Global Journal of Mechanical Engineering, Volume 3, Number 2, 2002
ISSN 1595 – 7578

DUCT DESIGN AND PERFORMANCE: A CASE STUDY OF A UNIVERSITY SENATE CHAMBER

Odesola Isaac F. Mnse
Department of Mechanical Engineering
Faculty of Technology
University of Ibadan, Nigeria

and
Ndubuisi Christian U.
Department of Mechanical Engineering
Faculty of Technology
University of Ibadan, Nigeria

and
Ogbolu N. Dominic
Department of Mechanical Engineering
Faculty of Technology
University of Ibadan, Nigeria

ABSTRACT

This paper present the design of a central air-conditioning / ducting system for a case study of a University Senate Chamber in Nigeria. The design aims at maintaining conditions that are conducive to human comfort within the senate chamber of the University. (24℃ – 25.5℃, 50% - 55% RH). The design month was March (6) and using the expert assistance found in the carrier system design ducting analysis manual the refrigeration tonnage obtained was 40 and the total air quantity required to remove the heat load was 9908cfm. Low velocity with equal friction method was used in the ducting analysis Equipment selection was carried out for the proposed system. A simulation was carried on the proposed system to estimate the relative humidity and temperature at various times. Another test was carried out on the actual (existing unit) to estimate relative humidity and temperature at various times and positions within the chamber. A comparison of the simulated and tested values with standard comfort conditions reveals that the new design will certainly give more comfort.

Keywords: Refrigeration, humidity, chamber, comfort, air-conditioning, ducting, design.

INTRODUCTION

Air conditioning is that branch of engineering science, which deals with the study of conditioning of air for human comfort. It implies temperature of the surrounding (ambient)

As a consequence of molecular structure, air can hold varying amount of water vapor according to its temperature. Extreme high or low level of relative humidity can

create discomfort. Level between 40-60 percent should however create a reasonable internal environment.

It can be classified broadly as follows

1 According to the purpose namely comfort and industrial air conditioning
2 According to the session of the year; viz winter and summer air condition.
3 According to arrangement of equipment viz central, unitary (package system)
4 According to the mode of cooling namely all air-cooled and all water-cooled.

For effective and perfect air distribution when the conditioned air cannot be supplied directly from the air-conditioning equipment to the space to be conditioned, the ducts are installed. To fulfill this function of the ducts in a practical manner, the system must be designed within the prescribe limits of available space, friction loss, velocity, sound level, heat and leakage and losses and gain.

METHODOLOGY

Before the load was estimated, a comprehensive survey of the building was carried out and relevant air conditioning Handbook was used to obtain an accurate evaluation of the load component for the particular case being considered (senate chamber). The general procedure involve:

(1) Obtaining the characteristics of building, building material, component sizes external surface color and shape from building plans and specification.

(2) Determine the building location, orientation and external shading.

(3) Obtain appropriate weather data and select outdoor design conditions. Weather data can be obtained from local data station. (From reference No. 6 Table (1) and(11) of Appendix)

(4) Select indoor design conditions, such as indoor dry bulb temperature, indoor wet bulb temperature.

(5) Obtain a proposed schedule of lighting, occupants internal equipment, and process that would contribute to the thermal load.

(6) Select the time of the day and months to do the cooling load calculation.

(7) Calculate the space-cooling load at design condition.

LOAD ESTIMATION:

Design condition

Location:	Ibadan (Nigeria)
Average wind velocity	Low
Wind direction	South west
Design month	March
Elevation	745ft
Design level	2 ½ %
Outdoor dry bulb temperature	89^0F (31.6^0C)
Indoor dry bulb temperature	77^0F(25.5^0C)
Outdoor wet bulb temperature	78 (25.5^0C)
Inside relative humidity	55%
Daily range	$19^{0\,0}$F
Latitude	$7^0 20^0$N
Number of people	300
Outdoor specific humidity (W^0)	0.0182
Indoor specific humidity (W^1)	0.0112
Activity of people	(seated very light work writing)

Assumption:

(1) Lattitude used 8^0N

(2) The block used are those of 8 inch

(3) U values are chosen with respect to design consideration

(4) For general comfort air conditioning 2½ % design level is used.

