EXPERIMENTAL INVESTIGATION OF LAMINAR NATURAL CONVECTION HEAT TRANSFER IN A RECTANGULAR ENCLOSURE WITH AND WITHOUT INSIDE PARTITIONS

Prof. Hussain Y. Mahmood
Mechanical Engineering Department
University of Baghdad

Ahmed Abdul Ameer Abdul Raheem
Mechanical Engineering Department
University of Baghdad

ABSTRACT

Experimental study has been conducted for laminar natural convection heat transfer of air flow through a rectangular enclosure fitted with vertical partition. The partition was oriented parallel to the two vertical isothermal walls with different temperatures, while all the other surfaces of the enclosure were insulated. In this study a test rig has been designed and constructed to allow studying the effect of Rayleigh number, aperture height ratio, partition thickness, the position of aperture according to the side walls and according to the height, the position of the partition according to the hot wall, and partition inclination. The experiments were carried out with air as the working fluid for Rayleigh number range ($5*10^7 – 1.3*10^8$) and aspect ratio of (0.5). 22 different configurations of partition were used in this study these are:

a) Undivided enclosure (no – partition).
b) (21) Cork partitions of different shapes.

Empirical correlations for average Nusselt number are obtained for the different cases tested. The results show that heat transfer is independent on the partition position according to the cold wall and according to the upper or lower walls, while it shows that heat transfer is sensitive to:

1. Rayleigh number (Ra), which increase with increasing Ra.
2. Aperture height ratio ($A_p=h_p/H$), which is found that when A_p= 5/6 (case 2,3), the reduction in heat transfer is 10.3%, while when A_p=1/2 (case 4,5), the reduction is 17.2% compared with the non partitioned enclosure.
3. Aperture position according to the height, which is found that when the aperture at the centre of the partition (case 13), the reduction in heat transfer is 16.7%, while when the aperture displaced to the upper surface (case 14), the reduction is 19% compared with the non partitioned enclosure.
4. Partition thickness (t), which is found that when t = 10 mm (case 4,5) the reduction in heat transfer is 17.2%, while when t = 150 mm (case 16) the reduction is 20.5% compared with the non partitioned enclosure.
5. Partition inclination (δ), which is found that the rate of heat transfer reduced with increasing δ as shown:
   a. For $\delta = 30^\circ$ toward the cold wall (case 22), the reduction in heat transfer is 18.2%.
   b. For $\delta = 45^\circ$ toward the cold wall (case 18), the reduction in heat transfer was 21.9%.
   c. For $\delta = 60^\circ$ toward the cold wall (case 20), the reduction in heat transfer is 30.2%.
   d. For $\delta = 30^\circ$ toward the hot wall (case 21), the reduction in heat transfer is 31.3%.
   e. For $\delta = 45^\circ$ toward the hot wall (case 17), the reduction in heat transfer is 40.7%.

f. For $\delta = 60^{\circ}$ toward the hot wall (case 19), the reduction in heat transfer is 42.1%.

الخلاصة

اجريتْ دراسة عملية لانتقال الحرارة بالحمل الطبيعي الطباقي لجريان الهواء في حيز مغلق بوجود حواجز ذات أشكال هندسية مختلفة وبعدم وجود حاجز، يتم وضع الحاجز بين سطحين أحدهما بارد والاخر ساخن بشرط ثبوت درجة الحرارة، بينما الاسطح الباقية معزولة حرارياً. الجانب العملي اشتمل على تصميم وبناء جهاز مختبري لدراسة تأثير رقم رايلي (Ra)، نسبة ارتفاع فتحة الحاجز الى ارتفاع الحيز (H_p)، سمك الحاجز (t)، موقع الفتحة في الحاجز، موقع الحاجز، و ميلان الحاجز. اجريت الدراسة العملية باستخدام الهواء كوسط ناقل للحرارة (مائع العمل) ولمدى عدد رايلي (٥×١٠^٧ – ١.٣×١٠^٨) ولنسبة ارتفاع الحيز الى طوله (النسبة الباعية) تساوي نصف. ولقد تم استخدام (٢٢) حاجزاً بمختلف الاشكال :

- ✓ غرفة مفردة بدون حاجز.
- ✓ حواجز من الفلين وعددها (٢١).

تم استنتاج علاقات تجريبية للحالات المختلفة التي درست، وقد بينت النتائج ان الحرارة المنتقلة لا تعتمد على موقع الحاجز نسبة الى الجدار البارد، بروز الحاجز من الاعلى او من الاسفل، وانما تعتمد على:

١) رقم رايلي (Ra) ، حيث وجد ان معدل انتقال الحرارة يزداد بزيادة رقم رايلي.
٢) نسبة ارتفاع فتحة الحاجز الى ارتفاع الحيز(A_p)، حيث وجد عندما تكون (A_p=5/6) (حالة ٢,٣)، ان الحاجز يقلل من انتقال الحرارة بنسبة %١٠.٣، بينما عندما (A_p=1/2) (حالة ٤,٥)، الحاجز يقلل من انتقال الحرارة بنسبة ١٧.٢%، مقارنة بانتقال الحرارة في الحيز بعدم وجود حاجز.
٣) موقع الفتحة نسبة الى الارتفاع، حيث وجد عندما تكون الفتحة في وسط الحاجز (حالة ١٣)، ان الحاجز يقلل من انتقال الحرارة بنسبة ١٦.٧%، بينما عندما تكون الفتحة قرب الجدار العلوي (حالة ١٤) الحاجز يقلل من انتقال الحرارة بمقدار ١٩%، مقارنة بانتقال الحرارة في الحيز بعدم وجود حاجز.
٤) سمك الحاجز (t)، حيث وجد عندما يكون سمك الحاجز ١٠ ملم (حالة ٤و٥)، ان الحاجز يقلل من انتقال الحرارة بمقدار ١٧.٢%، بينما عندما يكون سمك الجدار ١٥٠ ملم (حالة ١٦)، الحاجز يقلل من انتقال الحرارة بمقدار ٢٠.٥%، مقارنة بانتقال الحرارة في الحيز بعدم وجود حاجز.
٥) زاوية ميلان الحاجز (δ)، حيث وجد ان معدل انتقال الحرارة يقل بزيادة زاوية ميلان الحاجز.

