

Estimation of Air Conditioning Load for a Lecture Auditorium

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ABSTRACT

The estimation of air conditioning load for a lecture auditorium in the Department of Mechanical Engineering, University of Port Harcourt was carried out in this work. The auditorium is at latitude 4.9069°N and longitude 6.9170°E. The grand total heat load of the auditorium was estimated with the use of MATLAB software. The grand total heat load was estimated to pave the way for the installation of appropriate air conditioning units. Parametric analysis was also carried out to determine the effect of different parameters on the grand total heat load. The results showed that an increase in the outdoor temperature and number of glass windows in the building will lead to an increase in the grand total heat load posed on the building if other variable physical parameters are kept constant, while an increase in the indoor temperature of the building will lead to a decrease in the grand total heat load of the building when other variable physical parameters are kept constant. The methodology used in the air conditioning load estimation for this case study can be easily applied to other buildings. The work will be useful to air conditioning engineers, individuals and groups who intend to install air conditioning systems in homes and auditoriums.

Key words: Load estimation, Parametric analysis, MATLAB, Grand total heat.

INTRODUCTION

Air-conditioning involves maintaining a given space at a predetermined temperature and relative humidity by supplying clean and conditioned air to the given space for human comfort and / or product processing [1]. This treatment or control of the air deals basically with the temperature and humidity of the air and has become very important to man because of the harsh climate conditions and seasonal changes in the world. The cooling of the supply air follows basic refrigeration techniques using refrigerants. Some of the refrigerants used has ozone layer depletion potentials and has been of concern [2]. Issues concerning the use of different refrigerants as well as the performance of a number of refrigerants have been sufficiently studied [3-6]. There are thus several works aimed at exploiting thermoelectric cooling instead of refrigerant-based cooling systems [7-10].

Air-conditioning may involve either increasing or decreasing the temperature of the conditioned space. Cosnier et al. [7] presented an experimental and numerical study of a thermoelectric air-cooling and air-heating system. They have reached a cooling power of 50W

per module, with a COP between 1.5 and 2, by supplying an electrical intensity of 4A and maintaining the 5°C temperature difference between the hot and cold sides. Shen et al. [11] investigated a novel thermo-electric radiant air-conditioning system (TERAC). The system employs thermoelectric modules as radiant panels for indoor cooling, as well as for space heating by easily reversing the input current. Based on the analysis of a commercial thermoelectric module they have obtained a maximum cooling COP of 1.77 when applying an electric current of 1.2A and maintaining cold side temperature at 20°C.

Manoj et al [12] presented an experimental study of novel potential green refrigeration and air-conditioning technology. They analyzed the cause and effects of an existing air-conditioning system. Thermoelectric cooling provides a promising alternative refrigeration and air conditioning technology due to their distinct advantages. The available literature shows that thermoelectric cooling systems are generally only around 5-15% as efficient compared to 40-60% achieved by the conventional compression cooling system. The problem of global warming and emission of harmful gases due to use of refrigerants like Freon and Ammonia can be prevented by using thermoelectric modules air conditioner [13]. How right sizing of equipment helps to meet the accurate prediction of the cooling or heating loads was explained by Kumari et al [14]. Right-sizing of heating, ventilation and air conditioning (HVAC) system aids the selection of equipment and the design of the air distribution system to meet the accurate predicted heating and cooling loads of the house. Bharathi et al [15] analyzed the cooling load calculations, design calculations involved in selection of air conditioning systems. The sizing of an air conditioning system and how to find out the most appropriate temperature required for the selected room in which we put the AC was the subject of discussion by Raj and Soni [16]. Energy source for powering air-conditioning systems have been an issue as electricity from fossil fuels has been majorly used in powering air-conditioning systems. The use of solar energy for powering air-conditioning systems has been the focus in recent studies [17,18].

In Nigeria, the weather condition is mostly very hot, resulting from the intensity of solar radiation on this part of the world and makes the need for cooling a pertinent one in the country. Most universities in the country built structures without proper ventilation and airing systems, hence students are made to digest complex topics in the lecture hall amidst thermal inconveniences. The need for a cooling system in one of the lecture auditoriums in the department of Mechanical Engineering, University of Port Harcourt inspired this work. The auditorium is at latitude 4.9069°N and longitude 6.9170°E. The cooling load of the auditorium was estimated in this work so as to pave way for the selection and installation of appropriate air-conditioning units. The analysis could easily be applied to other lecture theaters by merely varying relevant input data.

