2018 ASHRAE Design Competition
HVAC Design Calculations

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This Report Followed the 2018 ASHRAE Design Competition Format.

1. Executive Summary

This report is prepared by mechanical engineering students from British Columbia Institution of Technology for 2018 ASHRAE Student Design Competition. The goal of this project is to design heating, ventilation, and air conditioning (HVAC) system for a 70,000 ft^2, four story mixed use building in north of Istanbul, Turkey. The building contains retail spaces, a restaurant, office spaces, as well as hotel area. The design of the system is based on owner's project requirement (OPR), complied with latest ASHRAE Standards 55, 62.1, 90.1, 189.1, and ASHRAE Handbooks with consideration of Turkey Building Codes. The HVAC design includes zoning, ventilation rate and load calculations, system selection, energy analysis and life cycle cost analysis over 50 year's life of the facility.

Istanbul's weather is considered as a warm and humid, according to ASHRAE Standard 169-2013, "Climatic Data for Building Design Standards", Istanbul is located a zone category of 3A. By finding the climate zone number and considering the construction material stated by OPR; the building envelope for roofs, walls, floors, and windows were determined. Furthermore, zoning was designed based on several factors such as occupancy type, orientation, occupancy variation, etc.

Ventilation, heating and cooling load were determined by using HAP software. There are four Variable Air Volume (VAV) Air Handling Units used in this building to provide thermal comfort along with minimum air quality requirements. The main part of the building is served by three VAV systems located on the roof of the building; one AHU dedicated for retail space on the first floor, one for the second floor office spaces and one designed for hotel area on third and fourth floor. Heat Recovery Unit (HRU) is selected for the restaurant to recover energy from hot kitchen exhaust air. The main heating and cooling requirements are provided by air source heat pump located on the main building roof. In addition, a gas fired condensing boiler is selected for back up purpose with capability of providing the peak building heating load. The total heat and cool capacity required for this facility are 385,000 BTU and 113.9 tons respectively.

eQuest software energy consumption result demonstrates the total energy required by this building per year. The anticipated electric consumption found to be 1,167,300 kWh per annum. The total amount of natural gas required for equipment and heating is estimated to be 3,577.2 Million BTU annually. With accordance to OPR for utility cost and escalation rate, the total utility cost over 50 year totals to 11,000,664 USD. Photoelectric Array is used to take advantage of renewable energy. The solar system were designed in order to provide the expected 7% return on capital investment. The solar system contributes in generating 18% of the total electricity which will be 219,219 kWh per year. By adding the capital cost and estimated maintenance needs, the total cost of the facility at the end of the 50 year's life of the project is 12,912,288 USD which is equivalent to 184.46 USD per square foot. The estimated cost is well below the owner's expectation of 200 USD per square foot.

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5. Introduction

The purpose of this project is to design the heating, ventilation, and air conditioning (HVAC) system for a 70,000 square foot, four story mixed use building in north of Istanbul, Turkey. The building consists of retail spaces, a restaurant, office spaces, as well as hotel area. The design of the system is based on Owner's Project Requirements (OPR), and complying with latest ASHRAE Standards with consideration of Turkey Building Code.

5.1 Owner's Project Requirements (OPR)

The owner has provided certain requirements which are summarized as following,

- Design an air handling unit (AHU) complete with variable air volume (VAV) to meet the thermal comfort and indoor air quality requirements to ASHRAE Standards
- The building must comply with the latest ASHRAE Standards[1] stated below
    - ASHRAE Standard 55 – Thermal Environmental Conditions for Human Occupancy (2013)
    - ASHRAE Standard 62.1 – Ventilation for Acceptable Indoor Air Quality (2016)
    - ASHRAE Standard 90.1 – Energy Standard for Buildings Except Low-Rise Residential Buildings (2016)
    - ASHRAE Standard 189.1 – Standard for the Design of High Performance Green Buildings (2014)
    - ASHRAE Handbook: Fundamentals – 1989 and 2017
    - ASHRAE Handbook: HVAC Applications – 2015
- Building assumptions
- Miscellaneous loads which are going to be used in different spaces
- Budget consideration and limitation by examining life cycle costs of the building

5.1.1 Building Assumptions

The HVAC system should be designed based on the design criteria provided by the owner such as hours of operation, interior design condition for summer and winter, and sound criteria for each building space. The stated assumptions are tabulated in Table 1 based on the space usage.

Table 1. Building Operation of Owner's Requirements

	Retail	Restaurant	Office & Administrative Support Spaces	Lodging
Occupancy	9 am – 10pm Monday – Saturday 11am – 7pm Sunday	7 am – 10 pm Monday – Friday	7 am – 6 pm Monday – Friday 8 am – 1 pm Saturday	24 hours / day 7 days / week
Summer Interior Condition	73.4 °F (23°C) DB 50%RH	73.4 °F (23°C) DB 50%RH	73.4°F (23°C) DB 50%RH	78.8°F (26°C) DB 55%RH
Winter Interior Condition	70°F (21°C) DB	70°F (21°C) DB	70°F (21°C) DB	73.4°F (23°C) DB
Sound Criteria	NC 30	NC 30	NC 35	N/A

5.1.2 Miscellaneous Loads

Miscellaneous loads or equipment and appliances have sensible and/or latent heat which have direct impact on cooling and heating load calculations; therefore it is necessary to recognize what miscellaneous

[1] *Codes as determined by the local Authority Having Jurisdiction (AHJ)*

loads each space will have. Table 2 demonstrates the expected miscellaneous loads in each space throughout the entire building.

Table 2. Miscellaneous Loads for Entire Building

Space Type	Miscellaneous Type
Break and Vending Areas	Refrigerator Microwave / Coffee Vending Machine
Kitchen	Fryer – Gas, 8 Burner Range with Oven Ice Maker with bin 2-Door Reach-In Freezer 2-Door Reach-In Refrigerator Gas Griddle (2), Steam Table
Conference	CPU / Monitor LCD TV Projector x 2
Mechanical / Electrical	Loads as per required equipment
Office, Individual	CPU / Monitor
Office, Executive	CPU / Monitor LCD TV
Office, Open Areas	CPU / Monitor per workstation / person High volume copy machine x 1

5.1.3 Budget Considerations and Limitations

As explained above, one of the owner's requirement is to do the life cycle cost analysis in order to verify the sustainability and efficiency of the selected HVAC system. The owner's budget for the entire life of this facility is 200 USD per square foot. The expected return on investment is 7% with consideration of 3% inflation rate each year. The life of the building is to be 50 years and any utility escalation rate should be based on a 10-year trend for utility providers such as water and gas in Istanbul.

5.2 Climate Zone and Weather

5.2.1 Istanbul Climate Zone

According to ASHRAE Standard 169 Table A-6, "International Stations and Climate Zones", Istanbul climate zone number is 3A. Furthermore, ASHRAE Standard 189.1 Appendix A states the cooling degree day required in Istanbul is between 4,500 and 6,300. The amount of energy to maintain the building at design temperature is proportional to the cooling degree days (CDD). The Istanbul climate zone is summarized in Table 3.

