CHAPTER 4

RESULTS AND DISCUSSIONS

Transient simulation of intermittent cooling load characteristics of the building with existing HVAC system have been done for a year and are presented in this chapter. However, since the building has many rooms, these rooms are classified into three according to the usage pattern i.e office, classroom and workshop. The cooling load patterns of rooms with same type would close to each other and thus, three rooms from each type of rooms are chosen to represents the cooling load patterns.

Two experimental works (indoor and outdoor experiments) were done and the results were compared to the simulation results to validate the model. The comparison was performed to shows the ability of the model to behave close to the real one. Using the validated model, the performance of current HVAC system and the effect of various types of rooms are discussed. The cooling load characteristics of adaptive cooling technique as proposed system were compared with the current system to determine the energy saving potential in terms of cooling load reduction. At the end of this chapter, the economic analysis of the proposed system was discussed to understand the feasibility of the system.

4.1 Cooling Load Characteristic

Simulation results of cooling load characteristics for office, classroom, and workshop are presented in the next sections. For better visualization, the simulation results are presented for one or two days.

4.1.1 Predicted Cooling Load Characteristic of the Office

An office room facing west located on ground floor of the building is analyzed as representation of other office type rooms. In this room, internal heat gains from lighting, electronic devices, computers and occupants exist during working hours while stored heat during nighttime exist since the beginning of HVAC system operation. External heat gain from ventilation air, infiltrated air and heat transfer from outdoor to indoor environment exist for the whole day.

The simulation results showed that the maximum of *RSH* and *RLH* were 17.53 MJ and 5.87 MJ while the averages of *RSH* and *RLH* were 10.31 MJ and 2.66 MJ, respectively. These results show that the latent cooling load is only 33.5% of the *RSH*. The reason for this is because the numbers of occupant as latent heat source is very low (only two staff) compare to the room volume which is 166.5 m^3. The maximum *RTH* (total *RSH* and *RLH*) were found to be 20.66 MJ with an average of 12.97 MJ, respectively. Based on the maximum *RSH* and *RTH*, the *RSHF* was found to be 0.85. This value is within the range of RSHF value for typical office room which is between 0.7 – 0.93 [116].

The maximum *GSH* and *GLH* were 19.08 MJ and 8.39 MJ while the averages *GSH* and *GLH* were 10.91 MJ and 4.17 MJ, respectively. It can be seen that the maximum of *GSH* and *GLH* values are 8.8% and 42.93% higher than the maximum *RSH* and *RLH*, respectively. These implied that, in this room, ventilation heat gain is dominated by latent heat component. The maximum grand total cooling loads (*GTH*) was 24.05 MJ with an average of 15.08 MJ. This cooling load represents cooling capacity of the cooling coil. Usage factor as percentage of total occupied hours of the room to the total HVAC operation hours was found to be 56.31% in this room. It indicated that the HVAC system is at minimum setting for almost half of the total operation hours. Typical hourly cooling load characteristic in the office is presented in Figure 4.1.

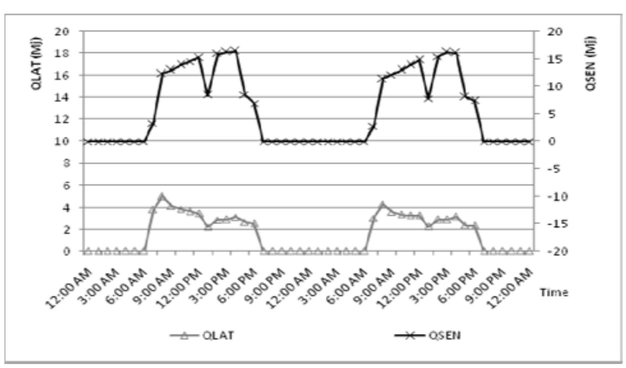
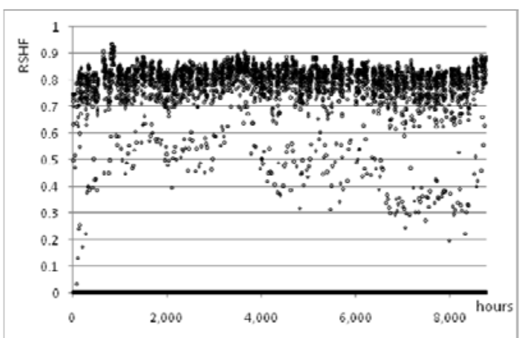
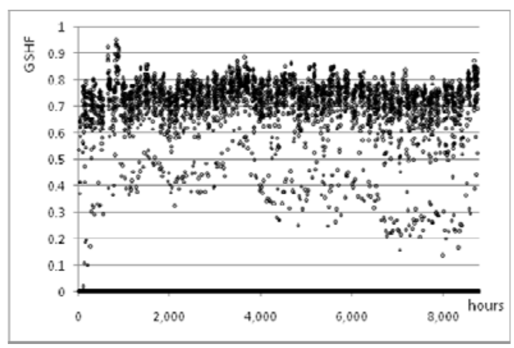


Figure 4.1 Typical hourly cooling load characteristic in the office

Room sensible heat factors (*RSHF*) were found to be between 0.7 to 0.9 while grand sensible heat factor (*GSHF*) were in the range of 0.6 – 0.85. In VAV system, the value of *RSHF* and *GSHF* is pre-determined to define the supply air condition and chilled water flow rate. The values remain fixed during HVAC operation. It indicated that there was a period where the supply air conditions were not properly set during HVAC operation (when actual *RSHF* was not same with the design value). Moreover, the results show that there were several hours when the RSHF and GSHF drop up to 0.2. This was happened mostly in the early morning or during rainy days with low ambient temperature and high ambient humidity. If the VAV system fails to provide supply air condition and chilled water flow rate over the RSHF and GSHF operation range, indoor temperature and humidity would exceed the set points. This phenomenon was in accordance with the indoor experimental measurement results that are explained in later section. The *RSHF* and *GSHF* for a year are presented in Figure 4.2.

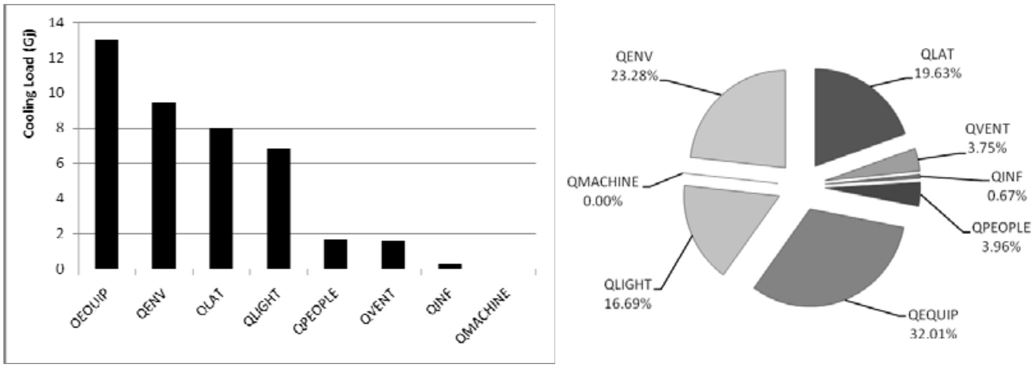




(a) (b)

Figure 4.2 a.) Annual *RSHF* value in the office, b.) Annual *GSHF* value in the office

Breakdown of the cooling load and the percentage of cooling load components as presented in Figure 4.3 showed that *QEQUIP* was the largest cooling load (32.01%) and was accounted up to 13.03 GJ. Heat gains from *QENV* (23.28%) and *QLAT* (19.63%) were the second and third largest cooling load followed by *QLIGHT* (16.69%), *QPEOPLE* (3.96%), *QVEN* (3.75%), *QINF* (0.67%), and *QMACHINE* (0%). *QMACHINE* was found zero because there was no machine in this type of room. Small portion of *QINF* reflects that the room is well insulated.



(a) (b)

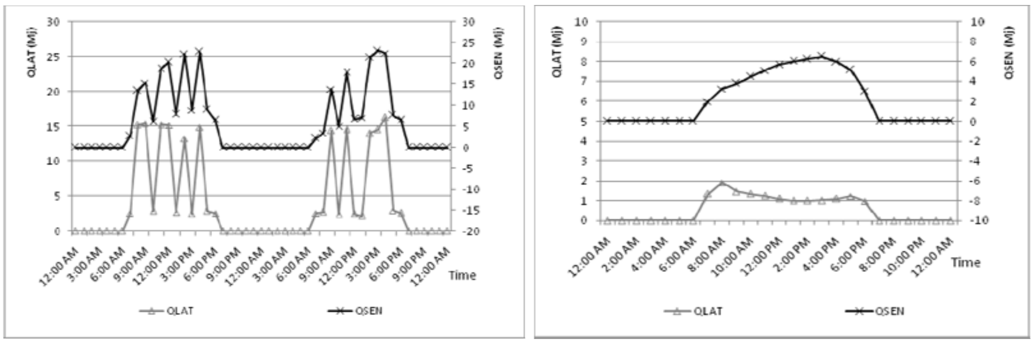
Figure 4.3 a.) Breakdown of cooling load in the office, b.) Percentage breakdown of cooling load in the office.

4.1.2 Predicted Cooling Load Characteristic of the Classroom

In order to understand the cooling load characteristic in a classroom, a 227.8 m^3 of classroom facing west located on first floor of the building is analyzed. Inside this room, sensible and latent heat rise due to internal and external cooling load which was driven by occupant and outdoor environment. However, occupancy in the classroom was not continuous as found in the office. In a day, the classroom are intermittently used for several hours (two to five hours) and left empty after the class. This condition enlarges the gap between maximum and minimum cooling load. During examination period, the exam was held in other room that made the class empty. However, the classroom was still conditioned by the centralized HVAC system. It implied that the cooling load characteristic during peak and off-peak period were significantly different.

The maximum *RSH* and *RLH* were accounted up to 24.35 MJ and 18.67 MJ, respectively. It can be seen that the maximum *RLH* is up to 76.67% of the *RSH* which is much higher than in the office room. This is because of number of the occupants in the classroom (30 occupants) which is significantly higher than in the office. Another reason is due to less number of internal sensible heat sources from electronic/electronic devices in the classroom than in the office. These conditions were represented by the *RSHF* which found to be 0.57 (based on the maximum *RSH* and *RLH*). This value is much lower than in the office room.

