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A PRELIMINARY STUDY ON HVAC SYSTEMS AND THERMAL COMFORT IN A TROPICAL UNIVERSITY BUILDING IN MALAYSIA

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ABSTRACT

This study aims to determine whether the Level 2 of Balai Ungku Aziz, a faculty building of the Oral Biology Department in the Dental Faculty of University of Malaya, exhibits the Sick Building Syndrome (SBS) and meet the standards for Indoor Air Quality as specified by ASHRAE. It is significant to conduct this research because no similar investigation has been carried out for a tropical building in Malaysia. The study covers only level 2 of the building, where the computer lab, dean's office and the library are located. Among the topics studied are the overview of the air conditioning system used in the building, load calculations, capacity measurements and other analyses that will provide Mechanical and Electrical (M&E) designers a better understanding of the air conditioning systems used in a typical tropical building. From the research conducted, it was shown, preliminary, that the Balai Ungku Aziz did not exhibit SBS as the ventilation was sufficiently high and thus, CO_2 concentration was low. But further investigation for CO, Formaldehyde and TVOC studies needed to confirm SBS effect on occupants as we had limited number of IAQ and IEQ equipment in hand during studies. The calculations have brought the measured specifications of the air conditioning system closer to the design specifications.

Based on the result obtained, the room temperature is slightly cooler than the thermal comfort zone recommended by ASHRAE. Therefore, it is suggested that some adjustments on the HVAC design, including the use of reheat coils, should be done on the system to meet the standard. This can lead to better energy efficiency and cost effectiveness.

Keywords: Thermal Comfort; HVAC Systems; Tropical Building; Humidity; Fresh Air Supply; Sick Building Syndrome (SBS)

NOMENCLATURE

ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc
A_{duct}	Duct area
ACH	Air change per hour
ACM	Air change method
AHU	Air Handling Unit
BF	Ballast factor
BUA	Balai Ungku Aziz
CFM	Cubic feet per minute
CLTD	Cooling Load Temperature Difference
CO_2	Carbon Dioxide
c_p	Specific heat capacity (kJ/kg.K)
DBT	Dry Bulb Temperature
EWT	Entering Water Temperature
GLF	Glass load factor
h	Specific enthalpy (kJ/kg)
H_f	Heat of fusion (kJ/kg)
HVAC	Heating, Ventilating & Air Conditioning
IAQ	Indoor Air Quality
IEQ	Indoor Environmental Quality
ITS	Ice Thermal Storage
k	Thermal conductivity (W/m.K)
LWT	Leaving Water Temperature
M	Mix air
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat transfer rate (W/s)
Q_l	Latent cooling load
Q_s	Sensible cooling load
R	Resistance factor, for cooling load calculation
%RH	Humidity Ratio percentage
SBS	Sick Building Symptom
T	Temperature (K)
TNB	Tenaga Nasional Berhad
x_{ice}	Ice fraction
$\dot{V}$	Diffuser Volumetric Flowrate
$Y_{air@DBT,\%RH}$	Specific Volume at certain DBT and %RH
υ_{mean}	Mean Velocity
υ_{max}	Maximum Velocity

Greek symbols

ρ	Density (kg/m3)
Δ	Increase
α	Heat transfer coefficient (W/m^2.K)
τ	Shear stress (Pa)
$\dot{\gamma}$	Shear rate (s^{-1})
μ	Viscosity (Pa.s)

Subscript

cf	Carrier fluid
ice	Ice
in	Inlet
m	Mean value
ma	Mix air
oa	Outdoor Air
out	Outlet
ra	Return air
sl	Ice slurry
sa	Supply Air

1 INTRODUCTION

There has been an increasing concern on thermal comfort and indoor air quality of buildings in the recent decades. While thermal comfort is mainly related to the comfort level of occupants, indoor air quality study is all about keeping gaseous and particulate contaminants below some acceptable level in the indoor environment to ensure the health of the occupants. Because of the thermal comfort and indoor air quality studies are so crucial in ensuring the occupants' well being, substantial research has been done in this area.

For example, Sekhar *et al.* (2002) investigated the ventilation characteristics of an air-conditioned office building in Singapore. The results indicated that the concentration levels of indoor air pollutants were within reasonable limits. In another research, Cheong *et al.* (2003) carried out a thermal comfort study of an air-conditioned lecture theatre in Singapore and found that the existing ventilation system failed to cope with the variation of occupancy load during the peak period. There are number of literatures on thermal comfort and indoor air quality studies in Singapore.

Kavgic *et al.* (2008) analysed the level of indoor air quality and thermal comfort in a theatre in Belgrade. He identified that over-ventilation in the theatre caused excessive energy consumption and the arrangement of the ventilation system resulted in complaints of cold discomfort. Increasing awareness on the indoor air quality issue has given rise to terminologies called Sick Building Syndrome (SBS). Buildings with an unusual number of occupants having physical problem is described as having SBS. SBS has proved to affect the health of occupants adversely as Gupta (2007) revealed in his research that headache, lethargy and dryness in body mucous are the main symptoms prevailing in a SBS building. Gupta (2007) further attributed these sicknesses to the concentration of CO_2. SBS is more apparent with an increased in the CO_2 concentration.

Malaysia, as a country located near to the Equator, is having a tropical climate. The air is humid and hot throughout the years due to its unique geographical location. Although researchers in Malaysia have long identified the thermal comfort and indoor air quality issues of buildings, there has not been any published study on this area yet in Malaysian buildings. Therefore, it is particularly significant to perform a research on thermal comfort and indoor air quality of the buildings in Malaysia, especially institutional buildings, such as Balai Ungku Aziz of University Malaya, where students spend much time in.

2 BALAI UNGKU AZIZ

Balai Ungku Aziz located south of 1st residential college of University Malaya, is the extension of the Faculty of Dentistry and is used primarily by the 1st and 2nd year dentistry students. The building consists of 3 levels, which are the lower ground (LG), level 1 (L1) and level 2 (L2). Examination hall, tutorial rooms, religious rooms and canteen are located in LG, research laboratory, lecture halls, and lecturer's rooms are located in L1, while the library, computer room and the administration office is located in L2. The Balai Ungku Aziz is primarily air-conditioned by central air conditioning systems, which is the hydronics system.

2.1 The HVAC system

The study covered only the level 2 (L2) of the Balai Ungku Aziz, which possesses the largest air handlers, cooling capacity and load, ductworks, and thermal zones. Figure 1 shows the HVAC hydronics system used in level 2 of the building. The chiller plant is located within 50 meters northwest from the center of the Balai Ungku Aziz. The plant stores 3 units of Carrier 30HXC190 chillers, 6 units of Aeroflo AE-80-32-D water pumps and power boards.

