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Enclosure Phenomena in Forced Convection

Pranjal Nag[1], Shivam Gupta[2], Pratik Tiwari[3] , Vinayak Malhotra[4*]

[1,2,3] B.Tech Student, Dept. of Aerospace Engineering, SRM University, Chennai, Tamil Nadu, India

[4] Professor (Assistant), Dept. of Aerospace Engineering, SRM University, Chennai, Tamil Nadu, India

ABSTRACT

Experiments on low forced convective heat transportation are carried out over a square plate within a confinement. Analysis is carried out for the interaction of heat transportation with enclosure in the aid of varying surface orientations. Enclosure effect is noted on the heat transportation from the pilot source for three different cases of fully closed exit, partially closed and fully open exit. The conditions for optimal transportation of heat and related implications are detailed under varying conditions of orientations and enclosure placement. Results shows that enclosures can be of magnificent assistance for varying conditions of heat energy conservation. Partially enclosed confinements are betters for purposes of additional heat conservation or heat transportation. At certain locations they shift role from heat conservation to enhanced heat transfer. The key controlling parameter enclosure distance optimizes the enclosure effect with surface orientation.

Keywords: Forced convection, heat transportation, enclosures, square flat plate, VRM_PPT point, devinn point.

1.INTRODUCTION

Heat transfer by convection is a physical phenomenon with wide range of engineering applications of practical and functional significance. Convective heat transfer is generally categorized as: Free and forced. Free convection refers to fluid motion by buoyant forces arising due to density gradients which are a result of temperature gradients. Whereas in, forced convection, the flow of the fluid is enhanced by external sources. The mechanism is found very commonly in everyday life and includes central heating, air conditioning, steam turbines, heat exchangers, pipe flow etc. Almost all of the applications are driven with the need of efficient energy transportation or energy conservation. The need in this field is related to the efficient heat transfer and heat conservation. Appreciable work in the form of experiments, computations, analytically, theoretically had been done in the past. However, the complexity of the problem have eluded of comprehensive understanding with wide range of applications. The characteristics viz., temperature gradients, heat transfer coefficients are investigated over diverse configuration to fundamentally understand the occurrence. The interest in this class of problems is specifically driven by the need to have better understanding of convective heat transfer for growing applicability.

Following the classical work of Tribus et al., [1] for forced convection over non-isothermal surfaces, last six decades research works have contributed significantly to the advancement in the understanding of forced convection. The contributions have been reported in several reviews like Cess [2], Szewczyk [3], Whitaker [4], Shah [5], Cooper et al., [6], Copeland [7], Kim et al., [8].The works provide an excellent review on the developments up to the end of the century. Cheng et al., [9] investigated unsteady forced convection on a flat plate with inertia effect and thermal dispersion. They noted that the rate of unsteady heat transfer can be accelerated by the thermal dispersion. Sartori [10] studied equations of the forced convection heat transfer coefficient over flat surfaces. He reasoned that there must be a decay of heat transfer coefficient along the plate dimension in the wind direction.

Lakhal et al., [11] numerically studied natural convection in an inclined rectangular enclosure with perfectly conducting fins attached to the heated wall. They work stated that the heat losses through the cold wall can be reduced considerably by using fins attached on the heated wall. This phenomenon becomes more pronounced when the enclosure inclination angle from the vertical is increased. Islam et al., [12] investigated natural convection in a tilted square enclosure containing internal energy sources. They noted that the diffusion heat transfer is prominent for the lower value of internal heat generation whereas the convection outweighs the diffusion for the higher value of internal energy. Furthermore, the work stated that the convective currents always prevail at the bottom part of the cavity whatever its magnitude is. Abu-Nada et al., [13] explored the influence of inclination angle for a square enclosure. Inclination angle of the enclosure was proposed a control parameter for fluid flow and heat transfer. In recently, Tiwari et al., [14] adapted a porous wall to study the effect of enclosures on forced convection. They concluded mainly that the enclosure effects are primarily in reducing forced convection strength.

Although much has been done but complexity of the problem has prevented a complete understanding due to interaction between flow, heat and mass transfer. Therefore, a systematic study is needed to understand mechanisms controlling the forced convective heat transfer. The present work is a consequence to the above mentioned preceding work [14] with deeper explorations. The work focuses on effect of an enclosure on forced convective heat transfer coefficient for varying surface orientations. The enclosure effect is investigated for the cases of partially enclosed to fully enclosed passage in comparison to the one without enclosure.

