CHAPTER 2

LITERATURE REVIEW

This chapter describes HVAC system in general, space cooling load and the purpose of HVAC system for indoor thermal comfort. The Malaysian standard for indoor thermal comfort is also considered in this chapter. Space cooling load is described to show the effect on indoor thermal conditions and the energy consumed. Previous research on developing HVAC system strategies for lower energy consumption and recent study on adaptive comfort are addressed to know current position of HVAC system technology. Previous studies that are discussed in this chapter focus on cooling and dehumidification process since the weather in Malaysia is hot and humid.

2.1 Centralized HVAC System

Centralized HVAC system is a central hydronic air conditioning system used to provide indoor thermal comfort in multi-zone buildings. The system can be divided into two loops: primary and secondary loop. Primary loop is a water system which produces cooling/heating effect through chilled/hot water production and distributes it to secondary loop. Secondary loop is an air system by which cooled/hot air is produced and transferred to the conditioned spaces to maintain indoor thermal set point temperature and humidity [31]. An example of centralized HVAC system schematic diagram is shown in Figure 2.1.

Vapor compression and vapor absorption are two thermodynamic cycles used to produce cooling or heating effects. In a vapor-compression refrigeration cycle, four processes are occurred: isentropic compression in a compressor, constant-pressure heat rejection in a condenser, throttling in throttle device, and constant-pressure heat absorption in an evaporator [32]. Schematic of the vapor-compression refrigeration cycle is presented in Figure 2.2.

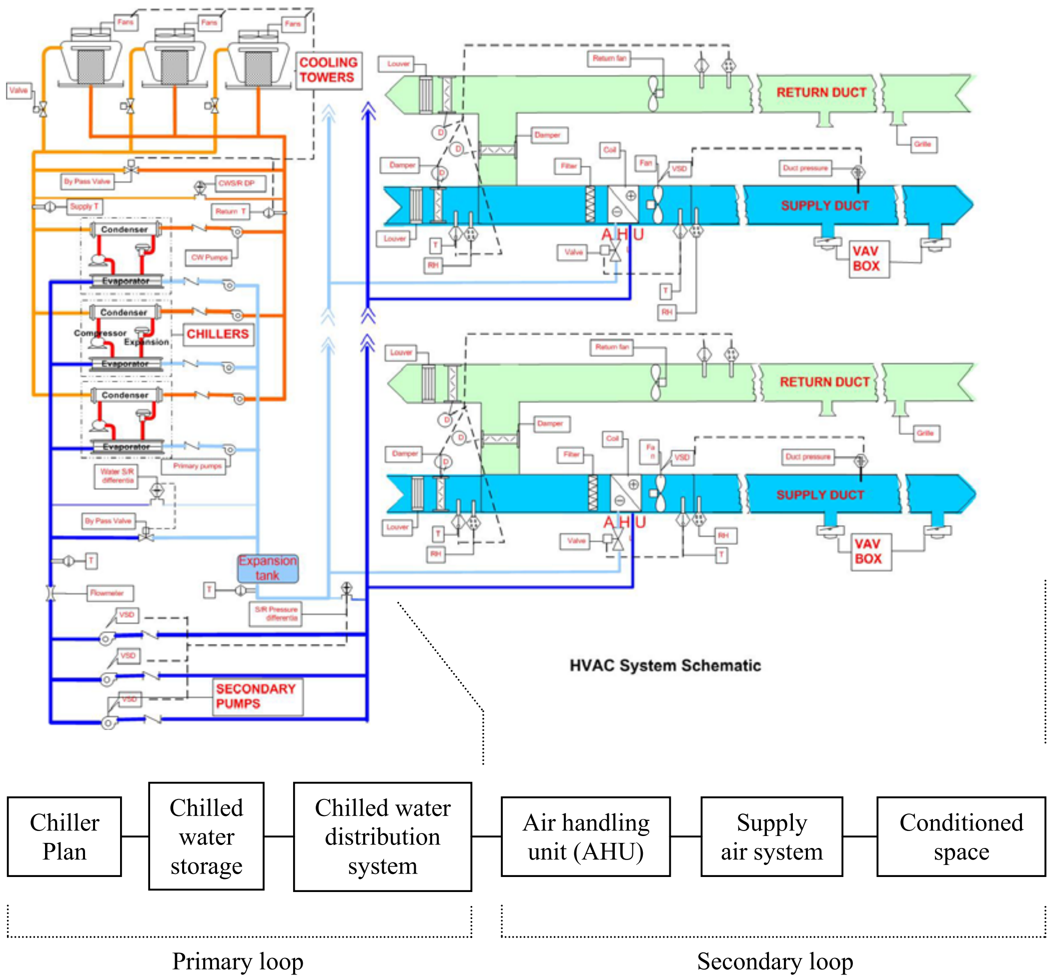


Figure 2.1 Example of schematic diagram of centralized HVAC system [33]

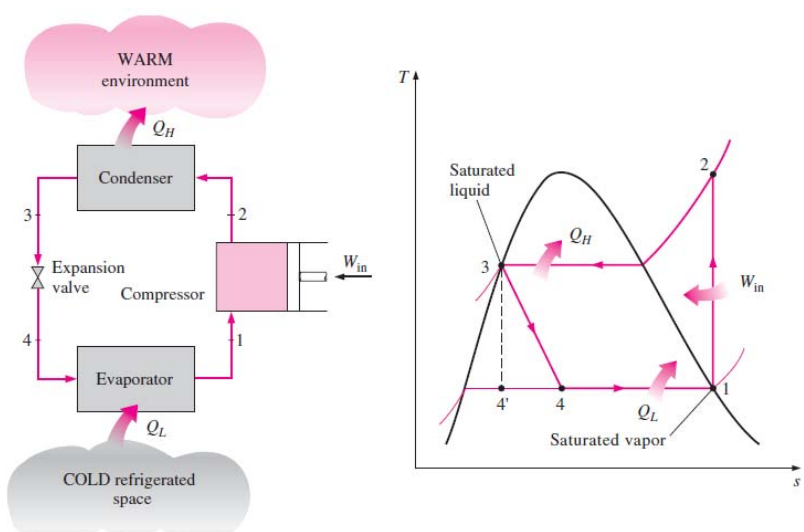


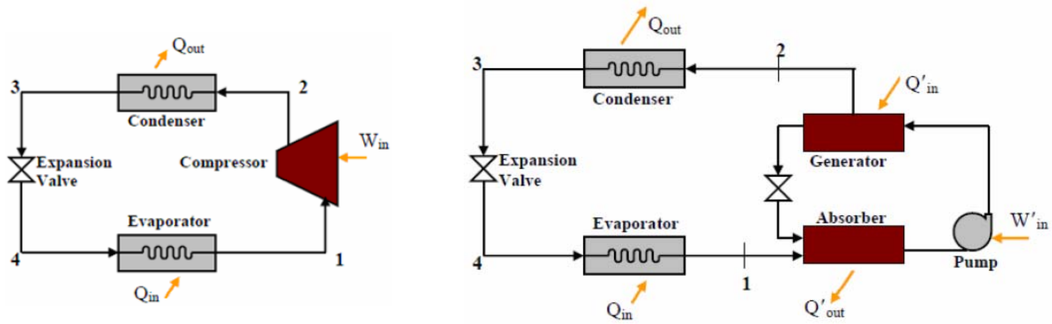
Figure 2.2 Schematic diagram of vapor-compression refrigeration cycle [32]

From Figure 2.2, state 1 – 2 is isentropic compression process where the refrigerant (as saturated vapor) is compressed to the condenser pressure. The temperature of the refrigerant increases during this process as the pressure increased. The refrigerant leaves compressor as super heated vapor and enters condenser at temperature above the temperature of the surrounding medium. In the condenser, the temperature of the refrigerant is decreased by heat rejection to the surrounding medium. The refrigerant leaves the condenser as saturated liquid at state 3. The refrigerant is then throttled by which reduces the pressure and the temperature drops below the conditioned space. At state 4, the refrigerant enters the evaporator as a low-quality saturated mixture. In evaporator, the refrigerant absorbs heat from conditioned space and completely evaporates. Leaving the evaporator, the refrigerant vapor re-enters the compressor at state 1 and finish the cycle [32].

The process in vapor absorption cycle and vapor compression cycle are basically the same. However, the way of the cycle increase the pressure of the refrigerants is different. In vapor absorption cycle, the compressor is replaced by absorption mechanism consisting of an absorber, a pump, a generator, a regenerator, a valve, and a rectifier [32]. Detail process of vapor absorption cycle will be described in the next section.

Vapor compression and absorption chillers are physical components that produce space cooling/heating and the two types of chillers used in many commercial buildings. Vapor compression chiller use motor-driven compressor to compress the refrigerant while absorption chiller depend on thermo-chemical process to get pressure difference for the compression process. Compared to vapor compression chiller, absorption chiller has lower coefficient of performance (COP), nevertheless, the operation cost is lower than vapor compression chiller because it is powered by available waste heat (at a temperature between $100^{o}C$ – $200^{o}C$) while vapor compression chiller is usually driven by motor or engine [33,34]. Illustration of vapor compression chiller and absorption chiller were presented in Figure 2.3.

With the ever increasing thermal power generation that produces waste heat and increase concern in reducing CO_2 emission, absorption chiller is a good choice as a low electricity consumption chiller that pretend additional CO_2 emission in providing cooling effect [34]. Primary loop using absorption chiller (as used in the studied building) would be explained further on the next section.



