COMPLEX ENGINEERING PROBLEM REPORT

"DESIGN OF CENTRAL HVAC SYSTEM FOR THE MECHANICAL HALL"

Submitted to

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Associate Professor/ Chairperson
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in partial fulfillment of the requirements for the

Course: HEAT VENTILATION AND AIR CONDITIONING

By

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1. Abstract:

This report aims to present a comprehensive design and analysis of a Central HVAC (Heating, Ventilation, and Air Conditioning) system for a Mechanical Engineering Auditorium Hall. The objective of this project was to create an energy-efficient and environmentally sustainable solution that provides optimal thermal comfort for occupants while minimizing energy consumption and carbon emissions.

The report begins calculation of the cooling load, considering factors such as occupancy, lighting, equipment, and outdoor weather conditions. This information forms the basis for the subsequent design and selection of a rooftop unit cooling system, tailored to meet the specific cooling demands of the auditorium. Sizing of the ductwork is a crucial element of the design process, ensuring efficient distribution of conditioned air throughout the space. This phase of the project takes into account the unique architectural features and layout of the auditorium, guaranteeing that the HVAC system operates at peak performance. Furthermore, an extensive analysis of annual electricity bills and CO2 emissions is conducted, comparing the proposed HVAC system's energy efficiency with conventional alternatives. This analysis provides valuable insights into the economic and environmental benefits of the design, demonstrating the long-term cost savings and reduced carbon footprint.

2. Problem statement:

The Mechanical Hall is extensively used for various events in the MUST. If the HVAC system is to be installed on the roof and the chilled air is to be supplied through ducts, design a HVAC system with the design indoor conditions of 23°C DBT and 50% RH. For outdoor conditions, make reasonable assumptions. Using your knowledge of HVAC, and working in a group of maximum four members

3. Introduction:

In present days the environmental problem is one of the most serious problems. Energy consumption by industries and buildings are responsible for this problem. About 72% of world energy is consumed by infrastructure, industry, commercial buildings, residential houses and markets. In a large building, which is air-conditioned, about 60% of the total energy requirements in the building is allocated for the air conditioning plant installed to use the cooling purpose. Exact prediction of the cooling and heating load, proper sizing of the heat ventilation air condition system and optimal control of the HVAC systems are important to minimize energy consumption. Factors that affect cooling loads are the external climate such as outdoor temperature, solar radiation and humidity. Local climatic conditions are important parameters for energy efficiency of buildings. Because the energy consumption in buildings depends on the climatic conditions and the performance of the HVAC system changes with them as well, better design in building HVAC application that take account of the right climatic condition will result in better comfort and more energy efficient buildings

4. Terminology:

Commonly used terms relative to cooling load calculation and heat transfer of the buildings according to the ASHRAE reference are given below:

1. **Refrigeration:** The term "refrigeration" means process of removing heat from a substance under the controlled conditions. It also includes process of reducing and maintaining the temperature of a body below the surrounding temperature.
2. **Air-Conditioning:** Controlling and maintaining environmental parameters such as temperature, humidity, cleanliness, air movement, sound level pressure difference between condition space and surrounding within prescribed limit.
3. **Cooling Load Temperature Difference (CLTD):** Cooling load temperature difference is an equivalent temperature difference used for calculating the external cooling load across the walls and roofs.
4. **Humidity:** It is the mass of water vapor present in 1 kg of dry air and is expressed in terms of gram per kg of dry air. It is also called specific humidity or humidity ratio.

5. **Relative humidity:** It is a ratio of actual mass of water vapor in a given volume of moist air to the mass of water vapor in the same volume of saturated air at the same temperature and pressure.
6. **Dry bulb temperature:** It is the temperature of air recorded by thermometer when it is not affected by the moisture present in the air. The dry bulb temperature is denoted by T_{DB}.
7. **Wet bulb temperature:** It is the temperature of the air recorded by the thermometer when its bulb is surrounded by a wet cloth exposed to the air. The wet bub temperature is denoted by T_{wb}.
8. **Dew Point temperature:** It is the temperature of the air recorded by the thermometer, when the moisture present it begins to condense.
9. **Heat transfer co efficient:** It is the rate of heat transfer through a unit area of building envelop material including its boundary films per unit temperature difference between the outside and inside air.
10. **Sensible heat gain:** It is the direct addition of heat to the enclosed space without any change in its specific humidity is known as sensible heat gain.
11. **Latent heat gain:** It is the heat gain of space through addition of moisture, without change in its dry bulb temperature is known as latent heat gain.
12. **Space heat gain:** It is the rate of heat gain at which heat enter into and generated within the conditioned space.
13. **Space cooling load:** It is the rate at which energy must be removed from a space to maintain a desired air temperature of a space.

Space Heat Gain V/s cooling load:

The heat received from the heat sources (conduction, convection, solar radiation, lightning, people, equipment) does not go directly to heating the room. Only some portion of the heat is absorbed by the air in the conditioned space leading to a minute change in its temperature. Most of the radiation heat especially from the Sun, lighting people is first absorbed by the internal surfaces which include ceiling, floor, internal walls, furniture etc. Due to large thermal capacity of the roof, floor, walls etc., their temperature increases slowly due to absorption of radiant heat.

Space Cooling V/s cooling load:

Space cooling is the rate at which heat is removed from the spaces to maintain air temperature at a constant value. Cooling load is the rate at which energy is removed at the cooling coil that serves one or more conditioned space in any central conditioning systems. It is the of the space cooling loads for all spaces served by the system plus any additional load imposed on the system external to the conditioned spaces items such as fan energy, fan location, duct heat gain, duct leakage, heat extraction lighting systems and type of return air systems all affect component sizing.
Cooling loads Classified by kinds of heat:
There are two distinct components of the air conditioning load.
(1) The Sensible load gain (2) The Latent load gain

Sensible loads:

Sensible heat gain is the addition of heat to a space which shall result in space temperatures. The factors including sensible cooling load:

1) Solar heat gain through building envelop (exterior walls, glazing, skylights, roof , floors)
2) Partitions that separate spaces of different temperatures
3) Ventilation air and air infiltration through cracks in the building, doors and windows
4) People in the building
5) Equipment and appliances operated in the summer
6) Lights

Latent loads:

A latent heat gain is the heat gained in the water vapor. Latent heat does not cause a temperature rise but it constitutes a load on a cooling equipment. Latent load is the heat that must be removed to condense the moisture out of the air. The sources of the latent heat are:

1) People Breathing
2) Cooking equipment
3) Housekeeping, floor washing
4) Appliances or machinery that evaporates water
5) Ventilation air and air infiltration through cracks in the buildings, doors and window

5. COOLING LOAD CALCULATION OF MECHANICAL HALL

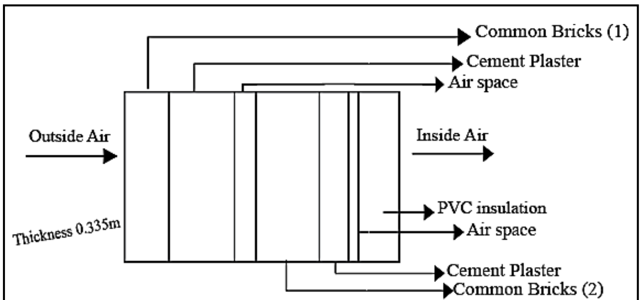
5.1. gain through building walls:

Heat gain through building walls can be calculated using following equations

$$Q = U\ X\ A\ X\ \Delta T$$

Q= heat load in watt
U= Overall heat transfer coefficient
A= Surface Area of wall
ΔT= Temperature difference internal and external

Calculating U-Factor of wall:



Structure of a wall

The wall is composite and consist of combination of different layers. It consists of 5mm cement plasters or mortar, 150 mm common bricks, Air space, 150mm common brick, Cement plaster 15mm, air space, 3mm Polyvinylchloride PVC insulation.