Table 3. Climate Zone of Istanbul, Turkey

Climate Zone Number	Name	Thermal Criteria (I-P)	Thermal Criteria (SI)
3A	Warm – Humid	$4500 < CDD50°F \leq 6300$	$2500 < CDD10°C \leq 3500$

5.2.2 Weather Data

Figures 1 to 4, respectively illustrates the average minimum and maximum temperature of each month, participation of each month, relative humidity, and the average monthly hours of sunshine in Istanbul Turkey. The weather can be considered to be relatively warm and humid. The relative humidity during the summer is about 60% with average maximum temperature of 85°F. Based on the cooling 2%, heating 99% criteria, evaporation 1%, and dehumidification 1% for the climate of Istanbul Turkey, the coldest month is on February with 31.7°F, and the hottest month is on August with 84.6°F DB, 74.5°F WB, and 72°F DP.

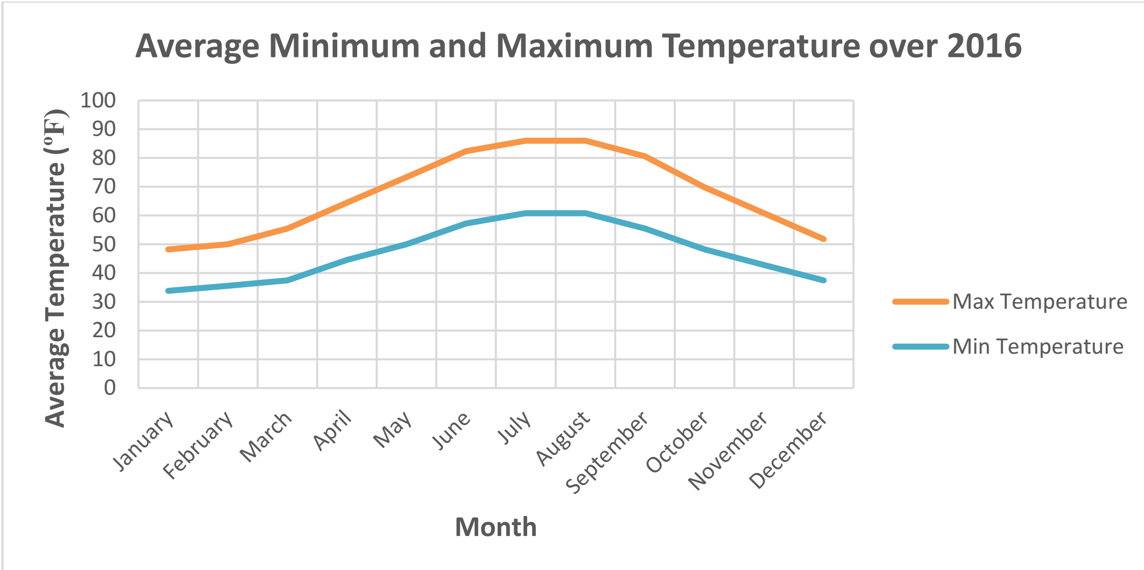

Figure 1. Average Minimum and Maximum Temperature (Climate Istanbul, 2016)

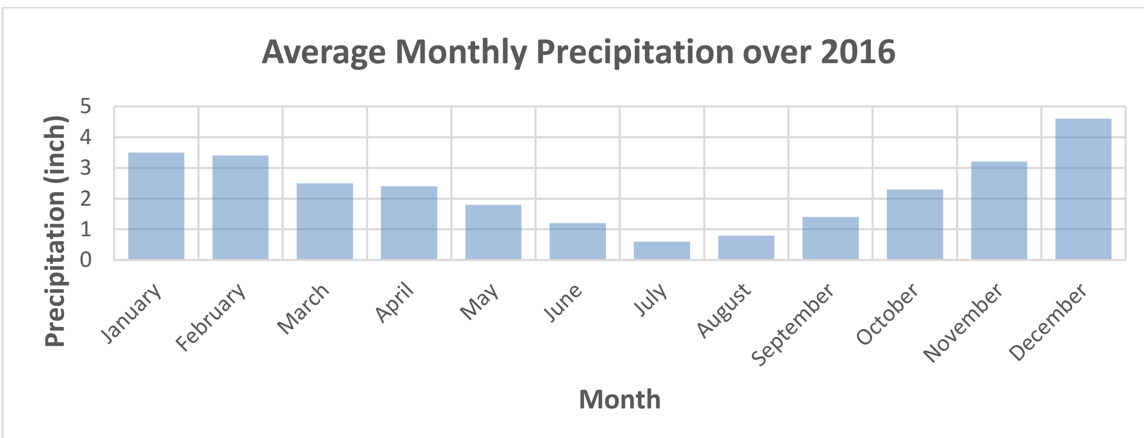

Figure 2. Average Monthly Precipitation (Climate Istanbul, 2016)

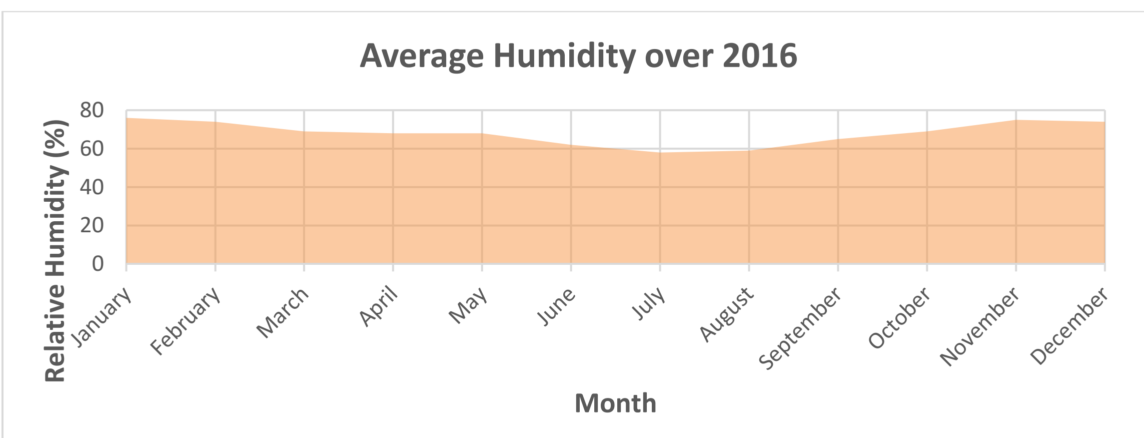

Figure 3. Average Humidity (Climate Istanbul, 2016)