(a) (b)

Figure 2.3 Schematic presentation of a) Vapor compression chiller, b) Absorption chiller [34]

2.1.1 Primary Loop With Absorption Chiller

Primary loop of the HVAC system consists of chiller plant, and chilled water system. In the chiller plant, the refrigerant is circulated in a close system following the refrigeration cycle to cool water as medium to store the cooling effect. The system involves the absorption of the refrigerant by an absorbent through thermo-chemical process and the evaporation of the refrigerant by hot exhaust gas from the gas district cooling (GDC). Illustration of absorption chiller as used in the studied building was shown in Figure 2.4.

There are 40 refrigerant compounds and 200 absorbent compounds available as suggested by Marcriss [35]. However, the refrigerant/absorbent solution which commonly used in absorption chiller are water/lithium bromide, and ammonia/water system [36]. The application of first system is limited to air conditioning purpose only because the minimum temperature is above the freezing point of water while the second system can be used to provide low temperature below the freezing point of water [37].

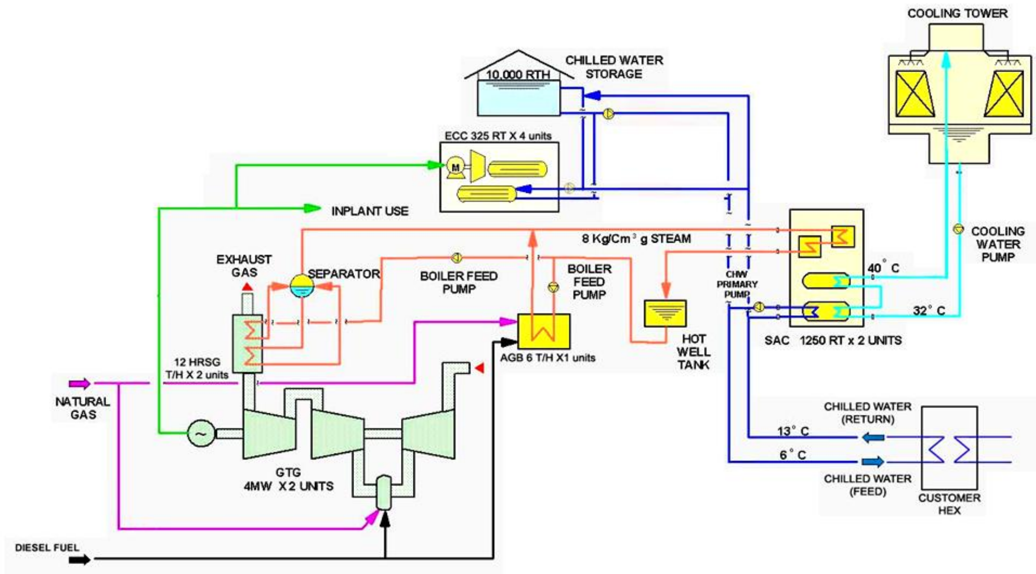
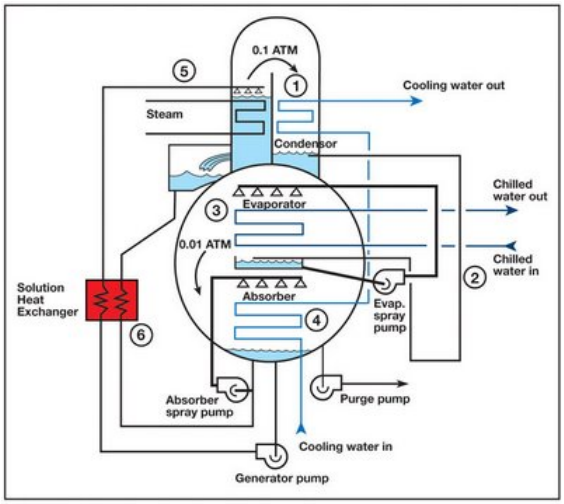


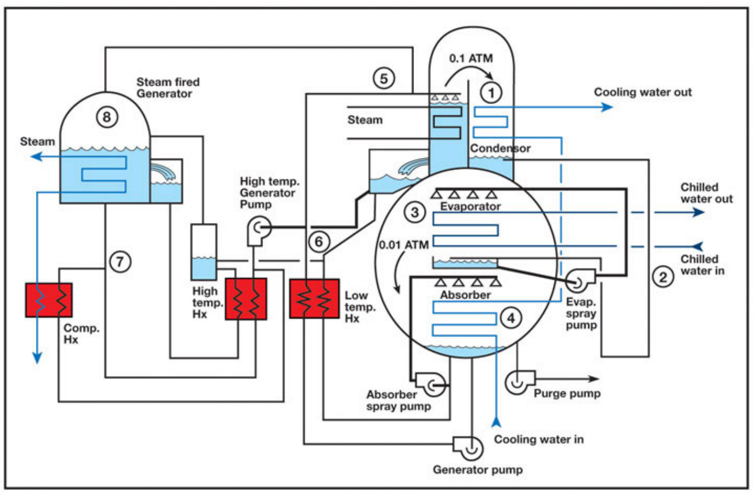
Figure 2.4 Schematic diagram of power generation and absorption chiller [38]

Absorption chillers available in the market can be categorized as direct- or indirect-fired and single, double, or triple-effect [34]. Single-effect means that the system use one generator while double-effect uses two generators: low temperature generator and high temperature generator. The triple-effect cycles are still under development to increase the chiller performance [39,40]. Direct-fired system use fired generator where fuel (oil/coal) is burned to provide the heat required. Indirect-fired system use available waste heat (steam/hot water) from the environment or other thermal systems [31]. Illustration of single- and double-effect absorption chiller is presented in Figure 2.5.

The double-effect chiller is 40% more efficient than the single-effect [40]. The double-effect system mainly consists of six components: condenser, expansion, evaporator, absorber, low temperature generator/concentrator and high temperature generator/concentrator. In the condenser, heat from vaporized refrigerant is transferred to the cooling water that in turn changes the refrigerant from vapor to liquid phase. The liquid refrigerant is then flowed to the evaporator pan through expansion pipe by which the pressure and temperature are decreased. The liquid refrigerant is then pumped to the top of heat exchanger (HX) and sprayed to the HX to remove heat from the chilled water.



(a)



(b)

Figure 2.5 Schematic diagram of: a) single-effect absorption chiller, b) double-effect absorption chiller [40]

After absorbing heat, the refrigerant evaporates and travels to the absorber where lithium bromide is sprayed to absorb the refrigerant. The solution temperature increases as a result of interaction between the absorbent and refrigerant. This heat is absorbed by cooling water to maintain the temperature of the solution below boiling point. The liquid solution is then pumped to the low temperature generator where the solution is heated to evaporate the refrigerant. The liquid solution from low temperature generator is then separated into two paths. First path flow the refrigerant to the absorber and being mixed with higher concentrated lithium-bromide solution coming from the high temperature generator. The other path flow the solution to the high generator where the solution is further heated to evaporate the refrigerant. The refrigerants vapor from low and high temperature generator flow to the condenser, finishing the absorption cycle [40,41]. Generally, coefficient of Performance (COP) of an absorption refrigeration system is described as [36]:

$$COP = \frac{\text{cooling capacilty obtained at evaporator}}{\text{heat input for the generator} + \text{work input for the pump}} \tag{2.1}$$

In order to increase COP as performance indicator of absorption chiller using $H_2O/LiBr$ solution (as widely used in many air conditioning system) Yoon and Kwon suggested new $H_2O/LiBr$□$HO(CH_2)3OH$ solution which result in COP approximately 3% higher than conventional $H_2O/LiBr$ working solution [42].

For cost optimization in an area where water is expensive, air-cooled absorption chiller can be utilized instead of water-cooled absorption chiller. However, it was possible when ambient temperature was below $40^{o}C$ and the evaporator temperature below $10^{o}C$ [43].

Jaruwongwittaya and Chen [44] found that in an area like Thailand where high solar radiation intensity was available, absorption chiller could use solar radiation if waste heat from another system was not available as alternative heat source. They also found that water/lithium bromide solution was the most suitable solution to be used in the chiller regarding to the solar radiation intensity. When excess solar radiation are available, phase change material (PCM) like Erythritol could be used as thermal storage during the day. The stored energy could have been used by the chiller at night since it could discharge the energy storage up to 70.9% [45].

In the water distribution system or chilled water system, the system pumps water to the chiller to store the cooling effect. The water is then stored in a water tank and distributed to the air handling unit (AHU) through water pipe network. Heat loss in transporting the chilled water to AHU cannot be avoided and thus, insulation on the water pipe is crucial to reduce the heat gain from outdoor environment [31].

2.1.2 Secondary Loop

The secondary system is basically an air handling system which consist of air-handling units, supply/return ductwork, fan-powered boxes, space diffusion devices, and exhaust systems. The purpose of an air handling system is to condition, to transport, to distribute the conditioned, re-circulating, outdoor, exhaust air, and to control the indoor environment according to the indoor thermal condition set point. On the basis of volume air displaced, air handling system is divided into two sub-systems i.e. constant air volume (CAV) and variable air volume (VAV) system. The first system maintains constant supply of air flow during the operation. It modulates the conditions of the supply air by varying the refrigerant flow rate to meet the cooling load [31]. The second system maintains constant supply air conditions. It varies the supply air flow rate using motorized-VAV damper in the VAV box based on indoor temperature measurement to meet the cooling load accordingly [46].

