COMPLEX ENGINEERING PROBLEM REPORT

"Designing of HVAC system for Mechanical Hall"

Submitted to

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Chairman
Department of Mechanical Engineering

in partial fulfillment of the requirements for the

Course: Heating, Ventilation and Air Conditioning

By

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DEPARTMENT OF MECHANICAL ENGINEERING
MIRPUR UNIVERSITY SCIENCE & TECHNOLOGY (MUST)

Outline

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11. Conclusion and Recommendations

1. Abstract:

This report presents a thorough examination and design proposal for the HVAC system designated for the Mechanical Hall at MUST. The focus is on achieving optimal indoor environmental conditions through rooftop installation and ducted chilled air supply. Detailed calculations of sensible and latent heat loads form the basis for the selection of HVAC components, including the Chiller, Air Handling Unit, Pumps, and Fans. Manufacturer details and technical specifications are meticulously outlined. The report further provides a visual representation of the HVAC system layout. Monthly variations in electricity consumption and associated costs are graphically presented, offering insights into operational efficiency. An environmental impact analysis, measured in tCO_2 emissions, is conducted for different months, with corresponding graphical illustrations. Through this holistic approach, the report aims to provide actionable insights for the implementation of an efficient, cost-effective, and environmentally conscious HVAC solution tailored to the unique needs of the Mechanical Hall.

2. Objective:

The primary objective of this report is to provide a detailed analysis and design proposal for the installation of an HVAC system on the roof of the Mechanical Hall at MUST. The focus is on achieving optimal indoor environmental conditions, considering a design set point of 23°C Dry Bulb Temperature (DBT) and 50% Relative Humidity (RH). The report aims to address key aspects, including the calculation of sensible and latent heat loads, the selection of appropriately sized HVAC components, and the presentation of manufacturer details and technical specifications. Additionally, the report aims to analyze the energy consumption and associated costs, providing a monthly variation graph. Furthermore, an environmental impact assessment in terms of tCO_2 emissions for different months will be conducted, with corresponding graphical representation.

3. Introduction:

The Mechanical Hall, serving as a hub for diverse events at MUST, demands a robust HVAC system to ensure optimal indoor conditions for various activities. This report outlines a comprehensive design approach, with the HVAC system planned for rooftop installation, delivering chilled air through ducts. The chosen design parameters, including indoor conditions of 23°C DBT and 50% RH, and outdoor conditions of 37°C DBT and 60% RH, set the stage for a meticulous analysis. This introduction provides a brief overview of the report's objectives, emphasizing the significance of a tailored HVAC system to meet the unique requirements of the Mechanical Hall. The subsequent sections delve into the technical details of the system design, component selection, layout, energy consumption, and environmental impact.

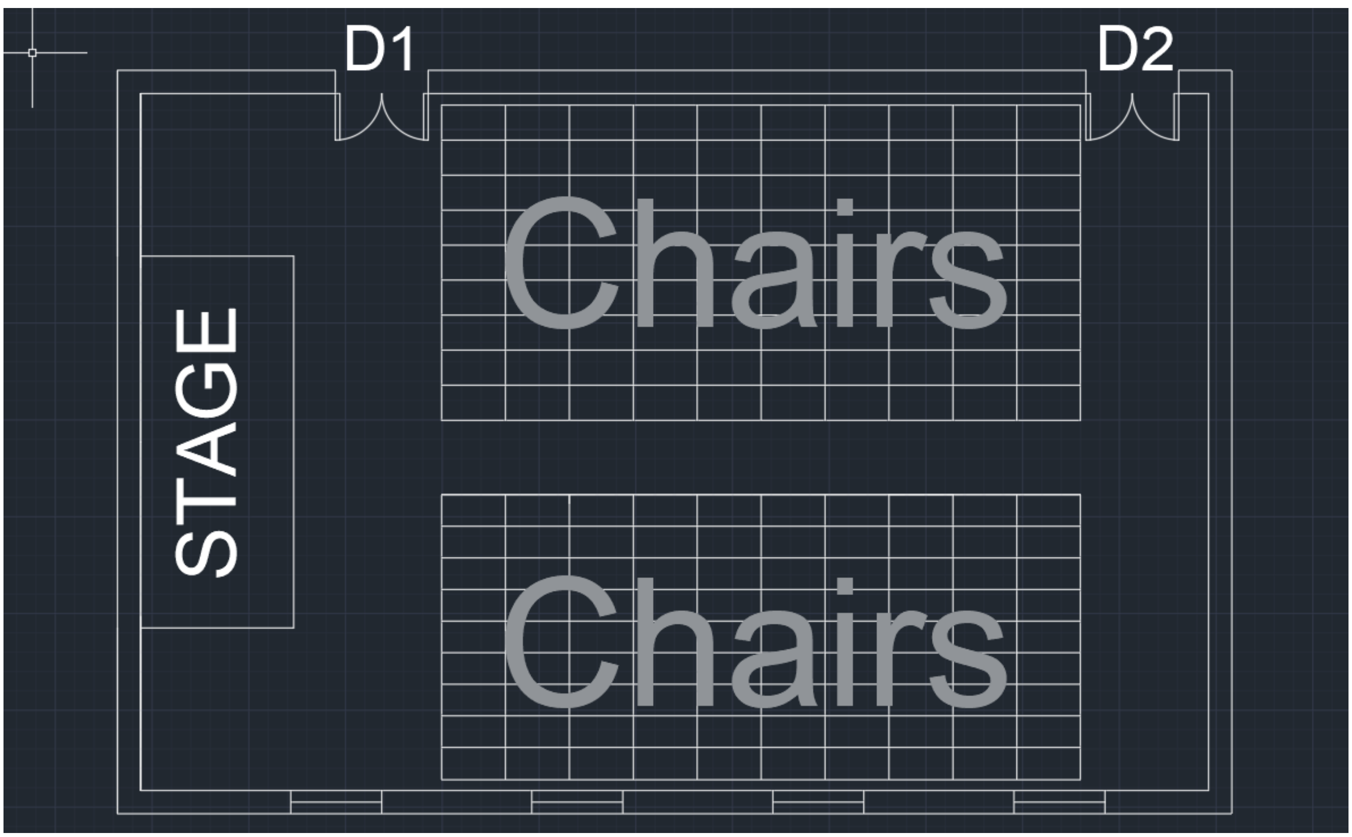
4. Calculation the total sensible heat load in kWs:

Sensible heat gain refers to the transfer of heat to a space that results in an increase in the dry bulb temperature (DBT) without any change in moisture content. In other words, it is the heat that causes a change in temperature but not humidity.

The factors in sensible cooling load include:

1. Sensible heat gain due to ventilation air and air infiltration through cracks in the building, doors and windows
2. Sensible heat gain due to lightning
3. Sensible heat gain due to projector
4. Sensible heat gain due to occupants in the building
5. Sensible heat gain due to walls
6. Sensible heat gain due to window glass
7. Sensible heat gain due to roof

Layout of Mechanical Hall



Dimensions of Mechanical hall

Length=60 ft. =18.288 m

Width=40 ft. =12.192 m

Height=22 ft. =6.7056m

Amount of in filtered air, (Vint) = $\frac{\text{LxWxHxAc}}{60}\ m^3\ /min$

Ac= Number of air changes per hour

For a hall with one wall exposed take Ac=1

Amount of in filtered air, (Vint) =$\frac{18.288\text{x}12.192\text{x}6.7056\text{x}1}{60}$=24.92 $m^3\ /min$

Mark point 1 (37°C DBT and 60% RH) and point 2 (23°C DBT and 50% RH) on psychrometric chart.

From psychrometric chart, we find that specific volume of air at point 1,

$v_{s1=}$0.909 m^3/kg of dry air

Enthalpy of air at point 1,

$h_{1=}$99.3 kJ/kg of dry air

Enthalpy of air at point 2,

$h_{2=}$45.6 kJ/kg of dry air

and enthalpy of air at point A,

$h_{A=}$ 60 kJ/kg of dry air

Mass of infiltrated air at point 1 is

$m_{1}=\frac{\text{Vint}}{\text{vs1}}=\frac{24.92}{0.909}$=27.4 kg/s

Sensible heat gain due to air infiltration

Q=m_1 (h_A-h_2)

Q=27.4 (60-45.6)

Q=394.56 kJ/min=6.6 kW

Sensible heat gain due to lightning

Total lights = 141 LED bulbs+ 8 tube lights

Q_{lights} = (Total LED bulbs× Watt of each bulb) + (Total tube lights× Watt of each tube light)

$Q_{lights\ =}$ (141×25) + (8×36)

$Q_{lights=}$3813 W=3.813kW

Sensible heat gain due to projector

Q_{proj}= Number of Projector × Watts of each projector

Q_{proj}=2×400

Q_{proj}=800 W=0.8kW

Sensible heat gain due to occupants

Q_{occup}= Qs per person × Number of occupants

Q_{occup}= 53×171=9063W=9.063kW

Sensible heat gain due to walls

Heat gain through building walls can be calculated using following equations

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

Q= heat load in watt

U= Overall heat transfer coefficient=1.53 W/m^2K (For Brick Wall)

A= Surface Area of wall

ΔT= Temperature difference

Area of windows

Length=1.447m

Height=1.1684m

Area of 1 window= 1.447×1.1684=1.6907m^2

Area of 9 windows= 1.6907×9=15.22 m^2

Surface Area of walls:

Surface area of East wall

Area= 18.288 × 6.7056

Area= 122.63 m^2

Surface area of West wall

Area= 18.288 × 6.7056

Area= 122.63 – 15.22 (window area)

Area= 107.41 m^2

Surface area of South wall

Area= 12.191 × 6.7056

Area= 81.75 m^2

Surface area of north wall

Area= 12.191 × 6.7056

Area= 81.75 m^2

Temperature	East Wall	South Wall	North Wall	West Wall
Indoor t_{d2} (K)	23˚C=296K	23˚C=296K	23˚C=296K	23˚C=296K
Outdoor t_{d1}(K)	25 ˚C=298K	30 ˚C=303K	30 ˚C=303K	40 ˚C=313K
Difference ΔT (K)	2	7	7	17

Total Heat gain through building walls

Wall-1 (East)

c= 1.53 × 122.63 × 2

Q_e = 375.248 W

Q_e = 0.375 kW

Wall-2 (South)

Q_s = 1.53 × 81.75 × 7

Q_s = 875.54 W

Q_s = 0.875 kW

Wall-3 (West)

$Q_w = 1.53 \times 107.41 \times 17$

$Q_w = 2793.73$ W

$Q_w = 2.793$ kW

Wall-4 (North)