Using ASHRAE fundamental handbook 1997, CH-28

(1) Thermal conductivity of bricks (Kb)= 0.77W/mK
(2) Thermal conductivity of cement plaster (K plaster) = 8.65 W/mK
(3) Thermal conductivity of air gap= 5.8 W/m^2K
(4) Outside film coefficient ho= 20W/m2K
(5) Inside film coefficient hi+ 8.5W/m2K
(6) Thermal conductivity of PVC paneling= 0.087W/m2K

Now using general equation

$$Uwall = \frac{1}{\frac{1}{ho} + \frac{x1}{K1} + \frac{x2}{K2} + \frac{1}{k3} + \frac{x3}{K4} + \frac{x4}{K5} + \frac{1}{K6} + \frac{x5}{K7} + \frac{1}{hi}}$$

$$Uwall = \frac{1}{\frac{1}{20} + \frac{0.015}{8.65} + \frac{0.15}{0.77} + \frac{1}{5.8} + \frac{0.15}{0.77} + \frac{0.015}{8.65} + \frac{1}{5.8} + \frac{0.003}{0.087} + \frac{1}{8.5}}$$

$$Uwall = \frac{1}{0.05 + 0.00173 + 0.1948 + 0.1724 = 0.1948 + 0.00173 + 0.1724 + 0.03448 + 01176}$$

$$Uwall = \frac{1}{0.93994}$$

U wall = 1.063 W/m2K (East wall)

U wall= 1.065 W/m2K (West wall)

U wall for South wall

$$Uwall = \frac{1}{0.1176 + 0.00173 + 0.1948 + 0.1724 + 0.1948 + 0.00173 + 0.1724 + 0.03448 + 0.1176}$$

$$Uwall = \frac{1}{1.00754}$$

$$Uwall = \ 0.9925\ W/m2K$$

Similarly, U wall for North wall

$$(Uwall) = \ \frac{1}{0.9668}$$

$$Uwall = \ 1.034\ W/m^2K$$

Surface Areas of walls:

Surface area of East wall

Area= 18.288X 6.7056

Area= 122.63 m2

Area= 109.94 m2

Surface area of West wall

Area= 18.288 X 6.7056

Area= 122.63 – 12.69 (window area)

Surface area of South wall

Area= 12.191 X 6.7056

Area= 81.75 m^2

Surface area of north wall

Area= 12.191 X 6.7056

Area= 81.75 m^2

Table 1: inside and outside temperature conditions of all four walls

Temperature	East Wall	South wall	North Wall	West Wall
Indoor (Ti)	23 C	23 C	23 C	23 C
Outdoor (to)	32 C	26 C	32 C	38 C
Difference ΔT	9 C	3 C	9 C	15 C

Total Heat gain through building walls;

Wall-1 (East) $Q = 1.063\ X\ 122.63\ X\ 9$

$Q = 1173.30$ W

$Q = 1.17\ KW$

Wall-2 (South) $Q = 0.925\ X\ 81.75\ X\ 3$

$Q = 243.41\ W$

$Q = 0.243\ KW$

Wall-3 (West) $Q = 1.065\ X\ 109.94\ X\ 15$

$Q = 1756.29$ W

$Q = 1.76\ KW$

Wall-4 (North) $Q = 1.034\ X\ 81.75\ X\ 9$

$Q = 760.76\ W$

$Q = 0.76\ KW$

Total load of walls = Sum of load of all walls

Q = 0.76+1.76+1.17+0.243

Q = 3.933 KW

5.2. Heat gain of roof:

The basic conduction equation for heat gain is

$$Q = UA(To - Ti)$$

Area= 18.288 X 12.191

Area= 222.94 m^2

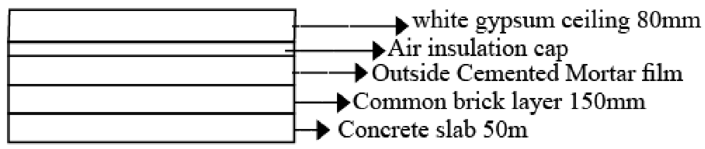
Overall heat transfer Coefficient:

$$U\,roof = \frac{1}{\frac{1}{ho} + \frac{x1}{K1} + \frac{x2}{K2} + \frac{x3}{K3} + \frac{1}{K4} + \frac{x4}{K4} + \frac{1}{hi}}$$

$$U\,roof = \frac{1}{\frac{1}{15} + \frac{0.015}{8.65} + \frac{0.15}{0.727} + \frac{0.05}{1.739} + \frac{1}{5.8} + \frac{0.003}{0.11} + \frac{1}{8.5}}$$

$$Uroof = \frac{1}{0.066 + 0.00173 + 0.2063 + 0.028 + 0.1724 + 0.02727 + 0.1176}$$

U_{roof} = 1.6147 W/m^{2}K



Roof Layers

Total heat gain:

Q= 1.6147 X 222.94 X (38-23)

Q= 1.6147 X 222.94 X 15

Q= 5399.7 watt

Q= 5.3 KW

5.3. Heat load through Windows:

There are total 9 glass windows

Solar load through glass has two components

(i) Conductive

(ii) Direct Solar transmission

Cooling load equations for each window,

Conductive:

$Q_{\text{(glass conduction)}}$ = U X A X T

Direct Transmission:

Q = A X SC X SCL

Area of Windows:

Area= length X Width

Area = 1.4935 X 0.944 = 1.41 m^2

U- Value of glass:

Using ASHRAE fundamental 2001,

Air gaps/ spaces	R= value in m^{2}K/W
Inside air film	0.1198
Insulation drapes	0.2144
Air gap (3in)	0.1938
Single glazed window	0.2144

Outside air film	0.04405

R_{total} = R_{film} + R_{drapes} + R_{air} + R_{glass} + $R_{out\ films}$
R_{total} = 0.1198 + 0.2144 +0.1938 +0.2144 + 0.04405
R_{total} = 0.7864 m^2K/W
U_{value} = 1/ R_{total}
U_{value} = 1/ 0.7864
U_{value} = 1.2716 W/m^2K

(1) Conductive heat load through windows

Q= U X A X T
Q= 1.2716 X 1.41 X (38-23)
Q= 26.89 W
For all windows
Q= 26.89 X 9
Q= 242 watt
Q= 0.242 KW

(2) Direct transmission

Q= A X SC X SCL
Almost all the windows are on West wall of the hall
So SCL for window glass at peak time has been calculated
Using ASHRAE hand book
Solar Cooling Load Factor (SCL) = 350 W/m^2
Shading Coefficient (SC) = 0.35
Q= 1.41 X 0.35 X 350
Q= 172.725 watt
For all windows
Q= 172.725 X 9
Q = 1554.525 W
Q= 1.55 KW

Total Solar heat gain through windows:

Q_t = Qc + Q_R
Q_t = 0.242 + 1.55
Q_t = 1.792 KW

Heat gains due to infiltration (door opening, cracks, leakages)

