



THE UNIVERSITY OF BRITISH COLUMBIA

Mechanical Engineering
Faculty of Applied Science



Shaping Tomorrow's Global
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ASHRAE Student Design Competition

HVAC Design Calculations 2025

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1 Executive Summary

This report is submitted by Mechanical Engineering students from The University of British Columbia for the 2025 ASHRAE Student Design Project Competition (HVAC Design Calculation). The project involves the complete HVAC design and load calculation for a three-story, 67,158 ft² outpatient medical office building located in Manchester, England, as per the competition's Owner's Project Requirements (OPR). The facility includes a wide variety of space types such as exam rooms, treatment areas, administrative offices, waiting rooms, laboratories, and a dental suite, all of which require careful zoning and environmental control.

The primary objective of this project was to accurately calculate heating and cooling loads and design appropriately sized HVAC systems that ensure compliance with the latest ASHRAE Standards 55, 62.1, 90.1, and 170, while also fulfilling OPR criteria for energy efficiency, indoor air quality, thermal comfort, and sustainable life cycle performance over a 50-year building lifespan.

Using Carrier HAP 5.11 and IES VE 2024, the team performed detailed thermal load simulations and cross-validated results. The building's peak design cooling load was calculated at 681,581 Btu/h (sensible) and 94,964 Btu/h (latent), while the peak heating load was 497,769 Btu/h. These loads informed the sizing of the mechanical systems, which include a four-pipe fan coil system, a modular Air Source Heat Pump (ASHP) rated at 840 MBH cooling, and a Water-to-Water Heat Pump (WWHP) rated at 600 MBH heating. A Renewable Natural Gas (RNG)-fired high efficiency condensing boiler provides redundancy and supplemental heating.

To overcome ventilation distribution challenges and comply with ASHRAE Standard 170, the design utilizes a rooftop Make-Up Air Unit (MAU) for the West Wing and ceiling-mounted Heat Recovery Ventilators (HRVs) in the East Wing. The building requires approximately 12,208 CFM of ventilation air, and all systems are equipped with economizer functions and energy recovery features. Psychrometric analysis confirmed compliance with ASHRAE Standard 55 for both ventilation air and zone-level thermal comfort, using Predicted Mean Vote (PMV) modeling in critical spaces.

The energy modeling, based on ASHRAE Standard 90.1 Appendix G, compared the proposed design to the baseline System Type 6 (Variable Air Volume System with Hot Water Reheat). Results showed a 73% reduction in total energy use over a baseline system. Additionally, incorporating a 10,000 ft² rooftop PV array yielded a 31% reduction in carbon emissions, saving approximately 54,800 lbs CO₂e per year.

A life cycle cost analysis over 50 years confirmed the proposed system as a cost-effective, resilient solution. The proposed system, with PV, costs \$ 8.4 M.



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4 Introduction & Inputs

4.1 Owner Project Requirements

The Owner's Project Requirements (OPR) [2] state the following general design goal for the project:

"The design team shall make every effort to provide a safe and sustainable design, taking into account: energy efficiency, occupant health and safety, occupant comfort, functionality, future flexibility as well as maintainability and a 50-year service life."

The following key requirements and needs are considered for the HVAC system design:

- The following indoor conditions must be maintained for all spaces, excluding mechanical/electrical rooms, as defined in Table 1. Mechanical/electrical rooms must be maintained between 55°F and 85°F dry bulb (DB).

Table 1 Indoor Design Conditions

	Summer	Winter
Temperature	75 °F DB	72 °F DB
Relative humidity (RH)	30-50%	

- Building must comply with the following ASHRAE standards and codes:
 - ASHRAE Standard 55 (2023) [8]
 - ASHRAE Standard 62.1 (2022) [4]
 - ASHRAE Standard 90.1 (2022) [3]
 - ASHRAE Standard 170 (2021) [5]
- The following ASHRAE standards and guidelines are also considered:
 - ASHRAE Standard 189.1 (2020) [6]
 - ASHRAE Advanced Energy Design Guide for Achieving Zero Energy in Small to Medium Office
- Propose an optimal design and building orientation that minimizes energy consumption, in accordance with the ASHRAE Advanced Energy Design Guide for Small to Medium Office Buildings.
- The building must be operated at positive pressure to prevent unwanted infiltration.
- The selected design must meet the owner's budget while considering life cycle costs, based on parameters defined in Table 2.
- The selected HVAC design provides excellent acoustical controls per ASHRAE Handbooks.

Table 2 OPR Requirements

Parameters	Values
Owner's Budget	\$10,000,000 USD
Life of HVAC System	30 years
Life Cycle Period	50 years
Owner's Rate of Return	4%
General Inflation Rate	3%
Utility Inflation Rates	Electrical: 3.5% Natural Gas: 7% Water & Sewer: 3%
Electric Rate	Day: 0.35 USD/kWh Night: 0.23 USD/kWh
Natural Gas Rate	3.42 USD/m ³

Parameters	Values
Hours of Operation	Weekday: 7am – 7pm Saturday: 7am – 12pm

4.2 Climate and Weather

The heating degree-days base 65°F (HDD65) is 5488, and the cooling degree-days base 50°F (CDD50) is 1437 for Manchester, UK. The climate zone of Manchester is defined as 5A according to the criteria in ASHRAE Standard 169 (2021)¹. For the load calculations and system design, we adopted the 99.6% heating dry bulb (DB) design temperature of 25.0°F and 0.4% cooling design DB temperature of 83.9°F. The average dry bulb temperature and relative humidity are shown in Figure 1.

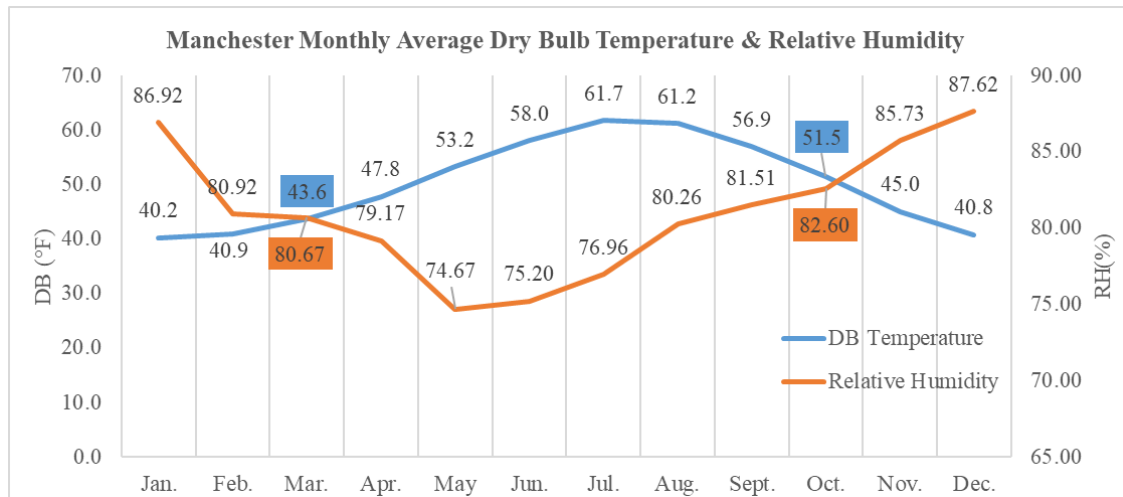


Figure 1 Monthly Average Dry Bulb Temperature and Relative Humidity in Manchester

4.3 Building Envelope

The building envelope U-values for each assembly were calculated based on the geometry and materials outlined in the competition drawings. Each U-value was then compared to the requirements specified in Table 5.5.5 (Zone 5) of ASHRAE Standard 90.1 and Table E-5 of ASHRAE Standard 189.1 to ensure that the calculated U values are lower than the required limits. Low U values are specified as the building is assumed to be comprised of high-performance materials.

For windows and walls, the U-values were referenced directly from Table 5.5.5 and used for load calculations in Hourly Analysis Program (HAP) version 5.11 and IESVE (2024) software. Table 3 below summarizes the calculated and maximum allowable building envelope values. The assembly R and U values were calculated using an online assembly calculator [1].

The F-factor for slab-on-grade and SHGC values were compared between ASHRAE Standards 90.1 and 189.1 and presented in Table 4. In this case, maximum SHGC and F-factor values were directly referenced from ASHRAE Standard 189.1 and were utilized in the load calculation software to apply SHGC variation depending on the orientation.

¹Weather data used for energy modeling is based on Manchester Airport (WMO: 033340), provided by ASHRAE as part of the 2025 Design Competition. Climate design conditions follow ASHRAE Standard 169-2021 and are formatted for compatibility with building energy simulation tools.



Table 3 Calculated and Minimum Envelope Requirements (Units: $R = \text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}/\text{Btu}$, $U = \text{Btu}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$)

Assembly	Layer	Depth (in)	R-Value	Total R-Value	Total U-Value	Max U-Value (ASHRAE 90.1)	Max U-Value (ASHRAE 189.1)
Walls	Brick Veneer	3.625	0.80	22.74	0.04	0.09	0.086
	Air gap	3	0.5				
	Rigid Insulation	3	20.3				
	Concrete Masonry Unit	7.625	1.14				
Floor	Concrete Slab	4	0.6	0.60	1.67	N/A	N/A
Roof	Fully adhered TPO	0.1	0.3	57.20	0.02	0.032	0.3
	Roof Cover Board	0.5	0.45				
	Rigid Insulation	10	56				
	Roof Cover Board	0.5	0.45				
Slab on Grade	Concrete Slab	4	0.6	8.60	0.12	N/A	N/A
	Rigid Insulation	2	8				
Windows	Fixed type	N/A	2.78	2.78	0.36	0.36	0.34
Doors	Swinging	N/A	2.7	2.7	0.37	0.37	0.352

Table 4 F-factor & SHGC Values (Units: SHGC = unitless, F-factor = $\text{Btu}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$)

Assembly	Type	ASHRAE 90.1			ASHRAE 189.1		
		F-factor	Max SHGC	Min SHGC	F-factor	Max SHGC	Min SHGC
Windows	Fixed	N/A	0.38	1.1	N/A	E & W - 0.36 S - 0.38 N - 0.48	1.1
Slab-on-grade	Unheated	0.520	N/A	N/A	0.494	N/A	N/A

5 Thermal Zoning and Room Classification

Thermal zoning is considered due to the diverse occupancy patterns, varying equipment loads, and strict ventilation and comfort requirements across different space types. Effective zoning enables control over heating, cooling, and ventilation, ensuring occupant comfort, and maintaining air quality standards.

The building is divided into multiple zones consisting of rooms with similar occupancy types, internal loads, ventilation needs, and perimeter exposures. This approach supports efficient distribution of conditioned air while minimizing energy consumption through targeted equipment deployment. Rooms with high perimeter losses, caused by windows, external walls, and infiltration, are grouped based on building orientation and occupant characteristics. For example, west-facing exam rooms on the first floor (Zone 1) are grouped due to similar occupancies (doctor and patient), shared solar gain patterns, and comparable envelope exposure.

Rooms of medical occupancy are zoned according to the air classification standards in ASHRAE Standards 62.1 and 170 (see Section 9.2), as well as ventilation and occupancy criteria. Zones 4.1, 4.2, and 4.3, for instance, are grouped due to similar occupancy levels and cooling demands.

Large open areas, including waiting rooms and entrance lobbies, are split into multiple zones to allow for precise temperature control based on orientation (e.g., north vs. south-facing). On the third floor, unoccupied shell space is divided into perimeter and core zones; larger loads are on the perimeter rather than a single distributed load through the large space. A full overview of the thermal zoning layout is provided in Figure 2.



Figure 2 Floor Zoning and Room Classification - Top Left: 1st Floor, Top Right: 2nd Floor, Bottom: 3rd Floor

6 Heating & Cooling Load Calculation

6.1 Introduction

To ensure accurate heating and cooling load calculations, both Carrier's Hourly Analysis Program (HAP) version 5.11 and IES VE (2024) are used. The tools were used to create a detailed simulation of the building's thermal performance under varying conditions. Inputs were guided by ASHRAE Standard 90.1 for energy efficiency and equipment loads, and ASHRAE Standards 62.1 & 170 for ventilation requirements. Occupant thermal comfort was a key design criterion, and software inputs were considered to align indoor conditions with the guidelines set by ASHRAE Standard 55. This section outlines the methodology and results of the load calculations that informed the overall HVAC design.

6.2 Calculation Methodology

The Hourly Analysis Program (HAP) version 5.11 was used for its ease of operation, paired with its accuracy. IES VE (2024) was used as validation as it is accurate, and complex, allowing for flexible inputs that are easily modifiable. Both software packages use the Steady-State Heat Balance Approach as outlined in the ASHRAE Handbook – Fundamentals.

The building was modeled in the software through manual inputs, room by room. Thermal zones were previously decided, discussed in Section 5 of this report. It is important to note that thermal zoning changes as the project progresses, reflected in the final CAD drawings of this submission.

In addition to the load calculations, a comprehensive solar shading analysis was performed using the software's advanced 3D modeling capabilities (as shown in Figure 3). This analysis evaluated solar gains and the impact of shading on the building's envelope, ensuring that the design optimizes both thermal performance and occupant comfort.

The loads were calculated using the owner's schedule, and infiltration occurred 100% of the time. The building is operating under positive pressure, but due to wind conditions, the team decided to safely assume infiltration still occurs at all times².

6.3 Results

Table 5 describes the calculated peak heating and cooling loads of the entire building. The internal loads consider the people, lighting, equipment, ventilation, and infiltration. The source equipment that produces total building heating & cooling is sized based on these total loads.

Other than the peak building heating and cooling loads, the individual peak loads for each thermal zone have also been determined using the software. The terminal air-side equipment, which handles the heating and cooling capacity for each thermal zone, was sized based on these peak loads to optimize efficiency while ensuring that the equipment can serve the peak load conditions.

Table 5 Heating and Cooling Load Calculation Checksum

Load Types	DESIGN COOLING			DESIGN HEATING		
	Aug. - 14:30 PM; Dry Resultant Temperature (°F) = 75.7; Relative Humidity (%) = 56.75			Jan. - 12:00 PM; Dry Resultant Temperature (°F) = 71.03		
	Details	Sensible (Btu/h)	Latent (Btu/h)	Details	Sensible (Btu/h)	Latent (Btu/h)
Solar Gain	5996 ft ²	181546	-	5996 ft ²	0	-
Conduction Gain - External Walls	29869 ft ²	-13141	-	29869 ft ²	-47768	-
Conduction Gain - Roofs	22386 ft ²	251	-	22386 ft ²	-17852	-
Conduction Gain - Ground/Exposed Floors	22386 ft ²	-14996	-	22386 ft ²	-13897	-
Conduction Gain - External Windows & Doors	5996 ft ²	22296	-	5996 ft ²	-100194	-
Equipment Gain	54961 W	187515	-	54961 W	0	-

² Infiltration was estimated using assumptions and methods prescribed in the ASHRAE Handbook – Fundamentals (2021), Chapter, 16 Section 9. Values were calculated using the assumptions outlined in Section 9 and known values from given data or prior calculations for load modelling.