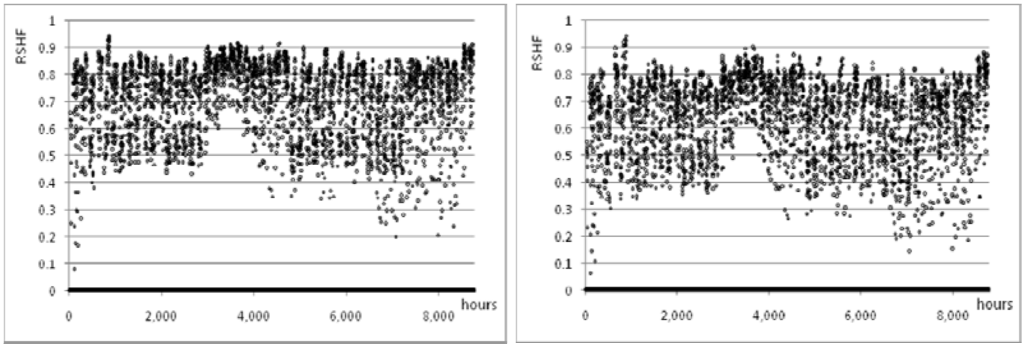
The maximum *GSH* and *GLH* were found to be 30.63 MJ and 29.53 MJ, respectively. These values are 25.8% and 58.17% higher than the *RSH* and *RLH* respectively. These indicate that the ventilation load was considerably high and dominated by latent cooling load component. The maximum *GTH* were found to be 56.23 MJ with an average of 11.99 MJ. Typical hourly cooling load characteristic in the classroom during peak and off-peak period are presented in Figure 4.4.



(a) (b)

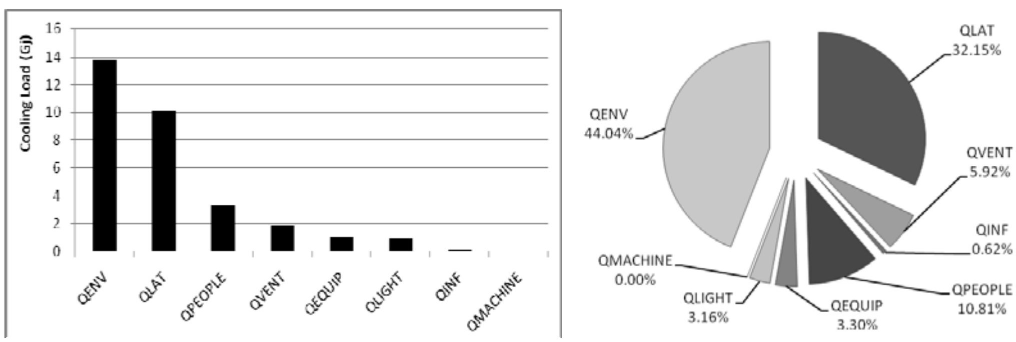
Figure 4.4 Typical hourly cooling load characteristic in the classroom. a.) during occupied period, b.) during unoccupied period.

RSHF was found to be between 0.6 to 0.95 while *GSHF* was found in the range of 0.5 – 0.9 (as presented in Figure 4.5). These ranges are wider than the ranges in the office and the minimum of *RSHF*/*GSHF* are lower than the values in the office. It indicated that latent load component was much higher in the classroom rather than in the office. Breakdown of the cooling load and the percentage of cooling load component as presented in Figure 4.6 showed that *QENV* was the largest cooling load (44.04%) and accounted up to 13.78 GJ. Heat gains from *QLAT* (32.15%) and *QPEOPLE* (10.81%) were the second and third largest cooling load followed by *QVEN* (5.92%), *QEQUIP* (3.3), *QLIGHT* (3.16%), *QINF* (0.62%), and *QMACHINE* (0%). Usage factor of the classroom was found to be only 11.68%. It implies that the HVAC system was mostly operated at minimum capacity.



(a) (b)

Figure 4.5 a) Annual *RSHF* value in the classroom, b) Annual *GSHF* value in the classroom



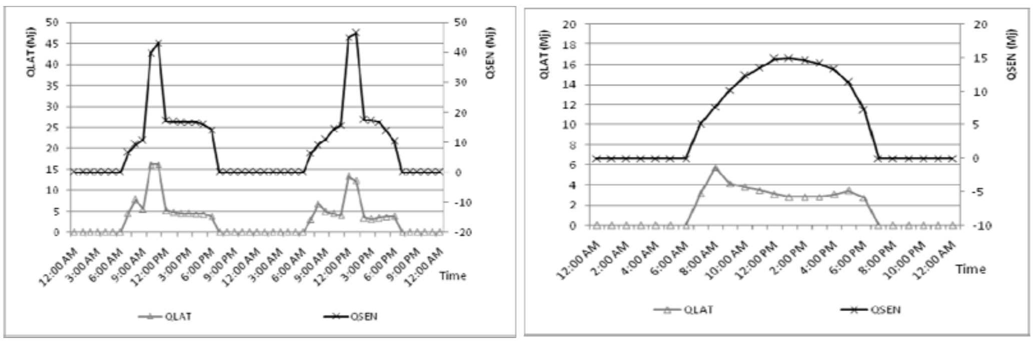
(a) (b)

Figure 4.6. a.) Breakdown of cooling load in the classroom. b.) Percentage breakdown of cooling load in the classroom.

4.1.3 Predicted Cooling Load Characteristic of the Workshop

A mechanical workshop facing east located on ground floor is selected to analyze the cooling load pattern. The workshop has volume 637.84 m^3 and three electric discharge machines (EDM) wire cut machines. The heat gain from these machines give additional internal sensible heat gain to the workshop. In The sensible and latent heat rise due to occupancy and practical activity inside the workshop.

The maximum *RSH* and *RLH* were 49.5 MJ and 21.5 MJ, respectively. The portion of *RLH* is 43.4% of the *RSH* and represented by the *RSHF* value which found to be 0.7 (from the maximum of *RSH* and *RLH*). This value is higher than in the classroom but lower than in the office. This is because in the workshop, the number of occupants higher than in the office but lower than in the classroom. In addition, there was additional internal sensible heat gain from the machine which contributes to the *RSH*. It should be noted that there was a sudden increase on both sensible and latent heat gain when the room was occupied for only several hours. It represents low usage factor in the workshop. However, the maximum of *RTH* were 66.34 MJ and to be the highest among the three of rooms under the analysis. Typical hourly cooling load characteristic in the workshop during peak and off-peak period were presented in Figure 4.7.

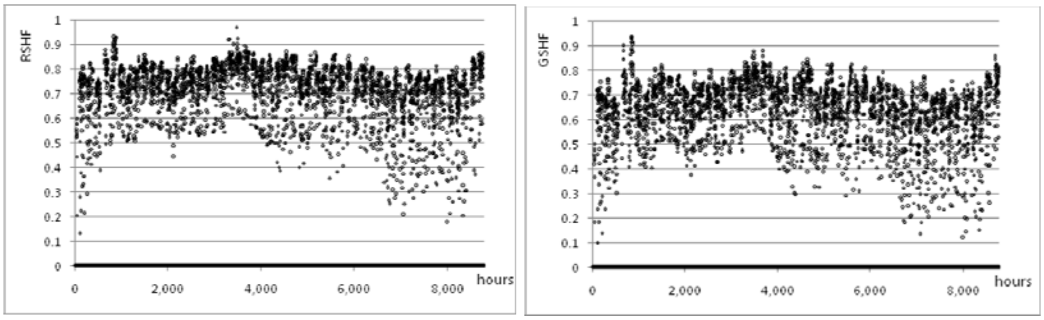


(a) (b)

Figure 4.7 Typical of hourly cooling load characteristic in the workshop. a.) during occupied period. b.) during unoccupied period.

The maximum *GSH* and *GLH* were 59.19 MJ and 39.72 MJ, respectively. These values are 19.58% and 84.74% higher than the *RSH* and *RLH* respectively. It implied that heat gain from ventilation has significant contribution to the cooling load especially the latent cooling load component. The percentage was also higher than in the classroom even though the numbers of occupants in this room is lower than in the classroom. The reason of this is because the workshop is much bigger than the classroom so that it needs much more ventilation air (building component) for air circulation. Moreover, since the occupant was doing mechanical work, the degree levels of activity were higher than in the classroom which results in more latent heat release to the room. The maximum *GTH* were 90.49 MJ and to be the highest among the rooms under analysis.

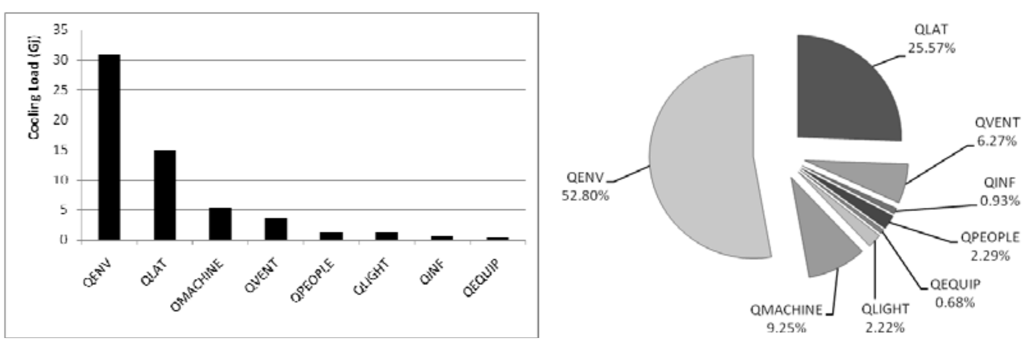
The *RSHF* was found to be between 0.50 to 0.85 while the *GSHF* was mostly between 0.45 to 0.8. Similar with the classroom, these results implies that the workshop has high latent load component and has wider range of *RSHF* and *GSHF* than in the office. The *RSHF* and *GSHF* for a year were presented in Figure 4.8.



(a) (b)

Figure 4.8 a.) Annual *RSHF* value of the workshop, b.) Annual *GSHF* value of the workshop

Breakdown of the cooling load and the percentage of cooling load component as presented in Figure 4.9 showed that *QENV* was the largest cooling load (52.8%) which was accounted up to 30.85 GJ for a year. Heat gains from *QLAT* (25.57%) and *QMACHINE* (9.25%) were the second and third largest cooling load followed by *QVENT* (6.27%), *QPEOPLE* (2.29%), *QLIGHT* (2.22%), *QINF* (0.93%), and *QEQUIP* (0.68%). Usage factor of the workshop was found to be only 7.55%. This value is the lowest among the three rooms and indicates that a lot of energy for cooling is wasted during unoccupied period.



(a) (b)

Figure 4.9 a.) Breakdown of cooling load in the workshop. b.) Percentage breakdown of cooling load in the workshop.

The simulation results of cooling load characteristic for office, classroom and workshop in the analysis is summarized in Table 4.1 and 4.2.