MATERIALS AND METHODS

System description

The air conditioner in a building is a cooling system that provides cool air via a process that draws out the warm air inside, removes its heat and replaces it with a cooler air. The Mechatronic Engineering building is located at a latitude of 4.9069°N and longitude 6.9170°E. Having both offices and auditorium compartments, it was erected in an open space where the sun can incident it from different angles and directions. The auditorium has a seating capacity of about 500 persons.

Design Conditions of Air Conditioning Systems

Before proceeding with an air conditioning design; the inside and outside conditions have to be specified. Also, the supply air condition has to be fixed.

Indoor Design Conditions

The choice of inside air conditions depends on the application. In this case it's a comfort air conditioning. Parameters for measuring comfort include: air temperature, relative humidity, air velocity, air purity, pressure differential between the space and surroundings. However, a parameter called the effective temperature is used to correlate the combined effects of air velocity, air purity, air temperature and relative humidity. In high occupancy applications, it may be economical and still comfortable to specify the summer indoor relative humidity as 55 to 60 percent. Table 1 shows the indoor design conditions, although parametric analysis was carried out.

Outdoor Design Conditions

Outdoor design conditions (temperature, humidity, wind conditions, solar radiations) are highly variable, both by season and by time of day. Table 2 shows the outdoor design conditions.

Table 1: Indoor design conditions

Design Parameters	Values
Effective temperature	22 °C
Dry bulb temperature	22±1 °C
Relative humidity	60±5 %

Table 2: Outdoor design conditions

Design Parameter	Values [°C]
Maximum dry bulb temperature (TDB_{max})	35 22
Minimum dry bulb temperature (TDB_{min})	13
Change in dry bulb temperature ΔTDB_o	
Wet bulb temperature	24

Cooling Load Calculations

The major components of cooling loads in buildings are due to solar radiation, transmission through the building fabric or structure and fresh air for ventilation. However, in design applications for theatres and auditoriums, the predominant cooling load is due to occupancy. The components of cooling load are classified into four (4) categories. They are external cooling load, cooling load due to infiltration and ventilation, internal (room) cooling load and system cooling load. Each of these is presented below.

External Cooling Load

The external cooling load is to a large extent, a product of the fabric heat gains such as heat gains from the walls, roof, partitions, windows, ceiling and floor. However, the exteriors (windows) of the mechatronic engineering building in the University of Port Harcourt (used as

a case study) comprise about 25% glass. Hence the solar heat gain through glass is still significant.

Periodic Heat Gain into the Building Structure

External weather conditions are not steady over a period of 24 hours. Heat transfer through walls and roofs at any time can be calculated using Equation (1),

$$Q_{\tau} = UA(T_{e_m} - T_i) + UA\lambda(T_{e_{\tau-\phi}} - T_{e_m}) \quad (1)$$

where Q_{τ} is heat transfer at time τ , U is overall heat transfer coefficient, A is surface area of the material, T_{e_m} is the mean outside sol air temperature, $T_{e_{\tau-\phi}}$ is sol-air temperature at time $\tau - \phi$, i.e., ϕ hours before the heat transfer is to be calculated, T_i is inside room temperature, and λ is decrement factor, ϕ is time lag.

Equivalent Temperature Differential

Due to the complexity of using Equation (1), a parameter called the equivalent temperature differential is used to account for solar radiation and outside air temperature. It is given as,

$$(\Delta T_E) = (T_{e_m} - T_i) + \lambda(T_{e_{\tau-\phi}} - T_{e_m}) \quad (2)$$

The solar heat gain through walls and buildings is given as,

$$Q = UA\Delta T_E \quad (3)$$

ΔT_E can be calculated with fixed values of inside and outside air temperature (T_i and T_o) for different types of constructions and as a function of latitude and time for roofs and latitude, time and orientation of surface for walls using Schmidt's method, with a correction temperature of 4.225 °C,

$$\Delta T_E = \Delta T_{E_{table}} + T'_e \quad (4)$$

where $\Delta T_{E_{table}}$ is the temperature differential value estimated using Schmidt's method while $T'_e = 4.225$ °C is the correction temperature. An average equivalent temperature differential value was used in this work.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient, U is estimated from Equation (5),

$$\frac{1}{U} = \frac{1}{f_o} + \sum \frac{\Delta x}{k} + \sum \frac{1}{c} + \frac{1}{f_i} \quad (5)$$

where K is thermal conductivity of the material, C is conductance (used for some materials as against the thermal conductivity and material thickness), Δx is thickness of material, f_o is outside wall film heat transfer coefficient, and f_i is inside wall film heat transfer coefficient.

