Revitalizing Comfort: Designing an Energy–Efficient HVAC System for the University Auditorium

Comodidad revitalizante: Diseño de un Sistema HVAC Energéticamente Eficiente para el Auditorio Universitario

Revitalizando o Conforto: Projeto de um Sistema HVAC com Eficiência Energética para o Auditório Universitário

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Summary. - Nowadays, thermal comfort is becoming a major problem for people due to increasing global warming and climatic changes, but it can be resolved by the concept of Heating, Ventilation, and Air Conditioning (HVAC) systems. The purpose of HVAC is to provide occupants with a comfort zone so that they can feel comfortable according to their thermal comfort. The core objective of this study is to design and propose an HVAC system as per actual design conditions for the University Auditorium located in Karachi, Pakistan. A direct Expansion (DX – Type) system is installed in the Auditorium that has exceeded the lifespan of twenty years, refrigerant R-22 which is currently being used has been obsolete due to its high GWP (Global Warming Potential) and ODP (Ozone Depletion Potential) values which are 1810 and 0.05 respectively. To achieve the objective of this study, two approaches are employed. Cooling Load Temperature Difference (CLTD) method & Hourly Analysis Program (HAP) software. The cooling load calculated from the CLTD method is 202 kW equivalent to 57.5 Ton of Refrigeration (TR). On the other side, the cooling load calculated from HAP software is 192.8 kW equivalent to 55 TR. By considering the calculated cooling load for the University Auditorium, two different HVAC systems are proposed, based on Water cooled and Air-cooled Vapor Compression Cycle. After this study, engineers will be able to design an HVAC system for any facility as per design conditions. Also, they can propose different cost-effective and energy-efficient HVAC systems for that particular space.

Keywords: HVAC; Auditorium; Duct sizing; Cooling load; Piping.

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Resumen. - *Hoy en día, el confort térmico se está convirtiendo en un gran problema para las personas debido al aumento del calentamiento global y los cambios climáticos, pero puede ser resuelto por el concepto de sistemas de Calefacción, Ventilación y Aire Acondicionado (HVAC). El objetivo de HVAC es proporcionar a los ocupantes una zona de confort para que puedan sentirse cómodos de acuerdo con su confort térmico. El objetivo central de este estudio es diseñar y proponer un sistema HVAC según las condiciones de diseño reales para el Auditorio Universitario ubicado en Karachi, Pakistán. En el Auditorio se encuentra instalado un sistema de Expansión Directa (Tipo DX) que ha superado la vida útil de veinte años, el refrigerante R-22 que se utiliza actualmente ha quedado obsoleto por su alto GWP (Global Warming Potential) y ODP (Ozone Depletion Potencial) valores que son 1810 y 0.05 respectivamente. Para lograr el objetivo de este estudio, se emplean dos enfoques. Método de diferencia de temperatura de carga de enfriamiento (CLTD) y software de programa de análisis por hora (HAP). La carga de refrigeración calculada a partir del método CLTD es de 202 kW equivalente a 57,5 Toneladas de Refrigeración (TR). Por otro lado, la carga de refrigeración calculada a partir del software HAP es de 192,8 kW equivalente a 55 TR. Al considerar la carga de enfriamiento calculada para el Auditorio Universitario, se proponen dos sistemas HVAC diferentes, basados en el ciclo de compresión de vapor enfriado por agua y enfriado por aire. Después de este estudio, los ingenieros podrán diseñar un sistema HVAC para cualquier instalación según las condiciones de diseño. Además, pueden proponer diferentes sistemas HVAC rentables y energéticamente eficientes para ese espacio en particular.*

Palabras clave: *HVAC; Auditorio; Dimensionamiento de ductos; Carga de enfriamiento; Piping.*

Resumo. - *Hoje em dia, o conforto térmico está a tornar-se um grande problema para as pessoas devido ao aumento do aquecimento global e às alterações climáticas, mas pode ser resolvido pelo conceito de sistemas de Aquecimento, Ventilação e Ar Condicionado (HVAC). O objetivo do HVAC é proporcionar aos ocupantes uma zona de conforto para que se sintam confortáveis de acordo com o seu conforto térmico. O objetivo principal deste estudo é projetar e propor um sistema HVAC de acordo com as condições reais de projeto para o Auditório Universitário localizado em Karachi, Paquistão. No Auditório está instalado um sistema de Expansão Direta (Tipo DX) que ultrapassou sua vida útil de vinte anos. O refrigerante R-22 atualmente utilizado tornou-se obsoleto devido ao seu alto GWP (Potencial de Aquecimento Global) e ODP (Depleção de Ozônio). Potencial) valores que são 1810 e 0,05 respectivamente. Para atingir o objetivo deste estudo, duas abordagens são utilizadas. Método de diferença de temperatura de carga de resfriamento (CLTD) e software de programa de análise horária (HAP). A carga de resfriamento calculada a partir do método CLTD é de 202 kW equivalente a 57,5 toneladas de refrigeração (TR). Por outro lado, a carga de refrigeração calculada a partir do software HAP é de 192,8 kW equivalente a 55 TR. Ao considerar a carga de refrigeração calculada para o Auditório Universitário, são propostos dois sistemas HVAC diferentes, baseados no ciclo de compressão de vapor refrigerado a água e arrefecido a ar. Após este estudo, os engenheiros serão capazes de projetar um sistema HVAC para qualquer instalação com base nas condições de projeto. Além disso, eles podem propor diferentes sistemas HVAC econômicos e energeticamente eficientes para esse espaço específico.*

Palavras-chave: *AVAC; Público; Dimensionamento de dutos; Carga de resfriamento; Tubulação.*

Nomenclature:

A	Area
ACH	Air Changes per Hour
$CLTD_{adj}$	Cooling Load Temperature Difference Adjusted
D	Diameter
E	Efficiency
F_b	Ballast Factor
F_u	Utilization Factor
g	Acceleration due to gravity
h_i	Inside Relative Humidity
h_L	Head Loss
h_M	Mixed air enthalpy
h_o	Outside air enthalpy
h_R	Recirculated air enthalpy
l	Length
L_{eq}	Equivalent Length
m_M	Mixed air flowrate
m_o	Outdoor air flowrate
m_R	Recirculated air flowrate
N	No. of Occupant
$\dot{Q}$	Heat Gain
$\dot{Q}_{il}$	Latent Heat Gain by Infiltration
$\dot{Q}_{is}$	Sensible Heat Gain by Infiltration
$\dot{Q}_{latent}$	Latent Heat Gain
$\dot{Q}_{sensible}$	Sensible Heat Gain
$\dot{Q}_{vl}$	Latent Heat Gain by Ventilation
$\dot{Q}_{vs}$	Sensible Heat Gain by Ventilation
t_a	Temperature below the floor of the Auditorium
t_{avg}	Outside average temperature
t_i	Inside design temperature
T_M	Mixed air temperature
t_0	Outside design temperature
T_S	Supply air temperature
R	Thermal Resistance
R_A	Area Outdoor Air rate
R_P	People's Outdoor Air rate
U	Thermal Transmittance
V	Volume
v	Velocity
$\dot{V}_i$	Infiltration Air Flowrate
$\dot{V}_m$	Minimum Outdoor Air Flowrate
$\dot{V}_o$	Outside Air Flowrate
$\dot{V}_r$	Recirculation Air Flowrate
$\dot{V}_v$	Ventilation Air Flowrate
W	Wattage
W_i	Inside Humidity Ratio
W_0	Outside Humidity Ratio

Greek Symbols:

ρ	Density
ΔP	Pressure drop
$\emptyset$	Relative Humidity
$q_{e,\theta}$	Heat gain through wall or roof, at calculation hour θ
θ	Time
δ	Time interval
$t_{e,\theta-n\delta}$	Sol-air temperature at time $\theta - n\delta$
$t_{r,c}$	Constant indoor room temperature
b_n, c_n, d_n	Conduction transfer function coefficients

Subscripts:

a	adjacent space
adj	adjusted
avg	average
i	inside
il	infiltration latent
is	infiltration sensible
M	mixed
m	minimum outdoor air
n	summation index (each summation has as many terms as there are non-negligible values of coefficients)
o	outside
P	people
R	recirculated
r	recirculation
S	supply
vl	ventilation latent
vs	ventilation sensible

Acronyms

AHU	Air Handling Unit
ASHRAE	American Society of Heating, Refrigeration & Air Conditioning Engineers
CLF	Cooling Load Factor
CLTD	Cooling Load Temperature Difference
DX	Direct Expansion
EER	Energy Efficiency Ratio
GWP	Global Warming Potential
HAP	Hourly Analysis Program
HFC	Hydrofluorocarbons
HVAC	Heating Ventilation and Air Conditioning
IAQ	Indoor Air Quality
LHG	Latent Heat Gain
ODP	Ozone Depletion Potential
PET	Polyethylene terephthalate
SCL	Solar Cooling Load Factor
SEER	Seasonal Energy Efficiency Ratio
SHG	Sensible Heat Gain
TFM	Transfer Function Method
TR	Ton of Refrigeration
VCC	Vapor Compression Cycle

1. Introduction. - Nowadays, global warming become one of the major issues. If temperature, pressure, and relative humidity in the ambient atmospheric conditions are observed, there is an uncomfortable situation for the people. There are many regions in the World where outside ambient conditions are too hot and humid. So, in this situation, people can't perform daily life activities and tasks due to their lower thermal comfort. Due to the environmental changes, the term "climate" comes into play [1]. Pakistan, the country in this World is now tolerating the summer season extending from April to November. At the global scale, it is significantly observed there was a greater number of hot days as compared to cold days over the past decade which is evident the frequency of hot and humid days is higher. Also, from the study conducted by the Pakistan Meteorological Department, a significant increase in heatwave days is observed which is now a major issue for Pakistan especially the occupants living in different cities due to thermal comfort zones [2].

Without people's thermal comfort, the occupants can't feel comfortable, and they can't perform daily activities. By considering the mentioned situation and conditions related to human comfort, especially in hot climatic conditions, scientists developed the concept of an HVAC system which is the most important requirement for people.

The concept of thermal comfort is a state of mind, the essential parameter for the occupant's comfort zone that provides satisfaction to the occupant so that one can perform his tasks in a comfortable environment [3-5]. The comfort zone for an occupant is predicted by relative humidity, air velocity, air & radiant temperatures, clothing insulation, and metabolic rate [6], it can be defined by a range of operative temperatures that will provide acceptable thermal conditions for a person's state of mind [7].

An HVAC system is mainly responsible for maintaining the desired IAQ by supplying adequate and acceptable fresh air [8, 9]. HVAC systems need to be much more efficient as it consumes around 60% of the building's total energy consumption [10]. In Pakistan, it is observed that the systems which are installed for human comfort are not designed on standard conditions. Either the system is overdesigned in that it produces too much cooling effect and is not economically feasible or the system is under designed so that the occupants are not thermally comfortable [11].
Heat gain is the heat generated by material or equipment in space. HVAC system performance depends on the heat generated by several pieces of equipment. Heat gain ultimately depends on several factors such as room orientation relative to solar radiation, electric devices or appliances, wall and roof materials, and the number of occupants present in a space [12].

Nowadays, due to increasing heat-generating sources and hot & humid climatic conditions, the need for an air conditioning system is a must. On a domestic level, air conditioning requirement is fulfilled by split air conditioners [13] but in large buildings, offices, or auditoriums, these are not feasible due to insufficient supply of air flowrate. So, to resolve this issue, the HVAC system plays an important role to supply adequate fresh air and maintain IAQ within the facility. Researchers and designers introduced some software to make energy-efficient systems. But the problem with this software is it doesn't completely fit the real-time data and the real-time data is something else that is different from the built-in values of the software. This is the reason why when researchers design a particular system that is only based on software and no manual method is used, there must be an error [14].

There are several ways to determine the cooling load of space. Some methods such as CLTD, CLF, SCL, and HAP (Transfer Function) are being used for cooling load calculation. Each of these methods possesses a different and unique methodological approach for calculating space cooling load. CLTD and HAP methods will be discussed in this paper [15].
CLTD is widely used for manual cooling load calculation, as proposed by ASHRAE. This method is used to calculate space cooling loads in which heat-dissipating devices are present. It is also used to calculate the load due to heat dissipation from the walls, windows, and roofs in the space [16]. For walls and roofs, CLTD uses the concept of heat transfer temperature difference but for internal load and windows, it uses CLF [17]. The tabulated CLTD and CLF data were calculated using the transfer function method, which yielded cooling loads for standard environmental conditions and zone types. The cooling loads for each component are then summed to obtain the total zone cooling load [18]. The cooling load of an auditorium space is determined using the CLTD/CLF method, developed as a manual calculation technique relying on tabulated CLTD and CLF values. These tabulated data were derived using the transfer function method, providing cooling load estimates for typical environmental conditions and zone types. These loads were subsequently standardized for easy hourly calculation by designers through normalization. Total zone cooling load is computed by summing the cooling loads of individual components.
HAP is a powerful computer-based tool developed by Carrier Corporation, designed for consulting engineers, HVAC contractors, facility engineers, and other professionals involved in the design and analysis of commercial building HVAC systems. The HAP uses the ASHRAE Transfer Function Method (TFM) for system load calculations and detailed 8,760 hour-by-hour simulation techniques for energy analysis. The TFM uses transfer functions to model transient heat transfer equations [19].