- عندما يميل الحاجز بزاوية 30° باتجاه الجدار البارد (حالة ٢٢)، الحاجز يقلل من انتقال الحرارة بمقدار ١٨.٢%.
- عندما يميل الحاجز بزاوية 45° باتجاه الجدار البارد (حالة 18)، الحاجز يقلل من انتقال الحرارة بمقدار ٢١.٩%.
- عندما يميل الحاجز بزاوية 60° باتجاه الجدار البارد (حالة ٢٠)، الحاجز يقلل من انتقال الحرارة بمقدار ٣٠.٢%.
- عندما يميل الحاجز بزاوية 30° باتجاه الجدار الحار (حالة ٢١)، الحاجز يقلل من انتقال الحرارة بمقدار ٣١.٣%.
- عندما يميل الحاجز بزاوية 45° باتجاه الجدار الحار (حالة ١٧)، الحاجز يقلل من انتقال الحرارة بمقدار ٤٠.٧%.
- عندما يميل الحاجز بزاوية 60° باتجاه الجدار الحار (حالة ١٩)، الحاجز يقلل من انتقال الحرارة بمقدار ٤٢.١%.

Keywords: Laminar, Natural Convection, Rectangular Enclosure, Partitions.

INTRODUCTION

In the free convection, fluid motion is due to buoyancy effect, whereas in forced convection it is externally imposed. Buoyancy is due to the combined presence of a density gradient within the fluid and a body force that is proportional to the fluid density. As energy costs have escalated, there has been an increasing awareness of the impact that building design decisions can have an effect on energy consumption in the resulting structure. In addition to energy issue, the designer must also take into account aesthetic, economic, and functional requirements of the building. The most effective design solution depends on proper weighting of all relevant factors. In order for energy to have an appropriate weight in the decision, adequate accuracy in energy calculations must be provided. **Emery (1969)** studied experimentally free convection in a narrow vertical two – dimensional layer with a vertical baffle of variable height (l), (l/H = 0.25,0.5,0.75,0.9), the baffle was located, halfway between the hot and cold walls with equal spaces at the top and bottom. The experiments were carried out with pure glycerin (USP), the range of aspect ratio A=H/L = 10 - 40, and Rayleigh number (Ra_L) 1.5×10^6. Overall heat transfer measurements and temperature profiles were obtained; the result showed that there was no appreciable reduction of the overall free convection heat transfer although fairly significant changes do occurred in the local temperature profile. **Lin and Bejan (1983)** studied experimentally and analytically the phenomenon of heat transfer by natural convection in a rectangular enclosure fitted with an incomplete internal partition. The experiments were carried out in a water – filled enclosure with adiabatic horizontal walls and vertical walls maintained at different temperatures. Heat transfer measurements and flow visualization were conducted in the Rayleigh number range ($10^9 – 10^{10}$), for aperture height ratio h/H = 1, 1/4, 1/8, 1/16 and 0, where h and H were the height of the internal opening (in the partition) and the height of the enclosure, respectively. It was demonstrated that the aperture ratio h/H had a strong effect on both the heat transfer rate and the flow pattern. The heat transfer data were correlated satisfactorily by the equation:

$$Nu=\frac{0.366\times Ra^{0.25}}{Ap^{-0.75}+0.5} \qquad \ldots\ldots\ldots\ldots(1)$$

The flow and temperature fields in this study were reported. **Sadiq Elias Abdullah (1997)** studied experimentally natural convection heat transfer in undivided and partially divided enclosures. The enclosure was fitted with vertical partition at midway between two vertical isothermal walls, one of which was heated and the other was cooled, while the other surface of the enclosure was insulated. Different configurations of partitions were used. The experiments were carried out using air as the working fluid over Rayleigh number range (Ra_H) of ($6*10^7 – 1.22*10^8$), and aspect ratio of (0.5). The results showed that the location of the opening has a significant effect on the heat transfer together with the aperture height ratio, while a little effect of aperture width ratio was noticed. **Ahmed Fakhrey Khudheyer (1999)** investigated experimentally the heat transfer by natural convection in a rectangular enclosure fitted with a vertical adiabatic partition. The partition was oriented parallel to the two vertical isothermal walls. One of which was heated by heaters and the other cooled by water while all the other surfaces of the enclosure were insulated. The experiments were carried out with air as the working fluid for Rayleigh number range (Ra_H) ($6\times 10^7 – 1.5 \times 10^8$) and an aspect ratio (height/ width) of 1/2, 14 different configurations of partition were used in this study.

For all these partitions the effect of their location was examined with respect to the hot wall by the ratio (x/L = 0.25, 0.5, 0.75), the results indicate that heat transfer was sensitive to the aperture height and it was independent of the aperture width and the upper or down extensions do not effect on the heat transfer. **Baïri, et-al (2007)** studied experimentally and numerically steady state natural convection in rectangular cavities filled with air. The active walls, hot and cold, of the cavity are maintained isothermal at temperatures T_h and T_c, respectively, and the other walls that close the cavity are adiabatic. Different angles of inclination $\emptyset$ of the cavity from 0º to 360º are considered. Two aspect ratios A= H/L = 0.75 and 1.5 are treated. The numerical study is carried out by means of the finite volume method and provides the thermal and dynamic maps of the fluid for several geometrical configurations obtained while varying $\emptyset$, A and $\Delta T = T_h - T_c$. the range of Rayleigh number, Ra_L from 10 to 10^8. The following relations was gotten for A=0.75 and 1.5, at different angles of inclination α, and $10^3 \leq Ra_L \leq 10^8$.

A (deg)	Correlation
0, 30, 360	$Nu_{ave} = 0.147\ Ra_L^{0.287}$
45, 135, 315	$Nu_{ave} = 0.130\ Ra_L^{0.305}$
60, 90	$Nu_{ave} = 0.133\ Ra_L^{0.304}$
270	$Nu_{ave} = 0.058\ Ra_L^{0.058}$

The main aim of the present investigation is to determine the effect of:

1. Rayleigh number (Ra) on heat transfer.
2. Different partition shapes on heat transfer.
3. Aperture height ratio (A_p) on heat transfer.
4. Partition thickness (t) on heat transfer.
5. The position of the aperture according to the side walls and according to the height on heat transfer.
6. The position of the partition according to the cold wall (L_p) on heat transfer.
7. Partition inclination (δ) on heat transfer.

