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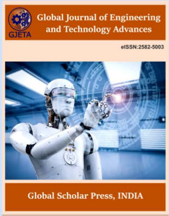

(Research Article)

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Numerical analysis for the design of a central air conditioning system for center of excellence

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Abstract

The temperature of the towing tank floor of the Center of Excellence in Marine and Offshore Engineering, Rivers State University needs regulation due to the high level of humidity during experiments. Since vapor evaporates from the towing-tank liquid column, the need for central air conditioning is timely. With the temperature regulated, it will help enhance the comfort of the students and improve the working condition of the staff and equipment. This is achieved by conducting a cooling load analysis, consistent with ASHRAE Guidelines, on the floor plan of the Laboratory building that provided useful information on the proper selection and sizing of fit-for-purpose air-conditioning equipment. Analysis revealed a total sensible and latent heat load of 59195.74 W with system capacity of 16.91TR (tons). Duct design utilizing the Velocity Reduction method resulted in a main supply duct length of 57m and flow are of 0.72m^2 incorporating four branched ducts measuring 6m each in length and flow area of 1.05m^2

Keywords: Ceramic Porcelain; Humidity; Latent heat; Sensible; Towing-tank

1. Introduction

The basic science of air conditioning has been known for many years, with refrigeration defined as the process of heat removal, and as that branch of science that deals with the process of reducing and maintaining the temperature of a space or material below the temperature of the surroundings. The temperature ranges within which people feel comfortable depend upon; clothing worn, the physical activity and the amount of moisture present in the atmosphere. There are four atmospheric conditions, which affects human comfort such as temperature of the surrounding air, the humidity of air, air purity and air movement. Hence, Air conditioning is defined as the process of producing air, controlling simultaneously its temperature, humidity, cleanliness, and distribution for the fulfillment of required condition of the confined space. Air conditioning is a process that simultaneously conditions air, distributes it combined with the outdoor air to the conditioned space; and at the same time controls and maintains the required space's temperature, humidity air movement, air cleanliness, sound level and pressure differential within predetermined limits for the health and comfort of the occupants, or for product processing or both [1].

The acronym HVAC&R stands for heating, ventilating, air-conditioning, and refrigerating. The combination of this process is equivalent to the functions performed by air-conditioning. An air conditioning system or HVAC&R system consists of components and equipment arranged in sequential order to heat or cool, humidify, or dehumidify, clean and purify, alternate objectionable equipment noise, transport the conditioned outdoor air and recirculate air to the conditioned space, and control and maintain an indoor or inclusive environment. The main objectives of this research work is to design a central air conditioning system for the towing tank floor of the new Marine Engineering laboratory in Rivers State University, Centre of Excellence (CoE). To achieve the above, the following shall be done; to have a site survey of the new marine laboratory to discover opportunities and limitations, to have the floor plan of the new marine

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laboratory in order to estimate the cooling loads of the air conditioning system to design with the laboratory and to design. The supply main duct and the supply branch duct of the air conditioning system. Due to the architectural plan of the new marine laboratory as the towing tank floor is the ground floor of the building, hence the design of the central Air Conditioning system will be limited to the ground floor plan. This research will also calculate the cooling loads of the new marine laboratory and design the duct for the air conditioning system.

1.1. HVAC System

The goals of HVAC system can be narrowed down to control temperature, fresh air circulation, air filtration and efficiency/economy. With the cost of fuel, HVAC system must also be efficient and economical. Energy efficient HVAC System might incorporate; Variable air volume designs, components designed for operation at low pressures, digital control systems, mechanisms for reclaiming heating/cooling benefits from re-circulated air and high efficiency ECM fan motors. The IVAC system installed in a commercial or office space differs greatly from the heating and cooling system which we are familiar with. Although the air handling equipment is centralized, different rooms and regions in the building will have different needs for heating and cooling, these needs are called loads because they place a load or demand on the HVAC system. These loads can come from the equipment, people, weather, and many other factors. In the simplest case, addressing this is by providing a constant supply of cool air which is then managed by the HVAC distribution system. Hence the basic parts of an HVAC system include the following.

- Chiller/Air conditioner

The chiller or air conditioner utilizes heat exchangers and circulated fluid or gas to cool the air that is passed through it.

- Air Handler

The air handler is a fan or blower that moves air throughout the building's ductwork. Axial or centrifugal fan types may be found in the air handler.

- Air Filters

Depending on the requirements of the occupants and the activities in the building, various grades of air filters may be used in the down-stream ductwork.

- Duct work

Round squared and rectangular ductwork provides a passage for the conditioned air for the air-handling unit to the environment.

- Damper

A damper consists of one or more blades which can be used to control the amount of air flow through a duct. Manual dampers are used to ensure that different parts of the building receive proportional ventilation, based on area and demand. Automatic dampers may be installed at fire-walls which closes in the case of a fire.

- Terminal units

A terminal unit is a device that uses an automated damper to control the amount of air which is delivered to a room or region, the damper is typically controlled by an electric or digital actuator, which is regulated by a thermostat.

- Zones

A set of regions in the buildings which have identical heating and air conditioning needs are called zones. It is typical to assign at least one terminal unit and corresponding thermostat to each zone.

- Heating Coils

Heating coils may be installed after a terminal unit to provide heat on a zone-by-zone bases. Buildings with a higher number of occupants may not require extensive heating even during the coldest months of the year.

Heating coils offer an efficient way of providing heat for those few areas that require heat and are controlled by the same thermostat system as the terminal unit.

- Linings /Attenuators

Various types of linings are available to dampen noise within a duct. A short length of line duct work called an attenuator is often installed following the terminal unit to dampen discharge noise.

Ultimately the test of an HVAC system lies in its ability to deliver conditioned airs to the occupants. Air from the duct work enters the occupant's space through grilles, registers or diffusers often called GRD's.