The two major functions of HAP are:

1. Estimating load and designing systems
2. Performing Energy and Cost analysis

The HAP software uses a specific built-in program called the "HAP System Design Load" program to calculate, design, and size the HVAC system. The following are some features of this program:

- ✓ Calculates design cooling and heating loads for spaces, zones, and coils in the HVAC system.
- ✓ Determines required airflow rates for spaces, zones, and the system.
- ✓ Sizes cooling and heating coils.
- ✓ Sizes air circulation fans.
- ✓ Sizes chillers and boilers.

Some previous case studies were performed by different researchers for cooling load calculation of particular facilities by different methods and there is a significant result variation after validation by researchers which leads to an inappropriate design of the HVAC system. ALameen awad Alameen et al. [20] performed a study related to the designing of an Air Conditioning System for a Sports Hall, with a capacity of 1000 occupants in Sudan by using CLTD and HAP methods. The results obtained from CLTD and HAP were 116 TR and 103 TR respectively, with a percentage error of 13%. Khakre et al. [21] conducted a study for cooling load calculation that incorporates the CLTD method for an evaporative cooling system. The cooling load obtained from the CLTD method was 42.35 TR and from the HAP program, it was 38.6 TR, contributing an error of 9% which is not allowed to design an accurate HVAC system. These

variations of calculated cooling load from two different methods make it impossible to design an accurate HVAC system.
Based on cooling load calculation, all previous research studies are focused on designing of HVAC system for any facility by using the CLTD method and HAP software. The purpose of this research study is to design and propose different possible HVAC systems for the University Auditorium. A Central Air Conditioning system is already installed in the Auditorium which is DX – type system. An HVAC system that is currently being operated has exceeded the lifespan of twenty years and Refrigerant R-22 (Chloro-Difluoro-Methane) is currently being used and has been obsolete due to high GWP and ODP values. In the existing HVAC system, R-22 enters AHU and cools the air without the need for secondary refrigerant but it's a danger zone for occupants if the refrigerant piping leaks for some reason, it will directly be mixed with cool air and goes into the space where potential health hazards for occupants will be taken place. So, R-22 (Freon) is not suitable for the cause rather it will be harmful to the occupants. To the knowledge of the authors, no study is available that proposed new HVAC systems with the refrigerant R-410 (A) for the University Auditorium. R-410 (A) is a zeotropic and its ODP is zero due to the absence of chlorine. Although it has a higher GWP than R-22 due to its higher SEER rating, and by reducing the power consumption of the system, it is overall more environmentally friendlier than R-22. The novelty of this research work is to fill this research gap. In this paper, the cooling load of the University Auditorium is calculated via the CLTD method and HAP software. The calculated cooling load is then used to select new equipment like Chillers for both proposed systems which are water-cooled and air-cooled. Other equipment like AHU, cooling towers, and pumps are also selected for both systems as per available data and calculations.

2. The University Auditorium. - The analysis involves the designing of an HVAC system for the University Auditorium located in Karachi. The Auditorium contains electrical equipment and devices that are dissipating continuous heat for which proper temperature control is necessary and IAQ must be maintained. The audience or working staff can't feel comfortable in such an environment when there is no proper control of inside dry bulb temperature and relative humidity due to occupants present at that particular time. DX – Type system has already been installed in the Auditorium and this system provides both air-conditioning as well as ventilation through a ducting network in which refrigerant directly enters AHU and cools the air without the need for any secondary refrigerant. This system has exceeded the lifespan of twenty years with the usage of R-22 which is now obsolete due to its high GWP and ODP values which are 1810 and 0.05 respectively. Montreal and Kyoto Protocols introduced HFCs for the replacement of chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) because of the low ODP of HCFCs [22]. However, the qualifying criteria for the selection of refrigerant is not only based on ODP but the potential alternative refrigerant has been selected based on a high energetic performance and GWP [23, 24]. Table I represents the specification of components installed in the HVAC system of the University Auditorium.

Components	**Specifications**
Compressor	Company: Carlyle Carrier *Type: Open-Drive Reciprocating* *Model: 5H66*
Condenser	Shell & Tube Type
Evaporator	DX - Type
Expansion device	Thermal Expansion Valve
Air Handling Unit	Size: 400 x 300 x 175 cm Coil Area: 300 x 170 cm
Cooling Tower	Induced draft Counter Flow
Refrigeration cycle	Vapor Compression
Refrigerant	R-22 (Chloro-Difluoro Methane)
Other capacities	Compressor: 75 hp AHU fan: 15 hp Cooling tower pump: 1.5 hp Condenser water pump: 9.8 hp

Table I. *Existing HVAC system of the University Auditorium*

The occupancy schedule is taken as 6 hours (from 0800 hours to peak time of 1400 hours). The geographical location of the Auditorium is as: Latitude is 24.9 °N, Longitude is 67.1 °E and Elevation is 51 meters. The orientation of all four walls of the Auditorium in a cardinal direction is shown in Figure I.

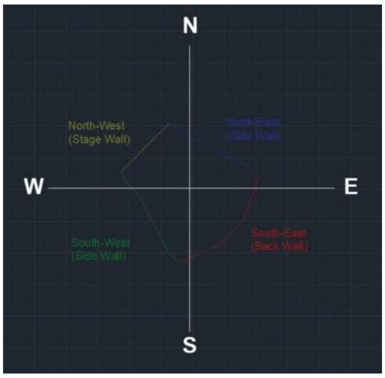


Figure I.- *Walls orientation of the University Auditorium*

The Auditorium consists of a Hall with roof and floor areas are 615.25 m^2 & 615.24 m^2 respectively. It consists of four walls South East (Back wall), South West (Sidewall), North East (Sidewall), and North West (Stage wall) with areas of 67.23 m^2, 168.84 m^2, 168.84 m^2 & 103.79 m^2 respectively. Although there is no window in the Auditorium, its cooling load won't be calculated. The outside weather data condition for the University Auditorium is obtained by keeping Jinnah International Airport as a reference [25]. The weather data shows an outside design temperature of 38.9 ℃ and an average humidity ratio of 19.8 g/kgda [25]. Figure II depicts how the existing HVAC system of the Auditorium works while Figure III shows the 3D model drawn on Autodesk Revit.

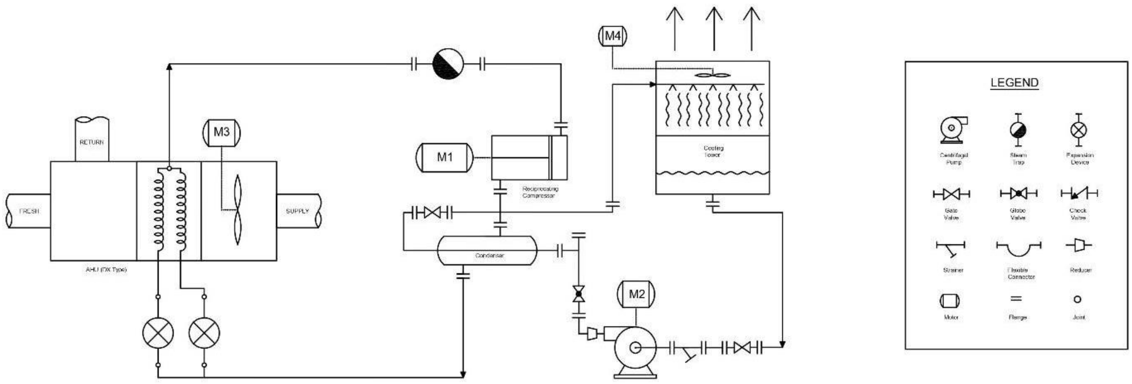
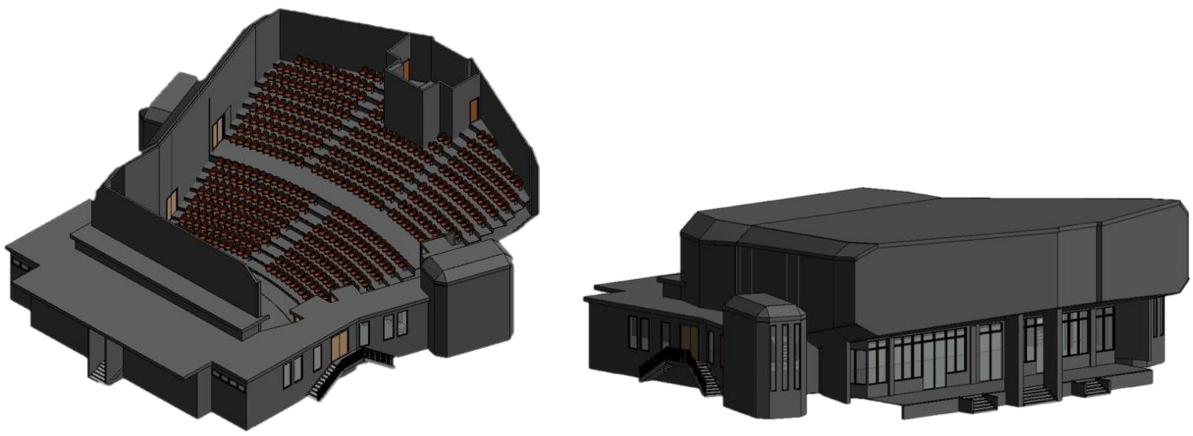


Figure II.- *Process flow diagram of the existing HVAC system of the University Auditorium*



(a) ***(b)***

Figure III.- *3D Revit model of the University Auditorium*

Figures IV, V, VI & VII represent the front view, back view, top view, and section view of the Auditorium.

