University of Nebraska – Lincoln
Team Master Builders

ASHRAE Student Design Competition
Design Calculations
May 4th, 2015

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Executive Summary

This report is submitted by the University of Nebraska-Lincoln in Omaha, Nebraska for the Design Calculations ASHRAE Student Design Competition. The objective of the competition is to perform the design calculations to correctly size the variable air volume HVAC system for a three-story classroom and office building located in Doha, Qatar. All equipment is selected based on Trane Trace 700 load calculations and cost effectiveness.

The building heating and cooling loads are 511,400 Btu/h and 157 tons, respectively. The total VAV airflow capacity needed to support these loads is 44,500 cfm. Three rooftop air handling units and 53 single duct terminal units supply the necessary airflow to the building zones, in addition to six fan coil units that serve the stairwell areas. The special ventilation needs of the workshop are provided by fume and dust collectors. The building primary system is a chiller with thermal ice storage and heat rejection fountains to save on energy costs.

A 50-year life cycle cost analysis of the building system priced the initial system cost at $1.9 million and operation and maintenance at $4.8 million, making the total price for the system over 50 years is $6.7 million. Costs were based on provided utility and inflation rates, as well as educated assumptions on equipment costs.

The three-member UNL student team spent the spring semester working on this project as part of a design course. The team members have previously taken two courses in mechanical systems, which made this a valuable experience in applying knowledge and learning many new topics that have not been covered previously.

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Section 1: Introduction

This report will outline the decisions, methodology, and solutions that were a part of designing the HVAC system for the three-story classroom and office building located in Doha, Qatar.

The Owner has asked for a variable air volume (VAV) air handling unit (AHU) to provide heating and cooling to the building. The system was designed to offer the most cost-effective solution while complying with ASHRAE Standards 55-2010, 62.1-2010, 90.1-2010.

- ASHRAE Standard 55-2010 outlines the requirements for indoor thermal comfort
- ASHRAE Standard 62.1-2010 provides requirements for ventilation in commercial buildings
- ASHRAE Standard 90.1-2010 is the energy standard for all buildings with the exception of low-rise residential.

In addition to the standards, there are also several directives from the owner that were factors in the system design.

A variety of software programs were used on this project. Trane Trace 700 was used for the system modeling and load calculations, and AutoCAD 2015 was used for the duct and pipe layouts. The most acoustically-sensitive duct paths were analyzed with Trane Acoustics Program (TAP) 4.1 for noise analysis. A 3D exterior model of the building was rendered using Autodesk Revit 2015.

This report includes sections covering the load calculations, design conditions, and design considerations that were a part of the VAV system selection. Following that, there is a section providing a detailed description of the VAV AHU and its components, as well as the chiller and thermal ice storage system. The report concludes with a section detailing the operations and maintenance of the system. Calculation checksums and schematics are appended.

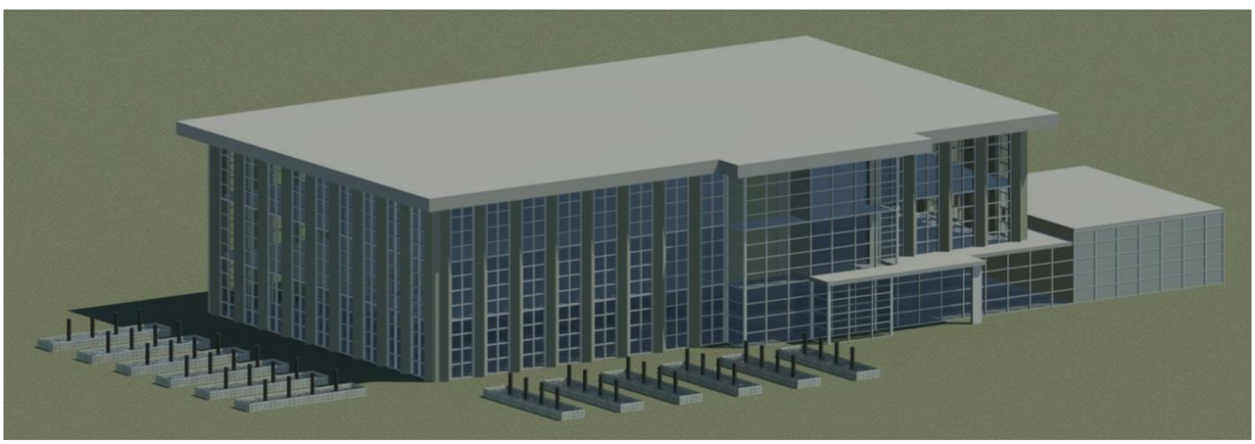


Figure 1. 3D Rendering of Proposed Building

Section 2: Load Calculations

A baseline model was created in Trane Trace 700 to calculate the overall building load. Internal load and airflow templates were made for each different type of room (office spaces, classrooms, etc.), incorporating values from ASHRAE Standards and Handbooks for occupancy, ventilation, lighting, envelope, and miscellaneous loads. The overall results from the baseline model and designed system analyses are shown in Table 1. The miscellaneous loads that were considered are outlined in Table 2.

Table 1. System Results

	Area (ft^2)	Cooling (tons)	Heating (Btu/h)	Outdoor Air (cfm)	Exhaust Air (cfm)	VAV Capacity (cfm)
Baseline System	46,700	171.6	785,400	7,522	8,757	51,184
Designed System	46,700	156.5	511,400	10,709	12,279	44,495

Table 2. Space Specific Loading

Space Type	Miscellaneous Loads	Load Values (W)
Break and Vending Areas	Refrigerator Microwave/Coffee Vending Machine	352 400/1050 240
Computer Room	4 typical racks of blade servers 2 typical racks of networking equipment	10,000
Conference	CPU/Monitor LCD TV 2 Projectors	97/29 90 200
Classrooms	Overhead Projector One laptop per student	100 36 per student
Mechanical/Electrical	Loads as per required equipment	--
Office, Individual	CPU/Monitor	97/29
Office, Executive	CPU/Monitor LCD TV	97/29 90
Office, Open Areas	CPU/Monitor per workstation/person One high-volume copy machine	97/29 per workstation/person 800
Library	CPU/Monitor: 1 workstation for every 10 people 2 LCD TV Copy Machine	97/29 per 10 people 180 800

Section 3: Design Conditions

3.1 Design Requirements

The main design criteria from the Owner's requirements for HVAC system selection are outlined in Table 3. Compliance with ASHRAE standards 55, 62.1, and 90.1 is also achieved for the system design. Climatic design conditions are based on the ASHRAE 2% design criterion.

Table 3. Design Criteria

	Office and Administrative Support Spaces	Classroom and Study Spaces	Library	Special Instruction Spaces	IT Support Spaces
Occupancy	M-F: 7am-6pm Sat: 8am-1pm	M-F: 8am-5pm	M-F: 8am-5pm	M-F: 8am-3pm	---
Summer DB	73.4˚F	73.4˚F	73.4˚F	78.8˚F	73.4˚F DB and 50% RH year-round, 24-hours a day
Summer RH	50%	50%	50%	55%	
Winter DB	70˚F	70˚F	70˚F	73.4˚F	
Noise Criterion	NC 35	NC 30	NC 30	---	---

For electricity generation, a photovoltaic array must be provided that will support 5% of the total annual electrical energy used by the building. Also, high efficiency lighting must be used to reduce the lighting power densities by 25-35%, with daylighting used whenever applicable.

3.2 Owner's Project Requirements

The Owner has requested a VAV system to provide heating, cooling, and ventilation for the building. The system shall be selected based on the lowest life cycle cost analysis within the owner's budget and incorporating as many of the Owner's goals as possible.

Several of the Owner's goals are achievable through compliance with the minimum ASHRAE standards. Furthermore, the Owner wants the system to save 15% more energy than what is required by ASHRAE standard 90.1 while remaining within budget. Other goals include the following: installation of equipment that allows for ease of access for regular service and maintenance, low utility and maintenance costs, low HVAC noise and background noise in sensitive spaces, minimize sound transmission from the shop area to adjacent noise-sensitive rooms, and indoor air quality controlled in order to promote occupant performance and productivity. The HVAC system designed must have a budget that falls reasonably within the Owner's overall construction budget of $200/ft^2.

3.3 Doha's Climate

The climate in Doha is very hot and dry for the majority of the year, classifying it as climate zone 1B in reference to ASHRAE Standard 90.1. Figures 2 through 4 show month-to-month annual data for average high and low temperature, relative humidity, and daylight hours.

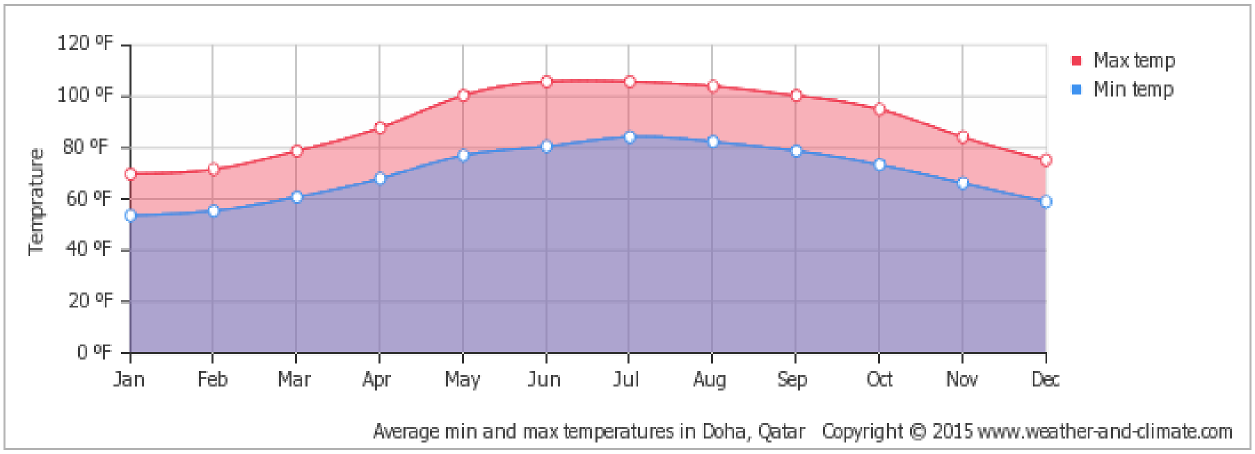


Figure 2. Average High and Low Temps in Doha (source: weather-and-climate.com)

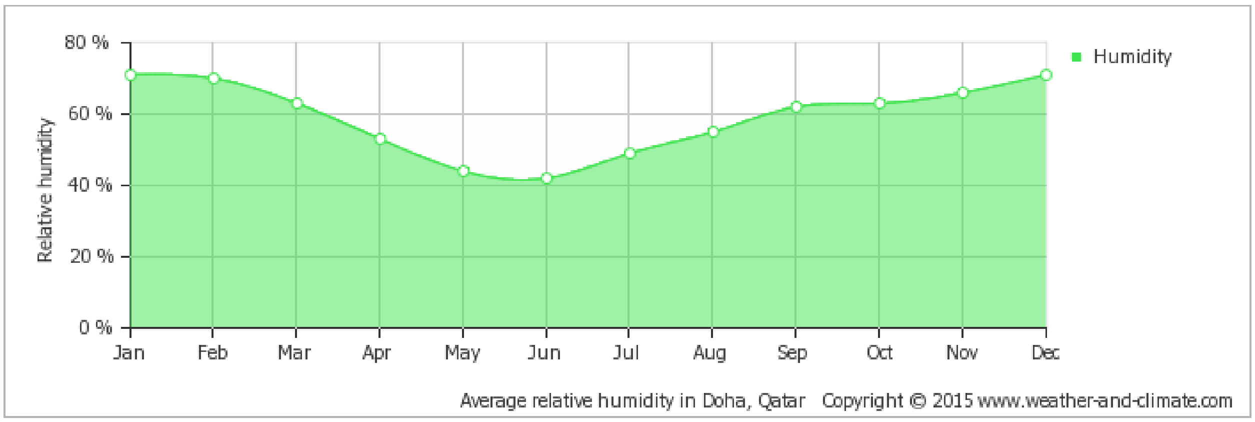


Figure 3. Average Relative Humidity in Doha (source: weather-and-climate.com)

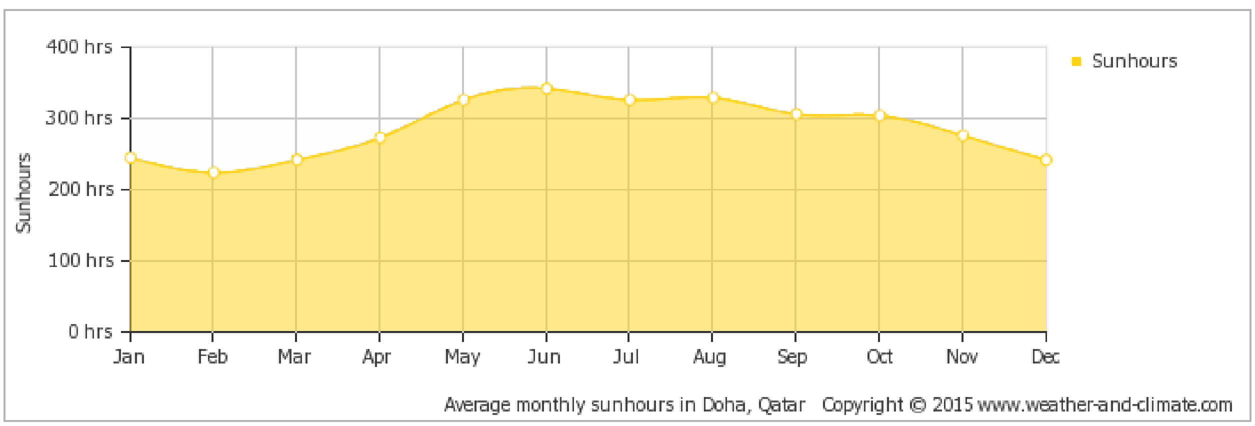


Figure 4. Average House of Sunshine in Doha (source: weather-and-climate.com)

Section 4: Design Considerations

4.1 ASHRAE Standards

4.1.1 Standard 55

The requirements outlined in ASHRAE Standard 55 were used to address the thermal comfort needs of the building occupants; this included acceptable operative temperature range and maximum draft. As per the guidelines in the Standard, calculations were done to target 80% occupant satisfaction, allowing 10% dissatisfaction from global thermal discomfort and 10% dissatisfaction from local thermal discomfort.