Amount of in filtered air, (V_{int}) = $\frac{Volume\ of space\ X\ Ac}{60}\ m^3/min$
Volume of Hall = 18.288 X 12.191 X 6.7056
Volume of Hall= 1495 m^3
Ac= Number of air changes per hour
For a hall with four walls let take Ac=1.5
(V_{int}) = $1495\ X\ \frac{1.5}{60}\ m^3/min$
(V_{int}) = 37.37 m^3/ min

5.4. Sensible heat gain due to infiltration:

Q_s = 20.44 X 37.37 X (38-23)
Q_s = 20.44 X 37.37 X 15
Q_s = 11459.17 watt
Q_s = 11.45 KW

Latent heat gain due to infiltration:

Q_l = 50000 X V_{int} X (W_o-W_i)

Q_l = 50000 X 37.37X (0.023-0.0085)

Q_l = 1868500 X 0.0145

Q_l = 27093.25

Q_l = 27 KW

Rate of infiltration per door opening:

Estimated rate of infiltration per door opening = $41.148\ X\ \frac{3.108}{2}$

= 63.94 m^3/min

5.5. Heat Gain from occupants:

Sensible heat gain from occupants

Q_s = Qs person X N X CLF (using ASHRAE standards, Qs=65 watt for each person)

Q_s = 65 X 180 X CLF

CLF =1 for auditorium or Hall

Q_s = 65 X 180 X 1

Q_s = 11700 watt

Q_s = 11.7 KW

Latent heat from occupants

Q_l = Ql X N (using ASHRAE standards, Ql = 30 at rest)

Q_l = 30 X 180

Q_l = 5400 W

Q_l = 5.4 KW

Total occupants load

$Q_t = Q_s + Q_l$

Q_t = 11.7 + 5.4

$Q_{t\ =\ 17.1}$ KW

5.6. Heat gain from lights

Total lights = 153 LED + 8 tube lights

Q_{lights} = total wattage X use factor X allowance factor

Use factor: ratio of actual wattage in use to installed wattage

Allowance factor: 1.2 for fluorescent bulbs

Total wattage = 153 X 25 (LED)

Total wattage = 3825 W

Tube lights = 8 X 86

Tube lights = 288 W

Q_{light} = 4113 X 0.92 X 11.8

Q_{light} = 4540.75 = 4.54 KW

5.7. Heat gain from electrical equipment's:

Fans: 168 W (ceiling fans 3) + 525 W (wall fans 7)

Total wattage = 693 watt

Projector: Q= 150 W X 2

300 Watt

Laptop: Q= 50 W X1

50 Watt

Total wattage = 693+ 300+ 50 = 1043 watt

Now, $Qs_{equipment}$ = total wattage of equipment X use factor X CLF

= 1043 X 1 X 0.71

CLF= Cooling load factor value is 1

$Q_{equipment}$ = 740.53 W

$E_{quipment}$ = 0.74053 KW

5.8. Heat gain from Ventilation:

T1=38 C Q1=65
T2= 23 C Q2=50
From Psychometric chart,
W1= 0.029 W2= 0.0088
Δ W= W1-W2 = 0.029-0.0088 =0.015 Kg/Kg
Latent heat gain= 24.91 X 0.0156 X 2500
= 960 W = 0.960 KW
Sensible Heat gain = 1.08 X Qair X ΔT
= 1.08 X 24.91 X 15
=403.54 W = 0.4 KW
Total load due to ventilation:
$\text{Load}_{\text{total}}$ = 0.4 KW+ 0.960 KW
$\text{Load}_{\text{total}}$ = 1.36 KW

5.9. Total Sensible Load:

Transmission through walls = 3.87 KW
Total Solar heat gain through roof = 1.16 KW
Sensible heat gain through windows = 1.792 KW
Sensible heat gain due to infiltration and ventilation = 7.63 KW
Sensible heat gain from occupants = 11.7 KW
Heat gain from lights= 4.54 KW
Heat gain from equipment = 0.74053 KW
Sensible Heat gain from ventilation = 0.4 KW
Total Sensible Load= 3.87+1.16+1.792+7.63+11.7+4.54+0.74053+0.4
= 31.83 KW

5.10. Total Latent load:

Latent load due to infiltration and ventilation= 18 KW
Latent load from occupants = 5.4 KW
Latent load from ventilation = 0.960 KW
Total latent load = 18 + 5.4+0.960 KW = 24.36 KW
Total Cooling Load: 56.19 KW

5.11. <u>Capacity of AC:</u>

TR= $\frac{\boldsymbol{Qtotal}}{\mathbf{3.5177}}$**=** $\frac{\mathbf{56.19}}{\mathbf{3.517}}$

TR= 15.97 Tons of Refrigeration

Table 2: sensible, latent and total load calculation for mechanical hall

particulars	Sensible (kW)	Latent (kW)	Total (kW)
Walls			
North South East West	0.76 o.243 1.17 1.76		3.933

Roof	5.3		5.3
Windows			
Conductive Direct transmission	0.242 1.55		1.792
Infiltration	11.45	18	38.45
Occupants	11.7	5.4	17.1
Lights	4.54		4.54
Fans	0.693		0.693
Projector	0.3		0.3
Laptops	0.05		0.05
Ventilation	0.4	0.96	1.36
Total	**31.83**	**24.36**	**56.19**

Installation Capacity of AC:

TR= $\frac{Qtotal}{3.5177}$ **=** $\frac{56.19}{3.517}$

TR= 15.97 Tons of Refrigeration

6. Cooling load calculation using HAP (hourly analysis program)

The required data for cooling load estimation has been provided to HVAC software HAP under required conditions. The output data for the cooling load has been compared with by hand calculations and also attached below:

Air System Sizing Summary for mechanical hall

Project Name: mechanical hall cep
Prepared by: get

09/26/2023
11:49AM

Air System Information

Air System Name .... **mechanical hall**
Equipment Class .... **PKG ROOF**
Air System Type .... **SZCAV**

Number of zones **1**
Floor Area .... **2400.0** ft²
Location .... **mirpur, Pakistan**

Sizing Calculation Information

Calculation Months .... **Jan to Dec**
Sizing Data .... **Calculated**

Zone CFM Sizing .... **Sum of space airflow rates**
Space CFM Sizing .... **Individual peak space loads**

Central Cooling Coil Sizing Data

Total coil load .... **15.2** Tons
Total coil load .... **182.9** MBH
Sensible coil load .... **147.0** MBH
Coil CFM at Aug 1700 .... **6655** CFM
Max block CFM .... **6655** CFM
Sum of peak zone CFM .... **6655** CFM
Sensible heat ratio **0.804**
ft²/Ton **157.5**
BTU/(hr-ft²) **76.2**
Water flow @ 10.0 °F rise .... **N/A**

Load occurs at .... **Aug 1700**
OA DB / WB .... **99.3 / 73.8** °F
Entering DB / WB .... **81.7 / 67.7** °F
Leaving DB / WB .... **60.2 / 58.9** °F
Coil ADP .... **57.8** °F
Bypass Factor **0.100**
Resulting RH .... **57** %
Design supply temp. .... **58.0** °F
Zone T-stat Check .... **0 of 1** OK
Max zone temperature deviation .... **0.1** °F