- Grilles

The term grille is commonly implied to any air outlet or intake that consists of a square or rectangular face and neck and whose facial appearance is made up of the louvers which may be used to deflect the air.

- Registers

A register is a grille that has one or more adjustable plates or dampers that controls the amount of air that flows through.

- Diffusers

A diffuser is an air outlet, which incorporates structures such as veins, louvers, proliferation, and other features for distributing and directing air. The diffuser job is to direct the airflow throughout the occupied space in the most efficient manner possible.

Once air enters the space, it circulates through the return inlets and returns to the air-handling unit unlike the air outlet, a return requires no sophisticated veins. However, the relative location of the air inlet and outlet can be critical to the efficiency of the system upon returning to the air handler, a certain portion of the return air is exhausted and replaced with fresh exterior air. In an average office building or commercial space, it is approximately 10 – 20%.

There are two basic types of central air conditioning system: central unitary and field erected. Design for the main and branch duct of the air conditioning system.

2. Materials and method

2.1. Site Survey

Site survey is a very important step in designing a central AC System. The site survey will give the following information. The true cooling or heating requirement includes: the possibilities of greatest load reduction with least cost, the most economical equipment selection and location and the most efficient air distribution [2].

2.2. Information Collected Under Site Survey

Physical plan and elevation of the area to be air conditioned. Figure 2 shows a building for the department of marine engineering in Rivers State University. The building has been built to standard to carry out marine engineering experiments and tests. Such as Stability test, ship proportion text, wave test etc. the building was also designed and built with classrooms for undergraduate and post graduate students, offices for lecturers and laboratories for experiment. It is sited in a way that none of the structures/facilities surrounding it provides a shade from the sun.

2.3. Occupancy

An approximate figure for occupancy is not known as the towing tank floor will only be used when necessary. Hence to estimate the average number of occupants, consideration will be given to the average number of final year students in undergraduate level, the average number of students in the Postgraduate level, the number of lab attendants/operators and the number of staffs/lecturers.

- The central unitary systems are used mainly in new or retrofit home installations. They may also be used in small commercial installations. The primary advantage in using the unitary system is the low cost and the ease of installation.
- the field erected systems are frequently used in large commercial structures. They may also be used to heat and cool various sections of a large building [3]

From the above literature review it is important to state that an efficient design of the central air conditioning system for the towing tank floor (ground floor) of the CoE center on the following: Site survey of the new marine laboratory and Cooling load analysis and calculation.

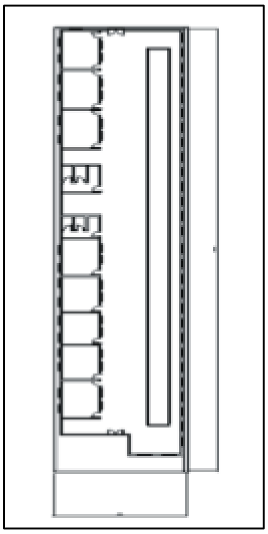

Figure 1 Plan view of the towing tank floor (ground floor) of the CoE

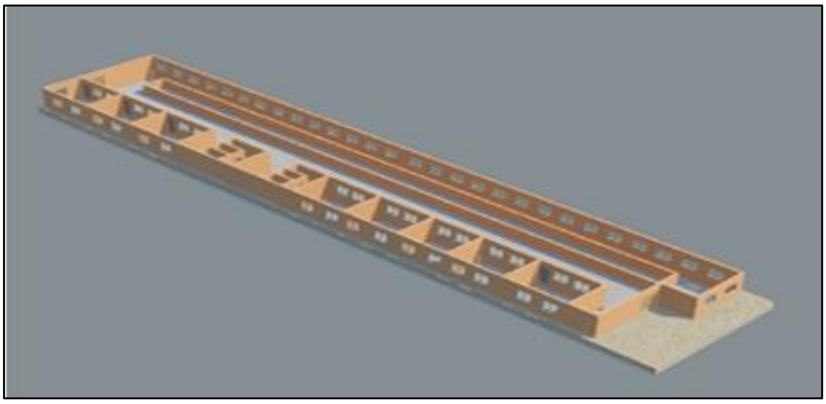

Figure 2 Elevations of the towing tank floor (ground floor) of the CoE.

2.4. Cooling Load Estimation

Cooling load is the total heat that is required to be removed from a space to bring it to the desired temperature by the air conditioning and refrigeration equipment [4].

Load estimation is necessary to determine the size of air conditioning and refrigeration equipment which is required to maintain the inside design conditions during periods of maximum outside temperatures.

2.5. Component of a Cooling Load

There are two main component of a cooling load which are imposed on an air conditioning plant, operating in summer (during hot weather) and they are;

- Sensible Heat Gain

Sensible heat gain is said to occur when there is a direct addition of heat to the enclosed space. This heat gain may occur by any one or all of the following sources of heat transfer.

- The heat flowing into the building by conduction through exterior walls, fans, ceilings, doors and windows due to the temperature difference on their two sides.
- The heat received from solar radiation, which consist of
- The heat transmitted directly through glass of windows, ventilations or doors.
- The heat absorbed by walls and roofs exposed to solar radiation and later on transferred to the room by

conduction.
- The heat conducted through interior partition from rooms in the same buildings, which are not conditioned.
- The heat given off by lights, motors, machinery, cooking operations, industrial processes etc.
- The heat liberated by the occupants.
- The heat carried by the outside air, which leaks in through cracks in doors, windows, and through their frequent openings.
- The heat gains through the walls of ducts carrying conditioned air through unconditioned space in the building.
- The heat gains from the fan work.

- Latent Heat Gain.

When there is an addition of water vapor to the air of enclosed space, a gain in latent heat is said to occur the heat gain may occur due to one or all of the following sources.