The chart for acceptable operative temperature and relative humidity (Figure 5) was used to determine the acceptable internal conditions for 80% satisfaction.

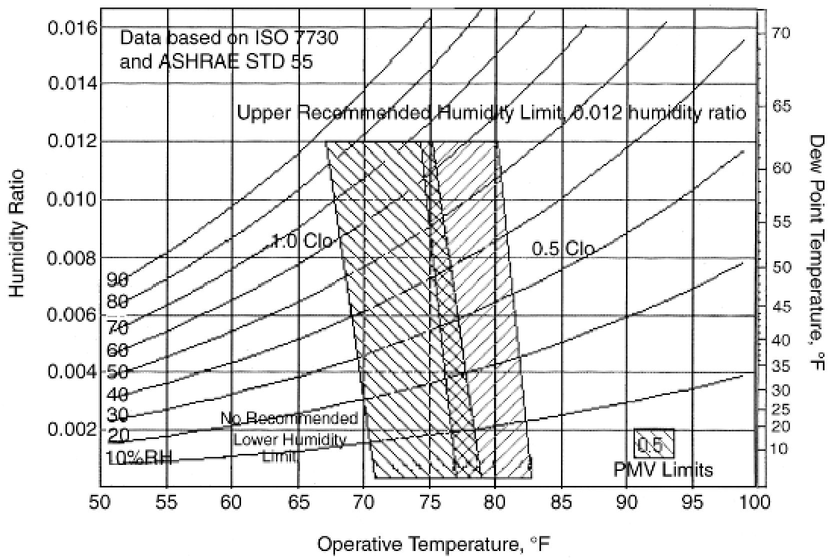


Figure 5. Acceptable Range of Operative Temperature and Relative Humidity (source: ASHRAE Standard 55-2004)

Although Doha has a hot climate all year round, cultural considerations were taken into account when assuming clothing values. Qatar has a strict dress code in which women typically wear a long dress that covers their whole body, as well as a headdress that covers their hair and face, and men wear long-sleeve shirts and long pants with a headdress that covers their hair. Based on these clothing guidelines, clo values were assumed to be around 0.8 to 1.0 clo year round. Equations 1 and 2 were used to make adjustments to the acceptable operative temperature range based on the ranges shown in Figure 6.

$$T_{max,Icl} = \frac{(I_{cl} - 0.5clo) * T_{max,1.0clo} + (1.0clo - I_{cl}) * T_{max,0.5clo}}{0.5clo} \qquad \text{(Equation 1)}$$

$$T_{min,Icl} = \frac{(I_{cl} - 0.5clo) * T_{min,1.0clo} + (1.0clo - I_{cl}) * T_{min,0.5clo}}{0.5clo} \qquad \text{(Equation 2)}$$

Where I_{cl} = Selected clo value
T_{min} = Minimum temperature from clo zone on chart, °F
T_{max} = Maximum temperature from clo zone on chart, °F

With an indoor relative humidity of 50%, the operative temperature range in which 80% of the building occupants would be satisfied is 70.8°F to 77.8°F.

The allowable mean air speed for acceptable draft was calculated by solving iteratively for v in Equation 3. Assuming DR = 20% (to meet 80% satisfaction), t_a = 73.4°F (from the Owner's directive), and Tu = 35% (for a room with mixing ventilation), the allowable mean air speed is approximately 40 fpm.

$$DR = [(93.2 - t_a) * (v - 10)^{0.62}] * [0.00004 * v * Tu + 0.066] \quad \text{(Equation 3)}$$

Where DR = predicted percentage of people dissatisfied due to draft
t_a = local area temperature, °F
v = local mean air speed, fpm
Tu = local turbulence intensity, %

4.1.2 Standard 62.1

ASHRAE Standard 62.1 provides a means to calculate minimum outdoor air rates for acceptable indoor air quality. The procedure for calculating the outdoor air intake can be simplified into three main steps.

Step 1: Breathing Zone Outdoor Airflow

The breathing zone outdoor airflow for each zone is calculated using Equation 4. Values for R_p and R_a are selected from Table 6-1 in Standard 62.1 based on room type.

$$V_{bz} = R_p * P_z + R_a * A_z \quad \text{(Equation 4)}$$

Where V_{bz} = Breathing Zone Outdoor Airflow, cfm
A_z = Zone Floor Area, ft^2
P_z = Zone Population (maximum expected), people
R_p = Outdoor airflow rate per person, cfm/person
R_a = Outdoor airflow rate per unit area, cfm/ft^2

Step 2: Zone Air Distribution Effectiveness

Values for the zone air distribution effectiveness, E_z, are found in Table 6-2 in Standard 62.1 and range from 0.5 to 1.2 depending on the configuration of the air distribution system. E_z provides a measure of how well ventilation air is distributed to zone occupants rather than being short-circuited.

Step 3: Zone Outdoor Airflow

Equation 5 is used to calculate the minimum required outdoor airflow to each zone based on V_{bz} and E_z.

$$V_{oz} = V_{bz}/E_z \quad \text{(Equation 5)}$$

Where V_{oz} = Zone Outdoor Airflow, cfm

If the air handling unit services a single zone, the total system outdoor airflow, V_{ot}, is directly equal to V_{oz}. For multiple-zone recirculating systems, a few more steps must be taken to account for recirculated outdoor air.

Step 4: Zone Primary Outdoor Air Fraction

Equation 6 is used to calculate the primary outdoor air fraction for each zone based on V_{oz} and the primary airflow to each zone.

$$Z_p = V_{oz}/V_{pz} \quad \text{(Equation 6)}$$

Where Z_p = Zone Primary Outdoor Air Fraction
V_{pz} = Zone Primary Airflow, cfm

Step 5: Uncorrected Outdoor Air Intake

The uncorrected outdoor air intake for each system is calculated using Equation 7.

$$V_{ou} = D\sum_{all\ zones}(R_p \times P_z) + \sum_{all\ zones}(R_a \times A_z) \quad \text{(Equation 7)}$$

Where V_{ou} = Uncorrected Outdoor Air Intake, cfm
D = Occupant Diversity

Step 6: System Ventilation Efficiency

Based on the calculated Z_p value, the system ventilation efficiency, E_v, is selected using Standard 62.1 Table 6-3. If Z_p is greater than 0.55, a procedure in Appendix A of the Standard is used to calculate E_z. First, the average outdoor air fraction is calculated using Equation 8 based on V_{ou}.

$$X_s = V_{ou}/V_{ps} \quad \text{(Equation 8)}$$

Where X_s = Average Outdoor Air Fraction
V_{ps} = System Primary Airflow, cfm

Second, the zone discharge outdoor air fraction is calculated using Equation 9 based on V_{oz}.

$$Z_d = V_{oz}/V_{dz} \quad \text{(Equation 9)}$$

Where Z_d = Zone Discharge Outdoor Air Fraction
V_{dz} = Zone Discharge Airflow, cfm

Finally, the zone ventilation efficiency for a single supply system is calculated using Equation 10 based on X_s and Z_d. The minimum calculated zone ventilation efficiency is the E_v that applies to the system.

$$E_{vz} = 1 + X_s - Z_d \quad \text{(Equation 10)}$$

Where E_{vz} = Zone Ventilation Efficiency

Step 7: Total Outdoor Air Intake

The total outdoor airflow for each system is calculated using Equation 11, based on V_{ou} and E_v.

$$V_{ot} = V_{ou}/E_v \quad \text{(Equation 11)}$$

A spreadsheet was made to enable simple calculation of the outdoor air ventilation for each zone and system to confirm the results from the Trace checksums. A sample zone calculation from the spreadsheet is shown in Table 4. Table 5 gives the system calculation results.

Table 4. Sample Outdoor Air Calculation for a Lecture Hall-Type Zone

Zone Area, A_z	860 ft^2
Zone Population, P_z	43 people
Area Outdoor Air Rate, R_a	0.06 cfm/person
People Outdoor Air Rate, R_p	7.5 cfm/person
Breathing Zone Outdoor Airflow, V_{bz}	374.1 cfm
Zone Air Distribution Effectiveness, E_z	1.0
Zone Outdoor Airflow, V_{oz}	374.1 cfm

Table 5. System Outdoor Airflow Results

	AHU-1	AHU-2	AHU-3
V_{ps} (cfm)	30,022	12,844	1,629
Max Z_p	0.70	0.63	0.49
E_v	0.56	0.61	0.6
V_{ou} (cfm)	9,081	3,437	796
V_{ot} (cfm)	15,149	5,351	1,358

4.1.3 Standard 90.1

The requirements presented in ASHRAE Standard 90.1 were used for equipment energy use and efficiency compliance, as well as building envelope compliance.

The owner wants the building to use 15% less energy than the Standard 90.1 baseline. The percentage of improvement is determined using Equation 12, found in Standard 90.1 Appendix G. The final system performance was compared to the baseline model results using Equation 12. Efficiency improvements were made to the final VAV system until the 15% requirement was met.

$$\%\,Improved = 100 \times \frac{Baseline\ Building\ Performance - Proposed\ Building\ Performance}{Baseline\ Building\ Performance} \quad \text{(Equation 12)}$$

The Standard 90.1 U-value and SHGC requirements for building envelope that are used in the Trace baseline model are shown in Table 6. U-values are the thermal transmission coefficients of building materials, and SHGC is the solar heat gain coefficient for fenestrations.

Table 6. Nonresidential Building Envelope Baseline Requirements for Climate Zone 1 (A,B)

Building Elements		Assembly Max. U-Value (Btu/h-ft^2-°F)	Assembly Max. SHGC
Roof	Insulation entirely above deck	U-0.063	--
Walls, Above-Grade	Steel-Frame	U-0.124	--
Floor	Steel-Joist	U-0.350	--
Windows	Metal Framing (curtain wall)	U-1.20	SHGC-0.25

The wall and floor envelope types for the construction templates of the designed system were adjusted to Mass building elements to meet the Owner's requirements. Windows with a U-value of 0.48 and SHGC of 0.52 were selected to contribute to the 15% energy reduction of the building.

4.2 Noise Control

4.2.1 Noise Criteria

In order to ensure that noise criteria (NC) requirements were being met, the most sensitive duct paths were analyzed in Trane Acoustics Program (TAP) for noise analysis. "Sensitive" in this case means a duct path that will likely prevent a room from meeting the required NC rating. If a duct path was unable to meet the requirements, various alternatives were analyzed, including the addition of acoustic lining or attenuators, and in some cases, rerouting the ductwork. Figures 6 and 7 show an example analysis report for an unsatisfactory duct path to one of the first floor conference rooms and for one analyzed alternative, respectively. The original supply duct path produced a background noise rating of NC 65 which is much too high for a meeting space. The alternative option added 1-inch acoustic lining to the ductwork directly over the room; this lowered the rating to NC 29. The portions of the path where changes were made are highlighted in each figure.