Table 4.1 Summary of predicted *RSH*, *RLH*, *RTH* and *RSHF*

Room Type	RSH (MJ)		RLH (MJ)		RTH (MJ)		Average RSHF	
	Maximum	Mean	Maximum	Mean	Maximum	Mean	Occupied	Unoccupied
Office	17.53	10.31	5.87	2.66	20.66	12.97	0.85	
Classroom	24.35	6.4	18.67	3.35	40.27	9.75	0.53	0.73
Workshop	49.5	13.1	21.5	4.98	66.34	6.19	0.73	0.7

Table 4.2 Summary of predicted *GSH*, *GLH*, *GTH* and *GSHF*

Room Type	GSH (MJ)		GLH (MJ)		GTH (MJ)		GSHF	
	Maximum	Mean	Maximum	Mean	Maximum	Mean	Occupied	Unoccupied
Office	19.08	10.91	8.39	4.17	24.05	15.08	0.79	
Classroom	30.63	7.02	29.53	4.98	56.23	11.99	0.46	0.66
Workshop	59.19	14.33	39.72	7.8	90.49	7.58	0.64	0.63

4.2 Experimental Data Measurement

Indoor and outdoor experimental measurements were conducted for 10 days starting from Friday (September, 17th 2010) at 4 p.m. until Monday (September, 27th 2010) at 4 p.m. The measurements were conducted to determine indoor/outdoor environment and indoor supply air condition which would be used to verify the building model.

4.2.1 Indoor Experimental Results

On weekends (HVAC system was off), indoor air temperature (T_{i}) and RH (RH_{i}) were floating according to the ambient condition. The next five days (weekdays), during HVAC operation (from 7 a.m. – 7 p.m.), T_{i} varied from 23^{o}C to 24^{o}C for the first two days and from 20^{o}C to 23.5^{o}C on the next three days (Figure 4.10). The reason why T_{i} varied on different range was because the HVAC operator changed the supply air temperature set point ($T_{\mathrm{s,sp}}$) into a higher value as shown in Figure 4.11. However, the setting was changed back to the design setting after two days. Using the design setting, T_i was found to be below the set point (24^{o}C) during HVAC operation on the last three days which mean that over cooling occurred.

For whole experimental period, RH_{i} varied according to T_{i} since there was no *RH* control on the HVAC system. On weekdays, during AC operation, RH_{i} varied from 73% to 80% when T_{i} varied from 23^{o}C to 24^{o}C. RHi varied from 73% to 75% when T_{i} varied from 20^{o}C to 23.5^{o}C. These results showed that when $T_{\mathrm{s,sp}}$ was increased into higher value, the ability of the supplied air to handle latent cooling load was decreased. It was due to fact that the supplied air would be more humid as the Ts increases. Sudden increase on indoor relative humidity also occurred at the beginning of HVAC operation due to sudden decrease on indoor temperature. During non-HVAC operation, indoor temperature and relative humidity raised due to heat and mass transfer with outdoor environment.

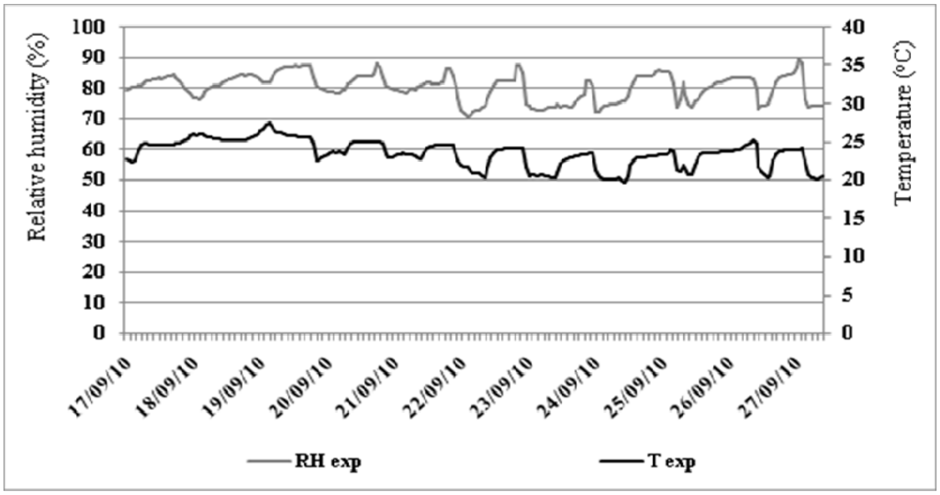


Figure 4.10 Measured indoor temperature and relative humidity from the experiments.

Measured supply air flow rate and temperature data were shown in Fig. 4.11. The results showed that there were differences on supply air temperature entering the zone (20^{o}C and 16^{o}C). On weekdays, during HVAC operation, overcooling was occurred (T_i<23^{o}C) when $T_{s,sp}$ was set according to the design value (16^{o}C). This condition increased complains from the occupants (5 persons) to increase the T_i (~24^{o}C). The complaints were mostly during morning (8 a.m – 10 a.m) and afternoon (3.30 p.m – 5 p.m) time. As a response, the maintenance operator increased the $T_{s,sp}$. The set points were changed to higher value when the complaints were made about the extra cooling and vice versa. Due to the fact, the $T_{s,sp}$ should has been fixed to a certain value to control overheating/overcooling problems.

Supply air flow rate entering the zone was found to be approximately the same with different $T_{s,sp}$. During HVAC operation, VAV dampers would regulate supply air flow rate entering the zone according to T_i which was fed by the indoor temperature sensor. After T_i reached the set point, VAV dampers should be completely closed (no supply air) and would start to open again, once T_i fall below the set point. However, current HVAC system could not completely close the VAV dampers since ventilation air was always needed regardless of the T_i value. This condition implied that ventilation air needed in the zone or group of zones should become the minimum supply air flow rate of the AHU.

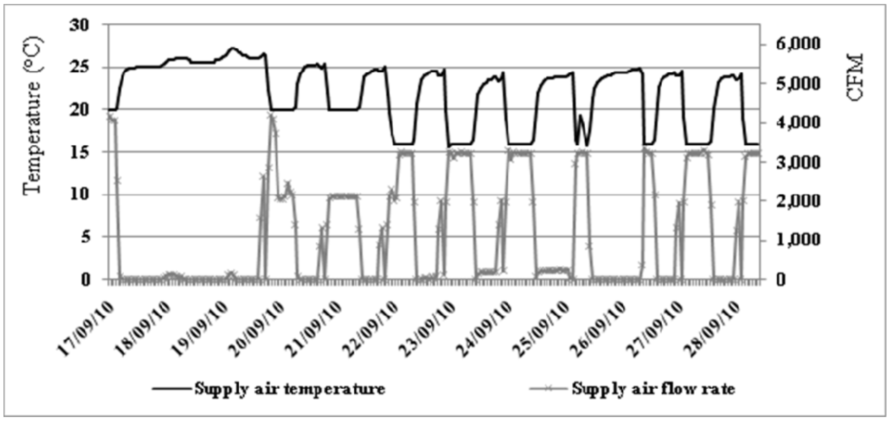


Figure 4.11 Measured supply air temperature and flow rate enter the zone.

Based on one of the current AHU setting and technical data serving two zones (office and store room), the maximum and minimum flow rate were 5,566 CFM and 1,900 CFM based on recommended friction loss (0.08 – 0.6 inH_2O/100ft). This information would be used to analize the effect of occupancy pattern on the cooling load and HVAC system performance in other zones.

4.2.2 Outdoor Experimental Results

Outdoor experimental results are presented in Figure 4.12. The results showed that highest global solar radiations are available between 11.00 a.m – 02.00 p.m. During this period, ambient temperature was observed to be the highest (30^{o}C – 35^{o}C) and the relative humidity was found to be the lowest (50% - 70%). This phenomenon occurred because of the characteristic of air where the capacity of vapor contents would increase as the temperature increases. Therefore, the air relative humidity decrease as the air temperature increase since relative humidity is the ratio of vapor content of the air over the capacity of vapor content that can be hold by the air. It can also be observed that 8^{o}C – 10^{o}C increase in ambient temperature results in a decrease of 38% - 42% of the relative humidity. This implies that relative humidity of ambient air is sensitive to ambient temperature. During nighttime, ambient temperature decrease periodically and reaches minimum value (23°C – 26°C). In contrary, the relative humidity increase periodically and reaches maximum during this time (90% - 98%). The experimental results represent the characteristic of weather in Malaysia that are hot and humid.

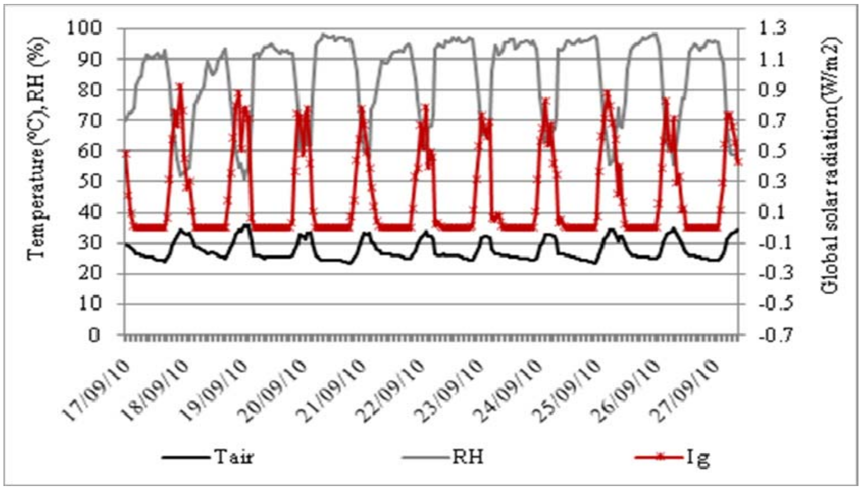


Figure 4.12 Measured outdoor environmental conditions (ambient temperature, ambient relative humidity and global solar radiation).

4.3 Model Validation

Indoor and outdoor experimental results were used to validate the simulation model. The simulation of the HVAC system with the building model is performed to calculate indoor temperature and relative humidity using weather data and supply air condition (supply air flow rate and temperature) data from the experiments. The simulation results: indoor temperature (T_{sim}) and indoor humidity (RH_{sim}) were then compared with the measured values as shown in Fig. 4.13 and 4.14.

The comparison for both weekends and weekdays periods, showed that 90.8% of the predicted indoor temperature values were fall within $\pm$1.5°C of the measured values with R^2 value to be 0.84. The graph also showed that 98.33% of the simulated humidity values were found to be within $\pm$10% of the measured values. The trend for both measured and simulated values were found to be closed to each other. However, on some hours there were some errors which result in 0.56 of R^2 value.