EXPERIMENTAL APPARATUS AND PROCEDURE

A rig was designed and constructed to investigate the heat transfer by natural convection due to a temperature difference between hot and cold walls forming a rectangular cavity with air as the working fluid. Three individual heaters were used, each having a separate power control to maintain a uniform "isothermal" temperature over the entire hot wall surface. The apparatus dimensions were selected to simulate a living zone with inside dimensions of (300 mm) height, (300 mm) width, (600 mm) long, aspect ratio (A = H/L = 0.5), as shown in plate (1), and figure (1). The rig consists of the following parts:

1- Styrofoam enclosure
2- Plexiglas sheets
3- Hot wall assembly
4- Cold wall assembly
5- Different types and shapes of partitions
6- Instrumentation.

The Partitions

The partitions were inserted in the ceiling to divide the enclosure into two parts connecting each other by aperture at the partitions. 22 partitions classified into five groups as shown in figure (2).

Test Procedure:

The following steps were made in each test:

1- The required type of partition is positioned in place.
2- The power supply and water flow rate were set at the desired level.
3- The temperature of the heating plate was adjusted by varying the current supplied to the heaters with the help of a Rheostat for each heater.
4- During the transient condition, before reaching the steady state, the temperature of the hot wall was examined a number of times by means of thermocouples to check for a uniform temperature distribution over the hot wall, the current supplied to each heater was controlled for this purpose.
5- The steady state was identified by little or no change in the readings for (20-30) minuets.
6- After the experiments reaching steady state, the front and top insulations surfaces of the apparatus are removed.
7- Burning incense sticks are hung from the top of the enclosure near the cold wall to generate the smoke required for flow visualization technique.
8- A strong diffuse light was found to be the most appropriate light source for visualization the flow inside the enclosure. A diffuse light source was required to avoid light reflections from the enclosure walls. The enclosure was illuminated from the top, with the light diffusively reflected into the enclosure.
9- Capturing the smoke from the front surface.

A total of 110 tests were carried out for 22 different cases.

Calculation Procedure:
Analysis of Heat Transfer:

The electrical power input to the hot wall was measured by using the current and the voltage across each heater. The heat is transferred in the following ways:

1- Heat loss by conduction to the surrounding.
2- Heat exchange by radiation in the enclosure.
3- Heat transfer by natural convection in the enclosure.

Heat Loss by Conduction:

The conduction heat loss is assumed to consist of the following components:

A) Heat loss from the enclosure (excluding the hot and cold walls) to the surrounding.

Conduction shape factor method was used to estimate this loss. The following general equation, Bejan, A., (1994), was used.

$$q_{c1} = k\,(T_r - T_a)\left[\frac{2(H*L+W*L)}{X} + 2.16\,(L + H + W) + 1.2X\right] \qquad (2)$$

It is assumed that the wall was constructed from (50 mm) thick layer of Styrofoam with k =0.025 W/m.K.

$$q_{c1} = 0.025\,(T_r - T_a)\left[\frac{2(0.3*0.6+0.3*0.6)}{0.05} + 2.16\,(0.6 + 0.3 + 0.3) + 1.2*0.05\right]$$

$$q_{c1} = 0.4263 * (T_r - T_a) \quad W \qquad (3)$$

B) Heat loss from back side of the hot wall. The following equation was used to calculate this loss.

C) $q_{c2} = U.\, A_h\, (T_h - T_c)$ (4)

D) $$\frac{1}{U} = \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{1}{h_\circ}$$

Where:

$h_\circ$ = Exterior convective heat transfer coefficient (ASHRAE 1981) = 3.08 $W/m^2.K$

x_1 = Styrofoam thickness = 50 mm

k_1 = Styrofoam thermal conductivity = 0.025 W/m.K

x_2 = Asbestos thickness = 20 mm

k_2 = Asbestos thermal conductivity = 0.17 W/m.K

$$\frac{1}{U} = \frac{0.05}{0.025} + \frac{0.02}{0.17} + \frac{1}{3.08}$$

q_{c2} = 0.0368 ($T_h - T_a$) W (5)

Total heat loss by conduction was

($q_c = q_{c1} + q_{c2}$) (6)

Heat Exchange by Radiation:

Absorption factor method was used in calculating the heat transfer by radiation from the hot wall to other walls. The general equation used was that given by Sucec, (1985):

$$(q_r)_j = E_j A_j - \sum_{i=1}^{n} B_{ij} * A_i * E_i \quad (7)$$

Where:

A_i = Area of i_{th} surface (m^2)

$$E_i = \sigma . e_i . T_i^4$$

Where:

ϵ_i = Emissivity of i_{th} surface

T_i = Temperature of i_{th} surface (K)

B_{ij} = Absorption factor which is defined as the fraction of radiant energy emitted by the i_{th} surface which is eventually absorbed by the j_{th} surface after complex reflection pattern.

Absorption factor is obtained from solving the equation:

$[k_{ij}]\ \{B_{ij}\} = \{ - \epsilon_i F_{ij} \}$

Where:

$k_{ij} = F_{ij}\ \rho_i$

ρ_i = Reflectivity of i_{th} surface

ρ = 0.8 for Aluminum surfaces

ρ = 0.06 for Styrofoam surfaces

F_{ij} = Shape factor between walls

Heat Transfer by Natural Convection in the Enclosure:

The net heat transfer by natural convection between two walls was obtained from the following equation:

$q_{conv} = I * V^{-} - (q_c + q_r)$ (8)

The convection heat transfer coefficient was calculated as follows:

$q_{conv} = h.\ A_h\ (T_h - T_c)$ (9)

$$h = \frac{q_{conv}}{A_h (T_h - T_c)} \quad (10)$$

Dimensionless Groups:

The results have been analyzed with the help of the well known dimensionless parameters given below:

Prandtl Number = $\frac{C_p \mu}{k_f}$

Rayleigh Number =

$$\frac{\beta * g * \Delta T * H^3}{\upsilon^2} * \frac{C_p * \mu}{K_f}$$

Nusselt Number = $\frac{h * H}{k_f}$

The properties were interpolated from table given by Sucec (1985), depending on a reference temperature given by:

$T_r = (T_h - T_c) * 0.5$

RESULTS AND DISCUSSION

A summary of the results and correlations of the present work are given in table (1). Natural convection heat transfer in enclosure is calculated by these empirical correlations. The general formula of these empirical correlations is:

$$Nu = C.Ra^n$$

Enclosure without Partition:

This case is the reference for the other cases. The heat transfer rate increases with increasing Rayleigh number due to increase temperature difference (ΔT). The present results are compared with those of **Sadiq Elias (1997)** since these studies are curried out in a small test cell using air as the working fluid and at the same range of Rayleigh number. The comparison is shown in figure (3). The comparison shows good agreement between the two experimental works, the percentage deviation was found to be about (0.5 %).

Group One:

The heat transfer results and correlations are shown in figure (4). The partition was found to reduce the heat transfer compared with the non partitioned room as shown:

A) Case (2,3); Ap = 5/6; Reduction = 10.3%
B) Case (4,5); Ap = 1/2; Reduction = 17.2%
C) Case (16); Ap = 1/2; Reduction = 20.5%

The reduction was produced as a result of relatively static pocket of hot fluid which was trapped in the upper zone of the hot zone of the test cell when the partition was at the upper wall. The trapped hot fluid inhibits convective heat transfer from upper zone of the hot wall, (see plate (2.a)). While when the partition was at the lower wall the cold fluid will trapped in the lower zone of the cold wall (see plate (2.b,c))

The comparison between case (2) and case (3), and between case (4) and case (5), explain that for the same distance between partition aperture and the upper surface or the lower surface, there is no effect on heat transfer.

The comparison between cases (2,3) and cases (4,5) represent the effect of the aperture height ratio (A_p) on natural convection heat transfer. It is found that the aperture height ratio has a significant effect on the heat transfer, the reason of that is the velocity of fluid is very low near the partition and it is larger at the aperture, therefore a part of fluid moving from the hot zone to the cold zone, heat is transferred with this moving fluid, due to buoyancy force which occurs because of the difference between hot and cold wall, because the amount of fluid transfer depends on aperture area so the amount of heat transfer also depends on aperture area on the partition.

The comparison between cases (4,5) and case (16) explain the effect of partition thickness on the convection heat transfer, the figure shows a little effect in heat transfer it shows that case (16), (t=150 mm) reduce heat transfer by about (3.3%) compared with cases (4,5), (t=10mm).

Group Two:

Figure (5) shows heat transfer results and correlations for group two.

The comparison between case (7) and case (8), explain the effect of the position of aperture according to the side wall it is found that for case (7) (aperture at side) the heat transfer reduction is about (23.2%) compared with case of non partition room, while for case (8) (aperture at centre) the reduction in heat transfer was about (20.1%). The drop in heat transfer in case (7) is larger than that for case (8), this reduction was expected since the blockage to air flow is greatest, in addition to the air trapped in the upper third of the hot zone of the test cell there is relatively static pocket of hot fluid which was trapped in opposed side of the aperture side.

In case (6) the partition found to reduce the heat transfer by about (9.1%) respect to the non partitioned room.

Group Three

Heat transfer results and correlations are shown in figure (6). The comparison between case (9) and case (10), and between case (13) and case (15) (in group four) represent the effect of aperture distribution according to the vertical axis for multi-aperture. It is found that for case (9) the heat transfer reduction is about 15.5%, while for case (10) the reduction was about 18% compared with the case of non partition. The reduction was expected since the fluid motion increase near the upper and lower surface and decrease near the centre of the enclosure.

Case (11) shows that the reduction in heat transfer is smaller than in case 9 and 10 it is about 12.2%, it is due to increase in aperture area.

Group Four:

Figure (7) shows the heat transfer results and correlations for this group. The partition was found to reduce the heat transfer compared with the non partitioned room as shown:

A) Case (15); Reduction = 10.6%

B) Case (12); Reduction = 13.5%

C) Case (13); Reduction = 16.7%

D) Case (14); Reduction = 19%

The comparison between case (13) and case (14), explain the effect of the distance between the aperture centre and the partition centre. The figure shows that the rate of heat transfer decreases when this distance increase. The reason of this effect is due to air falling and traps down in the cold zone.

Group Five:

Heat transfer results and comparison are shown in figure (8). This group explain the effect of the partition inclination at (δ = 30˚, 45˚,60˚) according to the hot wall, and at (30˚, 45˚, 60˚) according to the cold wall. The partition was found to reduce the heat transfer compared with the non partitioned room as shown:

- Case (22); Reduction = 18.2%
- Case (18); Reduction = 21.9%
- Case (20); Reduction = 30.2%
- Case (21); Reduction = 31.3%
- Case (17); Reduction = 40.7%
- Case (19); Reduction = 42.1%

The figure shows:

A- At the same angles [case (17, 18), case (19,20) and case (21, 22)] when the partition inclines toward the hot wall, the rate of heat transfer decrease more than that when the partition incline toward cold wall. Because in case when the partition inclines toward hot wall the partition obstructs the motion of fluid more than when the partition inclines to the cold wall. in addition to the air trapped in the upper part of the cold zone of the test cell there is relatively static pocket of cold fluid which was trapped in opposed side of the aperture side.

B- For cases when the partition inclines toward the hot wall, [case (17), case (19) and case (21)], the slope of the curve increase when the angle decrease. This mean that at low temperatures the rate of heat transfer is very low, and at high temperatures the rate of heat transfer become very high.

C- For cases when the partition inclines toward the cold wall, [case (18), case (20) and case (22)], the rate of heat transfer increase, when the angle decrease. The reason of that is due to the separation occured in the upper zone between the partition and the hot wall. And this separation increase when the angle increase (see plate 3, 4).