The chillers are used to produce chilled water at 6.7°C for the use of air handlers in the main building. Three of the Aeroflo AE-80-32-D water pumps are used to pump the chilled water through the chiller evaporator and subsequently to the air handlers in the main building. The other 3 pumps are used to pump the condenser water into the chiller condenser and subsequently to the cooling towers next to the plant. The remaining equipment in the chiller plant is the main power board, where the switches for all the chillers, pumps and cooling tower are contained.

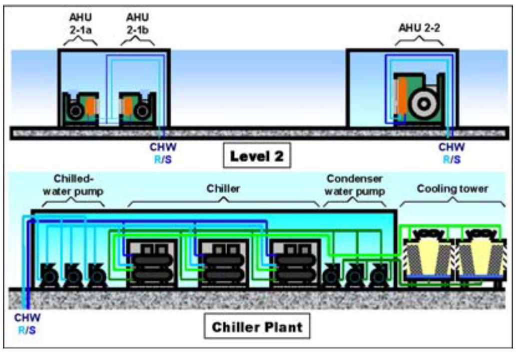

Figure 1: Chiller plant diagram

There are 2 units of Nihon CTA225UN cooling towers at the back of the chiller plant, where the condenser water is pumped through the chiller condenser from the plant and cooled in the tower.

The HVAC equipments in the chiller plant are described above, the chilled water produced by the chilled are pumped throughout the building to the air-handling units in the main building. There are a total of 11 AHUs by Carrier in the main building. Of the 11 AHUs in Balai Ungku Aziz, 3 of them are in level 2. The computer room is air conditioned by a model 39GH1018, while the library and administration office shared two AHUs, which are the model 39GH2127 and model 39GH1118. Table 1 shows the design specifications of these AHUs.

Table 1: AHU design specifications

Model	39GH1018	39GH1118	39GH2127
Cooling capacity	61.5 kW	74.7 kW	230.0 kW
Air flow rate	6750 cfm	7920 cfm	23200 cfm
EWT	6.7	6.7	6.7
LWT	12.2	12.2	12.2
Water flow rate	2.7 l/s	3.3 l/s	9.9 l/s

3 BACKGROUND THEORY

3.1 Supply air volumetric flow rate calculation

It is anticipated that supply air flows out from the round duct with a certain profile, and it is assumed to be parabolic (as shown in Figure 1) with the following relation:

$$y = ax^2 + b \qquad (1)$$

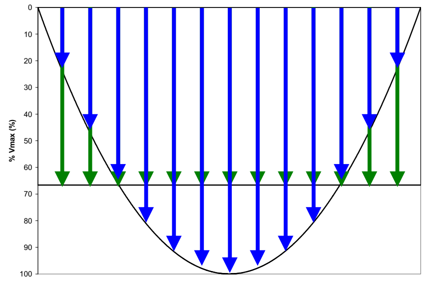

Figure 1 Supply Air Flow Profile

It is found that the maximum velocity in the profile is related to the mean velocity of the air flow through the following relation:

$$v_{mean} = \frac{2}{3} c v_{\max} \qquad (2)$$

Where, c is a constant.

The constant c can be determined empirically, by comparing the air flow rate of the centrifugal fan with the total air flow calculated from the site work results using the equations in this section. The air volumetric flow rate of each diffuser is then calculated using the equation:

$$\dot{V}_{diffuser} = A_{duct} v_{mean} \qquad (3)$$

3.2 Supply air cooling capacity calculation

The supply air cooling capacity is calculated using the supply air conditions (dry bulb temperature, relative humidity and the calculated volumetric flow rate) and space air conditions.

The mass flow rate of the supply air entering the space can be determined by dividing the volumetric flow rate by the specific volume of air can be determined using the dry bulb temperature (DBT) and Relative Humidity (RH). The following equation is used to calculate the mass flow rate of the supply air for each diffuser to the space:

$$\dot{m} = \frac{\dot{V}_{diffuser}}{\upsilon_{air\,@\,DBT,\%RH}} \qquad (4)$$

The cooling capacity from the supply air from each diffuser can be calculated by multiplying the mass flow rate of supply air with the enthalpy difference between the supply and space air. The following equation is used to calculate the cooling capacity of supply air for each diffuser to the space:

$$\dot{Q} = \dot{m}\left(h_{SA} - h_{RA}\right) \qquad (5)$$

The total part load cooling capacity could be found by summing the cooling capacity of each diffuser. This is conducted due to the difference in air flow in each diffuser and the unknown heat gain variations along the duct. That is the rationale why this method is used instead of using the average variables to determine the total cooling capacity.

The details regarding the calculation of cooling capacity of each diffuser could be obtained from Tables 1A to Table 6A in the Appendix and references (Foo *et al,* 2007; Pita, 2002).

3.3 Space cooling load calculation

Cooling load, which is the amount of heat that must be removed from a space or a building to maintain comfortable conditions, can be calculated by adding the effect of different items that may affect the conditions of air inside a building or room. Those items are as follows:

3.3.1 Cooling load due to heat transfer through external walls:

The cooling load caused by this part can be calculated using the equation:

$$Q = U \times A \times CLTD_c \qquad (4)$$

Where:

Q	=	cooling load (Btu/hr)
U	=	overall heat transfer coefficient (Btu/hr-ft^2-F)
A	=	Area of the wall or Roof (ft^2)
CLTDc	=	Corrected cooling load temp. difference (F)

The above CLTD is not the actual temperature difference between the outdoor and indoor air. It is the modified value that accounted for the heat storage due to the time lag.

The $CLTD_c$ was obtained from tables that list CLTD for different types of walls roofs and floors based on the following conditions:

1) indoor temperature is 78°F (25.5°C)
2) Outdoor average temperature on the design day is 85°F (29.4°C) DBT.
3) Date is July 21st.
4) Location is 40°N latitude.

If the actual conditions differ from any of the above, the CLTD must be corrected as follows:

$$CLTD_c = CLTD + LM + (78 - t_R) + (t_a - 85) \qquad (5)$$

$CLTD_c$	=	corrected value for CLTD, F
LM	=	correction for latitude and month
t_R	=	room temperature, F
t_a	=	Average outside temperature. Ta = to - (DR/2)

The U value for the walls was calculated using information from ASHRAE Handbook (2007) for:

A0	Outside surface resistance	R= 0.17
C4	4 in. Common brick	R= 0.79
E0	Inside surface resistance	R= 0.69

The total U value for the walls is 0.55 Btu/F.hr.ft^2 (0.312 W/m^2-C)

For the floor, the U value was taken from reference (Pita, 2002) for concrete deck, no insulation, which is 0.59 Btu/hr.ft^2.F (0.335 W/m^2-C).

For the roof, the U value was taken from the same table, which is 0.18 Btu/hr.ft^2.F (0.102 W/m^2-C) for roof-ceiling 2.5 in. wood deck (no insulation).