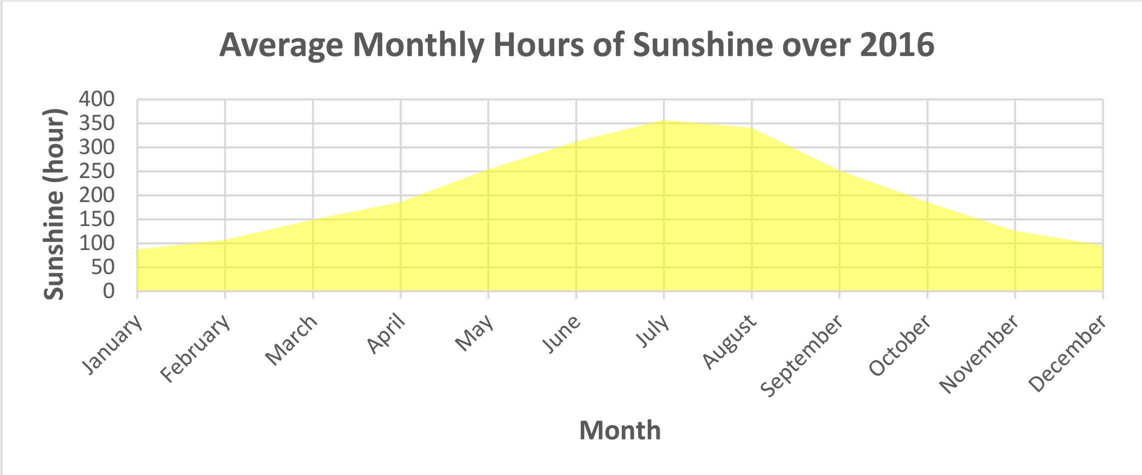

Figure 4. Average Monthly Hours of Sunshine (Climate Istanbul, 2016)

5.3 Building Envelope

ASHRAE Standard 189.1-2014 Table E-3, "Building Envelope Requirements for Climate Zone 3 (A, B, C)", provide the maximum U-values for different construction materials. Based on the construction materials provided in OPR and certain assumptions the U-value for different assemblies were calculated (for more details please refer to Appendix A). The maximum allowable building envelop as well as the calculated U-value used for the heating and cooling load calculations are tabulated in Table 4.

Table 4. Building Envelope of Design U-Value

Assembly	Construction Material	Maximum U-Value based on ASHRAE 189.1 for Non-Residential (Btu/h-ft^2-F)	Design U-Value (Btu/h-ft^2-F)
Roof	Synergy with surrounding architecture including red tile style roof	0.039	0.031
Floor (Ground Floor)	Concrete poured as slab on grade	0.074	0.074
Exterior Wall	Masonry Mass Wall Construction	0.123	0.068 (U_{eq} = 0.039)
Window	Double glazed, fixed windows, 1/2" air space, low emissivity coating on third surface, bronze tint	0.45	0.42
Spandrel Wall	Spandrel bronze-tinted glass, opaque, backed with air space, fiber and batt insulation	0.123	0.043
Door	-	0.7	0.7

In Table 4, the exterior wall U-value states the U-value of the masonry mass wall construction, however, due to the triangular geometry and existence of air trap in the middle of the geometry, the actual U-value may vary significantly. Therefore, the equivalent U-value is calculated (for more detail please refer to Appendix A) in order to replace the triangular geometry with a simple wall construction. The exterior wall U-value for the entire building is 0.039 BTU/h-ft^2-F with a thickness of 12 inches.

6. Zoning

The HVAC zones where selected based on occupancy type, air classification, and perimeter spaces. In addition, the zoning design such as area, number of occupant, and further the minimum ventilation rate complies with ASHRAE Standard 62.1.

Proper zoning design can significantly increase the efficiency of the HVAC system. On the other hand, poor zoning design could prevent thermal comfort, escalate energy consumption, higher maintenance cost and lastly overloading the HVAC system. Each zone has its own thermostat (which controls temperature, humidity) and requires a VAV terminal box. Each VAV box is equipped with a motorized damper, a reheat coil, and a differential pressure transmitter to measure the velocity of air passing through VAV terminal. Reheat coil within VAV box is provided to reheat the pre-heated air to the desired temperature for each zone. Since the cooling and heating loads required by each zone varies at different hours in a day, the perimeter zones demand more cooling and heating due to solar radiation.

The following are few key factors that were considered during the zoning design.

- Each zone has an area no more than 1,000 ft^2
- The perimeter depth[2] is 15 ft
- Spaces with the same orientation and occupancy type were grouped together as one zone
- Spaces with varying occupant loads have individual VAV terminal
- Spaces with no ventilation requirement will be considered as electric baseboard heating only

6.1 First Floor

Figure 5 below shows the zoning plan for the first floor. Two AHUs are used to serve the heating, cooling and ventilation requirements for the first floor. The reason that two AHUs are used in the first level is the fact that two spaces have different building schedule and occupancy type. One of the AHUs is used for the retail spaces including zones 1 to 19. All these zones are classified as Air Class 2 according to ASHRAE Standard 62.1-2016 Table 6.2.2.1, “Minimum Ventilation Rates in Breathing Zone”. Nonetheless, the washroom and mechanical room within the core spaces are ventilated by negative pressure.

Figure 5. Zoning Plan of First Floor

The restaurant area on east side of level 1 comprises of five different zones (20 to 24) which are all recognized as Air Class 2. The kitchen (zone 22 and 23) exhaust within the restaurant is considered to be Air Class 3 and the air is taken away from entering into restaurant by using induced ventilation. To take

[2] *Referring to ASHRAE Standard 90.1, a perimeter zone is defined as an area enclosing an exposed perimeter wall and depth of 15 ft*

advantage of the heat energy from exhaust air, AHU for restaurant is a Heat Recovery Unit (HRU) by integrated with an enthalpy wheel. There are three commercial type range hoods in the kitchen which the exhaust air is returned to HRU placed on the restaurant's roof.

6.2 Second Floor

In Figure 6, the second floor is for office usage and it is classified as Air Class 1. The zoning is done by considering the orientation of each office space as well as occupant variation loads. For instance, zone 1 and zone 7 each has its own VAV terminal unit due to having two walls exposed to exterior condition. Moreover, conference rooms have their own zones because of occupant variation loads in these spaces at different time of a day.

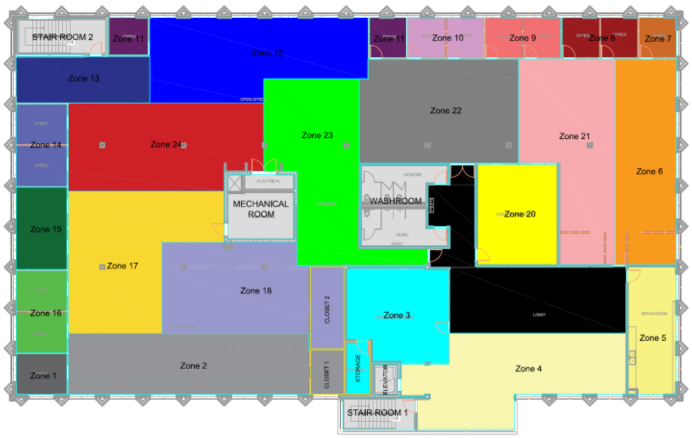

Figure 6. Zoning Plan of Second Floor

6.3 Third and Fourth Floor

In Figure 7, the top two floors are used for hotel purpose with an identical floor plan. The thermal comfort is achieved by one AHU serves both floors and each suite is designed to have its own VAV box. Each floor has 8 number of zones.

