THERMAL COMFORT OF A TYPICAL OFFICE BUILDING: A PARAMETRIC INVESTIGATION

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DECLARATION

I declare that this project report entitled “Thermal Comfort of a Typical Office Building: A Parametric Investigation” is the result of my own work except as cited in the references

Signature : ……………………………

Author : ……………………………

Date : ……………………………

SUPERVISOR'S DECLARATION

I hereby declare that I have read this project report and in my opinion this report is sufficient in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Honours.

Signature : …………………………

Name of Supervisor : …………………………

Date : …………………………

DEDICATION

To my beloved mother and father

Acknowledgement

Foremost, I would like to acknowledge and express my sincere appreciation to those who has provided me the assistance to complete this project. Without their help and support, this project would not have been successfully completed.

First of all, I would like to take this opportunity to express my deepest appreciation to my supervisor, Dr Tee Boon Tuan, for his professional knowledge and help in assisting me to complete my project. His patience, guidance and advices in guiding me through this project was greatly appreciated.

Besides that, I would like to extend my sincere thanks and appreciation to my university, Universiti Teknikal Malaysia Melaka (UTeM) and Faculty of Mechanical Engineering (FKM) for providing me this opportunity to complete my project.

Last but not least, a million thanks to my family and friends for their continuous mental support and encouragement throughout this project period.

Abstract

Air temperature and air velocity distribution are important parameters that can affect the thermal comfort level of the occupants in the building. The main purpose of this study is to investigate the air temperature and air velocity distribution inside a typical office building. Ground floor and first floor of Mechanical Engineering Faculty (FKM) building are selected for this study. The air temperature and air velocity distribution for both floors were analyzed to determine the suitable air-conditioning setting configuration. Physical measurement and Computational Fluid Dynamics (CFD) are the two main approaches in this study. Physical measurement was conducted by using air velocity meter. CFD simulation was developed by using IESVE software. The air-conditioning temperature setting configuration which is able to maintain the acceptable thermal comfort level is determined. Besides that, the energy consumption for the configuration is also being calculated. The velocity setting configuration for BK 4, 5, 6, 8, 17 and PA are 3 m/s, 4m/s, 3 m/s, 3m/s, 3m/s and 4m/s respectively. The temperature setting configuration for BK 4, 5, 6, 8, 17 and PA are 22 ^{o}C, 21 ^{o}C, 21 ^{o}C, 22 ^{o}C, 21 ^{o}C and 24 ^{o}C respectively. The energy consumption for suitable settings is lower than non-suitable settings for both ground floor and first floor.

Abstrak

Taburan suhu udara dan halaju udara adalah parameter penting yang boleh memberi kesan kepada tahap keselesaan termal penghuni di dalam bangunan. Tujuan utama kajian ini adalah untuk mengkaji taburan suhu udara dan halaju udara di dalam bangunan pejabat yang tipikal. Tingkat aras bawah dan tingkat satu Fakulti Kejuruteraan Mekanikal (FKM) telah dipilih untuk kajian ini. Taburan suhu udara dan halaju udara untuk kedua-dua tingkat dianalisis untuk menentukan konfigurasi pelarasan penghawa dingin yang sesuai. Pengambilan data fizikal dan kaedah Pengkomputeran Dinamik Bendalir (CFD) adalah dua kaedah utama dalam kajian ini. Pengambilan data fizikal akan dijalankan dengan menggunakan meter halaju udara. Simulasi CFD akan dijalankan menggunakan IESVE. Konfigurasi suhu pelarasan penghawa dingin yang boleh mengekalkan tahap keselesaan termal yang sesuai ditentukan. Selain itu, kegunaan tenaga untuk konfigurasi tersebut juga turut dikira. Taburan halaju udara untuk BK 4, 5, 6, 8, 17 dan PA adalah 3 m/s, 4m/s, 3 m/s, 3m/s, 3m/s dan 4m/s. Taburan suhu udara untuk BK 4, 5, 6, 8, 17 dan PA adalah 22 ^oC, 21 ^oC, 21 ^oC, 22 ^oC, 21 ^oC dan 24 ^oC. Kegunaan tenaga untuk konfigurasi yand sesuai adalah lebih rendah daripada konfigurasi yang tidak sesuai untuk kedua-dua tingkat aras bawah dan tingkat satu.

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List of Symbols

Symbols	Description
T_{comf}	Comfort Temperature
^oC	Degree Celcius
ε	Epsilon
α	Griffiths' constant
T_a	Indoor Air Temperature
V_a	Indoor Air Velocity
T_g	Indoor Globe Temperature
K	Kelvin
m	Meter
m^2	Meter Square
T_o	Outdoor Temperature
Pa	Pascal
%	Percentage
s	Second
Σ	Summation
T	Temperature
Θ	Theta
W	Watt

List of Abbreviations

Abbreviations	Description
ADPI	Air Diffusion Performance Index
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers
BK	Bilik Kuliah
CATIA	Computer Aided Three dimensional Interactive Application
CFD	Computational Fluid Dynamics
CFM	Cubic Feet per Minutes
CL	Mechanical Cooling
Clo	Clothing Unit
CO	Carbon Monoxide
CO_2	Carbon Dioxide
FKM	Fakulti Kejuruteraan Mekanikal
FPM	Feet per Minutes
FR	Free-Running
H	Height
HVAC	Heating, Ventilation, and Air-Conditioning
IEQ	Indoor Environmental Quality
IESVE	Integrated Environmental Solution- Virtual Environment
IGSES	Interdisciplinary Graduate School of Engineering Science
KU	Kyushu University
L	Length
MET	Metabolic Rate

MRT	Mean Radiant Temperature
MS	Malaysian Standard
PA	Postgraduate Area
PMV	Predicted Mean Vote
PPD	Predicted Percentage of Dissatisfied
PSM	Projek Sarjana Muda
RH	Relative Humidity
RPM	Revolutions per Minutes
TNB	Tenaga National Berhad
UFAD	Under-Floor Air Distribution
UiTM	Universiti Teknologi MARA
UTeM	Universiti Teknikal Malaysia Melaka
UTM	Universiti Teknologi Malaysia
W	Width

CHAPTER 1

INTRODUCTION

With the rising of awareness about maintaining good indoor environmental conditions, there are more and more people started to emphasize the importance of indoor thermal comfort. Indoor thermal comfort in general can affects occupants' comfort and this will further influence their emotional condition as well as productivity (Youssef et. al, 2015)

As defined by ASHRAE Standard 55-2010, thermal comfort is the occupants' satisfaction towards surrounding thermal conditions. Thermal comfort is used to determine whether the surrounding condition is in the occupants' acceptable range or not. Air temperature, metabolic rate, clothing insulation, radiant temperature, air velocity and humidity are the six primary factors that must be stated when defining thermal comfort level.

As the location of Malaysia is near to the equator and expose to sun. The country has a warm and humid climates condition throughout a year. Due to the high average air temperature, most Malaysia citizens choose to install a mechanical cooling system such as air-conditioning in order to achieve suitable indoor thermal comfort level (Zr & Mochtar, 2013).

As more and more buildings in Malaysia rely on mechanical cooling system, the average energy usage goes up as well. According to Hassan et. al (2014), 48% out of the total electricity generated in Malaysia consumes by buildings. Meanwhile, more than 50% of the electricity is used by mechanical cooling systems to provide suitable thermal comfort level to occupants. Hence, it can be seen that there is a huge potential in energy saving by optimizing the usage of mechanical cooling systems such as HVAC (Heating, Ventilation and Air Conditioning).

With the advancement of computer technology, Computational Fluid Dynamics (CFD) has becomes a popular tool to visualize and predict the motions of gas or liquids. CFD already exists since early 20th century which enables users to simulate complex fluid motion by using a computer. As explained by Chen (2009), about 70% of the researchers who study ventilation performance choose to use CFD in simulating the flow and temperature distribution in a building. Most of the indoor environment designs are getting more complicated and CFD comes in handy to predict the indoor thermal comfort.

CFD is widely used in everyday life to analyse the fluid flow such as meteorological phenomena, environmental hazards, HVAC industries, complex flows in furnaces, combustion in automobile engines and etc. Researchers do not need to perform difficult experiments hands on. They can obtained the simulation results as long as the physical data are available. This can save a lot of time as well as cost in performing experiments (Kuzmin, n.d.).

By optimizing the application of CFD, this project is aimed to simulate the current thermal comfort condition for university building.

1.1 Problem Statement

The main purpose of this study is to investigate the thermal comfort of UTeM's Faculty of Mechanical Engineering (FKM). A suitable and stable thermal comfort of the faculties is extremely important to students, lecturers, and staffs.

Humans are very sensitive to surrounding temperatures. According to the survey done by Izzat (2016), a majority of FKM's students feel that the surrounding temperatures is hotter and most of the temperatures level are beyond the acceptable range. The fluctuated temperature and air velocity distribution perhaps is due to the inefficient air-conditioning system of the building.

Air temperature and air velocity play an important role in thermal comfort. Optimizing air temperature and air velocity not only will provide thermal comfort, it will save the energy cost as well and reduce carbon dioxide emission. As mentioned by Tenaga Nasional Berhad (TNB) (n.d.), 24^{o}C – 26^{o}C is the optimum temperature to give the best thermal comfort in a building.

Hence, this project intends to find the answer for the following question:

a) How does the current setting for the air-conditioning system of the building affect the temperature and air velocity distribution?
b) Does the current thermal comfort level of the building within the acceptable standard?

c) What are the suitable air-conditioning system configuration in order to improve the current thermal comfort level and energy consumption?

1.2 Objective

i. To illustrate the changes of thermal comfort level under different circumstances and parameters by using the available building simulation software.

ii. To investigate the change in the energy demand of the case building in the different scenarios.

1.3 Scope

The scope of this study is to evaluate on the thermal comfort in a typical office building:

i. The study will be conducted in FKM building.

ii. The main parameters involved in the thermal comfort simulation are temperature and air velocity.

iii. The simulation will be conducted by using available CFD software.

1.4 The Importance of the Work

This simulation obtained in this investigation will give a good overview in term of temperature and air velocity distribution of the building, whether comply with the current standard or not. The information will be a good input for the building engineer or manager to find potential measures to improve the situation for the non-comply case. The modelling of the thermal comfort condition by using CFD software can provide another alternative in analysing indoor condition besides conducting actual measurement. The simulation result if applicable can be a benchmark for future building design at pre-design stage. The study also focuses on how different temperature and air velocity affect the outcome of thermal comfort as well as the changes in the energy demand for the different scenarios. Through the study, a typical office building can know the optimum indoor temperature and air velocity required for the building to provide suitable thermal comfort level can be determined. Thus, building energy consumption can be minimized through correct setting of air-conditioning temperature.

CHAPTER 2

THEORY

2.1 Thermal Comfort

ASHRAE Standard 55 (2010) defined thermal comfort as:

"*that condition of mind that expresses satisfaction with the thermal environment and is assessed by subjective evaluation*".

As explained by Setaih et. al (2014), thermal comfort level is an important factor to be considered either at the indoor or outdoor environment as thermal comfort level can bring positive and negative impact on the occupants using the places. Occupants will feel comfortable when the rate of heat loss and heat gain are balanced. In order to maintain the suitable thermal comfort level, heating, ventilation and air conditioning system (HVAC) of a building needs to be optimized in order to maintain the comfort level of occupants and decrease in the energy demand of a building.

Thermal comfort can be analysed by 2 categories which are personal factor and environment factor. Personal factor is subdivided into occupants' metabolic rate and clothing insulation while environment factor is subdivided into air temperature, air speed, radiant temperature and relative humidity.

2.2 Personal Factor

Personal factor is determined by the occupants themselves as it is the characteristics of the occupants. Personal factor includes occupants' metabolic rate and clothing insulation.

2.2.1 Metabolic Rate

Every human being has a different metabolic rate which affected by the activity level and environmental conditions (Toftum, 2005). The definition of metabolic rate given by ASHRAE 55-2010 is the rate in which the chemical energy in the body is transformed into heat and mechanical work through metabolic activities and its expression is in terms of unit area of the total body surface. In general, metabolic rate is the heat produced by our bodies when we are doing some physical activities.

The metabolism process is still going on even when people are resting. According to ASHRAE 55-2010, a person who is at rest and quietly seated tend to produce a MET value of 1, which equivalent to $60W/m^2$. Evaporation of sweat will occur when a person's metabolic rate is above 1.0 met. Some example of activities that have metabolic rate above 1.0 met listed by ASHRAE 55-2010 are standing (1.2 met), walking with 0.9m/s (2.0 met), cooking (1.6-2.0 met), house cleaning (2.0-3.4 met), playing basketball (5.0-7.6 met) and etc.

Szokolay and Steven (2010) stated that metabolic rate will influence by the eating habit and body shape of a person and therefore determine their thermal comfort level. They explained that a person with rounded body shape is harder to dissipate heat compare to tall and skinny person as the dissipation of heat is depend on the body surface area. Hence, a tall and skinny person is easier to maintain a comfortable thermal comfort level.

2.2.2 Clothing Insulation

Clothing insulation is one of the personal factor to determine thermal comfort level of people. People wear clothes because clothes act as a protection to protect people from being too cold. If clothing does not provide enough insulation, people may suffer cold injuries. Clothes insulation can provide a suitable thermal comfort level as it helps to trap the heat released by the body so that a person will feel warm. People may add layers of clothing to increase the clothing insulation if they feel cold and remove a layer of clothing if they feel warm. However, clothing insulation also will lead to overheating and influence the thermal comfort level of a person if there are too many layers of clothing (Auliciems and Szokolay, 1997). Clothing insulation is determined by clo unit in which 1 clo is equivalent to 0.155 m^2K/W.

2.3 Environmental Factors

Air temperature, mean radiant temperature, air speed and relative humidity are thermal comfort level determination factors which can be categorized as environmental factors. Those factors are not able be controlled by people. By this, the mechanical cooling system plays an important role to control the environmental factors to reach a desire thermal comfort level.

2.3.1 Air Temperature

As defined by ASHRAE 55-2010, the average temperature of the air surrounding an occupant is known as the air temperature. Air temperature is measured by dry bulb temperature because the temperature measured by dry bulb thermometer is not affected by the air moisture. Air temperature is the most important factor as it will determine the convection heat dissipation from the body. When combined with suitable air speed, mean radiant temperature and relative humidity, a wide range of temperature can provide a suitable thermal comfort level. In case one of the condition varies, thermal comfort level can be maintained through adjusting dry bulb temperature (Bradshaw, 2006).

The author further explained occupants will feel discomfort when the temperature difference is large between head level and feet level. The temperature difference is known as vertical air temperature difference. The recommended vertical air temperature difference should not exceed 3°C and the temperature of the floor should be between 18°C and 29°C.

2.3.2 Mean Radiant Temperature (MRT)

Mean radiant temperature is defined as the temperature of a uniform, black enclosure that exchanges the same amount of thermal radiation with the occupants as the actual enclosure (ASHRAE 55-2010). In general, the amount of radiant heat transfer from a surface is known as mean radiant temperature and it is a function of the surface temperature. Tredre (1964) explained that the MRT at a certain point in a room will change depending on the closeness of that point to the boundary surface which radiating with high or low intensity. All of the factors such as radiation received from occupants, furniture and equipment will affect the MRT of the point.

The equation given by Bradshaw (2006) is as below:

$$MRT = \frac{\Sigma \mathrm{T}\theta}{360} = \frac{\mathrm{T}1\theta 1 \ + \ \mathrm{T}2\theta 2 \ + \cdots . + \mathrm{Tn}\theta \mathrm{n}}{360}$$

Where,

T = surface temperature

θ = angle of surface exposure (degrees)

2.3.3 Air Speed

The definition of air speed given by ASHRAE 55-2010 is the average speed of air to which the body is exposed, with respect to location and time. It is the rate of air movement without considering the direction. Free and forced convection as well as the occupants' movements will result in the air movement (Bradshaw, 2006). There is a direct impact between air movement and the feeling and thermal preference of occupants inside a building (Oluwafemi and Adebamowo, 2010). Higher air velocity is allowed when the air and radiant temperature is higher (as shown in Figure 2.1). Hence, cooling system designed by HVAC engineers aimed to move energy and ventilated air through buildings. Air velocity is often expressed in feet per minutes (FPM).

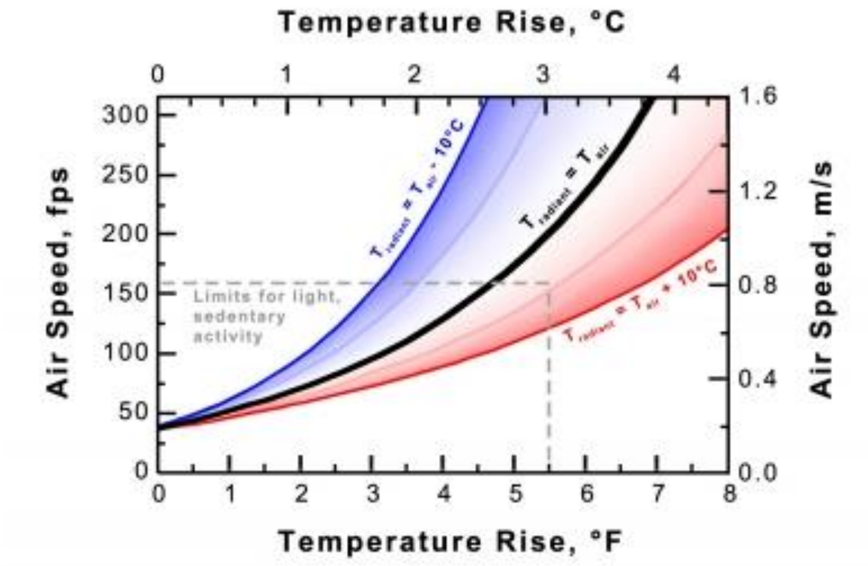


Figure 2.1: Air speed required to offset increased air and radiant temperature

(ASHRAE 55-2010)

2.3.4 Relative Humidity

Humidity is the moisture content in the air (ASHRAE 55-2010). Absolute density is the water vapor density per unit volume of air and it is expressed in pounds (water vapor) per cubic foot (dry air). Meanwhile, humidity ratio is given by the weight of water vapor in the air over the weight of dry air per unit and humidity ratio has no unit. The amount of moisture that air can hold is depending on the temperature. The higher the temperature of the air, the amount of moisture that air can hold is higher too.

The dryness of air is represented by percentage humidity. Percentage humidity is obtained through the degree of saturation in which the degree of saturation is the amount of water vapor in the air relative to the maximum amount of water vapor that air can hold without causing condensation. Relative humidity (RH) is the ratio of the actual vapor pressure of the air-vapor mixture to the pressure of saturated water vapor at the same dry-bulb temperature, multiply by 100.

Compare to temperature variation, human tend to have a higher tolerance for the humidity variation. However, humidity control is still as important as air temperature control to maintain a suitable thermal comfort level. The recommended relative humidity in a building is between 55% - 70% (Chen, 2009). Problems such as condensation on cold surface and hinders human heat loss through sweating or respiration are caused by high humidity. On the other hand, even though low humidity can increase the rate of heat dissipation from the skin, low humidity still causes problems such as dry throat which can cause discomfort to occupants in buildings (Bradshaw, 2006).

2.4 Predicted Mean Vote (PMV) and Predicted Percentage of Dissatisfied (PPD)

Predicted mean vote (PMV) model is used to determine the thermal comfort level in a building. The PMV model uses heat balance principles to relate all of the six parameters for the thermal comfort. PMV model consists of seven points thermal sensation scale which ranges from -3 to +3. -3 represent cold, +3 represents hot and 0 represents the neutral thermal sensation.

+3	Hot
+2	Warm
+1	Slightly warm
0	Neutral
-1	Slightly cool
-2	Cool
-3	Cold

Predicted Percentage of Dissatisfied (PPD) is related to PMV as shown in Figure 2.2. PPD is used to predict how many people are dissatisfied in a given thermal environment. According to ASHRAE 55-2010, the acceptable range for PMV range is within -0.5 to +0.5 and <10 for PPD value.

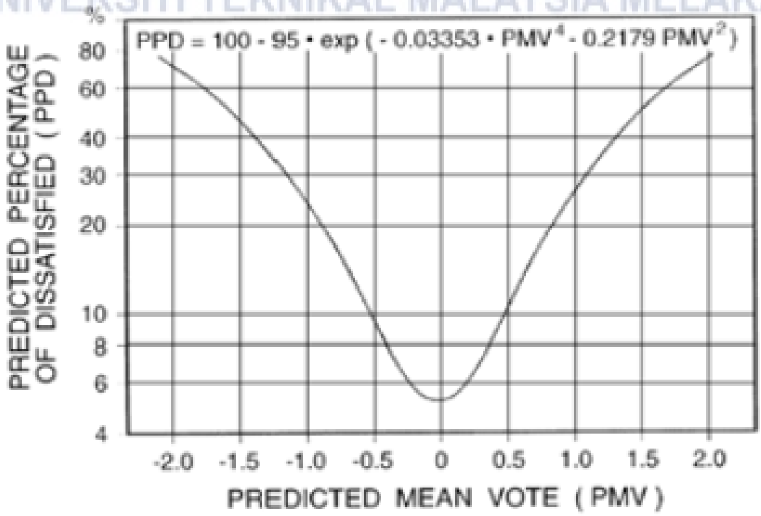


Figure 2.2: Predicted percentage dissatisfied (PPD) as a function of predicted mean vote (PMV) (ASHRAE 55-2010)

2.5 Air Conditioning System

Air conditioning system is known as heating, ventilation and air conditioning system (HVAC). HVAC is a type of mechanical cooling system and it plays an important role in maintaining a suitable thermal comfort level in a building. The main function of HVAC is to heat or cool, humidify or dehumidify, clean and purify, improve air circulation and maintain indoor environment in acceptable condition (Wang & Lavan, 1999). There are three basic types of air conditioners namely window air conditioner, split air conditioner and packaged air conditioner.

2.5.1 Window Air Conditioner

Window air conditioner consists of only one single unit in which all of the components such as compressor, evaporator coil, refrigerant tube and condenser coil are assembled together as a unit in a casing. Users are required to mount the window air conditioner in a slot made in the wall of a room. Figure 2.3 shows the window air conditioner parts (Vandervort, 2017)

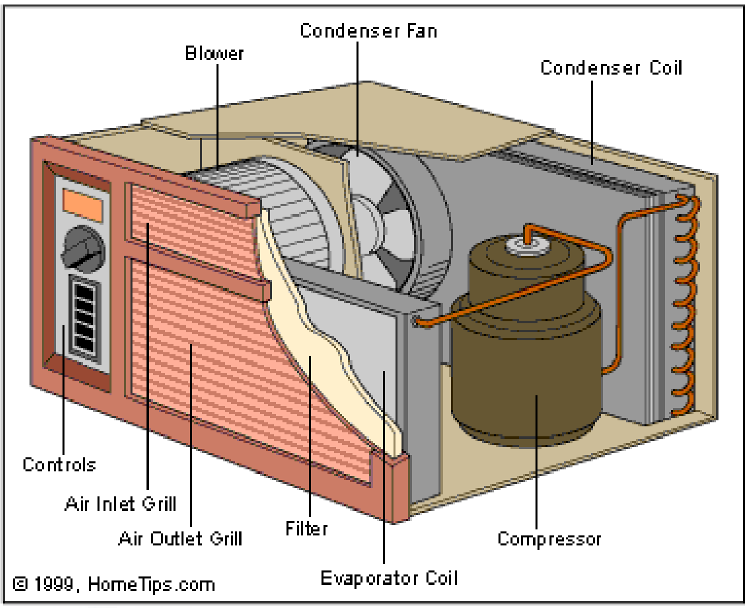


Figure 2.3: Window Air Conditioner Parts (Vandervort, 2017)

2.5.2 Split Air Conditioner System

Split air conditioner system is also known as room air conditioner. The term is given as split air conditioner is aimed to supply cool air to a room rather than a building. Split air conditioner consists of 2 separate parts that are indoor air handler (consist of evaporator and cooling fan) and outdoor condensing unit (consist of compressor, condenser and expansion valve). Indoor air handler is installed in a room while the condensing unit is installed outside a room. Both of the parts are connected with electrical wires and tubing. The function of tubing is to transport air between the 2 separated parts. Split air conditioner is cheaper in price and it is affordable. The maintenance fee is low as well (Wang & Lavan, 1999).

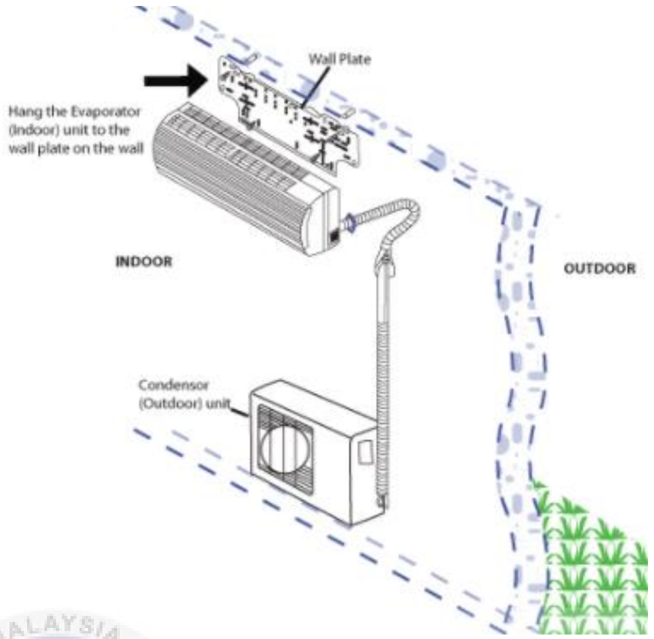


Figure 2.4: Split Air Conditioner System (Wang & Lavan, 1999)

2.5.3 Packaged Air Conditioner System

Packaged air conditioner is able to provide cool air to big buildings. The difference between split air conditioner and packaged air conditioner is that the components of packaged air conditioner such as evaporator, condenser and compressor are located together in one casing. Packaged air conditioner is ideal to cool a whole building rather than install split air conditioner in every room. The cooled air produced flows through the ducts to various rooms. The cost to install a packaged air conditioner is much cheaper than install multiple split air conditioner in every room. Packaged air conditioner capacity is available from about 5 tons up to 100 tons. A standard packaged air conditioner is able to supply air flow rate of 400 CFM (cubic feet per minutes). (Bhatia, n.d.)