- The heat-gain due to moisture in the outside air entering by infiltration.
- The heat-gain due to condensation of moisture from occupants.
- The heat-gain due to condensation of moisture from any process such as cooking foods which takes place within the conditioned space.
- The heat-gain due to moisture passing directly into the conditioned space through permeable walls or partitions from the outside or from adjoining regions where the water vapor pressure is higher.

The total heat load to be removed by the air-conditioning and refrigeration equipment is the sum of the sensible and latent heat loads. The sensible heat gain is due to the temperature difference between the fresh air and the air in the space whereas the latent heat gain is due to the difference in humidity.

2.5.1. Sensible Heat Gain Through Building Structure by Conduction

A building wall compound of a single homogeneous material shows that the heat passing through the wall is first received at the wall's surface exposed to the region of higher air temperature by radiation convection and conduction. It then flows through the material of the wall to the surface exposed at the region of a lower air temperature, and the heat is dispersed through the process of radiation, convection, and conduction. Thus, the heat transferred or gained through a wall under steady state condition is stated according to

$$Q = f_o(t_o - t_1)A + {}^{k}/_{x}\,(t_1 - t_2)A + f_1(t_2 - t_1)A \;\ldots\ldots\ldots\; (1)$$

$$= UA(t_o - t_1) \quad \ldots\ldots\ldots\ldots\; (2)$$

Where:
f_o = outside film or surface conductance
f_1 = inside film or surface conductance
A = outside area of wall
t_o = outside air temperature
t_1 = inside air temperature
x = thickness of wall
k = thermal conductivity for the material of the wall
U = overall coefficient of heat transmission of the wall [4].

2.6. Heat Gain from Solar Radiation

The amount of heat that flows toward the interior of a building due to solar radiation depends upon the following factors.

- Altitude angle of the sun
- Clearness of the sky
- Position of the surface with respect to the direction of the sun's ray
- Absorptivity of the surface
- Ratio of the overall coefficient of heat transfer of the wall to the coefficient of heat transfer of the outside air film.
- Temperature of the ground and surrounding objects with which the heated surface may interchange radiant heat.

The heat from the solar radiation is received by the building surfaces in two forms. i.e. direct radiation or sky or diffuse radiation.

The direct radiation is the impingement of the sun's rays upon the surface. The sky or diffuse radiation is received from moisture and dust particles in atmosphere which absorbs part of the energy of the sun's rays thereby becoming heated to a temperature above that of the air. The sky radiation is received by surfaces which do not face the sun.

2.7. Solar Heat Gain (Sensible) Through Outside Walls and Roofs

The transmission of heat through the walls exposed to the outdoors and roofs is not steady (i.e., the flow of heat is periodic) due to variation in the outside air temperature and the solar radiation intensity over a period of 24 hours. A little consideration will show that the temperature of walls rises with the rise in outside air temperature and the heat is stored in the wall which has a considerable storage capacity. Thus, the heat transferred to the room is reduced. The stored heat in the wall is given off to the room when the outside air temperature falls. Since the outside air temperature changes continuously every 24 hours, hence instantaneous heat gain from outside is not equal to the instantaneous heat gain inside the room. And the difference being stored or rejected by the wall.

The heat stored by the wall is given off later in the evening. Figure 3. shows the curves of instantaneous load coming from outside and the actual load felt inside. The area under the two curves is equal and the shaded area above the actual load shows the heat stored and below the actual load shows the heat released by the walls and other structures. The heat gain through outside walls and roofs is given by; [4].

$$Q = UAt_e \quad \text{……….} (3)$$

Where:
U = Overall heat transmission coefficient of roof or wall
A = Area of roof or wall
t_e = Equivalent temperature differential

2.8. Solar heat gains through glass areas

The heat gain through glass constitutes a major portion of the load on the cooling apparatus. When a sheet of glass is subjected to solar radiation (direct and diffuse) a part of it is absorbed, a part is reflected, and the remaining is transmitted directly to the interior of the building. In the absorption process, the temperature of the glass increases, until it can lose heat the same rate in an interchange of energy with surface inside and outside of the building. In addition to the radiant effect, the net heat gain into the interior of the building through a sheet of glass is affected by conduction. The complete net balance can be written as follows.

Net heat gain = transmitted solar radiation + heat flow by convection and radiation heat exchange between glass and indoor surface. The solar heat gain through glass areas varies from hour to hour from day to day and from latitude to latitude.

2.9. Heat gain due to infiltration

The infiltration air depends upon the temperature difference between the inside and outside air. There are two methods of estimating the infiltrated air

- Crack length method
- Air change method.

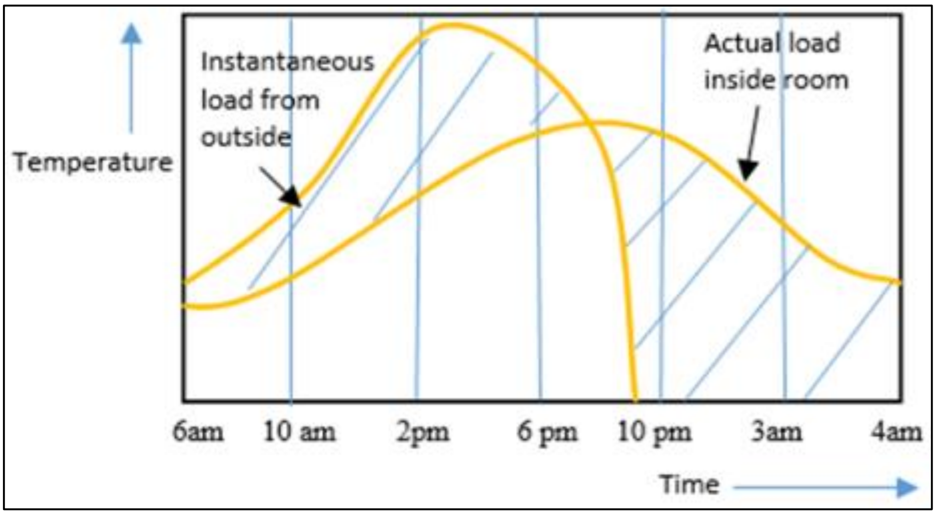

Figure 3 Curves of instantaneous load [4]