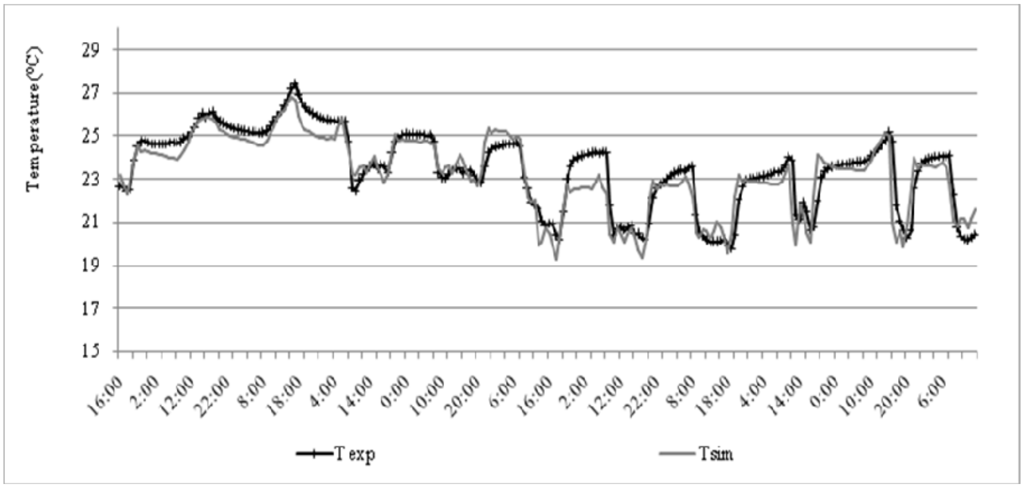


Figure 4.13 Comparison of indoor temperature between experimental and simulation results.

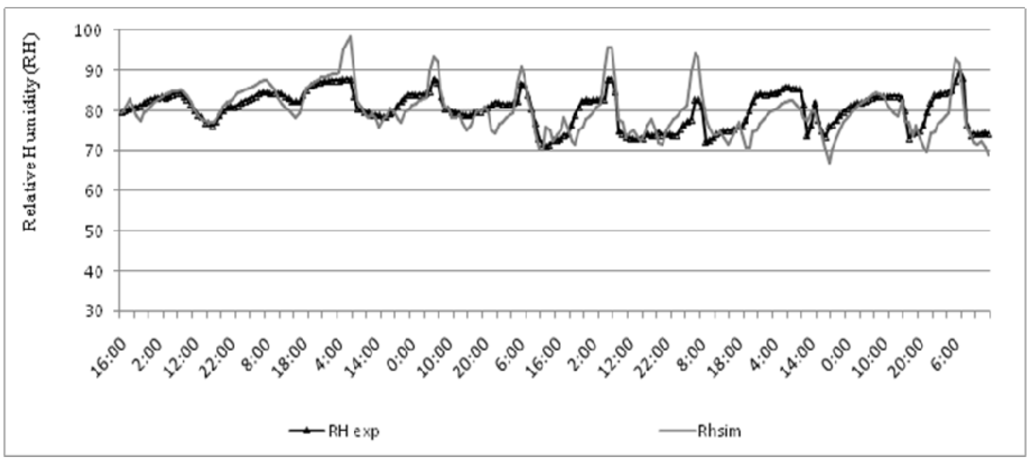


Figure 4.14 Comparison of indoor relative humidity between experimental and simulation results.

In order to give better explanation to the errors occurred during validation process, the comparisons are presented fortnightly during weekends and workdays in Figure 4.15 and 4.16, respectively. From these graphs, it can be seen that the errors occurred on the relative humidity is a function of temperature for both periods. The reason of this is due to the fact that relative humidity is influenced by air temperature and thus, the errors on indoor temperature give additional error to the indoor relative humidity.

During weekend (HVAC system off), the predicted indoor temperature and relative humidity are in good agreement with the experimental results (as shown in Figure 4.15). The simulation results of indoor temperature falls within $\pm 1^{o}$C of the experimental values while predicted indoor relative humidity was within $\pm$5% of the experimental values. However, the error occurred during weekdays (under HVAC operation) were higher than on weekends (as shown in Figure 4.16). The errors on the indoor temperature and relative humidity were found within $\pm 1.5^{o}$C and $\pm$10%, respectively. The reasons for this are due to the unpredictable infiltration air, effect of vapor storage and thermal mass which hard to be perfectly matched with the real one and effect of uniform indoor air temperature assumption that was used in the simulation.

The comparison presented above showed that the experimental measurements and the simulation results are having the similar trends, however, the instantaneous values have some minor difference which fall within $\pm 1.5^{o}$C (T_r) and $\pm$10% (RH_r). These results is found to be similar with the results obtained by Aynur et al. [92] where 90.6–94.7% of the indoor temperature experimental data fall within $\pm 1.5^{o}$C range of the simulation data, and 88.3–91.3% of the indoor relative humidity data fall within $\pm$18% range of the simulation data. Therefore, the simulation model has reasonable accuracy to be used for further analysis and investigations.

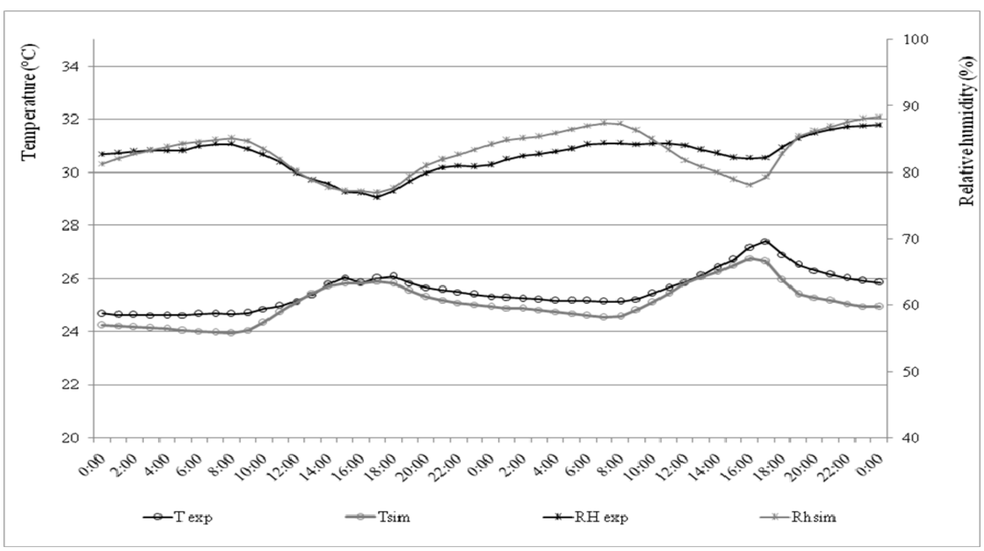


Figure 4.15 Comparison of predicted and measured indoor temperature and relative humidity during weekends

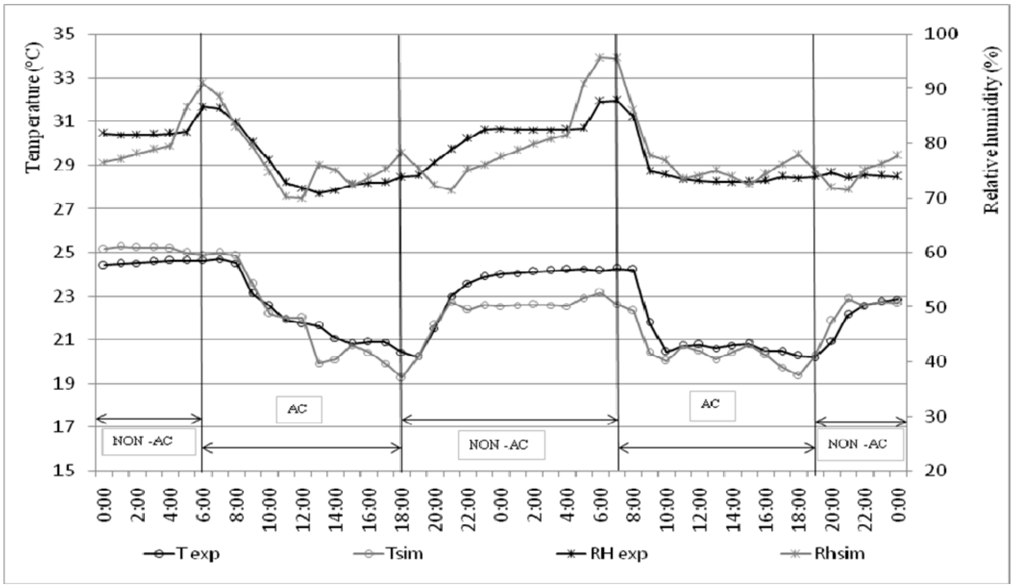


Figure 4.16 Comparison of predicted and measured indoor temperature and relative humidity during weekdays

4.4 HVAC System Performance Dependency on Design Cooling Load

Calculated cooling load known as design cooling load is an essential part in designing good performance of an HVAC system. In order to know the dependency of HVAC system performance on the design cooling load, an office and a store were considered. These rooms were served by an AHU and has no mechanical window. From the cooling load calculation, range of *RSHF* values for office and store rooms were 0.7 – 0.9 and 0.65 – 0.9. Using 0.65 of *RSHF* value, the required supply air conditions were 16^{o}C with humidity 0.0115 kgH_2O/kg of dry air, as shown in Figure 4.17. These supply air conditions are found to be exactly the same with the current conditions.The supply air flow rate needed were then calculated based on the supply air conditions and the cooling load using Equations 3.7 and 3.8. The maximum flow rates needed for the office and store rooms were found to be 1,049 CFM and 790 CFM while the minimum flow rates were 143 CFM and 132 CFM, respectively. According to this, the AHU should be able to provide air flow rates of between 275 and 1,839 CFM.

These information were then used to design the ducting. Equal friction method as a common method on ducting design was considered. It should be highlighted that the recommended friction loss was between 0.08 to 0.6 in.wg/100 ft. Using friction chart for round duct, the duct diameter was found to be 11.5 inch (as shown in Fig. 4.18) and the operating range of supply air flow rates of the system become 530 to 1,839CFM. For rectangular duct, the duct size (*a* x *b*) is determined using Equation 3.36. Assumed that *a* is 2 times *b*, then, the rectangular duct size is 15.1 x 7.55 inch. The supply air flow rate shows that the minimum value was found higher than the calculated value (275 CFM). The condition where the required supply air flow rate is below 530 CFM happen for 171 hours or 5.7% of total HVAC operation hours (3,000) in a year. Under this conditions, the office and the store were overcooled. The calculated supply air flow rate, supply air flow rate design, and duct diameter are presented in Table 4.3.

