NATURAL CONVECTION HEAT LOSS IN BUILDINGS

by

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ABSTRACT

Natural convection heat loss is an issue that needs to be considered when designing the building. Energy lost to the surrounding will increase the overall energy consumed by the building for heating, ventilation and air conditioning (HVAC) system. Understanding the relationship between window opening ratio and heat transfer rate will enable better prediction of heat loss through the window, thus improving the building design. This final dissertation will discuss about the project objective of finding correlation between windows-to-wall sizes ratio and heat transfer coefficient via natural convection, scope, relevancy and also feasibility of the study. Literature review is done on previous studies and journals on natural convection in building. Related information from *American Society of Heating, Refrigerating and Air-Conditioning* (ASHRAE) standard has been included as well. Research methodology is described briefly which mostly involves analyzing the data obtained from the experiments conducted. Experimental setup is based on the real life building compartment with different opening ratio of the window. Engineering theory used to integrate the building model to the heat transfer calculation is described in a detailed manner. Data analysis is done and shown to indicate how the opening ratio effect the correlation linking the main parameters involved in natural convection which are Nusselt and Rayleigh numbers. Discussion on the results obtained as well as errors that might hinders the accuracy of them is also included. Smaller opening ratio leads to a more turbulent flow natural convectional heat transfer is the conclusion that can be derived from this project results.

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INTRODUCTION

1.1 Background of study

Heat loss is a major concern for the engineers when designing the buildings. This is due to the fact that heating, ventilation and air conditioning (HVAC) for the building is the major contributor to its overall energy consumption. Ideally, HVAC works best in a completely insulated enclosed area. This is not possible unfortunately since there will be always some energy lost to the surrounding through designated openings such as doors and window as well as through the walls and the roofs.

Physically, heat will flow from the high temperature to the low temperature region. The energy will be transferred via three means; conduction, convection and radiation. However, most of the heat transfer in the building is in the form of conduction and convection. While heat lost by conduction through the walls and ceilings is significant enough, convectional heat transfer that takes place at the openings of the buildings is what the designer needs to look into in finding the ways to reduce heat loss.

Buildings designed for hot and cold climate area require different measure in terms of energy saving for the HVAC system. Preventing the heat loss is the main objective for the cold climate area where a lot of energy is spent on heating up the building. Meanwhile, reducing the heat gain is the name of the game for building in hot climate area since most of energy consumption is for cooling and air conditioning.

Even though heat loss cannot be stopped, it is still crucial to find the means to understand the mechanism of it and take suitable measures to reduce its effect on the amount of energy used for air conditioning of the building.

1.2 Problem statement

A building is always designed to be efficient in term of energy usage. That means the amount of energy wasted in the form of heat loss must be kept at the lowest possible level. Therefore, the building is properly insulated at the walls and roofs. Unfortunately, buildings have openings in the form of windows, doors, chimney, air ducts and many others. There is significant heat transfer taking place through those openings, especially windows since they are usually opened for natural ventilation.

Ideally, building designers need to consider the effects of different opening on the heat loss or gain when designing the buildings. However, there are no definite rules or guidelines that can be used by those engineers in their designing process to correctly predict the heat loss via natural convection through the windows. It is highly convenient for the designers if there is some correlation available connecting the window opening ratio and the amount of heat lost to the surrounding.

1.3 Objective of the study

The main objective of this project is to investigate the effects of changes in the window opening ratios with the correlation linking main parameters involved in natural convection taking place inside a building which are Nusselt and Rayleigh number. Finding that relationship will help engineers to have better prediction on the amount of heat loss through the windows by considering the windows opening ratio for the building. Different window opening ratio will has different correlation linking Nu and Ra, with Nusselt number providing basis to find heat transfer coefficient of natural convection.

Apart from that specific objective, there are also some side objectives of this study:

- To integrate the theoretical knowledge of heat transfer mechanism into the real life engineering application via experimental modeling
- To provide the foundation for future study in natural convection heat transfer in building

1.4 Scope of study

Building windows have many aspects that can be manipulated to find the desired correlation with the heat loss. In this project, the focus would be on the opening ratio of the window and heat transfer coefficient.

An experimental, scaled down model of a building will be built with some variation in the sizes of the window opening with respect to the wall size. This parameter is designed to see the effect of window sizing to the amount of heat lost to the environment, thus enabling the prediction to be done in an easier way in the future when considering the natural convection heat loss through the window.

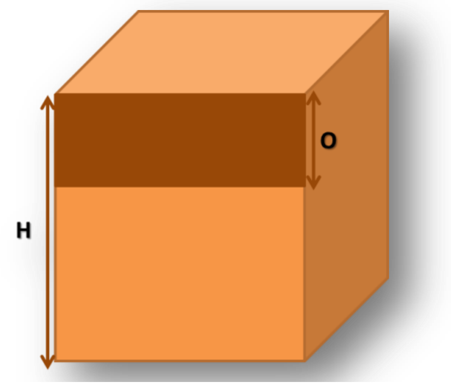


Figure 1: Experimental model for the project

$$Opening\ ratio = \frac{Window\ opening, O}{Wall\ height, H}$$

The study will be conducted to study the heat transfer coefficient variation for different opening ratio for the heat loss. In this case, heat will be supplied into the model and amount of heat loss to the surrounding will be calculated based on temperature rise outside the model.

1.5 Relevancy of study

Results of this study will enable the future designers to predict the heat loss through the windows for different opening ratios. It is important to get the predicted amount of heat loss to be as accurate as possible. This is due to the fact that under designing as well as overdesigning the heating load for the building are highly undesirable.

Past researches and studies mostly focused on the effects of heating surface variation; size and location of the surface and types of convection; natural and forced. There is a study which utilized computational fluid dynamic (CFD) software to simulate the effect of one-sided ventilation to the heat transfer throughout the building. The fact that this project use experimental model to simulate the effects of window opening ratios to the heat transfer coefficient offers some new inputs in helping the designers to predict the heat loss via natural convection better.

1.6 Feasibility of study

This project shall be completed within 28 weeks comprising of two semester of undergraduate study. The first part of the project which mostly involves literature reviewing and planning will be covered in the first semester while the execution of the experimental procedure will be conducted in the second semester. Fourteen weeks would be sufficient to build the model and run the experiment for desired settings and repetition.

In term of project complexity and scope of study, the factor affecting the heat transfer coefficient that will be studied is the opening ratios of the window. Different opening ratios can be achieved by simply changing the dimension of the opening with regard to the wall height. Financial constrain also has been considered in purchasing the materials for the model and the equipment required to collect the data needed. The budget has been controlled to be in compliance with the one provided by management of the Mechanical Engineering department.

2. LITERATURE REVIEW

Heat transfer can be divided into three means; radiation, conduction and convection. According to Cengel and Ghajar (2011), convection is a mechanism of heat transfer that is similar to conduction in term of both needs the medium for the heat movement to take place. The difference is convection requires the fluid motion. The movement of the fluid enhances the heat transfer rate since more energy will be transferred via the bulk fluid motion and no longer fully dependent of conduction alone.

Convection is a complicated mechanism due to the fact that it depends on many variables, mainly fluid properties, geometry and surface roughness of the solid and also the flow type of the fluid as stated by Cengel and Ghajar (2011). Even so, the rate of convectional heat transfer is proportional to the temperature difference, as expressed by Newton's Law of cooling;

$$\dot{Q}_{conv} = hA_s\,(T_s - T_\infty) \qquad \text{- Equation (1)}$$

$\dot{Q}_{conv}$ = rate of heat transfer, W

h = convectional coefficient of heat transfer, $\frac{W}{m^2}.K$

A_s = heat transfer surface area, m^2

T_s = temperature of the surface, ℃

T_∞ = temperature of the fluid sufficiently far from the surface, ℃

The value of *h* however will depend on the variables mentioned previously, rendering the calculation quite complex. Many properties of the fluid such as dynamic viscosity, μ, thermal conductivity, *k*, density, ρ, specific heat capacity, c_p, and fluid velocity, *V* as well as the type of fluid motion whether it is laminar or turbulent.

Based on the work done by Cengel and Ghajar (2011), there are two types of convectional heat transfer; forced and natural. Forced convection happens when there is a fluid flowing and in contact with solid bodies. This can be further divided into two categories; external and internal. External convection involves a fluid moving over a body, transferring heat to or from the body. That is the case for the cooling of the finned surface or the tube banks. In the other hand, internal convection is when the fluid motion is inside an enclosed body such as pipes and ducts, enabling the heat transfer to take place at the inside surface of the hollow body.

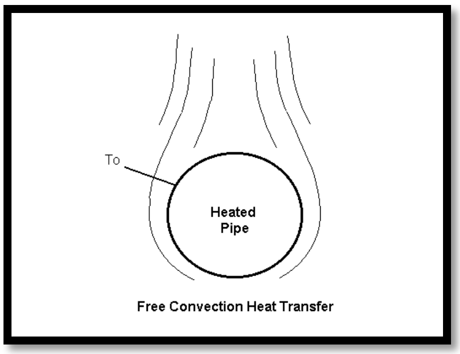


Figure 2: External convection

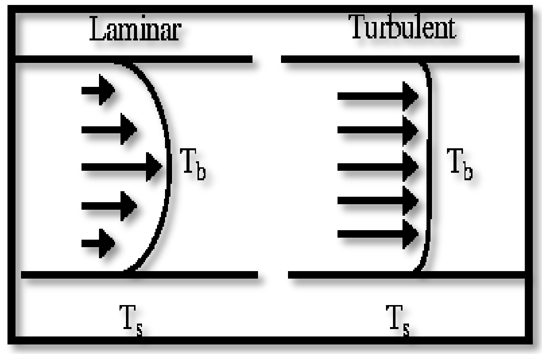


Figure 3: Internal convection

Meanwhile, natural convection is a phenomenon that occurs when the fluid moves around due to the change in its density, altering the buoyancy of the fluid. This change is caused by the change in the temperature of the fluid.

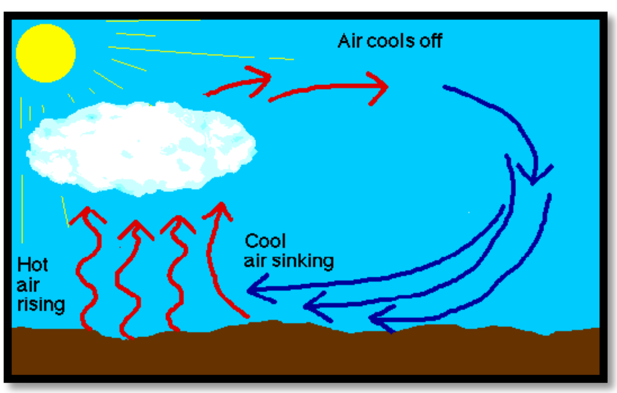


Figure 4: Natural convection phenomenon

When the hot body which in this case is soil is in contact with the air, heat will be transferred from the body to the air, raising the air temperature. The density of that heated air will be reduced, causing it to rise due to the buoyancy force generated by the difference in density. The cool air will replace the hot air and the cycle will goes on until the body and the fluid is in thermal equilibrium. This movement of cold and air will create the convection current which will enhance the heat transfer rate from the soil to the air as mentioned by Cengel and Ghajar (2011).