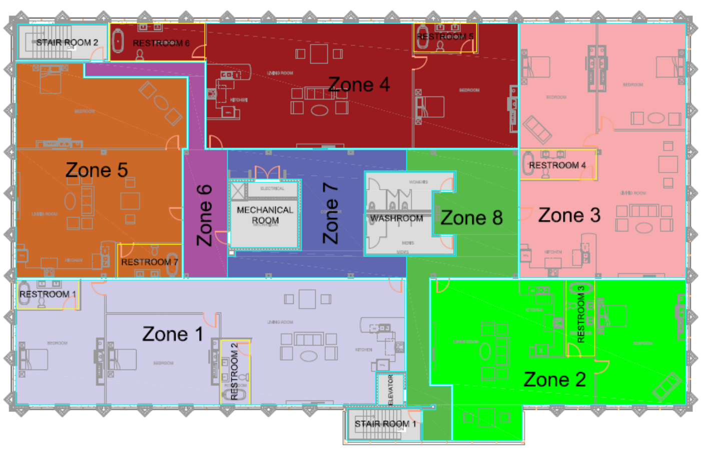

Figure 7. Zoning Plan of Third and Fourth Floors

7. Ventilation and Load Calculation

The ventilation and heating load calculations were performed in both Excel and HAP software. The cooling load calculations were analyzed by HAP software only. However, zone 1 on the second floor, as an example, was calculated by Excel further to verify the accuracy of the cooling load calculations obtained from HAP. Thermal loads in HAP are calculated using ASHRAE endorsed Transfer Function

Load methods. The ventilation and load calculations comply with the owner's requirements along with latest ASHRAE Standards 55, 62.1, 90.1, and 189.1.

7.1 Ventilation Rate

Ventilation is the process of bringing outdoor air to building space to achieve acceptable indoor air quality by means of diluting contaminants such as CO, NO and CO_2 to acceptable levels as well as removing air bone particulates, excess moisture along with unwanted heat. The ventilation rate calculations follow ASHRAE Standard 62.1. As stated earlier, the ventilation requirements for each space were analyzed by both Excel and HAP simultaneously for each zone to validate the accuracy of the results. There are various factors that can impact the minimum ventilation rate in the breathing zone such as zone floor area, occupancy category, people outdoor air rate, area outdoor air rate, occupant density, and zone population.

Table 5. Minimum Requirement of Ventilation Rate with Accuracy

Air Handling Unit #	Description	Ventilation Rate (CFM) Excel	Ventilation Rate (CFM) HAP	Error (%)
AHU – 1	1st Floor (Retail)	3194.9	3369	-5.4
AHU – 2	2nd Floor (Office)	1547.8	1382	10.7
AHU – 3	3rd & 4th Floor (Hotel)	977.1	918	6.1
AHU – 4	1st Floor (Restaurant)	1064.6	1062	0.2

Table 5 summarizes minimum ventilation rate required by each AHU to provide acceptable indoor air quality together with minimizing adverse health effects.

The total ventilation air required by the entire building according to the Excel calculation is 7761.5 CFM and the required ventilation rate for the building calculated by HAP is 7,649 CFM. The percentage difference between the two analyses is 1.4% which is in an acceptable range.

7.2 Heating Load

The heating load is needed to help sizing the heating system equipment, further determining building energy consumption and cost of providing the required heating. The building heat load refers to the amount of heat that heating equipment should offer to the building in order to overcome the building heat loss as well as ventilation load. The building heat loss depends on building envelope and infiltration loss.

The heating load for this project was determined by using both Excel and HAP to verify the accuracy of the calculations. Heating load on each AHU is displayed in Table 6 (for more details please refer to Appendix B).

Table 6. Heat Loss of Building with Accuracy

Air Handling Unit #	Description	Heating Load (BTU/h) Excel	Heating Load (BTU/h) HAP	Error (%)
AHU – 1	1st Floor (Retail)	68437.5	63300.0	7.5
AHU – 2	2nd Floor (Office)	47262.4	48100.0	-1.7
AHU – 3	3rd Floor (Hotel)	72579.3	72700.0	-0.2
	4th Floor (Hotel)	82210.2	78400.0	4.6
AHU – 4	1st Floor (Restaurant)	27584.3	28600	-3.6
Total		298073.7	291100.0	6.7

7.3 Cooling Load

The cooling load calculation were performed in HAP; however, cooling load calculation for an office located in the second floor (Zone 1) was calculated using Excel for verification purpose. Factors, such as CLTD and CLF, obtained from 1989 ASHRAE Fundamental Handbook – Chapter 26.

Space cooling load refers to the rate at which heat must be removed from space to maintain the space temperature at the desired indoor design conditions. There are several factors affect cooling load such as conduction through construction materials (sensible), solar radiation through windows (sensible), people (sensible and latent), lights (sensible), equipment or appliances (sensible and latent), infiltration (sensible and latent), as well as ventilation heat loss due to HVAC equipment.

Unlike heating load that was simple to calculate, cooling load is much more complex due to solar radiation. To size the equipment efficiently and effectively, the peak building cooling load for each air handling unit should be determined. The maximum building load occurs in August at 4 PM. Table 7 outlines the maximum cooling load required by each AHU.

Table 7. Heat Gain of Building

Air Handling Unit #	Description	Cooling Load (Tons) HAP
AHU – 1	1st Floor (Retail)	34.6
AHU – 2	2nd Floor (Office)	25.9
AHU – 3	3rd Floor (Hotel)	14.8
	4th Floor (Hotel)	15.1
AHU – 4	1st Floor (Restaurant)	23.5
Total		133.9

The Excel cooling load calculation for zone 1 on second floor is compared with HAP outputs for the same space. The Excel calculation were performed from June to October between 10 AM and 6 PM at two hours intervals. The maximum cooling load for the office is shown in Table 8 (for more details please refer to Appendix C).

Table 8. Accuracy of Heat Gain

Space	Cooling Load (BTU/h) Excel	Cooling Load (BTU/h) HAP	Error (%)
Zone #1, 2nd Floor, Office	4892.7	4669	4.6

8. System Selection

8.1 Overview

After determining the ventilation, heating and cooling loads required for the spaces in the building, the next step in the design process is to select the proper mechanical equipment. Mechanical equipment and systems serving the heating, cooling, and ventilation selected in accordance with AHSRAE Standard 90.1.

Owner requires to use air handling unit and VAV systems to meet HVAC needs. The HAVC requirements are serviced by using four different AHUs based on load requirements, occupancy type and building schedule. Two systems are used in first floor for serving the retail spaces and restaurant separately. One system is dedicated for the second floor to satisfy HAVC requirements to offices. Lastly, a system is used for the hotel on third and fourth floor.

All the HVAC systems in this building considered as centralized systems with All-Air units. The VAV terminals are equipped with hot water reheat coil to allow heating and cooling air supply at multiple zones simultaneously. To increase the efficiency of the system and reduce the waste energy, the indoor air is returned to each system respectively. By means of motorized dampers, the indoor return air is mixed with the minimum amount of outdoor air required to satisfy the minimum ventilation rate to dilute contaminants to the acceptable level. Since each AHU supplies zones that have the same air class, the indoor air can be returned back directly to the same AHU. In addition, each AHU is equipped with both heating and cooling coils to provide thermal comfort in the building. Figure 8 illustrates how a VAV system functions in general.

The heating water and chilled water are provided by the Air Source Heat Pump (ASHP) placed on the rooftop. In addition to the ASHP, a boiler is provided for backup heating purpose. The piping schematic design for serving hot water and chilled water to AHUs and VAV boxes is shown in Appendix E. In addition, the mechanical equipment schedule explained in the following subsections can be seen in Appendix F.