Heat Gain through the Wall

The building is located in an open area where the rays of the sun can incident it from all directions. However the North and South sides of the building are insulated by the office compartments on both sides, hence the heat gain through the wall is calculated for East and West only. The wall of the building is made of 225cm concrete and 1.25cm plaster. The heat gain through the wall is,

$$Q_{wall} = U_{wall} \times A_{wall} \times \Delta T_{E_{ave}} \quad (6)$$

where the parameters in Equation (6) are similar to those in Equation (3) but applicable to the building wall. The overall heat transfer coefficient through the wall is given as,

$$\frac{1}{U_{wall}} = \frac{1}{f_o} + \left[\frac{\Delta x}{k} \right]_{plaster} + \left[\frac{\Delta x}{k} \right]_{concrete} + \left[\frac{\Delta x}{k} \right]_{plaster} + \frac{1}{f_i} \quad (7)$$

where the parameters in Equation (7) are similar to those in Equation (5), defined with appropriate subscripts. There is also heat gain through the floor and through the roof. They are also estimated with relations similar to Equation (6). The floor is tiled while aluminium roofsheets are used for the roof. The airspace in the roof is also considered in the analysis.

Solar Transmission through Glass Windows

Heat gain through glass windows and doors constitute a major heat load on an air conditioning system due to the direct and diffuse solar radiation reaching the glass material. The solar heat gain used in this work is that gotten from the hottest day of the year in Port Harcourt which is around the 22nd day of March. The University of Port Harcourt is located at 4.9069°N. There are no tabulated values for 4°N or even 5°N, so values from 0°N (Table 3) are used. The solar heat gain values are in Watt per square metre. The average solar heat gain for any exposure side was used in this work.

Table 3: Solar heat gain through ordinary glass [19]

0° North Latitude		Am					Pm							
Time of year	Exposure	6	7	8	9	10	11	Noon	1	2	3	4	5	6
March 22	East	0	42 3	52 7	47 6	33 8	14 8	44	44	44	41	38	19	0
	North	0	19	38	41	44	44	44	44	44	41	38	19	0
	South	0	19	35	41	44	44	44	44	44	41	38	19	0
	West	0	19	35	41	44	44	44	14 8	33 8	47 6	52 7	42 6	0

Cooling Load Due to Infiltration

Infiltration is the unintentional or accidental introduction of outdoor air into the cooling space, typically through cracks in the building envelope and doors for passage. Infiltration has both sensible and latent heat portions given by Equations (8) and (9) respectively,

$$Q_{si} = nV \left[\frac{T_o - T_i}{3} \right] \quad (8)$$

$$Q_{Li} = 0.8nV[g_o - g_i] \quad (9)$$

where Q_{si} is the sensible portion of heat gain due to infiltration, n is number of air changes per hour, V is volume of space, T_o and T_i are outdoor and indoor temperatures respectively, g_o and g_i are outdoor and indoor moisture content respectively.

Ventilation entails maintaining the space air purity by introducing fresh air into the space. The heat load posed on the space due to ventilation has both sensible and latent components given by Equations (10) and (11) respectively,

$$Q_{Vs} = 0.0204 \times \dot{V} \times \Delta T \times BPF \quad (10)$$

$$Q_{VL} = 50 \times \dot{V} \times \Delta g \times BPF \quad (11)$$

where Q_{Vs} is the sensible heat load due to ventilation, Q_{VL} is the latent heat load due to ventilation, $\dot{V}$ is volume flow rate of fresh air, ΔT is change in dry bulb temperature between outside and inside air, Δg is change in moisture content between outside and inside air and BPF is by pass factor of the cooling coil.

