wall. The heat source occupied 1% of the total volume of the enclosure and the vertical walls in the enclosures were insulated. Results showed that the convection is influenced by the temperature difference between the left and right walls. The presence of different flow patterns in the enclosures and the flow and heat transfer was seen to be controlled by the external heating. Kandaswamy et al., [15] numerically explored unsteady laminar natural convection in an enclosure with partially thermally active side walls and internal heat generation. Nine different combinations of the hot and cold thermally active zones were considered. It was observed that the heat transfer rate increases with increasing the Grashof number due to an increase in buoyancy force and decreases with an increase in heat generation. The heat transfer was found to be the maximum when the hot and cold thermally active locations were placed at the middle of the side walls.

Abu-Nada et al., [16] explored the influence of inclination angle for a square enclosure. Inclination angle of the enclosure was detected as a control parameter for the fluid flow and heat transfer. Gdhaidh et al., [17] performed a numerical study of natural convection heat transfer in water filled cavity by using an array of parallel plate fins mounted to one wall of a cavity. A cold plate was used as a heat sink installed on the opposite vertical end of the enclosure. The fins were installed on the substrate to enhance the heat transfer. The results illustrated that as the fin number increases the maximum heat source temperature decreases. When the fin number was increased to a critical value the temperature started to increase as the fins were too closely spaced and that caused the obstruction of water flow. The introduction of parallel plate fins was noted to reduce the maximum heat source temperature by 10% compared to the case without fins. In recently, Heidary et al., [18] presented study on natural convection heat transfer fluid flow and entropy generation in a porous inclined cavity in the presence of uniform magnetic field. For control of heat transfer and entropy generation, one or two partitions were attached to the horizontal walls. The left wall of enclosure was heated and right wall was cooled isothermally with adiabatic horizontal walls. The influence of controlling parameters viz., inclination angle, partition height, irreversibility distribution ratio, and partition location was investigated on the heat transfer characteristics and the entropy generation. The results indicated that the partition, magnetic field and rotation of enclosure can be used as control elements for the heat transfer, fluid flow and entropy generation in porous medium.

In the light of above mentioned research efforts, as the case is widely observed, there is a needing requirement to address this issue for systems operating under diverse conditions. In most of the convection problems, the heat transfer features are explored on selected entities (viz., flat square plate) open to atmosphere. The interest in this class of problems is primarily driven by the need to have better understanding and efficient utilization of convective heat transfer. The heat transfer characteristics are expected to be altered with varying surface orientation and external enclosure implications. In the present work, the efforts are directed to understand non-linear heat transfer behavior over a square flat plate in the free convection configuration bounded by enclosures. Hence, a systematic study is needed to understand mechanisms controlling the free convective heat transfer under enclosure effect. The specific objectives of the present study are to:

a) Investigate the enclosure effect on confined free convective heat transfer.
b) Analyze the role of key controlling parameters for heat transfer optimization.
c) Enhanced understanding of the operating physics to implement for wide range of applications.

II. EXPERIMENTAL SETUP AND SOLUTION METHODOLOGY

A simple confined natural convection apparatus (Fig. 4(a)) was adapted for this study. The apparatus comprised of (a) base made of mild steel plates which supports the assembly (b) primary fixed enclosure (glass sheets which confine the square plate assembly from four sides and open to atmosphere from top and bottom) (c) digital display and a handhold handle with attached protractor to adjust the rate of heating (Fig.4(b)) and (d) the pilot heat source (a flat aluminum square plate (15 cm x 15 cm) (Fig. 4(c)) with smooth and rough surfaces on either side. A coil is sandwiched in between the plate surfaces and heated using electrical power at desired rate for 2 hours prior to the experimentation.

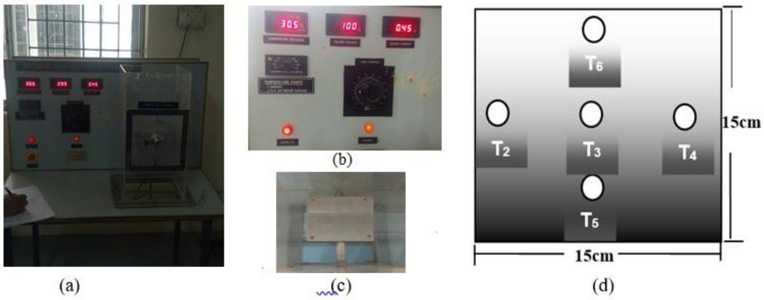


Figure 4. Pictorial view of the apparatus (a) Experimental setup (b) digital system (c) Top view of square plate (d) schematic of square plate with location of embedded equidistant thermocouples (shown by circles)

Entire experimentation was carried out at ambient temperature of 30.5°C for electric input of 100 Volt and 0.45 Ampere. The plate temperature is ensured to be uniform throughout. The heated plate temperature is ascertained with utility of Thermocouples (5 in numbers) embedded in plate (Fig. 4(d)) and located equidistance to embark average plate temperature. For the study, the external enclosures are placed at the top and bottom end of the fixed enclosure.