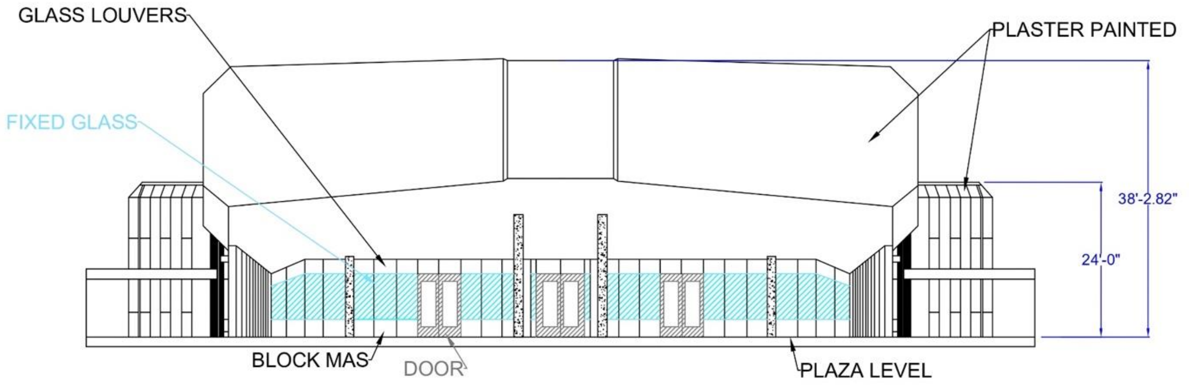


Figure IV.- *Front side view of the University Auditorium*

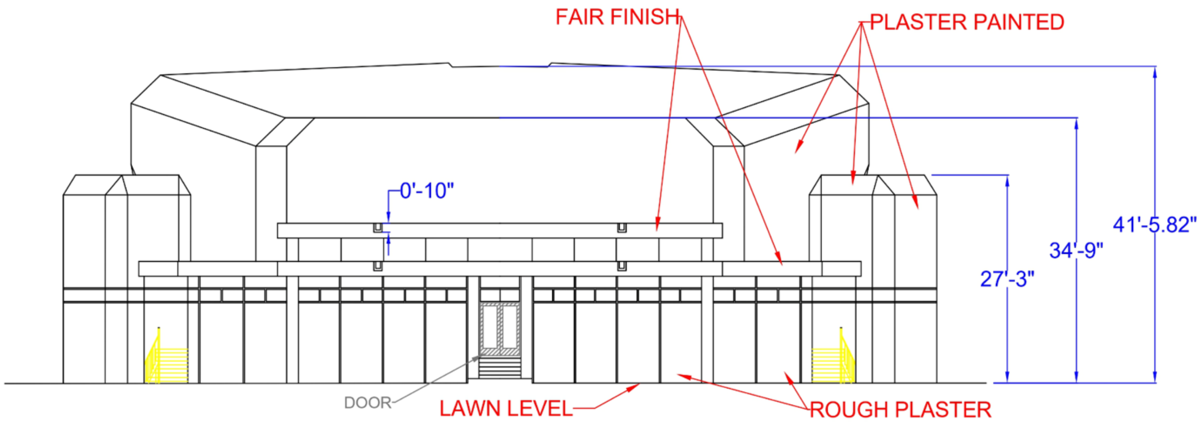


Figure V.- *Back view of the University Auditorium*

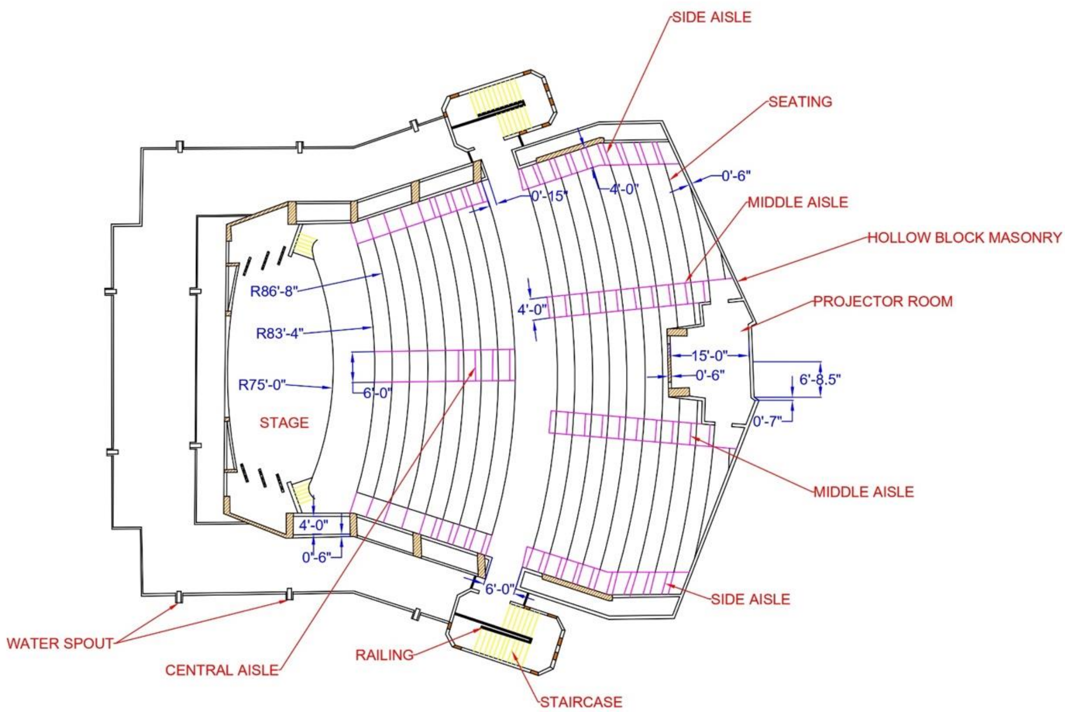


Figure VI.- *Top view of the University Auditorium*

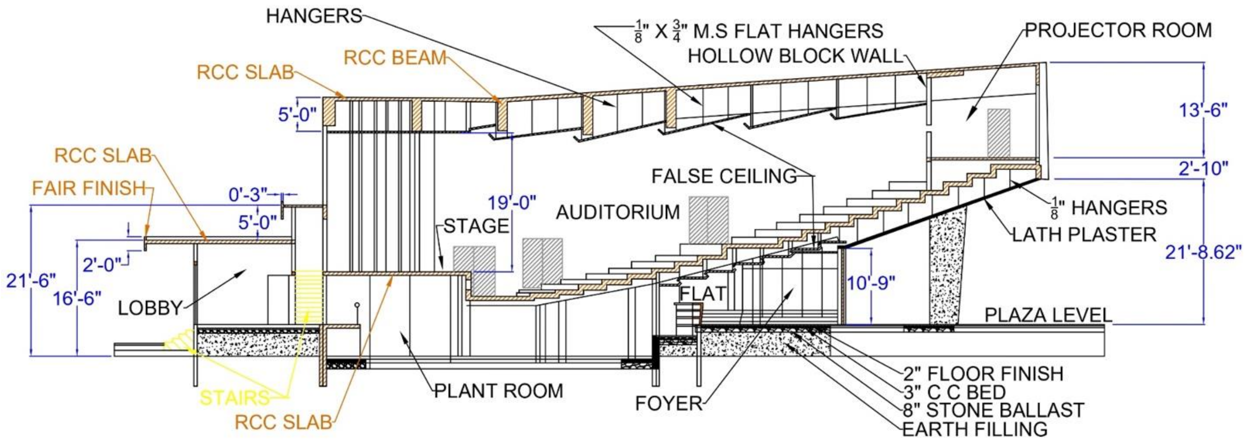


Figure VII.*- Section view of the University Auditorium*

3. Methodology. - This section consists of the calculations of the Roof, walls, and floor areas of the Auditorium manually. Inside weather conditions of the University Auditorium will be selected as per design conditions [26]. Figure VIII represents the methodological flow chart.

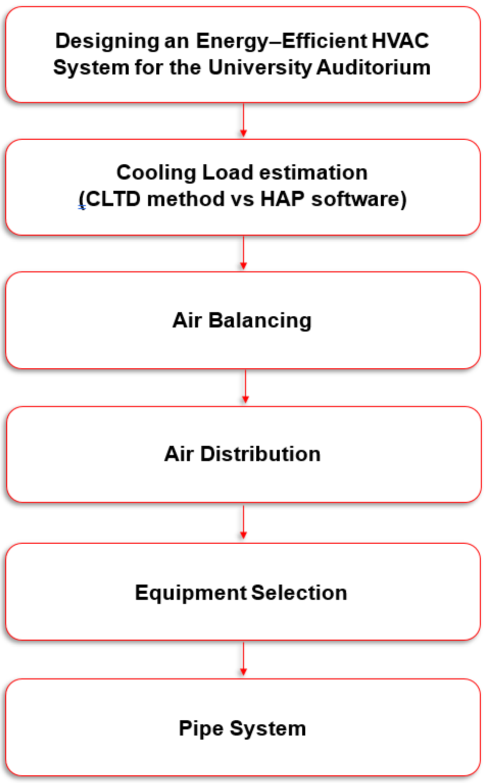


Figure VIII.*- Methodological approach*

3.1 Manual Cooling Load Estimation (CLTD Method). - Table II depicts the important design conditions for cooling load calculation.

Inside Condition	Symbols	Values	References
Dry bulb temperature	t_i	22 °C	[26]
Relative Humidity	$\emptyset$	60 %	[26]
Humidity Ratio	W_i	9.9 g/kgda	[26]

Outside Condition	Symbols	Values	References
Dry bulb temperature	t_o	38.9 °C	[25]
Average temperature	t_{avg}	36 °C	[25]
Humidity Ratio	W_o	19.8 g/kgda	[25]

Table II. *Inside and outside weather conditions*

Table III represents surface film coefficients/resistances [25]. Table IV represents the thermal resistances of Auditorium structural materials.

Table III. *Thermal Resistances of Surface Films*

Position of surface	The direction of heat flow	Thermal Resistance at Emissivity (90%) (m^2 k/W)	References
Indoor			
Vertical	Horizontal	0.12	[25]
Horizontal	Downward	0.16	[25]
Horizontal	Upward	0.11	[25]
Outdoor			
(any position for summer at 3.4 m/s)	Any	0.044	[25]

Material	Thermal Conductivity (W/ m.k)	Thickness of material (L)	Thermal Resistance (m^2 k/W)	Ref
The air between the roof and ceiling	26.24	60’’	0.058	[27]
The air between the side walls	26.24	54’’	0.052	[27]
Plaster	0.72	1’’	0.035	[25]
Concrete Slab	1.818	6’’	0.0825	[28]
Acoustic Tiles	0.052	5/8’’	0.288	[25]
Hardwood	0.18	1’’	0.141	[25]
Hollow Block	0.88	6’’	0.172	[29]

Table IV. *Thermal Resistances of Structure Materials*

3.1.1 Roof. - The cooling load from the roof can be calculated by eq (1) [28].

$$\dot{Q} = U\ A\ CLTD_{adj} \tag{1}$$

U-value for the surface of the roof can be calculated by eq (2). Figure IX shows the thermal network diagram for the construction material of the roof.

$$U = \frac{1}{\sum Resistance} \tag{2}$$

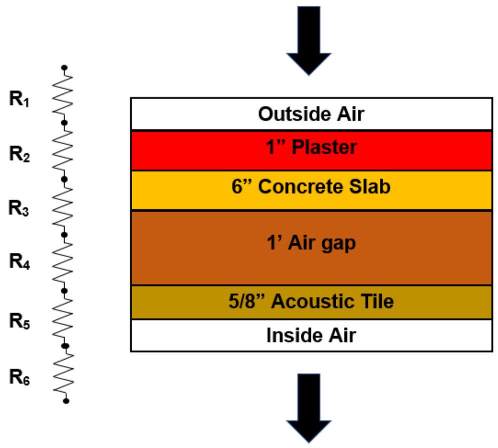


Figure IX.- *Thermal network diagram for the roof's construction*

Table V depicts the thermal resistance values of the material used in the construction of the Roof.

Resistance	Description	Value (m^2 k/W)	References
$\boldsymbol{R_1}$	Outside Air Resistance	0.044	[25]
$\boldsymbol{R_2}$	Plaster	0.035	[25]
$\boldsymbol{R_3}$	Concrete Slab	0.083	[28]
$\boldsymbol{R_4}$	Air Gap	0.058	[27]
$\boldsymbol{R_5}$	Acoustic tile	0.288	[25]
$\boldsymbol{R_6}$	Inside Air Resistance	0.160	[25]

Table V. *Thermal Resistances for Roof Material*

The total area of the roof is 615.25 m^2. Solar time of 1400 hours with Type-4 roof with suspended ceiling is selected. The CLTD value obtained is 16 °C [28]. $CLTD_{adj}$ value is determined from eq (3) [28].

$$CLTD_{adj} = CLTD + (25 - t_i) + (t_{avg} - 29) \qquad (3)$$

3.1.2 Walls. - The cooling load from the wall can be calculated from eq (1) [28]. U-value can be calculated for the surface of the wall by eq (2). Figures X, XI, XII, and XIII illustrate thermal network diagrams for walls oriented in the southeast, southwest, northeast, and northwest directions, respectively.

3.1.2.1 South East (Back Wall). - Table VI represents the thermal resistance values of the material used in the construction of South East (Back Wall).

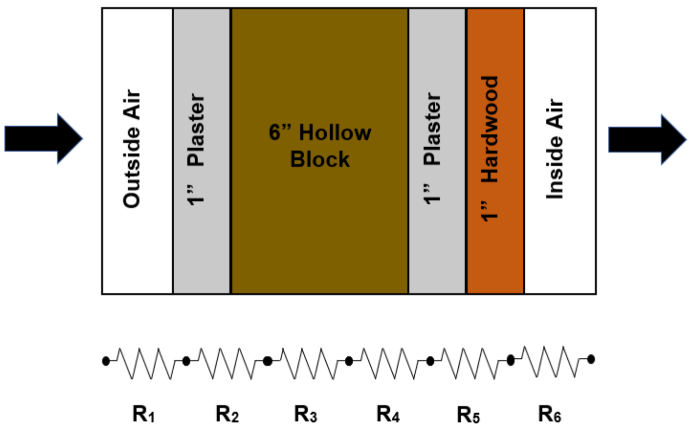


Figure X.- *Thermal network diagram for South East wall construction*

Resistance	Description	Value (m^2 k/W)	References

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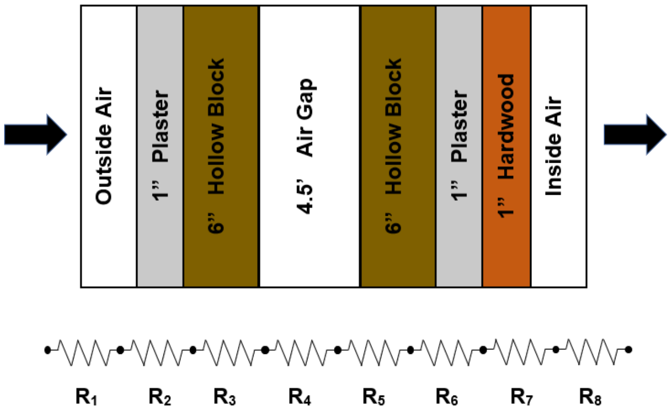
R_1	Outside Air Resistance	0.044	[25]
R_2	Plaster	0.035	[25]
R_3	Hollow block	0.172	[29]
R_4	Plaster	0.035	[25]
R_5	Hardwood	0.141	[25]
R_6	Inside Air Resistance	0.120	[25]

Table VI. *Thermal Resistances for South East material*

3.1.2.2 South West (Side Wall). - Table VII represents the thermal resistance values of the material used in the construction of South West (Side Wall).



Figur *on*

Resistance			References
R_1	Outside Air Resistance	0.044	[25]
R_2	Plaster	0.035	[25]
R_3	Hollow block	0.172	[29]
R_4	Air gap	0.052	[27]
R_5	Hollow block	0.172	[29]
R_6	Plaster	0.035	[25]
R_7	Hardwood	0.141	[25]
R_8	Inside Air Resistance	0.120	[25]

Table VII. *Thermal Resistances for South West (Side Wall) material*

3.1.2.3 North East (Side Wall). - Table VIII represents the thermal resistance values of the material used in the construction of North East (Side Wall).

