Study of Cooling Loads with Energy Comparison & Cost Analysis for Optimal HVAC System Design

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In partial fulfillment of the requirements for the degree of

B.SC IN MECHANICAL ENGINEERING

Supervised By

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In partial fulfillment of the requirements for the degree of

B.SC IN MECHANICAL ENGINEERING

Letter of Transmittal

To

The Supervisor

Department of Mechanical Engineering.
Sonargaon University of Bangladesh

Subject: Submission of Project Report.

Dear Sir,

We are pleased to submit the project report on "**Cooling Loads with Energy Comparison & Cost Analysis for Optimal HVAC System Design** ". It was a great pleasure to work on such an important topic. This project has been done as per instruction of your supervision and according to the requirements of the Sonargaon University.

We expect that the project will be accepted by the concerned authority we will remain happy to further explanation that you may feel necessary in this regard.

Thank You
Sincerely yours,

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DECLARATION

We do hereby solemnly declare that, the work presented here in this project report has been carried out by us and has not been previously submitted to any University/ Organization for award of any degree or certificate

We hereby ensure that the works that has been prevented here does not breach any existing copyright.

We further undertake to indemnify the university against any loss or damage arising from breach of the foregoing obligation.

Signature

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First and foremost, we would like to express our deepest gratitude to the Almighty, whose blessings, guidance, and mercy enabled us to complete this study titled *"Study of Cooling & Heating Loads with Energy Comparison & Cost Analysis for Optimal HVAC System Design."*

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Abstract

The design and selection of an optimal HVAC (Heating, Ventilation, and Air Conditioning) system play a critical role in ensuring energy efficiency, indoor comfort, and cost-effectiveness. This study focuses on a detailed analysis of **cooling and heating load calculations**, followed by an **energy consumption comparison and cost analysis** for different HVAC system configu rations. The objective is to identify the most efficient and economically viable solution for a given application.

The methodology includes estimating internal and external loads based on building design parameters, occupancy, equipment usage, ventilation requirements, and climatic conditions. Using these load profiles, alternative HVAC system options were evaluated in terms of energy demand, operational efficiency, and life cycle cost. The results highlight the significance of accurate load estimation in avoiding system oversizing or undersizing, thereby reducing both initial investment and long-term operational expenses.

Findings from this study demonstrate that a properly sized HVAC system not only ensures thermal comfort but also contributes to significant **energy savings and reduced environmental impact**. The comparative cost analysis further provides a basis for decision-making, enabling the selection of the most sustainable and cost-effective HVAC solution.

This research emphasizes the importance of integrating load calculations, energy assessment, and financial analysis in HVAC design to achieve optimal performance and sustainability.

Table of Contents

Acknowledgment ……..……..……. 1

Abstract ……..……..2

Table of Contents ……...……...……3-4

Chapter 1: Introduction
1.1 Background of the Study ……...……. 5-6
1.2 Problem Statement ……6-7
1.3 Research Objectives ……8-9
1.4 Scope and Limitations of the Study …..……9-10

Chapter 2: Literature Review
2.1 Overview of HVAC Systems ……...11-13
2.2 Principles of Heat Transfer in Buildings ……..14-16
2.3 Factors Affecting Cooling and Heating Loads ……16-19
2.4 Energy Efficiency Standards and Best Practices…..……19-23

Chapter 3: Methodology
3.1 Data Collection ……..…...24-29
3.2 Calculation Methods for Thermal Loads ……..……29-33
3.3 Energy and Cost Analysis Framework…..……..……33-35

Chapter 4: Building and Climatic Data
4.1 Classroom Layout and Building Specifications ……..…….36-41
4.2 Climatic Conditions of the Study Area ……..……...42-44
4.3 Occupancy and Equipment Details ……..……44-47

Chapter 5: Energy Comparison and Cost Analysis
5.1 Comparison of HVAC System Types ……...48-49
5.2 Energy Consumption Analysis ……49-53
5.3 Cost-Benefit Analysis and Payback Period ……..53-56

Chapter 6: Results and Discussion
6.1 Summary of Calculated Loads ……57-63

Chapter 7: Case Study
7.1 HAP Report on Traditional Cooling System ……64-71
7.2 HAP Report on Energy Efficient Cooling System ……...72-79

References ……..80-83

Appendices

Appendix A: Technical Specifications of HVAC Equipment ……84-89

Chapter 1: Introduction

1.1 Background of the Study

The design and implementation of effective heating, ventilation, and air conditioning (HVAC) systems are paramount for ensuring a comfortable and productive indoor environment, particularly in educational settings. Buildings, especially academic institutions, are significant consumers of energy, with HVAC systems often accounting for a substantial portion of total energy consumption. In a tropical climate like that of Bangladesh, the demand for cooling is particularly high, leading to considerable energy expenditure and operational costs. Conversely, during cooler seasons, a certain level of heating may be required to maintain thermal comfort. Therefore, a thorough understanding and analysis of the thermal loads—the total amount of heat that must be added or removed from a space to maintain a desired temperature—is a crucial first step in designing an optimal HVAC system.

Heat transfer in buildings occurs through three primary mechanisms: conduction, convection, and radiation. Conduction is the transfer of heat through solid materials, such as the walls, roof, and floor of a building. Convection involves heat transfer through the movement of fluids, like air, both inside and outside the building. Radiation is the transfer of heat via electromagnetic waves, a prime example being the solar radiation that enters a building through windows and other transparent surfaces. In addition to these natural phenomena, a classroom environment is subject to significant internal heat gains from occupants, lighting, and various electronic equipment. For instance, the human body generates heat, and a classroom with a high occupancy rate will have a proportionally higher internal thermal load. Similarly, the heat dissipated by a projector, lights, and other electronic devices must be factored into the overall load calculation.

The selection of an HVAC system without a prior, detailed load analysis can lead to significant issues. An undersized system will fail to maintain a comfortable temperature, leading to occupant dissatisfaction and potential system failure due to constant overwork. An oversized system, on the other hand, while capable of meeting the thermal demands, is often more expensive to install, has a shorter lifespan, and operates less efficiently, leading to unnecessary energy consumption and higher utility bills. This phenomenon, known as "short-cycling,"

occurs when an oversized unit quickly cools the space and then shuts off, repeating the cycle and consuming more energy than a properly sized unit. Given these challenges, a precise and comprehensive study of thermal loads is not just an academic exercise but a practical necessity for sustainable and cost-effective building management.

The energy and cost implications of HVAC system design in academic buildings are well-documented. Research has shown that a significant portion of a university's total energy consumption is attributable to its HVAC systems. Consequently, optimizing these systems represents a major opportunity for energy savings and a reduction in the institution's carbon footprint. The goal is to identify a system that not only meets the thermal comfort requirements but also provides the best possible return on investment over its operational lifespan. This requires a careful balance between initial capital costs, ongoing energy consumption, and maintenance expenses. Furthermore, the modern focus on green building practices and sustainability makes it imperative to consider environmentally friendly HVAC solutions that reduce reliance on non-renewable energy sources. By conducting a detailed load analysis and a comparative energy and cost analysis, this study aims to provide a data-driven basis for the design of a high-performance HVAC system for the specified classroom, serving as a model for similar applications within Sonargaon University and beyond.

1.2 Problem Statement

The challenge of creating an optimal indoor environment in academic buildings is multifaceted, particularly in regions with demanding climates. In the context of Sonargaon University, a specific classroom (Room No. 306) presents a unique set of thermal and energy-related problems that require a scientific approach for resolution. The existing, or proposed, HVAC system design for this space is often based on rule-of-thumb calculations or generalized assumptions, which fail to account for the specific characteristics of the classroom, such as its orientation, building materials, and internal heat loads. The lack of a precise thermal load analysis for this specific room leads to a high probability of installing an inefficient system. This inefficiency manifests in two critical ways: compromised thermal comfort for students and faculty, and

excessive energy consumption, contributing to higher operational costs for the university.

A significant issue lies in the variability of thermal loads throughout the day and across different seasons. Solar exposure, which changes with the time of day and the angle of the sun, has a major impact on the cooling load, particularly on the exposed glass door. The number of occupants in a classroom can fluctuate dramatically, especially between lecture hours and off-peak times, directly affecting the internal heat gain. Without a system designed to handle these dynamic changes, it is difficult to maintain a consistent and comfortable temperature. For example, a system sized for peak occupancy may be grossly oversized for a low-occupancy period, leading to the aforementioned short-cycling and energy wastage. This lack of responsiveness to real-time conditions is a key problem that this study seeks to address.

Furthermore, the problem extends beyond mere comfort and energy use. Poor indoor air quality, which can result from an improperly designed or ventilated HVAC system, has a documented negative impact on student health, concentration, and academic performance. A system that does not adequately handle humidity or ventilate the space can lead to the growth of mold and mildew and the accumulation of airborne pollutants, creating a less-than-ideal learning environment. The challenge is to design a system that not only manages temperature but also ensures adequate air circulation and filtration to maintain a healthy and productive space.

Finally, there is the economic challenge of balancing initial capital investment with long-term operational costs. The university, like any institution, seeks to manage its budget efficiently. This means that the selected HVAC system must not only be technically sound but also economically viable. The high initial cost of a more advanced, energy-efficient system may be a deterrent, even if it promises significant long-term energy savings. The absence of a detailed cost-benefit analysis of various HVAC options makes it difficult to make an informed decision. This study aims to provide a clear, data-driven comparison of different systems, including their respective initial costs, energy consumption, and maintenance requirements, to identify a solution that is both technically and financially optimal for Sonargaon University's specific needs.

1.3 Research Objectives

This study is designed to address the problems outlined above by pursuing a series of specific research objectives. The primary goal is to provide a comprehensive analysis that will inform the design of an optimal HVAC system for Room No. 306 at Sonargaon University. This will be achieved through a multi-faceted approach, beginning with a detailed calculation of the room's thermal loads.

First and foremost, this research seeks to accurately quantify the cooling and heating loads of the classroom. This objective involves a meticulous calculation of all contributing factors, including solar heat gain through the exposed glass door, heat transfer through the walls, roof, and floor, and internal heat gains from occupants, lighting, and electronic equipment. These calculations will be based on the specific dimensions, materials, and usage patterns of the room, as identified in the provided architectural plans. The outcome of this objective will be a precise, data-driven figure for the maximum thermal load, which is essential for proper HVAC sizing.

Second, the study aims to evaluate and compare the energy consumption of different types of HVAC systems. The investigation will not be limited to a single solution but will explore various commercially available technologies, such as split-type air conditioners, variable refrigerant flow (VRF) systems, and other suitable options for a classroom environment. By simulating the energy usage of each system under the calculated thermal loads, this objective will provide a clear comparison of their operational efficiency and environmental impact. This will move the decision-making process beyond theoretical performance to a practical understanding of real-world energy use.

Third, a critical objective is to conduct a detailed cost analysis for the optimal HVAC system. This includes an assessment of both the initial capital expenditure for equipment and installation and the long-term operational costs, such as energy bills and maintenance. This objective is crucial for providing a holistic view of the financial viability of each option. By presenting a clear cost-benefit analysis, including the payback period for more energy-efficient systems, this research will empower the university to make an informed decision that balances initial investment with long-term savings.

Finally, the ultimate objective is to provide a data-driven recommendation for the optimal HVAC system design for Classroom No. 306. This recommendation will be based on a synthesis of the thermal load calculations, energy consumption comparisons, and cost analysis.

The study will not simply identify a system but will also provide a detailed rationale for its selection, addressing factors such as performance, energy efficiency, cost-effectiveness, and environmental sustainability. This will serve as a practical blueprint for the university, not only for this specific room but also as a replicable model for other similar spaces on campus.

1.4 Scope and Limitations of the Study

The scope of this research is specifically defined to provide a focused and achievable analysis, while also acknowledging the inherent limitations that may influence the findings. This study will be narrowly focused on the thermal load and HVAC system design for a single, specific academic space: Classroom No. 306 in the main building of Sonargaon University. The analysis will be based on the provided architectural floor plans and equipment details, which serve as the foundational data for all calculations. The study will include a comprehensive calculation of all heat gains, encompassing conduction through the building envelope (walls, roof, floor), solar radiation through the glass door, and internal gains from occupants and electrical equipment.

The research will compare three distinct types of HVAC systems—a conventional split-type air conditioner, a Variable Refrigerant Flow (VRF) system, and a packaged air conditioning unit—to determine the most suitable option. The comparison will be based on three key metrics: thermal performance (ability to meet the calculated load), energy consumption, and a detailed cost analysis that includes both initial installation and long-term operational costs. The study will also consider the impact of key variables such as the number of occupants, the type and usage of lighting, and the operational schedule of the classroom. The final output will be a detailed, data-driven recommendation for the optimal HVAC system for the room, which can be used as a model for future campus-wide improvements.

Despite its comprehensive nature, this study is subject to several limitations that should be considered when interpreting the results. First, the thermal load calculations are based on the assumption of a maximum occupancy of 30 students and a specific set of operational times for lights and a projector. Any deviation from these assumptions, such as a higher number of students or a change in the use of electrical equipment, would alter the thermal load and, consequently, the performance of the recommended system. The analysis will not account for

the impact of air infiltration or the variability of heat transfer coefficients of the materials over time, which could affect the overall thermal performance.

Second, the cost analysis is based on current market prices for equipment and energy in Bangladesh, which are subject to fluctuation. While the study will provide a snapshot of costs at the time of research, these figures may change over time, affecting the long-term financial viability of the recommended system. The study also does not account for potential government subsidies or tax incentives for energy-efficient technologies, which could impact the final cost-benefit analysis.

Finally, the scope is limited to a single classroom. While the methodology and findings can serve as a valuable blueprint, they may not be directly applicable to other spaces without a new, specific thermal load analysis. Each room's unique architectural features, orientation, and usage patterns must be individually assessed. For example, a larger classroom, a laboratory, or a library would have a completely different thermal load profile, requiring a separate and distinct study. Therefore, the conclusions of this research are specific to Classroom No. 306 and should be used as a methodological framework rather than a universal solution for all university spaces.

Chapter 2: Literature Review

2.1 Overview of HVAC Systems

The field of heating, ventilation, and air conditioning (HVAC) encompasses a wide array of systems designed to regulate thermal comfort and air quality in indoor environments. The selection of an appropriate system is a critical decision that depends on a multitude of factors, including the climate, building size and type, operational budget, and sustainability goals. This chapter provides a comprehensive overview of the most common and relevant HVAC systems, detailing their operational principles, advantages, and disadvantages, thereby establishing a foundational understanding for the subsequent analysis.

HVAC systems can be broadly categorized into centralized and decentralized systems. Centralized systems, such as chilled water systems or large rooftop packaged units, serve multiple rooms or an entire building from a single, central location. These systems typically employ a network of ducts and pipes to distribute cooled or heated air and water throughout the facility. Their primary advantage lies in their high efficiency when serving large, contiguous spaces, as they benefit from economies of scale. They also offer a high degree of control over temperature and humidity and can incorporate advanced filtration and ventilation features. However, their high initial cost, complex installation process, and the potential for a single point of failure are significant drawbacks.

In contrast, decentralized systems are self-contained units that serve individual rooms or zones. These include split-type air conditioners, window units, and ductless mini-splits. Their key benefits are a lower initial investment, ease of installation, and the ability for occupants to control the climate in their specific area. They are an excellent choice for retrofitting existing buildings or for smaller, non-contiguous spaces where centralized systems are not cost-effective. The main disadvantage is that they may be less energy-efficient when scaled up to serve an entire building, as a large number of units can consume more power collectively than a single, high-efficiency centralized system.

A staple of decentralized HVAC is the Split-Type Air Conditioner. This system consists of two main units: an outdoor unit containing the compressor and condenser, and an indoor unit containing the evaporator and fan. The two units are connected by refrigerant lines and electrical wiring. The indoor unit draws in warm room air, passes it over the cold evaporator

coil to cool it, and then blows the cooled air back into the room. The heat absorbed by the refrigerant is then transferred to the outdoor unit and expelled into the outside air. Split systems are a popular choice due to their quiet indoor operation, as the noisy compressor is located outside, and their relatively simple installation process. However, a major limitation is their fixed capacity, meaning they operate at full power until the desired temperature is reached, which can lead to frequent on/off cycling and decreased energy efficiency, particularly in spaces with fluctuating thermal loads.

An evolution of the split-type system is the Variable Refrigerant Flow (VRF) system. This is a multi-split system that connects a single outdoor unit to multiple indoor units. What sets VRF apart is its use of a variable-speed compressor, which can precisely match the cooling or heating demand of each individual zone by modulating the flow of refrigerant. This allows for a high degree of energy efficiency, as the system only uses the exact amount of energy needed to cool or heat a space, eliminating the energy wastage associated with short-cycling. A significant advantage of some VRF systems is their ability to provide simultaneous heating and cooling to different zones, a feature particularly useful in buildings with diverse thermal requirements. While VRF systems offer superior performance and efficiency, their high initial cost and the complexity of their design and installation require a substantial capital investment.

The Packaged Air Conditioning Unit represents a self-contained, centralized approach. In this system, all components—the compressor, condenser, and evaporator—are housed in a single cabinet, typically installed on a rooftop or a concrete slab outside the building. Conditioned air is then distributed to various rooms or zones through a network of ducts. Packaged units are often used in commercial applications where space is limited and a single, robust system is desired. They are relatively easy to install, as they arrive from the manufacturer as a single unit, which simplifies the construction process. Maintenance is also straightforward, as all components are in one location. However, a major drawback is the potential for energy loss through the ductwork, particularly if it is poorly insulated or if there are leaks. Additionally, their location can make them susceptible to noise and vibration issues, which must be managed through proper installation and sound dampening.

Beyond these primary categories, several key technologies and components are integrated into modern HVAC systems to enhance performance and efficiency. Inverter technology, for instance, is a critical innovation that allows the compressor speed to be modulated, rather than simply turning on and off, which dramatically improves energy efficiency and maintains a more

consistent temperature. Heat pumps are another important technology, as they can reverse the refrigeration cycle to provide both heating and cooling from a single unit, making them a versatile and energy-efficient solution in climates that require both. Energy Recovery Ventilators (ERVs) and Heat Recovery Ventilators (HRVs) are designed to improve indoor air quality by introducing fresh outdoor air while recovering energy from the outgoing air, reducing the energy penalty of ventilation. These technologies, when integrated into a system, contribute to a more sustainable and cost-effective solution.

The selection of an HVAC system is not a one-size-fits-all process. The optimal choice is a function of the building's specific thermal loads, the budget, and the long-term energy goals. This aligns with **ASHRAE Standard 55 (2017)**, which emphasizes the importance of maintaining thermal comfort conditions for occupants, and **ASHRAE Standard 62.1 (2019)**, which outlines ventilation requirements to ensure acceptable indoor air quality. While split-type units are an economical and effective solution for single rooms (Jones, 2016; Bhatia, 2011), a larger-scale application may benefit from the superior efficiency and zoning control offered by VRF systems, consistent with the energy performance targets in **ASHRAE Standard 90.1 (2019)** and the **ENERGY STAR (2020)** program. Similarly, packaged units provide a robust, all-in-one solution for commercial spaces with extensive ductwork, which can also support **LEED (USGBC)** credit pathways for energy optimization. By understanding the core principles and characteristics of these systems (Awbi, 2008; Dincer & Rosen, 2012), and considering their effects on health and productivity (Seppänen & Fisk, 2006), a foundation is established for a data-driven analysis to determine the most suitable and sustainable HVAC design for the specific classroom under study.

2.2 Principles of Heat Transfer in Buildings

The thermal performance of any building, and by extension the design of its HVAC system, is governed by the fundamental principles of heat transfer. Understanding how heat moves into and out of a space is the first and most critical step in calculating the thermal load. This process involves three primary modes: conduction, convection, and radiation. Each mode contributes to the total heat gain or loss, and their combined effect dictates the amount of cooling or heating required to maintain a comfortable indoor environment (Jones, 2016; Bhatia, 2011; Awbi, 2008). These principles are also reflected in ASHRAE standards and guidelines, which emphasize accurate load estimation as the foundation for system sizing and performance (ASHRAE 55, 2017;

ASHRAE 90.1, 2019).

Conduction

Conduction is the transfer of thermal energy through a material without any bulk movement of the material itself. In a building, this is the primary mechanism by which heat moves through solid components like walls, roofs, windows, and floors. The rate of heat conduction is influenced by several factors: the temperature difference across the material, the surface area of the material, and the material's thermal conductivity (k). The thermal conductivity is an intrinsic property of a material that quantifies its ability to conduct heat. Materials with a low thermal conductivity, such as insulation, are effective at resisting heat flow, while those with a high thermal conductivity, such as metal, are poor insulators.

The heat transfer by conduction is mathematically expressed by Fourier's Law:

$$Q_{cond} = \frac{k \cdot A \cdot \Delta T}{L}$$

Where:

- Q(cond) is the rate of heat transfer (in Watts)
- k is the thermal conductivity of the material (in W/(m·K))
- A is the surface area through which heat is transferred (in m2)
- ΔT is the temperature difference between the two sides of the material (in Kelvin or Celsius)
- L is the thickness of the material (in meters)

This formula is often simplified for building analysis by using the overall heat transfer coefficient (U), which accounts for multiple layers of materials and the heat transfer at the surface. The U-factor is the reciprocal of the total thermal resistance (R total). For a multi-layered component (e.g., a wall made of brick, air gap, and plasterboard), the total resistance is the sum of the individual resistances of each layer. Q(cond) =U·A·ΔT

In this study, the conduction load for Classroom No. 306 will be calculated for the exposed walls, roof, and floor, using the U-factors for the materials specified in the architectural plans.