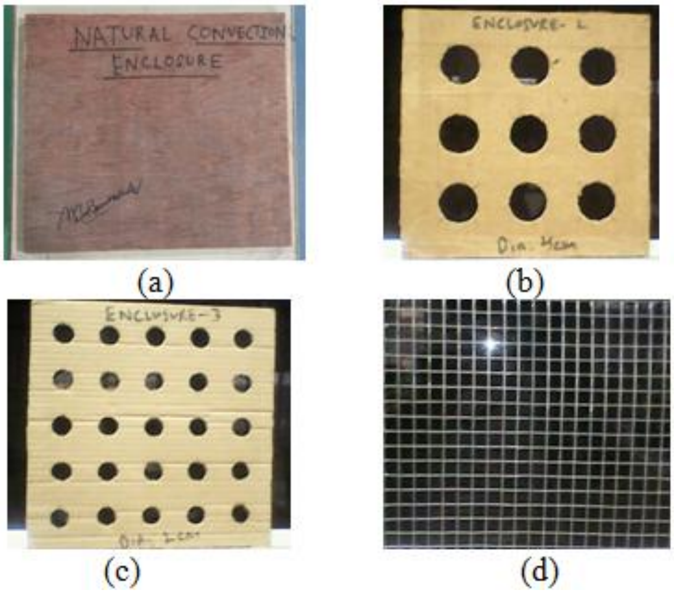


Figure 5. Pictorial view of the external heat sinks (a) wall enclosure (b) perforated enclosure (dia. 4 cm) (c) perforated enclosure (dia. 2 cm) (d) mesh enclosure (wire dia. 0.80 cm).

Figure 5 shows the different types of external enclosures utilized for the present study viz., solid cardboard plate or wall enclosure with specifications (35 cm x 40 cm x 2 cm), perforated plates (small and large diameter and wire mesh enclosure (wire diameter 0.80 cm).
The convective heat transfer coefficient is determined by the power balance as the heat power lost due to convection is equated to the electrical power supplied as:

$$h A \Delta T = V I \qquad (1)$$

Where,

$$\Delta T = (T_{av} - T_1) \qquad (2)$$

$$T_{av} = \left(\frac{T_2 + T_3 + T_4 + T_5 + T_6}{5}\right) \qquad (3)$$

h = Heat transfer coefficient (W/m^2-K)
V = Voltage supplied (Volt)
T_{av} = Average thermocouples temperature (K)
T_1 = Ambient temperature (K)

θ = Surface orientation (Degrees)
I = Current intensity (Ampere)
A = Area of square plate (m^2)

The work primarily focuses on the role of an external enclosure on convective heat transfer from a confined square plate. The primary enclosure is fixed for all the cases and implications of different external enclosures is explored. It is important to note that all the readings presented here represent the repeatability of the results obtained.

III. RESULTS

An experimental investigation was carried out to gain physical insight into the enclosure effect on convective transfer of heat from a confined heated flat plate. The resultant heat transfer is mainly looked upon as implications of flow behavior owing to buoyancy and convection currents governing heat transfer on square plate. The effect of controlling variables viz., surface orientation, enclosure distance, types of enclosure was searched as variation in heat transfer coefficient. Prior to the main study, the predictions of the experimental setup were validated with the bench mark heat transfer theory. A base study on heat transfer rate variation with plate orientation was carried out with primary enclosure(fixed) and without any external enclosure. It is important to note that, without enclosure in the present work refers to experimentation without any external enclosure. Figure 6 shows the variation of heat transfer coefficient with plate orientation extending from horizontal to vertical for the case of smooth plate surface facing upward. Experiments showed that the heat transfer rate exhibits a monotonically increasing behavior with increment in plate orientation. Highest heat transfer rate was noted for the case of plate kept vertical and minimum with the plate oriented horizontal. Results dictate that confinement in any form do affects the in natural convection heating rate.

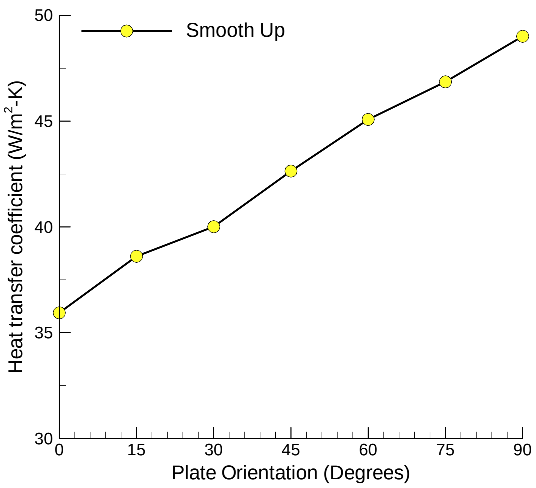


Figure 6. Effect of surface roughness on convective heat transfer for case of no enclosure

Confined buoyant flow subjected to varying conditions readjusts to regain equilibrium. As plate orientation increases, the buoyant flow from lower section of plate mixes with the successive upper section buoyant flow (flow assimilation) and increases the upward flow velocity which complements with enhanced heat transfer. Maximum heat conservation effect comes when the objects are in horizontal orientation whereas, in applications requiring enhanced heat transfer effect, vertical orientation are more suitable. The above mentioned results conform reasonably well to the heat transfer theory which conditions boosted heat transfer for plates kept vertical in a square cavity natural due to thick boundary layer formation. The experimental setup predictions match reasonably well so it is expected to provide better physical insight to the present explorations. As the governing dynamics is fluid and thermal in nature, prior to the experimentation the nature of flow is determined. The Grashof number (for normal gravity case; buoyant flow) were evaluated at the fuel surface to ensure that the flow is laminar.

Grashof number for flow over flat plate is determined as:

$$G_{R_L} = \frac{g\,\beta (T_S - T_\infty) L^3}{\nu^2} \qquad (4)$$

Where,
g = Acceleration due to earth's gravity (9.81 m/s^2)
β = Volumetric thermal expansion coefficient (K^{-1})

(Here, $\beta = 1/T_{av}$)

T_s = Surface temperature (Here, "T_{av}")

T_∞ = Free stream temperature (Here, 30.5°C = 303.5 K)

ν = Kinematic viscosity (Here, 1.6036e-5 m^2/s)

L = Characteristic length (Here, 0.15 m)

Note that the transition to turbulent flow occurs in the range of $10^8 < G_{R_L} < 10^9$ for natural convection from flat plates. At higher Grashof numbers, the boundary layer is turbulent; at lower Grashof numbers, the boundary layer is laminar.