Supply Fan Sizing Data

Actual max CFM .... **6655** CFM
Standard CFM .... **6329** CFM
Actual max CFM/ft² .... **2.77** CFM/ft²

Fan motor BHP .... **0.00** BHP
Fan motor kW .... **0.00** kW
Fan static .... **0.00** in wg

Outdoor Ventilation Air Data

Design airflow CFM .... **1494** CFM
CFM/ft² .... **0.62** CFM/ft²

CFM/person .... **8.30** CFM/person

System Psychrometrics for mechanical hall

Project Name: mechanical hall cep
Prepared by: get

09/26/2023
11:49AM

August DESIGN COOLING DAY, 1700

TABLE 1: SYSTEM DATA

Component	Location	Dry-Bulb Temp (°F)	Specific Humidity (lb/lb)	Airflow (CFM)	CO2 Level (ppm)	Sensible Heat (BTU/hr)	Latent Heat (BTU/hr)
Ventilation Air	Inlet	99.3	0.01299	1494	400	34869	8121
Vent - Return Mixing	Outlet	81.7	0.01206	6655	931	-	-
Central Cooling Coil	Outlet	60.2	0.01086	6655	931	146991	35891
Supply Fan	Outlet	60.2	0.01086	6655	931	0	-
Cold Supply Duct	Outlet	60.2	0.01086	6655	931	-	-
Zone Air	-	76.6	0.01179	6655	1085	112122	27760
Return Plenum	Outlet	76.6	0.01179	6655	1085	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.027 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4513.8 BTU/(hr-CFM)
Site Altitude = 1385.0 ft

TABLE 2: ZONE DATA

Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	CO2 Level (ppm)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	116168	Cooling	112122	76.6	6655	1085	0	0

Air System Design Load Summary for mechanical hall

Project Name: mechanical hall cep 09/26/2023
Prepared by: get 11:49AM

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Aug 1700 COOLING OA DB / WB 99.3 °F / 73.8 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 76.0 °F / 59.0 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	91 ft²	5950	-	91 ft²	-	-
Wall Transmission	1119 ft²	12769	-	1119 ft²	0	-
Roof Transmission	2400 ft²	10040	-	2400 ft²	0	-
Window Transmission	91 ft²	1090	-	91 ft²	0	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	68 ft²	426	-	68 ft²	0	-
Floor Transmission	0 ft²	0	-	0 ft²	0	-
Partitions	0 ft²	0	-	0 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	4549 W	15521	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	350 W	1194	-	0	0	-
People	180	41400	21600	0	0	0
Infiltration	-	22247	4838	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	5% / 5%	5532	1322	5%	0	0
>> Total Zone Loads	-	**116168**	**27760**	-	**0**	**0**
Zone Conditioning	-	112122	27760	-	0	0
Plenum Wall Load	0%	0	-	0	0	-
Plenum Roof Load	0%	0	-	0	0	-
Plenum Lighting Load	0%	0	-	0	0	-
Return Fan Load	6655 CFM	0	-	6655 CFM	0	-
Ventilation Load	1494 CFM	34869	8121	1494 CFM	-9206	0
Supply Fan Load	6655 CFM	0	-	6655 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	**146991**	**35881**	-	**-9206**	**0**
Central Cooling Coil	-	146991	35891	-	-9206	0
>> Total Conditioning	-	**146991**	**35891**	-	**-9206**	**0**
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

System Psychrometrics for mechanical hall

Project Name: mechanical hall cep 09/26/2023
Prepared by: get 11:49AM

WINTER DESIGN HEATING

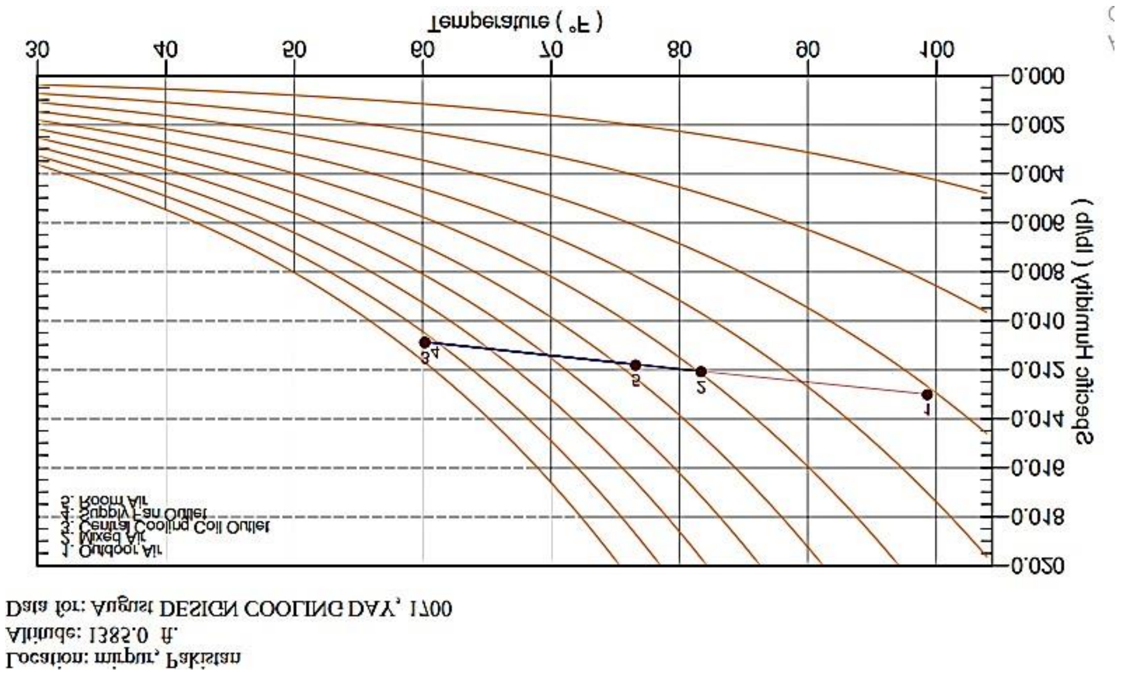
TABLE 1: SYSTEM DATA

Component	Location	Dry-Bulb Temp (°F)	Specific Humidity (lb/lb)	Airflow (CFM)	CO2 Level (ppm)	Sensible Heat (BTU/hr)	Latent Heat (BTU/hr)
Ventilation Air	Inlet	76.0	0.00730	1494	400	9206	0
Vent - Return Mixing	Outlet	71.3	0.00730	6655	431	-	-
Central Cooling Coil	Outlet	70.0	0.00730	6655	431	9206	0
Supply Fan	Outlet	70.0	0.00730	6655	431	0	-
Cold Supply Duct	Outlet	70.0	0.00730	6655	431	-	-
Zone Air	-	70.0	0.00730	6655	441	0	0
Return Plenum	Outlet	70.0	0.00730	6655	441	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.027 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4513.8 BTU/(hr-CFM)
Site Altitude = 1385.0 ft

TABLE 2: ZONE DATA

Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	CO2 Level (ppm)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	0	Deadband	0	70.0	6655	441	0	0



7. Description of conditioned space and HVAC system:

The space under consideration is a 180-seat mechanical engineering department hall, located at the ground floor of the department building and is used mostly for events, seminars, workshops and other academic activities and during the working days of the week. The hall is conditioned by a single-duct, single-zone, constant volume, all-air HVAC system, which is the most appropriate for a single room of this size, with uniform load behavior and considerable ventilation air needs. The HVAC system consists of an air handling roof top unit (RTU) system and the air duct distribution system. It follows the vapor compression refrigeration cycle in its working.