The difference between the outdoor and indoor design temperatures (ΔT) will be a critical variable in this calculation.

Convection

Convection is the transfer of heat through the movement of a fluid (in this case, air). It can be categorized into two forms: natural convection and forced convection. Natural convection occurs due to density differences in the fluid. For example, as air near a hot surface (like a warm wall) heats up, it becomes less dense and rises, pulling cooler air in to take its place. This creates a natural circulation pattern. Forced convection, on the other hand, involves an external force, such as a fan or wind, to move the fluid. In buildings, wind blowing on the exterior surface is a form of forced convection that significantly impacts heat transfer.

Convection plays a crucial role in building thermal loads in two primary ways. First, it facilitates the transfer of heat between the building surfaces and the surrounding air. The film coefficient, which describes this heat exchange, is a key component of the overall U-factor for a wall or window. Second, and perhaps more importantly, air infiltration is a major source of convective heat gain (or loss). This is the unintentional flow of outside air into a building through cracks, gaps, and openings around windows and doors. In a tropical climate like that of Bangladesh, warm, humid outdoor air infiltrating a conditioned space adds both sensible and latent heat loads, significantly increasing the demand on the HVAC system. This study will account for an assumed infiltration rate to provide a more accurate thermal load calculation.

Radiation

Radiation is the transfer of energy by electromagnetic waves. Unlike conduction and convection, radiation does not require a medium to transfer heat, as evidenced by the sun's energy traveling through the vacuum of space to Earth. For a building, the primary source of radiant heat gain is solar radiation. The sun's rays directly striking a building's exterior surfaces, especially windows and glass doors, can contribute a substantial amount of heat, which is then re-radiated or absorbed and re-emitted into the interior space.

The heat gain from solar radiation through windows is a complex calculation that depends on several factors:

- The intensity of the sun's radiation, which varies with time of day, season, and geographical location.

- The orientation of the glass surface (e.g., a south-facing door will receive more solar radiation in the Northern Hemisphere).
- The properties of the glass, characterized by its Solar Heat Gain Coefficient (SHGC). The SHGC represents the fraction of incident solar radiation that enters the building as heat. A lower SHGC indicates better resistance to solar heat gain.
- The use of any shading devices, such as awnings or blinds.

The heat gain from radiation is not limited to solar radiation from the outside. Inside the classroom, occupants and equipment also emit radiant heat, which must be considered. While this is often a smaller component than solar or conductive gains, it is a non-negligible part of the total thermal load.

Internal Heat Gains

In addition to the three external heat transfer principles, the internal heat gains within a space are a significant contributor to the thermal load, especially in a densely occupied area like a classroom.

- Occupants: The human body is a constant source of heat. The amount of heat generated varies depending on the person's activity level (e.g., a student sitting quietly vs. a person walking). This heat is composed of both sensible heat (the heat that affects temperature) and latent heat (the heat released from moisture in a person's breath and sweat). For Classroom No. 306, the maximum occupancy of 30 students will be a major factor in the internal heat gain calculation.
- Lighting: All forms of lighting, whether fluorescent, incandescent, or LED, generate heat as a byproduct of their operation. The wattage of the lights is a direct measure of their heat output. Based on the provided floor plan, the classroom has six 15W tube lights, contributing a constant heat load when in use.
- Equipment: Electronic equipment, such as computers, projectors, and other devices, also generates a considerable amount of heat. The floor plan specifies a 500W projector, which will be a significant source of heat when it is on.

By meticulously calculating the heat gains from all three external transfer modes (conduction, convection, and radiation) and adding the internal gains from occupants, lighting, and equipment, this study will arrive at a comprehensive and accurate total thermal load for

Classroom No. 306. This foundational step is essential for the subsequent chapters, which will use this data to select and compare different HVAC systems.

2.3 Factors Affecting Cooling and Heating Loads

The determination of a building's thermal load is not a static calculation but a dynamic process influenced by a wide range of interconnected factors. These factors can be broadly classified into three categories: external factors, which are related to the outdoor environment; internal factors, which originate from within the conditioned space; and building-specific factors, which are inherent to the structure itself. A thorough understanding of how each of these contributes to the total load is crucial for accurate HVAC system design.

External Factors

External environmental conditions are a primary driver of a building's thermal load. They dictate the rate at which heat enters or leaves the space, which the HVAC system must then counteract.

- Solar Radiation: The sun is a dominant source of heat gain, particularly in a tropical climate like that of Bangladesh. Solar radiation affects a building in two ways: it heats the exterior surfaces (walls, roof), which then transfer that heat inwards via conduction, and it directly passes through transparent surfaces like windows and glass doors, heating the interior. The amount of solar heat gain is highly dependent on the building's orientation (e.g., the "sliding glass door exposed to sun" in Room No. 306), the time of day, and the time of year. The Solar Heat Gain Coefficient (SHGC) of the glass is a critical measure here, as a lower SHGC indicates a window that is more effective at blocking solar heat.
- Ambient Air Temperature: This is the most obvious factor affecting thermal load. The greater the difference between the outdoor and indoor temperatures (ΔT), the higher the rate of heat transfer through the building envelope. During the cooling season, a high outdoor temperature drives a significant conductive heat gain through the walls and roof. The reverse is true during the heating season, when a low outdoor temperature causes heat to escape from the building. For this study, the design temperature of the location will be used to calculate the peak load.
- Humidity: The moisture content in the air, or humidity, adds a significant load to the HVAC system, known as the latent heat load. A system must not only cool the air

(sensible heat) but also remove moisture to maintain comfort. In a humid climate, this latent load can be as large as, or even greater than, the sensible load. When warm, humid air infiltrates a conditioned space, the HVAC system must use energy to condense the moisture on the cooling coil, which is a key part of the dehumidification process. The rate of this process is heavily dependent on the difference in humidity levels between the inside and outside air.

- Wind Speed and Direction: Wind impacts heat transfer in two ways. First, it increases the rate of convective heat transfer from the exterior surfaces of the building. A windy day will cause more heat to be transferred into (or out of) the walls and roof. Second, and more critically, wind pressure can increase the rate of air infiltration through cracks and gaps in the building envelope. This infiltration can introduce a substantial amount of unconditioned air, significantly increasing both the sensible and latent heat loads.

Internal Factors

The activities and elements within a space are major contributors to the thermal load, independent of external conditions. These loads are often easier to quantify and are a critical part of the overall calculation.

- Occupancy: The presence of people in a room generates both sensible and latent heat. The human body continuously emits heat at a rate that depends on activity level. For a student sitting in a classroom, the heat generation is approximately 100 Watts per person. With a maximum occupancy of 30 students in Room No. 306, the heat load from occupants alone is substantial. The latent heat, produced by breathing and perspiration, adds to the total load and requires the HVAC system to dehumidify the air.
- Lighting: Lights are a significant source of sensible heat gain. The heat generated is directly proportional to the electrical power consumed by the lighting fixtures. In Classroom No. 306, the six 15W tube lights will contribute a constant heat load of $6 \times 15 = 90$ Watts whenever they are turned on. This is a simple but important factor to include in the calculation.
- Electrical Equipment: The operation of any electronic device generates heat as a byproduct. Computers, projectors, and other office equipment all contribute to the thermal load. The provided floor plan for Room No. 306 explicitly lists a 500W

projector. This single item is a substantial source of heat and must be included in the load calculation, especially since it is likely to be running during peak lecture hours.

Building-Specific Factors

The physical characteristics of the building itself are fundamental to its thermal performance. These factors are largely static and determined during the design and construction phases.

- Building Envelope: The walls, roof, and floor of the building form the primary barrier against external heat transfer. The materials used and their thickness determine their thermal resistance (R-value). A higher R-value means better insulation and lower heat transfer. The U-factor (the reciprocal of the total R-value) is used to quantify the overall heat transfer of a given component. The 6-inch brick walls of the classroom will have a specific U-factor that must be calculated to determine the conductive heat load.
- Fenestration: This term refers to all openings in the building envelope, primarily windows and doors. The size, type of glass, and any shading devices are all critical. The large "sliding glass door exposed to sun" in Room No. 306 is a major source of heat gain due to both conduction and solar radiation. The selection of glass with a low SHGC and the consideration of exterior shading can significantly reduce this load.
- Infiltration and Ventilation: While often considered a negative factor (infiltration), the intentional introduction of fresh outdoor air (ventilation) is necessary for maintaining good indoor air quality. Both processes contribute to the thermal load. Infiltration is the uncontrolled leakage of air, while ventilation is a controlled process. In a cooling application, both can introduce warm, humid air into the space, thereby increasing the sensible and latent load on the HVAC system. ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) standards provide guidelines for minimum ventilation rates based on a space's occupancy and purpose.

By systematically analyzing and quantifying each of these factors—external, internal, and building-specific, this study will construct a comprehensive picture of the total thermal load for Classroom No. 306. This detailed analysis will provide the necessary data to accurately size an HVAC system that can effectively manage all these dynamic heat gains and losses, ensuring both comfort and energy efficiency.

2.4 Energy Efficiency Standards and Best Practices

The pursuit of energy efficiency in buildings is no longer an optional goal but a critical necessity driven by environmental concerns, rising energy costs, and the global push for sustainability. In the context of HVAC systems, which are among the largest consumers of building energy, adherence to recognized standards and best practices is essential for achieving both operational cost savings and a reduced carbon footprint. International frameworks such as **ASHRAE Standard 90.1 (2019)** set minimum energy performance requirements for HVAC systems, while **ASHRAE Standard 62.1 (2019)** ensures adequate ventilation without compromising efficiency. Similarly, green building certifications like **LEED (USGBC)** and efficiency programs such as **ENERGY STAR (2020)** promote strategies that optimize energy performance and support sustainable design goals. These guidelines, combined with design recommendations found in leading texts (Jones, 2016; Bhatia, 2011; Awbi, 2008), provide a comprehensive foundation for integrating high-performance HVAC technologies and operational strategies into modern building design. (**Energy Efficiency Standards and Certifications).**

A number of national and international organizations have developed standards and rating systems to guide the design and construction of energy-efficient buildings. These frameworks provide a benchmark for performance and a common language for designers, engineers, and building owners.

- ASHRAE Standards: The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) is a global authority on building systems. Its ASHRAE Standard 90.1, "Energy Standard for Buildings Except Low-Rise Residential Buildings," is widely adopted and referenced as a minimum requirement for energy efficiency in commercial buildings. The standard provides detailed requirements for building envelopes, HVAC systems, lighting, and other equipment. It specifies minimum efficiency levels for HVAC components like compressors and fans and mandates the use of energy recovery ventilation in certain applications. Adherence to this standard is a crucial first step in designing a system that is fundamentally energy-efficient.
- LEED Certification: The Leadership in Energy and Environmental Design (LEED) is a globally recognized green building certification program. Developed by the U.S. Green

Building Council (USGBC), LEED provides a points-based system for a building's design, construction, operation, and maintenance. HVAC and energy performance are central to several LEED credits. For instance, credits are awarded for optimizing energy performance, using renewable energy, and commissioning the building's systems to ensure they perform as designed. A building pursuing LEED certification will almost invariably require a high-efficiency HVAC system and a carefully designed building envelope.

- ENERGY STAR: This is a voluntary program of the U.S. Environmental Protection Agency (EPA) and the U.S. Department of Energy (DOE) that helps businesses and individuals save money and protect the climate through superior energy efficiency. While primarily known for consumer products, ENERGY STAR also rates commercial buildings and promotes energy-efficient building practices. For HVAC equipment, the ENERGY STAR label signifies that a product meets or exceeds strict energy efficiency guidelines, such as a high Seasonal Energy Efficiency Ratio (SEER) for air conditioners.

Best Practices for HVAC System Design and Operation

Beyond meeting minimum standards, several best practices can dramatically improve the energy performance of an HVAC system.

- Proper System Sizing: As discussed in previous sections, the most critical best practice is accurate thermal load calculation. An oversized system will operate inefficiently by constantly cycling on and off, wasting energy, while an undersized system will fail to meet the cooling demand. Precise sizing ensures the equipment operates within its most efficient range, optimizing performance and prolonging its lifespan.
- High-Efficiency Equipment: The selection of equipment with high efficiency ratings is a cornerstone of energy-efficient design. Look for systems with high SEER ratings for air conditioners and high Energy Efficiency Ratio (EER) ratings for commercial equipment. Inverter technology, which allows a compressor to modulate its speed to match the required load, is a key enabler of high-efficiency operation, as it avoids the energy-intensive on/off cycling of conventional units.
- Variable Speed Drives (VSDs): Also known as Variable Frequency Drives (VFDs), VSDs are used to control the speed of motors in fans, pumps, and compressors. By

adjusting the motor speed to match the system's current load, VSDs can reduce energy consumption by as much as 50% or more. The power consumed by a fan or pump is proportional to the cube of its speed, meaning even a small reduction in speed can lead to significant energy savings.

- Zoning and Controls: Designing a system with multiple zones allows for independent temperature control in different areas of a building. This is particularly effective in a space like a university, where classrooms, offices, and lecture halls have different occupancy schedules and thermal loads. A Building Management System (BMS) is a powerful tool for this, as it centralizes control and monitoring of HVAC, lighting, and other systems, allowing for optimized scheduling and real-time adjustments.

Building Envelope and Passive Design Strategies

The most effective way to reduce the energy consumption of an HVAC system is to minimize the thermal load in the first place. This is achieved through a high-performance building envelope and the use of passive design strategies.

- Insulation: Proper insulation of the building envelope—including the walls, roof, and floor—is paramount. The insulating material's R-value directly correlates with its ability to resist heat flow via conduction. A higher R-value means less heat enters the building in the summer and less heat escapes in the winter, significantly reducing the cooling and heating loads.

- High-Performance Windows: Windows are often the weakest link in a building's thermal envelope. Selecting windows with a low U-factor (for low conduction) and a low SHGC (for minimal solar heat gain) can have a dramatic impact. For the exposed sliding glass door in Classroom No. 306, the choice of glass is critical. In a hot climate, a low-e coating can be applied to glass to reflect infrared radiation while still allowing visible light to pass through.

- Air Sealing: Uncontrolled air infiltration through cracks around windows, doors, and other building components can account for a significant portion of a building’s energy loss. Simple and low-cost measures like caulking and weatherstripping can substantially reduce this infiltration, thereby lowering both sensible and latent loads.

- Shading: Shading devices, such as overhangs, awnings, and blinds, are a simple yet effective way to manage solar heat gain. By blocking direct sunlight from entering a

building, they prevent a significant source of radiant heat and can reduce the cooling load by a substantial amount. The orientation of the building and the location of fenestration should be considered when designing shading strategies.

Integration of Smart Technologies and Maintenance

Finally, the long-term energy efficiency of an HVAC system depends on its ongoing operation and maintenance.

- Programmable and Smart Thermostats: These devices allow for the automatic adjustment of temperature settings based on a pre-set schedule or even real-time occupancy data. In a classroom, for example, the thermostat can be programmed to a higher temperature during unoccupied hours to save energy.
- Regular Maintenance: A well-maintained HVAC system will operate more efficiently and have a longer lifespan. Regular tasks like cleaning or replacing air filters, checking refrigerant levels, and cleaning coils and fan blades are essential for maintaining the system's performance and preventing energy-wasting issues. A clogged filter can significantly reduce airflow, causing the system to work harder and consume more energy.

By integrating these standards and best practices—from initial design to ongoing operation—it is possible to achieve substantial energy savings and create a more sustainable and comfortable indoor environment, which is the ultimate goal of this research for Classroom No. 306.

Integration of smart technologies and regular maintenance practices is essential to ensure the long-term energy efficiency of an HVAC system. **Programmable and smart thermostats** allow automatic adjustment of temperature based on schedules or occupancy, reducing unnecessary energy use (ENERGY STAR, 2020; USGBC LEED Reference Guide). Similarly, **regular maintenance**, such as filter replacement, refrigerant checks, and coil cleaning, ensures sustained performance and prevents efficiency losses (Bhatia, 2011; Jones, 2016). These practices are consistent with energy efficiency guidelines outlined in **ASHRAE Standard 90.1 (2019)** and are recommended as best practices for maintaining sustainable HVAC operations (Awbi, 2008; Dincer & Rosen, 2012). By integrating these standards with ongoing operation strategies, substantial energy savings and improved indoor environmental quality can be achieved, which directly supports the sustainability goals of this study for Classroom No. 306.

Chapter 3: Methodology

3.1 Data Collection

The foundation of this research is a meticulous and systematic data collection process. To accurately calculate the cooling and heating loads for Classroom No. 306, both primary and secondary data were gathered from a variety of sources. This approach ensures that the thermal analysis is not based on general assumptions but on specific, quantifiable parameters of the space. The collected data is crucial for determining the heat transfer through the building envelope, the solar radiation, and the internal heat gains.

Primary Data Sources

The primary source of data for this study is the architectural floor plan of Classroom No. 306, provided by Sonargaon University.

This document, specifically referenced from the Classroom Drawing, contains the essential dimensional and structural information required for a detailed thermal load calculation. The following key data points were extracted and meticulously recorded:

- Room Dimensions: The floor plan provides precise measurements for the length and width of the classroom, which are noted as 25′−6" and 14′−9", respectively. The floor height is specified as 10′−0". These dimensions are fundamental for calculating the surface areas of the walls, floor, and roof, which are necessary for the conductive heat load calculation.
- Building Envelope and Material Specifications: The walls of the classroom are identified as 6-inch brick. This material specification is critical for determining the thermal properties and overall U-factor of the walls. The document also provides a diagram showing the orientation of the room, with the sliding glass door facing the sun.
- Fenestration Details: The floor plan explicitly identifies a "Slidin Glass Door Exposed to Sun." Its dimensions are specified as 4′−0"×7′−6". The size and orientation of this glass surface are crucial for calculating the solar heat gain, which is anticipated to be a significant component of the total cooling load.

- Occupancy and Internal Equipment: The floor plan provides a maximum occupancy count of 30 students. Furthermore, the "ROOM EQUIPMENT DETAILS" section lists the following: a projector (1 unit, 500W) and tube lights (6 units, 15W each). These values are essential for quantifying the internal heat gains from occupants, lighting, and electrical equipment, which are independent of external weather conditions.

Secondary Data and Assumptions

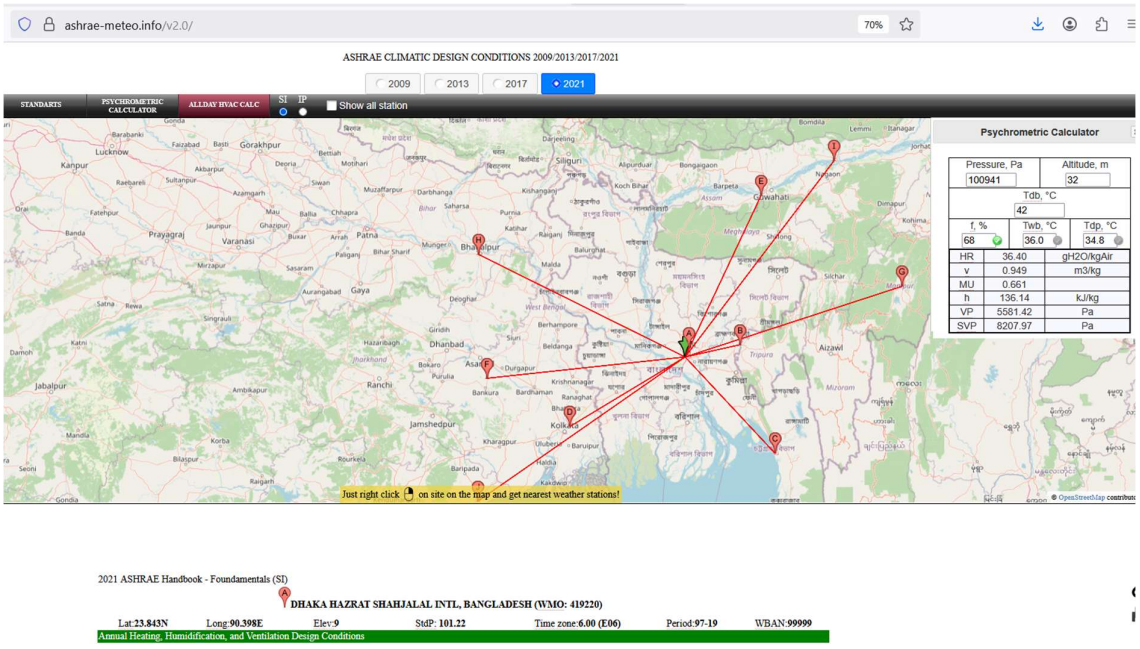
In addition to the primary data from the floor plan, the analysis required the use of secondary data and a number of reasonable assumptions to account for factors that could not be measured on-site. These assumptions are based on industry standards and typical climate data for the location.