CLTD Values were taken from ASHRAE Handbook 2007, in reference (ASHRAE Handbook, 2007)

3.3.2 Cooling load due to heat transfer through glass

The sensible cooling load due to heat gains through glass is found by using glass load factor (GLF). The GLF values accounts for both solar and radiation and conduction through glass. The glass sensible cooling load is determined using the equation:

$$Q = A \times GLF \qquad (6)$$

Where,

GLF	=	Glass load factor BTU/hr-ft^2
A	=	area of glass ft^2
Q	=	sensible cooling load due to heat gain through glass

GLF values for the glass could be found in reference (ASHRAE Handbook, 2007), for regular single glass with no inside shading and outdoor design temperature 95°F (35°C).

3.3.3 Cooling load due to occupancy and appliances

Cooling load due to occupancy and appliances was obtained directly from reference (ASHRAE Handbook, 2007) for people doing very light work. As for refrigerators, the cooling load may again be found from reference (ASHRAE Handbook, 2007).

3.3.4 Cooling load due to equipments

Cooling load due to equipments used may be obtained directly from ASHRAE Handbook (2007). Again, for the photocopier and personal computer brand (2), the heat gain could be acquired from reference (ASHRAE

Handbook, 2007) and converting the values from Watt to Btu.

3.3.5 Cooling load due to lighting
Equation (7) is used to determine the cooling load due to lighting as follows:

$$Q = 3.412 \times W \times BF \quad (7)$$

The Ballast Factor (BF) accounts for heat losses in the ballast in fluorescent lamps, a typical value of 1.25 may be taken for normal florescent lamps. The 3.412 is the conversion factor to convert from Watts to Btu/hr.

3.3.6 Infiltration and ventilation
The procedure used to determine the infiltration effect on cooling load is the Air Change Method (ACM), which depends on the number of air changes per hour (ACH) in a room caused by the infiltration. The number of ACH usually is determined based on experience and testing. The suggested values range from 0.5 to 2.0 for building ranging from "tight" to "loose" construction.

The following equation can be used to find the CFM

$$CFM = ACH \times \frac{v}{60} \quad (8)$$

Where,
v = volume of space to conditioned

Then sensible and latent cooling loads can be found using the following equations:

$$\begin{aligned} Q_s &= 1.1CFM(\Delta T) \\ Q_l &= 0.68CFM(\Delta W) \end{aligned} \quad (9a \ \& \ 9b)$$

For ventilation, both equations above could be used to calculate the total heat gain due to ventilation. However, the volumetric flow rate in CFM must be estimated initially so as to be used in equations 8 and 9.

The ventilation rate in CFM per person as recommended in reference (Pita, 2002) for offices MUST NOT BE LESS THAN 5CFM/person while the recommended value is 15 CFM/person. For other applications, the ventilation requirements are listed in CFM/ft^2 as shown in references (Pita, 2002; ANSI/ASJRAE Std, 2007).

The details on the calculation of cooling load of each diffuser could be obtained from Table 7A and Table 8A in the Appendix and reference (Foo *et al*, 2007)].

3.4 Ventilation
The outdoor air intakes were situated high up near the ceiling of the AHU room and the ladder cannot reach the spot. Therefore, an alternative approach was taken to obtain the readings by graphical method of the psychometric chart.

The method is to use the measurements of mixed air and return air conditions. By using equation (10):

$$M_{OA} + M_{RA} = M_{MA} \quad (10)$$

the air flow of the outdoor air readings could be obtained by applying the graphical method. By plotting the points for M_{MA} and M_{RA}, and drawing a straight line on the chart, we can pinpoint the point for OA conditions such as shown in the figure 2 below

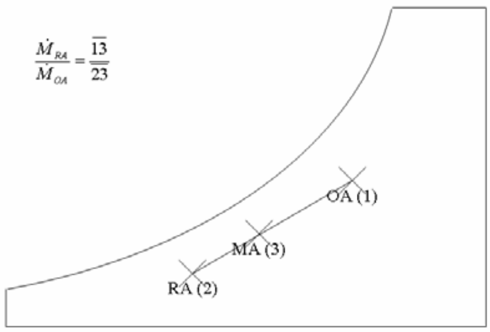

Figure 2: Graphical Method for Outdoor Air Intake

Taking the length of the conditions we can achieve M_{OA} and then the air flow for each AHU unit to seek the conditions for the library which consists of 3 parts, namely, library office and computer room.

4 FIELD MEASUREMENTS

4.1 Supply air condition measurement
Diffusers are detached from the ceiling diffuser frame. The dampers are left untouched as there may be previous balancing calibration conducted for the air flow. The thermal hygrometer is used to measure the dry bulb temperature, relative humidity, as well as the air velocity at the centerline of the round duct and the CO_2 analyzer is used to measure the concentration of carbon dioxide in the supply air. The readings from these instruments are recorded and tabulated for further analysis.

4.2 Space air conditions measurement
Space conditions are measured at the level of one meter below the diffuser, with the diffuser intact on the ceiling diffuser frame. The instruments used are the same as the instruments used in the supply air condition measurements. The readings from the instruments are recorded and tabulated for further analysis.

5 ANALYSIS ON DATA

5.1 Cooling capacity

There are 18, 22 and 58 diffusers from AHU 2-1a, AHU 2-1b and AHU 2-2 respectively. Table 2 shows the comparison of the cooling capacity of each AHU.

Table 2: Total cooling capacity for each AHU

	Total Cooling Capacity, Q (kW)
AHU 2-1a	26.014
AHU 2-1b	47.745
AHU 2-2	46.147

It can be noted that AHU 2-1b has the highest total cooling capacity, followed by AHU 2-2 and AHU 2-1a.

5.2 Ventilation

The ASHRAE standard (ANSI/ASHRAE Std, 2004) suggests the need of 8L/s per person for outdoor air requirement. Table 3 shows the comparison between the total the measured outdoor air intake and the total suggested outdoor air requirement.

Table 3: Comparison between the total measured and suggested air intake

Facility	Measured Outdoor Air Intake	Suggested Outdoor Air Requirement
Library (150 persons)	1530 L/s	8 L/s × 150 = 1200 L/s
Office (6 persons)	310 L/s	8 L/s × 6 = 48 L/s
Computer room (61 persons)	2890 L/s	8 L/s × 61 = 488 L/s

The measured outdoor air intake is sufficient to meet the suggested outdoor air requirement.

6 DISCUSSIONS

6.1 Cooling load versus cooling capacity

It is found in this project that the total cooling load under full load condition for the level 2 of the Balai Ungku Aziz is 100 tons of refrigeration, while part load condition gives a total cooling load of 55 tons of refrigeration. The definition of part load in this context is the total cooling load under the conditions when the measurements were made. The total calculated cooling load is in close agreement with the design cooling load, which is 104 tons of refrigeration. It is calculated that the system is at 34% of the full load when the site work is being carried out.