	DESIGN COOLING			DESIGN HEATING		
	Aug. - 14:30 PM; Dry Resultant Temperature (°F) = 75.7; Relative Humidity (%) = 56.75			Jan. - 12:00 PM; Dry Resultant Temperature (°F) = 71.03		
Load Types	Details	Sensible (Btu/h)	Latent (Btu/h)	Details	Sensible (Btu/h)	Latent (Btu/h)
People Gain	558 people	141066	115251	558 people	0	0
Lighting Gain	50196 W	171317		50196 W	0	-
Infiltration Gain	2547.6 CFM	20874	-19329	2547.6 CFM	-129758	0
Ventilation	12208	13800	0	12208	-188300	0
Miscellaneous	-	-28947	-958	-	0	0
Safety Factor	-	0%	0%	-	0%	0%
Total Loads	-	681581	94964	-	-497769	0
Notes: - Subtraction sign "-" means losing heat, and space require heating.						

6.4 Validation

To validate the IES VE load results, a manual heat loss calculation was performed for the exterior wall and exterior window of a room 1082 Staff Station in accordance with the ASHRAE Handbook – Fundamentals. As shown in Table 6, the team validated the results by comparing HAP, IES VE, and the hand calculation, with discrepancies within 10%.

Table 6 Heating Load Validation through Building Envelope

1082 STAFF STATION				
Elements	Ext. Surface (ft²)	Manual Calc Conduction Gain (Btu/h)	IES VE Conduction Gain (Btu/h)	Tolerance (%)
Ext. Wall	675	-1269	-1274	0.4
Ext. Window	200	-3384	-3175	-6.2

7 System Selection

7.1 Introduction and Overview

For small commercial buildings, Variable Air Volume (VAV) systems are often preferred due to their simplicity and ease of maintenance. However, in this medical office building, the limited vertical shaft space is not able to accommodate the airflow introduced by high Air Changes per Hour (ACH) requirements mandated by ASHRAE Standards 170 and 62.1, particularly for healthcare spaces.

To meet load and zoning control requirements while maintaining system efficiency and flexibility, the team decided to proceed with a four-pipe water source fan coil system. This configuration allows for independent zone-level heating and cooling with minimal duct for central ventilation air distribution. Ventilation air for the West Wing and atrium is supplied via a rooftop-mounted Make-Up Air Unit (MAU) equipped with an enthalpy recovery wheel. For the East Wing, ceiling-mounted Heat Recovery Ventilators (HRVs) were specified to supply minimum ventilation rates on a zone-by-zone basis following ASHRAE Standards 62.1 or 170 (whichever is more stringent).

Hot and chilled water for the fan coil loop is supplied by a two-pipe Air Source Heat Pump (ASHP), operating in tandem with a Water-to-Water Heat Pump (WWHP) for heat recovery, allowing for simultaneous heating and

cooling. A Renewable Natural Gas (RNG)-fired boiler provides both redundancy and supplemental heating during peak winter conditions. All equipment is specified for 50 Hz operation in compliance with UK grid standards. Equipment selection and performance rating can be found in Section 15, Mechanical Drawing Set M3.01.

7.2 Air Source Heat Pump and Water Source Heat Pump

The hourly building thermal load profile shown in Figure 4 was generated by averaging hourly loads across each representative week, from 7 AM to 7 PM. The plot explains that this facility consistently experiences cooling demand year-round, and simultaneous heating and cooling loads are frequently present during occupied hours.

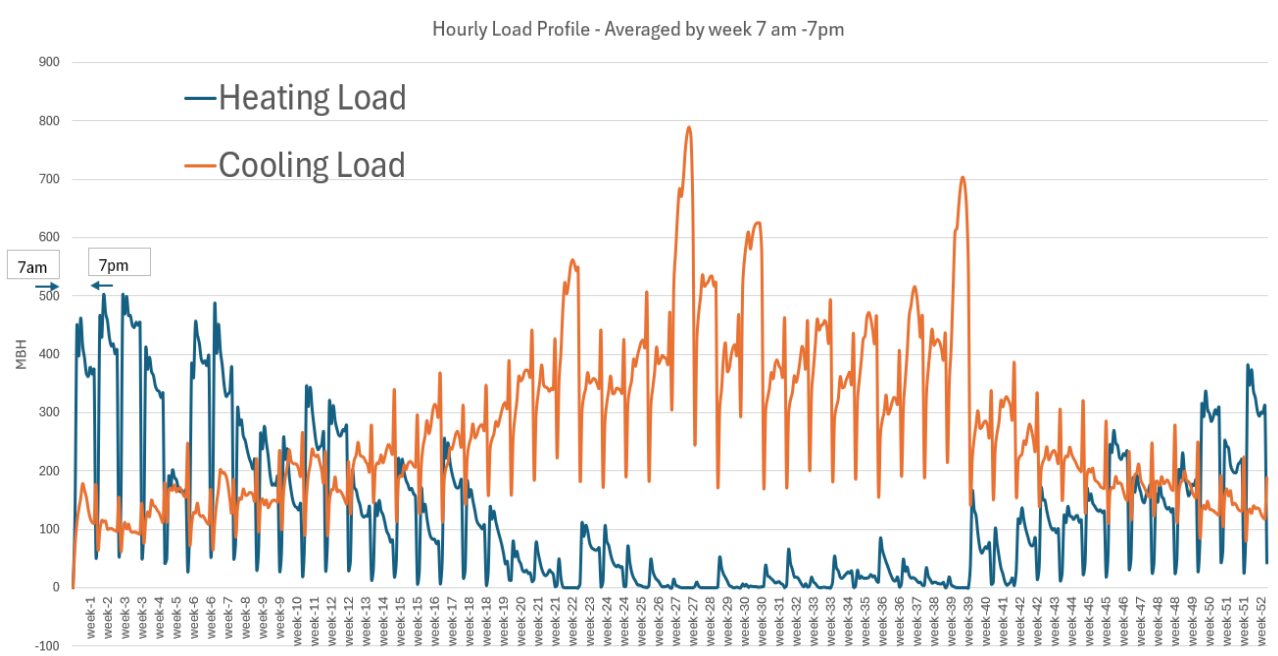


Figure 4 Hourly Building Thermal Load

As outlined in Section 6.2, the peak load requirements are approximately 800 MBH for cooling and 500 MBH for heating. Considering Manchester's mild climate and the building's cooling-dominant internal load profile, a hybrid system was selected to maximize energy efficiency and resilience. The system features a large ASHP sized to meet the cooling demand, paired with a smaller WWHP sized for the heating load, ensuring balanced and reliable performance throughout the year.

Although four-pipe ASHP systems are capable of providing simultaneous heating and cooling, they typically involve significantly higher capital costs and greater operational complexity. For smaller-scale buildings with limited overlapping heating and cooling demands, the cost-benefit ratio is less favorable. Therefore, the final system design features a four-stage MultiStack modular ASHP (840 MBH cooling) paired with a four-stage WWHP (600 MBH heating), offering a more efficient and cost-effective solution for the building's load profile.³

To enhance system performance, ensuring efficient heat transfer at fan coils and baseboard units, while maintaining energy efficiency with favorable COP⁴ values, the hydronic loop operating temperatures are set as follows:

- Heating loop: 110–95 °F (43.3–35 °C)
- Cooling loop: 45–55 °F (7.2–12.8 °C)

³ For this project 4 pipe ASHP cost \$308,000 USD, 2 pipe ASHP with WWHP cost \$308,000USD

⁴ Coefficient of Performance (COP), a measure of system efficiency: the ratio of energy output to input – higher means more efficient.

7.3 Boiler and Low Loss Header

Heating is a critical function in this building, especially during periods of extreme weather when the ASHP may operate less efficiently. To ensure system reliability and maintain occupant comfort, an RNG boiler is included as a supplemental heat source.

Given the presence of multiple heating and cooling sources, a low-loss header is used to decouple the system and load sides. This approach replaces the need for a traditional hydronic separator and provides flexibility for switching between different heating and cooling sources.

7.4 Make-Up Air Unit and Heat Recovery Ventilators

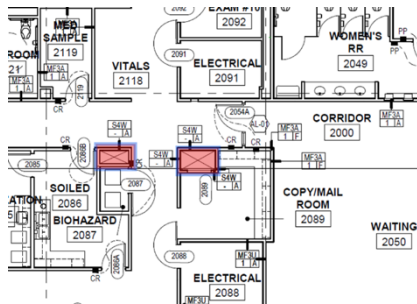


Figure 5 Building Shaft Location

Due to medical occupancy, many rooms are ventilated in accordance with the more rigorous ASHRAE Standard 170, resulting in a high ventilation demand of approximately 245,000 CFM. Due to spatial constraints, specifically the limited shaft size indicated in the architectural drawings, distributing all ventilation air through a single MAU presents significant challenges.

To address this, the ventilation system is split between two strategies. A rooftop MAU equipped with an enthalpy recovery wheel serves the West Wing and atrium. Both supply and exhaust air are distributed via the designated shaft.

The East Wing lacks adequate vertical shaft space to accommodate a centralized solution. Therefore, ceiling-mounted HRVs are specified for each zone. To

satisfy the OPR of maintaining a net positive building pressurization, all HRV and MAU systems are designed to slightly oversupply outdoor air compared to exhaust air. This also reduces the risk of cross-contamination across the heat recovery core.

In reference to Section 4.2 Weather and Climate, there is a significant amount of time that the building is in cooling and the exhaust air is warmer than outdoor air.

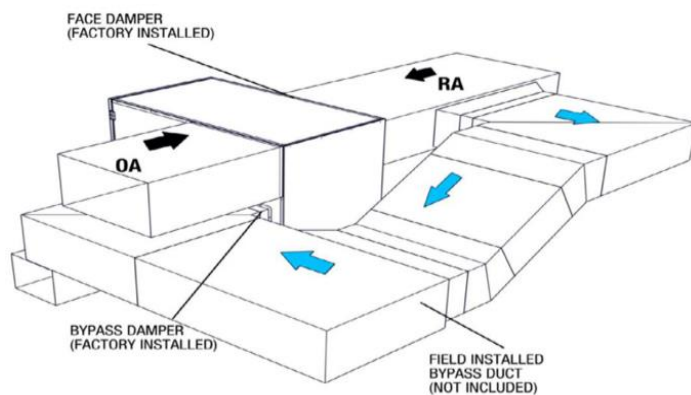


Figure 6 MAU Illustration

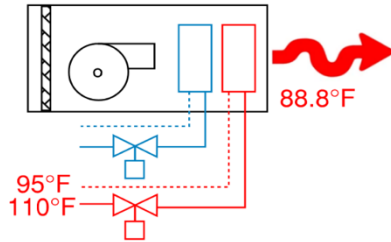
To comply with ASHRAE Standard 90.1 Section 6.5.1 and ASHRAE Standard 189.1, the ventilation strategy includes airside economizer functionality. The MAU achieves this by deactivating the enthalpy wheel motor when outdoor dry-bulb temperatures fall below 65 °F, enabling direct outdoor air intake and bypassing heat recovery. For HRVs, a dedicated bypass duct is included to disable energy recovery under the same conditions.

To prevent the contamination of the ventilation intake by exhaust air, all HRV intakes and exhaust termination points are spaced more than 10 feet apart, in accordance with the separation requirements of ASHRAE Standard 62.1 and the International Mechanical Code (IMC).

7.5 Terminal Units

7.5.1 Fan Coil

ACFM	400
Cooling	
Total Capacity (BTUH)	11961
Sensible Capacity (BTUH)	8849
Leaving Dry Bulb (°F)	59.9
Leaving Wet Bulb (°F)	57.6
Flow Rate (GPM)	2.4
Water Press Drop (FtWC)	14.1
Heating	
Total Capacity (BTUH)	8138
Leaving Dry Bulb (°F)	88.8
Flow Rate (GPM)	1.1
Water Press Drop (FtWC)	2.3



As noted in Section 7.2, the heating loop operates with a small approach temperature, supplying water at 110 °F to maximize the system's coefficient of performance (COP).⁵ However, when using standard dual-coil fan coils, the heating coil often lacks sufficient surface area to provide effective heat transfer at this lower temperature, resulting in leaving air temperatures below 90 °F (88.8 °F). This may increase the perception of drafts and contribute to occupant discomfort due to the wind chill effect.

Figure 7 Standard Fan Coil Performance

To improve heat transfer performance, the design incorporates larger, single heating coils paired with 6-way valves, which enable both heating and cooling through the same coil while ensuring effective thermal exchange. This solution enhances occupant comfort while maintaining high energy efficiency.

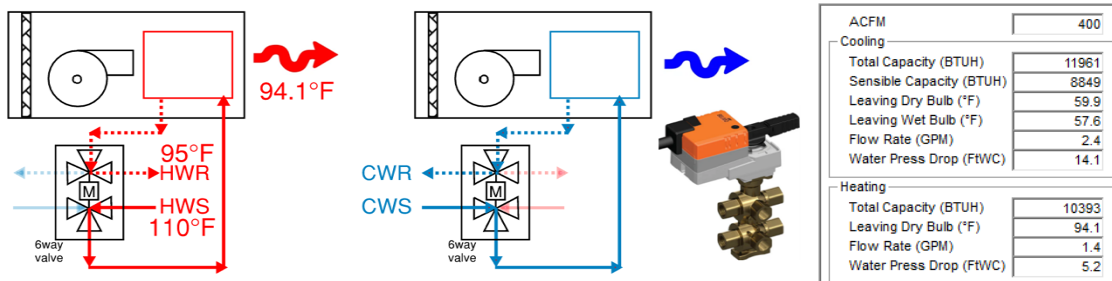


Figure 8 Fan Coil Performance with a 6-Way Valve

Fan coils are strategically located to minimize transmitted motor noise. In addition, diffusers have been selected and sized to limit the Noise Criteria (NC) level to a maximum of 30dB, following the E.H. Price grilles and registers application guidelines. Diffuser throw and spread patterns were also evaluated to prevent air stagnation and short-circuiting, ensuring even temperature distribution throughout the space.

7.5.2 Hydronic Baseboard

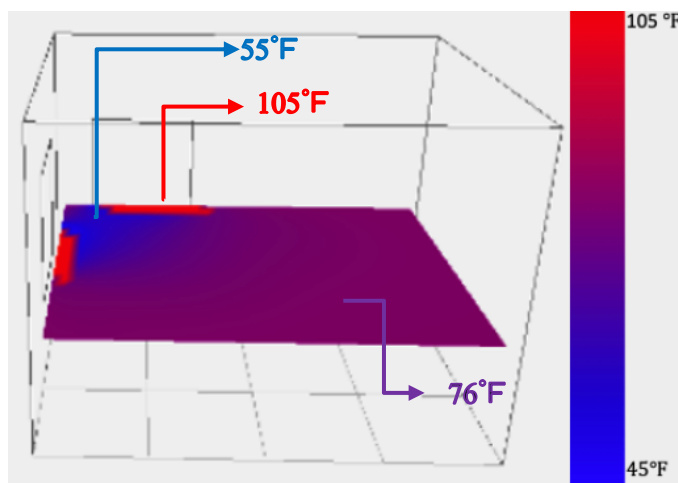


Figure 9 Exam Room 1083 Local Thermal Comfort Analysis

As per ASHRAE Standard 55 - 2023, Section 5.3.4, a local thermal discomfort analysis was performed and is detailed in Section 9.2. In zones where low mean radiant temperatures were predicted to negatively impact occupant comfort, such as Exam Room 1083, which features a large perimeter wall and extensive glazing, simulation results showed that the window-adjacent area fell below the acceptable comfort range based on the PMV gradient. To address this issue, hydronic baseboards were added along the perimeter to increase localized radiant heat and maintain thermal comfort for occupants.