Table 1 Overall Coefficient of Heat Transmission (U) for Structures with a Wind Velocity of 24km/h Outside [4]

Structure	**Overall coefficient of heat transmission (U) in W/m² K**
Brick wall, 20cm, bare	2.84
Brick wall, 20cm, plaster one side of brick	2.61
Brick wall, 20cm, plaster one side on metal lath-furred	1.82
Brick wall, 40cm, bare	1.59
Brick wall, 40cm, plaster one side of brick	1.53
Brick wall, 40cm, plaster one side on metal lath-furred	1.20
Hallow tile, stucco exterior, 20cm, bare.	2.27
Hallow tile, stucco exterior, 20cm, plaster on metal lath furred.	1.59
Hallow tile, stucco exterior, 30cm, bare.	1.70
Hallow tile, stucco exterior, 30cm, plaster on metal lath furred.	1.25
Cinder blocks, 24cm, bare	2.38
Cinder blocks, 20cm, plaster one side on metal lath-furred	1.59
Concrete blocks, 20cm, bare	3.18
Clap-board frame construction, plaster on wood lath	1.42
Wood shingle frame construction, plaster on wood lath	1.42
Stucco frame construction, plaster on wood lath	1.70
Brick veneer frame construction, plaster on wood lath	1.53

Table 2 Thermal Conductivity (K) of building materials [4]

Material	**Thermal conductivity (k)**

	in W/m K
Brick, low density	8.66
Brick, high density	15.93
Cement mortar	20.82
Cement plaster, typical	20.82
Concrete, typical	20.82
Concrete, typical fiber gypsum, 87.5% gypsum, 12.5% wood chips.	1.61
Sand and gravel	21.86
Limestone	18.72
Cinder	8.50
Cinder, boiler (12.5mm to 19mm)	2.13
Title or terrazzo, typical	20.82
Asbestos building board	4.71
Gypsum between layers of heavy paper	2.44
Gypsum, plaster, typical	5.72
Wood across grain, typical	1.73
Balsa	0.657
Maple, hard, 16% moisture	2.0
Pine, short yellow leaf, 16% moisture	1.45

Table 3 Thermal Conductivity (K) of insulating materials [4]

Material	**Thermal conductivity (k) in W/m K**
Asbestos packed	2.812
Asbestos loose	1.852
Asbestos, paper, thin layers' organic binder	0.85
Asbestos mill board	1.454
Asbestos wood	4.85
Asphalt roofing (felt)	1.212
Balsa	0.607
Cement wood (saw dust and Portland cement)	1.683
Charcoal (hard woods) coarse	0.617
Cork board, typical	0.468 to 0.486
Cork re-granulated, coarse	0.536
Cork re-granulated	0.468
Cotton	0.676
Diatomaceous earth	0.460

Eel grass	0.412
Glass wool, high grade	0.388 to 0.460
Glass wool, commercial grade	0.444
Hair felt, not compressed.	0.444
Insulation boards, various fibres	0.553 to 0.657
Kapok, loosely packed	0.416
Planer shaving, various woods	0.692
Rock wool	0.468 to 0.486
Rubber expanded.	0.364
Saw dust, various woods	0.710

The crack length method is usually used where greater accuracy is required. In most cases, the air change method is used for calculating the quantity of infiltrated air.

$$Q_s = 1.2Q\Delta t \text{………} \qquad (4)$$

The amount of infiltrated air through windows and walls is;

$$Q = \frac{L \times W \times H \times A_c}{60}\ m^3/min \quad \text{……….} \qquad (5)$$

Where Q_s = Sensible heat
Q = Volumetric flow rate
Δt = The temperature difference between the outdoor and inside temperature $(t_o - t_1)$ °C
L = Room length in meters
W = Room width in meters
H = Room height in meters
A_c = Air changes per hour [4]

For each person passing through a door leading to the outside or to an unconditioned space, the values given in the following table 5 for the door infiltration should be added to the infiltration air through windows and walls to find the total building infiltration.

The heat gain due to infiltration from doors is expressed as;

$$Q = U \times A(t_R - t_S) \qquad \text{…………..} \qquad (6)$$

Table 4 Number of Air changes per hour [4]

Kind of room or building	**Number of air changes per hour (A_c)**
Rooms with no windows or outside doors	0.5 to 0.75
Rooms, one wall exposed.	1
Rooms, two wall exposed.	1.5
Rooms, three wall exposed.	2
Entrance hall	2 to 3
Reception hall	2
Bath rooms	2

Table 5 Infiltrated Air [4]

Usage of door	Infiltrated air for 1.8 m revolving door in m^3 per person per passage	
	Freely revolving door	Door equipped with brake
Infrequent	2.5	2
Average	2	1.75
Heavy	1.5.	1.25

2.10. Heat gained due to ventilation

The ventilation (i.e. supply of outside air) is provided to the conditioned space in order to minimize odor, concentration of smoke, carbon dioxide and other undesirable gases so that freshness of air could be maintained. The quantity of outside air used for ventilation should provide at least one-half air change per hour in buildings with normal ceiling heights. Also, if the infiltration air quantity is larger than the ventilation quantity, then the later should be increased to at least equal to the infiltration air. The outside air adds sensible as well as latent heat.

2.11. Heat gain from occupants

The heat gain from occupants is based on the average number of people that are expected to be present in the conditioned space. The heat load produced by each person depends upon the activity of the person.

