2013 ASHRAE Competition

HVAC Design Calculations

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Table of Contents

Executive Summary 1
Location and Environmental Conditions 2
Design Criteria 2
System Sizing 3
Heating and Cooling Load Calculations 3
Zoning 3
Modeling 3
Alternative Energy Conservation Measures 7
Illustrated Cooling and Heating Sample Calculations 8
Summary of Ventilation and Duct Sizing Calculations 8
Summary of Pipe Sizing Calculations 8
Design and Analysis 9
System Selection 9
Equipment Selection 10
Energy Analysis 11
Economic Analysis 12
ASHRAE Standard Compliance 13
ASHRAE 55 13
ASHRAE 62.1 15
ASHRAE 90.1 15
Conclusion 15
Acknowledgments 16
References 16

Executive Summary

The contents of this report provide a summary of the process that we used to design the HVAC system for the Dallas Power and Light building using Ottawa, Ontario's weather conditions. Ottawa's climate conditions were used because it facilitates the design of an HVAC system with significant heating and cooling, rather than primarily cooling in the climate of Dallas, Texas. The main factors in the design were Location and Environmental Conditions, Design Criteria, System Sizing, and ASHRAE Standard Compliance. The Location and Environmental Conditions and Design Criteria sections of the report outline the framework that we used for our design given our choice of climate and the competition requirements from both the judging criteria and Owner's Project Requirements (OPR). The System Sizing section provides insight into the heating and cooling load calculations, our sample hand calculations used to verify the validity of the software, and the piping and ducting transit calculations. The Design and Analysis section details our final plant system type selection, major equipment selection, energy analysis of the building, and economic analysis of equipment choices. The ASHRAE Standard Compliance section elaborates on how we met ASHRAE 55-2010, ASHRAE 62.1-2010, and ASHRAE 90.1-2010. Mechanical drawings have also been provided to illustrate the final design.

Trane's software package with TRACE 700, Trane Pipe Designer, and VariTrane Duct Designer was used. TRACE 700 was used to determine the heating, cooling, and ventilation loads. The pipe and duct design software (verified with hand calculations) was used to determine the losses from transit for sizing and pressure loss calculations. AutoCAD was used to generate mechanical drawings detailing all of the HVAC related mechanical equipment within the building.

ASHRAE 90.1-2010 Zone 6A R-values were assumed for all constructions because there was not enough information given in the OPR to simulate the construction on a layer by layer basis, and a baseline model using the prescribed minimum efficiencies was used to determine the efficiency of the design. ASHRAE 62.1-2010 ventilation calculations were applied to the software to ensure minimum flow rates were maintained, and diffuser and grille placements were done with care to avoid discomfort and air contamination. ASHRAE 55-2010 compliance was demonstrated using an operative temperature calculation for the southern zone of the retail space that is heated with radiant panels.

Preliminary modelling using eQuest suggested that ground-source heat pumps would provide significant energy savings over a traditional boiler and chiller system, but further analysis demonstrated that the payback would be over 15 years for a vertical well system. Additionally, a horizontal well system would not be practical given the building's lack of underground parking and location in a downtown core. This led to the choice of a boiler and chiller based hydronic plant. The biggest design challenge for this building was the limited mechanical space available for large plant equipment and exhaust ducting. As a result, an air-cooled chiller was selected to be placed on the roof, and high efficiency, small footprint boilers were to be placed in Maintenance 113. Air-cooled chillers typically have lower Energy Efficiency Ratings (EERs) than the water-cooled variety, but in the case of the Dallas Power and Light building, there was little space for a cooling tower.

The results of our load calculations yielded building cooling and heating loads of 26 Btu/hr-ft$_2$ and 28 Btu/hr-ft$_2$ respectively. We applied these values when selecting the final design. The final design for the building was an air-cooled heat recovery chiller for cooling, and high efficiency condensing boilers for heating. Heat recovery was implemented via heat pipes to allow for washroom exhaust recovery. Hydronic radiant panels were used for skin heating in the first floor retail space to lower the room air temperature and maintain occupant comfort. When compared to the ASHRAE 90.1-2010 baseline our design was 8% more efficient. This is a considerable amount of savings given the constraints we had on mechanical space and terminal unit selection for the Ottawa climate.

Location and Environmental Conditions

This year's competition gave us the option of relocating the Dallas Power and Light building to the capital city of our country for weather conditions, which allowed us to relocate to the climate of Ottawa, Ontario Canada. Ottawa is situated in Zone 6A (as per ASHRAE 90.1-2010) and sees both humid summers and sub-arctic winters. This comparatively adverse climate offered our design team a chance to showcase a greater knowledge of HVAC equipment than if the building were located in cooling dominated Dallas, Texas. 2% Design criteria was used for the summer (as per the Owner's Project Requirements), and 99% criteria for the winter as outlined in Table 1. The choice of using 99% criteria for winter was to provide an additional safety factor for the cold winter that can reach -6˚F. For the purposes of software and hand calculations, Typical Meteorological Year (TMY) data from Ottawa International Airport was used.

Table 1: Design Temperatures for Ottawa, Ontario

Season	Degree Days	Design Temperatures
Winter	8571 HDD	-6˚F @99%
Summer	2045 CDD	81˚F DB @2% 67˚F WB @2%

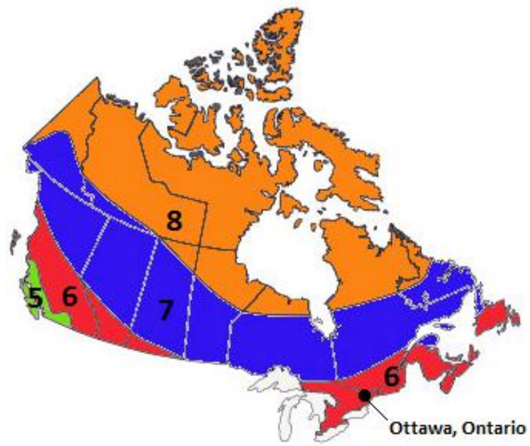


Figure 1: Canadian Climate Zones

Design Criteria

The Judging criteria outlines how the final design will be evaluated, but the Owner's Project Requirements (OPR) gives the framework for how the design should be implemented. In order to win the competition both sets of requirements must be satisfied. The Owner's Project Requirements are as follows:

- HVAC equipment must have low impact on energy and water consumption.
- Strict acoustic criteria must be met (less than NC 30 for the suites and NC 40 for the retail space).
- Ventilation must provide excellent air quality.
- Mechanical equipment must be "near invisible", but accessible for good maintenance.
- Each apartment must have individual comfort control.
- Ceiling height must be maximized.
- Mechanical space must be minimized.
- Retail space HVAC equipment must have a flexible design to account for changes in use and occupancy.

The judging criteria is shown in Table 2. As implied by the rubric, half of the total points are allocated to sound technical design (System Sizing, ASHRAE Standards), and the other half is allocated to creativity and presentation. The sections that follow will describe how our design fulfills these objectives.

Table 2: ASHRAE HVAC Design Calculations Competition Judging Criteria

Category	Maximum Points
System Sizing	300
ASHRAE Standards	200
Creativity	200
Communication of Results	300
TOTAL	**1000**

System Sizing

Heating and Cooling Load Calculations

Correct modeling of the building's heating and cooling loads is critical to producing a good design. Proper equipment sizing is important in keeping capital costs as low as possible while ensuring the majority of the building's occupants are comfortable.

Trane's TRACE 700 software and its accompanying pipe and duct designer software was selected to execute the heating and cooling equipment, pipe, and duct sizing for the Dallas Power and Light Building. Understanding the methodology behind the software's calculations is essential to good design. As such, TRACE 700's calculated loads were cross-checked with hand calculations as outlined in the Illustrated Cooling and Heating Sample Calculations section.

Terminal equipment was sized using peak loads, while the plant equipment was sized to meet the maximum block load of the building and to account for load diversity and setback conditions. This accounts for the fact that different faces of the building will see their respective peak loads at different hours of the day. The ratio of the block load to the peak load gives a diversity factor that can be applied to other portions of the HVAC system (pumping, fans, etc.).

Zoning

The figures below show the zone division of characteristic floors of the building. The first floor Variable Air Volume (VAV) system is divided based primarily on tenant and exterior wall orientation, while the other floors that contain suites are divided based on suite (with some bedrooms solely occupying a control zone to provide better granularity in cases where the exterior wall orientations of the main suite and bedroom are not concurrent).

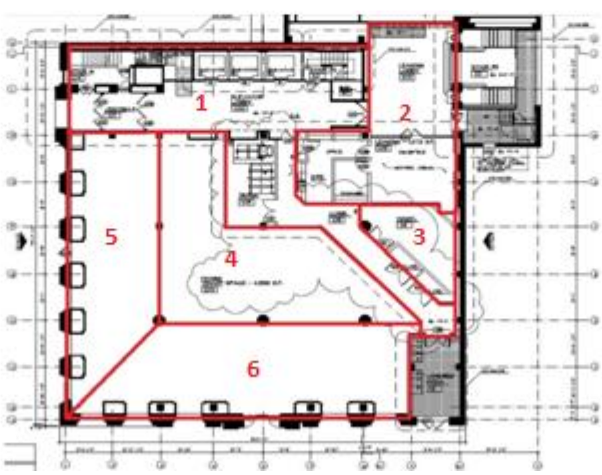


Figure 2: Retail Zoning Diagram

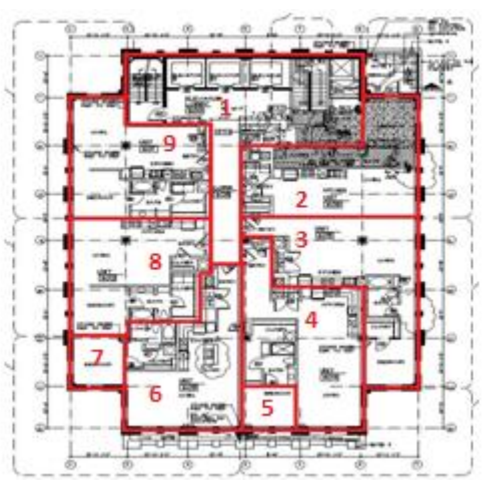


Figure 3: Residential Zoning Diagram

Zoning on the multi-floor suites (Level 17/18 and 19/20) was broken down into one zone per floor, unless high glazing loads required further zone breakdown (i.e. Suite 1903).

Modeling

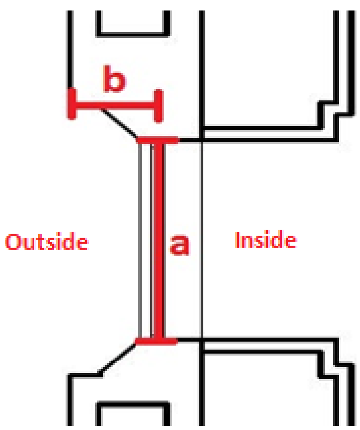
The first step to creating a good energy model is to verify the information that has been given by the project stakeholders, assemble the local climate conditions, and to state and validate design goals. Table 3 illustrates those values and assumptions.

Table 3: Input Summary Table

	Design Model Characteristics
General	
Location	Ottawa, Ontario
Simulation Weather File	Ottawa, Ontario
Climate Zone	ASHRAE Climate Zone 6
Modeling Software	TRACE 700
Hours of Operation	Residential: 24 hours (with reduced occupancy in working hours) Retail: 8am-9pm
Utility Rates	Electrical: On-peak Consumption - $0.085/kWh Off-peak Consumption - $0.045/kWh On-peak Demand - $5.75/kW Gas: Consumption - $0.75/Therm
Floor Area	108,000 ft^2
Envelope Performance	
Overall Roof U-value (Btu/hr·^{o}K·ft^2)	U-0.048 (R-20)
Overall Wall U-value (Btu/hr·^{o}K·ft^2)	U-0.071 (R-15.2)
Percentage Glazing	Overall: 27%
Overall Glass U-value including frame (Btu/hr·^{o}K·ft^2), and Solar Heat Gain Coefficient (SHGC)	U-0.478 $SHGC_{all}$-0.4 Double low-e air filled
Internal Loads	
Occupancy Density	Bedroom/Living: 100 ft^2/person Kitchen: 50 ft^2/person Lobby: 33 ft^2/person Mechanical: 200 ft^2/person Office: 200 ft^2/person Retail: 67 ft^2/person Storage: 500 ft^2/person Restrooms: 1 person/fixture
Lighting Power Density (LPD)	Bedroom/Living: 1.11 W/ft^2 Corridor: 0.66 W/ft^2 Kitchen: 0.99 W/ft^2 Lobby: 0.90 W/ft^2 Mechanical: 0.95 W/ft^2 Office: 1.11 W/ft^2 Restrooms: 0.98 W/ft^2 Retail: 1.68 W/ft^2 Storage: 0.63 W/ft^2
Plug-Loads (EPD)	Office: 0.50 W/ft^2
Domestic Hot Water Usage	2100 Btu/h.Person
Mechanical Systems	
Indoor Design Temperature	Occupied: 70°F/75°F Unoccupied: 64°F/81°F
Humidity Control	Humidification: Electric High-limit: 60% Low-limit: 30%
System Description	Residential: Rooftop Dedicated Outdoor Air System (DOAS) with fan coil terminal Retail: Air Handling Unit (AHU) with reheat
Fan Power & Control	Fans run continuously during occupied hours and cycle on/off to meet the heating/cooling loads during unoccupied hours. Design air flows are based on a supply-air-to-room-air temperature difference of 20°F AHU-1 (Suites) Fan total static pressure: Supply: 1.88 in.wg HR Return: 1.97 in.wg AHU-2 (1st Floor Retail) Fan total static pressure: Supply: 1.98 in.wg HR Return: 1.54 in.wg

	Design Model Characteristics	
	Fan total power: Supply: 7.5 HP HR Return: 7.5 HP Fan total air flow: Supply: 12650 cfm HR Return: 9500 cfm	Fan total power: Supply: 5 HP HR Return: 5 HP Fan total air flow: Supply: 6100 cfm (+1700 cfm relief) HR Return: 7800 cfm
Ventilation Controls	Zone level thermostat	
Heat Recovery	Air-to air heat recovery with 56.2% sensible effectiveness.	
Central Plant		
Heating Type	Natural gas condensing boilers with 91% efficiency	
Hot Water Loop	Temperature: Design supply: 160°F Design loop DT: 20°F Hot water supply temperature is reset based on load.	
Cooling Type	Air cooled heat recovery chiller: 9.93 EER	
Chilled Water Loop	Temperature: Design supply: 44°F Design loop DT: 10°F Chilled water supply temperature is reset based on outdoor air temperature.	
Domestic Water Heater	Gas-fired domestic water heater with 96% efficiency	
Pumps	Primary/Secondary pumps are modeled with Premium-efficiency motors and Variable-Speed drives.	

In addition to meeting all of the relevant ASHRAE standards (55-2010, 62.1-2010, and 90.1-2010), physical features of the building must be considered. The combination of punched windows and thick mass wall has left the majority of the building's windows with some usable shading represented as "overhangs" and "fins". The model accounts for the shading that is generated by these features on a face-by-face and floor-by-floor basis to provide the most accurate estimation of load reduction. An illustration of the effective portions of the overhangs can be seen in Figure 4.



- Depth of punched window recession from envelope edge is measured (b), along with window width (a).
- Measurement is repeated in the vertical plane.
- Shading scheme is used to generate shading coefficients that reduce direct solar loads.

Figure 4: Typical 1st Floor Punched Window Indentation (Plan View)

The heavy construction of the Dallas Power and Light building that was common in the 1930's also provides a "thermal mass effect" that delays and reduces the peak cooling load from the peak envelope load. This effect allows the building to store radiant heat that is projected into the building at peak solar angle times (noon for South faces, morning for East faces, evening for West faces, and sunrise/sunset for North faces), and use convection to release it later. The walls of the building act as "thermal masses" and store energy. As a result, the peak load for the HVAC system can be lower than the peak load of the space. The time-variant properties for various materials are documented in the ASHRAE Fundamentals handbook, and are the basis for Radiant Time Series (RTS) cooling load calculations used in the TRACE 700 software. We used the RTS method for this project because of its accuracy and popularity in industry.

In addition to standard infiltration caused by wind and pressure differences between both sides of the envelope, tall and relatively narrow buildings such as the Dallas Power and Light building tend to exhibit symptoms of the stack effect. The stack effect is a phenomenon where excessive infiltration combining with warm air rising causes the shafts of the building to draw air upward and generate a negative pressure in the lower portion of the building, and higher pressure at the top. It can be particularly prevalent in cold climates such as Ottawa. We used the stack effect calculation from ASHRAE Fundamentals 1997 to find the flow rate at design conditions. The equation we used is as follows:

$$Q = 60 \times C_d \times A \times \sqrt{2 \times g \times (H_n - H_b) \times (T_i - T_o/T_i)}$$

With the following variables:

Table 4: Stack Effect Inputs

Variable	Q	C_d	A	T_i	T_o	H_n	H_b	g
Meaning	Stack flow rate	Drag coefficient	Opening area (air gap)	Indoor temp	Outdoor temp	Height of "neutral plane"	Height of base opening	Gravity
Assumed Value	2450 CFM	0.65	2 ft^2	68˚F	-6˚F	110 ft	0 ft	32.2 ft/s^2

As Observed in Table 4, we found the stack effect flow rate to be just below 2500 CFM at heating design conditions. When compared with the general infiltration flow rate of 14,500 CFM obtained from the TRACE 700 model (also at heating design conditions), the stack effect can be neglected. Additionally, fan coils with corridor service were slightly oversized for ventilation to pressurize the corridors in an effort to reduce infiltration.

The monthly heating and cooling equipment loads follow the energy usage profile shown in Figure 5. The building's end use breakdown of energy consumption is shown in Figure 6. Note the high heating load in both figures. This was considered in the choice of high efficiency condensing boilers.

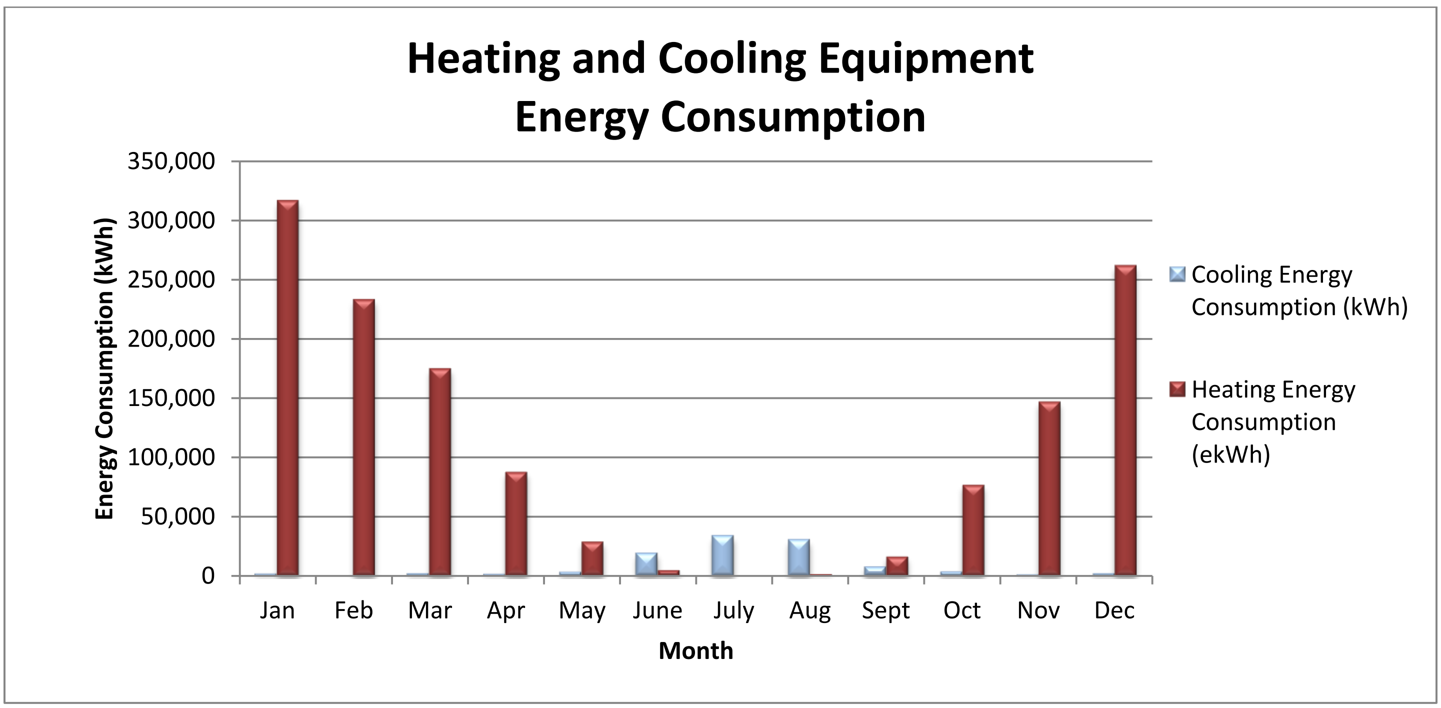


Figure 5: Monthly Breakdown of Heating and Cooling Equipment Energy Consumption

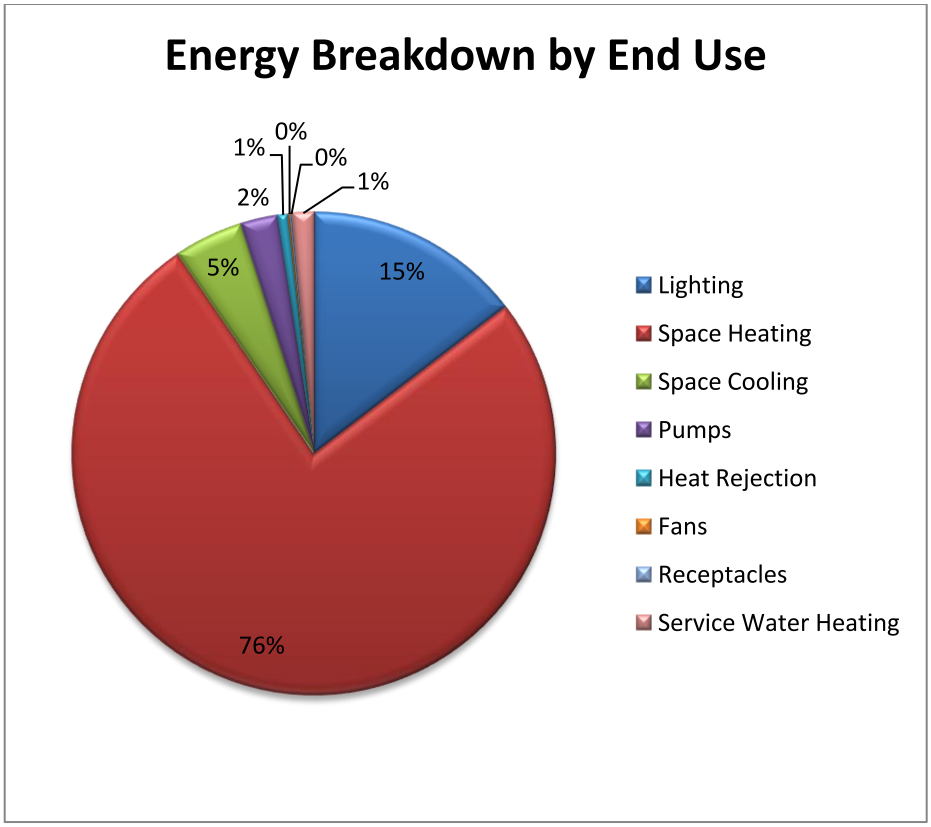


Figure 6: Breakdown End Use Energy Consumption

Due to the cold/dry winter climate and hot/humid summer climate of Ottawa, Ontario, it was found that there was both a significant heating and cooling season. The heating season's intensity initially brought the relative humidity of the supply air down to 20%, but the addition of domestic hot water line supplied humidifiers to the air handling units brought the Relative Humidity (RH) back to a reasonable level (30-60%).