$Q_n = 1.53 \times 81.75 \times 7$

$Q_n = 875.54$ W

$Q_n = 0.875$ kW

Total load of walls = Q_{e+} Q_{s+} Q_{w+} Q_n

Total load of walls =0.375+0.875+ 2.793+0.875

Total load of walls =4.918 kW

Sensible heat gain due to window glass

Sensible heat gain due to window glass consists of two components

- Conduction heat transfer
- Direct Solar transmission

By Conduction

$Q_{cond}=U\times A\times\Delta T$

Q= heat load in watt

U= Overall heat transfer coefficient=1.2716 W/m^2K (From ASHRAE fundamental 2001)

A= Surface Area of windows

ΔT= Temperature difference

For 1 window

$Q_{cond}=1.2716\times1.6907\times17=36.55W$

For 9 windows

$Q_{cond}=36.55\times9=328.95W$

$Q_{cond}=0.328$ kW

By direct solar transmission

$Q_{solar}=A\times SHGC$ $Q_{solar}=SR$

A= Surface Area of windows

SHGC=Sensible Heat Gain Coefficient=0.6

SR= Solar radiations=350 radiations

For 1 window

$Q_{solar}=1.6907\times0.6\times350=355.047W$

For 9 windows

$Q_{solar}=355.047\times9=3195.423W$

$Q_{solar}=3.195kW$

Sensible heat gain due to window glass= $Q_{cond}+$ Q_{solar}

Sensible heat gain due to window glass=0.328 kW+3.195kW

Sensible heat gain due to window glass=3.523kW

Sensible heat gain due to roof

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

Q= heat load in watt

U= Overall heat transfer coefficient=1.6147 W/m^{2}K (From ASHRAE fundamental 2001)

A= Surface Area of roof

ΔT= Temperature difference

Q_{roof} = 1.6147 × (18.288×12.192) ×2

Q_{roof} =720.05W

Q_{roof} =0.72 kW

Sensible heat gain due to ventilation

Q_{vent}=Q_{air}×ΔT×1.08

Q_{vent}= $\frac{L\times W\times H\times Ac}{60}$×$\Delta T$×1.08

Q_{vent}=24.92 × 12×1.08=322.96W

Q_{vent}=0.322kW

Total Sensible Heat Load

Total Sensible Heat Load= Sensible heat gain due air infiltration + Sensible heat gain due to lightning + Sensible heat gain due to projector + Sensible heat gain due to occupants in the building + Sensible heat gain due to walls + Sensible heat gain due to window glass + Sensible heat gain due to roof + Sensible heat gain due to ventilation air

Total Sensible Heat Load=6.6+3.813+0.8+9.063+4.918+3.523+0.72+0.322

Total Sensible Heat Load=29.759 kW

5. Calculation the total latent heat load in kWs

Latent heat gain refers to the transfer of heat to a space that results in a change in moisture content, typically an increase in humidity, without a corresponding change in dry bulb temperature. It represents the heat associated with the phase change of water vapor.

The factors including latent cooling load are:

1. Latent heat gain due to ventilation air and air infiltration through cracks in the building, doors and windows
2. Latent heat gain due to occupants in the building
3. Latent heat gain due to walls

Latent heat gain due to air infiltration

Q=m_1 (h_1-h_A)

Q=27.4 (99.3-60) = 1076.82 kJ/min

Q=17.95 kW

Latent heat gain due to occupants in the building

Q_{occup}= Q_L per person × Number of occupants

Q_{occup}= 44×171=7524W=7.524kW

Latent heat gain due to ventilation air

Q = Q_{air} ×ΔW×2500

Q_{air}= 24.92 m^3 / min (as calculated in sensible load)

ΔW= Specific Humidity Difference

From psychrometric chart,

Specific humidity of air at point 1,

W_1=0.024 kg/kg of dry air

Specific humidity of air at point 2,

W_2= 0.0086 kg/kg of dry air

ΔW= W_1- W_2=0.024-0.0086

ΔW= 0.0154 kg/kg of dry air

Now

Q_{vent} =24.92×0.0154×2500

Q_{vent} =959.42 W

Q_{vent}=0.959 kW

Total Latent Heat Load

Total Latent Heat Load= latent heat gain due air infiltration+ latent heat gain due to occupants in the building+ Latent heat gain due to ventilation air

Total Latent Heat Load= 17.95+7.524+0.959

Total Latent Heat Load=26.43 kW

6. Capacity of AC

Total cooling load = Total Sensible Load + Total Latent Load

Total cooling load =29.759+26.43

Total cooling load =56.189 kW

TR=$\frac{Total\ cooling\ load\ in\ kW}{3.5}$

TR=$\frac{56.189}{3.5}$

TR= 16.054 TR

Cooling Load Calculation by HAP Software:

Air System Sizing Summary for Mechanical hall

Project Name: HVAC — 09/27/2023
Prepared by: get — 12:15am

Air System Information

Air System Name	Mechanical hall		Number of zones	1	
Equipment Class	UNDEF		Floor Area	2400.0	ft²
Air System Type	SZCAV		Location	Mirpur, Pakistan	

Sizing Calculation Information

Calculation Months	Jan to Dec	Zone CFM Sizing	Sum of space airflow rates
Sizing Data	Calculated	Space CFM Sizing	Individual peak space loads

