ENERGY PERFORMANCE COMPARISON BETWEEN THE VARIABLE REFRIGERANT FLOW (VRF) AND CONVENTIONAL AIR – CONDITIONING AND MECHANICAL VENTILATION (ACMV) SYSTEMS FOR TROPICAL BUILDINGS

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2. Energy Analysis

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ABSTRACT

Generally, an enormous amount of energy is consumed by the large buildings due to the usage of low efficiency and poor thermal performance air - conditioning and mechanical ventilation (ACMV) system. The variable refrigerant flow (VRF) system has found its place in a building due to its high efficiency air - conditioning design. However, it is important to study the real performance of this system under actual operating situations before installing this system in new or retrofitted building in order to achieve energy savings goal. This research is focused on the annual energy consumption of VRF system in the tropical area (Malaysia) and compared it with other ACMV systems, which are split type unit and multi - split type unit systems under the same indoor and outdoor conditions. TRNSYS software is used to simulate the energy consumption for both VRF and ACMV systems based on the existing building's characteristic and operation. Note that the bin method is applied in finding the annual energy consumption for these systems, thus the baseline simulation in the research is created based on the bin method concept. This research is divided into two case studies according to the types of the building. In case study I, the VRF system is compared with the split type unit in a small building which needs cooling in single zone while for case study II, the VRF system is compared with the multi - split type unit in a large building which needs cooling in several zones. The reason for choosing these types of building is to identify whether the VRF system is suitable to use either in the building with several zones of cooling or building with single zone of cooling or both. For both case studies, simulation results have indicated that the VRF system saves about 15.57% and 13.62% when compared with the split type unit system and multi - split unit system, respectively. The operating cost for these systems is also pointed out and the results indicated that the annual operating cost for the VRF system is 15.64% and 13.63% lower than the split type unit system and multi - split unit system, respectively. This shows that the VRF system has a high potential for energy savings and could reduce significant electricity consumption in the building located in the tropic. Besides, the percentage uncertainty between the both simulation and manually calculation results on the annual energy consumption for these systems are too small which is approximately to 1%, where it shows that the bin method is appropriate to use in the simulation program for energy analysis.

Keywords: ACMV system; VRF system; Energy savings; Tropical area; Bin method.

ABSTRAK

Pada umumnya, bangunan besar menghasilkan sejumlah tenaga yang besar disebabkan penggunaan sistem pengudaraan mekanikal dan penghawa dingin (ACMV) yang mempunyai prestasi termal yang lemah serta kecekapan yang rendah. Penggunaan sistem *variable refrigerant flow* (VRF) telah mendapat tempat di dalam bangunan disebabkan reka bentuknya yang mempunyai kecekapan yang tinggi. Walaubagaimanapun, ianya sangat penting untuk mempelajari prestasi sebenar sistem ini dalam keadaan operasi sebenar sebelum menggunakan sistem ini di bangunan baru atau lama supaya misi penjimatan tenaga boleh dicapai. Penyelidikan ini tertumpu kepada penghasilan tenaga oleh sistem VRF dalam setahun di kawasan tropika (Malaysia), dan membandingkan ia dengan sistem ACMV yang lain, iaitu sistem *split type unit* dan *multi - split type unit* di bawah kondisi luaran dan dalaman yang sama. Perisisan TRNSYS digunakan untuk mensimulasikan penghasilan tenaga oleh sistem VRF dan ACMV berdasarkan ciri - ciri dan operasi bangunan yang sedia ada. Ianya penting untuk menjelaskan bahawa kaedah *bin* digunakan untuk mencari jumlah tenaga tahunan yang dihasilkan oleh sistem - sistem ini, maka garis asas simulasi di dalam penyelidikan ini dihasilkan berdasarkan konsep kaedah *bin*. Penyelidikan ini dibahagikan kepada dua kajian berdasarkan jenis bangunan di mana bagi kajian I, sistem VRF dibandingkan dengan sistem *split type unit* di dalam satu bangunan kecil yang memerlukan pendinginan dalam satu zon, manakala untuk kajian II, sistem VRF dibandingkan dengan sistem *multi - split type unit* di dalam sebuah bangunan besar yang memerlukan pendinginan dalam beberapa buah zon. Pemilihan dua jenis bangunan ini adalah untuk mengenalpasti sama ada sistem VRF sesuai untuk digunakan di dalam bangunan yang memerlukan pendinginan dalam beberapa buah zon atau bangunan yang memerlukan pendinginan dalam satu zon atau kedua - duanya sekali. Untuk kedua - dua kajian, keputusan simulasi telah menunjukkan bahawa sistem VRF berjaya mengurangkan penggunaan tenaga sebanyak 15.57% apabila dibandingkan dengan sistem *split type unit* dan 13.62% apabila dibandingkan dengan sistem *multi* - *split type unit*. Kos operasi untuk sistem - sistem ini juga turut dikaji dan keputusannya menunjukkan bahawa kos operasi tahunan untuk sistem VRF ialah 15.64% lebih kurang daripada sistem *split type unit* dan 13.63% lebih kurang daripada sistem *multi* - *split type unit*. Ini menunjukkan bahawa sistem VRF mempunyai potensi yang tinggi dalam penjimatan tenaga dan boleh mengurangkan penggunaan elektrik yang ketara di bangunan yang terletak di kawasan tropika. Selain itu, ketidakpastian peratusan antara hasil simulasi dan kaedah pengiraan secara manual bagi penggunaan tenaga tahunan untuk sistem - sistem ini terlalu kecil iaitu hampir kepada 1%, dimana ia menunjukkan bahawa kaedah *bin* sesuai untuk digunakan dalam program simulasi untuk analisis tenaga.

Keywords: Sistem ACMV; Sistem VRF; Penjimatan tenaga; Kawasan tropika; Kaedah bin.

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LIST OF SYMBOLS AND ABBREVIATIONS

Nomenclature

A	Area (m^2)
Cp	Specific Heat at Constant Pressure (J/kg.K)
CL	Cooling Load (kW)
CLF	Cooling Load Factor
$CLFT$	Sum of Cooling Load Factor for Solar Loads of 24 hours
$CLTDS$	Solar Transmission Contribution
COP	Coefficient of Performance
Dc	Degradation Coefficient
E	Energy (kJ)
$\overline{ESTD}$	Temperature Difference (°C)
FPS	Fraction of Possible Sunshine
h	Specific Enthalpy (kJ/kg)
K	Correction Factor
m	Mass Flow Rate (kg/s)
$MSHGF$	Maximum Value Heat Gain Factor at the Specified Latitude (W/m^2)
N	Number of Hour (h)
P_I	Power Input (kWh)
Q_c	Cooling Capacity (kW)
$\dot{q}$	Load (kW)
SC	Shading Coefficient
T	Temperature (°C)
TCL	Total Cooling Load (kW)
$TSCL$	Total Solar Cooling Load (W/m^2)

U	Heat Transfer Coefficient (W/m^2.K)
V	Volume Flow Rate (m^3/s)

Greek Symbols

η	Efficiency
ρ	Density
ω	Humidity Ratio of Air

Subscripts

a	Air
bin	Bin Interval
cond	Conduction
comp	Compressor
e	Building Envelopes (roof/ceiling or wall or door or window or glass)
eq	Equipment
evaA	Evaporator A
evaB	Evaporator B
evaC	Evaporator C
evaD	Evaporator D
g	Glass
i	Orientation Number
inf	Infiltration
isen	Isentropic
j	Homogenous Material Layer Number
Jan	January
Jul	July
l	Latent Heat
light	Lighting

out	Outdoor
oc	Occupied
occ	Occupancy
pc	Peak Cooling
ph	Peak Heating
r	Room
s	Sensible Heat
solar	Solar
t	Operation Time of ACMV System (h)
tot	Total
un	Unoccupied
vent	Ventilation

Abbreviations

ACMV	Air - Conditioning and Mechanical Ventilation
AHU	Air Handling Unit
CO_2	Carbon Dioxide
COP	Coefficient of Performance
DOAS	Dedicated Outdoor Air System
DX	Direct Expansion
EER	Energy Efficiency Ratio
EEVs	Electronic Expansion Valves
FAP	Fresh Air Processor
FORTRAN	Formula Translation Program
FPFA	Fan - Coil Plus Fresh Air
GSHP	Ground Source Heat Pump
HP	Horse Power

IAQ	Indoor Air Quality
JHVS	Heat Recovery Ventilator System
PLF	Partial Load Fraction
RH	Relative Humidity
SC	Shading Coefficient
SCHX	Sub Cooling Heat Exchanger
SDHP	Solid Desiccant Heat Pump
SEER	Seasonal Energy Efficiency Ratio
TMY	Typical Meteorological Year
TMY2	Typical Meteorological Year - 2
TNB	Tenaga Nasional Berhad
TRNSYS	TRaNsient SYStem
VAV	Variable Air Volume
VRF	Variable Refrigerant Flow
VRF HP	VRF Heat Pump
VRV-VAV	Combination of VRF and VRV System

LIST OF APPENDICES

CHAPTER 1: INTRODUCTION

1.1 Research Background

In today's modern society, where the people nowadays spend most of their time in the buildings, the usage of air - conditioning in the residential and commercial buildings is becoming necessary because of the enormous demand for indoor thermal comfort and healthy indoor environment. However, by obtaining the comfortable indoor environment, a large quantity of energy needs to be consumed. In the worldwide, energy consumed in the building sector including residential and commercial buildings is more than 30% and has steadily increased between 20 - 40% in the developed countries, which exceed the industrial and transportation sectors due to the spread of air - conditioning and mechanical ventilation (ACMV) (Hong et al., 2016; Zhu et al., 2014b).

Furthermore, commercial buildings as office buildings in the developed countries consume for almost 50% of the total energy due to the usage of ACMV system (Aynur et al., 2009, 2010; Hong et al., 2016; Li & Wu, 2010; Liu et al., 2015; Yildiz & Ersöz, 2015). In addition, the recent energy simulation studies and field surveys show that the total of electricity used by the ACMV system in the office building is about 37 - 60%, which based on the building's function (Zhou et al., 2007). In the United States (U.S), the energy consumption of the residential and commercial building is about 35%, where ACMV system consumes about one - third to one - half of the energy in these buildings (Kim et al., 2017). In China, energy consumption of the ACMV system is about 35% in the commercial, industrial, and residential building (Li et al., 2009; Zhou et al., 2007). - half of the energy in these buildings (Kim et al., 2017).

According to Yahaya et al. (2015), the ACMV system consumes more than 50% energy in the tropics. For example, in Singapore, energy consumption for supporting ACMV system in the single largest building is about 60% (Radhakrishnan et al., 2016).

Moreover, United Nations Development Programme (2006) states that the buildings in Malaysia consume for about 52% of energy and will continue to increase throughout the years in between the year 2010 and 2040 by 56% as mentioned in the report on the present situation of the global energy consumption by the International Energy Outlook (Hassan et al., 2014).

Energy savings for a building can be achieved by reducing the cooling energy in the building itself. This reduction can be done without sacrifice the indoor thermal comfort and indoor air quality by studying the details about the energy review on the variation of the ACMV designs, in which it can assist the designers or users to select the best one among them to be installed in the desired building.

1.2 Problem Statement

The previous studies reveal that when compared with other ACMV systems, the variable refrigerant flow (VRF) system consumes less energy without sacrificing indoor thermal comfort and air quality. Even though the initial costs for VRF system is quite expensive compared to other ACMV system, this system has great potential for energy savings and could reduce large electricity consumption in the building (Aynur et al., 2009; Hong et al., 2016; Liu et al., 2015).

However, not all VRF system is suitable for all building because the actual savings from VRF system would be different in certain country and building due to the several factors, including building layout, operation conditions, control strategies, and climate (Yu et al., 2016; Zhu et al., 2014a). Thus, before deciding whether to install the VRF system or any other ACMV systems, in the desired building, it is important to estimate the total of energy required for air - conditioning the desired building under usual weather conditions whether for short or long time periods of operation, evaluate the VRF system's performance, and compare compared it with the other ACMV system

(Zhou et al., 2007). Otherwise, the energy savings goal cannot be achieved because of the overload of the building itself.

The energy savings effect of the VRF system is worth studying. However, the research work on this system in the tropics is quite rare, where there are a few published literatures on the comparison of annual energy consumption especially in term of bin method between VRF system with other ACMV systems in the tropics including in Malaysia and Singapore. Therefore, in this research, the energy performance of VRF system in the tropic is studied by comparing it with other ACMV systems at the same indoor and outdoor conditions.

As for simulation, VRF system cannot be simulated as a built - in module by the simulation software including DOE - 2, BLAST, and EnergyPlus since these software do not provide the simulation capability for the VRF system due to the complexity of its operation. Therefore, in this research, TRNSYS software is used for the simulation purpose due to its ability to create new components. It is pertinent to mention that the baseline simulation in the research is created based on the bin method concept. Since the usage of bin method in the energy simulation is quite new, most of the components used in the simulation need to be created based on the bin method equations as adopted in previous studies (Elhelw, 2016; Wang et al., 2014) in order to ensure the bin method concept can be applied in the simulation.

Furthermore, this research is divided into two case studies in accordance with the types of the building, which are small building with a single zone of cooling and a large building with a several zones of cooling. The reason for choosing these types of building is to identify whether the VRF system is suitable to use either in the building with several zones of cooling or building with single zone of cooling or both.

1.3 Objectives

The objectives of this research are:

a) To estimate energy performance of VRF system at two different types of building located in the tropics, which are building with several zones of cooling and building with single zone of cooling by using bin method, and compared it with the other ACMV system.

b) To create the baseline simulation applied with the bin method concept in order to estimate the energy usage of the ACMV and VRF systems used in this research by using TRNSYS software.

c) To ensure the bin method is suitable to use in the simulation program such as TRNSYS program.

1.4 Scopes of Study

The scopes of study in this research are focused on:

a) Using the bin method to estimate the energy consumption of VRF and other ACMV systems.

b) Using TRNSYS simulation software to simulate the energy consumption of the target buildings installed with the VRF and ACMV systems by applying bin method concept in the simulation with the help of bin weather data over some period of time, which is from 2007 to 2016.

1.5 Significances of Research

The significances of this research are:

a) The findings of this research is applicable to buildings in Malaysia since this research is conducted in Malaysia, where these findings can be reference for the future studies in term of energy performance of the VRF and ACMV systems and also can be used as proof in order to commercialize the VRF system in Malaysia.

b) The energy performance by using bin method is quite new in tropical countries especially in Malaysia and Singapore. Therefore, the development of the energy simulation baseline based on the bin method concept by using TRNSYS software done in this research may contribute to a new way of research in term of energy performance of the VRF and ACMV systems.

CHAPTER 2: LITERATURE REVIEW

2.1 Variable Refrigerant Flow (VRF) System

The VRF system as a refrigerant system can match up with the building heating or cooling load by changing the refrigerant flow rate for maintaining the desired temperature at particular zones with the help of the inverter - driven and multiple compressors, and electronic expansion valves (EEVs), which are located in the indoor units of this system (Afify, 2008; Alshatti, 2011; Aynur et al., 2009, 2010; Bhatia, 2012; Goetzler, 2007; Hong et al., 2016; Jiang et al., 2013; Kim et al., 2017; Kim et al., 2016; Kwon et al., 2012; Liu et al., 2015; Liu & Hong, 2010; Wang, 2014; Wang et al., 2009; Yildiz & Ersöz, 2015). However, the VRF system needs a separate ventilation system because this system is ductless (Aynur et al., 2010; Goetzler, 2007; Kwon et al., 2012; Thornton & Wagner, 2012).

The design of the VRF system contributes to simultaneous cooling and heating at divergent zones with the help of heat recovery, and an individualized zoning control (Afify, 2008; Bhatia, 2012; Goetzler, 2007; Karr, 2011; Kim et al., 2017; Kwon et al., 2012; Liu & Hong, 2010; Thornton & Wagner, 2012; Wang, 2014; Yildiz & Ersöz, 2015). Generally, this system can be used for simultaneous cooling and heating at different zones with the help of heat recovery and also can be used either for heating or cooling purpose (Alshatti, 2011; Aynur, 2010; Bhatia, 2012; Kwon et al., 2012; Li & Wu, 2010; Li et al., 2016; Liu & Hong, 2010). It is pertinent to mention that the heat recovery in the VRF system assists this system to reject the heat from one zone in cooling mode, where this rejected heat can be used in other zones for heating mode rather than being rejected to the outdoor (Karr, 2011). According to research done by Li and Wu (2010), the usage of heat recovery in the VRF system can save a lot of energy compared to the VRF heat pump system, where the comparison of between both systems in term of energy simulation result under the same indoor and outdoor temperature shows that the energy savings potential for heat recovery VRF system is 15 - 17% higher than the VRF heat pump system. However, the price for heat recovery VRF system and its installation is higher compared to the VRF system used for either cooling or heating purpose only (Afify, 2008).

The same as other multi - split type unit of ACMV system, this system consists one outdoor unit which linked to several indoor units in parallel with the refrigerant pipes (Aynur, 2010; Bhatia, 2012; Goetzler, 2007; Hong et al., 2016; Kwon et al., 2012; Li & Wu, 2010; Li et al., 2016; Liu & Hong, 2010; Thornton & Wagner, 2012; Wang, 2014; Yan et al., 2012; Yildiz & Ersöz, 2015). However, different from multi - split type unit system, most of the VRF system manufacturers provide a centralized control option, where monitoring and controlling the entire system can be done either from a single location or by using the internet rather than using one master controller (Bhatia, 2012; Thornton & Wagner, 2012).

2.1.1 The Advantages and Disadvantages of VRF System

The main advantages of the VRF system are its great potential in energy savings and efficient performance. This system can operate efficiently during part load conditions due to the usage of variable speed compressor and multi - speed fans; minimizing noise during its operation due to the smaller indoor unit fans; minimizing heat losses and air leakage due to the minimal or none ductwork; and also maximizing the efficiency of refrigerant flow by minimizing the refrigerant path with the help of separation tube (Afify, 2008; Bhatia, 2012; Goetzler, 2007; Hong et al., 2016; Karr, 2011; Kim et al., 2017; Kim et al., 2016; Zhu et al., 2014b).

Furthermore, the capability of simultaneous cooling and heating at different zones, monitoring and controlling the entire system from a single location or by using the internet, and independent zoning control show that this system can accommodates better indoor thermal comfort to the user (Bhatia, 2012; Thornton & Wagner, 2012). The independent zoning control in the VRF system helps this system to adjust the thermostat (temperature) settings into the desired indoor temperature by different users in the different zones at the same time due to the automatic adjustment of the refrigerant flow by this system itself (Bhatia, 2012; Hong et al., 2016). Besides, its independent zoning control can helps in reducing the electricity consumption of the VRF system (Bhatia, 2012; Liu et al., 2015).

The flexibility of this system including simultaneous cooling and heating at different zones and independent zoning control make this system suitable for a building which needed to be cooled and heated in several zones at the same time, having difference heating and cooling loads across many different zones as well as complicated load conditions, and need to deliver individualized comfort conditioning such as offices, schools, apartment complexes, shopping centers, luxury single family houses, condos, multi - family residential building and hotels (Alshatti, 2011; Aynur, 2010; Bhatia, 2012; Hong et al., 2016; Wang et al., 2009). This system also is suitable to use in hospitals and nursing homes since these buildings avoid zone to zone air mixing (Aynur, 2010; Hong et al., 2016). Furthermore, VRF system is suitable to use in the buildings with a strict noise regulation is applied such as schools, hospital, places of worship, libraries, and nursing homes because the operating sound level for both indoor ductless unit and ducted unit are quite low, which are 27dB(A) and 29dB(A), respectively (Bhatia, 2012; Hong et al., 2016).

Moreovers, the VRF system has large capacity modulation and capability to make the desired indoor temperature in any zones can be reached extremely fast and minimize the inconsistency of this temperature despite of outside conditions, providing a total comfort for the occupants (Bhatia, 2012). In addition, since this system is self - contained, all the indoor units of this system can still run continuously even though the trouble occur at any of these indoor units in one system and the same goes for the outdoor unit, since this system uses multiple compressors, where if trouble occurs at any of these compressors, there is no instant shutdown because the other compressors will start to operate on an emergency basis (Bhatia, 2012).

The VRF system is light weight and has modular design which required less outdoor space and both outdoor and indoor units can be moved easily and fits into a standard elevator, enabling floor by floor installation and commissioning (Aynur, 2010; Bhatia, 2012; Goetzler, 2007; Karr, 2011; Kim et al., 2017; Thornton & Wagner, 2012; Yildiz & Ersöz, 2015). Besides, electric meter can be placed on one or a few condensing units of this system easily due to the modularity of this system. The submetered VRF system can provides separate billing, making the individualized billing lot easier especially in a multi - tenant buildings, where the total electricity costs can be charged specific to each tenant (Bhatia, 2012; Goetzler, 2007).

Moreover, since this system is lightweight and ductwork is required for the ventilation purpose only, it may minimize the requirements for structural reinforcement of the building without ventilation and also appropriate for adding the VRF system in the historical buildings without disturbing the structure, which clearly can reduce the renovation costs (Aynur, 2010; Bhatia, 2012; Goetzler, 2007; Karr, 2011; Thornton & Wagner, 2012). Furthermore, the bank prefers this system for security because smaller diameter ductwork can be added to this system, which it can minimize the egress path into the bank (Aynur, 2010).

The ductless VRF system operates on the direct expansion (DX) principle, where heat is directly delivered into or out the conditioned space by flowing the refrigerant to evaporators located near or within the conditioned space (Bhatia, 2012; Wang, 2014). Since the VRF system uses DX principle to operate, the maintenance costs for this system are lower when compared with the water - cooled chillers (Goetzler, 2007). Moreover, this system is easy to maintain because normal maintenance for a VRF system is quite the same with other ACMV system that use direct DX principle to operate, where the maintenance process are usually changing the filters and cleaning the coils (Goetzler, 2007; Kim et al., 2017).

The main disadvantage of the VRF system is the cost (Aynur, 2010; Bhatia, 2012; Hong et al., 2016). The initial cost for this system is quite expensive than the other ACMV system (Bhatia, 2012; Hong et al., 2016). The installation cost for this system highly depends on the building layout, construction, and application. In addition, the unfamiliarity with the VRF technology also should add up the installation cost (Afify, 2008; Aynur, 2010; Goetzler, 2007). Moreover, the VRF system usually uses an additional ventilation system to provide better indoor air quality because this system is ductless, and by adding the additional ventilation to this system, the installation cost for this system can be increased (Afify, 2008; Alshatti, 2011; Aynur, 2010; Thornton & Wagner, 2012).

Several cost comparison had been made between the VRF system and another ACMV system. For example, the total costs for VRF system are around 5% to 20% more expensive than the chilled water system with equivalent capacity, around 30% to 50% more expensive than single package ducted system with the similar capacity and seasonal energy efficiency ratio (SEER) of 13 to 14, and the VRF system cost more than twice when compared to the packaged terminal unit of the ACMV system (Aynur, 2010; Goetzler, 2007). Another comparison between VRF system, four pipe fan coils unit, and variable air volume (VAV) system with chilled ceiling in a new building, which is three storey office building shows that the costs for four pipe fan coils unit is 53% higher than the chilled ceiling; the costs for VAV system is 74% higher than the chilled ceiling; and the costs for VRF system is 111% higher compared to the chilled ceiling (Aynur, 2010).

According to the comparison between VRF system with water - cooled chiller and air - cooled chiller for a 200 ton cooling system in a commercial building, the installation cost for VRF system is 8% more expensive than water - cooled chiller and about 16% more expensive to the air - cooled chiller. However, VRF system saves about 30 - 40% of energy, where it can accommodates an estimated payback period of about 1.5 years compared to the air - cooled chiller and about 8 months compared to the water - cooled chiller (Aynur, 2010).

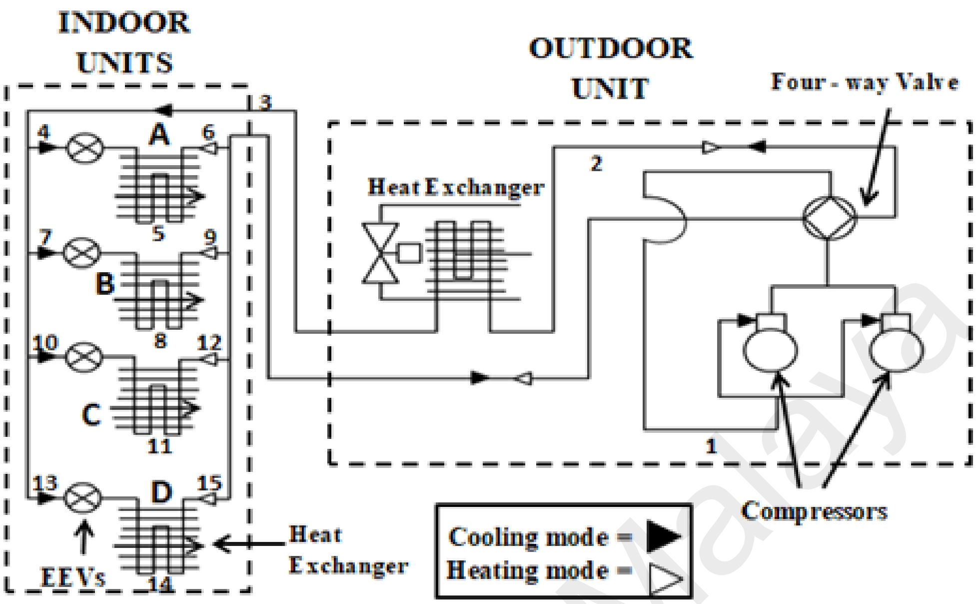
Furthermore, for a case study in 17 floors of 100,000 ft^2 (9300 m^2) office building in Brazil shows that the installation cost for VRF system is about 15 - 22% more expensive compared to the chiller, but the annual energy savings potential for VRF system during summer and winter is higher than the chiller, which is 30% and 60%, respectively (Aynur, 2010; Goetzler, 2007; Liu & Hong, 2010). Besides, in 1998, VRF manufacturer in Italy had compared the installation and operating costs for VRF system and chiller/boiler system in 7 buildings. The result shows that the installation cost for VRF system is more expensive when compared to the chiller/boiler system, but the maintenance costs and energy usage of this system is lower when compared with chiller/boiler system, which is 40% and 35%, respectively (Aynur, 2010).

From these comparisons, it clearly shows that even though the initial cost for VRF system is quite expensive, this system has high potential for energy savings, which can promote the possibility of the payback and also has a lower maintenance cost.

2.1.2 General Overview of VRF System

Since the VRF system is ductless, this system operates on the DX principle, where heat is directly delivered to or from the space by flowing the refrigerant to evaporators located near or within the conditioned space (Bhatia, 2012; Wang, 2014). Typically, the VRF system operates using the Inverse Carnot Cycle same as the other ACMV system which operates on DX principle (Liu et al., 2015). The flow of the refrigerant for each evaporator is stabilized accurately through EEVs associated with an inverter - driven and multiple - compressors control system based on the variations of the cooling or heating process in the conditioned space (Aynur, 2010).

Figure 2.1 shows the example of schematic diagram of the VRF system with four indoor units. In cooling mode, the refrigerants such as R22, R24, and R410A from the indoor units absorb heat produced in the indoor and extract it to the outdoor (Bhatia, 2012). The discharged refrigerant from the compressor in the outdoor unit pass through the four - way valve before entering the heat exchanger (condenser). During the movement of discharged refrigerant to the heat exchanger (condenser), this discharged refrigerant will has high pressure and low temperature. Thus, before this discharged refrigerant enters the indoor unit heat exchanger (evaporator), it will suppressed to a low pressure by the EEVs (Aynur, 2010). The EEVs help to control the quantity of refrigerant flow through the indoor units. The evaporation process takes place, where the heat is absorbed from the indoor air when the discharged refrigerant pass through the coil in the indoor units (Bhatia, 2012). Then, the heat is cooled down by the indoor unit. Lastly, the low pressure superheated discharged refrigerant returns to the compressor in the outdoor unit and completes the cycle (Aynur, 2010).