That is 2½ % is the tolerance usually specified for general comfort design.

$CLTD_{correction}$ (Roof) = { (CLTD + LM)K + (78 – TR) + (To – 85)}f

Lm (Table 3) Horizontal

$CLTD_{correction}$ (WALL) { (CLTD + LM) K + (78 – TR) + (To – 85)} f

Lm (Table 3) i.e. depends on orientation

(From CLF TABLE 7 1985 ASHRAE. Fundamental) Maximum Solar heat G: see Table 4.

Using ASHRAE 1985 fundamental, U for wall (wall coefficient of heat transfer)

TABLE (5) = 0.40 Btu/h, for LW + HW concrete block + finish group E wall

U (Roof coefficient of heat transfer) TABLE (2) = 0.126Btu/h, roof No. 8

SC (shading coefficient) = 1.0, for clear glass.

TABLE 1.1

HEAT LOAD AT VARIOUS TIME OF THE DAY

PEAK LOAD TIME	1100	1200	1300	1400	1500	1600
ROOF = U. A. $CLTD_C$	4229.9	5438.5	7251.3	9064.1	11481.2	13294.1
WEST GLASS Q = A.SC.MSHF.CLF	2386.6	2545.7	3341.2	4773.2	6364.3	7796.3
S.E GLASS A.SC. MSHF.CLF	6438.2	5830.8	4980.5	4373.16	4008.7	3644.3
WALL = U.A. $CLTD_C$						
EAST	4048.6	4376.9	4595.8	4486.3	4376.9	4158.1
WEST	1016.4	1201.2	1386.0	1663.2	2217.6	2864
NORTH. WEST	1094.2	1313.1	1531.9	1860.2	2188.48	2626.1
SOUTH. EASTWALL	2209.2	2682.1	2998.2	3156.0	3077.1	
NORTH EAST	3265.4	3382.1	3382.1	3498.7	3498.7	3498.7
TOTAL	24688.5	26770.5	29467.0	22874.8	37291.8	40959.1

PEAK LOAD TIME IS 1600 HOUR

DOORS: U = 0.087Btulh (See reference No. 11, Table 8)

= q = U.A. CLTD

S. E (Double door) = 0.087 x 33.75 x 39 = 37.58

N.W (single) 0.087 x 18 x 24 114.51

E " 0.087 x 18 x 38 59.5

211.59Btu / h

INTERNAL SOURCES:

(1) People; Assumption (300)

Sensible cooling load = no of people x SHF x CLF

300 x 230 x 1 = 69000 Btu / h

Latent cooling load = no of people x LHF x CLF

300 x 190 x 1 = 57000 Btu / h

(2) lights: Assumptions: light is on 12 hours

$q = \text{input} \times CLF \times \text{use-factor} \times \text{special allowance factor} \times 3.4$

$= 1200 \times 1 \times 3.4 \times 1 \times 1.2 = 4896 \text{ Btu / h}$

CLF = 1 because cooling system operate only during occupied hours.

Ventilation:

q_s (sensible heat load) $= 1.0 \times cfm \times \Delta t \times 1 - BF$

$$1.10 \times 300 \times 25 \times (89 - 77) \times (1\text{-}0.2)$$

$$79200 \text{Btu / h}$$

$q_l = 4840 \times cfm \times \Delta w \times 1 - BF$

$$4840 \times 300 \times 25 \times (0.0182 - 0.0.12) \times 1 - 0.2$$

$$= 203280 \text{ Btu/h}$$

OUT SIDE AIR BYPASS LOAD

$Q = 1.10 \times cfm \times \Delta t \times B.F$

$1.10 \times 300 \times 25 \times (98 - 77) \times 0.2 = 19800 \text{Btu / h}$

GRAND TOTAL HEAT LOAD

SENSIBLE HEAT LOAD	
People	69000
Light	4896
Ventilation	79200
Outside Air-bypass	19800
Roof	13294.1
West Glass	7796.3
South Glass	3644.3
Walls	16224.4
Door	211.59
	214066.69