Table 4.3 Calculated supply air flow rate, supply air flow rate design, and duct diameter

RSHF Design	RSHF	Room	Calculated CFM		CFM required		CFM Design		Duct	Friction
			Max.	Min.	Max.	Min.	Max.	Min.	Dia. (inch)	in.wg/100ft
0.65	0.60 - 0.90	Office	1,049	143	1,839	275	1,839	530	11.5	0.08 - 0.6
	0.65 - 0.90	Store	790	132						

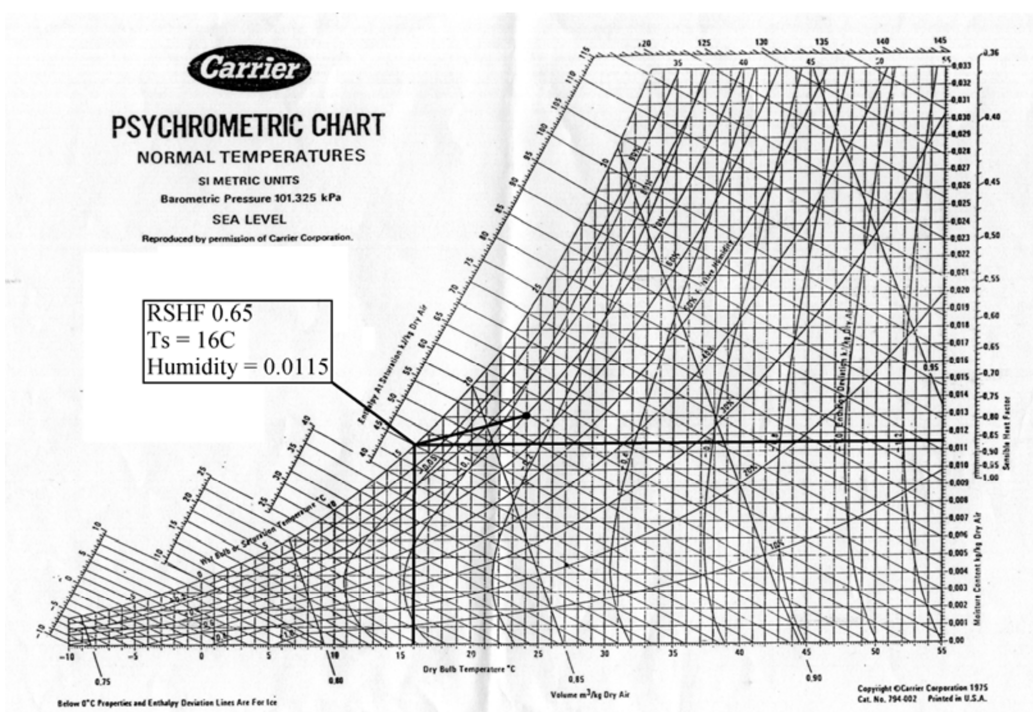


Figure 4.17 Plotted 0.65 and 0.8 *RSHF* line on the psychrometric chart [117]

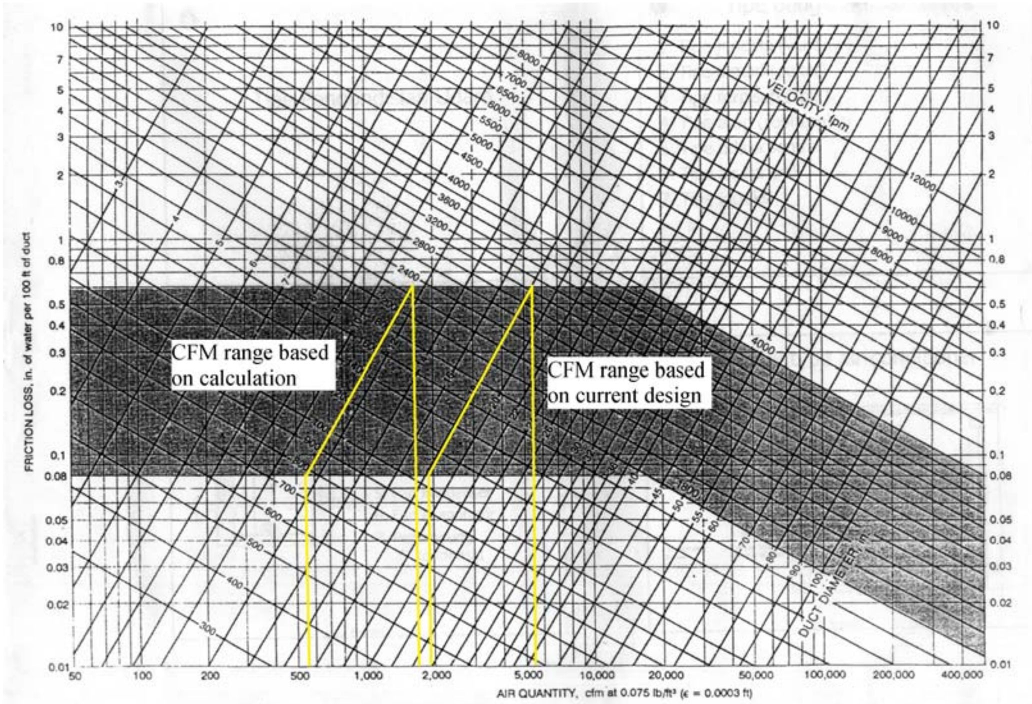


Figure 4.18 Ducting design using friction chart for round duct [31,118]

In real application, some designers apply safety factor to the design cooling load in order to overcome uncertainties (i.e. incresing number of occupant, additional internal heat gain, modification of the building envelope, etc.). This safety factor would increase the supply air flow rate and oversize the system. To understand the effect of the safety factor applied, the actual system is analyzed and discussed.

The fan selected has maximum airflow rate up to 5,566 CFM which means that the sytem is overdesigned. Using friction chart for round duct and considering recommended friction loss from 0.08 to 0.6 inH_2O/100ft [31], the minimum airflow rate is 1,900 CFM (as shown in Figure 4.18). However, the minimum friction loss set point was set to have 0.2 inH_2O/100ft [106] and shift the minimum supply air flow rate became 3,200 CFM. This value is even higher than the maximum airflow required. Due to this, indoor temperature would fall below the set point (overcooling) and the relative humidity would increase above the set point during HVAC operating hours. This analysis was in agreement with the collected data shown in Figure 4.19 where the indoor temperature falls below the set point.

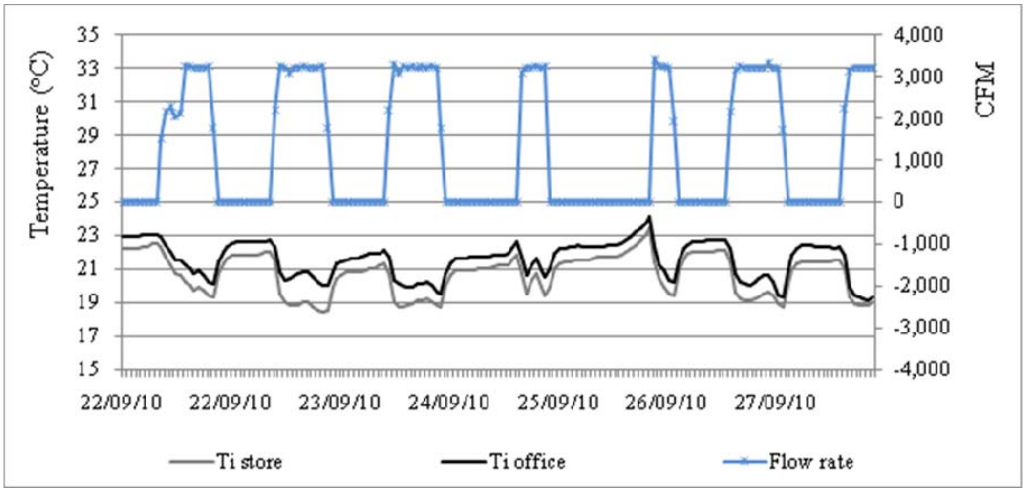


Figure 4.19 Actual indoor temperature and supply air flow rate

4.5 Effects of Various Type of Rooms on The HVAC System Performance

In order to analyze the effect of various type of room on the HVAC system performance, the cooling load characteristic data for three type of rooms: office, classroom and workshop are considered. The cooling load characteristic of these rooms are brought to the design state of the centralized HVAC system (secondary system).

Average *RSHF* values for office, classroom and workshop (as presented in Table4.4) were found to be 0.73, 0.70 and 0.77, respectively. Since the AHU serves these three type of rooms, the smallest *RSHF* value (in the workshop) are selected to determine supply air conditions. The reason of this is to assure that the AHU is able to handle both the sensible and latent cooling load in these three rooms. Using psychrometric chart, the required supply air temperature and humidity are 16.5^{o}C and 0.0117 kg H_2O/kg dry air, respectively.

Table 4.4 Average *RSHF* values and required supply air conditions

Type	Volume (m^3)	RSHF individual	RSHF Design	Supply air conditions	
				T (^{o}C)	Humidity ($kgH_2O/kgdryair$)
Workshop	637.84	0.73	0.73	16.5	0.0117
Classroom	227.8	0.70			
Office	321.32	0.75			

After determining supply air conditions, supply air flow rate needed were calculated based on cooling load in each room using Equations 3.7 and 3.8. In office room, the range of airflow required is from 152 to 1,119 CFM while in workshop, the airflow is from 82.65 to 2,180 CFM. In classroom, the range was found to be from 116 to 2,656 CFM which mean that the minimum airflow required was only 4.37% of the maximum airflow required. Since These rooms were served by single AHU, therefore it should be able to provide airflow rate in the range of 351 – 5,955 CFM. These results were summarized in Table 4.5.

Using equal friction method, the duct was then sized according to the maximum airflow required. Using friction chart for round duct and the recommended friction loss, the air flow rate of the system is in range of 2,100 – 5,955 CFM. It is evident that the minimum CFM is much higher than the calculated value i.e 351 CFM. Since the number of hours on which the airflow required below 2,100 CFM was up to 2,522hours in a year, overcooling would exist on 84% of total HVAC operation hours. Design airflow rate and the duct size were presented in Table 4.5.

Table 4.5 Design airflow rate and duct size

RSHF Design	RSHF individual	Type	Calculated CFM		CFM required		CFM Design		Duct	Friction
			Max.	Min.	Max.	Min.	Max.	Min.	Dia. (inch)	in.wg/100ft
0.70	0.73	Workshop	2,180	82.65	5,955	351	5,955	2,100	19	0.08 - 0.6
	0.70	Classroom	2,656	116						
	0.77	Office	1,119	152						

4.6 Energy and Cost Saving Potential by The Adaptive Cooling Technique

An adaptive cooling technique (as explained in the previous chapter) has been developed as a solution to effectively handle the cooling load from various type of room and to reduce the overcooling. Adaptive comfort temperatures (in Equations 3.30 and 3.33) are used to control the cooling load in the building. These temperatures were considered as indoor set point temperatures in the building when the building is occupied or unoccupied. Comparison of transient cooling load for the three types of rooms between current HVAC system and proposed system are discussed in the next paragraphs.