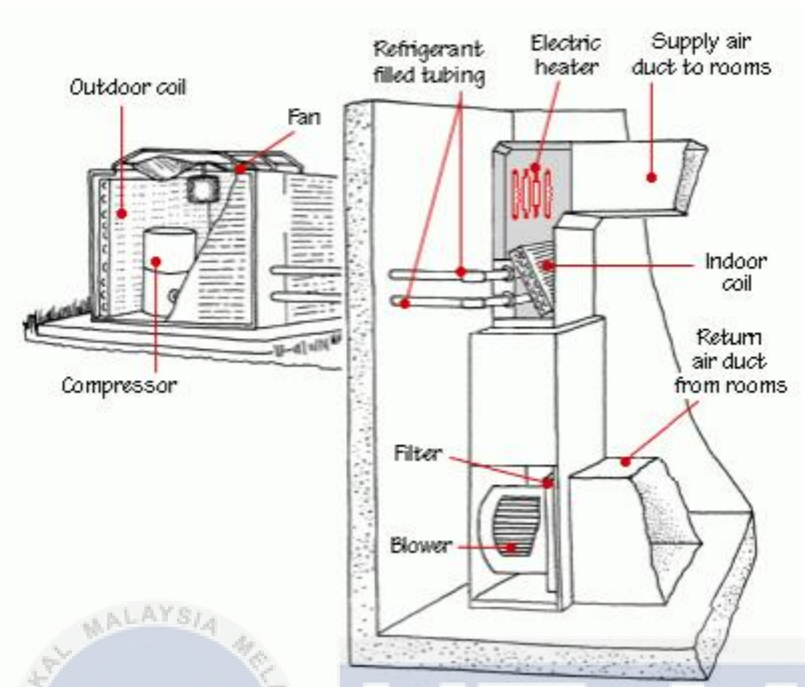


Figure 2.5: Packaged Air Conditioner System (Bhatia, n.d.)

2.6 ASHRAE Standard 55-2010

The American Society of Heating, Refrigerating and Air- Conditioning Engineers (ASHRAE) establishes standards and guidelines which is known as ASHRAE Standard 55. ASHRAE Standard 55 is important to HVAC and building services engineers as they can follow the guidelines provided to design and maintain the building environment. ASHRAE published the standard to express the minimum values and acceptable performance within the HVAC industry. The guidelines act as a benchmark for performance criteria to guide the engineers in measuring or testing ("The Importance of ASHRAE Standards", n.d). The recommended range for air temperature, air velocity and relative humidity by ASHRAE Standard 55- 2010 are describe in Table 2.1.

Table 2.1: Recommended range by ASHRAE Standard 55-2010

Parameters	Recommended range
Air temperature	20 ºC – 28ºC
Air velocity	0.1 m/s – 1.2 m/s
Relative humidity	40% – 60%

2.7 Malaysian Standard MS 1525-2014

Malaysian Standard is introduced by the Department of Standards Malaysia (STANDARDS MALAYSIA).The main purpose for STANDARDS MALAYSIA to create MS 1525 is to encourage the energy reduction design and construction, provide the criteria and minimum standard for energy efficiency in buildings, provide guidance for energy efficiency design, and encourage the application of renewable energy in new and existing buildings (MS 1525-2014). The recommended range for air temperature, air velocity and relative humidity by MS 1525-2014 are describe in Table 2.2.

Table 2.2: Recommended range by MS 1525-2014

Parameters	Recommended range
Air temperature	24 ºC – 26ºC
Air velocity	0.15 m/s – 0.50 m/s
Relative humidity	50% – 70%

2.8 CFD

As defined by Kuzmin (n.d.), Computational Fluid Dynamics (CFD) provides a qualitative and quantitative prediction of fluid flows by using mathematical modelling (partial different equations) and numerical methods (discretization and solution techniques). The flow of CFD can be described as Figure 2.6 (Zuo, n.d.)

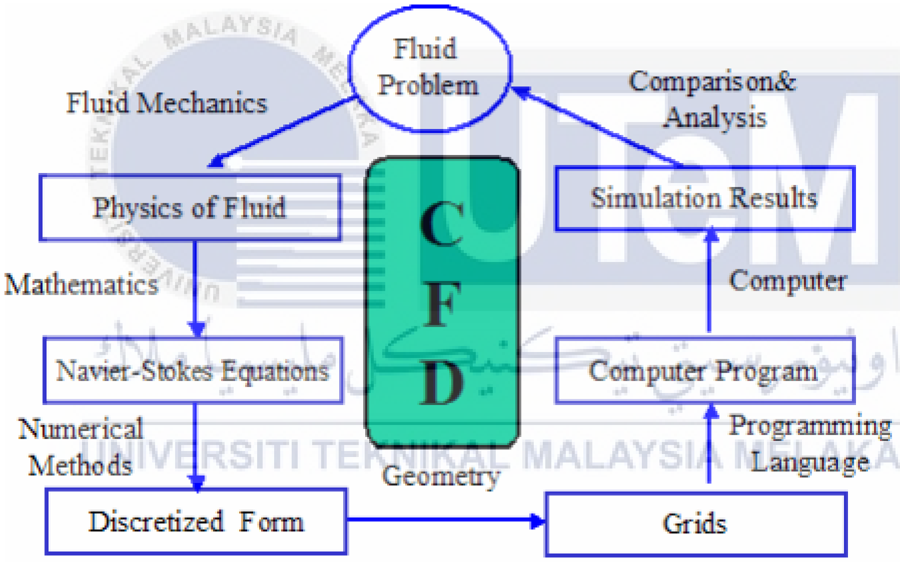


Figure 2.6: Process of Computational Fluid Dynamics (Zuo, n.d.)

Whenever we face a fluid problem, we should firstly know the physical properties of the fluid through Fluid Mechanics. The physical properties are then described in Navier- Stokes Equations. Navier-Strokes Equations are a set of nonlinear partial differential equations that can be used to describe the fluid flows. In order to solve the equation through the help of a computer, we have to convert the equation to discretized form by using numerical methods.

Finite Difference, Finite Element and Finite Volume methods are the example of numerical methods. Next, we have to create geometry and divide the problems into smaller parts known as grid. Computer program is needed to solve the problems. At the end of the steps, we will get our desired simulation results and compare it with the experiments and real problem (Zuo n.d.).

There are some flow patterns that are too complex and expensive to be performed by experimental method. In the case, CFD comes in handy to help researchers to analyse the flow patterns. However, the simulation results obtained from CFD are never 100% reliable because there may be too much guessing and lead to imprecision on the input data and the mathematical model of the problem at hand may be inadequate.

The main differences between experimental and simulations method are described as below by Kuzmin (n.d.).

Table 2.3: Differences between experimental and simulations method

Experiments	Simulations
Quantitative description of flow phenomena using measurements a) for one quantity at a time b) at a limited number of points and time instants c) for a laboratory- scale model d) for a limited range of problems and operating conditions	Quantitative prediction of flow phenomena using CFD software a) for all desired quantities b) with high resolution in space and time c) for the actual flow domain d) for virtually any problem and realistic operating conditions

Error sources: measurement errors, flow disturbances by the probes	Error sources: modelling, discretization, iteration, implementation

2.8.1 Working Flow Structure of CFD

The working flow structure of CFD shown in Figure 2.7 was described by Maysam et. al. (2012). The processes involves geometry, physics, mesh, solve, reports and post processing. The working flow structure of CFD are briefly described as follow.

1. Geometry – Determine the domain size and what kind of shapes needed to resolve the grometry.
2. Physics – Determine the flow conditios and fluid properties.
3. Mesh – Well designed meshes are important to resolve important flow features.
4. Solve – Setup suitable numerical parameters and choose suitable solvers.
5. Reports – All required results can be viewed in reports.
6. Post- processing – Analysis and visualization the simulation results

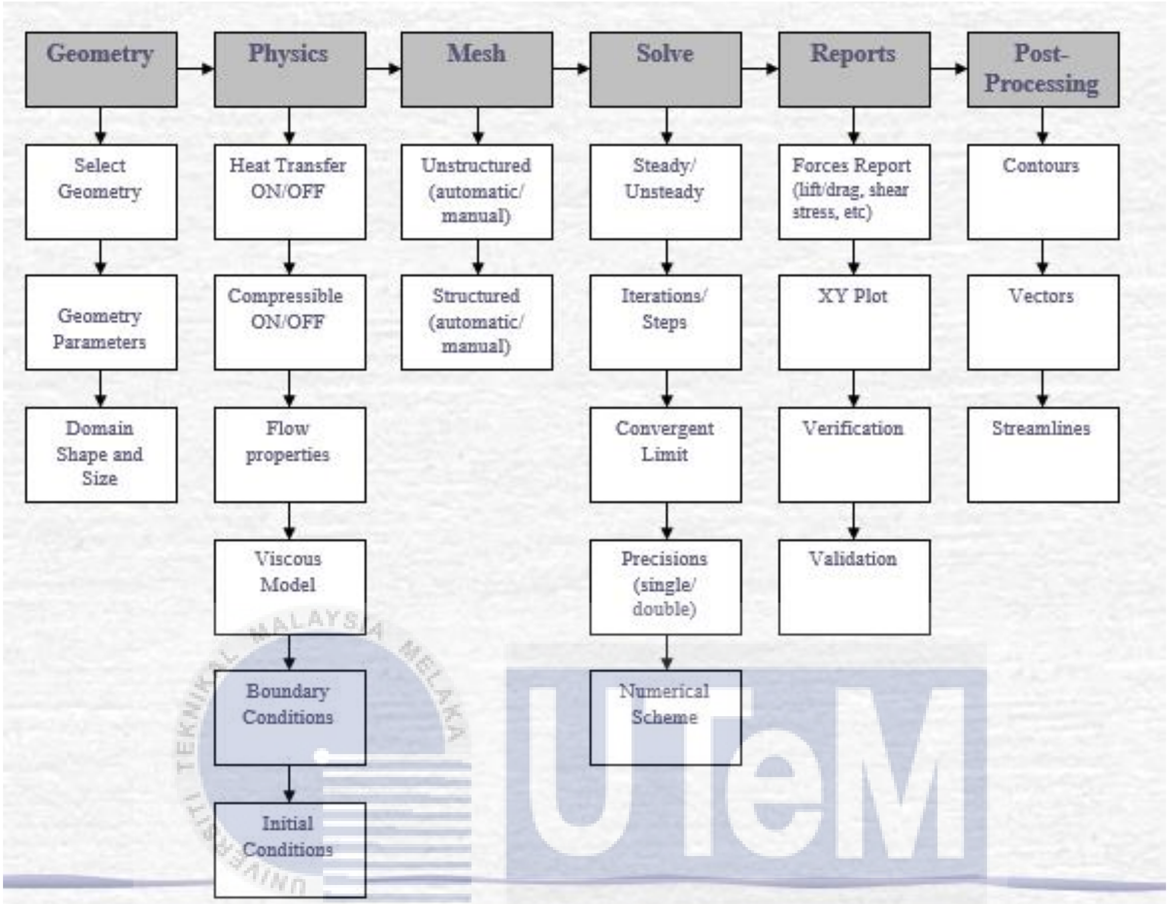


Figure 2.7: Working flow structure of CFD (Maysam et. al., 2012)

CHAPTER 3

LITERATURE REVIEW

This chapter discusses the literature review of previous related research. Several journal papers that are related to this field of study are reviewed and summarized. Comparison on the findings between researchers is included at the end of the chapter.

3.1 Comparison of air-conditioning systems with bottom-supply and side-supply modes in a typical office room by Zheng et. al. (2017)

This paper studied the performance of two different supply mode air-conditioning systems for a typical office building by physical experiment and numerical solution. The purpose for physical experiment was to validate the CFD model. The indoor thermal environment, human thermal comfort level and energy utilization efficiency of the two different supply mode were analysed and compared.

3.1.1 Methodology

An experimental laboratory for a simplified office with a dimension of 6m L x 4m W x 3.5m H was set up as shown in Figure 3.1. Bottom-supply and side-supply mode for the air-conditioning were created. There were eight air inlet for the bottom-supply air conditioning system and four inlet on the side walls. Both of the system used the same two outlet which located on the ceiling. Fourteen heat sources were included in the room, including four person (150W each), four computers (150W each) and six volumetric heaters (200W each). The cooling load of the office was 2400W. The supply velocity for both bottom-supply mode and side-supply mode were fixed at 0.44m/s and 0.21m/s, with the same $21^{\circ}C$ supply air temperature.

For physical experiment, there were a total of 100 measuring points measured with T-type thermocouples. T-type thermocouples was used to measure the air temperature and air velocity was measured by Testo air velocity transducers.

For the simulation analysis, ANSYS R 11.0 was used as the simulating software. The turbulence models used was RNG k-ε model. The boundary conditions for the simulation model was shown in Table 3.1. Mesh size of 6.5 million, 7.8 million and 9.6 million were used to verify the grid independence. Since all of the three type of mesh size result were close to each other, with difference under $0.5^{\circ}C$, 7.8 million mesh size was chosen as it was accurate enough. The model for both bottom-supply mode and side-supply mode were shown in Figure 3.2.

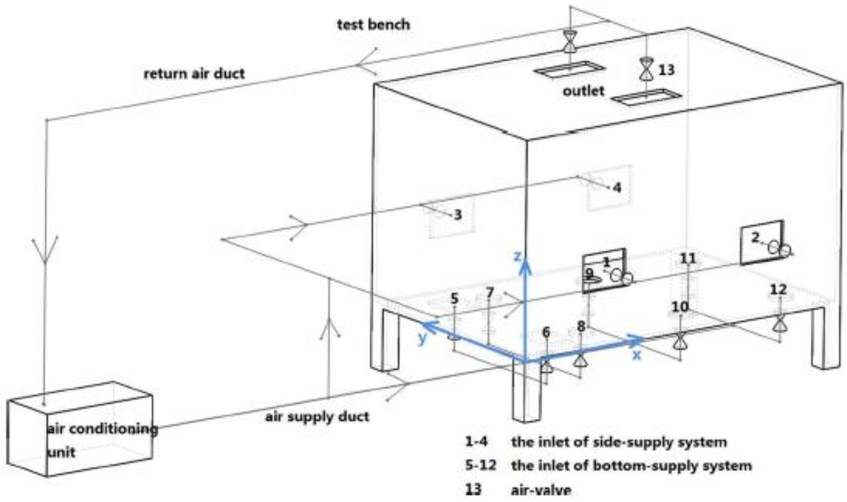
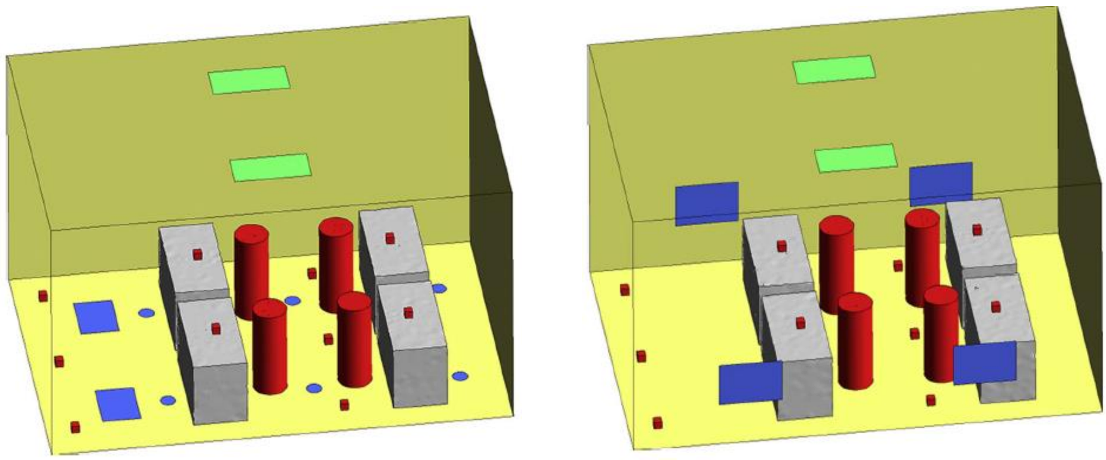


Figure 3.1: Schematic of the experimental system (Zheng et. al., 2017)

Table 3.1: Boundary conditions for the simulation (Zheng et. al., 2017)

Boundary	Boundary conditions
Inlet	Velocity inlet with 0.44 m/s and 21 °C under bottom-supply mode system, 0.44 m/s and 21 °C under side-supply mode system
Outlet	Pressure-outlet
Occupant	Four cylinders with constant energy source
Computer	Four cuboids with constant energy source
Other heat sources	Six cuboids with constant energy source
External wall	Constant temperature of 26 °C
Other walls	Adiabatic wall



(a) (b)

Figure 3.2: Model of (a) bottom-supply mode and (b) side-supply mode (Zheng et. al., 2017)

3.1.2 Results and discussions

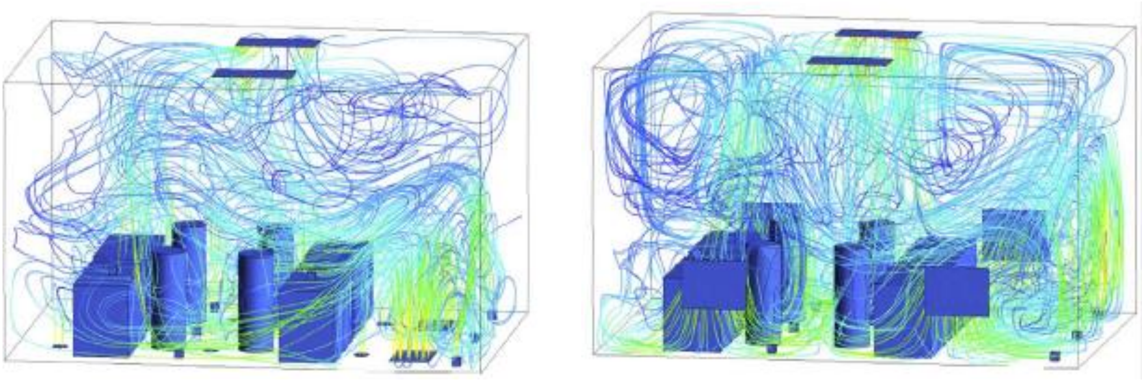
The simulation results for air temperature distribution for both mode were shown in Figure 3.3. There was an obvious temperature difference under both of the bottom-supply mode and side-supply mode. The temperature surrounding the heat source was higher compare to other region. The air temperature rose as the height increased. The indoor air formed stratification effect of temperature in height was due to the coupling effects of the buoyancy and pressure where the hotter air rose up while colder air fall down. The temperature at the bottom of the room for bottom-supply mode was relatively lower yet higher with the increasing height compared to the side-supply mode. The average temperature for the side-supply mode was 0.5^{o}C lower compared to bottom-supply mode due to its inlets' height.

Figure 3.4 shows the airflow distribution for both bottom-supply mode and side-supply mode. For the bottom-supply mode, whirlpool air flow pattern was observed due to the coupling of thermal and wind pressure. The cooled air flowed into the room through the inlet on the floor. The cooled air flow downwards due to gravity and changed direction after hitting the floor. Meanwhile for the side-supply mode, the cooled air supply fall down shortly after the cooled air flowed out from the side inlet. The cooled air then mixed with indoor air and discharged through the outlet.

The average value of air velocity simulated for bottom-supply mode and side-supply mode was 0.074 m/s and 0.064 m/s.

(a) (b)

Figure 3.3: Temperature distribution for (a) bottom-supply mode and (b) side-supply mode (Zheng et. al., 2017)



(a) (b)

Figure 3.4 :Airflow pattern for (a) bottom-supply mode and (b) side-supply mode (Zheng et. al., 2017)

3.2 Adaptive thermal comfort in university classrooms in Malaysia and Japan by Zaki et. al. (2017)

This study investigated the comfort temperature and adaptive behaviour of university students in Malaysia and Japan. The universities chosen were Universiti Teknologi Malaysia (UTM) Kuala Lumpur campus, Universiti Teknologi MARA (UiTM) Shah Alam campus and Kyushu University (KU) Chikushi campus. This research was carried out when Japan was at summer season to suit with Malaysia hot climate. The conditions of investigation were when air-conditioning was switched on for cooling purposes, known as mechanical cooling (CL) mode and when air-conditioning was switched off, known as free-running (FR) mode.

3.2.1 Methodology

Random classrooms were selected for all of the universities. A total of 6 classrooms ($57m^2$) were randomly selected in the building of the Malaysia Japan International Institute of Technology (MJIIT) for UTM, 14 classrooms ($38m^2$ to $47m^2$) in the building of the Faculty of Mechanical Engineering at UiTM and 4 classrooms ($55m^2$) in the Interdisciplinary Graduate School of Engineering Science (IGSES) buildings of KU.

Questionnaire survey was conducted simultaneously for each classroom. Thermal sensation was evaluated using the ASHRAE 7-point sensation scale, 5-point scale of thermal preference, thermal acceptability and 6-point scale of overall comfort. A total of 1415 respondents were used for analysis. Metabolic rate was assumed to be 1.2 met based on ASHRAE Standard 55 (students listening to lectures).

In this study, 5 objective parameters namely outdoor temperature (T_o), indoor air temperature (T_a), indoor globe temperature (T_g), indoor air velocity (V_a) and indoor relative humidity (RH) were measured. Measuring instruments such as Thermo recorder TR-77Ui (measure air temperature and relative humidity), Thermo recorder TR-52i (measure globe temperature), Hot-wire anemometer and VelociCalc 9565 (both measure air movement) were used.

3.2.2 Results and discussions

For the physical measurement results, both Malaysia and Japan had the similar mean indoor air temperatures regardless the ventilation mode. For relative humidity (RH), in overall the

FR mode locations had higher RH compared to CL mode locations. UTM had the lowest RH which operating under CL mode while KU had the highest value, operating under FR mode. Meanwhile for the air velocity, classrooms in KU had the lowest air velocity which is less than 0.10m/s. All of the four thermal indices (T_a, T_g, T_{mrt}, T_{op}) were almost similar in each location (difference less than 1°C), T_a was chosen for the analysis as it was used to determine the comfort range in international standards (ASHRAE). The overall result is shown in Table 3.1.

The comfort temperature were obtained through Griffiths' method by using the equation

$T_{comf} = T + (0\text{-}TSV) / \alpha$

Where, T_{comf} = comfort temperature (°C)

T = temperature (°C)

α = Griffiths' constant

Source: (Zaki et. al., 2017)

The value of α used was 0.50. It was found that the comfort temperature of FR mode for both Malaysia and Japan are 26.8°C and 25.1 °C respectively. For CL mode, the comfort temperature for Malaysia is 25.6 °C and 26.2 °C for Japan. The result was summarized in Table 3.3.

Table 3.2: Mean value of climatic parameters (Zaki et. al., 2017)

University	Mode	T_o (°C)	T_a (°C)	T_g (°C)	T_{mrt} (°C)	T_{op} (°C)	RH (%)	V_a (m/s)
UTM	CL (n = 677)	30.8 (2.4)	24.2 (0.8)	24.2 (0.9)	24.2 (1.1)	24.2 (0.9)	49.5 (6.8)	0.13 (0.11)
UiTM	FR (n = 106)	29.9 (3.1)	24.9 (0.3)	25.0 (0.5)	25.1 (0.9)	25.0 (0.5)	66.1 (3.0)	0.19 (0.05)
	CL (n = 196)	33.4 (2.9)	24.0 (1.0)	24.1 (1.0)	24.3 (0.8)	24.1 (0.9)	59.7 (6.5)	0.28 (0.09)
KU	FR (n = 152)	25.3 (1.6)	25.2 (0.7)	25.3 (0.7)	25.4 (0.7)	25.3 (0.7)	69.8 (5.0)	0.03 (0.02)
	CL (n = 284)	25.5 (2.5)	25.1 (0.8)	25.1 (0.8)	25.1 (0.8)	25.1 (0.8)	63.4 (6.7)	0.03 (0.02)

Table 3.3: Comfort operative temperatures based on the Griffiths' method (Zaki et. al., 2017)

Country	Mode	T_{comf} (°C)
Malaysia	FR	26.8
	CL	25.6
Japan	FR	25.1
	CL	26.2

3.3 Building Energy Performance and Indoor Environmental Quality (IEQ): Case Study Analysis in UTeM by Imran (2016)

This study aimed to investigate the existing indoor comfort parameters at FKM building located in UTeM technology campus. The existing indoor comfort parameters were measured and evaluated. The parameters were then compared with standards such as MS 1525 and ASHRAE Standard-55.

3.3.1 Methodology

The physical measurements were done by using measuring equipment. TSI's IAQ-Calc™ Indoor Air Quality Meter 7545 was used to measure CO, CO_2, air temperature and relative humidity. VelociCalc® Air Velocity Meter 9515 was used to measure air velocity and temperature.

FKM building consists of 7 floors. The average air temperature (^oC), average air velocity (m/s), average relative humidity (%), average flow (cfm) and average CO_2 in each floorwere measured by using the measuring equipment mentioned and recorded. The measurement had been repeated three times at 8:30 am to 10:00 am, 1:11 pm to 2:00 pm and 4.00 pm to 5:00 pm.

3.3.2 Results and discussion

The results for average air velocity and average air temperature for each floor in FKM building are shown in Table 3.4 and Table 3.5. For the average air velocity, the average data obtained for each floor was less than the recommended air velocity (0.15 m/s – 0.5 m/s) by MS1525: 2014. The same went for air temperature. The average air temperature for the each floor was less than the recommended air temperature (24^oC – 26^oC) by MS1525: 2014.