Table 1. Variation of Grashof number with plate orientation for without external enclosure

S.no	Plate orientation (Degrees)	Grashof number
1	0	0.199×10^8
2	15	0.187×10^8
3	30	0.182×10^8
4	45	0.172×10^8
5	60	0.164×10^8
6	75	0.158×10^8
7	90	0.152×10^8

Based on the above mentioned equation and parameters, the Grashof number value at different plate orientation in the domain was estimated. Table1 shows the variation of Grashof number for varying plate orientations. With the results, the flow was found to be well within the laminar range. In most of the heat transfer studies, the subject (heat source) is placed at an orientation (from θ = (0°-90°)) however, in most of practical cases, the placement is at an orientation. This adds to the complexity of the analysis by an additive factor with unpredictable heat transfer rates. Here, the issue is simplified by redefining heat source placement based on surface orientation. A novel heat source placement regime under observation is projected as **R-V Regime** comprising of three distinct zones.

1) **RV-I:** This zone signifies placement of heat source from surface orientation (θ)= (0°-30°).
2) **RV-II:** This zone signifies placement of heat source from surface orientation (θ)= (30°-60°).
3) **RV-III:** This zone signifies placement of heat source from surface orientation (θ)= (60°-90°).

Considering the frame of reference, this bifurcation is applicable to any heat source at any orientation.

First, the effect of an external enclosure on confined natural convective heat transfer is established. On the base configuration, a thick solid enclosure is placed enclosing the top opening completely representing fully enclosed case. Experimentation was performed for same variation of heat transfer rate with varying surface orientation. Figure 7 shows variation of heat transfer coefficient with plate orientation for enclosure effect. The experiments were conducted with top end fully closed and smooth surface of plate facing upward.

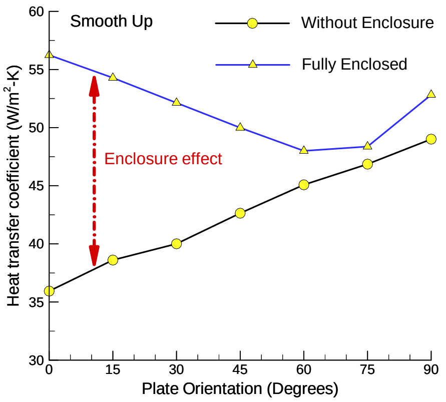


Figure 7: Enclosure effect on convective heat transfer.

Looking at the plot, one can note that the convective heat transfer rates vary from the base configuration (without external enclosure) emphasizing the very vital fact that external enclosures do affect the confined convective heat transfer. The gradient in convective heat transfer coefficient with and without external enclosure is stated as the **"Enclosure effect"**. Results indicate a non-monotonic reduction in heat transfer rates with surface orientation. However, at each plate orientation, enhanced heat transfer is noted than the one without enclosure highlighting external enclosure as potential heat sink. Maximum heat transfer is noted at horizontal plate orientation and minimum at 75^o. Steady gradual drop of heat transfer rates is noted in RV Zone I with similar trend followed in RV Zone II. In RV Zone III, the heat transfer rates approach least value at 75^o and up rises with further increase in plate orientation. RV Zonal variation reflects opposite trend in with and without enclosure. With fully enclosed top, RV Zone I signify significant rise in heat transfer rates with maximum at horizontal (**~55% rise**). RV Zone II indicates intermittent rise in heat transfer rates with maximum at 30^o (**30% rise**) and minimum at 60^o (**~7% rise**). RV Zone III shows the divergence trend in heat transfer rates with minimum at 75^o (**3% rise**) and with further increase in plate orientation rises to maximum at 90^o (~**8% rise**).

Figure 8: Pictorial view of the experimentation (a) Without enclosure (b) with enclosure.

With enhanced heat transfer, the external enclosures are meant to work as a potential heat sink. Within the confinement, the hot buoyant flow (primary fluid) rises up and interacts with the enclosure at top which takes a part of heat carried by the fluid. Owing to this heat loss, the fluid becomes denser and regresses downwards as the cold secondary fluid. The transition of hot to colder air within a confined space pertains to the mixing of two fluids. The mixing resorts to the heat transfer that occurs between primary and secondary fluid and the dominant fluid concentration redefines the convective heat transfer from the heat flat plate. Figure 8 shows the pictorial views of the experimentation with and without external enclosure.

As the results indicates, the secondary fluid strength varies with the plate orientation however, overcomes primary fluid in all zones. The RV-Zone I signifies minimum convective heat transfer without external enclosure and thus affirms more secondary fluid quantity resulting maximum connective heat transfer with presence of external enclosure. RV-Zone III without enclosure results maximum convective heat transfer without enclosure. Whereas, with enclosure, it results in minimum heat transfer rate rise as the buoyant flow is dominant minimizing cold air strength to carry more heat. RV-Zone II indicates intermittence in mixing of primary and secondary fluids with reduction in secondary fluid strength as surface orientation increases. The optimized heat transfer cases for fixed equidistant enclosure (here, without enclosure case) can be to keep surface vertical for maximum heat transfer and horizontal for maximum heat conservation. However, with fully enclosed condition, the cases can be optimized to horizontal surface orientation for maximum heat transfer and close to vertical for minimum heat transfer.