Components	Function
Heat exchanger (Indoor and outdoor units)	Transfer the heat from the indoor/outdoor units to the surrounding or vice versa, depending on the mode used
Compressors	Control the discharged of refrigerant mass flow rate
Four - way valve	Reversing the path of the system from cooling to heating or vice versa, depending on the mode used
EEVs	Control the flow of refrigerant to the evaporators (indoor units)

Figure 2.1: Schematic diagram of a VRF system with four indoor units (redrawn based on reference (Aynur, 2010))

For heating mode, the process is the same, but in the reversing path with the help of four - way valve (Aynur, 2010; Bhatia, 2012; Li et al., 2016). The heat from the discharged refrigerant is rejected to the indoor when this discharged refrigerant enters the heat exchanger (condenser) from the compressor in the outdoor unit. This process will make the indoor air becomes hot. Then, the discharged refrigerant will have high pressure and low temperature. Thus, before this discharged refrigerant enters the outdoor unit heat exchanger (evaporator), it will suppressed to a low pressure by the

EEVs and undergoes condensation process, which releases heat energy into the space (Aynur, 2010; Bhatia, 2012). Lastly, the low pressure superheated discharged refrigerant returns back to the compressor and finishes the cycle (Aynur, 2010).

Figure 2.2 shows the mass and energy balance at each component of the VRF system based on the schematic diagram of the VRF system in Figure 2.1 in form of P - h diagram.

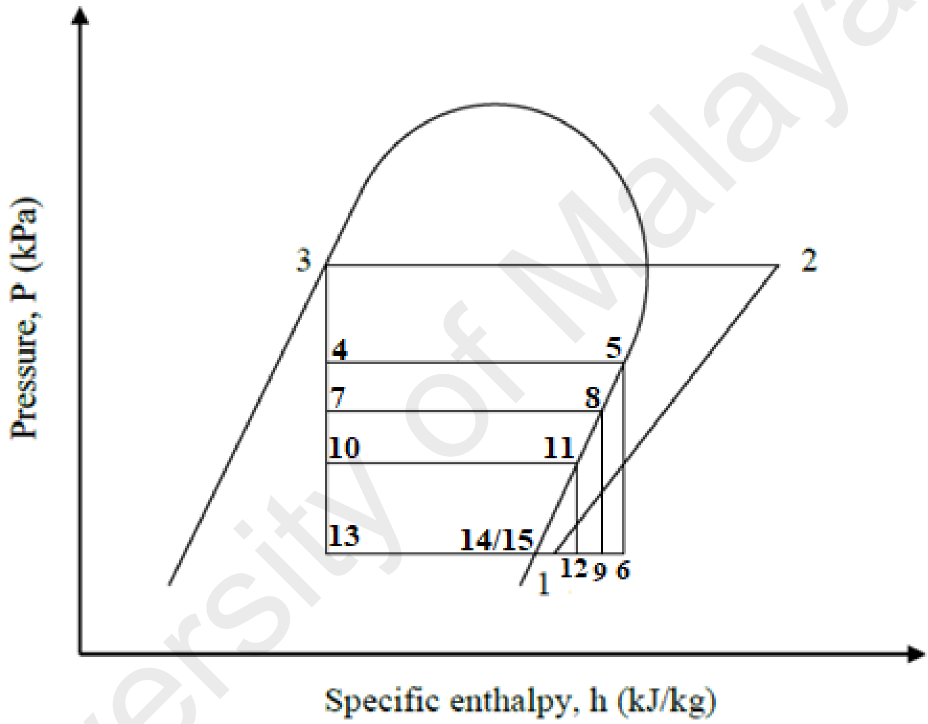


Figure 2.2: Pressure versus specific enthalpy diagram of the cycle (redrawn based on reference (Alshatti, 2011))

Based on Figure 2.2, the coefficient of performance (COP) of the VRF system can be obtained as shown below (Alshatti, 2011):

$$COP = \frac{Q_{c,evaA} + Q_{c,evaB} + Q_{c,evaC} + Q_{c,evaD}}{P_I} \quad (1)$$

Where the power input of the compressor can be calculated as follows (Alshatti, 2011):

$$P_I = m_{tot}(h_2 - h_1) \quad (2)$$

The mass flow rate for each indoor unit can be obtained by an energy balance as shown below (Alshatti, 2011):

$$m_{evaA} = \frac{Q_{c,evaA}}{h_5 - h_4} \tag{3}$$

$$m_{evaB} = \frac{Q_{c,evaB}}{h_8 - h_7} \tag{4}$$

$$m_{evaC} = \frac{Q_{c,evaC}}{h_{11} - h_{10}} \tag{5}$$

$$m_{evaD} = \frac{Q_{c,evaD}}{h_{14} - h_{13}} \tag{6}$$

The total mass flow rate can be represented as (Alshatti, 2011):

$$m_{tot} = m_{evaA} + m_{evaB} + m_{evaC} + m_{evaD} \tag{7}$$

The enthalpy of the compressor inlet can be computed by an energy balance nearby the connecting node (Alshatti, 2011):

$$h_1 m_{tot} = h_6\, m_{evaA} + h_9 m_{evaB} + h_{12} m_{evaC} + h_{14/15} m_{evaD} \tag{8}$$

The actual enthalpy of the compressor outlet can be computed by using its efficiency as follows (Alshatti, 2011):

$$\eta_c = \frac{h_{2isen} - h_1}{h_2 - h_1} \tag{9}$$

2.1.3 Configuration of VRF System

2.1.3.1 Configuration of Outdoor Unit

The outdoor unit of the VRF system is available in sizes up to 70 kW, where the outdoor unit of VRF system without heat recovery differ in size from 6 to 30 tons while the VRF system with heat recovery differ in size from 6 to 24 tons (Aynur, 2010;

Thornton & Wagner, 2012). This outdoor unit which has heat exchanger, multi - compressors, and inverter - driven compressor can be linked with the multiple indoor units due to the capability of having wide range capacity modulation with high part - load efficiency, where the inverter - driven compressor has a range of frequency from 20 - 30 to 105 - 120 Hz (20 - 30 to 105 -120 rps) (Aynur, 2010; Bhatia, 2012; Karunakaran et al., 2010; Liu & Hong, 2010; Thornton & Wagner, 2012; Tu et al., 2016).

The inverter - driven compressor is capable of changing the mass flow rate of discharged of refrigerant (capacity control range is between 6% to 100%), changing its speed to match up with the variations of total heating or cooling loads in the different zone, and also maintaining meticulous temperature control, typically within ±0.6°C (Alshatti, 2011; Aynur, 2010; Bhatia, 2012; Goetzler, 2007). Furthermore, the inverter - driven compressor in the VRF system assists this system by handling or responding the inconsistency of space load condition.

Figure 2.3: Inverter - driven compressor

Basically, the VRF system consists one inverter - driven compressor and one or two standard compressors, where the inverter - driven compressor operates at the beginning of the operation of VRF system while the standard compressor stay in the idle mode (Tu et al., 2016). Then, the inverter - driven compressor will increase the output capacity due to the increasing in the indoor thermal load. However, when the output capacity of this compressor does not reach the desired indoor thermal load, the standard compressor will begin to operate until the desired indoor thermal load demand can be reached (Tu et al., 2016).

The VRF system also has smart integrated control which located in the outdoor unit (Thornton & Wagner, 2012). The smart integrated control can make up to dozens of multi - tasking include automated diagnostic, multiple monitoring and control points, and coordinate the flow of refrigerant, while maintaining the operation stability such as controlling the processing of outdoor unit, indoor units, temperature, compressor frequency, fan motor speed, and switches simultaneously (Thornton & Wagner, 2012).

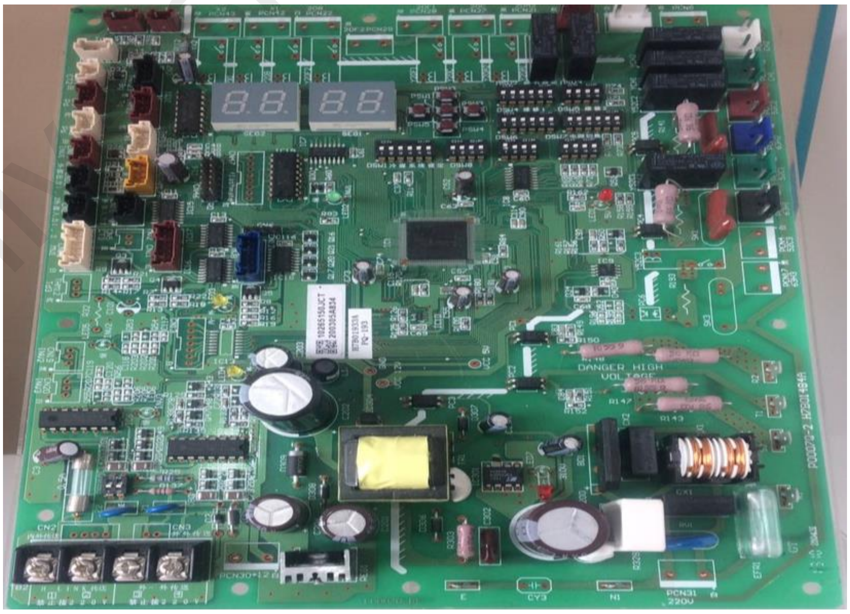


Figure 2.4: Smart integrated control

Usually, the VRF system undergoes heating or cooling process either by the air or water, where the air type VRF system undergoes heating or cooling process via the ambient air while the water type VRF system undergoes heating or cooling process via the water (Aynur, 2010; Li et al., 2009; Liu & Hong, 2010; Thornton & Wagner, 2012). The outdoor unit of air type VRF system consists of tube heat exchanger and fin while the outdoor unit of water type VRF system consists of plate type heat exchanger and usually linked with the cooling tower or boiler (Aynur, 2010; Li et al., 2009; Thornton & Wagner, 2012).

Same as air type VRF system, the liquid or gas refrigerant pipelines of the water type VRF system links the outdoor unit with a multiple indoor units, where this pipelines is not placed in the conditioned space to prevent the leakage problem (Aynur, 2010; Li et al., 2009). Besides, the plate heat exchangers in the water type VRF system help to link the water loop with refrigerant circuits and there is no limitation on the maximum length of water pipe needed to link this system with the cooling tower or boiler (Li et al., 2009; Thornton & Wagner, 2012).

2.1.3.2 Configuration of Indoor Unit

The indoor unit of VRF system has heat exchanger, temperature sensor, EEVs, direct expansion coil, and a multi - speed fans (Aynur, 2010; Karr, 2011; Liu & Hong, 2010). Besides, the indoor units of this system has different cooling or heating capacities which is from 1.4 kW to 17.5 kW, where these units come with different configurations such as ceiling mounted cassette, ceiling mounted built - in, ceiling mounted duct, wall mounted, and floor standing type (Afify, 2008; Aynur, 2010; Bhatia, 2012; Thornton & Wagner, 2012).

Furthermore, each indoor unit of this system has its own temperature sensor to differentiate the thermostat set temperature with the actual air temperature (Aynur, 2010; Hong et al., 2016). With this temperature difference, the EEVs controls the refrigerant flow rate through each indoor unit heat exchanger so that each indoor unit of this system can operates separately in the different zones and also some of them can be closed if the zones is unoccupied or the desired indoor temperature is obtained while the other continuous to operate (Aynur, 2010; Hong et al., 2016). Figure 2.5 shows the example of VRF system used in the building with several zones.

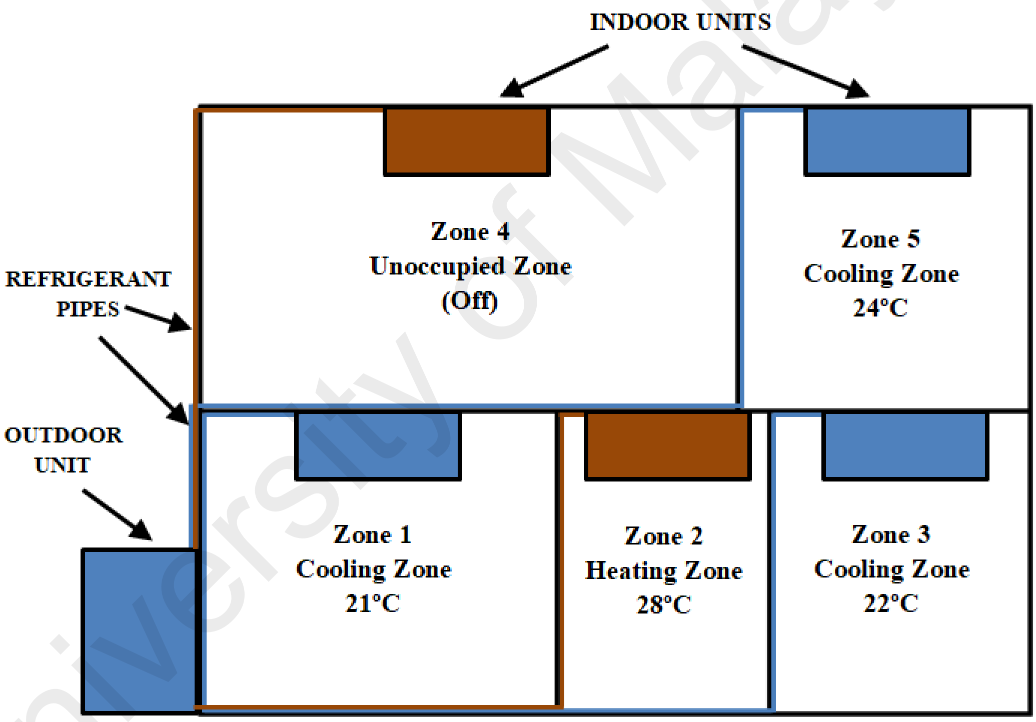


Figure 2.5: Example of VRF system used in the building with several zones

Generally, the VRF system used EEVs to maintain the pressure differential and also the flow rate of refrigerant in each indoor unit in accordance with the actual load of indoor units by reducing or stopping the flow of refrigerant to the individual indoor unit when superheat occur in the VRF system (Bhatia, 2012; Li et al., 2016). However, when the VRF system obtained the desired indoor temperature, the EEVs stay in the idle mode (Bhatia, 2012). The EEVs is provided with the synchronous electronic motor to isolate a full rotation into a large number of steps which is about 500 steps/rev (Bhatia, 2012).

Figure 2.6: Electronic expansion valves (EEVs)

The outdoor unit of the VRF system operates based on the demand from the indoor units, where the outdoor unit received information regarding the indoor temperature from the thermistor sensors in each indoor unit through the control wire that connected to the several indoor units. Then, the required indoor temperature is regulated by the microprocessor in the outdoor unit and varying the speed of its compressor in accordance with the actual load of indoor units with the helps of pulse modulating valve (PMV) (Bhatia, 2012; Thornton & Wagner, 2012).

The numbers of indoor unit which can be linked to one outdoor unit are increasing throughout the year. In the late 1980s, one outdoor unit is capable to link with 4 to 8 indoors units and increasing to 16 indoor units in 1990s. Then, in 1999, one outdoor unit is capable to operate with about 32 indoor units and increasing to 40 indoor units in 2003. Nowadays, the current VRF system has capability to link the single outdoor unit with 60 or more indoor units, where the total maximum capacity of the outdoor unit can exceed the maximum capacity of the indoor units, which allows the variation of the cooling or heating loads (Aynur, 2010; Bhatia, 2012; Hong et al., 2016; Thornton & Wagner, 2012).

2.1.3.3 Configuration of Refrigerant Pipe

The outdoor and indoor units for this system are linked with the refrigerant pipes and the piping length can reached up to 1000 m by the help of advanced oil circuitry, return, and control (Aynur, 2010; Bhatia, 2012; Liu & Hong, 2010). Typically, the VRF system is provided with either two or three pipe configurations to minimize the refrigerant path so that the efficiency of the refrigerant flow in this system can be increased (Aynur, 2010; Hong et al., 2016; Yildiz & Ersöz, 2015).

The VRF system which has two pipes including the high pressure gas pipe and low pressure liquid pipe is operated either for heating or cooling (Aynur, 2010; Hong et al., 2016; Yildiz & Ersöz, 2015). On the other hand, the VRF system with three pipes, which are high pressure gas pipe, low pressure gas pipe, and low pressure liquid pipe is used for the simultaneous cooling and heating at different zones (Aynur, 2010; Goetzler, 2007; Hong et al., 2016; Yildiz & Ersöz, 2015).

Figure 2.7 shows the schematic view for 3 pipe configurations of VRF. The branch selector box which located before each indoor unit helps the VRF system to operate in five different modes (Aynur, 2010; Yildiz & Ersöz, 2015):

a) Cooling mode (only for cooling operation).
b) Heating mode (only for heating operation).
c) Main cooling mode (cooling is the main mode in the simultaneously heating and cooling operation).
d) Main heating mode (heating is the main mode in the simultaneously heating and cooling operation).

e) Heat recovery mode (heat is balanced between indoor units while the outdoor unit heat exchanger is closed).

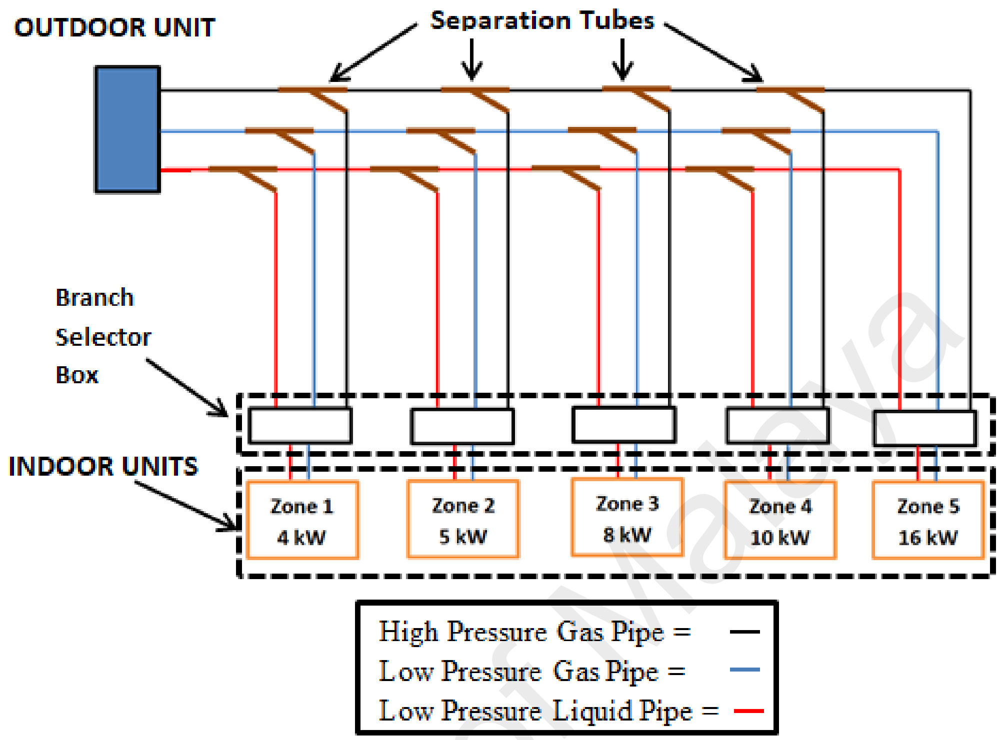


Figure 2.7: Schematic view for 3 pipe configurations of VRF system (redrawn based on reference (Yildiz & Ersöz, 2015))

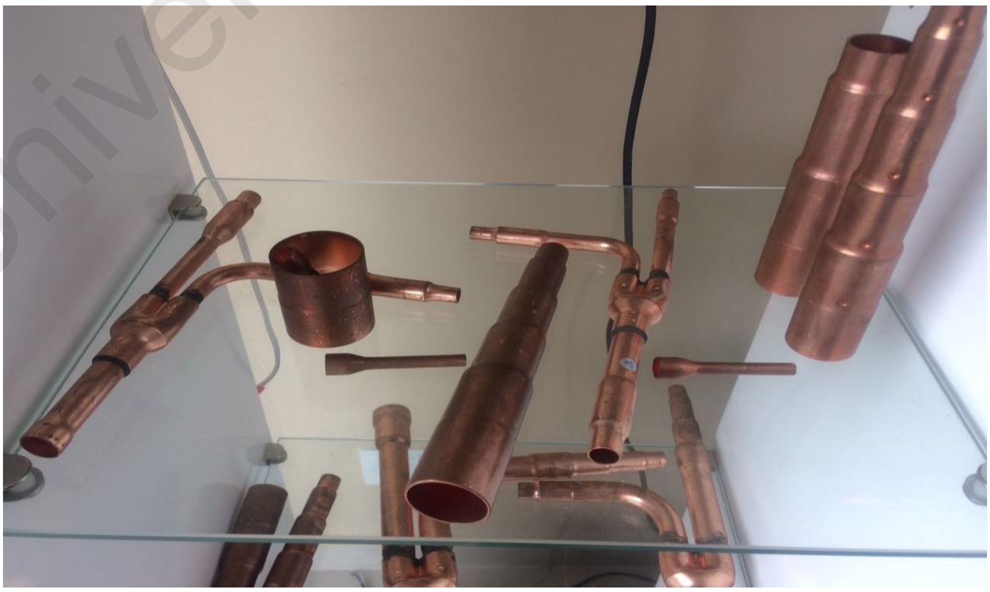


Figure 2.8: Separation tubes

Both separation tubes and headers are used to connect the refrigerant pipe from the outdoor unit to several indoor units, where the separation tube has 2 branches while the header has more than 2 branches. Besides, both separation tubes and headers can be installed together with the headers, but these separation tubes cannot be installed after the header to avoid the balancing issues (Bhatia, 2012).

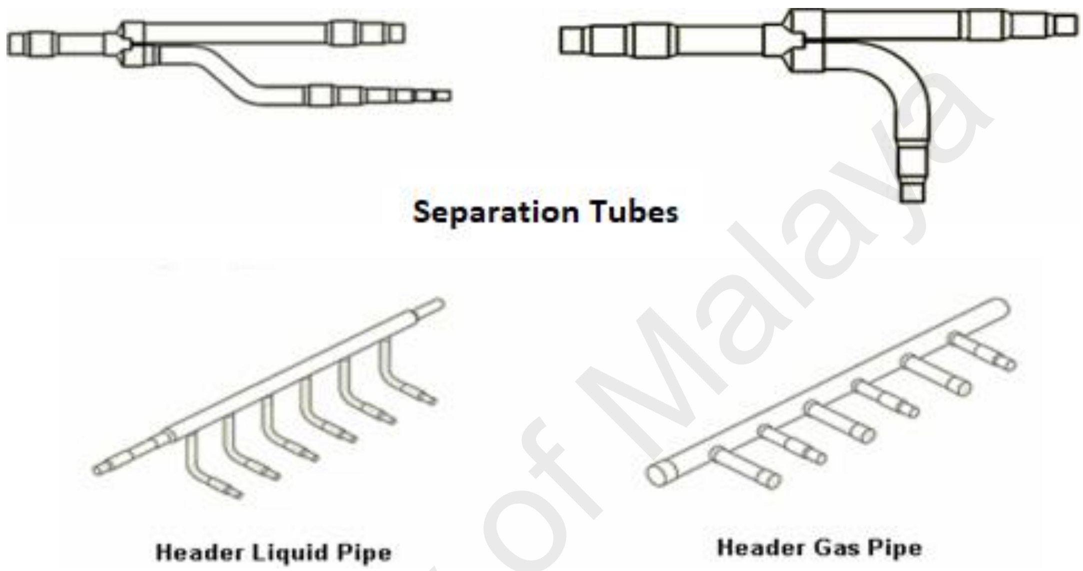


Figure 2.9: Example of separation tubes and headers (Bhatia, 2012)

2.1.4 History of VRF System

The VRF system was invented by Daikin Industries at Japan in 1982 (Aynur, 2010; Bhatia, 2012; Goetzler, 2007; Karunakaran et al., 2010; Wang, 2014; Yan et al., 2012). At first, in Japan, about 50% of VRF system was used in a medium - sized a commercial building with a net area up to 70,000 ft^2 (6500 m^2) and about one - third of this system was used in large commercial buildings with a net area bigger than 70,000 ft^2 (6500 m^2) (Aynur, 2010; Goetzler, 2007; Li et al., 2016).

After a few years, the VRF system starting to become popular in Asia and Europe because this system is provided with the high part - load energy efficiency, installation space savings, independent zoning control, and simple maintenance (Aynur, 2010;

Kwon et al., 2012; Li et al., 2016). However, even though the ductless air - conditioners have entered the U.S in the early 1980s, this country still has a limited market on the VRF system due to the differences in construction practices and regulatory environment, lack of support from the Japanese manufacturer, and also unfamiliarity with this system (Aynur, 2010; Goetzler, 2007). Besides, the ozone depletion issues and the high cost of the refrigerant used in the multi - split type unit system were the main concern by the U.S people at that time (Aynur, 2010).

Then, in the late 1990s, the VRF system was introduced in China and this system was widely used in the residential buildings, offices, hotels, and shopping centers (Li et al., 2016; Zhou et al., 2007). In the year 2003, about 85, 500 of VRF products were sold worldwide, where about 69% of VRF products were sold in Asia which is 46.8% and 22.2 % in Japan and China, respectively; 21.9 % and 6.3 % in Europe and Oceania, respectively; and only 2.8 % of VRF products were sold in the rest of the world. This is because of the improvement in a charge management, refrigerant developments, inverter technology, and controls of this system (Aynur, 2010).

As time goes by, due to the re – entering of Asian manufacturers in the U.S either individually or partnership with the U.S manufacturers, the VRF system's market in this country had been increased, where this country successfully sold about 10, 000 VRF systems in 2007 (Aynur, 2010). In the first half of 2014 in China, the VRF system has more than 40% of the share in sales of central air - conditioning products (Li et al., 2016). Nowadays, the VRF system is becoming more popular in China and widely used in the hospitals, nursing homes, school building, and residential buildings (Hong et al., 2016; Zhou et al., 2007).

2.1.5 Studies on the VRF system

2.1.5.1 Comparison Studies with other ACMV Systems

Previous literature reviews reveal that the variable refrigerant flow (VRF) system consumes less energy when compared with other ACMV system. For example, in China, VRF system, VAV system, and fan - coil plus fresh air (FPFA) system were compared in a 10 - storey office building by using the EnergyPlus software, where the simulation results shows that the VRF system has highest energy savings potential among the other two ACMV systems. The result shows that the energy savings for VRF system were about 22.2% and 11.7% when compared with the VAV system and the FPFA system, respectively (Aynur et al., 2009; Karunakaran et al., 2010; Liu & Hong, 2010; Zhou et al., 2007). Another comparison between the water type VRF system with FPFA system in the typical office building in Shanghai shows that during the cooling period, the energy savings for VRF was about 20% when compared with the FPFA system (Li et al., 2009; Liu & Hong, 2010).

Besides, another comparison between VRF system and other ACMV systems, which were VAV system, fan coil system, central chiller/boiler system, and chiller system shows that the energy savings for VRF system was about 20 - 58% when compared with VAV systems in the cooling season, 10% when compared with fan coil system in the cooling season, 35% when compared with central chiller/boiler system under the humid subtropical climate, and 30% when compared with chiller system under the tropical climate (Hong et al., 2016).