Path2
Floor 1 Path 1

Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Custom Element	96	91	88	92	85	82	82	
Straight Duct(RU1)	-1	0	0	0	0	0	0	
Junction (T,atten.)	-1	-1	0	0	0	0	0	
SubSum	**94**	**90**	**88**	**92**	**85**	**82**	**82**	
Junction (T,regen.)	57	54	48	43	37	30	23	Regenerated sound from junction.
Straight Duct(RU1)	0	0	0	0	0	0	0	
SubSum	**94**	**90**	**88**	**92**	**85**	**82**	**82**	
Junction (T,atten.)	-2	-2	-2	-2	-2	-2	-2	
SubSum	**92**	**88**	**86**	**90**	**83**	**80**	**80**	
Junction (T,regen.)	62	56	49	41	32	22	11	Regenerated sound from junction.
SubSum	**92**	**88**	**86**	**90**	**83**	**80**	**80**	
Straight Duct(RU1)	-1	-1	-1	0	0	0	0	
SubSum	**91**	**87**	**85**	**90**	**83**	**80**	**80**	
Junction (T,atten.)	0	0	0	0	0	0	0	
Junction (T,regen.)	39	36	32	27	23	16	10	Regenerated sound from junction.
SubSum	**91**	**87**	**85**	**90**	**83**	**80**	**80**	
Straight Duct(RU1)	-1	-1	-1	0	0	0	0	
Junction (T,atten.)	-12	-12	-12	-12	-12	-12	-12	
SubSum	**78**	**74**	**72**	**78**	**71**	**68**	**68**	
Junction (T,regen.)	37	33	30	25	20	13	7	Regenerated sound from junction.
SubSum	**78**	**74**	**72**	**78**	**71**	**68**	**68**	
Straight Duct(RU1)	-3	-2	-1	-1	-1	-1	-1	
Junction (X,atten.)	-6	-6	-6	-6	-6	-5	-5	
SubSum	**69**	**66**	**65**	**71**	**64**	**62**	**62**	
Junction (X,regen.)	1	0	0	0	0	0	0	Regenerated sound from junction.
SubSum	**69**	**66**	**65**	**71**	**64**	**62**	**62**	
Straight Duct(RU1)	-2	-1	0	0	0	0	0	
End Reflection	-17	-12	-7	-3	-1	0	0	
Sum	**50**	**53**	**58**	**68**	**63**	**62**	**62**	

NC 65 **RC 64(H)** **70 dBA**

Figure 6. Original Transmission to Conference Room

Path3
Floor 1 Path 1 Alternative 1

Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Custom Element	96	91	88	92	85	82	82	
Straight Duct(RU1)	-1	0	0	0	0	0	0	
Junction (T,atten.)	-1	-1	0	0	0	0	0	
SubSum	**94**	**90**	**88**	**92**	**85**	**82**	**82**	
Junction (T,regen.)	57	54	48	43	37	30	23	Regenerated sound from junction.
Straight Duct(RU1)	0	0	0	0	0	0	0	
SubSum	**94**	**90**	**88**	**92**	**85**	**82**	**82**	
Junction (T,atten.)	-2	-2	-2	-2	-2	-2	-2	
SubSum	**92**	**88**	**86**	**90**	**83**	**80**	**80**	
Junction (T,regen.)	62	56	49	41	32	22	11	Regenerated sound from junction.
SubSum	**92**	**88**	**86**	**90**	**83**	**80**	**80**	
Straight Duct(RU1)	-1	-1	-1	0	0	0	0	
SubSum	**91**	**87**	**85**	**90**	**83**	**80**	**80**	
Junction (T,atten.)	0	0	0	0	0	0	0	
Junction (T,regen.)	39	36	32	27	23	16	10	Regenerated sound from junction.
SubSum	**91**	**87**	**85**	**90**	**83**	**80**	**80**	
Straight Duct(RU1)	-1	-1	-1	0	0	0	0	
Junction (T,atten.)	-12	-12	-12	-12	-12	-12	-12	
SubSum	**78**	**74**	**72**	**78**	**71**	**68**	**68**	
Junction (T,regen.)	37	33	30	25	20	13	7	Regenerated sound from junction.
SubSum	**78**	**74**	**72**	**78**	**71**	**68**	**68**	
Straight Duct(RL)	-7	-9	-12	-24	-40	-40	-32	
Junction (X,atten.)	-6	-6	-6	-6	-6	-5	-5	
SubSum	**65**	**59**	**54**	**48**	**25**	**23**	**31**	
Junction (X,regen.)	1	0	0	0	0	0	0	Regenerated sound from junction.
SubSum	**65**	**59**	**54**	**48**	**25**	**23**	**31**	
Straight Duct(RL)	-4	-5	-7	-13	-33	-31	-18	
End Reflection	-17	-12	-7	-3	-1	0	0	
Sum	**44**	**42**	**40**	**32**	**5**	**5**	**13**	

NC 29 **RC 14(RH)** **34 dBA**

Figure 7. Alternative with Added Acoustic Lining

4.2.2 Sound Transmission

Zone-to-zone sound transmission is a particular issue for the connecting wall from the workshop to the rest of the building. This wall is adjacent to a hallway which in turn is adjacent to several acoustically-sensitive office spaces and classrooms. The ANSI/ASA S12.60-2010 standard for acoustical design criteria in school buildings recommends that corridors that are directly adjacent to classrooms do not exceed a background noise level of 45dBA. In order to ensure that the sound transmission from the workshop would not interfere with meeting the standard, a transmission loss analysis was performed in TAP with the intention of finding the minimum Sound Transmission Class (STC) rating the connecting wall needed in order to meet 45dBA in the hallway.

Generic sound data were collected for various types of workshop equipment; logarithmic addition was used to combine the octave-band data to make a single-source estimate of how much sound the workshop would generate per octave band. The estimated sound data and the dimensional data for the source room (workshop) and receiving room (hallway) were entered in TAP, and transmission loss values for different STC ratings were experimented with until the background noise met 44dBA. Figure 8 shows the final analysis report. The results show that an STC 50 rated CMU wall assembly will keep the transmitted noise at 44dBA which is acceptable.

	Element	63Hz	125Hz	250Hz	500Hz	1KHz	2KHz	4KHz	Comments
Path1									
	Custom Element	91	80	77	76	84	86	82	
	Wall or Floor	9	9	9	6	6	6	5	
	SubSum	**100**	**89**	**86**	**82**	**90**	**92**	**87**	
	Trans Loss Val	-31	-33	-38	-40	-40	-40	-40	STC 50
	Rec Rm Corr	-11	-11	-10	-11	-12	-12	-12	
	Sum	58	45	38	31	38	40	35	
	NC 41	RC 36(H)	44 dBA						

Figure 8. Sound Transmission from Workshop

4.3 Photovoltaic System

A photovoltaic (PV) array is composed of one or more PV modules that convert solar energy into direct current electricity. The direct current generated by the array is typically passed through a DC/AC inverter so alternating current electricity is supplied to the building. The whole PV system is connected to the building's main electrical control panel via balance of system equipment (BOS) which includes disconnecting means on either side of the inverter, ground-fault protection, and overcurrent protection. When several modules are a part of the PV array, a circuit combiner or junction box is included in the BOS to consolidate the wiring from the individual modules into a single bus. Additionally, one or two meters are wired into the system to measure the PV power supply. Figure 9 shows a diagram of a typical photovoltaic system.

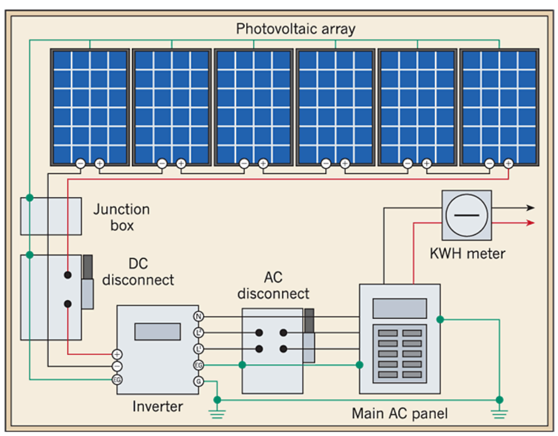


Figure 9. Photovoltaic System Diagram (Source: ecmweb.com)

According to the Florida Solar Energy Center, a typical commercial PV cell generates 0.5-0.6 volts of DC electricity. For a standard 25-in^2 cell, this translates to 0.8-2 watts of peak power depending on the sunlight intensity, meaning an industry standard 36-cell PV module produces approximately 28-72 watts of power.

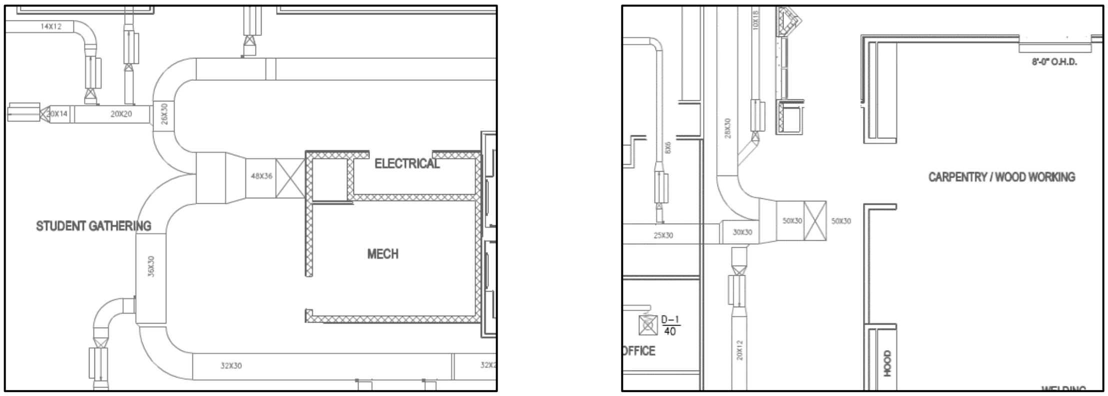
The PV array selected for the building is based on the 5% annual coverage requirement from the Owner. The total electricity consumption of the building based on the Trace annual analysis is 544,000 kWh, so the PV system must be able to cover 27,200 kWh per year. Using Sanyo mono-Si type PV modules with 215 W output, a 38-module PV array will be able to supply 5% of the building electricity usage. Full sizing calculations are appended.

Section 5: System Description

5.1 Air Handling Units and Ductwork

Three Temtrol® Nortek™ Air Solutions air handling units were selected to serve the building. Each air handling unit was sized differently according to the total load of the space it serves. AHU-1 serves Floors 2 and 3 and is sized for 38,200 cfm of actual airflow; AHU-2 serves Floor 1 except for the workshop area and is sized for 17,300 cfm; and AHU-3 serves the workshop area and is sized for 1,700 cfm.

The air handling unit placement was based on the most efficient means of routing the duct. AHU-1 was placed on the roof of the main building, with the duct penetrating the roof next to the mechanical and electrical rooms, as seen in Figure 10a. The duct continues to the second floor. AHU-2 and AHU-3 are both located on the roof of the workshop. The duct for AHU-2 punches through the first floor over the hallway space next to the workshop, as seen in Figure 10b. AHU-3 services the workshop directly with duct branching out from the center of the space.



(a) Third Floor **(b) First Floor**

Figure 10. AHU Duct Routing

All ductwork in the building is for the purpose of providing supply air to each zone. For the return air, a plenum system was chosen based on the benefits described by Taylor (2015), including reduced costs, reduced fan energy, better VAV system balance, and reduced noise transfer between rooms.

5.2 VAV Boxes and Fan Coil Units

Krueger® LMHS Single Duct Terminal Units with Electric Reheat were used; the different VAV box sizes and corresponding zones are shown in the appended equipment schedule. Each VAV box is sized according to its maximum required airflow of the zone with two exceptions: the library which has two VAV boxes instead of one, and the stairwells which use fan coil units instead.

Most zones consist of a single room, allowing each space to be conditioned as needed based on load and occupancy variance. For example, the load for a classroom can be considerably different from an adjacent classroom based on the daily class schedule and number of students in attendance. Individual room control increases system efficiency by reducing the occurrence of over-conditioning. The library is separated into two zones because of the room size and varied cooling needs. The south wall of the space contributed to roughly half of the peak cooling load, so it has its own dedicated VAV box.

Fan coil units (FCUs) are incorporated in the system to cool stairways and select entries. Krueger® HFEC type FCUs with 2-pipe cooling and electric reheat are used. The full schedule listing this equipment is appended to the report. The inclusion of a fan coil unit on the main entry can cause fogging on the glass during periods of high ambient humidity and continuous cold air delivered on adjacent glass. For this reason, a fan coil unit is not used on this particular entry; instead, fans are used for air circulation to mix entry and exfiltration when the doors are being used.

The use of electric reheat rather than heating coils in the VAV boxes and FUCs was decided upon for economic optimization of the VAV system. Electric reheat has lower first cost than the piping, coils, pump and boiler needed for a hydronic reheat system. Additionally, even though the energy cost of heating using electric reheat is higher, the amount of heating needed for this building is minimal so the increased first cost of a hydronic system doesn't pay off.

5.3 Chiller and Thermal Ice Storage

The selected primary system for the building is a chiller combined with thermal ice storage. The building location and proposed utility costs were the primary factors in implementation of the thermal ice storage system. Doha's hot and dry climate demands large amounts of cooling for buildings, especially during the peak hours of the day. This, in turn, largely increases the peak demand charge of the building operational cost. The thermal ice storage shifts some of the energy consumption from the peak hours during the day to the off-peak hours during the night. This is especially significant for Doha because the diurnal temperature swings are very significant and the cooling season is longer than average. By utilizing energy during off-peak hours, the peak demand charge is greatly decreased.