The system provides cooling, and dehumidification during the whole year. The fresh ventilation air is drawn in the RTU through an intake diffuser and is mixed, if needed, in a mixing chamber with air returning from the conditioned space. The mixed air is filtered and then conditioned to the desired inside conditions. If the system is in the cooling mode, the air is cooled and dehumidified before entering the space, while in the heating mode it is heated and humidified, at a suitable temperature and moisture content.

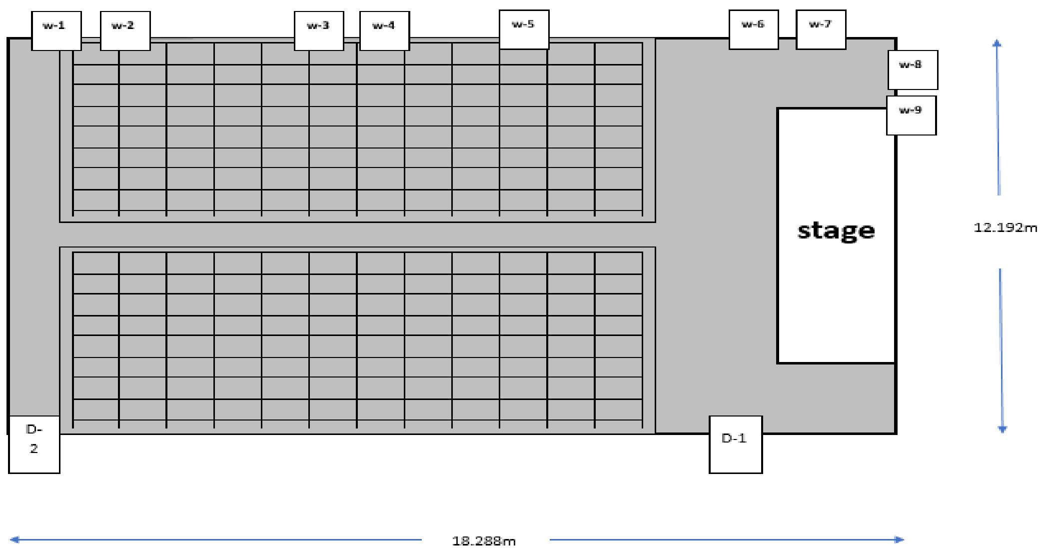


Figure 1 layout of mechanical hall

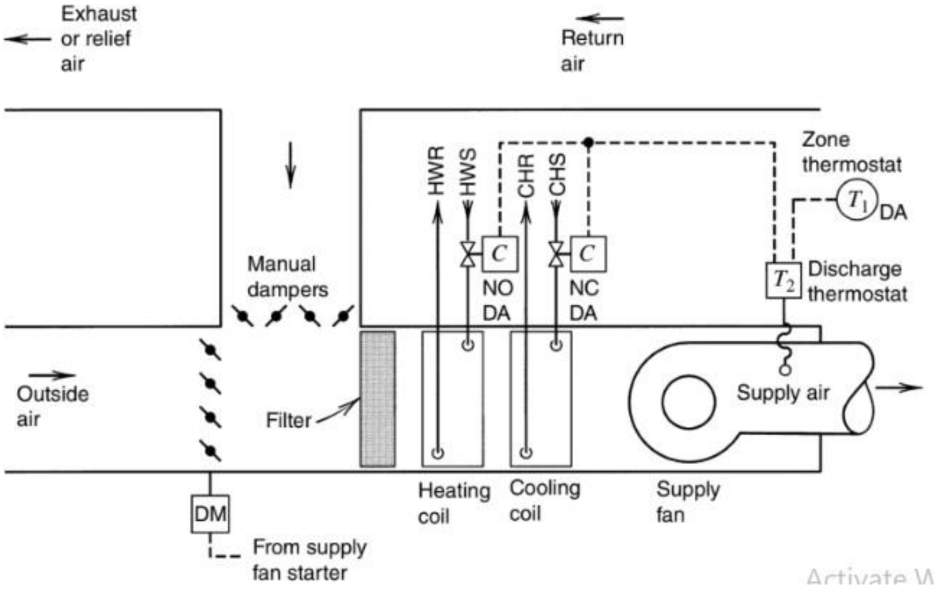


Figure 2: schematic diagram of roof top unit

Rooftop units are a type of air handler, the main difference is that they are usually more compact and they're always installed on the roof so they need to be more robust and weather proof to deal with the sun, rain, snow, wind etc. Additionally, AHU's will often be connected to central plant such as chillers and boilers to provide the heating and cooling, but RTU's are self-contained and have everything they need all in one unit. That's why they're called package air conditioners. This unit first pulls air in through the hood, the amount of air entering is controlled by the damper. The air then passes through a filter to catch the dust and dirt and protect the surface of the heat wheel, it then passes through the heat wheel. The heat wheel is a rotating heat exchanger which pickups the waste heat or coolth from the return discharge air and transfers this over to the incoming fresh air without the two air steams mixing. These heat wheels are not completely air tight so a little bit of air mixing will occur.

The heat wheel is used to offset the heating and sometimes cooling demand when conditions are right, saving energy and utility costs. After the heat wheel the air flows through another filter. Just before the filter we have a damper on the return air stream. This allows us to recirculate some of the return air into the fresh air and the quantity is varied using the dampers. Not all heat wheel rtu's will have this feature some only use 100% fresh air intake and extract. If it doesn't have the option to recirculate then the unit probably won't have this second filter bank.

After this the air will flow through the heat exchangers which heat or cool the air to the desired temperature.

The fan will then distribute the air through the building via the ductwork to the designated locations.

The return air is then pulled back into the RTU through the return ductwork. Once it re-enters the RTU it has the option to either recirculate some air back into the fresh air intake, otherwise it will all pass through a filter and then through the heat wheel to capture the waste heat. After the heat wheel we might find an extract fan otherwise the pressure caused by the main supply fan can be used to force the air out, in some designs.

The air then passes through the extract damper which is used to vary the volume of return air mixing as well as the pressure inside the building, after that it passes through a grille which just stops objects and wildlife from entering the unit where it will then be ejected from the system into the atmosphere.

8. HVAC system design criteria

In order to install and design a HVAC system first of all we have to calculate the heating and cooling loads of the desired space. We have already calculated the cooling load of mechanical hall according to the methods described in cooling load calculation section. Outdoor-indoor design parameters, namely dry-bulb (DB) temperature, wet-bulb (WB) temperature, relative humidity (RH) plus ventilation airflow requirements were adjusted according to the given conditions and are also given below in table. The weather conditions of the city were also held into account.