According to the study performed in the government building, where a VRF system was used on one side of the building while the rooftop VAV system was used on the other side in the same building shows that the VRF system saves more energy than the rooftop VAV system, which was about 38% (Aynur et al., 2009). It should be noted that the office space was mostly unoccupied during the evaluation (Aynur et al., 2009). Furthermore, the study on the simulation comparison between the VRF system and VAV system in the 3rd and 4th floors of the administrative office building at University of Maryland, Washington shows that the energy savings potential for VRF system was about 27.1 - 57.9% (Aynur et al., 2009; Liu & Hong, 2010).

Moreover, the study of performance comparison between the VRF system and VRF heat pump (VRF HP) system with the rooftop VAV system in the medium office building at the 16 U.S climate locations which conducted by using EnergyPlus software shows that the VRF system saves around 14 - 39% of energy while VRF HP system saves about 2 - 32% of energy when both systems were compared with the rooftop VAV system (Kim et al., 2017). Besides, from this study, it shows that the VRF HP system has lower energy cost than the VAV system in the hot and mild climates while the VAV system has lower energy cost in several cold climate zones because of the differences in the consumption of electricity and gas (Kim et al., 2017).

Liu et al. (2015) had analyzed the operational electricity consumption between the VRF system with the centralized air - conditioning system in the two office buildings at the campus based on the monitoring data obtained from a building energy monitoring and management system. From this research, the rate of electricity consumption obtained was 0.48 kWh/m^2 for VRF system and 0.42 kWh/m^2 for the centralized air - conditioning system, in which can be concluded that both systems have a great potential in energy savings. However, the electricity consumption of the VRF system is better than the centralized air - conditioning system due to the flexible adjustability of the VRF system (Liu et al., 2015).

A study on the comparison of energy savings between the ground source heat pump (GSHP) system with the air type heat recovery VRF system which conducted by using EnergyPlus in Atlanta, Baltimore, and Chicago shows that the GSHP system has a great potential in energy savings and can reduces more electric peak demand at cool humid, warm humid and mixed humid climates than the air type heat recovery VRF system. The result shows that the annual energy savings for GSHP system was 20% and reduces about 31 - 40% of the electric peak demand when compared to the air type heat recovery VRF system (Wang, 2014). Another comparison study on the air type heat recovery VRF system which comes with the standard rated refrigerant piping length and GSHP system in a small office at two selected U.S climates shows that the energy savings potential for GSHP system was expected to achieve 9.4 - 24.1% when compared with this VRF system. It should be noted that the GSHP system anticipated to has more energy savings if compared with the air type heat recovery VRF system with the longer refrigerant pipe (Liu & Hong, 2010).

Even though the GSHP system has greater energy savings potential compared to the VRF system especially in a building with high heating load, this system has high installation cost in which result in poor payback; need installation of specific design and ground loop engineering; and requires large space for ground coupling which can be quite challenging to install this system in densely built areas (Goetzler et al., 2009; Liu & Hong, 2010). However, according to Karr (2011), by combining the GSHP system with the VRF system, the advantages for both systems can become into a single system, which can increasing the efficiency of this combined system. In this combined system, the ground water loops from GSHP system is used with the multi - speed fans and variable speed compressors from VRF system to obtain better energy savings compared to either GSHP system or VRF system on their own (Karr, 2011).

2.1.5.2 Studies on Enhancing the Performance of VRF System

A few studies on enhancing the performance of VRF system are done in recent years for increasing the energy savings advantage and market potential. For example, Hong et al. (2016) had develop a new VRF heat pump model, where the new features of this model are the component level curves is introduced rather than using overall curves for the entire system, and the introduction of variable evaporating and condensing temperatures in the indoor and outdoor units as well as variable fan speed in the indoor unit. This new model can directly reducing the speed of the compressors in the outdoor unit by the help of inverter technology (Hong et al., 2016).

Jiang et al. (2013) had introduces a new model of solid desiccant heat pump (SDHP) combining with the VRF system in order to create a temperature and humidity independent control system. The comparison on the energy performance between this model and the combination of joint heat recovery ventilator system (JHVS) with the VRF system was done in the office room, which located in China and the simulation for this research was done by using EnergyPlus. The simulation results show that this new model saves about 18.7% of energy through the whole year when compared to JHVS combining with the VRF system. Furthermore, the indoor condition for thermal comfort provided by this new model is 85.5% while JHVS combining with the VRF system provides 35.4%. This clearly shows that this new model has greater indoor thermal comfort and energy savings potential than the JHVS combining with the VRF system. It is pertinent to mention that in this new model, the function of the SDHP is to handle the latent load while the function of the VRF system is to utilize the sensible load, where by handling these loads separately, higher COP and greater indoor thermal comfort can be obtained (Jiang et al., 2013).

Based on the Karunakaran et al. (2010) research on the application of intelligent fuzzy logic controller in the combination of VRF system with VAV system, it shows that this system consumes about 44% of energy during summer and 63% of energy during winter per day when compared with the conventional constant VAV system. This clearly shows that the intelligent fuzzy logic controller can helps this combined system to obtain better thermal comfort, indoor air quality (IAQ), and energy conservation in the modern ACMV applications. The fuzzy logic controller delivering the ventilation air volume in accordance with the detected CO_2 level in the conditioned space to ensure a better indoor thermal comfort can be obtained without sacrifiying IAQ for summer and winter season. Furthermore, this controller helps to control the compressor operating cycle of this combined system, ensuring the indoor temperature is at the satisfy indoor thermal comfort even at the part load conditions (Karunakaran et al., 2010). Besides, by using this controller in the ACMV system, the energy usage can be reduces because this controller helps in keeping the process variables (temperature and humidity) to their set points (Karunakaran et al., 2010; Rezeka et al., 2015).

In the installation of the VRF system especially in a large building, it is quite challenging to avoid a long refrigerant pipeline between the outdoor and indoor units due to the layout of this system is highly related to the building geometry (Afify, 2008; Li et al., 2016; Wang et al., 2009). However, the efficiency of this system can be reduces especially in the cooling mode if the length of horizontal refrigerant pipeline used between outdoor and indoor units is too long, where it will make the EEVs in the indoor units to has instability control and also produces noise because the refrigerant in form of liquid flashing before it enter the EEVs (Bhatia, 2012; Kwon et al., 2012; Li et al., 2016). Besides, the longer the refrigerant pipeline been used in the system, the higher the specific volume of refrigerant delivered into the compressors, reducing the cooling capacity and energy efficiency in this system (Li et al., 2016; Liu & Hong, 2010). In addition, according to Yan et al. (2012), the usage of longer refrigerant pipeline with many fittings can make this system to has high refrigerant pressure drop, where it can reducing the evaporating temperature, system's operating efficiency, and COP as well as increasing the condensing temperature.

Li et al. (2016) had develop a new model of the VRF system with sub cooling heat exchanger (SCHX) to overcome the problem with the length of refrigerant pipeline. From the simulation result, it shows that the maximum COP of the VRF system with the 190 m of main refrigerant pipelines can be improved by 1.7% - 1.0% successively if the length of SCHX is twice by 0.5 m, which is from 0.5 m to 1.5 m. This clearly shows that by increasing the length of SCHX, the cooling capacity and COP of the VRF system with longer refrigerant pipes can be improved. Another study on the impact of the SCHX in the VRF system during a cooling season shows that the VRF system with SCHX provides a greater performance when compared with the VRF system without the SCHX. The usage of SCHX can reduces the pressure ratio in this system, where by reducing the pressure ratio, the power consumption can be reduced and the compressor efficiency with enthalpy difference across the evaporators can be increased (Kwon et al., 2012). Besides, by using a SCHX, the refrigerant in the liquid circuit can be sub - cooled before it enters the evaporator (Kwon et al., 2012).

Energy conservation for VRF system can be increased by using proper insulation in refrigerant pipeline network, where with better insulation, the heat loss to the surroundings can be reduced. However, the optimization of the insulation thickness is necessary to reduce the insulation costs without sacrificing the energy conservation. The optimum insulation thickness is depends on the heat loads of the refrigerant pipeline, where the optimum insulation thickness is higher as the heat loads on the refrigerant pipeline is increase (Yildiz & Ersöz, 2015). Based on the study in finding the optimum insulation thickness for the refrigerant pipeline of VRF system with three pipe either for both heating and cooling mode, the optimum insulation thickness in heating mode is between 16 and 20 mm for the pipe sections of high pressure gas pipeline and between 11 to 13 mm for the pipe sections of low pressure liquid pipeline. In cooling mode, the optimum insulation thickness is between 7 and 8 mm for both pipe sections of low pressure gas pipeline and low pressure liquid pipeline (Yildiz & Ersöz, 2015).

The VRF system requires minimal or none ductwork to operate. Therefore, this system cannot ensure the quality of the indoor air since this system does not induce the outdoor air. However, this problem can be fixed by combining the VRF system with a fresh air processor (FAP) due to its simple structure and adjustable air flow. The FAP can helps the VRF system to obtain comfortable fresh air at different zones with different sizes and distances. Moreover, the usage of FAP in this system also is good for obtaining the clean air, where the produced indoor air pollutant can be removed continuously while maintaining the fresh healthy indoor air due to the introduction of fresh air from the outdoor (Tu et al., 2011).

According to Zhu et al. (2014b), the problem with IAQ in the building can be solved by the usage of the VRF system combining with the VAV system. This combined system can increases the energy savings potential without eliminating the indoor thermal comfort and IAQ. It should be noted that the VRF parts used in this combined system are one outdoor unit and multiple indoor units while the VAV parts used in this combined system are VAV boxes, direct expansion oriented outdoor air processing unit, supply fan, and air duct (Zhu et al., 2014b).

Kim et al. (2016) study on the VRF system combining with dedicated outdoor air system (DOAS) in the office building located in Seoul, in which the simulation for this system was done by using the EnergyPlus shows that the combination between these systems can helps in reducing the total energy consumption while obtaining better IAQ and indoor thermal and humidity comfort. It should be noted that this combined system features with DOAS with a plate type total heat exchanger and an electrically heated steam humidifier, and an air type VRF heat pump. Besides, the ability of DOAS in controlling the IAQ easily by separating ventilation air from cooling and heating air makes this system suitable to be used for centralized all air system in the buildings (Kim et al., 2016).

The investigation on the combination between VRF system and heat pump desiccant system for enhancing the potential of energy savings and indoor air condition was done during a cooling season by comparing three different operating modes of this system, which are non - ventilated, ventilation, and ventilation - dehumidification. Based on the result, the average of total cooling energy provided by this combined system was 97.6% for the ventilation mode and 78.9% for the ventilation - dehumidification mode. Besides, among these three operating modes, the combined system with the ventilation - dehumidification mode saves more energy than the ventilation mode, but lesser than the non - ventilated mode while produces better indoor thermal comfort and IAQ (Aynur et al., 2010).

2.2 Energy Analysis

Generally, the success of all economies either in the immediate or long term future is depending on the total usage of energy. Therefore, the building energy analysis is becoming an important tool in order to estimate the usage of energy and energy costs of the building, which can minimize the life cost of the building (Bulut et al., 2001; Elhelw, 2016; Jin et al., 2006; Papakostas et al., 2008; Papakostas, 1999; Peng et al., 2009; Thamilseran & Haberl, 1994). Usually, enormous energy consumption in the building is due to the low thermal performance of the building itself and the usage of low efficiency of cooling or heating system, where these problems always happens in the existing office building (Elhelw, 2016; Wang et al., 2014). Thus, the energy analysis is important for designing the building cooling or heating system, where it is crucial to find the optimum energy needed for cooling or heating system of the building at typical weather conditions whether for short or long time periods before making the final selection for installing this system, so that the energy savings goal can be achieved (Elhelw, 2016; Jin et al., 2006; Papakostas et al., 2008; Papakostas, 1999; Papakostas & Sotiropoulos, 1997).

There is many variations of energy analysis methods can be used from the simplest method to sophisticated methods, where the simplest methods require many assumption which has tendency in providing inaccurate results while the sophisticated methods require a few assumption which can provide adequate results (Bulut et al., 2001; Elhelw, 2016; Krarti, 2011; Papakostas et al., 2008; Papakostas & Sotiropoulos, 1997; Peng et al., 2009). Generally, the energy analysis for estimating the energy usage of a building and cooling or heating system used in the building can be done either by calculate manually or using energy simulation software such as DOE - 2, BLAST, EnergyPlus and TRNSYS (Bulut et al., 2001; Jin et al., 2006; Krarti, 2011).

Typically, the energy analysis can be classified into two types, which are forward method and inverse method. As for the forward method, the assumptions for energy calculation is in accordance with a physical description of the building itself such as geometry, location, and construction details, and also the types of the cooling or heating system used in the building with its operation while for the inverse method, the assumptions for energy calculation is based on the building loads and existing data of energy use, weather data, and any relevent data regarding the performance of the building itself and cooling or heating system used in the building (Krarti, 2011).

The inverse method are less complex to use compared to the forward method, but the flexibility of this method is depending on the building parameters and the accuracy of performance data of the building and cooling or heating system used in the building (Krarti, 2011). The forward method such as degree - day method and bin method which known as the steady - state energy analysis method (also known as single measure method) is easy to use because most of the calculations can be performed by hand or using spreadsheet programs, require few data, and has tendency in providing accurate results, especially in simple systems and applications (Bulut et al., 2001; Jin et al., 2006; Krarti, 2011; Özyurt et al., 2009; Papakostas et al., 2007; Papakostas et al., 2008; Papakostas, 1999; Peng et al., 2009). In addition, the forward method can be applied in most of the energy simulation softwares such as EnergyPlus, DOE-2, and TRNSYS (Krarti, 2011).

2.2.1 Bin Method

Bin method is a quite common method in many countries especially in U.S and Europe country, where this method is used for calculating either cooling or heating energy in the building (Elhelw, 2016; Krarti, 2011). Even though the bin method is quite similar to the degree - day method, this method needs reliable and detailed weather bin data to estimate total building and system heating or cooling energy consumption (Elhelw, 2016; Jin et al., 2006; Krarti, 2011; Papakostas et al., 2008; Wang et al., 2014).

This method using an instantaneous energy calculation based on the variation of outdoor temperature conditions (obtained by weather bin data), where these temperatures are divided into discrete groups (bins) in acccordance with the weather conditions to estimate either heating or cooling energy of the building and the heating or cooling energy of the system used in the building (Jin et al., 2006; Jones et al., 2009;

Krarti, 2011; University of Alabama). It should be noted that the divided temperature into discrete groups (bins) is known as bin temperature.

The heating or cooling energy can be obtained if the heating or cooling loads are known, where the heating or cooling loads can be obtained by multiply these loads with the total hours of each bin temperature within a month or year, which clearly shows that this method is suitable to use for a long term periods, either monthly or annually (Bulut et al., 2001; Cizik, 2009; Elhelw, 2016; Jin et al., 2006; McQuiston et al., 2005; Özyurt et al., 2009; Papakostas et al., 2007; Papakostas et al., 2008; Papakostas, 1999; Papakostas & Sotiropoulos, 1997; Peng et al., 2009; University of Alabama). In another word, the bin method utilizes the reference climate (bin weather data) as a basic table of outdoor temperature (bin temperature) and their frequency distribution to estimate heating or cooling loads at different outdoor temperature (Cizik, 2009; McQuiston et al., 2005; Schicktanza et al., 2014).

2.2.1.1 The Advantages of Bin Method

One of the advantages of the bin method is this method can be done separately for different outdoor temperature and time periods either for occupied or unoccupied building hours, where the bin temperatures are usually collected in six daily 4 hours shifts and these bin temperatures are often in 2.8ºC (5ºF) increments for the consideration of the different building loads and occupancy patterns with time as well as operating hours of heating or cooling system used in the building (Bulut et al., 2001; McQuiston et al., 2005; Papakostas et al., 2007; Papakostas et al., 2008; Papakostas, 1999).

For designing the building cooling or heating systems, various changes of outdoor climate, part load performance and COP of the ACMV equipment, and building load data must be taken into consideration (Elhelw, 2016; Papakostas et al., 2007; Wang et al., 2014). The performance and demand put on the cooling or heating equipment varies when the weather varies hour - by - hour (Bulut et al., 2001; Jin et al., 2006; Papakostas et al., 2007; University of Alabama). Meaning, the outdoor temperature might give impact either directly or indirectly to the efficiency, performance, and cooling or heating capacity for equipment that operates on the refrigeration cycle. However, since the bin temperature is expressed as a function in this method, it can be done separately for different outdoor temperature and time periods either for occupied or unoccupied building hours. This clearly shows that this method can obtains adequate results for the energy consumption for the building and the cooling or heating system used in the building, and also makes the analyzing of the energy change based on the variation of the outdoor temperature a lot easier, in which it can help in sizing the proper cooling or heating system equipment for desired buildings (Elhelw, 2016; Jin et al., 2006; Papakostas et al., 2007).

More accurate and precise total loads and energy efficiency ratio (EER) of a building and equipment that operates on the refrigeration cycle such as ACMV equipment can be obtained by using the bin method compared to the other certain methods (Elhelw, 2016; Schicktanza et al., 2014; Thamilseran & Haberl, 1994; University of Alabama; Wang et al., 2014). For example, Elhelw (2016) had comparing the modified bin method with the CLTD/SCL/CLF method on the load produced by the ACMV system. The results obtained shows that the modified bin method can obtained more adequate results compared to the CLTD/CLF/ SCL method. Furthermore, based on the research done by Wang et al. (2014), it is practical and suitable to use the bin method for analyzing the EER schemes of existing office buildings. Moreover, Thamilseran and Haberl (1994) had comparing the bin method with other existing savings calculation procedures in estimating the energy conservation retrofit savings for the Education Building at the University of Texas at Austin campus and it shows that both results obtained the same energy conservation. In this study, it should be noted that the bin weather data was taken from several agencies participating in the LoanSTAR program.

Another advantages of the bin method is this method can be carried out either by calculating it manually or using it in the simulation programs, where by using the simulation programs, the results can be obtained easily and faster compared with calculating manually especially when involves in energy or load calculation for a long periods at many variation of outdoor temperature (Krarti, 2011; Papakostas et al., 2008). Moreover, solar load for solar heating and cooling, and solar clearness index also can be obtained by using this method (Papakostas et al., 2007; Schicktanza et al., 2014).

2.2.1.2 Bin Method Calculation

The key in using bin method is the simplification of dynamic heat transfer into the steady - state and also the bin temperature must be expressed as a function in the bin method calculation (Cizik, 2009; Peng et al., 2009; Wang et al., 2014). In the bin method calculation, the building loads must be known and it is necessary to take a consideration of temperature variation in the cooling or heating loads of the building in order to identify the time - dependent load, where this time - dependent load are averaged according to the chosen period (occupied and unoccupied hours) or over multiple calculations (Elhelw, 2016; McQuiston et al., 2005; Papakostas, 1999; Wang et al., 2014).

(a) ***Solar Load***

As mention earlier, the bin temperature is expressed as a function in the bin method calculation. Thus, the equation for the solar load is expressed as the following (Elhelw, 2016):

$$\dot{q}_{solar} = aT_{bin} + b \tag{10}$$

Where a and b are constant. This linear equation is derived based on the target building envelopes, which are glass, door, window, wall, and roof/ceiling. According to Elhelw (2016), the solar load for glass is obtained based on the equation:

$$\dot{q}_{solar} = \sum_{i}^{n} TSCL_i \ x \ A_{gi} \ x \ SC_i \ x \ \left(\frac{FPS}{A_r}\right) \tag{11}$$

Where SC_i is a shading coefficient which can be obtained from the table of shading coefficient (Elhelw, 2016; Grondzik et al., 2010). $TSCL_i$ is a total solar cooling load (W/m^2) which can be obtained as following (Elhelw, 2016):

$$TSCL_i = \sum_{j=1}^{24} SCL_j \tag{12}$$

Based on the equation (12), the $TSCL_i$ is obtained by adding all the solar cooling loads per hour for every 24 solar hours (Elhelw, 2016). The fraction of possible sunshine (*FPS*) for any month can be calculated by (Elhelw, 2016):

$$FPS = \frac{Average\ monthly\ sun\ hours}{Number\ of\ days\ for\ the\ selected\ month\ x\ Maximum\ number\ of\ sunshine\ hours} \tag{13}$$

Then, a linear function of solar load for glass is expressed as equation (10). Another way to find the solar load for glass is based on the following equation (Wang et al., 2014):

$$\dot{q}_{solar,jul} = \frac{\sum_{i=1}^{n} \left(MSHGF_{jul,i} \ x \ A_{gi} \ x \ SC_i \ x \ CLFT_i \ x \ FPS_{jul}\right)}{t \ x \ A_r} \tag{14}$$

$$\dot{q}_{solar,jul} = \frac{\sum_{i=1}^{n} \left(MSHGF_{jan,i} \ x \ A_{gi} \ x \ SC_i \ x \ CLFT_i \ x \ FPS_{jan}\right)}{24 \ x \ A_r} \tag{15}$$

Based on equation (14) and (15), the solar load for glass in summer is in reference to the assumption of a clear sky while the solar load for glass in winter are not taken into the consideration since the sun is lower in the sky, where the sunlight is seldom reach the building (Wang et al., 2014). Then, a linear function of the solar load for glass is expressed as (Wang et al., 2014):

$$\dot{q}_{solar} = M \ x \left(T - T_{ph}\right) + \dot{q}_{solar,jan} \tag{16}$$

Where,

$$M = \frac{\dot{q}solar,jul - \dot{q}solar,jan}{T_{pc-}T_{ph}} \tag{17}$$

For each solar load of door, windows, wall, and roof/ceiling can be obtained by using the equation below (Elhelw, 2016):

$$\dot{q}_{solar,e} = U_{e,i} \ x \ A_{e,i} \ x \ \overline{ESTD_{\iota}} x \ \frac{FPS}{A_r} \tag{18}$$

Where the solar load for each building envelopes is totalized and then, a linear function of the totalized solar load is expressed as equation (10).

(b) ***Conduction Load***

The equation for the conduction load is expressed as the following (Elhelw, 2016):

$$\dot{q}_{cond} = aT_{bin} + b \tag{19}$$

Where *a* and *b* are constant. This linear equation is derived based on the target building envelopes which are glass, door, window, wall, and roof/ceiling. For each conduction load of door, windows, wall, and roof/ceiling can be obtained by using the equation below (Elhelw, 2016; Wang et al., 2014):

$$\dot{q}_{cond} = U_{e,i} \ x \ A_{e,i} \ x \ \frac{(T_{bin} - T_r)}{A_r} \tag{20}$$

According to Wang (2014), another way to find conduction load for wall, roof/ceiling, door, and window is based on the following equations:

$$\dot{q}_{cond,jul} = \frac{\sum_{i=1}^{n} \left(A_{e,i}\, U_{e,i}\, x\, K\, x\, CLTDS_{jul}\, x\, FPS_{jul}\right)}{A_r} \tag{21}$$

$$\dot{q}_{cond,jan} = \frac{\sum_{i=1}^{n} \left(A_{e,i}\, U_{e,i}\, x\, K\, x\, CLTDS_{jan}\, x\, FPS_{jan}\right)}{A_r} \tag{22}$$

Based on equation (21) and (22), a linear function of the conduction load can be written as (Wang et al., 2014):

$$\dot{q}_{cond} = M\, x\, \left(T - T_{ph}\right) + \dot{q}_{cond,jan} \tag{23}$$

Where,

$$M = \frac{\dot{q}cond,jul - \dot{q}cond,jan}{T_{pc-}\, T_{ph}} \tag{24}$$

(c) ***Internal Loads***

The internal loads are load of lighting, occupancy, and equipment used in the building. It is pertinent to mention that these loads have a constant value because these loads are not affected by the outdoor temperature.

The total installed lighting in the building might be operated discontinuously during the occupied hours, thus the lighting schedule is crucial in finding the realistic average usage of lighting in the building (Elhelw, 2016). The lighting load can be obtained as the following (Elhelw, 2016; Wang et al., 2014):

$$\dot{q}_{light} = (Average\ lighting\ usage)\ x\ \frac{(Max\ lighting\ load)}{A_r} \tag{25}$$

Another way to find lighting load is based on the following equation (SlideShare, 2014):

$$\dot{q}_{light} = (Average\ lighting\ usage)\ x\ Ballast\ Factor\ x\ CLF \tag{26}$$

Where ballast factor for fluorescent lights is 1.2 while ballast factor for incandescent lights is 1.0, and the value of cooling load factor, *CLF* is assumed to be 1.0 when the cooling or heating system used in the building is shut off at night (SlideShare, 2014).

The same as the lighting load, the occupancy pattern in the building might be different during the occupied hours, thus the occupancy pattern schedule is crucial in finding the realistic average occupancy in the building (Elhelw, 2016). In addition, the occupancy load comes in two components, which are sensible and latent, where both components are calculated separately (Elhelw, 2016). The occupancy load is calculated as the following (Elhelw, 2016; Wang et al., 2014):

$$\dot{q}_{occ} = (Average\ occupancy\ usage)\ x\ \frac{(Max\ occupancy\ load)}{A_r} \tag{27}$$

Another way to find both sensible and latent occupancy loads are based on the following equations, respectively (SlideShare, 2014):

$$\dot{q}_{occ,s} = No.of\ people\ x\ Sensible\ heat\ gain\ per\ person\ x\ CLF \tag{28}$$

$$\dot{q}_{occ,l} = No.of\ people\ x\ Latent\ heat\ gain/\ person \tag{29}$$

Where the sensible and latent heat loads per person can be obtained in ASHRAE Table (ASHRAE, 2014). Same as lighting load, the value of *CLF* is assumed to be 1.0 when cooling or heating system used in the building is shut off at night (SlideShare, 2014).

The total equipment which generate the heat in conditioned spaces might be operated discontinuously during the occupied hours, thus the equipment usage schedule is crucial in finding the realistic average usage of the equipment in the building (Elhelw, 2016). The equipment load is calculated by using the equation below (Elhelw, 2016; Wang et al., 2014):

$$\dot{q}_{eq} = (Average\ equipment\ usage)\ x\ \frac{(Max\ equipment\ load)}{A_r} \qquad (30)$$

Another way to find the equipment load is based on the following equation (SlideShare, 2014):

$$\dot{q}_{eq} = (Average\ equipment\ usage)\ x\ Sensible\ heat\ load\ of\ the\ equipment \qquad (31)$$

Where the sensible heat load of the equipment can be obtained obtained in ASHRAE Table (ASHRAE, 2014).

(d) ***Ventilation or Infiltration Load***

The ventilation or infiltration load is highly depends on the outdoor temperature and humidity. The sensible and latent components of this load are calculated separately (Elhelw, 2016; SlideShare, 2014; Wang et al., 2014). A linear function for ventilation or infiltration load for sensible and latent components can be described as the following, respectively (Elhelw, 2016):

$$\dot{q}_{inf,s}\ or\ \dot{q}_{vent,s} = aT_{bin} + b \qquad (32)$$

$$\dot{q}_{inf,l}\ or\ \dot{q}_{vent,l} = aT_{bin} + b \qquad (33)$$

Where a and b are constant. These linear equations are derived based on the following equation (Elhelw, 2016):

$$\dot{q}_{inf,s}\ or\ \dot{q}_{vent,s} = \rho_a\ x\ Cp_a\ x\ V_a x\ \frac{(T_{bin} - T_r)}{A_r} \tag{34}$$

$$\dot{q}_{inf,l}\ or\ \dot{q}_{vent,l} = \rho_a\ x\ h_a\ x\ V_a\ x\ \frac{(\omega_o - \omega_r)}{A_r} \tag{35}$$

Another way to derived the ventilation or infiltration load for sensible and latent components are based on the following equations, respectively (Wang et al., 2014):

$$\dot{q}_{inf,s}\ or\ \dot{q}_{vent,s} = \frac{0.34\ x\ V_a\ x\ (T_{bin} - T_r)}{A_r} \tag{36}$$

$$\dot{q}_{inf,l}\ or\ \dot{q}_{vent,l} = \frac{0.34\ x\ V_a\ x\ (\omega_{out} - \omega_r)}{A_r} \tag{37}$$

Where both equations can be expressed as equation (32) and (33), respectively. It should be noted that the ventilation load is considered to be zero if there is no ventilation provided in the building while the infiltration load is considered zero if the ventilation in the building is provided (SlideShare, 2014).