Table 3.4: The average air velocity (m/s) for each floor of the FKM building at three times (8:30 am to 10:00 am, 1:11 pm to 2:00 pm and 4.00 pm to 5:00 pm) and average data comparison

No.	Floor	Average Air Velocity (m/s)			
		8:30 am to 10:00 am	1:11 pm to 2:00 pm	4.00 pm to 5:00 pm	Average Data Comparison
1	Ground Floor	0.089	0.095	0.089	0.091 (less than standard)
2	First Floor	0.086	0.092	0.095	0.091 (less than standard)
3	Second Floor	0.074	0.079	0.072	0.075 (less than standard)
4	Third Floor	0.084	0.090	0.081	0.085 (less than standard)
5	Fouth Floor	0.104	0.111	0.114	0.110 (less than standard)
6	Fifth Floor	0.076	0.081	0.081	0.079 (less than standard)
7	Sixth Floor	0.107	0.114	0.097	0.106 (less than standard)
8	Seventh Floor	0.078	0.083	0.071	0.077 (less than standard)
Average		0.087	0.093	0.088	0.089 (less than standard)

Table 3.5: The average air temperature (^{o}c) for each floor of the FKM building at three times (8:30 am to 10:00 am, 1:11 pm to 2:00 pm and 4.00 pm to 5:00 pm) and average data comparison

No.	Floor	Average Air Velocity (m/s)			
		8:30 am to 10:00 am	1:11 pm to 2:00 pm	4.00 pm to 5:00 pm	Average Data Comparison
1	Ground Floor	24.2	24.1	22.5	23.60 (less than standard)
2	First Floor	24.4	23.2	21.6	23.07 (less than standard)
3	Second Floor	22.8	23.0	22.8	22.87 (less than standard)
4	Third Floor	22.3	23.5	21.8	22.53 (less than standard)
5	Fouth Floor	24.0	22.8	21.9	22.90 (less than standard)
6	Fifth Floor	23.2	23.1	21.9	22.73 (less than standard)
7	Sixth Floor	23.3	23.9	22.0	23.07 (less than standard)
8	Seventh Floor	23.2	22.9	22.1	22.73 (less than standard)
Average		23.425	23.313	22.075	22.94 (less than standard)

3.4 Indoor Airflow Simulation inside Lecture Room by Lin, Tee and Tan (2015)

This study investigated the two lecturer rooms in UTeM which consists of two types of air-conditioning configuration system which are split unit and centralized system. The air flow distribution between these two systems were analysed and compared. ANSYS Fluent software was used to develop the CFD simulation. From the results, lecture room 1 has a better streamline pattern than lecture room 2. However, the air velocity in lecture room 1 is below than the recommended standard stated by MS 1525-2007.

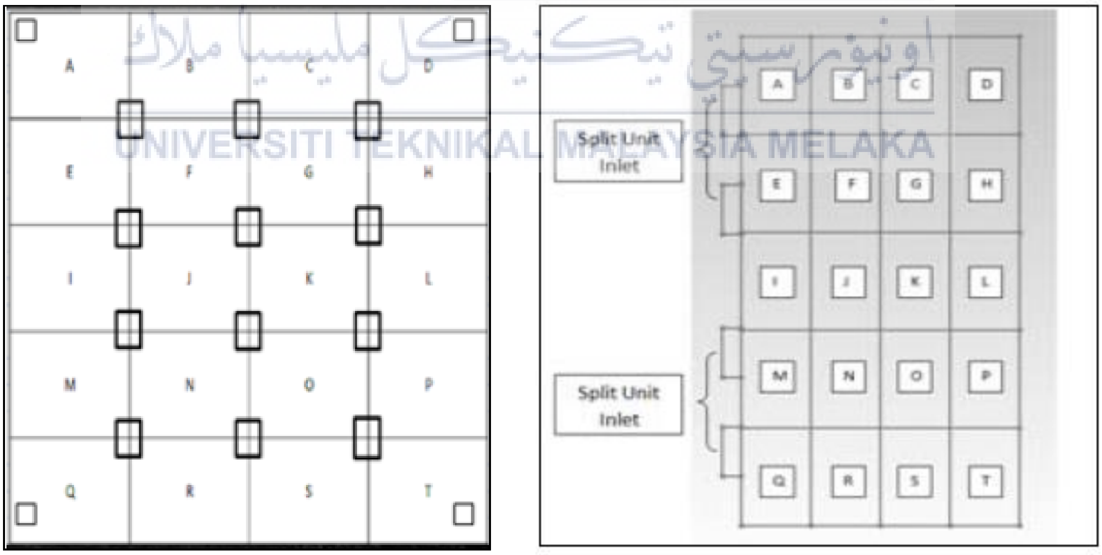
3.4.1 Methodology

This study was conducted in two lecture rooms in UTeM. Both lecture rooms has different dimensions and different air-conditioning configuration. Lecture room 1 has a dimension of 12m length, 9m width and 3m height. The type of air-conditioning configuration used in lecture room 1 is central unit. Lecture room 2 has a dimension of 14.85m length, 7.35m width and 3.5m height. The type of air-conditioning configuration used in lecture room 2 is split unit.

For physical measurement, air velocity meter was used to measure the air velocity and air temperature for both of the lecture rooms. The height of sensor was set to 1.1m above ground level. The air velocity meter can measure air velocity from 0m/s to 30m/s while the temperature range is from 17.8^{o}C to 93.0^{o}C. The lecture rooms were divided into 20 zones to obtain the desired measurement as shown in Figure 3.5.

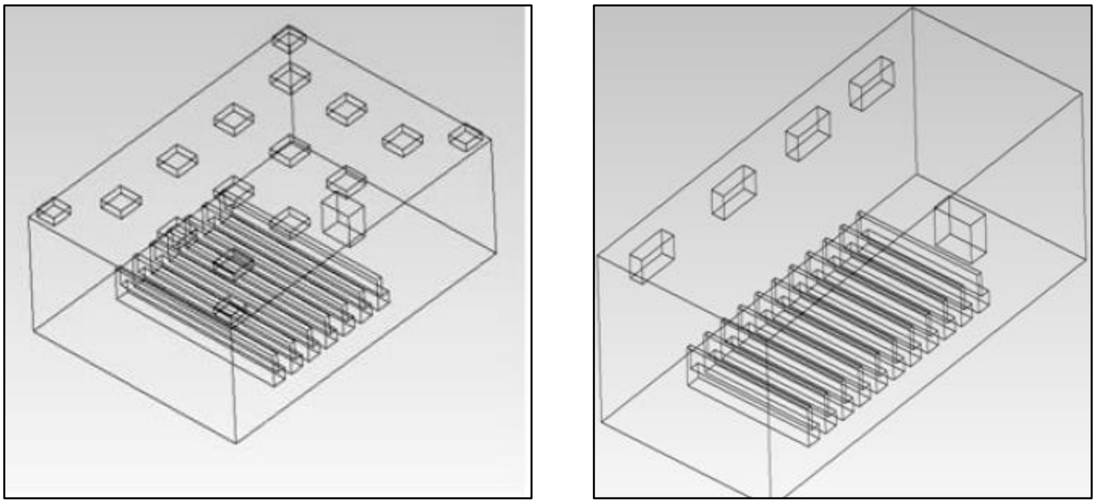
For CFD simulation, the simulation is conducted by using the ANSYS Fluent CFD software. The general room modeling procedures are defining geometry of the model, defining fluid domains, boundary conditions, initial guess, meshing and solver control. The Geometry sketch is shown in Figure 3.6. The authors also made few assumptions as follow:

a) The air conditioning system in selected lecture rooms is fully functioning and running well.
b) The rooms are fully sealed and enclosed without any holes or gaps (excluding doors, windows and exhaust vent).
c) The outside temperature on the surface of the room is constant
d) Internal heat source emitted from the digital devices and lights will be neglected due to minimal effect on the temperature.
e) The furniture (chairs and tables) are included in the simulation.



(a) (b)

Figure 3.5: Zonal Division of (a) Lecture Room 1 and (b) Lecture Room 2 (Lin, Tee and Tan, 2015)



(a) (b)

Figure 3.6: Geometry Sketch of (a) Lecture Room 1 and (b) Lecture Room 2 (Lin, Tee and Tan, 2015)

Table 3.6: Summary of Simulation Parameter (Lin, Tee and Tan, 2015)

Items	Lecture Room 1	Lecture Room 2
Domain		
Viscous model	k-epsilon (2 equation)	k-epsilon (2 equation)
k-ϵ Model	RNG	RNG
Near-wall treatment	Standard Wall Functions	Standard Wall Functions
Meshing		
Number of Nodes	30779	27886
Number of Elements	155986	141285
Solution Method		
Turbulent Kinetic Energy	Second Order Upwind	Second Order Upwind
Turbulent Dissipation Rate	Second Order Upwind	Second Order Upwind

3.4.2 Results and Discussions

In this study, the average air temperature and average air velocity for both lecture rooms were determined. Table 3.2 shows the experimental result and CFD simulation result in this study. For lecture room 1, the average air temperature and air velocity obtained for from experimental were 27.3^oC and 0.12m/s. While for lecture room 2, the results are 21.8^oC and 0.15m/s respectively. From the experimental data obtained, it is known that the results for lecture room 1 did not comply with the standard MS 1525:2007. At the opposite, the results for lecture room 2 shows a better air flow compare to lecture room 2.

For the CFD approach, the average air temperature and air velocity obtained for lecture room 1 were 26.9^oC and 0.10m/s. The average air temperature and air velocity obtained for lecture room 2 were 18.7^oC and 0.18m/s. The simulated results showed that data obtained for both lecture rooms did not comply with the standard MS 1525:2007.

The CFD stream plot for the air velocity of both lecture rooms are shown in Figure 3.7 and Figure 3.8. Based on the figures, it can be seen that lecture room 1 (centralized air conditioning unit) has a better air distribution pattern compare to lecture room 2 (split unit system).

Table 3.7: Experimental results and CFD simulation result for both lecture room 1 and 2 (Lin, Tee and Tan, 2015)

Items	Recommended Standard MS 1525:2007	Physical Measurement of lecture room 1	CFD simulation of lecture room 1	Physical Measurement of lecture room 2	CFD simulation of lecture room 2
Average Air Velocity (m/s)	0.15-0.5	0.12	0.10	0.15	0.18
Average Temperature (^{o}C)	22-26	27.3	26.9	21.8	18.7

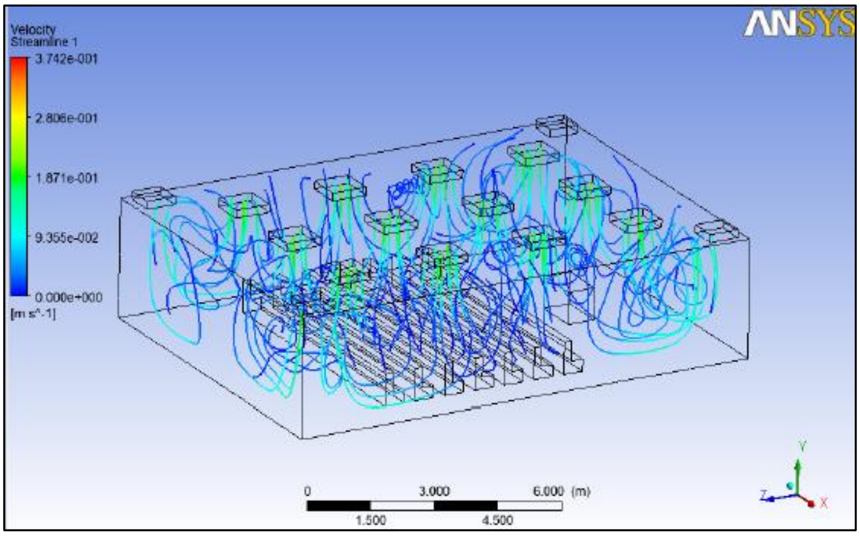


Figure 3.7: Stream plot for lecture room 1 (Lin, Tee and Tan, 2015)

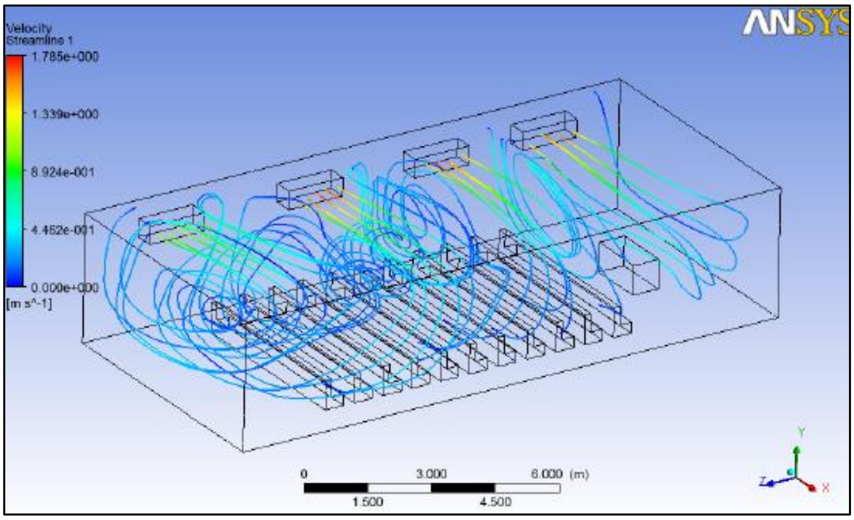
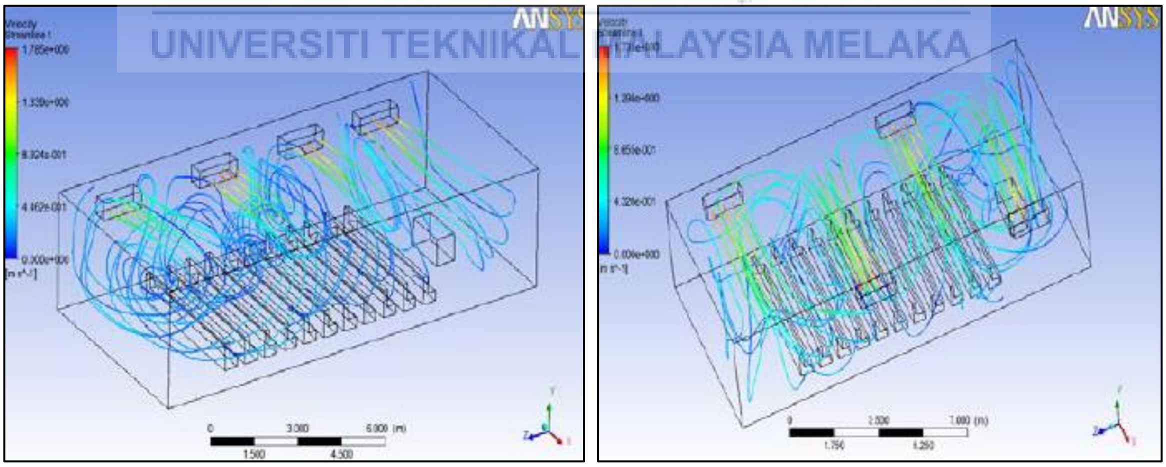


Figure 3.8: Stream plot for lecture room 2 (Lin, Tee and Tan, 2015)

The authors further modified the orientation and positioning of the split unit air conditioning unit in lecture room 2 to obtain a better air flow pattern. Figure 3.9 shows the different orientation of the air conditioning unit that gave a better air flow pattern in lecture room 2.



(a) (b)

Figure 3.9: Original position of the split units (a) and new orientation of the split units (b) with the air flow pattern (Lin, Tee and Tan, 2015)

The difference between the physical measurement result and CFD simulation results had a slight different from each other. For lecture room 1, the differences were 16.7% for air velocity and 1.4% for air temperature. For lecture room 2, the differences were 20.0% for air velocity and 14.22% for air temperature. The differences were mainly due to uncontrolled infiltration in actual room which are difficult to be quantified for CFD simulation. Overall, the air flow distribution in lecture room 1 was better than lecture room 2 as the air distribution was steadier and well distributed.

3.5 CFD Analysis on Thermal Comfort and Energy Consumption Effected by Partitions in Air-Conditioned Building by Aryal and Leephakpreeda (2015)

This paper studied about the effect of partitions on the thermal comfort level and energy consumption in an air-conditioned building by using CFD analysis. The variables of indoor air before and after installation of partitions were simulated. The simulated results were then validated by measurements with good agreement in which a case study was conducted in a library.

3.5.1 Methodology

To study the occupants' thermal comfort and energy consumption of an air-conditioned building, CFD analysis was conducted. The Predicted Mean Vote (PMV) was used as an index of thermal comfort. Library at Sirindhorn International Institute of Technology, Thammasat University was chosen as the study area. The library has a dimension of 38m L x 27.3m W x 3m H. There were a total of 23 ceiling type diffusers that were working during the time of experiment. Heat generated by occupants (140W), ceiling lights (60W) and computers (45W) were included in the simulation. The model of the library is shown in Figure 3.10. The partition was installed at the location represented in red box. The partition separated the library into reading area (left side) and resting area (right side).

The air temperature and air velocity were measured by using Testo 425 compact thermal anemometer. The relative humidity was measured by Testo 610 handy humidity meter. The boundary conditions at supply diffusers is shown in Table 3.8.

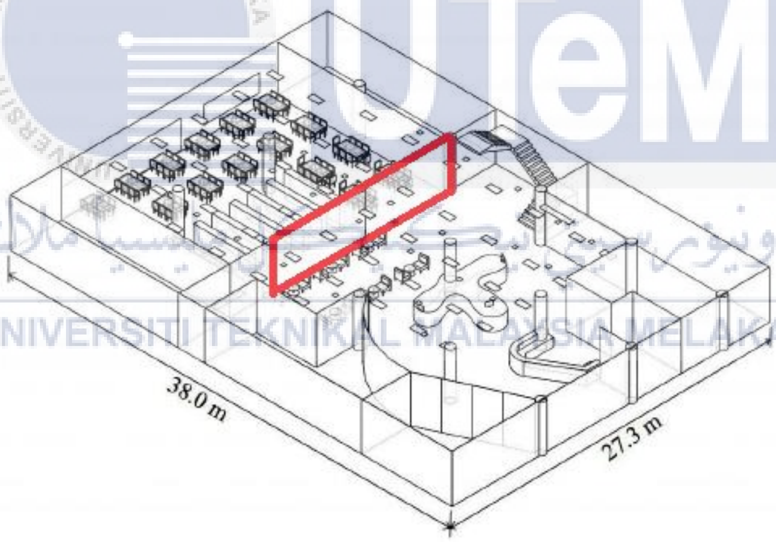


Figure 3.10: Model of the library (Aryal and Leephakpreeda, 2015)

Table 3.8: Boundary conditions at supply diffusers (Aryal and Leephakpreeda, 2015)

Diffuser	D1	D2	D3	D4	D5	D6	D7	D8	D9	D10	D11	D12	D13
Air temperature (℃)	23.3	23.3	23.3	23.3	-	-	23.3	23.3	23.3	23.3	22.6	22.6	22.6
RH (%)	60	60	60	60	-	-	60	60	60	60	63	63	63
Air velocity (m/s)	3.72	3.68	3.57	3.51	-	-	3.51	3.57	3.68	3.72	3.92	3.86	3.74
Diffuser	**D14**	**D15**	**D16**	**D17**	**D18**	**D19**	**D20**	**D21**	**D22**	**D23**	**D24**	**D25**	**D26**
Air temperature (℃)	-	-	22.6	22.6	22.6	-	-	-	-	-	-	-	-
RH (%)	-	-	63	63	63	-	-	-	-	-	-	-	-
Air velocity (m/s)	-	-	3.74	3.86	3.92	-	-	-	-	-	-	-	-
Diffuser	**D27**	**D28**	**D29**	**D30**	**D31**	**D32**	**D33**	**D34**	**D35**	**D36**	**D37**	**D38**	
Air temperature (℃)	21	21	21	21	21	21	21	21	20	-	-	-	
RH (%)	65	65	65	65	65	65	65	65	68	-	-	-	
Air velocity (m/s)	3.73	3.69	3.61	3.59	3.59	3.61	3.69	3.73	5.6	-	-	-	

3.5.2 Results and Discussions

The simulated results for the PMV value before and after installation of partition are shown in Figure 3.11. Before installation of partition, the result showed a neutral thermal sensation for most of the region at the reading area. Meanwhile for the resting area, the simulation result showed that the area had slightly cool and cool thermal sensation

After installation of partition, it can be seen that the overall PMV value in reading area had significantly increased but it was still remained below the threshold limit of thermal neutrality in most of the region. However, the location at the top left corner had a PMV value of 0.7-0.8 (slightly warm sensation). In the resting area, the overall PMV values decreased. It can be seen that the staircase area had changed from neutral sensation to slightly cool sensation.

Table 3.9 shows the results obtained from energy consumption. Based on the result, the energy consumption had increased from 10.23 kW to 12.73 kW, a total increment of

24%. This was due to the conditioned air at the reading side compensated a higher heat load after installation of partition.

Based on the simulation result obtained, it was not recommended to install the partition in current scenario.

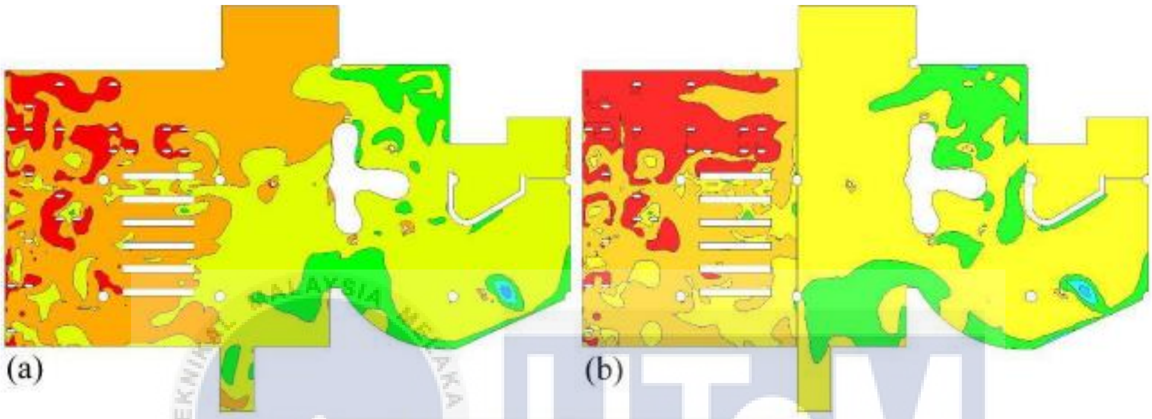


Figure 3.11: Comparison of PMV distribution (a) without partition, (b) with partition (Aryal and Leephakpreeda, 2015)

Table 3.9: Air temperature, air velocity and relative humidity at the extract grilles (Aryal and Leephakpreeda, 2015)

	Before installation of partition				After installation of partition			
Extract grilles	E1	E2	E3	E4	E1	E2	E3	E4
Air Temperature (°C)	24.6	24.4	24.5	24.2	22.9	24.9	25.1	22.6
RH (%)	52	52.4	52.7	54	58	53.2	53.2	59.9
Air velocity (m/s)	3.42	2.39	1.45	1.03	2	2.44	2.14	1.7
Energy consumption for making up air (kW) $\sum_{E1}^{E4} \dot{m}_o h_o - \sum_{D1}^{D38} \dot{m}_i h_i$	10.23				12.73			

E= Extract grilles, D= Supply diffuser,
$\dot{m} = \rho A v$, ρ = 1.225 kg/m^3, Area of extract grilles, $A_o = 1m^2$, Area of supply diffuser, A_i = 0.0961 m^2, velocity (v) and enthalpy (h).

3.6 Thermal comfort assessment of an office building served by under-floor air distribution (UFAD) system- A case study by Alajmi, Baddar and Bourisli (2014)

This study presented the occupants' thermal comfort in an office building using under-floor air distribution (UFAD) system. The occupants' thermal comfort level was investigated through field survey, physical measurement, subjective measurement and CFD analysis. Best operating conditions were determined by using CFD to maximize the occupants' thermal comfort level.

3.6.1 Methodology

Questionnaires were distributed to 40 occupants to determine their comfort level. The questionnaire was prepared based on ASHRAE 55 sample questionnaire. The tested room had a dimensions of 3.10 m L x 5.50m W x 2.90m H. A secretary assumed with 1.2 Met (70 W/m^2) and 0.5 Clo value stayed inside the room.

A physical measurement was done in the testing room. Relative humidity, air velocity, dry bulb temperature, wet bulb temperature and operative temperature were measured using an indoor climate analyser named Brüel and KjærTM (type 1221). ASHRAE seven-point thermal sensation scale was used to carry out the subjective measurement to measure the occupant's thermal comfort level in the tested room. CFD analysis was done by using DesignBuilder^{TM} software. The geometry of the tested room was shown in Figure 3.12 and the boundary condition was shown in Table 3.10.