VAV system is usually coupled with variable frequency drive (VFD) to vary the voltage as well as power consumed of the supply fan to increase fan efficiency at part load conditions when the air flow rate needed was below 100% capacity of the fan. Therefore, the fan was always kept at maximum efficiency and thus, the fan consumes less energy during whole operation time [47]. It also leads to cooling load reduction which in turn reduces chilled water consumption due to less heat released from the fan. If variable pump is used, VAV system also reduces the cooling energy required due to less heat released from the pump. Compare to CAV system, VAV system used lower energy and thus, save energy in a range of 30% - 50% [48]. However, for VAV system with 100% outdoor air, controlling the supply air temperature would results in a significantly lower HVAC energy use than with a constant supply air temperature [49].

There are two control modes in VAV system i.e. dependent and independent pressure controls. In dependent control system, the actuator dampers regulate supply air flow rate based on signal from indoor thermostat only without regard to system conditions [50]. Due to this, two rooms on the same system will influence one to another. In independent control system, the actuator damper is controlled by indoor thermostat and velocity of the air stream at the inlet of the unit. Indoor thermostat regulates the damper (in order to regulate the supply air flow rate) in a room where the cooling load is changing. It will increase/decrease the static pressure in the system duct. The velocity reset controller then regulates another damper (in a room with no cooling load changing) as response to the changing in the inlet pressure conditions to maintain the required air flow.

Supply air conditions are important parameters in the air-system and mainly depend on room sensible heat factor (RSHF). The RSHF keep changing during HVAC operation and difficult to be predicted due to dynamic changing on occupancy pattern, occupancy needs, occupancy density, and outdoor conditions [7]. An interactive system which accommodated occupants' needs for controlling the air-conditioning system could save 20% of energy compare to air-conditioning system which keep indoor temperature constant at 26^{o}C [51].

If RSHF design stage was considerably higher than RSHF operation, it would result in higher humidity than the set point. If RSHF design was considerably lower than RSHF operation, it might lead to a condition where the airflow needed would be below the minimum airflow of the system (design limitation) and thus, overcooling exists [11, 28].

Previous research [28] suggested that RSHF design value should be chosen based on the lowest RSHF operation. For multi-zone building, where the AHU handle more than one room, RSHF design value should be chosen based on the lowest RSHF operation range from group of zones. In case the RSHF operation higher than RSHF design value, the system would increase the supply air temperature to reduce latent capacity and therefore, the indoor temperature and humidity could be maintained within the comfort range.

2.1.2.1 Duct Design and Sizing

Air-handling system design begins with determining supply airflow rate needed from the cooling load that is used to design the ducting system. The main purpose in ducting design is to size the air duct on each run so as to minimize the pressure drop through the duct. There are three methods to design ducting recommended by ASHRAE: equal friction method, static pressure regain method, and T-method [31].

Equal friction method keeps pressure drop per unit length ($\Delta p_f/L$) the same in the main and branch duct. Suitable $\Delta p_f/L$ is within range 0.08 – 0.6 in.WG per 100 ft as described by ASHRAE [31,116]. The airflow rate in the main duct is equal to the sum of air flow rates to all the conditioned zones. Using air flow rate, $\Delta p_f/L$, and friction chart, duct diameter and air velocity of the main and branch duct are determined. The frictional losses are then calculated by multiplying $\Delta p_f/L$ with the length of each duct and the dynamic losses are calculated based on the layout of the duct [34]. This method can be used to determine supply or return duct and it is widely used in practice by many engineers.

The bernoulli's equation states that decrease in velocity pressure will convert dynamic pressure into static pressure, which increases the static pressure [52]. Static regain method use this principle to design and to size the ducting. Illustration of pressure distribution of main duct under static region method is presented in Figure 2.6. From the picture, a decreased on static pressure in the supply main duct at branch 2 (at P_{s2}) due to pressure loss in the succeeding duct section (from point 1s – 2) would be regain at point 2s due to the reduction of air velocity from point 2 to point 2s. The main difference between this method and equal friction method is that the first method use length of succeeding duct section while the other use same pressure drop per unit length ($\Delta p_f/L$). In addition, static region method can be applied only for designing supply duct. Despite of the differences, both methods are based on initial guess and these two methods are unable to select the most economically efficient duct design [31,52].

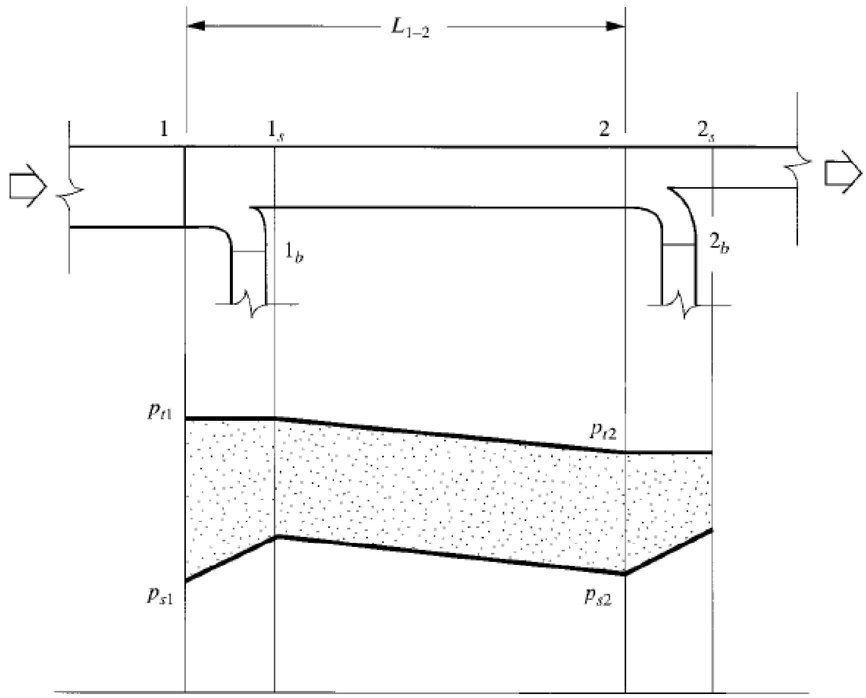


Figure 2.6 Pressure distribution of main duct under static region method [31]

The T method of duct sizing is an advanced optimization to size duct system by minimizing the life-cycle cost [53]. The method consists of three steps: system condensing, fan selection, and system expansion. The first step condense the various duct section of a duct system into single imaginary duct section which have same hydraulic characteristic and installation cost as the real one. The second step select appropriate fan and establish optimum system pressure loss. The last step expand the condensed imaginary duct section into the original system with optimum pressure loss distribution on each duct section as selected in step 2. The major difference between this methods with the other two methods described above is that this method consider constraint optimization. This method can be used to optimize both supply and return duct with air leakage as one system [31,53]. However, this method require complex calculation, while the simplified procedure called 1/3 boundary procedure is not accurate enough (rough approach).

2.1.2.2 Fan Selection and Low-SFP Design

Fan or blower is determined to provide the flow rate and to deliver the conditioned air to the conditioned rooms. The fan responsible to overcome highestpressure drop occurred (from ducting, cooling coil, filter, etc.) along the supply air path and peak airflow rate [54,55].

An air-duct distribution system is pressurized by three components pressure: static pressure, velocity pressure and total pressure. Total pressure is defined as summation of static pressure and velocity pressure. Static pressure is the air pressure of air stream flowing in a duct section that presses the duct wall. The value could be positive or negative. In air distribution system, the static pressure is used to overcome the various resistant (pressure drop of the system). Velocity pressure is the air pressure because of the air velocity and its weight that represents kinetic energy [56,57]. Static pressure, velocity pressure and total pressure measurements were illustrated in Figure 2.7.

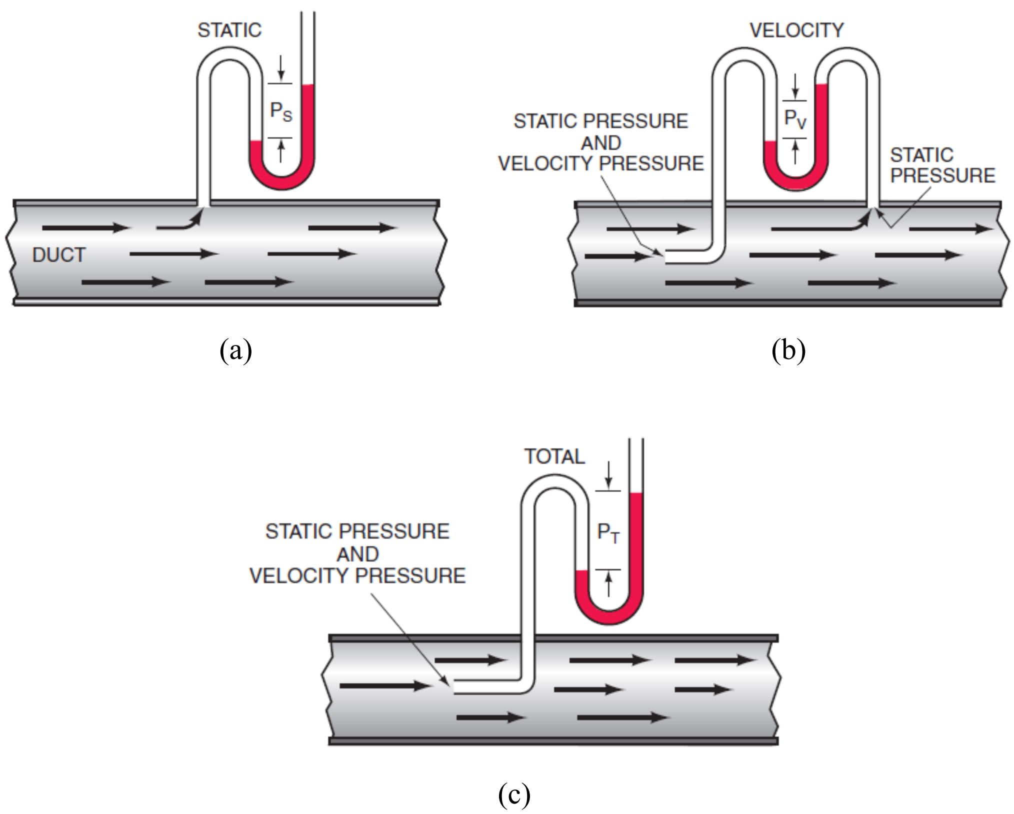
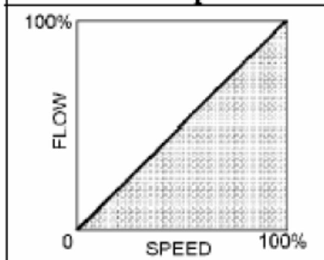
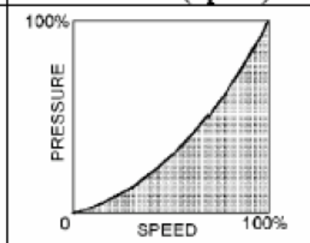
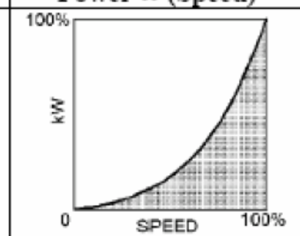


