CHAPTER 3

METHODOLOGY

This chapter discusses theoretical background that is used in the calculation and in the methodology to achieve the objectives as stated in chapter 1. Block 16 of academic building in Universiti Teknologi PETRONAS (UTP) as a typical academic building was chosen for the investigation. The building is facing 20° to North of East and located at Eastern longitude 101° and Northern latitude 4.383^o with an altitude of 32m above sea level. It has three levels and consists of offices, classrooms, laboratories, and workshops. The building model and its HVAC system were first described and explained in detail. Before the model is used for further analysis, it is validated by comparing the simulation results with experimental data. The experimental data were collected from two experiments. First experiment was to measure indoor thermal (indoor temperature and relative humidity) and supply air conditions (supply air flow rate and supply air temperature). Second experiment was to measure outdoor conditions i.e. global solar radiation, ambient temperature and ambient relative humidity. Following that, proposed algorithm as strategy to reduce the energy consumption is explained in detail. Finally, economic analysis as a tool to determine feasibility of the proposed system is presented.

3.1 Space Load Characteristic

Sensible heat (triggered by temperature difference) and latent heat (triggered by humidity difference) are two important components of heat transfer in any space conditioning project. The heat source could be located either inside or outside of the conditioned space. The instantaneous heat transfers between the conditioned space and environment can be classified into two categories i.e. convective and radiative heat transfers, as shown in Figure 3.1. Most of the cases the heat is conducted through a solid material used as wall, window, roof, or insulator before it is convected or

radiated. In order to maintain the indoor temperature set point, the heat transferred into the conditioned space (space cooling load/heat gain) or out to the environment (space heating load/heat loss) should be handled instantaneously.

There are two main processes which are provided by HVAC system: cooling and heating process. Cooling process is used to reduce space temperature while heating process is used to increase space temperature. Each process could be coupled with dehumidification (reduce moisture content) or humidification (increase moisture content) processes depend on the desired indoor humidity level. As in Malaysia where the weather conditions are hot and humid, the HVAC system is dominated with cooling and dehumidification process and thus, space heating and humidification process would not discussed in detail.
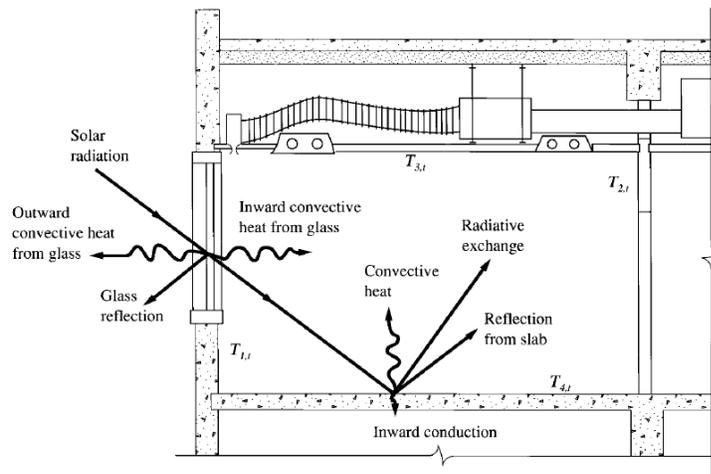

Figure 3.1 Convective and radiant heat in a conditioned space and the temperatures of the interior surfaces [31].

3.1.1 Cooling Load Calculation

Cooling load usually can be classified into two categories, i.e. external and internal. External cooling loads are the heat entering the conditioned space from external sources through the building envelope and the partition walls. The sources contributing in external cooling loads are conductive heat gain coming through the exterior walls and roofs, solar heat gain transmitted through the fenestrations, infiltration of outdoor air into the conditioned space, and heat gain entering from the partition walls and interior doors.

Internal cooling loads are the sensible and latent heat released from the heat sources inside the conditioned space. The sources contributing in internal cooling loads are heat released by the occupant, lights, equipment and appliances, etc. Infiltrated air, occupants, equipment, appliances, building structures and furnishings are cooling load components which have both sensible and latent cooling loads. The other remaining components have only sensible cooling loads. Indoor furniture would act as internal thermal mass that contribute to the cooling load. However, the additional cooling load is negligibly small (2%) [103]. Indoor furniture also has ability to store moisture that would affect the indoor relative humidity. If the storage effect of the moisture and heat are ignored, all heat gains released to the space air are instantaneous cooling loads [31]. Sensible and latent cooling load from internal and external sources are calculated using the formulae described in the section. Flow chart of cooling load calculation for the studied building is presented in Figure 3.2.
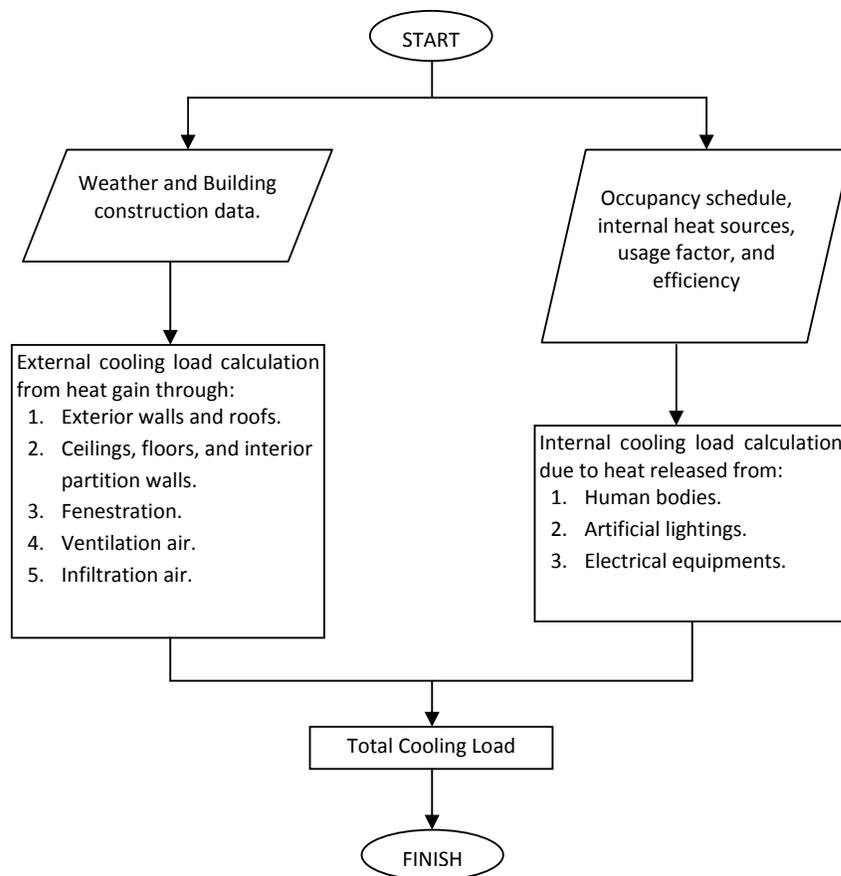

Figure 3.2 Flow chart of cooling load calculation for the studied building

3.1.1.1 Conduction Heat Gain Through Exterior Walls and Roofs

The sensible heat gains through an exterior wall or a roof are calculated by using heat balance method as recommended method by ASHRAE. Recommended outdoor surface combined heat transfer coefficient h_o = 4.4 Btu/hft^2°F (25 W/m^2°C), and an indoor surface combined heat-transfer coefficient h_i = 1.355 Btu/hft^2°F (7.692W/m^2°C) were used in the calculation [89]. This external heat gain at time t, denoted by $q_{e,t}$, Btu/h, is calculated as [31]:

$$q_{e,t} = U_{wall/roof} A_{wall/roof} (T_{o,t} - T_r)$$ (3.1)

3.1.1.2 Heat Gain Through Ceilings, Floors, and Interior Partition Walls

Heat can be transferred from adjacent room to the conditioned room due to the temperature difference. The sensible heat gain due to the heat transfer from the adjacent space through the ceiling, floor, or interior partition wall at time t, denoted by $q_{p,t}$ Btu/h, are calculated as [31]:

$$q_{p,t} = U_{c,f,p} A_{c,f,p} (T_{ad,t} - T_r)$$ (3.2)

3.1.1.3 Solar Heat Gain Through Fenestration

There are two component of heat flow through window glass: heat flow from the solar radiation absorbed by the window glass and the heat flow due to the outdoor and indoor temperature difference. These two components are actually combined, however it is simpler and acceptably accurate to separate this composite heat gain into solar heat gain and conductive heat gain which are both sensible heat gains. Solar heat gain through window glazing $q_{so,t}$ (Btu /h), is given as [31]:

$$q_{so,t} = A_{s,t}(SC)(SHGF_t) + A_{sh,t}(SC)(SGHF_{sh,t})$$ (3.3)

Usually, the SHGF$_t$ of the vertical surface facing north orientation is taken as SHGF$_{sh,t}$.

3.1.1.4 Conductive Heat Gain Through Fenestration

Conductive heat gain through fenestration due to the outdoor-indoor temperature difference $q_{win,t}$, Btu/h, is given as [31]:

$$q_{win,t} = U_{win} A_{win} \left(T_{o,t} - T_r \right) \tag{3.4}$$

3.1.1.5 Internal Heat Gains

The internal heat gain consists of the heat released by occupants, lights, equipments and appliances. Heat gain from artificial lights contributes on the space sensible cooling load only while heat gain from people contributes on both space sensible and latent cooling load. Other internal heat gain may contribute only on the space sensible or latent cooling load or both depending on the type of the equipments and appliances.

Human beings always release both sensible heat and latent heat to the conditioned space, even at rest. The amounts of these heats fluctuate according to the level of activity of the occupant. The space sensible heat gain for occupants staying in a conditioned space at time t, denoted by $q_{sp,t}$, Btu /h, are calculated as [31]:

$$q_{sp,t} = N_{p,t} \left(SHG_p \right) \tag{3.5}$$

Space latent heat gain for occupants staying in a conditioned space at time t, denoted by $q_{lp,t}$, Btu/h, is given as [31]:

$$q_{lp,t} = N_{p,t} \left(LHG_p \right) \tag{3.6}$$

ISO 7730 table lists both the sensible (SHG_p) and latent (LHG_p) heat gain from occupants based on degree level of activity of the occupant in conditioned space.