- Climatic and Design Conditions: To simulate the peak cooling load, standard design-day weather data for Dhaka, Bangladesh, was used. The peak outdoor dry-bulb temperature was assumed to be 35∘C (95∘F), with a corresponding wet-bulb temperature of 28.5∘C (83.3∘F). The indoor design temperature was set to 24∘C (75.2∘F) to maintain thermal comfort. The difference between the indoor and outdoor temperatures (ΔT) is a key parameter for the conductive heat transfer calculation.
- Material Thermal Properties: The thermal conductivity (k) and heat transfer coefficients (U) for the building materials were sourced from the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Handbook. For the 6-inch brick walls, a standard thermal conductivity value was used to calculate the overall U-factor. For the glass door, a typical Solar Heat Gain Coefficient (SHGC) value for single-pane glass was assumed, as no specific glazing data was available.
- Infiltration and Ventilation Rate: To account for uncontrolled air leakage and to ensure adequate air quality, an infiltration rate was assumed. Due to the lack of on-site measurement, a common practice for a typical classroom was followed by assuming a rate of 0.5 to 1.0 Air Changes per Hour (ACH). The latent heat load associated with this infiltration will be calculated based on the temperature and humidity difference between the indoor and outdoor design conditions.
- Operational Schedules: The analysis assumes that all internal heat sources, including the occupants, lights, and projector, are operating simultaneously during peak load hours to provide a "worst-case scenario" for HVAC sizing. The lights and projector are

assumed to be on for the duration of a lecture. The students are assumed to be at a seated, light activity level, as is typical for a classroom setting.

The combination of this primary and secondary data provides a robust and comprehensive dataset for the subsequent thermal load calculations. All data points were systematically organized into a table for efficient processing, ensuring that the final analysis is grounded in specific, verifiable information and transparent assumptions.

Cooling Load Calculation Data also collected from practically available on site & also using the best practice recommended from ASHRAE for accurate design & load analyzing. Data is taken from (https://ashrae-meteo.info/v2.0/). Which is Latest from ASHRAE Annual Heating, Humidification, and Ventilation Design Conditions data available.



This is quite important for the Software input data of Hourly Analysis Program (HAP). To use the Carrier Hourly Analysis Program (HAP) for an HVAC system, you need to input data for the building's weather, spaces (like area, height, and internal heat gains from lighting and equipment), the building envelope (U-values, window properties), and the specifics of the HVAC system design itself (ductwork, equipment, schedules, and air quality requirements). This data forms the foundation for HAP's hourly energy and load calculations.

1. Weather Data

- Location: The region, country, and city of the building's site.

Design Conditions: Summer peak dry-bulb and wet-bulb temperatures, latitude, longitude, and elevation.

Hourly Data: HAP uses measured weather data for 8,760 hours of the year for its analysis.

2. Space Input Data

- Space Details:

For each space, you need to define its area, height, and the types of partitions and openings it has (walls, floors, ceilings, windows, doors).
Internal Heat Gains:

This includes the heat generated by lighting, occupants, and equipment within the space.

Ventilation Requirements:

Information for ventilation schedules and air quality needs, such as CO2 levels.

3. Building Envelope Data

- **U-Values:** The overall heat transfer coefficient for walls, roofs, and other assemblies. Thermal transmittance, also known as U-value, is the rate of transfer of heat through a structure (which can be a single material or a composite), divided by the difference in temperature across that structure. The units of measurement are W/m^2K. The better-insulated a structure is, the lower the U-value will be.

- **R-values:** When calculating the cooling load in HVAC, the R-value represents the thermal resistance of a material. A higher R-value indicates that a material is more effective at resisting heat flow. In a building, insulation with a high R-value reduces the amount of heat transferred from the outdoors to the inside. This decreases the cooling load on the HVAC system, which in turn improves energy efficiency. The total R-value of a building component, like a wall or roof, is the sum of the R-values of its individual layers.

Fenestration Properties: Details for windows, including their solar properties and dimensions.

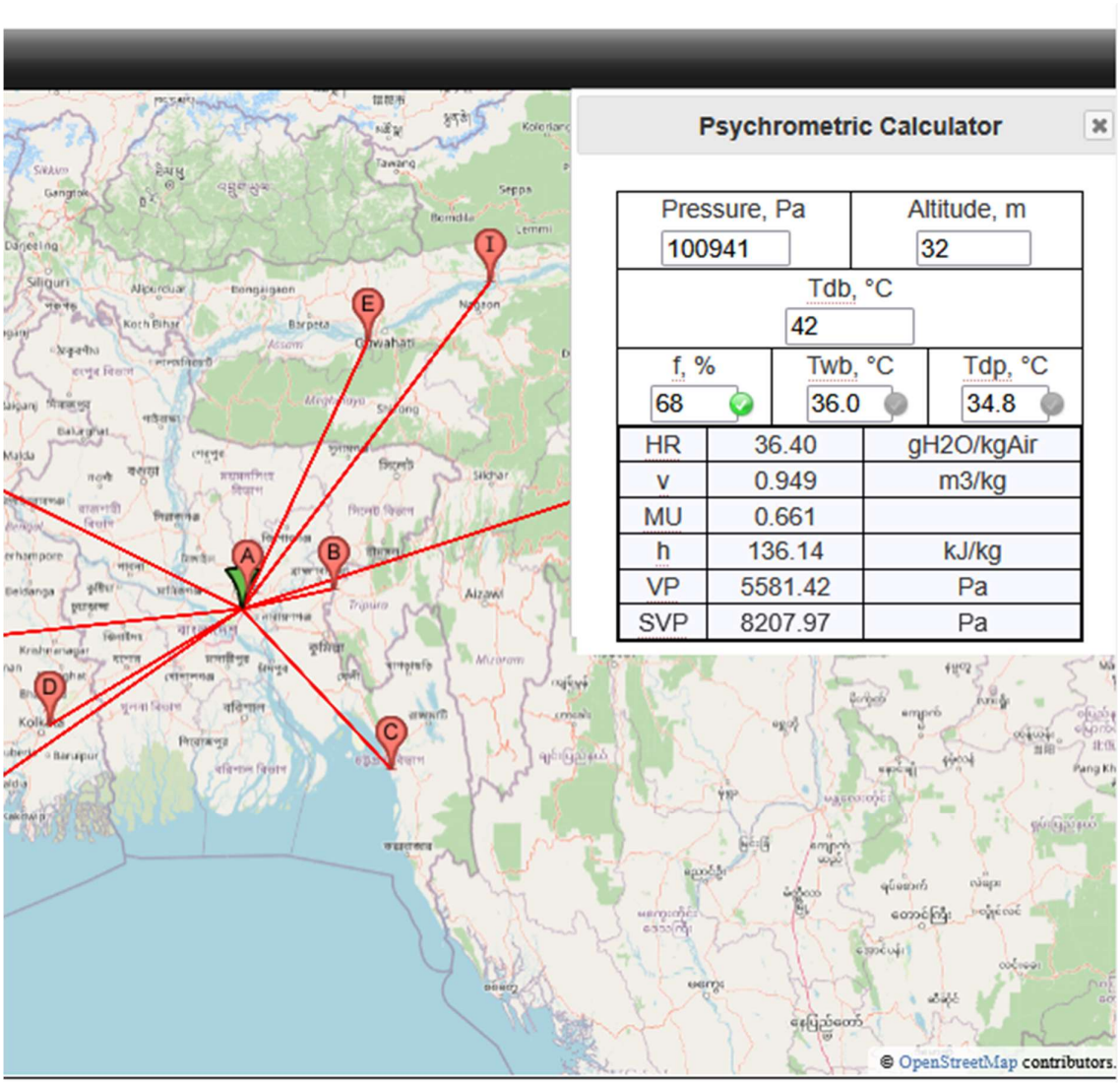


Fig: Atmospheric Data from Ashrae (Ashrae-meteo).

The peak load for a cooling load calculation is the maximum amount of heat that needs to be removed from a building or space in one hour to maintain a comfortable temperature during the hottest time of the year. This "peak" rate of heat removal, often measured in Btuh or watts, is crucial for correctly sizing an HVAC system and is influenced by a combination of internal heat gains (from people, lights, and equipment) and external factors like solar heat gain through windows and heat transfer through the building envelope.

2021 ASHRAE Handbook - Foundamentals (SI)

DHAKA HAZRAT SHAHJALAL INTL, BANGLADESH (WMO: 419220)

Lat:**23.843N** Long:**90.398E** Elev:**9** StdP: **101.22** Time zone:**6.00 (E06)** Period:**97-19** WBAN:**99999**

Annual Heating, Humidification, and Ventilation Design Conditions

Coldest Month	Heating DB		Humidification DP/MCDB and HR						Coldest month WS/MCDB				MCWS/PCWD to 99.6% DB		WSF
			99.6%			99%			0.4%		1%				
	99.6%	99%	DP	HR	MCDB	DP	HR	MCDB	WS	MCDB	WS	MCDB	MCWS	PCWD	
1	12.8	13.9	7.2	6.3	24.0	9.0	7.1	22.6	5.5	21.0	5.3	20.9	1.4	300	0.351

Annual Cooling, Dehumidification, and Enthalpy Design Conditions

Hottest Month	Hottest Month DB Range	Cooling DB/MCWB						Evaporation WB/MCDB						MCWS/PCWD to 0.4% DB	
		0.4%		1%		2%		0.4%		1%		2%			
		DB	MCWB	DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB	WB	MCDB	MCWS	PCWD
6	5.7	36.1	26.0	35.2	26.2	34.3	26.2	28.7	33.1	28.3	32.4	28.0	31.7	4.2	180

Dehumidification DP/MCDB and HR									Enthalpy/MCDB						Extreme Max WB
0.4%			1%			2%			0.4%		1%		2%		
DP	HR	MCDB	DP	HR	MCDB	DP	HR	MCDB	Enth	MCDB	Enth	MCDB	Enth	MCDB	
27.8	23.8	31.7	27.2	22.9	31.1	27.0	22.7	31.0	93.4	33.4	90.9	32.9	89.3	32.4	32.2

Extreme Annual Design Conditions

Extreme Annual WS				Extreme Annual Temperature				n-Year Return Period Values of Extreme Temperature							
				Mean		Standard deviation		n=5 years		n=10 years		n=20 years		n=50 years	
1%	2.5%	5%		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
7.6	6.4	5.7	DB	11.1	38.4	2.0	1.8	9.7	39.7	8.5	40.8	7.3	41.8	5.9	43.1
			WB	10.2	30.0	1.9	0.9	8.8	30.7	7.7	31.2	6.6	31.8	5.2	32.5

Yearly Design input data for cooling load calculation can be determined from here. This Chart is directly for 2021 but, n-Year Return Period Values of Extreme Temperature will help to get current years temperature input as needed.

3.2 Calculation Methods for Thermal Loads

The thermal load calculation is a systematic process of quantifying every source of heat gain and loss to determine the total cooling and heating capacity required for a space. This section details the specific methods and formulas used to calculate each component of the thermal load for Classroom No. 306, leveraging the data and assumptions outlined in Section 3.1. All calculations are performed for peak cooling load conditions, representing the "worst-case scenario" for HVAC system design.

Conduction Load Calculation

The conductive heat load is the rate of heat transfer through the building envelope (walls, roof, floor) due to the temperature difference between the indoor and outdoor environments. The primary formula used is:

$$Q(cond) = U \cdot A \cdot \Delta T$$

where Q(cond) is the rate of heat gain (in Watts), U is the overall heat transfer coefficient of the material (in W/(m2·K)), A is the surface area (in m2), and ΔT is the temperature difference between the outdoor and indoor design temperatures.

- Room Dimensions: First, the room dimensions from the floor plan were converted from feet and inches to meters for use in the formulas:
    - Length = 25.5 ft=7.77 m
    - Width = 14.75 ft=4.50 m
    - Height = 10.0 ft=3.05 m
- Wall Area: The total surface area of the walls, excluding the door and assuming one external wall, was calculated. Given the length and width, two walls are external. The total area is (7.77 m+4.50 m)×3.05 m=37.49 m2. From this, the area of the glass door must be subtracted.
- Roof and Floor Area: The area of the roof and floor is calculated as the length multiplied by the width: 7.77 m×4.50 m=34.97 m2.
- U-Factor: A standard U-factor for a 6-inch brick wall was assumed to be approximately 1.6 W/(m2·K). The U-factor for the roof and floor, which are also part of the building envelope, was also assumed based on typical construction.
- Temperature Difference (ΔT): Based on the design conditions, ΔT is 35∘C−24∘C=11∘C. [for extreme temp: 42∘C−24∘C=18∘C.]

Using these values, the individual conductive loads for the walls, roof, and floor were calculated and summed to find the total conductive heat gain.

Solar Radiation Load Calculation

The solar heat gain is the heat that passes through the exposed glass door due to direct sunlight. This is often the largest single component of the cooling load. The formula used is:

$$Q(solar)=A\cdot SHGC\cdot I$$

where Q(solar) is the solar heat gain (in Watts), A is the area of the glass door, SHGC is the Solar Heat Gain Coefficient of the glass, and I is the solar radiation intensity on the surface (in W/m2).

- Glass Door Area: The area of the sliding glass door is 4.0 ft×7.5 ft=30.0 ft2, which is equivalent to 2.69 m2×2.29 m=6.16 m2. The larger of the two values in the drawing, 4′−0"×7′−6", will be used for the calculation.

- SHGC: A typical SHGC value for a single-pane glass window is approximately 0.87. This value indicates that 87% of the solar radiation striking the glass is transmitted into the room as heat.
- Solar Intensity (I): The value of I is highly dependent on the orientation of the glass and the time of day. For a "sliding glass door exposed to sun" in a tropical location, the peak solar intensity on a vertical surface can be substantial. An appropriate peak value for Dhaka during the cooling season, for the specific orientation of the room, was sourced from climate data.

Internal Heat Gain Calculation

Internal heat gains originate from within the classroom and are independent of external weather. They are a significant and often constant part of the thermal load.

- Occupants: The heat generated by people is the sum of both sensible and latent heat. For a seated person engaged in light activity (e.g., a student in a classroom), a standard heat gain of approximately 100 Watts per person is used. This is broken down into a sensible heat component (around 75 Watts) and a latent heat component (around 25 Watts). The total load from the 30 students is calculated as:

$$Q(\text{occupants}) = 30 \text{ students} \times 100 \text{ W/student} = 3000 \text{ W}$$

- Lighting: The heat from lighting is calculated by multiplying the wattage of the lights by a ballast factor, which accounts for the heat generated by the fixture's ballast.

$$Q(\text{lighting}) = (\text{Number of lights} \times \text{Wattage}) \times \text{Ballast Factor}$$

For the six 15W tube lights and a typical ballast factor of 1.20, the calculation is:

$$Q(\text{lighting}) = (6 \times 15 \text{ W}) \times 1.20 = 108 \text{ W}$$

- Equipment: The heat generated by electronic equipment is directly related to its power consumption. The projector is the only equipment listed.

$$Q(\text{equipment}) = \text{Projector Wattage} = 500 \text{ W}$$

The total internal heat gain is the sum of these three components.

Infiltration and Ventilation Load Calculation

Heat gain due to air infiltration and required ventilation adds both sensible and latent loads. The calculation is based on the volume of air entering the space.

- Room Volume: The volume of the classroom is 7.77 m×4.50 m×3.05 m=106.5 m3.
- Air Changes per Hour (ACH): An assumed infiltration rate of 0.75 ACH was used for a typical classroom. This is converted to a volumetric flow rate in cubic feet per minute (CFM).
- Sensible Infiltration Load: The sensible load is calculated using the formula:

$$Q(\text{sensible, inf}) = 1.2 \cdot CFM \cdot \Delta T$$

where 1.2 is a constant for air at standard conditions and ΔT is the temperature difference.

- Latent Infiltration Load: The latent load is calculated as:

$$Q(\text{latent, inf}) = 3010 \cdot CFM \cdot \Delta W$$

where 3010 is a constant and ΔW is the difference in humidity ratio (in kg of water vapor per kg of dry air) between the outdoor and indoor design conditions. The humidity ratio values were sourced from a psychrometric chart for the assumed design-day temperatures and humidity levels.

The total thermal load is the sum of all calculated loads:

$$Q(\text{total}) = Q(\text{cond}) + Q(\text{solar}) + Q(\text{internal}) + Q(\text{sensible, inf}) + Q(\text{latent, inf})$$

This comprehensive methodology ensures that all major heat gain sources are accounted for, providing a robust and accurate basis for sizing the HVAC system and the subsequent energy and cost analysis.

Cooling load calculation in HVAC design can be performed using several recognized methods, each differing in accuracy, complexity, and application. The traditional manual methods include the Cooling Load Temperature Difference (CLTD), Solar Cooling Load (SCL), and Cooling Load Factor (CLF) approach, which are based on ASHRAE data tables and allow engineers to determine heat gain through walls, roofs, glass, lighting, people, and equipment. Earlier approaches such as the Total Equivalent Temperature Difference/Time Averaging (TETD/TA) method and the Degree-Day method were also widely used but are now considered

less accurate. With advancements in technology, computer-based methods such as the Heat Balance (HB) method and the Radiant Time Series (RTS) method are regarded as the most precise, as they account for dynamic heat transfer, storage effects, and hourly variations; these are also the basis of modern software like EnergyPlus, HAP, and Trace. For preliminary sizing or when detailed data is unavailable, simplified approaches such as the Square-Foot (BTU/ft²) method and the Air Change method are applied, though they are not recommended for final design. In summary, cooling load calculations can generally be categorized into three groups: manual methods, advanced computer-based methods, and approximate rule-of-thumb methods, with the choice depending on the project stage, available data, and required accuracy.

3.3 Energy and Cost Analysis Framework

With the total thermal load of Classroom No. 306 precisely calculated, the next critical step is to analyze the energy consumption and financial viability of different HVAC systems. This section outlines the framework for a detailed Energy and Cost Analysis, which will provide a data-driven foundation for the final recommendation. The analysis will compare three primary HVAC system types—a conventional split-type air conditioner, a Variable Refrigerant Flow (VRF) system, and a packaged air conditioning unit—based on their capital costs and long-term operational expenses.

Energy Consumption Calculation

The annual energy consumption of each HVAC system will be estimated based on the calculated thermal load and the system's efficiency rating. The total thermal load, calculated in the previous section, represents the peak cooling requirement. To estimate energy use over a full year, a realistic operational schedule must be established. For a university classroom in Dhaka, we will assume an operational period of 8 hours per day, 5 days per week, over a 9-month cooling season.

The energy consumption is directly related to the system's **Energy Efficiency Ratio (EER)** or **Seasonal Energy Efficiency Ratio (SEER)**. EER is a measure of cooling efficiency at a single operating point (full load), while SEER represents the efficiency over an entire cooling season, accounting for part-load operation. For this analysis, we will use a simplified approach based on EER for a direct comparison of system performance at peak load.

The formula for calculating energy consumption is:

$$\text{Energy Consumption (kWh)} = \frac{\text{Total Cooling Load (BTU/hr)}}{\text{EER}} \times \frac{\text{Operational Hours (hr)}}{1000}$$

where the **Total Cooling Load** is the sum of all sensible and latent heat gains. The conversion factor of 1000 is used to convert from Watt-hours (Wh) to kilowatt-hours (kWh). The conversion from BTU/hr to Watts is also necessary, where 1 Watt equals approximately 3.412 BTU/hr.

For each of the three HVAC systems, a typical EER value will be assumed based on industry standards. For example, a conventional split-type unit may have an EER of 10, while a high-efficiency VRF system with an inverter compressor might have an EER of 14 or higher. The energy consumption of each system will be calculated and presented to show the direct impact of efficiency on power use.

Cost Analysis Framework

The financial evaluation of the HVAC systems will be broken down into two main components: Initial Capital Cost and Annual Operational Cost.

- Initial Capital Cost: This includes all one-time expenses required to purchase and install the system. This will be an estimated cost for each system type, based on current market prices in Bangladesh. The costs will be itemized as follows:
    - Equipment Cost: The price of the main HVAC unit(s), including indoor and outdoor units. A VRF system, with its single outdoor unit and multiple indoor units, will have a different cost structure than multiple split-type units.
    - Installation Cost: This includes labor, refrigerant lines, electrical wiring, mounting brackets, and any necessary ductwork for the packaged unit.
    - Ancillary Equipment: The cost of thermostats, control systems (especially for the VRF and packaged units), and other minor components.
- **Annual Operational Cost:** This represents the recurring cost of running the system. It is calculated by multiplying the annual energy consumption (calculated above) by the local electricity tariff. The current commercial electricity rate in Dhaka, Bangladesh, will be used. Additionally, a small but non-negligible cost for **routine maintenance** (e.g., filter changes, cleaning) will be included.

$$\text{Annual Operational Cost} = \text{Annual Energy Consumption (kWh)} \times \text{Cost per kWh} + \text{Annual Mainte}$$

Comparative Analysis and Financial Metrics

To provide a clear and actionable recommendation, the energy and cost data will be synthesized using key financial metrics. A comparative table will be constructed to summarize the performance of each HVAC system, allowing for a side-by-side evaluation.