Another way of explaining convection is as stated by Lienhard IV and Lienhard V (2008), when the cold air moves past a warm body it constantly sweeps away warm air that is attached to the body and replace it with cool air. Unfortunately, the velocity of the fluid in the case of natural convection is quite small compared to the forced convection. According to Cengel (2009) the low velocity of the fluid will lead to a small heat transfer coefficient for natural convection.

In buildings, heat energy will be lost through the openings such as windows and doors. Heat loss through the windows however is more significant as mentioned by Al-Salaymeh, Alshboul and Hassouneh (2012), “Windows are considered as the major source of heat loss in winter and heat gain in summer.” Therefore, it is important to understand the mechanism of heat transfer through the opening of the building so suitable measures can be taken in designing the building to reduce the energy lost to the surrounding. This point is described briefly in the work of Salaymeh, Alshboul and Hassouneh (2012).

There are two aspects of the window that will affect the heat transfer rate through it; the size and the location of the windows. A study conducted by Persson et al. (2006) showed that the different placement of the windows will reduce energy consumption of the building. The result of their study which used the dynamic building simulation tool, DEROB-LTH indicated that the size of the windows has no major effect on the heating demand in the winter, but is significant for the cooling needed in the summer.

Based on the study done by Alloca, Chen and Glicksman (2003) using computational fluid dynamic (CFD) tool to study the trends of single sided-ventilation when the wind speed is changed, it is found that "For buoyancy-driven ventilation, the CFD model was very sensitive to how the boundary conditions are set up. The modeling requires the simulation of both the outdoor and indoor environment.". Therefore, for a more accurate result, experimental model is more preferable since it is easier to control the environment as mentioned in the study above.

Furthermore, due to the lack of study on natural heat loss through windows using experimental method, this project undertaken will be done using experimental model data analysis. A simple mathematical calculation will be developed to analyze the data that would be obtained from the experiments conducted. Small scale mathematical models allow the creation of experimental situations similar to reality through the development of flow parameters. Application of Rayleigh, Grashof and Prandtl numbers which are dimensionless make the model more applicable to real life situation since it will be more practical for scaling up. This is to ensure that the model generates useful results and insights regarding the natural convection process, which, in turn, yield valid conclusions as described in article by Awbi (1991). The result obtained in this project will possibly enhance the available sources for future researchers since convection heat transfer coefficient (CHTC) usually obtained from ASHRAE standard but those coefficients is based on the experiment using small free-edge heated plates and its application to room surfaces is debatable according to Awbi and Hatton (1999). Apart from that, there is no specific correlation published in the ASHRAE standards relating the window opening ratio to the heat loss via natural convection. Thus, this project shall enable better prediction for the natural heat transfer.

In designing the building model, heated surface is better placed on the floor to get a more accurate and reliable results. “When all the present CHTC results are compared with those from other sources, it is found that the floor results compare more favourably.” (Awbi and Hatton, 1999). When the heated surface is designed at the bottom of the model, the effect of air jet velocity which might be caused by standing fans in real-life situation is not that significant, as supported in their further works, by Awbi and Hatton (2000), “The effect of the jet velocity on CHTC is significant in all cases but this is more so when a jet acts over a heated ceiling.”

Parameters that will be having huge significance in this project would be Rayleigh and Nusselt numbers. This is especially true when the model is of rectangular shaped enclosures and heated from below. Previous model done by Calgani, Marsili and Paroncini in 2005 has shown that Rayleigh and Nusselt numbers are the two parameters relating air flow and heat transfer coefficient for natural convection.

3. RESEARCH METHODOLOGY

This project involves significant amount of data analysis works thus the project activities will mostly revolve around experimental analysis of a scaled model of a building compartment.

3.1 Project flow

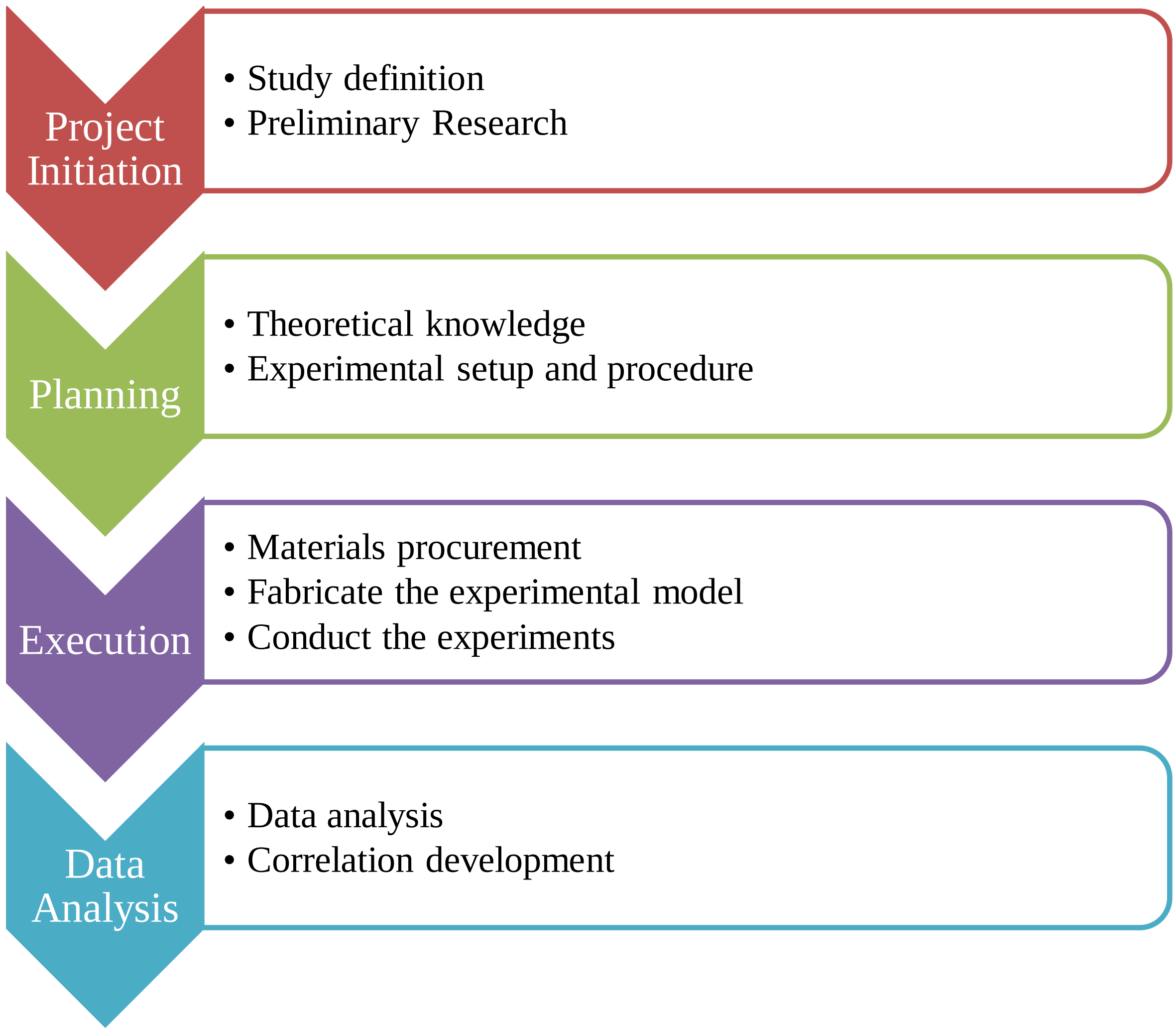


Figure 5: Project flow

3.2 Project Activities

First phase of the project would be *Project Initiation*. This is where project will be started with the problem identification, followed by the definition of the study. That includes the study background, its objectives, scope and feasibility. Preliminary research will be done to obtain theoretical knowledge required for the project. The information and data will be gathered from suitable and credible sources to enhance the integrity of the study results later on. Since this study is about the convection heat loss, the reference would be heat transfer textbooks together with the journals of past research and studies. A literature review will be done to extract the required information from those sources mentioned.

Next phase for the project is *Planning.*The parameters and variables involved in the model design and calculations will be identified and studied. Designing experimental procedure and setup also fall under planning process. A scaled down model of a building will be built with different opening ratios for the window. Once the materials and equipment required for the model design have been identified, the available materials and resources will be located to ensure the expenses done later will be within budget specified by the management. These planning processes will be aided by the lab technicians for heat transfer and fluid dynamics since their knowledge and input would be very valuable

In *Execution* phase, the experimental model will be fabricated according to the design followed by the experimental works as designated procedure and settings. The experiment will be done in three different settings with three times repetition to obtain more accurate results.

Results obtained from the experiment will be collected and analyzed using the mathematical calculation developed earlier. Several assumptions will be made to make the model simpler and solvable. Relationship between the window opening ratios and the heat transfer coefficient for natural convection will be synthesized based on the data analysis done.

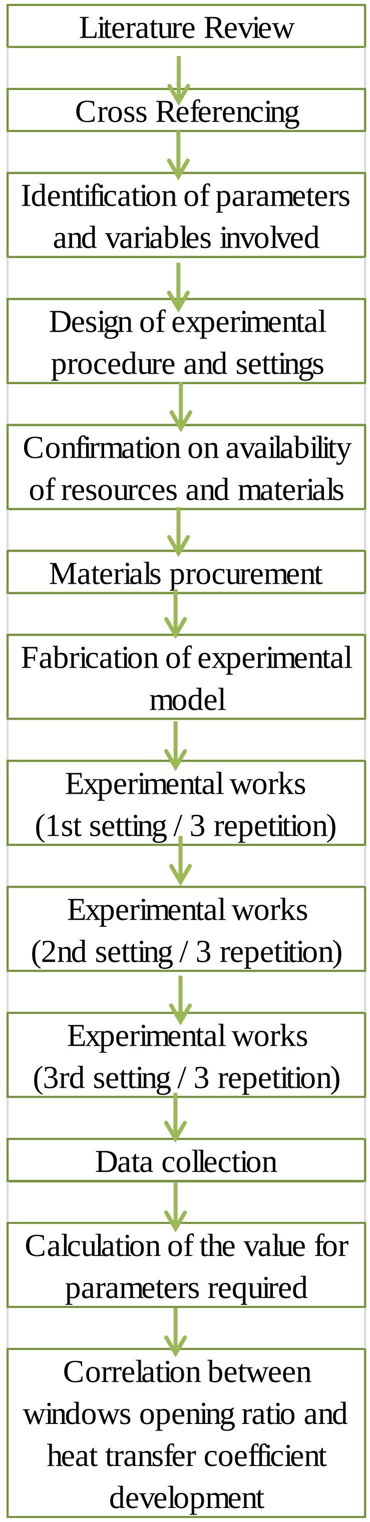


Figure 6: Key Milestones

3.3 Experimental model

This project use a model made from plywood, and with equal internal dimension. The measurements of the model will be 0.5m x 0.5m x 0.5m, with heater plate located at the bottom. The heating plate will consists of aluminium and metal coils properly fitted to ensure even distribution of heat across the heating surface. The experimental model will consist of three walls and a plywood ceiling, In addition, the model will consist of various opening ratios at one side of the wall.