The building's block loads and engineering checks from TRACE 700 are summarized in Table 5 and Table 6 respectively. Note that a safety factor of 10% was used on heating equipment, and the 2 boilers were sized so that each could provide 55% of the building load to prevent equipment damage in the case of a malfunction. The cooling equipment was not oversized because cooling is a less critical function in Ottawa.

Table 5: Building Block Loads

End Use	Block Load
Heating	3,100 MBh
Cooling	200 Tons
Ventilation (Total Supply)	110,000 CFM
Ventilation (Outdoor Air)	16,200 CFM

Alternative Energy Conservation Measures

Two alternative Energy Conservation Measure (ECM) models were run to demonstrate opportunities where significant lifetime savings could be obtained based on Dallas Power and Light management decisions. These alternatives included consideration of shading from the 37-storey Whitacre Tower located less than 50 feet West of the Dallas Power and Light tower, as well as the selection of triple-paned windows in response to the cold winters of Ottawa.

Table 6: Engineering Checks for a West Facing Suite (Suite 1001)

Check	Value
CFM/ft$_2$ (Cooling)	0.91
Btu/hr-ft$_2$ (Cooling)	26.32
CFM/ft$_2$ (Heating)	0.91
Btu/hr-ft$_2$ (Heating)	28.05

Shading from the Whitacre Tower could provide considerable reduction in cooling loads for the West exterior wall. This would affect roughly one third of the building's suites, and could reduce the capital cost of the HVAC system. The drawback would be that if the Whitacre Tower was unforeseeably removed, the equipment would no longer be adequate for the building's loads. If accounted for, the shading from the Whitacre Tower could have reduced our cooling equipment size by 8%.

Triple-paned windows are seeing frequent use in arctic and sub-arctic climates such as the Yukon Territory and Ontario. The energy savings from adding an additional inert space between the environment and the conditioned space are undeniable, but higher capital costs could give this ECM a negative Net Present Value (NPV). We found that the use of moderately tinted Triple-paned windows would reduce our heating and cooling equipment size by 14% and 25% respectively.

Illustrated Cooling and Heating Sample Calculations

Accurately evaluating heating and cooling loads is imperative as it governs whether equipment is sized adequately. As such, the Trace 700 software used to assess heating and cooling loads must be validated. The east-facing bedroom of suite 306 was chosen for a simple analysis for which calculations were performed.

Before cooling loads are calculated, the sol-air temperature must be determined for the hottest month using Ottawa's design conditions at 2% criteria. The Radiant Time Series (RTS) and Conductive Time Series (CTS) methods were applied to find the heating and cooling loads in both the model and the hand calculations. In the comprehensive analysis, infiltration, lighting, and occupant heat gains were also considered for cooling loads. Alternately for heating, solar and internal heat gains are neglected as building structures and contents are considered to have zero thermal inertia. The following table outlines the heating and cooling loads from manual calculation, Trace 700, and the difference.

As observed in the calculations, the differences are relatively small, and any deviation is likely caused by small rounding errors propagating throughout the computation. For a summary of the block loads please refer to Table 5 in Modeling.

Table 7: Comparison of TRACE 700 Loads and Hand Calculated Loads

Cooling Load	**Trace 700**	**Manual Calculation**	**Difference (%)**
Wall (Btu/hr)	190	198	4.1
Window (Btu/hr)	2077	2273	8.6
Heating Load			
Wall (Btu/hr)	501	495	1.1
Window (Btu/hr)	1008	1126	10.5

Summary of Ventilation and Duct Sizing Calculations

The ducting layouts were drawn keeping in mind the requirements of maximized ceiling height, almost invisible mechanical equipment, and low noise criterion. For the most part, the ducting was run through areas where a dropped ceiling would be acceptable, in the kitchens, bathrooms, closets and entry corridors. The maximum ceiling height for the residential space is 12 ft. In the areas where the ceiling was dropped, it reaches a height of 8 ft.

The duct sizing was done with the VariTRANE Duct Designer software, which allows the user to dictate the maximum section velocity, section elevation, inline fittings, duct roughness and many other important parameters. The resulting duct sizes were randomly checked using a hand calculator where the required air quantity was set with the desired velocity. When the duct sizes were compared using both the software and hand calculator they often differed by only half a size.

Summary of Pipe Sizing Calculations

The hydronic system is designed to minimize the number of branches into individual suites. As such, the main pipes run along the corridor and branch off into suites as smaller networks when necessary. Pipe sizing was performed using Trane Pipe Designer. The software optimizes the required size based on flow rates, frictional losses, and material cost. Additionally, it provides the critical path of the network with information on pressure drop and flow rates. A simple closed-loop system was implemented to verify the validity of the software. The Darcy-Weisbach equation and the methods outlined in Chapter 22 of ASHRAE Fundamentals 2009 were used to derive the head loss and pipe diameter.

Overall, the results had a difference of approximately 10%. This mostly originates from the assumptions made (e.g. fitting losses, steel pipe surface roughness). However, the findings did indeed show expected behavior when pipe diameter was changed, and prove that the software utilizes similar theories in determining pressure losses. To reduce pressure drop, pipes have been upsized to the next largest size wherever reasonable. Doing so will help reduce flow velocity and noise, and ultimately the lifetime energy consumption. This method of pipe upsizing is common practice in Europe where energy costs hold more weight in equipment sizing.

The head loss and flow rate were used as the primary factors in pump sizing. Due to the boiler rooms location on the ground floor, the additional hydrostatic pressure from the riser column needs to be accounted for. This approximates to an additional 235 ft w.g.

Additionally, Variable Speed Drive (VSD) pumps were chosen to reduce pumping utility costs. Sizing of VSD pumps is a greater challenge than sizing single speed pumps because there are several different systems curves. As such, the VSDs were optimally sized for the most common loads. This provides less than ideal efficiencies at peak loads, but in the most common operating band greater savings will be observed. Analysis of shoulder season pumping loads is the simplest way to locate this "sweet spot", and therefore the method we used.

Design and Analysis

After taking into account the building loads a system was selected to best fit the needs of the owner, while considering environmental impacts. Aside from the terminal fan coil and VAV units required by the OPR, the main distinguishing features of our HVAC system design are as follows:

- (x2) Viessmann Vitocrossal 1700 Mbh natural gas condensing boilers with 91% efficiency to reduce fossil fuel use.
- (x1) Airmec 200 ton air-cooled heat recovery chiller with 9.93 EER for reduced electrical consumption footprint.
- (x2) Air handling units with heat pipe air to air heat recovery at 56.2% sensible efficiency to reduce overall heating energy consumption.
- Hydronic radiant heating panels ceiling mounted in ground floor retail space along the perimeter to reduce heating consumption and improve occupant comfort.
- Retained "bonus bedroom" to increase saleable space by locating mechanical equipment in Maint. 113 and on Roof.

The following sections outline the methods behind design choices, energy analysis, and the economic impact analysis.

System Selection

To arrive at the conclusion that a gas-fired boiler and air-cooled chiller plant system is the best choice for the Dallas Power and Light building in the Ottawa climate some preliminary assessments of applicable systems were run. Some systems that are applicable for residential and retail plants in climate Zone 6 that had been considered were:

- Gas-fired boiler and air-cooled or water-cooled chiller
- Gas-fired boiler and absorption chiller
- Ground-source heat pump
- Water-source heat pump
- Air-source heat pump

To determine the most suitable candidate, preliminary energy models were run using eQuest. The early models did not have the room by room detail of the final model, so the zoning was instead broken into percentages of total building area by space type. Different plant equipment was interchanged for the same ASHRAE 90.1-2010 Zone 6 compliant building envelope to limit the variables in the model.

The air-source heat pump would not function at a higher Coefficient of Performance (COP) than 1 in the winter at the design condition (acting as an electric heater), and the water-source heat pump system had higher utility costs than a traditional hydronic boiler and chiller system with higher initial costs. Additionally, the absorption chiller did not pay back when compared to the hydronic boiler and chiller baseline. This narrowed the selection down to either a Ground-source Heat Pump (GSHP) or boiler and chiller system. It was found that due to the requirement of vertical ground wells for the Dallas Power and Light building's cityscape, the payback on the GSHP system over 15 years when compared to a boiler and chiller, which is not a reasonable payback for individually owned residential buildings. When selecting a chiller the limited space for HVAC equipment inside the building (and on the roof) necessitated the use of an air-cooled chiller to save the space that would be required by a cooling tower.

The decision to use a gas-fired boiler and rooftop air-cooled chiller combination was justified by both economics and space constraints. The largest challenge was meeting the needs of the building in the Ottawa climate under the space constraints of a building architecturally designed for Dallas, Texas.

Equipment Selection

As a result of the clients mandate of Fan Coil Units for the suites and VAV Units for the first floor retail, the hydronic plant equipment selection is one of the most critical choices. In the case of Ottawa, Canada, the choice of moving a building designed architecturally for a cooling dominated climate to a heating dominated climate with significant cooling loads presented additional challenges. One of the most prudent was fitting large hydronic equipment within small spaces. This was the motivation behind the choice of a rooftop air-cooled chiller. The limited roof space necessitated using an air-cooled system over a water-cooled one that required an additional cooling tower. In order to make up for our chiller's 9.93 EER, a heat recovery model was implemented to contribute to the heating loads in the shoulder seasons. Analysis of the cost of installing the necessary equipment for the heat recovery chiller gave us a payback period of 13 years and an NPV of $3,358 over the life of the building. This is with the consideration of additional piping costs and the fuel (natural gas) savings for when the chiller waste heat production was equal or greater than the building heating load so the boiler could be turned down.

Locating the chiller on the roof left room to locate the heating equipment (two Viessmann 1700 Mbtuh boilers) within the ground floor maintenance room. Using the suggested clearances provided by the Viessmann data sheets left little room for other equipment in the space. Additionally, the requirement of heat exchangers to protect the boiler loop from the hydronic loop pressure further pushed the limits of the space. The temperature output of the boilers was chosen to be 167˚F (162˚F off the heat exchanger) to accommodate both the four pipe fan coil units and the 1st floor radiant panels. As a result, the heating coils of the four pipe fan coil units were oversized to increase the volume of hot water in compensation for the lower temperature and to satisfy the first law of thermodynamics. Fortunately, the cooling coils were the dominant sizing factor in most of the fan coil units, so few had to be upsized from the standard 180˚F assumption.

Despite optimal performance of the fan coil units and radiant panels being designed at 180˚F or higher, a lower temperature condensing boiler outputting water at 167˚F was selected because it could observe up to 91% heating efficiency at lower loads (<50%). This is justified by the fact that the building's loads are heating dominated, so high efficiency heating equipment would provide the largest opportunity for savings. Each boiler (of the 2 boilers) was sized for ~55% of the design block load to allow for simultaneous operation to maintain higher efficiency at lower output, and provide minimum heating if one of the units were to go down.

The largest challenge of the heating equipment design was the ground floor hydronic radiant panels. This subsystem was chosen to complement the ground floor VAV because of the high skin loads from the floor-to-ceiling glazing. Running a lifecycle cost assessment between electric and hydronic radiant panels, it was found that even with combustion losses the hydronic panels were less than half the cost largely due to the higher cost of electricity. Due to the fact that radiant heating can maintain occupant comfort with lower ambient air temperatures, the bulk of the heating load can be moved from the convective VAV system to the higher efficiency radiant panel system. Note that the compromise of a boiler output temperature of 167˚F required a larger effective area of radiant panel than at traditional boiler output temperatures (~180˚F). In response to the flexibility concerns stated in the OPR, we divided the radiant panels into 3 separate "blocks" for each face of the building (PI-001). This would allow for multiple division scenarios for the Future Tenant Space and allow for sprinkler and light fixture installation. A detailed comfort analysis that conveys compliance to ASHRAE 55 is outlined for the south-facing portion of Future Tenant 104 in the ASHRAE 55 compliance section.

For air handling, Engineered Air custom air handling units were chosen. They are equipped with heat pipe sensible heat recovery coils that have 56.2% sensible efficiency for air to air heat recovery supplying both the suite fan coil makeup air and first floor VAV systems. This "zero contamination" heat recovery system allowed us to harvest the waste heat from both the kitchen and the washroom exhaust for preheat, despite the Class 4 Air designation of washroom exhaust, by mixing the airstreams at the rooftop plenum. This doubled the amount of exhaust air that could be recovered, which more than makes up for the loss of latent heat and lower efficiency compared to an enthalpy wheel. Additionally, the passive nature of heat pipes reduces their maintenance and control costs when compared to active heat recovery systems such as heat wheels. Due to the lower size of the first floor VAV system, the same style of air handling unit was used to reduce design costs and ease maintenance.

The cold Ottawa winters caused the water content of the outdoor air to be quite low. This brought the indoor Relative Humidity down to 20%. To mitigate the air dryness, and increase occupant comfort, an electric air handling unit humidifier was installed in the Dedicated Outdoor Air Supply (DOAS) unit supplying the suites to humidify the air between 30% and 60% RH. The hot water is supplied from the service water line. The first floor air handling unit was not equipped with a humidifier because of space constraints. This is generally acceptable because the retail spaces will see more infiltration than the suites and will have shorter periods of occupancy.

Seven different models of Trane fan coil units were selected to condition the suites. All of these units operate at or below Noise Criteria (NC) 30 at 8000 Hz. Six custom VAV units were selected for the first floor that operate below NC 45.

On the delivery side of the system, high efficiency VSD pumps and fans were selected to allow modulation and save energy over the lifetime of the building. VSDs will also allow lower operating pressures in the piping and ducting, which will decrease the amount of noise and increase their usable life. The selection of the optimal efficiency band was based on the shoulder seasons and most common loads, since that is where the pump and fan curves will most likely reside. Peak loads will experience lower efficiencies, but their occurrence is significantly lower in comparison.

Energy Analysis

With all end uses considered, the HVAC design's total (gas and electric) Energy Use Intensity (EUI) was 181 kWh/m^2 and the Greenhouse Gas (GHG) intensity was 0.0028 tons CO_2e/ft^2. Total energy consumption comparison and GHG breakdowns against our two alternatives can be seen in Figure 7 and Figure 8 respectively. For monthly heating and cooling breakdowns, as well as end use breakdown, please refer to Figure 5 and Figure 6 in the Modeling section on Page 3.

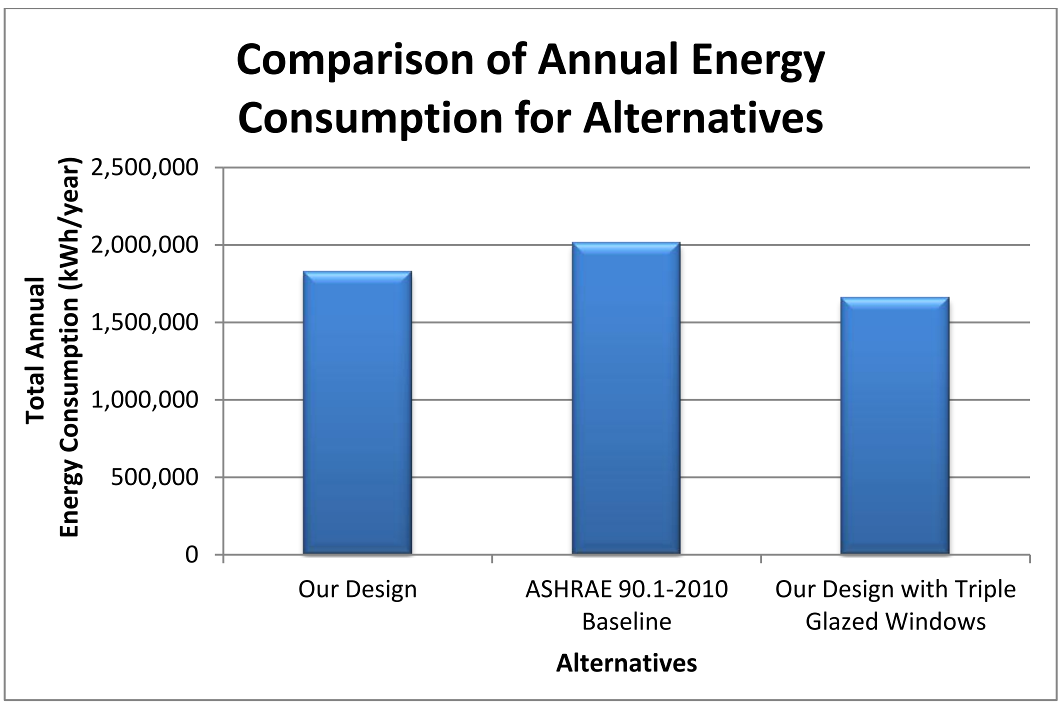


Figure 7: Comparison of Annual Energy Consumption for Our Design and Two Alternatives

As can be seen in Figure 7, 9.0% energy savings was achieved from the ASHRAE 90.1-2010 baseline. These savings can be entirely attributed to the HVAC system equipment choices because we used the assumption of ASHRAE 90.1-2010 minimum envelope construction. Our choice of a high efficiency boiler is the driving force behind these savings due to the building being located in a climate with large heating requirements. Additionally, the use of triple paned windows (in place of double paned) would see 17.5% energy savings from ASHRAE 90.1-2010, which could justify Leadership in Energy and Environmental Design (LEED) points if certification is desired by the owner.

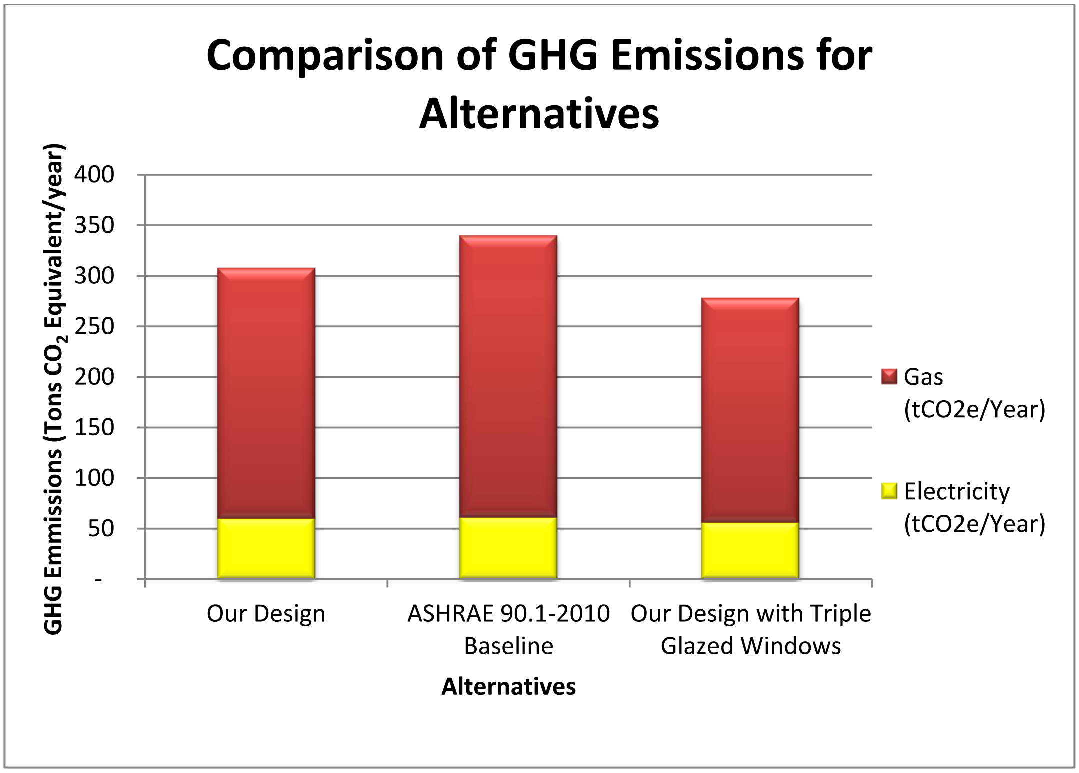


Figure 8: Comparison of Annual GHG Emissions for Our Design and Two Alternatives

The design also reduced GHG emissions by 9.5% when compared to the ASHRAE 90.1-2010 baseline. Adding triple paned windows to the design would bring the GHG emissions down by 18.0% when compared to the baseline. The bulk of these reductions are seen by either increasing boiler combustion efficiency or reducing heating loads because the emission factor for natural gas combustion is significantly higher than for Ontario Hydro electrical consumption.

Economic Analysis

When the utility rates provided in the Utility and Service Life Overview (see Input Summary Table in Modeling on Page 3) were applied, our design saved $4,800 (7.4%) annually from the ASHRAE baseline. While this currently doesn't meet LEED prerequisite requirements using ASHRAE 90.1 minimum envelope conditions, the addition of triple paned windows would yield 16.2% energy cost savings which would enable LEED NC 2009 credits. Using the guidelines in the OPR we saw significant savings, but in the sub-arctic climate of Zone 6, triple paned windows would be a worthwhile investment. Comparisons of the annual utility costs between our design and our two ECMs can be seen in Figure 9.