The infiltration is the air pass through into or out from the indoor through a small cracks, windows, and doors in the building. The outdoor air entering the conditioned space lead to the sensible and latent heat loads in that space since the outdoor air is usually more warm and humid compared to the indoor air (SlideShare, 2014).

Different from infiltration, the ventilation is the outdoor air enters the indoor through the usage of the building's ACMV system, where the outdoor air is cooled and dehumidified before it can be transferred to the indoor. The introduction of outdoor air through the ACMV system is usually for maintaining a positive pressure (relative to the outdoors) within the building to reduce or might even eliminate the infiltration load and will generate an additional load (ventilation load) on the ACMV equipment (SlideShare, 2014).

(e) ***Total Cooling Load***

All the loads as mention in section 2.2.1.2 (a) - 2.2.1.2 (d) are totalized in order to obtain the total cooling load, where the linear functions of cooling load can be represented as the following (Elhelw, 2016; Wang et al., 2014):

$$TCL = aT_{bin} + b \tag{38}$$

Where *a* and *b* are constant. Besides, this load are calculated separately for the occupied and unoccupied periods, where the total load obtained for both hours are multiply with the number of hours of occupied period and unoccupied periods, respetively as shown below (Elhelw, 2016):

$$TCLoc = CL\ x\ occupied\ hours \tag{39}$$

$$TCLun = CL\ x\ unoccupied\ hours \tag{40}$$

After obtaining the total cooling load for both periods, the value from both of them are totalized as shown below (Elhelw, 2016):

$$TCL = TCLoc + TCLun \tag{41}$$

(f) ***Total Energy***

In this research, the bin method is used for calculating the cooling energy of the cooling systems (VRF and ACMV systems) used in this research. The total energy obtained can be calculated by dividing the total cooling load which obtained by equation (41) with the COP of the cooling systems used in this research (Wang et al., 2014):

$$E_{bin} = \frac{TCL_{bin}}{COP_{bin}} \tag{42}$$

The COP of these system can be calculated based on the following equation (Elhelw, 2016):

$$COP_{bin} = Q_{c,bin} / P_{I,\, bin} \tag{43}$$

It should be noted that the COP of these system can be obtained if the power input and cooling capacity at different outdoor temperature (bin temperature) of these systems are known (Elhelw, 2016). The value of cooling capacity and power input of this system are expressed as a function of outdoor temperature, where the linear function for cooling capacity of cooling systems used in the building can be obtained by choosing two cooling capacities and outdoor temperatures based on its operating performance data. Besides, similar procedures are used to find the linear function for power input of these systems (Cizik, 2009; Elhelw, 2016).

Another way to obtain the total energy is by finding the partial load fraction (PLF) of the cooling systems used in the building. The PLF for occupied and unoccupied hours is computed by the following (Elhelw, 2016; McQuiston et al., 2005):

$$PLF = 1 - D_c \; x \; (1 - \frac{Building\ Load}{Unit\ Capacity}) \tag{44}$$

Where the value of degradation coefficient (D_c) can be obtained by the manufacturer or assume to be 0.25 as a default value as been specified by the Air - Conditioning and Refrigeration Institute (ARI) in the U.S (Elhelw, 2016). However, the PLF is assumes as 1 if the building load exceeds these systems's capacity due to the assumption that this system operates continuously in both hours (Elhelw, 2016). Then, the part load for both hours are calculated separately based on the following equation (Elhelw, 2016):

$$Part\ Load = PLF\ x\ TCL \tag{45}$$

The run time of the cooling system used in the building for both hours can be obtained based on the following equations (Elhelw, 2016; McQuiston et al., 2005):

$$Occupied\ run\ time = Occupied\ Part\ Load\ x\ \frac{Occupied\ Hours}{Capacity}\ x\ Occupied\ PLF \quad (46)$$

$$Unoccupied\ run\ time = Unoccupied\ Part\ Load\ x\ \frac{Unoccupied\ Hours}{Capacity}\ x\ Unoccupied\ PLF \quad (47)$$

After obtaining the run time of these systems, the energy can be calculated from the following equation (Elhelw, 2016; McQuiston et al., 2005):

$$Energy = Power\ input\ x\ (Occupied\ run\ time + Unoccupied\ run\ time) \quad (48)$$

According to Papakostas et al. (2008) and Bulut et al. (2001), the total energy also can be obtained by the equation below:

$$E_{bin} = N_{bin,i} = \frac{K_{tot}}{\eta}\ (T_{bin} - T_r)^{\pm} \quad (49)$$

Where the plus subscript on the equation (49) indicates that positive values for heating and negative values is for cooling. Then, the total energy consumption can be obtained as follows (Bulut et al., 2001; Papakostas et al., 2008):

$$E_{tot} = \sum_{i=1}^{m} E_{bin,i} \quad (50)$$

2.2.1.3 Bin Weather Data

To perform the cooling load estimation by using the bin method, the bin weather data in reference with the typical meteorological year (TMY) data such as hourly dry – bulb temperature and relative humidity need to be known (Bulut et al., 2001; Elhelw, 2016; Jin et al., 2006; Krarti, 2011; Papakostas et al., 2008; Wang et al., 2014). Generally, in classical bin method, only the outdoor temperatures are sorted into discrete groups (bins) of equal size, typically 5°F (2.8°C) based on the number of occurence of each bin.

The classical bin method is practical to use only for the buildings with sensible heat load and with no significant thermal mass effects and this method might not provide accurate energy predictions for buildings which have high latent heat load (Krarti, 2011). Besides, according to Jones et al. (2009), the classical bin method does not accurately represent in extremes weather data such as in hot and humid regions, where these regions are better represented by two variables, which are outdoor temperature and humidity. Figure 2.10 show the example of classical bin weather data.

Average of Outdoor Dry-Bulb Temperature Bin (°F)	Number of Hours of Occurrence	Average Coincident Humidity Ratio (lb/lb)
15	1	0.0020
20	42	0.0020
25	154	0.0020
30	291	0.0025
35	354	0.0031
40	641	0.0038
45	623	0.0044
50	665	0.0053
55	741	0.0065
60	882	0.0083
65	905	0.0100
70	1225	0.0128
75	1000	0.0133
80	672	0.0137
85	421	0.0143
90	133	0.0156
95	19	0.0170

Figure 2.10: Example of classical bin weather data for Atlanta, Georgia (Krarti, 2011)

However, the accuracy of the bin method can be improved by using two - dimensional bin weather data, which is available in the ASHRAE handbooks (Jones et al., 2009; Krarti, 2011). The two - dimensional (also referred to as joint - frequency) bin weather data are generated based on two variables, which are the dry - bulb temperature and humidity ratio (Jones et al., 2009; Krarti, 2011). Figure 2.11 show the example of two - dimensional bin weather data.

Average of Humidity Ratio Bin (lb/lb)	Average of Dry-Bulb Temperature Bin (oF)								
	50	55	60	65	70	75	80	85	90
0	0	0	0	0	0	0	0	0	0
0.0020	69	48	29	14	0	0	0	0	0
0.0040	229	183	123	54	35	11	1	0	0
0.0060	238	178	91	75	39	37	10	0	0
0.0080	129	210	196	122	71	85	28	14	0
0.0100	0	122	335	228	140	111	72	40	3
0.0120	0	0	108	312	202	131	156	53	5
0.0140	0	0	0	100	486	245	156	131	42
0.0160	0	0	0	0	237	289	184	142	57
0.0180	0	0	0	0	15	89	62	37	19
0.0200	0	0	0	0	0	2	3	4	7
0.0220	0	0	0	0	0	0	0	0	0

Figure 2.11: Example of two - dimensional bin weather data for Atlanta, Georgia (Krarti, 2011)

In the U.S, the required bin weather data can be obtained in the ASHRAE handbooks but there is a shortage in the information regarding of necessary bin weather data in certain countries such as China, Turkey, and Greek (Bulut et al., 2001; Jin et al., 2006; Özyurt et al., 2009; Papakostas et al., 2007; Papakostas et al., 2008; Papakostas, 1999; Papakostas & Sotiropoulos, 1997; Peng et al., 2009). Therefore, some researchers have generate a bin weather data for their countries based on the TMY data for a past years in a particular location located their countries because the bin weather data varies, depending on the location and year of analysis (Jones et al., 2009).

For example, Jin et al. (2006) has generates bin weather data for Nanjing, China by using measured hourly ambient temperature data for a year while Peng et al. (2009) has generates the bin weather data for 26 cities in China by using the long - term daily weather record and TMY data. This is because of the shortage in the information regarding of bin weather data for these cities (Jin et al., 2006; Peng et al., 2009). Furthermore, bin weather data in four hour periods for a few cities in Greece also had been generated based on monthly average outdoor temperatures (Papakostas et al., 2007; Papakostas et al., 2008; Papakostas, 1999; Papakostas & Sotiropoulos, 1997).

Besides, Bulut et al. (2001) and Özyurt et al. (2009) also have generate bin weather data in four hour periods for a few regions in Turkey due to the uncomplete and undetail bin weather data in Turkey.

The bin weather data can be generated using hourly data such as TMY - 2 files (Krarti, 2011). According to Hui and Cheung (1997), in generating a new bin weather data, the most latest weather data with a longer period of records need to be used in order to obtain more adequate result, where a period of 20 years is considered practical and sturdy.

2.2.2 TRNSYS

The bin method can been used in the computer energy simulation tools including TRNSYS (TRaNsient SYStem) (Krarti, 2011; Papakostas et al., 2008; Shrivastava et al., 2017). The TRNSYS software that was created by Duffy Beckman in 1935 had become a common computer energy simulation tools until today (Shrivastava et al., 2017). TRNSYS is usually use for modeling and simulating any transient systems behavior as well as analyzing the performance and the behaviour of the thermal for thermal solar systems, and electrical and mechanical energy building systems based on the modular structure of these systems (Megri, 2014; Shrivastava et al., 2017). In general, the components used in TRNSYS software are in reference to the mathematical models written in FORTRAN codes and the TRNSYS baseline simulation are usually assembled using IISIBAT 3.0 software (Krarti, 2011; Quesada et al., 2011; Solar Energy Laboratory, 2006; Yau, 2008).

The same as the other energy simulation softwares like EnergyPlus, BLAST, and DOE - 2, TRNSYS software can simulates the energy for both building and heating or cooling system used in the building within hour - by - hour by using hourly weather data for some periods of time, generally one year, where adequate simulation results for the long - term period can be obtained (Eguía et al., 2016; Jin et al., 2006; Krarti, 2011; Papakostas et al., 2008; Quesada et al., 2011). It should be noted that the simulation with hourly time steps is known as a dynamic method (detailed simulation methods) (Krarti, 2011; Papakostas et al., 2008).

Normally, forward method is applied in the TRNSYS simulation for estimating the energy consumption of a building and equipment that operates on the refrigeration cycle such as ACMV system in accordance with a physical description of the building itself including construction details, location, and geometry, and the type and operation of the equipment used in the building (Krarti, 2011). A typical calculation flowchart of detailed simulation for TRNSYS software is presented in Figure 2.12.

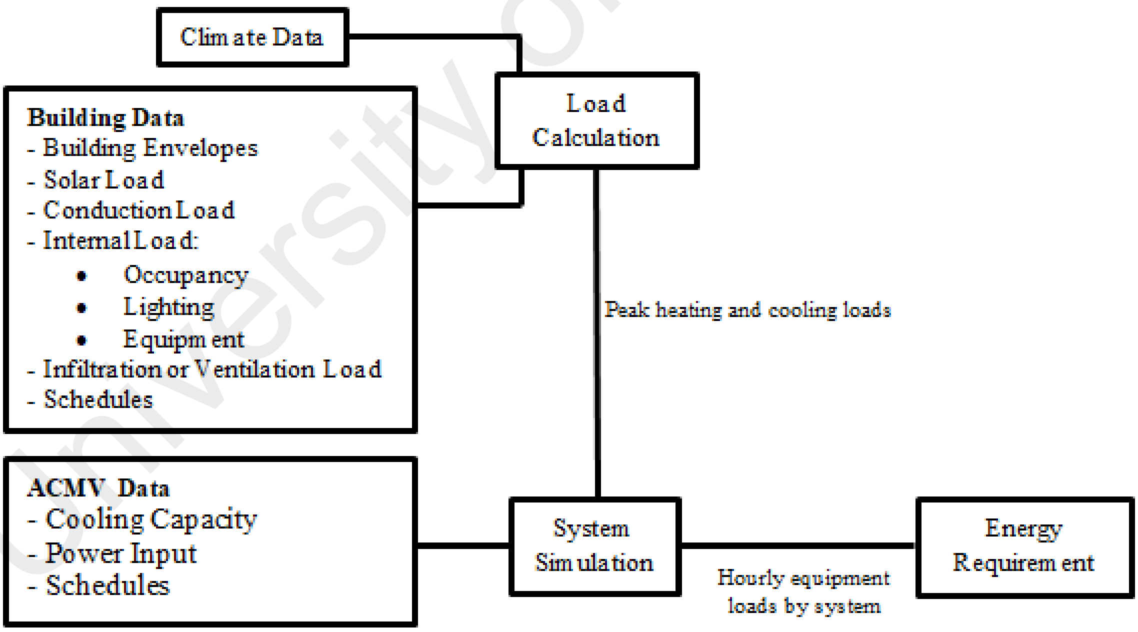


Figure 2.12: Example of flowchart of complete building model in TRNSYS (redrawn based on reference (Krarti, 2011))

In TRNSYS, a model is represented by a components link together based on their constant parameters and input/output variables (Megri, 2014; Riederer et al., 2009; Shrivastava et al., 2017). Besides, the modular nature of TRNSYS software makes the empirical codes of the components can be added into this software (Shrivastava et al., 2017; Yau, 2008). These components can be taken from TRNSYS library and each of these components are represented with one or more equations to estimate specific values of the output based on the inputs, variables, and parameters, where each of these components will communicate with the rest of the system by passing the inputs on to other components (Eguía et al., 2016; Megri, 2014; Quesada et al., 2011; Riederer et al., 2009; Shrivastava et al., 2017). Furthermore, it should be noted that each component is associated to a “Type” number according to their function (Megri, 2014; Riederer et al., 2009). Figure 2.13 shows the example of a connection between the components.

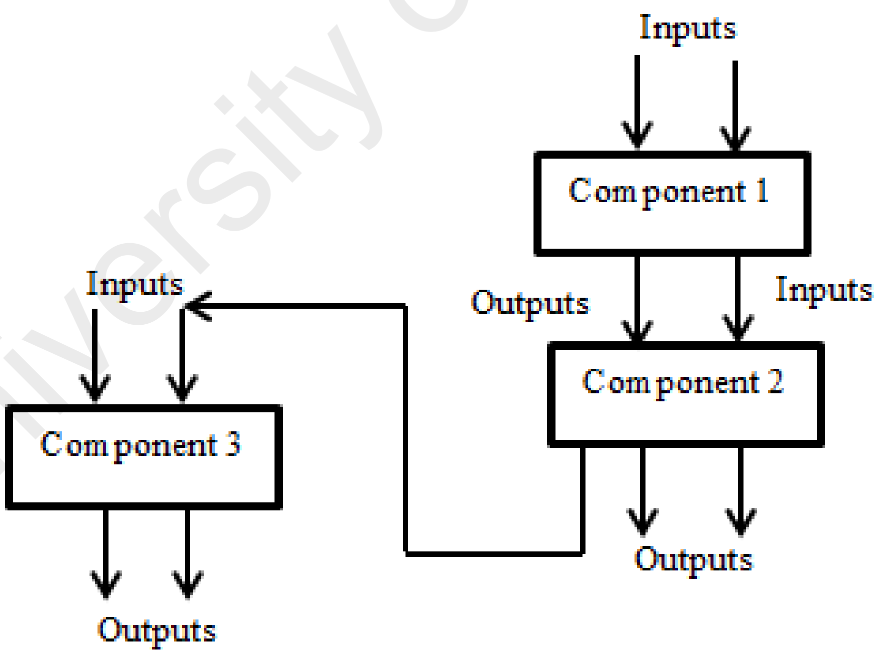


Figure 2.13: Example of a connection between the components (redrawn based on reference (Megri, 2014))

CHAPTER 3: METHODOLOGY

This chapter discusses the research methodologies used in the following chapters which is Chapter 4 and Chapter 5. This research is divided into two case studies according to the types of the building. In case study I, the VRF system is compared with the split type unit in a small building which provides cooling in one zone while for case study II, the VRF system is compared with the multi - split type unit in a large building which provides cooling in several zones. The overall workflow to achieve the objective of the energy performance estimation by using bin method in the target buildings is summarized sequentially as shown in Figure 3.1 below.

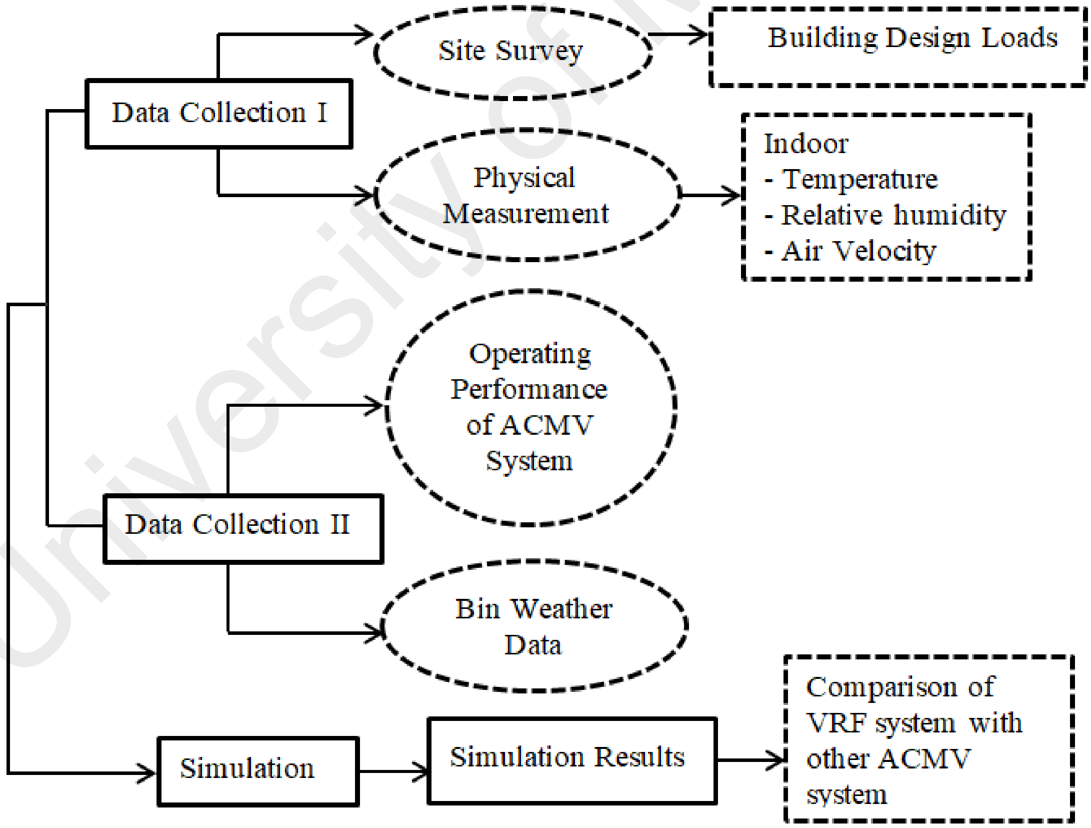


Figure 3.1: Overall methodology

In the present methodology, the data collection is divided by to categories, which are data collection I and data collection II. Data collection I is a data for building design loads and indoor environment including indoor air temperature, relative humidity, and air velocity. All these data are obtained by doing site survey and physical measurement. As for data collection II, the data is operating performance of the ACMV and VRF systems used in this research, which can be obtained by the factory or company, which manufactures these systems and also bin weather data, which adopted from other study by Yau et al. (2018).

After obtaining these data, the baseline simulation for estimating the energy usage of the ACMV and VRF systems used in this research can be created, where the bin method concept is applied in this simulation. It is pertinent to mention that the bin method applied in this simulation is based on the previous study by Elhelw (2016) and Wang et al. (2014). However, since the usage of bin method in the energy simulation is new, there are some components are needed to be created based on the bin method equations by using the FORTRAN software in order to ensure the simulation applied with the bin method can be run.

After the baseline simulation is created, the simulation results for both ACMV and VRF systems used in this research are compared in term of annual energy consumption. Besides, the total annual energy consumed for both systems also have been calculated manually and been compared to the simulation results in term of uncertainty in order to examine whether or not the errors present in the simulation results are at a satisfactory level.

3.1 Data Collection I

The building design loads including solar load, conduction load, internal loads, and infiltration or ventilation load of the target buildings are important since these loads will be used to estimate the energy consumption by using bin method and mostly acted as the inputs in the TRNSYS simulation. These loads can be obtained by doing site survey and physical measurement on the target building.

3.1.1 Site Survey

The solar and conduction loads of the target buildings can be obtained if the building envelopes characteristics including U - factor value and shading coefficient of the building components, which are walls, doors, roof/ceilings, floor, and windows of the these buildings are known, where these characteristics can be known based on the types of building envelopes. The types of target building envelopes (types of walls, doors, roof/ceiling, floor, and windows) can be known by conducting the site survey in the target buildings.

The sources of internal loads of the building are the occupancy, light, and equipment used in the building. These loads can be obtained if the schedule of the operating time of the target building; the number of occupant, their activity and cooling load factors (CLF); the quantity of the equipment and their heat gain which can be obtained from ASHRAE Table (ASHRAE, 2014); and the quantity, power usage, CLF and ballast factor of the lighting are known (SlideShare, 2014). This information can be obtained by doing a site survey on the target buildings.

3.1.2 Physical Measurement

The indoor air temperature, relative humidity, and velocity in the target buildings are measured by using the Alnor velometer AVM440 - A. Before performing a physical measurement, the Alnor velometer AVM440 - A is stabilized for about 5 minutes and then, in every thirty minutes, the indoor air relative humidity, temperature, and velocity for occupied hours of this building are measured and recorded. It should be noted that the procedures mentioned are based on the study by Yahaya et al. (2015). These data is taken for a day and assume to be the same for each day along the year.

Table 3.1: Equipment used in this research

Parameters	Equipment	Descriptions
Length Height	 Measuring Tape	Measuring tape is used to measure the length and the height of the target building. In addition, the length and height of the walls, doors, roof, floor, and windows of the target building is also measured by using this equipment.
Indoor air: 1.Temperature 2. Velocity 3.Relative humidity	 Alnor velometer AVM440 - A	The function of this meter is to find the indoor air temperature, velocity, and relative humidity in the target building. *Operating range* Temperature: -10 to 60 °C Velocity: 0 to 30 ms^{-1} RH: 5 to 95% *Accuracy* Temperature: ± 0.3 °C Velocity: ± 3% of reading or ± 0.015ms^{-1} RH: ± 3% *Resolution* Temperature: ± 0.1 °C Velocity: 0.01 ms^{-1} RH: 0.1%

The infiltration or ventilation load of the target buildings can be obtained if the volume flow rate and relative humidity of indoor air in these buildings are known. The indoor air relative humidity and temperature can be obtained directly by the Alnor velometer AVM440 - A. However, volume flow rate of air can be obtained by using a simple calculation based on the indoor air velocity obtained by this meter (Intelligent Energy Europe, 2011):

$$V_a = \textit{indoor air velocity x area of the target building} \quad (51)$$

Note that the ventilation load is considered to be zero if there is no ventilation provided in the building while the infiltration load is considered zero if the ventilation in the building is provided because the introduction of outdoor air through the ventilation can maintaining a positive pressure (relative to the outdoors) within the building which can reducing or might even eliminating the infiltration load and will generate an additional load on the ACMV equipment (SlideShare, 2014).

3.2 Data Collection II

Since the bin method is used for the energy analysis for the ACMV and VRF systems in this research, the bin weather data for the particular location necessary to ensure the energy analysis of these systems can be done by using the bin method. For this research, the bin weather data in Petaling Jaya, Malaysia in years 2007 to 2016 adopted from Yau et al. (2018) as shown in Figure 3.2 is used as the replacement of the typical meteorological year - 2 (TMY2) weather data files in Petaling Jaya, Malaysia.

The value of the COP of the VRF and ACMV systems used in this research is important for estimating the energy consumption of these systems by using the bin method as mention in section 2.2.1.2 (f). The COP of these systems can be obtained if the power input and cooling capacity at different outdoor temperature of these systems are known. This information can be obtained in their operating performance data. These data can be obtained by the factory or company which manufactured these systems, where they only give permission to use these data for academic purpose and nonprofit research only. It should be noted that the further detail on the usage of operating performance data of these systems will be shown in the Chapter 4 and Chapter 5 based on the case study of this research.

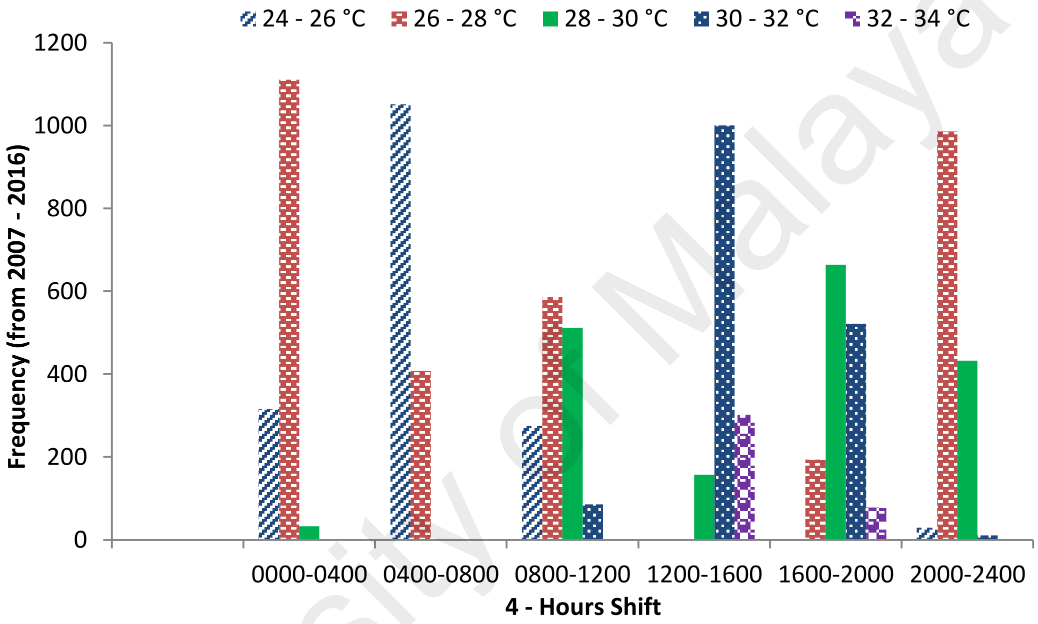


Figure 3.2: The bin weather data details in Petaling Jaya in years 2007 to 2016 (adopted from Yau et al. (2018))

3.3 TRNSYS Simulation

In this research, the condition of the VRF and ACMV systems used in this research is simulated in term of annual energy consumption for further analysis and design. With all the data obtained as mention in section 3.1 and 3.2, the appropriate TRNSYS baseline simulation applied with the bin method concept can be created. As mention earlier, the bin method applied in this simulation is based on the previous study by Elhelw (2016) and Wang et al. (2014), where the explanation regarding of this method in detail is discussed in the previous chapter, which is Chapter 2, section 2.2.1. Figure

3.3 shows the created TRNSYS baseline simulation of the target buildings. The arrows in this simulation indicating the energy calculation flow applied with the bin method concept. It should be noted that the red lines in Figure 3.3 represent the simulation of the target buildings in unoccupied operating hours.