CONCLUSIONS

The conclusions of this investigation can be summarized as follows:

1- The aperture height ratio (A_p) have a significant effect on heat transfer. Its increase with increases aperture height, which is found that when A_p= 5/6 (case 2,3), the reduction in heat transfer was 10.3%, while when A_p=1/2 (case 4,5), the reduction was 17.2% compared with the non partitioned enclosure.

2- The position of the aperture near the upper surface or near the lower surface didn't affect on heat transfer for the same distance.

3- The partition's thickness has a little effect on heat transfer, and it is found that when t = 10 mm (case 4,5) the reduction in heat transfer was 17.2%, while when t = 150 mm (case 16) the reduction was 20.5% compared with the non partitioned enclosure.

4- When the aperture position change from the centre toward the side wall the natural convection heat transfer decreases, and it is found that when the aperture at the centre of the partition (case 13), the reduction in heat transfer was 16.7%, while when the aperture displaced to the upper surface (case 14), the reduction was 19% compared with the non partitioned enclosure.

5- Natural convection heat transfer depend on the position of the aperture according to the height. Its increases when the aperture near the centre.

6- For multi-apertures the rate of heat transfer increases when the apertures distributed near the upper and lower walls.

7- The position of the partition according to the hot wall didn't affect on heat transfer.

8- For the case of partition inclination:

A) Heat transfer increases when the partition inclined toward the cold wall for the same angle.

B) Increases the angle of inclination according to the upper wall decrease the heat transfer.

C) When the partition inclined toward the hot wall, its found that the slope of curve increases when the angle of inclination decrease.

which is found that the rate of heat transfer reduced with increasing δ as shown:

- For $\delta = 30^{\circ}$ toward the cold wall (case 22), the reduction in heat transfer was 18.2%.
- For $\delta = 45^{\circ}$ toward the cold wall (case 18), the reduction in heat transfer was 21.9%.
- For $\delta = 60^{\circ}$ toward the cold wall (case 20), the reduction in heat transfer was 30.2%.
- For $\delta = 30^{\circ}$ toward the hot wall (case 21), the reduction in heat transfer was 31.3%.
- For $\delta = 45^{\circ}$ toward the hot wall (case 17), the reduction in heat transfer was 40.7%.
- For $\delta = 60$ toward the hot wall (case 19), the reduction in heat transfer was 42.1%.

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NOMENCLATURE

Latin Symbols

Symbol	Description	Units
A	Aspect ratio (H/L)	–
A_h	Area of hot wall = 0.09 m^2	m^2
A_c	Area of cold wall	m^2
A_p	Aperture height ratio (h_p/H)	–
A_w	Aperture width ratio (w_p/H)	–
E_i	Emissive power of i_{th} surface	W/m^2
g	Gravity acceleration	m/s^2
Gr_H	Grashof number	–
h	Heat transfer coefficient	$W/m^2.\ ^oC$
h_p	Aperture height	m
H	Enclosure height	m
I	Current	Amp.
k	Thermal conductivity	$W/m.\ ^oC$
L	length of enclosure	m
L_p	Distance between cold wall and partition	m
Nu_{ave}	Average Nusselt number	–
p	Pressure	N/m^2
Pr	Prandtl number	–
q_{conv}	Heat lost by convection	W
q_c	Rate of heat lost by conduction	W
q_r	Rate of heat lost by radiation	W
Ra_H	Rayleigh number	–

t	Partition thickness	m
T	Temperature	^oC
T_c	Temperature of cold wall	^oC
T_h	Temperature of hot wall	^oC
T_r	Reference temperature $(T_h+T_c)/2$	^oC
T_∞	Ambient temperature	^oC
U	The overall heat transfer coefficient	$W/m^2.\,^oC$
$\bar{V}$	Voltage	volt
w_p	Aperture width	m
W	Enclosure width	m
x,y	Cartesian coordinate	m

Creak Symbols

Symbol	Description	Units
α	Thermal Diffusivity	[$m^2 s^{-1}$]
δ	Angle of partition inclination	degree
Ø	Angle of enclosure inclination	degree
β	Coefficient of volume expansion	1/K
μ	Dynamic viscosity	kg/m.s
υ	Kinematic viscosity	m^2/s
σ	Stefan Boltizmann constant	W/m^2K^4
Δ	Difference between two values	–