Parameters	Values
Number of people in at full occupancy	180
Room DBT	23°
Relative humidity (RH)	50%
Outside conditions (T °C)	38°C
Relative humidity (RH)	65%

8.1. Flow rate calculation:

The flow rate of air is given by,

$m = Q / (cp \times \Delta t)$

$\rho v = Q / (cp \times \Delta t)$

Q = the cooling load of the room = 56.19(kW)

Cp=specific heat capacity of the air=1.026 kJ/kg.k

Δt = temperature difference between supply air and room air= 10

P= Density of air = 1.2 kg/m3

V= Volume of air movement per second (m3/s)

The volume of the air per second that passes through the duct

work is,

$V= 56.19KW / (1.2\times10\times1.026)$

$V=4.55\ m^3/s$

8.2. Air Change Rate:

How many times air needs to be change per second is given by:

ACR = V/ V_r

here, ACR is Air Change Rate (1/s),

V= Volume of air required per second (m3 /s) =4.55 m^3/s

V_r= volume of room (m^3) = 1495 m^3

ACR = (4.55 m^3/s) / (1495 m^3)

ACR = 0.003/s

9. Design of Ductwork:

There are many theories which define the size of the duct system including equal friction static regains total pressure velocity reduction, constant velocity from these methods, we elect to use the equal friction method because it gives us more correct values to keep the discharge constant. In the equal friction method, the system is sized for a constant pressure loss per unit length of duct. The equal friction method can be used for the design of supply and extract the resulting duct systems.

9.1. Duct sizing

To size the ducts, we used the duct sizing chart or duct pressure loss chart. We can obtain these from ductwork manufacturers or ASHRAE. These charts hold a lot of information. We can use them to find the pressure drop per meter, the air velocity, the volume flow rate and also the size of the ductwork. The layout of the chart does vary a little depending on the manufacturer but in this example the vertical lines are for volume flow rate. The horizontal lines are for pressure drop per unit length.

We start sizing from the first main duct which is section A. To limit the noise in this section we'll specify that it can only have a maximum velocity of 6m/s. We know that this duct also requires a volume flow rate of 4.55m3/s so we can use the velocity and volume flow rate to find the missing data.

We take the chart and move on x-axis from the bottom until we hit the volume flow rate of 4.55m3/s. Then we locate where the velocity line is of 6m/s and we draw a horizontal line across x axis to find pressure drop per unit length i.e., 3Pa So add this figure into the chart. As we're using the equal pressure drop method, we can use this pressure drop for all the duct lengths so fill those in too. Then we scroll up again and align our intersection with the upward diagonal lines to see this requires a duct with a diameter of 0.6m so we add that into the table also.

We know the volume flow rate and pressure drop so we can now calculate the values for branch ducts similarly and then the remaining ducts.

For the branch ducts, we use the same method.

Table 3: duct sizing parameters using data from above charts

Type	Volume flow rate (m^3/s)	Pressure losses (pa/m)	Velocity (m/s)	Diameter (m)	Length (m)	Duct losses (Pa) Length × pressure losses	Volume flow rate (m^3/s) From each diffuser
Main supply duct	4.54(m^3/s)	3 Pa	6 m/s	0.6m	7 m	21 Pa	
Branch duct 1	1.51 (m^3/s)	3 Pa	4.5 m/s	0.4 m	10 m	30 Pa	0.5
Branch duct 2	1.51 (m^3/s)	3 Pa	4.5 m/s	0.4 m	10 m	30 Pa	0.5

Branch duct 3	1.51 (m^3/s)	3 Pa	4.5 m/s	0.4 m	10 m	30 Pa	0.5

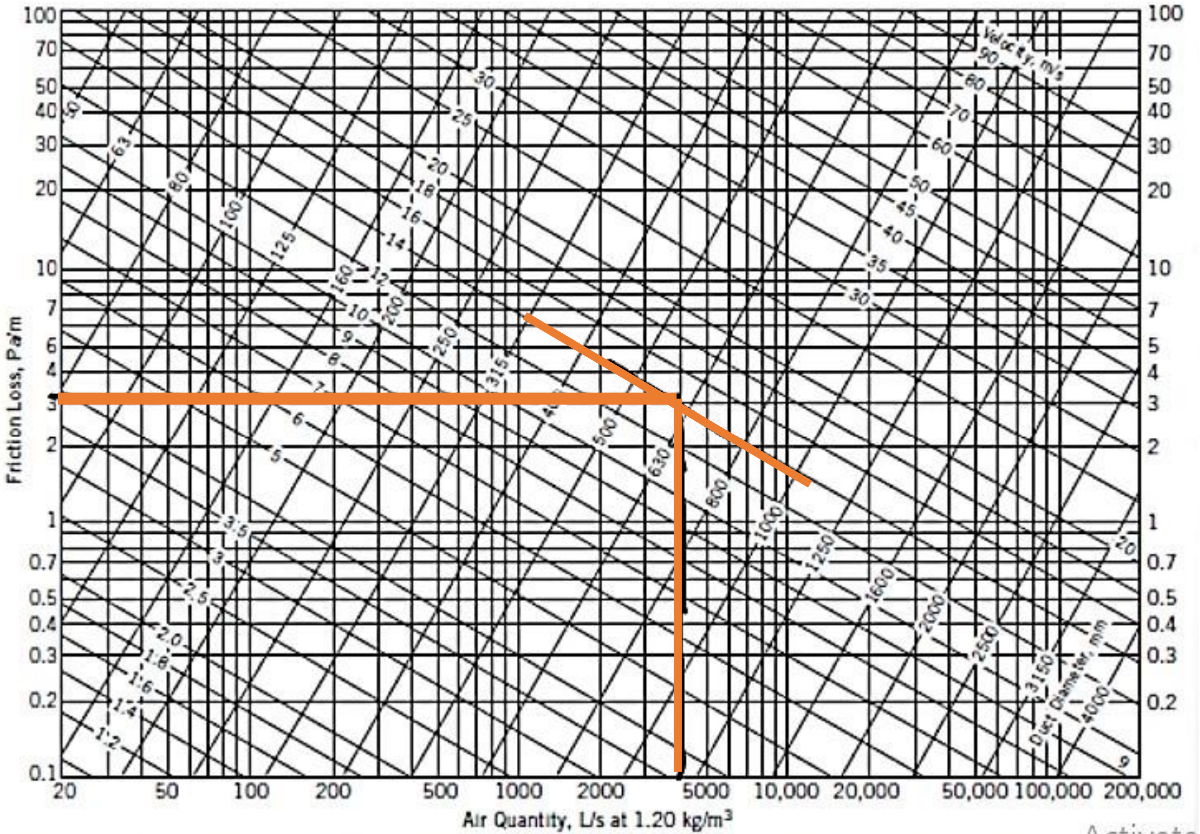


Figure 12-22 Pressure loss due to friction for galvanized steel ducts, SI units. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume SI*, 1997.)