The selected system includes a Daikin WGZ100DW Packaged Water-Cooled Scroll Chiller with 100 ton nominal capacity with two TSU-476M Modular BAC ICE CHILLER® Thermal Storage Units that each has a 476 ton-hour capacity. The system also implements a Taco G14416L U-tube heat exchanger that separates the glycol solution of the primary system from the water of the secondary system. The chiller with the glycol solution produces 22°F leaving temperature during ice build mode to de-rate the chiller to 65 ton capacity. The chiller will produce the 45°F leaving glycol solution during ice melt mode to send to the ice storage tanks to further cool the solution. After the solution leaves the ice tanks, it will run through the heat exchanger to cool the returning water from the system to a 40°F chilled water supply.

The thermal ice storage tanks apply an internal-melt ice-on-coil system (Figure 11). It is a form of static ice storage where a cylindrical coil is immersed in a tank full of water, where it acts as the phase changing medium and never leaves the tank. During ice build, a glycol/water solution from the cooling equipment is circulated through the coils at a temperature that freezes the water on directly on the coils and gradually propagates outward to create ice. During ice melt, the cooling equipment will then send back warmer solution through the coils of the ice tanks to exchange heat energy with the ice. The solution is returned to the system at its working, cool temperature while the ice melts from the inside first.

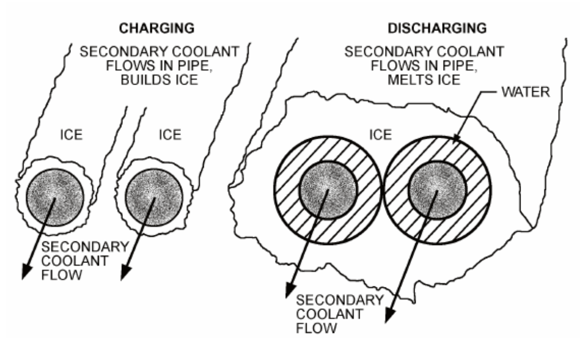


Figure 11. Internal-Melt Ice-On-Coil Diagram (source: ASHRAE Handbook: HVAC System and Equipment 2012)

The strategy of the thermal ice storage will change the selection and sizing of the equipment with different three strategies available to use shown in Figure 12. Avoiding electricity usage during expensive periods and reducing required chiller size were two main criteria for strategy selection. A partial load-leveling strategy incorporates both benefits by shaving high peak loads with a lower capacity chiller during the hot summer months. The partial load-leveling strategy in Figure 12b shows a smaller chiller sized to run at full capacity for 24 hours to charge the ice storage during off-peak and to meet cooling load during on-peak. The ice storage provides the remaining capacity when cooling load exceeds the chiller capacity. The partial load-leveling strategy was chosen over a full load strategy for this system due to the long period of on-peak electricity rates and its consequences of not properly utilizing the use of a thermal ice storage system.

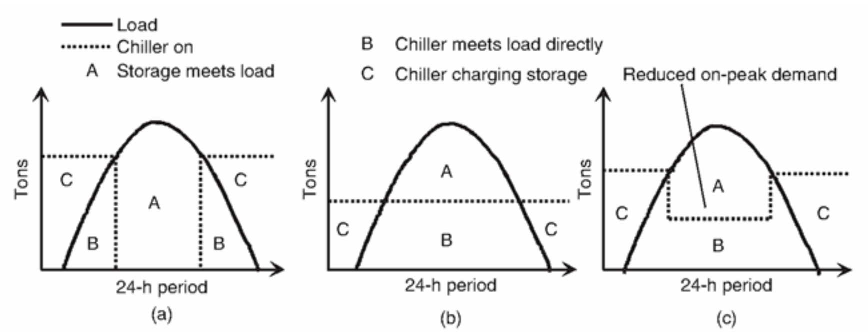


Figure 12. Thermal Ice Storage System Strategies (source: Dincer 2011)

A benefit of partial load-leveling is the option of utilizing a full load strategy during the non-summer months of the year. If the cooling load falls below the capacity of the chiller for longer periods of time, the control of system can shift from the combination of chiller and ice storage to only ice storage to handle the building load. During this time, the chiller is shut off during the on-peak hours and turned on during the off-peak hours to charge the ice storage (Figure 12a). This application can greatly reduce the yearly energy usage and better utilizes the implemented thermal ice system.

Trane Trace 700 was used to calculate total coil load. The largest 24 hour aggregated load was used in place of the peak building cooling design load to size the chiller and the storage equipment. The calculated design cooling load occurs in August with 1843 ton-hours of capacity that the cooling equipment needs to supply the building. The actual design strategy has the chiller running at full capacity for a full 24 hours, considering two hot days can occur in a row, so ice cannot be assumed to be carried over from a prior 24-hour period. Due to the production of below freezing temperatures from the chiller in order to create ice during the off-peak charging mode, in reality the chiller is de-rated because it is unable to hold the same capacity for all temperatures. The amount of capacity lost depends on how low the temperature falls below the conventional chiller leaving temperature of 45°F. Full calculations are included in the appendix.

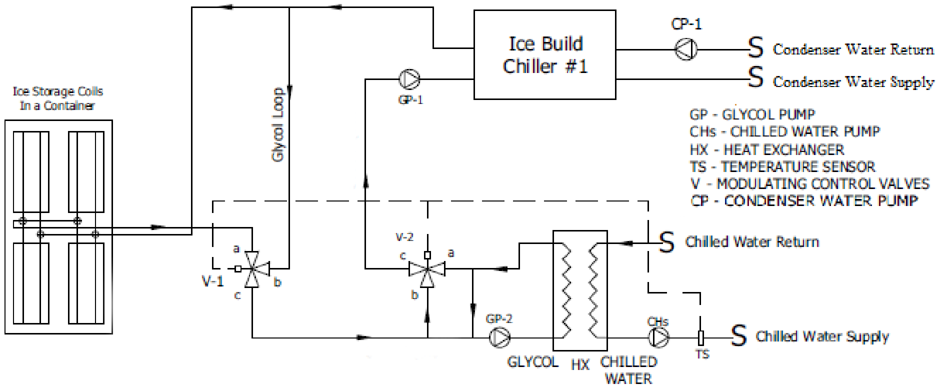


Figure 13. Schematic of Internal-Melt Ice Storage System (adapted from Evapco.com)

5.4 Condenser Water Heat Rejection Fountains

Doha is categorized in climate zone 1B which validated the use of a water-cooled heat rejection system. The humidity levels were low enough to implement a water-cooled system over air-cooled by utilizing the lower wet bulb temperature and lower-priced water rates. Focusing on implementing a creative and innovative design, the heat rejection fountains at the Museum of Islamic Art, as seen in Figure 14, became a source of inspiration. Using fountains for the building in Doha provides an innovative and easily integrated heat rejection system for the cooling dominant building, a visually pleasing architectural showcase for the university, and a means of relief to passing visitors in the hot and dry local climate.

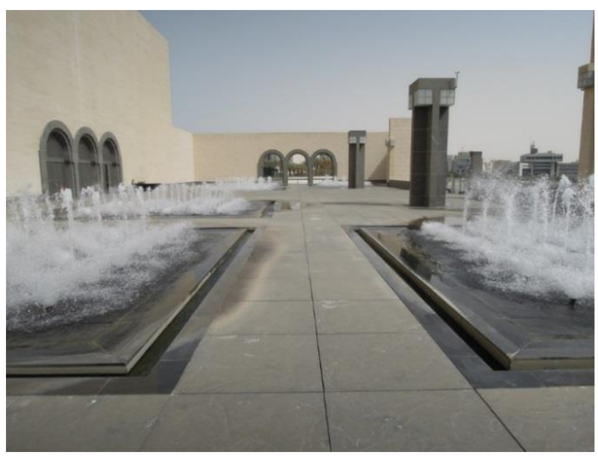


Figure 14. Water Fountains at the Museum of Islamic Art in Doha, Qatar (source: photo taken by industry advisor Joe Hazel)

Rather than using a standard cooling tower, several water fountains serve as the heat rejection system for the building, utilizing evaporative cooling by means of a large, relatively still pond with water jets to provide high velocity spray. Piping to the fountains is buried. Valves and piping control the flow to each fountain in order to ensure the spray elevation of the spray from the jets is uniform.

The design wet bulb temperature that was used is 76°F; this affects the rate of evaporation of the pond and the jets, hence the condenser water return temperature. The added heat from the condenser pump was taken into account for total heat rejection by the system. The total heat rejection is a function of the COP of the chiller, in which the worst case was used for sizing. The final system selection was determined to have 12 total fountains. Calculations can be found in the appendix.

Filtration is necessary to ensure the cleanliness of the fountain water returning to the chiller. Many different kinds of environmental debris will enter fountain ponds which can cause serious damage to the chiller, build-up in the return pipes, and potential health issues for occupants if left unfiltered. Sumps below and to the side of each of the fountains will house the filtration system, which will be easily accessible for maintenance. Another health risk is the development of legionella in the warm condenser water. Proper water treatment is necessary in order to prevent the serious and potentially fatal effects of these bacteria.

5.5 Special Instruction Area

The Special Instruction wing houses a space where students conduct welding, woodworking, and painting. Due to airborne contaminants created by such activities, proper exhaust must be installed to allow for the contaminants to exit the area without entering the neighboring spaces. The most effective means of achieving this is by providing an exhaust rate this is greater than the supply rate, creating negative room pressure. Return air from the main sector of the building was mixed into the supply air for the Special Instruction area to help increase efficiency due to similar temperatures. General ventilation is provided to the space by AHU-3, combined with three additional systems to provide proper exhaust rates: welding hood, individual flexible portable welding exhaust, and a dust collector.

A Donaldson Torit WB-2000 Weld Bench Fume Collector with a Greenheck SWD13 Exhaust Fan powered by a ½ HP Vari-Green Motor was selected to serve the permanent 4 foot wide bench. Equation 13 was used to confirm that the selected fume collector would meet the exhaust demand. The resulting rate of exhaust was 1400cfm which is sufficient.

$$Rate\ of\ exhaust = \left(350\,\frac{cfm}{ft}\right) * (length) \qquad \text{(Equation 13)}$$

Figure 15. Torit WB-2000 Welding Bench (source: Donaldson Company, Inc.)

Figure 16. Vari-Green Motor (source: Greenheck)

Six Sentry Air SS-400-PFS High Flow Portable Fume Extractors were selected to provide exhaust collection to individual stations in the space. They have the ability to be operated on variable volume control with a maximum flow rate of 700 CFM. The portable fume extractors were chosen to abide by owner's requirements for VAV control. The chosen model has multiple filter media options available to use as a safe option for collecting several types of contaminants.

Figure 17. SS-400-PFS High Flow Portable Fume Extractor (source: Sentry Air Systems, Inc.)

The woodworking area has table saws, rip saws, belt sander, planer, lathes, and floor sweeps. A Donaldson Torit Model 450 Unimaster Dust Collector was selected based on the required exhaust airflow as well as the air-to-cloth ratio. The air-to-cloth ratio is a dust collector rating that takes into account CFM with reference to the cloth filter. With a required ratio of 6.0-8.0, the dust collector selected provided the proper exhaust airflow of 3,100cfm and had a filter area of 444 ft^2. This can be confirmed using Equation 14, which gives a result of 6.98 fpm for the ratio.

$$\frac{Air}{cloth} Ratio = \frac{CFM}{filter\ area} \quad \text{(Equation 14)}$$

Figure 18. Torit Model 450 Unimaster Dust Collector (source: Donaldson Company, Inc.)

Section 6: System Analysis

6.1 Energy Consumption Summary

The total building energy consumption was analyzed in Trane Trace 700. Table 7 provides a summary and comparison of the energy consumption for the baseline model and the designed VAV system. Based on ASHRAE Standard 90.1 Appendix G (Equation 12), the designed system uses 15% less energy than the baseline model.

Table 7. Total Building Energy Consumption

	Baseline (kBtu/year)	Designed System (kBtu/year)
Primary Heating	28,404	101,219
Primary Cooling	1,310,174	1,014,560
Lighting	480,789	375,092
Receptacle	343,072	343,072
Total	**2,162,439**	**1,833,943**

6.2 Life Cycle Cost Analysis

A spreadsheet was created to perform a life cycle cost analysis on the designed system in order to make sure it falls within the Owner's $200/ft^2 budget. Table 8 shows the prices that were considered for the equipment and installation. Costs are based on assumptions recommended by the team industry mentors.

Table 8. System Budget

Equipment Type	Equipment Cost	Labor Cost	Total Cost
Air Handling Units	$171,600	$114,400	**$286,000**
VAV Units	$84,800	$42,400	**$127,200**
Fan Coil Units	$14,000	$7,000	**$21,000**
HVAC Ductwork	$186,800	$186,800	**$373,600**
Hydronic Piping	$93,400	$140,100	**$233,500**
Water Cooled Chiller	$45,000	$15,000	**$60,000**
Thermal Storage Tanks	$120,000	$90,000	**$210,000**
Heat Rejection Fountains	$60,000	$48,000	**$108,000**
Temperature Controls	$70,050	$93,400	**$163,450**

Including a 10% design contingency, 5% construction contingency, 7% sales tax, 5% overhead and profit, and 100% city multiplier for material and labor, the total initial cost for the building system is $1.9 million. This value was used to check budget compliance. Based on a building area of 46,700 ft^2, the price-per-area of the HVAC system is $41.24/ft^2, which is approximately 20% of the Owner's budget.