Internal Heat Gains

Internal cooling loads include the sensible and latent heat due to occupants, lights, appliances, machines, pipes etc within the space to be conditioned. For the sake of this project, we assume that the building does not have machines or pipe as it is used for just classroom activities. Hence heat gains due to the occupants and lights constitute the internal heat gains.

Occupancy Load

The human body emits sensible and latent heat due to their metabolic rates which largely depends on the type of activities carried out in the space, the dry bulb temperature of the ambient as well as the structure of the person. Due to the complexity of calculating the metabolic rate for a student in a class, we shall assume that the student gives out as much heat as a person in an office. The building has a seating capacity of 500 persons. The metabolic heat rate of humans is given in Table 4.

Table 4: Heat liberated due to occupancy [19]

Activity	Metabolic Rates [W]	Heat Liberated [W]											
		Room Dry Bulb Temperature, [°C]											
		20	21	22	24	25	26						
		S	L	S	L	S	L	S	L	S	L	S	L
Seated At Rest	115	90	25	85	30	80	35	75	40	7	45	6	50
Office Work	140	10	40	95	45	90	50	80	60	7	65	7	70
Standing	150	10	45	10	50	95	55	82	68	7	73	7	78
Light Factory Work	235	13	10	12	11	11	12	10	13	9	14	8	155
Dancing	265	14	12	13	13	12	14	10	16	9	16	9	175
		0	5	3	2	5	0	5	0	8	7	0	

Lighting Load

The heat gain from electrical light depends on the rating of the electrical bulb, use factor and allowance factor. The cooling load posed by lighting instruments is usually sensible, given for fluorescent bulbs and incandescent bulbs respectively in Equations (12 and (13),

$$Q_{fluo} = \text{total wattage} \times \text{use factor} \times \text{allowance factor} \quad (12)$$

$$Q_{inca} = \text{total wattage} \quad (13)$$

Allowance factor is used only for fluorescent tubes and to represent power used for blast. The value is given as 1.25.

System Heat Gains

These refer to the heat gain of an air conditioning system comprising its components due to operation or exposure of its components. In this work, we considered two major losses in the air conditioning system. The losses include duct heat gain and leakage loss, and heat gain from the air conditioning fan

Duct Heat Gain and Leakage Loss

The air-conditioning duct is an air delivery system. It delivers air at the required flow rate and velocity and as well carries return air from the conditioned space back to the system mostly for re-conditioning. Hence heat gain and leakage loss is as a result of the supply/return of air to and fro the system. Heat gain by the supply air duct is estimated as,

$$Q = UA(T_a - T_s) \quad (14)$$

where T_a is ambient temperature and T_s is supply temperature. However, it has been found that the value calculated using Equation (14) is usually within the range of 5% of the room sensible heat (RSH) if the whole supply duct is outside the building. The room sensible heat here refers total sensible heat gain in the space estimated from the different sources considered. For this work 4% of the RSH will be used as the duct heat gain. For leakages, 5% of the RSH will be used to compensate for leakage loss. For the return air duct 3% of the RSH will be used to compensate for both heat gain and leakage losses.

Heat Gain from Air-Conditioning Fan

For the fan, the draw through system is assumed in this work. The heat gain due to the fan operation is taken as 5% of RSH, same value designers usually assume.

Safety Factor

Safety factor is strictly a factor of probable error in the estimation of the load. To compensate for this error, additional 5% should be added to the Room Sensible and Latent heats. (Arora, 2000)

By Pass Factor (BPF)

This is a function of the physical and operating characteristics of cooling or heating coil. It represents that portion of the air which passes through the cooling coil completely un-altered. The portion of the outside air that was bypassed has both sensible heat, $OASH_{bpf}$ and latent heat, $OALH_{bpf}$ given by Equations (15) and (16) respectively,

$$OASH_{bpf} = OASH \times BPF \quad (15)$$

$$OALH_{bpf} = OALH \times BPF \quad (16)$$

where OASH and OALH are the outside air sensible heat and outside air latent heat respectively. These loads are posed on the room in exactly the same manner as the infiltration load. The remaining portions of the outside air which was altered by the cooling coil can be calculated using Equations (17) and (18) respectively,

$$OASH_{(1-bpf)} = OASH \times (1 - BPF) \quad (17)$$

$$OALH_{(1-bpf)} = OALH \times (1 - BPF) \quad (18)$$

where $OASH_{(1-bpf)}$ is the sensible portion of outside air that was not bye-passed and $OALH_{(1-bpf)}$ is the latent portion of outside air that was not bye-passed. For the purpose of this project, a bye pass factor of 0.15 was assumed.