The specific objectives of the work are:

1)To fundamentally understand the enclosure effect in forced convection and related implications.

2)To analyze the role of key controlling parameters.

The work is motivated by the scientific efforts to enhance the domain of heat transfer applications.

2.EXPERIMENTAL SETUP AND SOLUTION METHODOLOGY

A simple natural convection apparatus (Fig. 1(a)) was adapted for this study. The apparatus consisted of base made of mild steel plates which supported the assembly. The smooth plate assembly comprised of a glass enclosure bounded along the sides but open from both the ends to remove the external influences which can affect heat transfer rate. The aluminum plate specimen (Fig. 1(b)) is (15 cm x 15 cm) which was heated using electrical power at desired rate for 2 hours prior to experiments. The rate of heating the plate can be adjusted with the help of a handhold and a digital display. Thermocouples (5 in numbers) are embedded in plate (Fig. 1(c)) and located equidistance to embark average plate temperature. In order to facilitate the heat transfer at different orientations, the entire plate assembly can be adjusted with the help of a handle and an attached protractor.

Figure 1 Pictorial view of the apparatus (a) Front view (b) Top view of square plate (c) schematic of square plate with location of embedded equidistant thermocouples (shown by circles).

The assembly is bound by a confined passage along the sides. The input power supply to the pilot source (square flat plate) can be controlled with a knob (voltage and current) noted on a digital display. The confinement is attached to a centrifugal blower at lower end to supply air flow at preferred velocity and open from the other end to confiscate the hot working fluid carrying the heat. The opening from the top end is (24 cm x 24 cm).

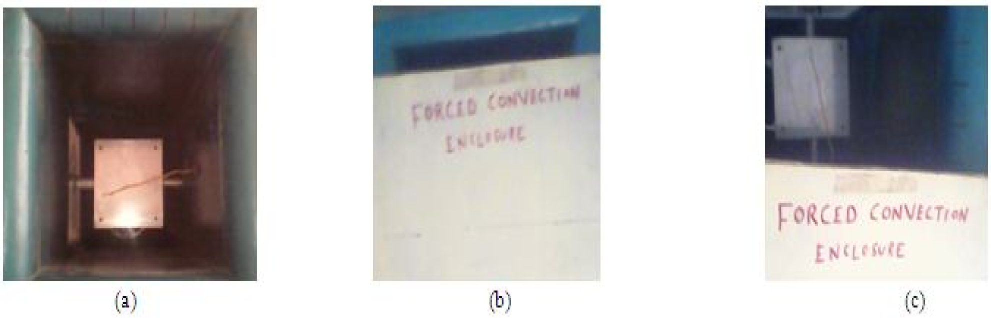

Figure 2 Pictorial view of the top (a) marked locations of enclosure placements (b) enclosure placed at 5/6th (c) enclosure placed at 2/6th.

A thick flat plate was utilized as an enclosure. The opening was subdivided into six parts (4 cm each) and every section was covered one after other to investigate the enclosure phenomenon (please see figure 2). The readings were taken systematically by stepwise increment in plate orientation in proper time interval and repeating the same for varying enclosure distance.

The convective heat transfer coefficient is determined by establishing power balance by equating electrical power supplied to heat the plate to the heat power lost by convection.

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$$hA\Delta T = VI$$

Where

$$h = \frac{V \times I}{A \times \Delta T}$$

$$\Delta T = (T_{(av)plate} - T_{(av)ambient})$$

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

$$T_{(av)ambient} = \left(\frac{T_1 + T_7}{2}\right)$$

h = Heat transfer coefficient (W/m^2-K)

V = Voltage supplied (Volt)

I = Current intensity (Ampere)

A = Area of square plate (m^2)

T_{av} = Average thermocouples temperature (K)

T_1 = Ambient temperature (K)

θ = Surface orientation (Degrees)

All readings were taken systematically by stepwise increment maintaining proper time interval. It must be noted all the data presented here represent the repeatability of results obtained.