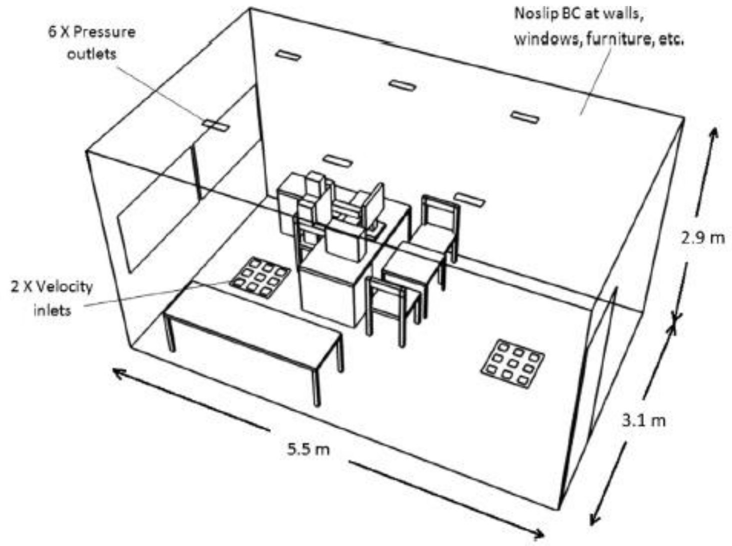


Figure 3.12: Geometry for the tested room (Alajmi, Baddar and Bourisli, 2014)

Table 3.10: Boundary conditions for the tested room (Alajmi, Baddar and Bourisli, 2014)

Parts of the room	Boundary condition
Walls, windows, furniture	Dirichlet (no slip) boundary condition
Velocity inlet	Dirichlet (velocity inlet) boundary condition
Pressure outlets	Neumann (zero gauge pressure) boundary condition

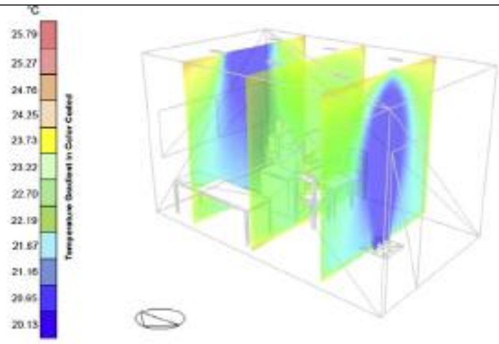
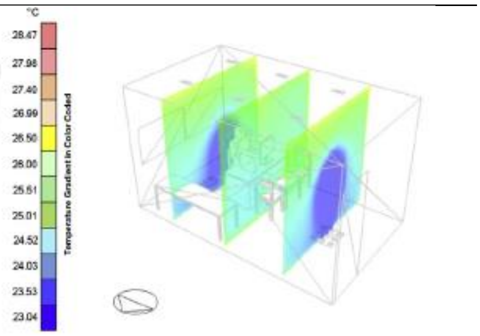
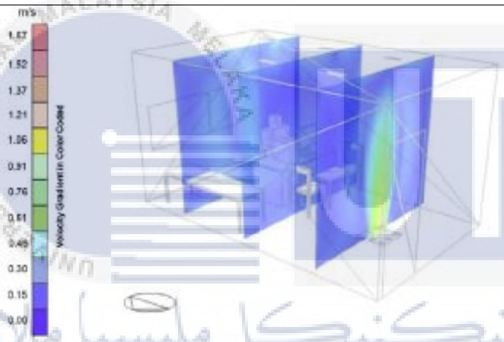
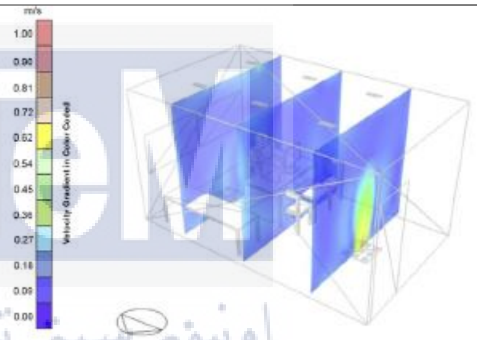
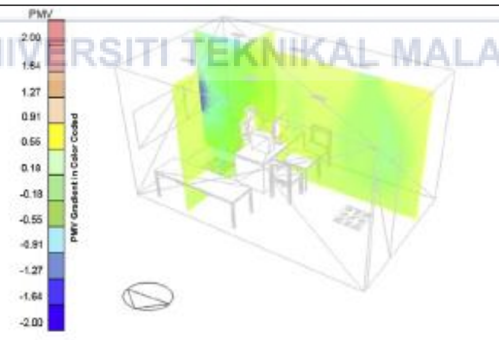
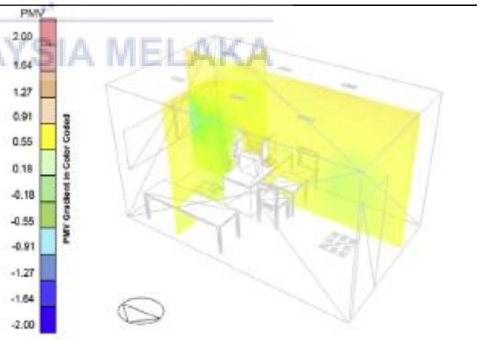
3.6.2 Results and Discussions

The result from the CFD simulation is shown in Table 3.11. The result showed that the ADPI is greatly affected by the air temperature and air velocity. The ADPI value of 96% indicated that 21^{o}C of air temperature and 1 m/s of air velocity were the best operation condition case to provide thermal comfort to occupants. The PMV value of the occupants showed a lesser degree by changes in the studied parameters. It was believe that the thermal comfort assessment based on PMV alone might be misleading as all of the PMV results shown were between -0.5 to +0.5 for all the cases and PMV was based on a single point measurement. The ADPI results tend to be more reliable to access the thermal comfort level as it was based on the multiple measurement points which showed variation. Table 3.12 shows the comparison of the parameters studied for the current setting (18°C and 1.5 m/s) and the best proposed setting (21°C and 1.0 m/s).

Table 3.11: ADPI and PMV values of the tested case using the validated CFD model (Alajmi, Baddar and Bourisli, 2014)

Simulation no.	Air temperature,−°C	Air velocity, m/s	ADPI	PMV
1	18	1	52	−0.2
2	19	1	76	+0.2
3	20	1	84	+0.3
4	21	1	96	+0.5
5 (actual)	18	1.5	0	−0.6
6	19	1.5	12	−0.1
7	20	1.5	44	+0.3
8	21	1.5	92	+0.4
9	19	2	36	−0.15
10	20	2	80	+0.3
11	21	2	84	+0.4

Table 3.12: Comparison between the current setting and the best proposed setting on temperature, air velocity and PMV (Alajmi, Baddar and Bourisli, 2014)

	Actual (18^{o}C and 1.5 m/s)	Proposed (21^{o}C and 1.0 m/s)
Temperature		
Velocity		
PMV		

3.7 Simulation of Thermal Comfort of a Residential House by Tap et. al. (2011)

This study investigates the thermal comfort level in a naturally ventilated residential house in Malaysia using computational fluid dynamics (CFD) method. The temperature distribution, air flow pattern and relative humidity (RH) were conducted through actual measurements. CFD simulations were then carried out to visualize the temperature distribution and air flow velocity and pattern in the house. Three type conditions of the house were modelled and simulated which are naturally ventilated house, naturally ventilated house with ceiling fan and naturally ventilated house with ceiling and extractor fans.

3.7.1 Methodology

The type of house model created was a residential single-storey terrace house with interior regions view by using Fluent CFD software. The house had regions such as hall, kitchen and stack. Front door, back door, front wall, mid wall and back wall were included as well (as shown in Figure 3.13).

Physical measurement was carried out to obtain the average air temperature, the relative humidity and average velocity of the air inside the house. The period for data recording was from 12a.m. to 9p.m. The data was then compared with ASHRAE standard to determine the thermal comfort level in the house.

For simulation analysis, CFD was used. First of all, virtual house was developed by using CFD under three different conditions. The first model represented the house with natural ventilation condition with no mechanical cooling. The second model represented the house equipped with a ceiling fan which located in the middle of the hall section. The third model represented the house equipped with a ceiling fan and two extractor fans, one equipped on the front wall while the other equipped at the real wall. The distribution of air temperature and air velocity inside the house were then performed by CFD.

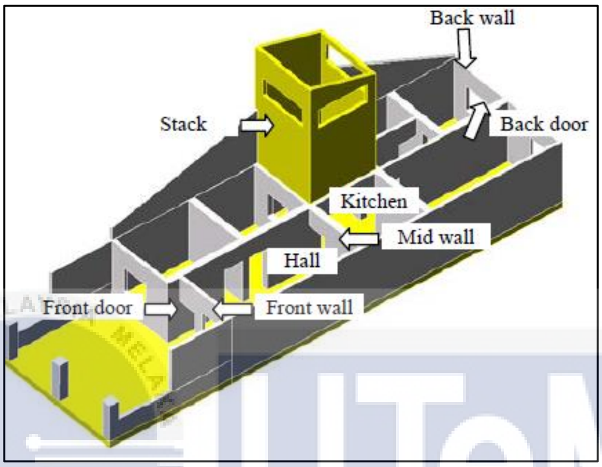


Figure 3.13: CFD model of a residential house (Tap et. al, 2011)

3.7.2 Results and Discussions

3.7.2.1 Comfort Conditions with Natural Ventilation

The average air temperature in the house was obtained through physical measurement. The highest value of air temperature recorded was 31°C at 6pm. The data obtained was then compared with ASHRAE standard and it was found that the average air temperature was not in the acceptable thermal comfort recommended by ASHRAE standard.

Relative humidity of the natural ventilated house was also not within the acceptable range of ASHRAE standard. The relative humidity of the air was ranged from 71% at 10 am to 81% at 3 am. This shows that the air in the house was too humid and reduces the thermal comfort level of the occupants. The third parameter measured was air velocity. The average air velocity in the house was below the acceptable level of air velocity stated by ASHRAE standard as well.

3.7.2.2 Comfort Conditions with Combined Natural Ventilation and a Ceiling Fan

Physical measurement was taken once again with the ceiling fan being turned on to investigate the effects of the working ceiling fan on the thermal comfort parameters. The air temperature in all regions managed to decrease to 27^{o}C by 10 am. The highest value recorded was 31^{o}C at 6 pm, which was the same as the value obtained from the natural ventilation condition. It can be concluded that even with the turning on of ceiling fan, the average air temperature was still outside the acceptable range stated by ASHRAE standard.

The highest value of relative humidity recorded was 80% at 1p.m. and the lowest was 65% at 6p.m. There was not much reduction in the relative humidity and it was out of the acceptable range stated by ASHRAE standard.

However, the working ceiling fan brought a significant effect to the air velocity. The air velocity for both of the hall section and kitchen section were acceptable according to the thermal comfort level stated by ASHRAE standard. Only the air velocity in the stack section was below the acceptable level of thermal comfort.

3.7.2.3 CFD Simulation on Naturally Ventilated House

For the CFD simulation, only air temperature distribution and air velocity were included. The boundary conditions for the house was shown in Table 3.13. Front door, middle door and back door were left opened. There were openings at the top of the stack to allow air flow from outside.

The simulation result for air flow distribution was shown in Figure 3.14. It can be seen that the air flow in a streamline condition, from front door to middle door at a decreasing velocity, and from middle door to back door with increasing velocity. The stack and kitchen had an uneven flow condition.

Figure 3.15 shows the simulation result for air temperature distribution inside the house. The air temperature distribution was taken at 1.5m height above floor level. It can be seen that in average, air temperature of the hall had a value of 302K (29^{o}C). The kitchen had an average air temperature of 303K (30^{o}C). The front wall had a slightly higher temperature at about 305K (32^{o}C), due to high heat gain through this wall.

Table 3.13: Boundary conditions for the house with natural ventilation (Tap et. al, 2011)

Types of BCs	Zone	Parameters
Inlet Air Velocity	Front door	Velocity = 0.4 m/s Temperature = 29.6°C
	Stack	Velocity = 0.4 m/s Temperature = 29.8°C
Outlet Air Pressure	Back Door	Pressure = 101 kPa Temperature = 29.3°C
Wall Thermal Conditions	Stack	Heat gain = 10 W/m^2 Temperature = 30°C
	Front wall	Heat gain = 29.8 W/m^2 Temperature = 29.3°C
	Back wall	Heat gain = 19.8 W/m^2 Temperature = 29.4°C

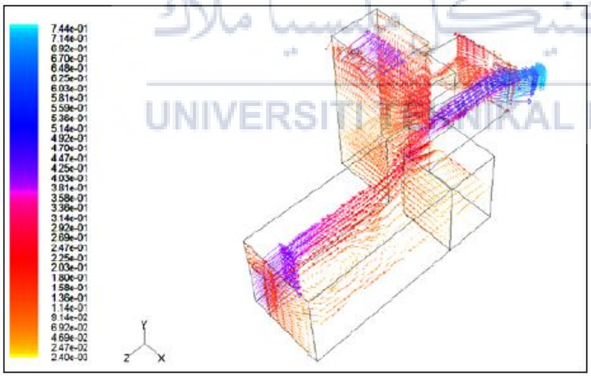


Figure 3.14: Distribution of air velocity (m/s) inside the house with natural ventilation condition (Tap et. al, 2011)

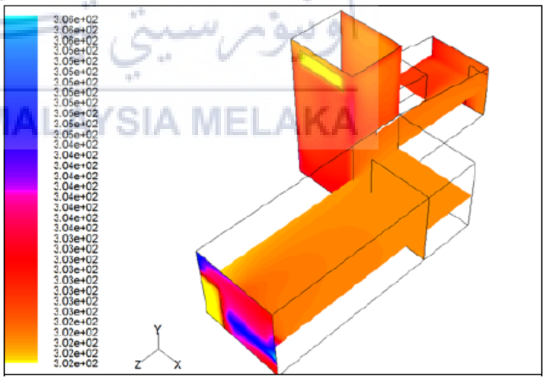


Figure 3.15: Temperature (Kelvin) distribution inside the house with natural ventilation condition (Tap et. al, 2011)

3.7.2.4 CFD Simulation on Naturally Ventilated House with Ceiling Fan

The next CFD simulation was done on the naturally ventilated house with ceiling fan condition. The boundary condition for this model was same as Table 3.3. Another boundary condition was added to represent the ceiling fan, rotating at speed of 150 RPM.

For the air velocity distribution, it can be observed from Figure 3.16 that the ceiling fan had caused swirling air flow. The highest velocity is 1.3m/s near the front wall. Stack section has a lower air velocity with uniform pattern. The working ceiling fan did increase the air flow velocity inside the house. However, the air velocity result obtained was much higher than the air velocity recommended by ASHRAE standard and it might cause discomfort to occupants.

Besides that, there was no significant different on the temperature distribution under the working ceiling fan. The temperature distribution in the hall and kitchen remain uniform at 302K (29^{o}C) and 303K (30^{o}C) as shown in Figure 3.17.

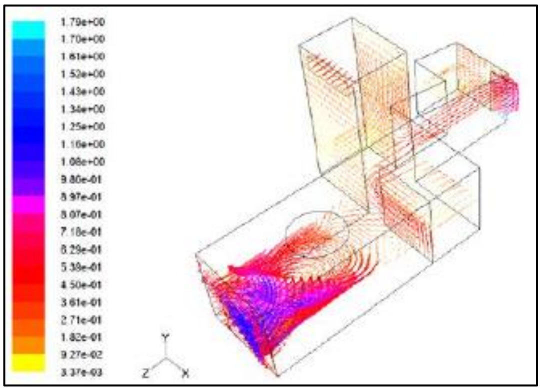


Figure 3.16: Distribution of air velocity (m/s) when the ceiling fan was turned on (Tap et. al, 2011)

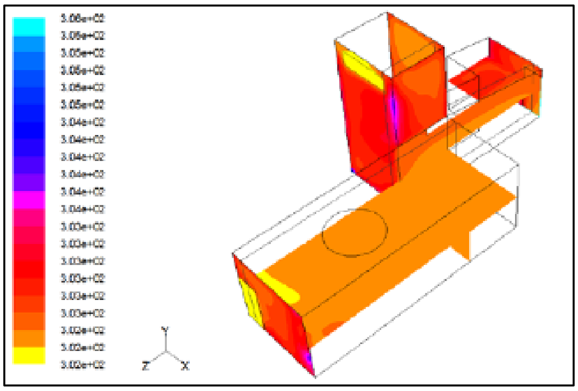


Figure 3.17: Temperature (Kelvin) distribution when the ceiling fan was turned on (Tap et. al, 2011)

3.7.2.5 CFD Simulation on Naturally Ventilated House with Ceiling and Extractor Fan

The third CFD simulation model was the naturally ventilated house equipped with working ceiling and extractor fan. Two extractor fans were equipped on the front and back wall. The introduced boundary conditions for the extractor fan were air pressure of 101 kPa and temperature of 29°C flowed with a 1.4m/s velocity.

The simulation result shown in Figure 3.18 shows that the front wall extractor fan did not have a significant effect on the air velocity distribution. This was probably due to the small amount and velocity of air compared to swirling flow condition of the air that created by the ceiling fan. However, the back wall extractor fan did affect the air flow condition in the kitchen and stack sections.

Meanwhile, the front wall extractor fan did not affect the hall section’s air temperature distribution as shown in Figure 3.19. The simulated results showed that the use of extractor fans had no significant effects on both of the air temperature distribution as well as air velocity distribution.

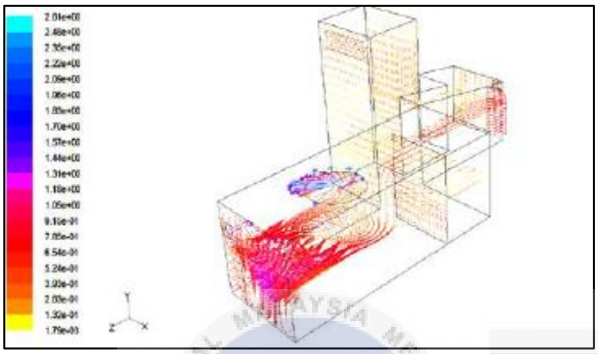


Figure 3.18: Distribution of air velocity (m/s) when the ceiling fan and extractor fans were turned on (Tap et. al, 2011)

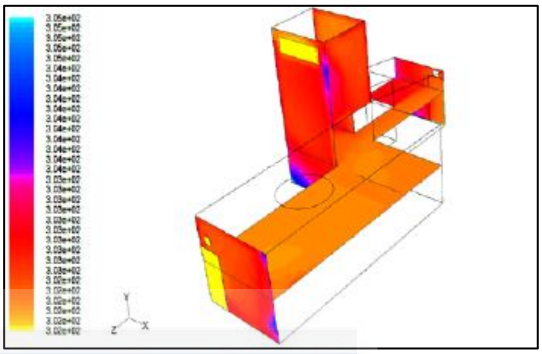


Figure 3.19: Temperature (Kelvin) distribution when the ceiling fan and extractor fans were turned on (Tap et. al, 2011)

3.8 Comparison of Study

Table 3.13 shows the comparison of study that has been summarized. The thermal comfort parameters which are air temperature and air velocity from the journals are compared with ASHRAE Standard 55-2010 and Malaysia Standard MS1525- 2014. The recommended air temperature and air velocity by ASHRAE Standard 55-2010 are 20ºC to 27ºC and 0.1 m/s to 1.2 m/s. On the other hand, the recommended air temperature and air velocity by Malaysia Standard MS1525- 2014 are 24ºC to 26ºC and 0.15 m/s to 0.50 m/s. The comparison of results for simulation and physical measurement are described in Table 3.14 and Table 3.15.

Table 3.14: Comparison of simulation results from journals

Researcher(s)	Title of study	Building Type	Type of study	Results	Result comply with standards	
					ASHRAE	MS1525
Lin, Tee and Tan (2015)	Indoor Airflow Simulation inside Lecture Room: A CFD Approach	Lecture rooms	Investigate the air flow for split unit and centralized system.	Lecture room 1: 27.3°C and 0.12 m/s	No	No
				Lecture room 2: 21.8°C and 0.15m/s	Yes	No
Tap et. al. (2011)	Simulation of Thermal Comfort of a Residential House	Residential house	Investigate the effect of ceiling fan and extractor fan on a naturally ventilated house.	Both ceiling fan and extractor fan did not affect much on the air temperature, air velocity and RH.	No	No

Aryal and Leephakpreeda (2015)	CFD Analysis on Thermal Comfort and Energy Consumption Effected by Partitions in Air-Conditioned Building	Library	Investigate the effect of partitions on the thermal comfort level.	The overall PMV value at reading area increased and the energy consumption increased by 24%. It was not recommend to install partition.	No	-
Zheng et. al. (2017)	Comparison of air-conditioning systems with bottom-supply and side-supply modes in a typical office room	Office building	Compare the performance of bottom-supply and side-supply air conditioning system.	The average value of air velocity simulated were 0.074 m/s and 0.064 m/s for bottom-supply mode and side-supply mode. Better thermal comfort level was obtained under bottom-supply mode with 21^{o}C supply air	Air velocity- No Air temperature - Yes	-

				temperature based on the air temperature distribution and air flow pattern.		
Alajmi, Baddar and Bourisli (2014)	Thermal Comfort assessment of an office building served by under-floor air distribution (UFAD) system- A case study	Office building	The best setting for air temperature and air velocity to provide suitable thermal comfort level.	The best setting obtained was 21°C and 1.0 m/s which contributed 96% improvement of thermal comfort.	Yes	-

Table 3.15: Comparison of physical measurement results from journals

Researcher(s)	Title of study	Building Type	Type of study	Results	Result comply with standards	
					ASHRAE	MS1525
Zaki et. al. (2017)	Adaptive thermal comfort in university classrooms in Malaysia and Japan	Universities Buildings	Investigate the comfort temperature and adaptive behaviour of university students in Malaysia and Japan	Comfort temperature for Malaysia was calculated as 26.8 ^{o}C for FR mode and 25.6 ^{o}C for CL mode.	FR- Yes CL- Yes	FR- No CL- Yes
				Comfort temperature for Japan was calculated as 25.1 ^{o}C for FR mode and 26.2 ^{o}C for CL mode.	Yes	-

Imran (2016)	Building Energy Performance and Indoor Environmental Quality (IEQ): Case Study Analysis in UTeM	University Building	Investigate the existing indoor comfort parameters at FKM building located in UTeM technology campus	The average air velocity for FKM building was 0.089 m/s	No	No
				The average air temperature for FKM building was 22.94^{o}C	No	No

CHAPTER 4

METHODOLOGY

This chapter illustrates the methods that were be applied in this study based on the scope. The important procedures to carry out this research were generally explained which includes the selection of suitable floor of FKM building, measurement of physical parameters in the selected floor as well as carry out simulation of the parameters using CFD software.

4.1 Description of the building

The purpose of this study is to evaluate on the thermal comfort in a typical office building. The case study building selected is the building of Faculty of Mechanical Engineering (FKM) which is located in UTeM's technology campus, situated in Ayer Keroh, Melaka, Malaysia. Figure 4.1 shows the bird view of FKM's building and Figure 4.2 shows the isometric view of FKM's building.

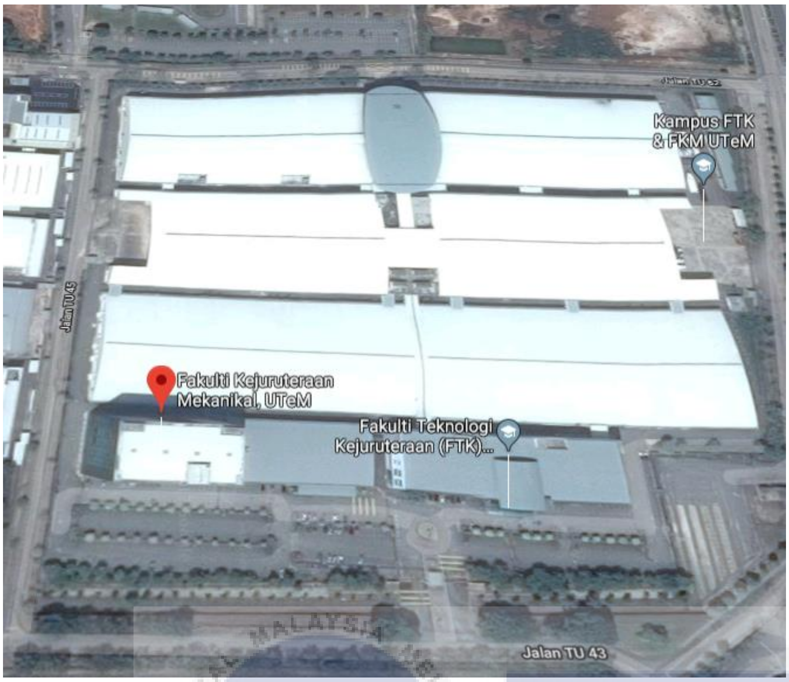


Figure 4.1: Bird view of FKM’s building

Figure 4.2: Isometric view of FKM’s building (FKM, n.d.)

4.2 Selection of suitable floor

To provide a suitable thermal comfort level to occupants in FKM building, the air-conditioning system plays an important role. In this research, ground floor and first floor were chosen as the case study floors. The reasons behind is that the floors are operating under running air-conditioning system and there are more occupants compare to other floors. The floors selected consist of lecture rooms with around 70-120 seat capacity. Since there are a lot of occupants, a suitable thermal comfort level for the both floors is extremely important.

According to the research by Imran (2016), the average air temperature and air velocity for ground floor are 23.60^{o}C and 0.091 m/s. Meanwhile, for the first floor, the average air temperature and air velocity are 23.07^{o}C and 0.091 m/s. The parameters obtained for both floors are less than the recommended range by Malaysia Standard MS 1525-2014. Hence, simulation study will be conducted to investigate how the outcome of thermal comfort changes under different parameters.

4.3 Physical measurements

To carry out this study, field data collection was conducted first to obtain desired data. Since the dimension size of the building section areas was not completed, a measuring tape was used to measure the rooms' dimension for ground floor and first floor. The length, width, and height for all the rooms in the ground floor and first floor were measured and recorded. The dimensions were required to create geometry model during CFD simulation step.

Physical measurement of air temperature and air velocity were conducted as a benchmark data in validating the CFD simulation. VelociCalc Air Velocity Meter 9545 was used to analyse the thermal comfort parameters which are air temperature and air velocity.

4.3.1 Measuring device

4.3.1.1 VelociCalc Air Velocity Meter 9545

VelociCalc Air Velocity Meter 9545 was used in this study to measure the average room air velocity. VelociCalc Air Velocity Meter 9545 has only one probe with multiple sensors. Air temperature, air velocity and relative humidity can be measured by using the air velocity meter. The technical specifications are described in Table 4.2. Figure 4.4 shows the VelociCalc Air Velocity Meter 9545 used in this study.

Table 4.2: Technical specifications of VelociCalc Air Velocity Meter 9545

Parameters	Range	Accuracy
Air velocity	0 to 30 m/s	± 0.015 m/s
Air temperature	-10 to 60°C	± 0.3°C
Relative humidity	5 to 95% RH	± 3% RH

Figure 4.4: VelociCalc Air Velocity Meter 9545 (TSI, n.d.)