Every artificial lights installed inside the conditioned space released sensible heat to the space air with the rate equal to the sensible heat released from the electric lights, the emitting element, and light fixtures $q_{s,l}$. The sensible heat released to the conditioned space denoted by $q_{es,l}$, Btu/h depend mainly on the criteria of illumination and the type and efficiency of electric lights and can be calculated as [31]:

$$q_{s,l} = 3.413 W_{lamp} F_{usl} F_{al} = 3.413 W_A A_{fl} \tag{3.7}$$

All equipment and appliances (e.g., in the motor windings, combustion chamber, rubbing surfaces, even at components where mechanical work is performed) convert all of their energy inputs into heat energy [31]. A portion of heat released may be exhausted locally by a mechanical ventilation system such as fan and aluminum fin. In practical and application, the space sensible heat gain due to the machine load when a motor is placed inside the conditioned space, denoted by $q_{s,e}$, Btu/h, are calculated as [31]:

$$q_{s,e} = 2546 P_{hp} F_{load} F_{use} \frac{1 - F_{exh}}{\eta_{mo}} \tag{3.8}$$

3.1.1.6 Heat Gains from Ventilation/Infiltration

Ventilation air taken from the environment is needed in a conditioned space and the amount varies with the numbers of occupant, internal odor sources, and outdoor air quality. Outdoor air may also infiltrate to the conditioned space through cracks and openings in the building envelope due to the pressure difference across the envelope caused by any of the following:

1. Wind pressure.

2. Stack effect due to the outdoor and indoor temperature difference.

3. Mechanical ventilation.

The infiltrated and ventilation air with the volume flow rate $V_{inf/vent}$, CFM, contribute to the space sensible heat gain $q_{inf/vent}$, Btu/h and is represented as [31]:

$$q_{s,inf/vent} = 60 V_{inf/vent} \rho_o c_{pa} (T_o - T_r) \tag{3.9}$$

The space latent heat gain component from infiltration $q_{l,inf/vent}$, Btu/h, is calculated as [31]:

$$q_{l,inf/vent} = 60 V_{inf/vent} \rho_o (\omega_o - \omega_r) h_{fg,32} \tag{3.10}$$

3.1.2 Parameters Controlling the Secondary System

Total space heat gains, local indoor and outdoor design conditions are used to find room sensible/latent heat (*RSH*/*RLH*), outside air sensible/latent heat (*OASH*/*OALH*),

and grand sensible/latent heat (*GSH*/*GLH*). *RSH*/*RLH* is amount of heat that should be removed from conditioned space to keep the indoor thermal set points. *OASH*/*OALH* is amount of heat gains from ventilation air. *GSH*/*GLH* is total heat gain that should be removed (total cooling load) that is calculated by summing *RSH*/*RLH* and *OASH*/*OALH*. The summation of *GSH* and *GLH* is total cooling capacity (*GTH*) that has to be provided by the cooling equipment (cooling coil). Room sensible heat factor (*RSHF*), outside air sensible heat factor (*OASHF*) and grand sensible heat factor (*GSHF*) are then determined, as defined below [56]:

$$RSHF = \frac{RSH}{RTH} \tag{3.11}$$

$$OASHF = \frac{OASH}{OATH} \tag{3.12}$$

$$GSHF = \frac{GSH}{GTH} \tag{3.13}$$

RSH and *RLH* are calculated according to internal heat sources: people, infiltration air, artificial lighting, electronic devices, machine, and computer while *OASH* and *OALH* are determined from the ventilation air required. In some cases, however, there are some portions of the air stream flowing across the cooling coil which have less or no contact at all (by-passed air) with the cooling coil and thus, this portion of air is not treated as it is so that it become additional *RSH* or *RLH*. This phenomenon is reflected by a factor called bypass factor ranging from 0 (100% contact) to 1 (100% of the air stream bypass the cooling coil). In this research, it was assumed that 100% of the air stream is in contact with the cooling coil and therefore, the bypass factor is equal to 1. With this assumption, only *RSHF* is used to determine supply air conditions needed by using psychrometric chart while the parameters *OASHF* and *GSHF* are used to size the cooling coil. The supply air flow rate needed was then calculated according to the *RSH*, *RLH*, and the supply air conditions using formulae as in the following [56]:

$$CFM = \frac{RSH}{1.1(\Delta T_{ss})} \tag{3.14}$$

$$CFM = \frac{RLH}{0.68(\Delta \omega_{ss})} \tag{3.15}$$

It can be seen that supply air conditions depend on *RSHF* in order to handle the space sensible and latent cooling load proportionally. Nevertheless, the *RSHF* keep changing during HVAC operation and hardly to be predicted due to dynamic changing on occupancy pattern, occupancy density, and outdoor conditions [7]. Resultantly, *RSHF* should properly be selected based on the cooling load characteristics so that the system could keep the indoor temperature and relative humidity within acceptable range.

If *RSHF* design 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 may lead to a condition where the airflow needed was below the minimum airflow of the system. Even though the blower can be modulated to supply low flow rate, the whole system (AHU), however, would be unable to deliver the airflow because of mechanical system limitation. In this case, the minimum airflow would be higher than it was required and thus, overcool the space [11, 28].

3.1.3 Space Heating Load

Space heating load is the rate of heat that must be added to the conditioned space so as to maintain indoor air temperature and relative humidity set point. Hot water, steam, or electric heating elements are usually used to add the required heat to the space (heating coil load).

In the literature, "The heating load is usually the peak heating load of a space, room, zone, floor, or building in a load profile. In heating load calculation, block load is not needed because the solar radiation and internal loads are not taken into account and thus, heating load depends entirely on indoor-outdoor temperature difference" [31].

In many western countries, space heating is required during winter and at night during summer since the temperature could be as low as 10^oC or less. The process is usually coupled with humidification process to humidify the typical dry air. In eastern countries near the equator, heating process in HVAC system is typically used to heat up dehumidified air leaving the cooling coil to maintain constant supply air temperature.

3.1.4 Heat Balance Method

Heat balance method is the exact method to calculate the space cooling load to simultaneously determine the temperature of the interior surfaces of the building structure at any time, *t*. The temperature is then used to calculate the space sensible cooling load, which is equal to the sum of the convective heat transfer from these surfaces, latent cooling loads, and the cooling load due to infiltration at time, *t*.

The following assumptions are taken to calculate the heat transfer between various internal surface:

- Heat flow through the building envelope is considered as one-dimensional transient heat flow only.

- The room air temperature is uniform.

- The materials of the building envelope are homogeneous.

- The surface temperature, the surface heat-transfer coefficient, and the absorption for each surface are uniform values. Sum of absorption, reflection and emittance is always equal to one.

- The cooling load is calculated based on the mean value between the time intervals.

In an air conditioned room, the internal surfaces may receive conduction heat at time *t*, denoted by $q_{i,t}$, through common walls of the adjacent buildings. The wall surfaces receive shortwave solar radiation through fenestration and long-wave radiation from other interior surfaces. Additional radiations are received from the lighting fixtures, appliances, equipment, and occupants, etc. Convective heat transfer also exists between these interior surfaces and space air. If the number of surfaces in the room are 1 to m across with the heat is transferred, then the sensible heat balance at the internal i_{th} surface (for $i = 1, 2, \ldots, m$) at time *t* are described as follows [31]:

$$q_{i,t} = \left[h_{c,i}\left(T_{r,t} - T_{i,t}\right) + \sum_{j=1}^{k} h_{i,j}\left(T_{j,t} - T_{i,t}\right) \right] A_i + S_{ir,t} + L_{ir,t} + E_{ir,t} + O_{ir,t} \qquad (3.16)$$

The conductive and radiative heat gain in Equation 3.1 are all continuous functions of time $f(t)$ and thus, the transient conductive heat transfer $q_{i,t}$ can be solved by using finite difference method or can be solved by means of the conduction transfer function.

In most of cooling load calculation, the room air temperature $T_{r,t}$ is considered as constant, and the conduction transfer function equation can be solved simultaneously to determine the interior surface temperatures $T_{i,t}$. Then sensible cooling load of the space air denoted by $Q_{rs,t}$, Btu /h, at time t, are calculated as [31]:

$$Q_{rs,t} = \sum_{i=1}^{m} h_{ci}\left(T_{i,t} - T_{r,t}\right)A_i + 60\dot{V}_{inf/vent,t}\rho_o c_{pa}\left(T_{o,t} - T_{r,t}\right) + S_{c,t} + L_{c,t} + E_{c,t} + SHG_p \quad (3.17)$$

Latent heat gains in the space air are converted to latent cooling load instantaneously. This amount of space latent cooling load at time t, denoted by $Q_{rl,t}$, Btu/h, are calculated using equation as below [31]:

$$Q_{rl,t} = q_{il,t} + 60\dot{V}_{if}\,\rho\left(\omega_{o,t} - \omega_{r,t}\right)h_{fg} + E_{l,t} + LHG_p \quad (3.18)$$

Then the total space cooling load at time t, denoted by $Q_{rc,t}$, Btu/h, is

$$Q_{rc,t} = Q_{rs,t} + Q_{lr,t} \quad (3.19)$$

In a space or room, the conditioned air is supplied from the ceiling diffuser at a mass flow rate at time t of $m_{s,t} = V_{s,t}\rho_s$, where, $V_{s,t}$ is the volume flow rate of supply air at time t, cfm (m^3/min), and ρ_s is the density of supply air, lb/ft^3 (kg/m^3). The supply air is then mixed with the space air and it absorbs convective sensible heat and latent heat from various internal surfaces. Due to leakage on the building envelope, the outdoor air infiltrates into the space at a volume flow rate $V_{inf/vent,t}$, cfm (m^3/min) at time t. The sensible and latent heat balance of the space air are described as [31]:

$$60\dot{V}_{s,t}\,\rho_s c_{pa}\left(T_{r,t} - T_{s,t}\right) = \sum_{i=1}^{m} h_{ci}A_i\left(T_{i,t} - T_{r,t}\right) + 60V_{inf/vent,t}\rho c_{pa}\left(T_{o,t} - T_{r,t}\right) + S_{c,t} + L_{c,t} + E_{c,t} + O_{c,t} \quad (3.20)$$

$$60V_{s,t}\rho_s\left(\omega_{r,t} - \omega_{s,t}\right)h_{fg} = q_{il,t} + 60\dot{V}_{inf/vent,t}\,\rho\left(\omega_{o,t} - \omega_{r,t}\right)h_{fg} + E_{l,t} + O_{l,t} \quad (3.21)$$

The heat balance method is more direct and clear in load calculation methodology. In the moisture transfer calculation for a building located in hot and humid climate, the heat balance method will give comparatively more accurate results. However, this method demands laborious work, more computing time, complicated computer programs, and experienced users. Therefore, the heat balance method is usually used for research and analytical purposes. However, transient system simulation software, TRNSYS that is used in this study calculate the entire space cooling load from the external and internal heat gains using transfer function method as simplification of the laborious heat balance method.

3.1.5 Transfer Function Method

The transfer function method or weighting factor method is a simplification of the laborious heat balance method. The wide application of the TFM is due to the user-friendliness of the inputs and outputs of the TFM software and the saving of computing time.

Space cooling load calculation procedure using the transfer function method consists of two steps. First, heat gains or heat loss from exterior walls, roofs, and floors are calculated using response factors or conduction transfer function coefficients. Following that, the solar and internal heat gains are calculated directly for the scheduled hour. Second, the heat gains are converted to cooling loads and the heat losses are converted to heating loads using room transfer function coefficients or room weighting factors. Weighting factors are transfer function coefficients to represent the importance of current and historical values of heat gain and cooling load on currently calculated cooling loads. These factors are presented in the form of z transform functions.

In TFM, The sensible heat gain through external wall or roof shown in Eq. 3.01 is calculated by the conduction transfer function [31]. The method use sol-air temperature at time, t ($T_{sol,t}$) to represent the combined temperature excitation of the outdoor temperature and solar heat on the exterior surface of an exterior wall or roof, and a constant indoor temperature T_r. Outdoor and indoor combined convective and radiative heat transfer coefficients (h_o and h_i) are used to calculate the conduction

transfer function coefficients. In TRNSYS, the h_o and h_i used are 4.403 Btu/ft²hf and 1.356 Btu/ft²hf [96].