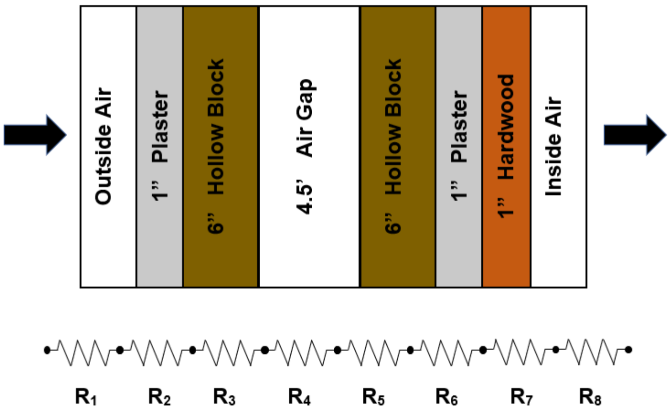


Figure XII.*- Thermal network diagram for North East wall construction*

Resistance	Description	Value (m^2 k/W)	References

$\boldsymbol{R_1}$	Outside Air Resistance	0.044	[25]
$\boldsymbol{R_2}$	Plaster	0.035	[25]
$\boldsymbol{R_3}$	Hollow block	0.172	[29]
$\boldsymbol{R_4}$	Air gap	0.052	[27]
$\boldsymbol{R_5}$	Hollow block	0.172	[29]
$\boldsymbol{R_6}$	Plaster	0.035	[25]
$\boldsymbol{R_7}$	Hardwood	0.141	[25]
$\boldsymbol{R_8}$	Inside Air Resistance	0.120	[25]

Table VIII. *Thermal Resistances for North East (Side Wall) material*

3.1.2.4 North West (Stage Wall). - Table IX represents the thermal resistance values of the material used in the construction of North West (Stage Wall).

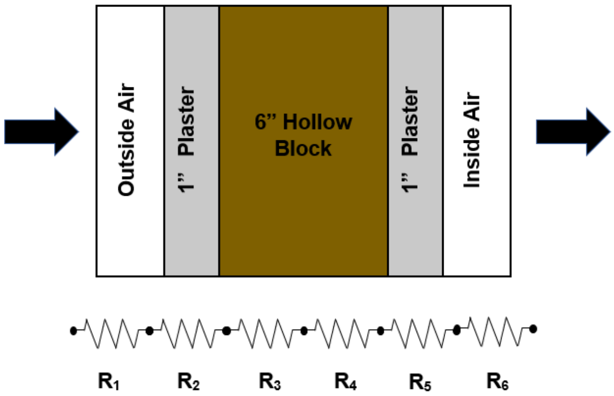


Figure XIII.- *Thermal network diagram for North West wall construction*

Resistance	Description	Value (m^2 k/W)	References
$\boldsymbol{R_1}$	Outside Air Resistance	0.044	[25]
$\boldsymbol{R_2}$	Plaster	0.035	[25]
$\boldsymbol{R_3}$	Hollow block	0.172	[29]
$\boldsymbol{R_4}$	Plaster	0.035	[25]
$\boldsymbol{R_5}$	Inside Air Resistance	0.120	[25]

Table IX. *Thermal Resistances for North West (Stage Wall) Material*

Solar time of 1400 hours with Wall Type - E is selected. $CLTD_{adj}$ value is determined from eq (3). Table X depicts the CLTD value of different walls along with their surface areas.

Wall orientations	CLTD (°C) [28]	Wall Area (m^2)
South East	20	67.23
South West	10	168.84
North East	14	168.84
North West	7	103.79

Table X. *Wall Areas & CLTD values*

3.1.3 Floor. - The cooling load from the floor can be calculated by eq (4) [28]. U-value is calculated for floor surface by eq (2). t_a is the temperature below the floor of University Auditorium, taken as 30 °C. The floor area is 615.24 m^2. Table XI represents the thermal resistance values of the material used in the construction of the Floor. Figure XIV represents the thermal network diagram for the construction material of the floor.

$$\dot{Q} = U\ A\ (t_a - t_i) \qquad (4)$$

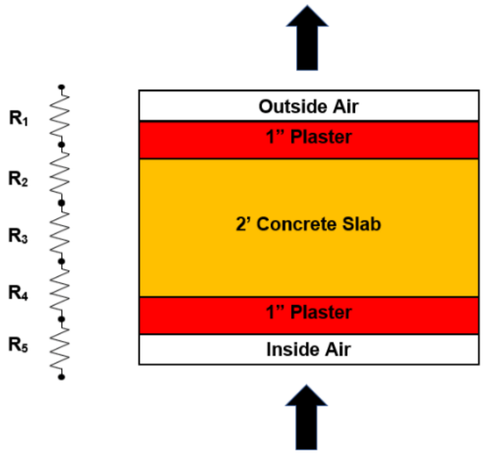


Figure XIV.- *Thermal network diagram for Floor Construction*

Resistance	Description	Value (m^2 k/W)	References
R_1	Inside Air Resistance	0.11	[25]
R_2	Plaster	0.035	[25]
R_3	Concrete Slab	0.335	[25]
R_4	Plaster	0.035	[25]
R_5	Outside Air Resistance	0.11	[25]

Table XI. *Thermal Resistances for Floor Material*

3.1.4 Occupants. - The sum of latent and sensible heat is the total heat generated by the people present within the space. Sensible Heat Gain (SHG) is the heat gained via conduction, convection, or radiation and can be calculated by eq (5) [28]. If the water molecules are incorporated, then the heat gain in this condition is Latent Heat Gain (LHG) and can be calculated by eq (6) [28].

$$\dot{Q}_{sensible} = No. of\ people\ (SHG)\ CLF \qquad (5)$$

$$\dot{Q}_{latent} = No. of\ people\ (LHG) \qquad (6)$$

The values of SHG & LHG are taken as per the desired condition whereas the CLF is selected for 8 hours in space (0800 hours to 1600 hours) and entry time is taken as 6 hours (from 0800 hours to peak time of 1400 hours). Table XII depicts the values of SHG & LHG by seated and unseated persons present within the Auditorium.

No. of people who are seated = 550

No. of people who are unseated = 25

Load	Seated (W / person)	Unseated	References
Sensible load	60	75	[28]
Latent load	40	75	[28]

Table XII. *Sensible & Latent heat gain by a person*

3.1.5 Lights. - The lighting load within the space can be calculated by eq (7) [28].

$$\dot{Q} = W.F_u.F_b.CLF \qquad (7)$$

Light is assumed to be operating for 10 hours and the duration of light between turning on and peak time is taken as 6 hours (from 0800 hours to peak time of 1400 hours). Table XIII represents the wattage calculation of lights installed in the Auditorium while Table XIV depicts the parameters of lighting load calculation.

Type of light	Quantity	Wattage per light	Total Wattage
Energy saver	25	25	625
Recessed light	16	60	960
		Total	1590

Table XIII. *Wattage of lights*

Parameters	Symbol	Value	Reference
Utilization factor	F_u	1	[28]
Ballast factor	F_b	1.2	[28]
Cooling Load factor	CLF	0.78	[28]
Wattage	W	1590	--

Table XIV. *Lighting load parameters*

3.1.6 Infiltration. - Infiltration is the uncontrolled introduction of outside air into a building and is represented by $\dot{V}_i$. The ACH (Air Changes per Hour) method is used to determine the ventilation requirement of the Auditorium. ACH value is taken for medium-construction buildings at an outside temperature condition of 38℃ [30]. V is the volume of the University Auditorium calculated from the drawing.

$$\dot{V}_i = \frac{V\,(ACH)}{3600} \qquad (8)$$

$\dot{Q}_{is}$ is the sensible heat gain while $\dot{Q}_{il}$ is the latent heat gain due to infiltration. These values can be calculated by eq (9) and (10) [28].

$$\dot{Q}_{is} = 1.23\,\dot{V}_i(t_o - t_i) \qquad (9)$$

$$\dot{Q}_{il} = 3000\,\dot{V}_i(W_o - W_i) \qquad (10)$$

The volume of the Auditorium (m^3)	ACH [30]
2876.5	0.52

Table XV. *Air Change Method Parameters*

3.1.7 Ventilation. - Ventilation air comprises a combination of fresh and recirculated air [28] and plays a crucial role in ensuring IAQ reaches acceptable levels, meaning the air contains no harmful concentrations of known contaminants [31]. This system functions by delivering fresh air to indoor areas while concurrently eliminating stagnant air [32]. Fresh outdoor air enters the building through ventilation ducts, where it undergoes filtration and conditioning via a cooling coil to meet comfort and health requirements. The conditioned air is then dispersed throughout the building via ductwork and strategically placed vents. Stale air, containing pollutants and excess moisture, is extracted from the building through exhaust vents, usually directed outside. Some of the air may be recirculated during the exhaust phase to mix with fresh air before being reintroduced into the space, completing the cycle. This entire process is depicted in Figure XV.

$\dot{V}_v$ is the supply air rate for ventilation purposes which can be calculated by eq (11) [28], $\dot{V}_r$ is the recirculation air rate can be calculated by eq (12) [28] and $\dot{V}_m$ is the minimum outdoor air rate.

$$\dot{V}_v = \dot{V}_r + \dot{V}_m \qquad (11)$$

$$\dot{V}_r = \frac{\dot{V}_o - \dot{V}_m}{E} \qquad (12)$$

$\dot{V}_o$ is the outdoor air rate and E is the efficiency of contaminant removal by cleaning device. $\dot{Q}_{vs}$ is the sensible heat gain while $\dot{Q}_{vl}$ is the latent heat gain due to ventilation. These values can be calculated by eq (13) and (14) [28].

$$\dot{Q}_{vs} = 1.23\,\dot{V}_v(t_o - t_i) \qquad (13)$$

$$\dot{Q}_{vl} = 3000\,\dot{V}_v(W_o - W_i) \qquad (14)$$

Parameters	Values	Reference
V_o	2.7	[33]
V_m	2.5	[33]

Table XVI. *Ventilation Load Parameters*

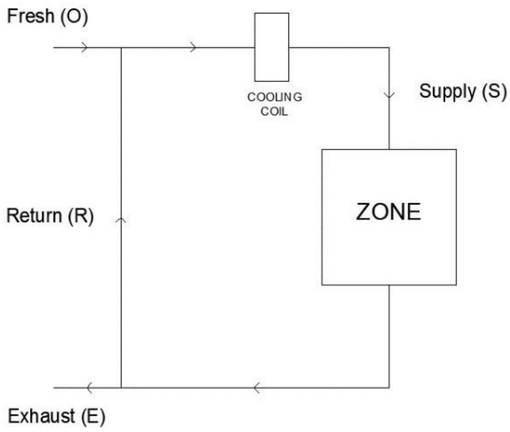


Figure XV.- *Airflow schematic diagram*

3.2 Cooling Load estimation using HAP (Transfer Function method)

The software used for the verification of cooling load calculation is Carrier's HAP version 4.9 as it has a user-friendly interface, uses the ASHRAE transfer function method for load calculations, and is the fastest way to get a solution and results rather than doing a manual calculation technique.

The weather properties such as design temperatures and humidity conditions are by default fed in HAP. The only selection will be of the desired location [34].

The distribution of the desired building into multiple units or sections for thermal comfort is known as Space in which further parameters like external cooling loads, internal cooling loads, infiltration, and ventilation loads are inserted [16].

As far as the schedule is concerned, it's the time and concern day for any activity that is to be performed in any section. There are some schedules made for lighting purposes, occupants, and other required conditions. Light is assumed to be operating for 10 hours and the duration of light between turning on and peak time is taken as 6 hours (from 0800 hours to peak time of 1400 hours).

For walls, its construction and structural details for the University Auditorium have been inserted and the R-value for each material to be used gives out the overall U-value of the wall. Similarly, for the roof, its construction and structural details have also been inserted, and the R-value for each material to be used gives out the overall U-value of the roof [35].

The transfer function method (TFM) is especially suitable for computer applications. It provides a straightforward calculation of the heat gain through a wall or roof, given by the eq (16) [36]:

$$q_{e,\theta} = A\left[\sum_{n=0} b_n\left(t_{e,\theta-n\delta}\right) - \sum_{n=0}\frac{d_n\left(q_{e,\theta-n\delta}\right)}{A} - t_{r,c}\sum_{n=0} c_n\right] \qquad (15)$$

3.3 Air Balancing

3.3.1 Fresh / Exhaust air flowrate (Ventilation). - Flow rates of fresh / exhaust air can be determined by eq (15) [33]. Table XVII represents ventilation rates and related parameters.

$$Outdoor\ Air = R_p\ (N) + R_A\ (Zone\ Area) \qquad (16)$$

Parameters	Symbol	Values	Units	Reference
People's outdoor air rate	R_P	5	cfm / person	[33]
Area outdoor air rate	R_A	0.06	cfm / ft^2	[33]
No of occupants	N	575	No.	--
Zone area (Area of Auditorium)	--	6619	m^2	--

Table XVII. *Ventilation rates in the breathing zone*

3.3.1.1 Condition of air at different stages. - Concerning the Figure XV, the condition of air at different stages can be categorized as:

- Outside Air - fresh air coming from outside.
- Mixed Air - this state is achieved when outside air is mixed with return air from the space.
- Supply Air - this state is achieved after mixed air passes through the cooling coil.
- Zone Air - this is the state of air inside the conditioned zone.