4.4 Computational Fluid Dynamics (CFD)

A simulation analysis is conducted in this study. The Computational Fluid Dynamics (CFD) used in this study is Integrated Environmental Solution- Virtual Environment (IESVE). IESVE is an energy analysis and performance modeling software that is used by most engineers and architectures to design energy efficient buildings in the world. IESVE was used in this study as the medium to research the existing parameters and determined the optimum parameters to provide the best thermal comfort level. By using the correct setting, air temperature and air velocity distribution were observed. IESVE has many functions to assist in the simulation. In this study, features such as ModelIT, ApacheHVAC, Apache and MicroFlo were used. The basic function of the four features are summarized in Table 4.3.

Table 4.3: IESVE features and functions

Feature	Function
ModelIT	Construct and model building geometry.
ApacheHVAC	Model and define the HVAC equipment and control system.
Apache	Simulate energy consumption of a building.
MicroFlo	Analyse air flow and air temperature distribution in a more detail way using computational fluid dynamics (CFD) model.

4.5 Procedures

4.5.1 Procedures workflow for PSM 1

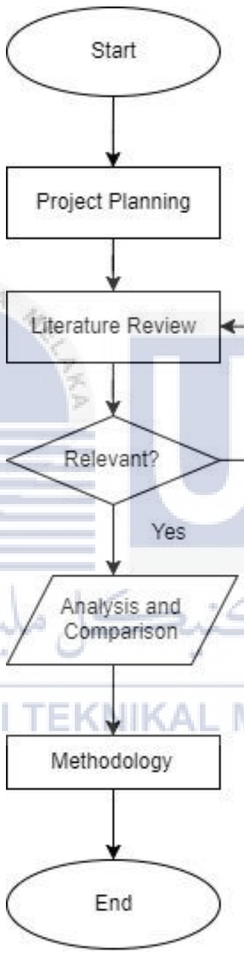
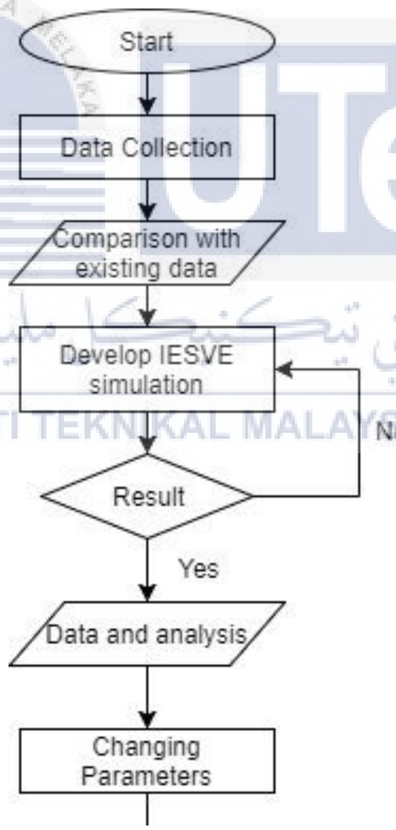


Figure 4.5: Flowchart of PSM 1

Figure 4.5 describes the flowchart of PSM 1. The procedures in carrying out PSM 1 are described as below:

1. Project planning was created so that this study can be done more systematically. A Gantt chart for the activities and timeline was included in Appendix A.
2. In literature part, relevant previous studies were identified and reviewed in order to have a better understanding regarding this study.
3. The information gains from the previous studies were then compared.
4. Suitable methodology to carry out this study in PSM 2 was proposed.

4.5.2 Procedures workflow for PSM 2



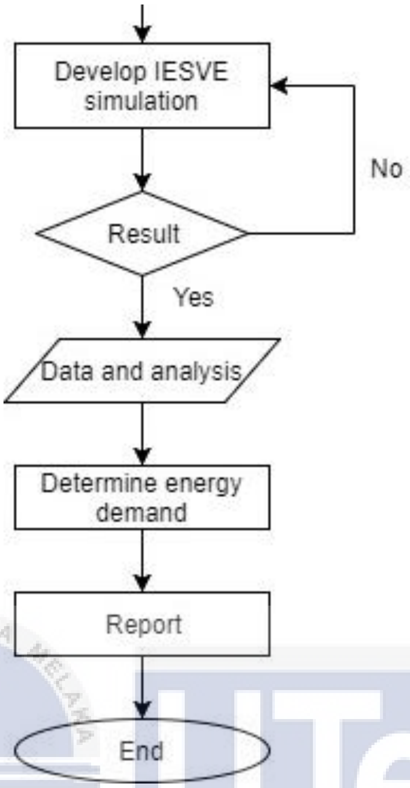


Figure 4.6: Flowchart of PSM 2

The flowchart of activities in PSM 2 was shown in Figure 4.6. The procedures in carrying out PSM 2 were described as below.

1. In data collection, the dimension of the rooms in FKM building ground floor and first floor were measured. The physical measurement of air temperature and air velocity in lecture rooms were conducted according to zone as shown in Figure 4.7 and Figure 4.8.

2. The data measured was used to compare with existing data obtained by Imran (2016). Since the percentage difference is small, the existing data was used for IESVE simulation.
3. Development of IESVE simulation. Ground floor and first floor were drawn and simulated. The geometry sketch of BK11 is shown in Figure 4.9. The red cylinder represent occupants as the heat source.
4. The simulated results were validated with the measurement data.
5. The IESVE simulation was continued by changing the parameter which were air temperature and air velocity. The results were observed and analysed.
6. The changes in energy demand for the air-conditioning system were calculated.
7. A report was written for the results.

Figure 4.7: Zone distribution for ground floor

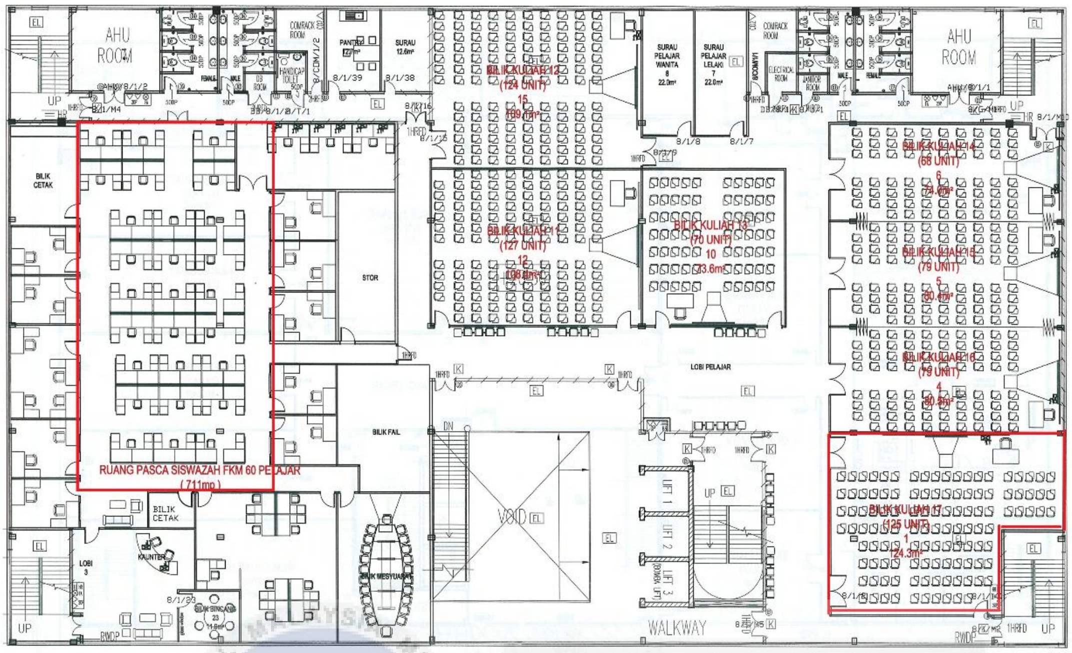


Figure 4.8: Zone distribution for first floor

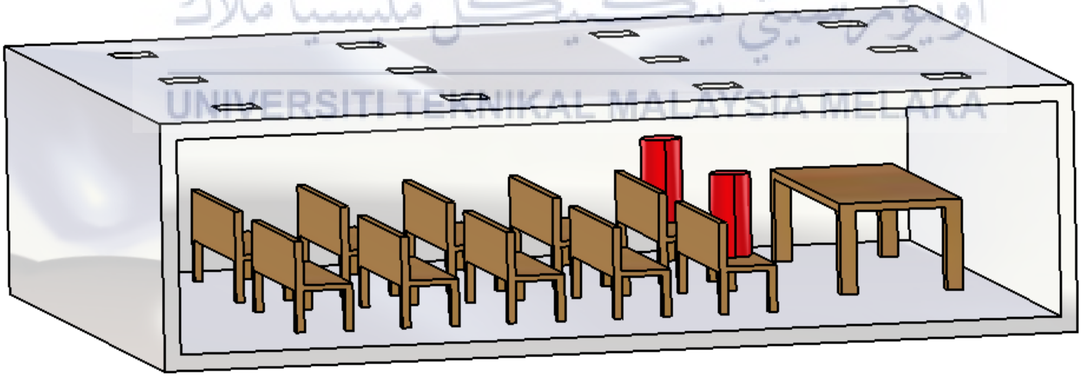


Figure 4.9: Geometry sketch of BK 11 using CATIA V5

4.6 Experiment Condition Assumption

In this study, some assumptions were made when conducting simulation as the accuracy and outcome of the simulation might affect by some external factors. The assumptions that were made were described as follow:

I. All of the air conditioning systems involved in this study are functioning well.

II. The rooms are fully enclosed without any other holes or gaps which might cause infiltration of air except the opening doors, windows and exhaust vent.

III. The chairs, tables, and occupants (heat source) are included during the physical measurement and simulation process.

IV. The electrical devices and lights are excluded as the heat source released has a minimal effect on the air temperatures.

CHAPTER 5

RESULT AND DISCUSSION

This section mainly discussed about the thermal comfort for the Faculty of Mechanical Engineering (FKM). Air temperature and air velocity are the main thermal comfort parameters that are being investigated. Physical measurement is carried out to take the actual measurements of lecture rooms. IESVE software is used to predict and simulate the ideal thermal comfort conditions of lecture rooms. Both of the physical measurement and simulation results are compared with MS 1525: 2014.

5.1 Malaysia Standard MS 1525: 2014

For a building to utilise its energy more efficiently, MS 1525: 2014 is one of the standards that can be used as a guide. It is essential to follow the recommended parameters that suggested by MS 1525 in order to provide the best thermal comfort to occupants. Both of the physical measurements and simulation results are compared with the standards recommended by MS 1525. The recommended air temperature and air velocity by MS 1525: 2014 are stated as below:

I. Recommended air temperature: 24^{o}C – 26^{o}C

II. Recommended air velocity: 0.15 m/s – 0.5 m/s

5.2 Case with No Occupants

5.2.1 Physical Measurements

The purpose to carry out these physical measurements with no occupants is to compare the real situation with ideal situation simulated by IESVE. The physical measurements were carried out at ground floor for Bilik Kuliah (BK) 4, BK 5, BK 6 and BK 8 as well as at first floor for BK 17 and postgraduate area (PA) with no occupants inside. The reason to choose BK 4, BK 5, BK 6, BK 8, BK17 and postgraduate area was because all of the 5 lecture rooms and postgraduate area have different room area. Table 4.1 shows the area of the selected lecture rooms.

The physical measurements were taken three times at different time period which were 8.30 a.m. - 10.30 a.m., 1.11 p.m. - 2.00 p.m. and 4.00 p.m. - 5.00 p.m. Before starting the measurements, the temperature was set as 20 ^{o}C and the air conditioner was running for 1 hour in order to obtain a steady air temperature distribution and air velocity flow. An air velocity metre was used to measure the parameters (air temperature and air velocity) inside the selected lecture rooms by placing the air velocity meter at the middle of the lecture rooms. The results obtained are the average of 100 data collected by air velocity meter. Table 5.1 until Table 5.7 show the results for the physical measurements in BK 4, BK 5, BK 6, BK 8, BK17 and PA

Table 5.1: The area of the selected rooms

Lecture Room	Area (m^2)
BK 4	66.4
BK 5	160.8
BK 6	113.7
BK 8	80.7
BK 17	124.3
PA	255.3

Table 5.2: Physical measurements in BK 4

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)
Results	22.9	0.04	22.5	0.02	21.4	0.06

Table 5.3: Physical measurements in BK 5

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)
Results	22.5	0.02	21.3	0.02	20.5	0.05

Table 5.4: Physical measurements in BK 6

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)
Results	21.7	0.06	21.1	0.06	19.4	0.10

Table 5.5: Physical measurements in BK 8

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)
Results	21.9	0.2	21.3	0.26	19.8	0.15

Table 5.6: Physical measurements in BK 17

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)
Results	22.0	0.14	20.7	0.09	20.3	0.07

Table 5.7: Physical measurements in PA

Time	8.30 a.m.-10.30 a.m.		1.11 p.m.- 2.00 p.m.		4.00 p.m.- 5.00 p.m.	
Parameters	T (°C)	V (m/s)	T (°C)	V (m/s)	T (°C)	V (m/s)

Results	23.2	0.09	22.2	0.07	22.3	0.05

Based on the physical measurement results from Table 5.2 to Table 5.7, it can be seen that the room temperatures in the selected lecture rooms are higher during the time period of 8.30 a.m.-10.30 a.m. The air temperature is then gradually decrease at time period of 1.11 p.m.- 2.00 p.m. and 4.00 p.m.- 5.00 p.m.. This could due to the reason that there is no any occupants inside the lecture rooms and there is no any heat energy generated. Hence, the average air temperature getting lower over time. For example, the average air temperature for BK 8 at 8.30 a.m.-10.30 a.m. is 21.9 ^oC and it decrease to 21.3 ^oC at 1.11 p.m.- 2.00 p.m. and finally decrease to 19.8 ^oC at 4.00 p.m.- 5.00 p.m.

For the air velocity, the results show that all of the selected lecture rooms have inconsistent air velocity flow. For example, the average air velocity for BK 8 at 8.30 a.m.-10.30 a.m. is 0.2 m/s and it increase to 0.26 m/s at 1.11 p.m.- 2.00 p.m. and finally decrease to 0.15 m/s at 4.00 p.m.- 5.00 p.m. The inconsistent air velocity flow is probably due to blocked or leaking ducts. Hence, in order to have a consistent air velocity, it is crucial to check the ducts to ensure that they are not blocked and leak free.

5.2.2 Simulation Results Relative to Actual Condition

This simulation is performed to compare with the actual condition. The simulation condition is considering as ideal condition as the following assumptions are made:

i. The air conditioning system in selected lecture rooms is fully functioning and running well.

ii. The rooms are fully sealed and enclosed without any holes or gaps (excluding doors, windows and exhaust vent).

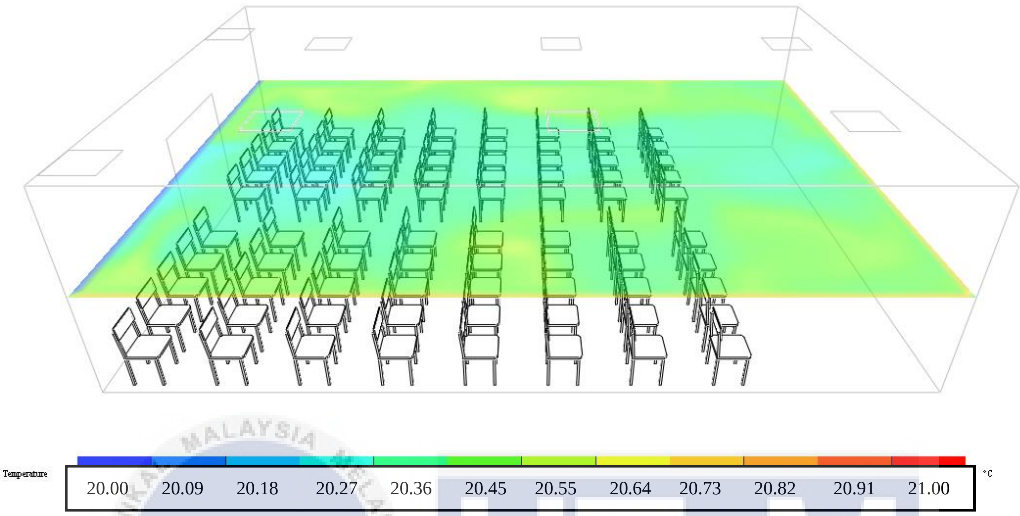
iii. The outside temperature on the surface of the room is constant.

iv. Internal heat source emitted from the digital devices and lights will be neglected due to minimal effect on the temperature.

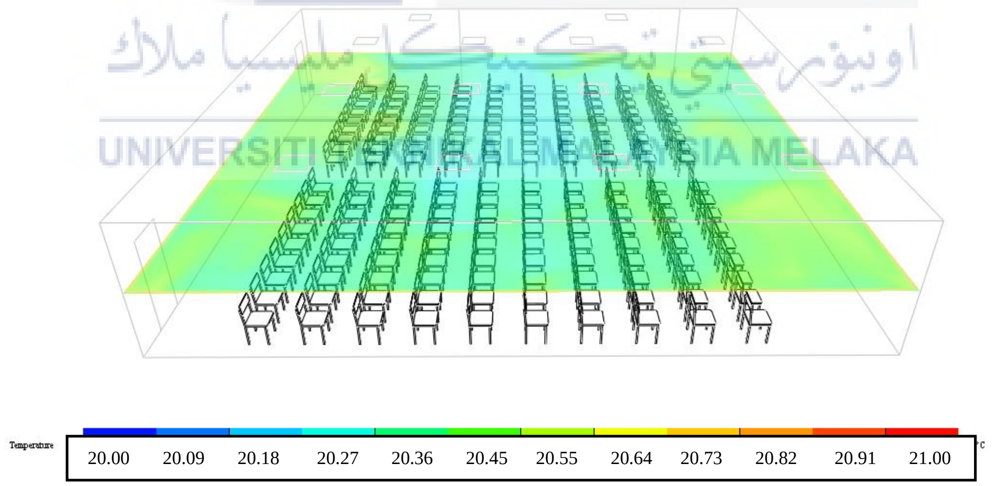
v. The furniture (chairs and tables) are included in the simulation.

Simulation results relative to actual condition when taking physical measurements are done. The temperature setting for simulation is set as 20 ^{o}C, which is identical to actual situation. This is to compare the results between the actual conditions with ideal condition without occupants. From the results, it can be known how much the deviation of actual measurement results compare to simulation results is.

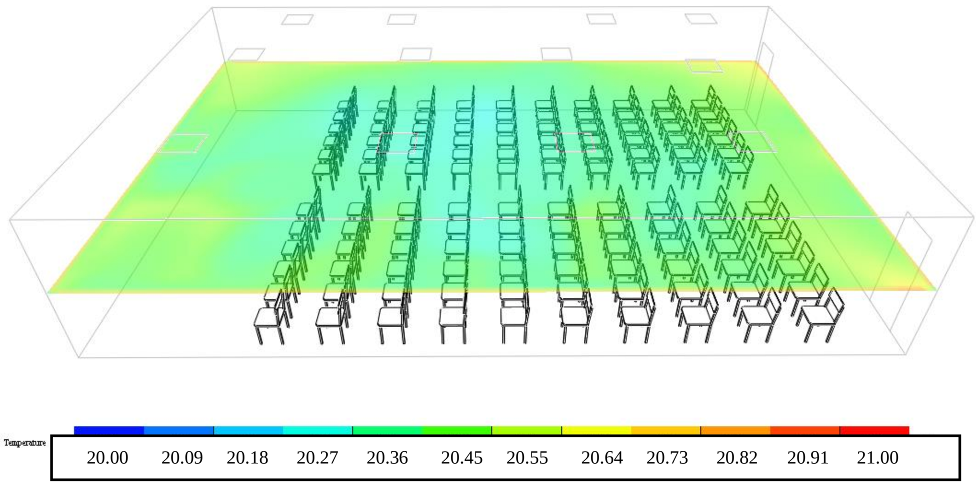
The results obtained from the simulation are shown in Figure 5.1 (a) - (f). Based on the result, it can be seen that in ideal condition, the average air temperature in every lecture rooms at the height of 1.5 m is about the same. The average air temperature at the middle of lecture room is around 20.27 ^{o}C while the average air temperature near to the wall is 20.55 ^{o}C. This means that in ideal condition, the biggest difference between the average air temperatures in lecture rooms and the air temperature coming out from diffusers is 0.55 °C. The deviation is small as there is no heat energy release to the surrounding.



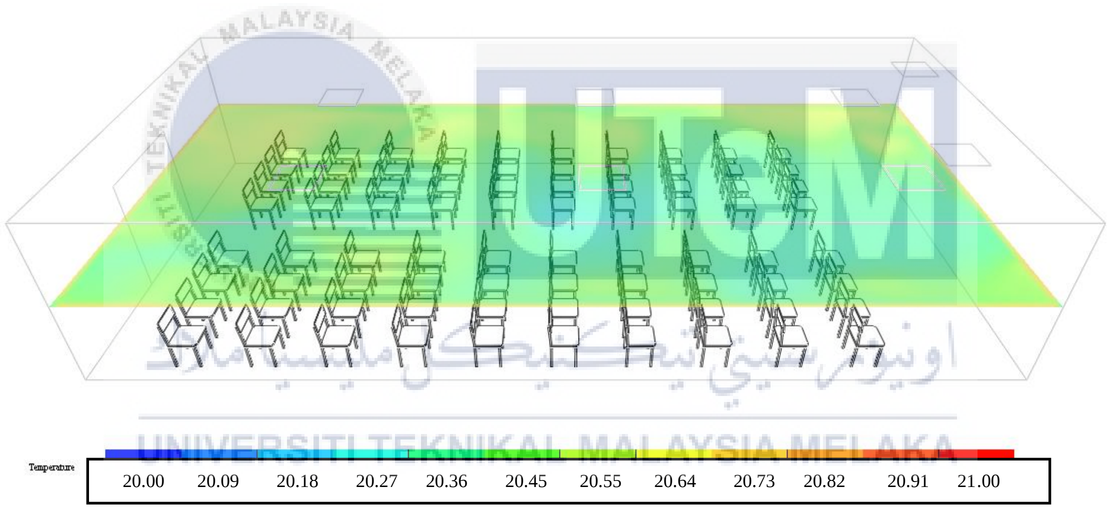
(a)



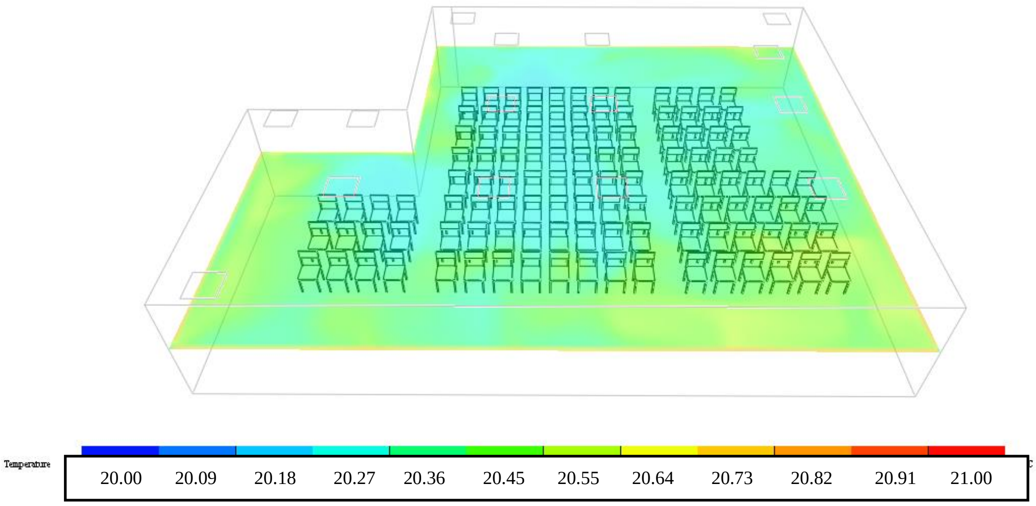
(b)


Temperature
20.00 20.09 20.18 20.27 20.36 20.45 20.55 20.64 20.73 20.82 20.91 21.00

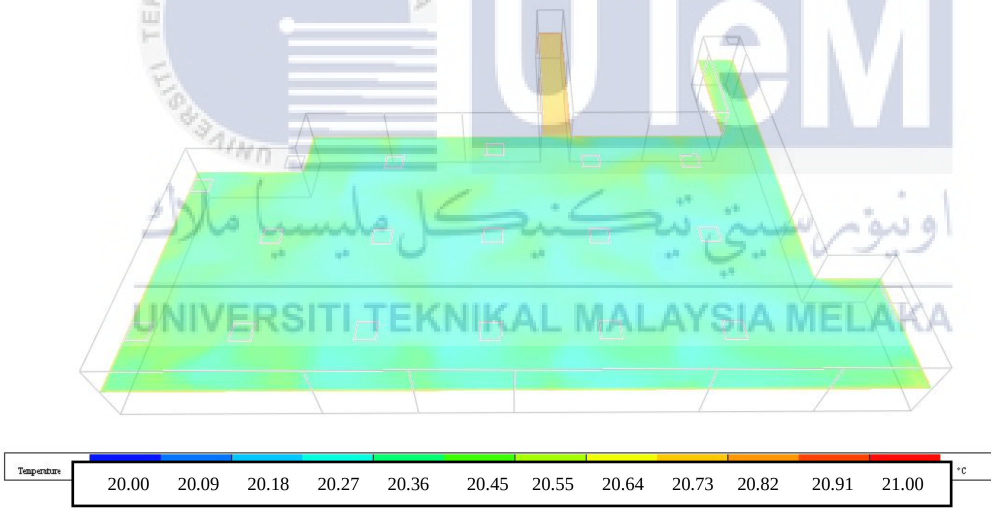
(c)


Temperature
20.00 20.09 20.18 20.27 20.36 20.45 20.55 20.64 20.73 20.82 20.91 21.00

(d)



(e)



(f)

Figure 5.1: Simulation results of air temperature distribution relative to actual condition for (a) BK 4, (b) BK 5 (c) BK 6 (d) BK 8 (e) BK 17 and (f) PA

The physical measurements data are then compared with the simulation results. Table 5.8 shows the difference between the actual air temperature in lecture rooms at 1.5m height compare to the air temperature coming out from diffusers. It can be seen that the largest difference between the actual air temperature is at PA with 3.2 °C difference at time period of 8.30 a.m.-10.30 a.m. The smallest difference is at BK 8 with -0.02 °C differences at time period of 4.00 p.m.- 5.00 p.m.