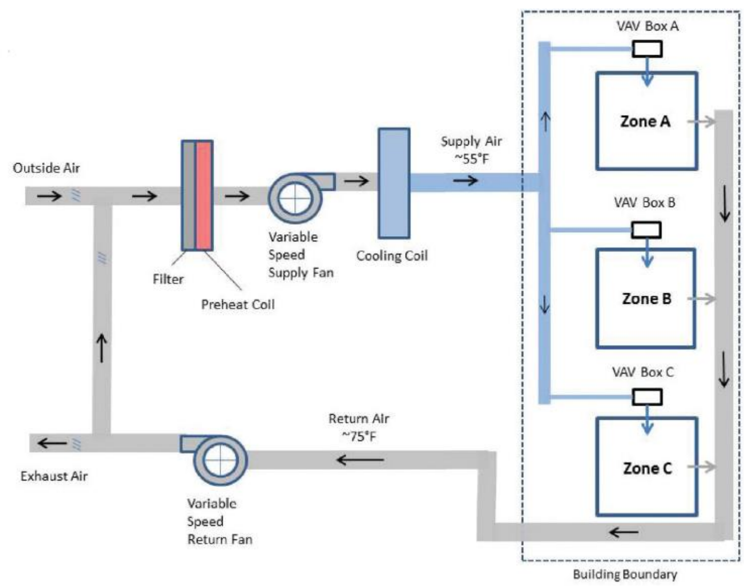

Figure 8. Function of VAV System (Crall, 2015)

8.2 VAV Air Handling Unit

The AHUs in this building consist of supply air fan, heating and cooling coils, filter, economizer, and mixed air compartment. The AHUs selected for retails, offices and hotel are very similar to one another with different fan power, as well as different heating and cooling coils capacity. On the other hand, the AHU in the restaurant is a HRU system which employs a cross flow between indoor air (exhaust) and outdoor air (intake) to take an advantage of the exhaust thermal energy to increase the efficiency as well as reducing the coil loads. All the air handling units and heat recovery unit for this project have the capability of dehumidification by placing heating coil upstream of the cooling coil. Thermostat is located within each zone to control the thermal comfort.

8.2.1 Filter

The AHU system is equipped with two filters that are selected to comply with ASHRAE Standard 62.1 – Module 3. Each AHU has a pre-filter (upstream of cooling coil) and a primary filter (downstream of cooling coil) with Minimum Efficiency Reporting Value (MERV) of 8 and 15 respectively.

8.2.2 Airside Economizer
Outdoor air damper, return air damper, and exhaust air damper are controlled in unison to provide free cooling by economizer cycle. Free cooling is provided when outdoor air is cooler and drier than design conditions. Having Economizer would diminish the use of cooling equipment; therefore less energy consumption is expected. ASHRAE Standard 90.1 requires airside economizer for any AHU greater than 54,000 BTU/h or 4.5 Tons.

8.2.3 Noise Limitation
OPR limits noise level in each space, the noise level in retail and restaurant areas should be less than 30 NC. Furthermore, noise level in office spaces must not exceed more than 35 NC. Air ducts need to be internally lined with acoustic insulation such as glass-wool to provide soundproof effect. Ducts are further sized and designed (Section 9) to limit air flow velocity as much as possible to achieve the stated noise requirements.

8.3 Air Source Heat Pump
The scroll Air Source Heat Pump is selected to provide required heating water and chilled water to the main coils and also serves the VAV boxes reheat coil for the entire building. The ASHP is placed on the rooftop of the main section of the building. The selected air source heat pump offers a combination of high efficiency, low sound levels and compact size to our application.

8.4 Boiler
The condensing boiler is designed based on maximum heat requirements (BTU/h) from all heating water coils for backup purpose in the case of heat pump shortage or failure of providing necessary heat. The selected condensing boiler is capable of overcoming all the necessary building heat loads.

8.5 Electric Base Board
The two staircases on each level in north and south part of the building, and the washrooms located in the hotel suites are equipped with electrical baseboard to satisfy heating requirements. The total number of 19 electrical baseboard heaters are selected to achieve the thermal comfort during the winter.

8.6 Forced Flow Heater
The main entry door of the entire building located at south side of the first floor is featured with a commercial type ceiling mounted forced flow heater in order to minimize the amount of heat loss due to high traffic of people accessing the building.

8.7 Exhaust System
The exhaust system is selected based on ASHRAE Standard 62.1. The exhaust system of the building is categorized under three different classes that need to be considered during the design procedure. The first category relates to the ventilation required for washroom spaces. The second exhaust category refers to residential kitchen exhaust located in hotel suits. Lastly, a separate exhaust system is design for the commercial kitchen space in the restaurant. The washroom doors are undercut to allow conditioned air from surrounding to make up for the exhausted air. Three range hoods for the exhaust air are located above the commercial kitchen appliances. The range hoods for the residential kitchens and the commercial kitchen are sized and selected in Section 10.2.1.

9. Duct Design
The duct design of the four-story mixed-use building was designed in REVIT as shown in Appendix D. The duct shape is chosen to be rectangular to save on space due to the limitation of plenum space. The ductwork for the entire building was sized using Duct Size Tool in REVIT. In order to size the duct work in REVIT, the tool required an input of friction in the duct that was assumed to be 0.08 in-wg/100ft. The duct work of the supply air for the entire building is insulated externally with 1” to 2” fiberglass blanket

backed with vapor barrier. This is for thermal insulation so the conditioned air within the box will not be affected by the surrounding air. In addition, the insulation prevent condensation to occur when cooling is provided to the space. Beside the thermal insulations, air ducts need to be internally lined with acoustic insulation such as glass-wool to provide soundproof effect and limits the noise generation.

The return air grills are placed on each floor ceiling, the return air travels through the plenum space (the space between the ceiling and above floor) to the return air duct designed to be in the mechanical room and finally return back to the AHUs on the rooftop.