Then, the external heat gain through an exterior wall or roof at time, t ($q_{e,t}$) is determine as shown below [31]:

$$q_{e,t} = \left(\sum_{n=0} b_n T_{sol,t-n\Delta} - \sum_{n=1} d_n \frac{q_{e,t-n\Delta}}{A} - T_r \sum_{n=0} c_n \right) A \qquad (3.22)$$

3.2 Building Simulation

3.2.1 Building Modeling

The main aim of simulation is to develop a model that behaves like the real one in terms of heat movement across its boundaries. Building construction details were used carefully to develop the virtual model of the building. Indoor environmental parameters were modeled using the real monitored data. Occupancy pattern in each room in the building were collected from the academic schedule and modeled in the simulation. The centralized HVAC system in the building was modeled according to its design and was integrated within the building model. Building construction details, occupancy pattern and centralized HVAC system of the building were described in the following sections.

3.2.1.1 Building Construction Detail

The building has three levels and consists of offices (O), classrooms (C), laboratories (L), store room (S) and workshops (W) except for the third level which is office rooms. The building is facing 19.90° to North of East and located at Eastern longitude 100.96° and Northern latitude 4.38° with an altitude of 32m above sea level. Simplified floor layouts of each level of the building can be seen in Figure 3.3 – 3.6 while the details are presented in Appendix B – E. The building has 11 air-conditioned rooms and 2 non air-conditioned rooms on ground floor. On first and second floor, it has 9 air-conditioned rooms and 2 non air-conditioned rooms. On third floor, it has two air-conditioned rooms and non air-conditioned rooms. Room number 16-00-01/02, 16-00-01/02, 16-00-03/04, 16-00-12, 16-01-01/02, 16-01-07/08, 16-01-09, 16-02-04, 16-02-09, and 16-02-10/11 were found to be practically empty.
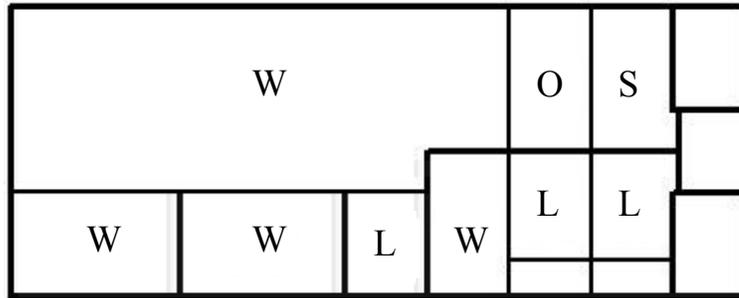
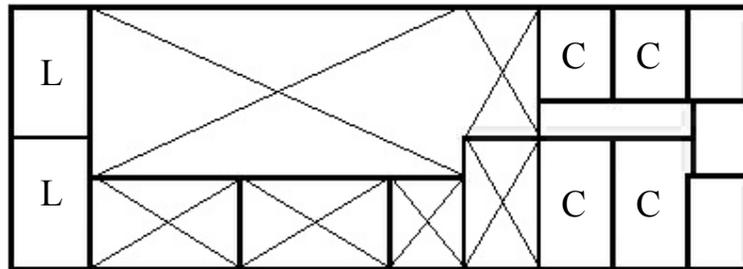Figure 3.3 Ground floor lay out of the building (simplified) [104]

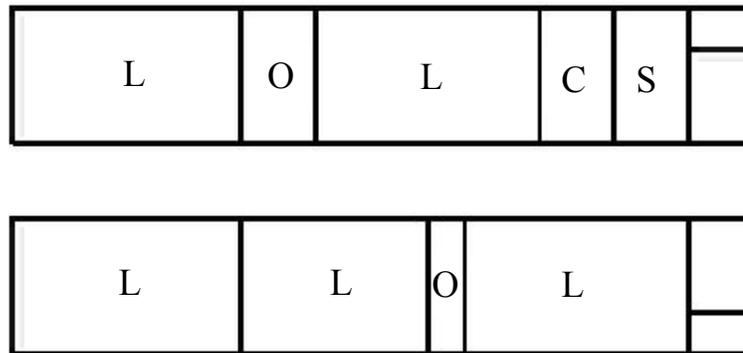Figure 3.4 First floor lay out of the building (simplified) [104]

Figure 3.5 Second floor lay out of the building (simplified) [104]
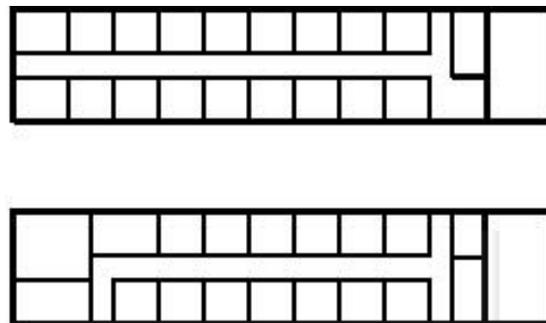

Figure 3.6 Third floor lay out of the building (simplified) [104]

Electrical/electronic devices are considered in the cooling load calculation using the equation described in previous section. List of electrical/electronic devices and artificial lighting are described in Appendix F – I.

External walls were modeled based on the wall construction detail. The exterior walls are dominated by 12 mm thick glass panels (±90%) with steel reinforcement. The portion of steel frame area to the glass panel area is 0.07. The steel reinforcements were modeled as the external walls while the glass panels were modeled as non-operable windows on the external walls. Transmittance of the construction glass has been reduced to a level where it is translucent. Hence, the contribution of such a glass panel in increasing the cooling load of the building would be decreased to a satisfied level. Detailed of external wall, partition wall, and roof construction is shown in Figures 3.7, 3.8, and 3.9.
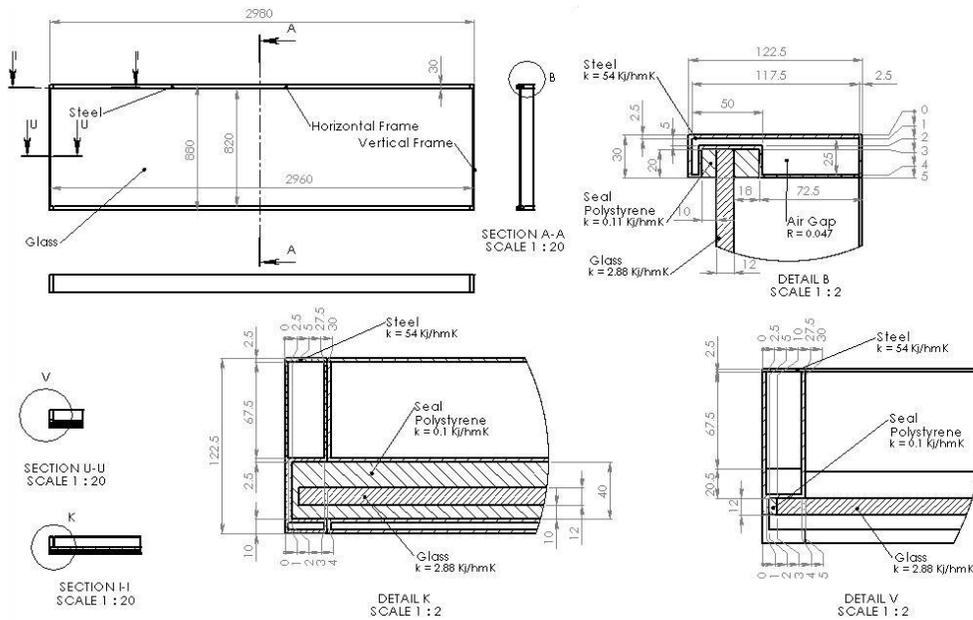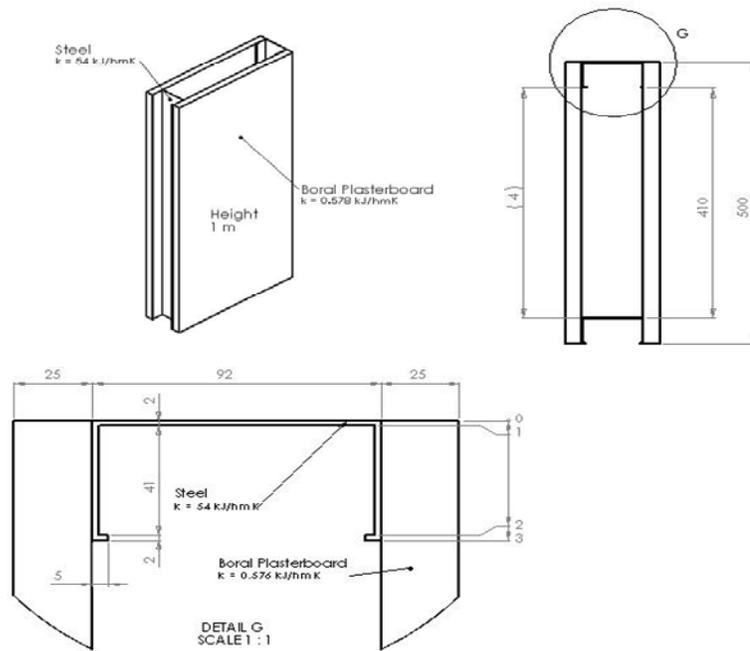
Figure 3.7 External glass wall constructions [104]

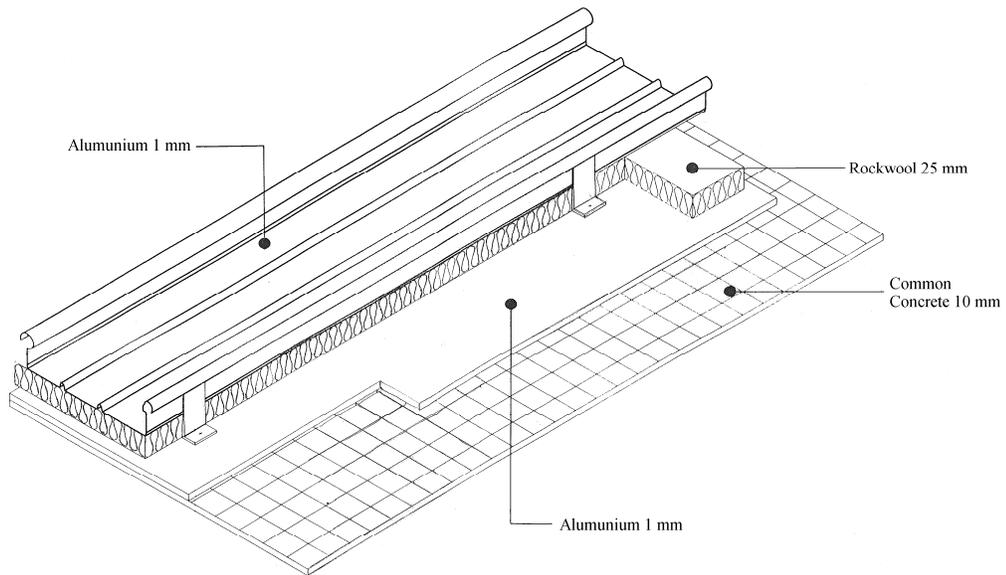Figure 3.8 Partition wall constructions [104]

Figure 3.9 Roof constructions [104]

Common concrete and concrete slab layers were used to construct the floor layers. This layer construction was based on assumption of typical floor layers. Double boral plasterboard with steel reinforcement is used for partition wall. Rock wool, aluminum

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and common concrete slab are used for roof layers. Building material specifications of the external wall, partition wall, floor, window and roof with the conductivity is presented in Table 3.1.