The two hydronic gas-fired boilers and rooftop air cooled chiller had capital costs of $106,000 and $222,000 respectively. The boilers were the more reasonably priced component given the capacity that we required and the heating savings. The air cooled chiller was a necessity to fit all of the mechanical equipment within the space given by the architectural drawings. Rooftop enclosures were not an option without significant architectural and structural changes to the building that were beyond the scope of this project. Part of the cost of the chiller was to add the heat recovery option. To confirm the validity of this design choice, we ran a net present value calculation using a lifetime of 25 years and the interest rates given in the Utility and Service Life Overview. Considering the heating savings and capital costs of the equipment required for an additional loop, we found the addition of the heat recovery chiller line to have a net present value of $3,350. A positive net present value justifies this design choice.

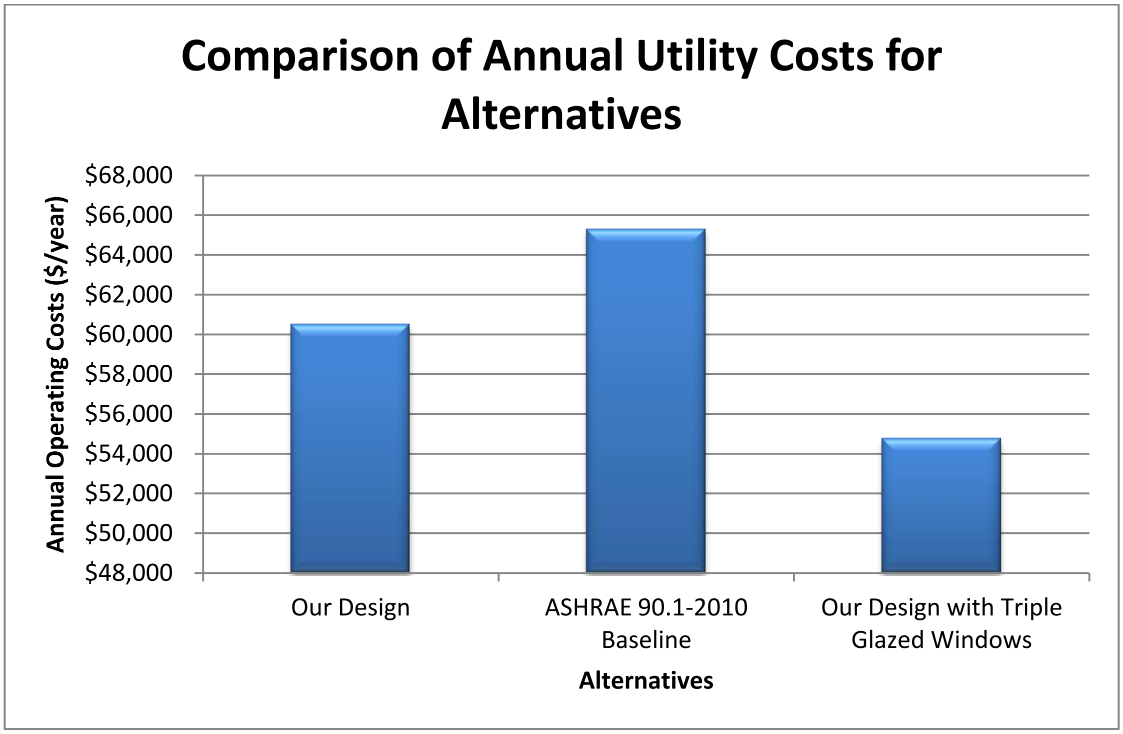


Figure 9: Comparison of Annual Utility Costs for Our Design and Two Alternatives

Economic analysis was also run for the 1st floor radiant panels to compare electric with hydronic versions. We found when maintenance, operating costs, and capital costs were considered, the electric panels had a lifetime cost of $180,500, and the hydronic panels had a lifetime cost of $83,700. This provided the justification for choosing hydronic panels, since they were less than half the cost of electric.

Addressing the building owner's desire for apartment owners to be responsible for individual energy usage, it is proposed that the utilities costs for the central plant be collected through a "maintenance fee". This fee would include the tenant's portion of the electrical and gas consumption determined by a ratio of suite square feet to the building total. This would then be applied to the building's total energy cost from the utility provider. The fee would account primarily for heating, cooling, and common area lighting utility costs, and would be administered monthly. This route was chosen to save the cost of installing hydronic energy meters, and avoid the requirement of the strata to be a Regulated Utility Provider. The electrical energy for the individual suite fan coils would be routed through the suite's electrical meter, so the tenant will pay for their own fan coils operation.

All economical choices were based on either lifetime costs, or in the case of the chiller, selecting a product that meets the unique needs of our project. If the Dallas Power and Light building were designed for the Ottawa climate, it is very likely that the mechanical space would have been larger than for Dallas. As a result, our choice to use the climate of our nation's capital provided an additional challenge that pushed the physical and economical feasibility of this project.

ASHRAE Standard Compliance

ASHRAE 55

To demonstrate our knowledge of ASHRAE 55-2010, a thermal comfort calculation was performed that demonstrates the first floor retail's radiant panel effectiveness. The temperature assumptions, and the governing equations are as follows:

$$T_{mean\ radiant}^4 = \sum F_{x-occupant} T_x^4 \qquad\qquad T_{operative} = (T_{mean\ radiant} - T_{ambient})/2$$

The angle factors (F_x) and temperatures (T_x) we assumed are:

	Exterior Wall	Exterior Window	Radiant Panel Surface	Imaginary Surfaces
Angle Factor	0.147	0.027	0.176	0.650
Temperature (˚F)	63.5	42.5	132.0	64.0

The geometries for the angle factor calculation were taken from Figure 10 and the elevation drawings under the assumption that the occupant is located in the center of the zone, and their center of mass is 2 feet above the floor (sitting position). The "imaginary surfaces" are a representation of the distant walls and roof that would be at roughly ambient air temperature which allows splitting of the space along the lines in Figure 10. The geometries were then used with the tables in Figure 11 to find the angle factors for the mean radiant temperature ($T_{mean\ radiant}$) calculation. The temperatures used were calculated using a combination of model output data and hand calculations to account for re-radiated heat and radiant panel thermal resistance. Note that the temperature gains of the floor from re-radiated heat were negligible due to the heavy thermal mass (similar to the walls), so it was assumed to be part of the ambient temperature surface (64˚F).

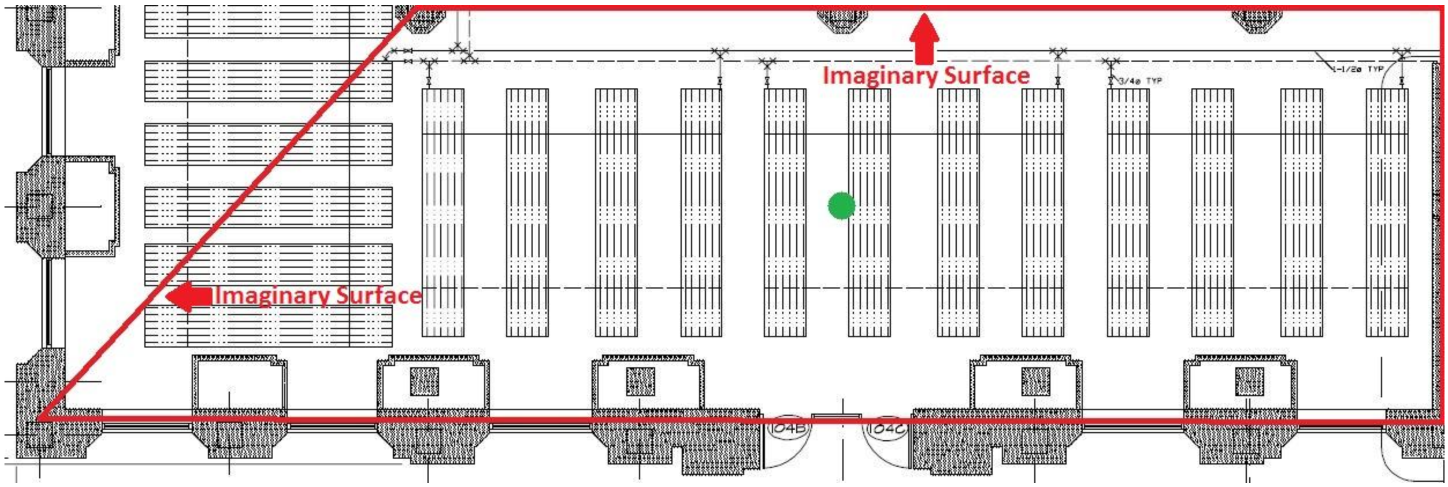


Figure 10: First Floor Retail South Zone Geometry Relative to Occupant

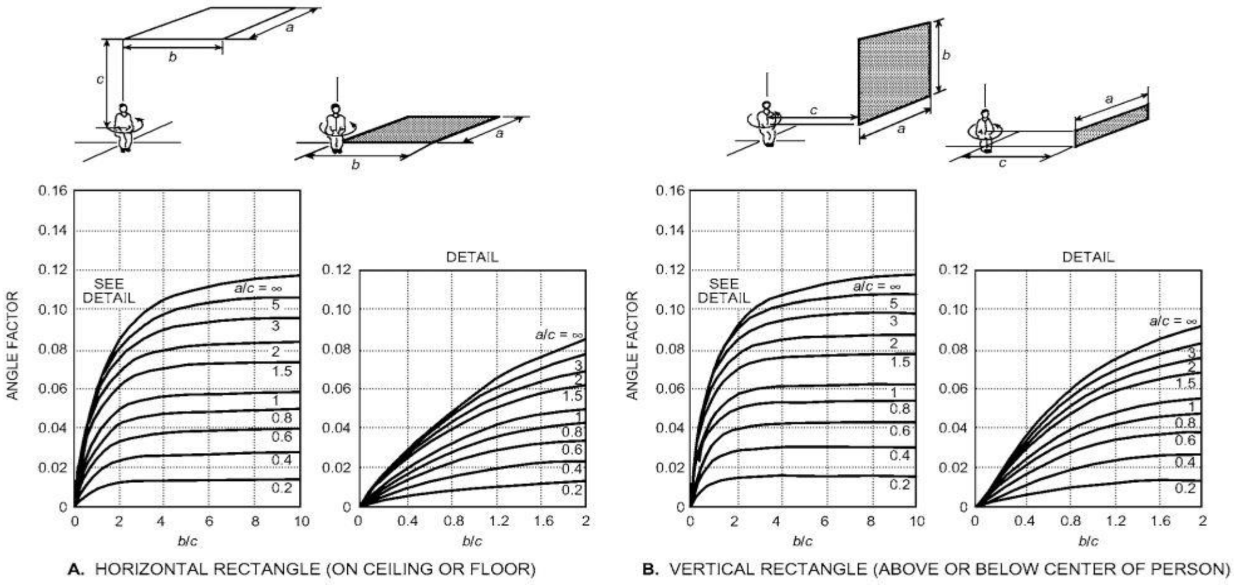


Figure 11: Angle Factor Tables from ASHRAE Fundamentals 2009

Applying the figures and calculations yielded an Operative Temperature ($T_{operative}$) of 71.5˚F at design conditions where the RH is 20% in the retail space. The resulting $T_{operative}$ and RH are represented by a red dot in Figure 12. This demonstrates that our design meets the criteria of ASHRAE 55-2010 for the heating season under the assumption of mild activity and winter clothing accumulating to 1.0 clo for the retail space.

ASHRAE 62.1

In order to verify the model's compliance with ASHRAE 62.1-2010, a spreadsheet was used to compare the model outdoor air flow rates with the rates prescribed by the standard. A summary of the outdoor air calculation results can be seen in Table 8. The 25% difference can be attributed to safety factors within the software, and infiltration losses. Additionally, having higher than minimum outdoor air is acceptable because the building needs to be positively pressurized.

Table 8: Comparison of Outdoor Air Requirement Between Model and Hand Calculation

	OA (cfm)
Calculated	9383
Model	12500
% Difference	25%

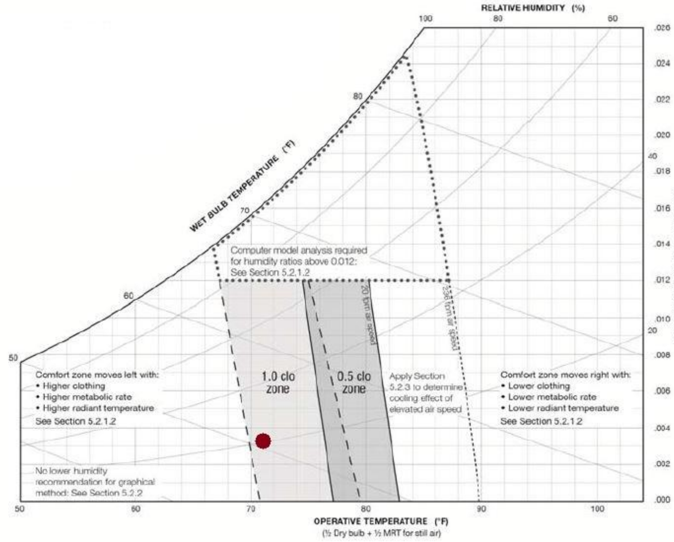


Figure 12: Acceptable Range of Operating Conditions for Spaces that Meet ASHRAE 55-2010 Section 5.2.1.1

ASHRAE 90.1

ASHRAE 90.1-2010 was used as a baseline standard for assumption about the building envelope and lighting loads, and for equipment efficiencies to compare performance. All of the hydronic and air handling equipment meet the minimum requirements of ASHRAE 90.1-2010 as per the OPR. The assumptions for lighting power density can be seen in Table 3. A brief comparison of equipment efficiencies and ASHRAE minima are outlined in Table 9.

Table 9: ASHRAE 90.1-2010 Equipment Efficiency Comparison

	Type	Design Efficiency	ASHRAE 90.1-2010 Minimum
Heating	Gas-fired hot water boiler	91%	80%
Cooling	Air cooled chiller	9.93 EER	9.562 EER
Heat Recovery	Heat pipe air-to air	56.2%	50%
Domestic Water Heater	Gas-fired storage tank heater	96%	80%

Conclusion

The UBC ASHRAE Student Design Team put the utmost care into designing a HVAC system that meets the needs of the Dallas Power and Light Building owner and the judging committee, and complies to ASHRAE 55-2010, ASHRAE 62.1-2010, and ASHRAE 90.1-2010. Energy efficiency and economics were also significant considerations. Where possible, payback and net present value analysis was applied to justify design decisions that could reduce operating and maintenance costs. The largest challenge of this project was retooling the HVAC system of a building architecturally designed for a cooling dominant climate to operate efficiently in a heating dominant climate. Mechanical spaces were clearly designed for much smaller heating equipment, which limited the space we had for cooling equipment. As a result, a rooftop air-cooled chiller was used to conserve the mechanical space on the first floor that houses the boilers and their support equipment. Air to air heat pipes were used for heat recovery because they provide minimal leakage and are a passive technology. These features allow energy recovery from washroom exhaust, and reduce maintenance costs. High efficiency boilers were also implemented to save energy on the building's largest energy load. With the annual energy savings from the boilers, they could pay for themselves within the lifetime of the building.

The condensing gas-fired boiler and air-cooled heat recovery chiller system provides a very energy efficient plant equipment solution given the limited mechanical space available and terminal unit requirements. When compared to heat pump based systems, its installation cost is significantly less because cooling towers are not required (water-source) and there is no need for ground well installation and geotechnical analysis (ground-source). The chiller selection did not result in a particularly high EER, but its rooftop placement allowed more significant savings from the high efficiency boilers. Some additional justification for the chiller selection is the heat recovery functionality that provides essentially free heating during the shoulder seasons.

In closing, the UBC ASHRAE Design Team learned a great deal about HVAC design from this project. The opportunity of working with different people in industry and within the team proved to be as valuable an experience as the technical learning aspect. In the world of HVAC design consulting, being skilled at all three of those aspects is essential to both the success of yourself and the industry as a whole.

Acknowledgments

We would like to thank Dr. Nima Atabaki, Dr. Steven Rogak, and Geoff McDonell for their help that led to the success of this project. Dr. Nima Atabaki is an instructor at UBC who provided in-depth technical insight and recommendations for our design. Dr. Steven Rogak is a professor at UBC who provided us guidance with respect to major milestones and general presentation. Geoff McDonell is an Associate Principle at Integral Group that provided valuable industry knowledge and regular technical review. Without their help this project would not have been possible.

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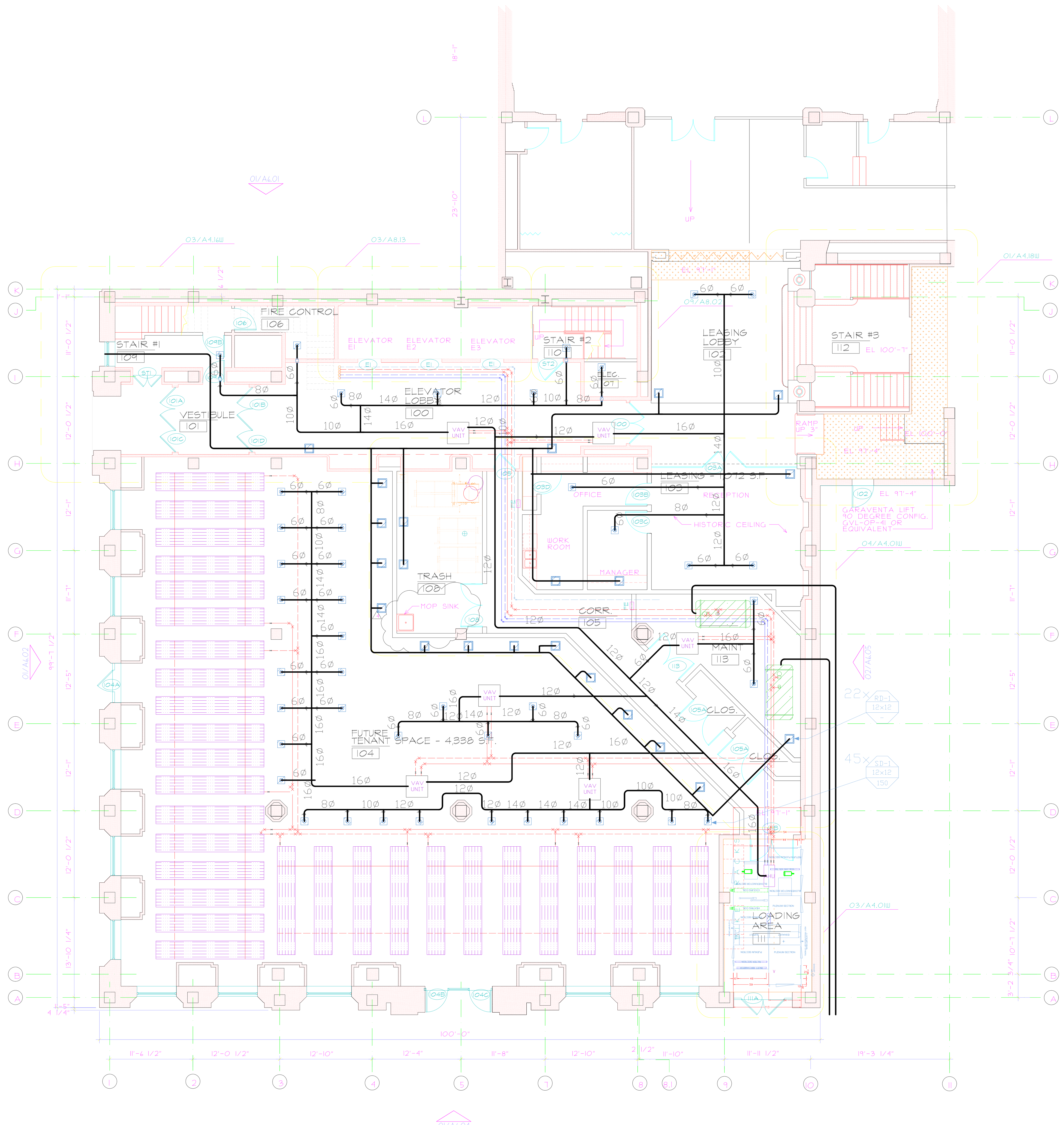

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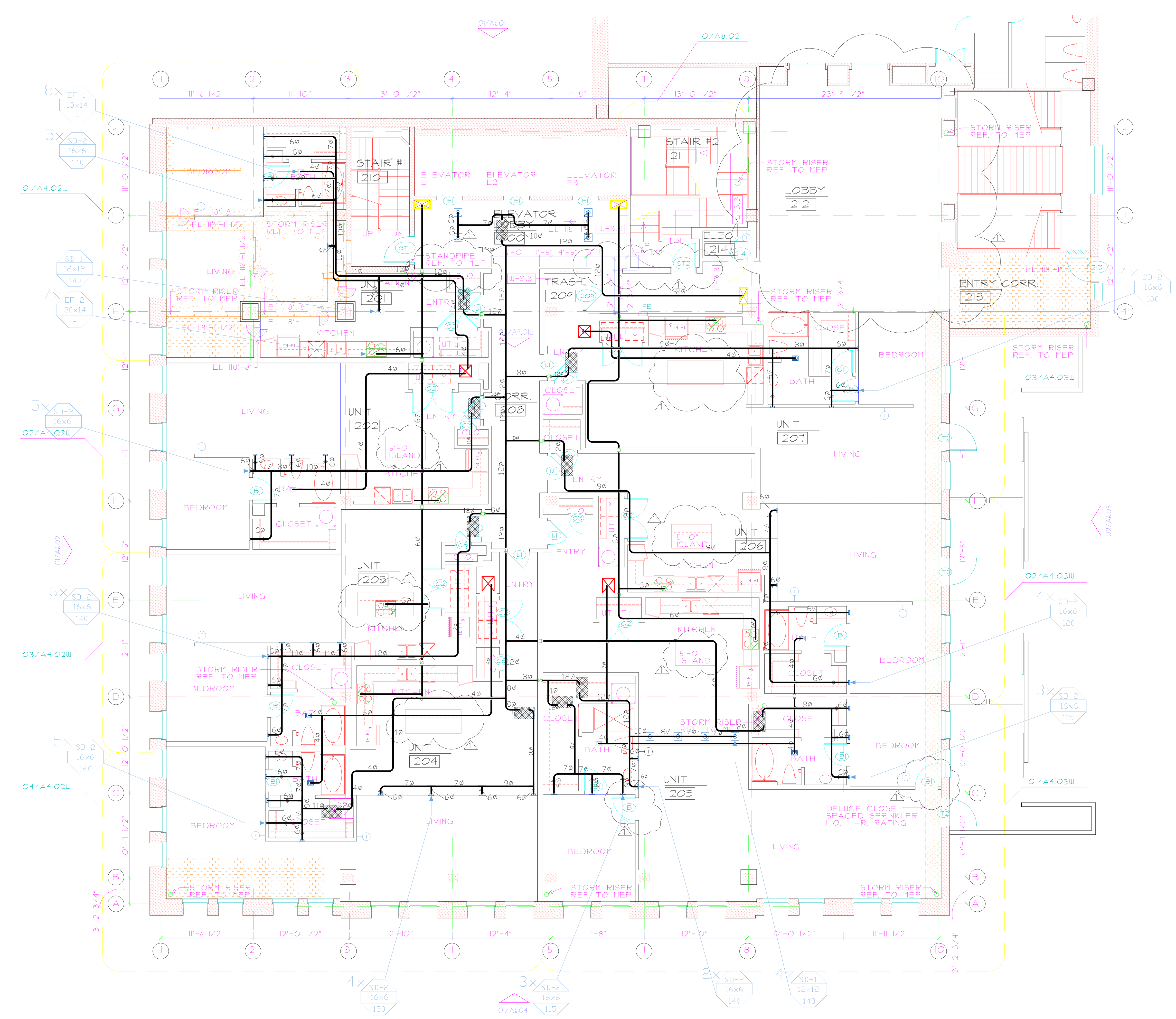