As the convective heat transfer rates are noted to be drastically affected by the presence of an external enclosure. The study was extended to note the enclosure effect for partially enclosed configurations. Of the entire confinement opening, the external enclosure was systematically placed by the proportion of area enclosed viz., 1/5 enclosed to 5/5(fully enclosed). Figure 9 shows the experimental configurations for systematically placed external enclosure at varying location from fully open (without enclosure) to fully enclosed.

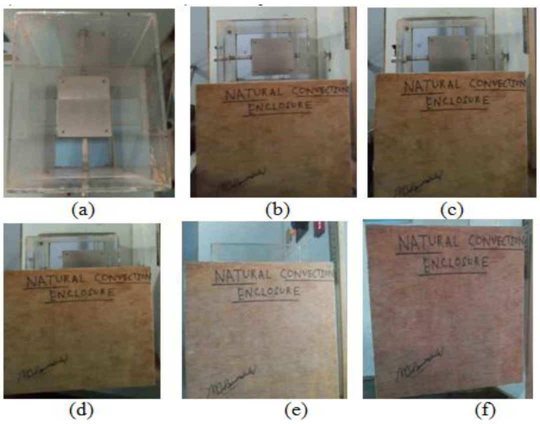


Figure9: Pictorial view of the wall enclosure placement at varying distances(a) fully open (b) 1/5 enclosed (c) 2/5 enclosed (d) 3/5 enclosed (e) 4/5 enclosed (f) fully enclosed.

The external enclosure obstruction to the primary hot fluid plays a vital role in defining the confined natural convective heat transfer. With partial external enclosure placement, the primary and secondary fluid interaction are expected to be altered with substantial implications on the heat transfer. Figure 10 details the enclosure effect for the above mentioned partially enclosed configurations with varying surface orientation. Looking at the plot, one can note the divergent variations in heat transfer rate for all the configurations. This indicates that the enclosure effect is a strong function of the enclosure placement. The results are compared with the cases of fully open and fully enclosed external enclosures. As the confinement is enclosed partially from (1/5 enclosed) to nearby fully enclosed (4/5 enclosed), the heat transfer rates in different RV Zones reflect non-monotonic trends with increasing surface orientation.

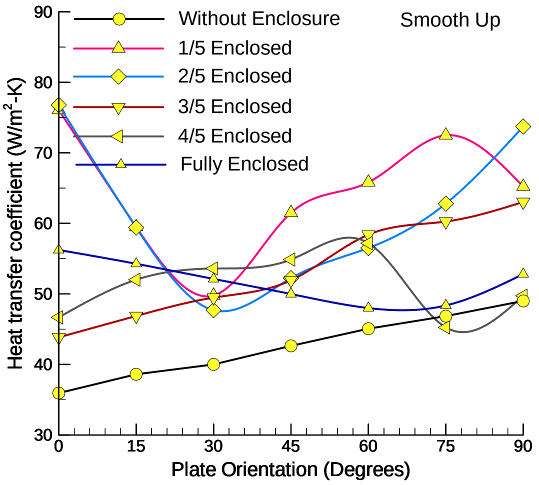


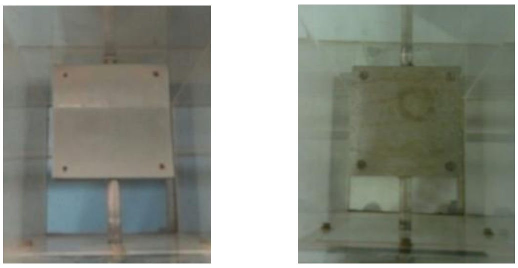
Figure10: Effect of partial enclosure on free convective heat transfer.

In RV-Zone I partially enclosed configurations (1/5 and 2/5 enclosed) are noted to result maximum heat transfer rates (higher than fully enclosed) which further drops with increase in orientation till the end of zone. For the cases of (3/5 & 4/5 enclosed) configurations, the heat transfer rates lower than fully enclosed were noted however, rises with increasing plate orientation. The maximum convective heat transfer occurs for the horizontal plate orientation with (1/5 enclosed) boosting heat transfer rate by (**111%**) and (2/5 enclosed) resulting in maximum (**113% rise**). With further increase in enclosed area the rise in heat transfer rate drops to (**22% rise**) for (3/5 enclosed) and (**29% rise**) for (4/5 enclosed). The values were noted to converge at the end of the zone to critical values lower than fully enclosed except for (4/5 enclosed). In RV-Zone II a uniqueness in the heat transfer rate trends is observed for all partially enclosed configurations.

All configurations exhibit gradual enhancement in heat transfer rates with surface orientation. For surface orientations above 45^o all partially enclosed configurations yield higher heat transfer rate than fully enclosed. The rate of heat transfer rise in this zone is maximum for the case of (1/5 enclosed) with (~**45% rise**). For remaining configurations, the rise reestablishes to (**20-30%**) in comparison to (**17% rise**) for fully enclosed. RV-Zone III shows the most unusual consequences of all partially enclosed configurations. At vertical plate orientation, all partially enclosed enclosures yield enhanced heat transfer rates than fully enclosed. Configurations (2/5 and 3/5 enclosed) follow similar increasing trend with orientation. (2/5 enclosed) is noted to yield maximum heat transfer rise (**50% rise**) whereas, (3/5 enclosed) gravitates to (**28% rise**). Assorted outcomes result in (1/5 & 4/5 enclosed) enclosures. Configuration (1/5 enclosed) peaks at 75^o (**~55% rise**) and drops to (**33% rise**) at vertical plate orientation. It is interesting to note that, configuration (4/5 enclosed) shields heat conservation effect at 75^o with heat transfer rate drops lower than without enclosure. This enclosure placement represents ***"Devinn point"*** (representing two diverse phenomenon of heat transfer and heat conservation). This also signifies that enclosures can also be utilized for the purpose of heat conservation along with heat transfer. With further increase in plate orientation, the heat transfer effect of configuration (4/5 enclosed) increases to little more than fully enclosed (**~2% rise**).