Table 3.1 Building Material Specifications [104]

Building Construction	Details (thickness-thermal conductivity)
External Wall	Steel (5mm-54 kJ/hmK), Air gap (0.047 hm^{2}K/kJ), Steel (11mm-54kJ/hmK)
Partition Wall	Plasterboard (25mm-0.576 kJ/hmK), Air Gap (92mm-0.047hm^{2}K/kJ), Plasterboard (25mm-0.576 kJ/hmK)
Flooring	Concrete slab (100mm-4.07 kJ/hmK), Common concrete (550mm-7.56 kJ/hmK)
Window	Optiwhite glass (12mm-3.24 kJ/hmK)
Roofing	Alumunium (1mm-846 kJ/hmK), Rockwool (25mm-0.162kJ/hmK), Alumunium foil (1-846 kJ/hmK), Common concrete (10mm-7.56kJ/hmK)

Manual calculation have been done to calculate U-values of conductivity and overall U-values for each constructions and these results (as presented in Table 3.2) were compared with the values obtained from TRNSYS. From the comparison, it can be seen that the value were match to each other.

Table 3.2 Comparison conductivity and U-values of the building construction

Type	U-values (conduction) (W/m^{2}K)		U-value (Overall) (W/m^{2}K)	
	Calculated	TRNSYS	Calculated	TRNSYS
Floor	1.872	1.871	1.42	1.42
Roof	1.785	1.786	1.37	1.369
Partition Wall	2.076	2.076	1.534	1.534
Window	75	75	5.458	5.458
External Wall	5.873	5.873	2.939	2.939

3.2.1.2 Occupancy Pattern

There are two occupancy patterns are used inside the building, i.e. student's occupancy schedule and staff occupancy's schedule. Student's occupancy schedule was modeled in detailed according to academic schedule which was available online in the university's website. In one semester, three different student's activities are observed that are study week, mid-semester or semester break and examination week.

During mid-semester or semester break, examination and national holidays, there are no activities in the classroom, workshop and laboratory. A sample of weekly schedule for the student is presented in Table 3.3.

Staff's occupancy schedule was modeled according to working schedule in the university. The working schedule has five working days from Monday to Friday and working hours from 8 a.m. to 5 p.m. with lunch time at 1 p.m. to 2 p.m. National holidays were taken into account for both occupancy patterns which was based on Malaysian calendar 2009. During weekends (Saturday and Sunday), national, and state holidays, no activity was recorded in the offices.

Based on observation of the occupant in the building, it was found that light working ensemble, typical business suit and light outdoor sportwear are the typical clothes of the occupants. It means that the clothing factor is in range of $0.6 - 1$ [96].

Table 3.3 Student academic schedule [105]

Month / Week	Jan	Feb	Mar	Apr	May	Jun
I	B	S	S	S	E	B
II	B	S	S	S	E	B
III	S	S	B	S	E	B
IV	S	S	S	S	E	B
V	-	-	S	S	-	B
Month / Week	Jul	Aug	Sep	Oct	Nov	Des
I	B	S	S	S	E	B
II	B	S	S	S	E	B
III	S	S	S	S	E	B
IV	S	S	B	S	E	B
V	-	-	S	S	-	B

B = Break period   S = Study period   E = Examination

3.2.1.3 Centralized HVAC System

Centralized HVAC systems with Gas District Cooling (GDC) were used to handle the cooling load from all UTP's academic blocks. The GDC produces chilled water and it is supplied (CHWS) to all air handling unit (AHU) on each block. The chilled water returned (CHWR) back to the GDC after absorbing heat from the supply air. There are 8 AHU on each block to handle cooling load and supply ventilation air in each zone. In each zone, there are motorized variable air volume (VAV) damper to control supply air flow rate enter the room and temperature sensor to monitor indoor temperature [104]. The supply air (SA) needed is treated in the AHU to have the supply air temperature set point before it is distributed to the zone through air ducting. Approximately 2/3 portion of return air (RA) from the zone is re-circulated back and the rest is exhausted to the environment. For chemical laboratory buildings (i.e. bloc 4 and bloc 5) which require 100% fresh air with no re-circulated air, energy recovery wheel are installed in front of the heat exchanger to pre-cool the fresh air using the exhaust air.

GDC and AHU's were modeled as the building component having the ability to deal with both sensible and latent cooling loads. These cooling equipments maintain the indoor temperature and relative humidity (RH) set points. The AC system was scheduled to operate from 07:00 hours to 19:00 hours from Monday to Saturday. Schematic diagram of the HVAC system and screen sort of the AHU was presented in Figures 3.10 and 3.11, respectively.
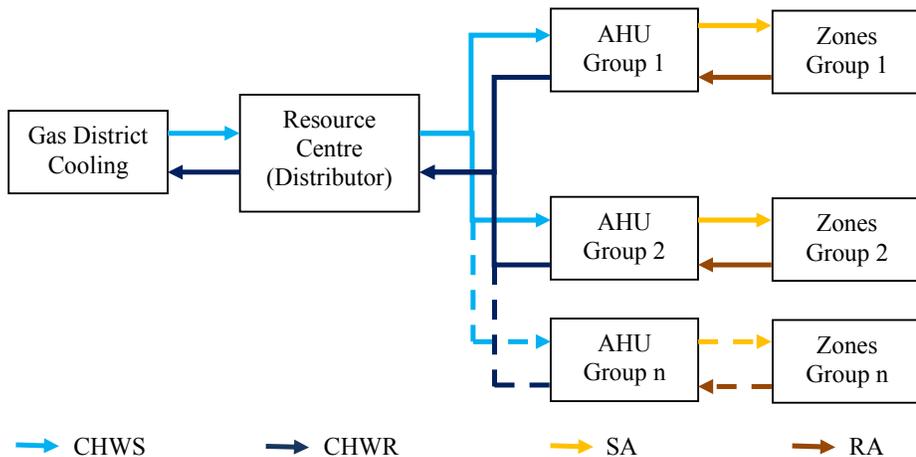

Figure 3.10 Schematic diagram of the HVAC system
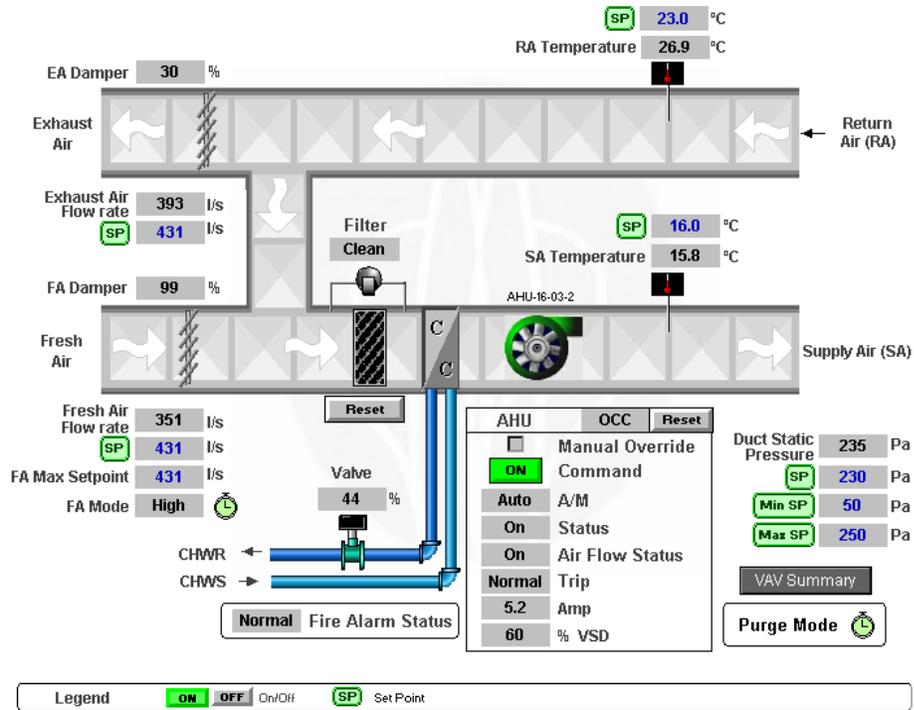
Each zone is well sealed with no mechanical window and fresh air was supplied by the air handling unit (AHU) only. Ventilation air flow rate was modeled based on ASHRAE Standard 62.1-2004 [106].   Two components of ventilation air were considered in the model; building component and occupancy component. Building component is to deal with odor released in the room while occupant component is to provide fresh air to the occupant. Detailed ventilation air rate components for the three types of rooms i.e. office, workshop and classroom were presented in Table 3.4.

Table 3.4 Ventilation rates components for the office, classroom and workshop

Ventilation air components	Room types		
	Office	Classroom	Workshop
Building component (Ra) (L/sm^2)	0.3	0.6	0.9
Occupancy component (Rp) (L/s)	2.5	3.5	5

3.2.2 Transient Simulation of The Centralized HVAC System

TRNSYS was used for the transient simulation of the building. It is in modular form and all the components have certain function to perform using certain parameters and inputs. These components were called in the main program as many times as it is

required. The components used in the simulation of academic building are summarized in Table 3.5

Table 3.5 Summary of components used in the simulation

No	Components	Description	Number of Component
1	Time dependent forcing function	This component is used to make daily schedule	45
2	Load profile sequencer – unique days for the week and holidays	The component is used to make weekly, monthly and yearly schedule.	4
3	Data reader and radiation processor	The component is used to read and generate weather data file.	1
4	Psychrometrics	The component is used to determine air properties based on psychrometric chart.	1
5	Effective sky temperature	The component is used to calculate fictive sky temperature.	1
6	Multi-zone building	The component is used to read and generate the building model (.bui file).	1
7	Calculator	The component is used to do calculation at outside the component.	1
8	Excel coupling	The component is used to call excel program.	6
9	Online plotter	The component is used to show the simulation results on the screen.	1
10	Simulation summary	The component is used to collect and save simulation results on an external file (.txt file).	9
11	Time values	The component is used to determine particular period of time during the simulation time.	1

Time dependent forcing functions were used to model daily occupancy pattern in each room and artificial lighting power. Load profile sequencers were then used to make a year occupancy schedule for each room based on daily occupancy pattern and Malaysian calendar 2009 which had 17 national holidays. The schedule would also be used to determine internal cooling load pattern from electric/electrical devices and artificial lighting in the building model. Outputs from these components were then input to the multi-zone building component.