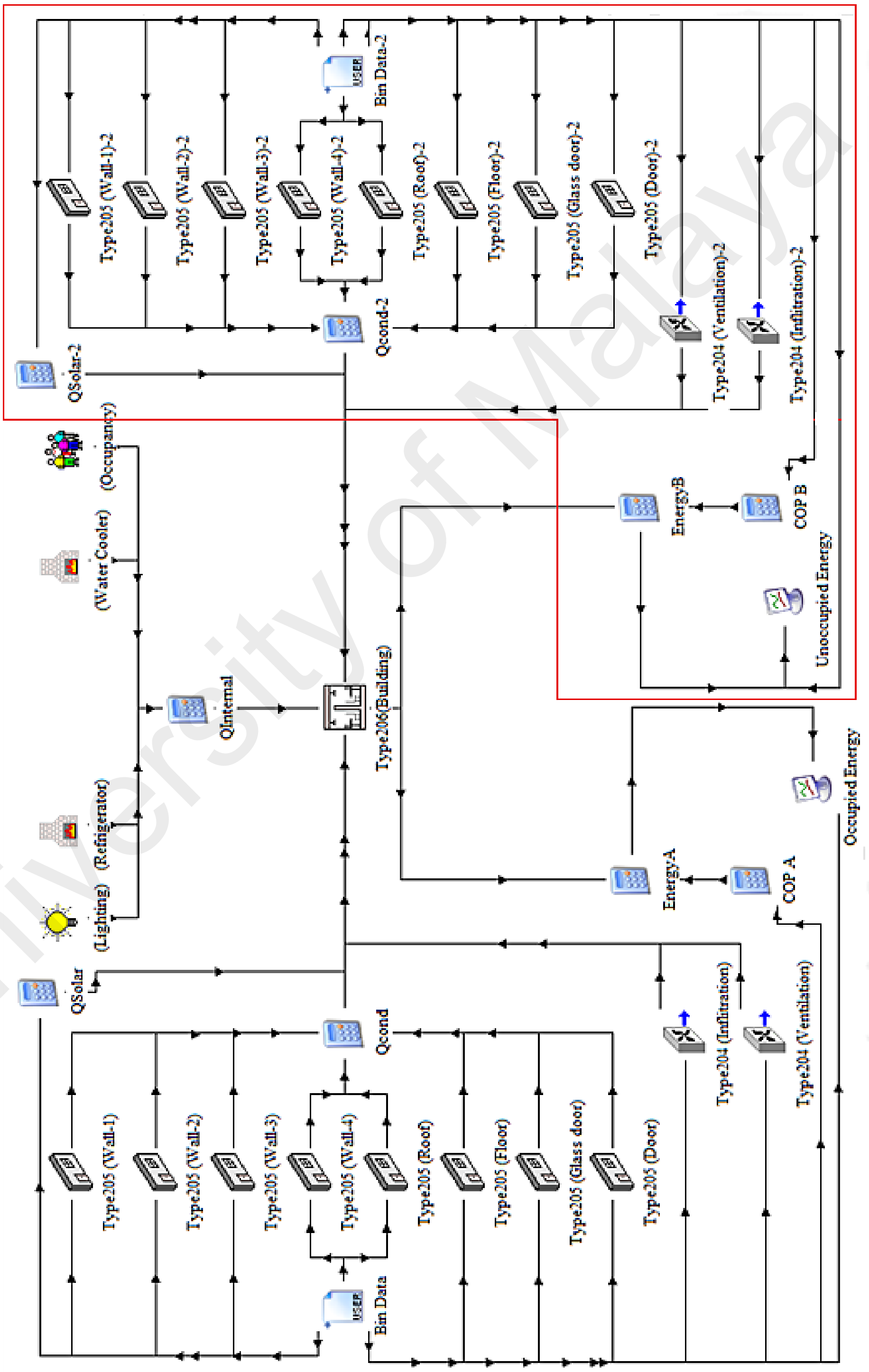


Figure 3.3: TRNSYS baseline simulation of the target buildings

It is pertinent to mention that the following assumptions have been made in this simulation:

1. The present work assumes the effects of transport delays and dynamics because of the thermal capacitance for refrigerant pipes (copper pipes), separation tubes and/or header, air ducts, and transitions are insignificant.
2. The present research assumes that heat load from the adjacent walls are negligible.

In order to make the simulation in Figure 3.3 to be easily understood by the reader, a simplified TRNSYS baseline simulation of the target buildings for occupied hours as shown in Figure 3.4 is created. It should be noted that the simplified TRNSYS baseline simulation of the target buildings for the unoccupied hours is the same as occupied hours.

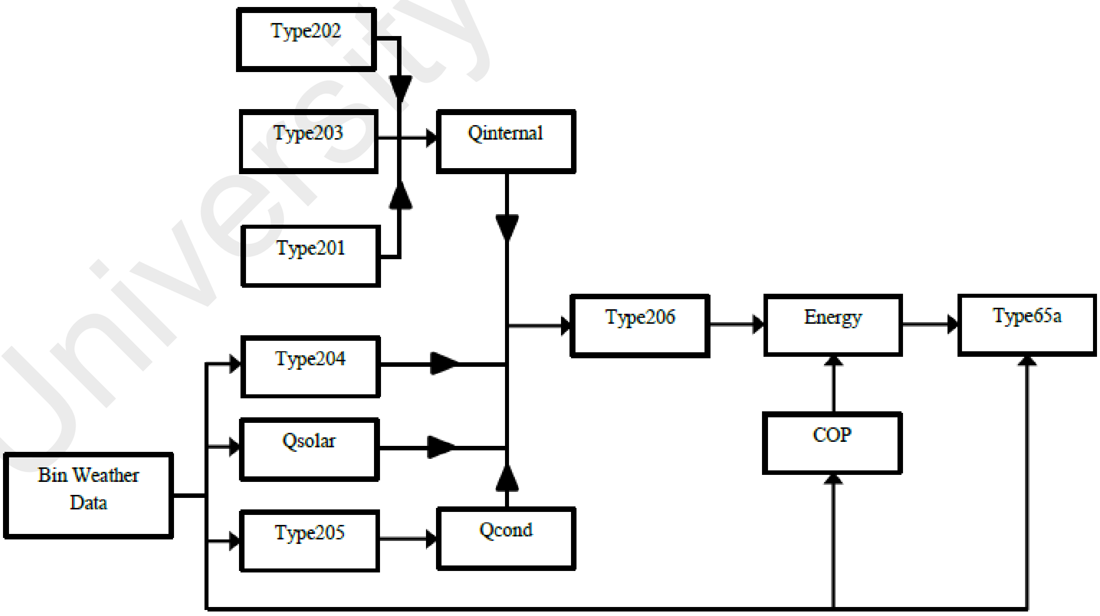


Figure 3.4: Simplified TRNSYS baseline simulation of the target building (occupied hour)

3.3.1 Creating TRNSYS New Components

Since the usage of bin method in the energy simulation is new, there are some components are needed to be created in order to ensure the bin method can be applied in the simulation. Thus, a good knowledge of computer programming, which is FORTRAN software is necessary in order to create these new components (Krarti, 2011; Solar Energy Laboratory, 2006). There are a few steps in creating a new component, which are (Solar Energy Laboratory, 2006):

1. Launch the TRNSYS software and go to File/New.
2. Select "Component" tab and then, fill in the data needed with the component's object and its Type number.
3. Add the component inputs, parameters and outputs in the "Variables" tab.
4. Save the component and export the created component as FORTRAN as shown in Figure 3.5.

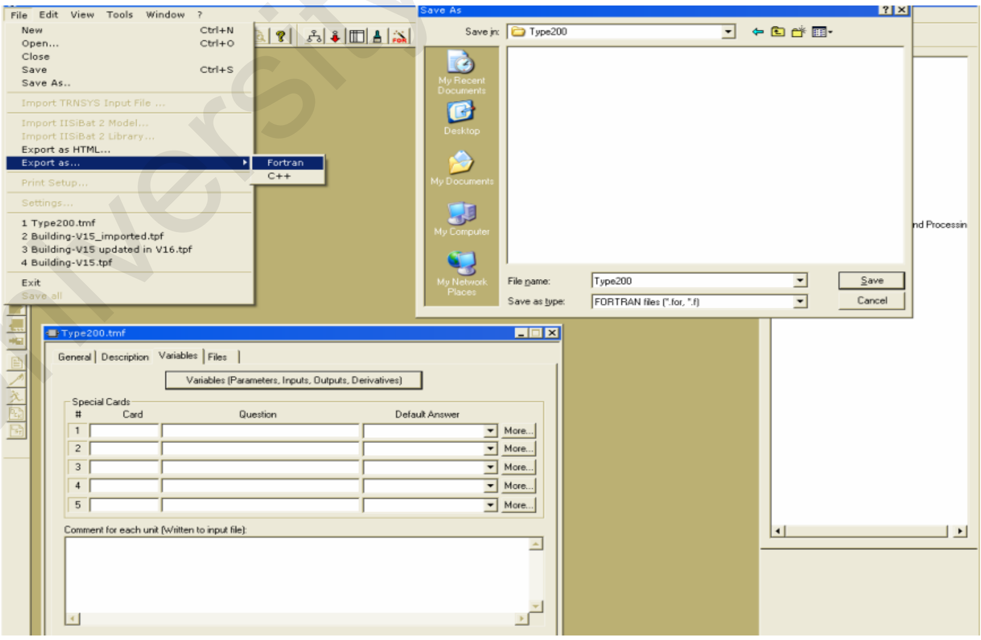


Figure 3.5: Exporting the created component as FORTRAN

5. Then, the TRNSYS software will automatically generate a FORTRAN skeleton for that component by the helps of the Compaq Visual FORTRAN 6.6. Figure 3.6 shows the example of FORTRAN code in the FORTRAN skeleton. It should be noted that Compaq Visual FORTRAN 6.6 is already installed on the machine/computer.

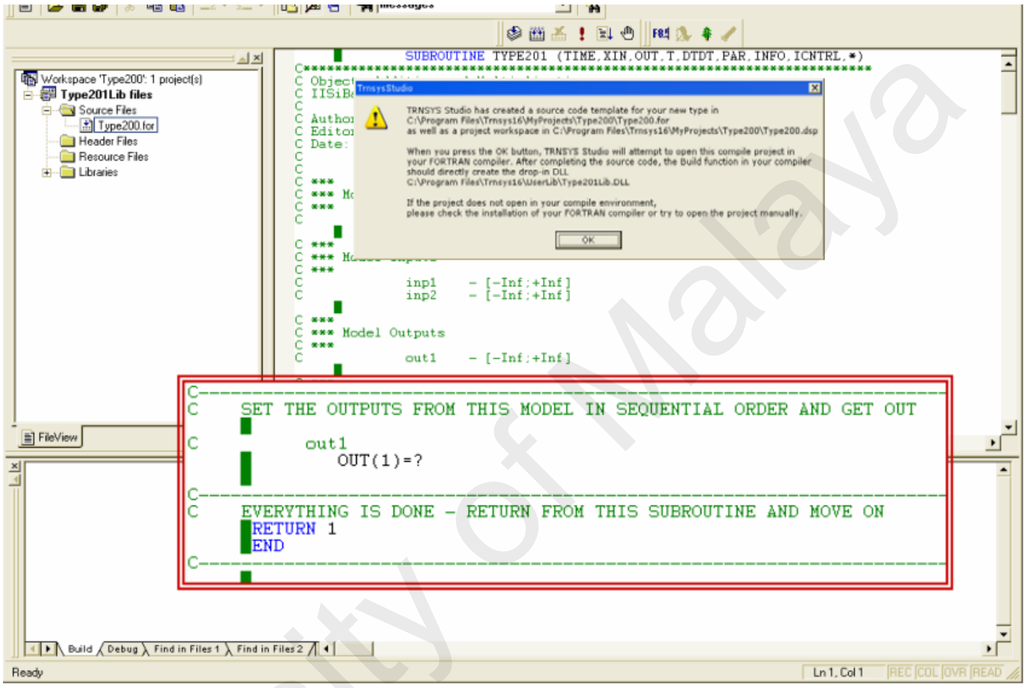


Figure 3.6: Example of FORTRAN code in the FORTRAN skeleton

6. Then, type a required codes in the FORTRAN skeleton as shown in Figure 2.15 and press F7 to build the 'DLL.' to generate "Typexxx.dll" file in the 'UserLib\ReleaseDLLs' folder. Note that the codes are the actual equations (bin method equation), which relate the inputs and parameters to the outputs.
7. Save the codes and now, the new component can be load and used in the TRNSYS software.

However, the equation model also is used instead of creating the new component because some of the components cannot be created due to the some complex bin method equation. Furthermore, the other components used in this simulation including Type9a (Data reader) and Type65a can be taken from TRNSYS library itself. Table 3.2 shows the created components and also existed TRNSYS components used in the simulation. Note that the FORTRAN codes for the created components will be shown in the Appendix section.

Table 3.2: Components and equation models used in TRNSYS simulation

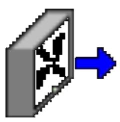
Created Components	
Types	**Descriptions**
 Type201	This component is used to calculate the heat generated by people in the target building. **Inputs:** Maximum load produced by the people in the target building and CLF **Parameter:** - **Output:** Occupancy load
 Type202	This component is used to calculate the heat generated by lighting installed in the target building. **Inputs:** Maximum load produced by the people in the target building, CLF and ballast factor **Parameter:** - **Output:** Lighting load
 Type203	This component is used to calculate the total load produced by the equipment in the target building. **Inputs:** Maximum load produced by the people in the target building and average usage **Parameter:** - **Output:** Equipment load used in the target building
 Type204	This component is used to calculate the ventilation or infiltration load of the target building **Input:** Bin temperature **Parameters:** density of air, specific heat of air, volumetric air flow, indoor and outdoor relative humidity, and indoor temperature **Outputs:** Sensible ventilation or infiltration load and latent ventilation or infiltration load

Table 3.2 continued

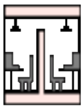
Created Components	
Types	**Descriptions**
 Type205	This component is used to calculate the conduction loads of the target building **Input:** Bin temperature **Parameters:** area and rate of heat loss of the elements of the target building, and indoor temperature **Output:** Conduction load
 Type206	Building structure subject to internal, solar, conduction and ventilation loads for occupied hour and unoccupied hour. **Inputs:** internal, solar, conduction and ventilation loads for occupied hour and unoccupied hour **Parameter:** - **Output:** Total building loads for occupied hour and unoccupied hour
Components Taken from TRNSYS Library	
Types	**Descriptions**
 Type9a (Data Reader)	This component reads bin weather data for both occupied and unoccupied periods which occurs in the form of bin temperature with the hours of occurrence for each bin. **Input:** Bin weather data (in form of .txt file) **Parameter:** - **Output:** Bin temperature
 Type65a	The online graphics component is used to display simulation results.
Equation Models	
Types	**Descriptions**
 Qinternal	This equation component is used to calculate the total of internal loads which are load generated by equipment, people and lighting. **Inputs:** All internal loads (Type201, Type202, and Type203) **Parameter:** - **Output:** Total of internal loads
 Qcond	This equation component is used to calculate the total of conduction loads which is conduction load of roof, walls, floor, windows and doors. **Inputs:** Conduction loads for roof, walls, floor, windows and doors of the target building (Type205) **Parameter:** - **Output:** Total of conduction loads

Table 3.2 continued

Equation Models	
Types	**Descriptions**
 Qsolar	This equation component is used to calculate the solar loads of the target building. **Input:** Bin temperature **Parameter: -** **Output:** Total of solar load
 COP	This equation component is used to calculate the COP of the ACMV system based on the cooling capacity and power input. **Input:** Bin temperature **Parameter: -** **Output:** Total of COP of ACMV system configuration which been used as the subject of this simulation
 Energy	This equation component is used to calculate the total energy consumed by the target building. **Input**: Total building load of the target building (Type206) **Parameter: -** **Output:** Total energy consumption

Type201, Type202, and Type203 are created for calculating internal loads including occupancy load, lighting load, and equipment load. These loads are calculated for occupied hours only because all lighting and equipment are assume to be off during unoccupied hours. Type201 component is developed for calculating the sensible and latent heat gains of the occupancy load, where both heat gains are calculated separately. This component is created based on equations (28) and (29). For calculating the energy gain by lighting, Type202 is created based on the equation (26). For calculating the energy gain by equipment, Type203 is created based on the equation (31). It is pertinent to mention that all the equations used in these components (Type201, Type202, and Type203) are mentioned in section 2.2.1.2 (c). Then, all these components are totalized in the equation model of "Qinternal" (refer Table 3.3) to obtain the overall total internal loads produced in the target buildings.

Type204 component is developed for calculating the sensible and latent components of the ventilation or infiltration load of the target building for occupied and unoccupied hours. The sensible ventilation or infiltration load can be obtained based on the equations (34) and (36), respectively while the latent ventilation or the infiltration load can be obtained based on the equations (35) and (36), respectively. These equations are mentioned in section 2.2.1.2 (d). Type205 component is also created to calculate the conduction loads for both hours based on the equation (20) as mention in section 2.2.1.2 (b). Then, these elements are totalized in the equation model of "Qcond" (refer Table 3.3). The equation model is used instead of creating the new component for the solar load calculation for both periods due to its complex calculation. The equation model for solar load, which is "Qsolar" (refer Table 3.3) are expressed as a function of the bin temperature as shown in equation (10) as stated in section 2.2.1.2 (a).

It should be mention that Type206 component is created to model a building structure subjected to internal, solar, conduction and infiltration or ventilation loads for both occupied and unoccupied hours. It differs from the simple building models (i.e. Type12 and Type56), where this component neglects the control scheme of the target building and is suitable to be used for calculating the building load of the targeted building by using the bin method. The components including equation models of "Qinternal", "Qcond", and "Qsolar", and Type204 are linked to Type206 as inputs, where this component will then total up the mentioned components for both occupied and unoccupied hours and then, will read them as the outputs. Eventually, the just mentioned outputs are linked to the equation model of "Energy" (refer Table 3.3) for occupied and unoccupied hours, in which they are used to calculate the energy consumption for both hours by using the bin method equation. The equation used in the current model is based on the equation (42) as mention in section 2.2.1.2 (f).

As shown in equation (42), the COP of the VRF and ACMV systems used in this research for occupied and unoccupied hours can be obtained using the equation model of "COP" (refer Table 3.3), where the equation used in this model is based on the equation (43). In this model, the outdoor temperature (bin temperature) is acted as the function for the equation of the power input and cooling capacity for both systems, where the function of cooling capacity of these systems can be obtained by choosing two cooling capacities and outdoor temperatures based on their operating performance curve. Note that the operating performance curve for both systems is based on the obtained operating performance data as mention section 3.2. The similar procedures are used to find the linear function of the power input for both systems. Then, the just mentioned "COP" model for both occupied and unoccupied hours are linked to the equation model of "Energy" for both hours, respectively.

In this simulation, the bin weather data in Petaling Jaya, Malaysia in years 2007 to 2016 adopted from Yau et al. (2018) as shown in Figure 3.2 in section 3.2 is used as the replacement of the typical meteorological year - 2 (TMY2) weather data files in Petaling Jaya, Malaysia. Type9a (data reader) is used to read the bin weather data for occupied and unoccupied hours in this simulation. This data reader read the user supplied data file (in form of .txt file), where the first line of data corresponds to the simulation start time. Note that the bin temperature from the bin weather data acts as an input for some of the components (Type204 and Type205) and equation models ("Qsolar" and "COP") for both hours in the present simulation.

The function of the output plotter (Type65a) is to analyze the final results from the simulation and also helps in fine tuning for the simulation. When the simulation results are obtained, the comparison between the VRF and ACMV systems used in this research in term of annual energy consumption and economical analysis as well as the comparison between the simulation results with the manually calculated results in term of uncertainty are conducted, in which will be discussed and analyzed in Chapter 4 and 5.

CHAPTER 4: CASE STUDY I - BEZAIRE BUILDING

This chapter focuses on the estimation of the annual energy consumption of mini VRF system used in the small building which needs cooling in single zone and the result obtained will be compared with the split type unit system under the same outdoor and indoor conditions, and followed by the comparison of the economical analysis for both systems. The comparison is made to evaluate whether or not the VRF system has a great potential for energy savings when compared with the split type unit system. In addition, the uncertainty between the simulation results with the manually calculation results for annual energy consumption for both systems also will be discussed and analysed in this chapter.

4.1 Target Building Overview

4.1.1 Target Building Description

Bezaire building as shown in Figure 4.1 is a shop lot building located in Petaling Jaya, Malaysia. This building has four floors, where the net area of this building is 138.75 m^2 per floor and the height for each floor is 3.11 m.

Figure 4.1: Bezaire building

This building has operated for 10 hours, which is from 8 AM until 6 PM on weekdays. The first floor of this building is used as the ACMV showroom while the other floors are used as the office. For this research, the assessments are only conducted on the first floor since only the first floor of this building is using the mini VRF system while the other floors are using the split type unit air - conditioning system. The first floor of this building consists of a large hall, one store, and toilet, where the only conditioned space is in the hall, which has an area of 82.38 m^2 and this floor accommodates two occupants which work as receptionists. Figure 4.2 shows a graphical description of the first floor of the target building function layout while Figure 4.3 shows a plan view of the first floor of the target building.

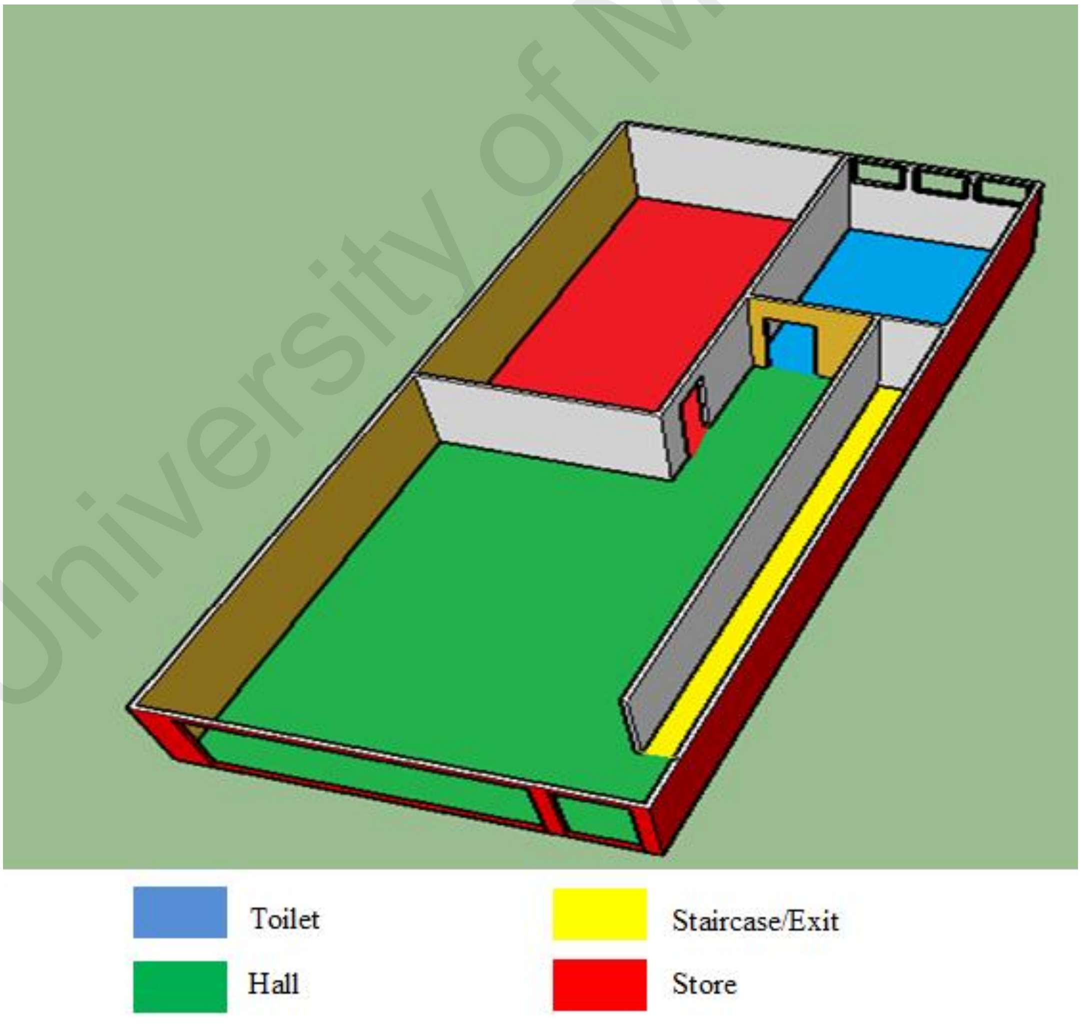


Figure 4.2: First floor of the target building function layout

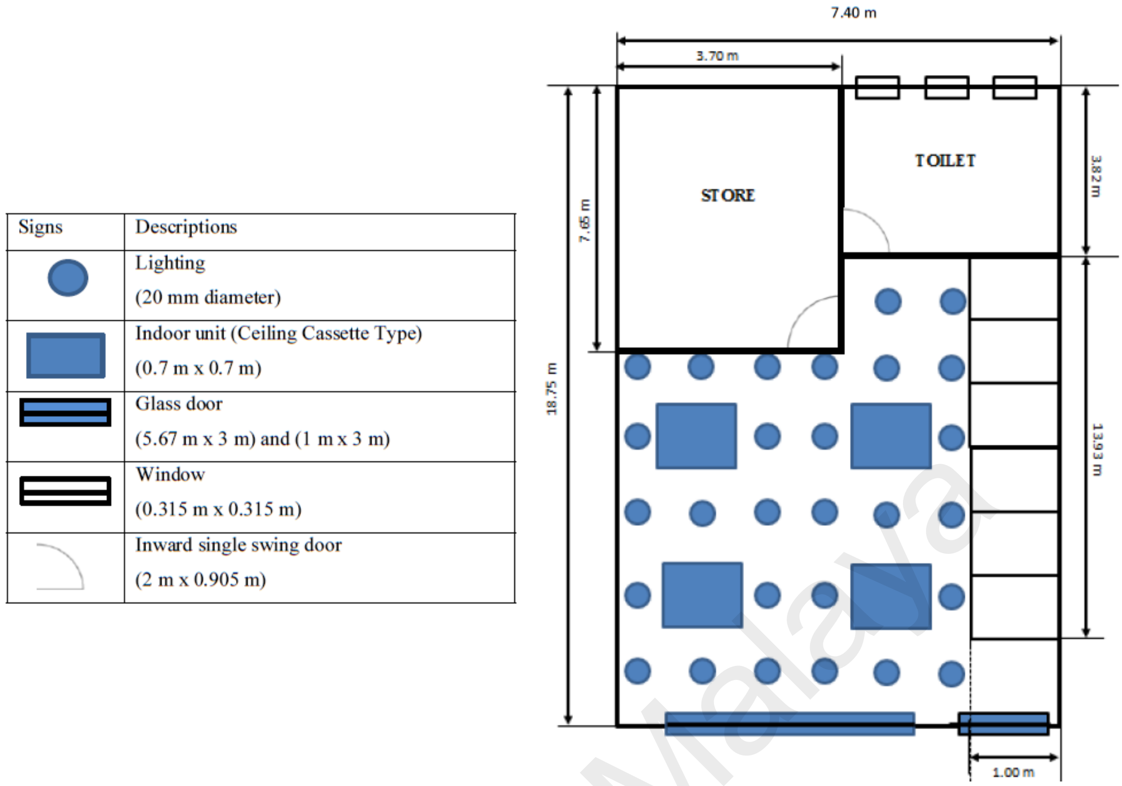


Figure 4.3: Top plan view of the first floor of target building

4.1.2 Target Building Design Loads

All the data/details regarding target building design loads including solar load, conduction load, internal loads, and ventilation or infiltration load will be used in the simulation as mentioned in section 3.3.