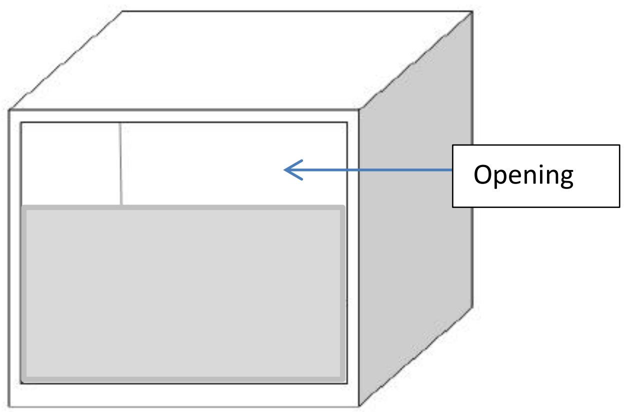


Figure 7: The model that will be used

The opening will facilitate air circulation, while the heated floor serves as the source of heat for the upward motion of air. Floor balancing will aid in achieving the steady state of the heat. It is assumed that the ceiling and walls do not let air and heat out, and that heat transfer will occur only through that one designated opening. The heat supplied will be varied, producing different heat flux by changing the input power to the heating pad. The experiment is expected to last for three to five hours to attain a steady state, after which, it will be possible to record the room temperatures with minimum errors.

The time elapsed for this experiment will be relatively long, so the usage of thermocouples with the data loggers will be implemented.

Figure 8: Data logger connected to thermocouples

Figure 9: Heater used as the heated surface

High resistance wire (usually copper) is spread across the heating pad to ensure even distribution of heat. The thermocouples used will be labeled properly for better identification. The model will be insulated according to the standard insulation of R-2.2 with the thermal resistance of 2.2 °C/W.

Figure 10: Bradford Gold blanket insulation

Meanwhile, thermal resistance of the wall can be found using this formula:

$$R_{wall} = \frac{Wall\ thickness\ (m)}{Thermal\ conductivity\ of\ wall, k \left(\frac{W}{m^2.°C}\right) \times Wall\ area\ (m^2)} \quad \text{------Equation (1)}$$

In order to ensure the safety of the experiment, some equipment will be used throughout the project works. Those include circuit breaker to prevent short circuit and excessive current flowing into the heater pad, transducer to measure power variance and power transferred to the heater pad and lastly power variance to control the power input, thus controlling the heat flux into the pad.

Figure 11: Circuit breaker

Figure 12: Transducer

Figure 13: Power variance

The equipment will borrowed from the Electrical and Electronic Engineering department at Block 22 and 23 of Universiti Teknologi Petronas.

Since the objective of this project is to get the correlation between the opening ratios of the window and heat transfer coefficient, three different ratios will be used throughout the experiment. The experiment will be repeated three times for each setting to obtain more accurate result.

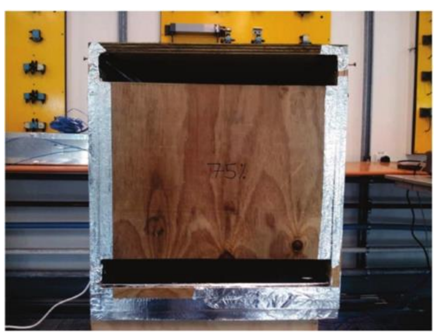


Figure 14: Possible setup of the model

3.4 Model insulation

Insulation of the building model needed to be changed to another material. This is due to the fact that the R-22 insulation is not feasible considering the scale of the model size. As a replacement, polysterene foam has been chosen based on its ability to resist heat transfer as well as its easiness of fabrication. The values computed shows that the heat resistance provided by the polysterene insulation is significantly higher than the resistance of the plywood itself. This is correct in the sense that the insulation must provide higher resistance than the wall material.

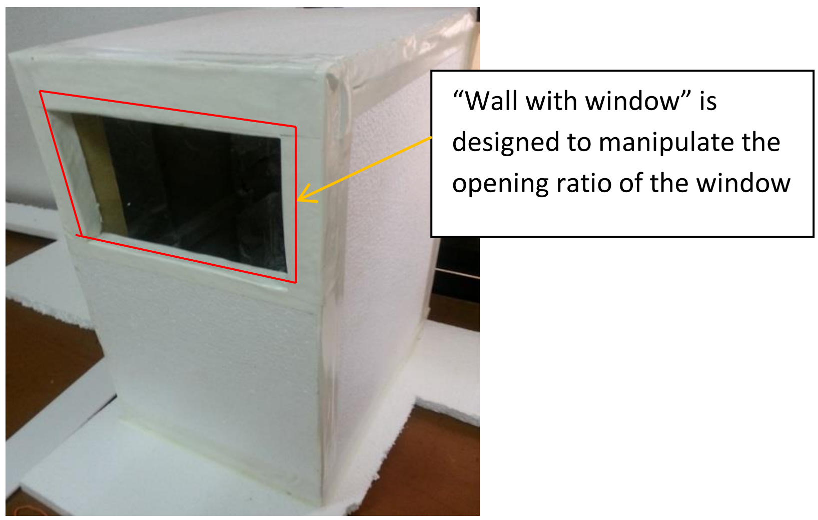


Figure 15: Building model after modification

3.5 Opening ratio

There are three different ratios which will be tested in this project. Those opening ratios would be 0.3, 0.2 and 0.1

Opening Ratio	Window Opening (m)	Wall Height (m)
0.3	0.15	0.5
0.2	0.10	0.5
0.1	0.05	0.5

Table 1: Opening Ratio Setting

3.6 *Theory used*

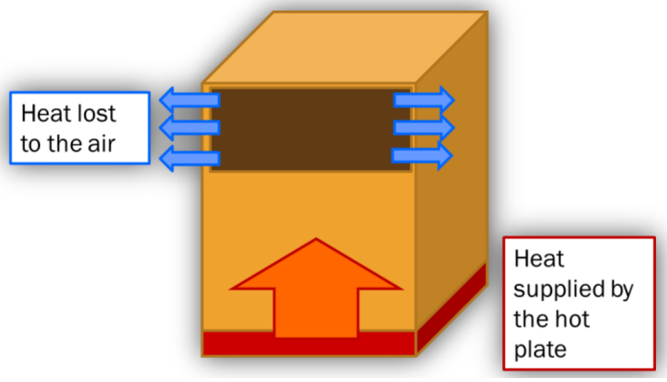


Figure 16: Schematic drawing of the model

$Power\ (P) = Voltage(V) \times Current\ (I)$ ------Equation (2)

Equation 2 shows the heat supplied by the heater used in this experiment. However, not all the heat supplied will be involved in natural convection. Some of them would be lost to the surrounding via the walls and the bottom of the model. The relationship between those mentioned heat is shown in Equation 3 below.

$Power, P = Q_{conv} + Q_{wall} + Q_{bottom}$----------Equation (3)

Heat lost through the wall and bottom plate is due conductional heat transfer and they can be calculated using Equation 4 while equation 5 show the difference between temperature of the heated surface and the temperature of the air.

$Q_{bottom}\ ,\ Q_{wall} = \frac{\Delta T}{R}$ ----------Equation (4)

$\Delta T = T_s - T_\infty$ ----------Equation (5)

Once the heat loss through natural convection has been obtained, the coefficient of convectional heat transfer, h can be determined using Equation 6.

$$Q_{conv} = h\, A_s\, \Delta T \text{ ----------Equation (6)}$$

The correlation between the window opening ratios and heat transfer coefficient for natural convection can be represented by these two parameters; Nusselt number, *Nu* and Rayleigh number, Ra. Nusselt number is the ratio of convective to conductive heat transfer through the gas or fluid while Rayleigh number is the ratio of buoyancy forces and the product of thermal and momentum diffusivities.

Heat transfer coefficient, h can be related to Nusselt number using this formula:

$$Nu = \frac{hL_c}{k} \text{ ------Equation (7)}$$

$h = Convective\ heat\ transfer\ coefficient, \mathrm{W/m^2.K}$

$k = Thermal\ conductivity, \frac{W}{m}.K$

$L_c = Characteristic\ length\ of\ geometry, m$

Based on Figure 16, the geometry of the heating would be horizontal with hot surface facing up. According to Cengel (2011), the characteristic length, L_c is the hot surface area, A_s divided by the perimeter of the surface, p.

$$L_c = \frac{A_s}{p} \text{ ---------Equation (8)}$$

Rayleigh number is the product of Grashof number,*Gr* and Prandtl,*Pr* number.

$$Ra = Gr.\mathrm{Pr} = \frac{g\beta(T_S - T_\infty)\,{L_C}^3}{\vartheta^2}\,Pr \text{ -------------Equation (9)}$$

$g = Gravitional\ acceleration, 9.81\ m/s^2$

$\beta = \ Volume\ expansion\ coefficient, \frac{1}{T_{Film}}\ , K^{-1}\ ,$

$T_s = Temperature\ of\ heater\ surface, ℃$

$T_\infty = Temperature\ of\ the\ air, ℃$

$\vartheta\ =\ Kinematic\ viscosity\ of\ the\ air, m^2/s$

Film temperature (K) is used to determine the air properties such as thermal conductivity, kinematic viscosity and Prandtl Number from the thermodynamics table of air properties and to establish Grashof number, Nusselt Number and Rayleigh number using equations stated above. Bulk temperature can be found using this formula:

$$T_{Film} = \frac{T_S + T_\infty}{2} \text{ ------Equation (10)}$$

Nusselt, *Nu* and Rayleigh number, *Ra* are related by this equation:

$Nu = C\ Ra_L{}^{n}$ ------Equation (11)

- n is dependent on the flow regime (1/4 for laminar flow and 1/3 for turbulent flow)
- C is dependent on the geometry of the heated surface

In this project, for each opening ratio, heat flux supplied will be varied producing different value for Rayleigh and Nusselt number for each value of heat flux. Those values for Nu and Ra will be plotted and the relationship between them will be analyzed to determine C and n for each opening ratio.