7.6 Pumps

In selecting pumps, the design team tailored motor technologies to system needs, employing Electronically Commutated Motor (ECM) and Variable Frequency Drive

⁵ At design condition full load ASHP operating at loop temperature 95-110°F COP is 2.6; operating at 125-140°F COP is: 1.8

(VFD) units based on flow and size requirements. ECM pumps, used in the low-head primary loops (P-1, P-5, P-6), efficiently support both the boiler and WWHP circuits, where mutual backup ensures service continuity despite non-redundancy. For load-side distribution (P-2.1/2.2 and P-4.1/4.2), higher-capacity VFD-controlled pumps modulate flow to match heating and cooling demands, offering both efficiency and redundancy. Critical ASHP circuits (P-3.1, P-3.2, P-3.3) rely on a duty/duty/standby VFD configuration to maintain variable flow—70 GPM in heating and 140 GPM in cooling—with automatic switching to maintain uninterrupted operation.

7.7 Reverse Return Hydronic Piping Design

We utilized a reverse return hydronic piping design to supply our fan coil units, ensuring balanced flow across all terminal devices. In this configuration, the first unit to receive supply water is the last to return it, and vice versa, which naturally equalizes pressure losses throughout the system. This approach simplifies balancing, improves flow distribution, and reduces the need for manual balancing valves, enhancing both system efficiency and operational reliability.

7.8 Psychrometric Process

To verify equipment compliance with ASHRAE Standards 55 and 62.1, as well as the OPR, we plotted the psychrometric process of our HVAC system. This analysis confirms that the system maintains appropriate temperature and humidity levels across all operating modes, meets ventilation and dehumidification requirements, and delivers air within the thermal comfort zone. It also validates the effectiveness of energy recovery and economizer functions under varying seasonal conditions.

7.8.1 Ventilation Air Process

Figure 10 shows the psychrometric process of the MAU located on the roof.

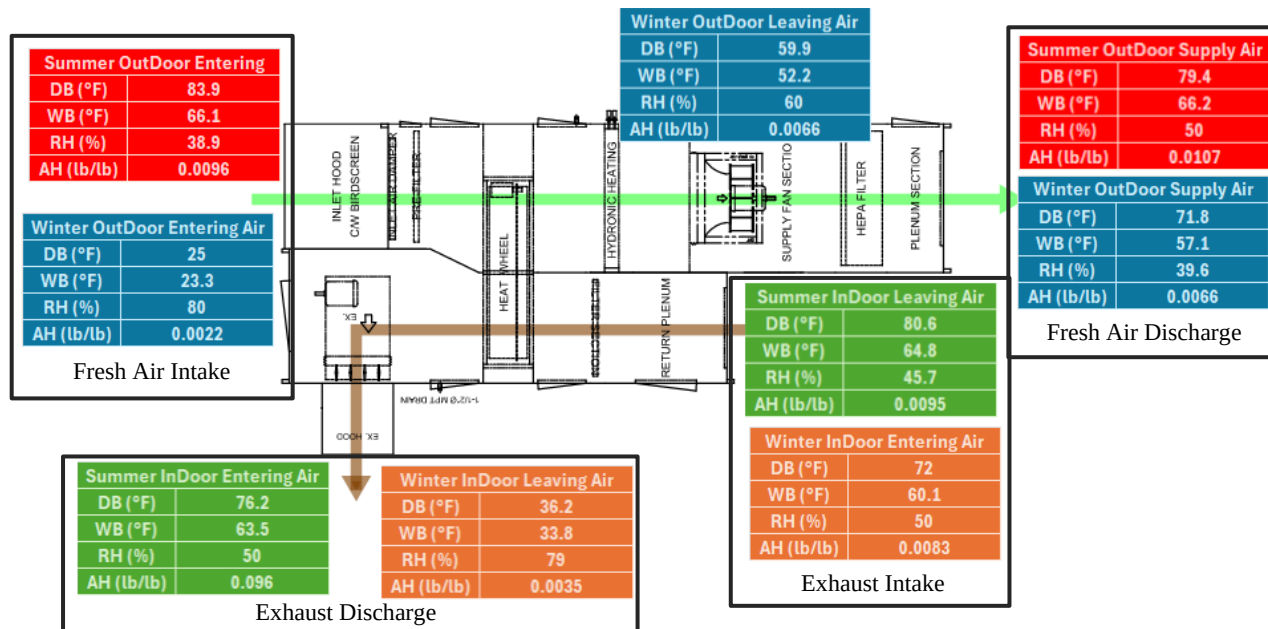


Figure 10 Psychrometric Process of MAU

The psychrometric process confirms that ventilation air is conditioned in line with ASHRAE Standard 55 and OPR. In winter, outdoor air at 25°F is preheated by the heat recovery wheel (75.8% effectiveness) to 59.9°F, then further tempered by a hydronic coil to meet the indoor supply condition of 72°F.

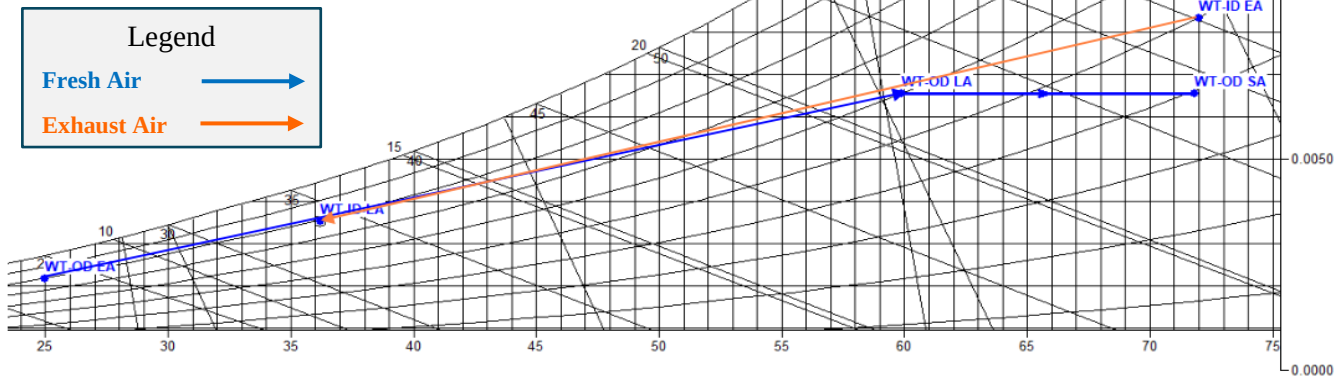


Figure 11 Summer MAU Psychrometric Process

In summer, the enthalpy wheel precools and reduces the moisture content of the incoming air before final conditioning to zone requirements. This reduces the load on mechanical systems while maintaining comfort and efficiency. And when outdoor air is cooler than indoor design conditions, the heat wheel stops operation and sends fresh air without heat exchange to help cool the indoor space.

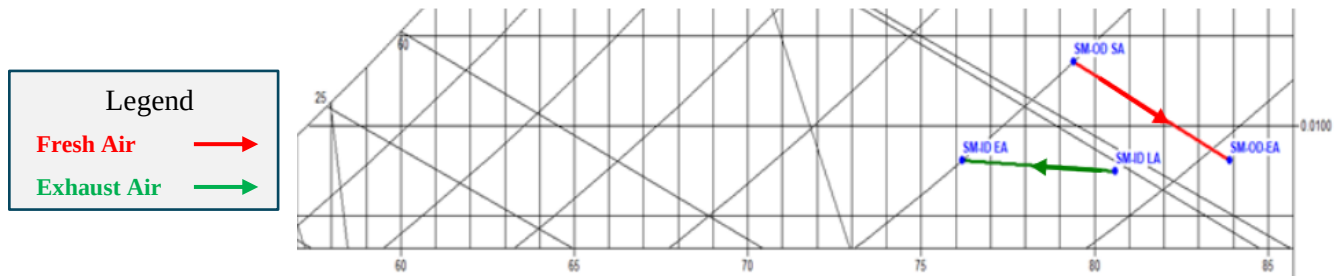


Figure 12 Winter MAU Psychrometric Process

7.8.2 Fan Coil Process

This psychrometric analysis evaluates the fan coil units' performance, specifically the impact of ventilation air mixing and internal heat gain from the fan motor on supply air conditions. Approximately 30% of the incoming air is ventilation air provided by the HRV, while the remaining 70% is return air from the conditioned space.

As illustrated on the psychrometric chart in Figure 13, the mixed air stream (Point 4) experiences a slight temperature increase due to fan motor heat before passing through the coil (Point 5). The resulting supply air (Point 6) is delivered at 55.4°F and 82.3% RH, effectively addressing both sensible and latent loads. This confirms the fan coil system operates within design expectations and supports zone-level thermal comfort in accordance with ASHRAE Standard 55. Figure 14 illustrates the connection points between the HRV and the fan coils.

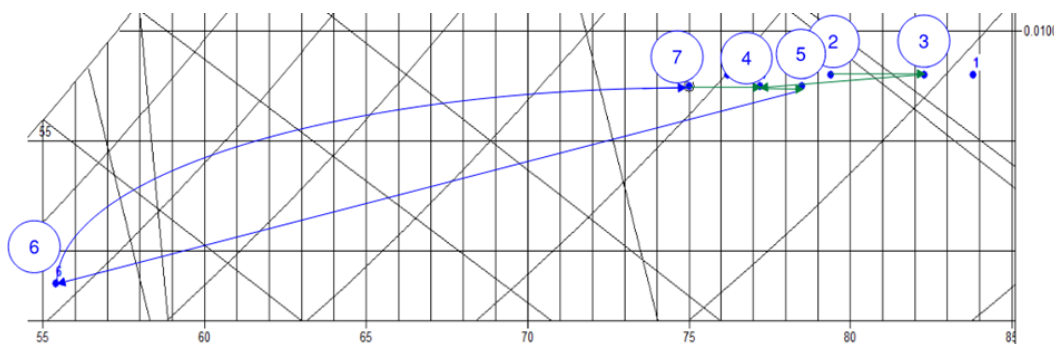


Figure 13 Summer Air Mixing Psychrometric Process at Fan Coil

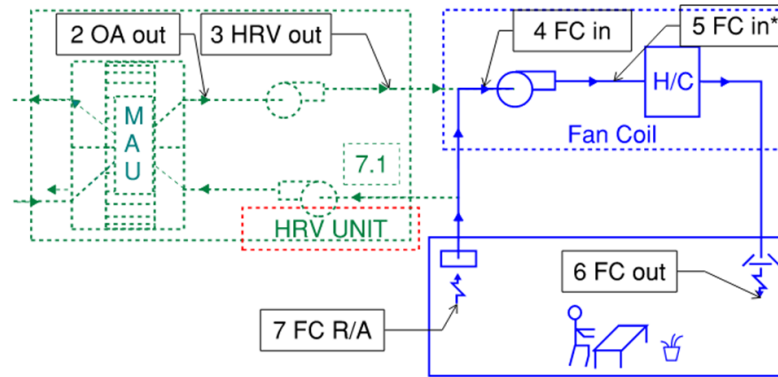


Figure 14 HRV to Fan Coil Schematic

8 Energy Modeling

8.1 Introduction

The energy modelling was performed using IES VE (2024). This software is ASHRAE Standard 140⁶ compliant and specifically chosen due to its ability to simulate in detail, thermal loads, daylighting analysis, and energy consumption; compliant with ASHRAE Standard 90.1 - Section 12 (Energy Cost Budget Method).

The primary objective of the energy modeling was to verify that the proposed HVAC design and building envelope meet or exceed the performance criteria set forth in ASHRAE Standard 90.1, with a focus on optimizing the balance between energy efficiency, occupant comfort, and life cycle costs. The model allowed the team to evaluate the building's predicted Energy Use Intensity (EUI), system performance under various operational loads, and the impact of design choices on heating and cooling demands.

8.2 Methodology

The IES VE model was constructed using architectural drawings provided by the competition and developed further using envelope parameters from ASHRAE Standard 189.1 for Climate Zone 5A (Manchester, UK). The internal loads, including lighting, plug loads, occupancy schedules, and infiltration rates, were defined based on ASHRAE Standards 90.1, 62.1, and 170. Operating hours were set from 7 AM to 7 PM Monday through Friday, and 7 AM to 12 PM on Saturday, as specified in the OPR. The building was assumed to be unoccupied on Sundays. The thermal zoning within the model was aligned with functional zones, including clinical areas, administrative zones, lobbies, laboratories, and support spaces. The energy model was then compared to the baseline system outline in ASHRAE Standard 90.1, shown in Figure 15. Per ASHRAE Standard 90.1 Appendix G, the baseline HVAC system is determined by building type, size, and climate zone. For a medical office in Climate Zone 5A, the baseline is typically System Type 6: VAV with Hot Water Reheat, featuring a central boiler and chiller. This system includes variable air volume distribution, terminal reheat, and the use of economizers where applicable. It serves as the reference for evaluating energy performance and code compliance with the proposed design.

⁶ ASHRAE Standard 140 (2020), defines procedures for validating building energy modeling software; modeling methods in this project align with its principles.

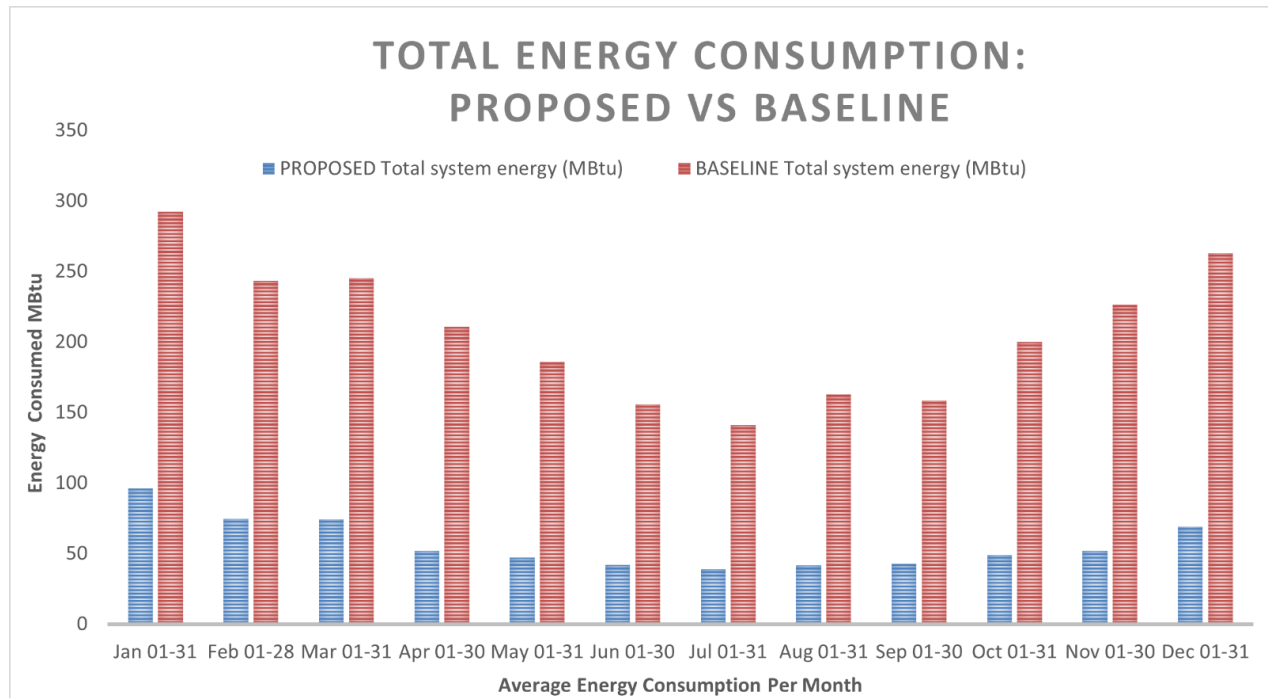


Figure 15 Energy Consumption: Proposed vs Baseline

8.3 Results & Validation

System performance was compared against the proposed design to ensure compliance with the Performance Cost Index (PCI) targets and to validate energy cost savings as outlined in Section 12.5 of ASHRAE Standard 90.1. Annual energy use simulations revealed that the proposed system consumes approximately 678.57 MBtu, compared to 2483.62 MBtu for the baseline system, resulting in an estimated 73% reduction in energy usage. This substantial improvement shows the effectiveness of the proposed design in meeting energy efficiency objectives and long-term operational cost reductions, as discussed in Section 10 & Section 11.