On the other hand, the measured cooling capacity is only approximately 34 tons of refrigeration. The causes of such deviations are discussed as follows:

1. The site work is done separately on 2 days. The conditions are ever changing with time. Thus, assuming an average value for the outdoor conditions proves to give rise to a significant error in the results.
2. The assumption of uniform U value across the ceiling is rendered inaccurate as there are portions of the plenum have been used as a dumping ground for the extra ceiling panels.
3. The fact that the equipments in the space do not continuously produce heat to the space, where there are energy saving features nowadays.
4. The use of inappropriate assumptions that may affect the cooling load calculations significantly.
5. The degree of accurateness that the methodology of the study could offer.

Hence, in order to produce better, more accurate results, the following steps shall be considered:

1. The measurements should be taken simultaneously, as to minimize the effect of change in the outdoor conditions. This can be done by mounting thermocouples at each diffuser and collects the data using a data logger. However, this method may not be economically feasible as the area of the library is relatively large, and the instrument price and effectiveness over the area of the site may be an issue.
2. The use of more sensitive instruments may cut down the measurement time, as part of the effort of minimizing the time difference between the measurements. A pair of RTD with an air sampler, for dry and wet bulb measurements should be able to give readings at a faster pace
3. A more thorough research methodology is to be devised to yield more accurate results.

6.2 Ventilation

From the calculated outdoor air supplied to the space, it can be concluded that the amount of ventilation is overly sufficient. This may increase the cooling load of the air handler, as there is more than required warm fresh air passing through the cooling coil. However, as far as the indoor air quality is concerned, it is good as more fresh air is supplied to dilute the contaminants. It is also found that the CO_2 concentration is below the ANSI/ASHRAE standard requirements (ANSI/ASHRAE Std, 2004).

6.3 Supply air distribution

Table 4 shows the calculated supply air distribution for all AHU units. It is noticeable that the total air flow for each air handlers are slightly lower than the total air flow stated in the initial testing and commissioning report; obtained from the O&M manual. This drop in air flow rate may be due to the assumptions, deterioration of fan or motor efficiency over time, leakages, increased in coil pressure drop due to coil fouling and the variation in testing conditions.

Table 4: Calculated supply air distribution for all AHU units

Unit		AHU2-1a	AHU2-1b	AHU2-2
Zone		C	B	A
Total air flow measured by MKP T&C (cfm)		11480	9085	25380
Total corrected air flow measured during site work (cfm)		10594	8243	24010
Mean measured air flow (cfm)		589	375	414
Quantity of diffusers supplying air at the given range	$\dot{V} \leq 50\% \dot{V}_{mean}$	0	1	4
	$50\% \dot{V}_{mean} < \dot{V} \leq 75\% \dot{V}_{mean}$	1	2	14
	$75\% \dot{V}_{mean} < \dot{V} \leq 125\% \dot{V}_{mean}$	16	13	26
	$125\% \dot{V}_{mean} < \dot{V} \leq 150\% \dot{V}_{mean}$	1	5	4
	$175\% \dot{V}_{mean} < \dot{V}$	0	1	10

From the analysis on the distribution of airflows around the average airflow from the diffusers, it is noticeable that there are a number of diffusers having less than half of the average airflow. These diffusers includes: B7, A17, A44, A51 and A52 as shown in Figure 3.

Figure 3: Problematic diffusers in Balai Ungku Aziz

Diffuser B7 and A17 are similar in position in the duct, and the reason of decreasing airflow is most probably due to the position of damper. Diffuser A44, A51 and A52 has lower than average air flow because of the pressure losses with the number of bends, and poor construction of the duct as shown in Figure 4. There are some deviations in terms of ducting design with the constructed ducting as shown in Figure 3. The branched-out duct is extended in an "S-bend" manner that increases the pressure drop, which subsequently results in the drop in the airflow.

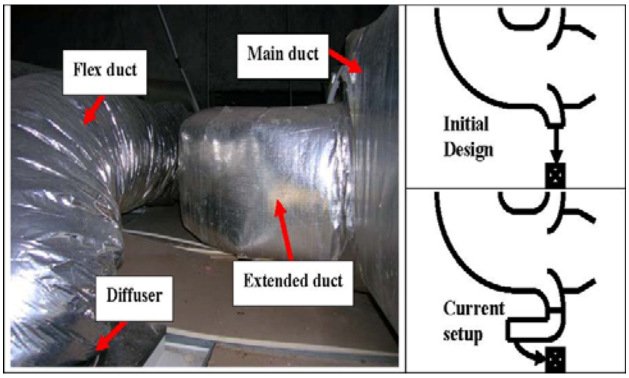

Figure 4: Problematic diffuser A52

Note that there are several remedial steps that could be taken to overcome this problem. The "S-bend" problem could be solved through minor renovation, where the ducting layout for diffuser A52 is reverted back to its initial design for less pressure losses. Air balancing could also be performed to ensure that the air is distributed evenly across the space, without any "dead spots".

7 CONCLUSIONS

Generally and preliminary, the library floor at Balai Ungku Aziz, does not exhibit sick building syndrome (SBS) as the ventilation is sufficiently high and CO_2 concentration is low. But in the other hand a thorough inspection with more parameters such as CO, Formaldehyde Respirable particulars and TVOC should be taken in account as at the time of investigation we have limited number of equipment to study SBS, IAQ and IEQ. The assumptions used in the calculations have brought the measured specifications of the air conditioning system closer to the design specifications. The space-cooling load under full load conditions matched the design specifications. However, the cooling capacity calculated from the measured site work data is lower than the calculated part-load cooling load. The discrepancies may be originated from the inappropriate assumptions used during the calculations of the cooling load.

It is also found that the room temperature in the range of 17.6°C to 20.9°C at level 2 is significantly cooler than the thermal comfort zone recommended by ASHRAE [8,9,10], which is 24°C, 50% RH. However, through observation, the majority of the occupants are not wearing additional clothing. Hence, it is concluded subjectively that the thermal comfort level in level 2 of the building is acceptable for the occupants of the Balai Ungku Aziz. In addition, the relative humidity in the range of 57.4 % to 66.8% is slightly higher than the standards requirements, thus a reheat coil must be used to reduce the relative humidity, as well as to bring the temperature back to 24°C, 50% RH.