Table 6 Outside air required per person [4]

Application	Smoking	Outside air on m^3/min/person	
		Recommended	Minimum
Departmental stores	None	0.23	0.15
Factories	None	0.3	0.23
Meeting room	Very heavy	1.5	0.9
Offices, general	Some	0.45	0.3
Offices, private	None	0.75	0.45
Theatres	None	0.23	0.15

2.12. Heat gain from lighting equipment

The heat gain of electric light depends upon the rating of lights in watts, use factor and allowance factor. Mathematically the heat gain from the electric lights is given by; [4]

$$Q = Total\ Wattage\ of\ light\ \times Use\ factor\ \times allowance\ factor \ldots\ldots\ldots \qquad (7)$$

The use factor is the ratio of actual wattage in use to the installed wattage. Its value depends upon the type of use to which the room Is put. In case of residence, commercial stores and shops, its value is usually taken as unity (1), whereas for industrial workshops it's taken below 0.5. the allowance factor is generally used in case of fluorescent tubes to allow for the power used by the ballast. Its value is usually taken as 1.25.

Heat gain through ducts

Heat gain due to supply-duct depends upon the temperature of air in the duct and the temperature of the space surrounding the duct.

$$Q_D = UA_D(t_a - t_s) \ldots\ldots\ldots\ldots\ldots\ldots \qquad (8)$$

Where U = Overall heat-transfer coefficient

A_D = Surface area of the duct
t_a =Temperature of ambient air
t_s =Temperature of supply air

when all the duct is in the air-conditioned space, then the heat gain is zero. If the duct is located in an unconditioned space, then there is a gain in heat and condensation takes place. In order to prevent condensation and to reduce duct heat gain, the duct must be insulated. The heat gain through supply duct is roughly taken as 5% of the room sensible heat. The losses due to supply air leakage is not easy to estimate because it depends upon the workmanship of duct construction and the length of run. It has been found that the duct leakages are of the order of 5 to 30%. The following are the minimum recommended value of air leakages through ducts.

- Long runs - - - 10%
- Medium runs - - - 5%
- Short runs - - Neglect

Duct Design

In the design of air conditioning duct, three methods are commonly used which are;

- Velocity reduction
- Equal friction
- Static re-gain [2]

In this research efforts is made to focus on the velocity reduction method in designing the duct.

Table 7 Heat gain from occupants (in watts) [4]

Degree of activity	**Typical application**	**Total heat Adults-male**	**Total heat adjusted**	**Sensible Heat load**	**Latent Heat load**
Seated – at rest	Theatre – Matinee Theatre – Evening	114 114	97 102	53 57	44 45
Seated – very light work	Office, Hotels, Apartments	132	117	57	67
Moderate active office work	Office, Hotels, Apartments	139	132	59	73
Walking, Seated, Standing, walking slowly	Drug store, Bank	161	146	58	88
Sedentary work	Restaurant	144	161	64	97
Light bench work	Factory	234	220	64	155
Moderate dancing	Dance hall	264	249	72	177
Walking 4.8km/h, moderately heavy work	Factory	293	293	88	205
Bowling	Bowling alley	440	425	136	288
Heavy work	Factory	440	425	136	288

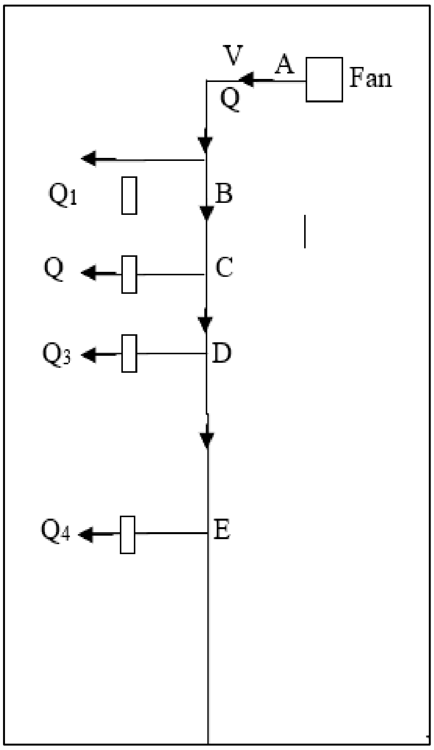

Figure 4a Duct layout

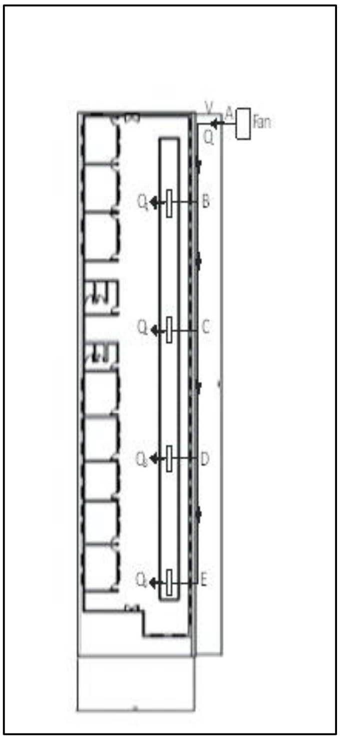

Figure 4b Duct layout placed on the plan

2.13. Velocity reduction method

Below are the steps to follow in designing the duct with this method.

- The velocity of the flow will be selected from Table 8. as shown in appendix A
- The capacity Q will be calculated by using the formula below.

$$Q = A \times V \text{........} \qquad (9)$$

Neglecting friction, the above relation gives

$$Q(How\ much) = A(How\ big) \times V(How\ fast) \qquad \text{........} \qquad (10)$$

Air supply requirements: the larger the air supply requirement in a conditioned space the larger will be the duct hence, the air supply requirement for each area must be carefully calculated. The air quantity required is calculated from the following equation (for standard air)

$$Q = RSH/[204.3(t_R - t_S)] \qquad \text{............} \qquad (11)$$

Where RSH = Room sensible heat load (watts)

t_R = Room DTB 0C

t_S = Supply air temperature 0C

- Q will be decided depending upon the requirements of the space to be air-conditioned.
- The velocities will also be selected from Table 8 as seen in appendix A
- And other values will be determined by using friction chart.
- Evaluate the losses due to area changes and hence calculate the losses due to change in direction using table 9 and table 10 from appendix B and C
- And finally, all the losses in the ducts will be added and the larger value will be adopted for fan selection [2].