PA has the largest air temperature difference which could due to the room area. PA has the largest room area compare to other lecture rooms. Hence, it required a longer time to cool the air temperature and the air temperature drop is smaller.

Table 5.8: Differences between actual air temperature and ideal air temperature comparative to the diffusers air temperature.

BK	Time	Actual Air Temperature (°C)	Diffusers Air Temperature (°C)	Actual Air Temperature difference (°C)	Ideal Air Temperature Difference (°C)
4	8.30 a.m.-10.30 a.m.	22.9	20	2.9	0.27
	1.11 p.m.-2.00 p.m.	22.5	20	2.5	
	4.00 p.m.-5.00 p.m.	21.4	20	1.4	
5	8.30 a.m.-10.30 a.m.	22.5	20	2.5	0.27
	1.11 p.m.-2.00 p.m.	21.3	20	1.3	

	4.00 p.m.- 5.00 p.m.	20.5	20	0.5	
6	8.30 a.m.- 10.30 a.m.	21.7	20	1.7	0.27
	1.11 p.m.- 2.00 p.m.	21.1	20	1.1	
	4.00 p.m.- 5.00 p.m.	19.4	20	-0.06	
8	8.30 a.m.- 10.30 a.m.	21.9	20	1.9	0.27
	1.11 p.m.- 2.00 p.m.	21.3	20	1.3	
	4.00 p.m.- 5.00 p.m.	19.8	20	-0.02	
17	8.30 a.m.- 10.30 a.m.	22.0	20	2.0	0.27
	1.11 p.m.- 2.00 p.m.	20.7	20	0.7	
	4.00 p.m.- 5.00 p.m.	20.3	20	0.3	
PA	8.30 a.m.- 10.30 a.m.	23.2	20	3.2	0.27
	1.11 p.m.- 2.00 p.m.	22.2	20	2.2	
	4.00 p.m.- 5.00 p.m.	22.3	20	2.3	

5.2.3 Simulation Result for Air Velocity Based On Ideal Condition

According to MS 1525: 2014, the recommended average air velocity inside a building is 0.15 m/s – 0.50 m/s. In order to achieve the recommended standards, simulation on the lecture rooms with no occupants are done. This is to determine the outcome of thermal comfort changes under different circumstances and parameters as well as determine the suitable setting which can provides the best thermal comfort.

In doing so, the different parameters are tested during the simulation. For the air velocity from the diffusers, 3m/s, 4m/s, 5m/s and 6m/s are tested. After several tries, it was found that air velocity with 5m/s and 6m/s are too high and at the height of 1.5m. For example, as shown in figure 5.2, the average air velocity for BK 4 at 1.5m height is 0.68 m/s, which is beyond the recommended range by MS 1525: 2014. Hence, 3m/s diffusers air velocity is selected for BK 4, BK 6, BK 8 and BK 17 while 4m/s diffusers air velocity is selected for BK 5 and PA.

This is because after several attempts on the simulations, it was found that 4.0 m/s is more suitable for BK 5 and PA as both of the rooms have a bigger room area in order to achieve the recommended air velocity by MS 1525 which is between 0.15 m/s – 0.50 m/s. The other lecture rooms has a smaller room area, hence, 3m/s is enough to achieve a suitable thermal comfort.

Figure 5.3 (a) – (f) show the average air velocity for BK 4, BK 5, BK 6, BK 8, BK17 and PA at 1.5m height. It can be observed that the average air velocity for all of the lecture rooms are around 0.27 m/s without occupants inside, which is within the recommended range by MS 1525: 2014.

Hence, it can be concluded that the suitable setting for air velocity for BK 4, BK 6, BK 8 and BK 17 are 3m/s while for BK 5 and PA are 4m/s.

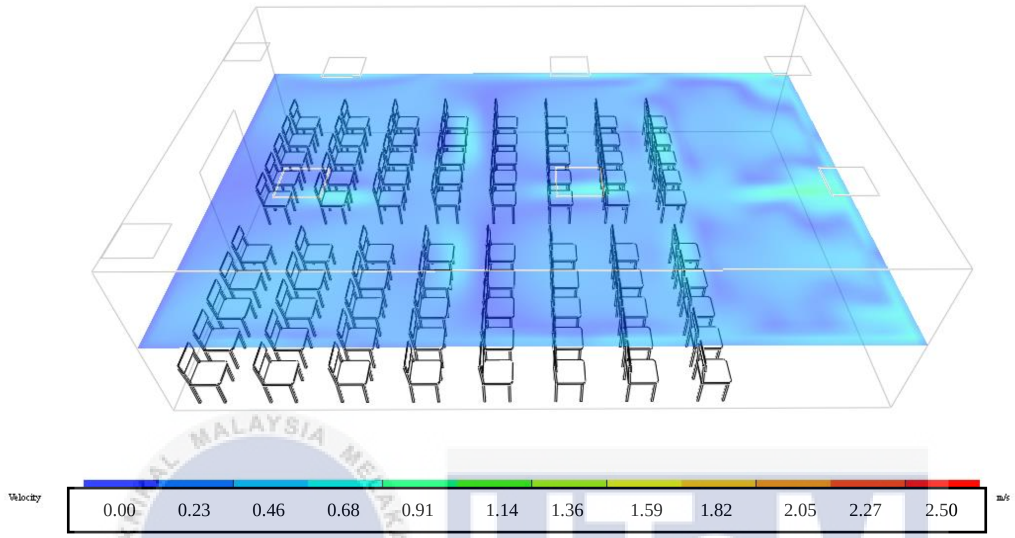
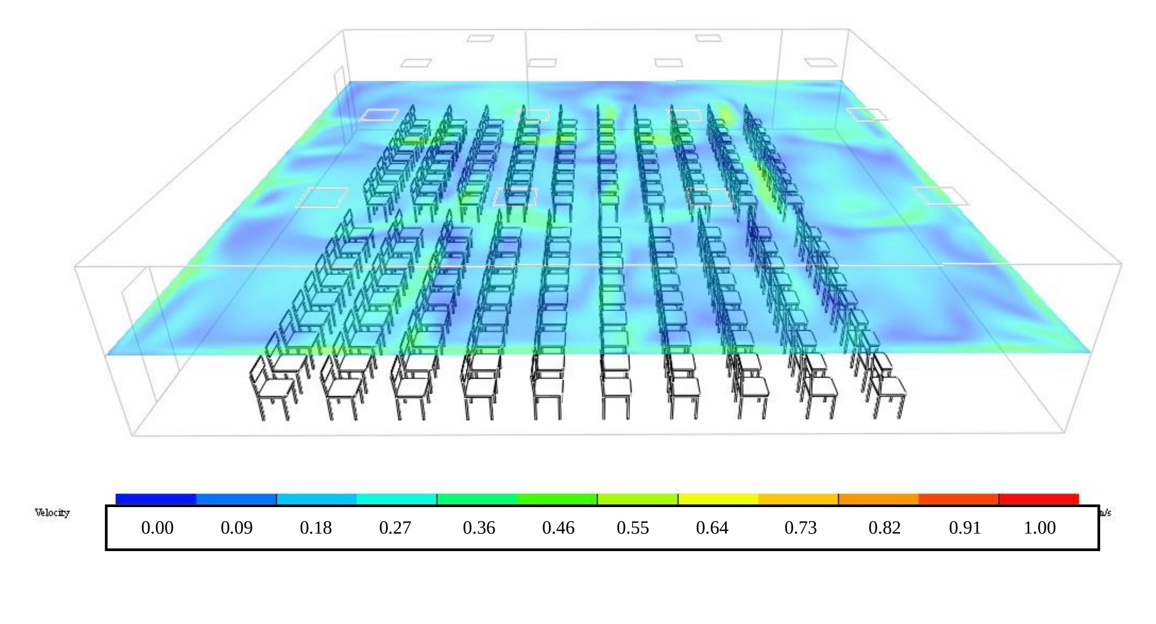
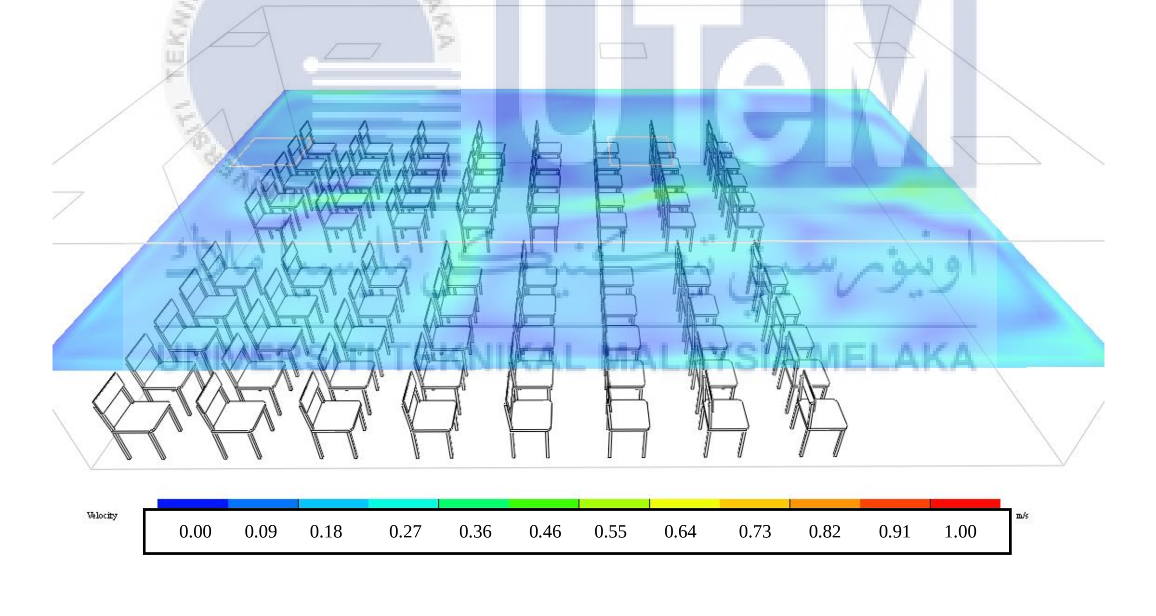


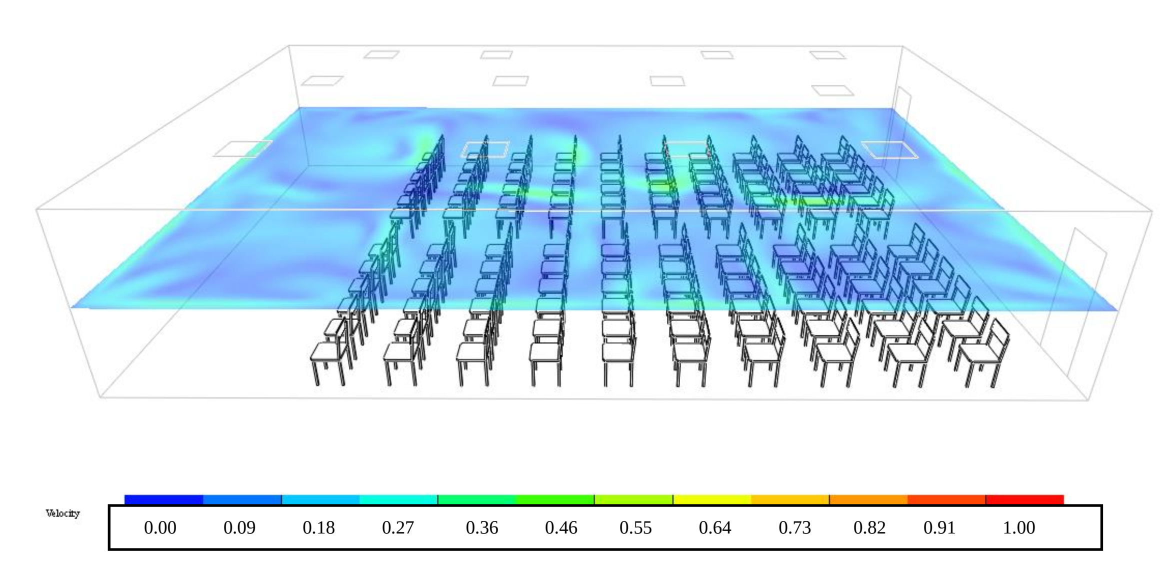
Figure 5.2: Average air velocity for BK 4 at 1.5m height with 6m/s diffusers air velocity.


Velocity
0.00 0.09 0.18 0.27 0.36 0.46 0.55 0.64 0.73 0.82 0.91 1.00
m/s

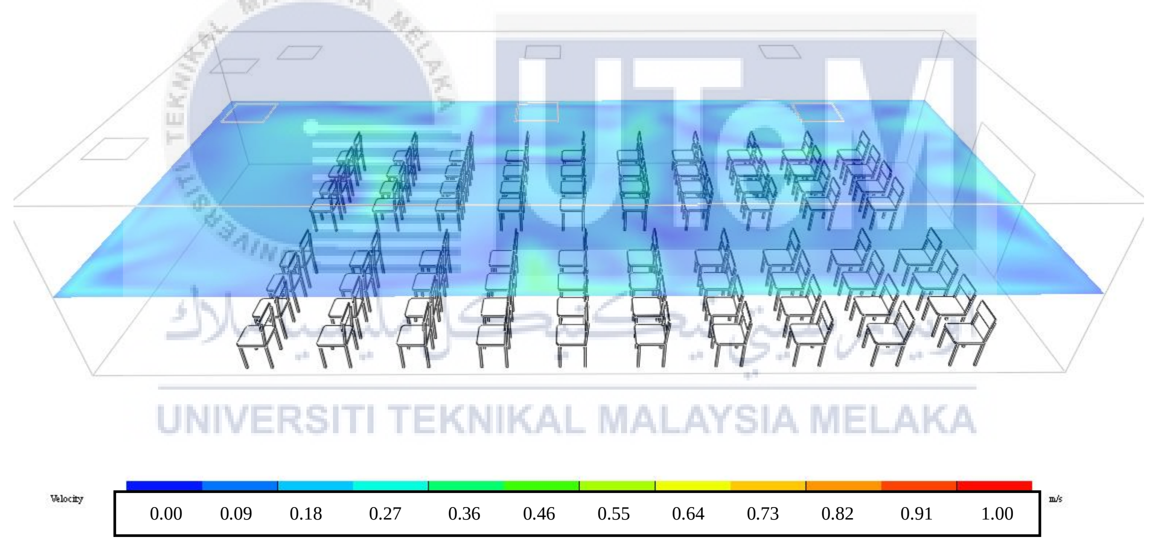
(a)


Velocity
0.00 0.09 0.18 0.27 0.36 0.46 0.55 0.64 0.73 0.82 0.91 1.00
m/s

(b)


Velocity
0.00 0.09 0.18 0.27 0.36 0.46 0.55 0.64 0.73 0.82 0.91 1.00

(c)


Velocity
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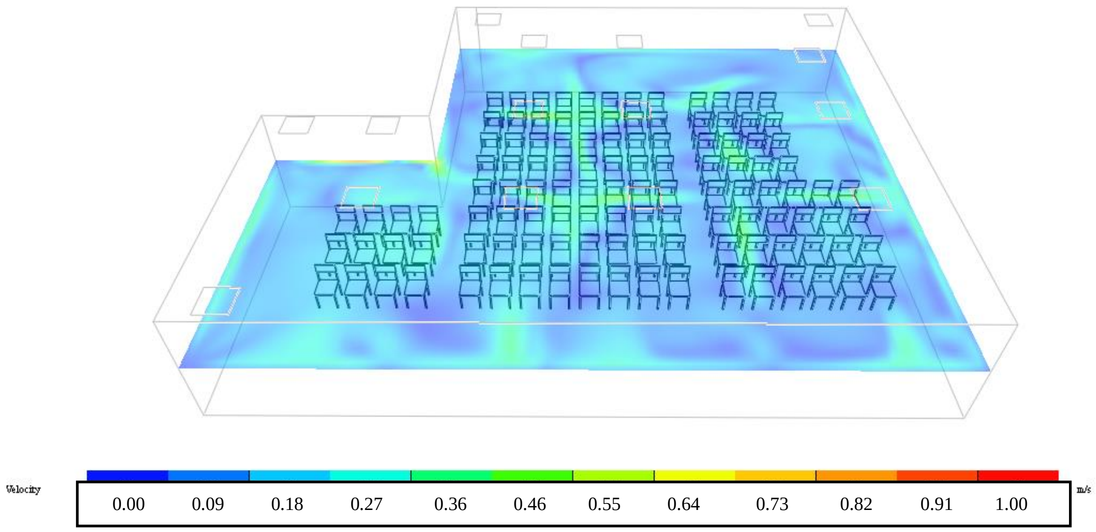
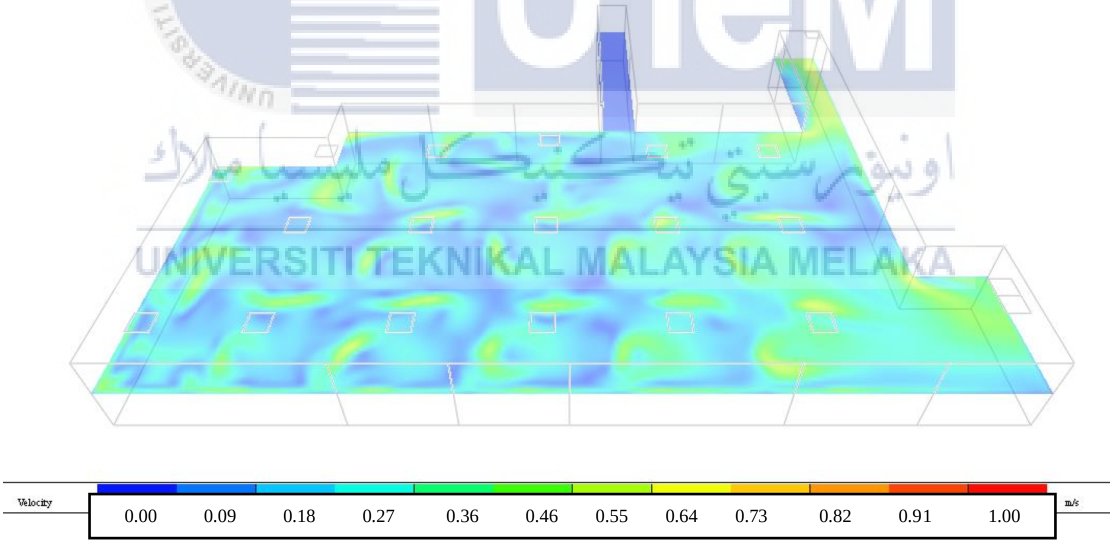
m/s

(d)



(e)



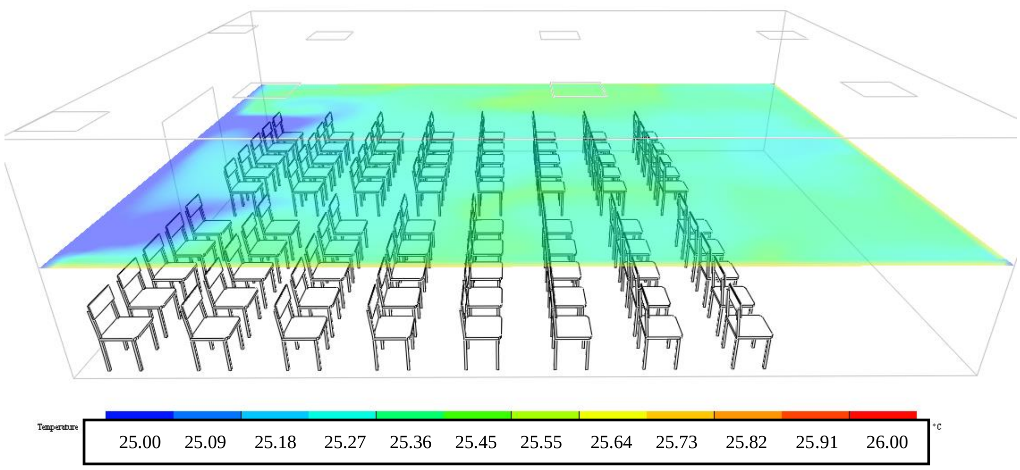
(f)

Figure 5.3: The average air velocity for (a) BK 4 (b) BK 5 (c) BK 6 (d) BK 8 (e) BK 17 and (f) PA (without occupants) at 1.5m height

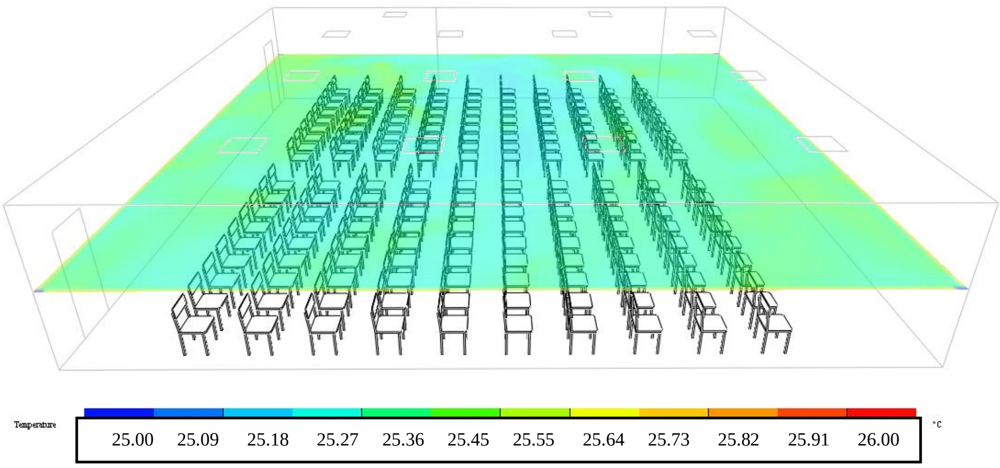
5.2.4 Simulation Result for Air Temperature Based On Ideal Condition

According to MS 1525: 2014, the recommended average air temperature inside a building is 24°C – 26°C. The simulations without occupants for the selected lecture rooms (BK 4, BK 5, BK 6, BK 8, BK 17 and PA) under perfect condition are done by using IES-VE software.

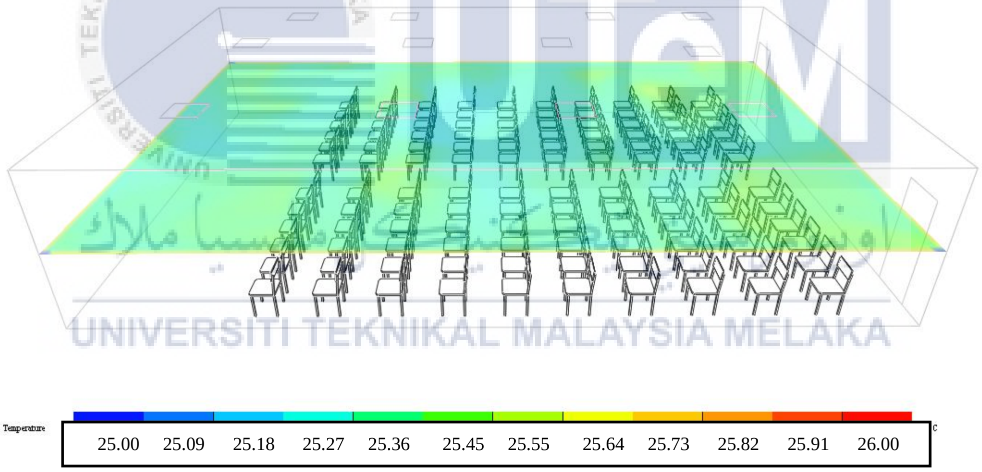
In order to obtain the suitable thermal comfort condition in lecture rooms without occupants, several tries are done. The selected lecture rooms are simulated by using the air temperature setting of 24 °C, 25 °C and 26 °C. The results show that the suitable setting for the selected lecture rooms are 25°C. The average air temperature for all of the lecture rooms without occupants is 25.30 °C at 1.5m height as shown in Figure 5.4 (a) – (f), which is within the range of recommended air temperature by MS 1525: 2014.