3. RESULT AND DISCUSSION

An experimental parametric study was carried out to study the heat transfer characteristics on a heated flat plate bounded by enclosures. The plate was subjected to the heat input of 100 volts and 0.44 amperes with flow velocity of 1 m/s throughout the experimentation. It is important to note that, all the reading were taken for the smooth surface of the flat plate facing upward. To begin with, the effectiveness of the experimental predictions was investigated with two benchmark cases viz., variation of heat transfer coefficient with surface orientation and effect of flow velocity on heat transfer coefficient (Figure 3

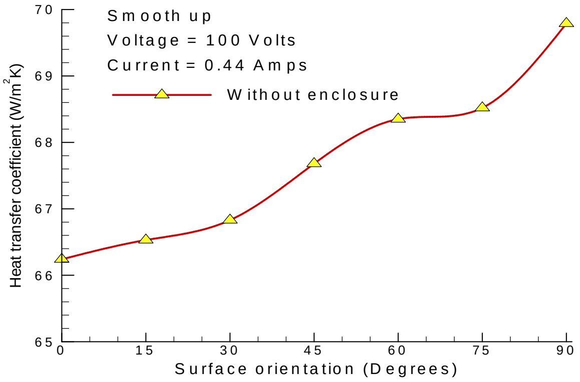

Figure 3 Variation of heat transfer coefficient with surface orientation for smooth surface facing upward.

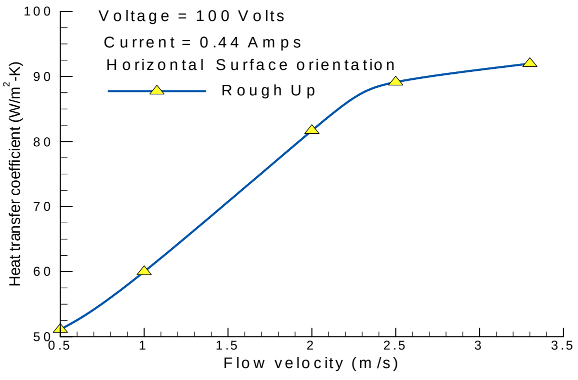

Figure 4 Variation of heat transfer coefficient with flow velocity and surface for rough surface facing upward.

The plate was oriented in the first quadrant and heat transfer coefficient was evaluated with a variance of 15 degrees from horizontal to vertical (Figure 2). The heat transfer coefficient was noted to increase monotonically with orientation. The increase is linear up to 60 degrees and drastically onwards till peak at vertical. Further, we looked at the variation of flow velocity on convective heat transfer (Figure 3). The flow velocity was varied till 3 m/s systematically and corresponding convective heat transfer coefficient was explored. Above mentioned two cases validates the benchmark heat transfer theory as they are widely known and proved. The trend predicted by the experimental setup matches reasonably well. Based on these predictions, the experimental setup is expected to offer accurate physical insight into the enclosure phenomenon and its implications on forced convective heat transfer. The exploration of the enclosure phenomenon on forced convection was initiated with consideration of extreme cases viz., fully open top and fully enclosed top.

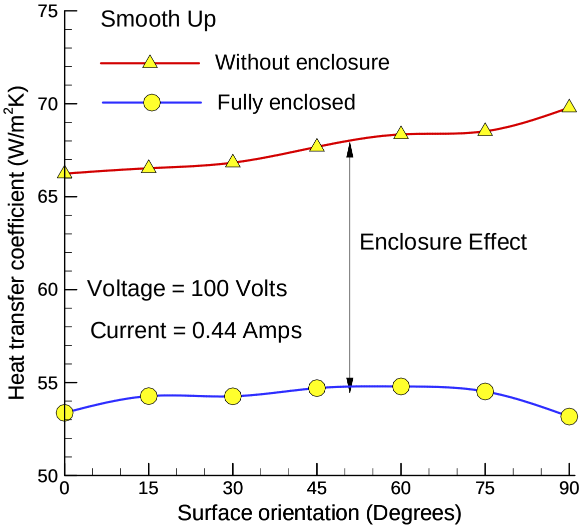

Figure 5 Variation of heat transfer coefficient with surface orientation highlighting enclosure