4.1.2.1 Solar and Conduction Loads

The solar and conduction loads of the target building can be obtained if the target building envelope characteristics are known. Table 4.1 shows the target building envelope characteristics.

Table 4.1: Target building envelope characteristics

Envelope	Description	U – factor ($W/m^2.K$)	SC	Unit	Area (m^2)
Roof	• Outside air film • 9.8'' RC beam • 1" Cement plaster (0.5" on both sides) • 2" Fiber glass • 0.5" Gypsum board • Inside air film	0.38	-	1	82.38
Floor	• Outside air film • 9.8'' RC beam • Inside air film	2.89	-	1	82.38
Wood Door	• 35'' Wood solid core flush door	1.48	-	2	1.81
Glass Door	• 1/4'' Double glazing uncoated clear glass	0.7	0.61	1	20.01
Exterior Wall a	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	44.58
Exterior Wall b	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	46.43
Exterior Wall c	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	18.03

4.1.2.2 Infiltration Load

For this target building, only the infiltration load is calculated and the ventilation load is considered to be zero because there is no ventilation provided in this building. Infiltration load calculation is depending on the volume flow rate of air entering the building or room. The specific heat (*Cp*) of air is obtained by the table of ideal - gas specific heats of various common gases (Çengel & Boles, 2014).

Table 4.2: Infiltration load details

ρ_{air} (kg/m^3)	Cp_a (KJ/kg.K)	Indoor Air Velocity (m/s)	Indoor V_a (m^3/s)	RH Indoor (%)	RH Outdoor (%)
1.225	1.005	0.0018	0.15	64.65	80

4.1.2.3 Internal Loads

The sources of internal loads of the building are the occupancy, lights, and equipment used in the building itself. These loads has a constant value due to these loads does not affected by the outdoor temperature.

Table 4.3: Internal loads details

<table>
<tr><th colspan="7">Occupancy Detail</th></tr>
<tr><td rowspan="2">Quantity</td><td rowspan="2">*CLF</td><td rowspan="2">Type of Activity</td><td colspan="2">*Energy / Person (kW)</td><td colspan="2">Total Occupancy Load (kW)</td></tr>
<tr><td>Sensible Load (kW)</td><td>Latent Load (kW)</td><td>Sensible Load (kW)</td><td>Latent Load (kW)</td></tr>
<tr><td>2</td><td>1</td><td>Light work and sit</td><td>0.075</td><td>0.055</td><td>0.150</td><td>0.110</td></tr>
<tr><th colspan="7">Lighting Detail</th></tr>
<tr><td>Quantity</td><td>Type</td><td>*CLF</td><td>*Ballast Factor</td><td>Power / Unit (kW)</td><td colspan="2">Total Lighting Load (kW)</td></tr>
<tr><td>28</td><td>Bulb light</td><td>1</td><td>1.0</td><td>0.032</td><td colspan="2">0.896</td></tr>
<tr><th colspan="7">Equipment Detail</th></tr>
<tr><td colspan="2">Equipment</td><td colspan="2">Quantity</td><td>Average Usage</td><td colspan="2">Heat Gain (kW)</td></tr>
<tr><td colspan="2">Water Cooler</td><td colspan="2">1</td><td>-</td><td colspan="2">0.350</td></tr>
<tr><td colspan="2">Refrigerator</td><td colspan="2">1</td><td>-</td><td colspan="2">0.690</td></tr>
</table>

*Data obtained from ASHRAE Table (ASHRAE, 2014)

4.2 Cooling System Overview

4.2.1 Mini VRF System

The mini VRF system with 12 HP serves the entire hall in the first floor of the target building without the ventilation system. This system is used for maintaining the indoor temperature at 24 °C and expected to operate from 8 AM until 6 PM on weekdays. This system is assumed to be fully operated at all time of the day throughout the entire year.

Since this system does not provides with the ventilation, direct expansion principle is used for operating this system, where heat is directly delivered into or out the conditioned space by flowing the refrigerant to evaporators located near or within the conditioned space (Bhatia, 2012; Wang, 2014). Furthermore, the outdoor unit of this system is connected with four indoor units and placed on the outdoor as shown in Figure 4.4.

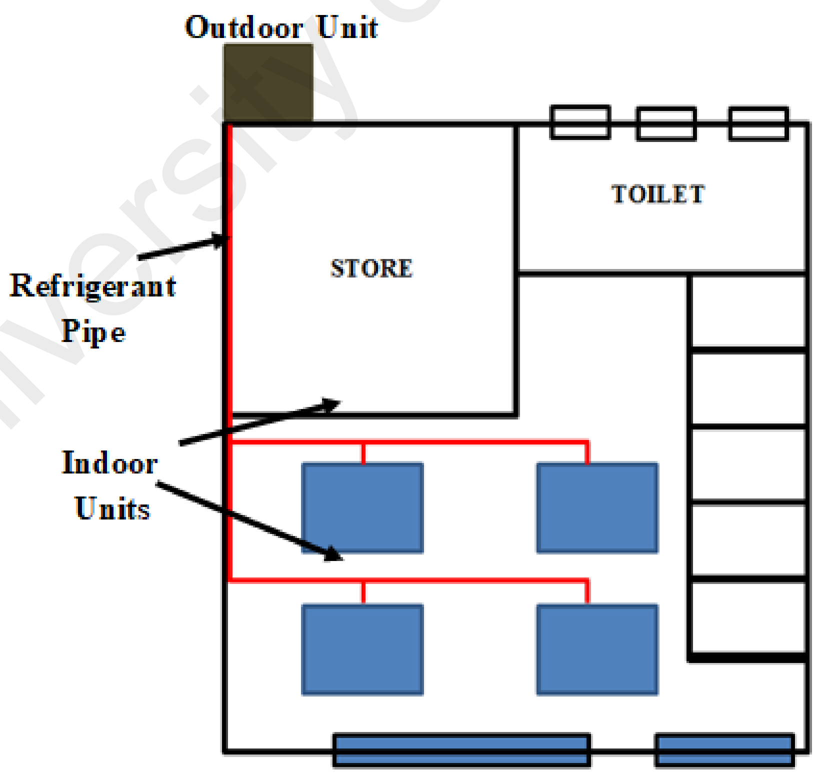


Figure 4.4: Schematic diagram of mini VRF system

The COP of the mini VRF system used in the target building can be obtained if the power input and cooling capacity at different outdoor temperature of this system are known. In this research, the bin temperature which also known as outdoor temperature is acts as the function in a linear function of the cooling capacity and power input for this system as shown in Table 4.4.

Table 4.4: Operating performance details of mini VRF system

Operating Performance	Linear Function (kW)
Cooling Capacity	40.85 - 0.11(T_{bin})
Power Input	5.12 + 0.165(T_{bin})

The linear function for cooling capacity and power input for this system can be obtained based on the operating performance curve of this system as shown in Figure 4.6. It should be noted that the operating performance curve for this system is based on the operating performance data of this system (obtained from the company which manufactured this system) as shown in Figure 4.5.

Model: AVWT-114TESSX Remarks: Q = Cooling Capacity; P = Power Input

Combination (%)	To \ Ti	14		16		18		19		20		22		23		24	
		Q	P	Q	P	Q	P	Q	P	Q	P	Q	P	Q	P	Q	P
100%	10	22.0	3.26	23.8	3.40	31.8	5.08	33.5	8.35	34.8	8.46	36.6	8.61	37.4	8.66	38.4	8.72
	12	22.0	3.26	23.7	3.40	31.8	5.08	33.4	8.37	34.8	8.48	36.6	8.61	37.4	8.66	38.3	8.72
	14	22.0	3.30	23.8	3.42	31.9	5.10	33.6	8.39	34.7	8.51	36.5	8.63	37.3	8.68	38.2	8.75
	16	22.0	3.34	23.8	3.47	31.8	5.17	33.5	8.43	34.6	8.55	36.5	8.68	37.3	8.73	38.2	8.79
	18	22.1	3.38	23.9	3.53	31.9	5.21	33.4	8.49	34.7	8.61	36.5	8.77	37.3	8.81	38.2	8.86
	20	22.1	3.47	23.9	3.61	31.8	5.31	33.5	8.58	34.7	8.70	36.5	8.83	37.2	8.88	38.1	8.93
	22	22.1	3.57	23.8	3.71	31.8	5.40	33.6	8.66	34.7	8.79	36.6	8.95	37.3	8.99	38.0	9.05
	24	22.0	3.67	24.0	3.81	31.8	5.52	33.6	8.79	34.7	8.91	36.5	9.06	37.3	9.11	38.1	9.16
	25	22.0	3.73	23.9	3.89	31.8	5.58	33.5	8.86	34.6	8.98	36.5	9.14	37.3	9.19	38.1	9.24
	26	22.0	3.85	23.9	4.01	31.8	5.71	33.5	8.99	34.6	9.10	36.5	9.27	37.3	9.31	38.1	9.38
	28	21.9	4.12	23.9	4.26	31.8	5.96	33.5	9.24	34.6	9.36	36.5	9.54	37.3	9.56	38.1	9.64
	30	21.9	4.37	23.9	4.51	31.8	6.21	33.5	9.49	34.6	9.61	36.5	9.81	37.3	9.81	38.1	9.91
	32	21.9	4.73	23.9	4.91	31.8	6.63	33.5	9.89	34.6	10.00	36.3	10.21	37.0	10.21	37.6	10.30
	34	21.9	5.12	23.9	5.30	31.8	7.04	33.5	10.30	34.6	10.40	36.1	10.60	36.7	10.60	37.2	10.70
	35	21.9	5.32	23.9	5.49	31.8	7.25	33.5	10.50	34.6	10.60	36.1	10.79	36.6	10.79	37.0	10.89
	36	21.7	5.61	23.7	5.79	31.4	7.51	33.0	10.76	34.0	10.83	35.4	10.99	35.8	10.99	36.2	11.07
	38	21.3	6.17	23.2	6.38	30.7	8.02	32.1	11.27	32.9	11.30	34.0	11.38	34.4	11.38	34.7	11.42
	40	20.9	6.75	22.8	6.97	30.1	8.53	31.2	11.78	31.8	11.78	32.6	11.78	33.0	11.78	33.3	11.78
	43	18.6	7.42	20.4	7.63	27.8	9.19	28.7	12.48	29.5	12.45	30.3	12.46	30.7	12.47	31.0	12.48

Figure 4.5: Operating performance of mini VRF system (obtained from Poh (5 April 2016))

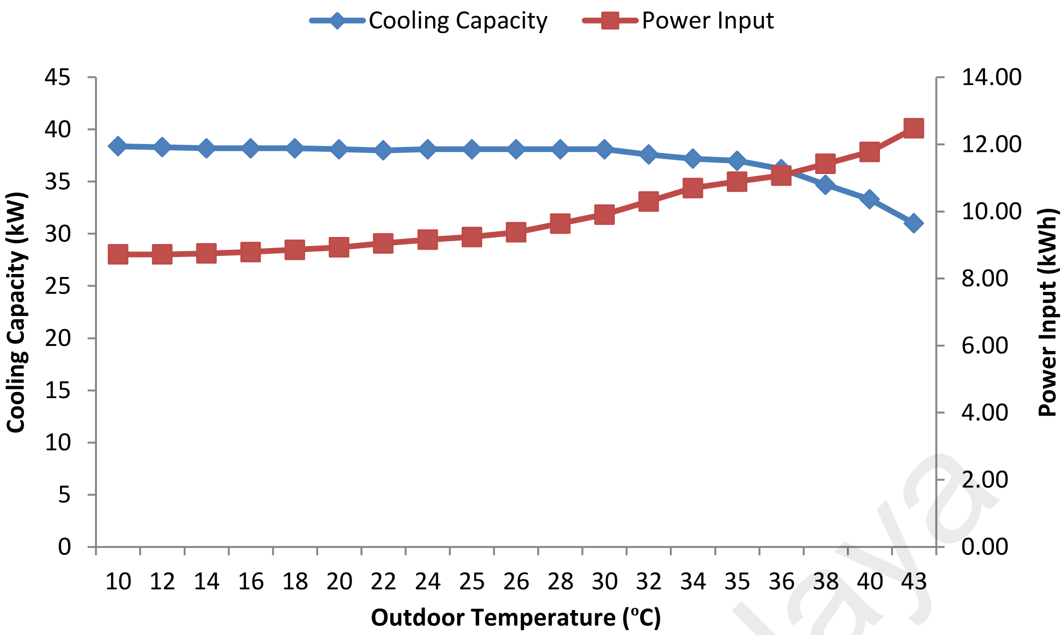


Figure 4.6: Performance curve of mini VRF system

From Figure 4.6, it clearly shows that the outdoor temperature can affect the power input of the system that operates on the refrigeration cycle, where the power input of this system increases when the outdoor temperature increases. However, the cooling capacity of this system is remains constant as the outdoor temperature increases due the ability of the multi - compressors and EEVs in this system to control the flow rate of refrigerant in each indoor unit by reducing or stopping the flow of refrigerant to each indoor unit when superheat occur in the VRF system (Bhatia, 2012; Li et al., 2016).

The cooling capacity of this system started to decrease at certain point of the outdoor temperature due to the decreasing of enthalpy difference of refrigerant at the inlet and outlet of evaporator in the indoor unit. This process occurred because the temperature of the refrigerant at the inlet of evaporator in the indoor unit increasing with the increasing of the outdoor temperature while the refrigerant at the outlet of the evaporator remains constant at any outdoor temperature (Zheng & Liang, 2010).

4.2.2 Split Type Unit System

Five units of split type unit system with 2.5 HP in each indoor unit are chosen to be compared with the mini VRF system, where the horse power of this system can be match up with the mini VRF system’s horse power. The configuration of the indoor units of this system is ceiling cassette type. The installation of this system in this building is assumed to be exactly the same as the mini VRF system, where this system operates on the direct expansion principle, maintaining indoor temperature at 24ºC, and the outdoor unit of this system is placed on the outdoor as shown in Figure 4.7. Furthermore, this system is expected to operate at the same time as the mini VRF system, which is from 8 AM until 6 PM on weekdays and assumed to be fully operated at all time of the day throughout the entire year.

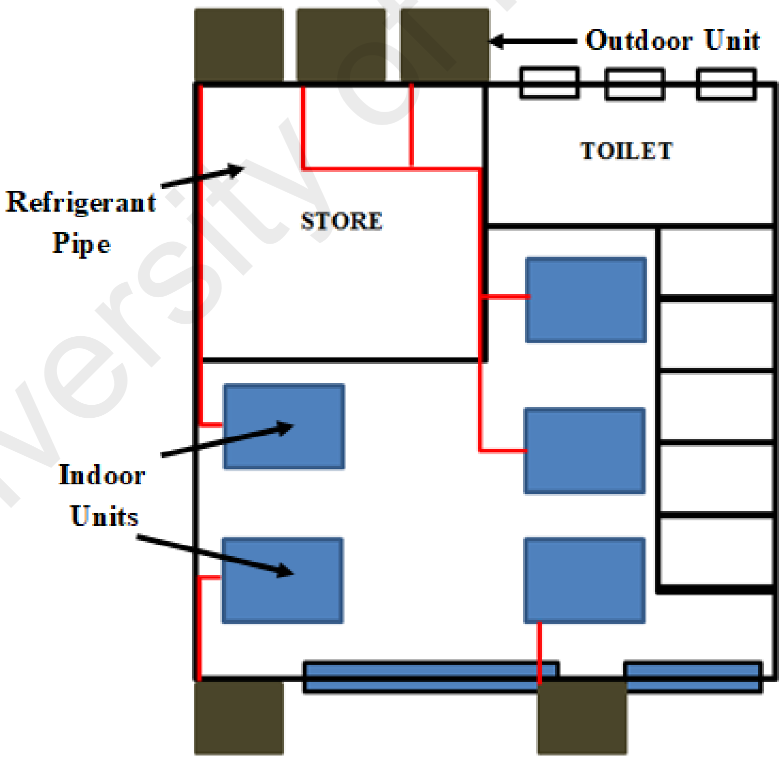


Figure 4.7: Schematic diagram of split - type unit system

The procedure to obtain the linear function of cooling capacity and power input of this system are the same as the mini VRF system. The operating performance detail for this system is shown in Table 4.5. The operating performance curve for this system as shown in Figure 4.9 is based on the operating performance data of this system (obtained from the company which manufactured this system) as shown in Figure 4.8.

Table 4.5: Operating performance details of split type unit system

Operating Performance	Linear Function (kW)
Cooling Capacity	5[9.435 - 0.061(T_{bin})]
Power Input	5[1.21 + 0.038(T_{bin})]

EDB	Outdoor temperature																	
	19°C			25°C			30°C			35°C			40°C			46°C		
	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI
24°C	8.21	4.83	2.00	7.91	4.71	2.16	7.61	4.59	2.34	7.30	4.46	2.54	6.70	4.16	2.75	6.19	3.93	3.03
27°C	8.23	5.59	2.00	7.94	5.46	2.16	7.64	5.33	2.34	7.33	5.20	2.54	6.73	4.86	2.75	6.22	4.60	3.04
30°C	8.30	6.87	2.00	8.01	6.72	2.17	7.72	6.57	2.35	7.42	6.41	2.55	6.84	5.98	2.76	6.35	5.64	3.05
33°C	8.48	8.48	2.01	8.23	8.23	2.18	7.98	7.98	2.36	7.72	7.72	2.57	7.15	7.15	2.79	6.68	6.68	3.08

Remarks:

TC = Total Cooling Capacity

SC = Sensible Cooling Load

PI = Power Input

Figure 4.8: Operating performance of split type unit system (obtained from Wong (10 April 2016))

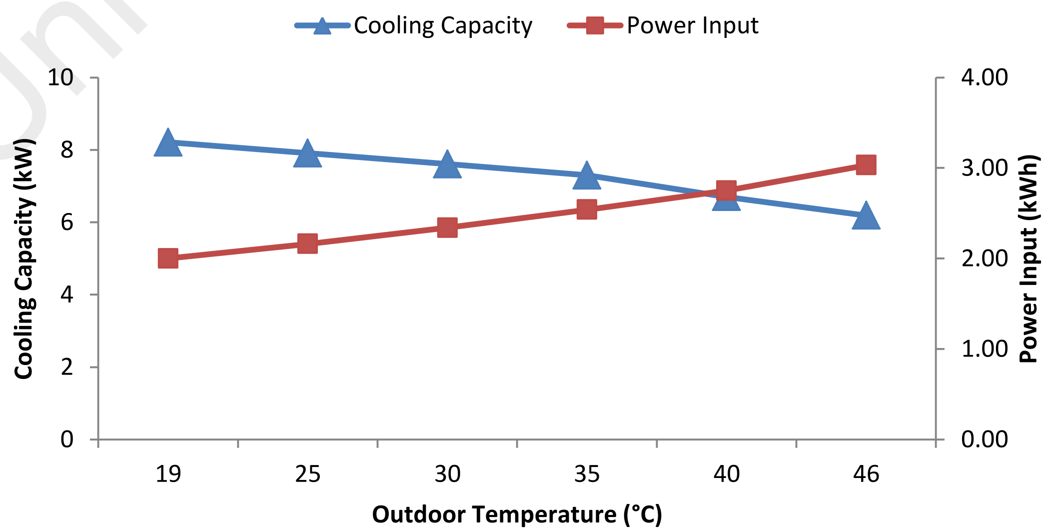


Figure 4.9: Performance curve of split type unit system

Same as Figure 4.6, Figure 4.9 shows that the power input this system increases when the outdoor temperature increases. However, the cooling capacity of this system decreases as the outdoor temperature increases. As mention earlier, this happen because of the decreasing of enthalpy difference of refrigerant at the inlet and outlet of evaporator (indoor heat exchanger) (Zheng & Liang, 2010).

4.3 Results and Discussion

4.3.1 Simulated Result

In this research, the bin method is applied in the simulation of estimating the annual energy consumption for the VRF and ACMV systems, in which the ACMV system for the case study I is split type unit system. The simulated result for both systems is based on the energy consumption at many outdoor temperatures (bin temperatures) distributions for every 4 hours shifts in Petaling Jaya, Malaysia in years 2007 to 2016 (refer Figure 3.2 on section 3.2 in Chapter 3).

As shown in Figures 4.10 and 4.11, the simulation result for both systems is divided into occupied and unoccupied periods, where each period is 4380 hours. It is pertinent to mention that the number of hour for occupied and unoccupied periods in this simulation are selected based on the relation between the bin weather data used in the present research with the target building operation hours. The red line in both figures represents the midpoint of the bin temperature interval based on the bin weather data while the blue line represents the amount of energy consumption of the VRF and ACMV systems.

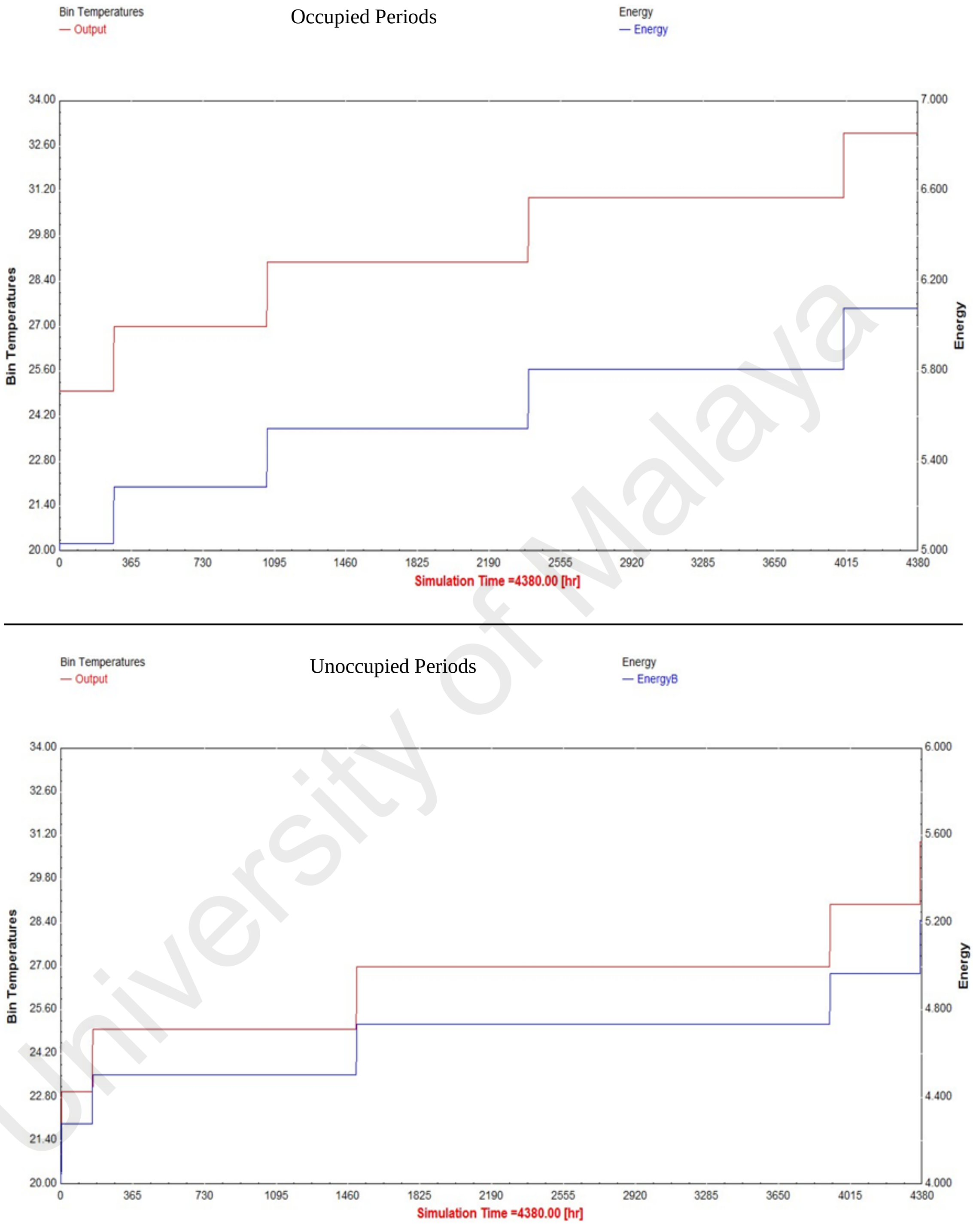


Figure 4.10: Simulated result for mini VRF system

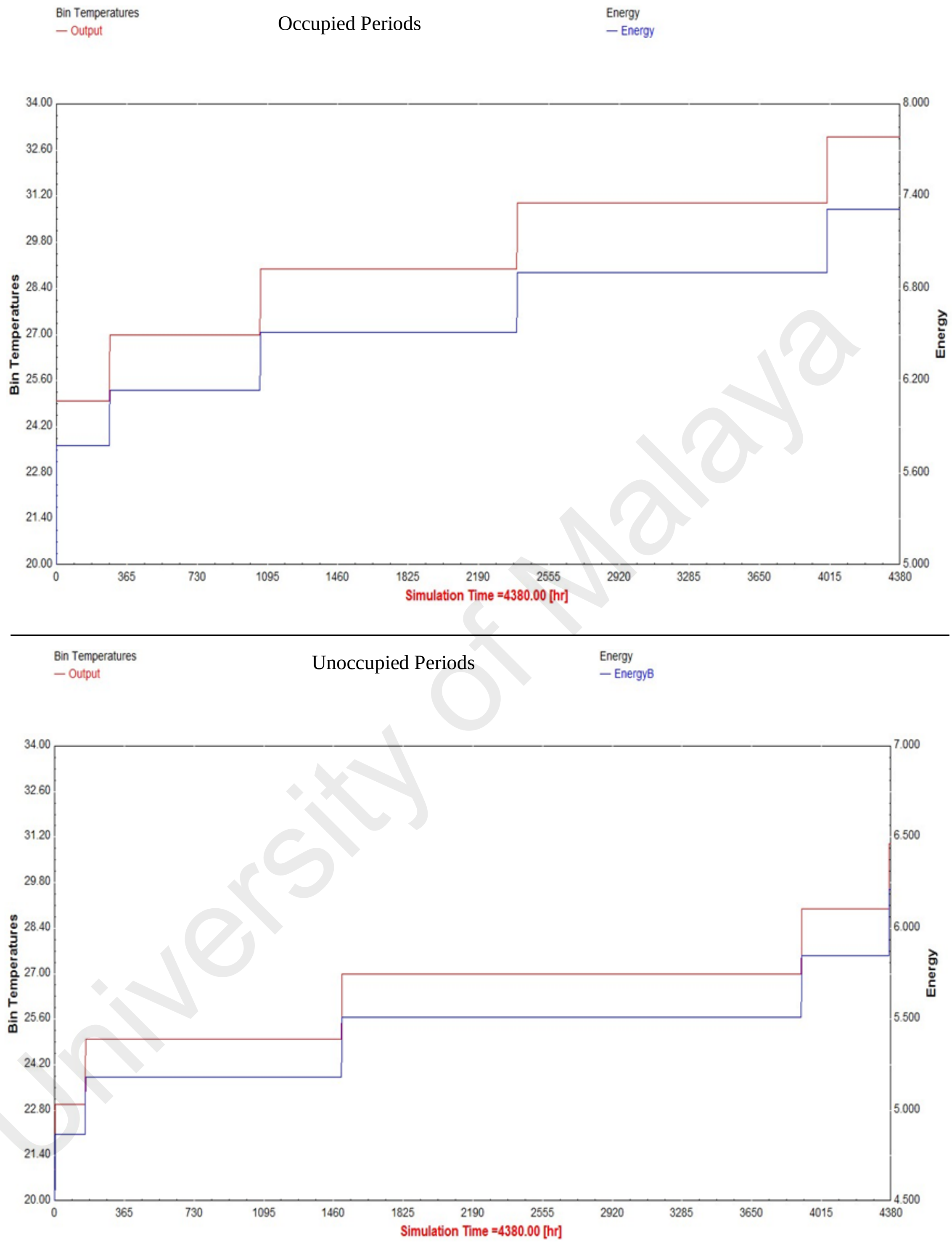


Figure 4.11: Simulated result for split type unit system

4.3.2 Annual Energy Consumption Analysis

The simulated results for both systems are simplified as shown in Figure 4.12 by multiple the frequencies for every midpoint of the bin temperature interval with the energy consumption at each of their own midpoint of the bin temperature interval and then, the result of occupied and unoccupied periods are totalized for each system.