Data reader and radiation processor was used to generate weather data file specified in the component. Perez model was used to calculate diffuse radiation on the walls. Tracking mode of the walls was set to 1 which indicated that the wall was fixed wall. Numbers of walls including the slope and azimuth angle were described in this component. Detail information about this component would be discussed in the next section (ambient condition model).

Psychometrics chart was used to calculate ambient dew point temperature from ambient temperature and relative humidity data provided by the data reader and radiation processor. Output from this component would be inputted to effective sky temperature for long-wave radiation exchange component.

Effective sky temperature for long-wave radiation exchange component was used to calculate fictive sky temperature. Beside dew point temperature as mention above, the component needs ambient temperature, beam radiation and diffuse radiation on horizontal surface for the calculation. These data would be provided by the data reader and radiation processor as input to this component. Fictive sky temperature as the output would be use for long-wave radiation heat transfer calculation in the building model.

Multi-zone building was used to generate the building model (.bui file) which had been created in TRNBuild and specified in the component. The inputs required would be based on the building model (.bui file) and additional input from other components used in the simulation studio. The outputs would be based on the outputs specified in TRNBuild. In this model, the outputs were indoor temperature, indoor relative humidity, cooling load (sensible and latent load) and additional output which would be used in the evaluation.

Some of standard procedure calculations are excluded in the component. To perform such calculation, a small component called calculator was used in the simulation. Inside the calculator, the inputs should be first described before it was used in the calculation. The outputs from this calculator are then fed to the components.

A component named 'excel coupling' was used to call Microsoft excel® program. In excel program, ventilation air needed in each zone was calculated. The calculation used inputs from other components and this component recommended coefficients as shown in Table 3.4.

Online plotter component was used to show the graphs of calculation results done by TRNSYS on the monitor screen. Number and names of parameters showed on the graph were first described in the component. The minimum and maximum values of left and right axis on the graph were also set on this component. Another component named 'simulation summary' was used to saved the results on an external file (.txt file). Simulation summary start time, and simulation summary stop time were set within the simulation start time, and simulation stop time range. Simulation summary time step was set according to the simulation time step. Print out of simulation model of the academic building in TRNSYS is presented in Figure 3.12.
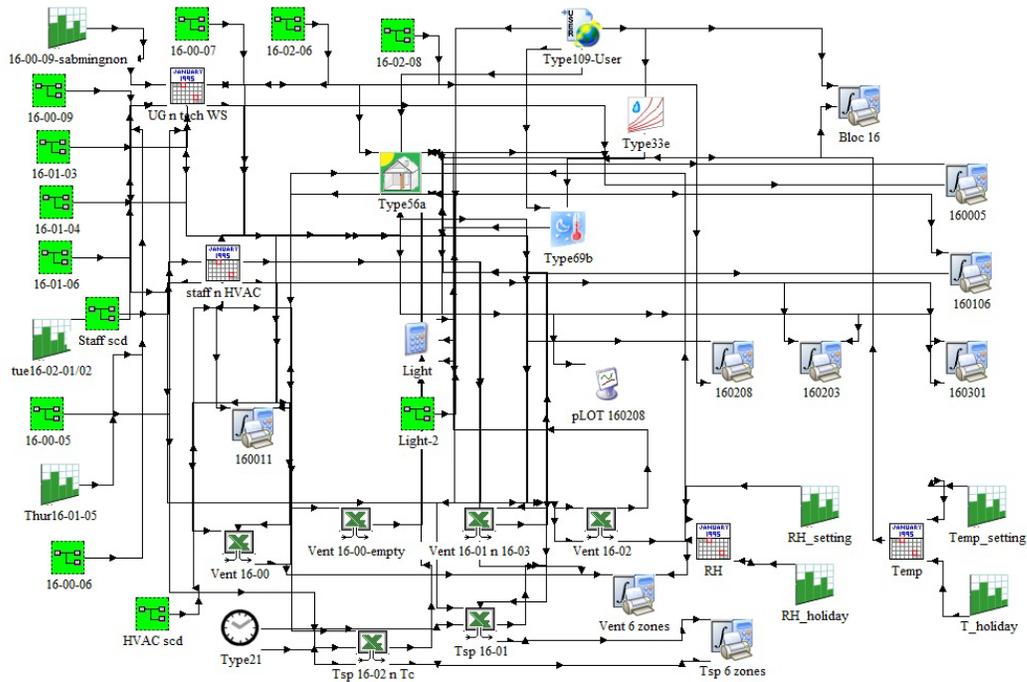

Figure 3.12 Print out of simulation model of the academic building

Thermal comfort zone with indoor temperature between 23.1°C to 25.6°C and 70% of relative humidity were considered for Malaysian region [18,72,77]. Electrical/electronic devices and artificial lighting inside the zone were assumed to be

operated during occupancy period only. Convective and irradiative fraction of heat gain from the electrical/electronic devices were assumed to be 0.7 and 0.3 while for artificial lighting, the values were 0.6 and 0.4 [107]. Infiltration air was assumed to be 0.05 and 0.15 of air change/h during HVAC operation and non-HVAC operation, respectively [31]. The value was smaller during HVAC operation because all zones were pressurized with positive pressure at that period. Activities without specific schedule were not considered in the simulation.

Activity level in classroom and office was set at level 4 with sensible and latent heat gain to be 269 kJ/hr, respectively. Activity level in workshop where practical work existed was set at level 5 with space sensible and latent heat gain of 332.33 kJ/hr and 342.88 kJ/hr, respectively. These values are based on ISO 7730 table provided by TRNBuild. The simulation has been done for a year with one hour time step.

3.2.3 Buffer Storage Humidity Model

Vapors are absorbed by the supply air or being stored in vapor mass storage (furniture, clothes, etc.) inside the room. This stored vapor would be only released to the room air if the humidity level is lower than the stored vapors and the relative humidity is below 100%. The released vapor from the vapors mass storage would increase the latent load in air-conditioned room and would increase the relative humidity of the room air when the room is not air-conditioned [108]. In this research, the buffer humidity storage effect of indoor furnishing on indoor air-conditioned space air is described by dividing it into surface and deep humidity storage (buffer storage humidity model).

There are three parameters used in this model i.e. κ, m and β [96]. The first parameter (κ) is the gradient of the sorptive isothermal line of the material which is a representation of the water storage capacity of the material. The second parameter (m) is the mass of the material while the third parameter (β) is a representation of the moisture transport from the storage to the room node. This method separates vapor storage into two i.e. surface storage and deep storage. In most of the cases, the influence of walls on the air humidity can be well described if β_{surf}/A is 3 kg_{air}/m^2h and β_{deep}/A is 1 kg_{air}/m^2h [96]. Therefore, this assumption is taking into account in the

buffer storage humidity model of the academic building. Buffer storage humidity model is presented in Figure 3.13.
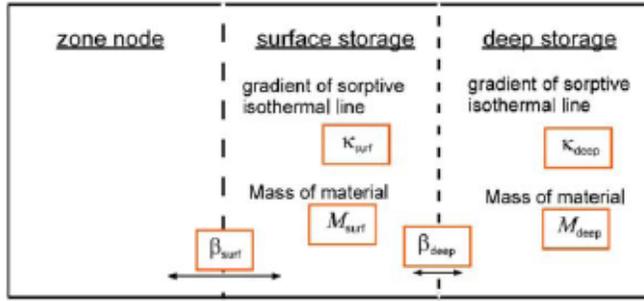
Figure 3.13 Buffer storage humidity model [96]

The differential equation of mass balance for the calculation of the room humidity by a term taking into account the exchange of moisture between the room node and the surface storage is described as follows [96],

$$M_{air,i}\frac{d\omega_i}{dt} = m_{\inf,i}(\omega_a - \omega_i) + \sum_k^{nvent} m_{v,k,i}(\omega_{v,k,i} - \omega_i) + W_{g,i} + \sum_{i-j} m_{cp\lg,s}(\omega_j - \omega_i) +$$
$$\beta_{surf}(\omega_{surf} - \omega_i) \tag{3.23}$$

Mass balance to describe the dynamics of the water contents of the surface and the deep storage are described as below:

$$M_{surf}\kappa_{surf}f(\varphi,\omega)\frac{d\omega_{surf}}{dt} = \beta_{surf}(\omega_i - \omega_{surf}) + \beta_{deep}(\omega_{deep} - \omega_{surf}) \tag{3.24}$$

$$M_{deep}\kappa_{deep}f(\varphi,\omega) = \beta_{deep}(\omega_{surf} - \omega_{deep}) \tag{3.25}$$

3.2.4 Malaysia Climate

The climate in Malaysia is hot and humid with average temperature 23.7ºC to 31.3ºC and average relative humidity 67% to 95% throughout a year [109]. The data was provided by Malaysian meteorological station from ten years data collection. There is small annual variation on the temperature which is less than 2ºC for most areas of Peninsular Malaysia and less than 3ºC for east coast areas of Peninsular Malaysia [108]. April and May are recorded to be the months with highest average monthly temperature in most places while December and January are the coldest months with lowest average monthly temperature. In Peninsular Malaysia, relative

humidity is normally minimum in the months of January and February. However, for the east coast states of Kelantan and Terengganu minimum relative humidity is recorded in March. Maximum relative humidity is generally found in the month of November [108].

Rainfall pattern showed that there are two period of maximum rainfall in Peninsular Malaysia. First period generally occurs in October - November while second period generally occurs in April - May. First period of minimum rainfall occurs in June - July while the second period occurs in February [108].

The one full year weather data was collected for the simulation by the Ipoh metrological data station which is located approximately 50 km from the building. The data was assumed as typical weather conditions in Ipoh and used to find cooling load characteristic of the building.

There are 5 parameters and 3 inputs which are used to simulate the ambient conditions. The parameters are global solar radiation, ambient temperature, ambient relative humidity, wind speed, and wind direction. The inputs are ground reflectance, slope and azimuth angle of the external wall. The data collected was saved in .txt file and was generated using weather data component (user format) in the TRNSYS simulation studio. The component calculates total solar radiation, beam radiation, diffuse radiation, and angle of incident of solar radiation on the external wall based on the slope and azimuth angle. These calculated values were then input to the building model for the simulation.

3.3 Environmental Data Monitoring

Indoor and outdoor environments were monitored for the validation of numerical results. The experimental measurements were conducted for 2 weeks. Space air temperature, indoor relative humidity, supply air flow rate and supply air temperature were measured during indoor measurement. Ambient temperature, relative humidity and global solar radiation were measured during outdoor measurements. Both indoor and outdoor measurements data were collected every 6 minutes and saved in Excel file.

Office type room with 6.75 m of width, 11.8 m of length and 4 m of height located on ground floor facing south was chosen for indoor experimental measurements. This room was selected because the occupancy pattern and the indoor activity were less dynamic than other rooms. The outdoor experimental measurement was located on an open area near the building to accurately measure the environmental conditions. The outdoor measurement was elaborated with Firmanda [110].