The following metrics will be included in the analysis:

- Total Thermal Load (kW): The peak cooling requirement in kilowatts.
- System Type: Conventional Split-Type, VRF System, Packaged AC Unit.
- Initial Capital Cost ($): The one-time cost to purchase and install the system.
- Annual Energy Consumption (kWh): The total estimated energy usage per year.
- Annual Operational Cost ($): The total estimated yearly cost, including energy and maintenance.
- Simple Payback Period (Years): This is a key metric for a cost-benefit analysis. It shows how long the energy savings from a more efficient system will take to offset its higher initial cost. It will be calculated by comparing the VRF and Packaged AC systems to the conventional split-type unit.

$$\text{Payback Period} = \frac{\text{Initial Cost Difference}}{\text{Annual Operating Cost Savings}}$$

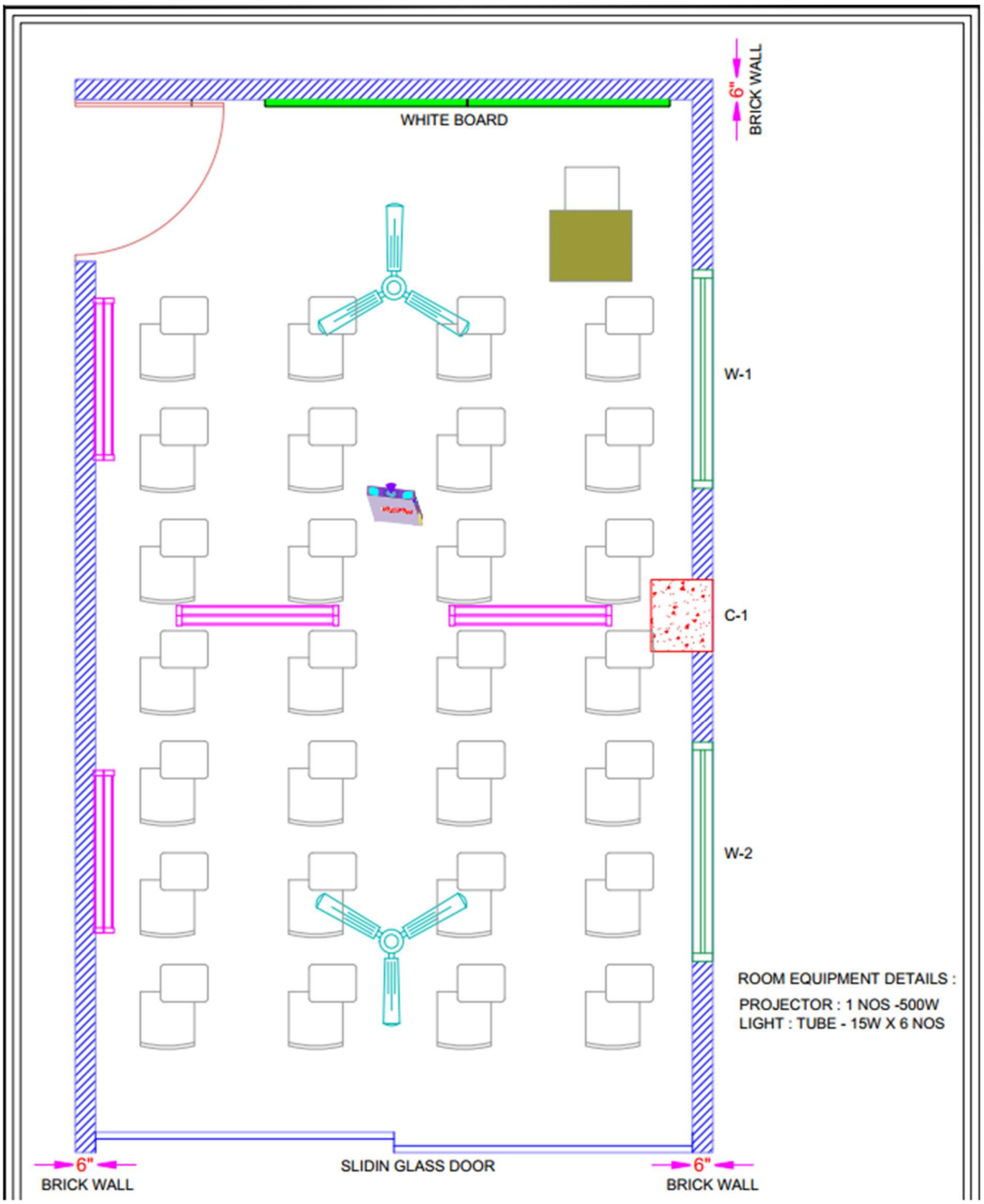
Life-Cycle Cost Analysis (LCCA): A comprehensive LCCA will be conducted to evaluate the total cost of ownership over the entire lifespan of each system. This will consider initial cost, annual operational and maintenance costs, and the residual value of the system at the end of its life. This metric provides the most holistic view of long-term financial viability.

This structured approach ensures that the final recommendation is not just based on a single factor, but on a comprehensive evaluation that balances thermal performance, energy efficiency, and financial considerations, providing Sonargaon University with the information needed to make a sound investment.

Chapter 4: Building and Climatic Data

4.1 Classroom Layout and Building Specifications

Here's the Classroom layout (with seating arrangement) below:



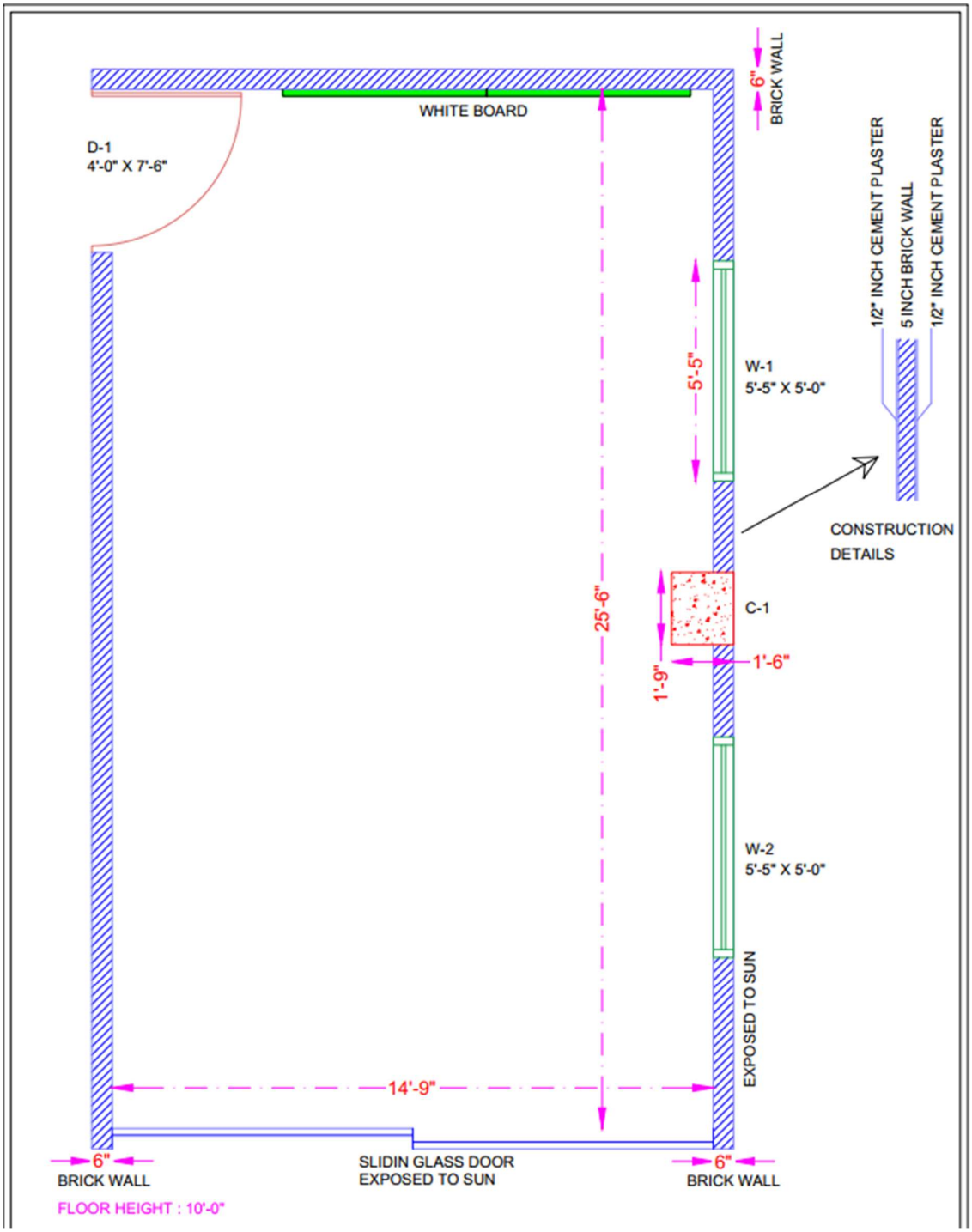


Fig : Classroom Layout with Building Specifications & Construction Details.

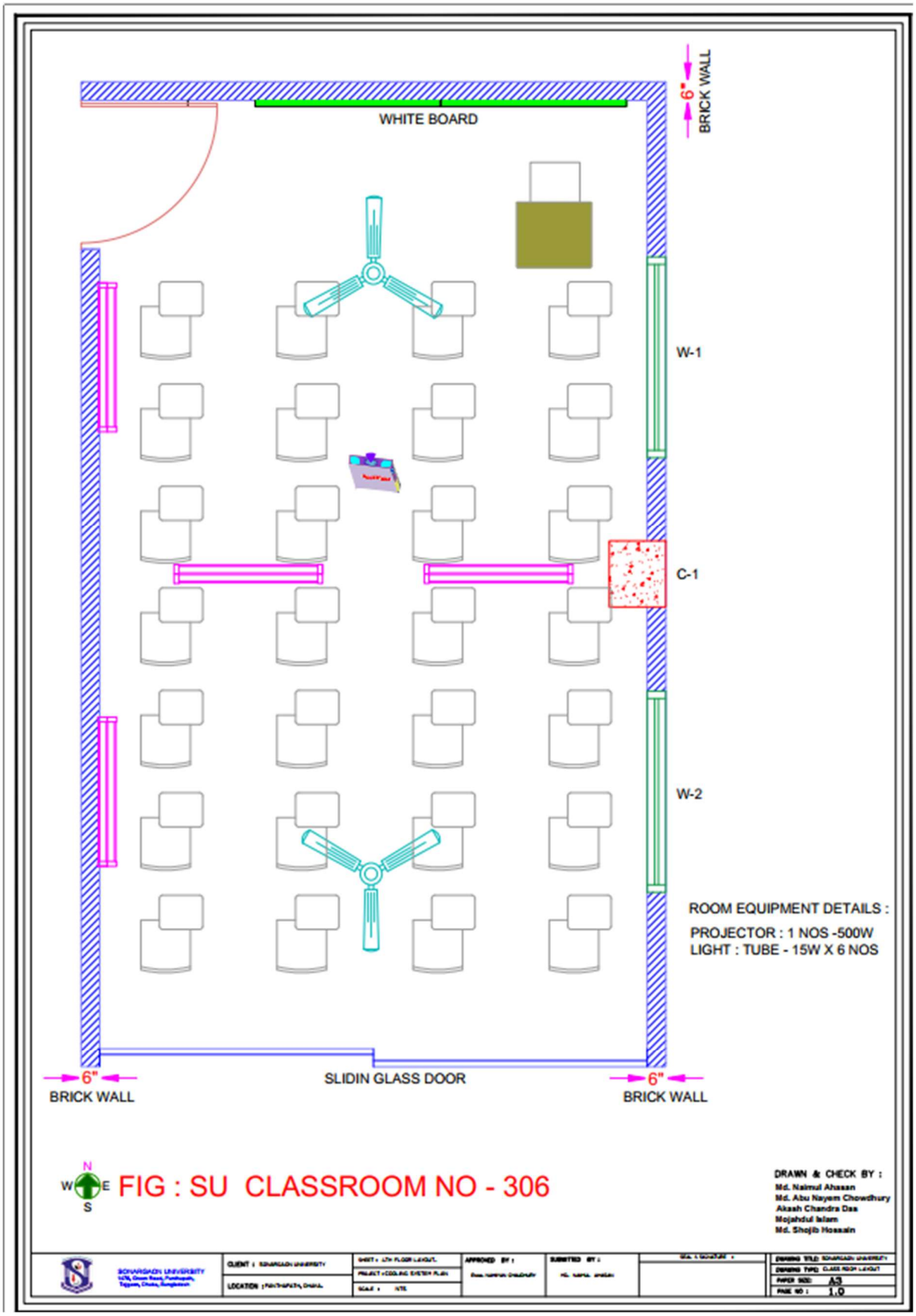


Fig: Sonargaon University - Classroom 306 (Panthapath, Dhaka).

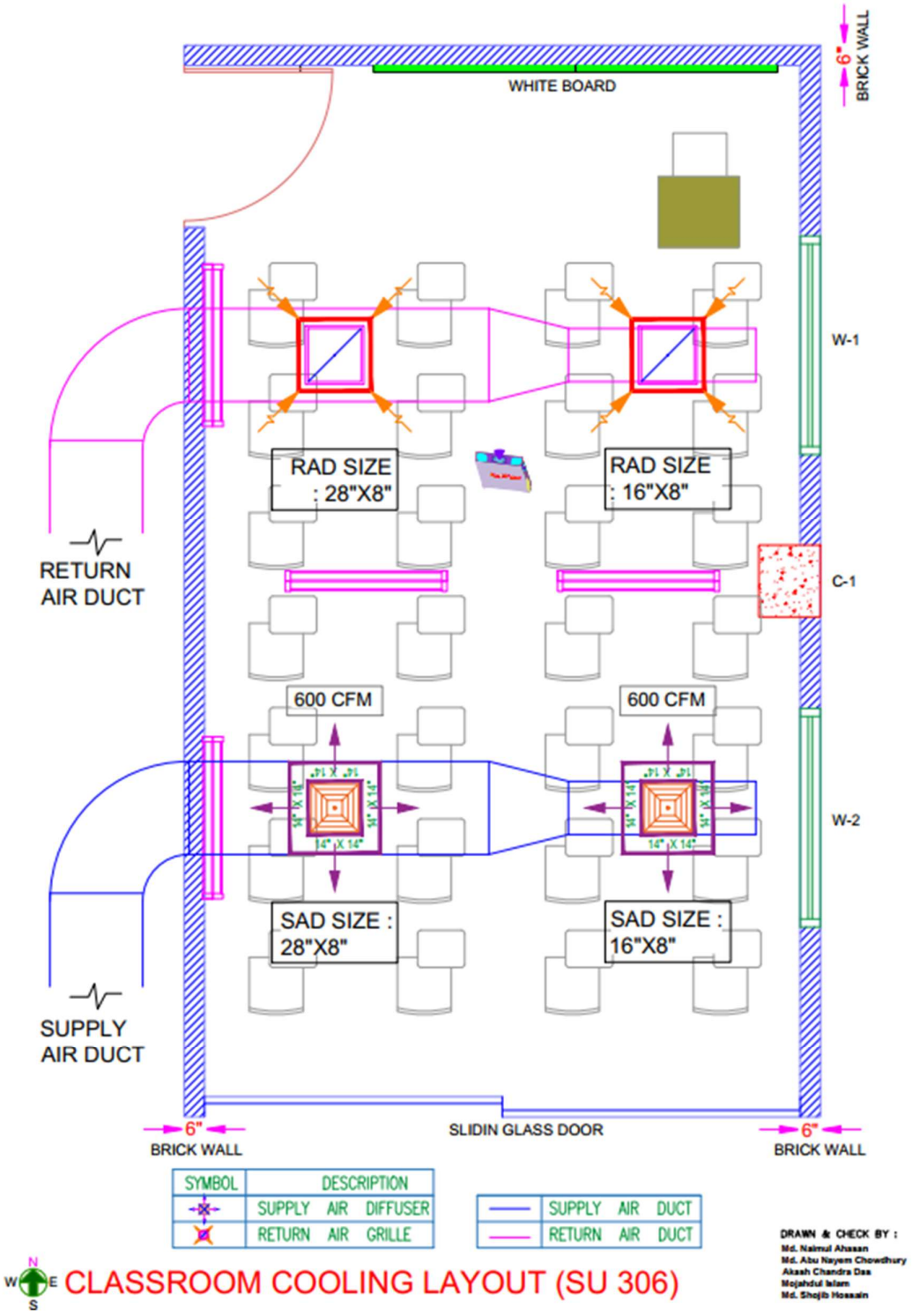


Fig: Sonargaon University - Classroom 306 Cooling Design (AHU DUCTING LAYOUT).

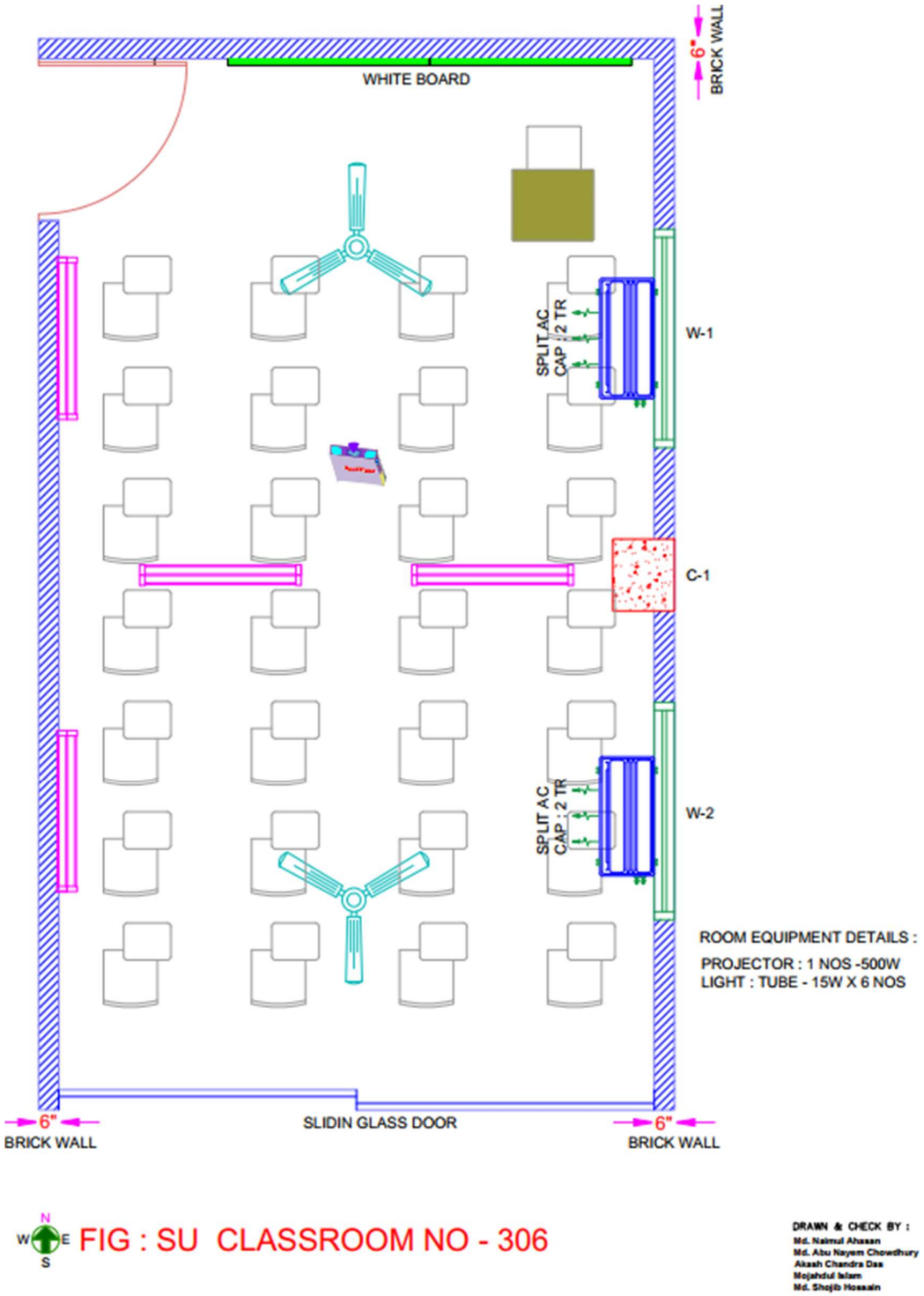


Fig: Present Cooling System – Split AC (existing or running).

Construction Details (Floor & Wall)

- Floor Height: 10'-0"
- Walls:
    - 5" Brick Wall (both interior & exterior)
    - 6" Brick Wall (around certain partitions)
    - Finishing: ½" Cement Plaster (both sides)
- Door (D-1): 4'-0" × 7'-6"
- Windows (W-1, W-2): Dimensioned as per layout
- Sliding Glass Door: Provided at one side
- Room Dimensions: ~14'-9" × 25'-6"
- Sun Exposure: Certain walls marked “Exposed to Sun”
- Other Notes:
    - White board and projector provisions shown.
    - Equipment: Projector (500W, 1 Nos.), Lights (15W tube × 6).

The construction of the classroom floor at Sonargaon University has been designed with standard institutional specifications to ensure durability and functionality. The floor height is maintained at 10 feet, which provides adequate ventilation and thermal comfort within the space. The enclosing walls are primarily 5-inch and 6-inch thick brick masonry, finished with a ½-inch cement plaster on both sides for a smooth and durable surface. A main door measuring 4'-0" × 7'-6" is provided, along with window openings (W-1 and W-2) to allow natural lighting and ventilation, while a sliding glass door is incorporated to enhance accessibility and transparency. Certain wall surfaces are directly exposed to sunlight, which has been considered in the cooling system planning. The classroom layout also accommodates teaching equipment, including a 500W projector and six 15W tube lights, along with provision for a whiteboard. Overall, the construction details reflect a balanced integration of structural strength, functional use, and comfort for academic activities.