In the workshop, the comparison results (as presented in Figures 4.20 and 4.21) showed that proposed algorithm reduced both the latent and sensible cooling load. The reductions are mainly due to higher indoor temperature set point during unoccupied hours that decreased the relative humidity as well. The peak cooling load shown in the graph was the time when the workshop was used for practical session. It implied that the workshop was used once in the morning and left empty after the session. During peak period, the maximum of latent and sensible cooling load using the proposed system was found higher than that of the current system. The reason of this was because the system need additional energy to shift indoor relative humidity and air temperature from $T_{c,\text{unoccupied}}$ (from Equation 3.30) to $T_{c,\text{occupied}}$ (from Equation 3.33) and to remove heat gain from the internal and external cooling load. Breakdown cooling load of the workshop using both current and proposed systems in a year are presented in Figure 4.22.

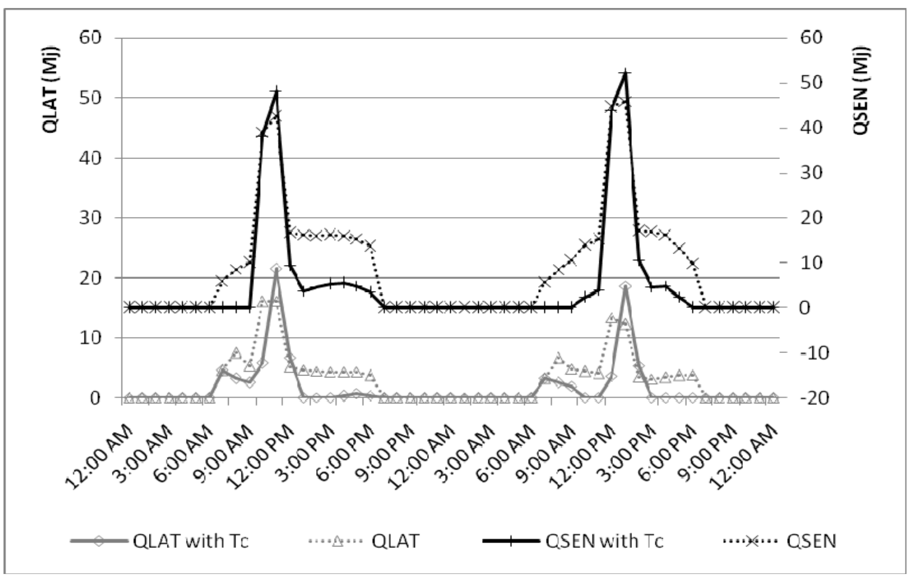


Figure 4.20 A sample of hourly cooling load characteristic in the workshop during peak period

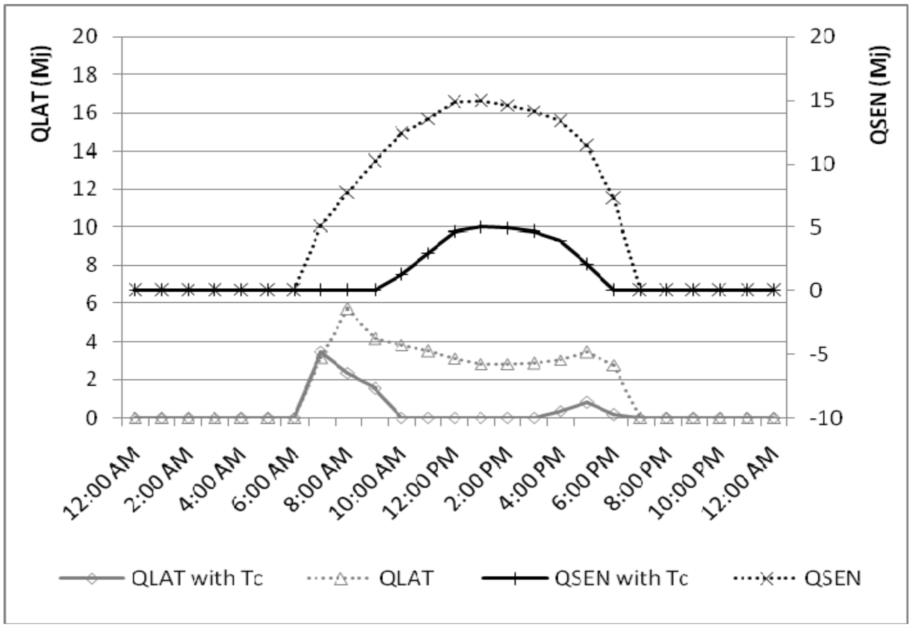


Figure 4.21 A sample of hourly cooling load characteristic in the workshop during off-peak period

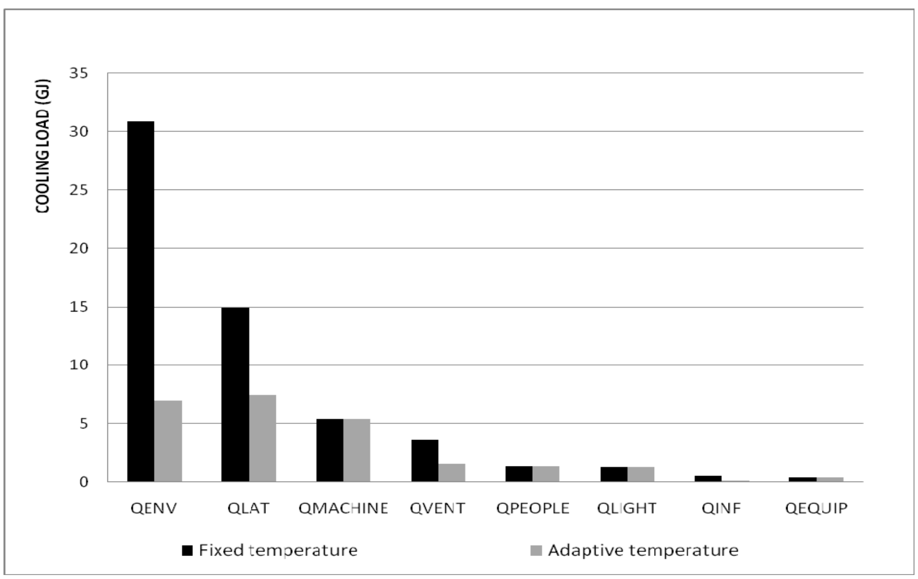


Figure 4.22 Breakdown of yearly total cooling load in the workshop

From Figure 4.22, it can be seen that heat gain from building envelope (*QENV*) and latent heat (*QLAT*) were the first two biggest cooling load components in the workshop. Other heat gains from machine (*QMACHINE*), ventilation (*QVENT*), occupants (*QPEOPLE*), lighting (*QLIGHT*), infiltration (*QINF*) and equipments (*QEQUIP*) gave minor contribution to the cooling loads. In this workshop, the cooling loads breakdown show that *QENV* and *QLAT* as the two major cooling loads are greatly reduced by the proposed system followed by another two minor cooling loads (*QVENT* and *QINF*). Using same occupancy pattern for the room, it was found that the maximum energy saving in terms of cooling load reduction is accounted up to 49.44% for a year compare to the current system.

Cooling load characteristic in the classroom was found to be different from that in the workshop. These different can be seen on the characteristic of the classroom as presented in Figures 4.23 and 4.24. During peak period, the classroom was intermittently used more often than the workshop. This is the reason why the characteristic has more peak than in the workshop. The sensible cooling load using current and proposed system were found almost similar which mean that there is no significant cooling load reduction during peak period. It implied that, the sensible cooling load reduction decrease as the frequency of intermittent usage pattern increase. During off-peak period, there was a big reduction on both sensible and latent cooling loads using the proposed system due to higher indoor temperature set point than the current system.

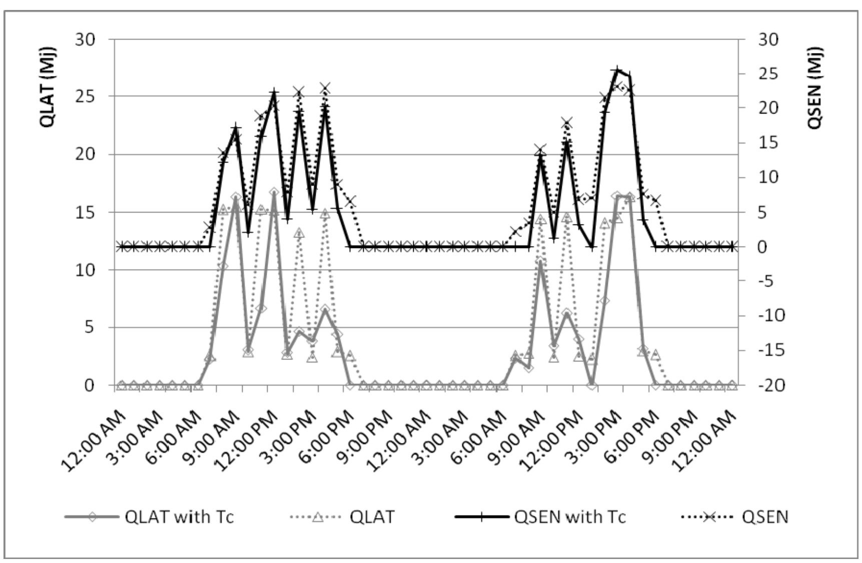


Figure 4.23 A sample of hourly cooling load characteristic in the classroom during occupied period

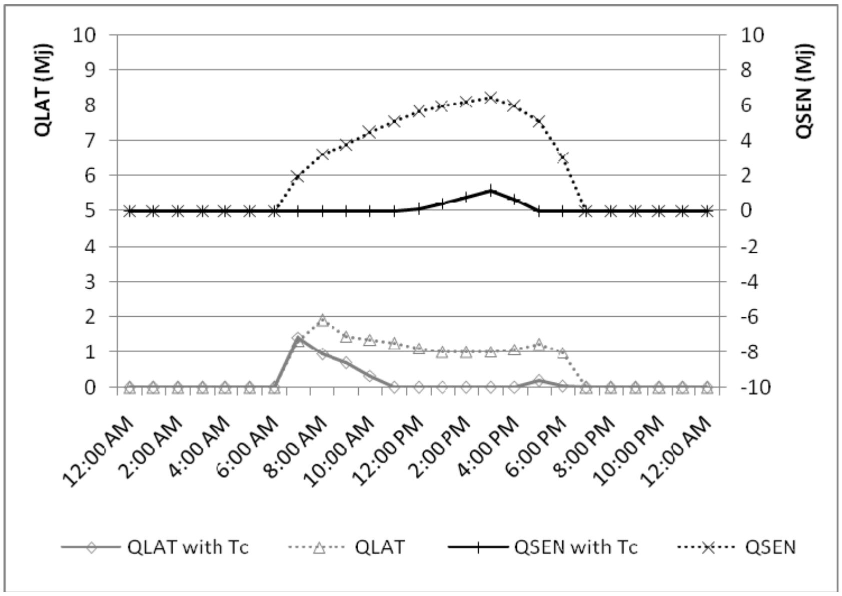


Figure 4.24 A sample of hourly cooling load characteristic in the classroom during unoccupied period

The cooling load reductions on each cooling load components are presented in Figure 4.25. The figure showed that the major cooling loads components were same as in the workshop (*QENV* and *QLAT*). However, *QPEOPLE* was become the third largest cooling load in this classroom. It differs from the workshop where *QMACHINE* was the third largest cooling load. The reason of this is because the classroom has high number of occupants and has no machines. Compare to the current system, the proposed system results in lower cooling load from building envelope, latent heat gain, ventilation and infiltration air. The total annual cooling load reduction was up to 42.79% in this classroom.