Central Cooling Coil Sizing Data

Total coil load	11.9	Tons	Load occurs at	Jul 1600	
Total coil load	142.3	MBH	OA DB / WB	107.9 / 74.2	°F
Sensible coil load	120.0	MBH	Entering DB / WB	76.4 / 65.4	°F
Coil CFM at Jul 1600	7126	CFM	Leaving DB / WB	59.9 / 58.9	°F
Max block CFM	7126	CFM	Coil ADP	58.1	°F
Sum of peak zone CFM	7126	CFM	Bypass Factor	0.100	
Sensible heat ratio	0.843		Resulting RH	57	%
ft²/Ton	202.4		Design supply temp.	58.0	°F
BTU/(hr-ft²)	59.3		Zone T-stat Check	1 of 1	OK
Water flow @ 10.0 °F rise	28.47	gpm	Max zone temperature deviation	0.0	°F

Supply Fan Sizing Data

Actual max CFM	7126	CFM	Fan motor BHP	0.00	BHP
Standard CFM	6746	CFM	Fan motor kW	0.00	kW
Actual max CFM/ft²	2.97	CFM/ft²	Fan static	0.00	in wg

Outdoor Ventilation Air Data

Design airflow CFM	0	CFM	CFM/person	0.00	CFM/person
CFM/ft²	0.00	CFM/ft²			

Zone Sizing Summary for Mechanical hall

Project Name: HVAC
Prepared by: get

09/27/2023
12:15am

Air System Information

Air System Name .... **Mechanical hall**
Equipment Class .... **UNDEF**
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**

Zone Sizing Data

Zone Name	Maximum Cooling Sensible (MBH)	Design Airflow (CFM)	Minimum Airflow (CFM)	Time of Peak Load	Maximum Heating Load (MBH)	Zone Floor Area (ft²)	Zone CFM/ft²
Zone 1	123.9	7126	7126	Jul 1700	0.0	2400.0	2.97

Zone Terminal Sizing Data

No Zone Terminal Sizing Data required for this system.

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (MBH)	Time of Load	Air Flow (CFM)	Heating Load (MBH)	Floor Area (ft²)	Space CFM/ft²
Zone 1							
Mechanical hall	1	123.9	Jul 1700	7126	0.0	2400.0	2.97

Air System Design Load Summary for Mechanical hall

Project Name: HVAC | 09/27/2023
Prepared by: get | 12:15am

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jul 1600 COOLING OA DB / WB 107.9 °F / 74.2 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 76.0 °F / 59.8 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	162 ft²	14572	-	162 ft²	-	-
Wall Transmission	4147 ft²	34809	-	4147 ft²	0	-
Roof Transmission	2400 ft²	6829	-	2400 ft²	0	-
Window Transmission	162 ft²	5305	-	162 ft²	0	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	20 ft²	181	-	20 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	2 W	8	-	0	0	-
Task Lighting	3813 W	13010	-	0	0	-
Electric Equipment	800 W	2730	-	0	0	-
People	180	41399	21600	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	3% / 3%	3565	648	0%	0	0
>> Total Zone Loads	-	122408	22248	-	0	0
Zone Conditioning	-	120003	22248	-	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	7126 CFM	0	-	7126 CFM	0	-
Ventilation Load	0 CFM	0	0	0 CFM	0	0
Supply Fan Load	7126 CFM	0	-	7126 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	120003	22248	-	0	0
Central Cooling Coil	-	120003	22282	-	0	0
>> Total Conditioning	-	120003	22282	-	0	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: HVAC 09/27/2023
Prepared by: get 12:15am

July DESIGN COOLING DAY, 1600

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	107.9	0.01145	0	400	0	0
Vent - Return Mixing	Outlet	76.4	0.01165	7126	5514	-	-
Central Cooling Coil	Outlet	59.9	0.01095	7126	5514	120003	22282
Supply Fan	Outlet	59.9	0.01095	7126	5514	0	-
Cold Supply Duct	Outlet	59.9	0.01095	7126	5514	-	-
Zone Air	-	76.4	0.01165	7126	5514	120003	22248
Return Plenum	Outlet	76.4	0.01165	7126	5514	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.023 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4493.9 BTU/(hr-CFM)
Site Altitude = 1506.0 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	122408	Cooling	120003	76.4	7126	5514	0	0

System Psychrometrics for Mechanical hall

Project Name: HVAC 09/27/2023
Prepared by: get 12:15am

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	0.00786	0	400	0	0
Vent - Return Mixing	Outlet	70.0	0.01300	7126	800	-	-
Central Cooling Coil	Outlet	70.0	0.01300	7126	800	0	0
Supply Fan	Outlet	70.0	0.01300	7126	800	0	-
Cold Supply Duct	Outlet	70.0	0.01300	7126	800	-	-
Zone Air	-	70.0	0.01300	7126	800	0	0
Return Plenum	Outlet	70.0	0.01300	7126	800	0	-