The uneven heat transfer behavior of partially enclosed configurations may be attributed to the conjunction of two distinct effects viz., heat sink effect of enclosure and formation of recirculation zone due to pressure gradient at the wall from obstructed buoyant flow. For partially enclosed (1/5 & 2/5 enclosed; covering lesser than half of confinement area) configurations, the recirculation effect dominates whereas for (3/5 & 4/5 enclosed) configurations, the heat sink effect of enclosure dominates. These effects result in enhancing the secondary fluid concentration which carries additional heat from heated flat plate. For the case of (1/5 & 2/5 enclosed) configurations, the reason for excess heat transfer may be attributed to the recirculation zone formed which instigates successive vortex formation till the plate surface. As the mixing strength increases, the strength of vortex and vorticity increases with distance till the plate surface. The faster moving fluid carries additional heat from the plate surface. The heat sink effect contributes little in generation of secondary fluid as only a part of main flow encounters the enclosure. The strength of fluid recirculating in an unbounded fluid varies with plate surface orientation for varying partial enclosure configurations owing to angled buoyant force. With increase in surface orientation, the strength of recirculation zone drops as glass enclosure stagnates a part of hot flow which reduces the secondary fluid concentration and thus results in weaker moving vortex formation and low vorticity till the end of RV-Zone I. With further increase in surface orientation, the angled buoyant force redirects flow more towards the enclosure end. The inclined plate surface redirects larger part of flow towards the partially enclosed area where secondary flow generation increases by heat sink action resulting stronger recirculation zones and successive vortex formations thus carrying more heat. For partially enclosed configurations (3/5 & 4/5 enclosed) the fluid motion relies more on heat sink effect of enclosures. The recirculation effect is present but, weak recirculation zones and their propagation is abstained by the fluid readjustment to the larger enclosed area. The heat transfer is low in RV-Zone I however, with increased plate orientation and flow readjustment, the recirculation effect starts assisting the heat sink effect in increasing secondary fluid concentration for carrying more heat. The effect increases in subsequent RV zones to the maximum value. A peculiarity is observed in the partially enclosed (4/5 enclosed) configuration yielding heat conservation effect. It indicates the reduced secondary fluid strength and increased hot fluid recirculation resulting plate reheating. The optimization of cases can be to keep horizontal surface orientation for maximum heat transfer and slightly inclined for minimum if top enclosure covers less than half of area. For utilizing solid enclosures at top covering more than half of area, it would be advisable to keep surface orientation horizontal for minimum heat transfer and close to vertical for maximum heat transfer.

As most of the surfaces are rough and consequentially generated fluid motion is expected to affect the convective heat transfer. Next, we analyze the effect of surface roughness on confined natural convection. For all cases till now, the maximum heat transfer with enclosure was noted for horizontal plate orientation, so the study was conducted on the same.



(a) (b)

Figure 11: Pictorial view of the experimentation (a) Smooth surface up (b) rough surface up.

Figure 11 shows the experimentation for the cases of horizontal flat plate orientation with smooth and rough surfaces facing upward under similar conditions. The fluid motion over a heated flat plate is governed by the presence of thermal boundary layer and the velocity boundary layer at the surface. Surface roughness increases flow disturbance by generating more velocity gradients thus carrying additional heat. The role of buoyant convection is expected to be same however, the amendments owing to surface roughness is investigated. Figure 12 shows the variation in heat transfer rate with enclosure distance for smooth and rough surfaces. For both cases, the heat transfer rates follow a non-monotonic trend with enclosure distance. Experiments shows that the heat transfer rate increases till peak value as the enclosure distance increases and then drops with further increase in enclosure distance. The results noted state that the enclosure placement in less than half of area viz., (1/5 & 2/5 enclosed) yields highest heat transfer rate for smooth surface. Similar trend is noted with rough surface as well. Rough surfaces are mostly seen imparting more heat transfer for varying enclosure placement except at nearby enclosure placement (1/5 & 2/5 enclosed).

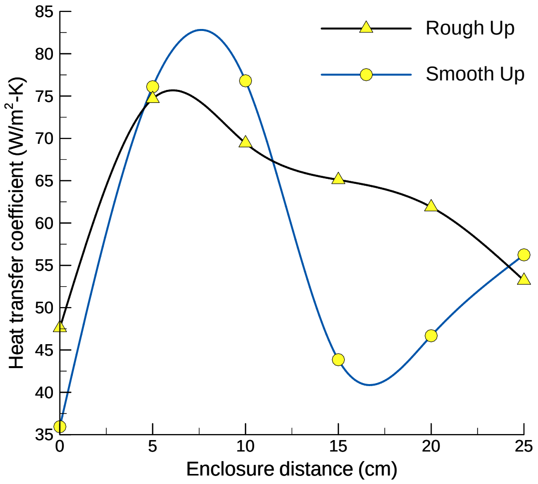


Figure 12: Effect of plate surface roughness on convective heat transfer in presence of varying enclosure.