Figure 2.7 Measurements of, a) static pressure, b) velocity pressure, c) total pressure [56]

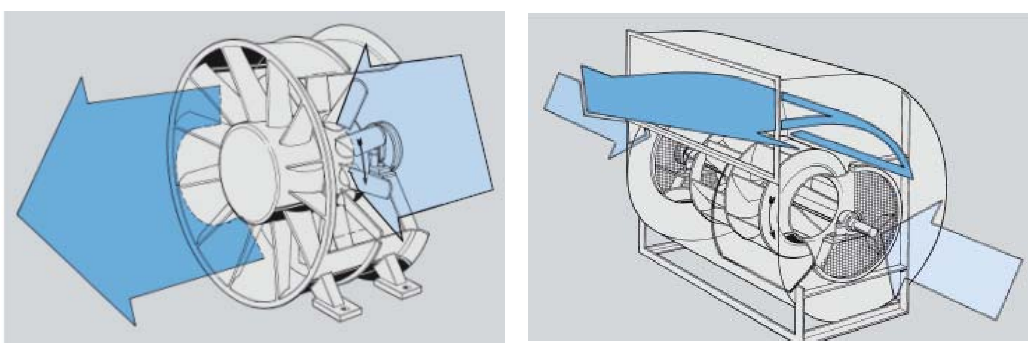
Fan operates with respect to certain sets of law which relates air flow rate, static pressure and fan power to fan speed. Any change on the fan speed will proportionally change the air flow rate, the static pressure and the fan power required to operate at the new fan speed [58]. The fan laws are presented in Table 2.1.

There are two main types of fan according to the direction of flow: centrifugal fan and axial fan. With centrifugal fan, the direction of airflow change twice: when the air entering and leaving the fan. With axial fan, the direction of air leaving the fan is the same with the direction of air entering the fan [59]. Illustrations of directions of airflow in centrifugal and axial fans are presented in Figure 2.8.

Table 2.1 Fan laws [58]

Flow ∞ Speed	Pressure ∞ $(Speed)^2$	Power ∞ $(Speed)^3$
		
$\frac{Q_1}{Q_2}=\frac{N_1}{N_2}$	$\frac{SP_1}{SP_2}=\left(\frac{N_1}{N_2}\right)^2$	$\frac{kW_1}{kW_2}=\left(\frac{N_1}{N_2}\right)^3$
Varying the RPM by 10% decreases or increases air delivery by 10%.	*Reducing the RPM by 10% decreases the static pressure by 19% and an increase in RPM by 10% increases the static pressure by 21%.*	*Reducing the RPM by 10% decreases the power requirement by 27% and an increase in RPM by 10% increases the power requirement by 33%.*

Where Q – flow, SP – Static Pressure, kW – Power and N – speed (RPM)



(a) (b)

Figure 2.8 Direction of air flow, a) axial fan, b) centrifugal fan [59]

Centrifugal fan is generally used in air-duct of an air conditioning system because it requires less input energy at higher static pressure. This fan has three types in common: radial (T-wheel), forward curved (F-wheel), and backward inclined fan (B and P-wheel) [58]. Radial fans, with flat blades is suitable for high static pressure and temperatures, nevertheless, the usage of this types of fan is limited for low – medium airflow rates. Forward curved fans, with forward curved blades has ability to move large air volumes against relatively low pressure. However, it has low energy efficiency (55% – 65%). The third types of the centrifugal fan have special ability to operate with changing static pressure and thus, it is used in a condition where the system behavior at high air flow is uncertain. The detail characteristics of the centrifugal fans are summarized in Appendix 2.1.

There are three terms used to define fan: system characteristics, fan characteristics, and system characteristics and fan curves [58,60]. System characteristics refer to total static pressure loss along the duct system including elbow, pickups and pressure loss across equipments (i.e. filter, heat exchanger, etc.). The pressure loss is considered as system resistance. In a duct system, the system resistance is keep changing according to the flow. If the flow is reduced, the system resistance will decreases and vice versa [60]. The system resistance curve (system curve) is usually plotted on a curve with certain range of airflow rate.

Fan characteristic is defined as a fan performance under a specific set of conditions [58]. Generally, the fan performance is plotted on a curve called fan performance curve. The curve is presenting a number of inter-related parameters including fan volume, system static pressure, fan speed, efficiency and brake horsepower required to drive the fan under the stated conditions. Illustration of fan performance curve including system resistance curve (system curve), brake horse power (BHP) curve and efficiency curve is shown in Figure 2.9.

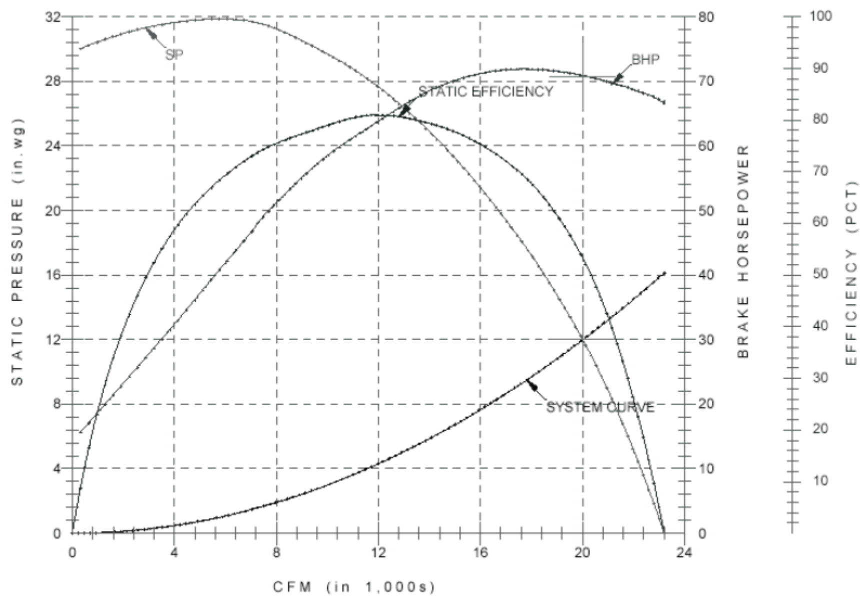


Figure 2.9 Illustration of fan performance curve [58]

On that curve, the operating condition is the intersection between static pressure curve and system curve [58]. Most of the cases, fan operates at reduced capacity. When the flow is reduced at part load condition, the system resistance decreases accordingly [60]. It results in different system resistance curve. There are two ways of reducing airflow from Q_1 to Q_2. First method is using damper to restrict the airflow that cause higher-pressure loss and change the system performance curve from SC_1 to SC_2. It increases the operating static pressure from P_1 to P_2. Second method is using variable speed drive device to reduce the rotational speed of the fan from N_1 to N_2. It results in different fan curve. With fan curve at N_2, the operating static pressure decreases from P_1 to P_3 as shown in Figure 2.10.

In sizing appropriate fan, there are three things must be considered: peak airflow rate, Static pressure at a peak volumetric flow rate, and fan efficiency. Peak airflow rate is first determined from the cooling load. After that, ducting lay out and sizings are determined according to the peak airflow rate. Using this information, duct static pressure required is calculated. Finally, the fan is selected based on the peak airflow rate and static pressure required. The selection process must refer to fan performance curve. It is to ensure the fan to have maximum efficiency at design volume rate [55,61].