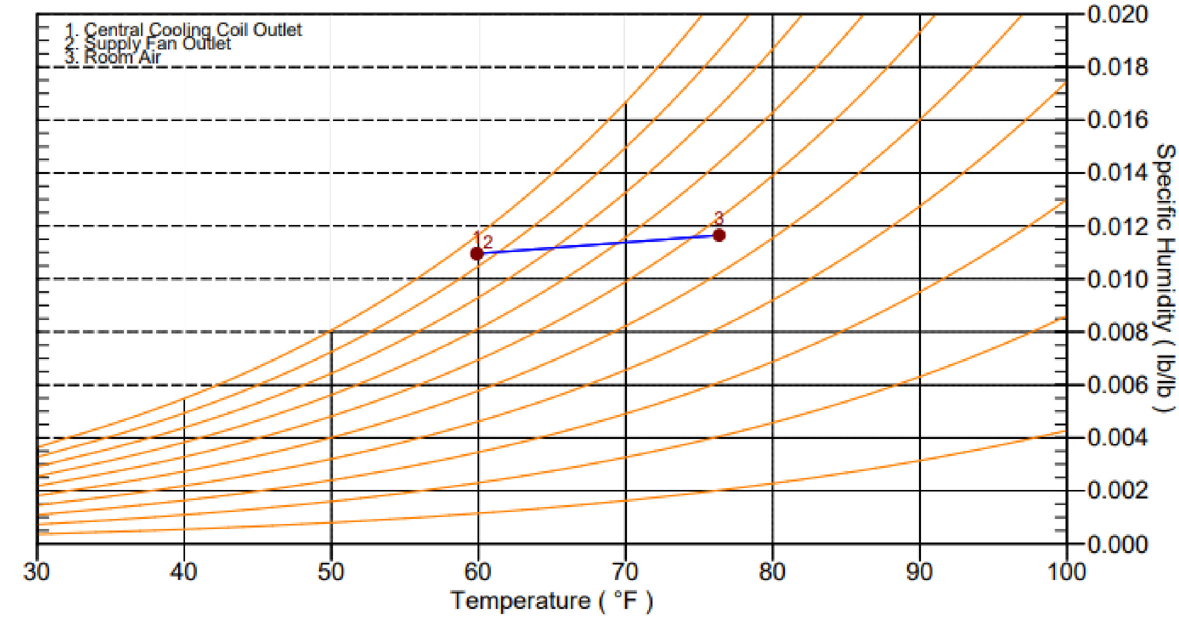
Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.023 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4493.9 BTU/(hr-CFM)
Site Altitude = 1506.0 ft

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	7126	800	0	0

Project Name: HVAC Prepared by: get	**Psychrometric Analysis for Mechanical hall**	09/27/2023 12:15am

Location: Mirpur, Pakistan
Altitude: 1506.0 ft.
Data for: July DESIGN COOLING DAY, 1600



7. Selection of appropriate size of HVAC system and its components:

In addressing the environmental and operational demands of the extensively utilized Mechanical Hall at MUST, a Variable Refrigerant Flow (VRF) cooling-only system has been meticulously designed. This VRF system, exemplified by the Samsung DVM S2, caters to the unique requirements of the facility, where the indoor conditions mandate a constant 23°C Dry Bulb Temperature and 50% Relative Humidity, amidst challenging outdoor conditions of 37°C Dry Bulb Temperature and 60% Relative Humidity. The modular design of the VRF system offers flexibility and scalability, ensuring efficient cooling tailored to diverse event scenarios. Employing inverter technology, the system's variable-speed compressors optimize energy consumption, contributing to elevated efficiency levels, particularly during partial-load conditions. Zoning capabilities enable precise temperature control in different areas, enhancing occupant comfort and energy savings. The Samsung DVM S2's cooling-only configuration aligns seamlessly with the project's specifications, offering a sustainable and eco-friendly solution for the Mechanical Hall's HVAC needs. The system's user-friendly controls, quiet operation, and potential integration with building management systems further position it as an ideal choice for meeting the cooling demands of this dynamic and multifunctional space.

Technical specifications of our selected system are as follows:

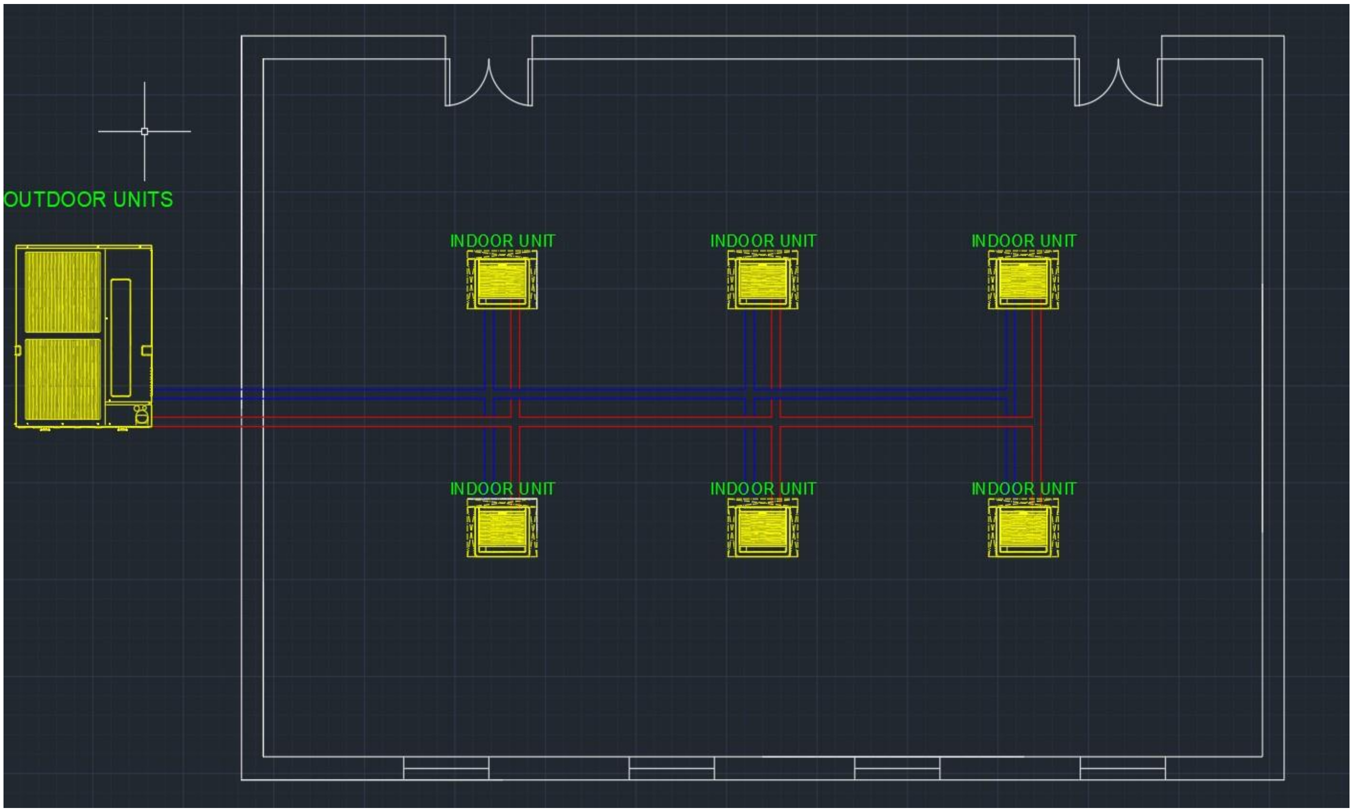
VRF DVM S2	
Manufacturer	Samsung Electronics Co. Ltd 129
Model Name	AM200AXVANC/TL
Capacity	56 kW
Power Input	12.18 kW
Current Input	19.6 A
C.O.P	4.6
Refrigerant Type	R410A
Operation Temperature Range	-5~53°C
Heat Exchanger	
Type	Fin & Tube
Material (Fin)	Aluminum (Al)
Material Tube	Copper (Cu)
ModelFin Treatment	Anti-corrosion
Compressor	
Type	Inverter Scroll x 1
Model	DS4GM7090FV* x 1
Fan	
Type	Propeller
Motor (Output)	620 W (0.6 kW)
Number of Unit (EA)	2 EA
Air Flow Rate	342 CMM
Piping Connections	
Liquid Pipe (Φ, mm)	15.88 mm
Liquid Pipe Type	Braze Connection
Gas Pipe (Φ, mm)	28.58 mm
Gas Pipe Type	Braze Connection
Installation Maximum Length (m)	220 m