The 50-year life cycle cost analysis was based on the $1.9 million initial cost, the provided utility costs and annual percentage increase, 5% PV system savings, chiller replacement at years 20 and 40, PV array replacement at year 25, other various component replacements starting at year 20, 3% inflation, 4% return on investment, and $14,000/year maintenance costs. The full analysis resulted in $4.8 million operation and maintenance costs, combined with the initial cost for a **total life cycle cost of $6.7 million**. Figure 19 provides a graphical representation of the life cycle cost analysis.

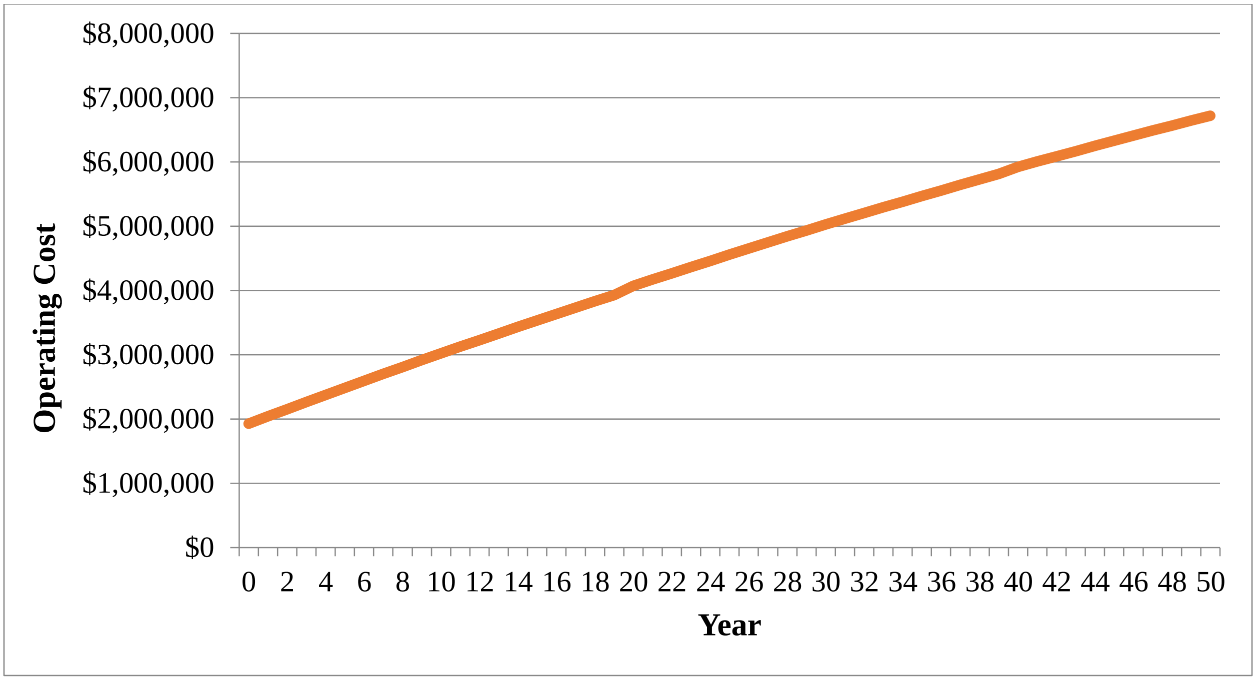


Figure 19. Life Cycle Cost Analysis

Section 7: Conclusion

The calculations conducted in this project show that the building experiences a peak **cooling load** of 157 tons. Calculations have been conducted to design a thermal ice storage system, which allows the chiller size to be reduced to 100 tons while still meeting the cooling load on the design day. Although the ice storage system adds initial cost, it is shown to reduce life cycle cost because of the reduction in initial cost for the chiller and reduction of energy used during the peak demand hours throughout the cooling season.

Calculations have been conducted to ensure compliance with the requirements of ASHRAE **Standards** 55 (thermal comfort), 62.1 (ventilation), and 90.1 (energy).

In accordance with the owner's project requirements, a **PV solar** collector system has been designed to produce 5% of the building's annual electrical consumption.

A creative **condenser water heat rejection** approach has been designed. This approach makes use of several decorative fountains on the grounds of the school to perform heat rejection. This approach is unusual, but it has been implemented in the Museum of Islamic Art in Doha, which shows that the practical difficulties can certainly be overcome. Heat and mass transfer calculations show that a total of 240 m^2 of fountains will be required.

Noise calculations have been conducted to demonstrate that the acoustically sensitive spaces in the building will have noise levels at or below the owner's requirements. These calculations include: a) transmission through the building structure from the woodworking space to the adjacent offices; b) noise transmitted through the ductwork from the air handling equipment to the classrooms; c) self-generated flow noise in the ductwork transmitted into the classrooms. Appropriate measures to prevent noise problems have been specified wherever necessary.

Design documents have been produced to show duct and piping layout, diffuser locations, equipment specifications, equipment selections, where appropriate. In addition, preliminary selections of the equipment for the school's shops has been provided, so that representative noise, airflow and heat load data could be gathered.

The final design shows a **life cycle cost** of $6.7 million.

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Appendix

Table 9. Sample Trace Load Calculations

System / Zone / Room **		Floor Area ft²	People #	Coil Cooling Sensible Btu/h	Coil Cooling Total Btu/h	Space Design Max SA cfm	Air Changes ach/hr	VAV Minimum SA cfm	VAV Minimum %	Main Coil Heating Sensible Btu/h	Heating Fan Max SA cfm	Percent OA Clg	Percent OA Htg
Zone - 047	Zn Peak	680	34.0	32,066	44,389	938			30	-5,139	0	25.2	100.0
Zone - 047	Zn Block	680	34.0	32,066	44,389	938			30	-5,139	0	25.2	100.0
309-Conference Room	Rm Peak	435	12.0	13,892	17,666	440	6.08	132	30	-4,669	0	13.7	65.2
Zone - 048	Zn Peak	435	12.0	13,892	17,666	440			30	-4,669	0	13.7	65.2
Zone - 048	Zn Block	435	12.0	13,892	17,666	440			30	-4,669	0	13.7	65.2
311 - Breakroom	Rm Peak	415	15.0	24,051	27,263	902	13.04	271	30	-8,789	0	3.3	36.9
Zone - 049	Zn Peak	415	15.0	24,051	27,263	902			30	-8,789	0	3.3	36.9
Zone - 049	Zn Block	415	15.0	24,051	27,263	902			30	-8,789	0	3.3	36.9
312 - Student Gathering	Rm Peak	2,070	41.4	50,482	66,281	1,602	4.64	481	30	-8,783	0	16.5	68.9
Zone - 050	Zn Peak	2,070	41.4	50,482	66,281	1,602			30	-8,783	0	16.5	68.9
Zone - 050	Zn Block	2,070	41.4	50,482	66,281	1,602			30	-8,783	0	16.5	68.9
313 - Classroom	Rm Peak	860	43.0	48,284	64,181	1,866	13.02	560	30	-14,146	0	16.0	66.8
Zone - 051	Zn Peak	860	43.0	48,284	64,181	1,866			30	-14,146	0	16.0	66.8
Zone - 051	Zn Block	860	43.0	48,284	64,181	1,866			30	-14,146	0	16.0	66.8
314 - Classroom	Rm Peak	680	34.0	39,306	46,021	1,256	11.08	377	30	-6,883	0	18.8	78.5
Zone - 052	Zn Peak	680	34.0	39,306	46,021	1,256			30	-6,883	0	18.8	78.5
Zone - 052	Zn Block	680	34.0	39,306	46,021	1,256			30	-6,883	0	18.8	78.5
315 - Classroom	Rm Peak	650	32.5	28,653	40,668	1,005	9.27	301	30	-5,507	0	22.5	93.8
Zone - 053	Zn Peak	650	32.5	28,653	40,668	1,005			30	-5,507	0	22.5	93.8
Zone - 053	Zn Block	650	32.5	28,653	40,668	1,005			30	-5,507	0	22.5	93.8
316 - Men Bathroom	Rm Peak	200	0.0	14,756	19,654	300	9.00	300	100	-1,851	0	80.0	100.0
317 - Women Bathroom	Rm Peak	200	0.0	14,756	19,654	300	9.00	300	100	-1,851	0	80.0	100.0
318 - Mechanical	Rm Peak	205	0.0	2,350	2,601	50	1.48	15	30	-489	0	24.4	81.2
319 - Electrical	Rm Peak	45	0.0	516	571	11	1.48	3	30	-107	0	24.4	81.2
320 - Corridor	Rm Peak	1,820	0.0	15,057	18,474	277	0.91	83	30	-3,368	0	27.6	100.0
Zone - 054	Zn Peak	2,470	0.0	47,436	60,954	938			75	-7,666	0	60.9	98.8
Zone - 054	Zn Block	2,470	0.0	47,436	60,954	938			75	-7,666	0	60.9	98.8
System - 001	**Sys Peak**	**28,152**	**893.9**	**1,109,025**	**1,446,152**	**37,296**				**-279,214**	**0**	**21.5**	**71.4**
System - 001	**Sys Block**	**28,152**	**893.9**	**1,023,881**	**1,386,538**	**34,391**				**-241,852**	**0**	**21.5**	**71.4**

Table 10. Example Ice Storage Operation Schedule

	Glycol Loop								Chilled Water Loop	
System Modes	Chiller # 1		Pump	Control Valve		Pump	Glycol/Water HX		Pump	
	Status	Set Point	Glycol GP-1	V-1 Flow	V-2 Flow	Glycol GP-2	Cold Side	Hot Side	Water CHs	Temp Sensor
Ice Build										
7 PM - 9 AM	on	22°F (-5.6°C)	on	a-c	b-c	off	no flow	no flow	off	x
Direct Cooling Chiller Only										
5 PM - 7 PM	on	37.4°F (3.0°C)	on	b-c	a-c	VFD on	37.4°F - 57.4°F (3.0°C - 14.1°C)	49.4°F - 40°F (9.7°C - 4.4°C)	VFD on	40°F (4.4°C)
Chiller and Ice Melt in Series										
9 AM - 7 PM	on	47.4°F (8.6°C)	on	a-b / c Mod	a-c	VFD on	37.4°F - 57.4°F (3.0°C - 14.1°C)	49.4°F / 40°F (9.7°C - 4.4°C)	VFD on	40°F (4.4°C)
Ice Melt Only										
9 AM - 7 PM	off	x	on	a-b / c Mod	a - c	VFD on	37.4°F - 57.4°F (3.0°C - 14.1°C)	49.4°F - 40°F (9.7°C - 4.4°C)	VFD on	40°F (4.4°C)
Ice Build with Cooling										
7 PM - 9 AM	on	20°F (-6.7°C)	on	a-c	a-b / c Mod	VFD on	> 32°F (> 0.0°C) Entering	40°F (4.4°C) Leaving	VFD on	40°F (4.4°C)

Table 11. Chiller and Ice Storage Capacity Calculations

PEAK 24-HOUR LOAD		
August	Ton	Design
1	20.7	60
2	39.7	60
3	39.1	60
4	37.5	60
5	31.7	60
6	34.5	60
7	131	100
8	123.1	100
9	119.3	100
10	113.9	100
11	129.2	100
12	137	100
1	145.8	100
2	153.3	100
3	146.8	100
4	143.4	100
5	137.8	100
6	15.7	100
7	10.5	100
8	9.4	60
9	21.4	60
10	40	60
11	39.3	60
12	38.5	60
Peak 24-hour load	1858.6	ton-hr

CALCULATIONS			
Chiller Design Before De-rate =	77.4417		$D$71/24
Chiller Design After De-rate =	97.3089		$D$71/(10+(0.65*14))
Ice Storage Design After De-rat	885.511		I61*0.65*14
DESIGN			
Chiller Size =	100	tons	
Chiller Reduced =	65	tons	I55*0.65
Capacity of Ice Makin	910	ton-hr	I56*14
Capacity of Chill =	1000	ton-hr	I55*10
Total Ton-Hr Capacity	1910	ton-hr	I57+I58
CONSIDERATIONS			
14 hours available to charge off-peak (7 PM to 9 AM)			
10 hours of on-peak cooling (9 AM to 7PM)			
1.5% De-rate for every 1° F below 45°F			
45 °F - 22 °F = 23°F			
*23 °F * 1.5% = 34.5%*			
100% - 34.5% = 65.5%			

Photovoltaic Calculations
5% annual electrical consumption = 27,200 kWh

Daily PV electrical energy = 27,200 kWh/365 days = 74.5 kWh/day

Average sunlight hours in Doha = approximately 12 hours

Hourly PV electrical energy = 74.5 kWh/day/12 sunlight hours = 6.2 kW

Industry Standard Derate Factor = 0.77

6.2 kW *1000/0.77 = 8063 W

Using Sanyo mono-Si type modules with 215-Watt output:

8063 W/215 W/module = 37.5 PV modules ⇒ **38-module PV array**

Example of Water Spray Fountain Calculations
Assuming $RH_{ambient}$ = 50%, T_{db} = 39℃
Find rate of evaporation per area $W_{jet,pond}$ $(\frac{kg}{s*m^2})$ to calculate total required pond area.

$$W_{jet} = \frac{(P_w - P_a)[0.089 + 0.0782(V_{tot})]}{Y}$$

Where P_w = Saturation vapor pressure at water temperature
P_a = Saturation vapor pressure at ambient air dew point
V_{tot} = Velocity accounting for water jet and wind
Y = Latent heat of evaporation

P_w: Determined via weather data ⇒ P_w = 6.993 kPa
P_a: Determined via weather data ⇒ P_a = 3.264 kPa
Y = Constant ⇒ Y = 2260 kJ/kg

$$V_{tot} = \sqrt{V_{avg}{}^2 + V_{wind}{}^2}$$

V_{wind}: Annual average via American Weather Service ⇒ V_{wind} = 4.10 m/s
V_{avg}: $P_1 + 0.5\rho V_1^2 + \rho g y_1 = P_2 + 0.5\rho V_2^2 + \rho g y_2$

With $Energy_{pot}$ = 0 @ ground
$Energy_{kin}$ = 0 @ peak
$P_1 = P_2 = P_{atm}$
Constant ρ

$$0.5{V_1}^2 = gy_2$$
$$V_1 = \sqrt{2 * gy_2}$$

$$V_1 = \sqrt{2 * \left(9.8\frac{m}{s^2}\right) * 2m} \quad \rightarrow V_1 = 6.26 \text{ m/s} \quad \rightarrow V_2 = 0 \text{ (velocity at peak)}$$

$$V_{avg} = \frac{V_1 + V_2}{2}$$

$$V_{avg} = \frac{6.26+0}{2} \quad \rightarrow V_{avg} = 3.13 \text{ m/s}$$

$$V_{tot} = \sqrt{3.13^2 + 5.17^2} \Rightarrow V_{tot} = 6.04 \text{ m/s}$$

$$W_{jet} = \frac{(6.993-4.10)[0.089+0.0782(6.04)]}{2260} \Rightarrow W_{jet} = 0.000716 \text{ kg/sm}^2$$

$$W_{pond} = \frac{(P_w - P_a)[0.089 + 0.0782(V_{wind})]}{Y}$$

$$W_{pond} = \frac{(6.993-4.10)[0.089+0.0782(5.17)]}{2260} \Rightarrow W_{pond} = 0.000631 \text{ kg/sm}^2$$

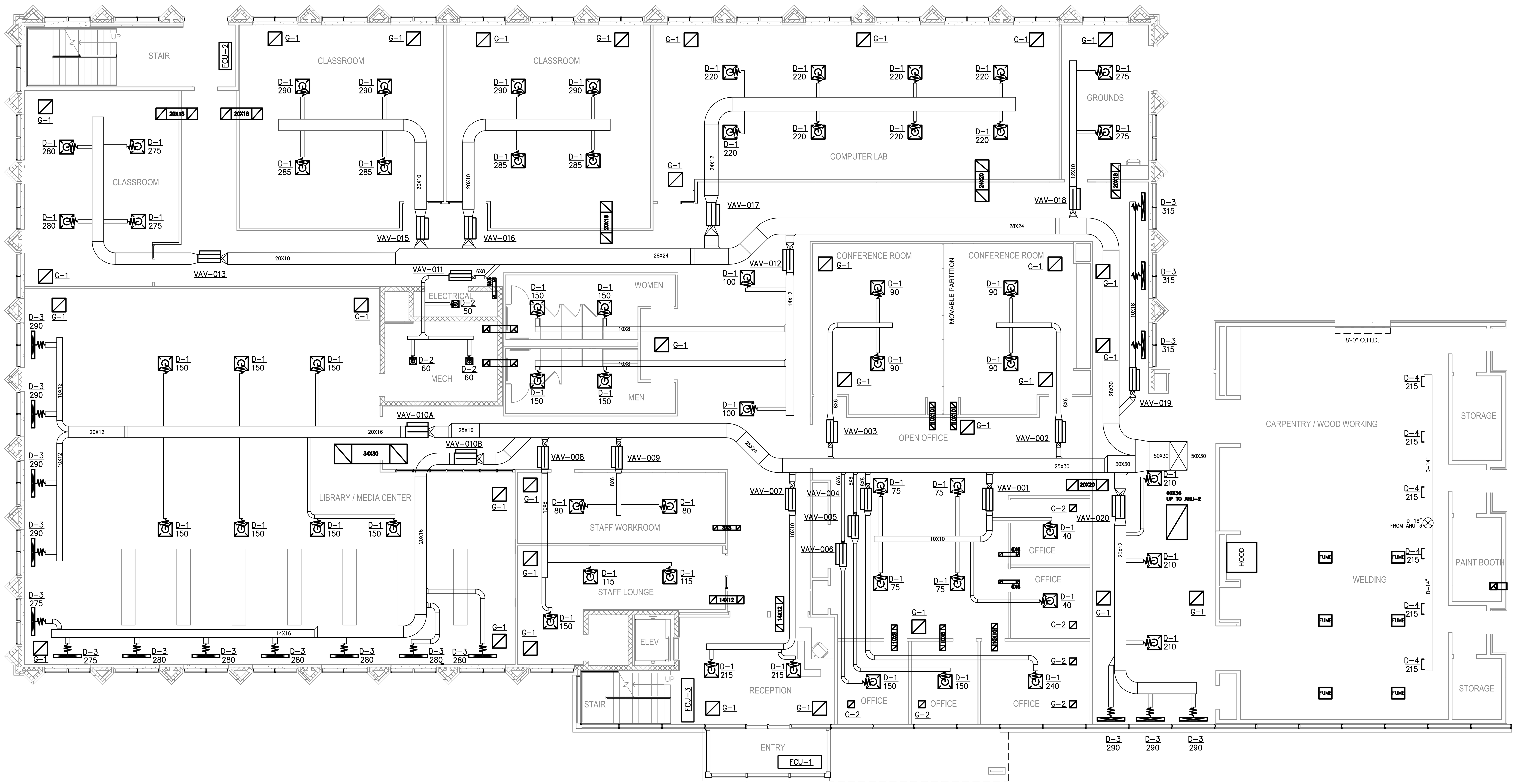
Find surface areas $SA_{jet,pond}$ (m^2) :

$SA_{jet} = 2\pi rh + 2\pi r^2$
$SA_{jet} = 2\pi(0.15)(2) + 2\pi(0.15)^2$
$SA_{jet} = 2.03\, m^2$
$SA_{jet,tot} = (48)(2.03)$
$SA_{jet,tot} = 97.4\, m^2$
$SA_{pond} = b * h$
$SA_{pond} = (10)(2)$
$SA_{pond} = 20\, m^2$
$SA_{pond,tot} = (12)(20)$
$\boldsymbol{SA_{pond,tot} = 240\, m^2}$

Find rate of evaporation R_{evap} (*kg/s*) to calculate makeup water required:

$$R_{evap} = (W_{jet} * SA_{jet,tot})(W_{pond} * SA_{pond,tot})$$

$$R_{evap} = (0.000716 * 97.4) + (0.000631 * 240) \Rightarrow \mathbf{R_{evap} = 0.221\ kg/s}$$



FIRST FLOOR PLAN

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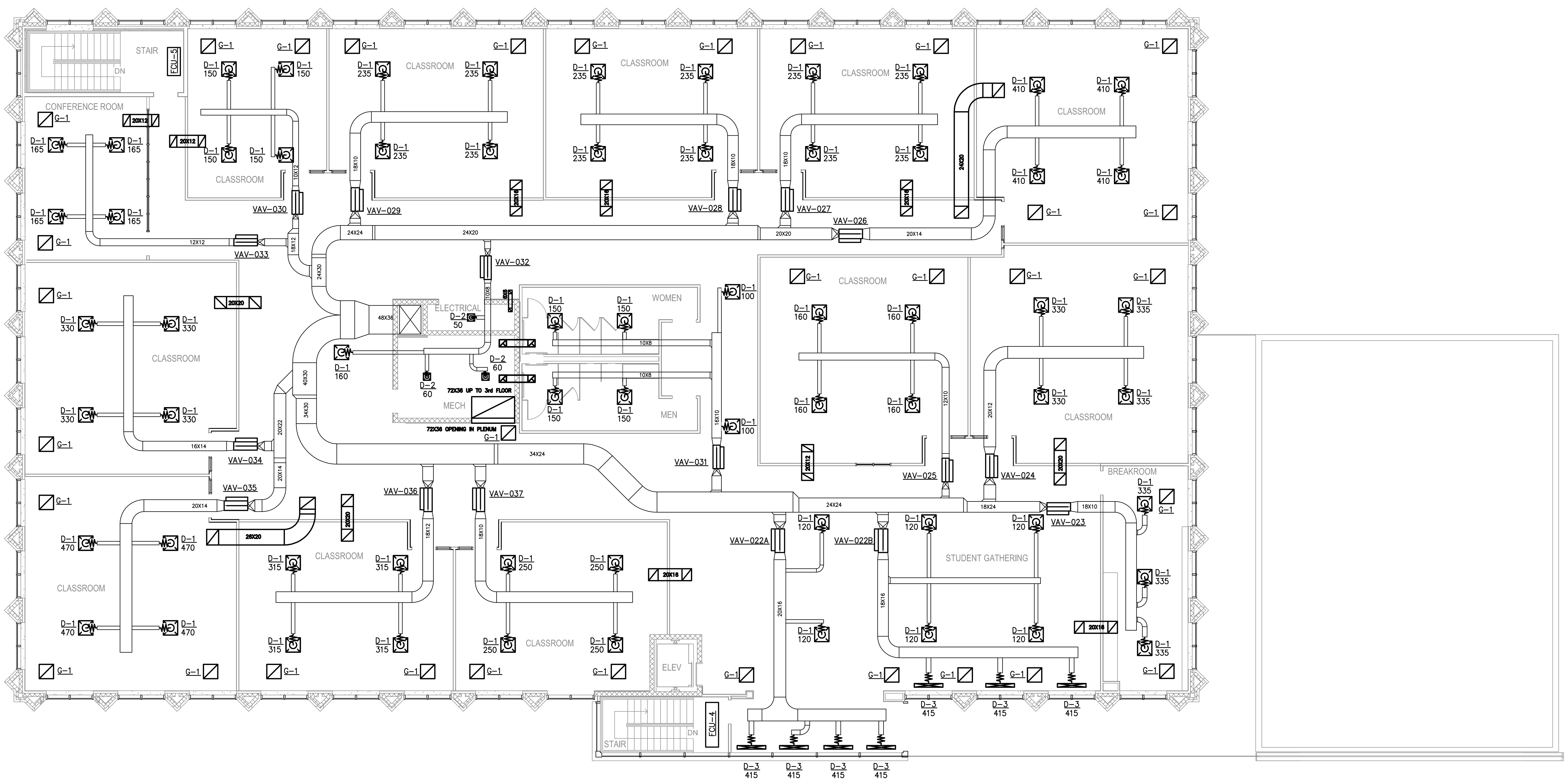