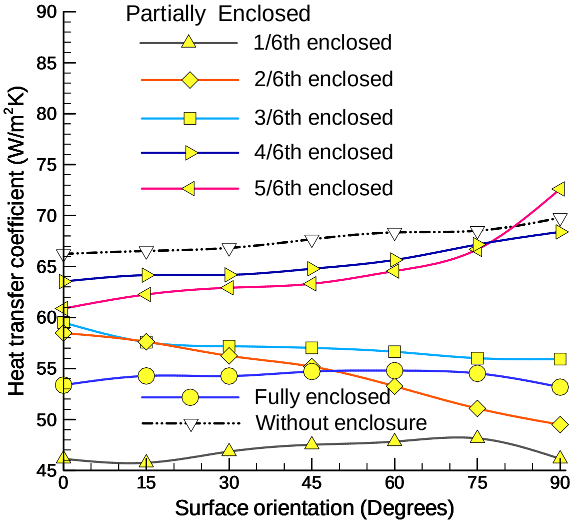

Figure 6 Variation of heat transfer coefficient with surface orientation for complete enclosure effect

Figure 5 shows the variation of heat transfer coefficient with surface orientation for above mentioned extreme cases of fully open and fully enclosed top. Looking at the plot one can note that, the presence of enclosure can cause significant change in the heat to be transferred. The enclosure effect is reflected in the reduction in heat transfer coefficient. While more heat transferred when the top was open however when fully enclosed, the enclosure acts as a heat sink and prevented heat transfer. It is interesting to note that with fully enclosed top the variation of heat transfer coefficient with orientation is marginal and follows a trend opposite to fully open (decreases with orientation). The enclosure restricts the heat transfer and ***VRM_PPT point (10% loss)*** is surpassed in multiple folds (close to 3 times) for all orientations. With the certainty of enclosure effect, next the work was extended to vitally understand the variation of characteristic parameters by systematically enclosing top partially till fully enclosed from fully open. Figure 6 shows varying partially enclosed the top (difference of 4 cm each) till fully enclosed and in comparison with the one without any enclosure. Looking at the plot, one can note that the enclosures in any position results in reduced heat transfer. In a forced convection configuration, a flat square plate transfers more heat without enclosures. As we start enclosing the top step wise from fully open case, drastic loss of heat transportation is noted with the maximum reduction at 1/6th enclosed (4 cm enclosed). The increasing variation of heat transfer coefficient with surface orientation is not followed but opposite is noted till enclosures are placed covering half of top (3/6th enclosed). Beyond the half enclosure till fully enclosed case, all placement of enclosure follows the increasing heat transfer coefficient trend with surface orientation. The maximum change is noted for the case of (5/6th) enclosed. Specifically the role of enclosure can be imitated in three different regimes viz., no change or marginal change, reasonable change and drastic changes. It was noted that the cases of little enclosure (1/6th enclosed) and fully enclosed (6/6th enclosed) project insensitiveness to the variation of heat transfer coefficient with the surface orientation. Whereas, the cases of 3/6th enclosed and 4/6th enclosed shows small changes in heat transfer coefficient with orientation. The cases of 2/6th enclosed and 5/6th enclosed depict significant changes in the heat transfer coefficient with orientation. The analysis of results confines to two cases of heat transfer and heat conservation.

Further interesting crossovers were noted, regarding placement of enclosure at 4/6th and 5/6th of open area, heat transfer coefficient is more for enclosure location 4/6th till 75^o source orientation and above that till vertical source orientation 5/6th results better heat transfer than fully open. Similarly, 2/6th location results more heat transfer than half (3/6th) if source is orientated till 15^o, beyond that it drops drastically to even lower than fully enclosed beyond 45^o source orientation. As the enclosure phenomenon primarily marks heat conservation however there is an interesting location of enclosure which results in occurrence of both the purposes. When the top is enclosed 5/6th of the totally open, both the phenomenon of heat conservation and heat transfer can be noted. While keeping the plate orientated till 75^o from horizontal serves heat conservation above that till vertical serves heat transfer more than fully open case. We call this particular location of enclosure as ***"devinn" point*** (point where two diverse phenomenon occurs simultaneously). With the enclosures verified as effective source of heat conservation with forced convection. The intensity of effectiveness also varies with enclosure distance. In the forced convection domain, for maximum heat transfer reduction the enclosure can be placed at 1/6th of open with source at any orientation. With the requirement of reasonable drop, half of the open area can be enclosed with source at any orientation. However, with varying requirement of heat conservation, the enclosure can be placed at 2/6th of the open area. For equitable drop the potential source can be placed at orientation close to horizontal and for more drop the orientations close to vertical.