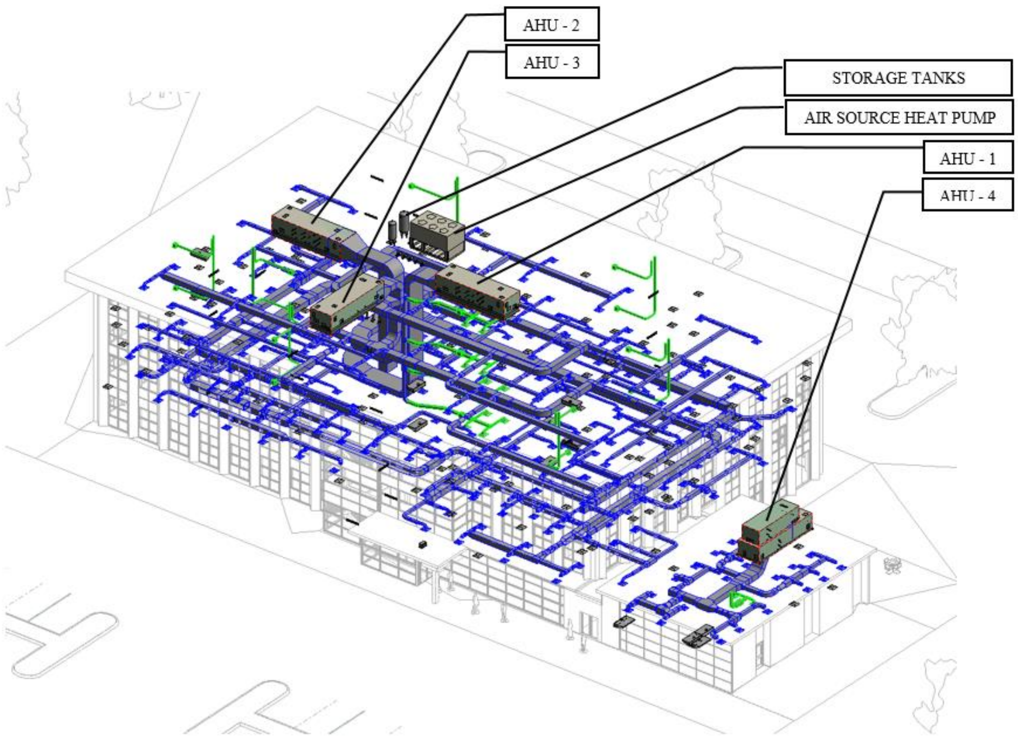

Figure 9. Duct Design

10. ASHRAE Standards

10.1 ASHRAE Standard 55

Thermal comfort for building occupants depends on the thermal balance between their body temperatures and the environmental conditions of their surroundings. Body temperature varies with activity and clothing as well with age, size, and gender. The environmental conditions in the surrounding are affected by air temperature, humidity, relative air velocity, and the radiant heat sources. Comfort air-conditioning systems provide simultaneous controls of air temperature, humidity, air cleanliness, and air distribution within the vicinity of building occupants. A well designed HVAC system will maintain acceptable thermal comfort conditions for the building occupants.

ASHRAE Standard 55 defines the environmental conditions required for comfortable indoor space for building occupants. The purpose of the standard is to establish specific conditions for indoor building space in which the majority of the sedentary or slightly active persons find their environment thermally acceptable.

The office spaces in the second floor are chosen to find the environment thermally comfortable. Referring to the ASHRAE Standard 55-2013 Table 5.2.1.2, “Metabolic Rates for Typical Tasks” and Table

5.2.2.2A, “Clothing Insulation Values for Typical Ensembles”, some assumptions regarding environment conditions are summarized in Table 9.

Table 9. Building Assumptions

	Value	Description
Clothing Insulation	0.96 clo	Occupants wear trousers, short-sleeve shirt, and suit jacket.
Metabolic Rate	1.0-1.1 met	Occupants’ activities are reading, writing, and typing.
Summer Indoor Temperature	73.4°F	Maintained temperature for office spaces by HVAC system.
Winter Indoor Temperature	70°F	Maintained temperature for office spaces by HVAC system.
Design Relative Humidity	50% RH	Maintained humidity for office spaces by HVAC system.
Range of Operative Temperatures	69.3-76.3°F	See Equations 1 and 2
Summer Outdoor Temperature	84.6°F	Based on the ASHRAE cooling 2% criteria.
Winter Outdoor Temperature	31.7°F	Based on the ASHRAE heating 99% criteria.

Operative temperature is rationally derived from the average air temperature and the mean radiant temperature. The range of the operative temperature for intermediate values of clothing insulation is defined by linear interpolation between the limits for 0.5 and 1.0 clo as follow,

$$t_{\min,Icl} = \frac{(I_{cl} - 0.5\text{ clo})t_{\min,1.0\ clo} + (1.0\text{ clo} - I_{cl})t_{\min,0.5\ clo}}{0.5\text{ clo}} \quad (1)$$

$$t_{\max,Icl} = \frac{(I_{cl} - 0.5\text{ clo})t_{\max,1.0\ clo} + (1.0\text{ clo} - I_{cl})t_{\max,0.5\ clo}}{0.5\text{ clo}} \quad (2)$$

where,

$t_{max,Icl}$ = Upper operative temperature limit for clothing insulation
$t_{min,Icl}$ = Lower operative temperature limit for clothing insulation
I_{cl} = Thermal insulation of the clothing

If the local air speed does not be controlled by occupants, the limits to average air speed, V, should meet the requirements as shown in Table 10, where t_a is the design temperature.

Table 10. Limits of Average Air Speed Regarding Operative Temperatures

Range of Operative Temperatures (t_o)	Maximum Average Air Speed (fpm)
$t_o \geq 77.9$°F	160
$t_o \leq 7.25$°F	30
72.5°F $\leq t_o \leq$ 77.9°F	$V = 31375.7 - 857.295t_a + 5.86288t_a^2$

The mean radiant temperature (MRT) of an environment is defined as the uniform temperature of an imaginary black enclosure which would result in the same radiation heat loss from the person and from the actual enclosure. Referring to the ASHRAE Fundamental Handbook, “Thermal Comfort”, MRT can be calculated from the measured temperature of surrounding walls and surfaces of the positions with respect to the person by the following equation.

$$\bar{T}_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \ldots + T_N^4 F_{p-N} \quad (3)$$

where,

$\bar{T}_r$ = Mean radiant temperature (°R)
T_N = Surface temperature of surface N (°R)
F_{p-N} = Angle factor between a person and surface N

Therefore, the definition of the operative temperature is based on average air temperature and mean radiant temperature as follow,

$$t_o = At_a + (1 - A)\bar{t}_r \quad (4)$$

where,

t_o = Operative temperature
A = Function of the relative air speed as shown in Table 11

Table 11. Factor, A, Regarding Relative Air Speed

v_r	< 40 fpm	40 fpm ~ 120 fpm	120 fpm ~ 200 fpm
A	0.5	0.6	0.7

The results are shown in Table 12 and will be used for a criteria of a diffuser selection, such as locations of the diffusers and a performance of air distribution. The range of the operative temperatures is 73.2°F to 75.7°F, which is met the range of the operative temperatures for clothing insulation (69.3°F to 76.3°F) based on the assumptions. Therefore, the majority of the occupants will find the environment thermally comfortable in the office spaces if the assumptions are valid.

Table 12. Result of Design Criteria

	Maximum Average Air Speed (fpm)	Operative Temperature (°F)
Winter Design Criteria	93.2	73.2
Summer Design Criteria	36.9	75.7

10.2 ASHRAE Standard 62.1

Ventilation is about air movement in a building, and it is critical in that it affects the health, life, and fire safety of occupants in the building. Ventilation has impacts on air quality, which must be maintained to acceptable levels for health safety of building occupants. Therefore, it can create different pressure zones for various levels of clean spaces within the building.

ASHRAE Standard 62.1 is intended to achieve an acceptable indoor air quality for 80% or more of building occupants in the space who do not express dissatisfaction with the ventilation in the surrounding environment. Air is classified into four categories:

Class 1: Air with low contaminant concentration, low sensory-irritation intensity, and inoffensive odor. The air is permitted to be transferred to any spaces in the building.
Class 2: Air with moderate contaminant concentration. The air can re-circulate within its own space or spaces with similar purpose and is permitted to be transferred to areas designated with Class 3 and Class 4.