LATENT HEAT LOAD

People	57000
Ventilation	203280
	260280

Grand total heat load = 214066.69 + 260280

= 474346.69

Refrigeration tonnage = $= \frac{474346.69}{12000}$

39.5 tons

40 tons

Ducting Analysis:

Using the room sensible heat obtained from design load estimation:

$q = \quad RSH$

$$1.08 \times \Delta\ t = \frac{214013.3}{1.08 \times 20} = 9908.02\text{cfm}$$

$\Delta t = 20^0c$

Total number of outlet = 14

Air quantity per outlet $= \frac{9908.02}{14}$

$= 707.7$ cfm

$= 0.334\text{m}^3/\text{sec}$

Using Table 13 of reference No. 7, the recommended maximum duct velocity for low velocity duct system (fpm) is obtained. Assumption: auditorium I chose;

Main duct: Supply = 1200 fpm

Return = 1000 fpm

Area of duct:

$$\frac{\text{air Quantity}}{\text{velocity}} (Q = \frac{9908.02}{1200} = 8.26\text{ft}^2\ (0.7674\text{m}^2)$$

EQUIVALENT CIRCULAR DIAMETER OF THE MAIN DUCT

$$D \frac{= 4 \times 8.26^{\frac{1}{2}}}{\Pi} \qquad = 3.24\text{ft}$$

$= (0.98\text{m})$

Friction loss in duct:

This is obtained by using diameter of main duct with the total air quantity on chart 1 of carrier handbook on air conditioning and ducting analysis.

0.045 in wg (water guage) per 100ft.

$= 11.21\text{N/m}^2$ (84075 mmHg per 30.48m)

(i.e.) 0.37N/m^2 per meter.

Perimeter of duct: (obtained form architectural plan)

= Length of Arc + perimeter of rectangle – width

85.21 + 154 – 55 = 184.21Ft (56.15m)

See fig (1) for dimension

STATIC PRESSURE IN THE DUCT

= Friction loss per meter x perimeter of duct

$0.37/lm^2 \times 56.15m = 20.78N/m^2$

VELOCITY PRESSURE:

Using table 14 reference No. 7 for low velocity of 1200 fpm the corresponding velocity pressure is 0.09 in wg. $= 22.41N/m^2$ (see conversion table in Appendix)

Total pressure in the duct = $P_T = Pv + Ps$

$$P_T = (22.41 + 20.78)\ N/m^2 = 43.19N/m^2$$

$$\text{Fan capacity} = \frac{RSH}{1.08 \times \Delta t} = \frac{214013.3}{1.08 \times 20} = 9908.02\text{cfm}\ (4.67m^3/\sec)$$

OUTLET VELOCITY: Using table 15

Velocity range is between 500 – 750 fpm, 750fpm was chose so as to remove the heat load from the space faster.

$$\text{Area of outlet } (A_0) = \frac{\text{Air quantity per outlet}}{\text{Outlet velocity}} = \frac{708}{750}$$

$$= 0.94Ft^2\ (0.087m^2) \text{ see conversion table}$$

From table 21 wall outlet rating, the only nominal size that could be used is 16" x 12" from carrier handbook.

The duct area are calculated using table (17) and duct sizes are determined from table (18). This is shown in table 1.2.

TABLE I.2

Duct section	Air quantity cfm	Cfm capacity %	Duct area %	Area (sq.ft)	Duct sizes
AHU – A	9908	100	100	8.26	35 x 35
A – 1	4954	50	58.0	4.79	34 x 22
1 – 2	4246	43	51.0	4.21	30 x 22
2 – 3	3538	36	44.0	3.63	28 x 20
3 – 4	2830	29	36.5	3.02	26 x 18
4 – 5	2122	22	29.5	2.44	24 x 16
5 – 6	1414	14	20.5	1.69	18 x 14
6 – 7	706	7	11.5	0.95	18 x 14
A – 8	4954	50	58.0	4.79	34 x 22
8 – 9	4246	43	51.0	4.21	30 x 22
9 – 10	3538	36	44.0	3.63	28 x 20
10 – 11	2830	29	36.5	3.02	26 x 18
11 – 12	2122	22	29.5	2.44	24 x 16
12 – 13	1414	14	20.5	1.69	18 x 14
13 –14	706	7	11.5	0.95	18 x 14