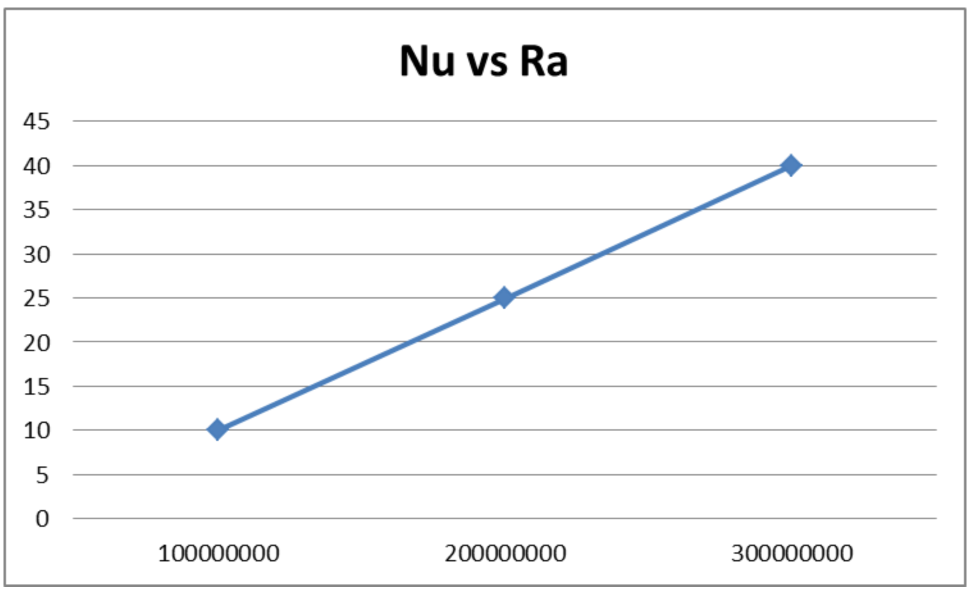


Figure 17: Probable plot of Nusselt against Rayleigh number

3.7 *Gantt chart and key milestones*

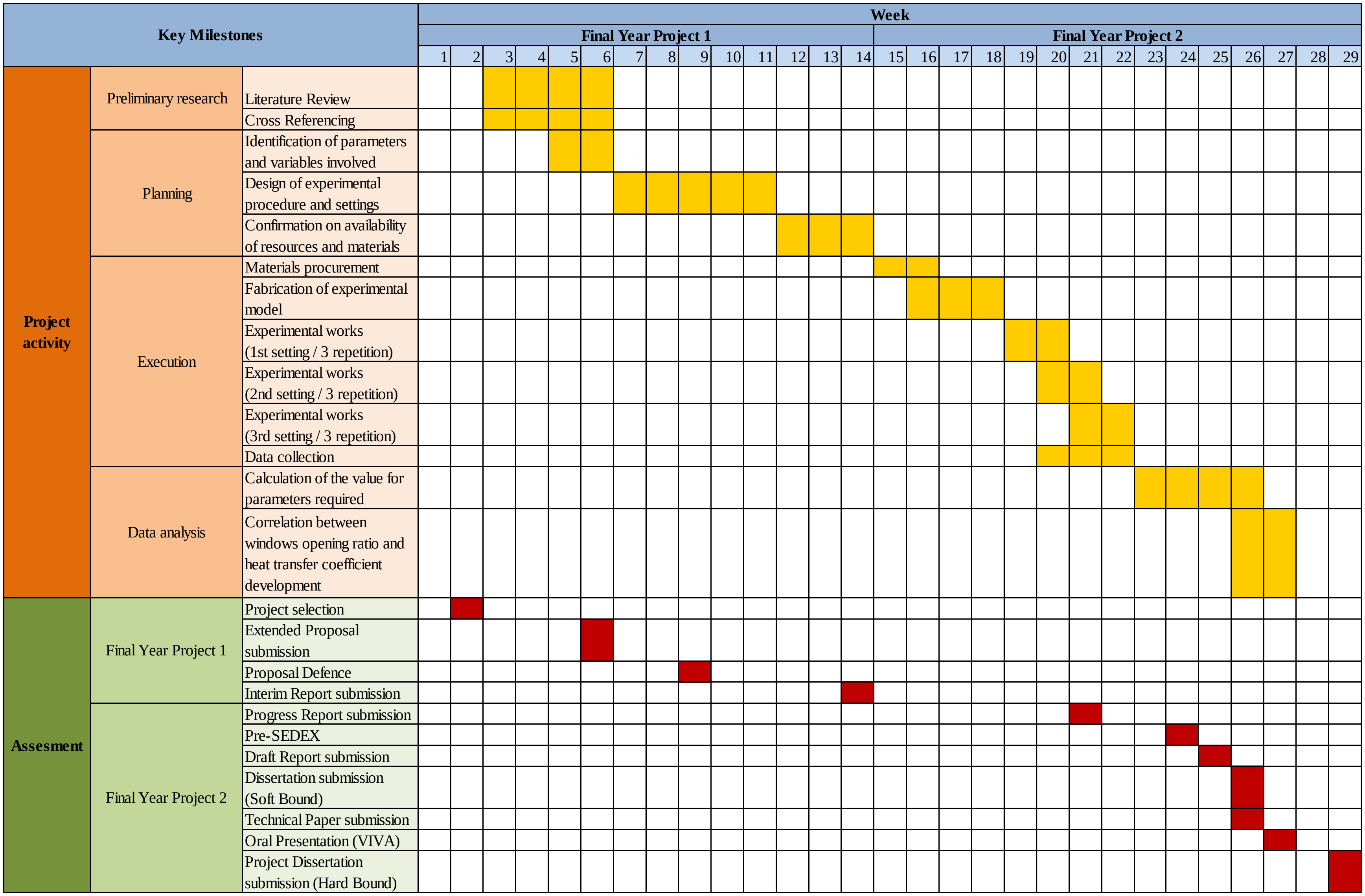
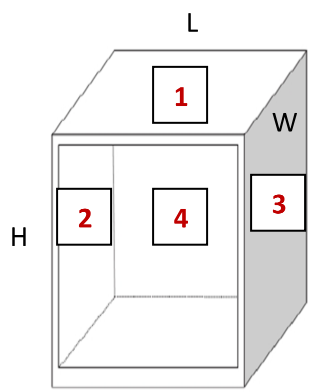


Figure 18:Study Plan of the project

4.0 RESULTS

4.1 Experimental model parameter

- Dimension of the model (together with insulation) :



30 cm (L) x 39 cm (W) x 51 cm (H)

Figure 19: Simple drawing of the model

Calculation for R-value of the wall (Equation 1)

$$R_{wall} = \frac{Wall\ thickness\ (m)}{Thermal\ conductivity\ of\ wall, k\ \left(\frac{W}{m.°C}\right) \times Wall\ area\ (m^2)}$$

Thermal conductivity of plywood = 0.13 W/m.°C

Plywood thickness = 0.005 m

Thermal conductivity of polysterene = 0.03 W/m.°C

Polysterene thickness = 0.025 m

For the walls ,

$$R_{wall\ 1} = R_{plywood} + R_{polysterene}$$

$$= \frac{0.005}{0.13\times(0.30\times0.39)} + \frac{0.025}{0.03\times(0.30\times0.39)}$$

$$= 0.3287 + 7.1225$$

$$= 7.45\ °C/W$$

$$R_{wall\ 2} = R_{plywood} + R_{polysterene}$$

$$= \frac{0.005}{0.13\times(0.51\times0.39)} + \frac{0.025}{0.03\times(0.51\times0.39)}$$

$$= 0.1934 + 4.1897$$

$$= 4.38\ °C/W$$

$$R_{wall\ 3} = R_{plywood} + R_{polysterene}$$

$$= \frac{0.005}{0.13\times(0.51\times0.39)} + \frac{0.025}{0.03\times(0.51\times0.39)}$$

$$= 0.1934 + 4.1897$$

$$= 4.38\ °C/W$$

$R_{wall\ 4}$ = $R_{plywood}$ + $R_{polysterene}$

$$= \frac{0.005}{0.13 \times (0.51 \times 0.30)} + \frac{0.025}{0.03 \times (0.51 \times 0.30)}$$

= 0.2514 + 5.447

= 5.70 °C/W

Meanwhile for the bottom plate,

R_{bottom} = $R_{plywood}$

$$= \frac{0.0075}{0.13 \times (0.39 \times 0.30)}$$

= 0.49 °C.m/W

The heater used has a circular surface. Using Equation 8, the surface area and the characteristic length of geometry for the heated plate can be determined.

$$L_c = \frac{A_s}{p}$$

$$L_c = \frac{\pi r^2}{2\pi r}$$

$$L_c = \frac{\pi r^2}{2\pi r} = \frac{\pi\ (0.09)^2}{2\pi(0.09)} = \frac{0.0254\ m^2}{0.5655\ m} = 0.045m$$

$$A_s = 0.0254\ m^2$$

4.2 Heat transfer analysis

Calculation towards getting the value of Nusselt number for the first run with opening ratio of 0.3:

Run	Voltage_heater (V)	Current_heater (A)	Power supplied (W)	Surface_Temp (°C)	T_s (K)	Air_Temp (°C)	T∞ (K)	T_film (°C)	T_film (K)	ΔT
1	240	0.59	141.6	65.2	338.2	31.8	304.8	48.5	321.5	33.4
2	240	1.22	292.8	97.9	370.9	34.1	307.1	66	339	63.8
3	240	1.48	355.2	112.4	385.4	37.3	310.3	74.85	347.85	75.1
4	240	1.81	434.4	128.3	401.3	38.2	311.2	83.25	356.25	90.1
5	240	2.07	496.8	140.1	413.1	39.4	312.4	89.75	362.75	100.7
6	240	2.43	583.2	155.9	428.9	40.5	313.5	98.2	371.2	115.4
7	240	2.66	638.4	167.9	440.9	42.3	315.3	105.1	378.1	125.6

Table 2: Sample calculation from Excel - Part 1

Eqn 2: $Power\ (P) = Voltage(V) \times Current\ (I)$

= 240V x 0.59A

= 141.6 W

Eqn 10: $T_{Film} = \frac{T_S + T_\infty}{2}$

= 65.2°C + 31.8°C / 2

= 48.5°C or 321.5K

Eqn 5: $\Delta T = T_s - T_\infty$

= 65.2°C - 31.8°C

= 33.4°C

Q_wall	Q_bott	Q_conv	h_conv	K (W/m.°C)	Nu	Pr	β	ϑ x 10^-5
25.5940122	68.1632653	47.84272	56.28313	0.027242	92.97191	0.72319	0.00311	1.7836
48.8891609	130.204082	113.7068	70.02843	0.028518	110.5014	0.7187	0.00295	1.9554
57.548213	153.265306	144.3865	75.5432	0.029159	116.5822	0.716585	0.002875	2.04447
69.0425298	183.877551	181.4799	79.14301	0.029761	119.6691	0.714685	0.002807	2.1308
77.1651803	205.510204	214.1246	83.5499	0.030222	124.4034	0.712581	0.002757	2.231975
88.4296108	235.510204	259.2602	88.27522	0.030822	128.8806	0.711478	0.002694	2.2871
96.2457463	256.326531	285.8277	89.4177	0.031307	128.5271	0.710131	0.002645	2.36108