The supply air flow rate has been taken from HAP software. Eq (17) & (18) are used to find properties at the mixing state. Table XVIII represents the condition of air at different stages.

$$m_M = m_O + m_R \qquad (17)$$

$$m_M h_M = m_O h_O + m_R h_R \qquad (18)$$

Parameters	Symbol	Values	Units
Total cooling load	Q_T	202	kW
Sensible load	Q_S	119	kW
Latent load	Q_L	82	kW
Fresh air rate	V_O	1.54	m^3/s
Supply air rate	V_S	7.59	m^3/s

Table XVIII. *Different Parameters for different stages of Air*

3.3.2 Supply & Return Air Flowrate (Heat Removal Method). - The supply & return air flow rate is calculated by eq (19) [28] and eq (20) respectively. Here T_M and T_S are the temperatures at mixed and supply air condition respectively.

$$Supply\ air\ flowrate = \frac{Q_{sensible}}{1.08\ (T_M - T_S)} \qquad (19)$$

$$Return\ air\ flowrate = Supply\ air\ flowrate - Fresh\ air\ flowrate \qquad (20)$$

3.4 Air Distribution

3.4.1 Diffuser / Return Grill Selection & Duct Routing. - Diffusers are selected based on flow requirements and the noise criteria permissible for the room [37]. To select a diffuser and return air grill, the flow rate of discharge and intake air respectively is required. The diffusers and return air grills were selected as per the catalog for price diffusers [38] and price louvered grilles [39] respectively. Return and Supply air diffusers are divided in such a way that they provide air to an equal number of outlets. Figure XVI depicts the placement of diffusers and routing of duct from AHU to Space.

__Figure XVI.__- Duct Routing

3.4.2 Duct Losses. - Duct pressure drop is calculated by using the Equal Friction method. The value of pressure drop per unit length must be determined from the friction chart [36] and kept constant among all segments. Major loss & minor loss in pipe segment is calculated by eq (21) & (22) respectively.

$$\Delta P = \left(\frac{\Delta P}{l}\right)_{seg} . l_{seg} \tag{21}$$

$$\Delta P = (sum\ of\ joint\ losses).\frac{1}{2}\ \rho v_{seg}^2 \tag{22}$$

Table XIX shows the loss coefficient values of some common joints [36].

Joint	Loss Coefficient
$\boldsymbol{K_{entrance}}$	0.82
$\boldsymbol{K_{tee-through}}$	0.04
$\boldsymbol{K_{tee-branch}}$	0.73
$\boldsymbol{K_{90^\circ}}$	1.27
$\boldsymbol{K_{45^\circ}}$	0.37
$\boldsymbol{K_{wye-branch}}$	0.52
$\boldsymbol{K_{wye-through}}$	0.05
$\boldsymbol{Pressure\ drop\ across\ supply\ diffuser}$	16 Pascals
$\boldsymbol{Pressure\ drop\ across\ return\ grills}$	17 Pascals

__Table XIX.__ Loss coefficients

Table XX represents the flow, dimensions, and head losses for each segment in the index run of the Supply air duct.

Major					Minor (K_L)						
					No. of Fittings						
Segment	$\frac{\Delta P}{l}$ (Pa/m)	Flow rate (L/s)	Velocity (m/s)	Length (m)	Entrance	Wye - through	Wye - branch	Tee - through	Tee - branch	K90	K45
A	0.15	8307	5.0	4.93	1	--	--	--	--	--	--
B	0.15	4153	4.2	13.47	--	1	--	--	--	1	--
B1	0.15	4153	4.2	10.01	--	--	--	--	--	1	--
D	0.15	4153	4.2	15.24	--	--	--	--	--	2	--
G	0.15	2076	3.6	6.83	--	1	--	--	--	--	--
I	0.15	1038	3.0	4.85	--	1	1	--	--	1	--
I1	0.15	519.2	2.5	0.30	--	--	--	--	--	--	--

***Table XX.** Pressure losses in different Supply air duct segments of the Index run*

Table XXI represents the flow, dimensions, and head losses for each segment in the index run of the Return air duct.

Major					Minor (K_L)						
					No. of Fittings						
Segment	$\frac{\Delta P}{l}$ (Pa/m)	Flow rate (L/s)	Velocity (m/s)	Length (m)	Entrance	Wye - through	Wye - branch	Tee - through	Tee - branch	K90	K45
A	0.08	6570	3.8	0.58	1	--	--	--	--	--	--
C	0.08	3285	3.3	5.08	--	--	--	--	--	1	--
G	0.08	2190	2.8	7.85	--	--	--	--	1	1	--
G1	0.08	2190	2.8	11.64	--	--	--	--	--	1	--
K	0.08	1095	2.4	1.04	--	--	--	--	--	1	--
K1	0.08	1095	2.4	1.65	--	--	--	--	--	1	--
K2	0.08	1095	2.4	10.98	--	--	--	--	--	1	--
K3	0.08	1095	2.4	0.66	--	--	--	--	--	1	--
K4	0.08	1095	2.4	1.65	--	--	--	--	--	1	--
K5	0.08	1095	2.4	0.66	--	--	--	--	--	1	--

***Table XXI.** Pressure losses in different Return air duct segments of the Index run*

3.4.4 Duct Sizing. - A rectangular duct for our design (except at the discharge where the cross-section will be circular) is selected, equivalent diameter is calculated by eq (23) [28].

$$D_h = \frac{1.3\,(ab)^{0.625}}{(a+b)^{0.25}} \qquad (23)$$

3.5 Equipment Selection. - We are proposing two different HVAC systems for the University Auditorium which are based on water cooled chiller and an air-cooled chiller.

3.5.1 Proposed System # 1. - This system primarily relies on a water-cooled chiller. Its main components consist of a compressor, condenser, expansion valve, and evaporator. Chilled water from the evaporator is directed to the AHU via two chilled water pumps (one operational and the other on standby). The AHU comprises various elements, including filters, a supply air duct, a fresh air duct, a return air duct, an air mixing chamber, and a cooling coil. Chilled water flows through the coil while air passes over it, becoming cooled in the process. The cooled air is then distributed into the space. Another essential part is the condenser water circuit. Hot water from the condenser is conveyed to the cooling tower through two condenser water pumps (one operational and the other on standby). After heat rejection, the water returns to the condenser, completing the cycle. The term "water-cooled" refers to the condenser water being cooled by the cooling tower. The entire system is illustrated in Figure XVII for reference.

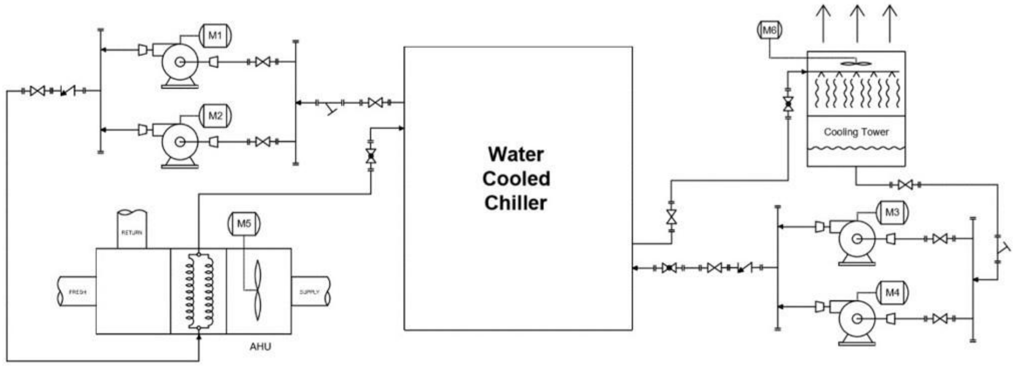


Figure XVII.- *Proposed System # 1*

3.5.2 Proposed System # 2. - This system operates on an air-cooled chiller. Its main components are identical to those of a water-cooled chiller, except for the absence of a cooling tower. Instead, an air-cooled condenser is used, eliminating the need for a cooling tower as the condenser water is cooled naturally by air. This system configuration is depicted in Figure XVIII.

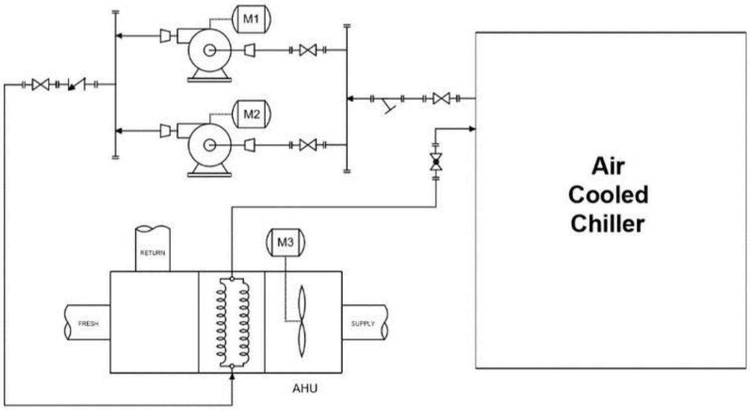


Figure XVIII.- *Proposed System # 2*

3.5.3 Chiller & AHU selection. - Typically, the chiller supplies chilled water at around 7 °C which makes the return water temperature 12 °C. The chilled water flow rate will be taken from water cooled chiller’s datasheet. The supply, return, and fresh air flow rates have been taken from the Air Balancing section. The fan selection performed in the AHU is based on the external and internal pressure drops. Internal pressure drop is due to the losses inside AHU while external pressure drop is the sum of pressure drop in the damper, ducting, and diffusers.

3.5.4 Cooling Tower & Pump selection. - The condenser water flow rate will be taken from the chiller’s datasheet. Cooling tower inlet and outlet water temperatures depend upon the selection of chiller because the condenser’s inlet and outlet water conditions vary with different chillers. For wet bulb temperature, outside air data will be used. We will select two pumps each for the condenser and chilled water circuits. One pump is working while the other is for backup. The cooling tower’s height, condenser & evaporator losses (from the chiller datasheet), pipes, fittings, and valve losses will be considered for the head. The flow rate will be taken from the selected water-cooled chiller datasheet.

3.6 Piping System

3.6.1 Water Cooled System. - For the application of HVAC, steel pipes are commonly used. We are selecting steel pipe of Schedule 40 for condenser and chilled water piping. For pipe sizing, we are using Carrier’s friction charts [36] for Schedule 40 Pipe. Table XXII depicts the Pipe design parameters for the cool and chilled water circuit.

Parameters	Condenser water	Chilled water	Unit	Ref
Velocity	5	6	Ft/sec	[40]
Flowrate	205	156	US gpm	[40]
Pipe size	4	3.5	inch	[40]
Friction loss	3.6	2.5	Ft of water per 100 ft	[40]

Table XXII. *Pipe design parameters for Cool and chilled water system*

The total piping length in the condenser and chilled water system is 161 ft & 79 ft respectively. Head losses due to valves and fittings for condenser and chilled water lines are taken [40] for 4” & 3.5” pipe diameters respectively. Table XXIII shows the equivalent lengths of the head, no. of valves, and fittings in the condenser and chilled water line.

Designing Criteria		Condenser water line			Chilled water line		
		L/D	Quantity	L_{eq} (ft)	L/D	Qty	L_{eq} (ft)
Fittings	Elbow 90°	10	20	66.7	9	16	48
	Tee (flow thru branch)	21	4	28	18	4	24
Valves	Gate	4.5	4	6	4	3	4
	Globe	120	2	80	100	1	33.3
	60° Strainer	60	1	20	48	1	16
	Swing Check	40	1	13.3	35	1	11.7
			Total	214			137
			Total equivalent length	375			216

Table XXIII. *Pipe fittings, valves & equivalent length in condenser & chilled water line*

Head loss due to piping is given by eq (24) [40].

$$h_L = f\frac{L}{D}\frac{v^2}{2g} \tag{24}$$

The pressure loss in the condenser and evaporator is calculated by eq (25) [40].

$$h_L = \frac{P}{\rho g} \tag{25}$$

3.6.2 Air Cooled System. - Steel pipe of Schedule 40 for Chilled Water Piping is selected. For pipe sizing, we select Schedule 40 Pipe [40]. Table XXIV depicts the Pipe design parameters for the chilled water circuit.