EQUIPMENT SELECTION

1. Condenser:

Based on the estimated refrigeration tonnage and the type of cooling required for the condenser the suitable condenser chosen is ELECTRA MODEL ECC 35 centrifugal air cooled condenser. (Reference No. 10) Table (19) the description and dimension are contained in the manual

2. Air Handling Unit:

This is obtained from CCEA quick selection by Reference No. 11, table 25, using the fan capacity of 9908cfm (4.67m^3/ sec)

3. Compressor Unit:

This selection is done using refrigeration tonnage of 40tonnes. Using Reference No. 12, table Recommended refrigerant is R12 because of its availability.

RESULTS AND DISCUSSION

Considering the standard conditions necessary room comfort (24 ^{O}C -25.5 ^{O}C, 50% - 55% RH) the proposed system (New design) would provide more comfort for the occupant all things being equal. Table 1.5 confirms this statement.

ISSN 1595 – 7578

In Table 1.4 the word actual is referring to the existing system performance characteristic while the word "simulated" refers to the proposed system (New design)

TABLE 1.3 TEST RESULT OF EXISTING CENTRAL AIR-CONDITIONING SYSTEM UNIT INDOOR DRY AND WET BULB TEMPERATURE AT 10 MINUTES INTERVAL

	DRY BULB TEMPERATURE (^{O}C)	WET BULB TEMPERATURE (^{O}C)	RELATIVE - HUMIDITY (%)
1	26.0	23	78
2	25.0	22	77
3	24.5	21.9	80
4	24.0	21.0	77
5	24.0	21.0	77
6	25.5	21.5	70
7	25.0	21.0	67
8	25.0	21.0	67

AVERAGE DRY BULB TEMPERATURE = 24.8 ^{O}C
AVERAGE WET BULB TEMPERATURE = 21.5 ^{O}C

TABLE 1.4

DESIGN CONDITIONS	ACTUAL	SIMULATED
Indoor dry bulb temperature	24.8^{O}C	25 ^{O}C
Outdoor dry bulb	29^{O}C	31.6 ^{O}C
Inside relative humidity	72%	55%
Indoor wet bulb temperature	21.5^{O}C	19 ^{O}C
Outdoor specific humidity	0.0184	0.0182
Indoor specific humidity	0.015	0.0112

Table 1.5

DESIGN CONDITIONS	ACTUAL	SIMULATED	STANDARD
Indoor dry bulb temperature	24.8 ^{O}C	25 ^{O}C	25.5 ^{O}C
Outdoor dry bulb temperature	29 ^{O}C	31.6 ^{O}C	31.6 ^{O}C
Inside relative humidity	72%	55%	55%
Indoor wet bulb temperature	21.5 ^{O}C	19 ^{O}C	19 ^{O}C
Outdoor specific humidity	0.0184	0.0182	0.0182
Indoor specific humidity	0.015	0.0112	0.0112

CONCLUSIONS AND RECOMMENDATIONS

It can be rightly concluded considering the information from tables1.3-1.5 that the new system will provide more comfort for the occupant in the chamber when compared with existing unit. The results obtained from the proposed system and the conditions necessary for comfort are very close.

Using the estimated refrigeration tonnage obtained (40tons), the following equipment were recommended.

a. Air Handling Unit (AHU) 3SE4 40 tons AHU by Reference No. 11, table 25,

b. Compressor: 40tonnes, 06L- 50Hz compressor by Reference No. 12.

c. Condenser: 40tonnes centrifugal condenser –type EEC, MODEL ECC 35 by Reference No. 10.