SECOND FLOOR
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CHECKED BY	DATE	DUCTING SYSTEMS - GROUND FLOOR AND 2ND FLOOR
	DRAWING NO. D-001	REV 0

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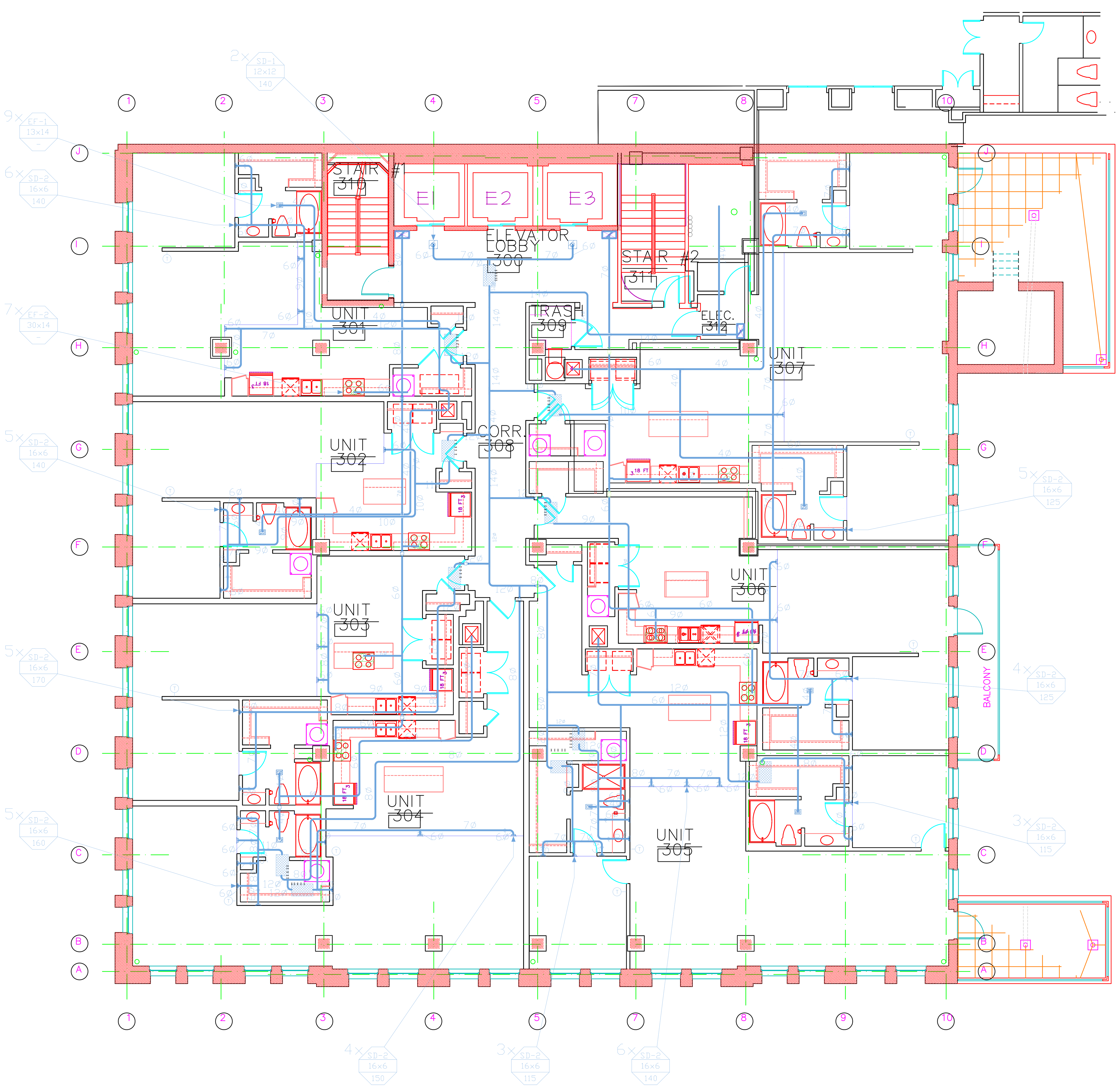

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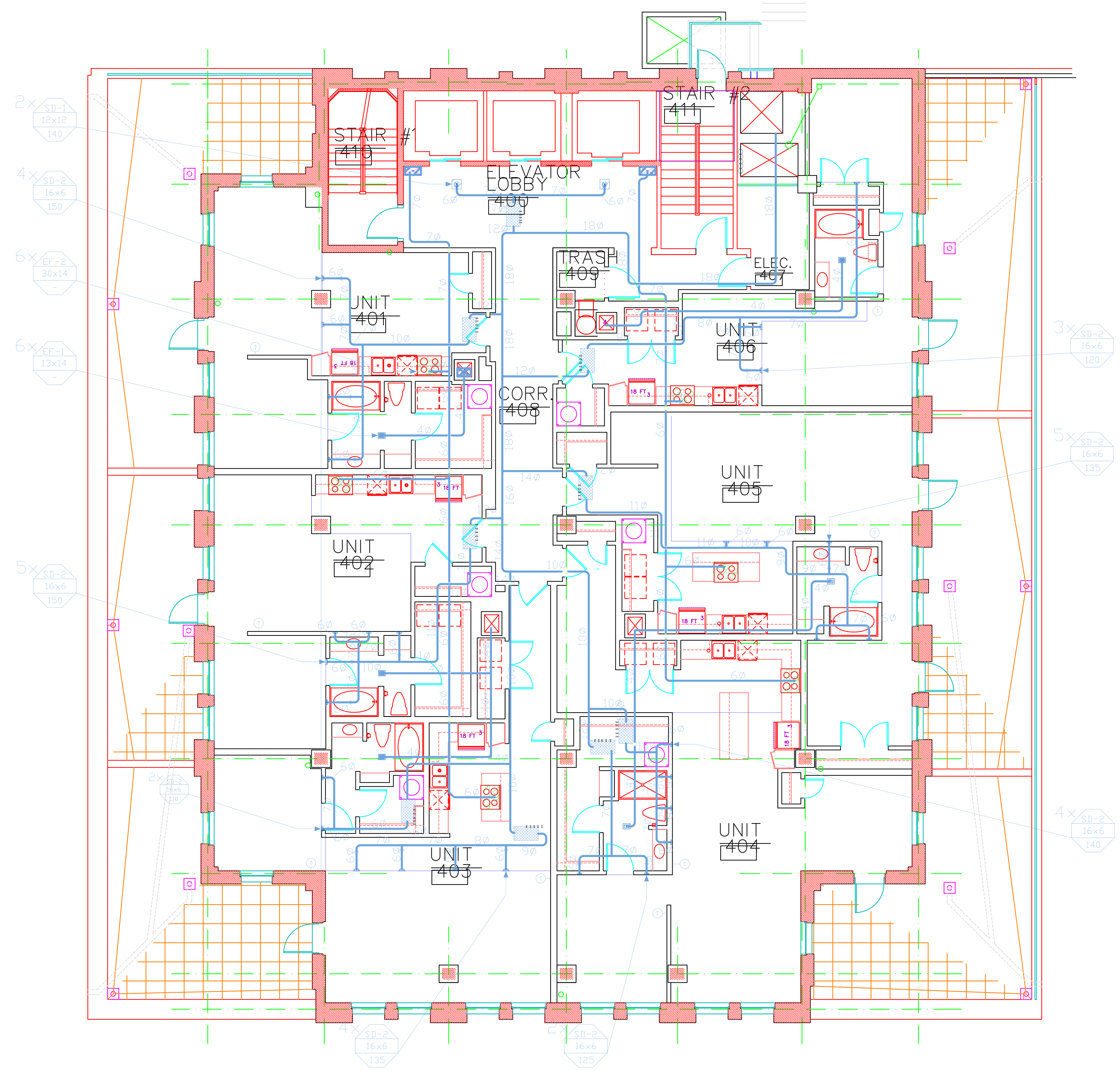

FOURTH FLOOR

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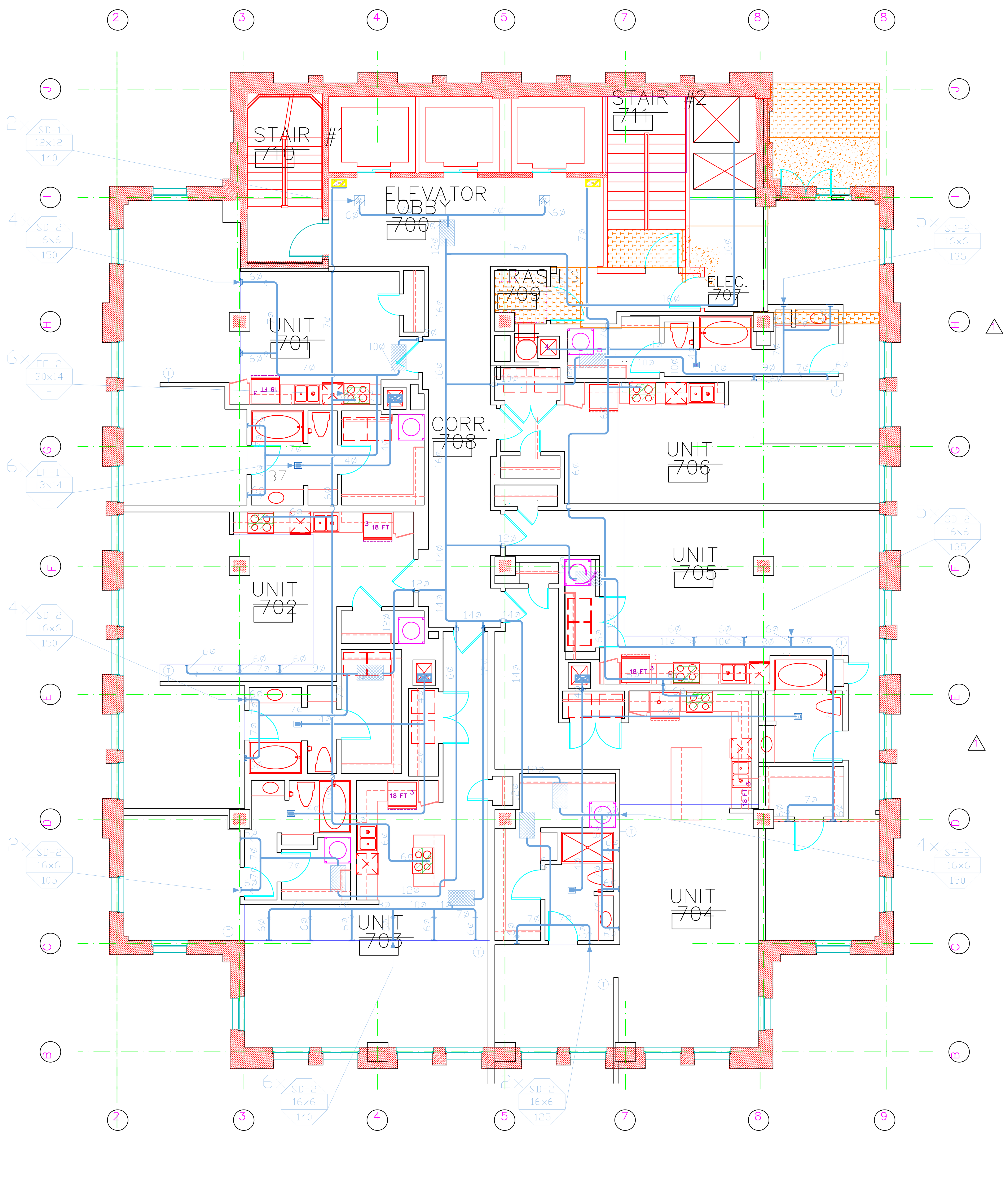

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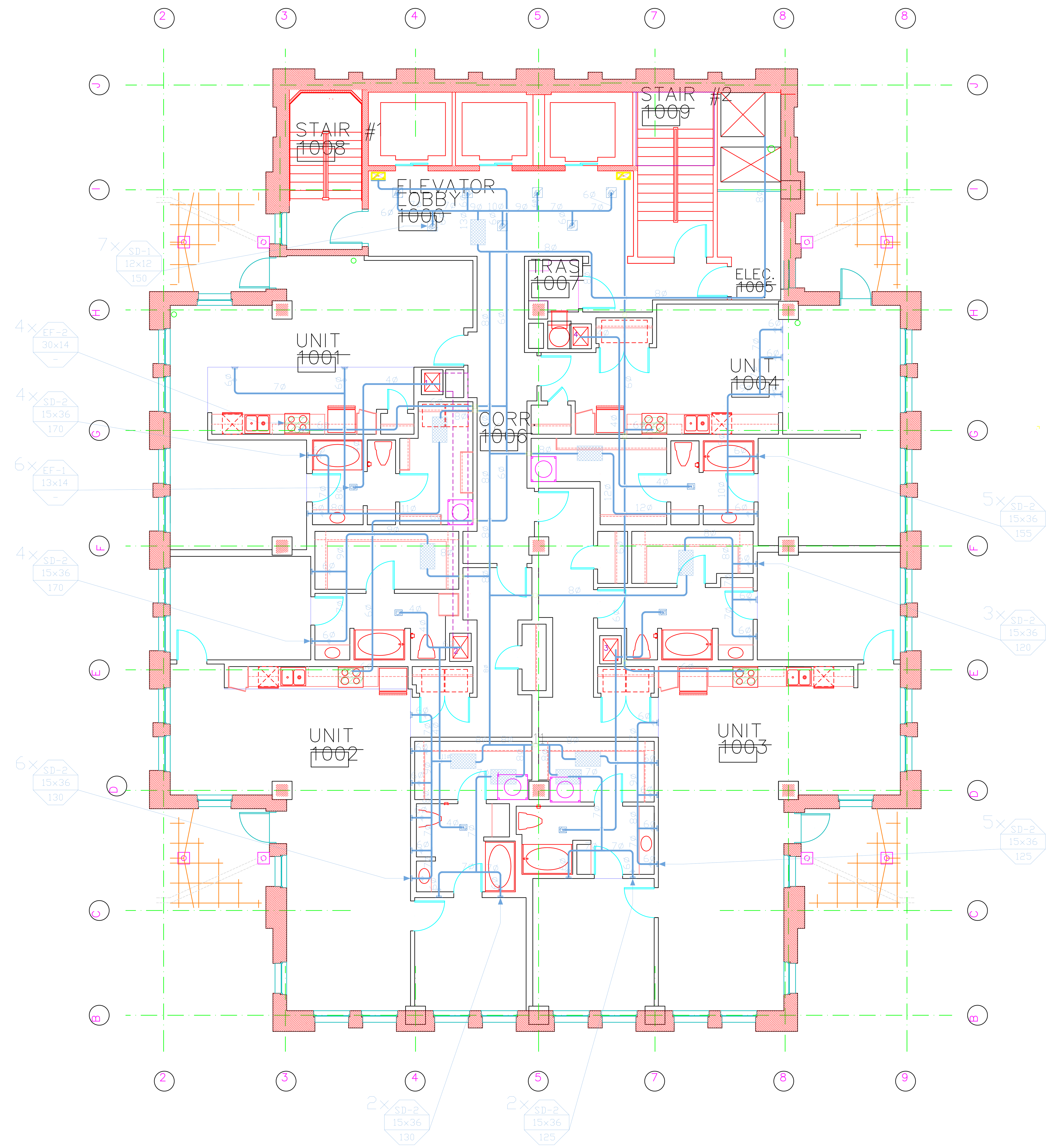

TENTH FLOOR
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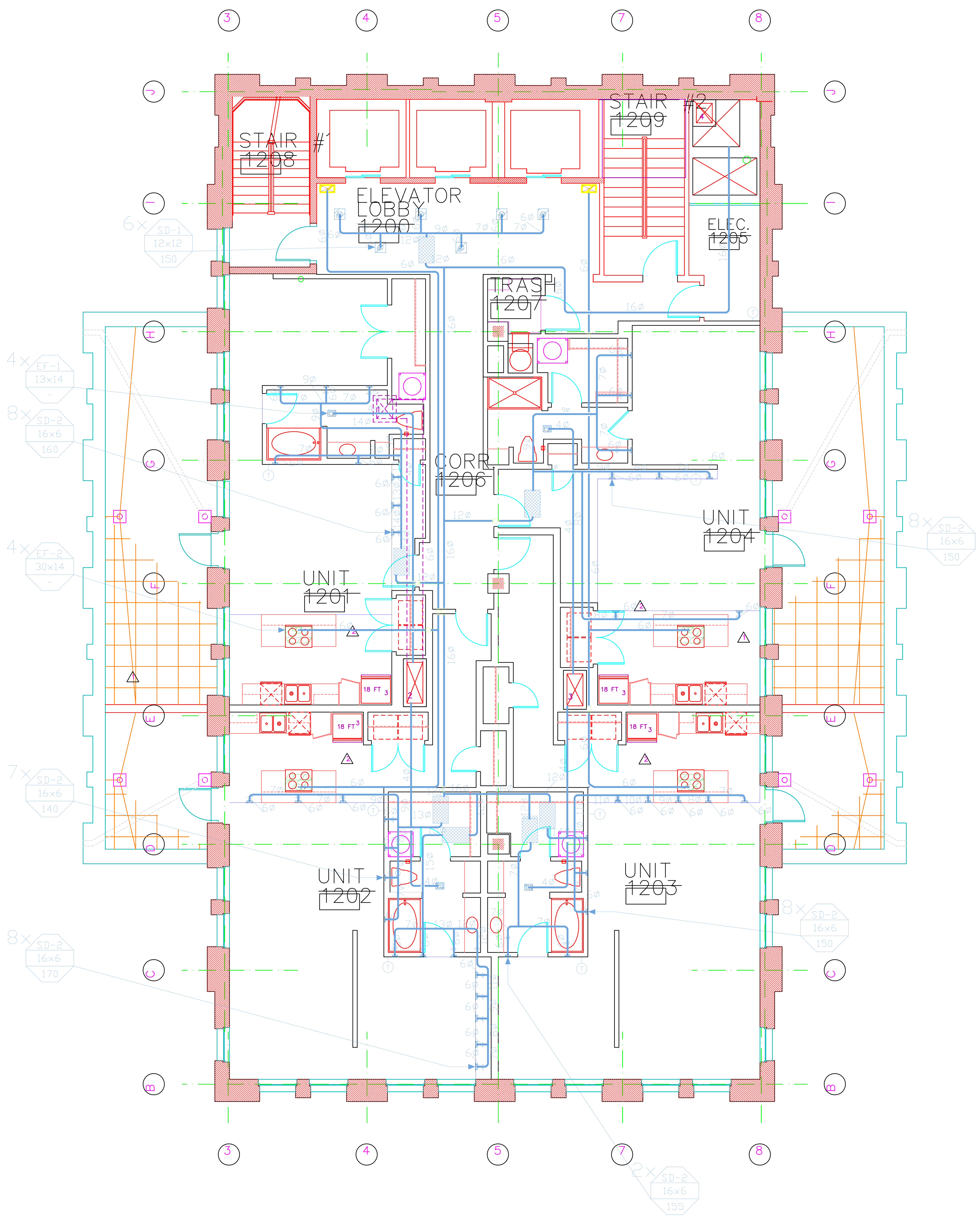

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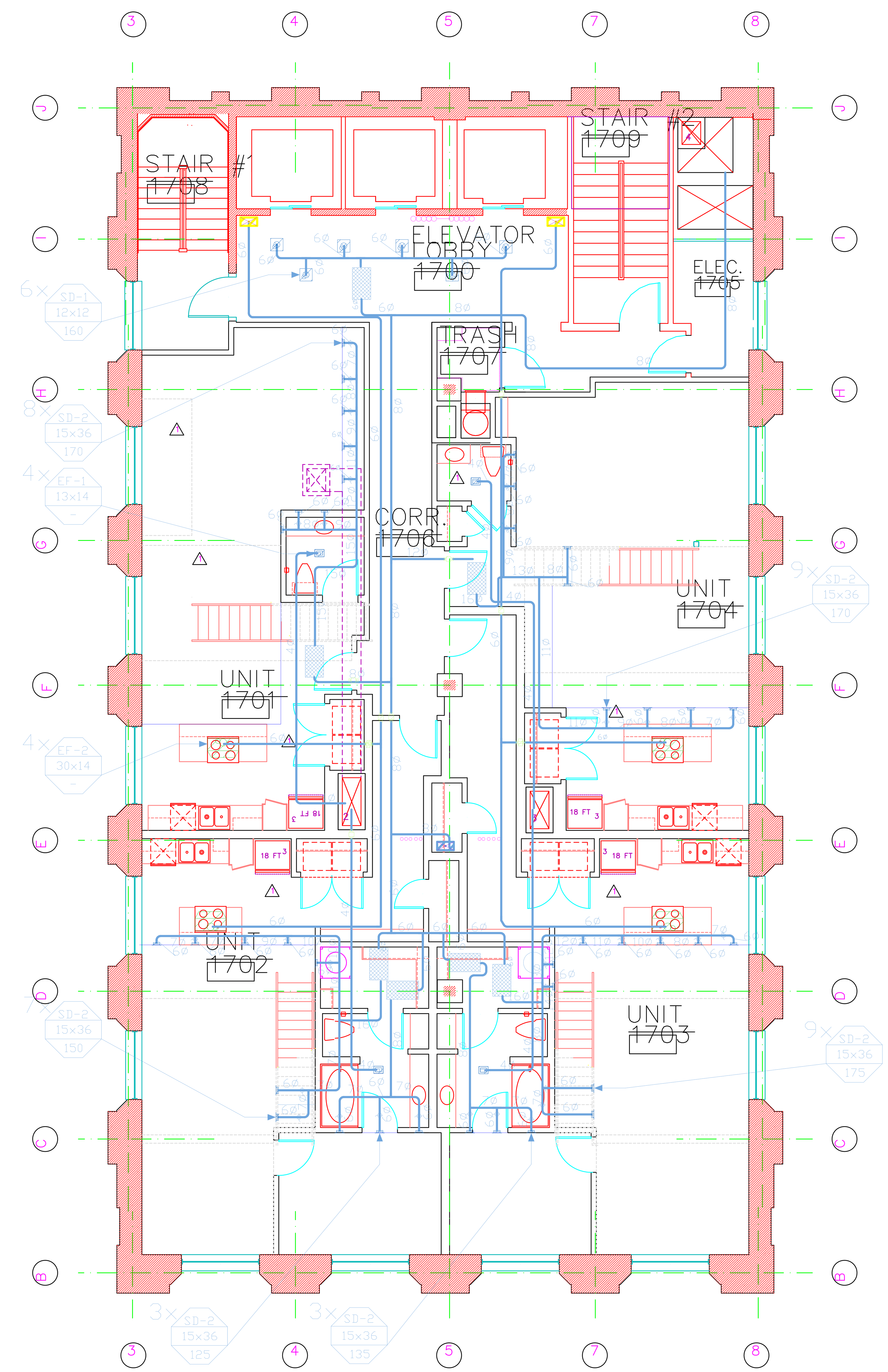