9 ASHRAE Standard Compliance

9.1 Introduction

The following section details how the team took measures to ensure compliance with ASHRAE Standards 55, 62.1, 170, and 90.1. It will detail the PMV⁷ analysis conducted to achieve thermal comfort, the ventilation calculations and considerations to provide the appropriate fresh air, and the performance rating method compliance checklist.

9.2 ASHRAE Standard 55 (2023): Thermal Environmental Conditions for Human Occupant

ASHRAE Standard 55-2023 defines the acceptable indoor environmental conditions required to achieve thermal comfort, based on six key factors: air temperature, mean radiant temperature, humidity, air speed, metabolic rate, and clothing insulation. The standard offers both analytical (PMV) and graphical methods to evaluate comfort in mechanically and naturally conditioned spaces, aiming for at least 80% occupant satisfaction which corresponds with a PMV score within ± 1 .

For this project, the design references Section 5: Conditions That Provide Thermal Comfort and Section 6: Design Compliance. A detailed room-by-room analysis was conducted to identify critical zones most susceptible to thermal

⁷ PMV (Predicted Mean Vote), defined in ASHRAE Standard 55, is a thermal comfort model that predicts the average occupant sensation on a 7-point scale from -3 (cold) to +3 (hot), based on temperature, humidity, air speed, clothing, and activity level.

discomfort due to environmental or spatial factors. These zones were evaluated using ASHRAE Standard 55 criteria, and the results were used to project thermal comfort performance across similar spaces in the building.

Table 7 Occupant Conditions

Space Type	Clothing Level (CLO)		Metabolic Rate (MET)
	Summer	Winter	
Office	0.5	1	1.1 (typing/writing)
Treatment Doctor	0.6	0.9	1.6 (light medical work)
Treatment Patient	0.3	0.6	1.0 (seated/resting)
Waiting/Common Area	0.5	1.1	1.0 (seated/quiet)

Table 8 Thermal Environmental Conditions

Maximum RH		Average Air Speed (fpm)	
Summer	Winter	Summer	Winter
50%	45%	25	22

PMV analysis was conducted to determine the optimal operative temperature for each zone and room type. Operative temperature is influenced by factors such as air speed, room air temperature, and mean radiant temperature, all of which can be adjusted through terminal unit controls.

For instance, in winter operation, the treatment room revealed differing thermal comfort preferences between the doctor and patient, resulting in a narrow-shared comfort zone. This insight informed the need for precise thermal control in spaces with mixed activity levels or clothing insulation.

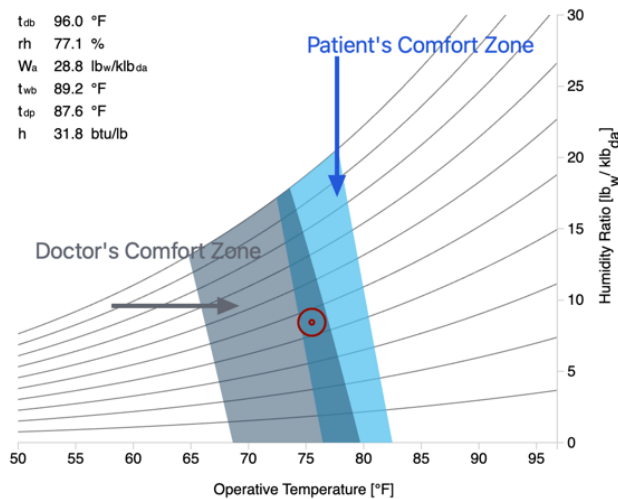


Figure 16 Process to Achieve the Desired Operative Temperature

Our analysis showed that, under typical conditions, patients are more sensitive to changes in mean radiant temperature. Increasing the mean radiant temperature effectively expanded the thermal comfort zone for patients. In rooms located in exterior zones with large windows, the fan coil system alone struggled to maintain an operative temperature that met the comfort needs of both the doctor and patient. To address this, a radiant heater was added to raise the mean radiant temperature, ensuring the desired operative temperature (red target symbol in Figure 16) was achieved for both occupants.

PMV practices were further assisted to define the design set point temperature using

$$t_o = At_a + (1 - A)\bar{t}_r$$



Where:

t_o = Operative Temperature, °F

t_a = Average Air Temperature (Set point temperature), °F

$\bar{t}_r$ = Mean Radiant Temperature, °F

Key thermal comfort parameters for most occupied spaces are summarized below:

Table 9 Operative Temperature in Critical Zones

Space Type	Operative Temperature (°F)		Set Temperature (°F)		MAX RH (%)		Average Air Speed (fpm)	
	Summer	Winter	Summer	Winter	Summer	Winter	Summer	Winter
Office/working Area	80	70	78.6	75	50%	45%	25	22
Treatment / exam (internal)	78	76	74.7	78	50%	45%	25	22
Common / Waiting Area	80	68	78.7	73	50%	45%	25	22
Treatment / exam (External)	78	75	74.7	75	50%	45%	25	22

9.3 ASHRAE Standard 62.1 (2022): Ventilation and Acceptable Indoor Air Quality & ASHRAE Standard 170 (2023): Ventilation of Health Care Facilities

ASHRAE Standard 62.1 provides the baseline for the ventilation rate that the HVAC system must meet and other considerations to maintain proper indoor air quality. Due to the function of the building, ASHRAE Standard 170 is also considered in the load calculation and system design process as a supplement for ASHRAE Standard 62.1.

ASHRAE Standard 62.1 requires the outdoor air to be treated properly before being supplied to the breathing zones. According to the Air Quality & Planning Technical Guidance by Manchester City Council, local PM_{2.5}, PM₁₀, and ozone levels are within the legal range. Thus, no particle filter or air cleaning device is required for ventilation air treatment in rooms of non-medical occupancy.

The standard provides three methods to determine the minimum ventilation rate to maintain the required indoor air quality: Ventilation Rate Procedure (VRP), Indoor Air Quality Procedure (IAQP), and Natural Ventilation Procedure (NVP). Our team has adopted VRP method because IAQP requires additional assumptions for contaminant types and concentrations, while the scope of the NVP method does not align with our design. The steps for the VRP method are presented below.

The Breathing Zone Outdoor Airflow (V_{bz}) is determined by Equation 6-1 from ASHRAE Standard 62.1. The minimum People Outdoor Air Rate (R_p) and Area Outdoor Air Rate (R_a) are extracted from Standard 62.1 Table 6-1 if available.

$$V_{bz} = R_p \times P_z + R_a \times A_z$$

Where:

R_p = People Outdoor Air Rate, cfm / person

P_z = Zone Population

R_a = Area Outdoor Air Rate, cfm / ft²

A_z = Zone Floor Area, ft²

For some room types such as general exam rooms, isolation exam rooms, blood draw rooms, etc., V_{bz} is calculated using an alternative method based on the Minimum Outdoor ACH provided in ASHRAE Standard 170 Table 8-1 and Table 8-2.

$$V_{bz} = A_z * H * ACH / 60$$

Where:

A_z = Zone Floor Area, ft²

H = Space Height, ft

ACH = Minimum Outdoor ACH

For rooms that appear in both ASHRAE Standards 62.1 and 170, V_{bz} will be calculated using both methods, and the higher value will be used for a conservative design.

Zone Air Distribution Effectiveness (E_z) is obtained from Table 6-4 based on the configurations of the supply air terminal units. Zone Outdoor Airflow (V_{oz}) is then calculated using Equation 6-2 in ASHRAE Standard 62.1.

$$V_{oz} = V_{bz}/E_z$$

Given the Zone Outdoor Airflow, the Zone Minimum Primary Airflow can be calculated with Equation 6.9.

$$V_{pz-min} = V_{oz} \times 1.5$$

Our design mainly consists of multiple-zone recirculating systems, for which the Uncorrected Outdoor Air Intake (V_{ou}) is calculated with Equation 6-5.

$$V_{ou} = D \sum_{all\ zones} (R_p \times P_z) + \sum_{all\ zones} (R_a \times A_z)$$

Where:

D = Occupant Diversity Ratio, determined by Equation 6-6.

$$D = P_s / \sum_{all\ zones} P_z$$

Where:

P_s = Total population in the area served by the system, persons

The System Ventilation Efficiency (E_v) also depends on occupant diversity and is calculated with Equation 6-7 and Equation 6-8.

$$E_v = 0.88 \times D + 0.22 \quad (D < 0.60)$$

$$E_v = 0.75 \quad (D \geq 0.60)$$

Then the Outdoor Air Intake can be corrected using Equation 6-10.

$$V_{ot} = V_{ou}/E_v$$

A sample calculation is provided for Zone 10 on the first floor. The terminal unit (fan coil in our design) must comply with the minimum ventilation rate at a zone level, and the HRV must provide the required ventilation rate at a system level.

Table 10 Minimum Ventilation Rates and Equipment Sizes

Zone	Zone Level				System Level				Equipment Size			
	V_{bz} (cfm)	E_z	V_{oz} (cfm)	V_{pz-min} (cfm)	D	V_{ou} (cfm)	E_v	V_{ot} (cfm)	Terminal Unit	Fan Coil Size (cfm)	AHU / HRV	AHU / HRV Size (cfm)
L1-Zone 10	275.84	1	275.84	413.76	1	275.84	0.75	367.79	L1-FC- 17	954	HRV 03	375

ASHRAE Standard 170 provides ventilation requirements specifically for healthcare environments, including outpatient and medical office buildings. The standard outlines minimum outdoor air change rates, pressure relationships, airflow directions, and filtration levels to support infection control, occupant health, and indoor air quality.

For this project, ASHRAE Standard 170 was used to define ventilation rates, air classifications, and space pressurization requirements for critical areas such as exam rooms and treatment rooms.

Positive or negative pressure differentials were implemented, for example, exam rooms were designed to be neutral or slightly positive, while soiled utility rooms and restrooms were maintained under negative pressure to prevent cross-contamination. Additionally, to comply with ASHRAE Standard 170, all outdoor air was filtered to meet minimum MERV-13 filtration, and no air recirculation was permitted in spaces classified as having "critical" air cleanliness requirements such as the Contamination Exam Room.



9.4 ASHRAE Standard 90.1 (2022): Energy Standard for Buildings Except Low-Rise Residential Buildings & ASHRAE Standard 189.1 (2020): Standard for the Design of High Performance, Green Buildings

ASHRAE Standard 90.1 has been a benchmark for commercial building energy codes and a key basis for energy standards worldwide. It establishes minimum energy efficiency requirements for the design and construction of commercial buildings, including new sites, building systems, and major renovations. For the purposes of system design, this project will only focus on the following section in ASHRAE Standard 90.1: 5 – Building Envelope, 6 – Heating, Cooling and Air Conditioning, 9 – Lighting, & 12 – Energy Cost Budget Method. The Baseline system was developed in compliance with ASHRAE Standard 90.1, ensuring optimized energy consumption, efficiency and regulatory adherence. Through iterative improvements and optimization, the finalized design not only met, but exceeded the standards recommendations.

9.4.1 Building Envelope

Minimum building envelope materials used were referenced from the architectural drawings, with U-values and R-values taken directly where specified. All building envelope values meet ASHRAE Standard 189.1 recommendations, which supplant Standard 90.1 for energy standards. The medical office is in Manchester, England, falling under Climate Zone 5A. Further details regarding the building envelope can be found in Section 4.3 of the report.

9.4.2 HVAC System

Both the baseline and proposed HVAC systems were modeled in accordance with ASHRAE Standard 90.1 requirements. While the baseline system reflects the prescribed configuration outlined in Appendix G⁸, the proposed system was verified to be compliant with ASHRAE Standard 90.1 performance criteria. Specifically, the ventilation strategy featuring a MAU and HRVs meets the requirements set forth in Section 6.5.1. Additional details regarding the selected system can be found in Section 7.4 of the report.

9.4.3 Lighting

Lighting Power Densities (LPD) have been selected based on building-specific space type using values from ASHRAE Standard 90.1, Section 9, Table 9.5.2.1-2 as a guide to inform LPD in various rooms and more accurately calculate the lighting load.

9.4.4 Energy Cost Budget Method

Whole-building energy simulations were conducted using IES VE (Compliant with ASHRAE Standard 140), this process allowed the team to verify that the design energy cost met ASHRAE Standard 90.1 Section 12, Table 12.5.1. Special consideration was taken to ensure compliance with the criteria mentioned in ASHRAE Standard Sections 12.2 – Compliance, and 12.4 – Simulation General requirements, including applicable exceptions as outlined in the standard where compliance was not required or has been listed as an assumption. Section 11 of this report provides further insight into the energy usage and associated costs of operation for the proposed system.

10 GHG Emissions Analysis

10.1 Introduction

In alignment with the OPR and the overarching goal of reducing environmental impact, our team conducted a comprehensive Greenhouse Gas (GHG) Emissions Analysis using IES VE. This section details the methodology, modeling parameters, results, and implications of the GHG emissions analysis, and demonstrates how our HVAC system design supports sustainable, low-carbon operations over the building's lifespan.

⁸Appendix G of ASHRAE 90.1 defines the Performance Rating Method, outlining Mandatory Provisions G1.2.1, & Performance Cost Index Calculation methodology G1.2.2, a part of the Performance Rating Method Reference Manual (PRMRM).



10.2 Methodology

Carbon emissions were calculated using IES VE software from the building model with the proposed system as discussed in Section 7, which allowed this team to model the energy consumption and apply fuel-specific emission factors to determine the annual equivalent carbon emitted by this system in lbCO₂e/year.

10.3 Results & Validation

IES VE simulation estimates 174,006 lbCO₂e/year for the proposed system's carbon emissions. Appendix I⁹ of ASHRAE Standard 90.1 details further information for carbon metrics and was used for comparisons in the next section for the introduction of renewables to decrease the equivalent emitted carbon of the proposed system.