ACKLOWLEDGEMENTS

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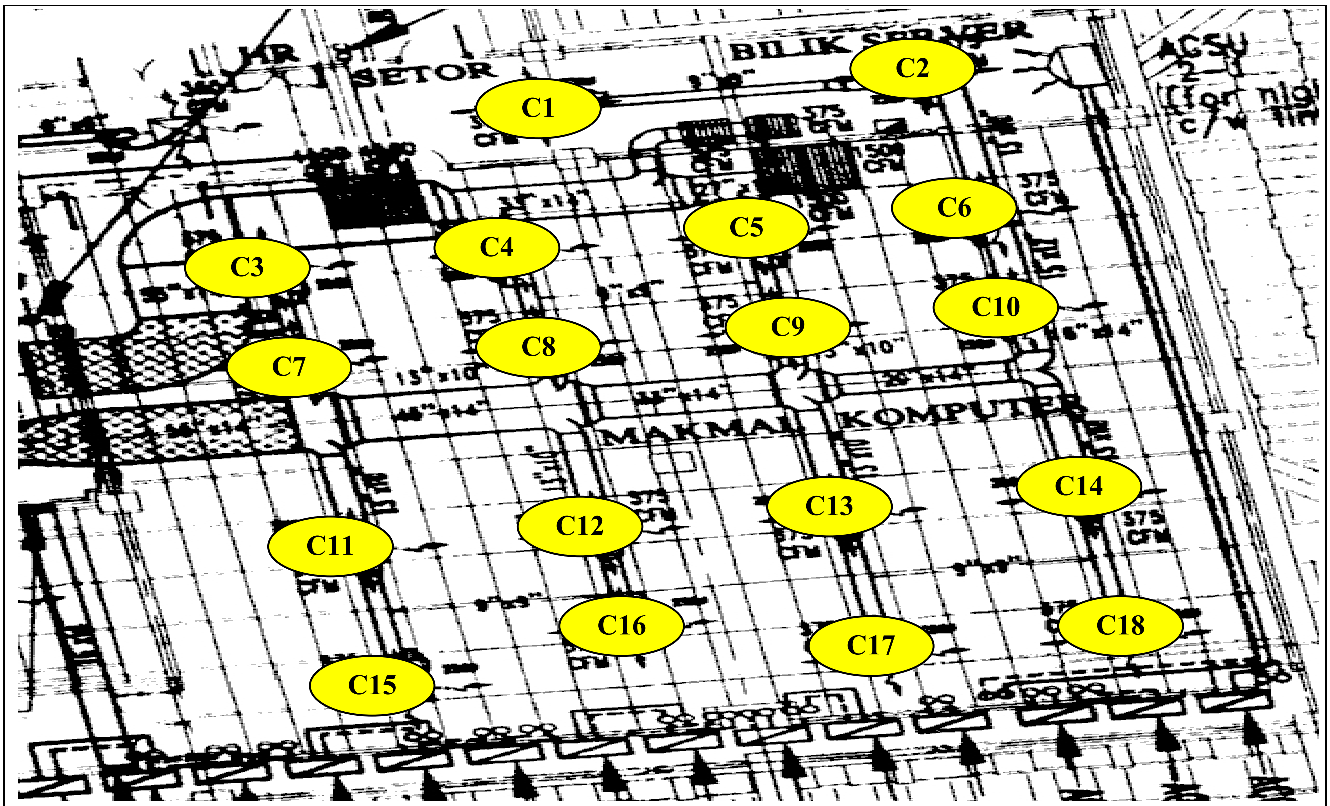
APPENDIX

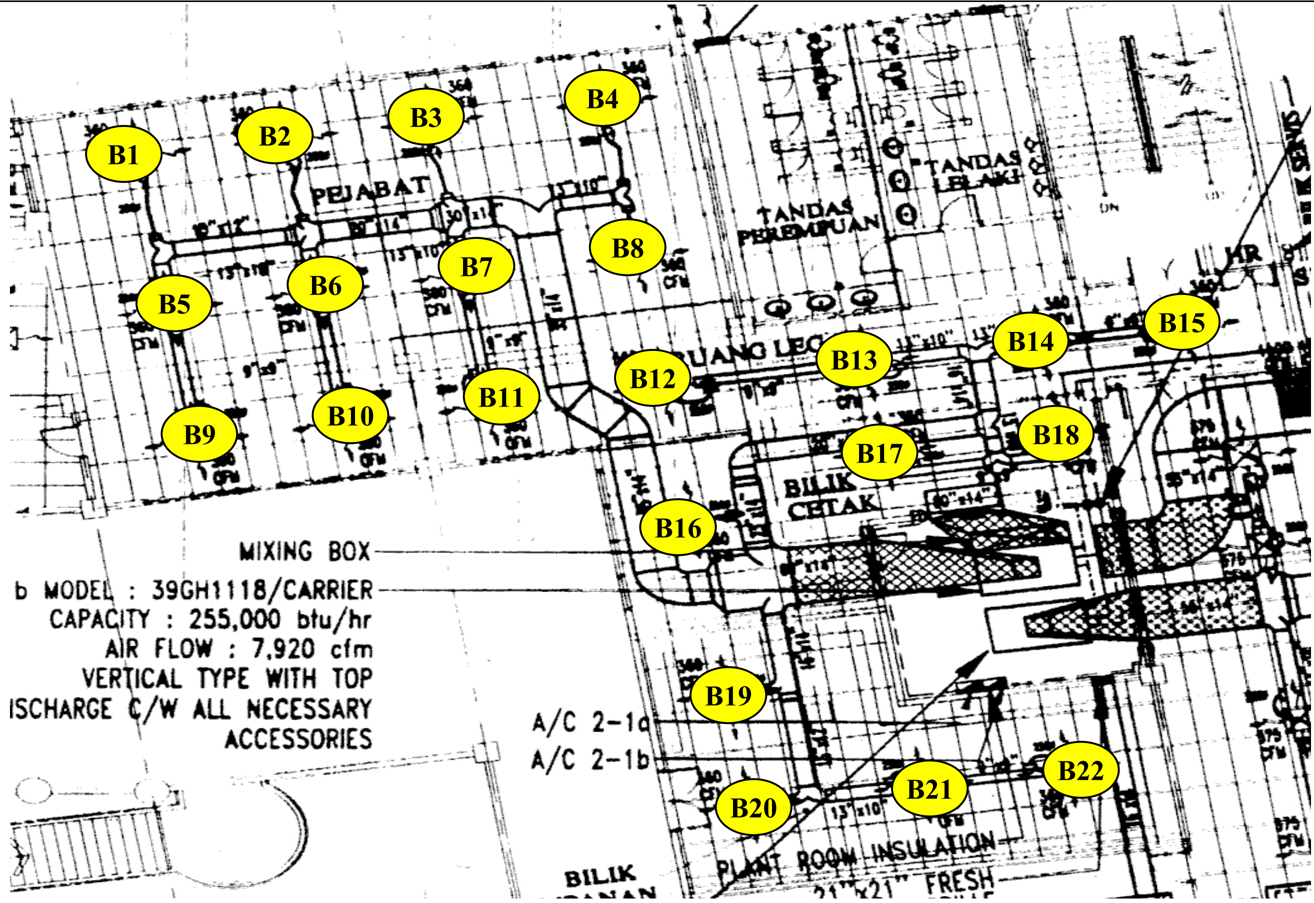
Table 1A: Supply air conditions of AHU2-1a