Table 3: Sample Calculation From Excel - Part 2

Eqn 4:

$$Q_{wall1} = \frac{\Delta T}{R1} = \frac{33.4}{7.45} = 4.483W$$

$$Q_{wall2} = \frac{\Delta T}{R2} = \frac{33.4}{4.38} = 7.626W$$

$$Q_{wall3} = \frac{\Delta T}{R3} = \frac{33.4}{4.38} = 7.626W$$

$$Q_{wall4} = \frac{\Delta T}{R4} = \frac{33.4}{5.70} = 5.860W$$

$$Q_{wall\_total} = Q_{wall1} + Q_{wall2} + Q_{wall3} + Q_{wall4} = 25.595W$$

$$Q_{bottom} = \frac{\Delta T}{Rbott} = \frac{33.4}{0.49} = 68.163W$$

Eqn 3:

$$P = Q_{conv} + Q_{wall} + Q_{bottom}$$

$$Q_{conv} = P - (Q_{wall} + Q_{bottom})$$

$$Q_{conv} = 141.6W - (25.6W + 68.2W) = 47.8W$$

Eqn 6:

$$Q_{conv} = h\, A_s\, \Delta T$$

$$h = \frac{Q_{conv}}{A_{\,s} \Delta T}$$

$$h = \frac{47.8W}{0.0254\, m^2\, (\,33.4℃)} = 56.34Wm^{-2}℃^{-1}$$

The value for thermal conductivity, k is obtained by interpolating the Table Of Properties of air at 1 atm according to specified temperature range. That value is then plugged into Eqn 7 to find Nusselt number.

$$Nu = \frac{hL_c}{k}$$

$$Nu = \frac{56.34\ (0.045)}{0.027242} = 93.0$$

Film temperature is used to find the values for the parameters needed to find the Rayleigh number. Those numbers are Prandtl number, coefficient of volume expansion and kinematic viscosity. That temperature is found using Equation 10.

$$T_{Film} = \frac{T_s + T_\infty}{2}$$

$$T_{Film} = \frac{65.2 + 31.8}{2} = 48.5℃$$

Interpolating the table at this temperature, the values obtained are:

$$\mathrm{Pr} = 0.72319$$

$$\vartheta = 1.7836 \times 10^{-5}\, m^2/s$$

$$\beta = \frac{1}{(48.5 + 273)K} = 0.00311\ K^{-1}$$

The last desired parameter is Rayleigh number. The value for this dimensionless number can be calculated using Equation 9.

$$Ra = Gr.\mathrm{Pr} = \frac{g\beta(T_s - T_\infty)\, {L_c}^3}{\vartheta^2}\, Pr$$

$$Ra = Gr.\mathrm{Pr} = \frac{9.81(0.00311)(65.2 - 31.8)\,(0.045)^3}{(1.7836 \times 10^{-5})^2}\, 0.72319$$

$$Ra = 211092$$

The rest of the run is calculated using Excel spreadsheet to get a more accurate and reliable results. The value of the Nusselt and Rayleigh number is then plotted into a graph and the correlation linking them is determined. The correlation should be in the same form as in Equation 11.

$$Nu = C\, {Ra_L}^n$$

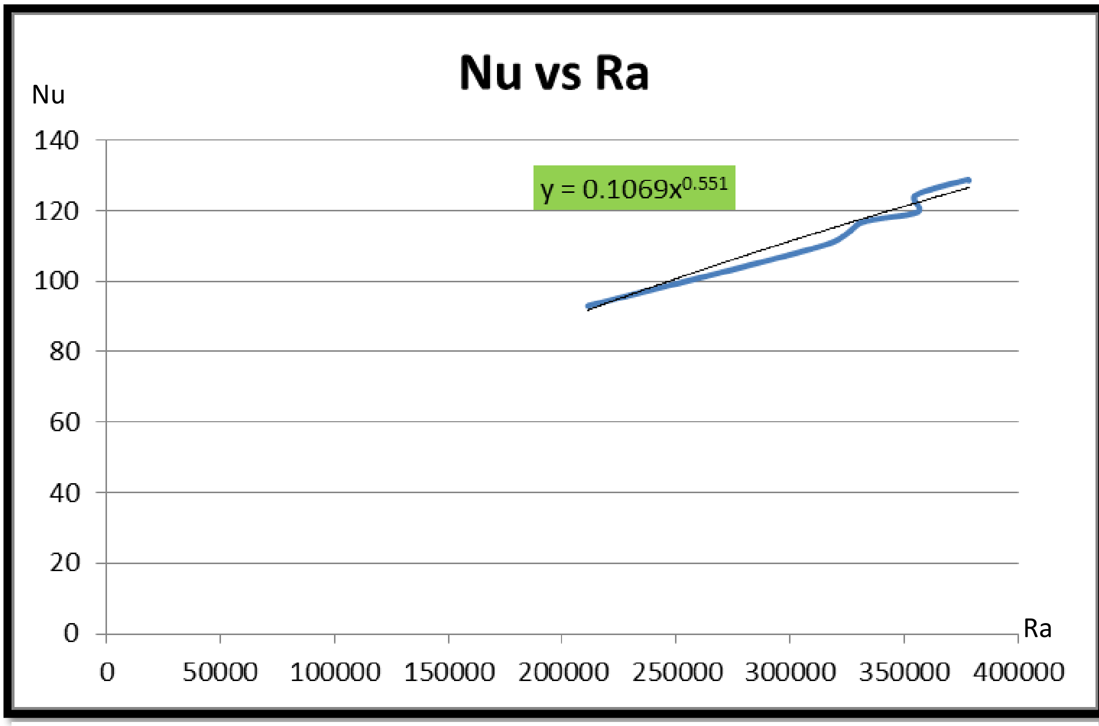


Figure 20: Nusselt against Rayleigh number for Opening Ratio of 0.3

Based on the graph obtained, it is found that the correlation between Nusselt and Rayleigh number when the opening ratio is set at 0.3 is:

$Nu = 0.1069\ Ra_L{}^{0.551}$ ------Equation (12)

where

$C\ = 0.1069$

$n\ = 0.551$

In order to justify the precision of the result and the accuracy of the correlation obtained, a plot of experimental value for Nusselt number against the theoretical value of Nusselt number is developed. The theoretical value is calculated by inserting the values for Rayleigh number into the correlation obtained (Equation 12).

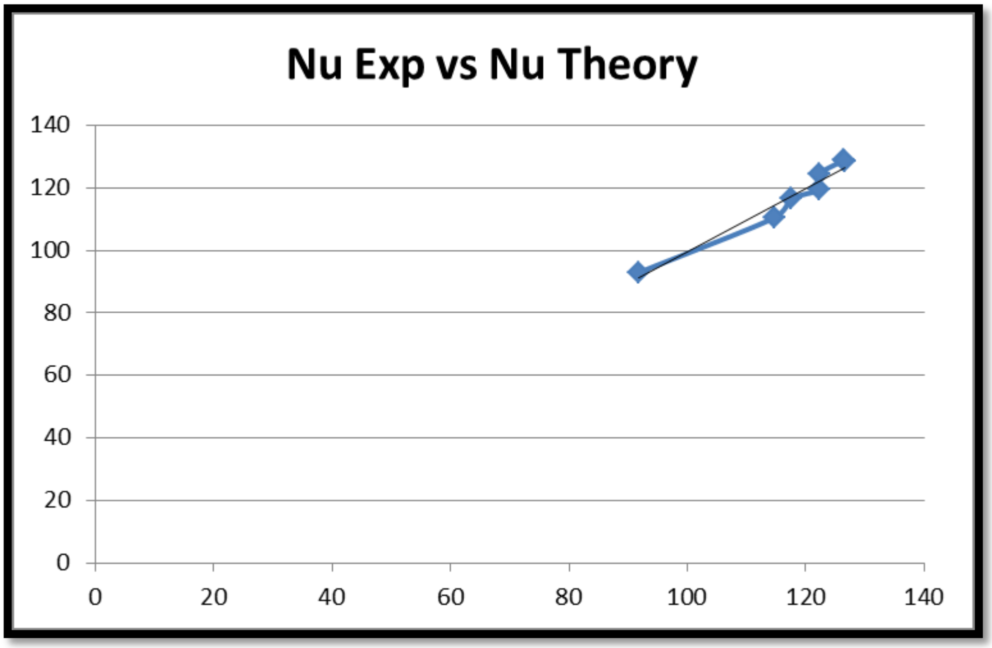


Figure 21: Experimental Nusselt vs Theoretical Nusselt for Opening Ratio of 0.3

The amount of heat transferred via natural convection should be related to the film temperature. The relationship between those two values can be observed in the graph plotted below.