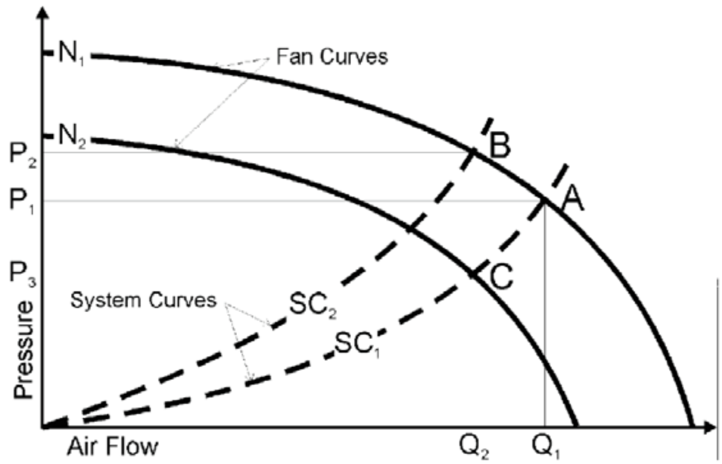


Figure 2.10 Fan performance curve with system curve at design and part load condition [60]

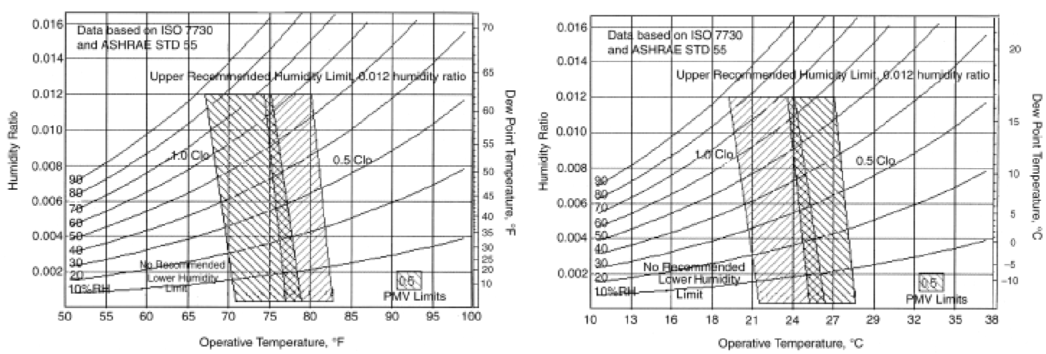
However, in practice, engineers oversized capacity of the fan to anticipate long-term demand growth in the future. This practice lead to higher operational cost because the fan operate at lower efficiency and thus, consumed more energy. With increasing of oil and electricity price, fan selection based on current design airflow rate and predicted short-term demand growth gave significant operational cost saving which could be used to buy new fan with larger capacity as the demand increased [61].

Specific fan power (SFP) is normally determined to represent the air system performance as stated in ASHRAE/IES Standard 90.1-1989 (A-90.1). It is defined as the power required by the motors for the combined fan system, i.e., all fans, divided by the supply flow at design conditions. Based on the standard, SFP for VAV system should not exceed 2.65 $kWm^{-3}s$ at design conditions or 1.6 $kWm^{-3}s$ under average operating conditions [62]. It implied that the pressure loss must be maintained as low as possible and the fan efficiency must be keep high to ensure the SFP below the standard [63].

2.2 Indoor Thermal Comfort

ISO 7730 standard [64] and American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) [65] define thermal comfort as "condition of mind which expresses satisfaction with the thermal environment". It is impractical to meet the thermal satisfaction of all the occupants since physiological and psychological perceptions are varying from person to person. Large experimental data from laboratory and field studies have been collected to provide enough statistical data. Based on these data, a condition is selected for thermal comfort of a specified percentage of occupants. ASHRAE define thermal comfort zone (as shown in Figure 2.11) that can satisfy a minimum of 80% occupants. The comfort zone is applicable for typical office room where [65]:

- The activity level result in metabolic rate between 1.0 and 1.3 met,
- People's clothing result in 0.5 and 1.0 clo of thermal insulation,
- The air speed is below 0.2 m/s.



(a) (b)

Figure 2.11 ASHRAE thermal comfort zone, a) operative temperature ^{o}F, b) operative temperature ^{o}C [65]

There are six primary factors influencing thermal comfort: metabolic rate, clothing insulation, air temperature, radiant temperature, air speed, and humidity [65]. Metabolic rate factor is used to determine sensible and latent heat released to the conditioned space from the occupants according to their activity. Clothing insulation factor is a factor that relates people's clothing with their thermal comfort. Air temperature is the average temperature of air in a conditioned space that respect to location and time. The temperature is the average temperature of air at 0.1, 0.6, and 1.1 m (the ankle, waist, and head level) and 0.1, 1.1, and 1.7 m for standing occupant.

Radiant temperature is representation of surface temperatures surrounding an occupant. Air speed is the average speed of air nearby exposed body of the occupant with respect to location and time. Humidity is general term to represent vapor content in the air. These six factors are considered in predicted mean vote (PMV) model to determine the response of people under specified conditions that is related to predicted percentage of dissatisfaction (PPD) from occupant [65]. PMV and PPD are used as indicators of indoor thermal comfort and will be further discussed in next section.

Indoor thermal comfort standard strongly influence energy consumption of the HVAC system. Lowering indoor temperature set point from 26^{o}C to 24^{o}C increased the energy consumption by 50% [66]. Since indoor thermal comfort is strongly influenced by the local weather condition, the standard is different by location [67].

2.2.1 PMV and PPD

Optimum comfort temperature is defined as 'the indoor operative temperature at which most of the occupant feels comfort (or neutral)' [22]. Comfort condition level is usually represented by PMV and PPD values. The acceptable limits of PPD which are widely used to describe a comfort zone are 10% and 20%. These values represent 90% and 80% of acceptability from the occupant [68].

The PMV index is calculated based on heat balance of human body involving the terms of internal generation and heat exchanges with the surrounding environment. The equation to calculate the PMV established by Fanger is a model of correlation between the subjective human perception and the difference between the heat generated and the heat released by the human body. Subjective human perception is expressed through the vote of comfort on a scale ranging from -3 (very cold) to +3 (very hot). The PMV equation established by Fanger is defined as [68]:

$$PMV = \left(0.303e^{-2.1\times M} + 0.028\right)\times\left[\left(M - W\right) - H - E_c - C_{res} - E_{res}\right] \qquad (2.2)$$

PMV model as shown above is predicted value for thermal sensation of the occupants by considering four physical variables (air temperature, air velocity, mean radiant temperature, and relative humidity), and two personal variables (clothing insulation and activity level) [68]. However, the predicted value is not always accurate to predict actual thermal sensation, particularly in field study settings. The discrepancies reflect the difficulties to get accurate measurements of clothing insulation and metabolic rate. In addition, the accuracy of the PMV model are more accurate for air-conditioned building rather than in natural ventilated building, in part because of the influence of outdoor temperature, and opportunities for adaptation [69]. Related to this, a study conducted by De Dear et al [70] found that neutral temperatures in air-conditioned and naturally ventilated buildings were under-predicted by 0.2^{o}C and over-predicted by 2.8^{o}C by using the PMV model.

Extended analysis of the PMV discrepancies done by Humphreys and Nicol's [71] on the ASHRAE RP-884 database found that the PMV model accurately predict the actual thermal sensation for clothing insulation in the range 0.3 to 1.2 clo, for activity levels below 1.4 met, and for air-conditioned buildings.

PPD (Predicted Percentage of Dissatisfaction) is the other index proposed in ISO Standard 7730 [64] that represent the expected percentage of dissatisfied people in a given thermal environment. According to Fanger's study, the variation of PMV index can be approximated by an analytic expression that corresponds to a curve whose appearance is similar to an inverted Gaussian distribution (see Figure 2.12), thus [68]:

$$PPD = 100 - 95e^{-(0.03353PMV4 + 0.2179PMV2)} \tag{2.3}$$

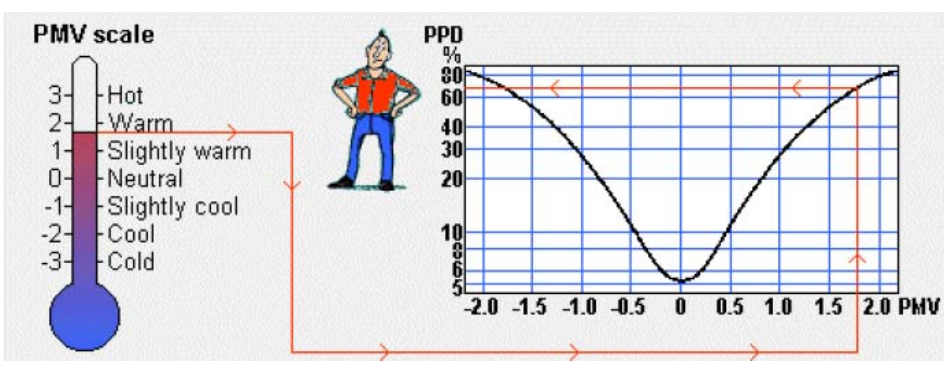


Figure 2.12 PPD – PMV scale [68]

It can be concluded that even for optimum selection of the situation where majority people feel comfort, the percentage of dissatisfaction is 5%. It is due to fact that the subjective perception of thermal comfort differs from person to person.

2.2.2 Indoor Thermal Comfort in Malaysia

Study on thermal comfort in a campus building in Malaysia showed that people are more sensitive to the variation of temperature rather than relative humidity [72]. This finding is in contrast with other findings about thermal comfort at high humidity that showed that "there were no significant psychological or physiological differences in human response to exposure of between 60% to 90% of relative humidity for the temperature range from 20ºC to 26ºC effective temperature while sedentary" [73].

Daghigh et. al. [74] conducted fields' survey for indoor thermal comfort in office building (Malaysia). The results found that for naturally ventilated building, the occupants felt thermally comfortable when the indoor temperature was 28.6^{o}C with 68.1% of indoor relative humidity. For air-conditioned (AC) building, the occupants felt thermally comfortable when the indoor air temperature was 23.5^{o}C with 56.1% of relative humidity. In 2009, the research for naturally ventilated (NV) building was repeated for different office room that considers 14 opening arrangements of the window and door. The results found that the neutral temperature under the 14 opening arrangements was between 25.2ºC -27.5ºC [75].