SECOND FLOOR TENANT PLAN

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

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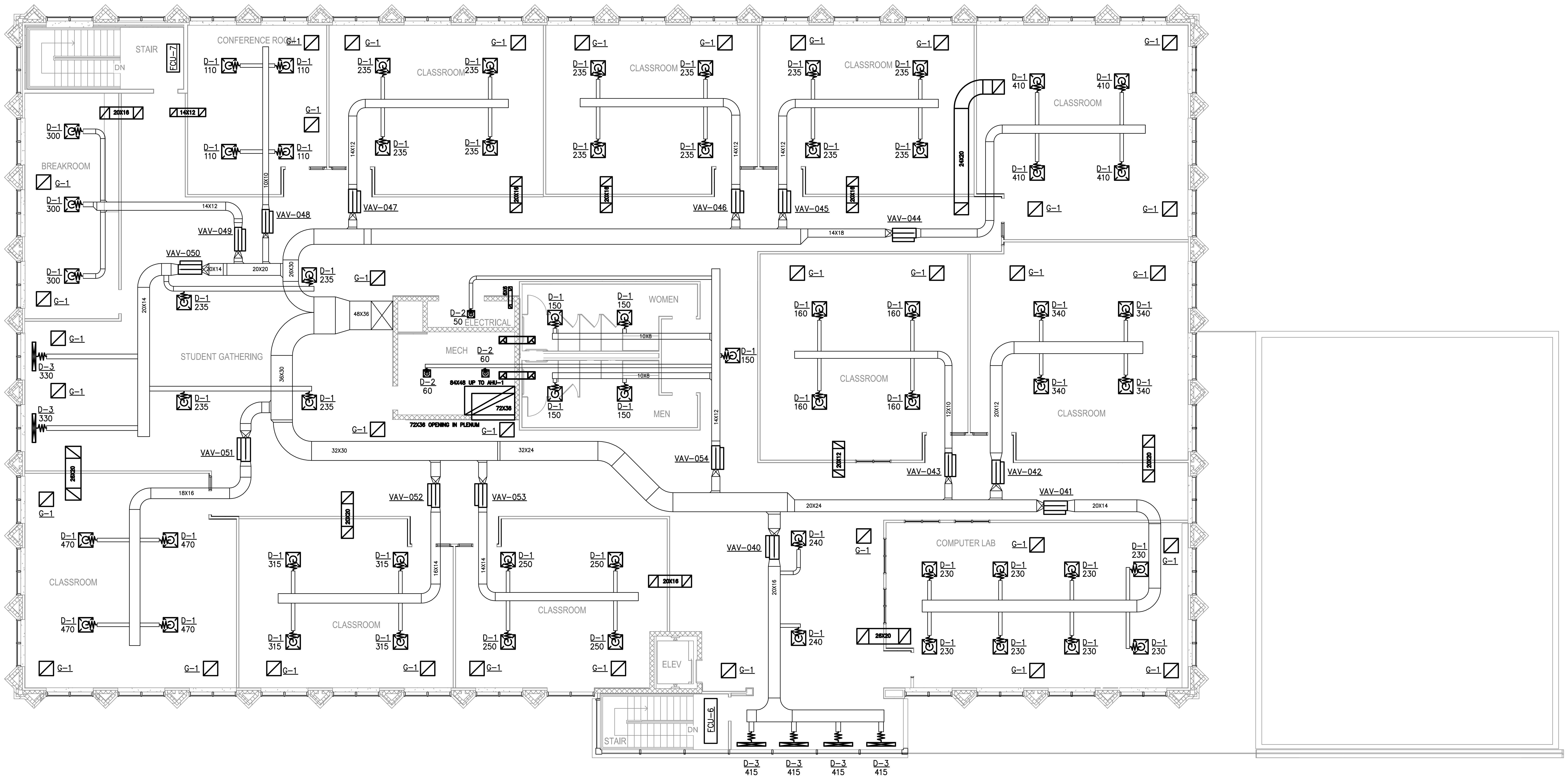

THIRD FLOOR PLAN

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

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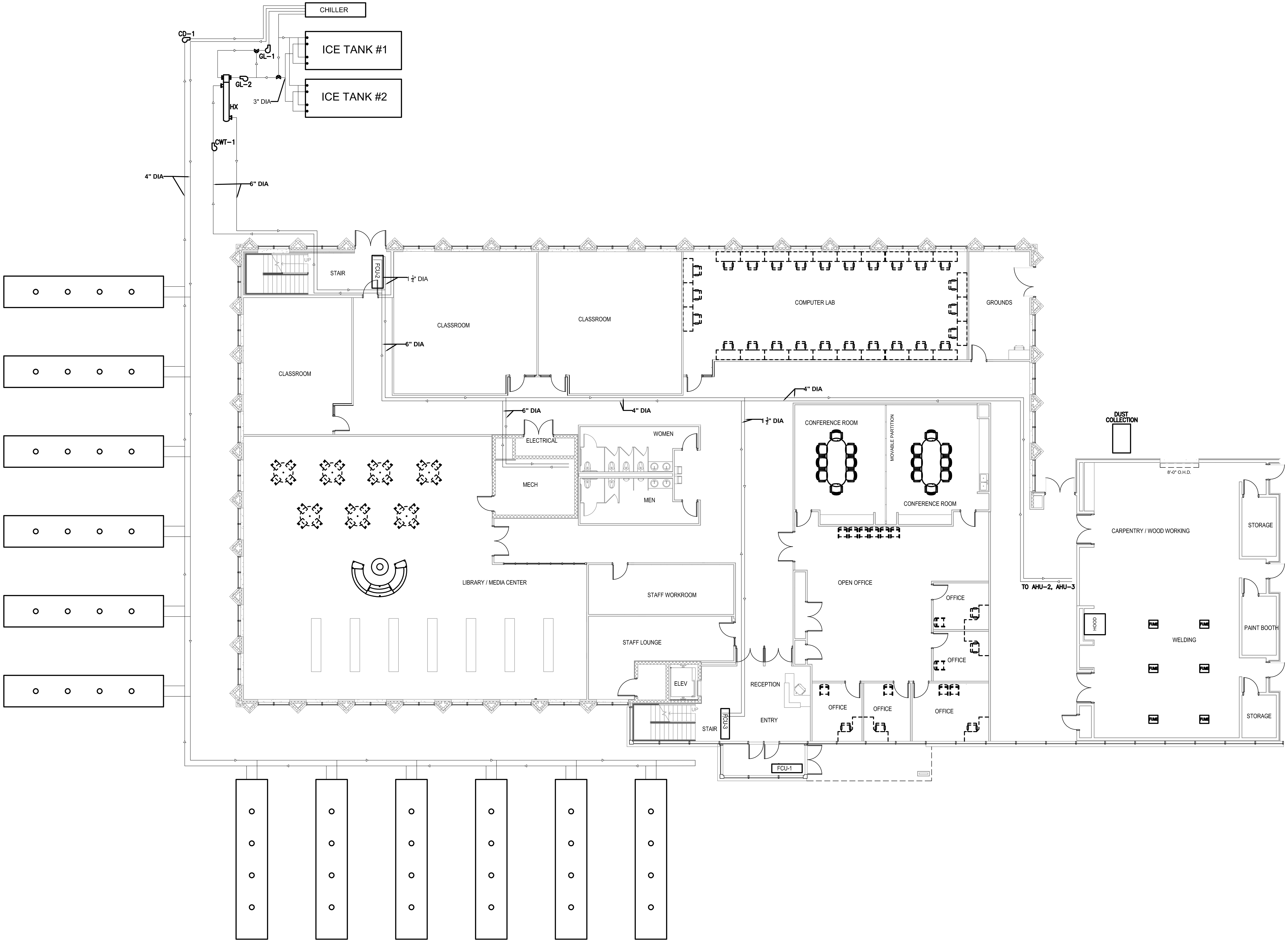

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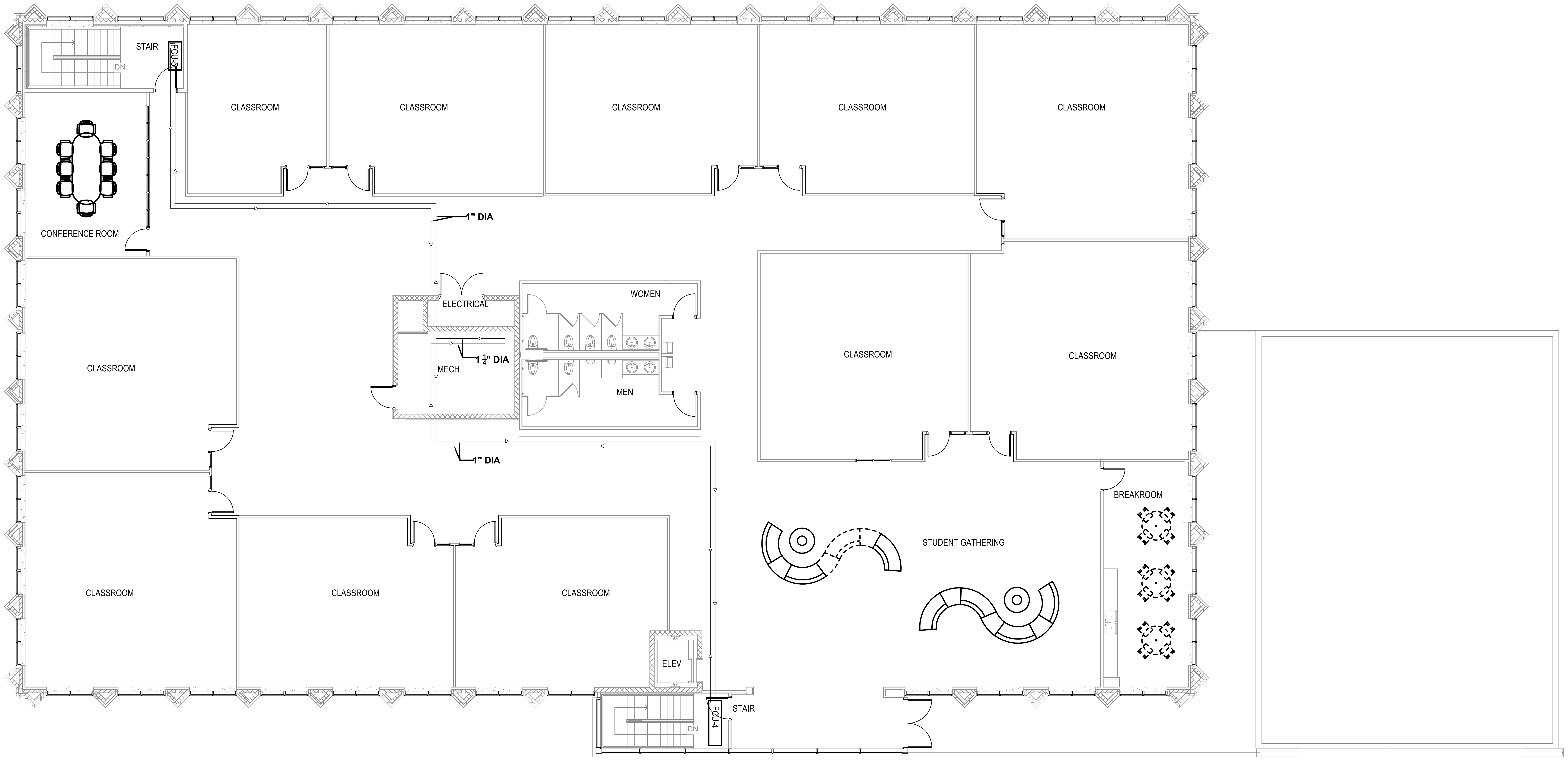

SECOND FLOOR TENANT PLAN
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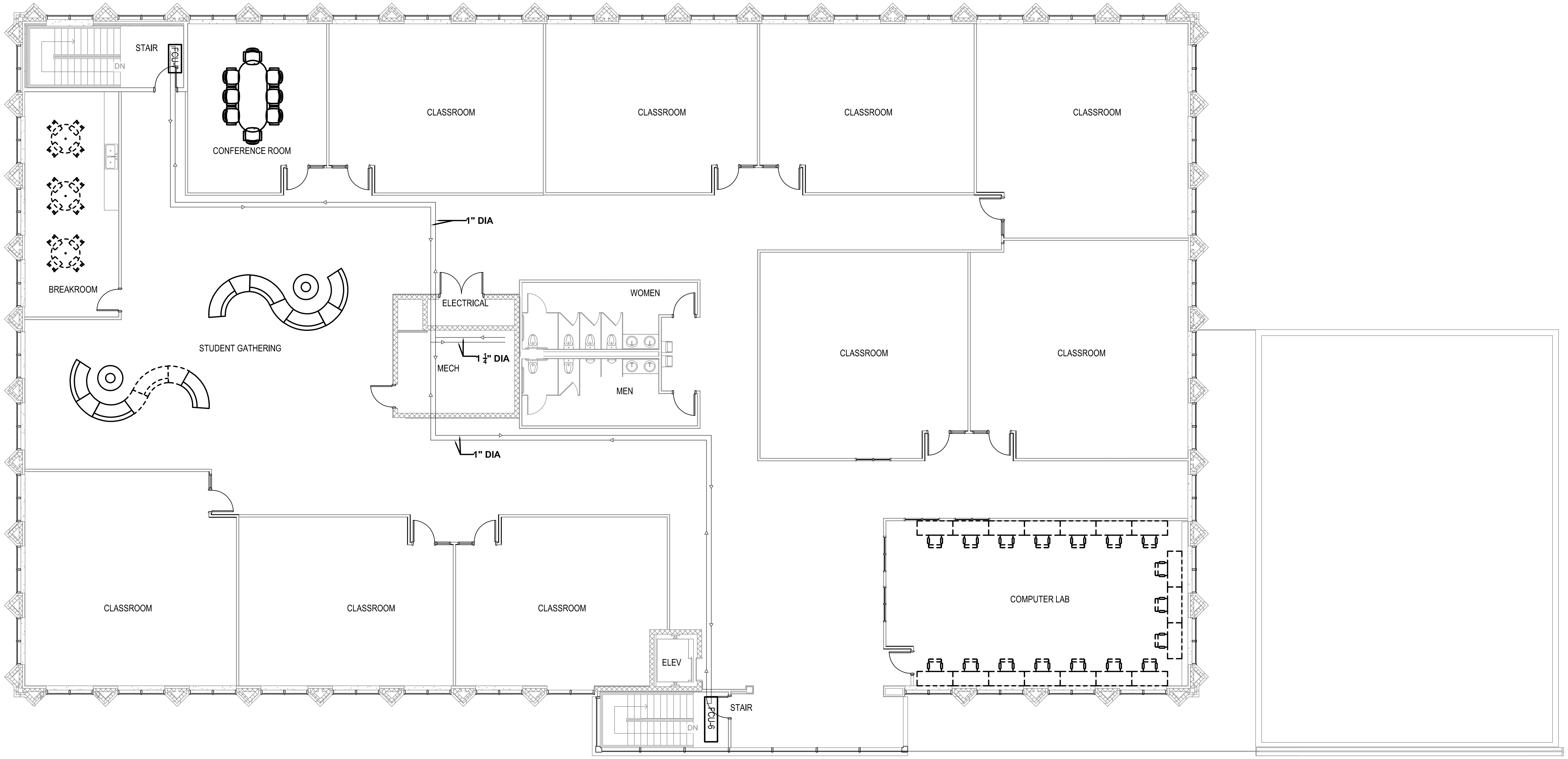