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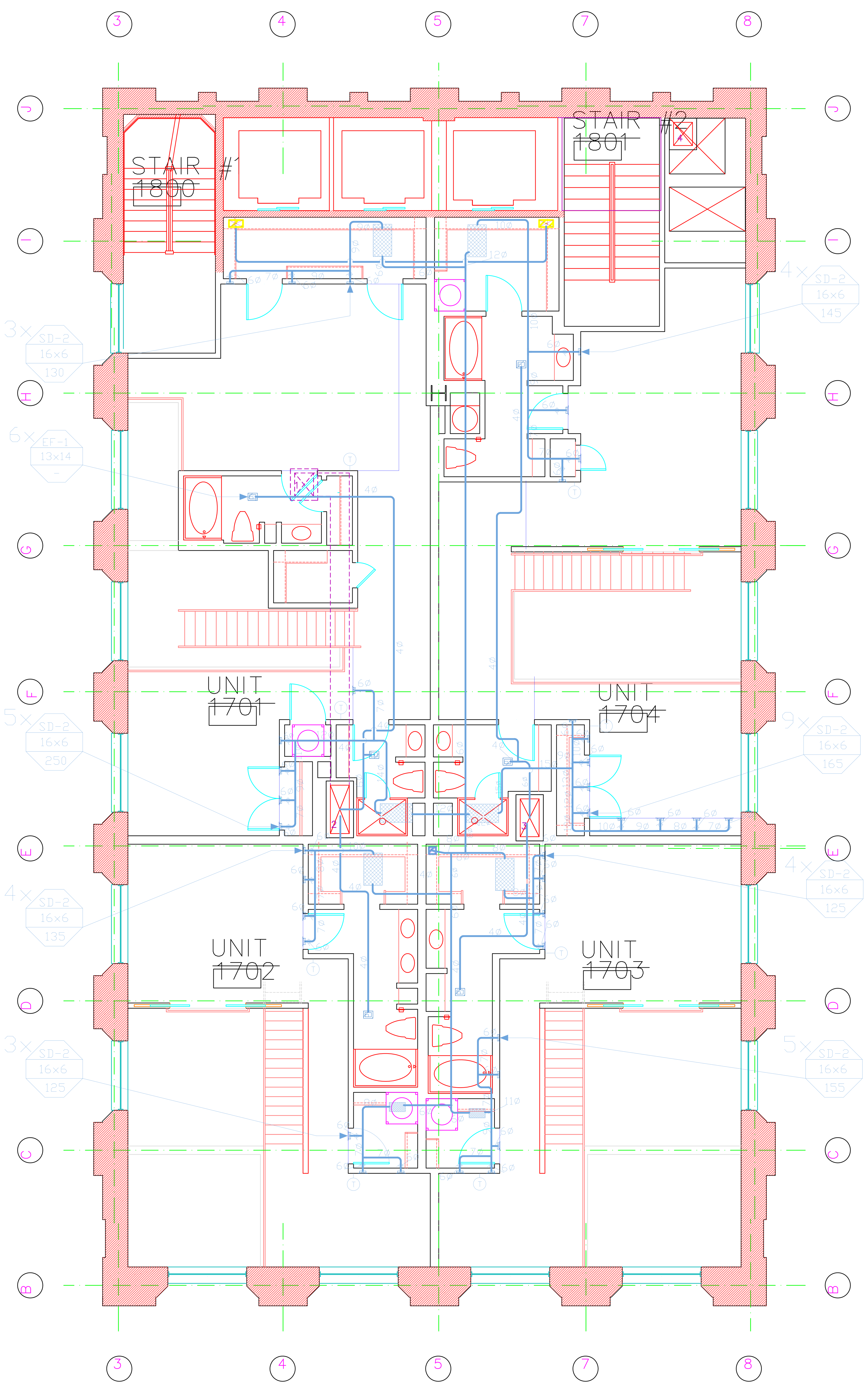

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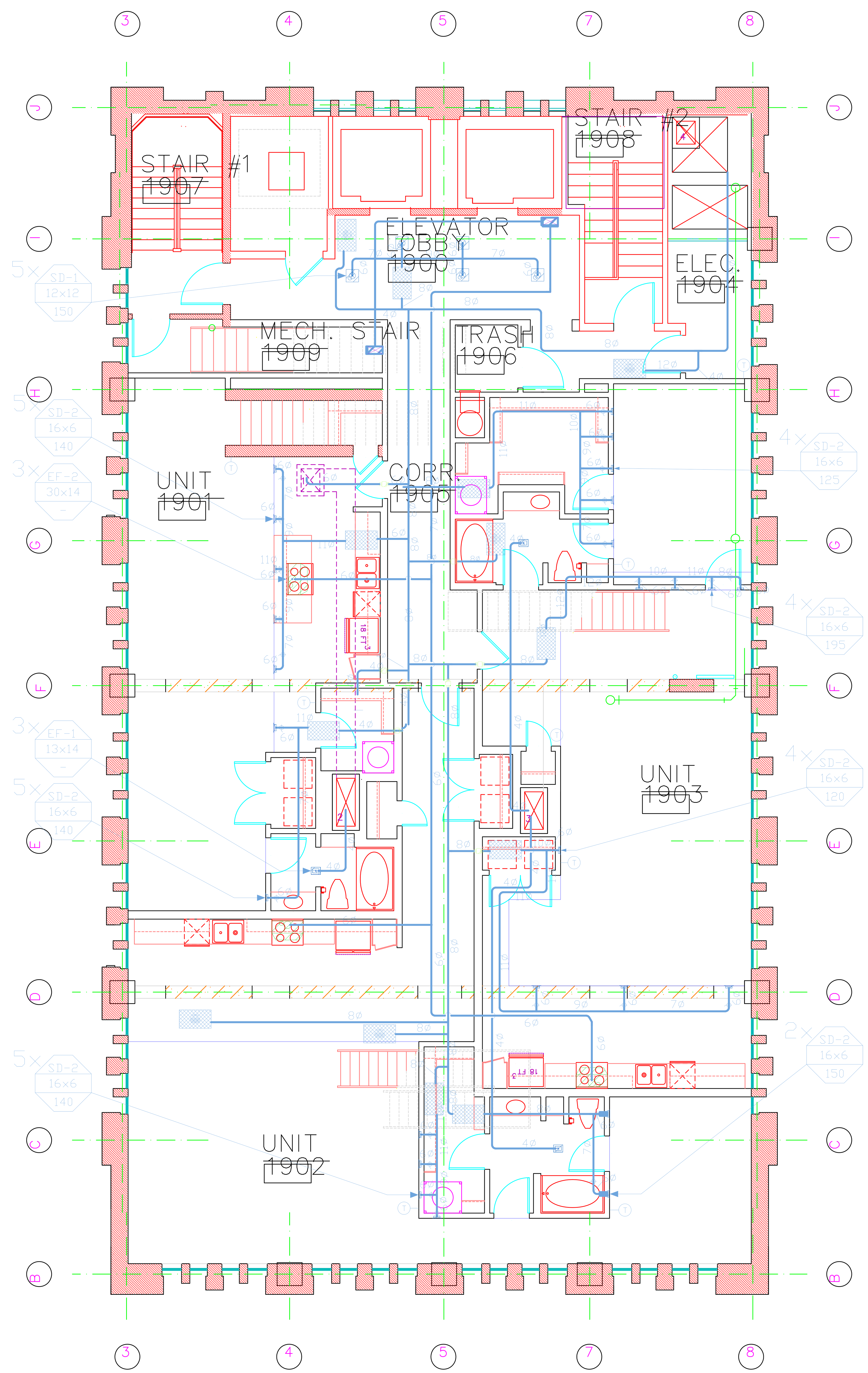

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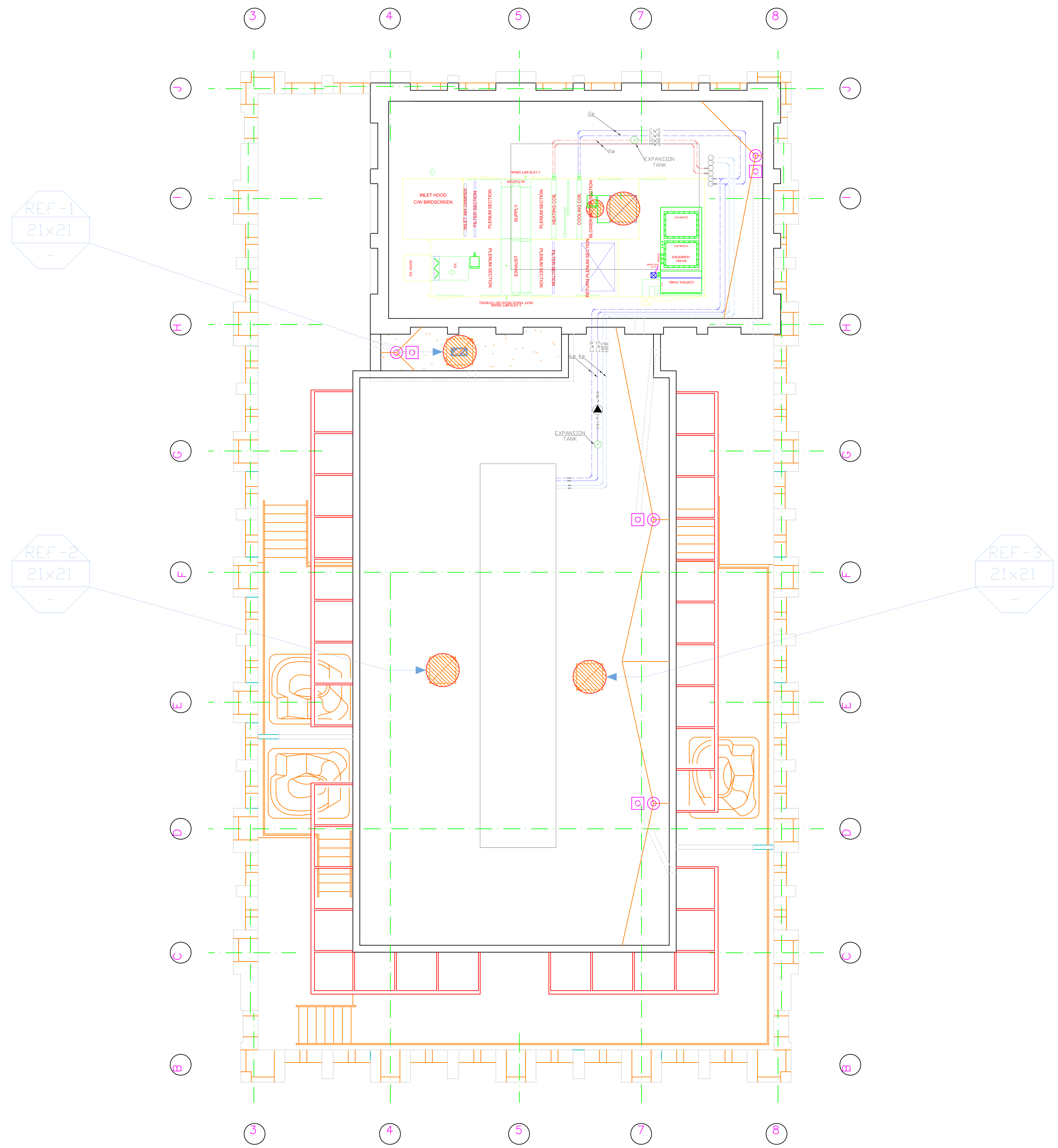

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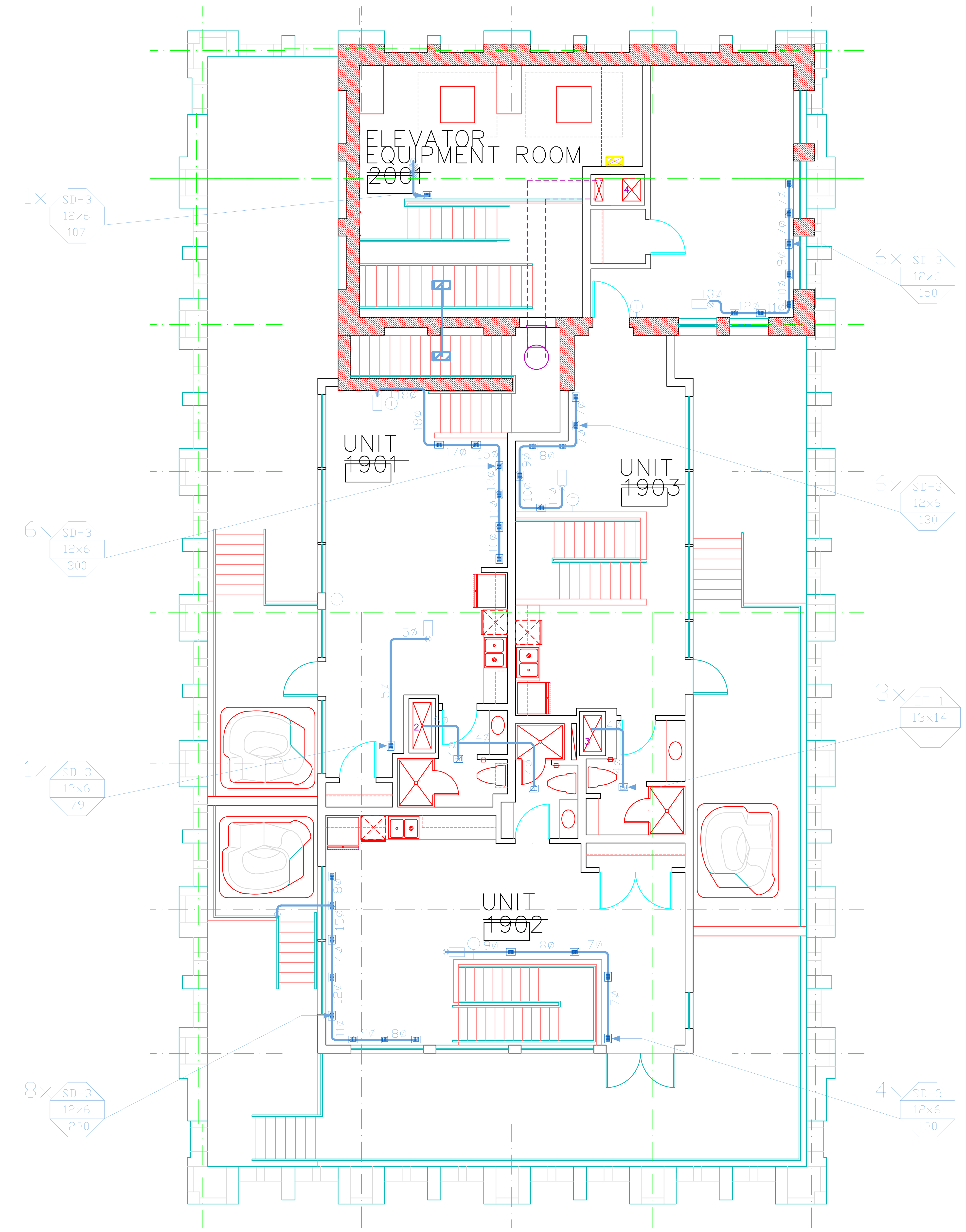

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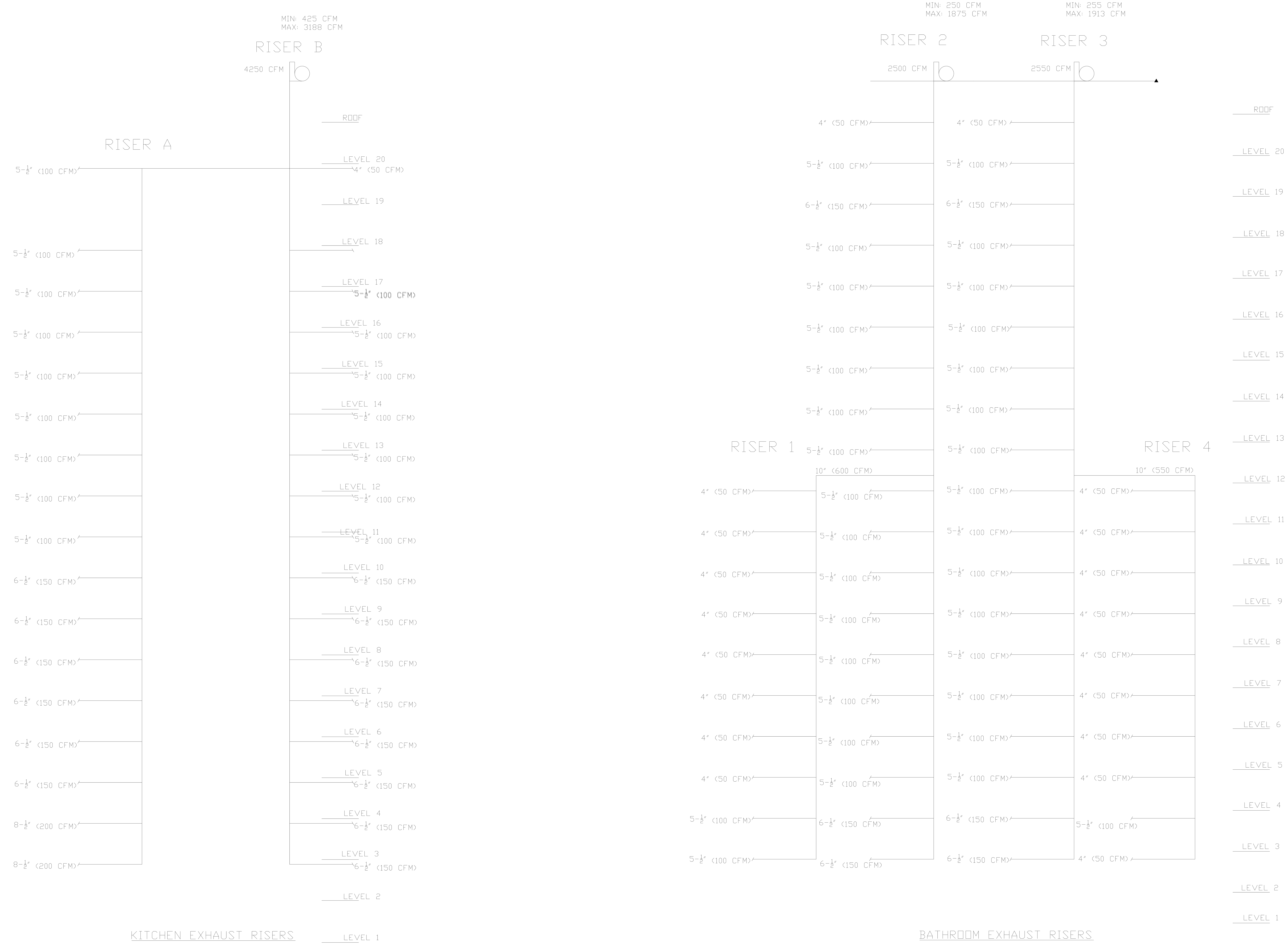

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DRAWING NO.: D-007	REV: 0

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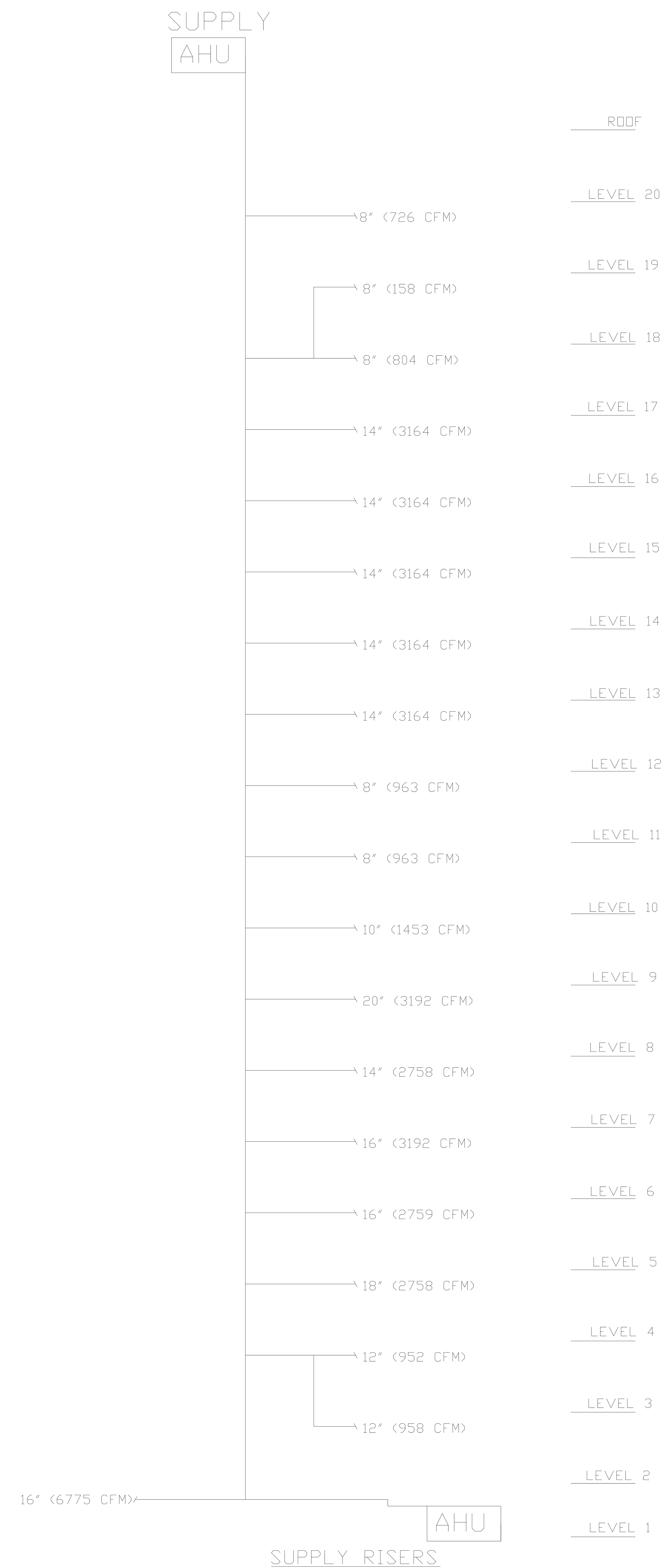
Exhaust Fans