Unlike smooth surfaces, the rough surface exhibit gradual drop till minimum for fully enclosed case. The surface roughness effect is more noted for enclosure placement covering more than half viz., (3/5 & 4/5 enclosed). Without enclosure the rough surface profits **(32.47%)** heat than smooth surface. For enclosure placement at (3/5 enclosed) the rough surface yields a maximum of **(48.46%)** higher heat transfer rate followed by **(32.56%)** higher at (4/5 enclosed) than the smooth surface. It is interesting to note that, at the intermediate enclosure placement (enclosing half of area), the heat transfer rates match for smooth and rough surfaces. The smooth surfaces dominate at below intermediate enclosure placement viz., (1/5 & 2/5 enclosed) with **(~10%)** higher for (2/5 enclosed) and **(~2%)** higher for (1/5 enclosed) than the rough surface.

For smooth surfaces, the flow readjusts accordingly with the conjunction of the heat sink effect and the recirculation effect which dominates in respective placements assisted by the subsequent other. Similar trend is observed for rough surfaces for varying enclosure placement. The surface roughness augments the momentum transfer to the flow which remains connected to surface for longer time and carries more heat. In case of enclosure, this effect significantly affects the primary and secondary fluid motion which redefines the enhanced heat transfer rates. The surface roughness effect can be stated with enhanced heat transfer with primary fluid flow. Additionally, the heat sink effect enhances concentration of secondary fluid and the recirculation effect enhances the momentum of the secondary fluid. The surface roughness effect peaks for the cases of lesser than intermediate enclosure placement viz., (1/5 and 2/5 enclosed) and drops with the heat sink effect dominant for greater than intermediate enclosure placement. The fact is justified by the linear gradual drop in the heat transfer rate with increasing enclosure placement. The heat transfer can be optimized by little roughness in surfaces with enclosure covering less than half of area at top.

The need of the obligatory heat transfer in most of the operational systems varies with varying conditions. The enclosures are validated to play an important role with the generation of secondary fluid redefining air cooling. To fulfill the objective, it is therefore necessary to test, design and validate different types or selections in enclosures. Consequentially, the enclosure effect on confined natural convection was further extended to testing widely utilized enclosures viz., perforations, meshes.

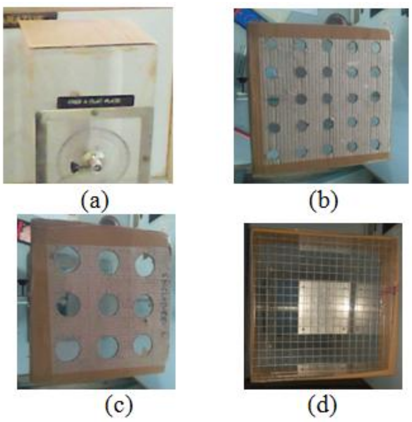


Figure13: Pictorial view of diverse enclosure studies(a) full solid cover (b) perforated (dia. 2cm) (c) perforated (dia. 4cm) and (d) mesh (wire dia. 0.80 cm).

The flow behavior and resulting convective heat transfer are likely to be affected by the presence of different types of enclosures. Figure 13 shows the pictorial view of diverse enclosures. The study comprises of extensive comparison of a solid enclosure at top with perforated enclosure of varying diameter and a wire mesh enclosure for the smooth surface facing upward. The effect of each enclosure is systematically experimented with the varying surface orientation and related implications on resultant heat transfer coefficient. Figure 14 divulges the variation of heat transfer coefficient with the surface orientation for the above-mentioned cases. Sequentially, confined natural convection (here, without enclosure) yields higher heat transfer rates than the open-air cases. With respect to the without enclosure cases, the solid enclosure enhances the heat transfer rates with different rates. Looking at the plot one can note that, the perforated enclosures follow a similar trend. Perforations in solid enclosures exhibit competitive and higher heat transfer rates than the solid cover at all orientations. The mesh enclosure shows insensitiveness to the heat transfer rates till plate orientation of 50^o (heat transfer rates equal to without enclosure) and drops with further increase in plate orientation with minimum value at vertical plate orientation.

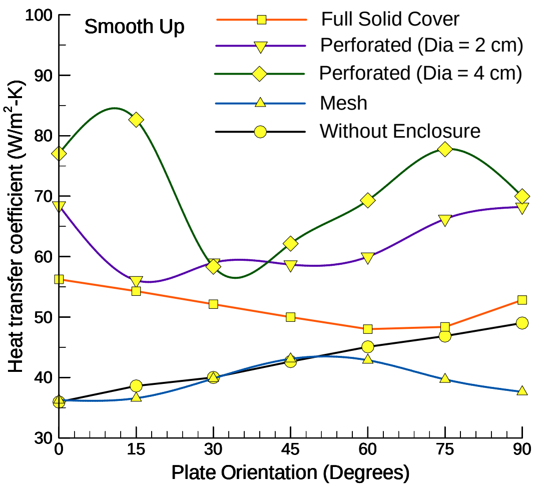


Figure14. Effect of multiple enclosure types on confined natural convective heat transfer.

It is important to note that all enclosures are noted to observe a non-monotonic trend in heat transfer rate variation with plate orientation. Statistically, small diameter perforations (here, 2 cm) yields maximum heat transfer rate rise (**90%**) at horizontal orientation and minimum of (**33%**) at plate orientation of 60^o. The large diameter perforated enclosure (here, 4cm) delivers maximum heat transfer rate rise of (**115%**) at horizontal orientation and minimum of (**45%**) at plate orientation of 30^o. Beyond insensitiveness of mesh enclosures (till 50^o), the drop in heat transfer rates is maximum (**23%**) at vertical plate orientation. Qualitatively, RV-Zone I deals with highest heat transfer rates for all external encloser configuration. Large diameter perforation shows slight increase in heat transfer rate value with orientation whereas, the small diameter perforation predicts a drop. However, the values for both perforated enclosures converge at the end of zone. The mesh enclosure details insensitiveness with accompanied by a small drop with further convers with without enclosure as orientation increases. RV-Zone II quantifies further increase in heat transfer rates for large diameter perforates enclosure and reduced rates for small diameter perforated enclosure. The rate of heat transfer rate increase is higher for large diameter perforated enclosures and drops for smaller. The mesh enclosure endures insensitiveness in heat transfer rates however, the rate of change drops from the without enclosure at the end of zone. RV-Zone III notices the enhanced heat transfer rates for large and small diameter perforates enclosures. The increasing rate peaks in the middle of zone (**at 75^o**) and drops with further orientation increase. The mesh enclosure specifies the origination of the concept to use enclosures for heat conservation. The heat transfer rate in this zone drops with increasing orientation with the rate of drops increasing significantly with increasing plate orientation.