3. Results and discussion

3.1. Internal sensible load calculation for the towing tank floor

3.1.1. Inside design condition

The inside design condition varies in accordance with the degree of activity of the occupants and intended use of the conditioned space.

The recommended inside design condition for comfort in the new marine laboratory in Rivers State University is given as;

- Dry bulb temperature 25.6 ^oC
- Relative humidity 60% [5].

3.1.2. Outdoor design condition

After taking the survey in section II, it was recorded that the highest mean monthly maximum outdoor dry bulb temperature for Rivers State University environment in the month of July was given as 32^oC. And this forms the basis for the outdoor design temperature.

The humidity ratio is calculated as follows.

The weight of water vapor for saturation is 0.454 of dry air.

3.1.3. Doors

The door used in the new Marine Engineering laboratory is a double exterior security steel door with U = 1.75 W/m^2 K which is gotten from table 5.

3.1.4. Floor construction

The floor is constructed with 152.4 mm concrete slab on grade.

3.1.5. Roof Construction

Steel with 25mm insulation with code number B5.

The thickness and thermal properties are;

- L = 25mm
- k = 0.043 W/m^oC
- Density = 91 kg/m^3
- SH = 0.233 KJ/Kg^oC
- R = 0.586 m2 $^oC/W$
- Mass = 2.3kg/m^2 [6]

The thermal conductance of the material is obtained by dividing the thermal conductivity of the material by its thickness.

$$C= K/L$$

Where; K = Thermal conductivity (W/m^oC)
L = Thickness of the material (mm)
C = Conductance (w/m^2°c)

$$= 0.043/0.0254=1.693 \text{ w/m}^2 \text{°c}$$

$$\text{Thermal resistance} \qquad R= 1/C = 1/1.693=0.591 \text{ m}^2 \text{°c/w}$$

$$\text{Hence;} \qquad U= 1/R = 1/0.591=1.69 \text{ w/m}^2 \text{ k}$$

3.1.6. Wall Construction

The wall has an inside finish of cement mortar, and the outside is finished with cement plaster and sand aggregate.

The following parameters were measured from the wall.

$$\text{Length} = 0.3048 \text{ m}$$

$$\text{Width} = 0.27$$

$$U_1 = 1/R_1 = 1/2944 = 0.34 \text{ w/m}^2 \text{ k}$$

$$\text{And } U_1 = U_2$$

3.1.7. Occupancy

Considering the assumptions made in section II, the average occupancy of the New Marine Engineering Laboratory in Rivers State University can be calculated using table 8 as shown below. The recommended outside air required per person as selected under factory from table 6 as 0.3

Therefore, the heat load produced by each person is.

$$Q = (0.3)/126 = 0.0024 \text{ m}^3/\text{min}$$

$$Q = (0.0024)/60 = 4\times10^{-4} \text{ m}^3/\text{s}$$

Table 8 The R Values Building Element in Determining Coefficient Heat Transfer

Element	R (Insulation)	R (Framing)
Outside surface exterior wind velocity for summer 3.4m/s	0.044	0.044
Inside surface (still air)	0.12	0.12
Inside finish (Cement motar)	1.39	1.39
Outside finish of cement plaster, Sand aggregate	1.39	1.39
Total	R_1=2944	R_2=2944

Table 9 Average Number of Occupants

Average number of Final year undergraduate students	**100**
Average number of post graduate students	15
Average number of Operators/Lab attendants	3
Average number of Staffs/Lecturers	8
Total	**126**

3.1.8. Appliances and light

From (7), the heat gain from the electric lights is given as;

$$Q = Total\ Wattage\ of\ light\ \times Use\ factor\ \times allowance\ factor$$

$$Q = W \times f_{ul} \times f_{sat}$$

It is considered that the towing tank floor uses a total of thirty-two (32), 26 Watts bulbs.

For industrial workshop such as the New Marine Engineering laboratory, the lighting use factor will be taken as 0.4 and allowance is 1.25.

$$Q=26 \times 32 \times 0.4 \times 1.25=416 \text{ W}$$

3.2. Infiltration

3.2.1. Main Hall

From (4), Infiltration into the main hall

$$Q_s = 1.2Q\Delta t$$

Where
Q = Volumetric flow rate (m^2/s)
Qs = Sensible heat (KW)

Δt = The temperature difference between the outdoor and inside temperature $t_0 - t_1$ °C

To find the air exchange; from Table 4 number of air changes per hour AC = 0.5. and from (5)

$$Q = \frac{L \times W \ \times H \times A_c}{60} \ m^3/min$$

Where;
L = 67 m
W = 13.3 m
H = 2.4 m

$$Q = (67 \times 13.3 \times 2.4 \times 0.5)/60=17.82 \text{ m\^{}3/min}$$

$$Q=17.82/60=0.297 \text{ m\^{}3/s}$$

$$Q_s=1.2 \times 0.297 \times (32-25.6) =2.281 \text{ W}$$

3.2.2. Occupancy

$$Q_s = no\ of\ occupants\ \times Total\ heat\ adjusted$$

From Table 7 the following can be obtained in terms of occupancy for Light bench work (Factory);

Total heat adjusted = 220 W

Average number of occupants = 126

$$Q_s=126 \times 220= 27720 \text{ W}$$

3.2.3. Toilets

Also, from (4);

$$Q_s=1.2 \text{ Q } \Delta\text{t}$$

There are two toilets in the New Marine Engineering Laboratory hence (5) becomes.

$$Q = \frac{L \times W \ \times H \times A_c}{60} \ m^3/min$$

Where; L = 8 m, W = 4 m, H = 2.4 m

$$Q=((0.5 \times 8 \times 4 \times 2.4)/60) \times 2=1.28 \text{ m\^{}3/min}$$

Q=1.28/60=0.0213 m^3/s

Q_s=1.2 ×0.0213×(32-25.6)=0.164 W

Area of Toilet Roof = 6 x 4 = 24m^2

3.2.4. Offices

Also, from (4);

$$Q_s = 1.2Q\Delta t$$

There is a total of seven (7) office spaces in the towing tank floor of the new marine laboratory hence, equation (5) becomes.