Tag	EF-1	EF-2	REF-1	REF-2	REF-3
Service					
Location	Bathroom	Kitchen	Roof-Kitchen	Roof-Bathroom 1	Roof-Bathroom 2
Airflow (CFM/ l/s)	50	50	3188	1875	1913
Total SP. (ins.WC/Pa)	0.05	1.05	0.445	0.501	0.501
External (Discharge) SP (ins.WC/Pa)			0.612	0.665	0.665
Fan Hp (Kw)			½	1/4	1/4
Fan Speed (RPM)			888	562	755
Electrical (Volts/Ph/hz)		120			
Sound Level (Sones)	0.8	4.1	11	4.1	6.5
Make	BROAN	BROAN	GREENHECK	GREENHECK	GREENHECK
Model	QTRE080C2S	BDF302SS	LB-18-5	LB-18-4	LB-18-4
Notes/Options					

Air Terminal and Louvres Schedule

Tag:	Make	Model	Installation	Notes
SD-1	E.H. PRICE	PDS Series	Lay-in	
SD-2	E.H. PRICE	STG1 Series	In wall	
SD-3	E.H. PRICE	DFGL Series	Underfloor	
RD-1	E.H. PRICE	PDR Series	Lay-in	



REV 0	ISSUED TO ASHRAE COMPETITION
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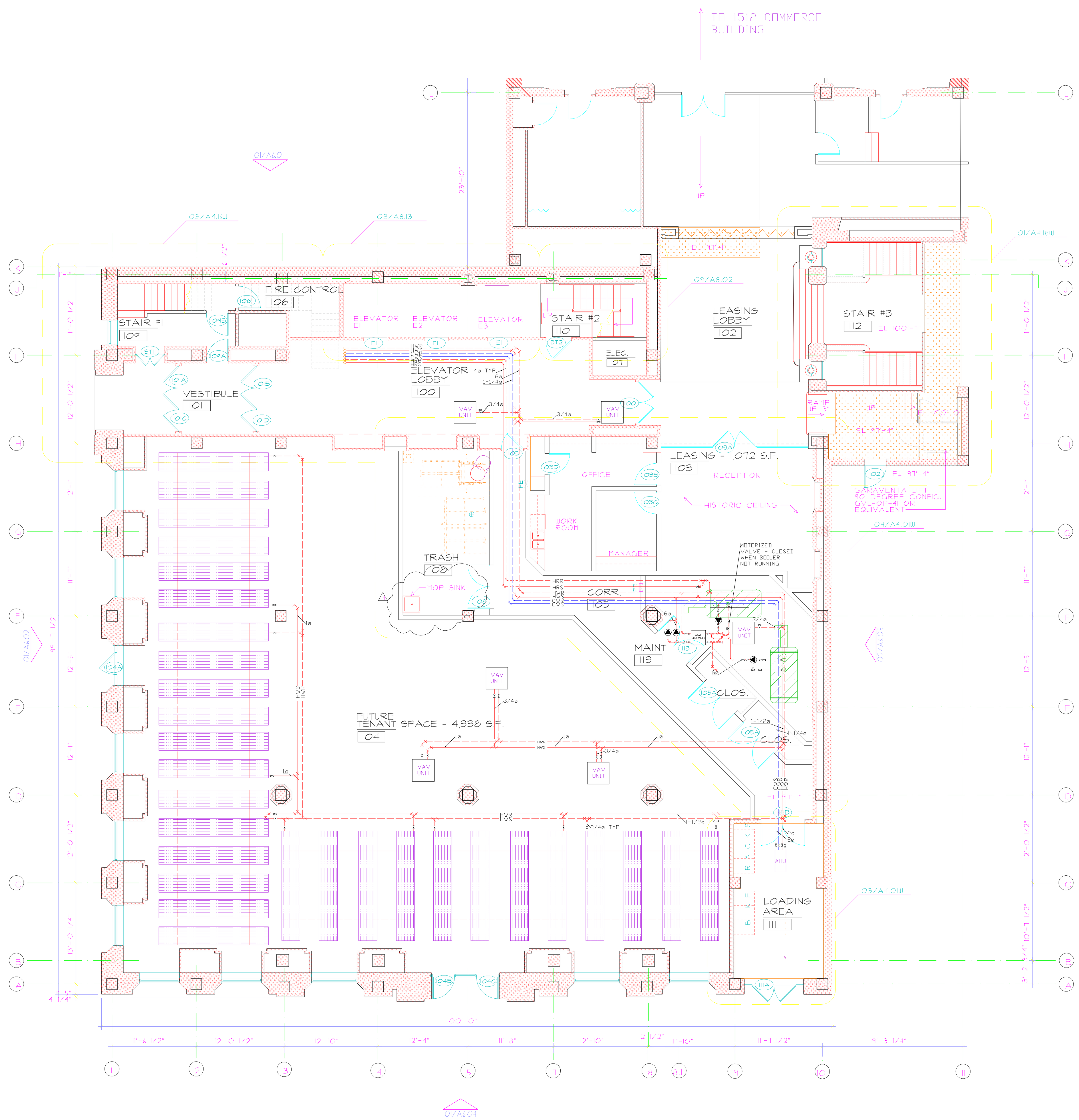
CHECKED BY: DATE:	DUCTING SYSTEMS - SCHEMATIC SUPPLY FLOW DIAGRAM & SCHEDULE
DRAWING NO.: D-008	REV: 0



GROUND FLOOR
NTS

SECOND FLOOR
NTS

TRUE NORTH
PLAN NORTH

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION
DRAWN BY: GW — DATE: 13/04/26	UBC ASHRAE TEAM
CHECKED BY — DATE	PIPING SYSTEMS - GROUND FLOOR AND 2ND FLOOR
ASHRAE University of British Columbia Student Branch	DRAWING NO.: PI-001 — REV: 0

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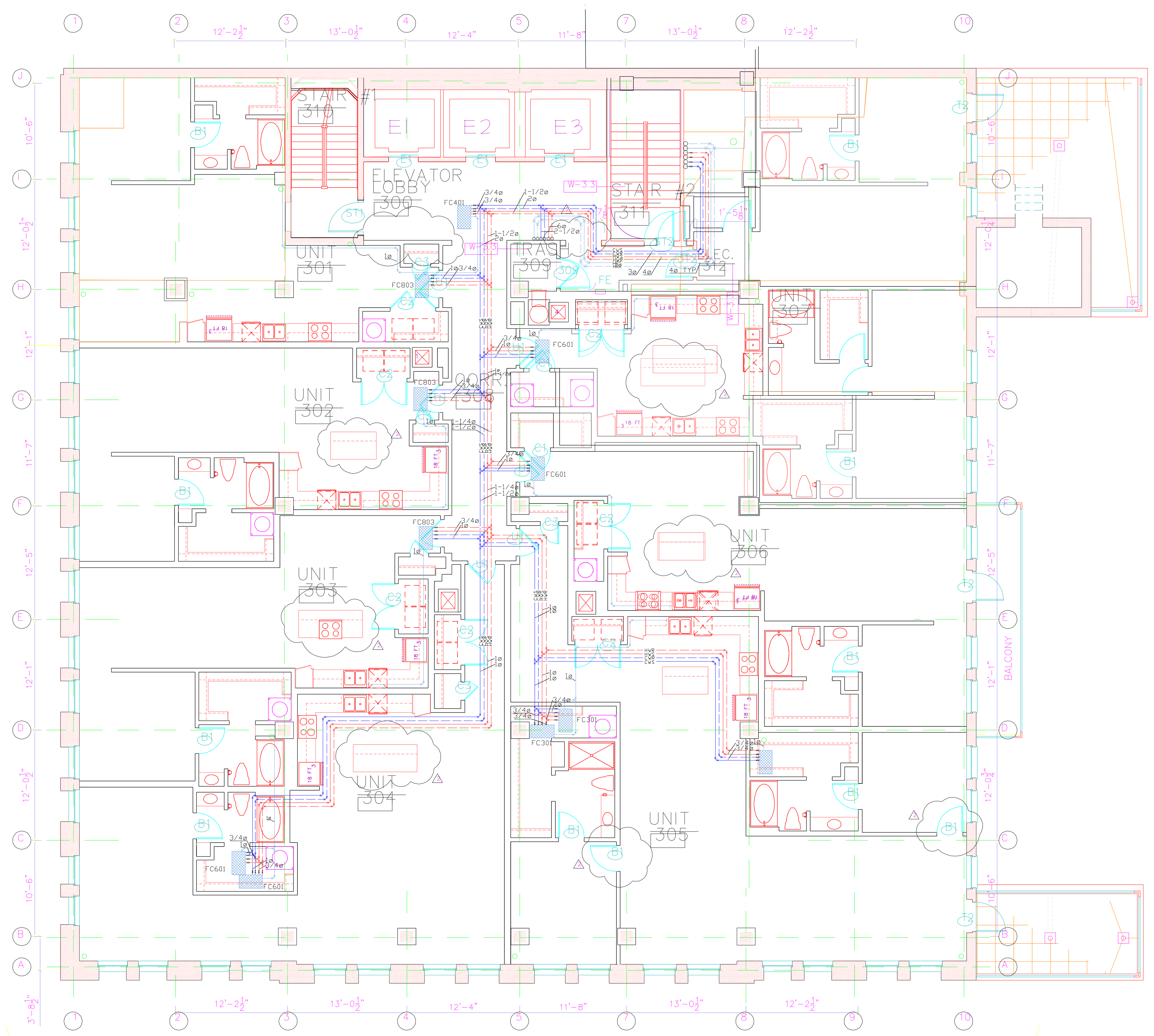
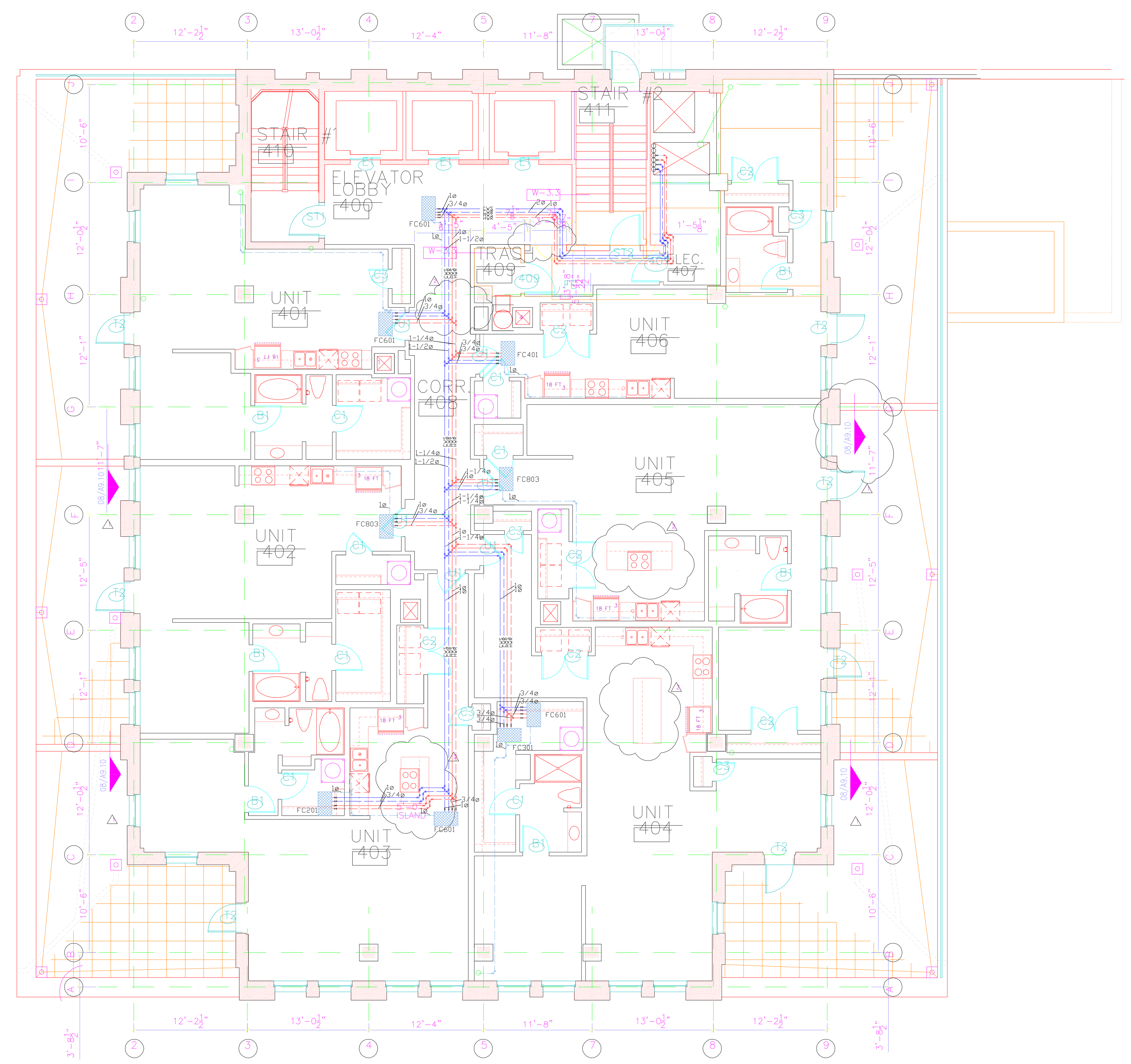


THIRD FLOOR
NTS



FOURTH FLOOR
NTS

TRUE NORTH
PLAN NORTH

No.	REVISION
REV 0	ISSUED TO ASHRAE COMPETITION

DRAWN BY	DATE	
GW	13/04/26	**UBC ASHRAE TEAM**
CHECKED BY	DATE	PIPING SYSTEMS - THIRD FLOOR AND FOURTH FLOOR

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Student Branch

DRAWING NO.	REV
PI-002	0

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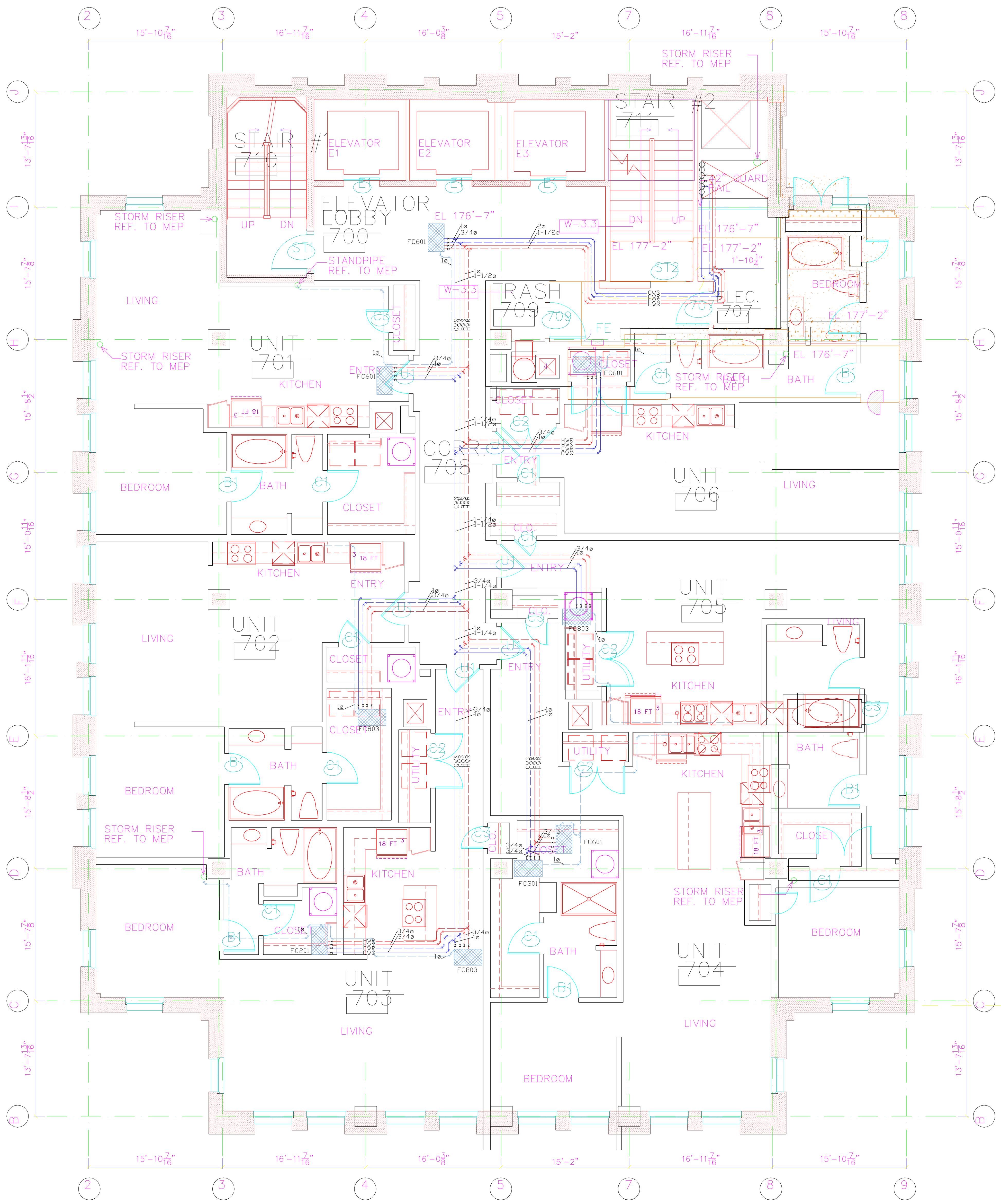
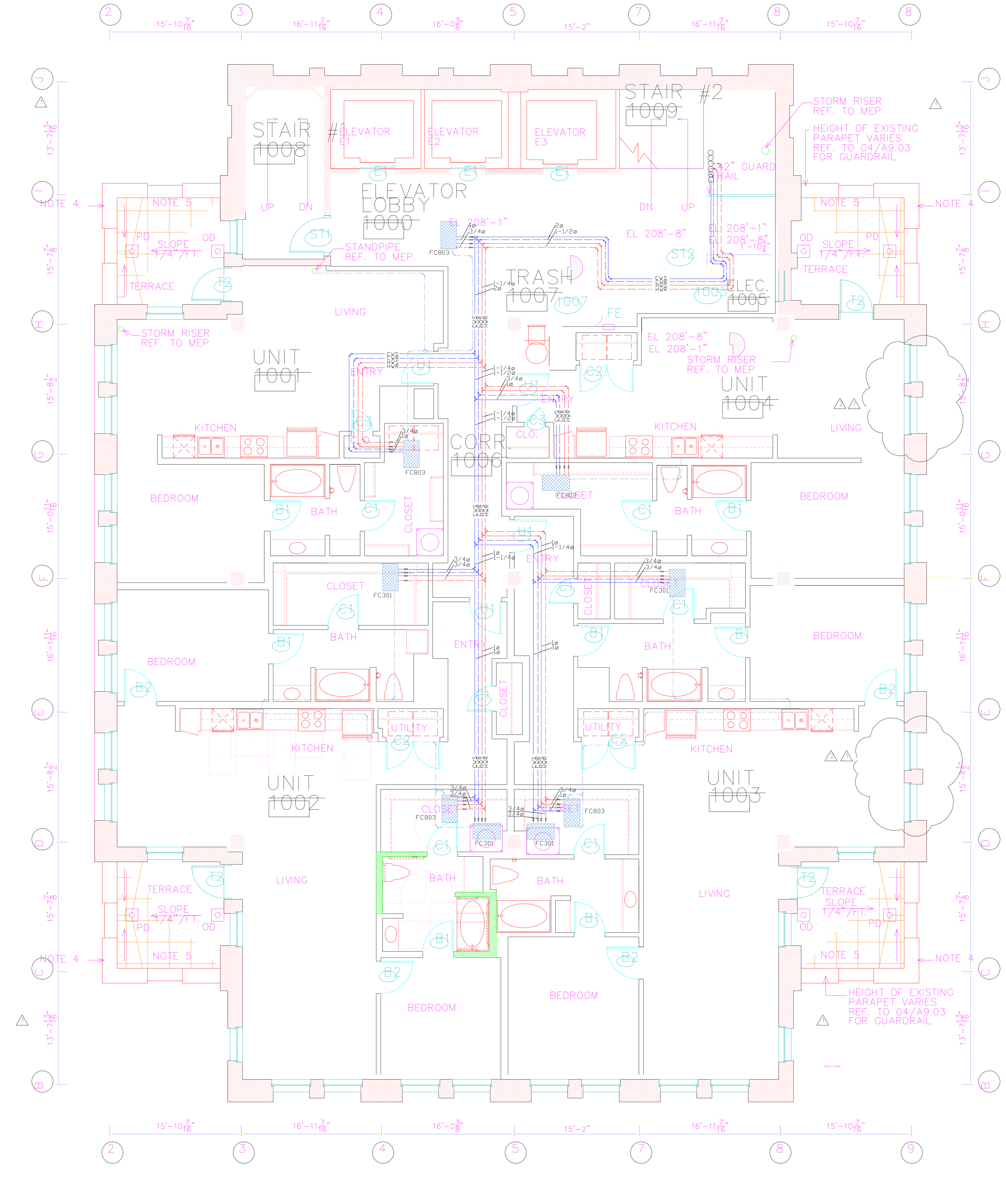


SEVENTH FLOOR
NTS



TENTH FLOOR
NTS

TRUE NORTH
PLAN NORTH

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION
DRAWN BY: GW · DATE: 13/04/26	UBC ASHRAE TEAM
CHECKED BY · DATE	PIPING SYSTEMS - SEVENTH FLOOR AND TENTH FLOOR
ASHRAE University of British Columbia Student Branch	DRAWING NO. PI-003 · REV 0