Occupancy pattern in the office was continuous during working hours (8 a.m to 5 p.m). During HVAC operation, this room was empty from 7 a.m to 8 a.m, during lunch time (1 p.m until 2 p.m) and after working hours (5 p.m to 7p.m). The cooling load comparison between current and proposed system are presented in Figure 4.26. The figure clearly shows that there is very little latent heat to be removed from the office whereas the sensible heat is considerable and also the energy requirement is more consistent during the office hours as compared to a classroom, where it is intermittent. During occupied hours, the proposed system resulted in higher cooling load than the current system. This is due to the fact that the proposed system has to remove cooling load from internal and external heat gain and additional cooling load to shift indoor temperature from $T_{c,unoccupied}$ (from Equation 3.30) to $T_{c,occupied}$ (from Equation 3.33). However, during unoccupied hours, the proposed system has lower cooling load than the current system.

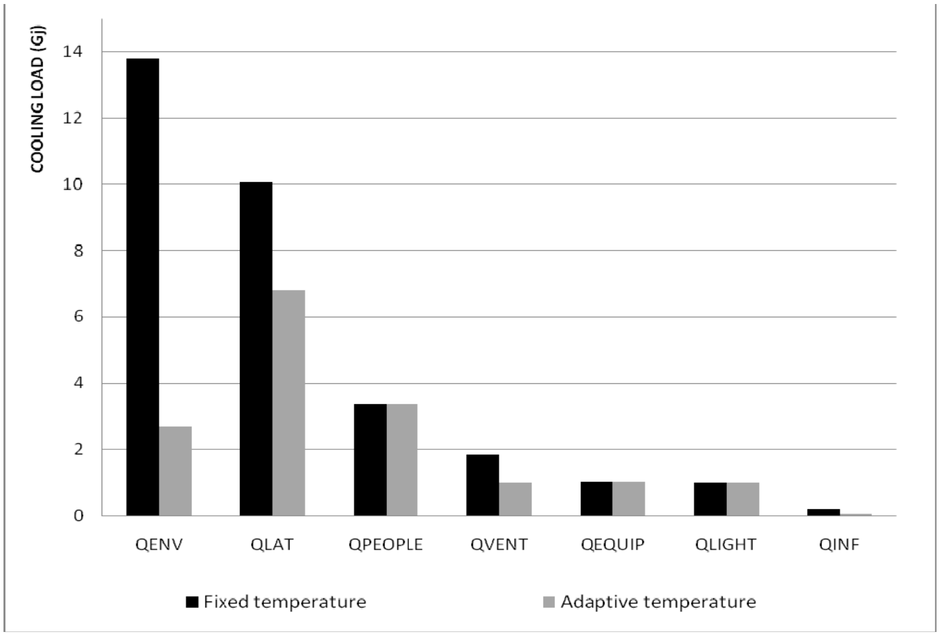


Figure 4.25 Breakdown of yearly total cooling load in the classroom

In the classroom and office room, the algorithm reduces the sensible and latent cooling load on the same way with that in the workshop (due to higher indoor temperature set point). Breakdown cooling load component of the office (as presented in Figure 4.27) shows that *QENV*, *QEQUIP*, *QLAT* and *QLIGHT* were the major cooling load followed by *QPEOPLE*, *QVENT* and *QINF*. Unlike the workshop or the classroom, the proposed system did not reduce greatly the *QENV* and *QLAT*. The main reason if this is because the occupancy pattern in the office is continuous every day whereas the unoccupied hours is minimum. It implies that the cooling load reduction decrease as the unoccupied hours during HVAC operation decreases. Total annual cooling load in the office using the proposed system was found to be 5.4% lower than the current system.

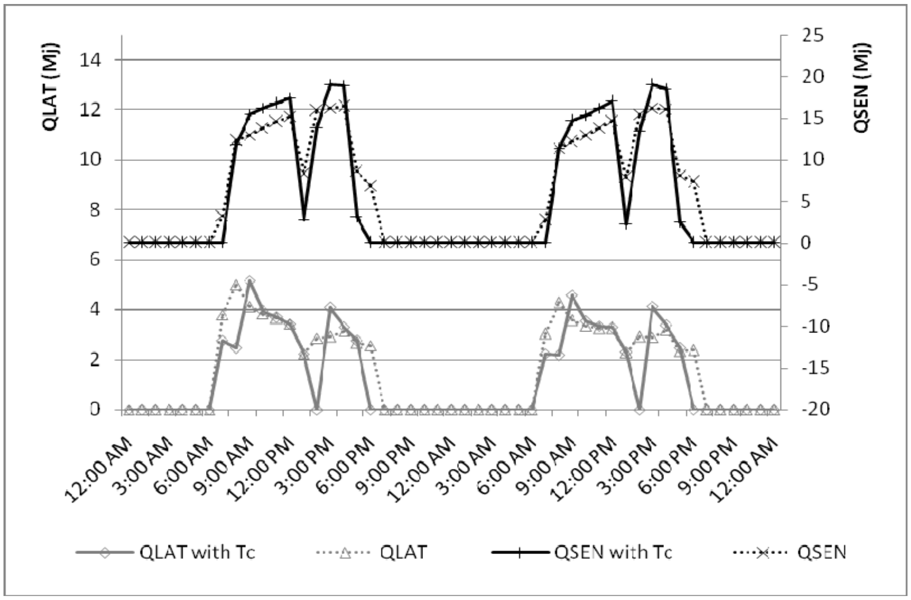


Figure 4.26 A sample of hourly cooling load characteristic in the office

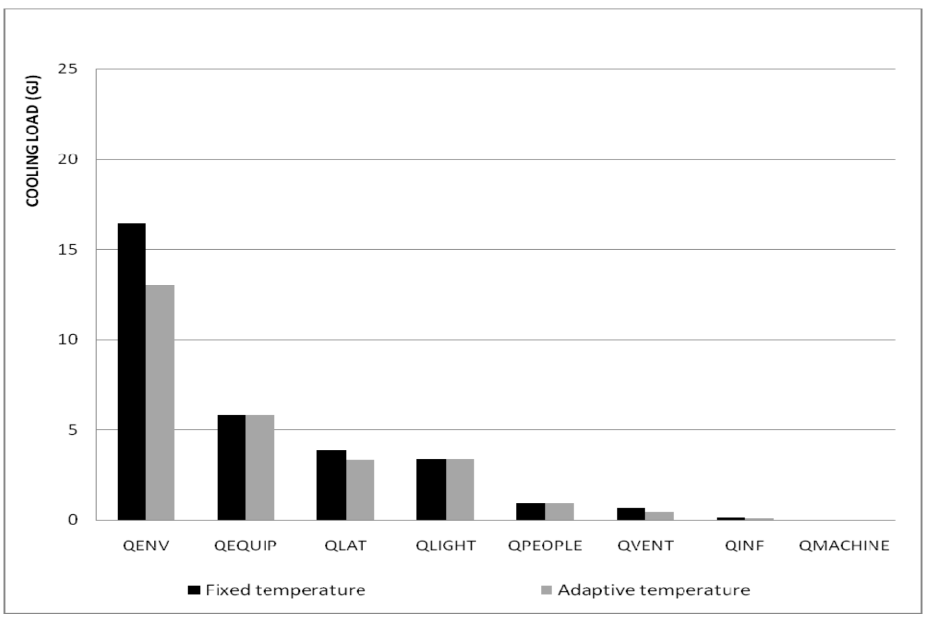


Figure 4.27 Breakdown of yearly total cooling load in the office

Hourly cooling load characteristic of the whole building is presented in Figure4.28. It clearly shows that the proposed system results in lower cooling load and thus consume less energy than the current system. Using the proposed system, the total cooling load reduction for the building was calculated up to 305,150 kWh (44.66%) per year.

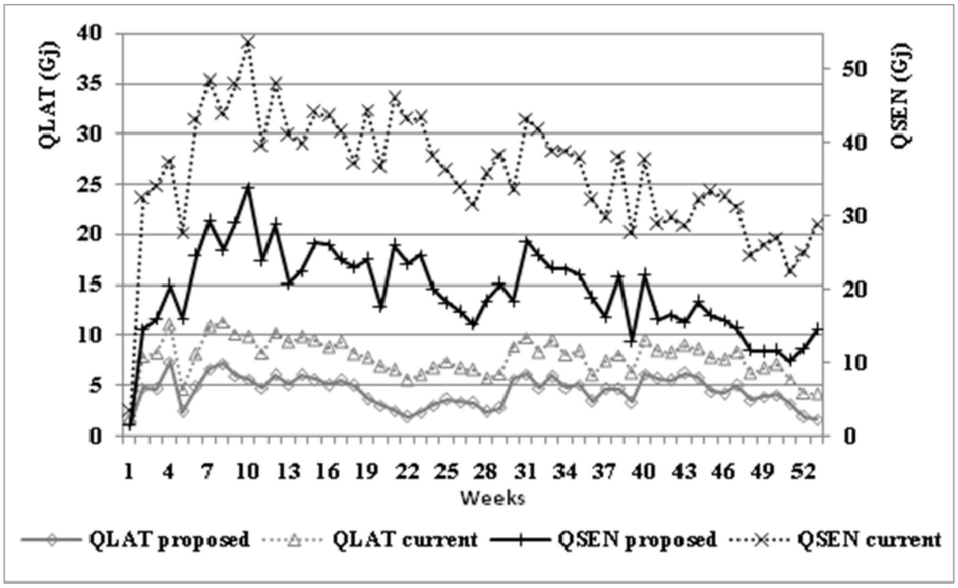


Figure 4.28 Hourly cooling load characteristic of the building (academic building)

Based on the commercial electricity tariff for the year 2009, electricity rates for the building was RM0.38/kWh. Since the cooling load reduction directly decrease energy consumption of the HVAC system, therefore, the reduction lead to RM115,957 cost saving per year. The annual cooling load reduction and cost saving for the building and various types of dwellings are presented in Table 4.6.