Next, we look at the variation of heat transfer coefficient at orientations experiencing maximum change viz., vertical and horizontal in comparison to the one intermediate. Figure 7 shows the variation of heat transfer coefficient with varying enclosure distance for heat source oriented vertically in comparison to the extreme cases of fully open and fully enclosed.

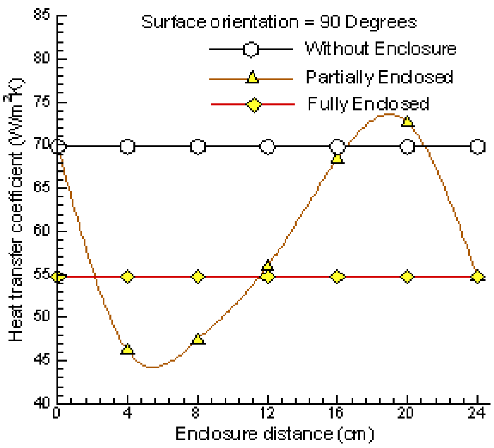

Figure 7 Variation of heat transfer coefficient with enclosure distance for vertical orientation.

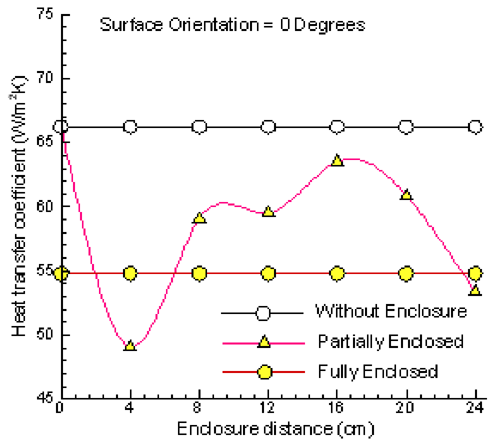

Figure 8 Variation of heat transfer coefficient with enclosure distance for horizontal orientation.

Looking at the plot one can note that, the enclosure causes a drastic change in heat transfer coefficient with variation in the form of a sinusoidal wave. With enclosure, the heat transfer from the plate in this orientation varies more than 6 times the *VRM_PPT point* (~ 64%) with maximum occurring at 5/6th enclosed and minimum occurring at 1/6th enclosed. Without enclosure, the increase in heat transfer coefficient is about 3% at 5/6th enclosed. This is the only location of enclosure which marks *"devinn point"*. However, the enclosure effect can be seen in terms of drop in heat transfer at 1/6th of the location which is about 6 times (~ 59%) the *VRM_PPT point*. In comparison to the fully enclosed case, the maximum heat transfer increase in heat transfer is about 3 times the *VRM_PPT point* (fully open can cause 30% drop in heat transfer coefficient as compared to increasing heat transfer coefficient at 5/6th enclosed). However, in the light of reduction, 1/6th enclosed can conserve close to 2.5 times the *VRM_PPT point* (reduced heat

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transfer coefficient by 25%). It is important to note that, this shifting follows a trend with value of heat transfer coefficient equal to fully enclosed at half enclosure and equal to fully open at $4/6^{th}$ enclosed. So for applications under forced convection domain in low speed flows requiring more heat transfer can be placed at $5/6^{th}$ of the total confinement. Whereas for the applications needing reduced heat transfer or needing to conserve heat, the enclosure can be placed at $1/6^{th}$ of the open end when the source is oriented vertically.