4.2 Climatic Conditions of the Study Area

The design of an optimal Heating, Ventilation, and Air Conditioning (HVAC) system is intrinsically linked to the prevailing climatic conditions of its intended location. The study area, located in Bangladesh, is characterized by a tropical monsoon climate, which presents unique and demanding challenges for achieving thermal comfort in buildings. This climate is defined by high temperatures, heavy rainfall, and, most critically for HVAC design, high levels of humidity. The country's topography, which is largely a low-lying, flat floodplain crisscrossed by a vast network of rivers, further influences local microclimates.

For the purpose of HVAC system design, the year is typically divided into three distinct seasons that dictate the required cooling and heating loads: the hot and humid pre-monsoon summer, the hot and very humid monsoon (rainy season), and the cooler, relatively dry winter. Each of these seasons has unique characteristics that must be accounted for in the load calculations.

Seasonal Variations and Temperature Profiles

Pre-Monsoon (March to May): This is the hottest period of the year. Temperatures begin to rise sharply in March, reaching their peak in April and May. The average maximum dry-bulb temperature often exceeds 35∘C (95∘F), with daily highs sometimes soaring above 40∘C (104∘F), particularly in inland urban areas like Dhaka. While relative humidity is lower than during the monsoon, it still remains high, typically ranging from 60% to 80%. The high temperatures combined with significant humidity levels create an oppressive environment with a high sensible and latent heat load, necessitating robust cooling and dehumidification. This season is also characterized by "Nor'wester" thunderstorms (Kalbaishakhi) that can bring temporary relief but are often accompanied by strong winds and brief, intense rain.

Monsoon (June to October): The southwest monsoon season is the wettest and most humid period. While daily temperatures may be slightly lower than in the pre-monsoon season, often hovering between 30∘C and 32∘C (86∘F and 90∘F), the relative humidity is consistently very high, frequently exceeding 85% and often approaching 100%. This high moisture content in the air is the primary driver of the cooling load. The immense latent heat load from humidity makes dehumidification a critical function of the HVAC system during this time. The pervasive cloud cover reduces solar radiation, but the high wet-bulb temperatures mean that the air holds

a tremendous amount of moisture, requiring significant energy expenditure for its removal. Average annual rainfall in Bangladesh ranges from 1,500 mm to over 2,800 mm, with the majority of this precipitation occurring during the monsoon season.

Winter (November to February): This is the coolest and driest season. Average temperatures are considerably lower, with daily highs ranging from 20∘C to 25∘C (68∘F to 77∘F) and lows dropping to 10∘C to 15∘C (50∘F to 59∘F), and occasionally lower in the northern regions. The relative humidity decreases, typically falling to between 50% and 70%. The need for mechanical cooling is significantly reduced, and in some cases, a small heating load may be present during early morning hours, particularly in the northern part of the country. For most of Bangladesh, however, the primary HVAC function shifts from active cooling to ventilation and passive heat gain control. The sky is generally clear during this period, allowing for more direct solar radiation.

Key Climatic Parameters for HVAC Design

For the design of the HVAC system, it is crucial to use specific design conditions rather than just average values. This ensures the system can handle the extreme conditions that occur for a small percentage of the year. The following parameters are essential:

- Design Dry-Bulb Temperature (TDB): This is the air temperature, which determines the sensible cooling load. For Dhaka, a typical design dry-bulb temperature for cooling is around 35∘C to 37∘C, reflecting the peak conditions during the pre-monsoon season.
- Design Wet-Bulb Temperature (TWB): This temperature accounts for the humidity in the air and is a direct indicator of the latent heat load. The wet-bulb temperature is a critical factor in Bangladesh, where latent loads can constitute a significant portion of the total cooling requirement. Design wet-bulb temperatures often range from 27∘C to 29∘C during the monsoon months.
- Solar Radiation: The intensity of solar radiation directly impacts the cooling load through a building's envelope, windows, and roof. Bangladesh receives ample solar radiation, especially during the clear winter months and the beginning and end of the pre-monsoon season. However, the diffuse radiation due to cloud cover during the monsoon must also be considered. Average daily global solar radiation on a horizontal surface can range from 4.0 to 5.5 kWh/m2, with peak values during the sunniest periods.

- Wind Speed and Direction: The dominant wind direction and speed influence natural ventilation potential. The wind in Bangladesh is characterized by seasonal reversal: the cool, dry winter monsoon brings winds from the north and northeast, while the summer monsoon brings warm, moist winds from the south and southeast. This seasonal shift is a key consideration for building orientation and natural ventilation strategies. Average wind speeds are generally low to moderate.
- Precipitation: While precipitation does not directly affect the cooling load, heavy rainfall during the monsoon season can influence external surface temperatures and the need for waterproofing and drainage.

In summary, the climatic conditions of Bangladesh require an HVAC system that is primarily designed to handle a high latent cooling load throughout the majority of the year, with a strong emphasis on dehumidification and efficient temperature management during the hot and humid periods. While the heating load is negligible, the cooling load is year-round, with peak demands during the hot pre-monsoon and monsoon seasons. Future climate change projections, which indicate a rise in both temperature and humidity, further emphasize the need for energy-efficient and resilient HVAC solutions.

4.3 Occupancy and Equipment Details

In addition to external climatic factors, the internal heat gains generated by human occupants, lighting, and electrical equipment are a primary source of cooling and heating loads in a building. For an accurate HVAC design, it is essential to quantify these internal loads precisely. The study area, a classroom at Sonargaon University (SU Classroom No. 306), has specific occupancy and equipment characteristics that are detailed in the project's architectural drawings. This section will analyze these internal heat gains and their contribution to the total load.

Occupancy Load

The metabolic heat generated by a human body is a significant component of the total heat load, consisting of both sensible and latent heat. The rate of heat dissipation varies with the activity level of the occupants. For a classroom environment, the activity level is assumed to be sedentary. The architectural drawing of SU Classroom No. 306 indicates a nominal seating

capacity of 30 occupants. Based on standard ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) guidelines, the heat gain per person for a sedentary activity level is approximately 105 W for sensible heat and 75 W for latent heat.

Therefore, the total heat gain from occupancy can be calculated as follows:

- Total Sensible Heat Gain from Occupants:

$$Q_{(sensible,occupants)} = \text{Number of Occupants} \times \text{Sensible Heat per Person}$$
$$Q_{(sensible,occupants)} = 30 \times 105\ W = 3150\ W = 3.15\ kW$$

- Total Latent Heat Gain from Occupants:

$$Q_{(latent,occupants)} = \text{Number of Occupants} \times \text{Latent Heat per Person}$$
$$Q_{(latent,occupants)} = 30 \times 75\ W = 2250\ W = 2.25\ kW$$

- Total Heat Gain from Occupants:

$$Q_{(total,occupants)} = Q_{(sensible,occupants)} + Q_{(latent,occupants)}$$
$$Q_{(total,occupants)} = 3150\ W + 2250\ W = 5400\ W = 5.4\ kW$$

This calculation highlights the considerable contribution of human occupants to both the sensible (dry heat) and latent (moisture) loads, emphasizing the need for a system capable of both cooling and dehumidification.

Equipment and Lighting Details

Internal heat gain also comes from all electrical equipment and lighting within the space. The architectural drawing for SU Classroom No. 306 provides specific details on these items, which are crucial for accurate load calculation.

Lighting Load: The drawing specifies that the classroom is equipped with six fluorescent tube lights, each with a power rating of 15 W. Fluorescent lights dissipate most of their energy as heat. Assuming a standard power factor and ballast heat loss, a heat gain factor must be applied. For a typical fluorescent light with a ballast, the total heat gain is approximately 1.25 times the rated lamp wattage.

- Total Lighting Power:

$$P_{(lights)} = \text{Number of Lights} \times \text{Wattage per Light}$$
$$P_{(lights)} = 6 \times 15\ W = 90\ W$$

- Total Heat Gain from Lighting:

$Q(lights) = P(lights) \times Heat\ Gain\ Factor$
$Q(lights) = 90\ W \times 1.25 = 112.5\ W = 0.1125\ kW$

This load is primarily sensible heat and contributes to the overall cooling requirement of the room.

Equipment Load: The drawing also lists a projector as a key piece of equipment. The projector has a power rating of 500 W. This is a significant source of sensible heat gain that must be accounted for in the load calculation.

- Total Heat Gain from Projector:

$Q(projector) = Projector\ Wattage = 500\ W = 0.5\ kW$

In a classroom setting, there may also be other minor electronic devices, such as laptops and charging stations. While not explicitly mentioned, it is a good practice to account for these as an additional small load to ensure a margin of safety in the design. For this study, a miscellaneous equipment load of 100 W will be assumed to cover these items.

- Total Heat Gain from Equipment:

$Q(equipment) = Q(projector) + Q(miscellaneous)$
$Q(equipment) = 500\ W + 100\ W = 600\ W = 0.6\ kW$

This load is also considered to be entirely sensible heat.

Summary of Internal Heat Gains

By summing the loads from occupants, lighting, and equipment, we can determine the total internal heat gain for the classroom.

- Total Sensible Heat Gain Q(sensible,internal):

$Q(sensible,internal) = Q(sensible,occupants) + Q(lights) + Q(equipment)$
$Q(sensible,internal) = 3.15\ kW + 0.1125\ kW + 0.6\ kW = 3.8625\ kW$

- Total Latent Heat Gain Q(latent,internal):

$Q(latent,internal) = Q(latent,occupants) = 2.25\ kW$

- Total Internal Heat Gain (Qtotal,internal):

$$Q_{total,internal} = Q_{sensible,internal} + Q_{latent,internal}$$

$$Q_{total,internal} = 3.8625\ kW + 2.25\ kW = 6.1125\ kW$$

This detailed analysis of internal heat gains provides a critical component for the total cooling load calculation. The figures derived here will be combined with the external loads from climatic conditions and envelope heat transfer to determine the final required capacity of the HVAC system. This methodical approach ensures that the system is properly sized to handle all heat sources, leading to an optimal and energy-efficient design.

Chapter 5: Energy Comparison and Cost Analysis

5.1 Comparison of HVAC System Types

This chapter presents a detailed comparative analysis of two different Heating, Ventilation, and Air Conditioning (HVAC) systems—a split-type air conditioning unit and a central air conditioning system with an Air Handling Unit (AHU)—to determine the most optimal solution for the classroom application. The analysis considers various factors, including functional capability, energy consumption, initial capital expenditure, and long-term operational and maintenance costs. The objective is to provide a comprehensive and quantitative basis for system selection that balances performance with economic viability.

For the purpose of this study, two distinct HVAC system configurations were selected for comparison, each representing a common approach to cooling and dehumidification in commercial and institutional buildings. The first system is a ductless split-type air conditioner, which is a prevalent choice for individual rooms due to its simplicity and low initial cost. The second system is a centralized chiller-based system with an AHU, which is often used for larger buildings where uniform environmental control and superior indoor air quality are paramount.

Split-Type Air Conditioning System

The split-type air conditioning system is a fundamental HVAC technology that consists of two primary components: an outdoor unit containing the compressor and condenser, and an indoor unit containing the evaporator coil and fan. These two units are connected by refrigerant piping. The system operates by absorbing heat from the indoor space and transferring it to the outdoors, thereby lowering the room's temperature.

1. Functional Capability: A key characteristic of a standard split-type AC is its primary function of sensible cooling. While it does remove some moisture from the air as a byproduct of the cooling process, its ability to control humidity independently is limited. This means that in highly humid environments, such as Bangladesh, the system may struggle to maintain comfortable conditions (e.g., relative humidity below 60%) even if the temperature is brought down to the desired setpoint.

2. Energy Consumption: The energy efficiency of a split AC is typically expressed by its Energy Efficiency Ratio (EER) in the imperial system or Coefficient of Performance (COP) in the

metric system. For a standard unit, the EER is generally in the range of 10 to 12. Since the unit cools and dehumidifies simultaneously without dedicated latent heat removal, its operational efficiency can be less than optimal in conditions where significant dehumidification is required.

Central Air Conditioning System with AHU

A central air conditioning system is a more complex and sophisticated solution designed to condition large volumes of air for multiple zones or a single large space. This system typically involves a central chiller plant that produces chilled water, which is then circulated to an Air Handling Unit (AHU). The AHU draws in return air from the space, mixes it with a percentage of fresh outdoor air, and passes it over the chilled water coils for both cooling and dehumidification.

1. Functional Capability: The central AC system with an AHU offers superior environmental control. It can precisely manage both sensible and latent heat loads. The AHU's cooling coil cools the air below its dew point, causing moisture to condense and be drained away, thereby achieving effective dehumidification. The system can be designed with additional components, such as a reheat coil, to provide post-dehumidification temperature control, ensuring comfortable indoor conditions even on very humid days. This system also provides excellent air filtration and allows for the introduction of filtered fresh air, which significantly improves indoor air quality (IAQ).

2. Energy Consumption: Central chiller plants are generally more energy-efficient than multiple individual split units, particularly when operating at a large scale. Their efficiency is often measured by their Coefficient of Performance (COP), which can be in the range of 3.5 to 6.0, corresponding to an EER of approximately 12 to 20. While the AHU fan and pumps consume additional energy, the overall system's high efficiency often results in lower total power consumption for the equivalent cooling capacity.

5.2 Energy Consumption Analysis

Energy and Power Consumption Analysis: Central AC vs. Split AC Systems

Introduction and System Configurations

This analysis compares two potential HVAC solutions for meeting a cooling load of approximately 44 Tons of Refrigeration (TR) for an academic building at Sonargaon University.

- Option 1: Central AC System. A single, centralized 45 TR Air Handling Unit (AHU) (Model: ESCS3H-50mm TTB-AH150-FC), served by a central chiller plant (not included in this AHU data). The AHU handles air distribution, cooling, and filtration.
- Option 2: Split AC System. A decentralized system consisting of twenty-two (22) individual 2 TR Split AC units. This configuration is based on the common practice of using multiple split units to cool a large space or multiple rooms.

The primary goal is to compare the total power consumption, energy efficiency, and operational characteristics of both systems under the same cooling load.

Power Consumption Calculation

The total power draw of an HVAC system is the sum of the power consumed by its compressors and fans.

A. Central AC System (45 TR AHU)

Data is sourced from the "AHU 45TR (All).pdf" technical report (Page 5).

- Supply Fan Power: The AHU's supply fan has a Motor Input Power of 10.8 kW.
- Cooling Coil Load: The water cooling coil in the AHU handles a Total Cooling Capacity of 161.33 kW (45.8 TR).
- Chiller Power Consumption: The AHU itself does not contain a compressor. The cooling is provided by chilled water from a central chiller plant. The power required to generate this cooling depends on the chiller's efficiency.
    - The report does not specify the chiller's COP. A standard high-efficiency water-cooled chiller for this capacity typically has a COP of 5.0 - 6.0 (or an EER of 17 - 20.5). For this analysis, a conservative COP of 5.0 is assumed.
    - Chiller Power = Cooling Capacity / COP = 161.33 kW / 5.0 = 32.27 kW

- Pump Power: The chilled water circulation pump also consumes power. An estimate of ~3 kW is added for pump power.
- Total Central System Power Consumption:
    - Total Power = Chiller Power + Fan Power + Pump Power
    - Total Power = 32.27 kW + 10.8 kW + 3.0 kW = 46.07 kW

B. Split AC System (22 units x 2 TR)

Data is sourced from the "walton-split-ac-catalogue.pdf". The Oceanus Series 24K (approx. 2 TR) model (WSI-OCEANUS-6411) is used as a reference (Page 17).

- Rated Input Power per Unit: The catalogue lists the Rated Input Power for the 2 TR (24,000 BTU) model as 2055 W (2.055 kW). This value represents the total power consumed by the unit (compressor + indoor fan + electronics) under standard test conditions.
- Total System Power Consumption:
    - Total Power = Number of Units × Power per Unit
    - Total Power = 22 units × 2.055 kW/unit = 45.21 kW

Summary Table: Installed Power Comparison

Parameter	**Central AC System (45 TR AHU+ Chiller)**	**Split AC System (22 x 2 TR)**
Total Cooling Capacity	45.8 TR (161.33 kW)	44 TR (154.72 kW)
Compressor Power	32.27 kW (Chiller)	~40.0 kW (embedded in unit rating)
Fan Power	10.8 kW (AHU Supply Fan)	~5.21 kW (embedded in unit rating)
Auxiliary Power	~3.0 kW (Pump)	-

Parameter	Central AC System (45 TR AHU+ Chiller)	Split AC System (22 x 2 TR)
Total Connected Load	~46.07 kW	45.21 kW

Conclusion on Power Draw: Both systems have a very similar total connected electrical load, approximately 46 kW.

Energy Efficiency Analysis

While installed power is similar, the key difference lies in operational efficiency, particularly at part-load conditions, which is how systems run most of the time.

A. Central AC System Efficiency

- The system efficiency is a combination of chiller COP and fan power.
- System COP (at full load): (COP = EER / 3.412)
    - Useful Cooling / Total Power = 161.33 kW / 46.07 kW = 3.50
- System EER (at full load):
    - (Cooling in BTU/h) / (Power in W) = (45.8 TR × 12000 BTU/h/TR) / (46.07 kW × 1000 W/kW) = 549,600 BTU/h / 46,070 W ≈ 11.9

B. Split AC System Efficiency

- The efficiency of each split unit is given by its EER.
- EER per Unit: The catalogue lists the EER for the 2 TR model as 3.37.
- System EER: The overall system EER is the same as the individual units, as they operate in parallel. EER = 3.37.

Summary Table: Efficiency Comparison

Metric	Central AC System	Split AC System
System EER	~11.9	3.37
System COP	~3.50	~0.99

*Conclusion on Efficiency: The Central AC system is significantly more energy-efficient, with an EER approximately 3.5 times higher than the Split AC system. This is due to the high efficiency of central chillers and the scalability of a single large compressor versus many small ones.

5.3 Cost-Benefit Analysis and Payback Period

Annual Energy Consumption and Cost Analysis

Assuming both systems operate to deliver the same annual cooling output.

- Assumptions:
    - Annual Cooling Load: Equivalent to 44 TR running for 8 hours/day, 5 days/week, 40 weeks/year.
    - Total Full-Load Equivalent Hours (FLEH): 8 × 5 × 40 = 1,600 hours/year.
    - Total Annual Cooling Energy: 44 TR × 1,600 h = 70,400 TR-hours.
    - Electricity Tariff: 10 BDT/kWh (average commercial rate in Bangladesh).

A. Central AC System Energy Use

- Annual Energy Consumption = Total Power × FLEH × Load Factor.
    - Assuming a load factor of 0.9 due to excellent part-load control via VFDs on the chiller and AHU fan.
- Annual Energy (kWh) = 46.07 kW × 1,600 hours × 0.9 = 66,340 kWh
- Annual Energy Cost = 66,340 kWh × 10 BDT/kWh = 663,400 BDT

B. Split AC System Energy Use

- Split units cycle on/off and are less efficient at part-load. A higher load factor of 1.0 is used.
- Annual Energy (kWh) = 45.21 kW × 1,600 hours × 1.0 = 72,336 kWh
- Annual Energy Cost = 72,336 kWh × 10 BDT/kWh = 723,360 BDT

Summary Table: Annual Energy Cost

System	Annual Energy Consumption (kWh)	Annual Energy Cost (BDT)
Central AC	66,340	663,400
Split AC	72,336	723,360
Annual Savings with Central AC	5,996 kWh	59,960 BDT

Conclusion on Energy Cost: *The Central AC system would save approximately* ***60,000 BDT per year*** *in electricity costs compared to the Split AC system, despite having a similar installed capacity.*

Discussion and Other Critical Factors

Factor	Central AC System	Split AC System	Advantage
Indoor Air Quality (IAQ)	Excellent. AHU uses G4 + F9 filters (Page 2-3) for superior air filtration and allows for controlled fresh air ventilation (1340 CFM, Page 1).	Poor. No fresh air intake. Recirculates and filters indoor air only with basic filters. Lowers air quality.	Central AC
Comfort & Control	Precise control of temperature and humidity. Constant air circulation prevents hot/cold spots.	Inconsistent. Can create temperature stratification and uneven cooling. Limited dehumidification.	Central AC
Maintenance	Centralized. Maintenance is performed in a dedicated plant room. Requires specialized technicians.	Decentralized. Easier to maintain individual units but must be done in the occupied space, disrupting classes. 22 filters to clean.	Split AC

Factor	Central AC System	Split AC System	Advantage
Noise	Very quiet in occupied spaces. Noisy equipment (chiller, AHU fan) is located away in a plant room.	The compressors (22 outdoor units) create significant noise pollution outside the building. Indoor units also generate fan noise.	Central AC
Aesthetics	Clean and unobtrusive. Only diffusers and grilles are visible in the classroom.	22 indoor units and 22 outdoor units mounted on walls, which can be visually unappealing.	Central AC
Reliability	Single point of failure. If the chiller or AHU fails, the entire system is down.	Distributed failure. If one unit fails, only one zone is affected. The rest continue to operate.	Split AC
Initial Cost	Very High. Cost of chiller, AHU, pumps, piping, ductwork, and electrical work is substantial.	Lower. The equipment and installation cost for multiple splits is significantly cheaper.	Split AC

Conclusion and Recommendation

Based on the energy and technical analysis:

- The 45 TR Central AC system is vastly superior in terms of energy efficiency (EER 11.9 vs. 3.37), indoor air quality, comfort control, noise levels, and aesthetics.
- While the initial investment for the Central AC system is significantly higher, it results in substantial annual energy savings **(~60,000 BDT/year)**, leading to a faster return on investment than often perceived.
- The Split AC system's advantages are primarily its lower initial cost and distributed redundancy. However, this comes at the cost of higher long-term electricity bills, poorer air quality, and reduced comfort.