THIRD FLOOR PLAN
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TERMINAL BOX AIRFLOW SUMMARY

MARK	TYPE	SERVING	COOLING MAX AIRFLOW CFM	MIN AIRFLOW CFM	HEATING MAX AIRFLOW CFM
AHU-1					
VAV-022A	H	ENTRY 200	1900	425	569
VAV-022B	H	STUDENT GATHERING 203	1725	425	526
VAV-023	F	BREAKROOM 202	1005	240	302
VAV-024	G	CLASSROOM 205	1330	300	400
VAV-025	D	CLASSROOM 204	640	140	190
VAV-026	H	CLASSROOM 206	1640	425	491
VAV-027	F	CLASSROOM 207	940	240	281
VAV-028	F	CLASSROOM 208	940	240	278
VAV-029	F	CLASSROOM 209	940	240	281
VAV-030	D	CLASSROOM 210	600	140	182
VAV-031	E	MEN RR 217/WOMEN RR 218/CORRIDOR 222	800	190	658
VAV-032	B	MECH 219/ELECTRICAL 220/CORRICOR 221	330	85	98
VAV-033	D	CONFERENCE ROOM 212	660	140	198
VAV-034	G	CLASSROOM 213	1320	300	394
VAV-035	H	CLASSROOM 214	1880	425	560
VAV-036	G	CLASSROOM 215	1260	300	377
VAV-037	F	CLASSROOM 216	1000	240	301
VAV-040	H	ENTRY 300/CORRIDOR 321	2140	425	643
VAV-041	H	COMPUTER LAB 302	1840	425	547
VAV-042	G	CLASSROOM 303	1360	300	408
VAV-043	D	CLASSROOM 304	640	140	190
VAV-044	H	CLASSROOM 305	1640	425	491
VAV-045	F	CLASSROOM 306	940	240	281
VAV-046	F	CLASSROOM 307	940	240	278
VAV-047	F	CLASSROOM 308	940	240	281
VAV-048	C	CONFERENCE ROOM 309	440	110	132
VAV-049	E	BREAKROOM 311	900	190	271
VAV-050	H	STUDENT GATHERING 312	1600	425	481
VAV-051	H	CLASSROOM 313	1880	425	560
VAV-052	G	CLASSROOM 314	1260	300	377
VAV-053	F	CLASSROOM 315	1000	240	301
VAV-054	F	MEN RR 316/WOM RR 317/MECH 318/ELEC 319/CORR 320	920	240	701
TOTAL			37350		
AHU-2					
VAV-001	C	OPEN OFFICE 100/OFFICE 101/OFFICE 102/STORAGE 134	380	110	114
VAV-002	A	CONFERENCE ROOM 111	180	55	56
VAV-003	A	CONFERENCE ROOM 112	180	55	55
VAV-004	B	OFFICE 103	240	85	85
VAV-005	A	OFFICE 104	150	55	55
VAV-006	A	OFFICE 105	150	55	55
VAV-007	C	RECEPTION 106	430	110	129
VAV-008	C	STAFF LOUNGE 108/STORAGE 110	380	110	114
VAV-009	A	STAFF WORKROOM 109	160	55	55
VAV-010A	I	LIBRARY 113	2210	580	660
VAV-010B	I	LIBRARY 113	2230	580	660
VAV-011	A	MECH 114/ELECTRICAL 115	170	55	55
VAV-012	E	MEN AND WOMEN RR 116/CORRIDOR 132	800	190	661
VAV-013	F	CLASSROOM 117	1110	240	333
VAV-015	F	CLASSROOM 120	1150	240	345
VAV-016	F	CLASSROOM 121	1150	240	345
VAV-017	H	COMPUTER LAB 122	1760	425	528
VAV-018	D	GROUNDS 123	550	140	166
VAV-019	E	CORRIDOR 135/CORRIDOR 133	945	190	283
VAV-020	H	STORAGE 128/CORRIDOR 136	1500	425	448
TOTAL			15825		

TERMINAL BOX SCHEDULE

TYPE	BASIS OF DESIGN Manucturer "Model"	RUNOUT SIZE DIA in.	INLET SIZE DIA in.	DESIGN MAX AIRFLOW cfm	MIN AIRFLOW cfm	MAX AIR SPD in. wg	MAX NC DIS/RAD AT 0.75" DELTA-P	ELECTRIC HEAT 277 V 1 PHASE MIN kW	MAX kW	480 V 3 PHASE MIN kW	MAX kW	MAX. DIMENSIONS L x W x H in.
A	KRUEGER "LMHS"	4	4	230	55	0.1	32/24	1.0	3.0	2.5	3.0	39.5 x 12 x 8
B	KRUEGER "LMHS"	5	5	360	85	0.1	31/21	1.0	5.0	2.5	5.0	39.5 x 12 x 8
C	KRUEGER "LMHS"	6	6	515	110	0.1	27/25	1.0	7.5	2.5	7.5	39.5 x 12 x 8
D	KRUEGER "LMHS"	7	7	700	140	0.1	24/23	1.0	9.5	2.5	9.5	39.5 x 12 x 10
E	KRUEGER "LMHS"	8	8	920	190	0.1	26/24	1.0	13.0	2.5	13.0	39.5 x 12 x 10
F	KRUEGER "LMHS"	9	9	1160	240	0.1	23/24	1.0	13.0	2.5	16.0	39.5 x 14 x 12.5
G	KRUEGER "LMHS"	10	10	1430	300	0.1	23/26	1.0	13.0	2.5	21.0	39.5 x 14 x 12.5
H	KRUEGER "LMHS"	12	12	2060	425	0.1	26/26	1.0	13.0	2.5	30.0	39.5 x 16 x 15
I	KRUEGER "LMHS"	14	14	2800	580	0.1	28/23	1.0	13.0	3.0	36.0	39.5 x 20 x 17.5

AIR OUTLET AND INLET SCHEDULE

MARK	MANUF. & MODEL	TYPE	MODULE SIZE, IN	NECK SIZE (W X H OR DIA), IN	MAX AIRFLOW CFM	MATERIAL	MAX. NC
D-1	KRUEGER 1400	LOUVERED SQUARE CEILING DIFFUSER 4-WAY THROW	24 X 24	6	160	STEEL	12
				8	350		27
				10	440		26
				12	550		27
D-2	KRUEGER 1400	LOUVERED SQUARE CEILING DIFFUSER 4-WAY THROW	12 X 12	6	160	STEEL	26
				8	280		28
D-3	KRUEGER 1900	LINEAR SLOT, 3/4" SLOT, 4 SLOTS VERTICAL WITHOUT BLADES	48 LONG	6 (OVAL)	205	STEEL	27
				8 (OVAL)	290		31
				10 (OVAL)	330		32
				12 (OVAL)	415		33
D-4	KRUEGER	4" LOUVER	4 X 18		235	ALUMINUM	11
	DMD						
G-1		PERFORATED LAY-IN	24 X 24			STEEL	
G-2		PERFORATED LAY-IN	12 X 12			STEEL	

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AIR HANDLING UNIT SCHEDULE

MARK	AREA SERVED	MANF & MODEL	SUPPLY FAN: SYSTEM AIRFLOW (CFM)	EST EXT SP (IN WG)	# OF FANS	BELT OR DIRECT	FAN SPEED (RPM)	FAN DRIVE	BHP (HP EA)	MOTOR SIZE (HP EA)	MOTOR SPEED (RPM)	ELECTRIC HEAT: HEATING AIRFLOW (CFM)	OUTPUT (MBH)	DELTA T (°F)	FLA (AMPS)	VAC/PH/HZ (VAC/PH/HZ)	COOLING COIL: MAX FACE VELOCITY (FPM)	MAX FINS PER INCH	MIN TOTAL CAP (MBH)	MIN SENS CAP (MBH)	EAT DB/WB (°F)	LAT DB/WB (°F)	MAX AIR PD (IN WG)	VENTILATION AIR: MAX AIRFLOW (CFM)	ELECTRICAL: FLA (AMPS)	MCA (AMPS)	V/PH/HZ (V/PH/HZ)	MAX TOTAL WEIGHT (LBS)
AHU-1	2nd FLOOR/ 3rd FLOOR	TEMTROL	38,200	5.00	6	DIRECT	2,341	VFD	7	7.5	1770	38,200	341	8.4	125.6	460/3/60	492.9	8	1,515.40	1,044.70	80/67	54.2/53.5	0.62	16,714	76.6	80	460/3/60	15,926
AHU-2	1st FLOOR	TEMTROL	17,300	5.0	4	DIRECT	2,472	VFD	5	5	1,750	17,300	171	9.2	62.8	460/3/60	506.3	8	680.1	470.2	80/67	54.4/53.7	0.65	6,086	30.8	32.70	460/3/60	9,353
AHU-3	WORKSHOP	TEMTROL	1,700	5.0	1	DIRECT	3,806	CONSTANT	2	2	3450	1,700				460/3/60	377.8	8	72.9	49.1	80/67	52.7/52.3	0.43	1,358	NA	85.2	480/3/60	2,394

FAN COIL UNIT SCHEDULE

MARK	MANUFACTURER & MODEL	SERVES	TYPE	FAN INFORMATION: HIGH (CFM)	MEDIUM (CFM)	LOW (CFM)	COOLING COIL: REFRIG.	TYPE	TOTAL CAPACITY (MBH)	SENSIBLE CAPACITY (MBH)	ELECTRICAL DATA: HP	VOLT/PH/HZ	NOTES
FCU-1	KRUEGER HFEC12	ENTRY 107	HORIZONTAL STANDARD EXPOSED CABINET	1284	893	717	NONE	N/A	N/A	N/A	(2) 1/10	115/1/60	
FCU-2	KRUEGER HFEC06	STAIR 118/ENTRY 119	HORIZONTAL STANDARD EXPOSED CABINET	658	535	396	WATER	CHILL WATER	11.2	9.3	(1) 1/10	115/1/60	1
FCU-3	KRUEGER HFEC12	STAIR 137	HORIZONTAL STANDARD EXPOSED CABINET	1284	893	717	WATER	CHILL WATER	23.2	18.6	(2) 1/10	115/1/60	1
FCU-4	KRUEGER HFEC12	STAIR 201	HORIZONTAL STANDARD EXPOSED CABINET	1284	893	717	WATER	CHILL WATER	23.2	18.6	(2) 1/10	115/1/60	1
FCU-5	KRUEGER HFEC06	STAIR 211	HORIZONTAL STANDARD EXPOSED CABINET	658	535	396	WATER	CHILL WATER	11.2	9.3	(1) 1/10	115/1/60	1
FCU-6	KRUEGER HFEC12	STAIR 301	HORIZONTAL STANDARD EXPOSED CABINET	1284	893	717	WATER	CHILL WATER	23.2	18.6	(2) 1/10	115/1/60	1
FCU-7	KRUEGER HFEC06	STAIR 310	HORIZONTAL STANDARD EXPOSED CABINET	658	535	396	WATER	CHILL WATER	11.2	9.3	(1) 1/10	115/1/60	1

NOTES:
1. ELECTRIC HEAT

PUMP SCHEDULE

MARK	BASIS OF DESIGN Manuacturer and Model	SERVICE	LOCATION	TYPE OF FLOW	TYPE OF FLUID	PUMP: TYPE	FLOW gpm	HEAD ft. H2O	IMPELLER IN	MIN EFF %	CONTROL TYPE	MOTOR: SIZE hp	SPEED rpm
CD-1	TACO F2007	CONDENSER PUMP	OUTSIDE	VARIABLE	GLYCOL	BASE-MOUNTED, END SUCTION	300	175	7.25	70	VFD	20	3500
CWP-1	TACO 4009C	PRIMARY CHILLED WATER PUMP	OUTSIDE	VARIABLE	WATER	BASE-MOUNTED, END SUCTION	500	75	9.25	76	VFD	15	1760
GL-1	TACO 3009C	GLYCOL WATER PUMP – 1	OUTSIDE	VARIABLE	GLYCOL	BASE-MOUNTED, END SUCTION	250	25	8.0	78	VFD	2	1160
GL-2	TACO 3009C	GLYCOL WATER PUMP – 2	OUTSIDE	VARIABLE	GLYCOL	BASE-MOUNTED, END SUCTION	250	25	8.0	78	VFD	2	1160

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