Also based on the ducting analysis the total outlet required in the senate chamber is 14. Two 90 degree elbows is required for ducting layout. The mounting of this equipment requires 2cm diameter cupper pipe, 45 meter for the total length of the pipe is estimated. The low-pressure side of the compressor is to be lagged using polyethylene material so that the heat load removed from space will not be increase, as a result of interaction with the surrounding, thereby increasing the workload on the chosen compressor.

REFERENCES

1. W. P Jones (1980), "Air conditioning Application and design", third edition by Edward Arnold, A division of Hodder and Stoughton London New York, Melbourne Auckland

2. W. F. Stoecker (1958) "Refrigeration and Air conditioning" by McGraw Hill Book Company. New York Toronto, London.

3. Norman .C. Harris (1983) "Modern Air Conditioning Practice" Third Edition McGraw Hill Book Company. New York.

4. The New Book of Knowledge A Volume 1 (1972) by Grolier incorporated, New-York. Copyright.

5. The World Book Encyclopedia A Volume 1 (1982) by World Book Childcraft International, Income. Merchandise Mart Plaza, Chicago, Illinois 60654, U. S. A.

6. Code of Practice, Design and Installation for Comfort Air-Conditioning in Buildings. By The Nigerian Society of Engineers (1964) P. O. Box 2299 Lagos, Nigeria.

7. Carrier Air Conditioning Company (1968) "Carrier System Design Manual" Eight Edition

8. ASHRAE Handbook 1985 Fundamentals by American Society of Heating, refrigeration and air- conditioning engineers Inc. 1791 Tullie Circle, N. E, Atlanta, GA 30329.

9. ASHRAE Handbook 1981 Fundamentals by American Society of Heating, Refrigeration and Air- Conditioning Engineers Inc. 1791 Tullie Circle, N. E, ATLANTA, GA 30329.