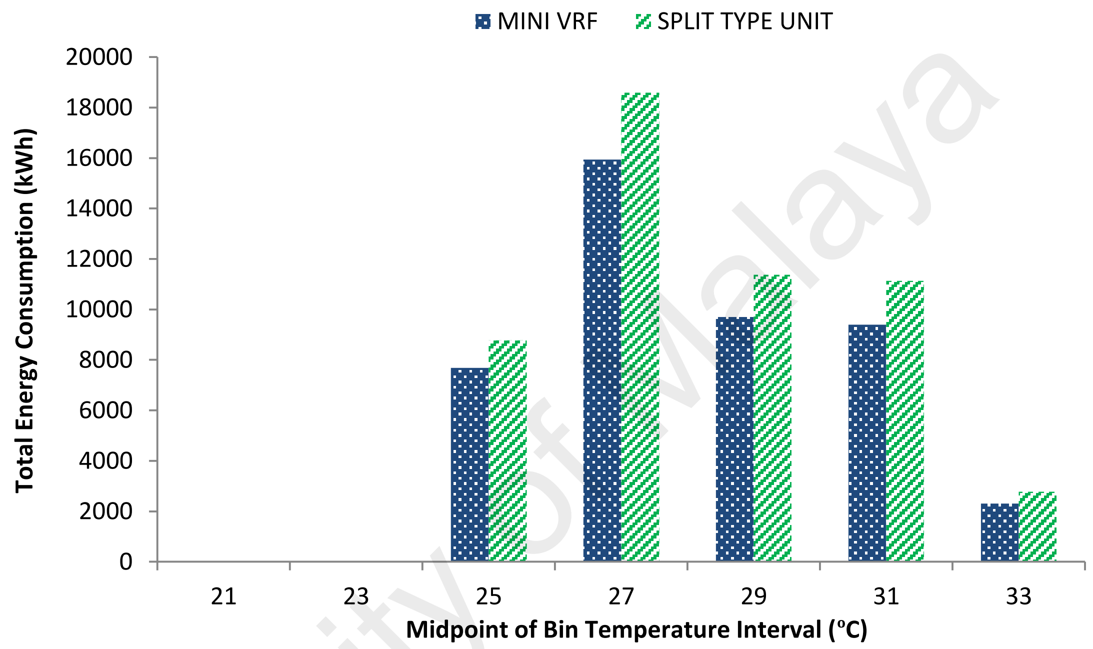


Figure 4.12: Comparison of the total annual energy consumption (kWh) of mini VRF system and split type unit based on the midpoint of bin temperature interval (°C)

From Figure 4.12, it shows that the highest frequency of the midpoint of bin temperature interval is 27°C. This shows that the average temperature in Malaysia is 27°C. Furthermore, the lowest frequency of the midpoint of bin temperature interval is 25°C followed by 33°C. The result suggests that sometime the temperature in Malaysia can reach its highest peak at about 33°C, which is extremely warm, and it rarely goes below 25°C. Thus, it can be concluded that Malaysia has the high daytime temperatures, which is between 25°C to 33°C. It should be noted that the zero values at 21ºC and 23º C in Figure 4.12 is because there is no energy consumption in that temperatures due to the temperature in Malaysia never reach below 23ºC as shown in the bin weather data in Petaling Jaya, Malaysia in years 2007 to 2016 (refer Figure 3.2 on section 3.2 in Chapter 3).

Based on Figure 4.12, it shows that the annual energy consumption of the mini VRF system is lower compared to the split type unit system in almost every midpoint of the bin temperature interval. The overall total of the annual energy consumption for the mini VRF system and split type unit system is 45065.25 kWh and 52675.87 kWh, respectively, where the difference is 7610.62 kWh (i.e. around 15.57%). This clearly shows that the mini VRF system can saves more energy when compared with the split type unit system.

4.3.3 Uncertainty Analysis

The total annual energy consumption for the mini VRF system and split type unit system also have been calculated manually and then, compared it with the simulation results in term of uncertainty in order to examine whether or not the errors present in the simulation results are at a satisfactory level. Table 4.6 shows the uncertainty analysis for both results. As mention earlier, the zero values at 21ºC and 23ºC in Table 4.6 is because the temperature in Malaysia never reach below 23ºC. Thus, there is no energy calculation involved at the midpoint of bin temperature interval at 21ºC and 23ºC.

As shown in Table 4.6, the simulation results for the overall total of annual energy consumption for mini VRF system is 45065.25 kWh while the computed result is 45123.66 kWh, where the percentage uncertainty is 0.07%. As for the split type unit system, the simulation result for the overall total of annual energy consumption is 52675.87 kWh while for computed result is 51590.24 kWh, where the percentage uncertainty is 1.04 %.

Table 4.6: Uncertainty analysis for the simulation and calculation results for mini VRF and split type unit systems

Midpoint of Bin Temperature Interval (°C)	Mini VRF System				
	TRNSYS	Calculation	Average Total Energy (kWh)	Bias Uncertainty	Percentage Uncertainty (%)
	Annual Total Energy (kWh)	Annual Total Energy (kWh)			
21	0	0	0	0	0
23	0	0	0	0	0
25	7690.63	7694.69	7692.66	2.032	0.026
27	15949.38	15994.92	15972.15	22.768	0.143
29	9708.17	9712.87	9710.52	2.350	0.024
31	9405.52	9409.77	9407.65	2.124	0.023
33	2311.54	2311.40	2311.47	0.069	0.003
Total	45065.25	45123.66	45094.46	29.342	0.065
Midpoint of Bin Temperature Interval (°C)	Split Type Unit System				
	TRNSYS	Calculation	Average Total Energy (kWh)	Bias Uncertainty	Percentage Uncertainty (%)
	Annual Total Energy (kWh)	Annual Total Energy (kWh)			
21	0	0	0	0	0
23	0	0	0	0	0
25	8783.15	8664.06	8723.61	59.545	0.683
27	18595.90	18177.16	18386.53	209.370	1.139
29	11376.21	11143.25	11259.73	116.480	1.034
31	11139.77	10901.14	11020.46	119.317	1.083
33	2780.84	2704.63	2742.74	38.105	1.389
Total	52675.87	51590.24	52133.06	542.817	1.041

The uncertainty analysis indicates that the error is not significant and the findings are reliable, where the percentage uncertainty obtained for both systems is quite small, which is approximately to 1%. This clearly shows that the manual calculated results have agreed well with the findings of the simulation model, meaning the bin method is suitable to be applied in the simulation in term of energy analysis.

4.3.4 Economical Analysis

Table 4.7 shows the economical analysis of the mini VRF and split type unit systems. Note that the installation, renewal, and maintenance costs for both systems have not been taken into consideration for this analysis since the original VRF system has been used for a few years in the target building. The initial cost for both systems is based on the current market price in Malaysia. Besides, the annual operating cost for both systems is based on the new current tariff provided by Tenaga Nasional Berhad (TNB) website and the calculation to estimate the annual electricity usage for both systems is acts as a guideline only (Tenaga Nasional Berhad, 2018). It is pertinent to mention that the amount of actual bill might be different as stated in this thesis.

Table 4.7: Economical analysis of mini VRF and split type unit systems

Cooling System	Price per Unit (RM)	Unit	Total Initial Cost (RM)
Mini VRF System	12000	1	12000
Split Type Unit System	3200	5	16000
Cooling System	Total Energy per Year (kWh)	Operating Cost per Year (RM) *(Current Tariff + 6% GST)	
Mini VRF System	45065.250	26420.36	
Split Type Unit System	52675.874	30904.14	
Total Difference		4,483.78	

*Based on Tenaga Nasional Berhad (2018)

From the results obtained, the price for the mini VRF system with 12 HP is cheaper compared to the 5 units of split type unit system with 2.5 HP each. The difference is RM 4000 (USD 1000), namely around 28.57%. However, for a single unit, the split type unit system is much cheaper than the mini VRF system. The difference is RM 8800 (USD 2200), which is around 115.79%. This indicates that a single unit cost for the mini VRF system is quite expensive compared to the split type unit system.

However, the results obtained also show that the operating cost for the mini VRF system is cheaper than the split type unit system. The operating cost for the mini VRF system is RM 26420.36 (USD 6700) while the operating cost for the split type unit is RM 30904.14 (USD 7900), where the difference is RM 4,483.78 (USD 1200), which is about 15.64%. This clearly shows that the mini VRF system can reduces significant electricity consumption in the building when compared with the split type unit system.

4.3.5 Research Discussion

According to the results on the comparison of annual energy consumption and economical analysis between the mini VRF and split type unit system, it shows that the mini VRF system has greater energy savings potential, where this system can saves more energy and reduces significant electricity consumption in the building located in the tropic when comparing with the split type unit system.

This is mainly due to the usage of EEVs in the VRF system, where this EEVs can control the flow of refrigerant to the evaporators by reducing or stopping the refrigerant flow to the individual evaporator unit when this system reaching the required superheat. In addition, when this system obtained the desired indoor temperature, the EEVs can change from fully open to totally closed and close down the system automatically. In other word, the EEVs can make the precise flow adjustment automatically in accordance with the actual load of indoor units, meaning it can control its output to the conditioned space and maintain its space temperature set point.

Generally, same as the VRF system, the split type unit system allows occupants to turn on the air conditioner only when needed and occupied. Thus, even though the VRF system comes with an obvious energy savings potential, this system is not suitable for buildings which require less cooling load and need cooling in one zone because this system comes with more than one indoor unit, where a single unit cost for the VRF system is quite expensive compared to the single unit of split type unit system.

CHAPTER 5: CASE STUDY II - DAIKIN R&D CENTRE BUILDING

This chapter focuses on the estimation of the annual energy consumption of the existing ACMV system used in the large building which needs cooling in several zones. Note that the existing ACMV system is a multi - split type unit system. The annual energy consumption of this system will be compared with the VRF system at the same indoor and outdoor conditions, and followed by the comparison of the economical analysis for both systems. The comparison is made to evaluate whether or not the VRF system can provide better energy consumption compared with the existing ACMV system. In addition, the uncertainty between the simulation results with the manually calculation results for annual energy consumption for both systems also will be discussed and analysed in this chapter.

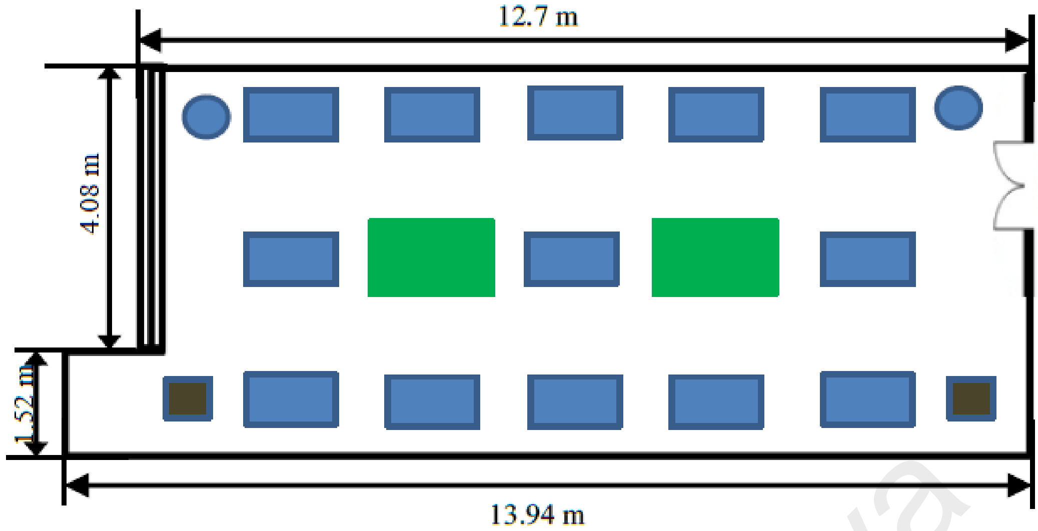
5.1 Target Building Overview

5.1.1 Target Building Description

R&D Centre Daikin building is located at the Sungai Buloh, Malaysia, where the investigation activities on improving the existing ACMV products and procedures as well as the development of new ACMV products and procedures are conducted. This building operates for 9 hours which is from 8.30 AM until 5.30 PM on weekdays. This building has four floors and the assessments are only conducted at the third floor. The third floor of this building consists of large hall, two toilets, air handling unit (AHU) room, and 7 rooms. A graphical description of target building function layout is shown in Figure 5.1

Figure 5.1: Third floor of the target building function layout

The only conditioned area is in the rooms, which each room accommodates 24 occupants which work mostly in front of the computer. A multi - split type unit system serves these rooms with the helps of ventilation system, which is AHU. The net area for each room is 73 m^2 and the height for each room is 3 m. A top plan view of one of the room in the third floor of the target building is shown in Figure 5.2.



Signs	Descriptions
	Circular ceiling diffusers (200 mm diameter)
	4 way diffuser grilles (0.46 m x 0.46m)
	Ceiling light panels (1.22 m x 0.48 m)
	Indoor unit (Ceiling Cassette Type) (0.7 m x 0.7 m)
	Window (3.08 m x 1.75 m)
	Inward double swing door (2.1 m x 1.81 m)

Figure 5.2: Top plan view of one of the room in the third floor of the target building

5.1.2 Target Building Design Loads

All the data/details regarding target building design loads including solar load, conduction load, internal loads, and ventilation or infiltration load will be used in the simulation as mentioned in section 3.3. The only conditioned area is in the rooms, thus the estimation of the energy consumption by using bin method is done only on these rooms by using room - by - room method, assuming the size and the room design detail are the same for all rooms.

5.1.2.1 Solar and Conduction Loads

The solar and conduction loads of the target building can be obtained if the target building envelope characteristics are known. Table 5.1 shows the target building envelope characteristics.

Table 5.1: Target building envelope characteristics

Envelope Element	Description	U – factor ($W/m^2.K$)	SC	Unit	Area (m^2)
Roof	• Outside air film • 9.8'' RC beam • 1" Cement plaster (0.5" on both sides) • 2" Fiber glass • 0.5" Gypsum board • Inside air film	0.38	-	1	73
Floor	• Outside air film • 9.8'' RC beam • Inside air film	2.89	-	1	73
Door	• 35'' Wood solid core flush door	1.48	-	1	3.80
Window	• 1/4'' Double glazing, uncoated clear	0.71	0.61	4	1.786
Exterior Wall a	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	41.82
Exterior Wall b	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	38.10
Exterior Wall c	• Outside air film • 4.5" Brick wall • 1" Cement plaster (0.5" on both sides) • 0.5" Gypsum board • 2" Fiber glass • Inside air film	0.54	-	1	9.66

Table 5.1 continued

Envelope Element	Description	U – factor ($W/m^2.K$)	SC	Unit	Area (m^2)
Exterior Wall d	• Outside air film • 4.5” Brick wall • 1” Cement plaster (0.5” on both sides) • 0.5” Gypsum board • 2” Fiber glass • Inside air film	0.54	-	1	13

5.1.2.2 Ventilation Load

For this target building, only the ventilation load is calculated. This is because the positive pressure produced by the ventilation used in the target building reduces or might even eliminates the infiltration of unconditioned air from the outdoor and will generate an additional load (ventilation load) on the ACMV equipment (SlideShare, 2014).

Ventilation load calculation is depending on the volume flow rate of outdoor air entering the building or room. The *Cp* of air is obtained by the table of ideal - gas specific heats of various common gases (Çengel & Boles, 2014).

Table 5.2: Ventilation load details

ρ_{air} (kg/m^3)	Cp_{air} (KJ/kg.K)	Indoor Air Velocity (m/s)	Indoor V_a (m^3/s)	RH Indoor (%)	RH Outdoor (%)
1.225	1.005	0.01375	1.00375	64.65	80

5.1.2.3 Internal Loads

The sources of internal loads of the building are the occupancy, lights, and equipment used in the building itself. These loads has a constant value due to these loads does not affected by the outdoor temperature.

Table 5.3: Internal loads details

Occupancy Detail						
Quantity	*CLF	Type of Activity	*Energy / Person (kW)		Total Occupancy Load (kW)	
			Sensible Load (kW)	Latent Load (kW)	Sensible Load (kW)	Latent Load (kW)
24	1	Light work and sit	0.070	0.045	1.680	1.08

Lighting Detail					
Quantity	Type	*CLF	*Ballast Factor	Power / Unit (kW)	Total Lighting Load (kW)
24	Fluorescent light	1	1.2	0.058	2.714

Equipment Detail			
Equipment	Quantity	Average Usage	Heat Gain (kW)
Desktop	25	0.75	1.219
Screen Monitor	25	0.75	1.313
Laptop	3	0.75	0.124
Water Cooler	1	-	0.350

*Data obtained from ASHRAE Table (ASHRAE, 2014)

5.2 Cooling System Overview

5.2.1 Existing ACMV System (Multi - Split Type Unit System)

A multi - split type unit system serves the entire rooms in the third floor of the target building with the helps of ventilation system, which is AHU. This system is used to maintain the indoor temperature at 24 °C in each room. The outdoor unit of this system is placed on the rooftop of this building and connected with several indoor units with 2.5 HP for each unit. Each room in the third floor of this building consists of two indoor units, where the configuration of these indoor units is ceiling cassette type. The outdoor air is introduced at this floor through the AHU. Figure 5.3 shows the schematic diagram of this system in each room. Furthermore, this system is operated from 8.30 AM until

5.30 PM on weekdays and assumed to be fully operated at all time of the day throughout the entire year.

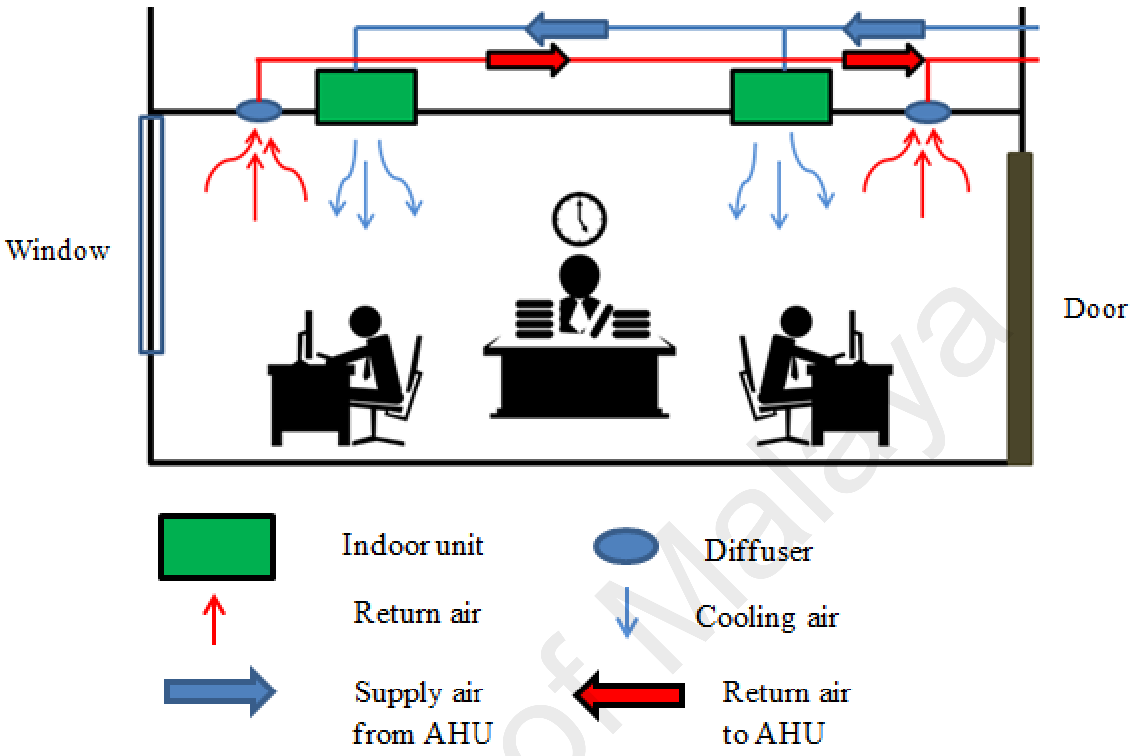


Figure 5.3: Schematic diagram of multi - split type unit system in one of the room in the third floor of the target building

The COP of the multi - split unit system can be obtained if the power input and cooling capacity at different outdoor temperature of this system are known. In this research, the bin temperature which also known as outdoor temperature is acts as the function in a linear function of the cooling capacity and power input for this system as shown in Table 5.4.

Table 5.4: Operating performance details of multi - split type unit system

Operating Performance	Linear Function (kW)
Cooling Capacity	14[9.435 - 0.061(T_{bin})]
Power Input	14[1.21 + 0.038(T_{bin})]

The linear function for cooling capacity and power input for this system can be obtained based on the operating performance curve of this system as shown in Figure 5.5. It should be noted that the operating performance curve for this system is based on the operating performance data of this system (obtained from the company which manufactured this system) as shown in Figure 5.4.

EDB	Outdoor temperature																	
	19°C			25°C			30°C			35°C			40°C			46°C		
	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI	TC	SC	PI
24°C	8.21	4.83	2.00	7.91	4.71	2.16	7.61	4.59	2.34	7.30	4.46	2.54	6.70	4.16	2.75	6.19	3.93	3.03
27°C	8.23	5.59	2.00	7.94	5.46	2.16	7.64	5.33	2.34	7.33	5.20	2.54	6.73	4.86	2.75	6.22	4.60	3.04
30°C	8.30	6.87	2.00	8.01	6.72	2.17	7.72	6.57	2.35	7.42	6.41	2.55	6.84	5.98	2.76	6.35	5.64	3.05
33°C	8.48	8.48	2.01	8.23	8.23	2.18	7.98	7.98	2.36	7.72	7.72	2.57	7.15	7.15	2.79	6.68	6.68	3.08

Remarks:
TC = Total Cooling Capacity
SC = Sensible Cooling Load
PI = Power Input

Figure 5.4: Operating performance of multi - split type unit system (obtained from Wong (10 April 2016))

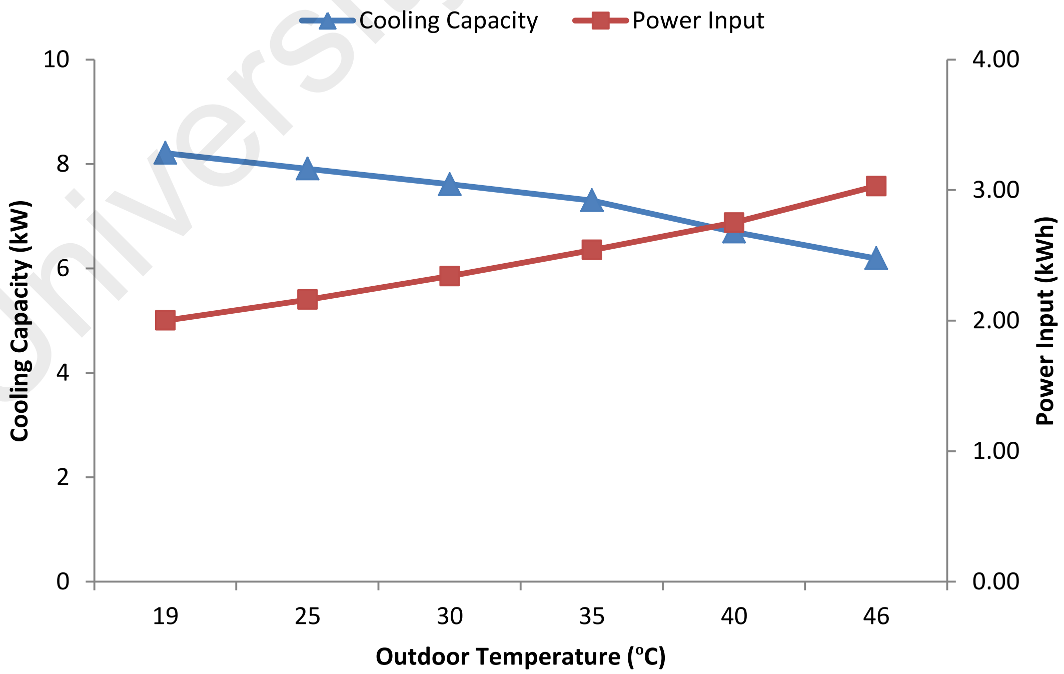


Figure 5.5: Performance curve of multi - split type unit system

As shown in Figure 5.5, the power input of this system increases when the outdoor temperature increases. However, the cooling capacity of this system decreases as the outdoor temperature increases. This happen because the temperature of the refrigerant at the inlet of evaporator in the indoor unit increasing with the increasing of the outdoor temperature while the refrigerant at the outlet of the evaporator in the indoor unit stays constant at any outdoor temperature, causing the decreasing of enthalpy difference of refrigerant at the inlet and outlet of evaporator in the indoor unit (Zheng & Liang, 2010).

5.2.2 VRF System

VRF system is used to compare with the existing ACMV system, which is multi - split unit system. A VRF system with 36 HP is chosen to match to match up with the existing ACMV system's horse power. Furthermore, the VRF system is expected to operate at the same time as existing ACMV system which is from 8.30 AM until 5.30 PM on weekdays and assumed to be fully operated at all time of the day throughout the entire year. The installation of the VRF system is assumed to be exactly the same as the existing ACMV system which is the outdoor unit of this system is placed on the rooftop and outdoor air is introduced at this floor through the AHU.

The procedure to obtain the linear function of cooling capacity and power input of this system are the same as the existing ACMV system. The operating performance detail for this system is shown in Table 5.5. The operating performance curve for this system as shown in Figure 5.7 is based on the operating performance data of this system (obtained from the company which manufactured this system) as shown in Figure 5.6.