Grand Total Heat (GTH)

The grand total heat which is the total cooling load is given as,

$$GTH = TSH + TLH \quad (19)$$

where TSH is the total sensible heat and TLH is the total latent heat. The TSH and the TLH are given respectively by Equations (20) and (21),

$$TSH = ERSR + OASH_{(1-bpf)} + \text{Heat Gain Due to Return Air Duct} + \text{Return Duct Leakage Gain} \quad (20)$$

$$TLH = ERLH + OALH_{(1-bpf)} + \text{Return Duct Leakage Gain} \quad (21)$$

where ERSR and ERLH are the effective room sensible heat and effective room latent heat. They are given by Equations (22) and (23) respectively,

$$ERSR = RSH + (OASH \times BPF) \quad (22)$$

$$ERLH = RLH + (OALH \times BPF) \quad (23)$$

where RLH is the room latent heat, computed from the different load sources.

RESULTS AND DISCUSSION

The various computations were done using MATLAB R2015b. The input data for the analysis are shown in Table 5 while the equivalent temperature differential values obtained using Schmidt's method are presented in Table 6.

Table 5: Input data for cooling load estimation

Parameter	Symbol	Unit	Value
Maximum dry bulb temperature	$TDB_{O_{max}}$	[°C]	35
Minimum dry bulb temperature	$TDB_{O_{min}}$	[°C]	22
Indoor dry bulb temperature	TDB_i	[°C]	22
Thickness of airspace	$\Delta x_{airspace}$	[m]	0.90
Thermal conductivity of airspace	$K_{airspace}$	[W/mK]	0.026
Thermal conductance of tiles	C_{tile}	[W/m ² K]	5.23
Surface area of West wall	$A_{W_{wall}}$	[m ²]	71.225
Outside wall film coefficient	f_o	[W/m ² K]	23.3
Inside wall film coefficient	f_i	[W/m ² K]	8.5
Thickness of plaster	$\Delta x_{plaster}$	[m]	0.0125
Thermal conductivity of plaster	$K_{plaster}$	[W/mK]	8.65
Thickness of concrete	$\Delta x_{concrete}$	[m]	0.225
Thermal conductivity of concrete	$K_{concrete}$	[W/mK]	1.73
Thickness of aluminum	$\Delta x_{aluminum}$	[m]	0.002
Thermal conductivity of aluminum	$K_{aluminum}$	[W/mK]	237
Thickness of plastic	$\Delta x_{plastic}$	[m]	0.008
Thermal conductivity of plastic	$K_{plastic}$	[W/mK]	0.25
Surface area of East wall	$A_{E_{wall}}$	[m ²]	71.225
Surface area of the floor	A_{floor}	[m ²]	483.14
Surface area of the roof	A_{roof}	[m ²]	483.14
Surface area of West glass	$A_{W_{glass}}$	[m ²]	63.52
Surface area of East glass	$A_{E_{glass}}$	[m ²]	63.52
Height of the space	H_{space}	[m]	5
Number of occupants	N_p	-	500
outdoor moisture content	g_o	[g/kg]	0.0216
Indoor moisture content	g_i	[g/kg]	0.01
Change in moisture content	Δg	[g/kg]	0.0116
Number of fluorescent bulbs	N_f	-	20
Power absorbed by each fluorescent bulb	P_f	[W]	40
Allowance factor of the fluorescent bulb	AF	-	1.25
Bye Pass Factor of cooling coil	BPF	-	0.15

Table 6: Temperature differential values

Time (pm)	$\Delta T_{E_{table}}$ (°C)
1	4.25
2	4.75
3	5.5
4	6.32
5	8.34
6	9.69

The results of the cooling load calculation are shown in Table 7.