Parameters	Chilled water	Unit	Reference
Velocity	6	Ft/sec	[40]
Flowrate	168	US gpm	[40]
Pipe size	3.5	inch	[40]
Friction loss	3.6	Ft of water per 100 ft	[40]

Table XXIV. *Pipe design parameters for chilled water system*

The total piping length in a chilled water system is 112 ft. Head losses due to valves and fittings for chilled water lines are taken [40] for a 3.5” pipe diameter. Table XXV depicts the equivalent lengths of the head, no. of valves, and fittings in the chilled water line.

Designing Criteria		Chilled water line		
		L/D	Quantity	L_{eq} (ft)
Fittings	Elbow 90°	9	20	60

	Tee (flow thru branch)	18	4	24
Valves	Gate	4	3	4
	Globe	100	1	33.3
	60° Strainer	48	1	16
	Swing Check	35	1	11.7
			Total	149
			Total equivalent length	261

Table XXV. *Pipe fittings, valves & equivalent length in a chilled water line*

3.6.3 Duct & Piping Layout. - Figure XIX & XX shows the ducting and piping layouts for the two proposed HVAC systems for the Auditorium while Figure XXI depicts a ducting layout of the plant room for both the proposed HVAC systems.

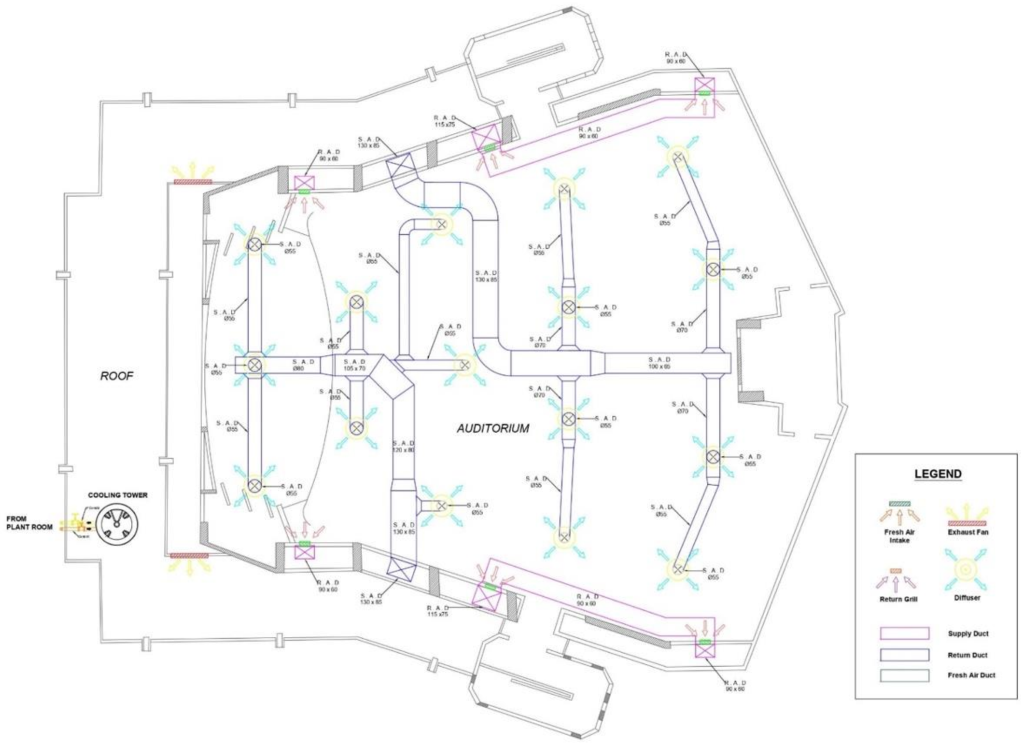


Figure XIX.- *Duct & Pipe Layout for Proposed System # 1*

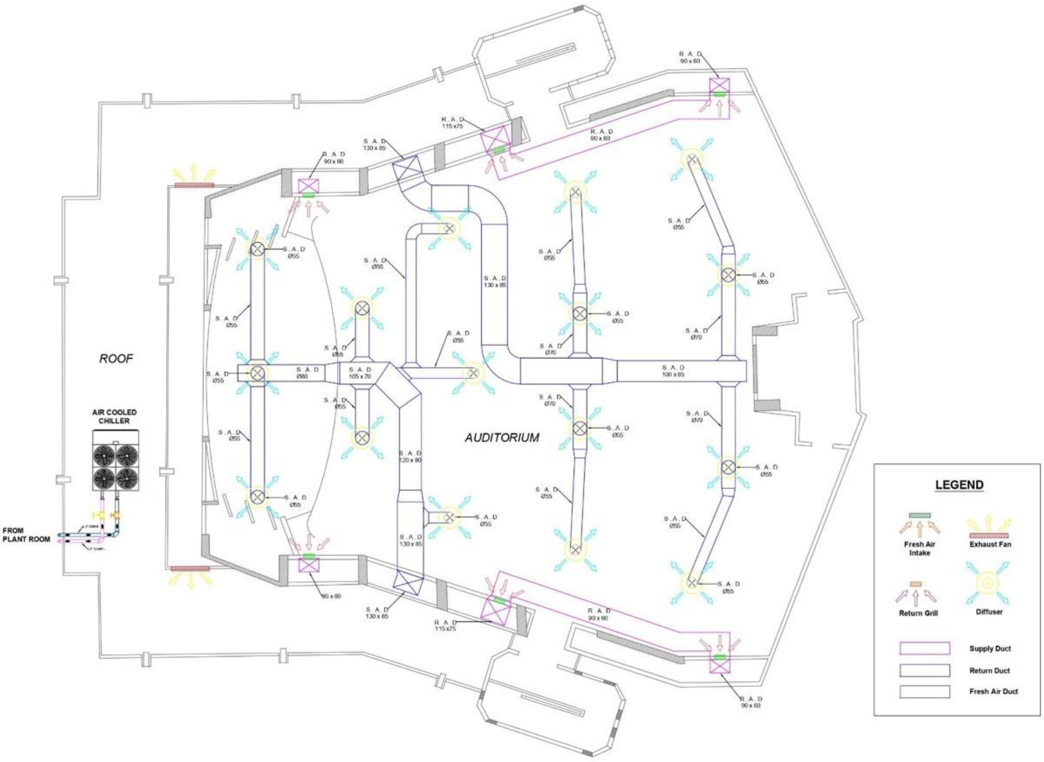


Figure XX.- *Duct & Pipe Layout for Proposed System # 2*

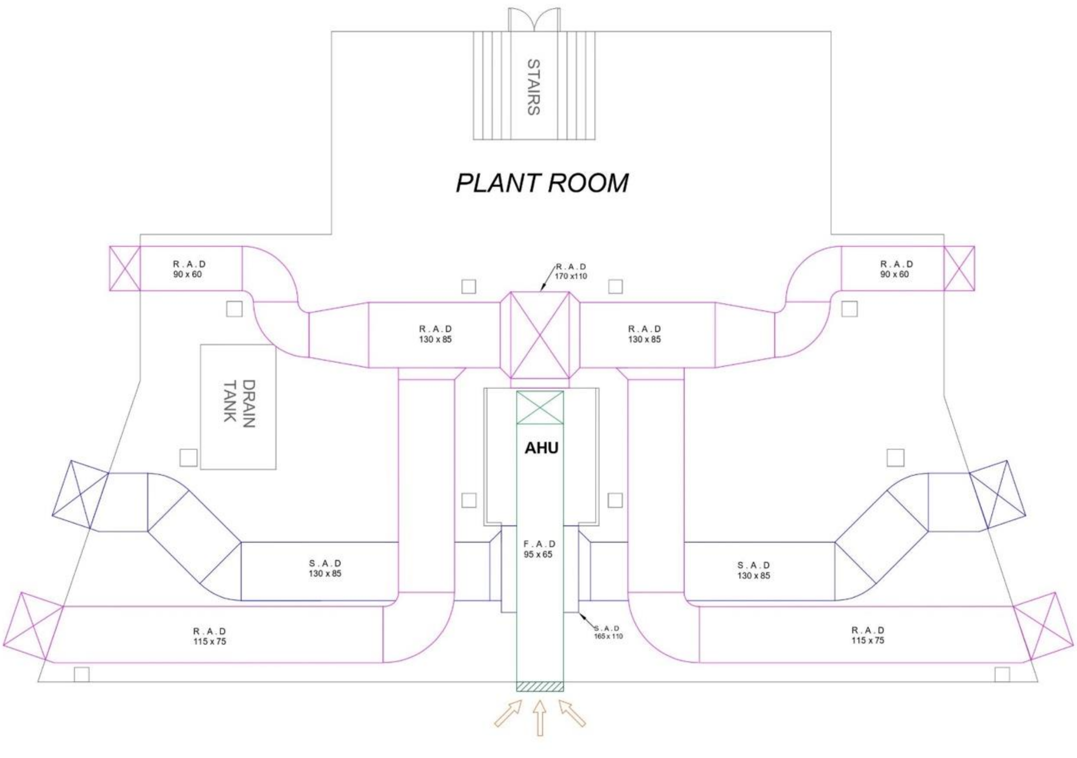


Figure XXI.- *Duct Layout of Plant Room for Proposed Systems # 1 & 2*

4. Results. - In this study, we calculated the cooling load of the University Auditorium first by manual calculation using the CLTD method then the results obtained were validated by using CARRIER's HAP (version 4.9) software.

4.1 Cooling Load by CLTD Results. - The results of the cooling load obtained after calculation by using CLTD methods are listed in Table XXVI. All the cooling loads are taken in kW.

External Cooling Load		**Roof**	23.96
	Walls	**South East (Back wall)**	3.7
		South West (Sidewall)	4.4
		North East (Sidewall)	5.28
		North West (Stage wall)	4.37
		Floor	7.87
Internal Cooling Load		**Occupant**	51.78
		Lights	1.48
		Infiltration Load	20.98
		Ventilation Load	78.38
		Total Cooling Load of the Auditorium	**202**

Table XXVI. *Cooling Load (CLTD method)*

The cooling load obtained from the CLTD-based calculation was 202 kW which is equivalent to around 57 TR. Figure XXII shows the condition of air at different stages on a psychrometric chart calculated by the CLTD method.

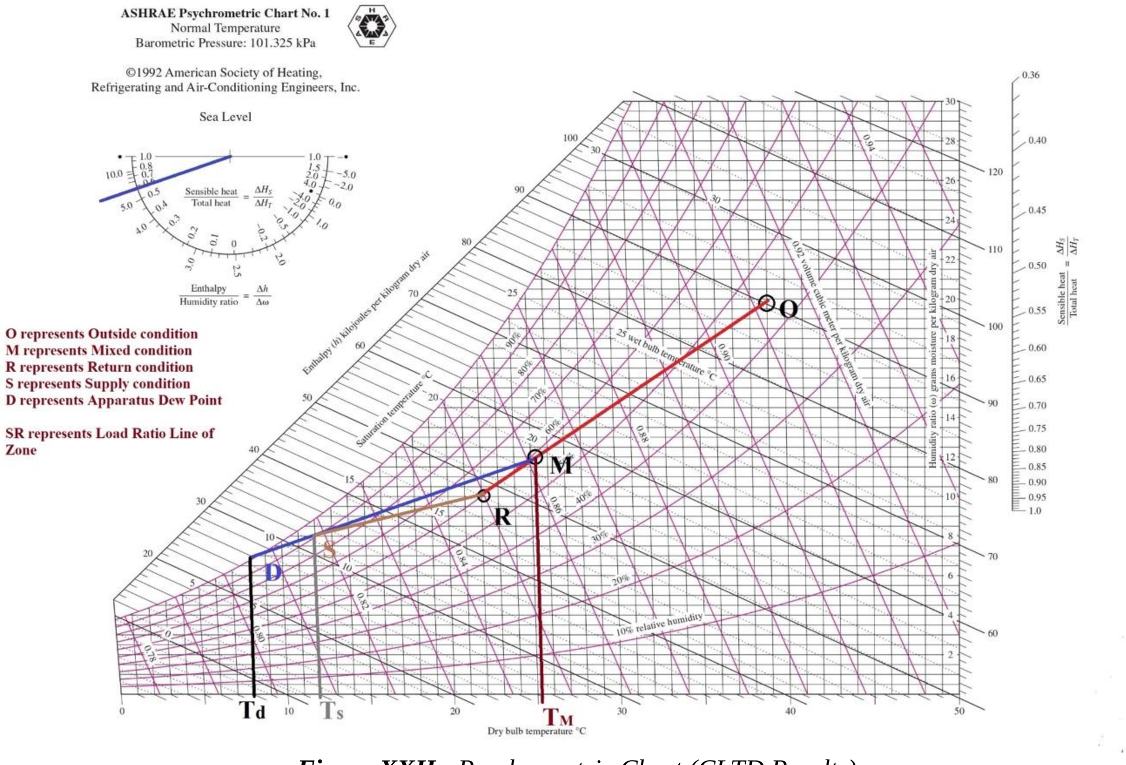


Figure XXII.- *Psychrometric Chart (CLTD Results)*

4.2 Cooling Load by HAP Results

- The HAP result depicts the cooling load of the University Auditorium as 192.8 kW which is equivalent to around 55 TR.
- The calculated ventilation (fresh air) rate for desired air conditioning in the air balancing section is 3273 cfm which is close to the value determined from the HAP result which is 1622 L/s (3436 cfm).
- The calculated supply air flow rate from the Heat Removal method is 16071 cfm which is close to the value given by the HAP result as 7589 L/s (16080 cfm).