The selected VRF cooling-only system, exemplified by the Samsung DVM S2, brings forth several benefits that make it an optimal choice for addressing the HVAC needs of the Mechanical Hall at MUST:

a. **Energy Efficiency:** With inverter-driven compressors, the VRF system enhances energy efficiency by adjusting its speed to match the specific cooling requirements, ensuring optimal performance and reduced energy consumption.
b. **Modular Design and Scalability:** The modular design of the VRF system enables easy customization and expansion based on the changing needs of the Mechanical Hall. This scalability ensures adaptability to the evolving requirements of various events and activities.
c. **Precise Temperature Control:** The system's ability to maintain a constant indoor condition of 23°C DBT and 50% RH, even in the face of challenging outdoor conditions, ensures a comfortable and controlled environment for events and gatherings.
d. **Quiet Operation:** The VRF system, including the Samsung DVM S2, is designed for quiet operation, minimizing disturbances during events and enhancing the overall user experience.
e. **Smart Controls:** The system's advanced control features provide user-friendly interfaces for monitoring and adjusting settings. This can include smart control options that allow for remote management, scheduling, and optimization.
f. **Integration Capabilities:** The VRF system can integrate seamlessly with building management systems (BMS), enabling centralized control and monitoring. This integration enhances overall system efficiency and facilitates comprehensive facility management.
g. **Consistent Performance:** The cooling-only configuration of the Samsung DVM S2 aligns precisely with the specified requirements, ensuring consistent and reliable performance without unnecessary complexity.
h. **Cost-Effective Operation:** Through its energy-efficient design, zoning capabilities, and precise control, the VRF system contributes to cost savings in both operational and maintenance aspects, making it a financially prudent choice for the Mechanical Hall.

In summary, the chosen VRF system offers a holistic solution that not only meets the cooling needs of the Mechanical Hall efficiently but also brings forth advantages in terms of flexibility, sustainability, and user comfort.

8. Drawing of the layout of HVAC system:



9. Calculation of annual electricity consumption and electricity bill associated with this HVAC system:

Annual Electricity Consumption

Hall operates for 6 hours a day

Daily latent load power consumption= latent load in kW × Operating hours

Daily latent load power consumption=26.43×6=158.58 kWh

Daily sensible load power consumption= sensible load in kW × Operating hours

Daily sensible load power consumption=29.759×6=178.554 kWh

Total Daily Power Consumption= Daily Latent load Power consumption + Daily sensible load Power consumption