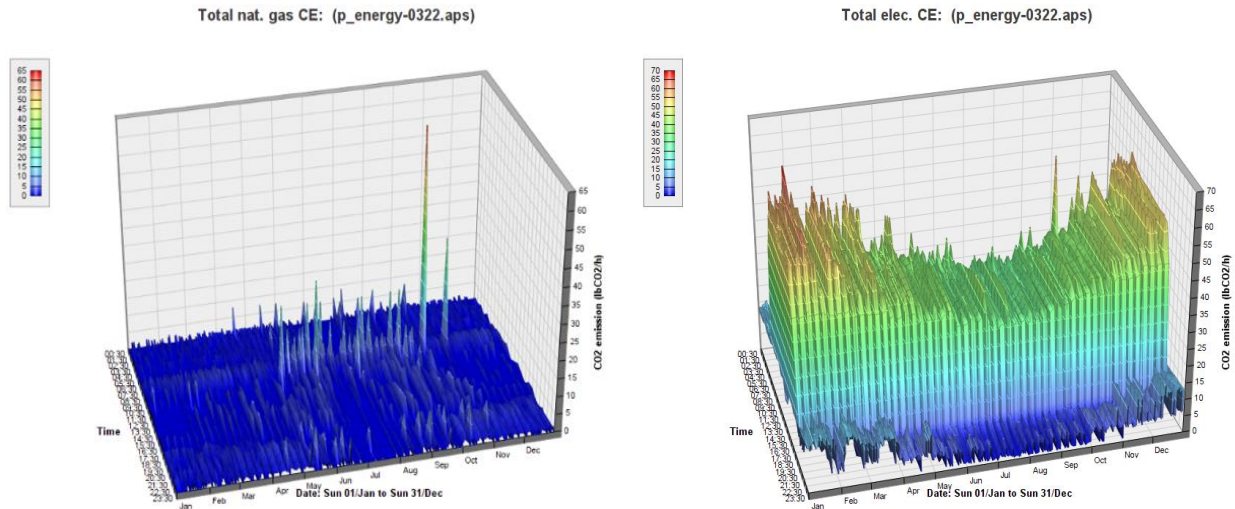


Figure 17 Total Natural Gas & Electricity in equivalent CO₂ emitted by proposed system (lbCO₂e/year)

10.4 Renewable Energy Calculation

Given the solar availability, a rooftop Photo-Voltaic (PV) array was chosen as a suitable renewable energy source. To quantify the impact of renewable integration, annual global solar radiation data for Manchester was obtained from IES VE simulations. This included hourly radiation values (in Btu/ft²) over the course of a full year. The radiation profile clearly peaks between May and July as shown in Figure 18, with the most useful solar gain occurring between 08:00 and 18:00 — aligning well with daytime operational loads. The annual PV energy generation was calculated using the simple PV power output linearized equation:

$$E = A * r * H * PR$$

Where:

A = Area, ft²

R = Panel Yield

H = Solar Radiation, Btu/ft²

PR = Performance Ratio

⁹Appendix I of ASHRAE 90.1, permits carbon emissions as an alternative metric in the Performance Rating Method; this study used cost-based and carbon-based comparisons.

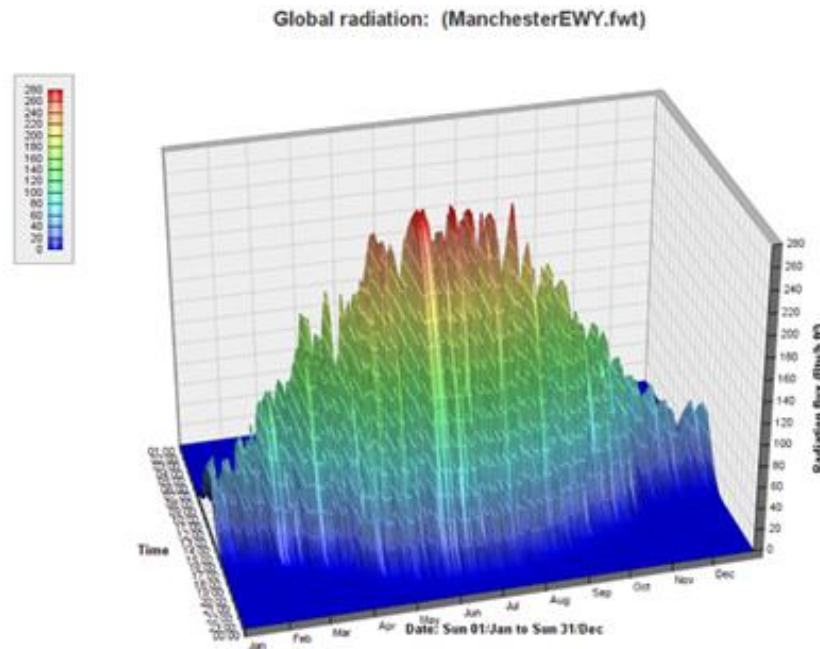


Figure 18 Radiation Profile

Table 11 Assumptions for PV Array and Calculated Results

Metric	Assumed/Calculated
Panel Efficiency	18%
Roof Coverage Area	10,000 ft ²
Performance Ratio (PR)	0.75
Hourly Solar Radiation	IES VE/ASHRAE Weather Data
Power Generated	120,052.87 kWh ~ 409.63 MBtu
CO ₂ e Savings	0.20705 kg CO ₂ e/kWh ¹⁰
CO ₂ e saved (lbCO ₂ e/year)	54,800 lbCO ₂ e/year (31% Reduction)

The integration of on-site PV panels reduces the proposed system's carbon emissions by approximately 54,800 lbCO₂e/year, achieving a 31% reduction in equivalent emitted carbon. In addition to making the building footprint greener, this strategy also results in significant savings on operating costs, as discussed in Section 11.

11 Life Cycle Cost Analysis

11.1 Introduction

A core objective of this project, as defined in the OPR, is to deliver a sustainable, cost-effective HVAC system design for a new three-story outpatient medical office building in Manchester, England. To that end, a comprehensive Life Cycle Cost Analysis (LCCA) was performed in alignment with ASHRAE Standard 90.1. The objective is to evaluate the total cost of ownership over a projected 50-year period, encompassing initial capital costs, operational costs, maintenance, energy consumption, and potential replacement costs.

¹⁰ Emissions factor based on UK Government GHG Conversion Factors for Company Reporting (2024); Electricity Generated: 0.20705 kg CO₂e/kWh. Source: <https://www.gov.uk/government/publications/greenhouse-gas-reporting-conversion-factors-2024>

11.2 Methodology

The total upfront costs for each system alternative were developed initially using IES VE cost reports and were refined to reflect only components relevant to the ASHRAE 2025 Design Competition. The cost of the proposed system was generated from detailed bill-of-materials (BOM) outputs and included mechanical system equipment, distribution systems, lighting, electrical, and installation charges. To maintain consistency across comparisons, installation-related surcharges (contractor's prelims, overhead, inflation, risk, and consultant fees) were proportionally applied to the proposed PV system using the same methodology. For the base system, costs were manually estimated and cross-checked using unit rates and equipment costs from ASHRAE and industry guidance. The assumptions used for present worth calculations, including inflation, escalation rates, and system replacement intervals—were aligned with the OPR as mentioned in Section 4.1 of this report. The methodology follows the principles of ASTM E917-15, accounting for replacement cycles, capital costs, energy usage, and discounted future value of cash flows over a 50-year study horizon.

Table 12 Base System Cost Comparison

System Type	Energy Sources	Major Replacement Cycles	On-site Power Generation	Total Upfront Cost (\$)
Base System	Gas + Grid Elec.	20 Years	-	\$3,114,506.85
Proposed System	Grid Elec.	20 Years	-	\$8,052,368.09
Proposed System w/PV Array	Grid Elec. + PV	20 Years	Available-Solar	\$ 8,423,051.22

The total upfront cost of the proposed HVAC system with PV is estimated to be \$8.4 million, and with roughly 68,000 ft² of occupied space, the total cost equates to \$128.87 per square foot. For electricity, the cost was calculated using a time-of-day rate structure, aligning with the OPR. For natural gas, hourly consumption was converted to volumetric flow (m³) and multiplied by the given rate of \$2.69/m³. See Table. 13 for all parameter assumptions and sources.

11.3 Assumptions

Table 13 is a list of assumptions and parameters used within the life cycle cost analysis.

Table 13 Parameter Assumptions

Parameters	Assumption/Source
Annual Maintenance Costs	Industry Advisors & Research ~ \$30,000/year
Natural Gas – Convert IES VE kBtu/h output into hourly volume of Natural Gas	IES VE ($\frac{1}{10.55} * \frac{m^3}{kWh} * kWh = m^3$)
Electricity – Convert from kBtu/h to kWh	IES VE
Water & Sewage	Assumed \$0.18/ft ² ~ Table 12-1, ASHRAE Design Manual for Hospitals and Clinics
Escalation Rate	Taken as 3% - Escalated Cost _n = Cost _n *(1+0.03)
PV System Power Generated	IES VE Global Radiation Data, assumed 18% PV efficiency, and approx. 10,000 ft ² coverage – Replace every 20 years.
PV Cost per ft ²	\$27.87/ft ² , UK average PV costs ¹¹

¹¹ Typical solar panel system costs are referenced from Renewable Energy Hub UK: <https://www.renewableenergyhub.co.uk/main/solar-panels/how-much-do-solar-panel-systems-cost>

11.4 Results & Validation

As shown in Table 14, the baseline system shows slightly lower electrical costs compared to the proposed heat pump system due to its reliance on natural gas boilers for heating. However, this comes at the expense of much higher natural gas usage, which not only increases total utility costs but also significantly raises the building's carbon emissions. In contrast, the proposed system uses high-efficiency ASHP, which slightly increases electricity consumption but greatly reduces gas dependency. When comparing systems, the proposed HVAC design with photovoltaic (PV) reduced total annual electricity costs by approximately 23.15% compared to the same system without PV. This represents significant operational savings over the system's lifecycle and justifies the inclusion of renewable energy strategies in the design.¹² This not only optimizes operational cost but significantly reduces the building's carbon footprint over its life cycle.

Table 14 Electrical and Gas Costs

System	Electricity	Natural Gas
Baseline	\$157,367.51	\$194,505.42
Proposed	\$191,746.23	\$4,991.47
Proposed w/PV	\$147,362.54	\$4,991.47

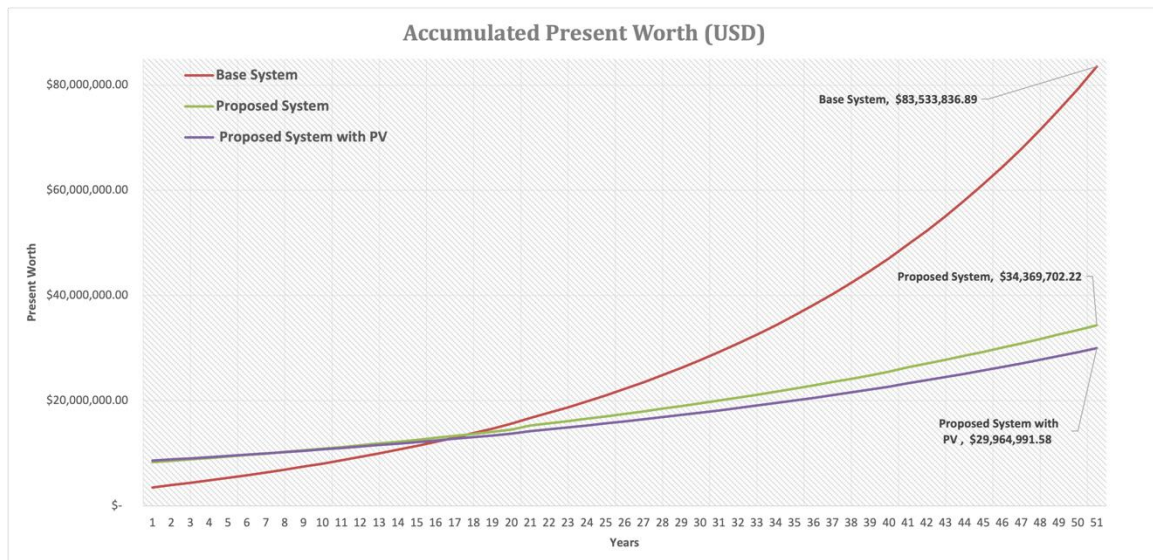


Figure 19 System Comparisons Brought Back to Present Value Sum

12 Conclusion

Through detailed analysis, simulation, and adherence to ASHRAE standards, our team has developed an HVAC system that is both accurate in its load calculation and efficient in operation. Utilizing Carrier HAP and IES VE, we validated our heating and cooling load calculations across multiple tools and hand-checks, achieving discrepancies of less than 10%, confirming the accuracy and reliability of our methodology.

The proposed system, comprising a modular ASHP and WWHP with hydronic distribution, decentralized HRVs, and high-efficiency fan coils, was designed in direct response to the OPR and the building's unique spatial constraints. By integrating reverse return hydronic piping, ECM and VFD pumps, localized radiant heating, and airside economizers, the system achieves high thermal comfort, low energy use, and operational resilience.

¹² The project also considers the Renewable Energy Guarantees of Origin (REGO) scheme in England, which provides information regarding the proportion of electricity sourced from renewables. Thus, considerations given to renewable energy adhere to section 12.4.3.1 – On-Site Renewable Energy and Site-Recovered Energy, and section 12.4.3.2 – was used in determination of Annual Energy Costs.



Our solution demonstrates full compliance with ASHRAE Standards 55, 62.1, 170, and 90.1, confirmed through psychrometric analysis, PMV evaluations, and detailed ventilation and energy modeling. Compared to the ASHRAE Standard 90.1 baseline, our system achieves a 73% reduction in annual energy use, and the inclusion of PV panels further reduces carbon emissions by 31%, aligning with sustainability goals outlined in the OPR.

The life cycle cost analysis supports the implementation of our design, showing significantly lower total present value costs over 50 years compared to the baseline system. Given the strong performance in energy efficiency, occupant comfort, code compliance, and long-term economic benefit, we confidently recommend this HVAC system for implementation in the proposed medical office building.



13 Acknowledgement

The design team extends its gratitude to Dr. Nima Atabaki (Professor at UBC) and Baki Cvijetinovic, P.Eng. (Team Lead at AtkinsRéalis) for their indispensable guidance, detailed technical reviews, and insightful recommendations throughout various project stages. In addition, the team appreciates Justin T.K. Chin, P.Eng. (Principal at Introba) for his valuable design input and industry expertise.



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7. American Society of Heating, Refrigerating, and Air-Conditioning Engineers. (2021). *ASHRAE Standard 169-2021: Standard for Climatic Data for Building Design*
8. American Society of Heating, Refrigerating and Air-Conditioning Engineers. (2023). *ANSI/ASHRAE Standard 55-2023: Thermal Environmental Conditions for Human Occupancy*.