Table 7: Results of cooling load estimation

Parameter	Symbol	Unit	Value
Change in outside air temperature	ΔT_{DB_O}	[°C]	13
Change in dry bulb temperature	ΔT_{DB}	[°C]	13
Equivalent temperature correction	T'_e	[°C]	4.225
Average ΔT_E	$\Delta T_{E_{ave}}$	[°C]	10.7
Overall thermal resistance of the wall	$R_{th_{wall}}$	[m ² K/W]	0.2935
Overall thermal resistance of the floor	$R_{th_{floor}}$	[m ² K/W]	0.4389
Overall thermal resistance of the roof	$R_{th_{roof}}$	[m ² K/W]	34.81
Overall heat transfer coefficient of the wall	U_{wall}	[W/m ² K]	3.41
Overall heat transfer coefficient of the floor	U_{floor}	[W/m ² K]	2.28
Overall heat transfer coefficient of the roof	U_{roof}	[W/m ² K]	0.029
Volume flow rate of fresh air	$\dot{V}$	[m ³ /S]	105
Volume of space	V	[m ³]	2415.7
Heat gain through the West wall	$Q_{W_{wall}}$	[W]	2596.5
Heat gain through the East wall	$Q_{E_{wall}}$	[W]	2596.5
Total heat gain through the wall	Q_{wall}	[W]	5193
Heat gain through the floor	Q_{floor}	[W]	11778
Heat gain through the roof	Q_{roof}	[W]	148.5166
Solar heat gain through West glass	$Q_{W_{glass}}$	[W]	10467
Solar heat gain through East glass	$Q_{E_{glass}}$	[W]	10467
Total solar heat gain through glass	Q_{glass}	[W]	20934
Bye-passed outside Air Sensible Heat	$OASH_{BPF}$	[W]	4.1769
Bye-passed outside Air Latent Heat	$OALH_{BPF}$	[W]	9.1350
Outside Air Sensible Heat that is not bye-passed	$OASH_{(1-BPF)}$	[W]	23.6691
Outside Air Latent Heat that is not bye-passed	$OALH_{(1-BPF)}$	[W]	51.7650
Sensible heat liberated due to occupancy	Q_{P_s}	[W]	45000
Latent heat liberated due to occupancy	Q_{P_l}	[W]	25000
Heat due to fluorescent light	Q_f	[W]	1000
Initial room sensible heat before addition of system heat gains	RSH_i	[W]	84054
Initial room latent heat before addition of system heat gains	RLH_i	[W]	25000
Heat gain due to supply air duct	Q_{SD}	[W]	3362.1000
Leakage loss due to supply air duct	Q_{LSD}	[W]	4202.7000
Heat gain due to return air duct	Q_{RD}	[W]	2521.6000
Leakage loss due to return air duct	Q_{LRD}	[W]	2521.6000
Heat gain from air conditioning fan	Q_{fan}	[W]	4202.7000
Safety factor	SF	[W]	4202.7000
Room Sensible Heat	RSH	[W]	100020
Room Latent Heat	RLH	[W]	33405
Effective Room Sensible Heat	$ERSH$	[W]	100030
Effective Room Latent Heat	$ERLH$	[W]	33414
Total Sensible Heat	TSH	[W]	105090
Total Latent Heat	TLH	[W]	35988
Grand Total Heat	GTH	[W]	141080