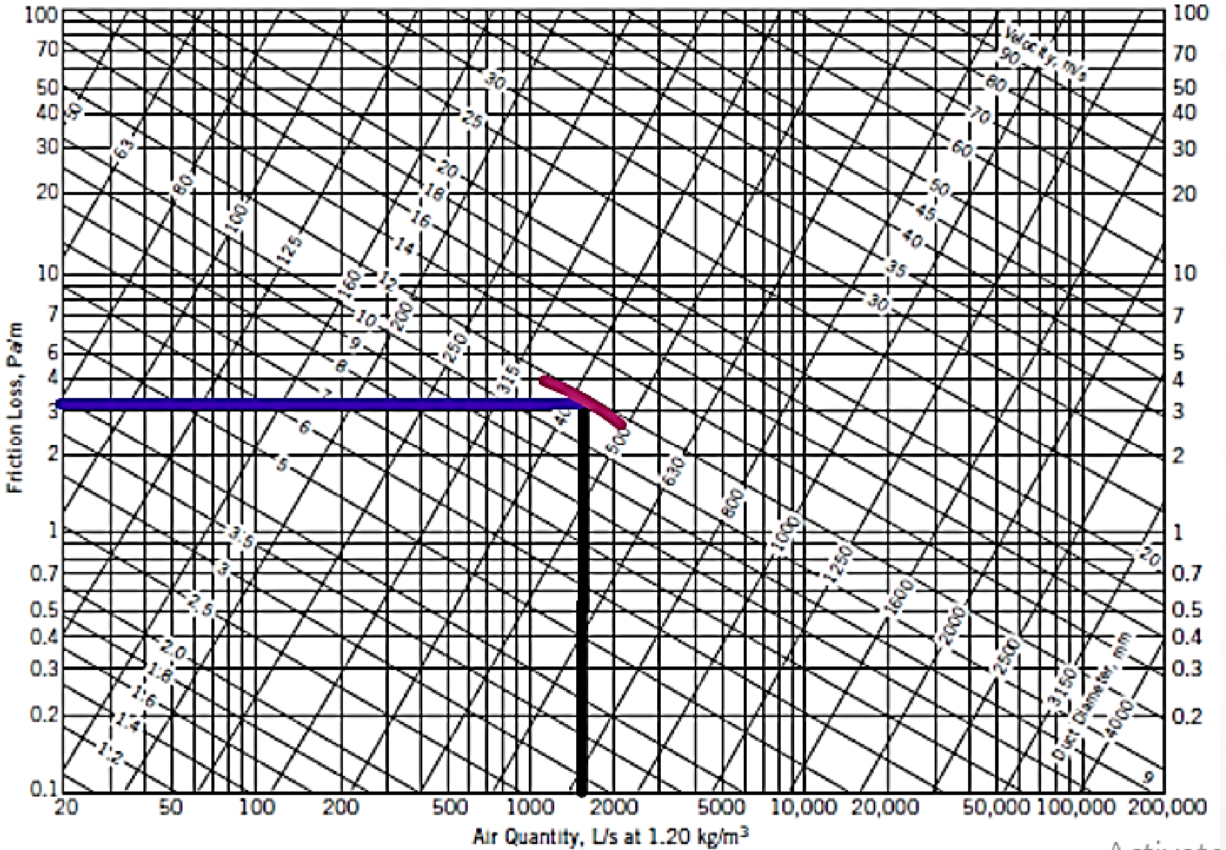


Figure 12-22 Pressure loss due to friction for galvanized steel ducts, SI units. (Reprinted by permission from *ASHRAE Handbook, Fundamentals Volume SI*, 1997.)

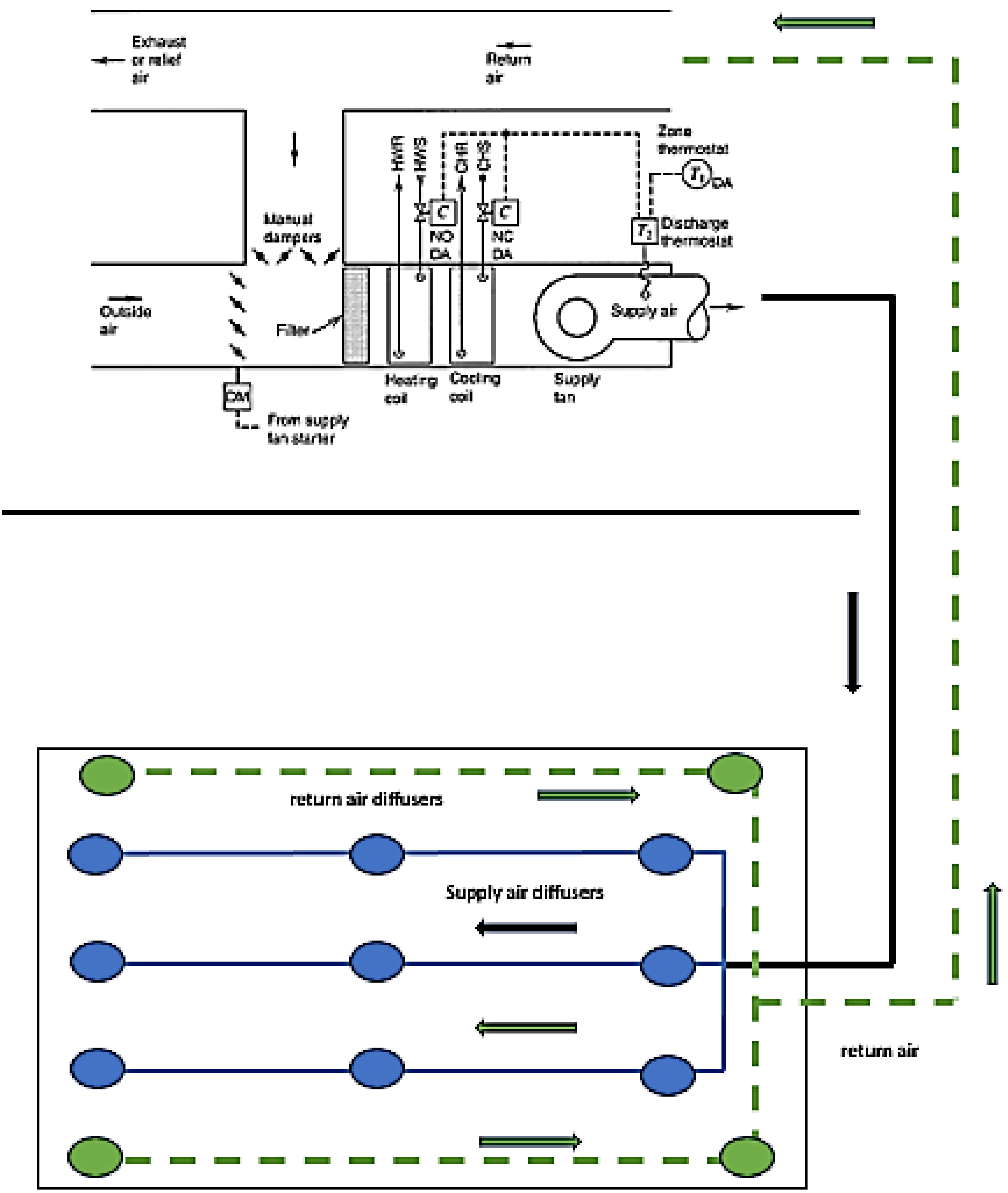


Figure 3: layout of HVAC cooling system

Annual Electricity Consumption:

Daily latent load power consumption = latent load in KW × operating hours

= 24.36 × 6=146.16KWh

Daily sensible load power consumption = sensible load in KW × operating hours

=31.83 × 6=190.98KWh

Total daily power consumption= 146.16+190.98 = 337.14KWh

11.1. Monthly Power consumption

Hall operating for 12 days in a month

Monthly power consumption = total daily power consumption × 12

= 337.14 × 12

= 4045.68KWh

11.2. Annual Power Consumption

= monthly power consumption × 12

= 4045.68 × 12

= 48548.16KWh

In Pakistan 1 unit = 60

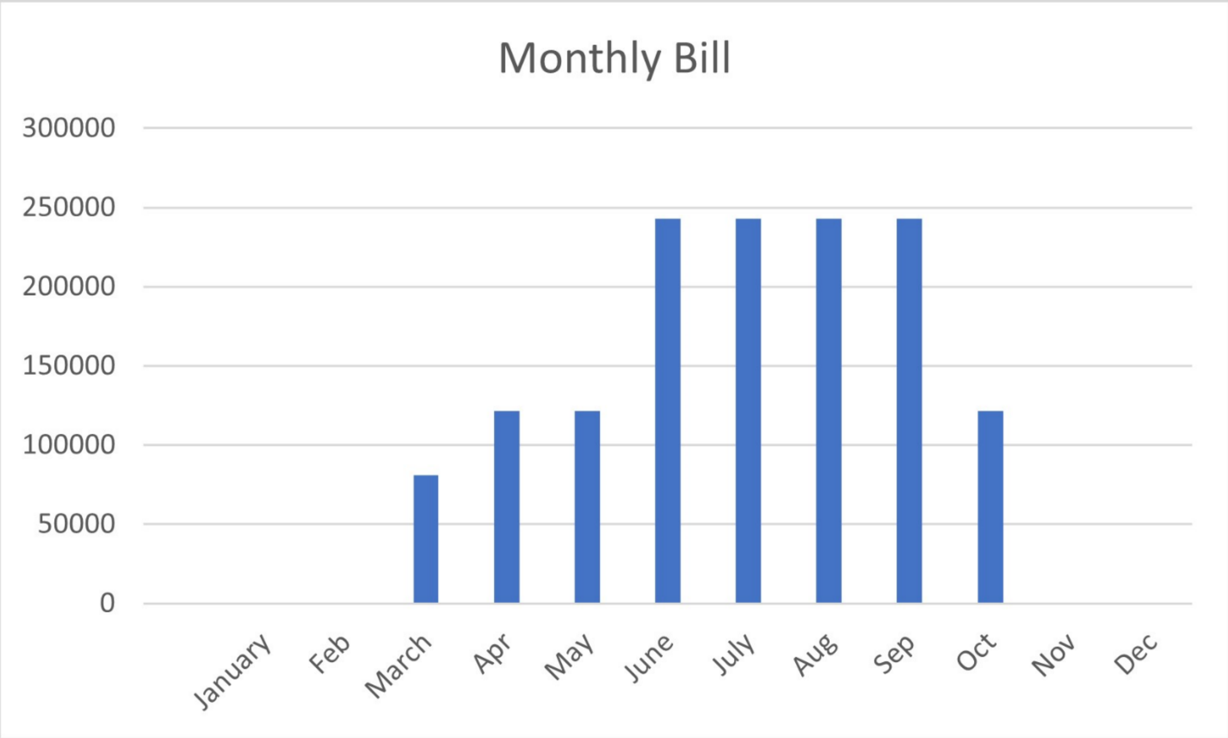
Table 4: power consumption details with months

Months	Operating hours	Daily latent load P.C(KWh)	Daily sensible load P.C(KWh)	Daily total (KWh)	Monthly (KWh)
January	0	0	0	0	0
Feb	0	0	0	0	0
March	2	48.72	63.66	112.38	1348.56
Apr	3	73.08	95.49	168.57	2022.84
May	3	73.08	95.49	168.57	2022.84
June	6	146.16	190.98	337.14	4045.68
July	6	146.16	190.98	337.14	4045.68
Aug	6	146.6	190.98	337.14	4045.68
Sep	6	146.16	190.98	337.14	4045.68
Oct	3	73.08	95.49	168.57	2022.84
Nov	0	0	0	0	0
Dec	0	0	0	0	0

11.3. Bill:

Table 4: monthly electricity bills

Months	Monthly electricity power consumption	Monthly Bill (1 unit = 60pkr)
January	0	0
Feb	0	0
March	1348.56	80913.6
Apr	2022.84	121370.4
May	2022.84	121370.4
June	4045.68	242740.8
July	4045.68	242740.8
Aug	4045.68	242740.8
Sep	4045.68	242740.8
Oct	2022.84	121370.4
Nov	0	0
Dec	0	0



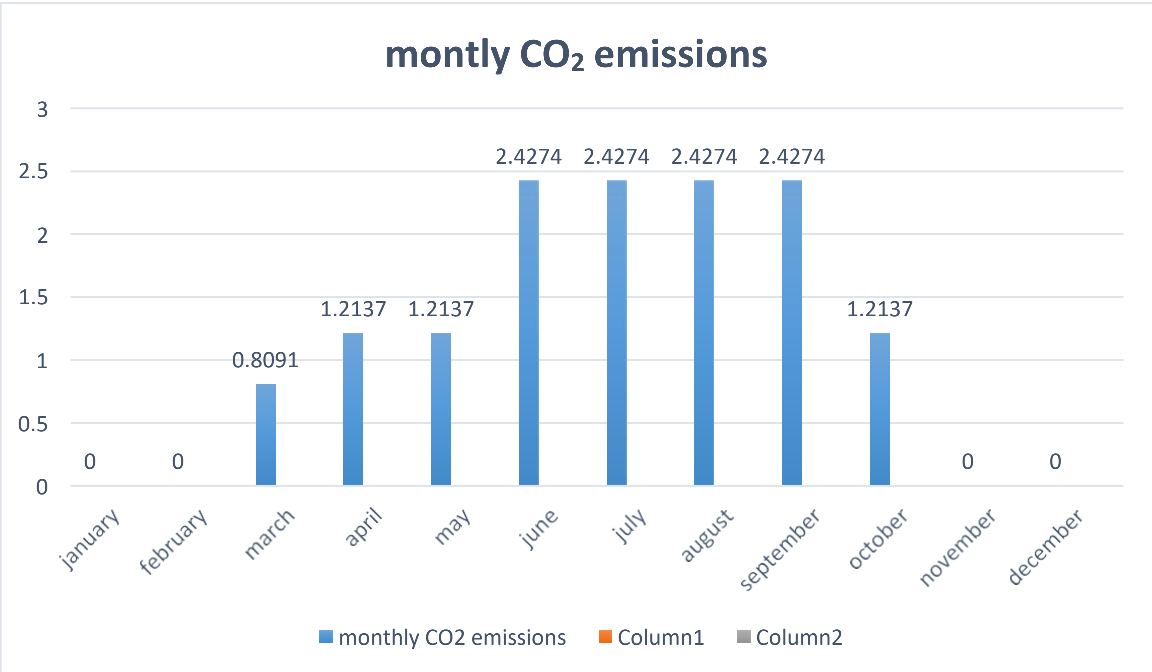
12. Monthly CO_2 emissions:

monthly CO_2 emissions = monthly energy consumption × CO_2 emissions per KWh

CO_2 emissions per hour = 0.6 kg of CO_2 per KWh

Table 5: environmental impact of this HVAC system in tCO_2 for different months of year

Months	**Monthly electricity /power consumption**	**Monthly CO_2 emissions (tCO_2)**
January	0	0
Feb	0	0
March	1348.56	1348.56 × 0.6/ 1000= 0.8091
Apr	2022.84	2022.84× 0.6/ 1000= 1.2137
May	2022.84	2022.84× 0.6/ 1000=1.2137
June	4045.68	4045.68× 0.6/ 1000= 2.4274
July	4045.68	4045.68× 0.6/ 1000= 2.4274
Aug	4045.68	4045.6.8× 0.6/ 1000=2.4274
Sep	4045.68	4045.68× 0.6/ 1000=2.4274
Oct	2022.84	2022.84× 0.6/ 1000=1.2137
Nov	0	0
Dec	0	0



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