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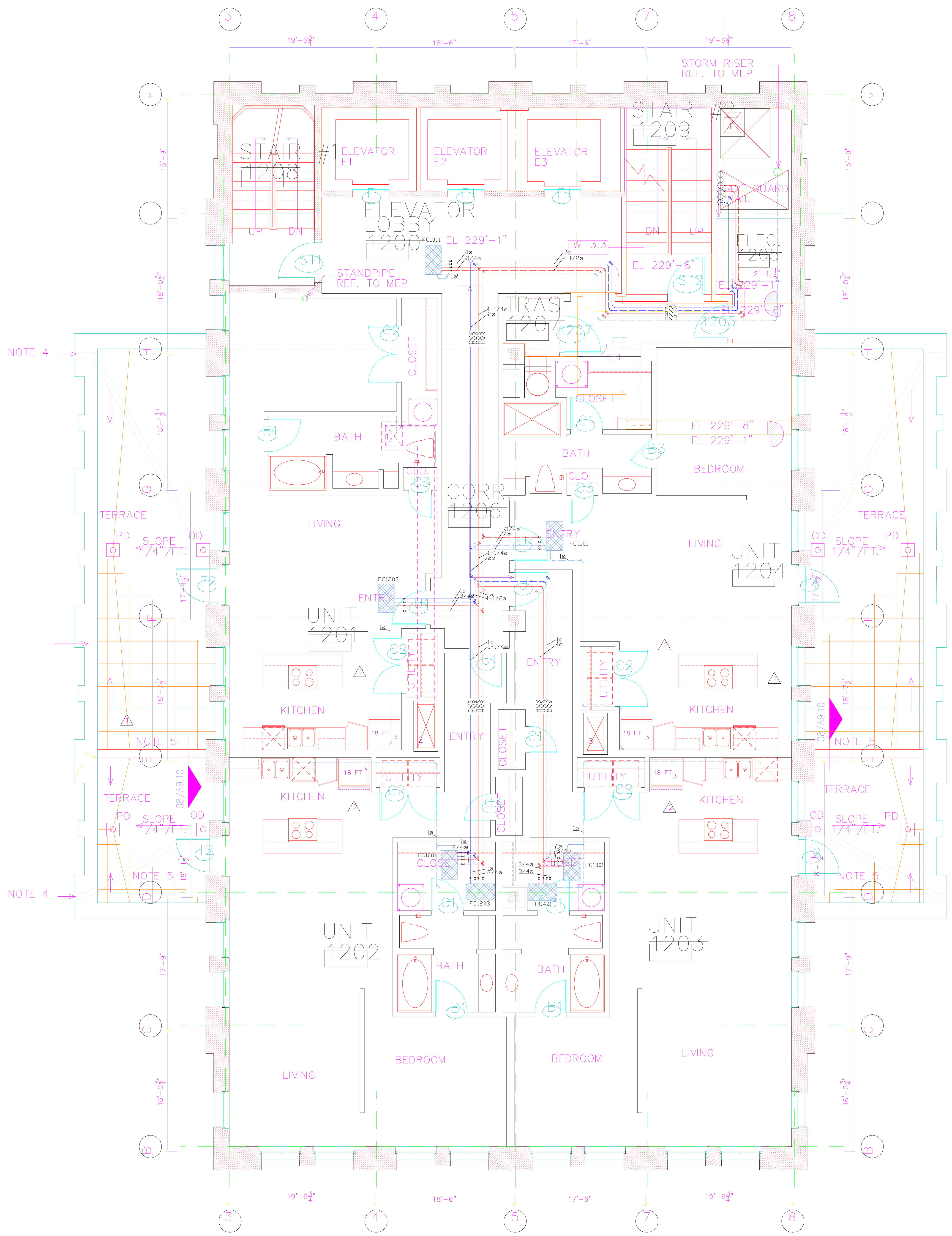
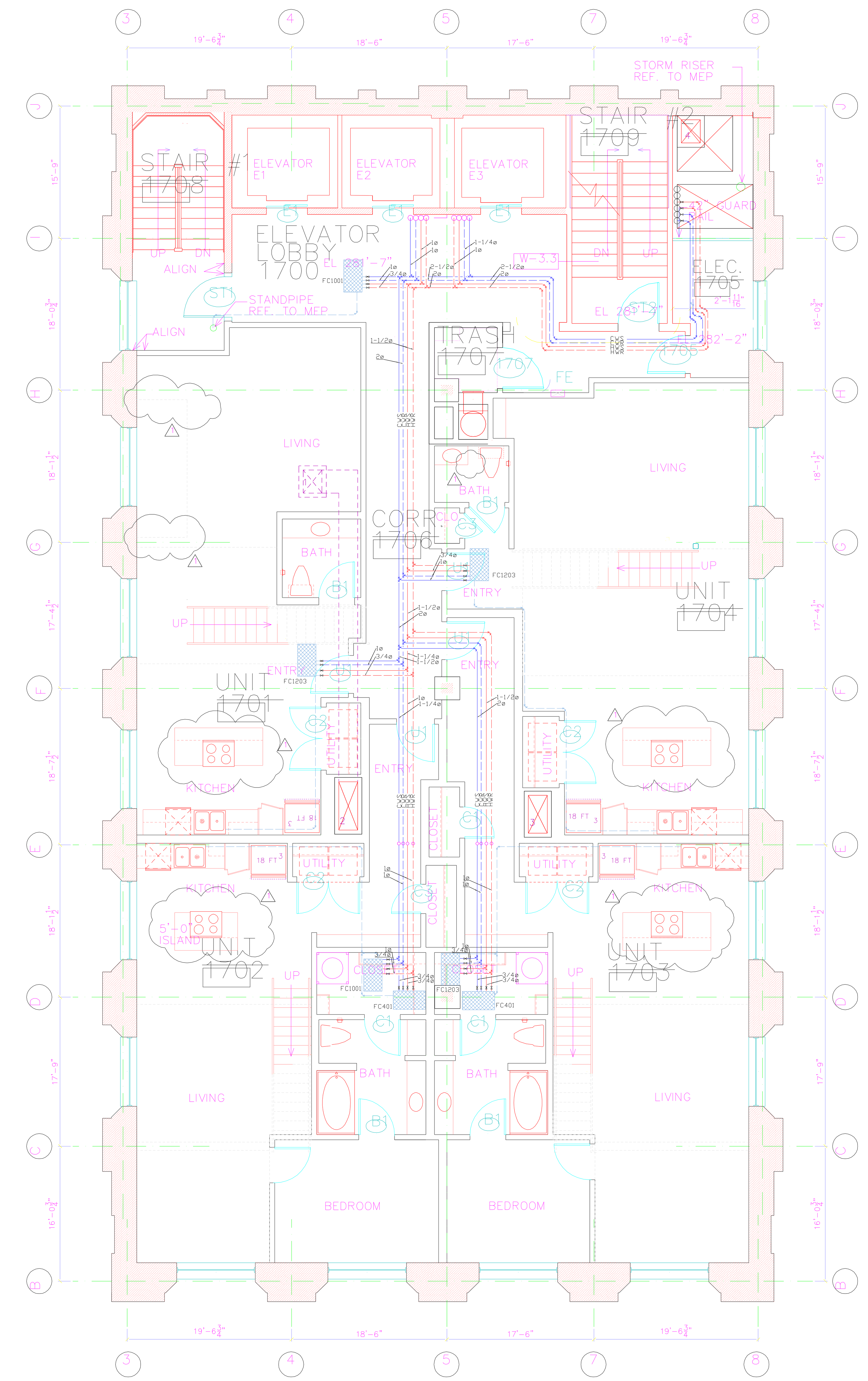


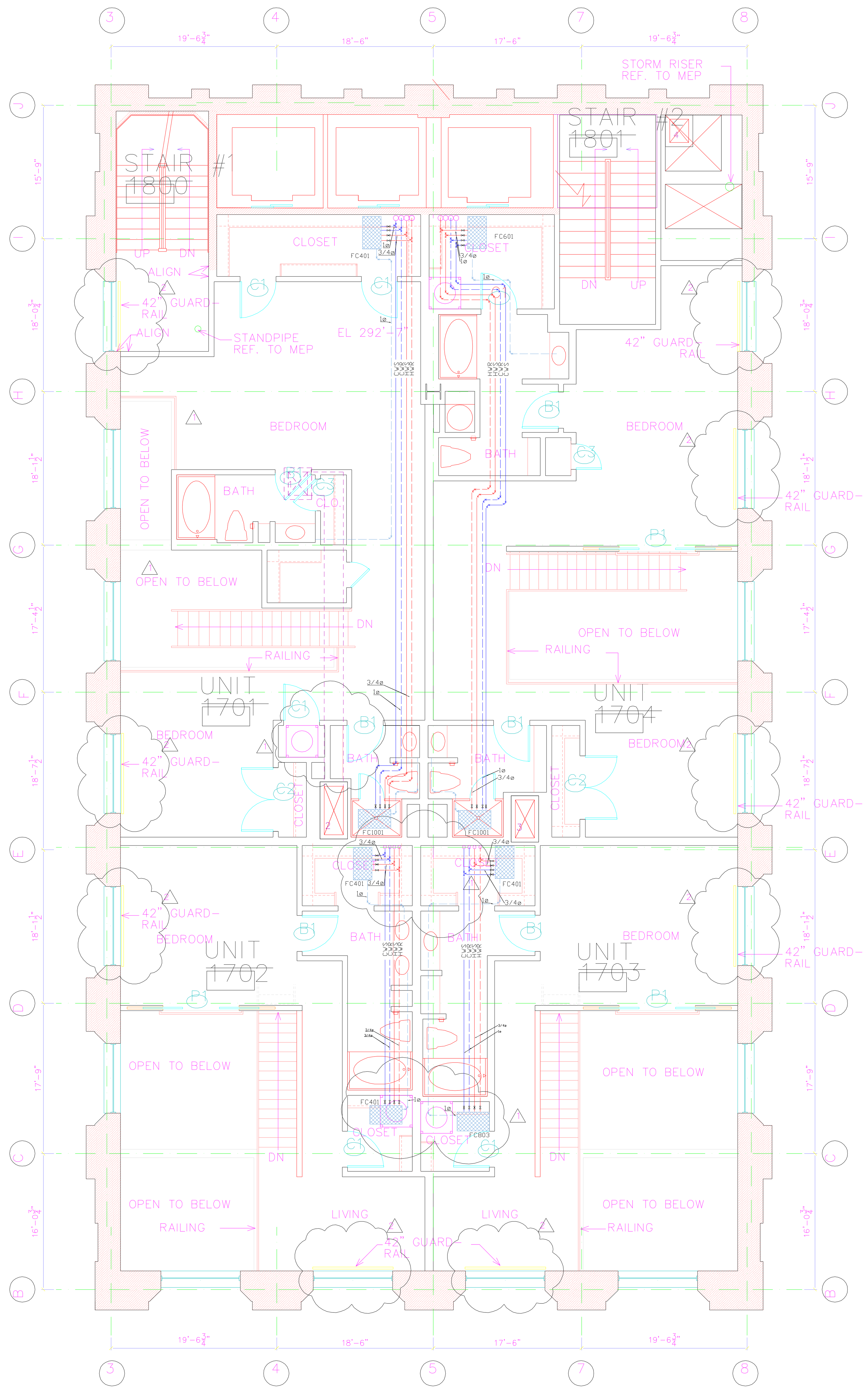
TWELFTH FLOOR
NTS



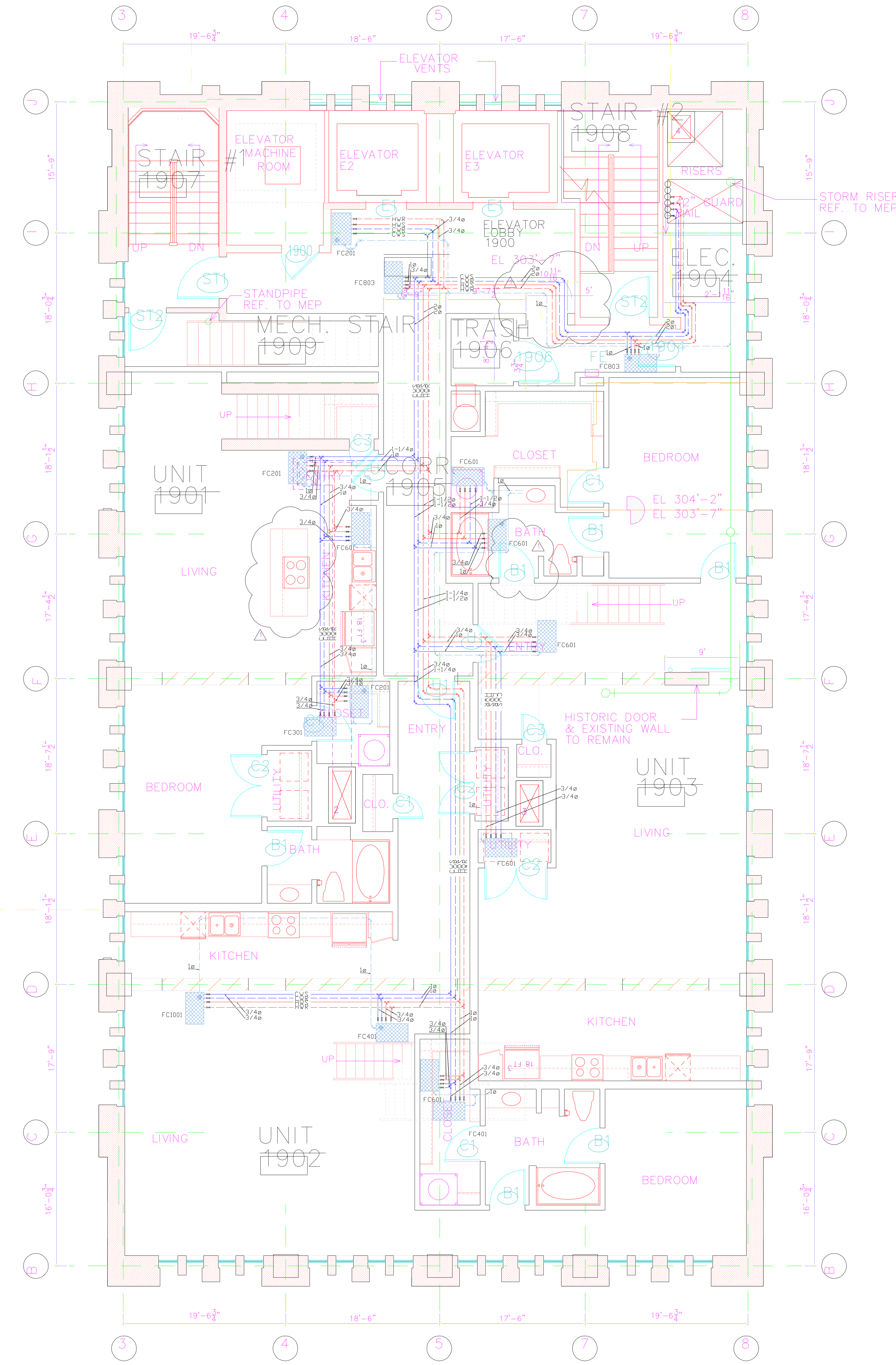
SEVENTEENTH FLOOR
NTS

TRUE NORTH
PLAN NORTH

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION
DRAWN BY: GW — DATE: 13/04/26	UBC ASHRAE TEAM
CHECKED BY — DATE	PIPING SYSTEMS - TWELFTH FLOOR AND SEVENTEENTH FLOOR
ASHRAE University of British Columbia Student Branch	DRAWING NO. PI-004 — REV 0



EIGHTEENTH FLOOR
NTS



NINETEENTH FLOOR
NTS

TRUE NORTH
PLAN NORTH

REV 0		ISSUED TO ASHRAE COMPETITION
No.		REVISION
DRAWN BY	DATE	
GW	13/04/26	UBC ASHRAE TEAM
CHECKED BY	DATE	

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University of British Columbia Student Branch

PIPING SYSTEMS - EIGHTEENTH FLOOR AND NINETEENTH FLOOR

DRAWING NO.	REV
PI-005	0

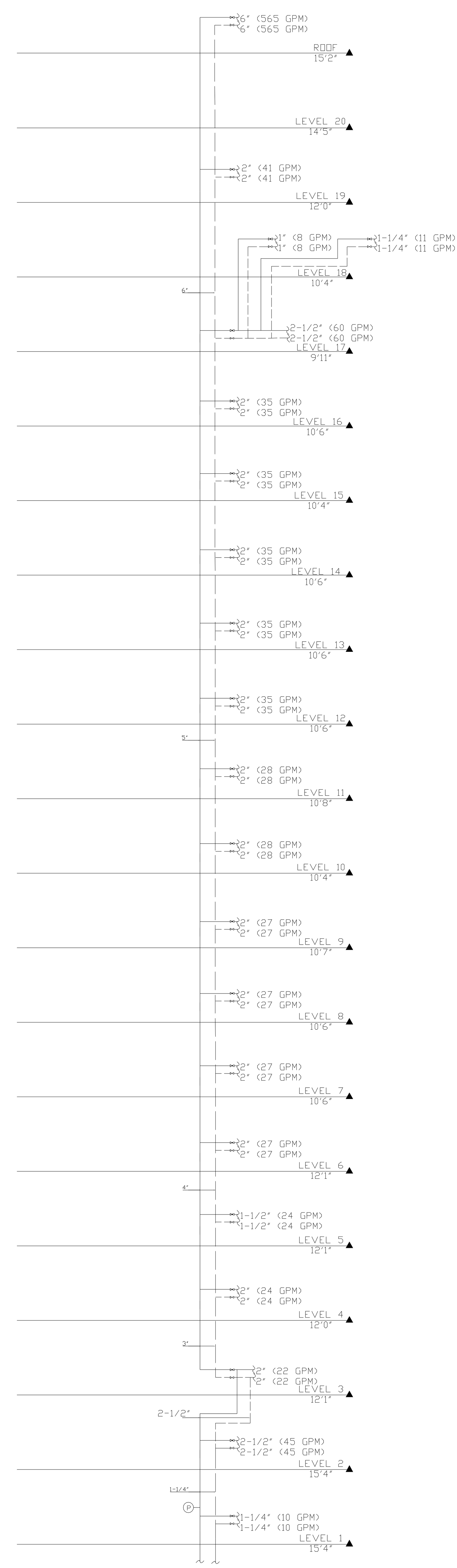
MECHANICAL SYMBOLS AND ABBREVIATIONS

	SUPPLY PIPE
	RETURN PIPE
	FAN COIL
	RADIANT PANEL
	PUMP
	GATE VALVE
	GLOBE VALVE
	CHECK VALVE
	MOTORIZED VALVE
	PRESSURE SENSOR FOR MODULATING PUMP
HWS	HOT WATER SUPPLY LINE
HWR	HOT WATER RETURN LINE
CWS	COLD WATER SUPPLY LINE
CWR	COLD WATER RETURN LINE
HRS	HEAT RECOVERY SUPPLY LINE
HRR	HEAT RECOVERY RETURN LINE

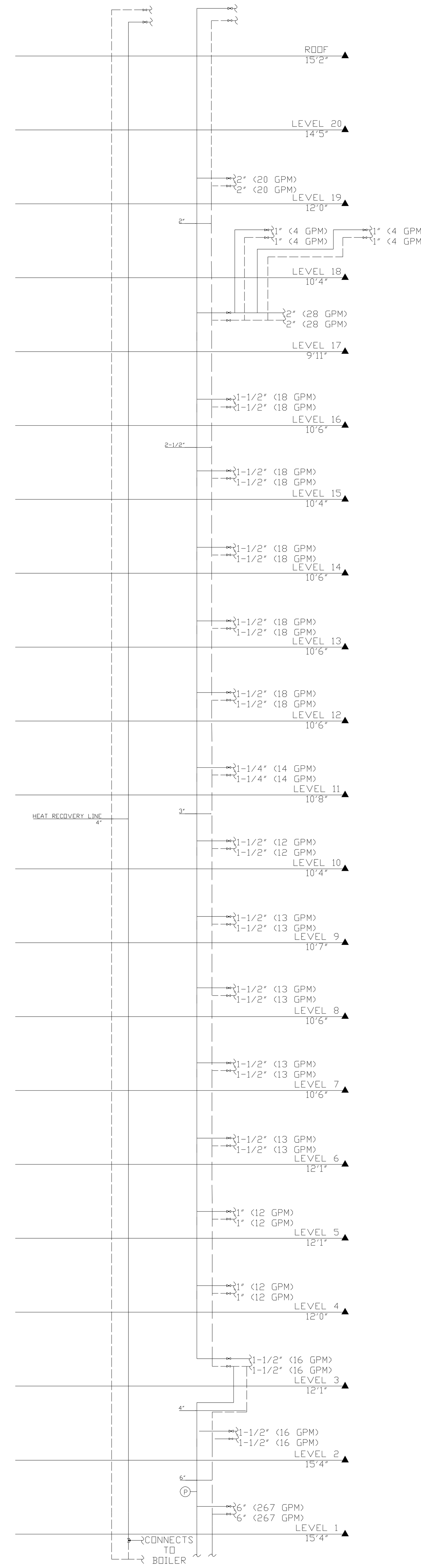
Pumps

Tag	P-1	P-2	P-3
Service	HWL	CWL	HWL Heat Recovery
Location	Maint. 113	Roof	Maint. 113
Type	Vertical Inline Centrifugal with VSD	Vertical Inline Centrifugal with VSD	Vertical Inline Centrifugal with VSD
Flow (gpm)	277	96	153 gpm
Head (Ft.)	285	37	9.84
Motor Hp. (Kw)	27	3.5	0.44
Pump Speed (RPM)	3550	2990	1450
Electrical (Volts/Phase/Hz)	110/3/60	110/3/60	110/3/60
Impeller Size (Ins.)	8.7	4.84	4.84
Efficiency	58%	76%	70%
Make	Armstrong 4302	Armstrong 4302	Armstrong 4302
Model	150-250	100-150	150-150
Pump Size	150-250	100-150	150-150
Notes/Options			

Notes/Options:
1. All bronze construction.
2. C/w quick connect electrical connection (twist-lock or equal).