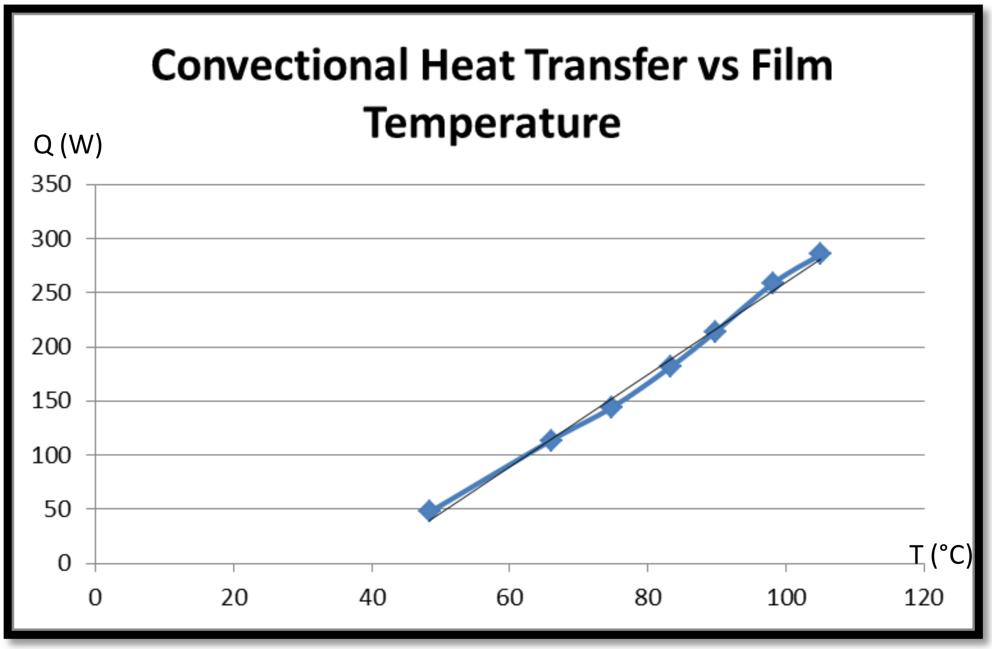


Figure 22: Heat Transferred vs Film Temperature for Opening Ratio of 0.3

Opening Ratio of 0.2

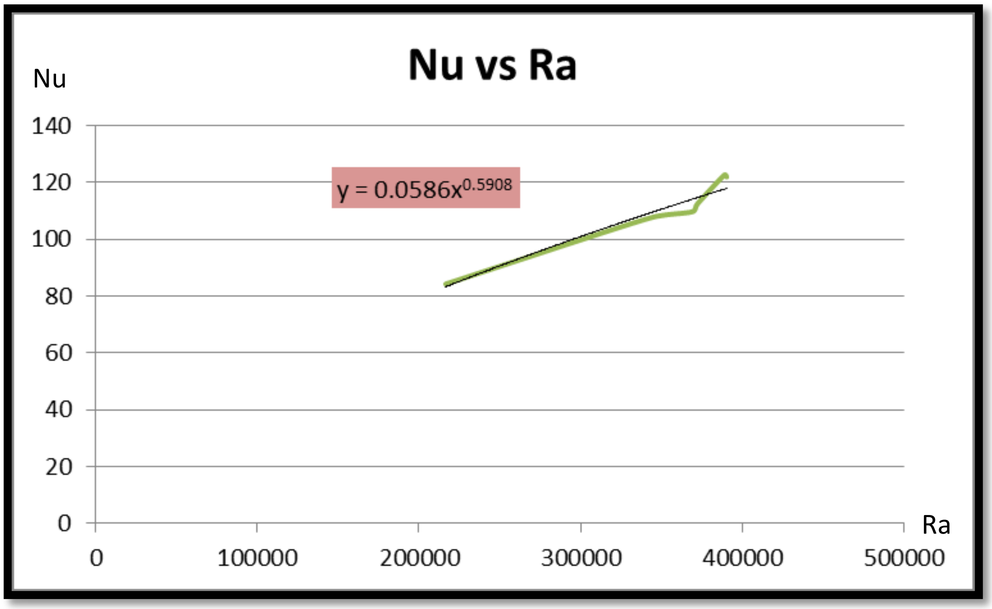


Figure 23: Nusselt vs Rayleigh number for Opening Ratio of 0.2

Based on the graph obtained, it is found that the correlation between Nusselt and Rayleigh number when the opening ratio is set at 0.2 is:

$Nu = 0.0586\, Ra_L{}^{0.5908}$ ------Equation (13)

where

$C \ = 0.0586$

$n \ = 0.5908$

Using Equation 13, the plot of experimental against theoretical values for Nusselt number is obtained

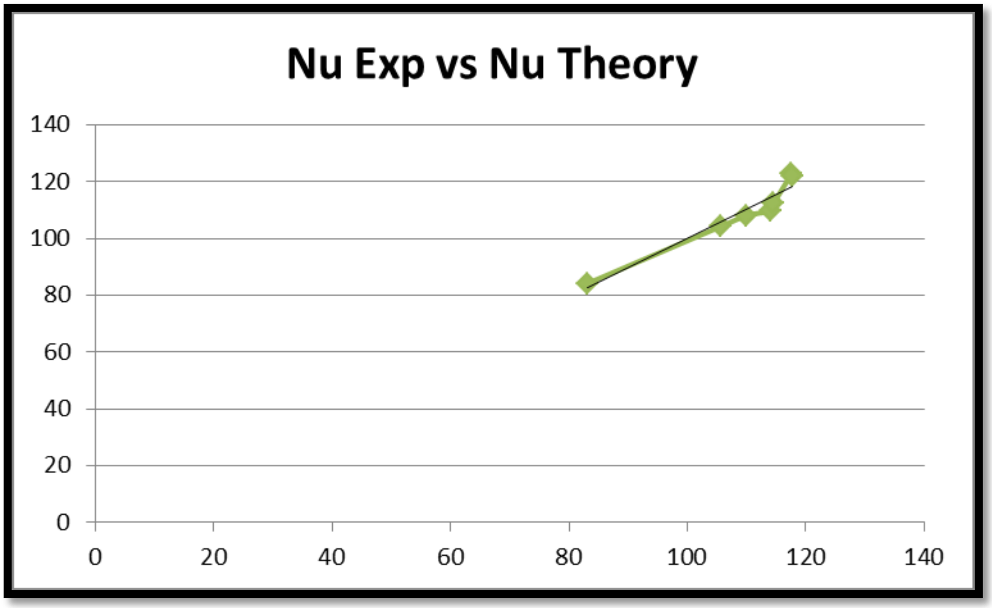


Figure 24: Experimental Nusselt vs Theoretical Nusselt for Opening Ratio of 0.2

The relationship between convectional heat transfer against film temperature can be observed in the graph plotted below.

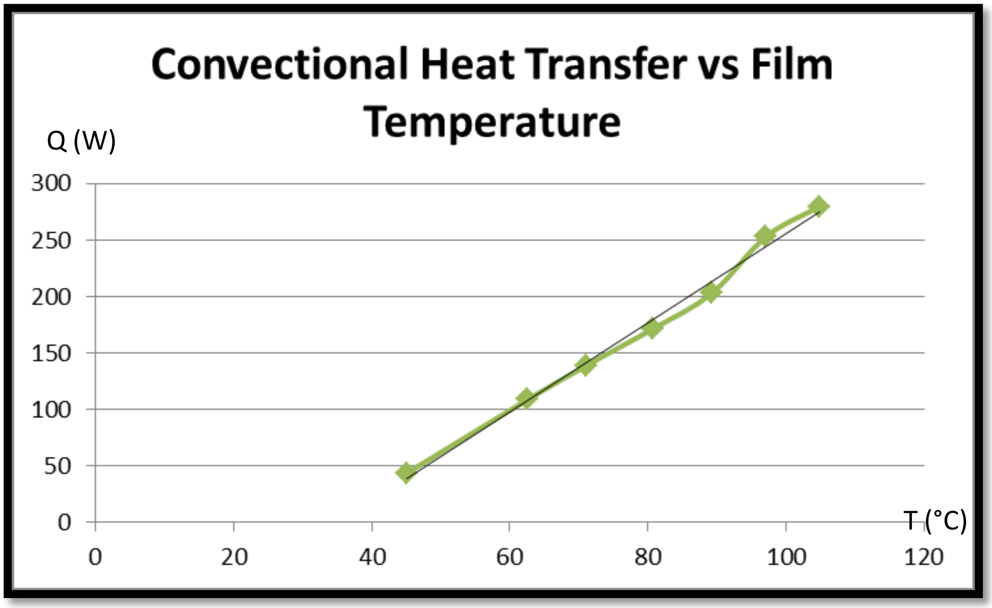


Figure 25: Heat Transferred vs Film Temperature for Opening Ratio of 0.2

Opening Ratio of 0.1

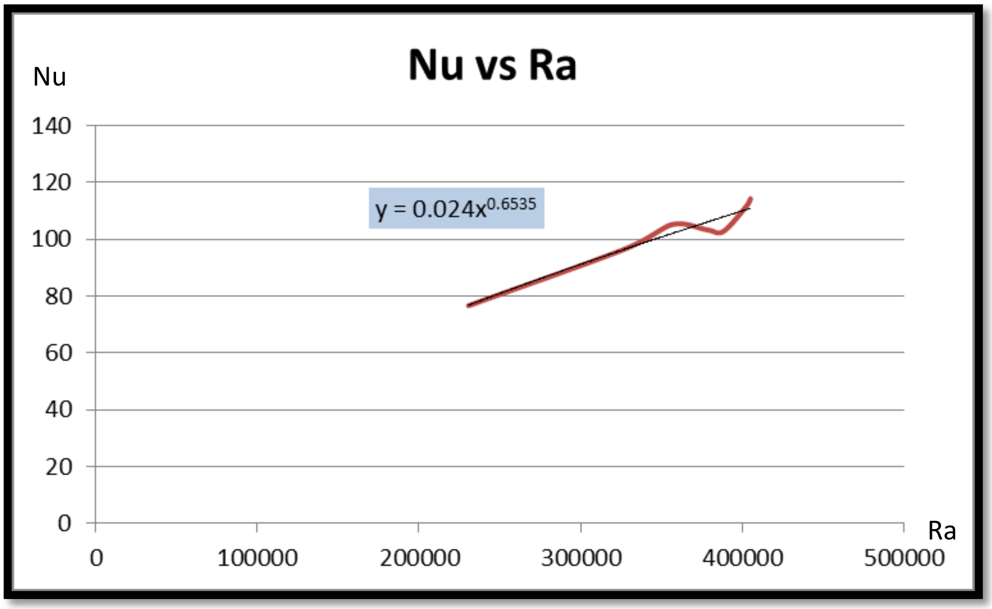


Figure 26: Nusselt vs Rayleigh number for Opening Ratio of 0.1

Based on the graph obtained, it is found that the correlation between Nusselt and Rayleigh number when the opening ratio is set at 0.1 is:

$Nu = 0.024\ Ra_L{}^{0.6535}$ ------Equation (14)

where

$C\ = 0.024$

$n\ = 0.6535$

Using Equation 14, the plot of experimental against theoretical values for Nusselt number is obtained

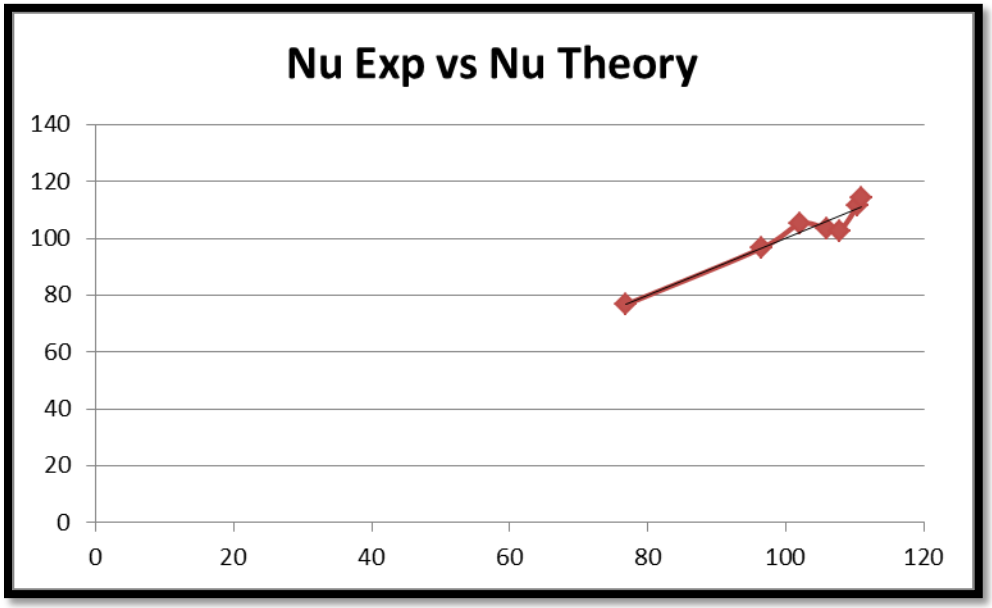


Figure 27: Experimental Nusselt vs Theoretical Nusselt for Opening Ratio of 0.1

The relationship between convectional heat transfer against film temperature can be observed in the graph plotted below.