Q=((L ×W×H×A_c)/60)*7 m^3/min

Where; L = 6 m, W = 4 m, H = 2.4 m

Q = ((0.5 ×6×4×2.4)/60) × 7=3.36 m^3/min

Q = ((3.36)/60) × 7=0.056 m^3/s

Q_s=1.2 ×0.056 × (32-25.6) = 0.43 W

3.2.5. Glass (Windows)

The new marine laboratory has a total of 41 windows.

Area of one window = L X B = 1.5 x 1.2 =1.8 m

=L ×B = 1.5 × 1.2 =1.8 m^2

Therefore, total area of window

$$A_T = (L \times B) \times number\ of\ windows$$

= (1.5 x 1.2)×41 =73.8 m^2

Q=0.043×73.8×(32-25.6)=20.31W

3.2.6. Doors

North Door, Breadth = 1.74 m, Height = 2.1 m

Area; A=B×H = 1.74 ×2.1 =3.65 m^2

From (6);

$$Q = U \times A(t_R - t_S)$$

From table 3.5, u = 1.75 w/m2 k

Hence, Q=1.75×3.65 × (32-25.6) = 40.88 W

And South Door = North Door = 40.88 W

Total Latent Heat Load in the towing tank floor

The total latent heat load in the towing tank floor can be calculated as;

$$Q_{el}(total) = Occupancy + Electric\ Lighting$$

Where;
Occupancy=Q_{el}=126 ×220= 27720 W
Electric lighting= Q_{el}=26 ×32 ×0.4×1.25=416 W

Q_{el} (total)=27720+416=28136 W

RSHF=(RSHL)/RTHL

Where;
RSHL = Room sensible heat load
RTHL = Room Total heat load
RLHL = Room Latent heat load

But RTHL = RSHL + RLHL

Tons of refrigeration = (Total cooling load)/3500

Therefore, RSHF=(RSHL)/RTHL=29652.94/59195.74=0.5

Capacity of the plant=RTHL/3500=59195.74/3500 =16.91 TR

Hence, A plant having a capacity of 16.91 TR should be installed.

Table 10 Summary of the Total Sensible Cooling Load

Particulars	Sensible (W)	Latent (W)	Total (W)
North Door	40.88		40.88
South Door	40.88		40.88
Glass (Window)	20.31		20.31
Occupants	27720		27720
Toilets	0.164		0.164
Towing tank space (main hall)	2.281		2.281
Offices	0.43		0.43
Electric Lighting	416	28136	28552
Total	28240.9	28136	56376.9
Heat gain through duct (5% of total)	1412.04	1406.8	2818.84
Total	29652.94	29542.8	59195.74

3.3. Duct Design

Following the method stated and explained in section II in designing the duct and considering table 8 for industrial application, the velocity of the flow in the duct is selected as:

For the main duct.

Supply velocity=1000/60=16.67mls

for the branch duct.

Supply velocity=700/60=11.67mls

The capacity can be calculated by using (9)

$$Q=A\times V$$

Recall (11); $Q=RSH/(204.3\ (t_R-t_s))$

Where
RSH = Room sensible heat load (watts) = 29652.94 W
t_R = Room DTB 0C,
t_s = Supply air temperature 0C
Assuming the building temperature is to be maintained at 22oC dbt and the supply air temperature is 10oC [4]

Therefore.

$$Q=29652.94/(204.3\ (22-10))=12\ m^3/s$$

and making A the subject.

$$A=Q/V$$

Hence, Area for supply main Duct.

$$A=Q/V=12/16.67=0.72\ m^2$$

Area for supply branch Duct.

$$A=Q/V=12/11.67=1.03\ m^2$$

Since the duct will have four (4) diffusers, therefore, the flow rate for each diffuser.

$$=Q/(\text{no of diffusers})=12/4=3\ m^3/s$$

Simulation of the supply main duct and branch duct

Overall length of duct = 57 m

Length of branch duct = 6 m

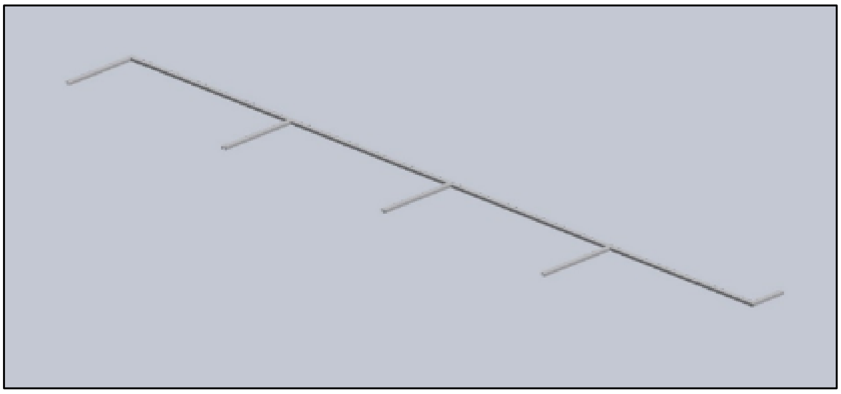

Figure 5 The Supply and branch ducts

4. Conclusion

The design of a central air conditioning system for the towing tank floor of the new Marine Engineering laboratory in Rivers State University is timely due to the high level of humidity as water evaporates from the towing tank liquid column. In designing the central Air Conditioning system, a proper site survey needs to be carried out which will aid in calculating the heat loads of the laboratory. The loads can be carefully obtained by calculating the sensible heat gain through the building structure, heat gain from solar radiation, solar heat gain through outside walls and roofs, solar heat gain through glass areas, heat gain due to infiltration, heat gain from occupants, heat gain from lighting equipment and heat gain through duct. The capacity of the air conditioning system to be installed can be estimated if the above heat load is known. With the capacity of the air conditioning system calculated, the duct is hence designed using the areas obtained from circulation and having in mind that pressure losses will occur due to the duct branches leading to the diffusers.