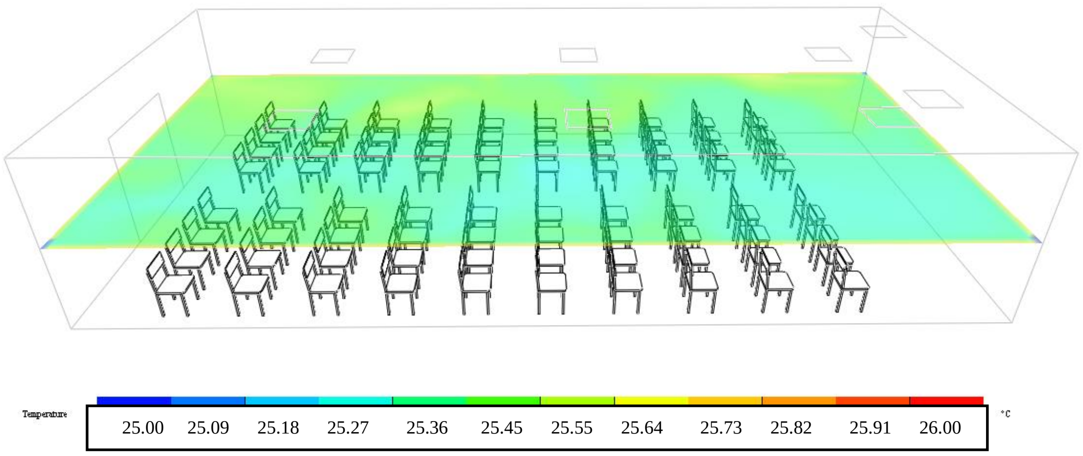
(a)


Temperature
25.00 25.09 25.18 25.27 25.36 25.45 25.55 25.64 25.73 25.82 25.91 26.00
°C

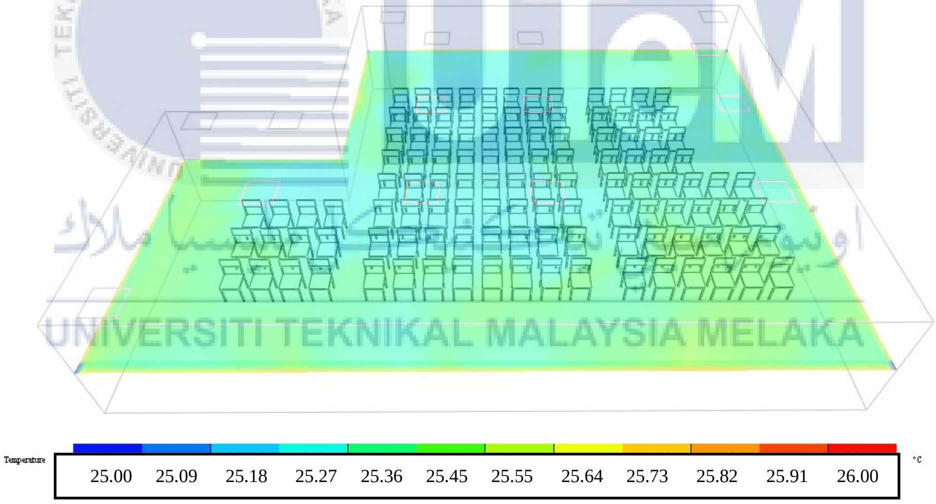
(b)


Temperature
25.00 25.09 25.18 25.27 25.36 25.45 25.55 25.64 25.73 25.82 25.91 26.00

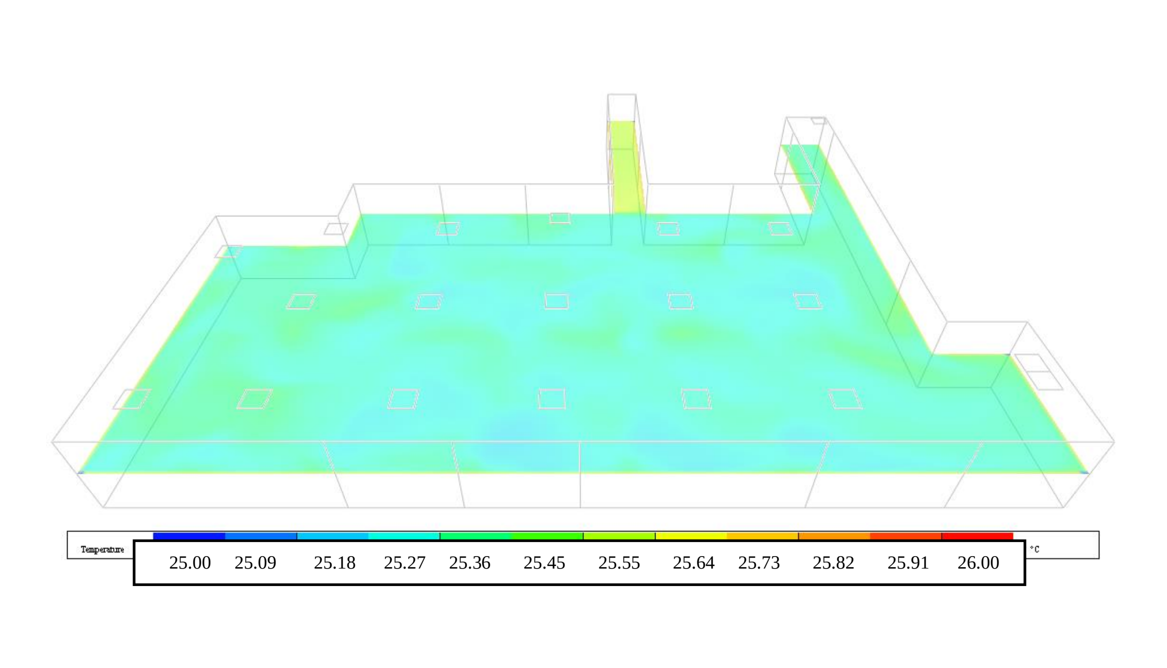
(c)


Temperature
25.00 25.09 25.18 25.27 25.36 25.45 25.55 25.64 25.73 25.82 25.91 26.00
°C

(d)


Temperature
25.00 25.09 25.18 25.27 25.36 25.45 25.55 25.64 25.73 25.82 25.91 26.00
°C

(e)



(f)

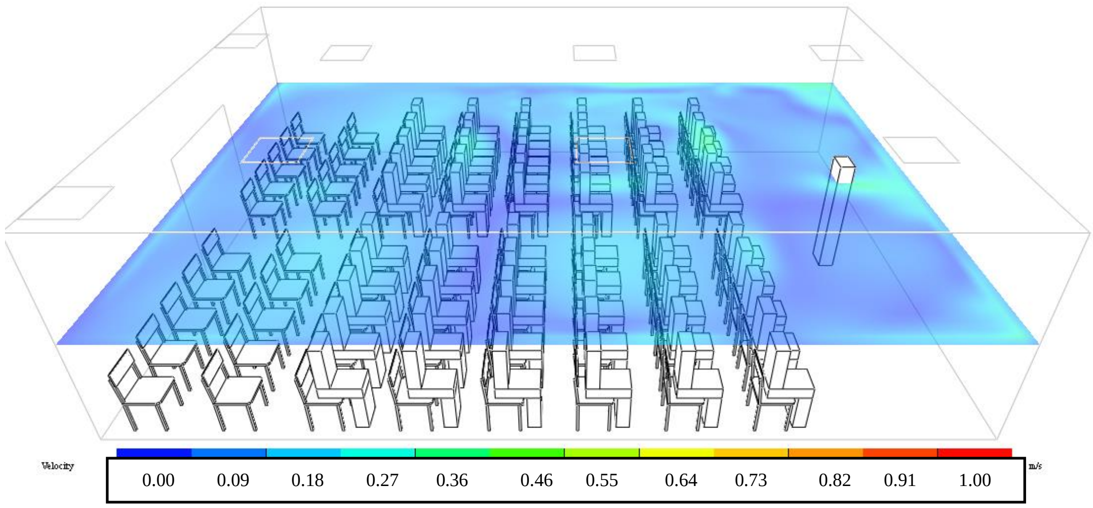
Figure 5.4: Average air temperature for (a) BK 4 (b) BK 5 (c) BK 6 (d) BK 8 (e) BK 17 (f) PA (without occupants) at 1.5m height

5.3 Case with Occupants

5.3.1 Simulation Result for Air Velocity Based On Ideal Condition

For the case with occupants, the simulation results of air velocity for the selected lecture rooms with occupants show no significant difference from the simulation results with no occupants. The air velocity setting used in with occupants case is the same with the setting used in without occupants case in which BK 4, BK 6, BK 8 and BK 17 use 3 m/s setting while BK 5 and PA uses 4 m/s setting. The average air velocity remain unchange, which is around 0.27 m/s as shown in Figure 5.5 (a) – (f).

Only a few spots in the room will have an air velocity of 0.46 m/s and it is still within the recommended air velocity by MS1525. The air flow in the rooms are steady and the average air velocity is 0.27 m/s. Hence, it can be concluded that the present of occupants do not have significant effect on the average air velocity. The suitable air velocity setting for rooms with occpuants will be the same as the case with non occupants.


Velocity
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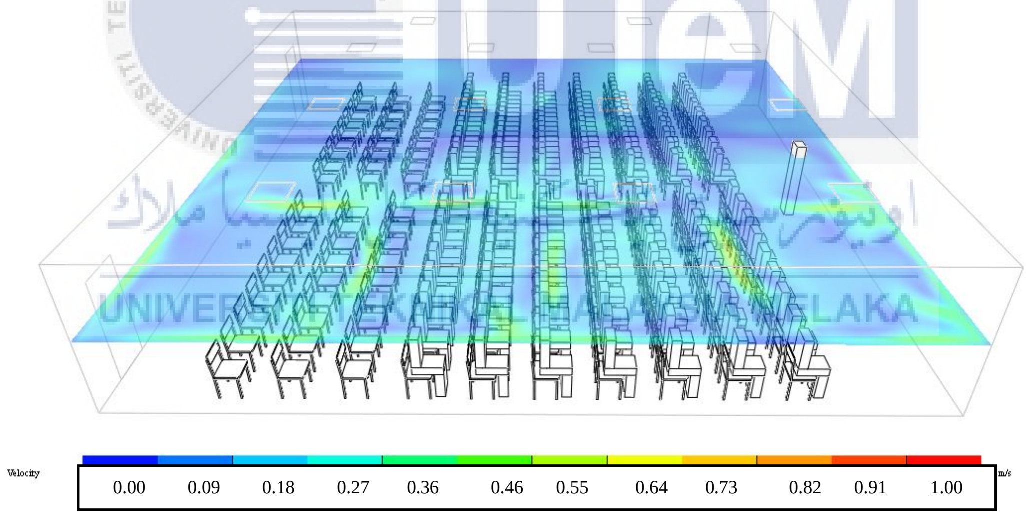
m/s

(a)


Velocity
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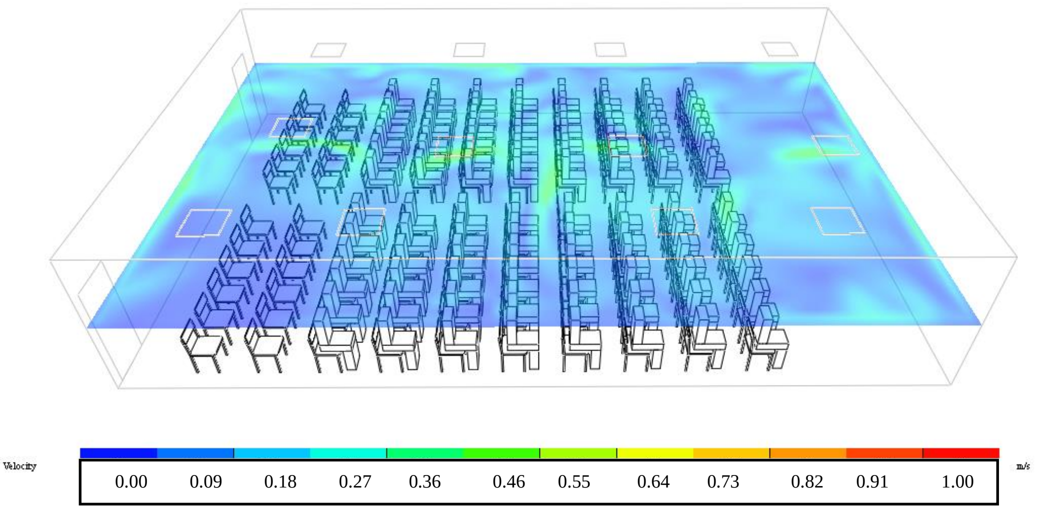
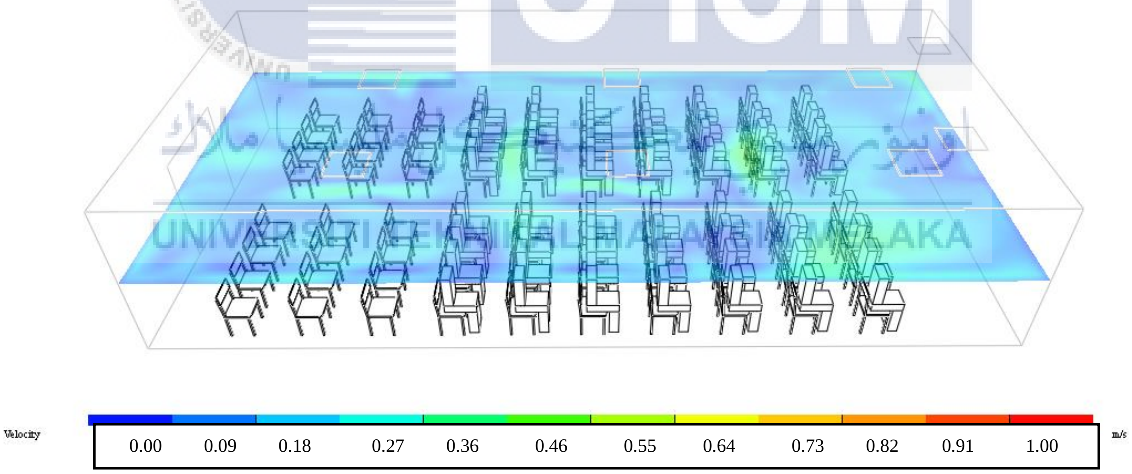
m/s

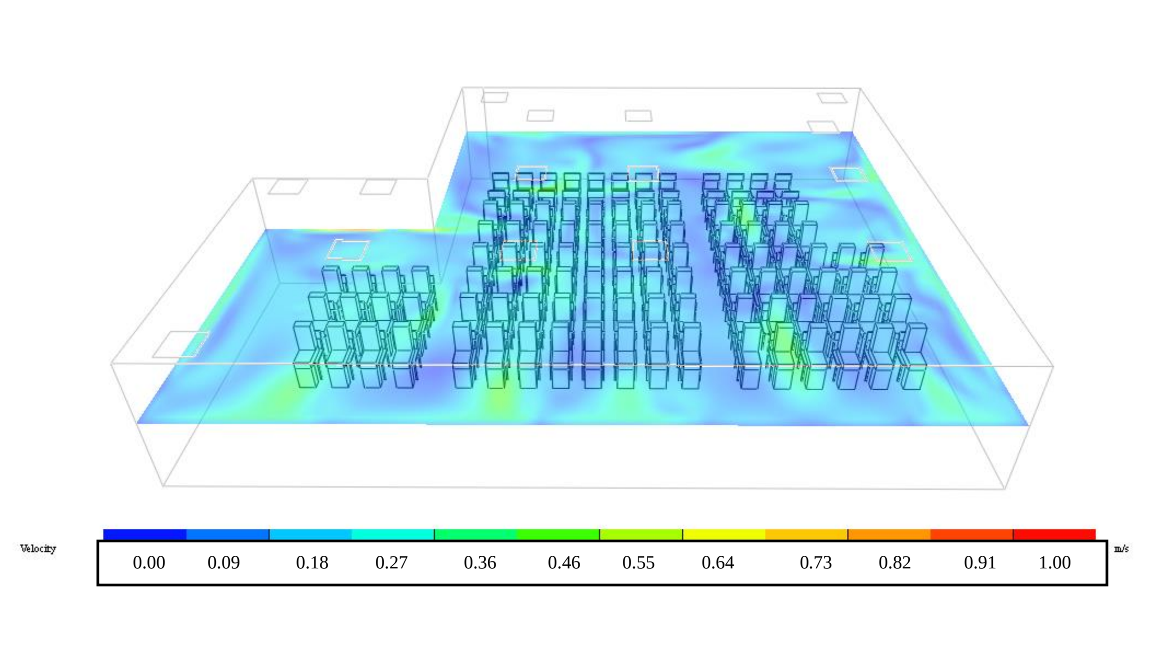
(b)


Velocity
0.00 0.09 0.18 0.27 0.36 0.46 0.55 0.64 0.73 0.82 0.91 1.00
m/s

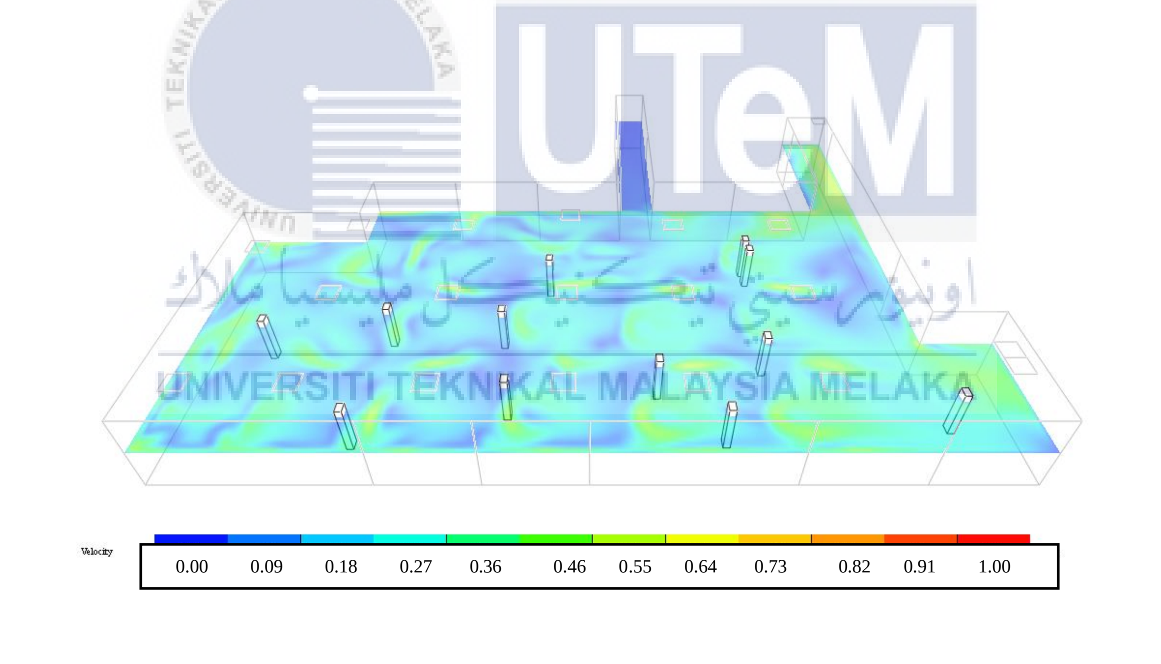
(c)


Velocity
0.00 0.09 0.18 0.27 0.36 0.46 0.55 0.64 0.73 0.82 0.91 1.00
m/s

(d)



(e)



(f)

Figure 5.5: Average air velocity for (a) BK 4 (b) BK 5 (c) BK 6 (d) BK 8 (e) BK 17 (f) PA (with occupants) at 1.5m height

5.3.2 Simulation Result for Air Temperature Based On Ideal Condition

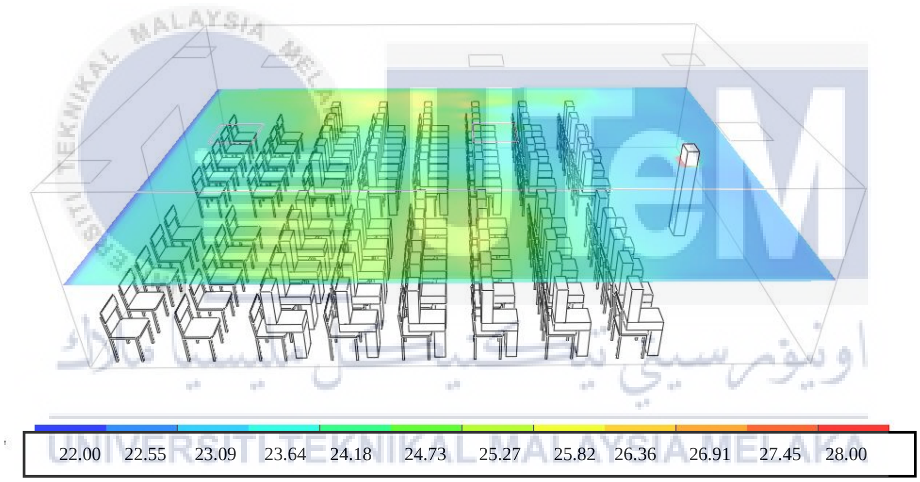
The simulations with the present of occupants under perfect condition are done for the selected lecture rooms. This is to study the impact of occupants towards the selected lecture rooms' thermal comfort. As the heat generated by occupants will raise the average air temperature in the lecture rooms, it is crucial to determine the suitable temperature setting of diffusers in order to achieve the acceptable thermal comfort.

The results are shown in Table 5.9 and Figure 5.6 (a) – (f). Based on the simulation results, the suitable temperature setting for BK 4 and BK 8 are 22 ^oC, for BK 5, BK 6 and BK 17 are 21 ^oC and for PA is 24 ^oC. The average air temperature for BK 4, BK 5, BK 6, BK 8, BK 17 and PA are 25.30 ^oC, 24.82 ^oC, 24.80 ^oC, 25.10 ^oC, 24.82 ^oC and 24.45 ^oC respectively.

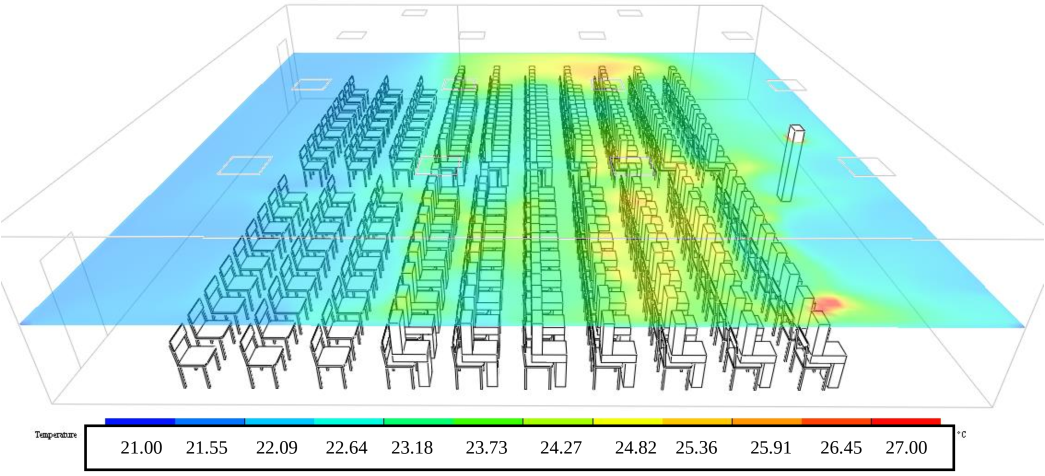
Table 5.9: The suitable temperature setting and average air temperature of selected lecture rooms

BK	Suitable Temperature Setting (^oC)	Average Air Temperature (^oC)
4	22	25.30
5	21	24.82
6	21	24.80
8	22	25.10
17	21	24.82
PA	24	24.45

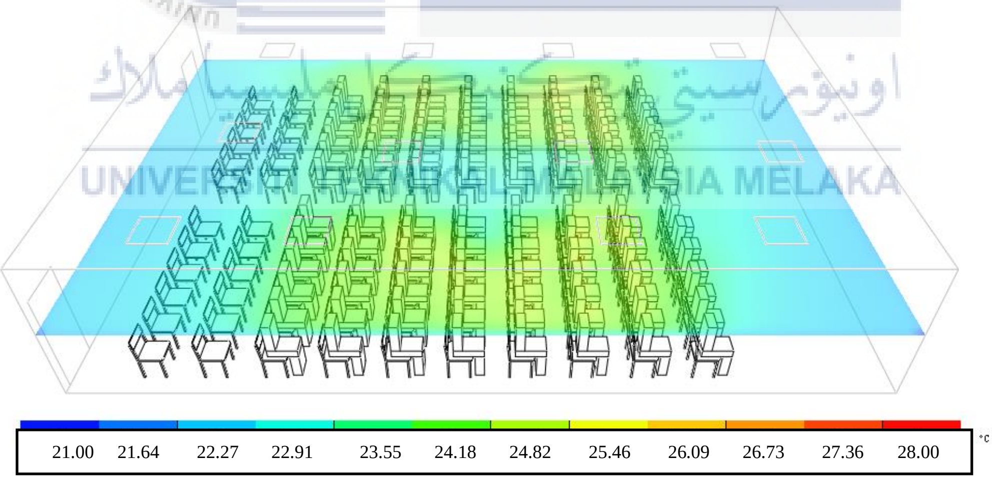
The reason to use 21 ^{o}C for BK 5, BK 6 and BK 17 is due to the lecture rooms have a bigger room area compare to BK 4 and BK 8. The number of occupants allowed in BK 5, BK 6 and BK 17 are more compared to BK 4 and BK 8. The total heat generated by greater number of occupants is larger as well. As the heat generated increase, the cooling load increase as well. Hence, it required a lower temperature setting in order to provide a suitable average air temperature. Meanwhile for the PA, the number of master students inside PA is not many. Hence, 24 ^{o}C is enough to provide the acceptable thermal comfort condition.