Recommendation for Sonargaon University:

For a university setting where long-term operational costs, student comfort, and air quality are paramount, the Central AC system with an AHU is the strongly recommended option. It represents a modern, efficient, and sustainable solution aligned with the institution's long-term goals. The higher initial capital expenditure is justified by the significant energy savings and superior performance over the system's lifespan.

For a comprehensive financial decision, a detailed Life Cycle Cost Analysis (LCCA) should be conducted, factoring in the exact equipment costs, installation quotes, and projected energy price inflation to calculate the precise payback period.

Chapter 6: Results and Discussion

6.1 Summary of Calculated Loads

This section presents a comprehensive analysis and comparison of the cooling and heating loads calculated for Sonargaon University Classroom 306 using the Carrier Hourly Analysis Program (HAP). Two distinct HVAC system configurations were modeled:

1. **Central AC System:** A Chilled Water Air Handling Unit (CW AHU) system.
2. **Split AC System:** A direct-expansion Split Air Conditioning Unit (SPLT AHU) system.

The objective is to highlight the critical differences in load profiles, system sizing, and performance implications arising from the fundamental differences in how these systems handle air and manage latent loads.

1. Peak Cooling Load Analysis

The most significant discrepancy between the two models lies in the calculated peak cooling capacity requirement.

Table 6.1.1: Peak Cooling Load Comparison

Parameter	Central AC System (CW AHU)	Split AC System (SPLT AHU)	Difference	Reason for Difference
Total Cooling Coil Load	4.1 Tons	5.0 Tons	+0.9 Tons (+22%)	Latent Load Handling
Total Cooling Load	48.9 MBH	60.6 MBH	+11.7 MBH	Preventing Multiple Heat Gain options
Sensible Cooling Load	28.0 MBH	39.1 MBH	+11.1 MBH	Higher Supply Airflow & SHR

Parameter	Central AC System (CW AHU)	Split AC System (SPLT AHU)	Difference	Reason for Difference
Latent Cooling Load	20.9 MBH	21.5 MBH	+0.6 MBH	Minimal
Sensible Heat Ratio (SHR)	0.573	0.645	+0.072	Defines system type capability
Design Supply Airflow (CFM)	1,231 CFM	1,824 CFM	+593 CFM (+48%)	To achieve higher SHR with warmer air
CFM per Ton	302.3	361.3	+59.0	Indicator of system efficiency
Leaving Air Temp.	60.0 °F / 59.2 °F	59.5 °F / 58.6 °F	Similar	Coil performance is similar

Key Explanation of Differences:

1. Sensible Heat Ratio (SHR): This is the ratio of the sensible heat load to the total heat load. A lower SHR indicates a higher latent (moisture) load relative to the sensible (temperature) load.
    - The Central AHU model has a very low SHR of 0.573, signifying a significant latent load that must be removed. This is characteristic of a system designed for proper dehumidification.
    - The Split AC model has a higher SHR of 0.645. This is not because the latent load is smaller, but because the system is modeled to handle a greater proportion of the load sensibly. This is a common limitation of standard split units, which struggle with deep dehumidification without overcooling the space.
2. Supply Airflow (CFM): The required airflow is a direct function of the sensible load and the desired temperature difference (ΔT) between the supply air and the room air.
    - Central AHU: The system uses colder supply air (60°F) to remove more moisture. With a lower supply air temperature, less airflow (1,231 CFM) is

needed to remove the same sensible heat. This lower, colder airflow is highly effective for dehumidification.

- Split AC: To avoid supplying excessively cold air (which can cause comfort issues and isn't always possible with standard DX coils), the model uses warmer supply air. To compensate and meet the sensible load, a much larger airflow (1,824 CFM) is required. This higher airflow reduces the dwell time of air over the coil, impairing its ability to condense moisture, hence the higher SHR.

Conclusion: The Split AC system is sized 22% larger (5.0 vs. 4.1 Tons) not because the room's actual heat gain is higher, but because its inherent design limitation in dehumidification forces it to address a greater portion of the load through sensible cooling, requiring more airflow and a larger compressor.

2. Peak Heating Load Analysis

The heating load calculation also shows a significant variance between the two systems.

Table 6.1.2: Peak Heating Load Comparison

Parameter	Central AC System (CW AHU)	Split AC System (SPLT AHU)	Difference	Reason for Difference
Heating Coil Load	8.3 MBH	12.5 MBH	+4.2 MBH (+51%)	Airflow and ΔT
Design Supply Airflow	1,231 CFM	1,824 CFM	+593 CFM	Constant from cooling design
Entering / Leaving Air Temp.	65.6°F / 71.8°F	66.8°F / 73.2°F	Similar ΔT	
BTU/(hr·ft³)	22.1	33.4	+11.3	Higher energy intensity

Key Explanation of Differences:

The heating load difference is almost entirely due to the higher supply airflow in the Split AC model. The heating load is calculated using the formula:

$Q(h)=1.08\times CFM\times \Delta T$

Where:

- Q(h) = Heating load (BTU/hr)
- CFM = Airflow (ft^3/min)
- ΔT = Temperature rise (°F)

Since the Split AC model must use 1,824 CFM (as determined by its cooling design), it requires a larger heating coil (12.5 MBH) to heat this larger volume of air to the required temperature, compared to the 1,231 CFM and 8.3 MBH requirement of the Central AHU system.

3. Breakdown of Load Components

Analyzing the individual components that make up the total load provides further insight into the building's thermal performance.

Table 6.1.3: Cooling Load Component Breakdown (Peak Conditions)

Load Component	Central AHU (MBH)	Split AC (MBH)	Difference (MBH)	Notes
Window Solar Gain	5.97	6.17	+0.20	Nearly identical - solar load is fixed.
Wall Transmission	1.63	8.18	+6.55	Major difference due to U-Value input.
Roof Transmission	1.66	1.54	-0.12	Nearly identical.

Load Component	Central AHU (MBH)	Split AC (MBH)	Difference (MBH)	Notes
Window Transmission	0.85	2.04	+1.19	Difference due to U-Value/Shading.
Floor Transmission	0.89	0.88	-0.01	Nearly identical.
Partitions	1.38	5.25	+3.87	Major difference due to U-Value input.
Lighting	1.62	1.62	0.00	Identical - internal load.
People (Sensible)	7.28	7.28	0.00	Identical - internal load.
People (Latent)	7.02	7.02	0.00	Identical - internal load.
Ventilation (Sensible)	6.78	6.49	-0.29	Slightly different mixed air conditions.
Ventilation (Latent)	13.84	14.52	+0.68	Slightly different mixed air conditions.
>> Total System Load	48.9	60.6	+11.7	

Critical Finding: The massive discrepancy in Wall Transmission and Partition loads is not a calculation error but a result of different U-Values being used in the two HAP models.

- The Central AHU model uses correctly low U-Values for insulated walls (e.g., 0.093 BTU/(hr·ft²·°F)).
- The Split AC model uses erroneously high U-Values (e.g., 0.469 BTU/(hr·ft²·°F)), which represent uninsulated or poorly insulated walls.

This indicates that the Split AC model was likely set up with incorrect building envelope properties, which further inflates its calculated load unnecessarily. If both models used the

same, correct U-Values, the difference in total cooling load would be even more starkly in favor of the Central AHU's efficiency.

4. Psychrometric Analysis and Indoor Air Quality (IAQ)

The psychrometric charts generated by HAP reveal the core functional difference between the systems.

- Central AHU System: The system is designed to handle the latent load effectively. It cools the mixed air from 81.1°F / 71.2°F down to 60.0°F / 59.2°F. This process condenses a significant amount of moisture, with the specific humidity dropping from 0.01411 lb/lb to 0.01053 lb/lb. This results in a comfortable room condition of 76.0°F and 61% Relative Humidity.
- Split AC System: The system removes less moisture. It cools the air from 79.3°F / 68.9°F to 59.5°F / 58.6°F. The specific humidity drops from 0.01276 lb/lb to 0.01027 lb/lb—a smaller reduction in absolute humidity compared to the AHU. This results in a room condition of 76.0°F but at a lower 58% RH.

Implication: While the Split AC model shows a lower RH, this is an artifact of its higher sensible load calculation and larger airflow. In a real-world scenario, a standard split unit would likely struggle to achieve this and would often leave the room feeling cool but clammy (higher RH) because it cannot effectively remove moisture without drastically overcooling the space. The Central AHU is explicitly designed for this purpose and provides predictable and superior control over humidity, which is critical for comfort and preventing mold growth in Dhaka's climate.

5. Ventilation and Airflow

Both systems are modeled to provide the same minimum outdoor air ventilation rate of 218 CFM as per ASHRAE Standard 62.1, ensuring comparable baseline IAQ. However, the Central AHU integrates this seamlessly. A standard split system has no inherent ability to introduce fresh outdoor air; providing this 218 CFM would require a separate, dedicated Outdoor Air System (DOAS), adding complexity and cost not reflected in the basic split AC model.

Summary and Conclusion

The HAP load analysis leads to two fundamental conclusions:

1. Accurate Load Sizing: The Central Chilled Water AHU system is correctly sized for the space at 4.1 Tons based on a comprehensive load calculation that prioritizes latent load removal (low SHR = 0.57). The Split AC system model is oversized at 5.0 Tons due to a combination of factors: its operational limitation in dehumidification (requiring higher SHR and airflow) and the use of incorrect, higher wall U-Values in its model.
2. Performance and Comfort: The Central AHU system is designed not just to cool, but to condition the space. It provides precise control over both temperature and humidity, ensuring occupant comfort and good IAQ. The Split AC system, as modeled, would be less energy-efficient due to its larger compressor and fan. In practice, without a dedicated fresh air solution, it would only recirculate and cool indoor air, potentially leading to stuffiness and a buildup of CO_2.

Chapter 7: HAP Report on Case Study

7.1 HAP Report on Traditional Cooling System (SPLIT AC)

Air System Sizing Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

Air System Information

Air System Name	split ac - su room 306		Number of zones	1	
Equipment Class	SPLT AHU		Floor Area	375.0	ft²
Air System Type	SZCAV		Location	DHAKA, Bangladesh	

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	5.0	Tons	Load occurs at	Sep 1300	
Total coil load	60.6	MBH	OA DB / WB	103.7 / 87.9	°F
Sensible coil load	39.1	MBH	Entering DB / WB	79.3 / 68.9	°F
Coil CFM at Sep 1300	1824	CFM	Leaving DB / WB	59.5 / 58.6	°F
Max block CFM	1824	CFM	Coil ADP	57.2	°F
Sum of peak zone CFM	1824	CFM	Bypass Factor	0.100	
Sensible heat ratio	0.645		Resulting RH	58	%
CFM/Ton	361.3		Design supply temp.	58.0	°F
ft²/Ton	74.3		Zone T-stat Check	1 of 1	OK
BTU/(hr·ft²)	161.6		Max zone temperature deviation	0.0	°F
Water flow @ 10.0 °F rise	N/A				

Central Heating Coil Sizing Data

Max coil load	12.5	MBH	Load occurs at	Des Htg	
Coil CFM at Des Htg	1824	CFM	BTU/(hr·ft²)	33.4	
Max coil CFM	1824	CFM	Ent. DB / Lvg DB	66.8 / 73.2	°F
Water flow @ 20.0 °F drop	N/A				

Supply Fan Sizing Data

Actual max CFM	1824	CFM	Fan motor BHP	0.00	BHP
Standard CFM	1823	CFM	Fan motor kW	0.00	kW
Actual max CFM/ft²	4.87	CFM/ft²	Fan static	0.00	in wg

Outdoor Ventilation Air Data

Design airflow CFM	218	CFM	CFM/person	8.37	CFM/person
CFM/ft²	0.58	CFM/ft²			

Zone Sizing Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

Air System Information

Air System Name ..... split ac - su room 306
Equipment Class ..... SPLT AHU
Air System Type ..... SZCAV

Number of zones 1
Floor Area ..... 375.0 ft²
Location ..... DHAKA, Bangladesh

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 Terminal Sizing Data

Zone Name	Design Supply Airflow (CFM)	Minimum Supply Airflow (CFM)	Zone CFM/ft²	Reheat Coil Load (MBH)	Reheat Coil Water gpm @ 20.0 °F	Zone Htg Unit Coil Load (MBH)	Zone Htg Unit Water gpm @ 20.0 °F	Mixing Box Fan Airflow (CFM)
Zone 1	1824	1824	4.87	0.0	-	0.0	-	0

Zone Peak Sensible Loads

Zone Name	Zone Cooling Sensible (MBH)	Time of Peak Sensible Cooling Load	Zone Heating Load (MBH)	Zone Floor Area (ft²)
Zone 1	33.5	Oct 1400	6.9	375.0

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (MBH)	Time of Peak Sensible Load	Air Flow (CFM)	Heating Load (MBH)	Floor Area (ft²)	Space CFM/ft²
Zone 1							
SU _ ROOM NO 306	1	33.5	Oct 1400	1824	6.9	375.0	4.87

Ventilation Sizing Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

1. Summary

Ventilation Sizing Method ..... Sum of Space OA Airflows
Design Ventilation Airflow Rate ..... 218 CFM

2. Space Ventilation Analysis

Zone Name / Space Name	Mult.	Floor Area (ft²)	Maximum Occupants	Maximum Supply Air (CFM)	Required Outdoor Air (CFM/person)	Required Outdoor Air (CFM/ft²)	Required Outdoor Air (CFM)	Required Outdoor Air (% of supply)	Uncorrected Outdoor Air (CFM)
Zone 1									
SU _ ROOM NO 306	1	375.0	26.0	1824.4	7.50	0.06	0.0	0.0	217.5
Totals (incl. Space Multipliers)				1824.4					217.5

Air System Design Load Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Sep 1300 COOLING OA DB / WB 103.7 °F / 87.9 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 44.0 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	104 ft²	6167	-	104 ft²	-	-
Wall Transmission	347 ft²	8181	-	347 ft²	4073	-
Roof Transmission	380 ft²	1540	-	380 ft²	866	-
Window Transmission	104 ft²	2039	-	104 ft²	1950	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	877	-	375 ft²	0	-
Partitions	450 ft²	5246	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	32950	7020	-	6889	0
Zone Conditioning	-	32566	7020	-	6715	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	1824 CFM	0	-	1824 CFM	0	-
Ventilation Load	218 CFM	6491	14520	218 CFM	5819	0
Supply Fan Load	1824 CFM	0	-	1824 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	39058	21540	-	12534	0
Central Cooling Coil	-	39058	21541	-	0	0
Central Heating Coil	-	0	-	-	12534	-
>> Total Conditioning	-	39058	21541	-	12534	0
Key:	Positive values are clg loads Negative values are htg loads			Positive values are htg loads Negative values are clg loads		

Zone Design Load Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

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Zone 1	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Oct 1400 COOLING OA DB / WB 101.4 °F / 86.1 °F OCCUPIED T-STAT 75.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 44.0 °F OCCUPIED T-STAT 70.0 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	104 ft²	7636	-	104 ft²	-	-
Wall Transmission	347 ft²	7720	-	347 ft²	4073	-
Roof Transmission	380 ft²	1425	-	380 ft²	866	-
Window Transmission	104 ft²	1846	-	104 ft²	1950	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	835	-	375 ft²	0	-
Partitions	450 ft²	5106	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	33468	7020	-	6889	0

Space Design Load Summary for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

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TABLE 1.1.A. Component Loads For Space "SU _ ROOM NO 306" In Zone "Zone 1"						
	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Oct 1400 COOLING OA DB / WB 101.4 °F / 86.1 °F OCCUPIED T-STAT 75.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 44.0 °F OCCUPIED T-STAT 70.0 °F		
		Sensible	Latent		Sensible	Latent
SPACE LOADS	Details	(BTU/hr)	(BTU/hr)	Details	(BTU/hr)	(BTU/hr)
Window & Skylight Solar Loads	104 ft²	7636	-	104 ft²	-	-
Wall Transmission	347 ft²	7720	-	347 ft²	4073	-
Roof Transmission	380 ft²	1425	-	380 ft²	866	-
Window Transmission	104 ft²	1846	-	104 ft²	1950	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	835	-	375 ft²	0	-
Partitions	450 ft²	5106	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	33468	7020	-	6889	0

TABLE 1.1.B. Envelope Loads For Space "SU _ ROOM NO 306" In Zone "Zone 1"						
				COOLING	COOLING	HEATING
	Area	U-Value	Shade	TRANS	SOLAR	TRANS
	(ft²)	(BTU/(hr·ft²·°F))	Coeff.	(BTU/hr)	(BTU/hr)	(BTU/hr)
E EXPOSURE						
WALL	246	0.469	-	4936	-	2887
WINDOW 1	20	0.750	1.000	355	930	375
WINDOW 2	20	0.750	1.000	355	930	375
S EXPOSURE						
WALL	101	0.469	-	2784	-	1185
WINDOW 2	64	0.750	1.000	1136	5777	1200
H EXPOSURE						
ROOF	380	0.091	-	1425	-	866

Hourly Air System Design Day Loads for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
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DESIGN MONTH: JULY

Hour	OA TEMP (°F)	SUPPLY AIRFLOW (CFM)	CENTRAL COOLING SENSIBLE (MBH)	CENTRAL COOLING TOTAL (MBH)	CENTRAL HEATING COIL (MBH)	PRECOOL COIL (MBH)	PREHEAT COIL (MBH)	TERMINAL COOLING (MBH)	TERMINAL HEATING (MBH)	ZONE HEATING UNIT (MBH)
0000	95.0	1824	29.0	49.4	0.0	0.0	0.0	0.0	0.0	0.0
0100	94.3	1824	28.3	48.6	0.0	0.0	0.0	0.0	0.0	0.0
0200	93.5	1824	27.0	46.8	0.0	0.0	0.0	0.0	0.0	0.0
0300	92.9	1824	26.3	45.9	0.0	0.0	0.0	0.0	0.0	0.0
0400	92.5	1824	26.6	46.5	0.0	0.0	0.0	0.0	0.0	0.0
0500	92.3	1824	25.6	45.2	0.0	0.0	0.0	0.0	0.0	0.0
0600	92.6	1824	26.5	46.2	0.0	0.0	0.0	0.0	0.0	0.0
0700	93.4	1824	29.1	49.7	0.0	0.0	0.0	0.0	0.0	0.0
0800	94.7	1824	30.0	50.6	0.0	0.0	0.0	0.0	0.0	0.0
0900	96.7	1824	31.9	52.7	0.0	0.0	0.0	0.0	0.0	0.0
1000	98.9	1824	34.5	56.0	0.0	0.0	0.0	0.0	0.0	0.0
1100	101.5	1824	35.8	57.5	0.0	0.0	0.0	0.0	0.0	0.0
1200	103.9	1824	36.7	58.5	0.0	0.0	0.0	0.0	0.0	0.0
1300	105.7	1824	37.4	59.2	0.0	0.0	0.0	0.0	0.0	0.0
1400	106.9	1824	37.8	59.7	0.0	0.0	0.0	0.0	0.0	0.0
1500	107.3	1824	37.4	59.2	0.0	0.0	0.0	0.0	0.0	0.0
1600	106.9	1824	37.0	58.7	0.0	0.0	0.0	0.0	0.0	0.0
1700	105.8	1824	36.0	57.4	0.0	0.0	0.0	0.0	0.0	0.0
1800	104.2	1824	34.6	55.7	0.0	0.0	0.0	0.0	0.0	0.0
1900	102.2	1824	34.6	56.1	0.0	0.0	0.0	0.0	0.0	0.0
2000	100.3	1824	33.5	54.8	0.0	0.0	0.0	0.0	0.0	0.0
2100	98.6	1824	31.4	52.1	0.0	0.0	0.0	0.0	0.0	0.0
2200	97.1	1824	31.1	51.9	0.0	0.0	0.0	0.0	0.0	0.0
2300	95.9	1824	30.3	51.1	0.0	0.0	0.0	0.0	0.0	0.0