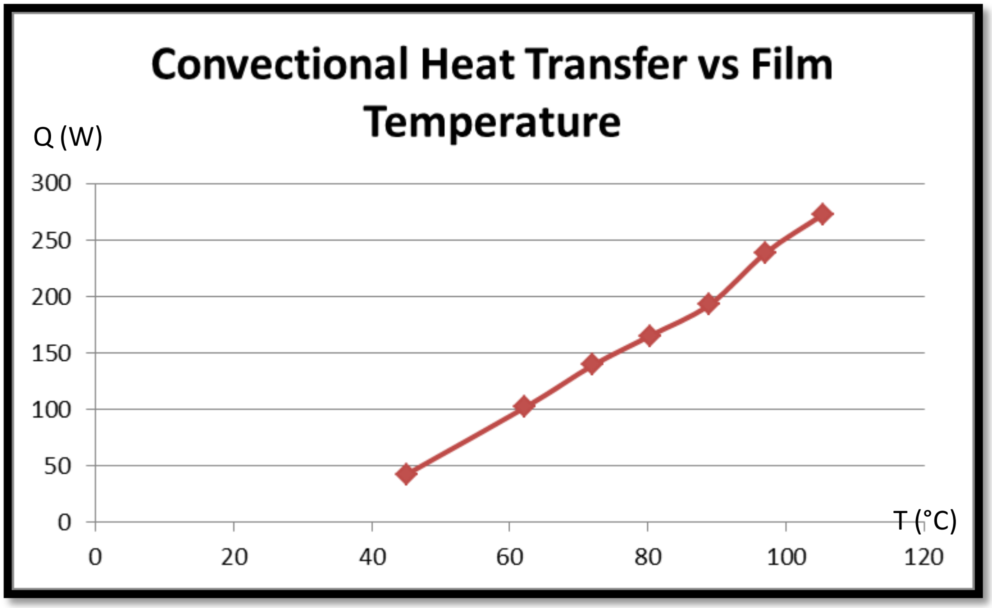


Figure 28: Heat Transferred vs Film Temperature for Opening Ratio of 0.1

For comparison purpose, a plot of Nusselt against Rayleigh number is drawn for all three opening ratios, as shown in the figure below.

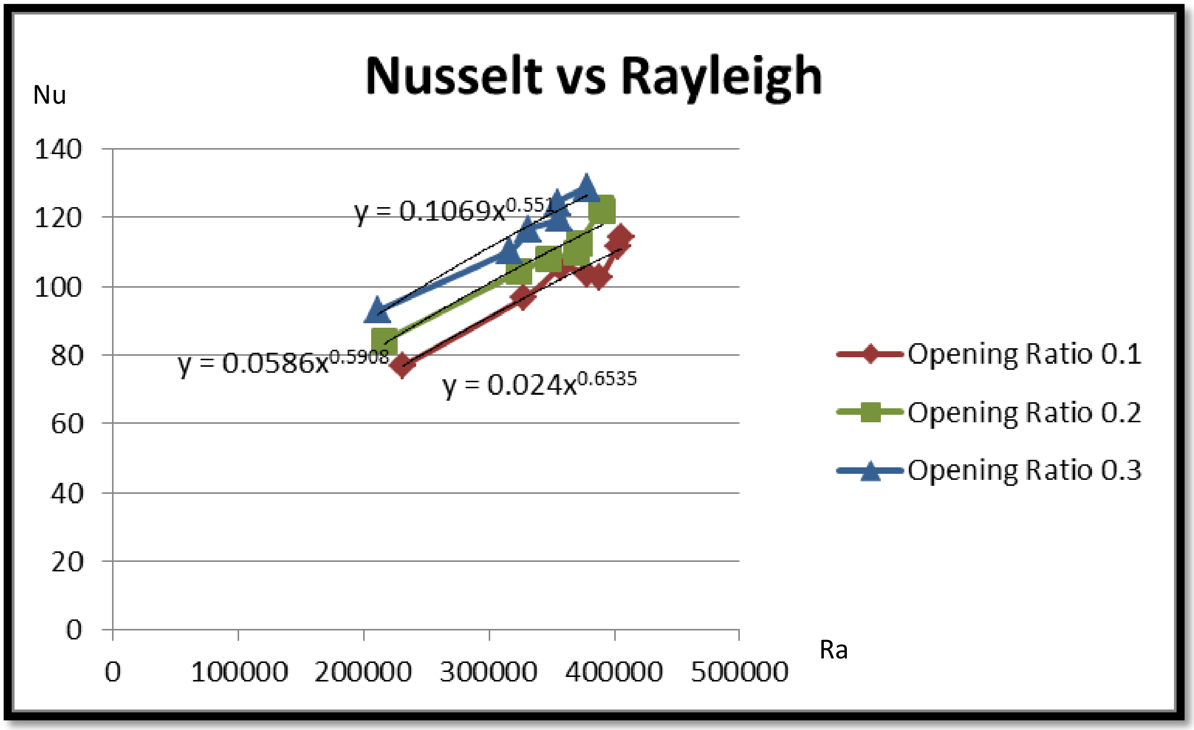


Figure 29: Plot of Nusselt vs Rayleigh For All Opening Ratio

A plot for the theoretical against experimental values for all the opening ratio is drawn to see how the value varies with different opening ratios

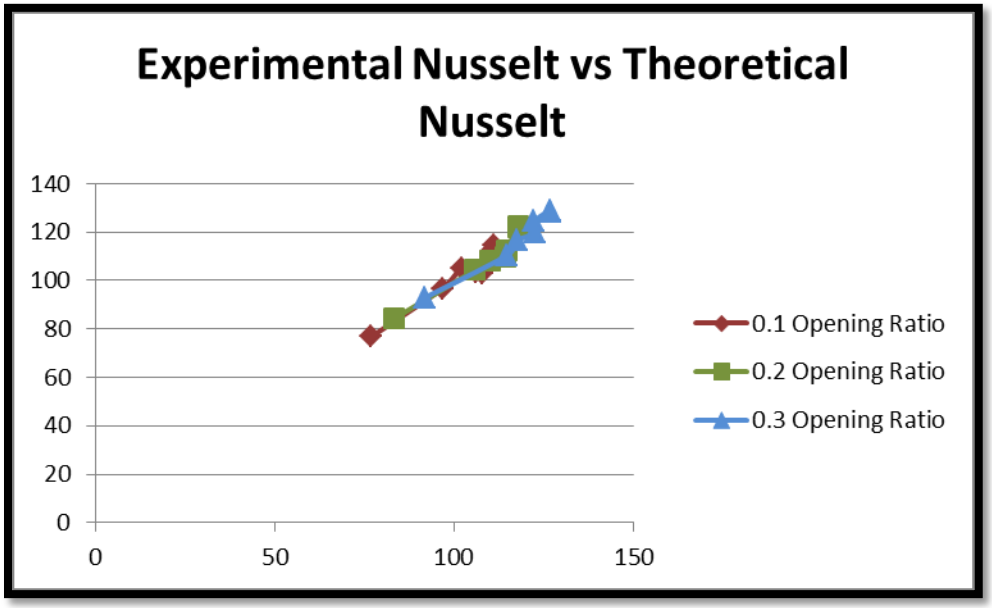


Figure 30: Plot of Experimental Nusselt vs Theoretical Nusselt

4.3 Discussion

4.3.1 Explanation on calculation method

First part of the calculation done shown that the value for thermal resistance of the wall and the bottom plate is quite significantly higher than the initial designated resistance. This is because polysterene foam which has lower thermal conductivity is used to replace the standard wall insulation, thus increasing the value for the thermal resistance of the model.

Meanwhile, calculation for convective heat transfer coefficient shows that the value which is above 50 W/m.K is quite high for natural convection. This high coefficient affected the value of Nusselt number calculated, which in return is also quite high for natural convection phenomenon. This slight deviation is due to some errors which occurred throughout the experiment which will be discussed further in the next section.

On the other hand, Rayleigh number depends much on the properties of the air at the film temperature. Film temperature is used to estimate the air properties by interpolating the value from the thermodynamic table as shown in the Appendix. Evaluating a certain property at film temperature gives a more accurate result since the properties of air both at the heated surface and at room temperature are put into consideration.

4.3.2 Significance of results

4.3.2.1 Correlation between Nu and Ra

Based on the plot obtained, the value for C which resembles the geometry of the surface is quite low since the size of the heated surface is relatively tiny compared to the building model. Meanwhile, for n value which represents flow regime, it is higher than the normal value used to evaluate the turbulent flow heat transfer. It can be deduced that the flow of the hot air in this experiment is highly turbulent.

As the opening ratio become smaller, the value for C decreased while the n value increased. The changes in those values might be due to the fact that the flow of the air became more turbulent as the size of the opening decreased. This pattern can be explained by looking at the changes in value of Grashof number as the opening ratio decrease. This is because Grashof number, Gr describes the relationship between buoyancy and viscosity within the fluid (Cengel,2011). Gr value increased when the opening became smaller, indicating some increment in the buoyancy of the hot air generated at the heated surface.

4.3.2.2 Precision and accuracy of the results

The graph of theoretical value against the experimental value for the Nusselt number showed that there is linear relationship between them for all opening ratios tested. This indicated that the correlation obtained is quite accurate since theoretical Nusselt number is obtained by plugging in the calculated value for Rayleigh number into the correlation linking Nu and Ra.

Apart from that, based on the plots of convective heat transfer and against film temperature shown previously, it is found that they are linearly proportional to each other. Considering the fact that film temperature depends on heated surface temperature and room temperature, those linear plot resembled the direct relationship between convectional heat transfer and temperature of the room since the temperature of the hot plate is almost the same for each experiment.

4.3.3 Errors

Some precautions have been taken to avoid errors from influencing the result of the experiment. Unfortunately, errors during experiments are inevitable thus the result obtained is still affected by flaws and faults during the experiment.

There would be two main sources of error in this experiment; heat loss through leakages that are unaccounted for and the temperature fluctuation throughout the experiment. Even though the building model has been sealed with aluminum foil and silicon glue sealant, there are still some risks of leaking especially at the joints of the walls. There is also some heat loss when the heat is supplied by the heater. This occurred when the heat is trying to overcome the thermal resistance of the heater plate itself which is quite thick in nature.

Furthermore, temperature fluctuation made it quite difficult to obtain an accurate reading for the temperature. This is especially true for the room air temperature since small changes in the external air flow will affect the reading of the thermocouple used.

5. CONCLUSION

5.1 Project conclusion

The objective of the project which to study the effects of changing the window opening ratio to the correlation linking the main parameters involved in natural convection which are Nusselt and Rayleigh number is achieved.

$$Nu = C\, {Ra_L}^n$$

As the opening ratio becomes smaller, the correlation associating those two parameters resembles a more turbulent flow heat transfer. The value of n increases as the window opening size decreases. This is due to the fact that the buoyancy of the air increases which is indicated by the increment in Grashof number value. The other part of the correlation, C is quite small since the size of the heated surface used is relatively small compared to the building model.