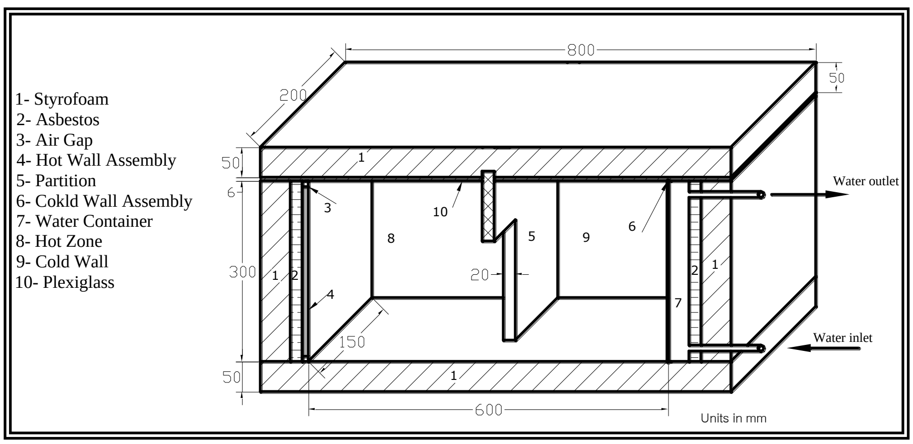


Fig.1: Cross Section of the Test Rig

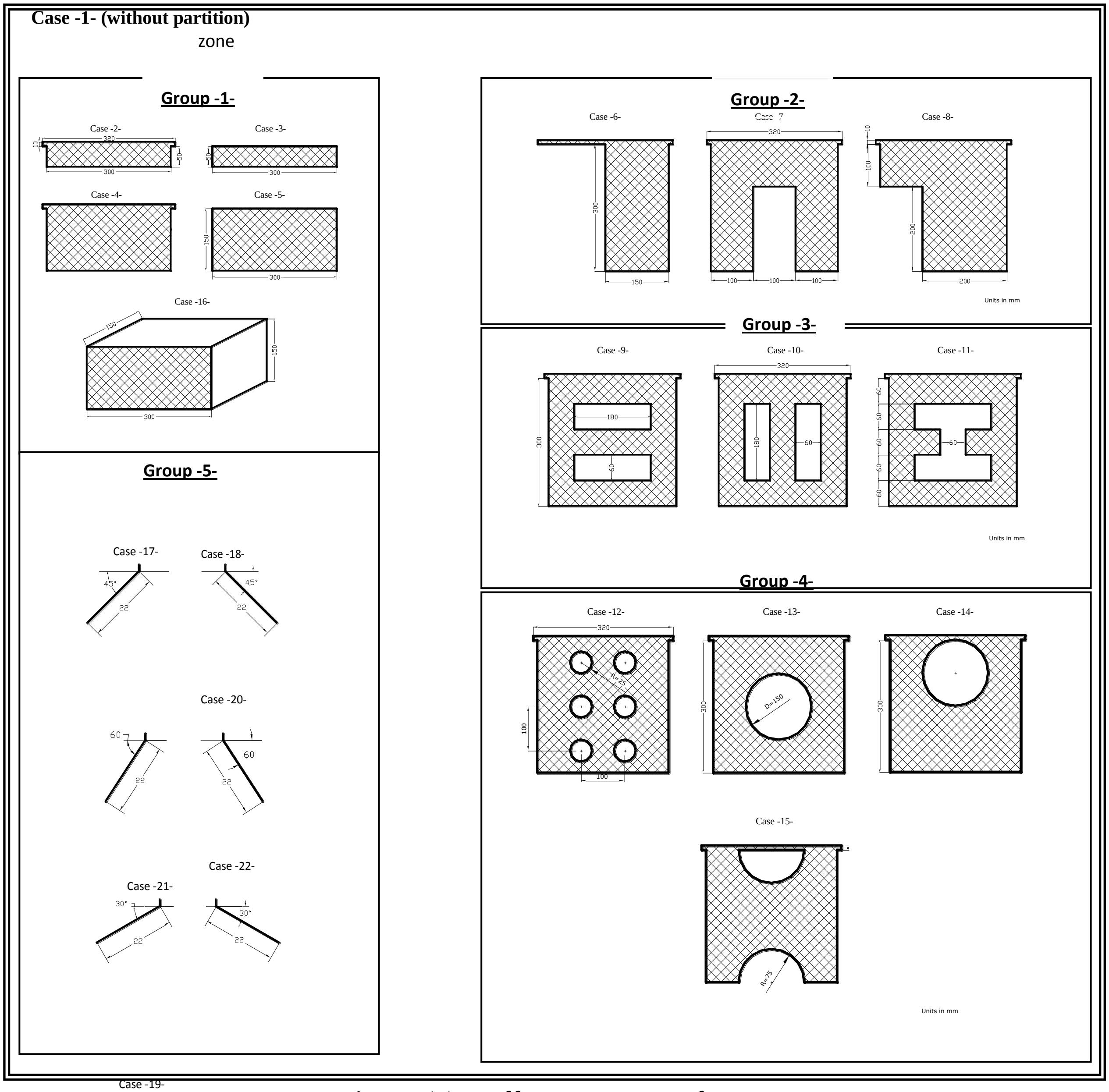
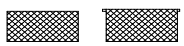
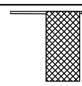
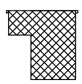
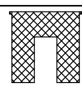
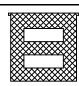
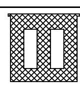
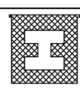
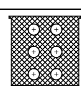
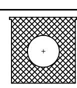
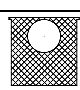
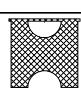
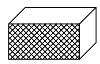
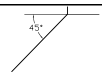
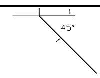
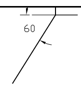
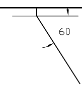
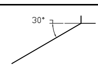
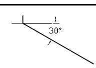


Figure (2): Different Patterns of Partitions

Table (1) Summary of Results

Case No.	Case Study	Nu = C.Ra^n C	n	$A_p = \frac{h_p}{H}$	$A_w = \frac{w_p}{W}$
1	Without Partition	0.00282	0.5027	1	1
2,3		0.00045	0.5972	1/6	1
4,5		0.00031	0.6139	1/2	1
6		0.00097	0.5555	1	0.5
7		0.00011	0.6659	2/3	1/3
8		0.00018	0.6397	2/3	1/3
9		0.00101	0.5496	2/5	3/5
10		0.00132	0.5334	3/5	2/5
11		0.00078	0.5659	3/5	3/5
12		0.00329	0.4864	1/2	1/3
13		0.00159	0.5240	1/2	1/2
14		0.00166	0.5204	1/2	1/2
15		0.00172	0.5237	1/2	1/2

١٦		0.00036	0.6035	1/2	1
17		1.50638E-6	0.8860	1/2	1
18		0.00044	0.5904	1/2	1
19		1.2276E-5	0.7702	2/3	1
20		0.00017	0.6352	2/3	1
21		3.53956E-7	0.9733	1/3	1
22		0.00055	0.5813	1/3	1