- Figure XXIII & XXIV shows the results obtained by HAP.

Air System Sizing Summary for University

Project Name: University

Air System Information

Air System Name	Auditorium		Number of zones	1	
Equipment Class	CW AHU		Floor Area	615.0	m²
Air System Type	SZCAV		Location	Karachi, Pakistan	

Sizing Calculation Information

Calculation Months	Jan to Dec	Zone L/s Sizing	Peak zone sensible load
Sizing Data	Calculated	Space L/s Sizing	Zone L/(s-m²)

Central Cooling Coil Sizing Data

Total coil load	192.8	kW	Load occurs at	Jun 1600	
Sensible coil load	117.2	kW	OA DB / WB	38.1 / 27.7	°C
Coil L/s at Jun 1600	7589	L/s	Entering DB / WB	25.9 / 20.1	°C
Max block L/s at Jul 1700	7589	L/s	Leaving DB / WB	13.1 / 12.6	°C
Sum of peak zone L/s	7589	L/s	Coil ADP	11.7	°C
Sensible heat ratio	0.608		Bypass Factor	0.100	
m²/kW	3.2		Resulting RH	61	%
W/m²	313.5		Design supply temp.	12.0	°C
Water flow @ 5.6 °K rise	8.31	L/s	Zone T-stat Check	1 of 1	OK
			Max zone temperature deviation	0.0	°K

Supply Fan Sizing Data

Actual max L/s at Jul 1700	7589	L/s	Fan motor BHP	0.00	BHP
Standard L/s	7585	L/s	Fan motor kW	0.00	kW
Actual max L/(s-m²)	12.34	L/(s-m²)	Fan static	0	Pa

Outdoor Ventilation Air Data

Design airflow L/s	1622	L/s	L/s/person	2.82	L/s/person
L/(s-m²)	2.64	L/(s-m²)			

Figure XXIII.- *HAP Results*

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Air System Design Load Summary for University

Project Name: University

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Jun 1600 COOLING OA DB / WB 38.1 °C / 27.7 °C			HEATING DATA AT DES HTG HEATING OA DB / WB 9.4 °C / 5.1 °C		
ZONE LOADS	Details	Sensible (W)	Latent (W)	Details	Sensible (W)	Latent (W)
Window & Skylight Solar Loads	0 m²	0	-	0 m²	-	-
Wall Transmission	510 m²	14748	-	510 m²	10439	-
Roof Transmission	615 m²	25465	-	615 m²	12242	-
Window Transmission	0 m²	0	-	0 m²	0	-
Skylight Transmission	0 m²	0	-	0 m²	0	-
Door Loads	24 m²	610	-	24 m²	479	-
Floor Transmission	615 m²	6789	-	615 m²	0	-
Partitions	0 m²	0	-	0 m²	0	-
Ceiling	0 m²	0	-	0 m²	0	-
Overhead Lighting	1902 W	1559	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	575	30106	20240	0	0	0
Infiltration	-	9670	12998	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	**88949**	**33238**	-	**23160**	**0**
Zone Conditioning	-	86802	33238	-	-6085	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	7589 L/s	0	-	7589 L/s	0	-
Ventilation Load	1622 L/s	30390	42381	1622 L/s	6310	0
Supply Fan Load	7589 L/s	0	-	7589 L/s	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	**117193**	**75619**	-	**224**	**0**
Central Cooling Coil	-	117193	75621	-	0	0
>> Total Conditioning	-	**117193**	**75621**	-	**0**	**0**
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

Figure XXIV.- *Sensible & Latent heat gains*

Figure XXV depicts the HAP result which shows a psychrometric diagram of the Air conditioning process for the University Auditorium.

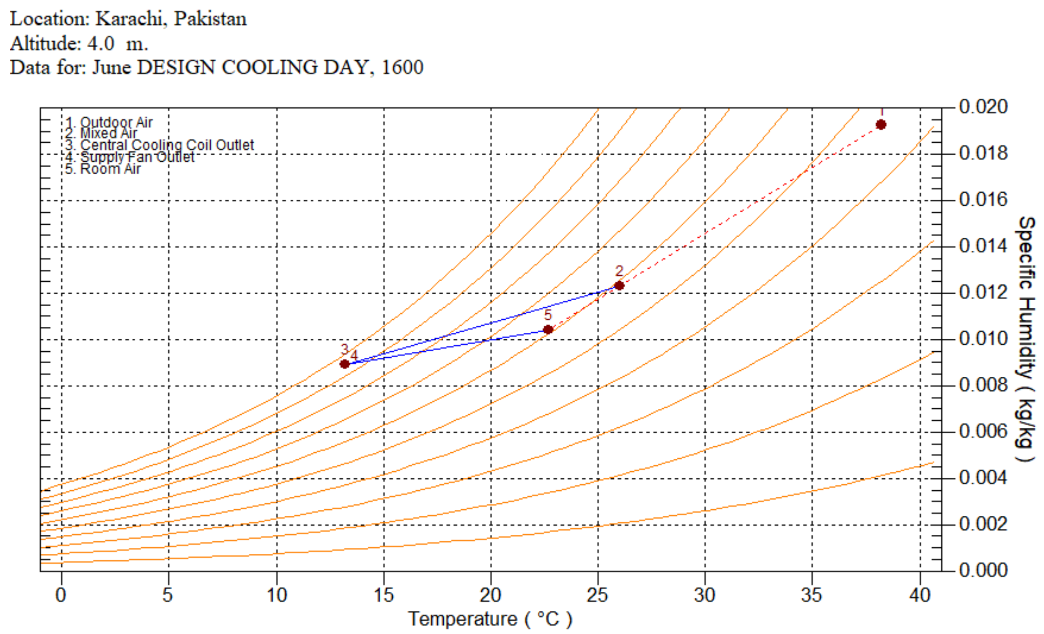

Figure XXV.- *Psychrometric chart obtained from HAP software*

Figure XXVI shows the comparison of the University Auditorium cooling loads between CLTD and HAP. Figure XXVII depicts the comparison of Total Heat Gain by CLTD & HAP software.

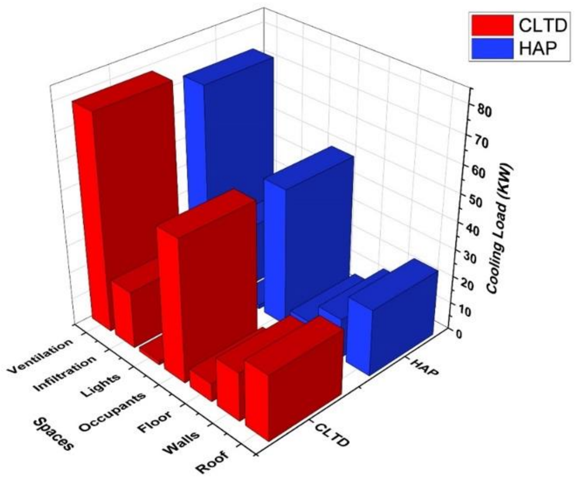

Figure XXVI.- *Cooling load (CLTD vs HAP)*

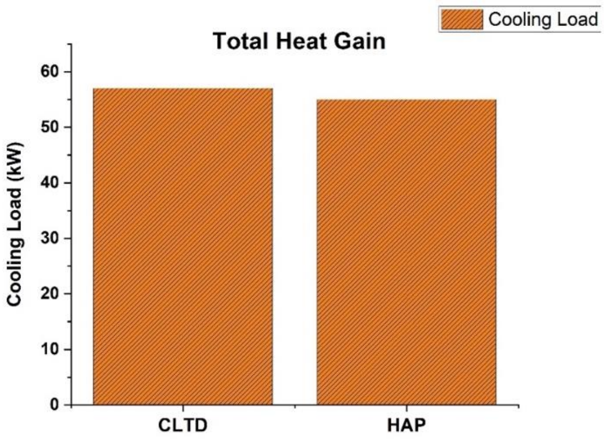

Figure XXVII.- *Total Heat Gain (CLTD vs HAP)*

4.3 Air Balancing. - Figure XXVIII shows the Fresh air, return air & Supply air flow rates by using the CLTD method.

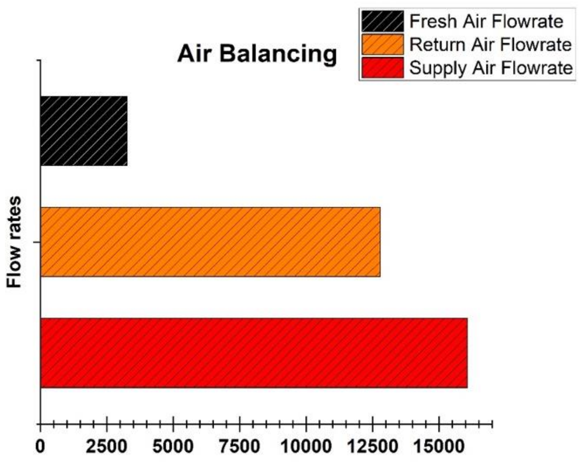

Figure XXVIII.- *Air Balancing Flowrates*

4.4 Diffuser & Return Air Grill Selection. - A total sixteen number of diffusers in the University Auditorium have been calculated. We have selected Round Diffusers [38] because they meet the low NC requirement for the Auditorium. The diffuser that meets our requirement is Price's 20" Round Cone Diffuser (RCD) providing supply air of 1100 cfm at a minimal NC value of 15. Air velocity is 500 fpm whereas the throw at 150 mm is 14 ft.
Similarly, we will install six return air grills in the Auditorium space and select Price's 40" x 16" Rectangular Louvered Grille, whose intake flow rate is 2320 cfm [39].

4.5 Duct Losses. - Table XXVII represents the results obtained for the major and minor losses through different Supply air duct segments of the Index run. Table XXVIII represents the results obtained for the major and minor losses through different Return air duct segments of the Index run.

Segment	Major Loss (Pa)	Minor Loss (Pa)	Head Loss (Pa)	Total Pressure Drop (Pa)
A	0.72	12.7	--	13.43
B	1.98	14.43	--	16.41
B1	1.47	13.88	120.29	135.65
D	2.24	27.76	--	30.00
G	1.00	0.39	--	1.39
I	0.71	7.36	--	8.07
I1	0.04	2.01	--	2.06
		Pressure drop across supply diffuser		16
			Total	223.01

***Table XXVII.** Supply air duct losses*

Segment	Major Loss (Pa)	Minor Loss (Pa)	Total Pressure Drop (Pa)
A	0.05	7.34	7.39
C	0.41	8.31	8.73
G	0.64	9.37	10.01
G1	0.95	5.95	6.90
K	0.09	4.35	4.43
K1	0.13	4.35	4.48
K2	0.90	4.35	5.24
K3	0.05	4.35	4.40
K4	0.13	4.35	4.48
K5	0.05	4.35	4.40
	Pressure drop across the return grill		17
		Total	77.46

***Table XXVIII.** Return air duct losses*

4.6 Duct Sizing. - Supply & return air duct sizing results for an aspect ratio of 1.5 are shown in Table XXIX & Table XXX respectively.

Segment	Dia. (cm)	Size		Duct Shape
		a (cm)	b (cm)	
A	144	164	109	Rectangular
B	111	126	84	Rectangular
B1	111	126	84	Rectangular
C	111	126	84	Rectangular
C1	111	126	84	Rectangular

D	111	126	84	Rectangular
E	66	75	50	Circular
E1	51	59	39	Circular
E2	51	59	39	Circular
F	66	75	50	Circular
F1	51	59	39	Circular
F2	51	59	39	Circular
G	86	98	65	Rectangular
H	66	75	50	Circular
H1	51	59	39	Circular
H2	51	59	39	Circular
I	66	75	50	Circular
I1	51	59	39	Circular
I2	51	59	39	Circular
J	111	126	84	Rectangular
J1	51	59	39	Circular
J2	106	120	80	Rectangular
K	93	105	70	Rectangular
L	66	75	50	Circular
M	51	59	39	Circular
N	51	59	39	Circular
O	51	59	39	Circular
P	51	59	39	Circular
Q	77	89	59	Circular
R	51	59	39	Circular
S	51	59	39	Circular
T	51	59	39	Circular
Fresh	82	93	62	Rectangular

Table XXIX. *Supply air duct sizing*

Segment	**Dia. (cm)**	**Size**	
		a (cm)	**b (cm)**
A	147	166	110
B	112	127	84
C	112	127	84
D	76	86	57
D1	76	86	57
D2	76	86	57
E	76	86	57
E1	76	86	57
E2	76	86	57
F	100	112	75
F1	100	112	75
G	100	112	75
G1	100	112	75
H	76	86	57
I	76	86	57

J	76	86	57
J1	76	86	57
J2	76	86	57
J3	76	86	57
J4	76	86	57
J5	76	86	57
K	76	86	57
K1	76	86	57
K2	76	86	57
K3	76	86	57
K4	76	86	57
K5	76	86	57

Table XXX. *Return air duct sizing*

4.7 Chiller Selection

4.7.1 Proposed System # 1. - As per the design requirement and the consultant's provided design condition datasheet, the chiller we have selected is YORK's 65-ton scroll compressor-type water-cooled chiller.