Better prediction on heat loss through the windows can be done in the future with the relationship between windows opening ratio and main correlation for the parameters involved in natural convection already determined in this particular project.

5.2 Recommendation for future works

However, considering that natural convection in building is still lacking in term of studies using the scaled-down model, it is suggested that more experiments should be conducted to study the other parameter involved in natural convection phenomenon.

For instance, orientation of the heated surface can be manipulated to see the effect of heating the room from vertical or horizontal angle on the heat transfer coefficient of the natural convection. Apart from that, studying the impact of heating from above or bottom of the room to the heat transfer mechanism of natural convection seems to be an interesting subject.

6. REFERENCE

Al-Salaymeh, A., Alshboul, A., & Hassouneh, K. (2012). Influence of infiltration on the energy losses in residential buildings in Amman. *Sustainable Cities and Society*.

Allocca, C., Chen, Q., & Glicksman, L. R. (2003). Design analysis of single-sided natural ventilation. *Energy and Buildings 35* (pp. 785-795).

Awbi, H. B. (1991). Design considerations for naturally ventilated buildings. *United Kingdom: Elsevier science limited.*

Awbi, H. B., & Hatton, A. (1999). Natural convection from heated room surfaces. *Energy and Buildings 30* (pp. 233-244).

Awbi, H. B., & Hatton, A. (2000). Mixed convection from heated room surfaces. *Energy and Buildings 32* (pp. 153-166).

Calcagni, B., Marsili, F., & Paroncini, M. (2005). Natural convective heat transfer in square enclosures heated from below. *Applied Thermal Engineering 25* (pp. 2522-2531).

Cengel, Y. A. (2009). Introduction to Thermodynamics and Heat Transfer. *McGraw Hill Science Engineering.*

Cengel, Y. A., & Ghajar, A. J. (2011). Natural Convection. *Heat and Mass Transfer:Fundamentals and Applications* (p. 520).

Cengel, Y. A., & Ghajar, A. J. (2011). Fundamentals of Convection. *Heat and Mass Transfer:Fundamentals and Applications* (pp. 374-375).

Cengel, Y. A., & Ghajar, A. J. (2011). External Forced Convection. *Heat and Mass Transfer:Fundamentals and Applications* (p. 418).

Cengel, Y. A., & Ghajar, A. J. (2011). Internal Forced Convection. *Heat and Mass Transfer:Fundamentals and Applications* (p. 466)

Lienhard IV, J. H., & Lienhard V, J. H. (2008). Convective heat transfer. *A Heat Transfer Textbook:3rd Edition.*

Persson, M.L., Roos, A., & Wall, M. (2006). Influence of window size on the energy balance of low energy houses. *Energy and Buildings* (pp.181-188.)

7. APPENDIX

i) Excel Spreadsheet used for Opening Ratio of 0.1

Run	Voltage_heater (V)	Current_heater (A)	Power supplied (W)	Surface_Temp (°C)	T_s (K)	Air_Temp (°C)	T∞ (K)	T_film (°C)	T_film (K)	ΔT
1	240	0.6	144	63.1	336.1	27	300	45.05	318.05	36.1
2	240	1.19	285.6	94.9	367.9	29.5	302.5	62.2	335.2	65.4
3	240	1.52	364.8	112.2	385.2	31.9	304.9	72.05	345.05	80.3
4	240	1.8	432	127.8	400.8	32.8	305.8	80.3	353.3	95
5	240	2.09	501.6	143.9	416.9	34	307	88.95	361.95	109.9
6	240	2.43	583.2	158.4	431.4	35.8	308.8	97.1	370.1	122.6
7	240	2.71	650.4	172.6	445.6	38	311	105.3	378.3	134.6

Q_wall	Q_bott	Q_conv	h_conv	K (W/m.°C)	Nu	Pr	β	ϑ x 10^-5	Gr	Ra	Nu (Theory)
27.6629892	73.6734694	42.66354	46.4364	0.027242	76.70648	0.72319	0.003144	1.7836	318950.5999	230661.9	76.7213
50.1152214	133.469388	102.0154	61.29101	0.028518	96.71419	0.7187	0.002983	1.9554	456151.4559	327836.1	96.5368
61.5329095	163.877551	139.3895	68.20613	0.029159	105.2593	0.716585	0.002898	2.04447	497712.534	356653.3	102.0009
72.79734	193.877551	165.3251	68.3792	0.029761	103.3935	0.714685	0.00283	2.1308	529420.9711	378369.2	106.0179
84.215028	224.285714	193.0993	69.03856	0.030222	102.7965	0.712581	0.002763	2.231975	544850.0668	388249.5	107.819
93.9468829	250.204082	239.049	76.6135	0.030822	111.8547	0.711478	0.002702	2.2871	566118.8633	402781.1	110.4394
103.142336	274.693878	272.5638	79.5668	0.031307	114.3676	0.710131	0.002643	2.36108	570550.2006	405165.4	110.8662

Constant parameters which has been calculated

R_wall (C/W)	
1	7.45
2	4.38
3	4.38
4	5.7

R_bott (C/W)
0.49

L_c	A_s (m2)
0.045	0.0254502

g (m/s2)
9.81

ii) Excel Spreadsheet used for Opening Ratio of 0.2

Run	Voltage_heater (V)	Current_heater (A)	Power supplied (W)	Surface_Temp (°C)	T_s (K)	Air_Temp (°C)	T∞ (K)	T_film (°C)	T_film (K)	ΔT
1	240	0.58	139.2	62.1	335.1	28.2	301.2	45.15	318.15	33.9
2	240	1.21	290.4	94.9	367.9	30.2	303.2	62.55	335.55	64.7
3	240	1.49	357.6	110	383	32.1	305.1	71.05	344.05	77.9
4	240	1.8	432	127.1	400.1	34.3	307.3	80.7	353.7	92.8
5	240	2.08	499.2	142	415	36.5	309.5	89.25	362.25	105.5
6	240	2.44	585.6	156.3	429.3	37.9	310.9	97.1	370.1	118.4
7	240	2.68	643.2	169.5	442.5	40	313	104.75	377.75	129.5

Q_wall	Q_bott	Q_conv	h_conv	K (W/m.°C)	Nu	Pr	β	ϑ x 10^-5	Gr	Ra	Nu (Theory)
25.977156	69.1836735	44.03917	51.04442	0.027242	84.31828	0.72319	0.003143	1.7836	299419.0251	216536.8	83.20124
49.5788199	132.040816	108.7804	66.0625	0.028518	104.2434	0.7187	0.00298	1.9554	450798.3985	323988.8	105.5647
59.6938188	158.979592	138.9266	70.07397	0.029159	108.1418	0.716585	0.002907	2.04447	484240.3332	346999.4	109.9319
71.1115068	189.387755	171.5007	72.61508	0.029761	109.7984	0.714685	0.002827	2.1308	516575.838	369189	114.0324
80.8433617	215.306122	203.0505	75.62414	0.030222	112.6022	0.712581	0.002761	2.231975	522603.0771	372396.8	114.6167
90.7284742	241.632653	253.2389	84.04028	0.030822	122.6977	0.711478	0.002702	2.2871	546724.9055	388982.7	117.6057
99.2342687	264.285714	279.68	84.8595	0.031307	121.9752	0.710131	0.002647	2.36108	549731.2673	390381.2	117.8553

Constant parameters which has been calculated

R_wall (C/W)	
1	7.45
2	4.38
3	4.38
4	5.7

R_bott (C/W)
0.49

L_c	A_s (m2)
0.045	0.0254502

g (m/s2)
9.81

iii) Excel Spreadsheet used for Opening Ratio of 0.3

Run	Voltage_heater (V)	Current_heater (A)	Power supplied (W)	Surface_Temp (°C)	T_s (K)	Air_Temp (°C)	T∞ (K)	T_film (°C)	T_film (K)	ΔT
1	240	0.59	141.6	65.2	338.2	31.8	304.8	48.5	321.5	33.4
2	240	1.22	292.8	97.9	370.9	34.1	307.1	66	339	63.8
3	240	1.48	355.2	112.4	385.4	37.3	310.3	74.85	347.85	75.1
4	240	1.81	434.4	128.3	401.3	38.2	311.2	83.25	356.25	90.1
5	240	2.07	496.8	140.1	413.1	39.4	312.4	89.75	362.75	100.7
6	240	2.43	583.2	155.9	428.9	40.5	313.5	98.2	371.2	115.4
7	240	2.66	638.4	167.9	440.9	42.3	315.3	105.1	378.1	125.6

Q_wall	Q_bott	Q_conv	h_conv	K (W/m.°C)	Nu	Pr	β	ϑ x 10^-5	Gr	Ra	Nu (Theory)
25.5940122	68.1632653	47.84272	56.28313	0.027242	92.97191	0.72319	0.00311	1.7836	291928.9135	211120.1	91.7896
48.8891609	130.204082	113.7068	70.02843	0.028518	110.5014	0.7187	0.00295	1.9554	440003.6788	316230.6	114.6779
57.548213	153.265306	144.3865	75.5432	0.029159	116.5822	0.716585	0.002875	2.04447	461735.2103	330872.5	117.5739
69.0425298	183.877551	181.4799	79.14301	0.029761	119.6691	0.714685	0.002807	2.1308	497956.1385	355881.8	122.3904
77.1651803	205.510204	214.1246	83.5499	0.030222	124.4034	0.712581	0.002757	2.231975	498138.314	354963.6	122.2163
88.4296108	235.510204	259.2602	88.27522	0.030822	128.8806	0.711478	0.002694	2.2871	531292.9856	378003.3	126.5254
96.2457463	256.326531	285.8277	89.4177	0.031307	128.5271	0.710131	0.002645	2.36108	532682.1033	378274.1	126.5753

Constant parameters which has been calculated

R_wall (C/W)	
1	7.45
2	4.38
3	4.38
4	5.7
R_bott (C/W)	
0.49	
L_c	A_s (m2)
0.045	0.0254502
g (m/s2)	
9.81	

iii) Excel Spreadsheet used to combine all the opening ratios in one plot

0.1			0.2			0.3		
Nu	Ra	Nu (Theory)	Nu	Ra	Nu (Theory)	Nu	Ra	Nu (Theory)
76.70647516	230661.8843	76.7212995	84.31828362	216536.84	83.2012414	92.971908	211120.07	91.789599
96.71419003	327836.0514	96.53680069	104.2433778	323988.81	105.564707	110.50142	316230.64	114.67794
105.2592685	356653.3362	102.0009444	108.1418079	346999.36	109.931916	116.58222	330872.53	117.57385
103.3935352	378369.2267	106.017906	109.798445	369189	114.032378	119.66908	355881.78	122.39035
102.796458	388249.5331	107.8189858	112.6022002	372396.76	114.6167	124.40343	354963.65	122.21627
111.8546918	402781.1166	110.4393647	122.6976795	388982.74	117.605723	128.88064	378003.27	126.52541
114.367586	405165.3845	110.8661515	121.9751975	390381.21	117.85534	128.52706	378274.07	126.57535