Hussein et al. [18] conducted another field study on thermal comfort in Malaysia. The research took place in UNITEN, primary school, secondary school, and public waiting area in a health clinic in Johor Bahru. Questionnaires and surveys were conducted for occupants in AC (total 184 respondents) and NV (total 375 respondents) buildings to collect their vote on the thermal sensation. The scale used was 7 point ASHRAE thermal sensation scale (-3 (cold), -2 (cool), -1 (slightly cool), 0 (neutral), 1 (slightly warm), 2 (warm) and 3 (hot). For humidity assessments, the scale used was -3 (much too dry), -2 (too dry), -1 (slightly dry), 0 (just right), 1 (slightly humid), 2 (too humid) and 3 (much too humid). Thermal sensation vote data (TSV) for AC and NV buildings were then plotted on a graph against the operative air temperature as shown in Figures 2.13 and 2.14.

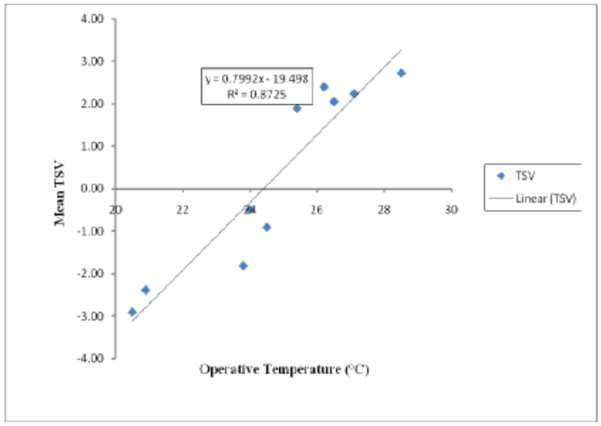


Figure 2.13 Regression of TSV on operative temperatures for air-conditioned buildings [72]

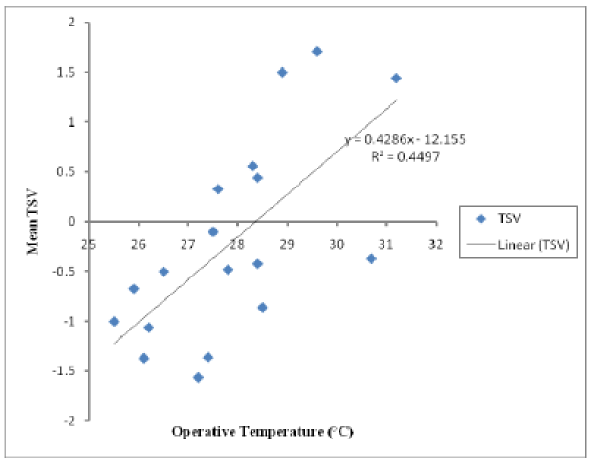


Figure 2.14 Regression of TSV on operative temperatures for non air-conditioned buildings [72]

Regression of the TSV and operative temperature for AC building showed that indoor neutral temperature was 24.4°C with comfort range 23.1°C – 25.6°C. The humidity assessments found that most of the occupant (>80% voted) felt comfort with relative humidity within 74% - 83% range [72]. The findings differ with standard set by Department of Standards Malaysia (MS 1525) [76] where indoor thermal comforts were 23°C – 26°C with relative humidity 60% - 70% [77]. Regression of the TSV and operative temperature for NV building showed that indoor neutral temperature was 28.4°C with comfort range 26°C – 30.7°C [18,72]. In designing HVAC system, PTM set outdoor design conditions for Malaysia to be 33.3°C of dry bulb temperature and 27.2°C of wet bulb temperature [77].

The results [72] were in good agreement with field study conducted by Busch [78] in Thai offices in Bangkok, Thailand which found that neutral temperature for the air-conditioned buildings and naturally ventilated buildings were 24.5^{o}C and 28.5^{o}C. The results also closed to another results from field study done by De Dear [70] which found that the neutral temperatures in the air-conditioned buildings and naturally ventilated buildings were 24.2^{o}C and 28.5^{o}C. Daghigh et al. [75] summarized the neutral temperatures and comfort ranges for occupants in the hot-humid regions as shown in Appendix A.

2.3 Space Cooling Load

The terminology of sensible and latent heat transfer between the space air and the surroundings can be explained as follows [31]:

1. Space heat gain q_e, in Btu/h, represents the rate of heat that enters a conditioned space from an external source whereas heat released to the surrounding from an internal source during a given time interval is termed as heat loss.

2. Space cooling load, often simply called the cooling load Q_{rc}, Btu /h, is the rate of heat that must be removed from a conditioned space so as to maintain indoor air temperature and relative humidity set point.

3. Space heat extraction rate Q_{ex}, Btu /h, is the rate of heat that is actually removed from the conditioned space by the air system.

4. Coil load Q_c, Btu /h, is the rate of heat transfer at the heat exchanger in AHU. The cooling coil load Q_{cc}, Btu/h, is the rate of heat that is absorbed by the chilled water or refrigerant flowing through the coil.

5. Refrigerating load Q_{rl}, Btu /h, is the rate of heat that is absorbed by the refrigerant at the evaporator. For central hydronic systems, the refrigerating load is the sum of the coil load plus total heat transferred to the chilled water from the components installed in the system. The refrigerating load is equal to the coil load for individual air conditioning system using direct expansion (DX) coil(s).

Storage of part of the radiative heat inside the building structures would result in lower instantaneous sensible cooling load than the instantaneous sensible heat gain. If the space relative humidity is kept approximately constant, the storage effect of the moisture in the building envelope and furnishings can be ignored. In this case, the instantaneous space latent heat gain will be equal to the the instantaneous space latent cooling load [31].

A load profile is usually used to illustrate the load variation of an air conditioned space—a room, a zone, a floor, a building, or a project over a period of time. The shape of the profile would be strongly affected by the outdoor climate, the operating characteristics and the variation of the internal loads. “The load duration curve is the plot of number of hours versus the load ratio. The load ratio is defined as the ratio of cooling or heating load to the design full load, both in Btu/ h, over a certain period. The period may be a day, a week, a month, or a year” [31].

The zone peak load is the maximum space cooling load in a load profile of a control zone according to the zone orientation, the internal load characteristics, and the outdoor design conditions containing summer and winter. For a zone cooling load with several components, such as solar load through window glass, heat transfer through roofs, or internal load from electric lights, the zone peak load is always the maximum sum of these zone cooling load components at a given time. The block load is the maximum sum of several zones cooling loads of a group of control zones in a building at the same time. The block load of a space, room, floor, or building is the maximum cooling load in that space, room, floor, or building at a given time.

For air systems, the supply volume flow rate required is calculated based on the cooling load in that zone, space, room, area, floor, or building. For a control zone, the supply volume flow rate is calculated based on the zone peak load while for a specific area the rate is calculated based on the block load (cooling) of this specific area, floor, or building. For conditioned space using variable-air-volume systems and space air conditioning systems, the required cooling coil load or refrigeration load can be calculated based on block load of the corresponding specific area that air system serves.

Cooling load derived from external sources and internal sources [12]. External sources consist of heat gains through building envelope, solar radiation, or due to ventilation/infiltration. Internal sources consist of heat gain from occupancy, machines, lights and electric appliances, etc. It implies that cooling load characteristic is specific according to the type of the room [12,13], weather conditions [12], and building envelope [14,15,16,79].

The characteristic of cooling load in any rooms consists of sensible (*RSH*) and latent (*RLH*) components [31,56]. The portion of these components is represented by the room sensible heat factor (*RSHF*) which is the ratio of sensible heat to the room total heat (*RTH*). *RSHF* was used to determine the supply air conditions in order to maintain indoor thermal set points. Ventilation air that was supplied to the room from outside air gave additional both sensible (*OASH*) and latent (*OALH*) heat. The total of *OASH* and *RSH* is grand sensible heat (*GSH*) while the total of *OALH* and *RLH* is grand latent heat (*GLH*). The ratio of *GSH* to grand total heat (*GTH*) is represented by the grand sensible heat factor (*GSHF*). In design stage of HVAC system, *GTH* is used to size the cooling coil while the *GSHF* is used to determine the portion of sensible and latent heat capacity of the cooling coil.

In the design stage of air conditioning system, cooling load calculations are mainly used to determine the volume flow rate of the air system as well as the coil and refrigeration load of the equipment. The cooling load is then used as a base to size the HVAC&R equipment and to select optimal design alternatives. There are three heat transfer modes by which heat is transferred from external or internal heat sources to the conditioned space i.e. conduction, convection and radiation. Conduction is heat transfer occurred in a solid medium. The heat rate (q_x) is calculated as [41]:

$$q_x = \frac{k}{l} A \Delta T = q'' A \tag{2.4}$$

where, k is thermal conductivity (W/m.°k), l is thickness of the medium (m), A is surface area (m^2), q'' is heat transfer rate per unit area (W/m^2), and ΔT is the temperature difference between surfaces in the medium (°k). Convection heat transfer is heat transfer occurred between a surface and a moving fluid. The heat rate (q_c)is calculated as below [41]:

$$q_c = hA\Delta T = q''A \tag{2.5}$$

where, h is convection heat transfer coefficient (W/m^2.^{o}k), and ΔT is temperature difference between the surface and the moving fluid. Radiation heat transfer is heat transfer between two surfaces in the form of electromagnetic waves and thus, it requires no medium. The heat rate (q_r) is calculated as below [41]:

$$q_r = \varepsilon\sigma A\left(T_s^{\ 4} - T_{sur}^{\ \ 4}\right) = q''A \tag{2.6}$$

where, ε is emissivity, σ is Stefan Boltzmann constant (5.67 x 10^{-8} W/m^2.^{o}k^4), T_s is surface temperature (^{o}k), and T_{sur} is the surrounding temperature (^{o}k). Illustration of these heat transfers was presented in Table 2.2.