Another important orientation of the source which experiences drastic change is horizontal. Observations from the variation of heat transfer coefficient (please see figure 8) shows a movement opposite to the source oriented vertically with increasing enclosure distance. At this orientation of source, no enclosure placement results in enhanced heat transfer than any of enclosure case. However, heat conservation is drastically noted. *VRM_PPT point* shifts down to more than 4 times (42% drop) at $1/6^{th}$ of enclosure in comparison to without enclosure for maximum drop. Heat transfer does rises with varied enclosure distance and is maximum observed at 6% less than without enclosure. Though, in comparison to the fully enclosed with source oriented horizontally in low speed confined forced convection domain, heat transfer coefficient can drop to a minimum value close to 7% at $1/6^{th}$ enclosed and rises to a maximum of 1.4 times the *VRM_PPT point* at $4/6^{th}$ enclosed.The diverse trends reported needed the variation of heat transfer coefficient to be investigated at some intermediate orientation of the source. Heat transfer coefficient variation with enclosure distance was investigated at 45^{o} source orientation. Figure 9 accounts the variation more profound than the one at horizontal however, follows similar trend. With source orientated at 45^{o} in low speed forced convective confined configuration, the *VRM_PPT point* drops the minimum heat transfer coefficient at $1/6^{th}$ to more than 4 times (~44%) in comparison to no enclosure. However, the heat transfer coefficient rises with enclosure distance and maximum value occurs at drop of close to 6% than the without enclosure. In comparison to with fully enclosed top, minimum values of heat transfer coefficient occurs at $1/6^{th}$ enclosed with *VRM_PPT point* shifting 1.5 times. The maximum case occurs at $4/6^{th}$ enclosed with *VRM_PPT point* shifting 1.8 times at $4/6^{th}$ enclosed.

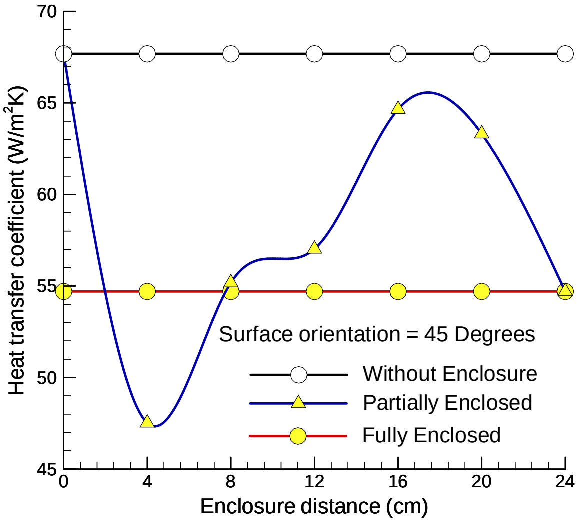

Figure 9 Variation of heat transfer coefficient with enclosure distance for 45^{o} surface orientation.

Applications with heat conservation as primary requirement can be looked at with intermediate orientation of the source. Keeping enclosure at $1/6^{th}$ position will allow maximum conservation of heat. However, as we keep increasing the enclosure distance more heat will start transferring though not equal to the fully open case. Considering the entire system, low forced convection through a confined channel, a heated square flat plate and top of the channel enclosed. Heat transfer is bound to the square plate and reduction in heat transfer signifies the flow not carrying sufficient amount of heat or redirected heat externally into the plate. From the main governing equation, heat transfer equation is inversely proportional to the average plate temperature or surface temperature. This states that, heat transfer coefficient can attain low values only in two cases viz., flow not carrying sufficient heat from the plate surface (plate surface remains hot) or heat is redirected externally into the plate which increases plate surface temperature thus reduced heat transfer coefficient. It is important to note that plate is uniformly heated in all cases and flow carries same heat for all cases. Entire change is owing to enclosure which is explored here for changing heat transfer coefficient under varying conditions.

When the top is enclosed fully, the heated flow carried from the plate interacts with the enclosure at the top which acts a potential heat sink. The enclosure takes away heat making hot flow denser which comes down and mixes with the hot flow coming up from plate. This interaction in time with plate increases the surface temperature reflected in the drastic fall of heat transfer coefficient. The continuous hot flow moving up owing to forced flow and buoyancy interacting with a heat sink (enclosure) at the top acts as an integral system. When the enclosure is partially open ($5/6^{th}$ enclosed) one can note that major portion of the top is still covered with a small opening for the flow to move out. Hence a small portion of flow escapes however, major portion undergoes forced interaction with a heat sink and turns back getting cold (denser) which forms a vortex. This vortex comprises mixing of hot flow moving up from plate (primary flow) and the denser fluid coming down (secondary flow). The enclosure effect in the form of heat transportation or heat conservation confines or depends primarily on the concentration of the two fluids. In vertical orientation, secondary flow (denser fluid) being more concentrated (sink takes more heat) will mix with primary fluid (hot fluid), subsequently reduce its temperature and will interact and reduce the hot plate surface temperature so the amount of heat transferred will increase. However, at other orientations, the intensity of hot flow overtakes and owing to limited trespassing and hot vortex formation, a part of heat is redirected back in to the source which results in plate surface temperature rising imitated in reduced heat transfer. Please note that for $4/6^{th}$ and $5/6^{th}$ enclosed locations the variation of heat transfer coefficient with surface orientation follows same trend as the one without enclosure. However, the effect stagnates when placed at the half ($3/6^{th}$ enclosed) where the variation of heat transfer coefficient with surface orientation is almost marginal. With more open area, more fluid can interlope and the mixing effect of two fluids comes down. The cold fluid concentration reduces and the fluid in immediate vicinity of source plate increases surface temperature reasonably. It is important to note that the heat transfer coefficient values for half enclosure lies between near fully enclosed and near fully open.