The reason for varied changes with diverse enclosure configurations can be attributed to two factors viz., blockage area and thermal conductivity. For confined natural convection, the flow readjustment with the plate orientation is well defined owing to momentous buoyant flow assimilation as plate orientation gravitates towards vertical. It is important to note that for the present study the different enclosures were made of different materials which represents varying thermal conductivity however, the compensation was made with the blockage area. The thermal conductivity of working fluid viz., air is 0.02 W/m-K for present operating conditions. The flat plate is made up of aluminum with thermal conductivity of 0.50 W/m-K. Within confinement and different enclosures with varying material thermal conductivity, the flow behavior

readjustment relates to a mixed convective-conductive heat transfer. The selection of material is based on the very idea that primary enclosure material (glass), solid enclosure (hardboard) and perforated enclosure (cardboard) have thermal conductivities in the same range (0.15-0.21 W/m-K) to facilitate same secondary fluid generation. The mesh enclosure (steel) has higher thermal conductivity of 50 W/m-K. The conductive part of heat transfer relates primarily to the primary fluid (hot air; fixed thermal conductivity) interaction with the diverse enclosure materials (varying thermal conductivity) and resulting in generation of the secondary fluid.
It is widely known that material with higher thermal conductivity are potential heat sinks and the lower one are thermal insulators. In the present study, for a particular enclosure thermal conductivity remains same and the flow behavior is compensated with the blockage area (higher thermal conductivity with low blockage area). The perforations represent uniformity in the diameter size and thus varying number of holes. The small diameter perforated enclosure offers lesser blockage area to the upcoming primary fluid however, the flow is slow and results in formation of small localized recirculation zones which strengthens with time for the secondary fluid generation till approaching plate surface. This flow adjustment varies in all the RV zones as the flat plate orientation changes. The cumulative effect is noted maximum for horizontal plate orientation and minimum with plate oriented at 15^o. The effect is magnified with the large perforations offering large blockage area and thus stronger localized recirculation zone formation which propagates strongly till plate surface. It is worth noting that, the perforation diameter is a critical parameter in perforated enclosure utilization. With increasing plate orientation, the effect of localized recirculation zones and flow assimilation in the respective regions generating localized secondary fluid reduces in RV-I owing to flow readjustment to the changing plate orientation. The effect picks up in RV-II and RV-III zones and resulting in increasing heat transfer rates. The mesh enclosure has relatively higher thermal conductivity however, the blockage area is low and results in insensitiveness till plate orientation of 50^o. Post that, as the plate is oriented towards vertical where one plate end is closer to the mesh, plate heating effect is noted. In view of the geometry, the generation of secondary fluid is minimal of all cases however, the role of hot fluid in the nearby regions owing to the heated mesh (higher thermal conductivity) have reheating effect on plate which drops the heat transfer rates lower than without enclosure. It is important to note that, this peculiar consequence is present only in the RV-III zone when one plate end is near to the mesh enclosure. The result also validates the enclosure effect for minimum flow blockage yet higher thermal conductivity as the results matches with the one without enclosure. The heat transfer can be optimized by utilization of perforated enclosure at top with horizontal surface orientation. Similarly, mesh enclosure at top would be effective in heat conservation.

As different enclosures significantly result in different heat transfer characteristics. Next, the work attempts to verify the possibility of using combination of enclosures for optimal heat transfer conditions. The present study explores combination of different enclosures as a configuration viz., solid and perforated enclosures, two different perforated enclosures, mesh and perforated enclosures on heat transfer rates with varying plate orientation. The results were compared to the base case of no enclosure.

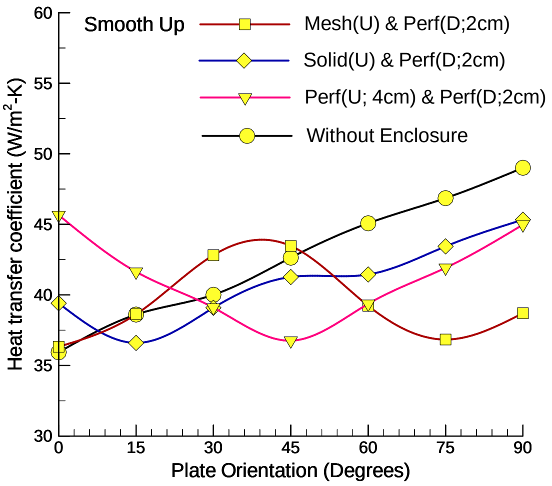


Figure15: Role of embedded enclosures on convective heat transfer.