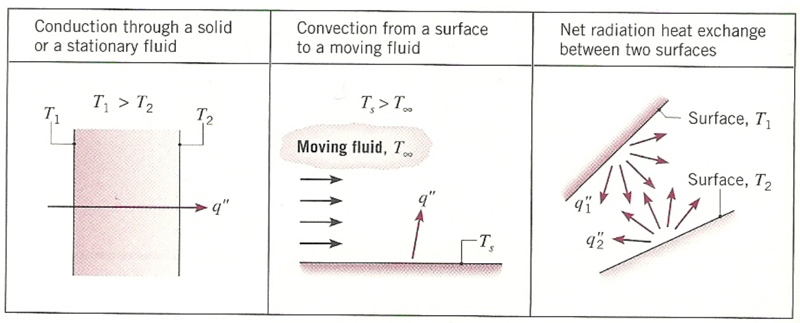


Table 2.2 Conduction, convection, and radiation heat transfer modes [41]

All of the heat transferred to the condition space is space cooling load or heat that must be removed from the space in order to maintain certain set points. There are two components of space cooling load: latent and sensible cooling load. Latent load is cooling load triggered by humidity difference while sensible load is cooling load triggered by temperature difference between conditioned room and the environment [31]. In tropical countries, the portion of latent load was higher than in many European countries due to the weather conditions which are hot and humid. Due to the fact, the sensible heat ratio (SHR) standard of room air conditioner (RAC) was lower than SHR in European countries [12]. This implied that dehumidification capacity of the RAC or HVAC system should be increased.

In most of the cases, the major space cooling load was heat gain from building envelope which was triggered by the weather conditions. Properly design building envelope would significantly reduce the building energy [14,15,16,79].

2.4 Energy Saving Potential in HVAC System

Energy saving strategies are generally achieved through system optimization, system modification, and internal/external heat gain reduction. A number of such strategies are discussed in the literature and are summarized in the following paragraphs.

HVAC system optimization, with CO_2 concentration control, get energy saving from optimization of how much outside and return air will be inserted into the mixing box according to the temperatures of both airs, and the concentration of CO_2 of the outside and return air. If CO_2 concentration of return air was lower than the desired set point, the outdoor air damper would be maintained at minimum position and therefore, reduce the energy consumed to condition outdoor ventilation air. This system prevented over-ventilated air and ensure that the amount of ventilation air is as much as it is required [80,81].

With desired CO_2 concentration of 400 ppm, 600 ppm, and 800 ppm, the system could achieve 11.5%, 14.2%, and 21.6% of energy saving [81]. According to this saving, CO_2 based ventilation control system are likely to be desired ventilation strategy. The system was not merely reducing the energy consumption but also maintain indoor air quality (IAQ) which is important for indoor thermal comfort [80]. However, this system would difficult to remove odors generated from inside the zone since the system would re-circulate more return air when the CO_2 concentration level is below the set point.

Another research [82,83] on economizer showed that the economizer could further increase energy saving of a VAV system by utilizing free cooling capacity of outdoor air to directly offset the space cooling load. Optimal On-Off control of an air conditioning and refrigeration system also offered energy saving by reducing on-off cycling frequency during part load operation. The system optimized the on-off cycle frequency to avoid high power consumption during start up of the air conditioning or refrigeration system. The system could give approximately 5.6% of energy saving in one hour [84].

An optimization research integrated genetic algorithm, and Artificial Neural Network (ANN) to optimize multi objectives in building design. The work considered two design variables: building envelope, and HVAC system [85]. Building envelope-related variables included heating/cooling temperature set points, RH set point, supply airflow rate, and thermostat delays. Building envelope-related variables were focused on getting advantages of passive solar design. The objectives of the function were thermal comfort which indicated by PMV value and energy consumption of the cooling/heating process. The results showed that the control system able to find optimum setting for the design variables, leading to lower energy consumption. On HVAC system optimization, thermal mass variable was set to constant value to get optimum set point for the other design variables.

Another optimization control suggests neural network based for optimal operation of VAV system. Typical two-zones VAV system as shown in Figure 2.15 is considered. The system maintained desired indoor temperature under variable operating conditions using 5 controllers which control: fan and compressor speed (by modulating normalized voltage input), air and chilled water flow rate (by modulating damper in VAV box and chilled water valve opening), and outdoor air damper. The main operation strategy is to find the optimal set points for chilled water supply temperature, supply air temperature and VAV system fan in a way that the indoor environment was maintained with the least chiller and fan energy consumption. Two dynamic models were first developed: two-zone VAV system model and the chiller model. The overall system was then developed by integrating these two models. The simulation results using the overall system and operation strategy showed that the operation scheme could lower energy consumption up to 10% during full load condition and 19% under partial condition [86].

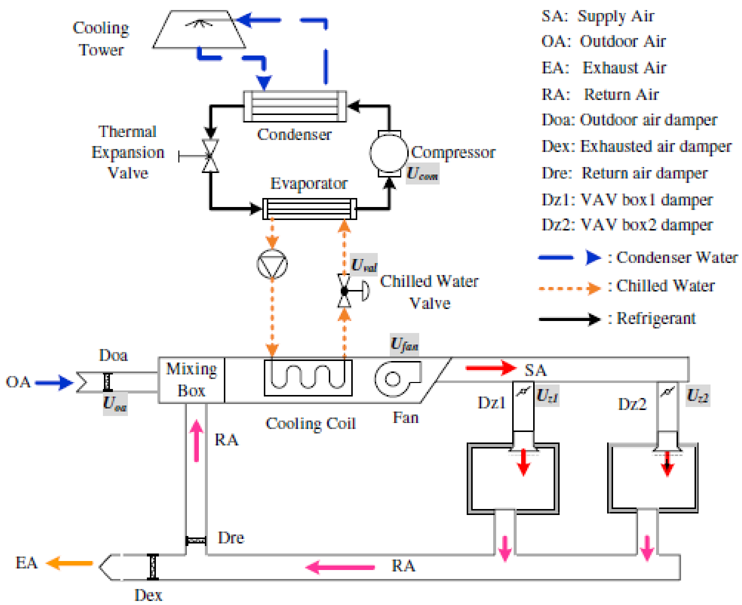


Figure 2.15 Schematic diagram typical two-zone VAV system [86]

A temperature set back algorithm which uses two indoor temperature set points for occupied and unoccupied period could reduce energy consumption of the HVAC system. This system require additional occupancy sensor to track occupancy inside the room [87]. The selection of indoor temperature set points during unoccupied period and the effect of these two temperature set points to the HVAC system performance were not discussed in detail in the previous research and thus, extensive research to accommodate these matters is needed.

Simulation study on dedicated outdoor air system (DOAS) also showed energy saving potential by treating outdoor air separately before it enters the building. Simple DOAS could save energy in range 14% - 37% while full DOAS could save energy in slightly higher range 21% - 38% [88].

Properly designed building envelope to make use of solar radiation as natural lighting would reduce electricity consumption and peak cooling load as much as 13% and 11% [79]. Study on trade off between thermal comfort and energy building consumption showed that there was possibility to slightly reduce thermal comfort in order to get considerable energy saving. Reducing predicted mean vote (PMV) as representative of thermal comfort from 0.08 to 0.11 would reduce building energy consumption up to 13% [85].

2.5 Building Simulation Program

Building simulation program (BSP) is an important tool in designing HVAC system and evaluating building thermal or HVAC performance [89]. Load calculation performed by the BSP were based on heat balance method as preferred method as stated by ASHRAE [90]. The tools are able to perform (hourly or sub-hourly) simulations (Carrier HAP, Trane TRACE 700, DOE-2, eQUEST, EnergyPlus, ESP-r, IDA ICE, TRNSYS, HVACSIM+, VA114, SIMBAD, etc.) based on a studied system that define thermal performance of buildings and systems, with given boundary conditions/assumptions, and with operation strategies and controls [91]. These simulation programs have been used by many researchers around the world to perform numerous calculations in order to analyze energy performance of a building [79], to model and simulate studied system [92], and to simulate performance of proposed systems [85,93].

A building under consideration for simulation may have multiple rooms with different thermal conditions and various orientations. A complex building may require the user to simplify the building into simple models, dividing into different zones on the basis of their orientation and thermal similarities. It is believed that the zoning of a building is often one of the toughest tasks in building simulation [91]. Another simplification was to neglect the influence of moisture content in indoor furnishings to the indoor RH level since the influence is negligible at less than 2% RH [12].

TRNSYS is a transient simulation program that is capable to integrate building and HVAC systems. Level of details of building and system models can vary from simple to very complex depending on desired accuracy levels and capability of the users. TRNSYS also allows users to make their own modules which can then be linked into the system [94]. This feature differentiates it from other simulation programs as it can be extended or customized by users to cover some special applications. In addition, the program allows to be directly embedded with components developed using other software (e.g. Matlab/Simulink, Excel/VBA, and EES) [95]. TRNSYS consist of three main parts: simulation studio, simulation engine, and building visual interface.

TRNSYS simulation studio is the main visual interface where users can drag and drop components required, make links between components, and set global simulation parameters for the project. The project file is saved in .tpf file. When the project is executed, TRNSYS create additional information file (.inf) about the detail of the project. This information can be printed for analysis or documentation. TRNSYS simulation studio provides output manager from where the user determines the output required. The simulation studio also provides error manager from where the user is able to monitor the simulation process [96].