15 Mechanical Drawing Set

SYMBOL SCHEDULE	
PIPING	SYSTEM MONITORING
NEW	NEW
DOMESTIC COLD WATER (DCW)	ROOM TEMPERATURE SENSOR
DOMESTIC HOT WATER (DHW)	REVERSE ACTING TEMPERATURE SENSOR
DOMESTIC HOT WATER RECIRC. (DHW-R)	SLAB TEMPERATURE SENSOR
SANITARY VENT	HUMIDITY SENSOR
SANITARY SEWER ABOVE GRADE	CO ₂ SENSOR
SANITARY SEWER BELOW GRADE	AIR FLOW METER
STORM SEWER ABOVE GRADE	FLOW SWITCH
STORM SEWER BELOW GRADE	PIPE TEMPERATURE SENSOR
PIPE CLEAN-OUT	PRESSURE SENSOR
PIPE CLEAN-OUT TO GRADE	
PIPE LINE	
WET SPRINKLER LINE	
DRY SPRINKLER LINE	
HYDRONIC HEATING WATER SUPPLY	
HYDRONIC HEATING WATER RETURN	
CHILLED WATER SUPPLY	
CHILLED WATER RETURN	
CONDENSATE DRAIN	
HEAT RECOVERY SUPPLY	
HEAT RECOVERY RETURN	
CONDENSER WATER SUPPLY	
CONDENSER WATER RETURN	
FITTINGS AND VALVES	
DIRECTION OF FLOW	
PIPE DROP	
PIPE RISE	
PIPE TEE UP	
PIPE TEE DOWN	
PIPE UNION	
ISOLATION VALVE (NORMALLY OPEN)	
ISOLATION VALVE (NORMALLY CLOSED)	
CHECK VALVE	
2-WAY CONTROL VALVE	
3-WAY CONTROL VALVE	
BALANCING VALVE	
PRESSURE REDUCING VALVE (PRV)	
POOL FLOW CONTROL VALVE	
STRAINER	
RELIEF VALVE	
BACKFLOW PREVENTOR (BFP)	
AUTOMATIC AIR VENT (AAV)	
SEISMIC GAS SHUT-OFF VALVE	
TEMPERATURE GAUGE	
PRESSURE GAUGE	
THERMOMETER	
PUMP	
ENERGY METER	
BTU METER	
OUTLETS AND DRAINS	
OPEN DRAIN	
HOSE-BIBB (HB)	
FLOOR DRAIN (FD)	
FUNNEL FLOOR DRAIN	
ROOF DRAIN (RD)	
AREA DRAIN	

MECHANICAL ABBREVIATIONS	
AD AREA DRAIN	INV INVERT
AFF ABOVE FINISHED FLOOR	JS JANITOR SINK
AHU AIR HANDLING UNIT	KW KILOWATT
ARCH ARCHITECTURAL	KS KITCHEN SINK
BB BASEBOARD HEATER	LV LAVATORY
BOD BACKDRAFT DAMPER	LAT LEAVING AIR TEMPERATURE
BF BOTTLE FILLER	LBS POUNDS
BFP BACKFLOW PREVENTER	LWT LEAVING WATER TEMPERATURE
BHP BREAK HORSEPOWER	MAU MAKE-UP AIR UNIT
BTUH BRITISH THERMAL UNIT - HOUR	MAX MAXIMUM
CD CONTROL DAMPER	MH MANHOLE
CE CLEANOUT	MBSH 1000 BRITISH THERMAL UNITS/HOUR
CFM CUBIC FEET PER MINUTE	MD MOTORIZED DAMPER
CLG CEILING	MECH MECHANICAL
CM MINIMUM	MU MAKE-UP MECHANICAL WATER
CON CONNECTION	NFHB NON FREEZE WALL HYDRANT
COW COMPLETE WITH	NC NOT IN CONTRACT
CONT CONTINUATION	NC NOISE CRITERIANORMALLY CLOSED
DB DRY BULB	NO NORMALLY OPEN
CTE CONNECT TO EXISTING	NTS NOT TO SCALE
DCV DOMESTIC COLD WATER	OIA OUTDOOR AIR
DDC DIRECT DIGITAL CONTROL	ODD OPPOSED BLADE DAMPER
DEG DEGREE	EAD EXHAUST AIR DUCT
DF DRINKING FOUNTAIN	OD OUTSIDE DIAMETER
DHW DOMESTIC HOT WATER	POI PLUMBING AND DRAINAGE INSTITUTE
DA DIAMETER	POC POINT OF CONNECTION
DN DOWN	PRV PRESSURE REDUCING VALVE
DW DISH WASHER	PSI POUNDS PER SQUARE INCH
DWG DRAWING	RA RETURN AIR
DWHR DOMESTIC HOT WATER RECIRCULATION	RF RETURN FAN
EA EXHAUST AIR	RM ROOM
EAT ENTERING AIR TEMPERATURE	RRM REVOLUTIONS PER MINUTE
EF EXHAUST FAN	RWL RAIN WATER LEADER
EFF EFFICIENCY	SIA SUPPLY AIR
ELEC ELECTRICAL	SH SHOWER
ENT ENTERING	SK SINK DRAIN ABOVE
ESP EXTERNAL STATIC PRESSURE	SS STAINLESS STEEL
EWT ENTERING WATER TEMPERATURE	SP STATIC PRESSURE
EXH EXHAUST	SPEC SPECIFICATION
F FIRE MAIN	SR SANITARY RISER
FD FLOOR DRAIN	ST STORM MAIN
FE FIRE EXTINGUISHER	TIA TRANSFER AIR
FLA FULL LOAD AMPS	TAD TRANSFER AIR DUCT
FLOOR	TBC TO BE CONFIRMED
FT FEET/FOOT	TD TO BE DETERMINED
GAL GALLONS	TD TRENCH DRAIN
GPM GALLONS PER MINUTE	THRU THROUGH
GWB GYPSUM WALL BOARD	TS TAMPER SWITCH
HD HUB DRAIN	TSP TOTAL STATIC PRESSURE
HB HOSE BIBB	TYP TYPICAL
HP HORSEPOWER	UR URINAL
HCR HEATING COIL RETURN	V VENT
HCS HEATING COIL SUPPLY	VFD VARIABLE FREQUENCY DRIVE
HRR HEAT RECOVERY RETURN	VTR VENT THROUGH ROOF
HRS HEAT RECOVERY SUPPLY	W WATER MAIN
HWR HEATING WATER RETURN	WB WET BULB
HWS HEATING WATER SUPPLY	WC WATER CLOSET
ID INSIDE DIAMETER	WCO WALL CLEANOUT
IE INVERT ELEVATION	SAD SUPPLY AIR DUCT
IN INCH	

LEGAL ADDRESS		
THE BUILDING IS LOCATED IN MANCHESTER, ENGLAND		
DRAWING LIST		
DRAWING NO.	DRAWING TITLE	SCALE
M0.00	COVER	NTS
M1.00	LEVEL 1 HVAC	1/8" = 1'-0"
M1.01	LEVEL 2 HVAC	1/8" = 1'-0"
M1.02	LEVEL 3 HVAC	1/8" = 1'-0"
M2.00	LEVEL 1 HYDRONIC	1/8" = 1'-0"
M2.01	LEVEL 2 HYDRONIC	1/8" = 1'-0"
M2.02	LEVEL 3 HYDRONIC	1/8" = 1'-0"
M3.00	ROOF LEVEL HVAC & HYDRONIC	1/8" = 1'-0"
M3.01	VENTILATION SCHEMATIC & DETAILED MECH	NTS
M3.02	HYDRONIC SYSTEM SCHEMATIC	NTS
M4.00	SCHEDULES	NTS
M4.01	DETAILS	NTS
M4.02	SPEC	NTS
CODE COMPLIANCE		
THIS PROJECT IS IN COMPLIANCE WITH ASHRAE 55 (2023), 62.1 (2022), 90.1 (2022), 189.1 (2020), & 170 (2021)		





UBC

CLIENT: 

PROJECT TITLE:
2025 ASHRAE STUDENT COMPETITION

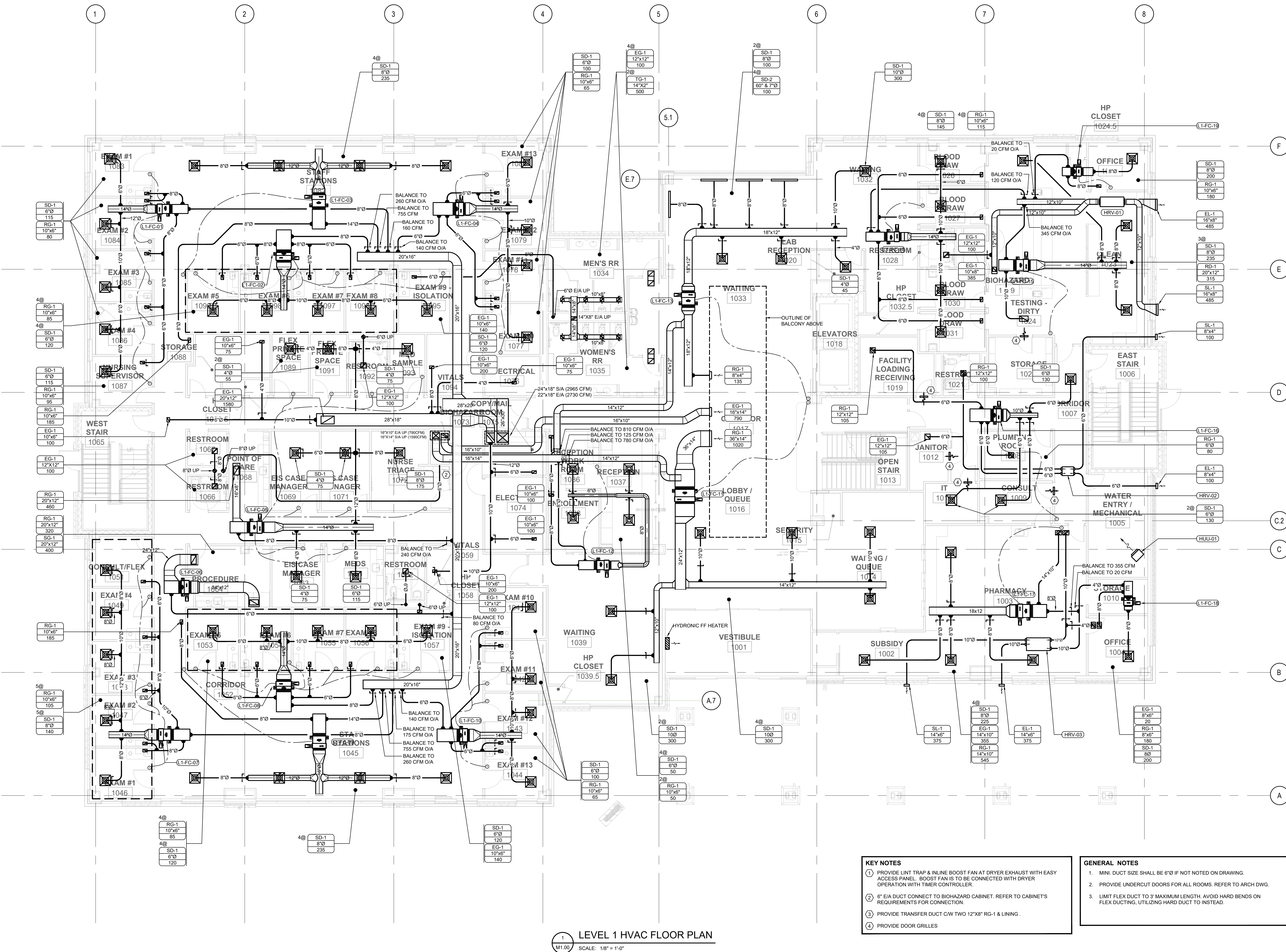
UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

DRAWN BY: L2. M.M. H.Y.
CHECKED BY: N.A.
SCALE: AS NOTED
DATE: May 3rd, 2025

DRAWING TITLE:
COVER PAGE

PROJECT NO.: 001
DRAWING NO.: M0.00





UBC

CLIENT: 

PROJECT TITLE:
2025 ASHRAE STUDENT COMPETITION

UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

DRAWN BY: L2. M.M. H.Y.
CHECKED BY: N.A.
SCALE: AS NOTED
DATE: May 3rd, 2025

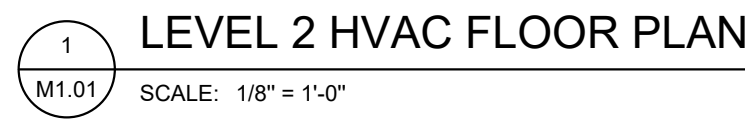
DRAWING TITLE:
LEVEL 1 HVAC FLOOR PLAN

PROJECT NO.: 001
DRAWING NO.: M1.00

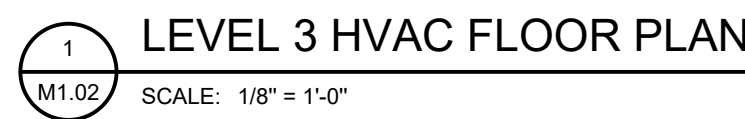




PROJECT NO.	DRAWING NO.
001	M1.01



1. MINI. DUCT SIZE SHALL BE 6"Ø IF NOT NOTED ON DRAWING.
2. PROVIDE UNDERCUT DOORS FOR ALL ROOMS. REFER TO ARCH DRAWING.
3. LIMIT FLEX DUCT TO 3' MAXIMUM LENGTH. AVOID HARD BENDS ON FLEX DUCTING, UTILIZING HARD DUCT TO INSTEAD.



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2. PROVIDE UNDERCUT DOORS FOR ALL ROOMS. REFER TO ARCH DRAWING.
3. LIMIT FLEX DUCT TO 3' MAXIMUM LENGTH. AVOID HARD BENDS ON FLEX DUCTING, UTILIZING HARD DUCT TO INSTEAD.





PROJECT NO.	DRAWING NO.
001	M1.02



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REV.	DATE	DESCRIPTION
1.	2024-09-23	ISSUED FOR DRAFT ZONING
2.	2025-03-12	ISSUED FOR RFS REVIEW
3.	2025-04-02	ISSUED FOR REVIEW
4.	2025-05-03	ISSUED FOR ASHRAE SUBMISSION

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CLIENT:

ASHRAE

SCALE:

PROJECT TITLE:

2025 ASHRAE STUDENT COMPETITION

UBC DESIGN TEAM

PROJECT ADDRESS:

Manchester, England

DRAWN BY:

L.I. M.M. H.Y.

CHECKED BY:

N.A.

SCALE:

AS NOTED

DATE:

May 3rd, 2025

DRAWING TITLE:

LEVEL 1 HYDRONIC PLANS

PROJECT NO.:

001

DRAWING NO.:

M2.00



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SCALE:

PROJECT TITLE:

2025 ASHRAE STUDENT COMPETITION

UBC DESIGN TEAM

PROJECT ADDRESS:

Manchester, England

DRAWN BY:

L.I. M.M. H.Y.

CHECKED BY:

N.A.