Table 4.6 Cooling load reduction and cost saving in a year for the building and various types of dwellings

Type of rooms	Volume (m^3)	Usage Factor (%)	Cooling Load reduction kWh	Cooling Load reduction %	Electricity (RM/kWh)	Saving RM/year
Workshop	637.84	7.55	9,370.58	49.44	0.38	3,561
Classroom	227.8	11.8	4,267.11	42.79		1,622
Office	321.32	56.31	871.80	5.40		331
The building			305,150	44.66		115,957

However, the proposed algorithm would further enlarge the gap between maximum and minimum supply air flow rate required. As discussed before, this gap makes the ducting system unable to cover all range of the air flow rate required and thus, overcooling would exist when the supply air flow rate required is lower than the minimum air flow rate of the ducting system (known as design limitation of the ducting system). This limitation make the proposed system fail to achieve the cooling load reduction as much as expected. Due to this, double fan double duct system is introduced to be coupled with the proposed system.

4.7 Adaptive Cooling Technique With Double Fan Double Duct System

As a solution to the problem discussed in the previous section, double duct double fan system is proposed with wider gap of supply air. Overall impact of this new algorithm would lie in the increase of ducting configurations. The new ducting configuration have to accommodate wider range of supply air required.

The office and the store with single AHU were considered to demonstrate how double fan double duct system increase the range of supply air flow rate. Using proposed system, the average of *RSHF* in these rooms were 0.68 and 0.36, respectively. Since the chiller capacity limit the supply air temperature to be not lower than 16^{o}C, *RSHF* 0.65 was used to determine supply air temperature (16^{o}C) and humidity (0.0115 kg H_2O/kg of dry air). The maximum and minimum airflow rate needed were found to be 1,362 CFM and 85 CFM. Using friction chart and the recommended friction loss (0.08 – 0.6 in.wg per 100 ft), the duct diameter was 16 inch and the maximum and minimum airflow rate were 430 and 1,362 CFM. It should be highlighted that the supply air flow rate needed below 430 CFM was accounted up to 1,613 hours (in a year) or 54% of the total HVAC operation hours. If common supply ducting system which used single supply ducting was applied, the cooling load reduction by the proposed system in these two rooms would be decreased from 55% to 39%.

The second ducting was selected using minimum airflow rate of first duct as the maximum airflow rate. Using the friction chart, the duct diameter was 7 inch and the maximum and minimum rate became 150 - 430 CFM. It can be seen that the minimum flow rate still higher than it was required. However, it would reduce the numbers of overcooling hours from 54% to 43% of the total HVAC operation hours (3,000 h) and the system could achieve cooling load reduction by 52% which was higher than that with single fan single duct system. It means that there was 3% decreased on the expected cooling load reduction due to the design limitation. The comparison of hourly supply air flow rate between single and double duct is presented in Figure 4.29.

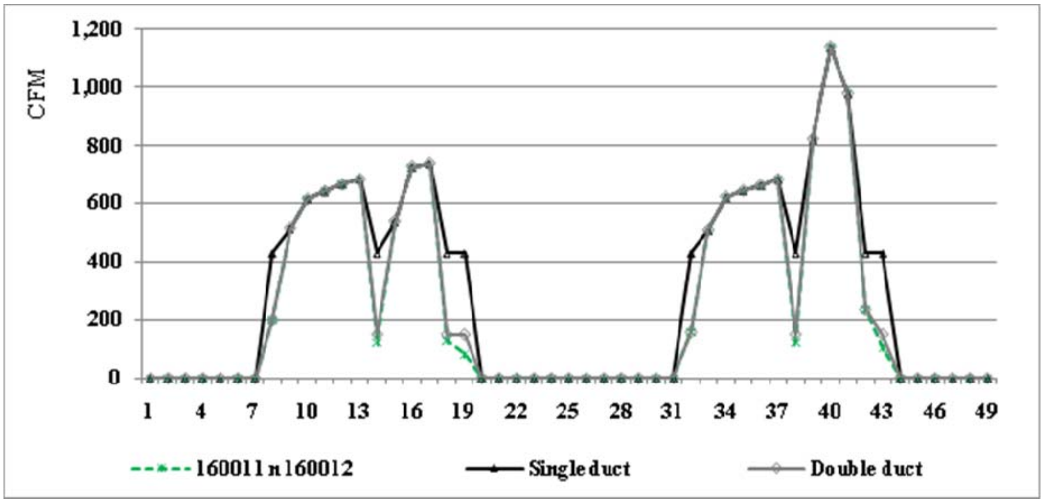


Figure 4.29 A sample of comparison of hourly supply air flow rate between single and double duct

In the workshop and classroom, the airflow rate required were in range of 83 – 3,168 CFM and 83 – 1,974 CFM, respectively. Unlike in the office, the first ducting was determined using the supply air flow rate during occupied hours while the second ducting was determined using the supply air flow rate during unoccupied hours. However, similar to the office, double fan double duct system still unable to cover the whole range of supply air flow rate. As a consequence, in the workshop, the expected cooling load reduction decrease by 8.62% while in the classroom, it decrease by 19%. The percentage of reduction in the classroom was the highest due to low usage factor and considerably small heat gain from building envelope (*QENV*). For the building, dual fan dual duct system decrease 16.12% of the expected cooling load reduction by the adaptive cooling technique and thus, the annual cost saving decrease to only RM97,265. These results are summarized in Table 4.7.

Table 4.7 Airflow capacity of double ducting and percentage of cost saving reduction

		CFM		First duct (CFM)		Second duct (CFM)		%
RSHF	Room	Occupied	Unoccupied	Min.	Max.	Min.	Max.	reduction
0.65	Office	85 - 1,362		430	1,362	150	430	3%
	Store							
	Workshop	1,505 - 3,168	83 - 477	1,150	3,168	170	477	8.62%
	Classroom	626 - 1,974	83 - 445	690	1,974	180	445	19%
Percentage decrease on the expected cooling load reduction in the building								16.12%

4.8 Economic Analysis of The Adaptive Cooling Technique

The proposed system showed energy saving potential for centralized HVAC system to handle intermittent cooling load in the building. In this section, simple economic analysis is performed to know whether the proposed system is economically feasible or not. In order to realize the proposed system, additional supply ducting, fan and occupancy sensors are required to update the existing centralized HVAC system. Dual technology occupancy sensors are used (ultrasonic and passive infrared) to detect any motion inside the room. DT 300 is used for classrooms, workshops, and big offices while W-500A is used for small office [119,120]. Additional relay, modul, and power pack are needed to install the sensors. The prices of these equipments are based on information gathered from local contractor and the suppliers. Installation cost for installing the sensors, wiring from the building to the control room, and controller upgrade are considered according to the local contractors.

The cost to install additional supply duct was based on estimated HVAC budget pricing from an air conditioning specialist (CT mechanical) [121]. The price of additional supply fan was based on industrial centrifugal fans price chart from Canada blower. The additional fan was selected based on the capacity needed which was determined using maximum flow rate of the first supply fan. Additional cost for maintenance of the proposed system is assumed to be RM 10,000/year. The breakdown of the cost was presented in Table 4.8.

Table 4.8 Cost breakdown of the proposed system

No.	Item	Price (RM)	Quantity(s)	Total (RM)
1	Occupancy sensors			
	a. DT 300	473	36	17,028
	b. W-500A	211	42	8,862
2	BZ 150 power pack	91	78	7,098
3	Relay for W-500A	120	42	5,040
4	Humidity sensor	120	78	9,360
5	Wiring from site to control room	27 / meter	1,000	27,000
6	Controller upgrade		5,000	5,000
7	Additional modul	800	39	31,200
8	Sensor's installation	100	78	7,800
9	Fan (1,000 CFM, 3" WG)	1,980	8	15,840
10	Ductwork	15,086	8	120,688
TOTAL				254,916

Simulation net cash flow of the proposed system is performed to find the payback period. 10% of rate of return, 2.75% of interest rate, and 25% corporate tax were considered on this analysis. The corporate tax was included because minimum attractive rate of return (MARR) as profit from the project is taken into account. The result showed that the proposed system needs less than 5 years to cover the cost and also recover the funds it has invested to install the proposed system. Simulation of the economic analysis is presented in table 4.9. From the table, it can be seen that the proposed system give profit from the invested money during payback period.

To determine feasibility of the proposed system, net present worth (NPW) analysis is performed. The proposed system is assumed to have 10 years of service lifetime. According to the amount of invested money at the beginning of the project and yearly maintenance fee, the present worth (PW) of outflows were found to be RM354,916. On the other hand, the PW of inflows according to ten years of service lifetime and expected MARR was found to be RM531,320. From these two value of PW, the NPW was found to be RM176,404. Since NPW is greater than 0, the proposed system is economically acceptable.

Table 4.9 Simulation net cash flows of the proposed system

		Outflows						
Year	Inflows	Initial Cost	Maintenance	I (2.75%)	MARR (10%)	Tax (25%)	Net cash	ROI
0	-	(254,916)	-	-	-	-	(254,916)	-
1	97,265		(10,000)	(7,010)	(25,492)	(6,373)	(206,526)	(0.81)
2	97,265		(10,000)	(5,679)	(20,653)	(5,163)	(150,756)	(0.73)
3	97,265		(10,000)	(4,146)	(15,076)	(3,769)	(86,481)	(0.57)
4	97,265		(10,000)	(2,378)	(8,648)	(1,730)	(11,972)	(0.14)
5	97,265		(10,000)	(329)	(1,197)	(239)	73,527	**6.14**

Summary

The cooling load characteristics of the current HVAC system have been discussed. The results showed that the simulation results follow closely the experimental results which mean that the model can behave as the real without major error. It was found that the effect of the design values on current HVAC system resulted in overcooling for 5.7% of total HVAC operation hours in the office and the store. Effect of various types of room on the HVAC performance showed worst case where overcooling occurred for 84% of total HVAC operation hours. Energy saving potential of the adaptive cooling technique in terms of cooling load reduction has been explained in detail. It was found that the expected cooling load reduction using this technique cannot be achieved 100% due to wide range of supply air flow rate. However, dual fan dual duct system improves the performance of the adaptive cooling technique and gives annual cooling load reduction as much as 255,960 kWh which equal to annual cost saving RM97,265. The cost saving resulted in payback period (discounted) less than 5 years and has positive NPW (economically acceptable).