TRNSYS simulation engine is engine to read all information in the TRNSYS input file (.dck file) including additional input file (e.g. weather data) and perform necessary calculations to create the output file. The engine also implement online plotter where the user would be able to monitor all desired output variables during a simulation [96].

Building visual interface or TRNBuild is a tool to model multizone building under investigation. It allows the user to describe in detail the building construction and to enter all inputs needed to represent thermal behavior of the building such as wall material, glazing type, cooling/heating schedule, etc. When the building model is saved (in .bui file), TRNBuild create another additional information file (,inf) on which all parameters, inputs, and outputs information of the model are described. The information file is provided to the user to easily trace or analyze the model [96].

However, even though BSP like TRNSYS can provide results corresponding to what the user inputs, but they cannot provide suggestions to improve design. It means that knowledge-based of the users about the system is crucial to find strategies which will improve the design of the building and its HVAC systems, and optimize the operation of HVAC systems in order to reduce the energy consumption [91,95,97].

2.6 Adaptive Comfort

2.6.1 Adaptive Comfort Model

Previous work [98] suggested adaptive comfort model as a model to find desired comfort temperature (T_c) based on adaptive comfort studies. It found that T_c is strongly correlated to the outdoor air temperature and behavioral adaptation of the occupant. The work also collected data about indoor comfort temperature according to the outdoor monthly mean temperature (as presented in Figure 2.16) from surveys conducted world-wide.

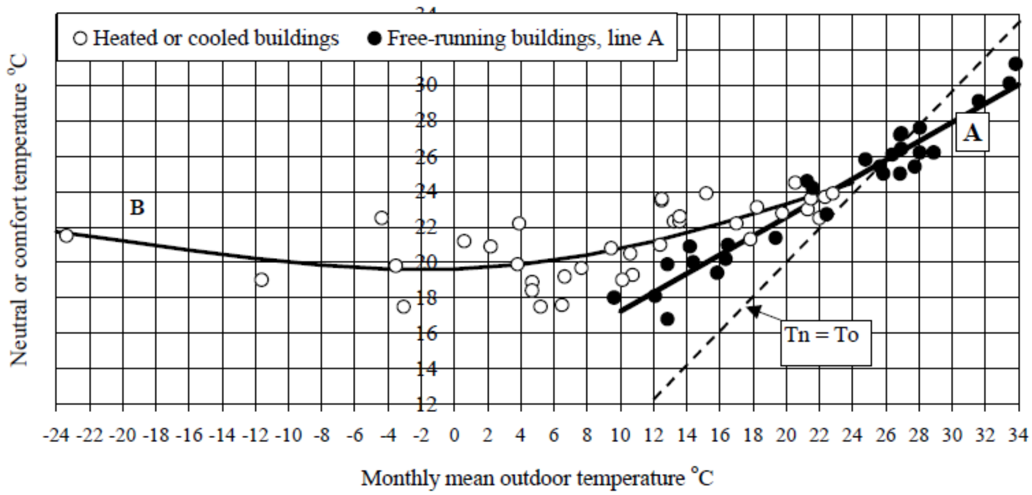


Figure 2.16 Comfort temperature according to the outdoor monthly mean temperature [98]

Based on the Figure 2.16, Humphrey [98] suggested that desired T_c (with 100% of acceptability) could be derived from an equation in the form :

$$T_c = aT_{a,out} + b \tag{2.7}$$

where, $T_{a,out}$ is outside temperature index which can be determined as monthly mean outside temperature. The equation was then become general adaptive comfort model to find T_c from $T_{a,out}$ for all climate region in the world. Constants a and b should be determined from laboratory or field study of comfort conditions from where the optimum T_c would be calculated. The ratings of T_c for 80% and 90% of acceptability was $\pm$ 3.5°C and $\pm$ 2.5°C on either side of the optimum T_c. These ratings were not derived from experimental work or survey. Instead, they came from a widely assumed relationship between the group mean thermal sensation votes (Fanger's PMV) and thermal dissatisfaction (Fanger's PPD) [67].

2.6.2 Application of Adaptive Comfort Model

Adaptive comfort model described above is used to find the optimum comfort temperature. The optimum temperature is defined as neutral temperature or an indoor temperature where the PMV value is 0. Using formula suggested by Humprey, for naturally ventilated (NV) buildings, De dear and brager [67,99] found that general linear equations for adaptive comfort standard can be written as:

$$T_c\ (^{o}C) = 0.31T_{rm} + 17.8 \quad (2.8)$$

$$\text{Upper 80\% acceptable limit } (^{o}C) = 0.31T_{rm} + 21.3 \quad (2.9)$$

$$\text{Upper 90\% acceptable limit } (^{o}C) = 0.31T_{rm} + 20.3 \quad (2.10)$$

$$\text{Lower 80\% acceptable limit } (^{o}C) = 0.31T_{rm} + 14.3 \quad (2.11)$$

$$\text{Lower 90\% acceptable limit } (^{o}C) = 0.31T_{rm} + 15.3 \quad (2.12)$$

People's clothing and adaptation behavioral already consider under this adaptive model, so no need to take humidity, air speed limits and people's clothing into consideration when the above standards are applied. The standard becomes guidance to indicate optimum and acceptable indoor comfort temperature range for different climate zones of the world (based on the mean monthly outdoor air temperature). These adaptive comfort temperatures were presented in Figure 2.17.

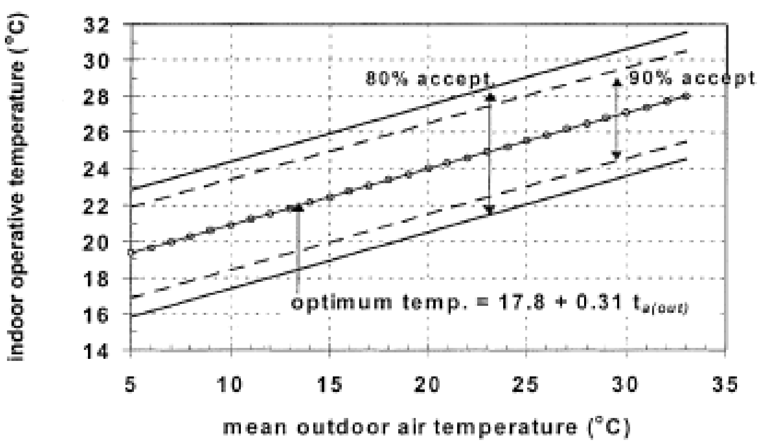


Figure 2.17 Adaptive comfort temperature vs mean outdoor air temperature [67]

Dahlan et. al. [100] conducted field measurement and survey on thermal comfort in Malaysia. It was set in 18 naturally ventilated rooms of two high-rise university hostels located in Universiti Malaya, Petaling Jaya (HH1) and Universiti Putra Malaysia, Serdang (HH2). The difference between these two buildings was described as:

- HH1 dimensions: 4.90 x 3.30 x 3.00 (m^3) with window-to-wall ratio (WWR)=0.35
- HH2 dimensions: 4.30 x 3.60 x 2.90 (m^3) with window-to-wall ratio (WWR)=0.26

The thermal sensation vote (TSV) from 208 students was collected and used to determine the neutral temperature. It was then compared with neutral temperature obtained from PMV regression [68] and optimum comfort temperature as describe above with 80% of acceptability [67].

Based on TSV, the neutral temperatures for HH1 and HH2 were 30.93^{o}C and 28.63^{o}C [100]. The results were close to the neutral temperatures obtained from equation 2.8: 29.87^{o}C in HH1 and 29.83^{o}C in HH2. However, these results were significantly higher than the neutral temperature obtained from PMV regression: 26.58^{o}C in HH1 and 25.48^{o}C in HH2. The difference between neutral temperature in HH1 and HH2 was due to the WWR. Higher WWR resulted in higher neutral temperature and vice versa. The comparison of the results obtained from TSV and Equation 2.8 implied that the equation was better suited for Malaysia rather than the PMV regression.

For air-conditioned or cooled building, Humphreys and Nicol [101] found that equivalent relationship between the mean monthly outdoor air temperatures (T_{rm}) with the comfort temperature (T_c) is:

$$T_c(°C) = 0.093T_{rm} + 22.6 \qquad (2.13)$$

They also determined comfort temperature model for NV building based on their field survey for European countries as follow:

$$T_c(°C) = 0.33T_{rm} + 18.8 \tag{2.14}$$

This model differs from the model obtained by De dear and Brager [67] which used research results in building comfort studies from all over the world.

Adaptive comfort studies showed that indoor comfort temperature was higher than the recommended value regarding the adaptive behavioral and outdoor conditions. Application of this adaptive comfort temperature (ACT) on the HVAC system would reduce the energy consumption mainly due to smaller temperature difference between indoor and outdoor temperature than if recommended indoor temperature used. In Australia, 40% - 45% energy can be conserved [19] and 7% of energy saving can be achieved from air-conditioned building in Hong Kong by using the ACT [20]. Another simulation study of application ACT on the VAV HVAC system showed energy saving potential up to 12.9% [102]. Even though there was offset on the indoor temperature set point (higher indoor temperature), the occupant would still feel comfortable because of their behavioral adaptation [27, 22].

Summary

Typical centralized HVAC system used in many building and the studied building have been explained. Heat gains which contribute in space cooling load were explained to understand how the heat gain affected the energy consumed in the cooling process. Indoor thermal standard from ASHRAE and Malaysian standard were also presented. Recent researches' in HVAC system strategy to reduce the energy consumed were addressed to know the position of the research. Adaptive comfort study which showed potential in energy saving of the HVAC system was explained to know how it would reduce the energy consumed and the potential to be applied in the real system in the future.