Kipp and Zonen pyranometer SPLite series is used to measure global solar radiation. The sensor has an ability to measure global solar radiation up to 200 W/m^2 with operating temperature from -30°C to 70°C. The sensors has directional error (up to 80° with 1000 W/m² beam) less than 5 W/m^2 [112]. LM 35 precision centigrade temperature sensors were used to measure the temperature with accuracy $\pm0.75^o$C over a full temperature range (-55°C – 150°C) temperature range and linear +10mV/^{o}C scale factor. This sensor needs supply voltage -0.2 VDC to +35 VDC [113]. HIH-5030 low voltage humidity sensors were used to measure the RH with accuracy $\pm$3%. The temperature and relative humidity sensors were shown in Figure 3.14. These sensors need supply voltage +2.7 VDC to +5.5 VDC. Voltage measured by the RH sensors are converted into RH value using the following equation [113],

$$V_{out} = (V_{sup\,ply})(0.00636\,(sensorRH\,) + 0.1515) \tag{3.26}$$

$$TrueRH = (sensorRH\,)/(1.0546 - 0.00216\,T) \tag{3.27}$$
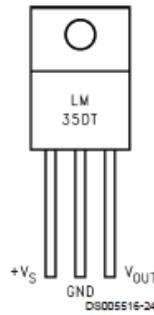
(a)   (b)

Figure 3.14 a.) HIH 5030 humidity sensor [113], b.) LM 35DT temperature sensor [112]

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AC (alternating current)/DC (direct current) adaptor was used to supply DC voltage from AC voltage source. DC voltage divider circuit was placed on each sensor to reduce voltage from 6 VDC to 5 VDC. During indoor measurements, the sensors were placed on four different locations with different height (0.5 m, 1 m, 1.5 m and 2m). Indoor temperature and RH have been calculated as the average value of the four sensors at different height. All of the sensors were connected to picolog data logger using cooper cable with 1 mm^2 of area. Voltage drop due to internal resistant from the cable can be calculated using the equation below [114]:

$$V_{drop} = IR_{int\,ernal} \qquad (3.28)$$

$$R_{int\,ernal} = \rho \frac{L}{A} \qquad (3.29)$$

Using above equation, measured internal resistant for the longest cable (11m) was only 0.185 ohm. Maximum electric current for both temperature and humidity sensors was only 560 µA and thus, the voltage drop was only 10^{-4} V or 0.1 mV. Hence, the effect of the voltage drop on input voltage from the power supply to the sensors and output voltage from the sensors to the data logger is minimal, thus, can be neglected.

The data logger (as shown in Figure 3.15) has 16 channels of analog inputs and 0.5% of accuracy. The data logger was connected to a computer to read the sensors signals and calculate the true value using software provided by the manufacturer. Formula used to calculate real indoor/outdoor temperature and relative humidity were described in the data logger using the software. The data logger collected data measured every 6 minutes and saved the data in .txt file. Due to technical difficulties, measurement data cannot be collected continuously during experimental period. The measured data would be shifted to the data processing computer fortnightly.

The humidity and temperature sensors were covered with 4 stage of white-painted plastic plate to avoid error due to bright light (as shown in Figure 3.16). The sensors were calibrated every day using hygro thermo-anemometer for the first two weeks before being used for the measurement. Lay out of the experiment location and sensors arrangement is presented on Figure 3.17.

Figure 3.15 Picolog data logger

Figure 3.16 Construction of temperature and humidity sensors covered with 4 stage of white-painted plastic plate

Figure 3.17 Lay out of the experiment location and sensors arrangement

Comfort temperature is different from person to person and from location to location. It depends upon degree level of activity, metabolic rate, clothing factor and behavioral adaptation of the occupants and their outdoor environment. Adaptive cooling technique is one of the possible techniques which able to determine the comfort temperature and use it as indoor temperature set point. This technique is described in the next section.

3.4 Adaptive Cooling Technique

Adaptive cooling technique has been developed as a proposed technique to use comfort temperatures (T_c) for indoor temperature set point. The comfort temperature is determined from adaptive comfort model which relate behavioral adaption of the occupants and the outdoor air temperature (mean monthly maximum and minimum daily air temperature) as discussed in Chapter 2. Since these temperatures are higher than the standard set point temperature, the application would reduce the cooling load and thus, decrease the overall energy consumption.

The technique considered two comfort temperatures for air-conditioned (AC) and naturally ventilated (NV) building from previous research. These temperatures are used as indoor temperature set points during occupied and unoccupied period, respectively. The reason for this was due to the behavioral activities of the students and staff. Before the students and lecture enter classrooms, workshops or office, they were exposed to the ambient conditions expecting comfort temperature as in NV building.

When they enter the air-conditioned room, they experience sudden cooling and feel cold for a period of time. After sometimes, their bodies totally adapt with the new environment and expect comfort temperature as in AC building. At the end of the session, they would leave the room and exposed back to the ambient conditions - demanding comfort temperature as in NV building after sometimes.

To control the comfort temperature inside a building using adaptive cooling technique, there are three steps required as listed below:

1. Calculating mean monthly maximum and minimum daily air temperature

2. Developing adaptive comfort model

3. Developing adaptive cooling algorithm

The time needed by the occupants to adjust their neutral temperature from air-conditioned zone to naturally ventilated zone is assumed to be negligible and therefore, be neglected. The technique first calculated the mean monthly maximum and minimum daily ambient air temperature from weather data collected from the weather station. Secondly, adaptive comfort model was developed by adopting general adaptive comfort model and local adaptive comfort temperature. Detail explanation of each step of the technique would be presented in the following sections.

3.4.1 Adaptive Comfort Model

There were three parameters considered on developing adaptive comfort model for Malaysia: comfort temperature (T_c) for Malaysia [18], adaptive comfort model

(ACM) for air-conditioned (AC) building [101], and ACM for naturally ventilated (NV) building [73,99]. The general ACM was used to find correlation between the indoor and outdoor temperature at site. However, the ACM need to be matched with ACT at site as reported by Hussein et al. [18,72].

Modifications should be made according to the T_c before it was used for further work. By assuming that the slope between Malaysian comfort model and the previous model is the same, adaptive comfort temperature for Malaysia was found to be:

$$T_c = 0.31 T_{a,out} + 19.74 \qquad \text{for NV building} \qquad (3.30)$$

$$T_{cmax} = 0.31 T_{a,out} + 22.04 \qquad \text{for NV building} \qquad (3.31)$$

$$T_{cmin} = 0.31 T_{a,out} + 17.34 \qquad \text{for NV building} \qquad (3.32)$$

$$T_c = 0.093 T_{a,out} + 21.8 \qquad \text{for AC building} \qquad (3.33)$$

$$T_{cmax} = 0.093 T_{a,out} + 23 \qquad \text{for AC building} \qquad (3.34)$$

$$T_{cmin} = 0.093 T_{a,out} + 20.5 \qquad \text{for AC building} \qquad (3.35)$$

In this work, optimal comfort temperatures (T_c) in Equations 3.30 and 3.33 were chosen as indoor temperature set points during unoccupied and occupied periods in the proposed algorithm. Average of mean monthly maximum and minimum daily air temperature ($T_{a,out}$) were calculated based on Ipoh weather data. The temperature ($T_{a,out}$) was then used to calculate monthly T_c [67] as shown in Table 3.6.

Table 3.6 Monthly mean of indoor comfort and outdoor temperature

Month	Mean T_{min}	Mean T_{max}	$T_{a,out}$	T_c of 100% acceptability	
				Occupied	Unoccupied
Jan	24.14	32.42	28.28	24.43	28.51
Feb	24.34	33.34	28.84	24.48	28.68
Mar	24.45	33.75	29.10	24.51	28.76
Apr	24.21	32.47	28.34	24.44	28.53
may	24.31	32.62	28.46	24.45	28.56
Jun	23.81	31.82	27.82	24.39	28.36
Jul	23.64	31.48	27.56	24.36	28.28
Aug	23.84	31.67	27.75	24.38	28.34
Sep	23.80	31.64	27.72	24.38	28.33
Oct	23.30	30.27	26.79	24.29	28.04
Nov	23.51	31.22	27.37	24.35	28.22
Dec	23.31	31.06	27.19	24.33	28.17
Average			27.94	24.40	28.40

The results show that maximum gap from calculated monthly T_c in a year were only 0.22^{o}C and 0.72^{o}C for occupied and unoccupied periods. It can be seen that during hot period (Jan – Jun), T_c was lower than that during rainy period (Jul – Dec).

3.4.2 Adaptive Cooling Algorithm

Adaptive cooling technique has been developed for the HVAC system to effectively handle cooling load during occupied and unoccupied period using adaptive comfort temperature. The technique controls the system to achieve set points of indoor temperature and relative humidity. Occupancy sensor, temperature sensor and relative humidity sensors were used inside the room. These signals were sent to the controller to regulate the VAV damper in order to give the required supply air. A pressure sensor was placed inside the supply duct. The signal was used by the controller to maintain the duct static pressure by varying the fan speed. Another temperature sensor were also placed in the supply duct just after the cooling coil and inside the Chilled water supply (CHWS) pipe to measure supply air (T_s) and chilled water temperature. The controller used these signal to regulate the CHWR flow rate and T_s required. The process flow diagram (PFD) is shown in Figure 3.18.

FA : Fresh air   OE : Occupancy sensor
EA : Exhaust air   AE : RH sensor
RA : Return air   PE : Pressure sensor

CHWR : Chilled water return   C : Controller
CHWS : Chilled water supply   CC : Cooling coil
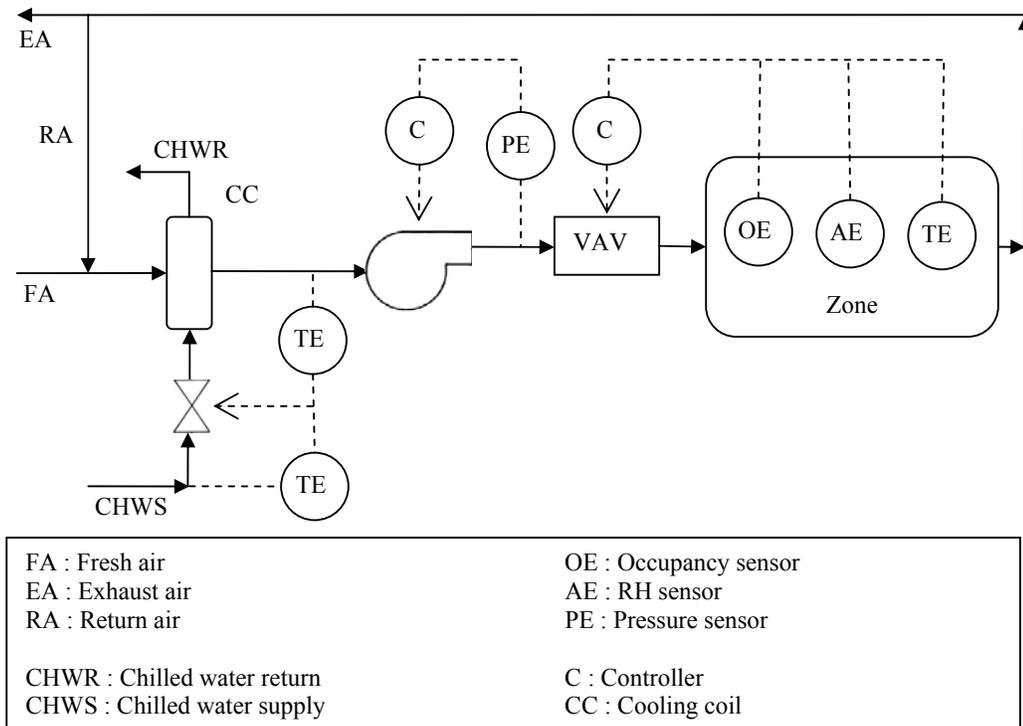
Figure 3.18 Process flow diagram of adaptive cooling technique

HVAC operation was set from 7.00 a.m. – 7.00 p.m. every day, except during holidays. Occupancy sensors in each room of the building were used to detect occupancy. The sensor send signal to the HVAC system to decide indoor thermal set points ($T_{r,sp}$ and $RH_{r,sp}$) prior to occupancy. If the room is occupied, the $T_{r,sp}$ and $RH_{r,sp}$ are as stated in Equation 3.33 while if the room is unoccupied, these values are as stated in Equation 3.30. Initially the system determines the required supply air flow rate based on currently measured indoor temperature (T_r), indoor relative humidity (RH_r), $T_{r,sp}$, $RH_{r,sp}$, and supply air conditions (T_s and RH_s). In this case, RH_s is always set to 100% since dehumidification process is always needed under humid climate. Due to this, the measurement of RH_s is not needed. Then, the next signal from online measurement of T_r, and RH_r would be used to decide what the system has to do. The algorithm of the adaptive cooling technique is shown in Figure 3.19.