YORK's 65 tons scroll compressor type water-cooled chiller specification	
Model	YCWL0261HE
Refrigerant	R-410 (A)
EER	14.90 (Btu/W·h)
Chilled water supply/return temp.	7 °C / 12 °C (44 °F / 54 °F)
Condenser water supply/return temp.	32 °C /38 °C (90 °F /100 °F)
Condenser / chilled water flow rates	205 / 156 US gpm
Condenser/evaporator pressure drop	8.01 / 5.74 ft. of water
Maximum operating power	52 kW

Table XXXI. *Specification of Water-cooled chiller*

This chiller consists of four scroll compressors which provide a part load condition. All four compressors are operating at maximum cooling load conditions but as the cooling requirement of the zone decreases, the system automatically turns off one of the compressors. Therefore, the number of compressors in operation depends on the zone cooling requirement resulting in saving power. Moreover, even at the maximum capacity, the system operating power is still lower than the existing installed system whose compressor's operating power is 55 kW. GWP of R-410 (A) is 2088.

4.7.2 Proposed System # 2. - Another chiller that is nearest to our design capacity and as per the consultant's provided design condition datasheet is YORK's 71 tons Air-cooled Scroll Chiller.

YORK's 71 tons Air-cooled Scroll Chiller specification	
Model	YLAA0286SE
Refrigerant	R-410 (A)
EER	7.47 (Btu/W·h)
Chilled water supply/return temp.	7 °C / 12 °C (44 °F /54 °F)
Chilled water flow rates	168 US gpm
Pressure loss	10.6 ft. of water
Maximum operating power	115 kW

Table XXXII. *Specification of Air-cooled chiller*

This chiller consists of six scroll compressors, R-410 (A) as a refrigerant, and works on the same part load condition as water cooled chiller does. As compared to water cooled chiller, this chiller has a lower value of energy efficiency and its power input is more than double that of the water-cooled chiller. The chiller has an approx. size of 3400 mm x

2250 mm x 2400 mm. The roof of the University Auditorium where the current cooling tower is installed has sufficient space to accommodate this chiller.

4.8 AHU Selection. - Table XXXIII depicts the important parameters for selection of AHU which we have previously calculated.

AHU Selection	
Supply air flow rate	16071 cfm
Return air flow rate	12800 cfm
Fresh air flow rate	3272 cfm
Off-coil temperature (dbt / wbt)	26 °C / 20 °C
On coil temperature (dbt / wbt)	13 °C / 11 °C
External press. drop	223 x 1.25 (Factor of safety) = 277 Pa
Chilled water flow rate	156 / 168 US gpm
Chilled water supply/return temperature	7 °C / 12 °C

Table XXXIII. *AHU selection parameters*

By considering the above parameters and as per the consultant's provided datasheet, we have selected AHU from YORK, having Model # YMA(T)1730H-2450W. This AHU provides a supply air flow rate of 8.30 m^3/s (17600 cfm approx.) having a size of 3000 mm x 2450 mm x 1830 mm (smaller than the AHU of an existing system whose dimensions are 3940 mm x 3090 mm x 1740 mm) and will easily be fitted inside the plant room. The fan must overcome a total pressure drop of 750 Pa. (internal Pressure Drop = 277 Pa, external Pressure Drop = 473 Pa). For this purpose, a fan of 15 kW of nominal power is selected. This AHU model is applicable for both proposed System # 1 and System # 2.

4.9 Cooling Tower Selection. - Table XXXIV depicts the important parameters for the selection of a cooling tower which we have previously calculated.

Cooling Tower Selection	
System capacity	65 tons
Condenser water flow rate	205 US gpm
Cooling tower inlet/outlet temperatures	32 °C / 38 °C (90 °F /100 °F)
Design wet bulb temperature	23.2 °C (73 °F)

Table XXXIV. *Cooling tower selection parameters*

By considering the above parameters and as per the consultant's design condition datasheet, we have selected Liang Chi's bottle type counter flow induced draft cooling tower of Model # LBC-60-S. This model can provide a flow rate at 205 US gpm and meets the required condenser temperatures. It has a direct motor drive axial-flow fan that runs at 750 rpm with a motor input power of 2 hp. Furthermore, the inlet and outlet connection have a 3" dia. so a reducer must be required for the installation of the condenser water pipe. The roof of the University Auditorium is a suitable place for its installation.

4.10 Pump Selection. - Table XXXV depicts the important parameters for the selection of Chilled/condenser water pumps which we have previously calculated.

Parameters	**Unit**	**Water cooled system**		**Air-cooled system**
		Chilled water pump	Condenser water pump	Chilled water pump
Flowrate	US gpm	156	205	168
Pressure head	Ft	32.43	36.37	51.26
Pipe dia.	inch	3.5	4	3.5

Table XXXV. *Pump selection parameters*

By considering the above parameters and as per the consultant's design condition datasheet, we have selected KSB's low-pressure centrifugal pumps. Table XXXVI depicts the model numbers and quantity of selected chilled & cool water pumps for both proposed systems.

System	Pump	Quantity	Model No.
Water cooled system	Chilled Water Pump	2	ETN 065-050-200 GBSAA11GD200224B
	Condenser Water Pump	2	ETN 080-065-200 GBSAA11GD200224B
Air-cooled system	Chilled Water Pump	2	ETN 080-065-200 GBSAA11GD200404B

Table XXXVI. *Pumps Model Number*

These models meet our required head and flow rates. Motor input power and speed of chilled water and condenser water pumps are 2.95 hp each & 1400 rpm for water cooled system. The motor input power and speed of the chilled water pump are 5.36 hp & 1400 rpm for the air-cooled system.

4.11 Piping System. - Table XXXVII depicts the head loss from the water-cooled and air-cooled chilled & cool water piping system. All losses are measured in ft.

Head Loss	Water cooled system		Air-cooled system
	Condenser water circuit	Chilled water circuit	Chilled water circuit
Pipe Friction	15.72	7.19	18.01
Condenser / Evaporator	8.03 / --	-- / 5.75	-- / 10.62
Cooling Tower / AHU	6.56 / --	-- / 14.09	-- / 14.09
Total Head Loss (with 20% Factor of Safety)	36.4	32.43	51.26

Table XXXVII. *Piping head loss*

4.12 Piping Layout. - Figure XXIX & XXX depicts the piping layout of the plant room for both the proposed HVAC systems.

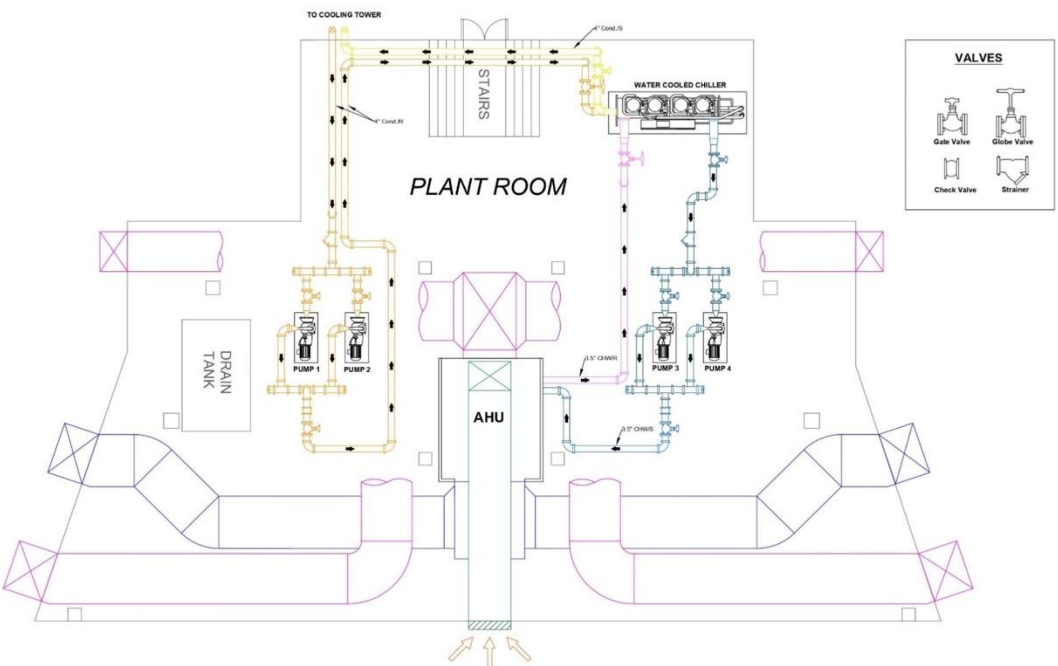

Figure XXIX.- *Pipe Layout of Plant Room for Proposed System # 1*

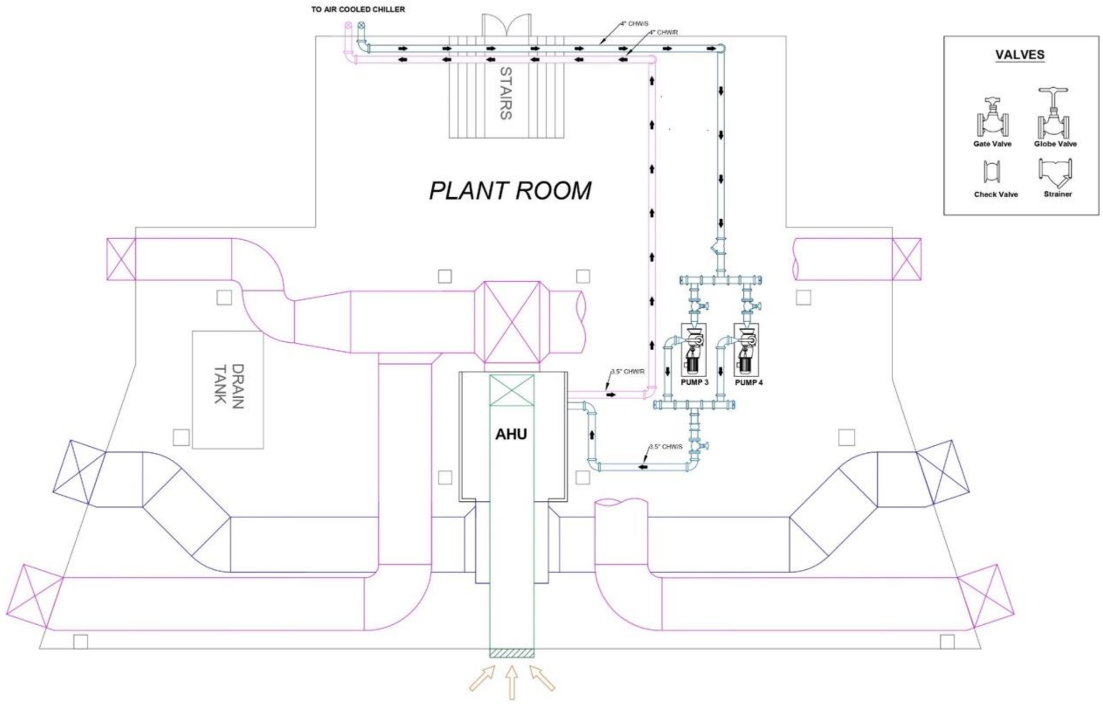

Figure XXX.- *Pipe Layout of Plant Room for Proposed System # 2*

5. Conclusion & Discussion. - This study outlines the design of HVAC systems for the University Auditorium, proposing both water-cooled and Air-Cooled Vapor Compression systems. Cooling loads were calculated using the CLTD method and HAP software, resulting in 57 TR and 55 TR respectively, with a small discrepancy of only 3.5%. The fresh air flow rate was manually calculated at 3272 cfm, closely matching HAP's 3436 cfm, while for the Supply air flow rate, the manual calculation (Heat Removal method) yielded 16071 cfm compared to HAP's 16080 cfm. Equipment selections include sixteen Price’s 20-inch Round Cone Diffusers, six Price’s 40” x 16” Rectangular Louvered Grille, YORK’s 65 tons scroll compressor type water-cooled chiller, YORK’s 71 tons Air-cooled Scroll Chiller, YORK’s AHU, and Liang Chi’s bottle type counter flow induced draft cooling tower, along with two KSB’s centrifugal pumps. The study highlights the importance of designing cost-effective and energy-efficient HVAC systems, addressing the increasing demand driven by global warming and humid climates, and offering guidance for engineers and researchers to propose new systems for spaces, including replacing obsolete ones, based on selection criteria and research findings.

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2. Adquisición de datos
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