Compliance with ethical standards

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Disclosure of conflict of interest

No conflict of interest to be disclosed.

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APPENDICES

Appendix A: Recommended maximum duct velocities for low velocity systems (m/min) [2]

Application	*Controlling factor noise generation main ducts*	*Controlling factor duct friction*			
		Main duct		*Branch ducts*	
		Supply	*Return*	*Supply*	*Return*
Residences	200	300	250	200	200
Apartments Hotels Bedrooms Hospital Bedrooms	300	500	400	400	300
Private Offices Director's rooms Libraries	400	600	500	500	400
Theatres Auditoriums	250	400	400	300	300
General Offices, High Class Restaurants, High Class Stores and Banks	500	700	500	500	400
Average Stores Cafeterias	600	700	500	500	400
Industrial	800	1000	600	700	500

Appendix B: Pressure loss due to area changes [2]

Type	Illustration	Conditions	Loss Coefficient C_1	Loss Coefficient C_2
Abrupt expansion		A_1/A_2	C_1	C_2
		0.1	0.81	81
		0.2	0.84	18
		0.3	0.49	5
		0.4	0.36	2.25
		0.5	0.25	1.00
		0.6	0.16	0.45
		0.7	0.09	0.18
		0.8	0.04	0.08
		0.9	0.01	0.01
Gradual expansion		θ	C_r	
		5°	0.17	
		7°	0.22	
		10°	0.28	
		20°	0.45	
		30°	0.59	
		40°	0.73	
Abrupt exit	$A_2 = \infty$	$A_1/A_2 = 0.0$	1.00	
Square edge orifice exit		A_0/A_1	C_0	
		0.0	2.50	
		0.2	2.44	
		0.4	2.26	
		0.6	1.96	
		0.8	1.54	
		1.0	1.00	
Bar across duct		E/D	C	
		0.10	0.7	
		0.25	1.4	
		0.50	4.0	
Pipe across duct		E/D	C	
		0.10	0.20	
		0.25	0.55	
		0.50	2.0	
Streamlined strut across duct		E/D	C	
		0.10	0.07	
		0.25	0.23	
		0.50	0.90	

Type	Illustration	Conditions	Loss coefficient
Abrupt contraction square edge		A_2/A_1	C_2
		0.0	0.34
		0.2	0.32
		0.4	0.25
		0.6	0.16
		0.8	0.06
Gradual contraction		θ	
		30°	0.02
		45°	0.04
		60°	0.07
Equal area transformation		$A_1 = A_2$	C
		θ ≤ 14°	0.15
Flanged entrance		$A = \infty$	C
			0.34
Duct entrance		$A = \infty$	C
			0.85
Formed entrance		$A = \infty$	C
			0.85
Square edge orifice entrance		A_0/A_2	C_0
		0.0	2.50
		0.2	1.90
		0.4	1.39
		0.6	0.96
		0.8	0.61
		1.0	0.34
Square edge orifice in duct	$A_1 = A_2$	A_0/A	C_0
		0.0	2.50
		0.2	1.86
		0.4	1.21
		0.6	0.84
		0.8	0.20
		1.0	0.0

Appendix C: Pressure loss due to Elbows. (Additional equivalent losses in excess of friction to intersection of centerlines).

- a value based on f values of approximately 0.02.
- b values calculated from L/D and L/W values for f = 0.02. [2]

Type	Illustration	Conditions	Pressure loss C^a	Pressure loss L/D	Pressure loss L/W
N-Deg.		Rectangular or round; with or without vanes	$\frac{N}{90}$ times value for similar 90 – deg. elbow		
90-Deg. round section		Miter	1.30^b	65	
		R/D = 0.5	0.90		
		0.75	0.45	23	
		1.0	0.33	17	
		1.5	0.24	12	
		2.0	0.19	10	
90-deg. rectangular section		H/W 0.25, R/W Miter	1.25^b		25
		0.25, 0.5	1.25		25
		0.25, 0.75	0.60		12
		0.25, 1.0	0.37		7
		0.25, 1.5	0.19		4
		0.5, Miter	1.47		49
		0.5, 0.5	1.10		40
		0.5, 0.75	0.50		18
		0.5, 1.0	0.28		9
		0.5, 1.5	0.13		4
		1.0, Miter	1.50		75
		1.0, 0.5	1.00		50
		1.0, 0.75	0.41		21
		1.0, 1.0	0.22		11
		1.0, 1.5	0.09		4.5
		4.0, Miter	1.38		110
		4.0, 0.5	0.98		65
		4.0, 0.75	0.37		43
		4.0, 1.0	0.19		17
		4.0, 1.5	0.07		6
90-deg. square section with splitter vanes		R/W, R_1/W, R_2/W			
		Miter, 0.5			28
		0.5, 0.4	0.70		19
		0.7, 0.4			12
		1.0, 1.0	0.13		7.2
		1.5	0.12		
		Miter, 0.3, 0.5			22
		0.5, 0.2, 0.5	0.45		18
		0.75, 0.4, 0.7	0.12		
		1.0, 0.7, 1.0	0.10		
		1.5, 1.3, 1.6	0.15		
Miter with turning vanes	PLATE FORMED	C = 0.10 to 0.35 Depending on manufacture			
Miter tee with vanes		Consider equal to a similar elbow, Base loss on entering velocity.			
Radius tee					