START

HVAC schedule

Operation?

Yes — Occupancy sensor detects occupants

No — T_i and RH_i are floating

Occupied?

Yes — Occupied mode
$RH_{i,sp} = 70\%$
$T_{i,sp} = 0.093T_{rm} + 21.8$

No — Unoccupied mode
$RH_{i,sp} = 70\%$
$T_{i,sp} = 0.31T_{rm} + 19.74$

$RH_{i,sp}$, $T_{i,sp}$, T_s, T_i, RH_i

If $T_i > 24\,^{\circ}C$ $RH_i \geq 70\%$ — No
Yes — Increase supply Air flow rate (m_{sa})

If $T_i < 24\,^{\circ}C$ $RH_i \geq 70\%$ — No
Yes — Decrease supply Air flow rate (m_{sa})

If $T_i < 24\,^{\circ}C$ $RH_i < 70\%$ — No
Yes — Increase T_s

If $T_i > 24\,^{\circ}C$ $RH_i < 70\%$ — No
Yes — Decrease T_s

Supply air conditions Supply air flow rate (m_{sa})

No action — No

If $T_i = 24\,^{\circ}C$ $RH_i < 70\%$ — No

If $T_i = 24\,^{\circ}C$ $RH_i > 70\%$

Increase humidity of supply air — Yes

Decrease humidity of supply air — Yes

If $m_{sa,min} < m_{sa} < m_{sa,mi}$ — No

If $m_{sa} < m_{sa,min.}$ — No

$m_{sa.\,max.}$

Zone T_i, RH_i, T_s, m_{sa} measurement

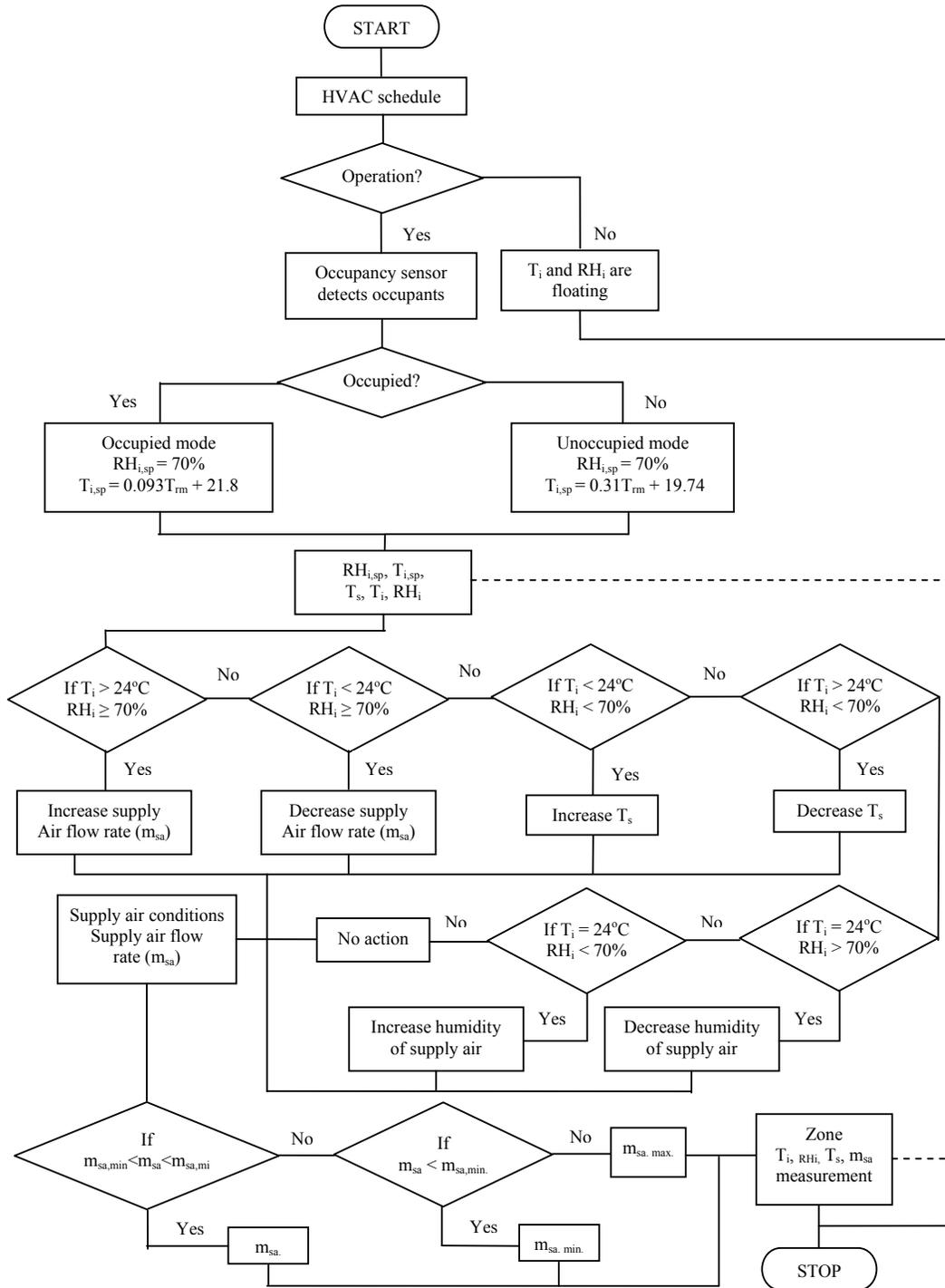Yes — $m_{sa.}$

Yes — $m_{sa.\,min.}$

STOP

Figure 3.19 Algorithm of adaptive cooling technique used for HVAC application

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There are six combinations of measured T_r, nd RH_r that result in different action from the system. These actions are based on the rules as described in the following text:

1. If $T_r > 24^{\circ}$C and $RH_r \geq 70\%$, then, the system would increase m_{sa}

2. If $T_r < 24^{\circ}$C and $RH_r \geq 70\%$, then, the system would decrease m_{sa}

3. If $T_r < 24^{\circ}$C and $RH_r < 70\%$, then, the system would increase T_s

4. If $T_r > 24^{\circ}$C and $RH_r < 70\%$, then, the system would decrease T_s.

5. If $T_r = 24^{\circ}$C and $RH_r > 70\%$, then, the system would reduce the humidity of supply air using dehumidifier, otherwise the system would keep the current settings.

6. If $T_r = 24^{\circ}$C and $RH_r < 70\%$, measurements are outside the seven possibilities describe above, the system would keep current supply air conditions at current flow rate if humidifier is not available, otherwise it could control the indoor parameter by using the humidifier.

The supply air flow rate required is then used to control motorized VAV damper to open and close accordingly. The fluctuation of percentage opening damper results in higher and lower duct static pressure than the maximum and minimum set points. If the static pressure is lower than the minimum set point, static pressure sensor would send signal to the VAV system to increase the air volume flow rate by increasing its frequency using variable frequency drive (VFD) and vice versa. According to this, the minimum/maximum supply air flow rate ($m_{sa,min.}/m_{sa,max.}$) is a rate at which the duct static pressure is equal to the lower/upper set point. These different air flow rates become the mechanical limitations of the VAV system.

Three online measurements were performed during cooling process i.e. supply air temperature, indoor air temperature and indoor relative humidity. These measurements are feed back to the system as basis to make required action (as described above). The supply air flow rate, chilled water supply and return temperature were then used to decide how much chilled water flow rate is needed.

The energy saving in terms of cooling load reduction is the difference between cooling load before and after the application of the algorithm (from the simulation results). Parametric analysis was performed on the simulation results to find the effect of mechanical limitation on the energy saving using the equations described in previous section. Economic analysis of the proposed system was performed to know whether the proposed system was economically feasible or not. The method used to do the economic analysis is presented in the next section.

3.5 Dual Supply System

In a VAV-HVAC system, dual ducting system is used to separate space cooling and heating process. The system may also be combined to set the required supply air temperature at a certain supply volume flow rate. The main advantages of dual duct system are that the supply air temperature and the variation of supply volume flow rate can be adjusted better according to the space load changes and outdoor ventilation air requirements. The system may use single fan or dual fans that are arranged in series or pararel [31]. Dual fans are widely used in the current dual duct VAV system because of its advantages to separate cold and warm air control in the cooling and heating process. Moreover, dual fans allow the system to the use of different ratios of outdoor air to supply air for the cold duct and the warm duct during cooling and heating process. Schematic diagram of typical dual duct dual fan system is shown in Figure 3.20.