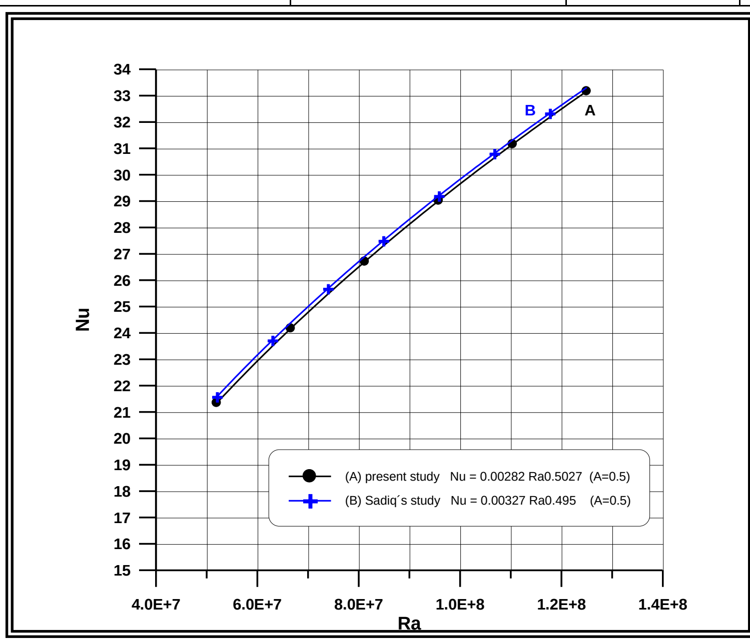
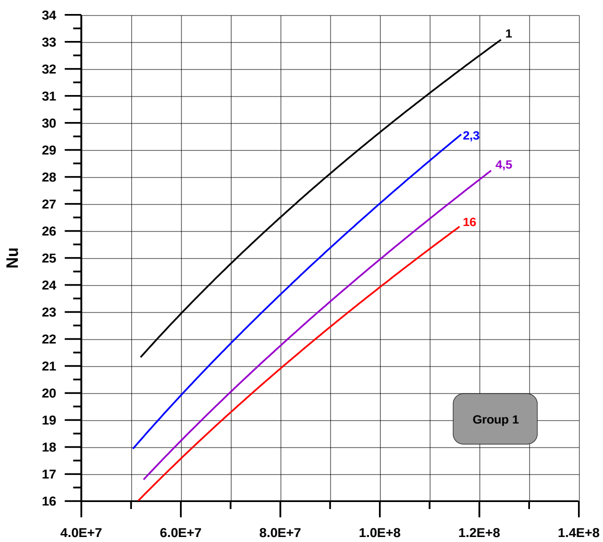


Figure (3) A comparison of average Nusselt Number Results in a single room



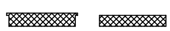
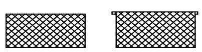
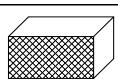
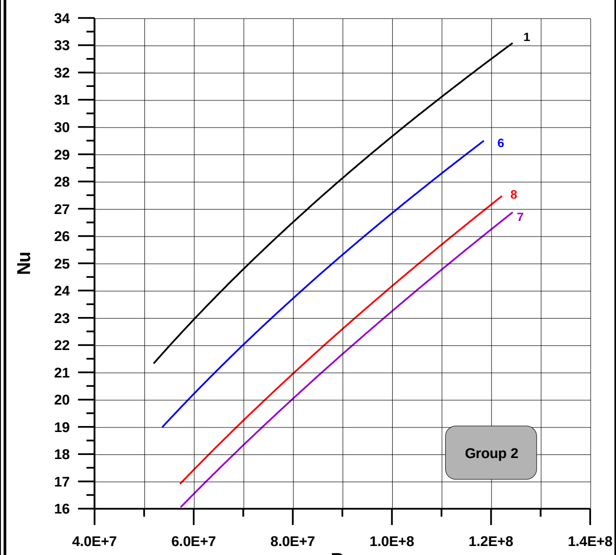
Case	Case Study	Correlation
1	Without Partition	Nu = $0.00282Ra^{0.5027}$
2,3		Nu = $0.00045Ra^{0.5972}$
4,5		Nu = $0.00031Ra^{0.6139}$
16		Nu = $0.00036Ra^{0.6035}$

Figure (4) a Comparison of Heat Transfer Results for Group One.



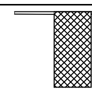
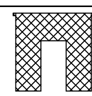
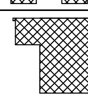
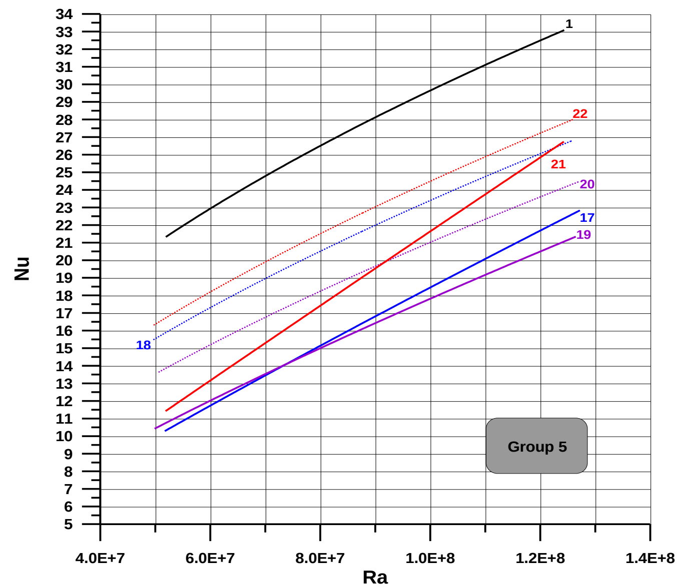
Case	Case Study	Correlation
1	Without Partition	Nu = $0.00282Ra^{0.5027}$
6		Nu = $0.00097Ra^{0.5555}$
8		Nu = $0.00018\ Ra^{0.6397}$
7		Nu = $0.00011\ Ra^{0.6659}$

Figure (5) a Comparison of Heat Transfer Results for Group Two.



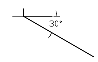
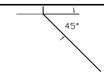
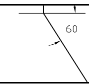
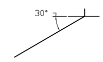
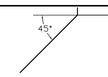
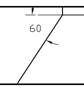
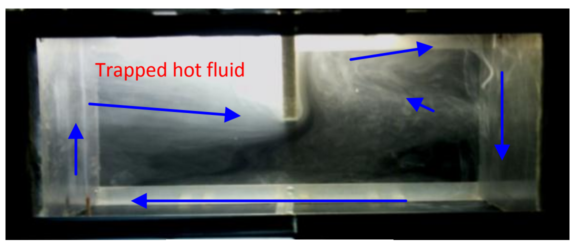
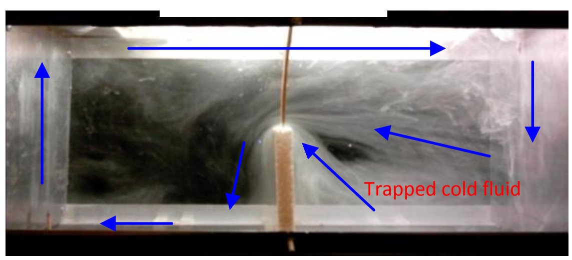
Case	Case Study	Correlation
1	Without Partition	Nu = $0.00282Ra^{0.5027}$
22		Nu = $0.00055Ra^{0.5813}$
18		Nu = $0.00044Ra^{0.5904}$
20		Nu = $0.00017Ra^{0.6352}$
21		Nu = $3.5396E\text{-}7Ra^{0.9733}$
17		Nu = $1.5064E\text{-}6Ra^{0.8860}$
19		Nu = $1.2276E\text{-}5Ra^{0.7702}$

Figure (8) a Comparison of Heat Transfer Results for Group Five.