For different enclosure configurations, one enclosure is placed at the top and selected another at bottom end. For effective comparison, the slam diameter perforated enclosure is fixed at the bottom end. Figure 15 shows the variation in heat transfer rate with surface orientation for diverse collaborative enclosure combinations. Looking at the plot one can note that, the different enclosure configurations follow similar non-monotonic trend in heat transfer rate variation with plate orientation. An important aspect of heat transfer is

addressed with enclosures in collaboration becoming effective thermal insulators for in RV-II and RV-III. The combined enclosure configuration of mesh enclosure at the top and perforated enclosure (dia. 2cm) at bottom predicts comparable outcome as with the mesh alone at the top end. This validation verifies the thermal insulation effect of mesh enclosure in RV-III along with certifying the redundancy in the role of enclosure at the bottom. Combination of a mesh and small perforated enclosure results in maximum heat transfer rate rise (**7%**) at 30^o plate orientation and maximum thermal insulation effect (**21% drop**) for vertical plate orientation. The enclosure configuration with solid at top end and small diameter perforated enclosure at bottom end predicts significant role as thermal insulator at all plate orientations except at horizontal. The combination results maximum heat transfer rate rise (**~10%**) for horizontal and highest insulation effect (**8% drop**) at 60^o plate orientation. Perforated enclosures amplify the heat transfer rates in collaboration also however, with increasing plate orientation the thermal insulation effect overtakes. The combination of large diameter perforated enclosure at top and small diameter perforated enclosure at bottom results in maximum heat transfer rate (**27% rise**) for horizontal plate orientation and maximum thermal insulation effect (~**14% drop**) at 45^o plate orientation. By zonal divisions, in RV-Zone I the heat transfer rates are higher at horizontal plate orientation for all collaborated configurations. With increase in plate orientation, the heat transfer rate drops for large diameter perforation and solid whereas for mesh it drops below no enclosure. At the end of zone, the solid and large diameter perforation enclosure matches and approaches little no enclosure. Interestingly, mesh enclosure depicts maximum rise at the end of zone. In RV-Zone II, with increase in plate orientation, the large diameter perforated enclosure gives maximum drop. The solid enclosure configuration shows rise and mesh enclosure drops to match no enclosure. With increase in plate orientation, the large diameter perforated enclosure gives maximum drop. The solid enclosure configuration shows rise and mesh enclosure drops to match no enclosure. At the end of zone, all configurations acts as potential thermal insulations with perforated and mesh enclosures predicting a match.
RV-Zone III shows similar results with all configurations resorting to heating rate well below no enclosure. With increase in orientation, perforated and solid shows rise and mesh drops to the least values. It is important to note that, with the presence of an external enclosure at bottom, the heat transfer rate variation for diverse configurations outcomes in a close range of (here, 35-50 W/m^2-K). This fact signifies the importance of bottom enclosure in optimized thermal insulation effect.

To fundamentally understand the effect, we begin with the sequence of cases viz., an increasing heat transfer trend with plate orientation for both ends open followed by the enhanced heat transfer rate without bottom enclosure for most of configurations. However, incorporation of a selected bottom enclosure results in drastic drop in heat transfer rates within a confined extent. The experimentation result validates the preceding results (mesh enclosure study). The role of secondary fluid, intensity of mixing, heat sink and recirculation effect remains intact however, the reduced heat transfer from the plate can be addressed to the flow redirected to the plate reheating owing to the trapped surrounding fluid within a confined area. The significant lower side of plate encloses the hot surrounding fluid resulting in two singularities viz., reheating the top end and thermally neutralizing the secondary fluid to carry heat from heated plate. The result can be justified by the limited changes in heat transfer rate for different configurations. The intensity of this effect do varies with the plate orientation however, the changes drop to a limited extent. The strength of reheating is low for horizontal orientation but as the plate inclination increases, the effect picks up owing to enhanced mixing within a confined volume. The optimized heat transfer cases would be to utilize perforated enclosure at top and bottom with horizontal surface orientation for maximum heat transfer and to resort to different enclosure configuration for inclined surfaces for maximum thermal insulation.

IV. CONCLUSIONS

Experimental simulations were carried out to investigate effectiveness of external enclosures on heat transfer characteristics of confined natural convection under diverse conditions. An effective zonal system is defined with respect to surface orientation for effective heat transfer analysis. Under varying conditions, the physics of enclosure phenomenon directs to the flow behavior and adjustment to the presented change. Based on results obtained following conclusions may be drawn:

a) Confined natural convection heat transfer is more effective in vertical orientation due to stronger buoyant forces leading to better cooling applications.
b) Presence of external enclosure significantly affect the transfer of heat.
c) Fully enclosed top results in enhanced heat transfer at all orientations owing to stronger heat sink effect in generating the denser secondary fluid. The strength of secondary fluid varies with plate orientation. Partial enclosure placement returns with enhanced heat transfer rates. For less than half placement, additional heat is transferred owing to dominant recirculation effect and for greater than half placement, the heat sink effect is dominant with assisted recirculation effect.
d) With external enclosure, rough surfaces yield higher heat transportation than smooth surfaces.

e) Perforated enclosures are highly effective than solid or mesh enclosures for transfer of heat. Large diameter perforated enclosures are better than small diameter. Wire mesh enclosures are not suitable for heat transfer however; they are efficient for heat conservation characteristics. Solid enclosures result in-between perforated and mesh. The flow readjustment redefines the heat transfer.
f) Different external enclosure configurations (top and bottom placement) are primarily potential thermal insulators.
g) The predictions of the experimental setup were validated with the benchmark heat transfer theory and matches reasonably well.

The enormous rise in the system controlled environment have necessitated the need of heat transfer in wide range of practical, functional, scientific, Industrial and engineering applications. The nature of the heat transfer is two folds in transfer and in conservation for efficient utilization. The difficulties have always been to produce simplistic and efficient solution. The work proposes use of enclosures as an effective and productive method for emergent heat transfer and conservation.

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