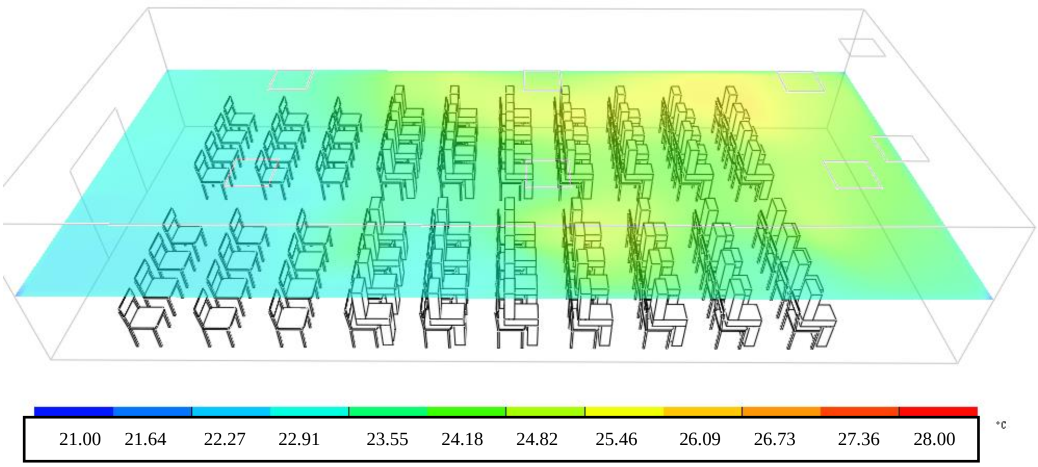
(a)


Temperature
21.00 21.55 22.09 22.64 23.18 23.73 24.27 24.82 25.36 25.91 26.45 27.00
°C

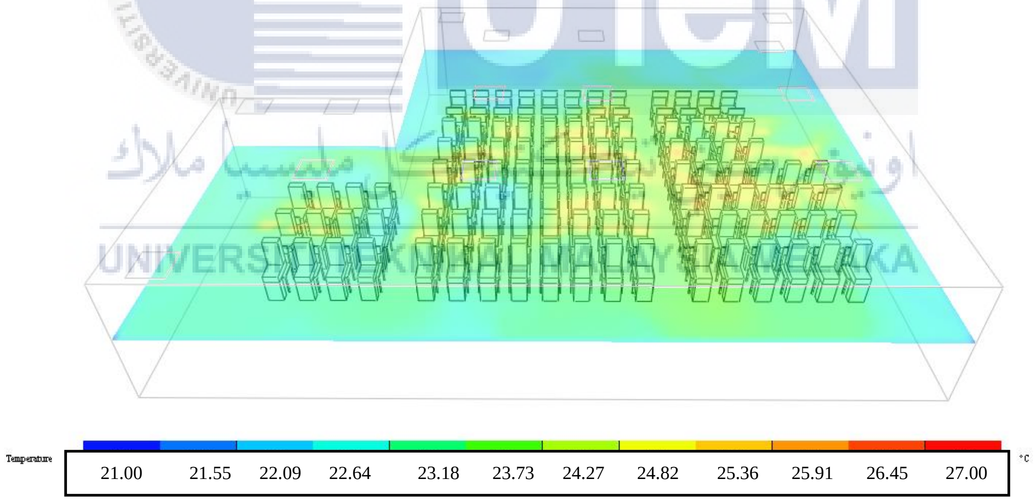
(b)


21.00 21.64 22.27 22.91 23.55 24.18 24.82 25.46 26.09 26.73 27.36 28.00
°C

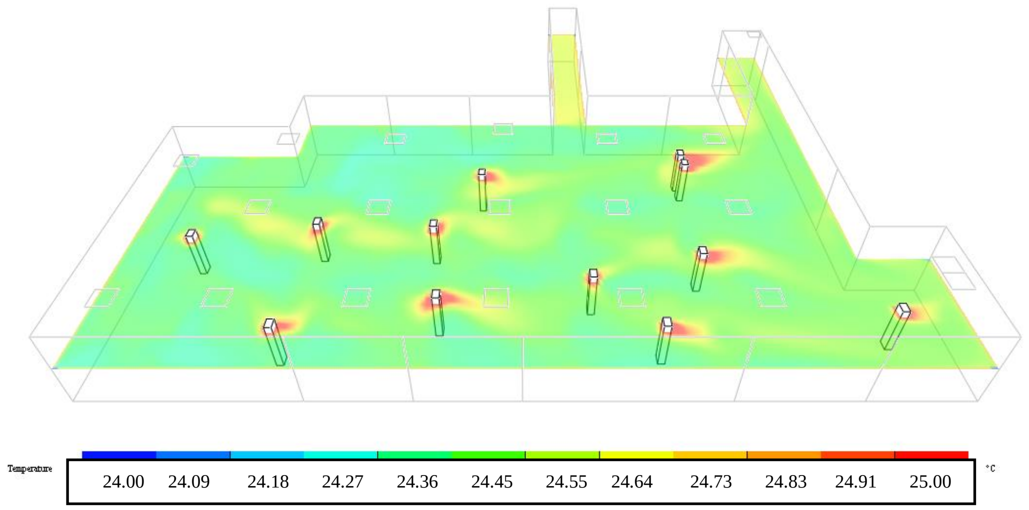
(c)


21.00 21.64 22.27 22.91 23.55 24.18 24.82 25.46 26.09 26.73 27.36 28.00
°C

(d)


Temperature
21.00 21.55 22.09 22.64 23.18 23.73 24.27 24.82 25.36 25.91 26.45 27.00
°C

(e)



(f)

Figure 5.6: Average air temperature for (a) BK 4 (b) BK 5 (c) BK 6 (d) BK 8 (with occupants) at 1.5m height

5.4 Comparison between Existing Result, Simulation Results without Occupants and Simulation Results with Occupants

The simulations results without occupants and with occupants are compared with existing results taken by Imran (2014). A pilot test was done to measure the air temperature in selected lecture rooms with occupants. As there is no large significant difference between the measured air temperature and existing results, the existing results are accepted and compared with simulation results. All of the results are then evaluated whether comply with MS 1525: 2014 or not.

5.4.1 Air Velocity

The recommended air velocity range by MS 1525: 2014 is 0.15 m/s – 0.50 m/s. Table 5.10 shows the comparison between existing results and simulation results with occupants and without occupants for the average air temperature.

The existing result taken by Imran (2014) shows that the average air velocity for the selected lecture rooms mostly do not comply with MS1525: 2014. The only lecture rooms which comply to the standard is BK 17 with 0.169 m/s. BK 5 and PA have the lowest average air velocity which is 0.063 m/s out of the 6 selected rooms. The average air velocity for both BK 6 and BK 8 are 0.069 m/s and 0.084 m/s respectively. BK 4 has an average air velocity of 0.095 m/s.

Oppositely, the simulation results for average air velocity for the selected lecture rooms are within the recommended range by MS 1525: 2014. For both of the cases (without occupants and with occupants), the suitable setting for air velocity are the same. BK 4, BK 6, BK 8 and BK 17 use 3 m/s while BK 5 and PA uses 4 m/s as the most suitable air velocity setting. The average air velocity for the selected lecture rooms at 1.5m height are 0.27 m/s for both cases.

It can be known that with the suitable air velocity setting, the occupants will feel more comfortable staying inside the rooms. It is crucial to determine the suitable air velocity setting inside a room in order to provide the best thermal comfort to occupants. The occupants will feel uncomfortable if the air velocity is too low or too high.

Table 5.10: The comparison between existing results and simulation results for average air velocity (Without occupants and with occupants)

BK	Existing Result (°C)	Comply With MS 1525	Simulation Results (without Occupants) (°C)	Comply With MS 1525	Simulation Results (With Occupants) (°C)	Comply With MS 1525
4	0.095	NO	0.27	YES	0.27	YES
5	0.063	NO	0.27	YES	0.27	YES
6	0.069	NO	0.27	YES	0.27	YES
8	0.084	NO	0.27	YES	0.27	YES
17	0.169	YES	0.27	YES	0.27	YES
PA	0.063	NO	0.27	YES	0.27	YES

5.4.2 Air Temperature

The recommended air temperature range by MS 1525: 2014 is 24 °C – 26 °C. Table 5.11 shows the comparison between existing results and simulation results with occupants and without occupants for the average air temperature.

Based on the existing result, it can be seen that only BK 8 with 25.95°C average air temperature is comply with MS 1525: 2014. It is just 0.05 °C away to reach the limit of the recommended air temperature. The other three lecture rooms which are BK 4 (23.92 °C), BK 5 (23.60 °C) and BK 6 (23.67°C) are not within the recommended range stated by MS 1525: 2014.

On the other hand, the average air temperature simulation results for both cases which are without occupants and with occupants are comply with MS 1525: 2014. By using the suitable temperature setting discussed (BK 4- 22 ^{o}C, BK 5- 21 ^{o}C, BK 6- 21 ^{o}C, BK 8- 22 ^{o}C, BK 17- 21 ^{o}C and PA- 24 ^{o}C), the average air temperature in the selected lecture rooms are within the recommended standard by MS 1525: 2014.

It can be concluded the suitable temperature settings are able to provide a suitable thermal comfort to the occupants in the lecture rooms in which all of them are able to provide average air temperature which comply with MS1525: 2014. The average air temperature lower than the recommended standard might cause the occupants to feel slightly cold and uncomfortable. Therefore, it is vital to maintain the average air temperature within the standard so that occupants will feel comfortable staying in a room.

Table 5.11: The comparison between existing results and simulation results for average air temperature (Without occupants and with occupants)

BK	Existing Result (^{o}C)	Comply With MS 1525	Simulation Results (without Occupants) (^{o}C)	Comply With MS 1525	Simulation Results (With Occupants) (^{o}C)	Comply With MS 1525
4	23.92	NO	25.30	YES	25.30	YES
5	23.60	NO	25.30	YES	24.82	YES
6	23.67	NO	25.30	YES	24.80	YES
8	25.95	YES	25.30	YES	25.10	YES
17	23.61	NO	25.30	YES	24.82	YES
PA	23.82	NO	25.30	YES	24.45	YES

5.5 ENERGY CONSUMPTION

The energy consumption is analysed for the ground floor and first floor of FKM building using IES-VE software. After the suitable setting for both air velocity and air temperature are determined, it is important to determine the changes in the energy consumption, whether it will increase or decrease. The total energy consumption for the ground floor is then compared with the existing data and with the Malaysia Standard.

5.5.1 ApacheHVAC and Apache

The energy consumption estimation is made using IES-VE software. The function used in IES-VE are ApacheHVAC module and Apache module. These two functions are able to estimate the energy consumption of a building. ApacheHVAC function is used to model and define the HVAC equipment and control system. A simple HVAC model is created using ApacheHVAC function. All of the energy consumption by the inlet fan, outlet fan and cooling coil can be defined. Figure 5.7 shows the ApacheHVAC interface, figure 5.8 shows the HVAC equipment model used and table 5.12 shows the details of the HVAC model.

Apache function is then used to simulate the energy consumption. The HVAC equipment model created in ApacheHVAC is input into the Apache simulation.

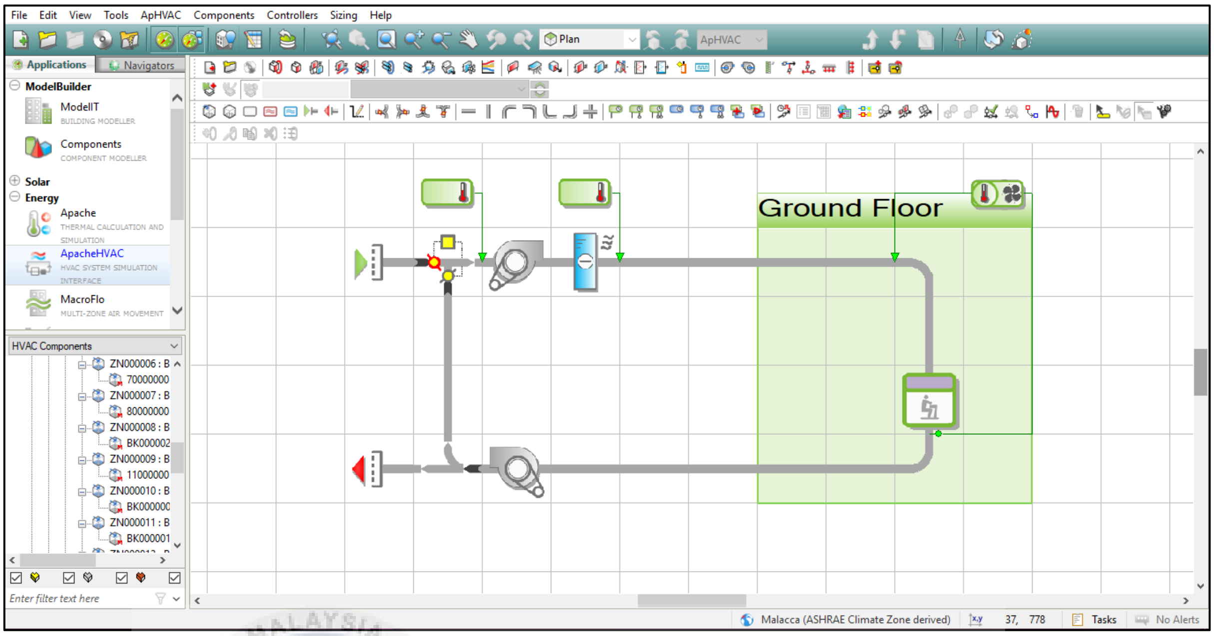


Figure 5.7: ApacheHVAC interface

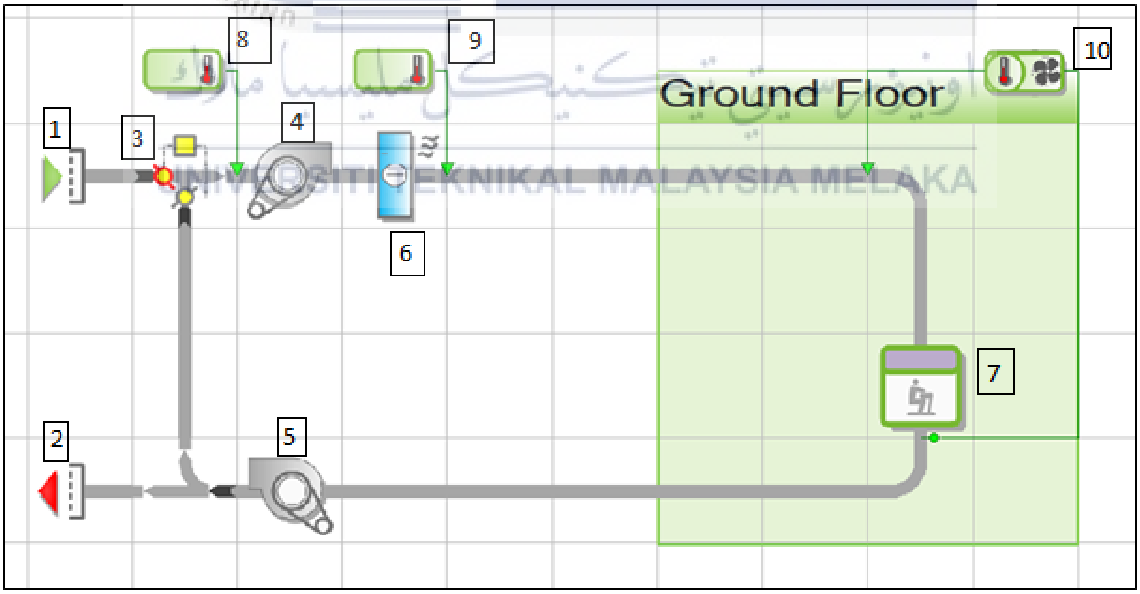


Figure 5.8: HVAC equipment model

Table 5.12: Details of the HVAC model

Parts No.	Parts Name	Description
1	Air Inlet	The place where air is brought into the HVAC system.
2	Air Outlet	The place where air is released from the HVAC system,
3	Mixing Damper Set	Mix the fresh air with recirculated air.
4	Inlet Fan	Draw the mixing air into the HVAC system.
5	Outlet Fan	Draw the air out from the lecture rooms.
6	Cooling Coil	Cool the mixing air then release to the lecture rooms.
7	Zones	Specify the lecture rooms into a single zone.
8	Independent Time Switch 1	Specify the air temperature of the mixing air.
9	Independent Time Switch 2	Specify the air temperature of the cooled air.
10	Independent Controller With Sensor	Specify the air flow into the lecture rooms.

5.5.2 Energy Consumption Analysis Based On The Suitable Setting

The energy consumption for both ground floor and first floor are analysed based on the suitable setting recommended for both of the air velocity and air temperature. Table 5.13 tabulated the results of the energy consumed by ground floor and first floor in a year. The energy consumption per month is also listed in the table 5.13.

Table 5.13: The estimated energy usage for ground floor and first floor based on the suitable setting

Estimated Energy Usage for the suitable setting (kwh)		
Month	Ground Floor	First Floor
Jan	21199.3	17761.4
Feb	19046.6	15735.4
Mar	26410.4	22280.1
Apr	22359.6	18919.0
May	22726.8	18942.4
Jun	23427.4	20379.5
Jul	24055.9	20354.6
Aug	23425.1	19641.8
Sep	22183.2	18526.7
Oct	21688.1	18437.2
Nov	22596.0	18485.1

Dec	22403.6	18432.0
Total	271521.9	227895.3

From the table 5.13, it can be seen that the estimated energy consumption for ground floor is higher than first floor. The estimated energy consumption for ground floor is 271521.9 kwh while for first floor is 227895.3. The difference is 43626.6 kwh. This could due to the difference between the number of lecture rooms presented. Ground floor has more lecture rooms compared to first floor. Ground floor has 11 lecture rooms while first floor has only 7 lecture rooms and one postgraduate area. This could lead to a more frequent usage of energy in ground floor compare to first floor and hence the estimated energy consumption is higher. The estimated energy consumption is only represent the HVAC system. Other equipments such as lightnings, computer system and projector are not included in the estimation.

5.5.3 Energy Consumption Analysis Without The Suitable Setting

Another energy consumption analysis is done by not using the suitable setting discussed. This is to determine the significance difference of energy consumption between suitable setting and without the suitable setting. The air temperature setting is set as 16 ^{o}C, which is lower than the acceptable air temperature setting. The estimated energy consumption for both ground floor and first floor without using the suitable setting is shown in table 5.14.

Table 5.14: The estimated energy usage for ground floor and first floor without the suitable setting

Estimated energy usage without the suitable setting (kwh)		
Month	Ground Floor	First Floor
Jan	24102.2	23605.1
Feb	22832.6	22358.6
Mar	27685.9	27081.8
Apr	25888.1	25286.5
May	24791.3	24153.2
Jun	25453.4	24276.0
Jul	25771.5	25004.0
Aug	25679.5	25193.0
Sep	25262.9	24847.6
Oct	24630.3	24491.8
Nov	26317.0	26163.0
Dec	26303.4	26807.8
Total	304718.0	299268.5

Based on the result, it can be seen that without the suitable setting, the estimated energy consumption is much higher. The differences for both ground floor and first floor are 33196.1 kwh and 71373.2 kwh. If the air temperature setting is set lower than the suitable air temperature setting recommended, the HVAC system is required to use more energy in order to cool the air to a lower temperature.

Based on the observation, FKM students tend to set the air temperature setting to the lowest. This action is not recommended as the average air temperature in the lecture room will be much lower than the recommended air temperature by MS1525. It is not only let the occupants feel uncomfortable as it is too cold, it also consume unnecessary energy. Moreover, higher energy consumption brings higher electricity cost. This will cost UTeM to pay more on the electricity bill. If the FKM students are concern about this matter, the unnecessary energy consumption can be reduced and the cost of electricity can be reduced as well.

CHAPTER 6

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

Based on the existing data, most of the air temperature and air velocity in the selected lecture rooms in FKM are not within the recommended standard. The best thermal comfort condition cannot be achieved in the lecture rooms. Unacceptable thermal comfort condition will cause discomfort to the occupants which can decrease their productivity and influence their emotional. Suitable air velocity and air temperature setting is vital to keep the building in a suitable range of thermal comfort.

After the simulations on the selected lecture rooms were done, it can be known that different room area requires different settings in order to provide the best thermal comfort to the occupants. This is due to more occupants can be fit into a bigger room and the heat energy generated is much higher. Hence, a bigger room with more occupants required a lower air temperature setting and higher air velocity setting. Oppositely, the air temperature setting can be set higher with a lower air velocity for a smaller room with less occupants. The suitable air temperature and air velocity setting for the selected lecture rooms are summarized in table 6.1.

Table 6.1: Suitable settings for air temperature and air velocity

BK	Air Temperature (^oC)	Air Velocity (m/s)
4	22	3
5	21	4
6	21	3
8	22	3
17	21	3
PA	24	4

Moreover, with the suitable air temperature and air velocity setting, the estimated energy usage can be lowered down for both the ground floor and first floor compare to the one estimated in existing result. The estimated energy consumption for ground floor is 271521.9 kwh while for first floor is 227895.3 kwh. The estimated energy consumption for ground floor is higher than first floor is due to ground floor has more lecture rooms than first floor. As the air temperature setting goes lower and the air velocity setting goes higher, the energy consumption of the HVAC equipment will increase. This will increase the cost of electricity and it is a burden for UTeM.

In conclusion, using the suitable air temperature and air velocity setting in a building is important. It will not only provide the best thermal comfort to occupants, it also will reduce the energy consumption of a building and thereby saving energy. The result gained from the simulation in this research is important as it can be used as a reference for further studies by other researchers. The simulation result also can act as a guidance for the company who want to utilize the building energy usage.

6.3 Recommendation on the project

This project is focusing on the indoor thermal comfort of occupants and out of the six parameters that defining thermal comfort, air temperature and air velocity are involved. Recommendation after a project is important as it is able to give a better suggestion and improvement for further research. After finishing this project, a few recommendation have been made as follow:

i. The HVAC system of the lecture rooms are not in proper maintenance. In a few lecture rooms, some of the diffusers are not working. The air velocity release from the diffusers is not consistent as well. The HVAC system of FKM should be regularly maintained so that the equipment can function at their optimum performance.
ii. Future research can include other four thermal comfort parameters into the research which are clothing insulation, metabolic rate, humidity and mean radiant temperature.
iii. Future researcher can do the similar research by using other available modelling technique such as using ANSYS FLUENT.
iv. The heat gain from other sources such as leakage of outside air, projector and lamps should be considered for a better accuracy of result.
v. Future researcher should consider the worst case scenario when the lecture rooms are fully occupied, the heat generated will be higher than the half-occupied. The suitable air temperature setting for fully occupied and half-occupied lecture rooms might be different.

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20range%20of%20temperatures%20can%20provide%20comfort%20when%20combined%20with%20the%20proper%20combination%20of%20relative%20humidity%2C%20MRT%2C%20and%20air%20flow.&f=false

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Appendix A

1.0 Project timeline Gantt Chart for PSM 1

No	Task	Month			
		Sept	Oct	Nov	Dec
1	PSM Topic Selection				
2	Information Gathering				
3	Journals Review				
4	Literature Review				
5	Methodology				
6	Report Submission				
7	PSM 1 presentation				

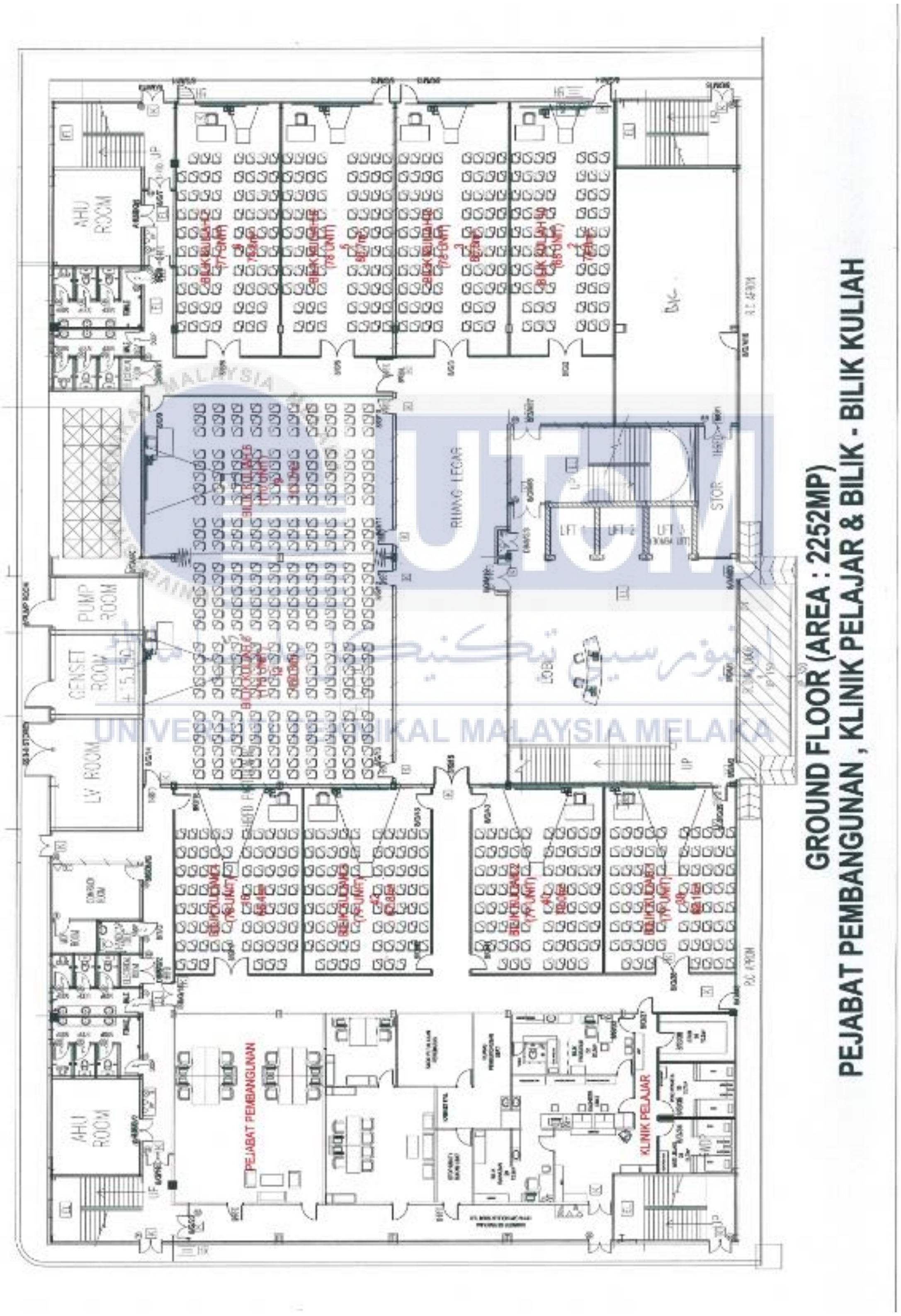
Appendix B

2.0 Project timeline Gantt Chart for PSM 2

No	Task	Month			
		Feb	March	April	May
1	Data Gathering				
2	Analyse Data				
3	Develop IESVE Simulation				
4	Changing Parameters				
5	Preparing PSM Report				
6	PSM Report Submission				
7	PSM presentation				

Appendix C

3.0 Floor plan for ground floor



4.0 Floor plan for first floor