SCALE:

AS NOTED

DATE:

May 3rd, 2025

DRAWING TITLE:

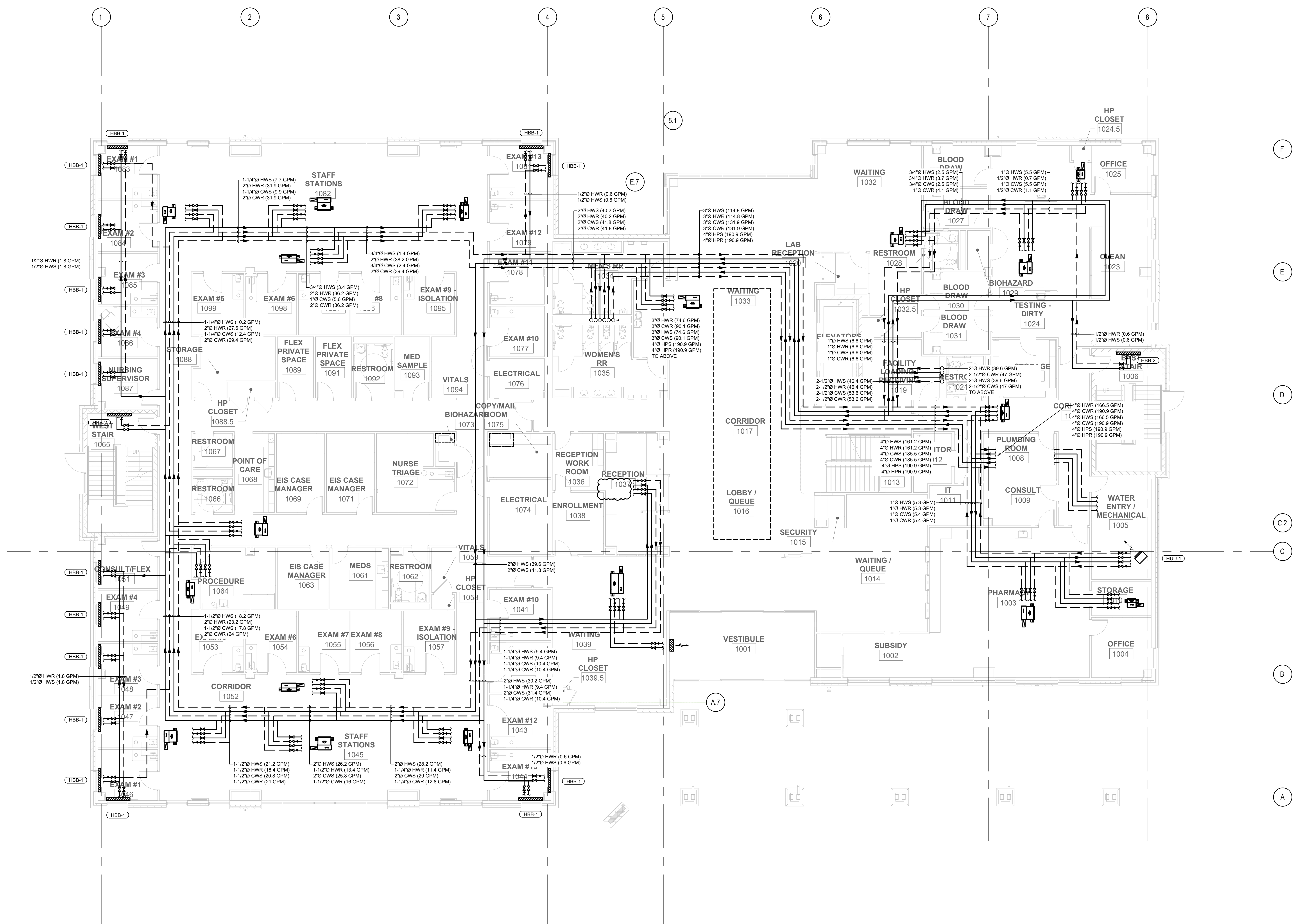
LEVEL 2 HYDRONIC PLANS

PROJECT NO.:

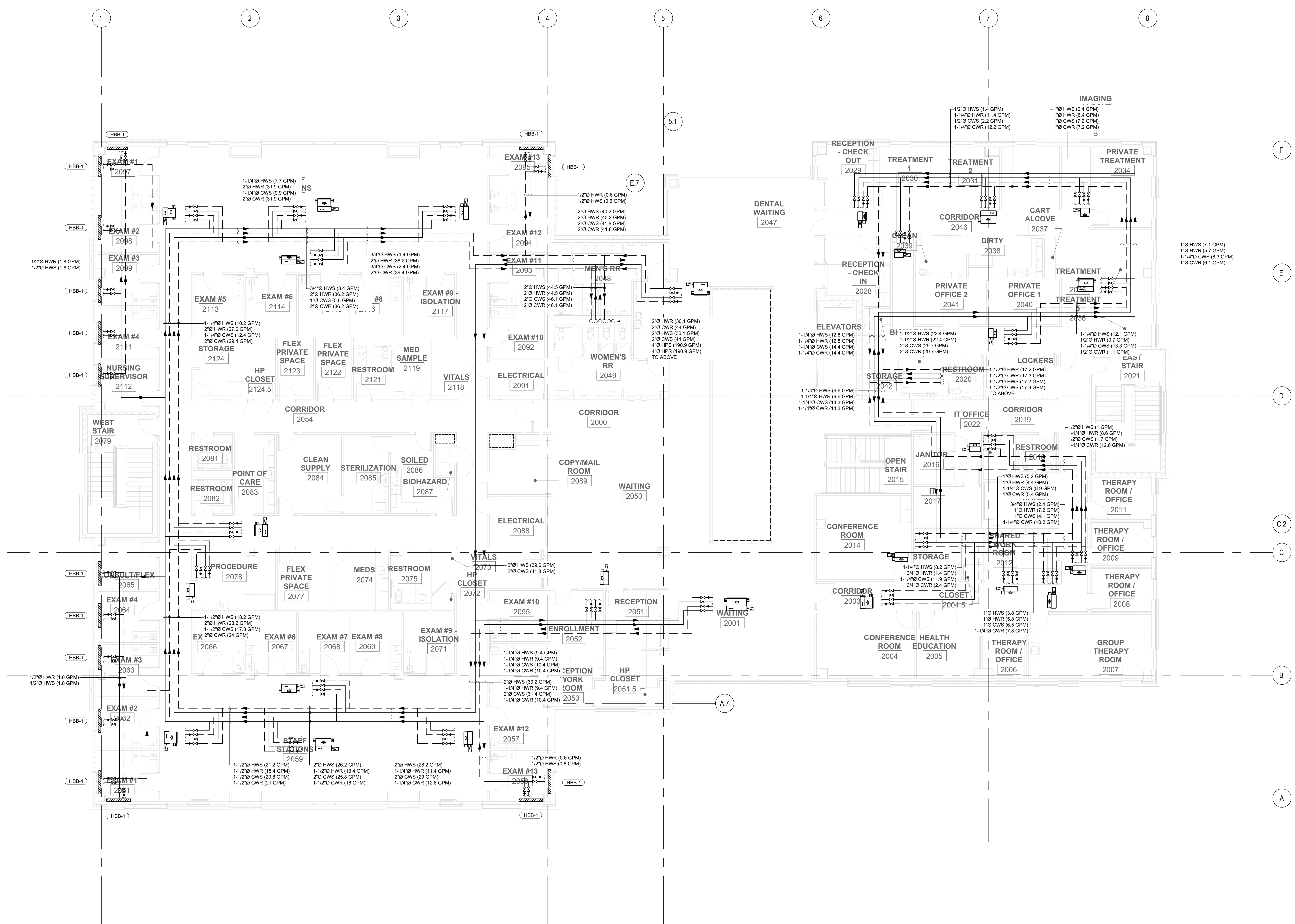
001

DRAWING NO.:

M2.01



LEVEL 1 HYDRONIC PLANS
SCALE: 1/8" = 1'-0"



LEVEL 2 HYDRONIC PLANS
SCALE: 1/8" = 1'-0"



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3.	2025-04-02	ISSUED FOR REVIEW
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CLIENT:



SEAL:

PROJECT TITLE:
2025 ASHRAE STUDENT
COMPETITION

UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

DRAWN BY	L.Z. M.M. H.Y.
CHECKED BY	N.A.
SCALE	AS NOTED
DATE	May 3rd, 2025

DRAWING TITLE:
LEVEL 3 HYDRONIC
PLANS

PROJECT NO.	DRAWING NO.
001	M2.02



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CLIENT:



SEAL:

PROJECT TITLE:
2025 ASHRAE STUDENT
COMPETITION

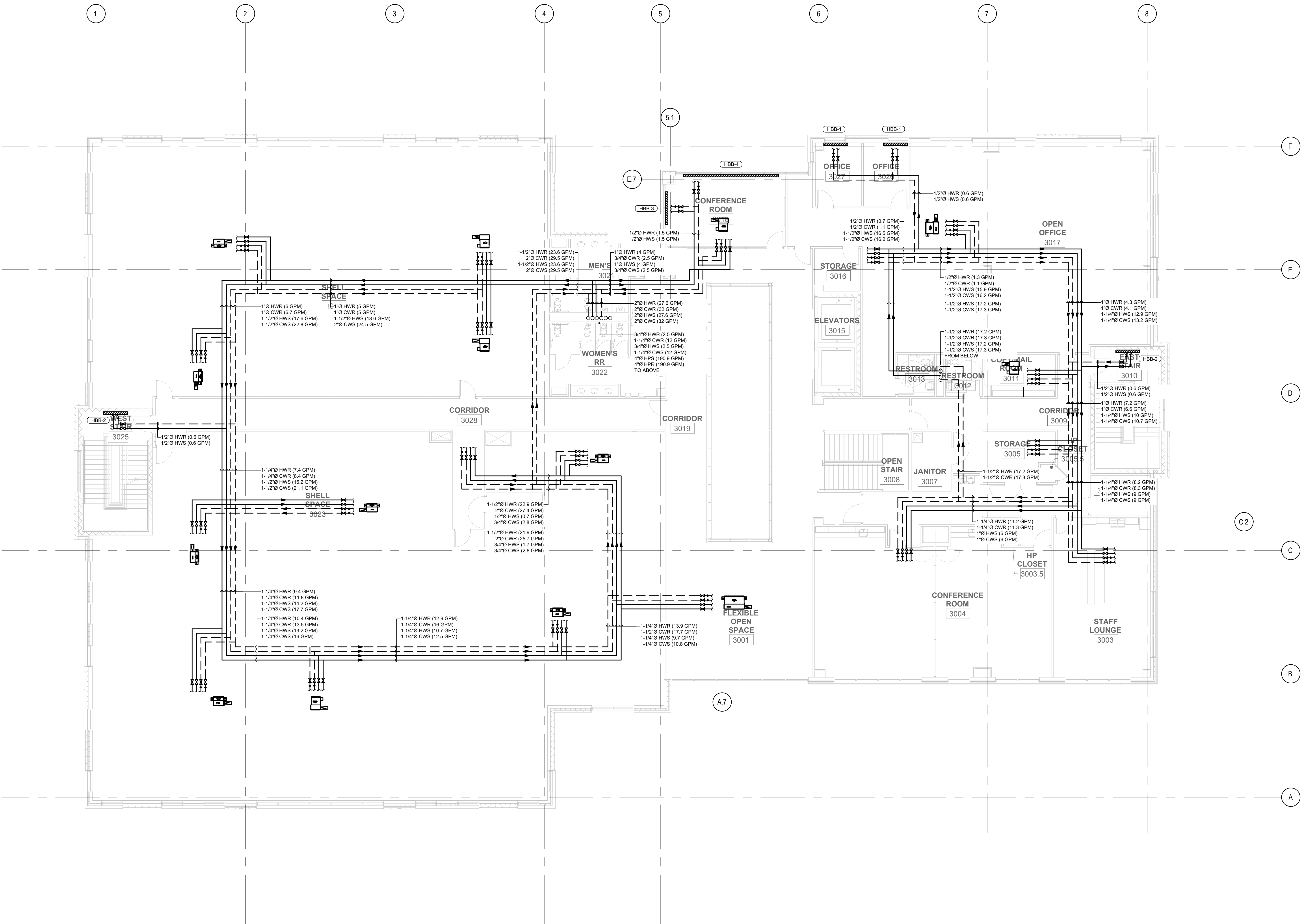
UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

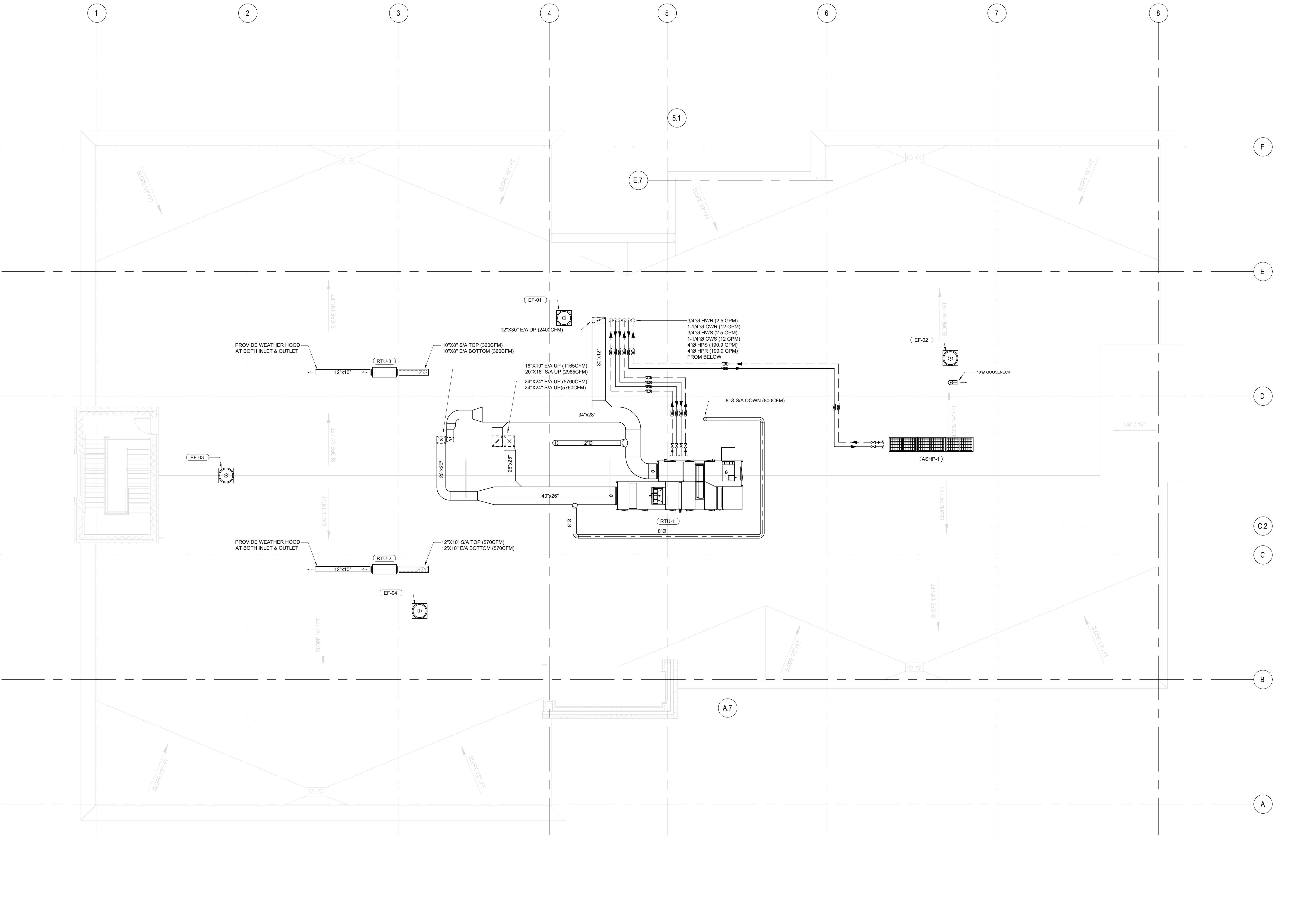
DRAWN BY	L.Z. M.M. H.Y.
CHECKED BY	N.A.
SCALE	AS NOTED
DATE	May 3rd, 2025

DRAWING TITLE:
ROOF PLAN

PROJECT NO.	DRAWING NO.
001	M2.03



1 LEVEL 3 HYDRONIC PLANS
M2.02 SCALE: 1/8" = 1'-0"

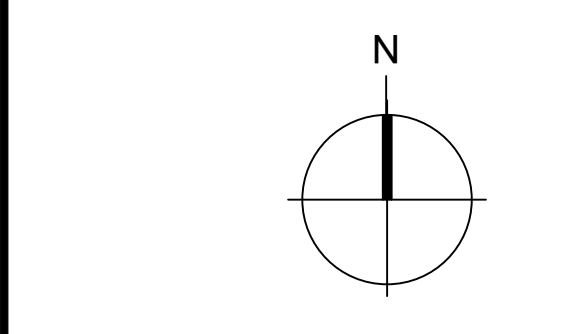


1 ROOF PLAN
M2.03 SCALE: 1/8" = 1'-0"



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CLIENT:
ASHRAE

SEAL:

PROJECT TITLE:
2025 ASHRAE STUDENT COMPETITION

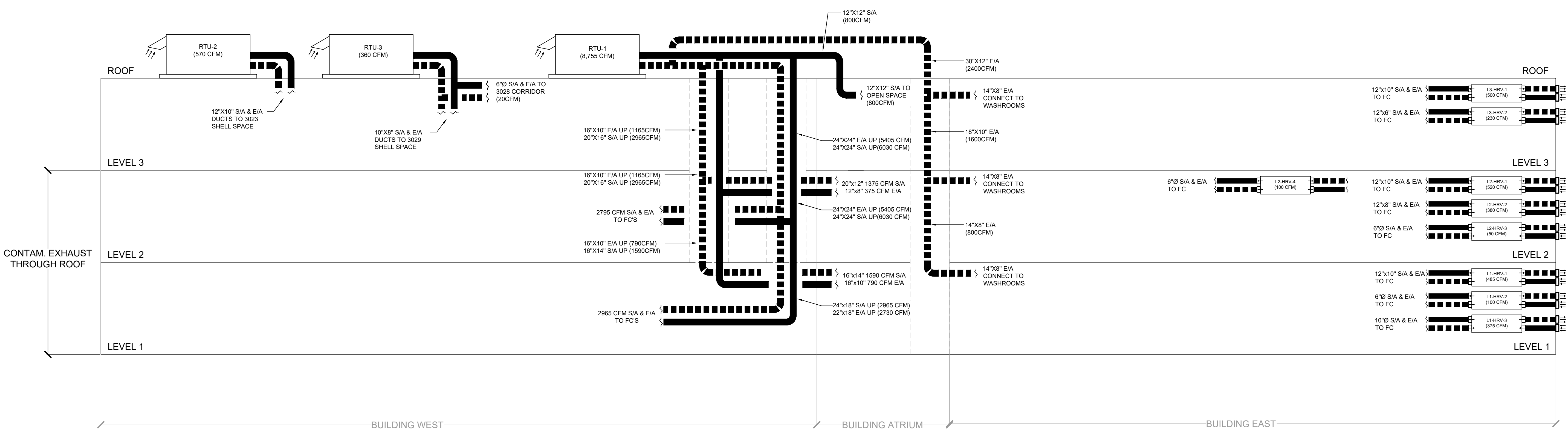
UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

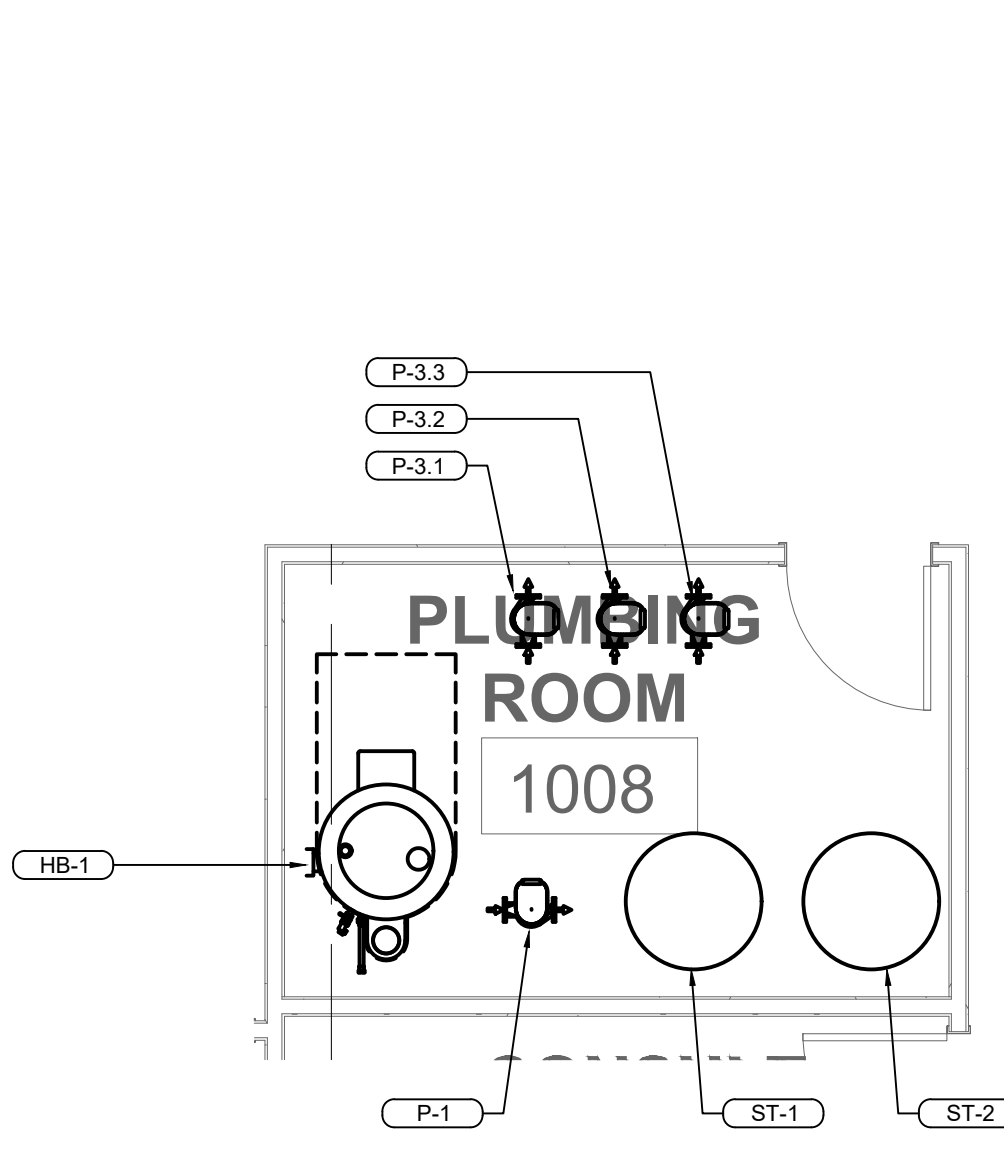
DRAWN BY: L2: M.M. H.Y.
CHECKED BY: N/A
SCALE: AS NOTED
DATE: May 3rd, 2025

DRAWING TITLE:
VENTILATION SCHEMATIC & DETAIL MECH ROOM

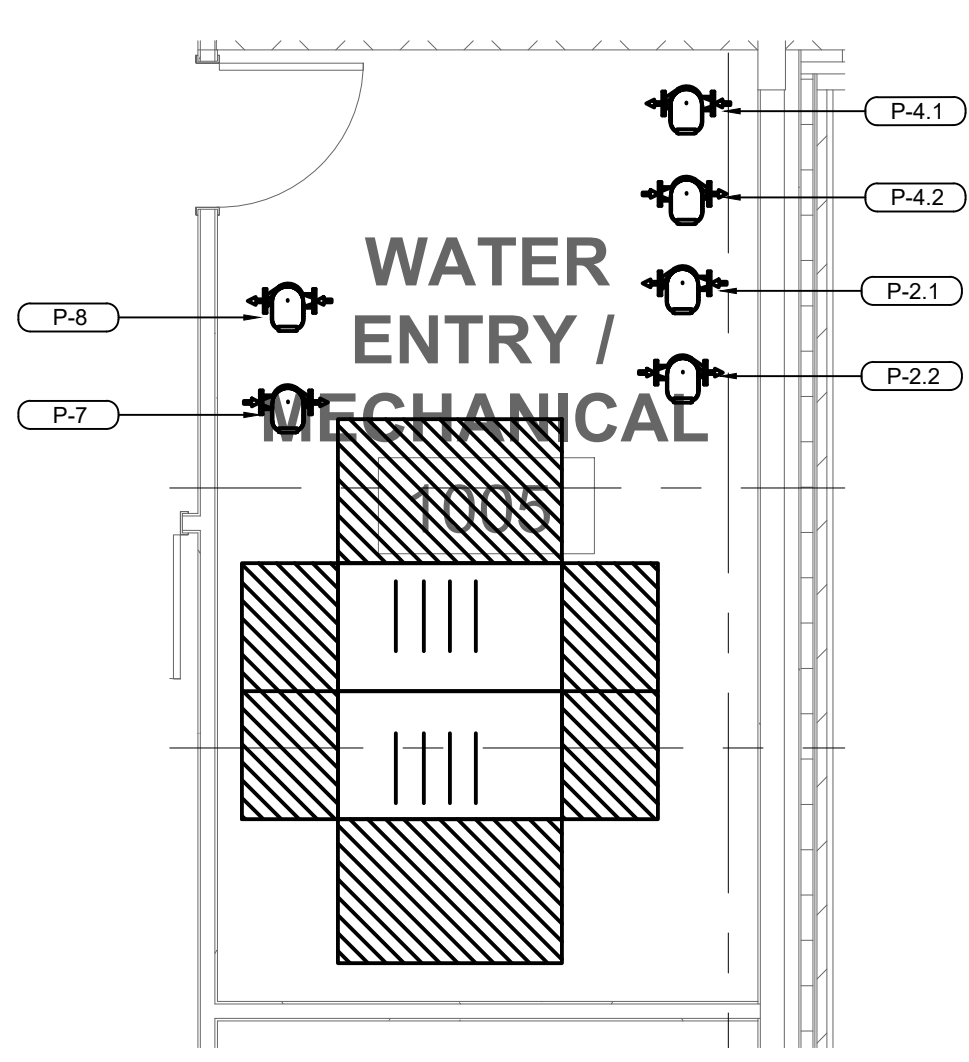
PROJECT NO. 001
DRAWING NO. M3.00



VENTILATION SCHEMATIC
SCALE: 1/8" = 1'-0"

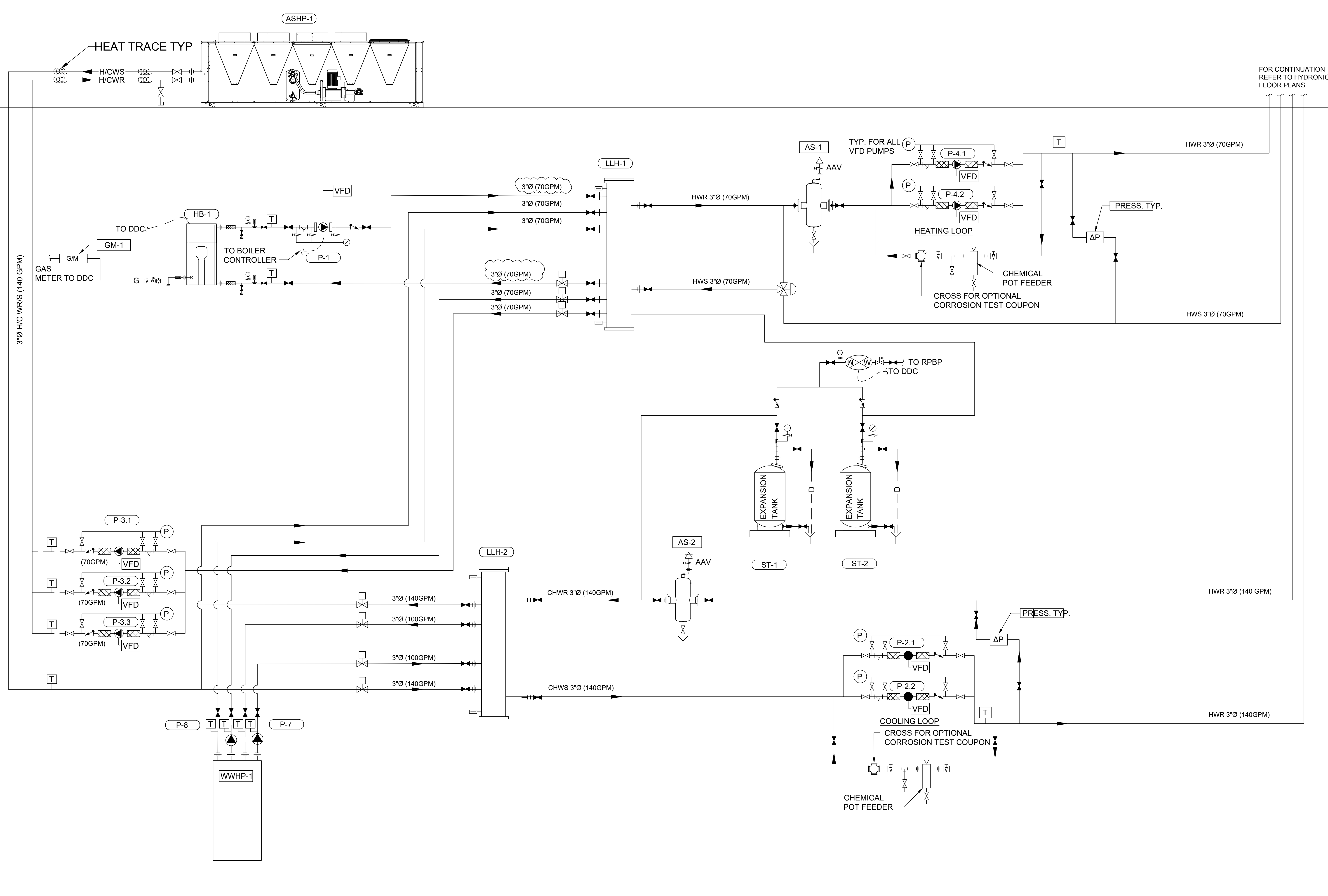


PLUMBING ROOM DETAIL
SCALE: 1/4" = 1'-0"



MECHANICAL ROOM DETAIL
SCALE: 1/4" = 1'-0"

GENERAL NOTES
1. REFER TO HYDRONIC SCHEMATIC FOR DETAIL PIPE SIZING & PIPING.
2. CONTRACTOR SHALL PROVIDE AND INSTALL ALL NECESSARY ACCESSORIES FOR EACH SHOWN EQUIPMENT.
3. REFER TO ARCH & STRUCTURAL DRAWINGS FOR CONCRETE PAD LOCATION.

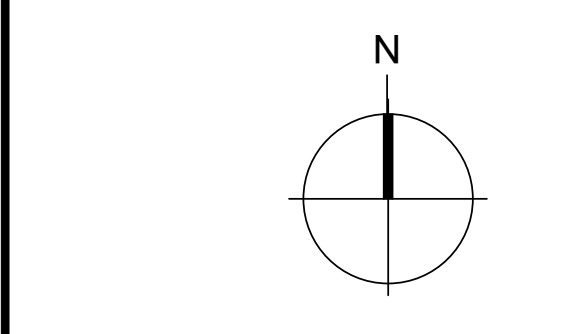


HYDRONIC SCHEMATIC
SCALE: N.T.S.



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SEAL:

PROJECT TITLE:
2025 ASHRAE STUDENT COMPETITION

UBC DESIGN TEAM

PROJECT ADDRESS:
Manchester, England

DRAWN BY: L2: M.M. H.Y.
CHECKED BY: N/A
SCALE: AS NOTED
DATE: May 3rd, 2025

DRAWING TITLE:
SYSTEM SCHEMATIC

PROJECT NO. 001
DRAWING NO. M3.01



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COOLING										HEAT										METER				REMARKS																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																																					
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