Total Daily Power Consumption=158.58 kWh+178.554 kWh

Total Daily Power Consumption=337.134 kWh

Hall operates for 12 days a month

Monthly Power Consumption= Total daily Power Consumption × Operating days

Monthly Power Consumption=337.134×12

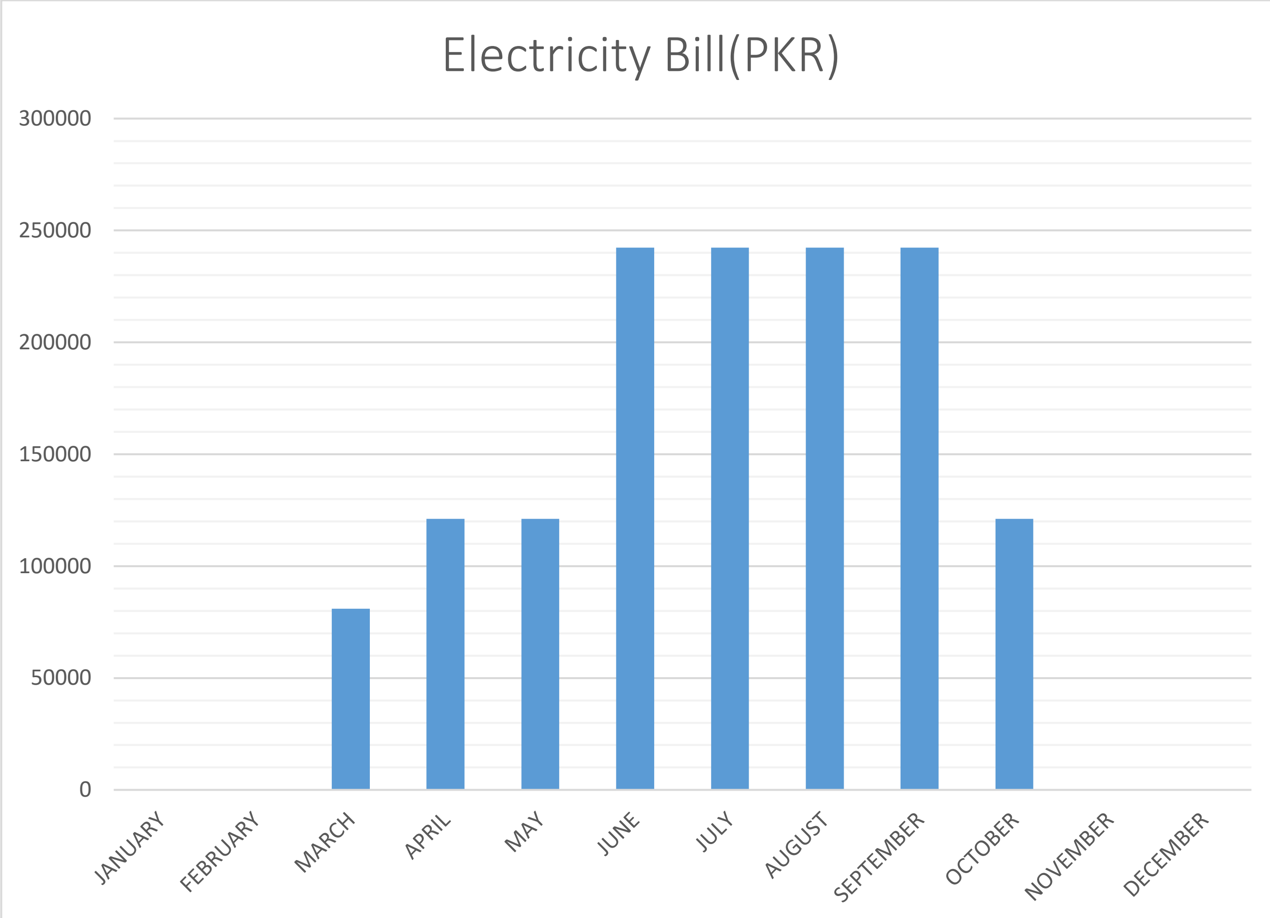
Monthly Power Consumption=4045.608 kWh

Annual Power Consumption= Monthly Power Consumption×12
Annual Power Consumption=4045.608×12
Annual Power Consumption=48547.3 kWh

For monthly variation of electricity bill with the electrical power consumption in the form of graph, we have to calculate data as shown in table below.

FORMULAS USED IN TABLE
Daily latent load power consumption= latent load in kW × Operating hours
Daily sensible load power consumption= sensible load in kW × Operating hours
Total Daily Power Consumption= Daily Latent load Power consumption + Daily sensible load Power consumption
Monthly Power Consumption= Total daily Power Consumption × Operating days
Monthly total Electricity Bill= Monthly Power Consumption×60 (price of one unit in PKR)

Months	**Jan**	**Feb**	**Mar**	**Apr**	**May**	**Jun**	**Jul**	**Aug**	**Sep**	**Oct**	**Nov**	**Dec**
Operating Hours	0	0	2	3	3	6	6	6	6	3	0	0
Daily latent Power Consumption (kWh)	0	0	53	79	79	158	158	158	158	79	0	0
Daily Sensible Power Consumption(kWh)	0	0	59.5	89.3	89.3	178.5	178.5	178.5	178.5	89.3	0	0
Daily total Power Consumption(kWh)	0	0	112.5	168.3	168.3	336.5	336.5	336.5	336.5	168.3	0	0
Monthly total Power Consumption(kWh)	0	0	1350	2019.6	2019.6	4038	4038	4038	4038	2019.6	0	0
Monthly total Electricity Bill(PKR)	0	0	81000	121176	121176	242280	242280	242280	242280	121176	0	0



10.Environmental impact of HVAC system in tCO2 for different months of the year:

The environmental impact of the VRF HVAC system, specifically the Samsung DVM S2 cooling-only system, can be assessed in terms of its carbon footprint, measured in metric tons of carbon dioxide equivalent (tCO2), for different months of the year. The analysis takes into account the energy consumption associated with the system's operation. Here's an outline of the monthly variation in tCO2 emissions:

1. **Seasonal Variation:** The environmental impact would likely show variations across different seasons. For instance, during hotter months where the system operates more frequently, the energy consumption and associated tCO2 emissions may be higher.
2. **Energy Efficiency Measures:** If the VRF system is equipped with energy-saving features or if there are measures in place to optimize its efficiency, certain months may exhibit lower tCO2 emissions.
3. **Maintenance and Optimization:** Regular maintenance and optimization of the HVAC system can impact its energy efficiency, directly influencing the environmental footprint. This may lead to variations in tCO2 emissions over time.
4. **Graphical Representation:** Represent the monthly variation in tCO2 emissions in the form of a graph. This visual representation will make it easier to identify trends and patterns, allowing for a comprehensive understanding of the system's environmental impact throughout the year.