Class 3: Air with significant contaminant concentration, significant sensory irritation intensity, or offensive odor. The air is only permitted to be re-circulated within its own space but cannot be transferred to any other spaces.
Class 4: Air with highly objectionable fumes or gases, with potentially dangerous particles, bioaerosols, or gases at concentration high enough to be considered harmful. The air cannot be re-circulated or transferred to any space.

ASHRAE Standard's Ventilation Rate Procedure (VRP) prescribes ventilation rates required in a "breathing zone". The VRP has four basic steps: (1) Satisfaction of the National Ambient Air Quality Standard (NAAQS), (2) Calculation of the ventilation in the breathing zone, V_{bz}, (3) Calculation of the outdoor air ventilation airflow in the ventilation zone, V_{oz}, and (4) Calculation of the outdoor air ventilation rate, V_{ot}.

In accordance with ASHRAE Standard 52.2, an air filter with a Minimum Efficiency Reporting Value (MERV) of 6 is required in the mechanical ventilation system when outdoor air drawn into the ventilation system has particulate matter that exceeds the national standard for PM10. In additional, an air filter with a MERV of not less than of 11 is required in the mechanical ventilation system when outdoor air drawn into the ventilation system has particulate matter that exceeds the national standard for PM2.5. Air cleaning is required whenever the outdoor ozone levels are expected to exceed 0.107 ppm.

Referring to the ASHRAE Standard 62.1-2016 Table 6.2.2.1, "Minimum Ventilation Rates in Breathing Zone", specifies the outdoor air ventilation rate per person, R_p, and per unit area, R_a. Zone population P_z, shall be equal to the largest number of people expected to occupy the zone during typical usage. The largest number can be estimated from the number of seats shown on the design plan. The definition for the ventilation in the breathing zone, V_{bz}, is,

$$V_{bz} = R_p \times P_z + R_a \times A_z \tag{5}$$

where,

V_{bz} = Ventilation in the breathing zone
R_p = Outdoor airflow rate required per person
P_z = Zone population
R_a = Outdoor airflow rate required per unit area
A_z = Zone floor area

The effectiveness of the air distribution system (deliver the outdoor air to the breathing zone), V_{oz}, is,

$$V_{oz} = \frac{V_{bz}}{E_z} \tag{6}$$

where,

V_{oz} = Ventilation outdoor airflow in the breathing zone
E_z = Effectiveness of the air distribution system (ASHRAE Standard 62.1-2016 Table 6.2.2.2)

The outdoor air ventilation rate can be considered in two situations: single zone systems and 100% outdoor air systems. The single zone system defines the supply air is a mixture of outdoor air and recirculated air to a single zone, and the 100% outdoor air system defines the supply air is only outdoor air to one or more zones. The definitions of the outdoor air intake flow are,

Single Zone Systems: $$V_{ot} = V_{oz} \tag{7}$$

100% Outdoor Air Systems: $$V_{ot} = \sum_{all\ zones} V_{oz} \tag{8}$$

A ventilation system can be designed to serve one or more zones. The outdoor air required at the outdoor air intake opening depends on the occupant diversity and the system ventilation efficiency. The definition of the occupant diversity, D, is,

$$D = \frac{P_s}{\sum_{all\ zones} P_z} \tag{9}$$

where,

P_s = Total population in the area served by the system

For a multiple zone system, the uncorrected outdoor air intake flow is modified from the ventilation outdoor air for a single zone system. The definition of the uncorrected outdoor air intake flow, V_{ou}, is,

$$V_{ou} = D \times \sum_{all\ zones}(R_p \times P_z) + \sum_{all\ zones}(R_a \times A_z) \tag{10}$$

The definition of the outdoor air intake flow, V_{ot}, is,

$$V_{ot} = \frac{V_{ou}}{E_v} \tag{11}$$

where,

V_{ot} = Design outdoor air intake flow
E_v = System ventilation efficiency (ASHRAE Standard 62.1-2016 Table 6.2.5.2)

The definition of the primary outdoor air fraction, Z_{pz}, is,

$$Z_{pz} = \frac{V_{ou}}{V_{pz}} \tag{12}$$

where,

Z_{pz} = Primary outdoor air fraction
V_{pz} = Zone primary airflow

The purpose of the exhaust ventilation is to ensure acceptable level of indoor air quality for healthy environments in a building. Any combination of outdoor air, re-circulated air, or transferred air can be used to make up for the exhausted air. The design exhaust airflow follows the ASHRAE Standard 62.1-2016 Table 6.5, "Minimum Exhaust Rates". Some units in the building should meet the minimum of the exhaust rates as shown in Table 13.

Table 13. Minimum Requirements of Exhaust Rates in Building

Occupancy Category	Exhaust Rate[3]	Unit	Air Class
Copy, printing rooms	0.50	cfm/ft^2	2
Kitchens – commercial	0.70	cfm/ft^2	2
Residential kitchens	50/100	cfm/unit	2
Toilets – private	25/50	cfm/unit	2
Toilets – public	50/70	cfm/unit	2

10.2.1 Commercial Kitchen Exhaust

Referring to 2015 ASHRAE Handbook: HVAC Applications, Chapter 33, to ensure a comfortable environment and the safety of personnel working in the kitchen, system design of kitchen ventilation has to be concerned. Typically, the kitchen ventilation includes reducing heat from cooking appliances, replacing air during cooking operation, and removing heat and effluent by cooking appliances.

There are kitchen spaces in the hotel (on 3rd and 4th floors), they are considered as residential kitchen. The design of exhaust systems for residential kitchen mostly is wall-mounted, conventional range hoods. A built-in duct connector of the hood should be same size as the duct, whether round or rectangular. In our application, the exhaust duct is a 2"x10" rectangular duct within the wall and up to gooseneck on the roof.

[3] *The higher rate where periods of use are expected to occur, but otherwise the lower rate should be used*

The need for replacement air is generally from supply air and natural infiltration. The range hood is controlled by an ON/OFF switch. Overall, the residential hoods are efficient due to the few running hours and the low rate of exhaust.

The commercial restaurant contains an 8-burner gas range, a steam table, two gas fryers, and two gas griddles that require a suitable type commercial exhaust hood to exhaust air. The style of all commercial exhaust hoods is selected to be Wall-Mounted Canopy as it is the most common for these appliances. Then, the hood type for each appliance and the duty category are determined based on the appliance type from ASHRAE Handbook: HVAC Applications – 2015 Section 33.9 Table 1 and Section 33.10 Table 2.

Next, the exhaust flow rate of each commercial exhaust hood are determined and designed based on the hood type of each appliance from ASHRA Handbook: HVAC Applications – 2015 Section 33.10 Table 3 for type I hood and Section 33.15 Table 7 for type II hood. Because it is not practical to place a separate commercial hood for each piece of appliance, therefore, the commercial hood (RH-1A) is designed to cover both of the 8-burner range and the two gas fryers. Similarly, the commercial hood of the two gas griddles (RH-1C) designed to cover both gas griddles. This is shown in Appendix D. Table 14 shows the list of appliances that require commercial exhaust hoods and their determined and designed parameters, such as the hood type, duty category, the code exhaust flow rate (CFM/ft), and the designed exhaust flow rate (CFM/ft).