Room conditions: 20.9°C, 66.8% RH

Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)	Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)
C1	5.3	17.7	76.4	174	C10	6.4	16.3	84.2	105
C2	4.6	18.2	87.0	169	C11	7.7	16.6	84.0	132
C3	6.3	16.3	82.0	156	C12	6.0	16.6	83.7	156
C4	7.7	16.3	81.0	174	C13	6.2	16.6	84.0	169
C5	6.5	16.2	83.0	181	C14	7.2	16.6	84.2	181
C6	4.9	16.3	84.2	201	C15	6.4	19.4	88.5	171
C7	8.8	16.3	84.0	156	C16	7.6	17.5	84.0	168
C8	6.2	16.4	84.0	198	C17	7.6	18.0	91.0	174
C9	5.2	17.9	93.1	174	C18	7.0	16.7	84.0	181

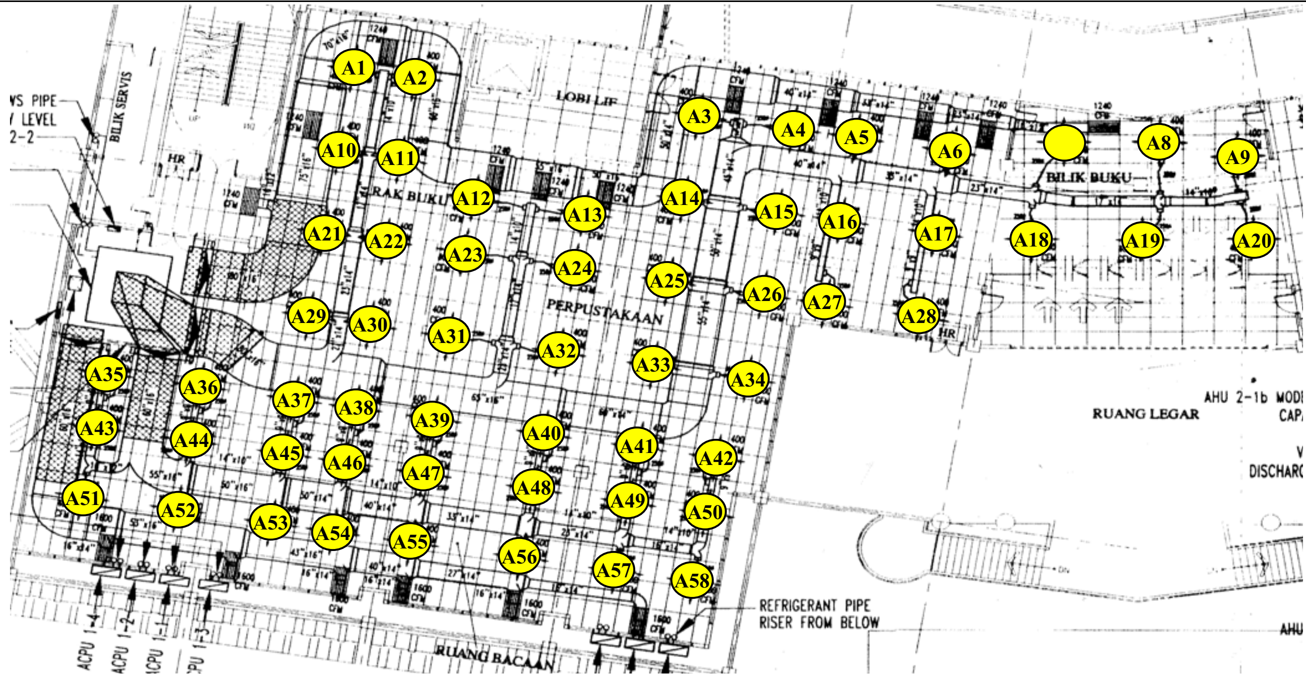
Table 2A: Supply air conditions for AHU2-1b



Room conditions: 20.5°C, 57.4% RH

Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)	Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)
B1	3.8	13.4	79.5	175	B12	3.8	13.9	77.0	275
B2	5.6	13.3	79.6	186	B13	3.5	14.0	76.0	260
B3	3.7	13.4	79.5	300	B14	2.8	13.3	73.0	215
B4	6.3	13.3	80.5	260	B15	4.4	13.5	78.3	150
B5	5.0	13.5	76.0	285	B16	3.6	13.1	77.2	379
B6	3.5	13.6	78.0	219	B17	3.6	13.1	79.2	378
B7	1.5	15.0	71.0	310	B18	2.5	13.7	77.4	350
B8	4.2	13.4	80.0	230	B19	3.5	13.3	78.0	300
B9	5.2	13.7	78.6	220	B20	4.1	13.9	73.2	248
B10	6.2	13.7	78.7	290	B21	5.5	13.2	79.0	182
B11	5.4	13.7	77.5	300	B22	3.8	14.1	74.0	210

Table 3A: Supply air conditions for AHU2-2



Room conditions: 17.6°C, 62.6% RH

Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)	Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)
A1	4.2	14.7	80.0	402	A25	7.4	13.6	80.5	378
A2	4.1	14.9	80.0	378	A26	5.2	14.0	80.5	356
A3	6.7	13.6	80.2	298	A27	5.1	14.2	77.2	327
A4	8.0	13.6	81.0	378	A28	4.9	13.9	78.7	136
A5	6.1	13.6	83.0	301	A29	4.4	14.4	81.0	423
A6	7.8	13.7	80.0	261	A30	7.0	14.4	82.4	340
A7	3.1	13.7	78.0	405	A31	3.3	14.6	79.5	342
A8	5.4	13.6	78.5	327	A32	7.0	13.6	81.0	135
A9	4.4	13.9	78.0	342	A33	5.7	13.5	80.0	278
A10	5.0	14.4	81.0	449	A34	4.4	12.5	80.5	224
A11	4.7	14.7	80.6	370	A35	2.7	13.7	80.5	187
A12	7.3	13.5	82.2	278	A36	3.0	14.7	80.0	327
A13	6.4	13.6	81.4	251	A37	3.1	13.8	79.5	398
A14	7.4	13.6	80.5	316	A38	2.9	14.7	80.5	342
A15	5.1	13.5	81.3	316	A39	3.8	14.8	79.0	316
A16	5.5	13.5	81.1	378	A40	3.4	15.0	78.3	278
A17	1.4	13.4	81.0	169	A41	3.0	13.2	78.0	287
A18	4.3	13.6	79.0	327	A42	3.2	14.0	80.5	256
A19	3.8	13.8	79.6	342	A43	2.3	13.1	78.5	378
A20	7.5	13.3	80.1	167	A44	2.0	14.7	79.5	219
A21	4.5	14.7	80.0	427	A45	4.1	14.4	79.8	327
A22	7.0	14.5	82.3	380	A46	3.1	13.7	78.5	342
A23	6.3	14.0	81.0	189	A47	2.8	14.1	77.2	316
A24	8.0	13.7	81.3	278	A48	2.4	14.3	78.0	224