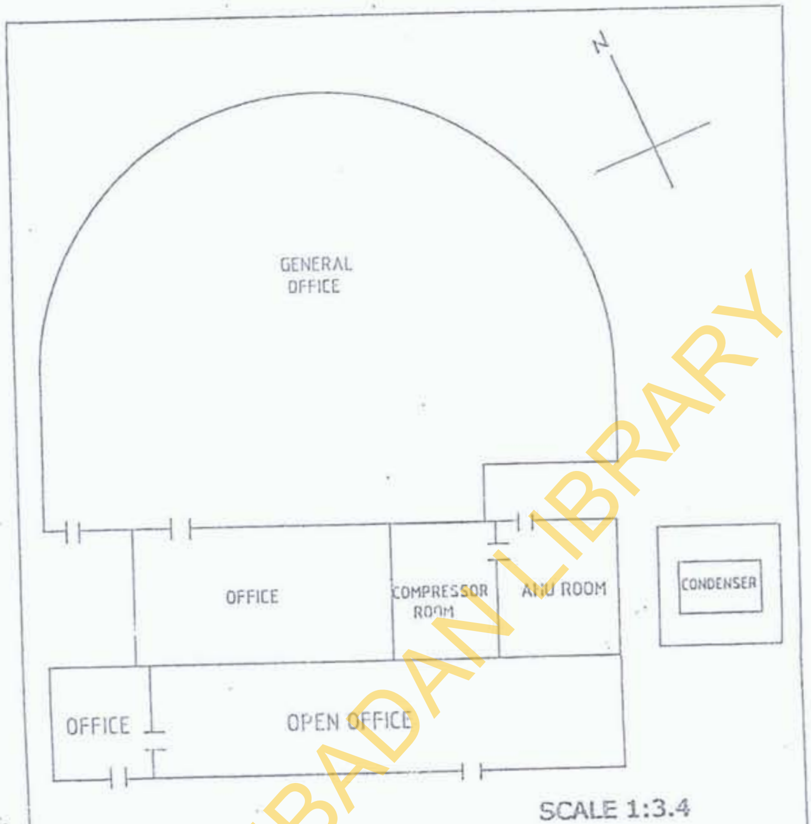

Fig1.0 SENATE CHAMBER GROUND FLOOR PLAN

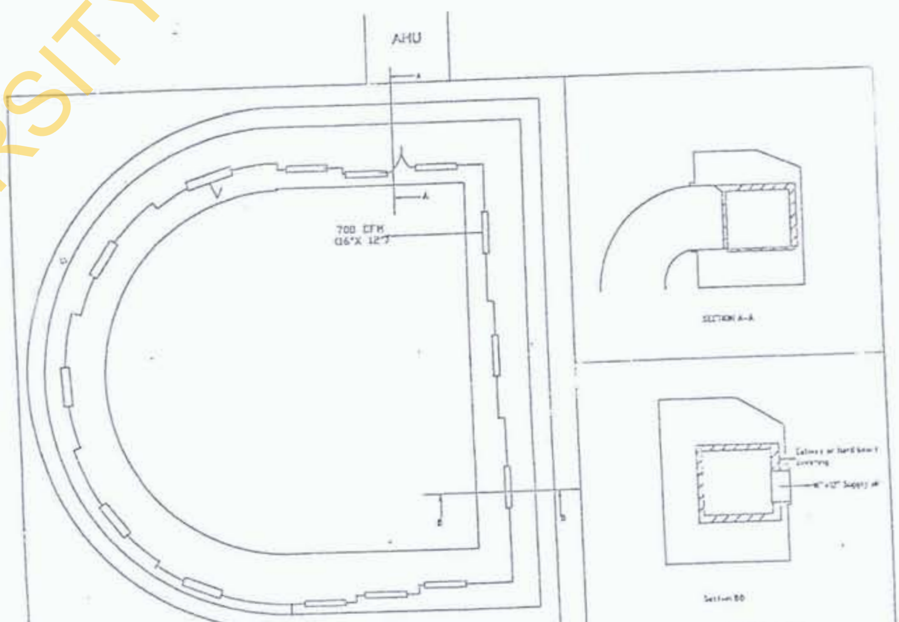

Fig2.0 SENATE CHAMBER DUCT LAYOUT

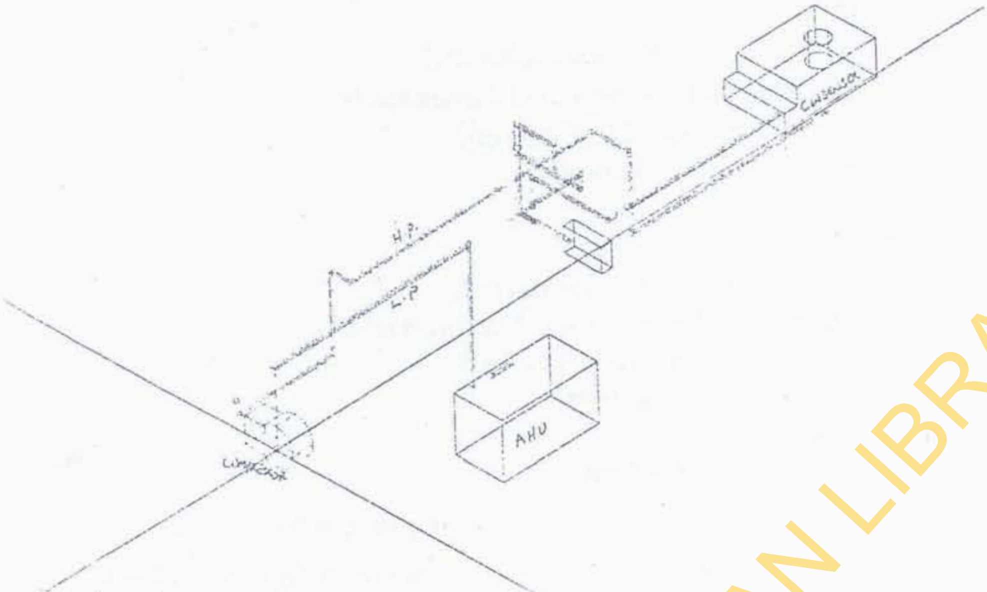

Fig 3.2 EQUIPMENT ARRANGEMENT PIPING LAYOUT SCALE 1:34