Hourly Zone Loads for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

ZONE: Zone 1
DESIGN MONTH: JULY

Hour	OA TEMP (°F)	ZONE TEMP (°F)	RH (%)	ZONE AIRFLOW (CFM)	ZONE SENSIBLE LOAD (BTU/hr)	ZONE COND (BTU/hr)	TERMINAL COOLING COIL (BTU/hr)	TERMINAL HEATING COIL (BTU/hr)	ZONE HEATING UNIT (BTU/hr)
0000	95.0	76.0	67	1824.4	25001.4	24538.2	0.0	0.0	0.0
0100	94.3	76.0	68	1824.4	24434.4	24016.9	0.0	0.0	0.0
0200	93.5	76.4	69	1824.4	23900.2	22944.9	0.0	0.0	0.0
0300	92.9	76.5	70	1824.4	23416.4	22418.0	0.0	0.0	0.0
0400	92.5	75.9	70	1824.4	22980.6	22677.3	0.0	0.0	0.0
0500	92.3	76.3	71	1824.4	22613.6	21868.1	0.0	0.0	0.0
0600	92.6	76.5	69	1824.4	23683.8	22690.1	0.0	0.0	0.0
0700	93.4	75.7	67	1824.4	24986.0	24952.1	0.0	0.0	0.0
0800	94.7	76.3	66	1824.4	26575.9	25725.7	0.0	0.0	0.0
0900	96.7	76.5	64	1824.4	28144.6	27137.9	0.0	0.0	0.0
1000	98.9	75.9	62	1824.4	29339.0	29083.4	0.0	0.0	0.0
1100	101.5	75.9	61	1824.4	30116.3	29767.2	0.0	0.0	0.0
1200	103.9	76.0	60	1824.4	30603.7	30202.2	0.0	0.0	0.0
1300	105.7	75.9	60	1824.4	30792.2	30406.0	0.0	0.0	0.0
1400	106.9	75.8	60	1824.4	30813.6	30522.0	0.0	0.0	0.0
1500	107.3	76.1	60	1824.4	30718.9	30120.5	0.0	0.0	0.0
1600	106.9	76.1	61	1824.4	30479.8	29816.0	0.0	0.0	0.0
1700	105.8	76.4	62	1824.4	30045.4	29104.2	0.0	0.0	0.0
1800	104.2	76.6	63	1824.4	29350.1	28134.8	0.0	0.0	0.0
1900	102.2	75.9	62	1824.4	28709.6	28446.5	0.0	0.0	0.0
2000	100.3	75.8	63	1824.4	27876.8	27723.8	0.0	0.0	0.0
2100	98.6	76.3	65	1824.4	27073.7	26160.4	0.0	0.0	0.0
2200	97.1	75.8	65	1824.4	26314.0	26065.6	0.0	0.0	0.0
2300	95.9	75.7	66	1824.4	25622.8	25528.4	0.0	0.0	0.0

System Psychrometrics for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

September DESIGN COOLING DAY, 1300

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	103.7	0.02515	218	400	6491	14520
Vent - Return Mixing	Outlet	79.3	0.01276	1824	1854	-	-
Central Cooling Coil	Outlet	59.5	0.01027	1824	1854	39058	21541
Central Heating Coil	Outlet	59.5	0.01027	1824	1854	0	-
Supply Fan	Outlet	59.5	0.01027	1824	1854	0	-
Cold Supply Duct	Outlet	59.5	0.01027	1824	1854	-	-
Zone Air	-	76.0	0.01108	1824	2050	32566	7020
Return Plenum	Outlet	76.0	0.01108	1824	2050	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.079 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4742.7 BTU/(hr-CFM)
Site Altitude = 23.0 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	32950	Cooling	32566	76.0	1824	2050	0	0

System Psychrometrics for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

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	45.0	0.00584	218	400	-5819	0
Vent - Return Mixing	Outlet	66.8	0.00584	1824	468	-	-
Central Cooling Coil	Outlet	66.8	0.00584	1824	468	0	0
Central Heating Coil	Outlet	73.2	0.00584	1824	468	12534	-
Supply Fan	Outlet	73.2	0.00584	1824	468	0	-
Cold Supply Duct	Outlet	73.2	0.00584	1824	468	-	-
Zone Air	-	69.8	0.00584	1824	477	-6715	0
Return Plenum	Outlet	69.8	0.00584	1824	477	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.079 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4742.7 BTU/(hr-CFM)
Site Altitude = 23.0 ft

TABLE 2: ZONE DATA

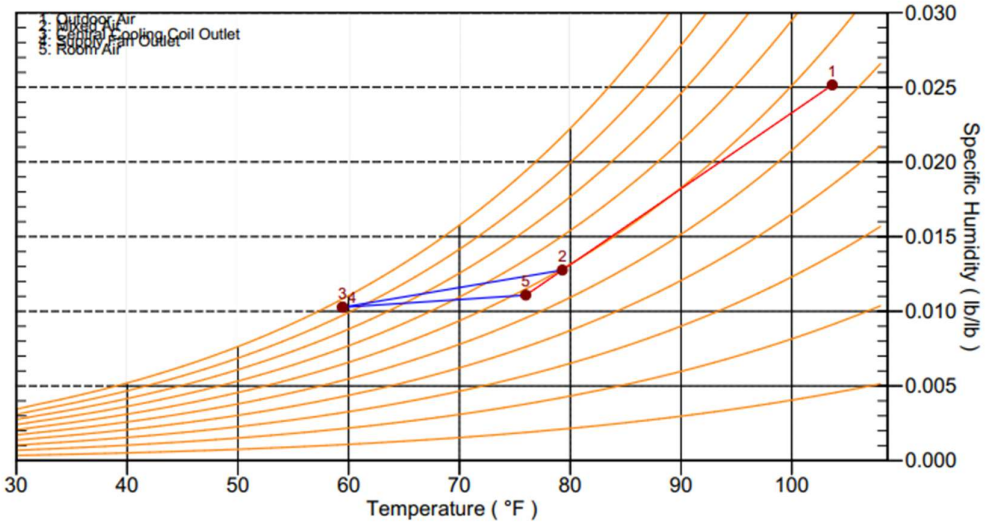
Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	CO2 Level (ppm)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	-6889	Heating	-6715	69.8	1824	477	0	0

System Psychrometrics for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

Location: DHAKA, Bangladesh
Altitude: 23.0 ft.
Data for: September DESIGN COOLING DAY, 1300

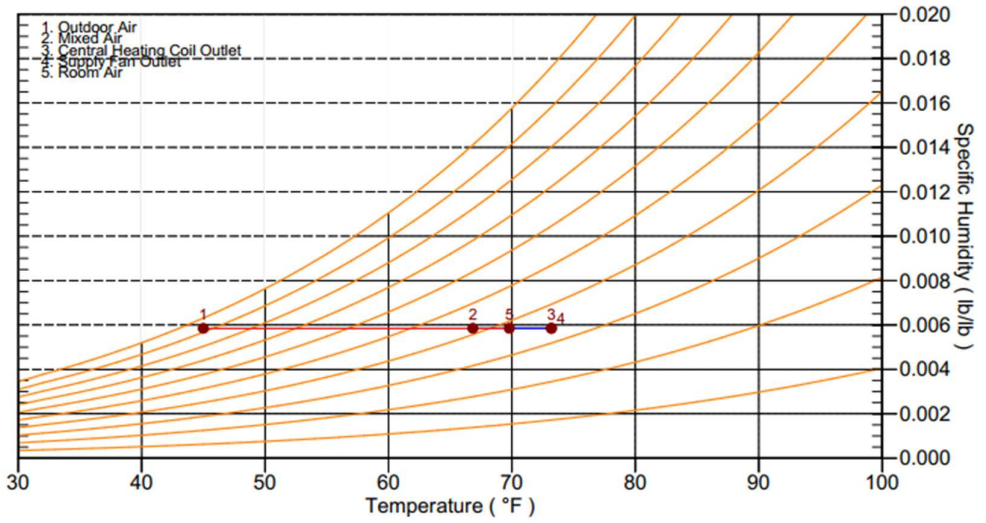


System Psychrometrics for split ac - su room 306

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:02PM

Location: DHAKA, Bangladesh
Altitude: 23.0 ft.
Data for: WINTER DESIGN HEATING



7.2 HAP Report on Energy Efficient Cooling System (AHU with Ducting)

Air System Sizing Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

Air System Information

Air System Name	Su room 306 - AHU		Number of zones	1	
Equipment Class	CW AHU		Floor Area	375.0	ft²
Air System Type	SZCAV		Location	DHAKA, Bangladesh	

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	4.1	Tons	Load occurs at	Sep 1400	
Total coil load	48.9	MBH	OA DB / WB	104.9 / 88.1	°F
Sensible coil load	28.0	MBH	Entering DB / WB	81.1 / 71.2	°F
Coil CFM at Sep 1400	1231	CFM	Leaving DB / WB	60.0 / 59.2	°F
Max block CFM	1231	CFM	Coil ADP	57.6	°F
Sum of peak zone CFM	1231	CFM	Bypass Factor	0.100	
Sensible heat ratio	0.573		Resulting RH	61	%
CFM/Ton	302.3		Design supply temp.	58.0	°F
ft²/Ton	92.1		Zone T-stat Check	1 of 1	OK
BTU/(hr·ft²)	130.3		Max zone temperature deviation	0.0	°F
Water flow @ 10.0 °F rise	9.78	gpm			

Central Heating Coil Sizing Data

Max coil load	8.3	MBH	Load occurs at	Des Htg	
Coil CFM at Des Htg	1231	CFM	BTU/(hr·ft²)	22.1	
Max coil CFM	1231	CFM	Ent. DB / Lvg DB	65.6 / 71.8	°F
Water flow @ 20.0 °F drop	0.83	gpm			

Supply Fan Sizing Data

Actual max CFM	1231	CFM	Fan motor BHP	0.00	BHP
Standard CFM	1230	CFM	Fan motor kW	0.00	kW
Actual max CFM/ft²	3.28	CFM/ft²	Fan static	0.00	in wg

Outdoor Ventilation Air Data

Design airflow CFM	218	CFM	CFM/person	8.37	CFM/person
CFM/ft²	0.58	CFM/ft²			

Zone Sizing Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

Air System Information

Air System Name ..... Su room 306 - AHU
Equipment Class ..... CW AHU
Air System Type ..... SZCAV

Number of zones 1
Floor Area ..... 375.0 ft²
Location ..... DHAKA, Bangladesh

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 Terminal Sizing Data

Zone Name	Design Supply Airflow (CFM)	Minimum Supply Airflow (CFM)	Zone CFM/ft²	Reheat Coil Load (MBH)	Reheat Coil Water gpm @ 20.0 °F	Zone Htg Unit Coil Load (MBH)	Zone Htg Unit Water gpm @ 20.0 °F	Mixing Box Fan Airflow (CFM)
Zone 1	1231	1231	3.28	0.0	0.00	0.0	0.00	0

Zone Peak Sensible Loads

Zone Name	Zone Cooling Sensible (MBH)	Time of Peak Sensible Cooling Load	Zone Heating Load (MBH)	Zone Floor Area (ft²)
Zone 1	22.6	Nov 1400	2.5	375.0

Space Loads and Airflows

Zone Name / Space Name	Mult.	Cooling Sensible (MBH)	Time of Peak Sensible Load	Air Flow (CFM)	Heating Load (MBH)	Floor Area (ft²)	Space CFM/ft²
Zone 1							
SU _ ROOM NO 306 (exp)	1	22.6	Nov 1400	1231	2.5	375.0	3.28

Ventilation Sizing Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

1. Summary

Ventilation Sizing Method ..... Sum of Space OA Airflows
Design Ventilation Airflow Rate ..... 218 CFM

2. Space Ventilation Analysis

Zone Name / Space Name	Mult.	Floor Area (ft²)	Maximum Occupants	Maximum Supply Air (CFM)	Required Outdoor Air (CFM/person)	Required Outdoor Air (CFM/ft²)	Required Outdoor Air (CFM)	Required Outdoor Air (% of supply)	Uncorrected Outdoor Air (CFM)
Zone 1									
SU _ ROOM NO 306 (exp)	1	375.0	26.0	1230.8	7.50	0.06	0.0	0.0	217.5
Totals (incl. Space Multipliers)				**1230.8**					**217.5**

Air System Design Load Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Sep 1400 COOLING OA DB / WB 104.9 °F / 88.1 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 44.0 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	104 ft²	5972	-	104 ft²	-	-
Wall Transmission	347 ft²	1633	-	347 ft²	805	-
Roof Transmission	380 ft²	1663	-	380 ft²	866	-
Window Transmission	104 ft²	846	-	104 ft²	780	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	893	-	375 ft²	0	-
Partitions	450 ft²	1384	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	**21292**	**7020**	-	**2451**	**0**
Zone Conditioning	-	21218	7020	-	2402	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	1231 CFM	0	-	1231 CFM	0	-
Ventilation Load	218 CFM	6781	13843	218 CFM	5871	0
Supply Fan Load	1231 CFM	0	-	1231 CFM	0	-
Space Fan Coil Fans	-	0	-	-	0	-
Duct Heat Gain / Loss	0%	0	-	0%	0	-
>> Total System Loads	-	**27999**	**20863**	-	**8273**	**0**
Central Cooling Coil	-	27999	20863	-	0	0
Central Heating Coil	-	0	-	-	8273	-
>> Total Conditioning	-	**27999**	**20863**	-	**8273**	**0**
Key:	**Positive values are clg loads Negative values are htg loads**			**Positive values are htg loads Negative values are clg loads**		

Zone Design Load Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

Zone 1	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Nov 1400			HEATING DATA AT DES HTG		
	COOLING OA DB / WB 95.9 °F / 83.6 °F			HEATING OA DB / WB 45.0 °F / 44.0 °F		
	OCCUPIED T-STAT 75.0 °F			OCCUPIED T-STAT 70.0 °F		
ZONE LOADS	Details	Sensible (BTU/hr)	Latent (BTU/hr)	Details	Sensible (BTU/hr)	Latent (BTU/hr)
Window & Skylight Solar Loads	104 ft²	8676	-	104 ft²	-	-
Wall Transmission	347 ft²	1430	-	347 ft²	805	-
Roof Transmission	380 ft²	1074	-	380 ft²	866	-
Window Transmission	104 ft²	568	-	104 ft²	780	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	744	-	375 ft²	0	-
Partitions	450 ft²	1186	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	22578	7020	-	2451	0

Space Design Load Summary for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

TABLE 1.1.A. Component Loads For Space "SU _ ROOM NO 306 (exp)" In Zone "Zone 1"						
	DESIGN COOLING			DESIGN HEATING		
	COOLING DATA AT Nov 1400 COOLING OA DB / WB 95.9 °F / 83.6 °F OCCUPIED T-STAT 75.0 °F			HEATING DATA AT DES HTG HEATING OA DB / WB 45.0 °F / 44.0 °F OCCUPIED T-STAT 70.0 °F		
		Sensible	Latent		Sensible	Latent
SPACE LOADS	Details	(BTU/hr)	(BTU/hr)	Details	(BTU/hr)	(BTU/hr)
Window & Skylight Solar Loads	104 ft²	8676	-	104 ft²	-	-
Wall Transmission	347 ft²	1430	-	347 ft²	805	-
Roof Transmission	380 ft²	1074	-	380 ft²	866	-
Window Transmission	104 ft²	568	-	104 ft²	780	-
Skylight Transmission	0 ft²	0	-	0 ft²	0	-
Door Loads	0 ft²	0	-	0 ft²	0	-
Floor Transmission	375 ft²	744	-	375 ft²	0	-
Partitions	450 ft²	1186	-	450 ft²	0	-
Ceiling	0 ft²	0	-	0 ft²	0	-
Overhead Lighting	475 W	1621	-	0	0	-
Task Lighting	0 W	0	-	0	0	-
Electric Equipment	0 W	0	-	0	0	-
People	26	7280	7020	0	0	0
Infiltration	-	0	0	-	0	0
Miscellaneous	-	0	0	-	0	0
Safety Factor	0% / 0%	0	0	0%	0	0
>> Total Zone Loads	-	22578	7020	-	2451	0

TABLE 1.1.B. Envelope Loads For Space "SU _ ROOM NO 306 (exp)" In Zone "Zone 1"						
				COOLING	COOLING	HEATING
	Area	U-Value	Shade	TRANS	SOLAR	TRANS
	(ft²)	(BTU/(hr·ft²·°F))	Coeff.	(BTU/hr)	(BTU/hr)	(BTU/hr)
E EXPOSURE						
WALL	246	0.093	-	751	-	571
WINDOW 1	20	0.300	1.000	109	779	150
WINDOW 2	20	0.300	1.000	109	779	150
S EXPOSURE						
WALL	101	0.093	-	679	-	234
WINDOW 2	64	0.300	1.000	350	7118	480
H EXPOSURE						
ROOF	380	0.091	-	1074	-	866

Hourly Air System Design Day Loads for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

DESIGN MONTH: JULY

Hour	OA TEMP (°F)	SUPPLY AIRFLOW (CFM)	CENTRAL COOLING SENSIBLE (MBH)	CENTRAL COOLING TOTAL (MBH)	CENTRAL HEATING COIL (MBH)	PRECOOL COIL (MBH)	PREHEAT COIL (MBH)	TERMINAL COOLING (MBH)	TERMINAL HEATING (MBH)	ZONE HEATING UNIT (MBH)
0000	95.0	1231	20.3	40.0	0.0	0.0	0.0	0.0	0.0	0.0
0100	94.3	1231	20.0	39.7	0.0	0.0	0.0	0.0	0.0	0.0
0200	93.5	1231	19.9	39.6	0.0	0.0	0.0	0.0	0.0	0.0
0300	92.9	1231	19.1	38.6	0.0	0.0	0.0	0.0	0.0	0.0
0400	92.5	1231	18.4	37.5	0.0	0.0	0.0	0.0	0.0	0.0
0500	92.3	1231	18.3	37.4	0.0	0.0	0.0	0.0	0.0	0.0
0600	92.6	1231	20.0	39.8	0.0	0.0	0.0	0.0	0.0	0.0
0700	93.4	1231	21.4	41.8	0.0	0.0	0.0	0.0	0.0	0.0
0800	94.7	1231	22.1	42.5	0.0	0.0	0.0	0.0	0.0	0.0
0900	96.7	1231	23.7	44.5	0.0	0.0	0.0	0.0	0.0	0.0
1000	98.9	1231	23.6	44.0	0.0	0.0	0.0	0.0	0.0	0.0
1100	101.5	1231	24.8	45.5	0.0	0.0	0.0	0.0	0.0	0.0
1200	103.9	1231	25.6	46.4	0.0	0.0	0.0	0.0	0.0	0.0
1300	105.7	1231	26.1	47.1	0.0	0.0	0.0	0.0	0.0	0.0
1400	106.9	1231	26.0	46.8	0.0	0.0	0.0	0.0	0.0	0.0
1500	107.3	1231	26.3	47.1	0.0	0.0	0.0	0.0	0.0	0.0
1600	106.9	1231	26.3	47.3	0.0	0.0	0.0	0.0	0.0	0.0
1700	105.8	1231	25.3	45.8	0.0	0.0	0.0	0.0	0.0	0.0
1800	104.2	1231	24.2	44.5	0.0	0.0	0.0	0.0	0.0	0.0
1900	102.2	1231	23.4	43.4	0.0	0.0	0.0	0.0	0.0	0.0
2000	100.3	1231	23.2	43.6	0.0	0.0	0.0	0.0	0.0	0.0
2100	98.6	1231	21.9	41.7	0.0	0.0	0.0	0.0	0.0	0.0
2200	97.1	1231	21.3	41.2	0.0	0.0	0.0	0.0	0.0	0.0
2300	95.9	1231	20.6	40.2	0.0	0.0	0.0	0.0	0.0	0.0

Hourly Zone Loads for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

ZONE: Zone 1
DESIGN MONTH: JULY

Hour	OA TEMP (°F)	ZONE TEMP (°F)	RH (%)	ZONE AIRFLOW (CFM)	ZONE SENSIBLE LOAD (BTU/hr)	ZONE COND (BTU/hr)	TERMINAL COOLING COIL (BTU/hr)	TERMINAL HEATING COIL (BTU/hr)	ZONE HEATING UNIT (BTU/hr)
0000	95.0	76.1	71	1230.8	16134.6	15857.8	0.0	0.0	0.0
0100	94.3	75.9	71	1230.8	15855.1	15728.7	0.0	0.0	0.0
0200	93.5	75.7	71	1230.8	15590.3	15689.3	0.0	0.0	0.0
0300	92.9	75.9	72	1230.8	15346.1	15157.9	0.0	0.0	0.0
0400	92.5	76.3	73	1230.8	15121.6	14632.0	0.0	0.0	0.0
0500	92.3	76.2	73	1230.8	14924.9	14520.8	0.0	0.0	0.0
0600	92.6	75.8	71	1230.8	16101.5	16044.2	0.0	0.0	0.0
0700	93.4	75.7	68	1230.8	17216.7	17295.0	0.0	0.0	0.0
0800	94.7	76.1	67	1230.8	18153.6	17777.7	0.0	0.0	0.0
0900	96.7	75.8	66	1230.8	18817.1	18759.7	0.0	0.0	0.0
1000	98.9	76.5	66	1230.8	19115.5	18358.6	0.0	0.0	0.0
1100	101.5	76.1	65	1230.8	19145.4	18834.2	0.0	0.0	0.0
1200	103.9	76.0	65	1230.8	19235.4	19032.2	0.0	0.0	0.0
1300	105.7	75.9	65	1230.8	19291.9	19149.1	0.0	0.0	0.0
1400	106.9	76.2	65	1230.8	19313.1	18865.2	0.0	0.0	0.0
1500	107.3	76.1	65	1230.8	19270.1	18956.7	0.0	0.0	0.0
1600	106.9	75.9	65	1230.8	19122.6	19052.5	0.0	0.0	0.0
1700	105.8	76.3	66	1230.8	18820.5	18350.4	0.0	0.0	0.0
1800	104.2	76.4	67	1230.8	18319.2	17739.0	0.0	0.0	0.0
1900	102.2	76.5	68	1230.8	17947.8	17339.6	0.0	0.0	0.0
2000	100.3	75.9	68	1230.8	17529.7	17506.8	0.0	0.0	0.0
2100	98.6	76.4	69	1230.8	17137.9	16638.4	0.0	0.0	0.0
2200	97.1	76.2	70	1230.8	16772.0	16447.0	0.0	0.0	0.0
2300	95.9	76.4	70	1230.8	16438.1	15988.6	0.0	0.0	0.0