Table 3A: Supply air conditions for AHU2-2 (cont)

Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)	Diff #ID	v (m/s)	DBT (°C)	%RH	CO2 (ppm)
A49	3.6	14.8	79.0	301	A54	4.2	14.0	79.0	327
A50	5.0	14.7	81.0	195	A55	4.8	14.4	80.7	278
A51	1.6	14.1	71.0	215	A56	4.8	15.0	81.5	267
A52	1.9	14.7	78.0	378	A57	3.0	14.9	79.8	342
A53	3.7	14.4	80.0	316	A58	3.7	15.0	81.0	327

Table 4A: Cooling capacity for each diffuser in AHU2-1a

Diffuser ID	Volumetric flow rate		Enthalpy		Cooling Capacity, Q (kW)
	CFM	m3/s	hSA (kJ/kg)	hRA (kJ/kg)	
C1	477	0.225	42.2	47.2	1.3750
C2	414	0.196	47.1	47.2	0.0239
C3	568	0.268	40.4	47.2	2.2228
C4	694	0.327	40.1	47.2	2.8366
C5	586	0.276	40.4	47.2	2.2934
C6	441	0.208	41.0	47.2	1.5763
C7	793	0.374	41.0	47.2	2.8309
C8	559	0.264	41.2	47.2	1.9302
C9	468	0.221	48.3	47.2	-0.2968
C10	577	0.272	41.0	47.2	2.0588
C11	694	0.327	41.7	47.2	2.1974
C12	541	0.255	41.7	47.2	1.7122
C13	559	0.264	41.7	47.2	1.7693
C14	649	0.306	41.8	47.2	2.0173
C15	577	0.272	51.2	47.2	-1.3283
C16	685	0.323	44.2	47.2	1.1830
C17	685	0.323	47.9	47.2	-0.2760
C18	631	0.298	42.0	47.2	1.8886
Total	10594	5.001			26.0145

Table 5A: Cooling capacity for each diffuser from AHU2-1b

Diffuser ID	Volumetric flow rate		Enthalpy		Cooling Capacity, Q (kW)
	CFM	m3/s	hSA (kJ/kg)	hRA (kJ/kg)	
B1	342	0.162	32.7	42.7	1.9717
B2	504	0.238	32.5	44.7	3.5448
B3	333	0.157	32.7	42.6	1.9006
B4	568	0.268	32.7	41.8	2.9746
B5	450	0.213	32.0	42.4	2.6981
B6	315	0.149	32.7	42.1	1.7070
B7	135	0.064	34.1	41.9	0.6071
B8	378	0.179	32.8	42.5	2.1138
B9	468	0.221	33.1	43.5	2.8060
B10	559	0.264	33.1	43.4	3.3134
B11	486	0.230	32.8	42.1	2.6057
B12	342	0.162	33.2	41.9	1.7153
B13	315	0.149	33.2	42.5	1.6889
B14	252	0.119	30.9	42.3	1.6562
B15	396	0.187	32.6	42.6	2.2830

B16	324	0.153	31.4	42.0	1.9800
B17	324	0.153	31.9	42.5	1.9800
B18	225	0.106	32.8	42.5	1.2582
B19	315	0.149	32.1	42.5	1.8886
B20	369	0.174	32.2	42.5	2.1911
B21	495	0.234	32.1	42.5	2.9679
B22	342	0.162	32.9	42.5	1.8928
Total	8243	3.891			47.7448

Table 6A: Cooling capacity for each diffuser from AHU 2-2

Diffuser ID	Volumetric flow rate		Enthalpy		Cooling Capacity, Q (kW)
	CFM	m3/s	hSA (kJ/kg)	hRA (kJ/kg)	
A1	378	0.179	35.8	37.5	0.3705
A2	369	0.174	36.3	37.5	0.2553
A3	604	0.285	33.3	39.8	2.2596
A4	721	0.340	33.5	37.5	1.6603
A5	550	0.259	34.0	37.5	1.1078
A6	703	0.332	33.5	37.5	1.6188
A7	279	0.132	33.0	37.5	0.7238
A8	486	0.230	32.9	37.5	1.2888
A9	396	0.187	33.4	37.5	0.9360
A10	450	0.213	35.4	37.5	0.5448
A11	423	0.200	36.0	37.5	0.3658
A12	658	0.310	33.6	37.5	1.4772
A13	577	0.272	33.6	37.5	1.2951
A14	667	0.315	33.4	39.0	2.1501
A15	459	0.217	33.3	37.5	1.1114
A16	495	0.234	33.3	37.5	1.1986
A17	126	0.060	33.0	37.5	0.3269
A18	387	0.183	33.0	37.5	1.0040
A19	342	0.162	33.6	37.5	0.7689
A20	676	0.319	32.6	37.5	1.9068
A21	405	0.191	35.8	39.6	0.8872
A22	631	0.298	36.0	37.5	0.5448
A23	568	0.268	34.4	37.5	1.0133
A24	721	0.340	33.8	37.5	1.5358
A25	667	0.315	33.4	37.5	1.5742
A26	468	0.221	34.4	37.5	0.8364
A27	459	0.217	33.9	37.5	0.9526
A28	441	0.208	33.6	37.5	0.9915
A29	396	0.187	35.4	37.5	0.4794
A30	631	0.298	35.7	37.5	0.6538
A31	297	0.140	35.5	37.5	0.3424
A32	631	0.298	33.5	37.5	1.4528
A33	513	0.242	33.0	37.5	1.3309
A34	396	0.187	30.9	37.5	1.5068
A35	243	0.115	33.6	34.2	0.0841
A36	270	0.128	35.8	38.1	0.3580
A37	279	0.132	33.6	37.5	0.6273
A38	261	0.123	36.0	37.5	0.2257

A39	342	0.162	35.8	37.5	0.3352
A40	306	0.145	36.1	37.5	0.2470
A41	270	0.128	31.8	37.5	0.8872

Table 6A: Cooling capacity for each diffuser from AHU 2-2 (cont)

Diffuser ID	Volumetric flow rate		Enthalpy		Cooling Capacity, Q (kW)
	CFM	m3/s	hSA (kJ/kg)	hRA (kJ/kg)	
A42	288	0.136	34.3	37.5	0.5313
A43	207	0.098	31.7	37.5	0.6922
A44	180	0.085	35.7	38.1	0.2491
A45	369	0.174	35.1	37.5	0.5106
A46	279	0.132	33.1	37.5	0.7077
A47	252	0.119	33.7	37.5	0.5521
A48	216	0.102	34.4	37.5	0.3860
A49	324	0.153	35.8	37.5	0.3175
A50	450	0.213	36.1	37.5	0.3632
A51	144	0.068	32.1	33.8	0.1411
A52	171	0.081	35.3	37.5	0.2169
A53	333	0.157	35.1	37.5	0.4607
A54	378	0.179	33.9	37.5	0.7845
A55	435	0.205	35.3	37.5	0.5513
A56	432	0.204	37.0	37.5	0.1245
A57	270	0.128	36.3	37.5	0.1868
A58	333	0.157	36.8	37.5	0.1344
Total	24010	11.335			46.1470