Depending upon the geographic locations, HVAC system has three operation modes i.e. cooling mode, mixing mode and heating mode (as shown in Figure 3.21). In cooling mode, the system has maximum cold air supply volume flow rate (100%) during summer season and reduces to minimum (0%) during winter season. In heating mode, the process is on reverse direction as in cooling mode. In mixing mode, the reduction on cold supply volume flow rate is equal to the increase on warm supply volume flow rate. Due to this, the total supply air flow rate is constant and it becomes the minimum flow rate of the system [31].
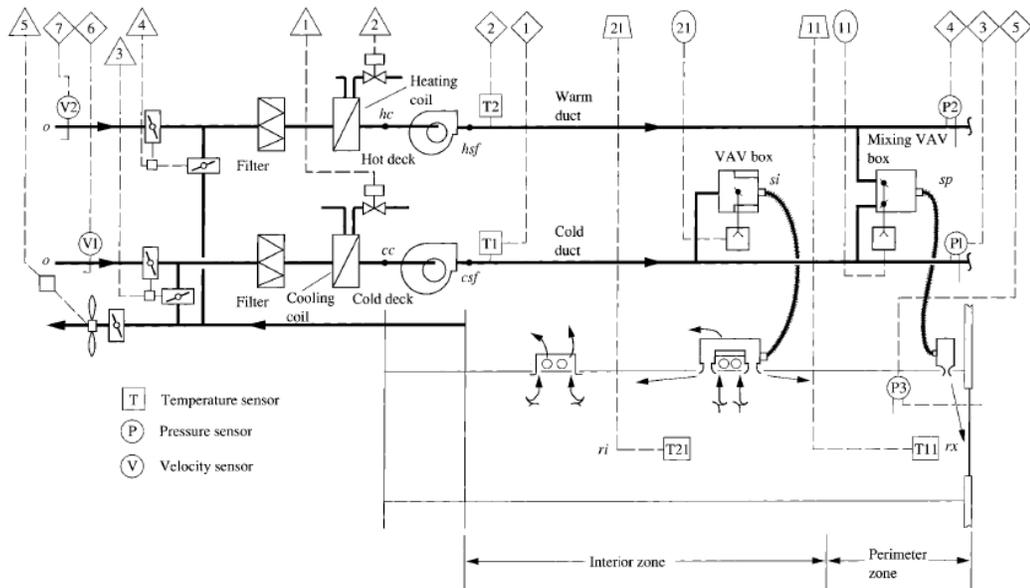
Figure 3.20 Schematic diagram of typical dual duct dual fan system [31]
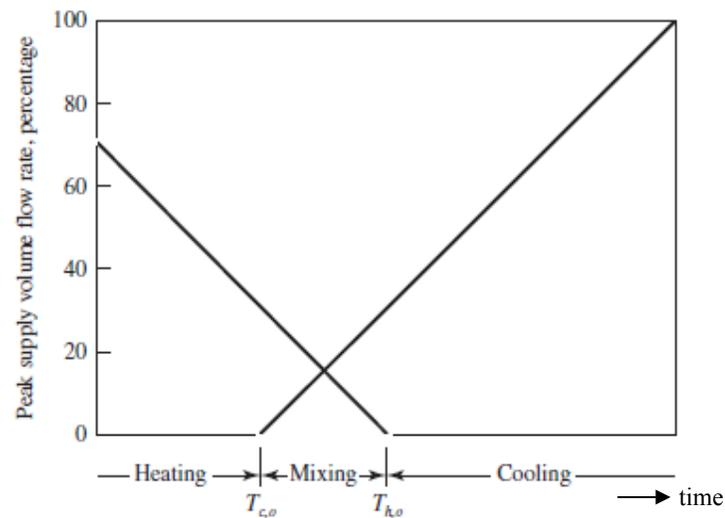
Figure 3.21 Cooling, mixing and heating mode [31]

In this research, dual duct dual fan system is used for cooling process only since the climate in Malaysia is hot and humid. The system that has wider range of supply volume flow rate is used to reduce negative effect of mechanical limitation of the system on handling the intermittent cooling load.

First, annual cooling load (hourly) is calculated using TRNSYS software, and the results were used to calculate the annual required supply air flow rate (CFM). From the calculations, the operation range of CFM required by the HVAC system during

occupied and unoccupied hours are determined. Following that, equal friction method is applied to size the (first) supply duct. Recommended pressure loss from 0.08 to 0.6 in.H_2O/100ft is considered during the sizing. In this step, the dimension (diameter) of the (first) supply duct is determined by plotting the recommended maximum pressure loss (0.6 in.H_2O/100ft) and the CFM maximum required during occupied hours on the friction loss chart (Appendix J). The CFM operation range of the (first) supply duct is then determined from the diameter of the duct and recommended minimum pressure loss (0.08 in.H_2O/100ft). If rectangular duct was used, the diameter found was equivalent diameter (D_e) of the rectangular duct. The actual sizes of the rectangular duct ($a×b$) were then determined using the equation in the following [31]:

$$D_e = \frac{1.3(a \times b)^{0.625}}{(a+b)^{0.25}}$$

(3.36)

If the minimum CFM of the (first) duct is still higher than the CFM required during unoccupied hours, then, the second supply ducting is designed using maximum CFM required during unoccupied hours. If the minimum CFM of the (first) duct is less than or equal to the CFM required by the system, then there is no need to use dual ducting system. Design flow diagram of dual ducting system is presented in Figure 3.22.

3.6 Economic Analysis

In any engineering project, economic analysis is needed to estimate the profits that the project would generate during its period of service [115]. The results from the economic analysis would be a basis for any engineering decision made. Predictions or forecasting is usually made in the economic analysis. However, the predictions are never completely accurate. Therefore, "economic decisions have to be based on the best information available at the time of the decision and a thorough understanding of the uncertainties in the forecasted data" [115]. In practice, this decision is the major of the project cost. Two common methods that are used to whether accept or reject a project proposal are payback method and discounted cash flow techniques (DCFs). These methods are described in the following sections.
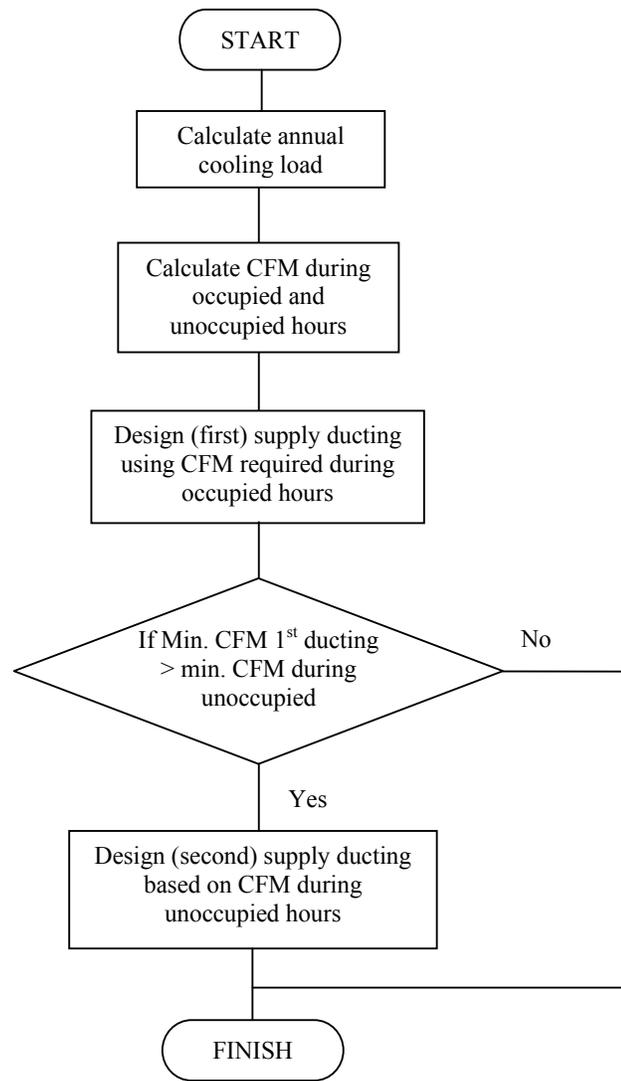
```
                        ┌─────────┐
                        │  START  │
                        └────┬────┘
                             │
                  ┌──────────┴──────────┐
                  │  Calculate annual   │
                  │    cooling load     │
                  └──────────┬──────────┘
                             │
                  ┌──────────┴──────────┐
                  │  Calculate CFM during │
                  │      occupied and     │
                  │   unoccupied hours    │
                  └──────────┬──────────┘
                             │
                  ┌──────────┴──────────┐
                  │ Design (first) supply ducting │
                  │  using CFM required during     │
                  │        occupied hours          │
                  └──────────┬──────────┘
                             │
                        ◇ If Min. CFM 1ˢᵗ ducting ◇───── No
                          > min. CFM during
                             unoccupied
                             │ Yes
                  ┌──────────┴──────────┐
                  │ Design (second) supply ducting │
                  │   based on CFM during           │
                  │      unoccupied hours           │
                  └──────────┬──────────┘
                             │
                        ┌────┴────┐
                        │ FINISH  │
                        └─────────┘
```

Figure 3.22 Design flow diagram of dual ducting system

3.6.1 Payback Method

Payback method is used to decide whether a project should be accepted or rejected based on how long it takes for net receipt to equal investment outlays (payback period) [115]. Project with payback period shorter than the maximum acceptable payback period will be accepted and vice versa. There are two types of payback periods under this method i.e. conventional payback period and discounted payback period. The first type ignores time value of money while the second type is taking it into consideration.

Conventional payback period calculation determines the payback period merely based on the initial cost and the annual benefit [115]. The weakness of this method is that the method gives no benefit during payback period and does not taking into account the time value of money from the invested money. As an answer to this, discounted payback period is developed as a better method than conventional payback period that overcomes the problems. This method able to determine the period necessary to recover both the capital investment and the cost of fund required. However, this measurement was unable to show the profitability of the project for its whole period of service especially after the payback period. Therefore another method called DCFs is developed and used as another economic measurement to strengthen the decision made for the project.

3.6.2 Discounted Cash Flow Techniques (DCFs)

DCFs have 4 ways to do a project analysis. However, user only needs to choose one of these DCFs in order to analyze the project. In this work, net present worth (NPW) criterion is used by the author to economically analyze the proposed system.

Under NPW criterion, the time value of money from the annual cash inflows is included in the analysis [115]. The total present worth of cash inflow is compared with the total present worth of cash outflow (as invested in the project). Present worth of cash inflow or outflow is the equivalent value of cash inflow or outflow at time 0 (when the project is proposed or started). Application procedures of NPW criterion are described in the following.

1. Determine the expected rate of return or minimum attractive rate of return (MARR) of the company or institution.

2. Predict how long the service of the project will stand.

3. Estimate the annual cash inflow for each period over the service lifetime.

4. Predict the annual cash outflow over each service period.

5. Calculate the present worth (PW) of each net cash flow at the MARR using formula as below:

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$$PW(i) = \frac{AA_o}{(1+i_{rate})^0} + \frac{AA_1}{(1+i_{rate})^1} + \frac{AA_2}{(1+i_{rate})^3} + ... + \frac{AA_n}{(1+i_{rate})^n} = \sum_{n=0}^{N} \frac{AA_n}{(1+i_{rate})^n} \quad (3.37)$$

6. Determine the net present worth (the difference between present worth of cash inflow and outflow).

7. Make decision based on the following decision rule:

If PW (i) > 0, accept the investment.

If PW (i) = 0, remain indifferent.

If PW (i) < 0, reject the investment.

A positive NPW indicates that the project makes profit since the equivalent worth of the inflows is greater than the equivalent worth of outflows and vice versa. NPW equal to zero means that the project able to exactly fulfill the expected rate of return from the invested money without making additional profit. Therefore, a project should be accepted when NPW is equal to zero or greater than zero.

Summary

Methodology to achieve objectives stated in Chapter 1 has been discussed in detailed in this chapter. Basic theory used to analyze the performance of HVAC system was presented as fundamentals to do the analysis in next chapter. The information of the studied building as an important part in the simulation was described and used to make the building model. The building model was then used in the simulation with other component to simulate the cooling load characteristic of the building. Following that indoor and outdoor experimental works were explained as justification to the simulation model. Adaptive cooling technique was proposed and discussed as solution for effective and efficient cooling system. Finally, economic analysis method was presented to determine feasibility of the proposed system.