Table 5.5: Operating performance details of VRF system

Operating Performance	Linear Function (kW)
Cooling Capacity	$127.7 - 0.54\ (T_{bin})$
Power Input	$16.43 + 0.45(T_{bin})$

Model: AVWT-343TESZX Remarks: Q = Cooling Capacity; P = Power Input

Combination (%)	To \ Ti	14		16		18		19		20		22		23		24	
		Q	P	Q	P	Q	P	Q	P	Q	P	Q	P	Q	P	Q	P
100%	10	65.9	10.20	71.1	10.64	95.4	15.26	100.4	25.05	104.5	25.42	109.8	25.87	112.3	25.97	115.2	26.17
	12	66.3	10.20	70.9	10.64	95.3	15.26	100.3	25.11	104.4	25.49	109.7	25.87	112.1	25.97	115.0	26.17
	14	65.8	10.33	71.2	10.70	95.6	15.33	100.7	25.18	104.3	25.55	109.4	25.93	111.8	26.04	114.7	26.24
	16	65.9	10.45	71.3	10.83	95.4	15.51	100.5	25.30	103.8	25.68	109.6	26.07	111.9	26.18	114.7	26.37
	18	66.0	10.58	71.5	11.01	95.8	15.64	100.3	25.49	104.2	25.87	109.6	26.34	111.9	26.41	114.6	26.58
	20	66.0	10.83	71.6	11.26	95.3	15.95	100.5	25.74	104.2	26.13	109.5	26.54	111.7	26.62	114.3	26.79
	22	66.1	11.14	71.7	11.58	95.4	16.20	100.7	26.00	104.3	26.39	109.8	26.87	111.9	26.96	113.9	27.14
	24	65.9	11.45	71.7	11.89	95.5	16.58	100.7	26.37	104.2	26.78	109.4	27.21	111.9	27.30	114.3	27.48
	25	65.7	11.64	71.4	12.11	95.3	16.77	100.5	26.60	103.8	26.97	109.4	27.44	111.9	27.54	114.2	27.73
	26	65.7	11.91	71.4	12.41	95.3	17.15	100.5	26.97	103.8	27.35	109.4	27.82	111.9	27.92	114.2	28.10
	28	65.6	12.57	71.4	13.01	95.3	17.90	100.5	27.73	103.8	28.10	109.4	28.58	111.9	28.67	114.2	28.86
	30	65.6	13.17	71.4	13.62	95.3	18.65	100.5	28.48	103.8	28.86	109.4	29.33	111.9	29.43	114.2	29.61
	32	65.5	14.08	71.4	14.60	95.3	19.90	100.5	29.69	103.4	30.03	108.3	30.50	110.2	30.56	112.0	30.67
	34	65.5	15.06	71.4	15.58	95.3	21.14	100.5	30.90	103.0	31.20	107.2	31.67	108.6	31.69	109.9	31.73
	35	65.6	15.55	71.4	16.07	95.3	21.77	100.5	31.50	102.8	31.78	106.6	32.25	107.8	32.25	108.8	32.25

Figure 5.6: Operating performance of VRF system (obtained from Poh (5 April 2016))

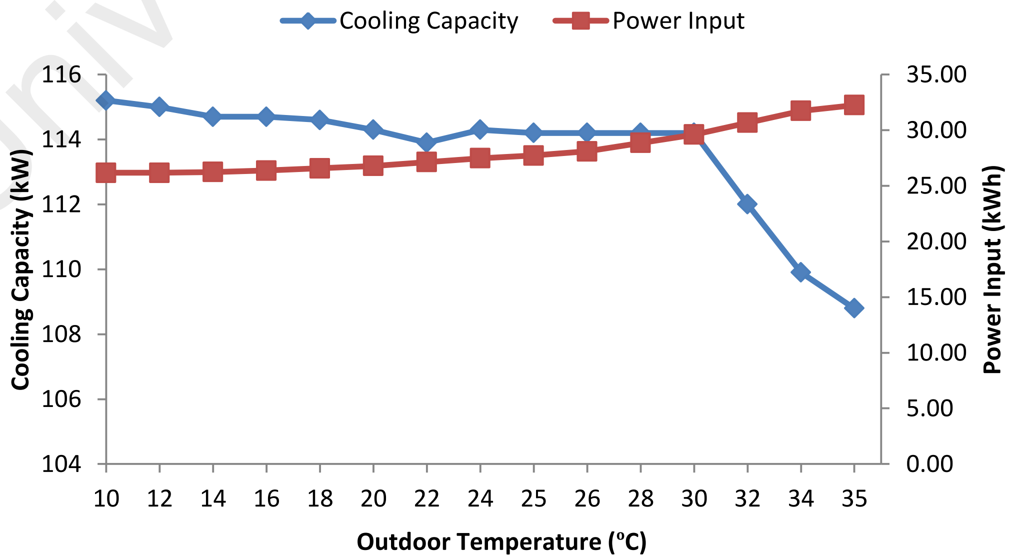


Figure 5.7: Performance curve of VRF system

Same as Figure 5.5, Figure 5.7 shows that the power input of this system increases when the outdoor temperature increases. However, the cooling capacity of this system is remains constant as the outdoor temperature increases due the ability of the multi - compressors and EEVs in this system to control the flow rate of refrigerant in each indoor unit by reducing or stopping the flow of refrigerant to each indoor unit when superheat occur in the VRF system (Bhatia, 2012; Li et al., 2016).

The cooling capacity of this system started to decrease at certain point of outdoor temperature due to the decreasing of enthalpy difference of refrigerant at the inlet and outlet of evaporator in the indoor unit. As mention earlier, this process occurred because the temperature of the refrigerant at the inlet of evaporator in the indoor unit is increasing with the increasing of the outdoor temperature while the refrigerant at the outlet of the evaporator in the indoor unit remains constant at any outdoor temperature (Zheng & Liang, 2010).

The extreme drop of the cooling capacity of this system is due to the large size/horse power of this system used for conditioning the large building which needs cooling in several zones. At a certain outdoor temperature, the large size/horse power of this system can make the temperature of the refrigerant at the inlet of indoor heat exchanger increasing too high while the refrigerant at the outlet of the evaporator remains constant at any outdoor temperature, causing the enthalpy difference of refrigerant at the inlet and outlet of indoor heat exchanger reduced in extremely manner.

5.3 Results and Discussion

5.3.1 Simulated Result

As mention earlier, the assessments are only conducted at the third floor of the target building, where the only conditioned area is in the rooms. Thus, the estimation of the annual energy consumption is done only for these rooms by using room - by - room method, assuming the size, room design detail, and building design loads are the same for all rooms.

In this research, the bin method is applied in the simulation of estimating the annual energy consumption for the existing ACMV system and VRF systems, in which the existing ACMV system for the case study II is multi - split type unit system. The simulated result for both systems is based on the energy consumption at many outdoor temperatures (bin temperatures) distributions for every 4 hours shifts in Petaling Jaya, Malaysia in years 2007 to 2016 (refer Figure 3.2 on section 3.2 in Chapter 3).

As shown in Figure 5.8 and Figure 5.9, the simulation result for both systems is divided into occupied and unoccupied periods, where each period is 4380 hours. It is pertinent to mention that the number of hour for occupied and unoccupied periods in this simulation are selected based on the relation between the bin weather data used in the present research with the target building operation hours. The red line in both figures represents the midpoint of the bin temperature interval based on the bin weather data while the blue line represents the amount of energy consumption of the VRF and ACMV systems.

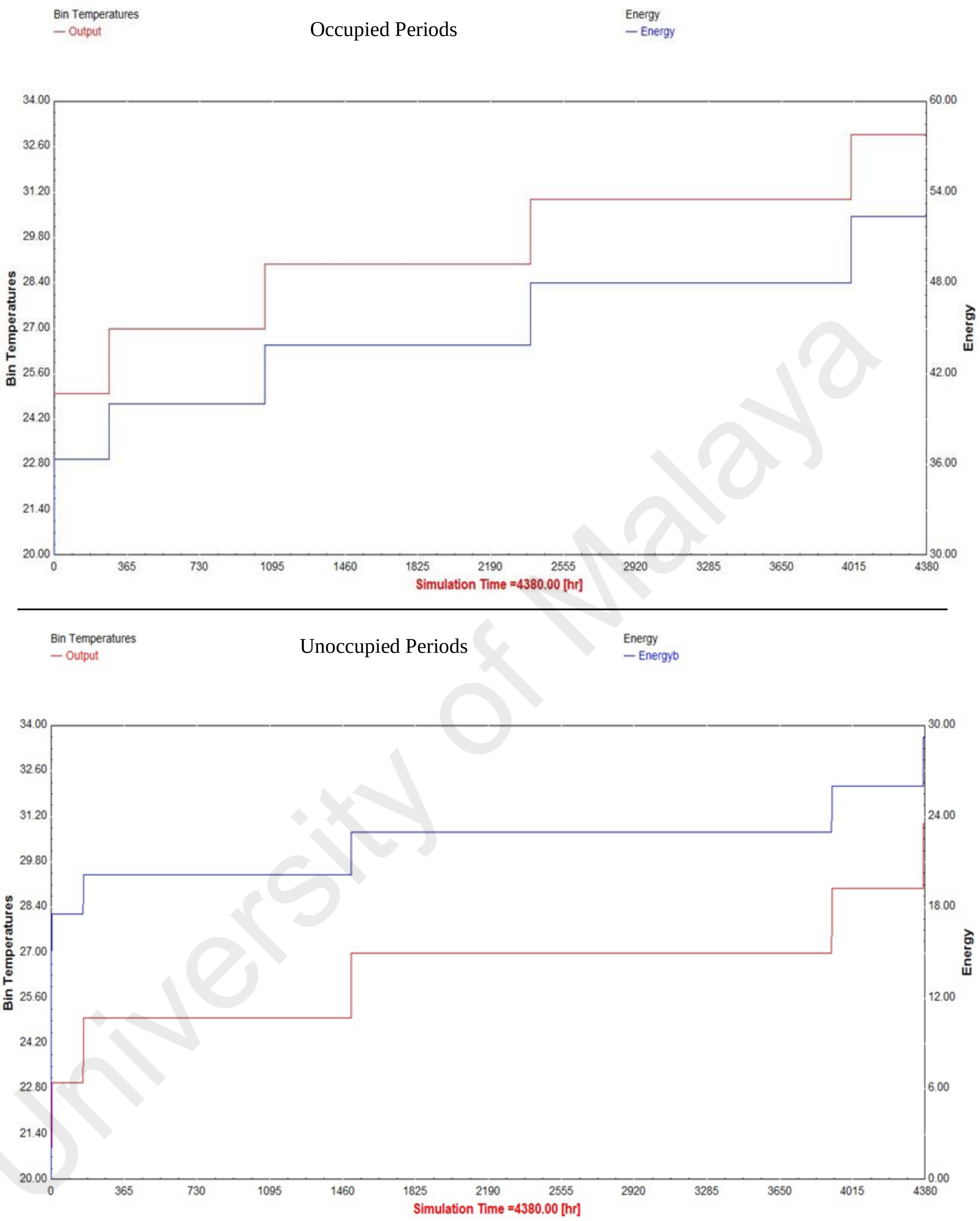


Figure 5.8: Simulated result for existing ACMV system (multi - split type unit system)

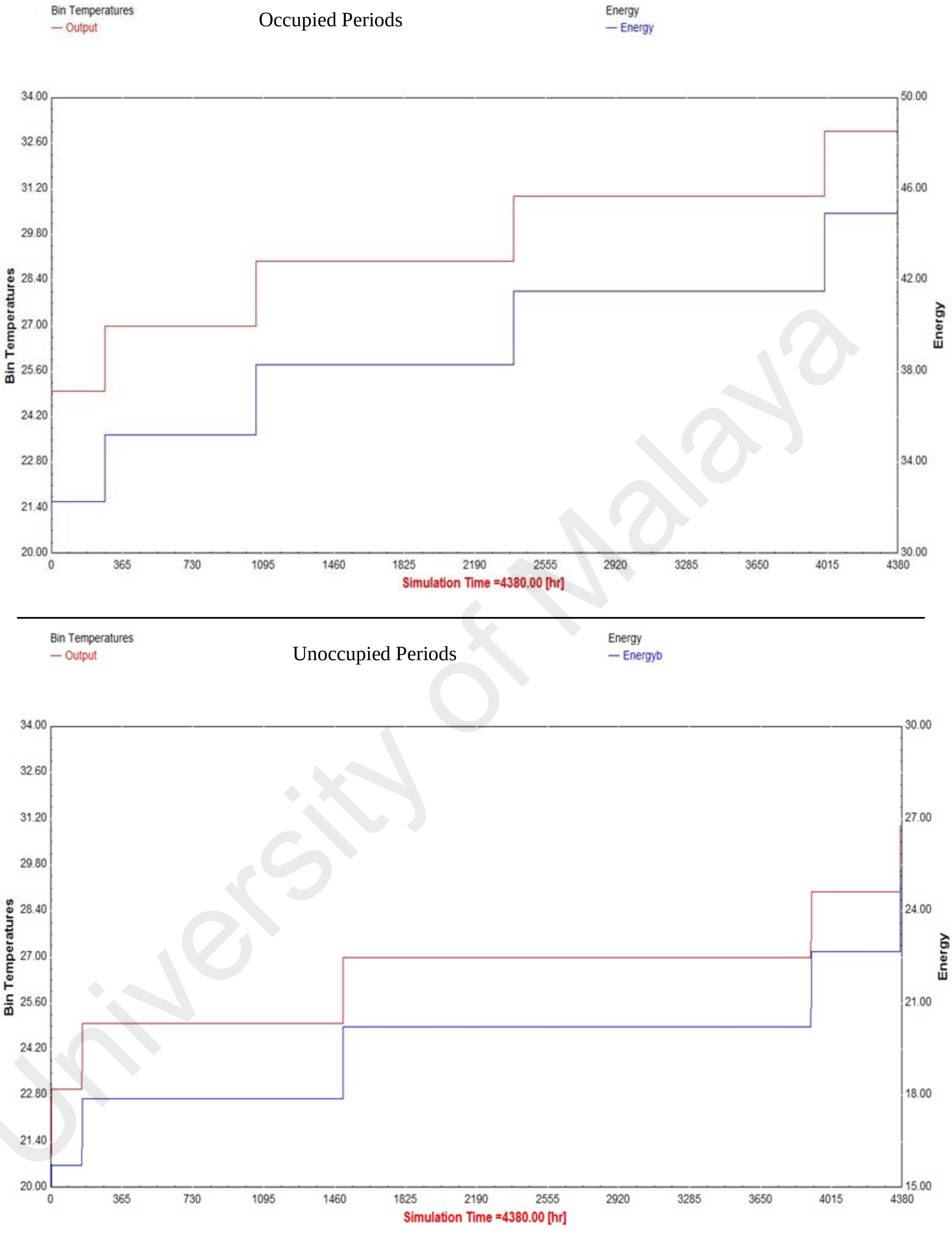


Figure 5.9: Simulated result for VRF system

5.3.2 Annual Energy Consumption Analysis

The simulated results for both systems are simplified as shown in Figure 5.10 by multiple the frequencies for every midpoint of the bin temperature interval with the energy consumption at each of their own midpoint of the bin temperature interval and then, the result of occupied and unoccupied periods are totalized for each system.

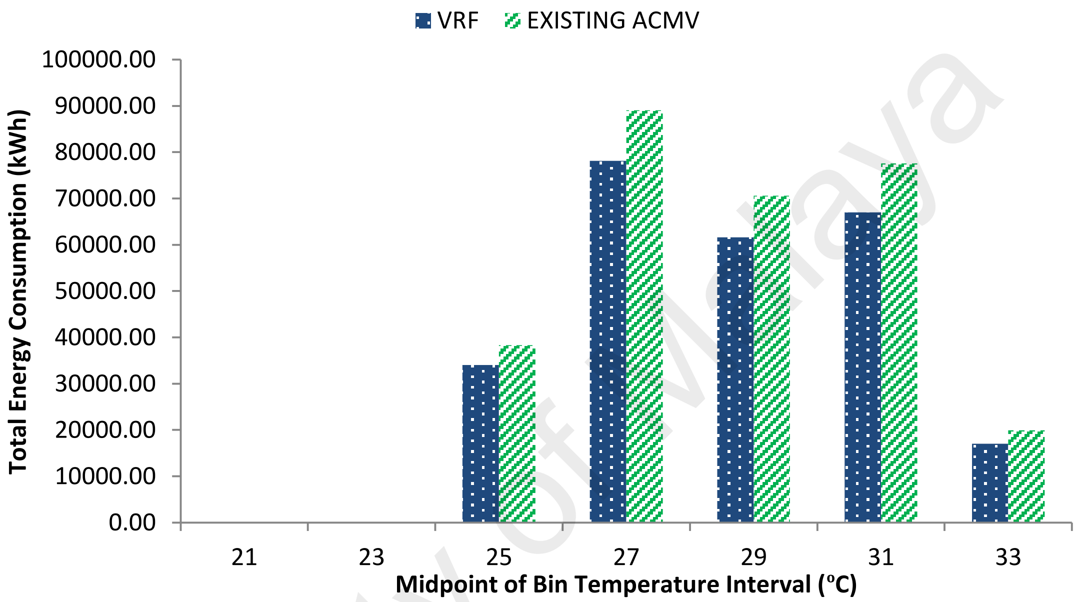


Figure 5.10: Comparison of the total annual energy consumption (kWh) of VRF system and existing ACMV system based on the midpoint of bin temperature interval (°C)

From Figure 5.10, it shows that the highest frequency of the midpoint of bin temperature interval is 27°C. This shows that the average temperature in Malaysia is 27°C. Furthermore, the lowest frequency of the midpoint of bin temperature interval is 25°C followed by 33°C. The result suggests that sometime the temperature in Malaysia can reach its highest peak at about 33°C, which is extremely warm, and it rarely goes below 25°C. Thus, it can be concluded that Malaysia has the high daytime temperatures, which is between 25°C to 33°C. It should be noted that the zero values at 21ºC and 23º C in Figure 4.12 is because there is no energy consumption in that temperatures due to the temperature in Malaysia never reach below 23ºC as shown in the bin weather data in

Petaling Jaya, Malaysia in years 2007 to 2016 (refer Figure 3.2 on section 3.2 in Chapter 3).

Based on Figure 5.10, it shows that the annual energy consumption of the VRF system is lower compared to the existing ACMV system, which is multi - split type unit system in almost every midpoint of the bin temperature interval. The overall total of the annual energy consumption for the VRF system and multi - split type unit system is 257916.22 kWh and 295615.21 kWh, respectively, where the difference is 37698.982 kWh (i.e. around 13.62%). This clearly shows that the VRF system can saves more energy when compared with the multi - split type unit system.

5.3.3 Uncertainty Analysis

The total annual energy consumption for the multi - split type unit and VRF system also have been calculated manually and then, compared it with the simulation results in term of uncertainty in order to examine whether or not the errors present in the simulation results are at a satisfactory level. Table 5.6 shows the uncertainty analysis for both results. As mention earlier, the zero values at 21ºC and 23ºC in Table 5.6 is because the temperature in Malaysia never reach below 23ºC. Thus, there is no energy calculation involved at the midpoint of bin temperature interval at 21ºC and 23ºC.

As shown in Table 5.6, for simulation result, the annual energy consumption for VRF system is 257916.22 kWh while for computed result is 257977.68 kWh, where the percentage uncertainty is 0.03%. As for the existing ACMV system (multi - split unit), the simulation result for the annual energy consumption is 295615.21 kWh while for computed result is 295269.96 kWh, where the percentage uncertainty is 0.06 %.

Table 5.6: Uncertainty analysis for the simulation and calculation results for existing ACMV and VRF systems

Midpoint of Bin Temperature Interval (°C)	Existing ACMV System				
	TRNSYS	Calculation	Average Total Energy (kWh)	Bias Uncertainty	Percentage Uncertainty (%)
	Annual Total Energy (kWh)	Annual Total Energy (kWh)			
21	0	0	0	0	0
23	0	0	0	0	0
25	38322.40	38297.01	38309.70	12.697	0.033
27	89104.78	88839.53	88972.15	132.621	0.149
29	70642.72	70632.82	70637.77	4.950	0.007
31	77618.11	77587.081	77602.60	15.514	0.020
33	19927.20	19913.52	19920.36	6.842	0.034
Total	295615.21	295269.96	295442.58	172.625	0.058
Midpoint of Bin Temperature Interval (°C)	VRF System				
	TRNSYS	Calculation	Average Total Energy (kWh)	Bias Uncertainty	Percentage Uncertainty (%)
21	0	0	0	0	0
23	0	0	0	0	0
25	34025.82	33992.85	34009.34	16.487	0.048
27	78132.28	78177.76	78155.02	22.741	0.029
29	61627.77	61622.87	61625.32	2.449	0.004
31	67051.64	67108.59	67080.11	28.476	0.042
33	17078.72	17075.61	17077.17	1.554	0.009
Total	257916.22	257977.68	257946.95	71.707	0.028

The uncertainty analysis indicates that the error is not significant and the findings are reliable, where the percentage uncertainty obtained for both systems is quite small which is below 1%. This clearly shows that the manual calculated results have agreed well with the findings of the simulation model, meaning the bin method is suitable to be applied in the simulation in term of energy analysis.

5.3.4 Economical Analysis

Table 5.7 shows the annual economic analysis of the VRF and the existing ACMV systems. The total cost of material and labor associated with the existing ACMV system installed in the target building such as ductwork, AHU, and mechanical piping are neglected since this information is unknown. Thus, the comparison for equipment cost for existing ACMV system and VRF system cannot be made.

Besides, the annual operating cost for both systems is based on the new current tariff provided by Tenaga Nasional Berhad (TNB) website and the calculation to estimate the electricity usage for a year of both systems is acts as a guideline only (Tenaga Nasional Berhad, 2018). It is pertinent to mention that the amount of actual bill might be different as stated in this thesis.

Table 5.7: Economical analysis of existing ACMV and VRF systems

ACMV System	Total Energy per Year (kWh)	Operating Cost per Year (RM) *(Current Tariff + 6% GST)
Existing ACMV	295615.21	174031.36
VRF	257916.22	151821.08
Total Difference		22210.28

*Based on Tenaga Nasional Berhad (2018)

From Table 5.7, it shows that the operating cost for VRF system is cheaper than the existing ACMV system installed in the target building. The operating cost for VRF system is RM 151821.08 (USD 38000) while the operating cost for existing ACMV system is RM 174031.36 (USD 44000). The difference of the overall operating cost for both systems is RM 22210.28 (USD 5600) which is about 13.63%. This indicates that a single unit cost for the mini VRF system is quite expensive compared to the split type unit system.

5.3.5 Research Discussion

According to the results on the comparison of annual energy consumption and economy analysis between the VRF system and multi - split type unit system, the VRF system has greater energy savings potential, where this system can saves more energy and reduces significant electricity consumption in the building located in the tropic when comparing with the multi - split type unit system.

This is mainly due to the usage of EEVs in the VRF system, where this EEVs can control the flow of refrigerant to the evaporators by reducing or stopping the refrigerant flow to the individual evaporator unit when this system reaching the required superheat. In addition, when this system obtained the desired indoor temperature, the EEVs can change from fully open to totally closed and close down the system automatically. In other word, the EEVs can make the precise flow adjustment automatically in accordance with the actual load of indoor units, meaning it can control its output to the conditioned space and maintain its space temperature set point.

In addition, the VRF system can provides different indoor temperatures at different zones and also can be shut down if one or more zones is unoccupied or the desired indoor temperature is obtained while the multi - split type unit operated with a fixed schedule, where it do not allow occupants to shut down the system if one or more of the zones is unoccupied or the desired indoor temperature is obtained, resulting in much longer operation time and thus consuming much more energy.

The usage of ventilation system is important in the large building in order to system to provide better indoor air quality. However, the VRF system usually recirculate indoor and needs a separate ventilation to provide better indoor air quality and “free cooling” because this system is ductless. Thus, by adding the additional ventilation to this system, the installation cost for this system can be increased. Different from the VRF system, the multi - split type unit system usually provided with the ventilation for “free cooling” when the outdoor temperature is lower than the recirculation air temperature.

CHAPTER 6: CONCLUSION AND RECOMMENDATIONS

6.1 Conclusion

Both case studies show that the VRF system has greater energy savings potential, where this system can saves more energy and reduces significant electricity consumption in the building located in the tropic when compared with the split type unit and multi - split type unit systems. In case study I, the results indicate that for conditioning the same building with same outdoor and indoor condition in one year, mini VRF system save about 15.57% energy and 15.64% on operating cost when compared with the split type unit system. For case study II, the results indicate that for conditioning the same building with same outdoor and indoor condition in one year, VRF system save about 13.62% energy and about 13.63% on operating cost when compared with the multi - split type unit system.

VRF system has great energy savings potential due to the ability of the multi - compressors and EEVs in this system to control the flow rate of refrigerant in each indoor unit by reducing or stopping the flow of refrigerant to each indoor unit when this system obtained the desired indoor temperature. Beside, the capability of simultaneous cooling and heating in different zones, monitoring, and controlling the entire system from a single location or by using the internet, and independent zoning controls show that this system can accommodates better indoor thermal comfort to the user. These advantages clearly shows that the VRF system can be considered as the priority selection for a building in accordance with the compendious analyses of energy saving, economic benefits, and thermal comfort capabilities.

Generally, same as the VRF system, the split type unit system allows occupants to turn on the air conditioner only when needed and occupied. Therefore, even though the VRF system comes with an obvious energy savings potential, this system is not suitable for buildings which require less cooling load and need cooling in one zone because this system comes with more than one indoor unit, where a single - unit cost for the VRF system is quite expensive compared to the single unit of split type unit system.

Typically, the VRF system can provides different indoor temperatures at different zones and also one or more indoor units of this system can be shut down if one or more zones is unoccupied or the desired indoor temperature is exceeded (occupants feel too cool) while the other continuous to operate. However, as for multi - split type system, it operates with a fixed schedule, where all the indoor units of this system needs to be operated through the occupied period (working hour) even though some of the zones are unoccupied or the desired indoor temperature is exceeded (occupants feel too cool) since this system used the centralized conditioning concept. This clearly shows that the VRF system is appropriate to use in the building which require large cooling load and need cooling in several zones simultaneously with different temperature at different zones.

Energy analysis is important in designing the cooling system in the building in order to achieve energy savings goal. In this research, bin method is used to estimate the total of annual energy consumption of the VRF, split type unit, and multi - split type unit systems, where the simulation baseline created in this research is applied with bin method concept. The bin method can be done either by calculating it manually or used it in the simulation programs. Thus, the comparison of the simulation and manually calculation results is compared in term of percentage uncertainty. For both case studies, the percentage uncertainties are too small, which is approximately to 1%. It can be concluded that the manual calculation results have agreed well with the findings of the simulation model.

The bin method can helps in making full prediction for sizing the proper ACMV or VRF equipment in the desired building in any weather conditions because by using this method, the results regarding the energy consumption is obtained based on the many outdoor temperatures (bin temperatures) distributions. Meaning, the energy consumption of the VRF and ACMV system can be known at any different outdoor temperatures. Note that the outdoor temperature might give impact either directly or indirectly to the efficiency, performance, and cooling or heating capacity for equipment that operates on the refrigeration cycle.

Besides, this method can be done separately for different time periods either for occupied or unoccupied building hours for the consideration of the different building loads and occupancy patterns with time as well as operating hours of cooling system in the building since the bin temperatures are usually collected in six daily 4 hours shifts.

6.2 Recommendation

The conclusions of this research are made based on the two types of ACMV system, which are split type unit and multi - split type unit systems. Besides, the number of target building is limited to two buildings, in which this research is not universal. Thus, more samples (other types of ACMV system and different types of target building) are required in order to obtain better data for the detailed comparisons for the future studies in term of energy performance of the VRF and ACMV systems.

Since all two target buildings are located in Malaysia, so the findings are more applicable to buildings in Malaysia. In addition, the methodology and simulation in this research are generic and suitable to use in other countries as a new way of research in term of energy performance of the cooling system. This is because the estimation of the energy usage for cooling system by using bin method are depending on the bin weather data on particular location/country which can be obtained by ASHRAE handbooks or weather forecasting organizations on its own country.

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LIST OF PUBLICATIONS AND PAPERS PRESENTED

Submitted paper (accepted)

1. Yau, Y. H. and 'Amir, M. (2019). Energy Use Analysis of the Variable Refrigerant Flow (VRF) System versus the Multi - Split Unit Using TRNSYS. *Heat and Mass Transfer*. doi: 10.1007/s00231-019-02726-7 (Impact Factor: Q3-1.551).