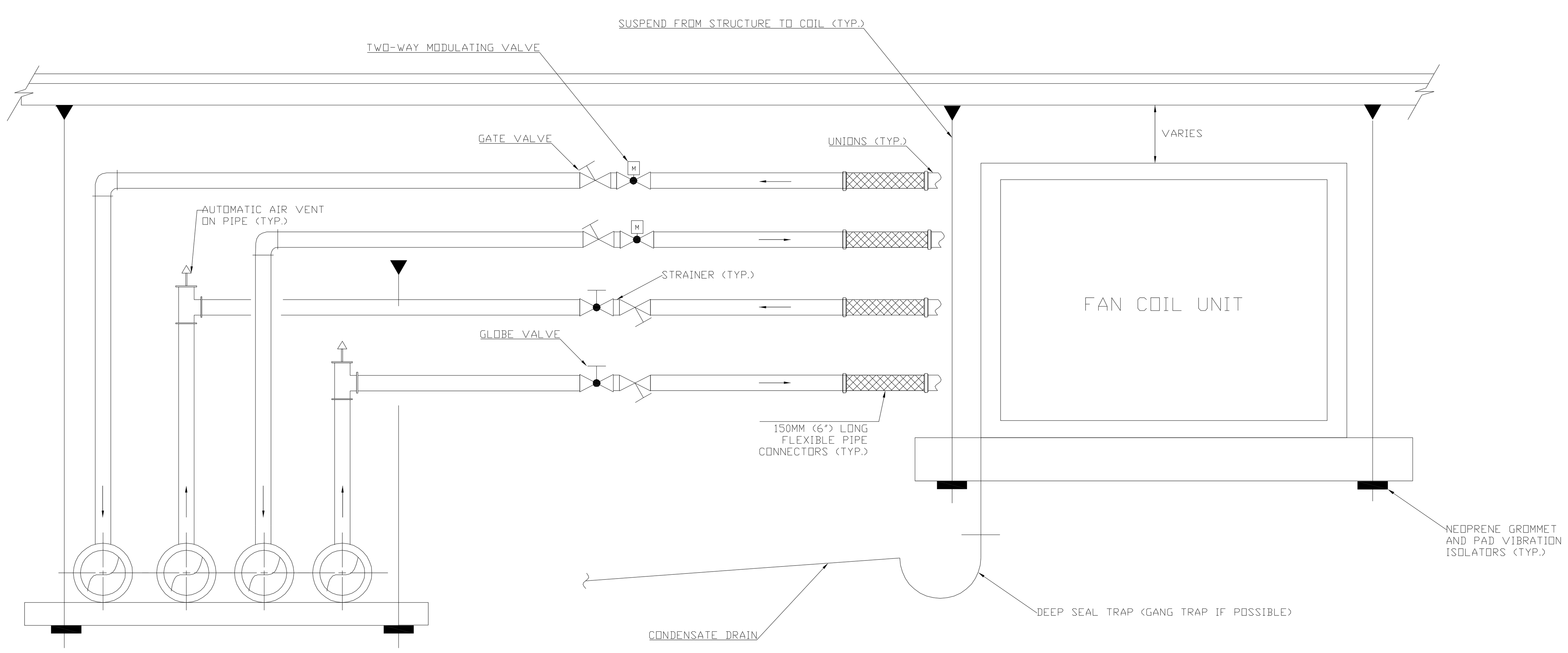
COOLING WATER RISERS
NTS



HEATING WATER RISERS
NTS

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION
DRAWN BY GW DATE 13/05/02	UBC ASHRAE TEAM
CHECKED BY DATE	PIPING SYSTEMS - SCHEMATIC FLOW DIAGRAM
ASHRAE University of British Columbia Student Branch	DRAWING NO. PI-006 REV 0

GENERAL NOTES:
- NOMINAL COOLING SUPPLY AIR TEMPERATURE BASED ON 56°F WITH 78°F RETURN AIR TEMPERATURE
- NOMINAL HEATING SUPPLY AIR TEMPERATURE / CAPACITY BASED ON 72°F RETURN AIR TEMPERATURE
- NOMINAL TOTAL MAXIMUM AIR PRESSURE DROP THROUGH COIL SHALL BE NOT MORE THAN 0.75" W.C. (187 Pa)



FAN COIL UNIT INSTALLATION DETAIL
SCALE NTS

VAV Terminal Units

Tag	VAV-A	VAV-B	VAV-C	VAV-D	VAV-E	VAV-F
Service	L1 Auxiliary Spaces	L1 Mech Rm.	L1 Tenant 103	L1 Tenant 104 Core	L1 Tenant 104 South	L1 Tenant 104 West
Location	Elevator Lobby 100	Maint. 113	Elevator Lobby 100	Future Tenant 104	Future Tenant 104	Future Tenant 104
Air Side Data:						
Airflow (cfm)	1000-350	100-50	650-200	650-200	1400-500	2400-700
External SP (ins.WC)	0.26	0.02	0.32	0.32	0.26	0.35
Cooling Mode Ent. Air Temp. DB/WB (F)	57/56	57/56	57/56	57/56	57/56	57/56
Heating Mode Ent. Air Temp. DB (F)	55	55	55	55	55	55
Heating Mode Lvg. Air Temp. DB (F)	77	99	79	81	80	86
Water Side Data:						
Heating Flow (USgpm)	0.6	0.3	0.43	0.38	0.81	1.68
Heating EWT (F)	160	160	160	160	160	160
Heating LWT (F)	130	140	130	130	130	130
Press. Drop (Ft. head)	0.08	0.07	0.16	0.13	0.16	3.15
Make	EH Price	EH Price	EH Price	EH Price	EH Price	EH Price
Model	SDV5	SDV5	SDV5	SDV5	SDV5	SDV5

Boiler(s)

Tag	B-1	B-2
Service	All	All
Location	Maint. 113	Maint. 113
Input Capacity (Btuh)	2160000	2160000
Output Capacity (Btuh)	1953000	1953000
Fuel (Gas)	Gas	Gas
Efficiency (%)	90.4	90.4
Water Flow (USgpm)	148	148
EWT (F)	140	140
LWT (F)	167	167
Pressure Rating (Psig)	30	30
Operating Wt. (lbs)	2322	2322
Electrical (Volts/Phase/Hz)	115/1/60	115/1/60
Notes/Options	Veissmann Vitocrossal 300 CT3-57	Veissmann Vitocrossal 300 CT3-57

Notes/Options:
1. C/w DDC Controls connection (dry contacts or equivalent) to monitor boiler operation.
2. C/w all standard trim: low water cutoff, pressure and temperature gauges at boiler connections, pressure relief valve(s), flow switch(s), disconnect switch (for field mounting by Div.16).

Heating Coils

Tag	HC-1	HC-2
Service	AHU-1	AHU-2
Location	Roof	Loading Area 111
Airflow (CFM)	12650	6100
Air Pressure Drop (ins.WC)	0.09	0.05
Face Velocity (Fpm)	448	379
Air On Temperature DB (F)	27.4	38.5
Air Off Temperature DB (F)	72.2	70.6
Total Load (Btuh)	612490	211300
Water Flow (gpm)	60.0	21.1
EWT (F)	160.0	160.0
LWT (F)	139.2	139.6
WPD (Ft. head)	2.8	2.7
Make	Engineered Air	Engineered Air
Model	Custom	Custom
Notes/Options		

Notes/Options:
Common to all coils: Coil supplier shall select rows and fin spacing based on the above performance data.

Chiller – Air Cooled

Tag	CH-1	
Service	All	
Location	Roof	
Total Cooling Load (Tons)	200	
Ambient Air DB/WB (F)	95	
Min. Stages of Cooling	1	
EWT (F)	57	
LWT (F)	44	
CHW Flow (USgpm)	490	
Water PD (Ft. Head)	25	
Running Load Amps	313.1	
Electrical (Volts/Phase/Hz)	575/3/60	
Refrigerant Type	R410a	
Refrigerant Compressor Type	Scroll	
Sound Power Level- Radiated (dbA)	66	
Full Load Efficiency (Kw/Ton)	1.208	
Make	AERMEC	
Model	NRL280000TA00800	
Notes/Options	Full Heat Recovery Included	

Notes/Options:
1. C/w solid state control panel with starter(s) with dry contact (or equal) terminal strip for Building Automation System (DDC) connection to read all operating characteristics of unit.
2. C/w matched seismically rated/enclosed vibration isolators for field mounting.
3. C/w factory mounted removable/hinged screening/louvred panels around condenser coils and compressor sections.
4. C/w insulated and heat traced "wet" sections.
5. C/w hot gas bypass or other acceptable low ambient/low load operation kit to allow operation at low loads.

Air Handling Units

Tag	AHU-1	AHU-2
Service	Condo Suites (L2-L20)	1st Floor Retail (L1)
Location	Roof	Loading Area 111
Supply Air Side :		
Airflow (CFM)	12650	6100 (+1700 relief)
Total Static Press. (ins.WC)	1.88	1.98
Fan Hp	7.5	5
Exhaust Air Side		
Airflow (CFM)	9500	7800
Total Static Press. (ins.WC)	1.97	1.54
External SP (ins.WC)	0.5	0.5
Fan Hp	7.5	5
Make	Engineered Air	Engineered Air
Model	Custom	Custom
Notes/Options	Custom w Heat Pipes	Custom w Heat Pipes

Notes/Options:
1. Complete with one set of spare filters to be turned over to Owner at Substantial Performance point.
2. C/w extended lube lines (pre-charged at factory) to exterior of AHU Casing with labels.

Fan-Coil Units

Tag	FC201	FC301	FC401	FC601	FC803	FC1001	FC1203
Service	Various Suites	Various Suites	Various Suites	Various Suites	Various Suites	Various Suites	Various Suites
Location	In Suite	In Suite	In Suite	In Suite	In Suite	In Suite	In Suite
Air Side Data:							
Airflow (cfm)	246	313	381	609	676	1015	993
External SP (ins.WC)	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Cooling Mode Ent. Air Temp. DB (F)	67	67	67	67	67	67	67
Heating Mode Ent. Air Temp. DB (F)	70	70	70	70	70	70	70
Fan Motor Power (Watts)	37	39	58	79	122	145	160
Electrical (Volts/Phase/Hz)	115/1/60	115/1/60	115/1/60	115/1/60	115/1/60	115/1/60	115/1/60
Water Side Data:							
Cooling Flow (USgpm)	1.2	1.4	1.7	3.0	4.0	5.3	6.1
Cooling EWT (F)	45	45	45	45	45	45	45
Press. Drop (Ft. head)	7.75	9.79	3.39	11.45	6.95	11.15	10.39
Heating Flow (USgpm)	0.6	0.7	0.9	1.5	1.8	2.8	2.7
Heating EWT (F)	160	160	160	160	160	160	160
Press. Drop (Ft. head)	0.95	1.23	2.25	6.82	11.35	36.37	32.29
Make	Trane	Trane	Trane	Trane	Trane	Trane	Trane
Model	Hor. Concealed	Hor. Concealed	Hor. Concealed	Hor. Concealed	Hor. Concealed	Hor. Concealed	Hor. Concealed

Heat Recovery

Tag	HR-1	HR-2
Service	Condo Suites (L2-L20)	1st Floor Retail (L1)
Location	AHU-1	AHU-2
Type	Heat Pipe	Heat Pipe
Heating Side:		
Incoming Temperature DB/RH(F/%)	72.0/50	72.0/50
Leaving Temp. DB/RH(F/%)	36.9/100	38.0/100
Flow (cfm)	9500	6100
Pressure Drop (ins.WC)	0.67	0.49
Heated Side :		
Incoming Temperature DB/RH(F/%)	-12.0/80	-12.0/80
Leaving Temp. DB/RH(F/%)	27.4/11	38.5/06
Flow (cfm)	12650	6100
Pressure Drop (ins.WC)	0.52	0.38
Total Load (Btuh)	538400	332800
Cooling Side:		
Incoming Temperature DB/RH(F/%)	74/60	74/60
Leaving Temp. DB/RH(F/%)	83.8/43	81.7/46
Flow (cfm)	9500	6100
Pressure Drop (ins.WC)	0.60	0.44
Cooled Side :		
Incoming Temperature DB/RH(F/%)	87.1/40	87.1/40
Leaving Temp. DB/RH(F/%)	79.7/57	79.7/57
Flow (cfm)	12650	6100
Pressure Drop (ins.WC)	0.60	0.44
Total Load (Btuh)	100600	50800
Make	Engineered Air	Engineered Air
Model	Custom	Custom

Notes/Options:
1. C/w stainless steel drain pan under all coil/plate sections.
2. C/w minimum 24"x24" (600x600mm) access doors at connecting ductwork for cleaning access.
3. C/w air filters on incoming air side and leaving air side, see Specifications and drawings.

Cooling Coils

Tag	CC-1	CC-2
Service	AHU-1	AHU-2
Location	Roof	Loading Area 111
Airflow (CFM)	9500	6100
Air Pressure Drop (ins.WC)	0.09	0.39
Face Velocity (Fpm)	355	373
Air On Temperature DB/WB (F)	79.7/68.6	79.4/68.5
Air Off Temperature DB/WB (F)	72.0/65.5	57.2/56.6
Total Load (Btuh)	103160	236230
Sensible Load (Btuh)	78910	146350
Water Flow (gpm)	20.8	49.9
EWT (F)	45	45.0
LWT (F)	54.9	54.4
WPD (Ft. head)	2.5	5.4
Make	Engineered Air	Engineered Air
Model	Custom	Custom
Notes/Options		

Notes/Options:
Common to all coils: Coil supplier shall select rows and fin spacing based on the above performance data. All Refrigerant coils for variable air volume systems shall be alternate row arrangement for matched # of refrigerant circuits/stages.
1. C/w stainless steel drain pan.
2. C/w epoxy painted steel drain pan.

Heat Exchangers - Fluid Type

Tag	HEX-1	HEX-2
Service	Boiler 1	Boiler 2
Location	Maint. 111	Maint. 111
Type	Plate	Plate
Heating Side:		
Medium	Hot Water	Hot Water
Incoming Temp. (F)	167	167
Leaving Temp. (F)	140	140
Flow (gpm)	148.0	148.0
Pressure Drop (Ft. Head)	8.08	8.08
Heated Side :		
Medium	Hot Water	Hot Water
Incoming Temp. (F)	136	136
Leaving Temp. (F)	162	162
Flow (gpm)	153.2	153.2
Pressure Drop (Ft. Head)	8.54	8.54
Total Load (Btuh)	1950000	1950000
Make	GEA	GEA
Model	NT50X	NT50X
Pressure Rating (Psig)	300	300
Notes/Options:		

Notes/Options:
1. C/w pressure relief valves on primary fluid and secondary fluid pipes.
2. C/w one spare set of gaskets to be turned over to Owner at Substantial Performance.
3. C/w floor or wall rack/stand as required for specific installation.

Expansion Tanks

Tag	ET-1	ET-2
Service	CW Line	HW Line
Location	Roof	Roof
Acceptance Volume (USGal)	2.4	11.3
Total Volume (USGal)	8	21.7
Pre-Charge Pressure (Psig)	12	12
Height (ins.)	19.5	29.5
Diameter (ins.)	12	16.5
Make	Amtrol	Amtrol
Model	AX-15(V)	AX-40(V)
Notes/Options:		

Notes/Options:
1. C/w floor stand and seismic restraint strap to wall anchors.
2. C/w hanging saddles for suspended horizontal installation.
3. C/w spare gasket and/or bladder to be turned over to Owner at Substantial Performance.

Suspended Radiant Panels

Tag:	RP-1
Service	Retail Space
Location	Future Tenant 104
Ambient air (F)	68
Heating Data :	
Heating EWT (F)	160
Heating LWT (F)	75
Press. Drop (Ft. head)	0.5
Heating Capacity (Btuh per sq.ft)	360
Notes/Options:	Heating Only

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION

DRAWN BY: GW DATE: 13/05/02
CHECKED BY: DATE:

UBC ASHRAE TEAM

PIPING SYSTEMS - FAN COIL DETAIL & EQUIPMENT SCHEDULE

ASHRAE University of British Columbia Student Branch

DRAWING NO.: MI-002 REV: 0

System Checksums
By ACADEMIC

L1 Tenant 104 S — VAV w/Baseboard Skin Heating

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 8 / 15; Outside Air: OADB/WB/HR: 83 / 74 / 111

CLG Space Peak — Mo/Hr: 8 / 15; OADB: 83

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	0	0	0	0	0
Glass Solar	20,675	0	20,675	24	20,675	48
Glass/Door Cond	5,107	0	5,107	6	5,107	12
Wall Cond	4,389	599	4,988	6	4,389	10
Partition/Door	0		0	0	0	0
Floor	-1,316		-1,316	-2	-1,316	-3
Adjacent Floor	0	0	0	0	0	0
Infiltration	11,689		11,689	14	2,492	6
Sub Total ==>	40,544	599	41,143	49	31,347	73
Internal Loads						
Lights	2,424	606	3,031	4	2,424	6
People	17,998	0	17,998	21	9,052	21
Misc	0	0	0	0	0	0
Sub Total ==>	20,423	606	21,029	25	11,477	27
Ceiling Load	239	-239	0	0	239	1
Ventilation Load	0	0	21,239	25	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-449	-449	-1		
Sup. Fan Heat			1,451	2		
Ret. Fan Heat		0	0	0		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	61,205	517	84,412	100.00	43,062	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: -9

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	0	0.00
Glass Solar	0	0	0.00
Glass/Door Cond	-20,279	-20,279	16.96
Wall Cond	-10,617	-12,067	10.09
Partition/Door	0	0	0.00
Floor	-26,991	-26,991	22.57
Adjacent Floor	0	0	0
Infiltration	-24,292	-24,292	20.31
Sub Total ==>	-82,178	-83,629	69.94
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-541	0	0.00
Ventilation Load	0	-36,860	30.82
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	0	0	0.00
Exhaust Heat		910	-0.76
OA Preheat Diff.		0	0.00
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-82,719	-119,578	100.00

TEMPERATURES

	Cooling	Heating
SADB	48.0	0.0
Ra Plenum	75.5	0.0
Return	75.5	0.0
Ret/OA	78.2	0.0
Fn MtrTD	0.1	0.0
Fn BldTD	0.2	0.0
Fn Frict	0.6	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	1,431	429
Terminal	1,431	429
Main Fan	1,431	429
Sec Fan	0	0
Nom Vent	514	429
AHU Vent	514	429
Infil	283	283
MinStop/Rh	429	429
Return	1,714	712
Exhaust	797	0
Rm Exh	0	0
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	35.9	100.0
cfm/ft²	0.96	0.29
cfm/ton	203.49	
ft²/ton	211.75	
Btu/hr·ft²	56.67	-28.31
No. People	45	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	7.0	84.4	49.6	1,431	78.2	67.0	81.6	47.1	46.9	47.2
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	7.0	84.4								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	1,490		
Part	0		
Int Door	0		
ExFlr	3,862		
Roof	0	0	0
Wall	2,311	462	20
Ext Door	54	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	0.0	0	0.0	0.0
Aux Htg	-82.7	0	0.0	0.0
Preheat	-32.2	514	-9.0	47.1
Reheat	-10.0	429	47.1	68.0
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-124.9			

Project Name: DP&L Building
Dataset Name: DP&L_v11.trc

TRACE® 700 v6.2.8 calculated at 06:15 PM on 04/01/2013
Alternative - 1 System Checksums Report Page 5 of 105

System Checksums
By ACADEMIC

L2 S201 — Fan Coil

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 7 / 18; Outside Air: OADB/WB/HR: 81 / 73 / 112

CLG Space Peak — Mo/Hr: Sum of; OADB: Peaks

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	0	0	0	0	0
Glass Solar	10,714	0	10,714	56	10,714	72
Glass/Door Cond	1,629	0	1,629	8	1,629	11
Wall Cond	457	190	647	3	457	3
Partition/Door	0		0	0	0	0
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	1,334		1,334	7	167	1
Sub Total ==>	14,134	190	14,324	75	12,966	87
Internal Loads						
Lights	176	44	220	1	618	4
People	1,663	0	1,663	9	1,300	9
Misc	0	0	0	0	0	0
Sub Total ==>	1,840	44	1,884	10	1,918	13
Ceiling Load	53	-53	0	0	68	0
Ventilation Load	0	0	2,183	11	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-19	-19	0		
Sup. Fan Heat			851	4		
Ret. Fan Heat		0	0	0		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	16,027	162	19,223	100.00	14,952	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: -9

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	0	0.00
Glass Solar	0	0	0.00
Glass/Door Cond	-5,746	-5,746	30.89
Wall Cond	-923	-1,306	7.02
Partition/Door	0	0	0.00
Floor	0	0	0.00
Adjacent Floor	0	0	0
Infiltration	-3,309	-3,309	17.79
Sub Total ==>	-9,978	-10,362	55.70
Internal Loads			
Lights	34	42	-0.23
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	34	42	-0.23
Ceiling Load	-79	0	0.00
Ventilation Load	0	-524	2.81
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	-41	-41	0.22
Exhaust Heat		28	-0.15
OA Preheat Diff.		-7,748	41.65
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-10,064	-18,604	100.00

TEMPERATURES

	Cooling	Heating
SADB	59.0	80.8
Ra Plenum	75.2	69.7
Return	75.2	69.7
Ret/OA	75.2	69.2
Fn MtrTD	0.1	0.0
Fn BldTD	0.2	0.0
Fn Frict	0.6	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	840	840
Terminal	840	840
Main Fan	840	840
Sec Fan	0	0
Nom Vent	94	94
AHU Vent	94	94
Infil	38	38
MinStop/Rh	0	0
Return	825	825
Exhaust	79	79
Rm Exh	53	53
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	11.2	11.2
cfm/ft²	1.09	1.09
cfm/ton	488.93	
ft²/ton	448.30	
Btu/hr·ft²	26.77	-24.15
No. People	8	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	1.6	19.2	15.2	840	75.5	63.8	69.8	58.4	56.3	64.3
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.1	1.4	1.1	94	85.0	74.9	114.6	75.0	71.2	109.5
Total	1.7	20.6								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	770		
Part	0		
Int Door	0		
ExFlr	0		
Roof	0	0	0
Wall	344	149	43
Ext Door	0	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-10.9	840	69.2	80.8
Aux Htg	0.0	0	0.0	0.0
Preheat	0.0	0	0.0	0.0
Humidif	0.0	0	0.0	0.0
Opt Vent	-7.8	94	-9.0	65.0
Total	-18.6			

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TRACE® 700 v6.2.8 calculated at 06:15 PM on 04/01/2013
Alternative - 1 System Checksums Report Page 56 of 105

NOTE: THESE TWO CHECKSUM SHEETS (FLOOR 1 ROOM 104 SOUTH & FLOOR 2 SUITE 201) ARE REPRESENTATIVE SAMPLE DATA TO SHOWCASE THE TWO SYSTEMS

REV 0	ISSUED TO ASHRAE COMPETITION
No.	REVISION
DRAWN BY: GW — DATE: 13/04/26	UBC ASHRAE TEAM
CHECKED BY — DATE	TRANE CHECKSUM
ASHRAE University of British Columbia Student Branch	DRAWING NO. MI-001 — REV 0