When the enclosure is placed covering top area less than half (here $2/6^{th}$ enclosed) the general trend reverses. The heat transfer coefficient varies with surface orientation but in reverse order with maximum in horizontal orientation and minimum in vertical. In this case, the concentration of hot fluid in the immediate vicinity of flat plate overtakes gradually making plate surface little hotter till plate surface orientation is 45^o and drastically beyond that to bring transportation of heat to a low value. The most interesting finding comes in the form of enclosure being placed at $1/6^{th}$ of top surface. Here, the variation of heat transfer coefficient with surface orientation is marginal and heat transfer values are noted lowest of all. More area is open so most of the flow is undisturbed. The little part enclosed may be attributed to the strong external influence on the square plate in the form of deflected flow which redirects to the adjoining wall and comes back on the plate forming a heated vicinity and increasing external heat effect on the plate which increases plate surface temperature heavily and thus drops heat transfer coefficient to a low value. Application wise the work can be effectively used as a potential mean for applications necessitating heat conservation in low forced convection environment within a confinement. The heat source can be placed at an optimized orientation and enclosure location as per heat transfer requirement of conservation or transfer.

4. CONCLUSION

An experimental investigation was carried out to understand the physics of enclosure effect on low speed confined forced convection in the aid of selected parameters viz., heat transfer coefficient, enclosure distance and surface orientation. Based on results obtained following conclusions may be drawn.

1) Enclosure primarily results in reduction of heat transfer from the heat source.
2) Enclosure effect is mostly noted within extreme cases of fully open and fully enclosed top. Partially enclosed top covers variety of physics noted with diverse variation in “**VRM-PPT point**” as per application requirements of heat conservation or heat transfer.
3) With top surface almost enclosed ($5/6^{th}$) “**devinn point**” is noticed as both phenomenon of heat transfer and heat conservation are present at this particular location. However, it results in diminishing returns when placed close to fully open ($1/6^{th}$) where heat transfer values falls drastically to even lower than fully enclosed.
4) Intermediate locations of enclosure viz., $2/6^{th}$, $3/6^{th}$, $4/6^{th}$, forms a wide range for heat conservation applicability.
5) Enclosure phenomenon works with augmentation of source surface temperature to justify minimum heat transfer in the form of external effect which reduces heat transfer less than without enclosure cases. This is noted in the balance of hot and cold fluid concentration. Phenomenon rests on the fluid with higher intensity viz., cold or hot.
6) The results predicted by experimental apparatus were validated and matched reasonably well. The presence of enclosures can prevent excess heat loss and is expected to be useful in energy conservation applications.

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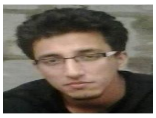

Pranjal Nag is undergraduate student in Department of Aerospace Engineering, SRM University Chennai. He is interested in the broad field of propulsion.

Shivam Gupta is undergraduate student in Department of Aerospace Engineering, SRM University Chennai. He is interested in the basic Fluid and Thermal sciences.

Pratik Tiwari is undergraduate student in Department of Aerospace Engineering, SRM University Chennai. He is interested in broad field of heat transfer and combustion and has worked on convective mode of heat transfer along with exploring physics in impinging jets and smoldering combustion.

Vinayak Malhotra received the B.E. from The Aeronautical Society of India and Master of Science degree in Aerospace Engineering from Indian Institute of Technology. During his Master program, he worked on flame spread in microgravity. He now professes in Department of Aerospace Engineering, SRM University Chennai.