Table 7A: Cooling load calculation for full load condition

Cooling Load Calculations

Indoor Design Conditions							Outdoor Design Conditions						
DB (F)	WB (F)	RH %	W (gr/IB)	v (cuFt/IB)	h (Btu/Ib)	DP (F)	DB (F)	WB (F)	RH	W	v (cuft/Ib)	h (Btu/Ib)	DP (F)
75	61	50	95	20	33	55	94	82	63	227	21	58	79
											Elev.	Lat.	DR
											127	3.0 N	20

Cooling Load Due to heat transfer through External walls							Cooling Load Due to heat Gain through windows			
WALL	Area	U	CLTDc	CLTD	LM	Qt (Btu/hr)	Facing	Area	GLF	Qt (kJ/hr)
North	2310	1	23	24	-3	29222	North	14	49	686
West	939	1	55	54	-1	28405	West	61	183	11163
South	1463	1	44	35	7	35405	South	796	96	76416
East	730	1	44	43	-1	17666	East	270	139	37530
ROOF	12643	0	71	70	-1	161578	South Door	349	96	33468
FLOOR	12643	1	15	14	-1	39162				
Total cooling Load through walls (Btu/hr) =						311436	Cooling Load through Glass (Btu/hr) =			159263

Table 7A: Cooling load calculation for full load condition (cont)

Cooling Load Due to occupancy and appliances							Cooling Load Due to Lighting Equipments						
Cooling Load Due to occupants							Computer Lab				Office & Lib		
Location	N (person)	Qs (Btu/hr)	Ql (Btu/hr)	Qt (Btu/hr)			No.	Wattage	Qt (Btu/hr)		No.	Wattage	Qt (Btu/hr)
Off. & Lib	156	38220	11232	49452			82	40	13989		294	40	50156
Comp. Room	61	14945	4392	19337			Total Lighting Cooling Load (Btu/hr) =				64146		
	Qt (Btu/hr)	53165	15624	68789									
Cooling Load Due to equipments							Cooling Load Due to Infiltration						
	Computer Lab			Office & Lib			ACH	V (Ft3)	CFM	Qs(Btu/hr)	QL(Btu/hr)	Qt (Btu/hr)	
Type	No.	Q/ each	Qt (Btu/hr)	No.	Q/ each	Qt (Btu/hr)	1	126430	2107	44040	189359	233399	
PC	62	426	26412	25	375	9375	Total cooling Load due to infiltration					233399	Btu/hr
OHP	2		0			0	Cooling Load Due to ventilation						
LCD-P	2		0			0	CFM/ per.	No.	CFM total	Qs(Btu/hr)	QL(Btu/hr)	Qt (Btu/hr)	
Photocopier				3	3750	11250	13	217	2821	58959	253507	312466	
Refrigrator				1	1100	1100	Total cooling Load due to ventilation					312466	Btu/hr
TV				1	273	273							
Qtotal for comp. room =			26412	Qtotal for Lib & off. =		21998							
Appliances and equipments total Cooling Load (Btu/hr) =						48410							
Total Cooling Load for building (Btu/hr) =							1197908						
Total Cooling Load for building (ton) =							100						

Table 8A: Cooling load calculation for part load condition

Cooling Load Calculations													
Indoor Design Conditions							Outdoor Design Conditions						
DB (F)	WB (F)	RH %	W (gr/IB)	v (cuFt/IB)	h (Btu/Ib)	DP (F)	DB (F)	WB (F)	RH	W	v (cuft/Ib)	h (Btu/Ib)	DP (F)
74	60	50	92	20	32	54	85	69	50	133	21	41	64
											Elev.	Lat.	DR
											127	3.0 N	13
Cooling Load Due to heat transfer through External walls							Cooling Load Due to heat Gain through windows						
WALL	Area	U	CLTDc	CLTD	LM	Qt (Btu/hr)	Facing	Area	GLF	Qt (kJ/hr)			
North	2310	1	19	24	-3	23504	North	14	49	686			
West	939	1	51	54	-1	26081	West	61	183	11163			
South	1463	1	40	35	7	31784	South	796	96	76416			
East	730	1	40	43	-1	15859	East	270	139	37530			

ROOF	12643	0	67	70	-1	176559	south Door	349	96	33468
FLOOR	12643	1	11	14	-1	27413				
Total cooling Load through walls (Btu/hr) =						301201	Cooling Load through Glass (Btu/hr) =			159263

Table 8A: Cooling load calculation for part load condition (cont)

Cooling Load Due to occupancy and appliances							Cooling Load Due to Lighting Equipments						
Cooling Load Due to occupants							Computer Room				Office & Lib		
Location	N (person)	Qs (Btu/hr)	Ql (Btu/hr)	Qt (Btu/hr)			No.	Wattage	Qt (Btu/hr)		No.	Wattage	Qt (Btu/hr)
Off &Lib	10	2450	720	3170			82	40	13989		294	40	50156
Comp. Room	5	1225	360	1585			Total Lighting Cooling Load (Btu/hr)=				64146		
	Qt(Btu/hr)	3675	1080	4755									
Cooling Load Due to equipments							Cooling Load Due to Infiltration & ventilation						
	computer Lab.			office & Lib			ACH	V (Ft3)	CFM	Qs (Btu/hr)	QL (Btu/hr)	Qt (Btu/hr)	
Type	No.	Q/each	Qt (Btu/hr)	No.	Q/each	Qt (Btu/hr)	1	126430	2634	31871	73645	105516	
PC	5	375	1875	25	375	9375	Total cooling Load due to infiltration & ventilation					105516	Btu/hr
OHP	1		0			0	the changes were						
LCDP	1		0			0	OD DBT changed to 85 which is equal to 29.5 C						
photocopier				2	3750	7500	ID DBT decreased by only 1 F						
Refrigrator				1	1100	1100	Occupants assumed to be 15 people 10 in the lib. And 5 in the computer lab						
TV				1	273	273	only 5 computers are on						
Qtotal for comp. room =			1875	Qtotal for Lib & off. =		18248	only one OHP is on						
Appliances and equipments total Cooling Load (Btu/hr) =						20123	only 2 of the 3 photocopiers are on						

Total Cooling Load for building (Btu/hr) =	655003
Total Cooling Load for building (ton) =	55