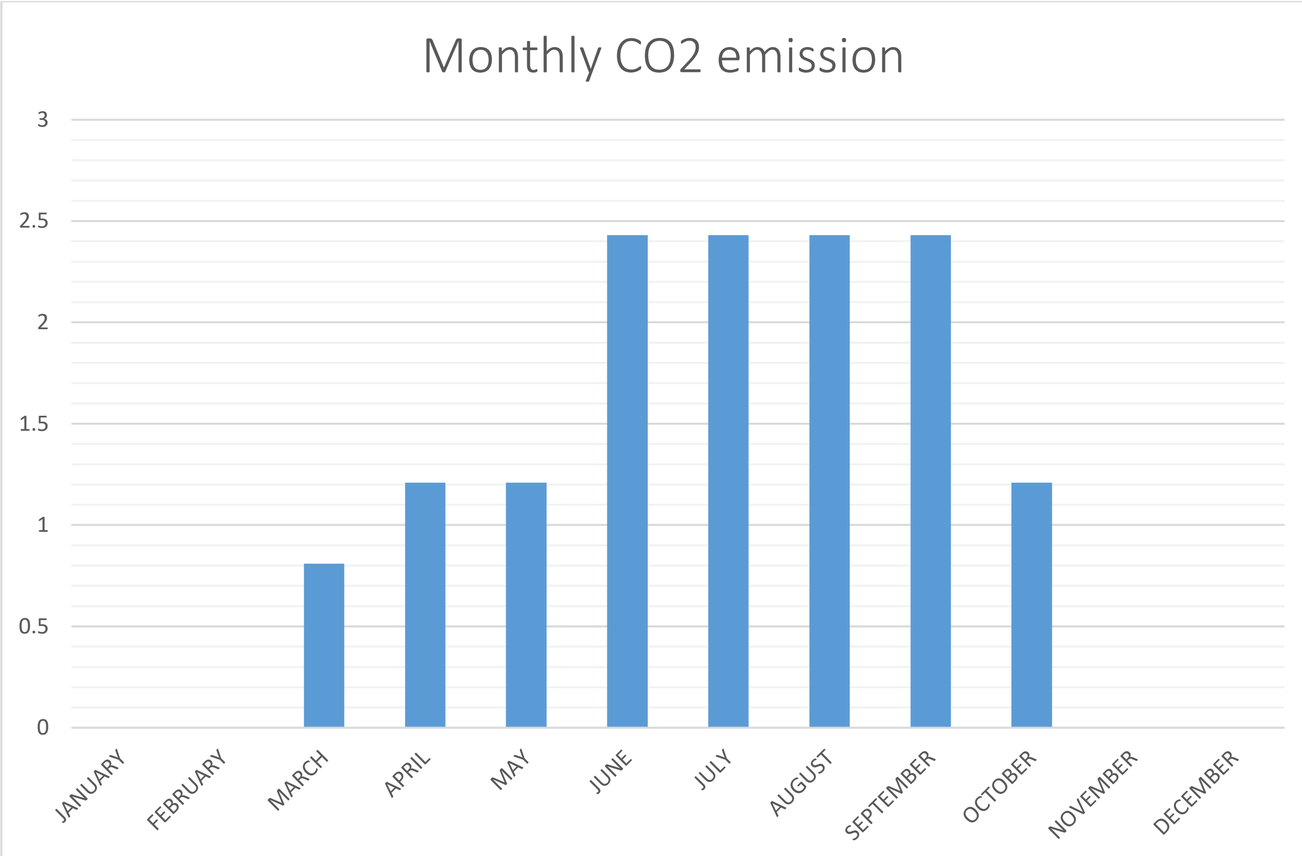
5. **Comparison and Analysis:** Compare the tCO2 emissions across different months to identify peak periods and areas for potential improvement. Analyze the data to draw insights into the system's performance and its alignment with environmental sustainability goals.

Monthly CO_2 emission is calculated for all months by using following formula:

$$\text{Monthly } CO_2 \text{ emission} = \frac{Monthly\ Energy\ Consumption \times \mathrm{CO2\ emission\ per\ kWh}}{1000}$$

CO_2 emission per hour= 0.6 kg of CO_2 per kWh

Months	Monthly Power Consumption (kWh)	Monthly CO_2 emission (tCO_2)
January	0	0
February	0	0
March	1350	0.81
April	2019.6	1.21
May	2019.6	1.21
June	4038	2.43
July	4038	2.43
August	4038	2.43
September	4038	2.43
October	2019.6	1.21
November	0	0
December	0	0



11. Conclusion and Recommendations: Concluding the report, this section synthesizes the findings and offers actionable recommendations for further enhancing the HVAC system's efficiency and environmental performance. It emphasizes a holistic approach that balances operational requirements with sustainability goals. This comprehensive report provides a detailed roadmap for implementing an HVAC system that not only meets the unique needs of the Mechanical Hall but also aligns with the principles of efficiency and environmental responsibility.