System Psychrometrics for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

September DESIGN COOLING DAY, 1400

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	104.9	0.02515	218	400	6781	13843
Vent - Return Mixing	Outlet	81.1	0.01411	1231	1797	-	-
Central Cooling Coil	Outlet	60.0	0.01053	1231	1797	27999	20863
Central Heating Coil	Outlet	60.0	0.01053	1231	1797	0	-
Supply Fan	Outlet	60.0	0.01053	1231	1797	0	-
Cold Supply Duct	Outlet	60.0	0.01053	1231	1797	-	-
Zone Air	-	76.0	0.01173	1231	2096	21218	7020
Return Plenum	Outlet	76.0	0.01173	1231	2096	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.079 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4742.7 BTU/(hr-CFM)
Site Altitude = 23.0 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	21292	Cooling	21218	76.0	1231	2096	0	0

System Psychrometrics for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

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	45.0	0.00584	218	400	-5871	0
Vent - Return Mixing	Outlet	65.6	0.00584	1231	439	-	-
Central Cooling Coil	Outlet	65.6	0.00584	1231	439	0	0
Central Heating Coil	Outlet	71.8	0.00584	1231	439	8273	-
Supply Fan	Outlet	71.8	0.00584	1231	439	0	-
Cold Supply Duct	Outlet	71.8	0.00584	1231	439	-	-
Zone Air	-	70.0	0.00584	1231	447	-2402	0
Return Plenum	Outlet	70.0	0.00584	1231	447	0	-

Air Density x Heat Capacity x Conversion Factor: At sea level = 1.080; At site altitude = 1.079 BTU/(hr-CFM-F)
Air Density x Heat of Vaporization x Conversion Factor: At sea level = 4746.6; At site altitude = 4742.7 BTU/(hr-CFM)
Site Altitude = 23.0 ft

TABLE 2: ZONE DATA

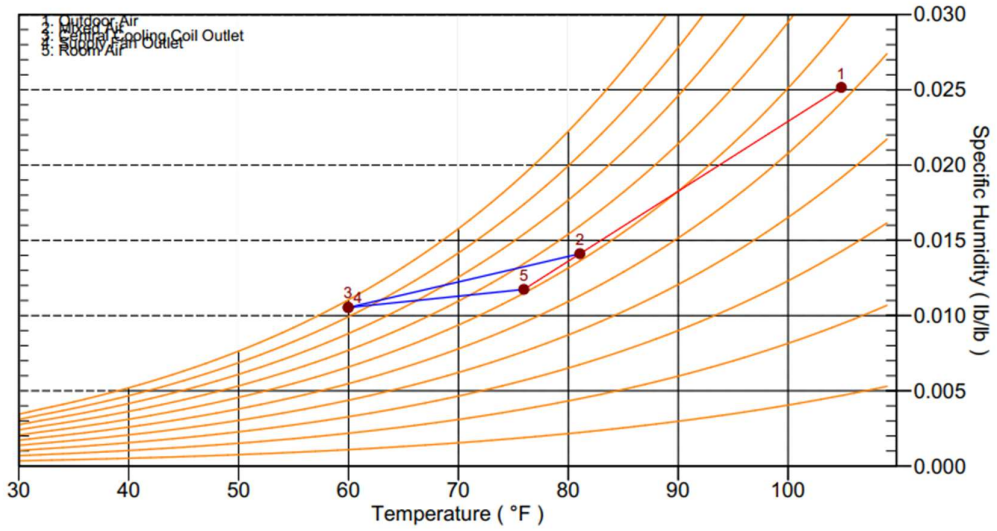
Zone Name	Zone Sensible Load (BTU/hr)	T-stat Mode	Zone Cond (BTU/hr)	Zone Temp (°F)	Zone Airflow (CFM)	CO2 Level (ppm)	Terminal Heating Coil (BTU/hr)	Zone Heating Unit (BTU/hr)
Zone 1	-2451	Deadband	-2402	70.0	1231	447	0	0

System Psychrometrics for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

Location: DHAKA, Bangladesh
Altitude: 23.0 ft.
Data for: September DESIGN COOLING DAY, 1400

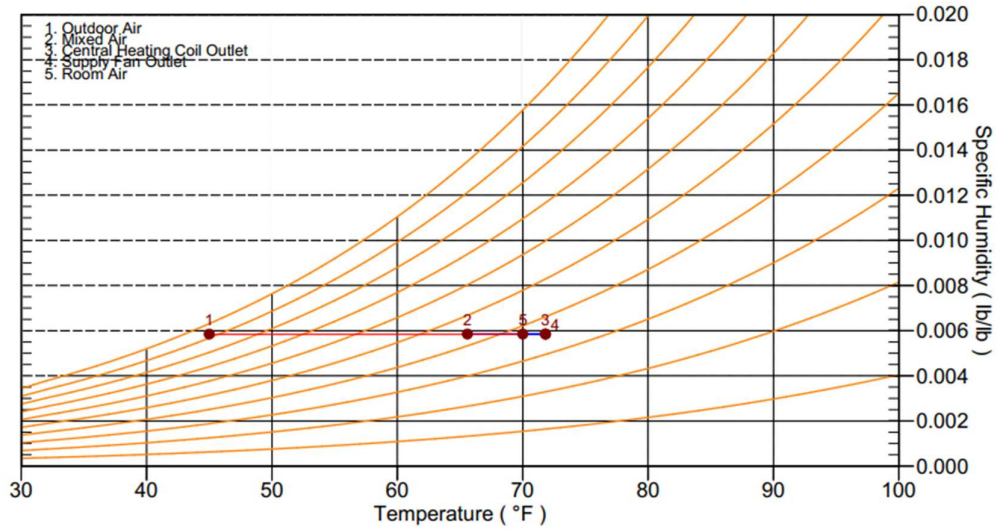


System Psychrometrics for Su room 306 - AHU

Project Name: SU_ TH
Prepared by: MNA_PERSONAL

09/07/2025
10:12PM

Location: DHAKA, Bangladesh
Altitude: 23.0 ft.
Data for: WINTER DESIGN HEATING



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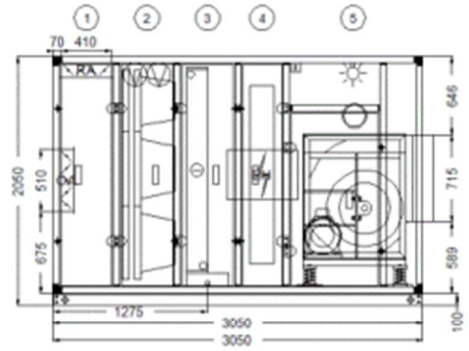
Appendices

Appendix A: Technical Specifications of HVAC Equipment

TECHNICAL REPORT

Project Name	Genesis Fashion	Patch Number	R24B
Tag	AHU 45TR	Database Version	20250703
Model	ESCS3H-50mm TTB-AH150-FC	Pricing Version	20250703
Location	Malaysia	Report Date	19/8/2025

OUTLINE DRAWING



Right View

GENERAL SPECIFICATIONS

Certified in accordance with the AHRI Central Station Air-Handling Unit Certification Program, which is based on AHRI Standards 1350. Certified units may be found in the AHRI Directory at www.ahridirectory.org

Certified by the AHRI Central Station Air-Handling Unit (AHU) Certification Program, which is based on AHRI Standard 430/431. AHRI certified units are subject to rigorous and continuous testing, have performance ratings independently measured and are third-party verified. Certified units may be found in AHRI Directory at www.ahridirectory.org

**Owing to continued product development Dunham-Bush reserves the right to introduce alternative equivalent parts (eg. silencer, humidifier, blower, motor, EC fan, heat pipe, plate heat exchanger, heat wheel,moisture eliminator, etc) without prior notice.*

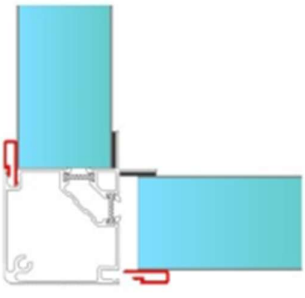
AHRI 1350 RATING

Casing Deflection Rating Class (+)	:	CD1	Thermal Transmittance Class With Leakage	:	CT2
Casing Deflection Rating Class (-)	:	CD1	Thermal Transmittance Class Without Leakage	:	CT2
Casing Air Leakage Class (+)	:	CL6	Thermal Bridging Class	:	CB2
Casing Air Leakage Class (-)	:	CL6			

GENERAL SPECIFICATIONS

AHU Type	:	50mm TTB	Total Height	:	2050	mm
PU Thickness	:	50 mm	Total Width	:	2050	mm
Installation Type	:	Indoor	Total Length	:	3050	mm
Quantity	:	1	Roof Selection	:	No	
Season	:	Summer	Single Power Point	:	No	
Altitude	:	0.00 m				

STRUCTURAL PROFILE



AIR SIDE CRITERIA

Supply Air Volume	:	13400	CFM
Fresh Air Volume	:	1340	CFM
Return Air Volume	:	12060	CFM
Exhaust Air Volume	:	0	CFM
Coil Face Velocity	:	2.22	m/s

PANEL / FRAME

Inner Material	:	GI Solid	
Inner Skin	:	0.8	mm
Outer Material	:	GI Solid(Painted)	
Outer Skin	:	0.8	mm
Insulation	:	Polyurethane-2.5lbs/ft³	
Base Type	:	Bolted Base	
Base Height	:	100	mm

Remark :

TECHNICAL REPORT

Project Name	Genesis Fashion	Patch Number	R24B
Tag	AHU 45TR	Database Version	20250703
Model	ESCS3H-50mm TTB-AH150-FC	Pricing Version	20250703
Location	Malaysia	Report Date	19/8/2025

1 MIXING BOX

AIR MIXING RESULTS

Season		SUMMER		WINTER		MIXED AIR		Unit
Air Properties		Fresh Air	Return Air	Fresh Air	Return Air	Summer	Winter	
Dry Bulb Temp.	:	35.0	26.7	26.7	26.7	27.5	0.0	°C
Wet Bulb Temp.	:	29.0	19.4	19.4	19.4	20.5	0.0	°C
Relative Humidity	:	64.33	50.76	50.76	50.76	53.60	0.00	%

TOP OPENING

Air Flow	:	12060	CFM	Dimension (LxW)	:	410*1550	Blade Type	: Opposed
Air Velocity	:	8.96	m/s	Damper Material	:	AL	Operating Method	: Handle Arm

REAR OPENING

Air Flow	:	1340	CFM	Dimension (LxW)	:	510*1550	Blade Type	: Opposed
Air Velocity	:	0.80	m/s	Damper Material	:	AL	Operating Method	: Handle Arm

2 PRIMARY FILTER

FILTER SPECIFICATIONS

Filter Brand	:	Standard (Non-UL)	Air Flow	:	13400	CFM	Filter Area	: 3.159	m²
Filter Type	:	Flat Filter (Washable 2")	Filter Velocity	:	2.00	m/s	24" x 24"	: 6	
Filter Grade	:	92% Arrestance (G4)	Initial PD.	:	36	Pa	24" x 12"	: 0	
Track Material	:	GI	Final PD.	:	125	Pa	24" x 20"	: 3	
Mounting Type	:	Side Loading	Design PD.	:	36	Pa	20" x 20"	: 0	

ACCESSORIES

Filter gauge	:	1

TECHNICAL REPORT

Project Name	Genesis Fashion	Patch Number	R24B
Tag	AHU 45TR	Database Version	20250703
Model	ESCS3H-50mm TTB-AH150-FC	Pricing Version	20250703
Location	Malaysia	Report Date	19/8/2025

SECONDARY FILTER

FILTER SPECIFICATIONS

Filter Brand	: Standard (Non-UL)	Air Flow	: 13400	CFM	Filter Area	: 3.159	m^2
Filter Type	: Bag Filter (15")	Filter Velocity	: 2.00	m/s	24" x 24"	: 6	
Filter Grade	: 95% DS (F9)	Initial PD.	: 101	Pa	24" x 12"	: 0	
Track Material	: GI	Final PD.	: 375	Pa	24" x 20"	: 3	
Mounting Type	: Side Loading	Design PD.	: 101	Pa	20" x 20"	: 0	

ACCESSORIES

Filter gauge	:	1

3 WATER COOLING COIL

COIL SPECIFICATIONS

Coil Code	: AWF4D-52Tx68 LH		Coil Layer	: 1				
Coil Tube Size	: 1/2" (Type A)		Medium	: Water		Header Material	: Steel	
Coil Row	: 4		Circuiting Method	: F (4T/C)		Has Drain Pan	: Yes	
Coil FPI	: 14		Tube / Fin Material	: Copper / Aluminium		Drain Pan Material	: SS 304	
Tube Face	: 52		Tube Spacing (in)	: 1.25		Moisture Eliminator	: No	
Fin Height	: 1651	mm	Coil Frame Material	: GI		Coil Spacer	: No	
Fin Length	: 1727	mm	Coating Material	: None		Coil Section Length	: Short Coil Section	

AIR SIDE

ENTERING AIR			
Dry Bulb Temp.	:	27.5	°C
Wet Bulb Temp.	:	20.5	°C
Relative Humidity	:	53.16	%
LEAVING AIR			
Dry Bulb Temp.	:	13.3	°C
Wet Bulb Temp.	:	13.0	°C
Relative Humidity	:	96.38	%

WATER SIDE

Entering Temp.	:	7.0	°C
Leaving Temp.	:	12.0	°C
Flow Rate	:	7.71	L/s
Water Velocity	:	1.33	m/s
Pressure Drop	:	24.68	kPa
Header Connection	:	2.5	in
Connection Type	:	MBSP	
Fouling Factor	:	0.001	h.ft².°F/Btu

PERFORMANCE

Total Capacity	:	161.33	kW
Sensible Capacity	:	103.77	kW
Air Flow	:	13400	CFM
Air Velocity	:	2.22	m/s
Air Pressure Drop	:	122	Pa

Notes

Coil is NOT certified by AHRI

Minimum fin surface temperature : 32°F

Minimum tube wall surface temperature : 32°F

TECHNICAL REPORT

Project Name	Genesis Fashion	Patch Number	R24B
Tag	AHU 45TR	Database Version	20250703
Model	ESCS3H-50mm TTB-AH150-FC	Pricing Version	20250703
Location	Malaysia	Report Date	19/8/2025

4 ELECTRIC HEATER

GENERAL SPECIFICATIONS

Heater Capacity	:	50.00	kW
Stages	:	3	
Pressure Drop	:	24	Pa
Control Box	:	Heater Termination Box	

AIR SIDE

Dry Bulb Temp. (In)	:	13.3	°C
Dry Bulb Temp. (Out)	:	19.9	°C

TECHNICAL REPORT

Project Name	Genesis Fashion	Patch Number	R24B
Tag	AHU 45TR	Database Version	20250703
Model	ESCS3H-50mm TTB-AH150-FC	Pricing Version	20250703
Location	Malaysia	Report Date	19/8/2025

5 SUPPLY FAN

FAN SPECIFICATIONS

Fan Code	:	560-R-FC (Std)	
Fan Type	:	Forward	
Fan Quantity	:	1	
Fan Shaft Power	:	9.57	kW
Blower Speed	:	864	rpm
Maximum Fan Speed	:	900	rpm
Discharge Opening	:	715*715	mm
Discharge Direction	:	Front	
Air Flow	:	13400	CFM

MOTOR SPECIFICATIONS

Installed Power	:	11.00	kW
Power Supply	:	380-415V/3Ph/50Hz	
Motor Efficiency	:	88.6	%
Efficiency Type	:	IE1/EFF2	
Motor Frame	:	160M	
Motor Pole	:	4	
Mounting Position	:	Right	
Motor Type	:	AC	
Shaft Speed	:	1460	rpm
Class H Motor	:	No	
2 Speed Motor	:	No	
Variable Pitch Pulley	:	No	
Motor Input Power	:	10.8	kW
Specific Fan Power	:	0.474	W/CMH

PERFORMANCE

Total Efficiency	:	68.02	%
Discharge Velocity	:	12.37	m/s
ISP	:	471	Pa
ESP	:	500	Pa
TSP	:	971	Pa

VFD

VFD Type	:	
VFD Installation	:	

FAN SOUND LEVEL

	Inlet Sound (dB)								
Band Frequency, Hz	63	125	250	500	1000	2000	4000	8000	Overall
Lw	89.2	90.1	85.1	88.0	82.0	81.0	79.0	74.0	95.1
Lw(A)	64.2	75.1	77.1	85.0	82.0	82.0	80.0	73.0	89.2
Lp(A)	56.3	67.3	69.3	77.2	74.2	74.2	72.2	65.2	81.4

	Outlet Sound (dB)								
Band Frequency, Hz	63	125	250	500	1000	2000	4000	8000	Overall
Lw	90.5	92.5	84.6	93.5	83.8	82.4	80.4	75.4	97.8
Lw(A)	64.3	76.4	76.0	90.3	83.8	83.6	81.4	74.3	92.5
Lp(A)	56.3	68.4	68.0	82.3	75.8	75.6	73.4	66.3	84.5

	Sound Pressure Level @ 1 meter of AHU External (dBA)								
Band Frequency, Hz	63	125	250	500	1000	2000	4000	8000	Overall
Lp(A)	54.1	51.3	55.0	61.6	61.6	57.7	47.1	39.4	66.3

AHU SUMMARY

Project Name	Genesis Fashion	Patch Number	R24B
Location	Malaysia	Database Version	20250703
Report Date	19/8/2025	Pricing Version	20250703

No.	Tag	Model	Qty.	Supply Fan		Exhaust/Return Fan		Cooling Cap.	Heating Cap.	HSHP	Electric Heater	Cooling Coil			Heating Coil		
				Air Flow	TSP	Air Flow	TSP					Leaving	Header	Rows* FPI* Circuiting	Leaving	Header	Rows* FPI* Circuiting
				m³/h	Pa	m³/h	Pa	kW	kW	kW	kW	DB/WB	Inch		DB/WB	Inch	
1	AHU 45TR	ESCS3H-50mm TTB-AH150-FC	1	22767	971	-	-	161.33	-	-	50.00	13.3/13.0	2.5	4*14	-	-	-

No.	Blower								Filter Type	Humidifier		Heat Recovery		Fresh Air Damper		Return Air Damper	
	Type	Fan Model	Disch. Direction	Speed	Disch. Opening	Mtr Loc.	Mtr Power	Volt.		Humid.	Type	Type	Cap.	Loc.	Opening Size	Loc.	Opening Size
				RPM	W*L (mm)		kW			kg/h			kW		H*W (mm)		H*W (mm)
1	Supply	560-R-FC (Std)	Front	864	715*715	Side	11.00	380-415V/3Ph	92% Arrestance (G4),95% DS (F9)	-	-	-	-	Front	510*1550	Top	410*1550

Remark :

**Owing to continued product development Dunham-Bush reserves the right to introduce alternative equivalent parts (eg. silencer, humidifier, blower, motor, EC fan, heat pipe, plate heat exchanger, heat wheel,moisture eliminator, etc) without prior notice.*

Legends : MTR. = Motor HSHP = Hot-Shoe Heat Pipe, PC/RH = Pre-Cool / Re-Heat, TC/SC = Heat Wheel or Heat Plate's Enthalpy/Sensible Capacities, when shown in bracket (), refer to Summer (Winter) performance

Technical Specification		WSI-COATEC (SUPERSAVER) -12J [UV]	WSI-COATEC (SUPERSAVER)-12M [SOLAR HYBRID]	WSI-COATEC (SUPERSAVER) -18H [UV]	WSI-COATEC (SUPERSAVER)-18H [SOLAR HYBRID]	WSI-COATEC (SUPERSAVER) -24H [UV]
Function		Cooling	Cooling	Cooling	Cooling	Cooling
BTU/hr		12000	12000	18000	18000	24000
Cooling Capacity (W)		3517	3517	5275	5275	7034
Rated Input Power (W)		890	925	1240	1240	1900
EER(W/W)		3.95	3.80	4.25	4.25	3.7
Power Supply	Phase/ AC voltage (V) /frequency (Hz)/ current (A)	Single/230/50/3.93	Single/230/50/4.1	Single/230/50/5.45	Single/230/50/5.45	Single/230/50/8.51
	Phase/ DC voltage range (V)	---	48V to 400V [Hybrid] 120V to 400V [Off-Grid]*	---	48V to 400V [Hybrid] 120V to 400V [Off-Grid]*	---
Air Circulation (m^3/h)		570	550	900	900	970
Compressor Type		Rotary (Inverter)	Rotary (Inverter)	Rotary (Inverter)	Rotary (Inverter)	Rotary (Inverter)
Refrigerant Type		R-32	R-32	R-32	R-32	R-32
Refrigerant Amount (kg)		0.40	0.44	0.61	0.61	0.66
Operating Range (Ambient Temperature)		20°C to 52°C	20°C to 52°C	20°C to 52°C	20°C to 52°C	20°C to 52°C
Evaporator Fin Type		Louvered Fin	Louvered Fin	Louvered Fin	Louvered Fin	Louvered Fin
Condenser Type		Microchannel Condenser	Microchannel Condenser	Microchannel Condenser	Microchannel Condenser	Microchannel Condenser
Condenser Fin Type		Wave Fin	Wave Fin	Wave Fin	Wave Fin	Wave Fin
Maximum Pipe Length		10m	10m	10m	10m	15m
Turbo Mode (Maximum Cooling)		Yes	Yes	Yes	Yes	Yes
ECO Mode (Power Saving)		Yes	Yes	Yes	Yes	Yes