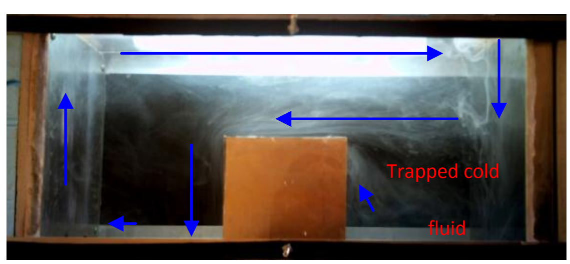
Plate (1) Experimental Apparatus



A) Ra = 8.24234*10^7

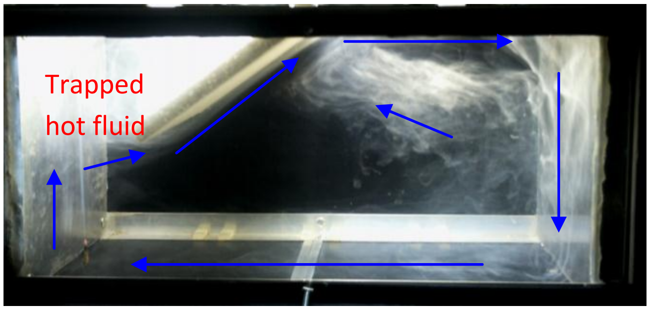


B) Ra = 5.25016*10^7

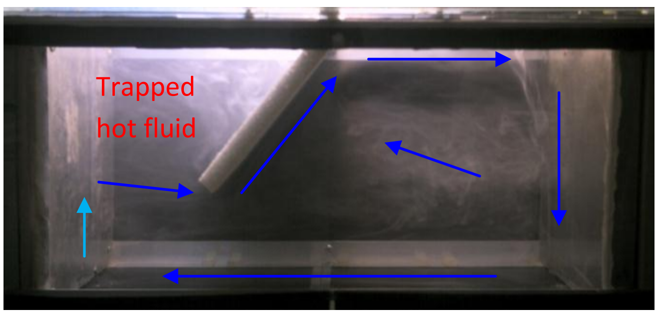


C) Ra = 4.97793*10^7

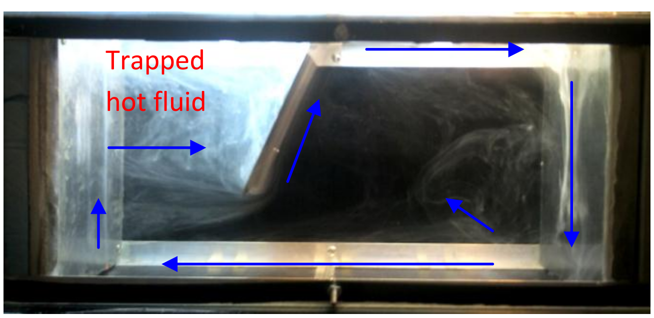
Plate (2) Flow Patterns using smoke visualization



A) Ra = $5.18627*10^7$

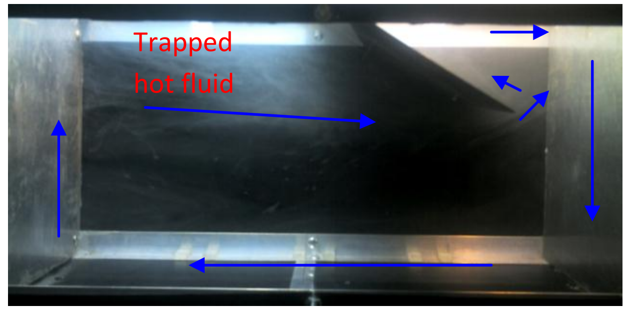


B) Ra = $8.84108*10^7$

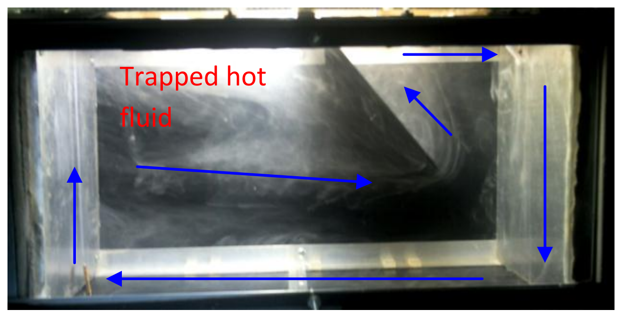


C) Ra = $4.98698*10^7$

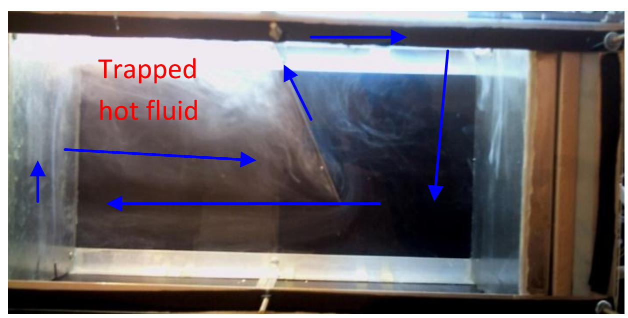
Plate (3): Flow Patterns Using Smoke Visualization (Partitions Inclined Toward Hot Wall)



A) Ra = $4.96998*10^7$



B) Ra = $7.2608*10^7$



C) Ra = $8.97188*10^7$

Plate (4): Flow Patterns Using Smoke Visualization (Partitions Inclined Toward Cold Wall)