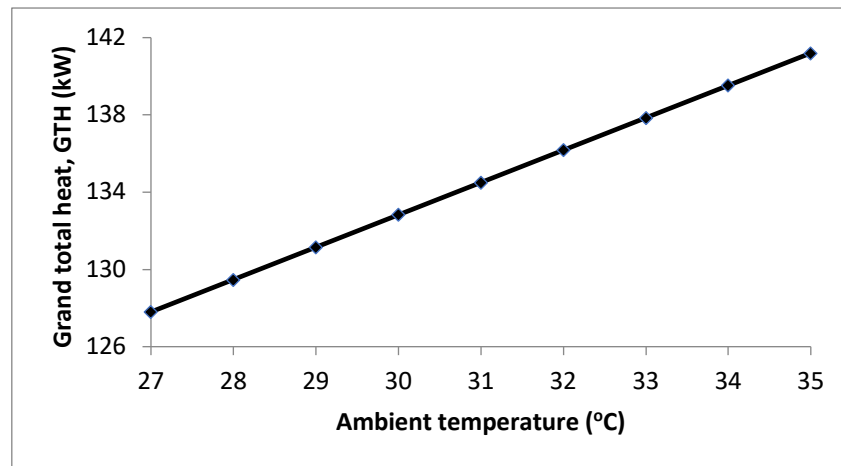


Figure 1: Effect of outdoor temperature on the grand total heat

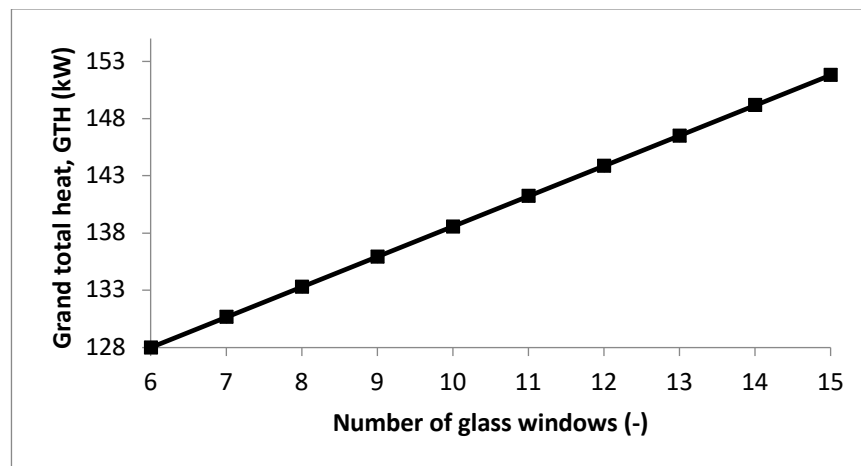


Figure 2 Effect of number of glass windows on the grand total heat

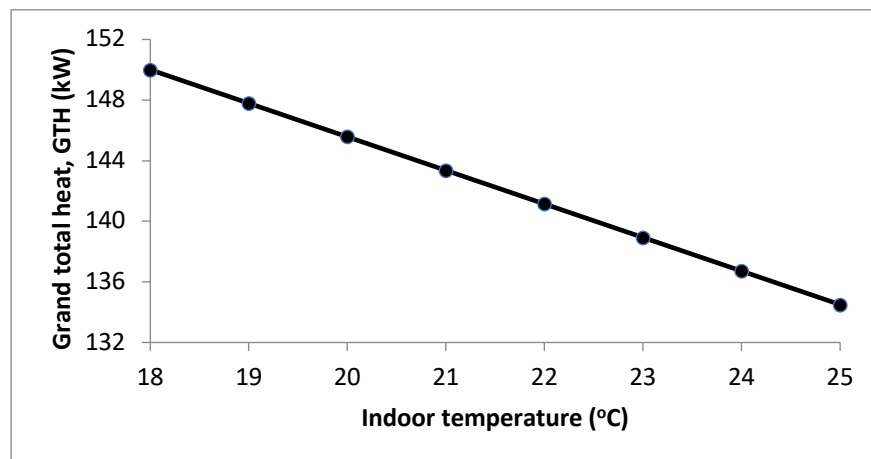


Figure 3: Effect of Indoor Temperature on the grand total heat

From the results above, the major components of the cooling load is that due to occupancy which is expected of a lecture auditorium. Hence the number of students per lecture period can largely determine the amount of cooling load posed on the system. The results of the parametric analysis showed that an increase in outdoor temperature of the building and an increase in the

number of glass windows will lead to an increase in cooling load for the building, as shown in Figures 1 and 2 respectively. On the other hand, if higher indoor temperature is required, the cooling load will decrease as presented in Figure 3.

CONCLUSION

An air conditioning system has been designed for a lecture auditorium in this work and the design procedure could be exploited in designing similar systems for different structures, especially those in hot regions as the case study used in this work. The choice of inside air condition temperature was based on the consideration of occupants' comfort. The equivalent temperature differential method was employed in calculating the fabric heat gains and design data were obtained from standard textbooks. Heat gain through the North and South sides of the building were neglected, because the North and South sides of the building were properly insulated by the office compartments on both ends. The results showed that the sensible heat is greater than the latent heat load for the system and the estimated cooling load is within the range of values expected. To keep the cooling load within design values, all doors must be shut after entering, to minimize infiltration, and glass windows facing the West should have blinds or trees planted in order to reduce the amount of sensible heat gain.

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