Table 14. Requirement of Appliances

Appliance	Hood Type	Duty Category	Code Exhaust Flow Rate Per Foot (CFM/ft)	Designed Exhaust Flow Rate per Foot (CFM/ft)
Burner Range	1	Heavy Duty (600°F)	200-400	400
Gas Fryer	1	Medium Duty (400°F)	200-300	250
Steam Table	2	Light Duty (400°F)	200	200
Gas Griddle	1	Medium Duty (400°F)	200-300	240

10.3 ASHRAE Standard 90.1

The purpose[4] of ASHRAE Standard 90.1 is to stablish the minimum energy efficiency requirements of buildings for design, construction, a plan for operation and maintenance as well as renewable energy resources.

10.3.1 Climate Zone

The first step in designing the HVAC system for a building is to determine the climate zone which the building is located at. According to Table Annex1-1 which references ASHRAE Standard 169-2013 Table B-1 Istanbul, Turkey is considered as zone 3A.

10.3.2 Building Envelope

After determining the climate zone, the next step in the design procedure is to determine the building envelope. The building envelope selected for this building in Section 5.3 is with regard to Table 5.5-3, "Building Envelope Requirements for Climate Zone 3 (A, B, C)".

10.3.3 HVAC System

ASHRAE Standard 90.1 – Chapter 6 defines certain requirements regarding mechanical equipment and systems serving ventilation, heating and cooling loads. The following subsections summarize the key factors that were considered during the system design.

[4] *Service water heating and power distribution are not in the scope of this project as outlined in OPR*

10.3.4 Economizer
According to ASHRAE Standard 90.1 Section 6.5.1 states that any cooling system having capacity more than 54,000 BTU/h shall include air economizer to provide free cooling to the building. Furthermore, Table 6.5.1.1.3 requires high-limit set point based on climate zone. The economizer should be off when the outdoor air temperature exceeds 18 C (64.4 F).

10.3.5 Exhaust Air Energy Recovery
The VAV Air Handling Unit selected for the restaurant features energy recovery system. ASHRAE Standard 90.1 Section 6.5.6.1 requires energy recovery systems result in an enthalpy recovery ratio of at least 50%. In addition, the system should be able to bypass the energy recovery system. The unit selected for the restaurant is certified in accordance with the AHRI Standards and complies with ASHRAE Standards. According to manufacturer data, the total effectiveness of the system is 91.84%.

10.3.6 Minimum Equipment Efficiency
ASHRAE Standard 90.1 Section 6.8.2 defines the minimum efficiency for different HVAC systems. Table 15 below summarizes the minimum efficiency required by the Standard and compared with the equipment selected for this project. According to the manufacture (TRANE) ASHP complies with ASHRAE Standard 90.1 as well as they are certified by AHRI. The boiler is certified under CSA 4.9 testing protocols and procedure which are consistent with ASHRAE Standard as well as with US Department of Energy (DOE) provisions in 10 CFR-430 which requires minimum efficiency of 80%.

Table 15. Requirement of Minimum Efficiency of Equipment

Equipment	Size Category	Minimum Efficiency Required by ASHRAE	Efficiency of the Selected Equipment at Full Load
Air Side HP	65.81 Tons > 40kW	3.2 COP_H	12.4 EER = 3.56 COP_H
Gas Boiler	> 88kW and < 733 kW	80% 10 CFR - 430	95.7% CSA 4.9

10.3.7 Lighting
Lighting loads were determined by finding power density allowances from ASHRAE Standard 90.1 Section 6 – Table 9.6.1 using space-by-space method.

11. Energy and Life Cycle Cost Analyses
11.1 Energy Analysis
The total energy consumption of the building were analyzed by using eQuest. The total energy consumption of the building is divided into two categories; electricity and natural gas. The total electricity consumption of the building is 1,167,300 kWh annually. The solar system selected in Section 11.3 is capable of generating 219,219 kWh in a year which is equivalent to 18.8% of the annual electricity consumption. Figure 10 illustrates the electric consumption of the building in each month.

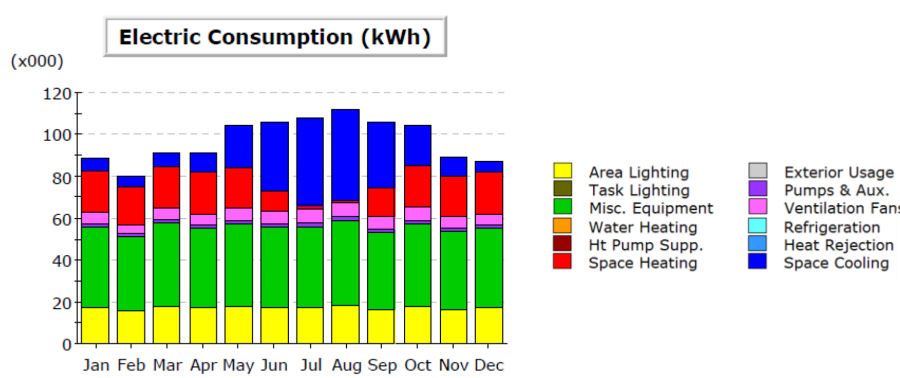

Figure 10. Electric Consumption of the Building

The gas consumption of the building mostly account for kitchen equipment since the main heating and cooling needs are provided by Air Source Heat Pump. However, boiler is assumed to work for two full months in a year. The total gas consumption of the building found to be 3,577.2 Million BTU per annum.

11.2 Life Cycle Cost Analysis

Life cycle cost analysis is a tool used to assess the total cost of the facility including capital cost of all equipment, materials, labor cost, maintenance cost, utilities, potential replacement costs and all other anticipated future costs.

The life cycle cost analysis is based on owner's requirements and in United State currency unit (USD). The maintenance and capital cost for all materials are obtained from manufacturers. The minimum man-hour required for this job obtained from industry experts and labor wage is based on current average salary for a technician in Turkey.

Utility rate for electrical and natural gas are from the given owner's requirement for this competition. Based on the given requirement the utility rate structure will rise at an annual rate of 3.5% for electrical cost, 3% for propane gas cost and 2.5% for water and sewer. The building service life is assumed a "Long Life" service building and is classified by ASHRAE Standard 189.1-2014 Table 10.3.2.3 which is indicated as a minimum service life of 50 years. General inflation rate for replacement, maintenance and anticipated future costs is 3%. In addition, the owner expects a minimum 7% return on capital investment.

11.2.1 Capital Cost

Capital cost includes the cost of HVAC equipment, materials, as well as labor cost to perform the required tasks. The capital cost of HVAC equipment is expected to be 285,758 USD for the purpose of this project. Moreover, the capital cost of PV array (Section 11.3) determined to be 105,600 USD. Therefore, the total capital cost of the project sums to 391,358 USD.

11.2.2 Labor Cost

Labor cost is estimated based on a technician income in Turkey. The average wage of a technician in Turkey is about 3 USD per hour which is a huge advantage for the owner to save on capital cost. The anticipated man hour wage required to perform ductworks, piping and installation of the equipment is estimated to be 11,760 USD.

11.2.3 Equipment and Material Cost

The cost of the equipment determined based on manufacturers, suppliers and industrial experts. The expected cost of the equipment and material for HVAC system is 273,998 USD.

11.3 PV Array

In order to achieve 7% return on investment specified by OPR, electric solar panel system is designed to produce portion of building electricity requirements in each year. Istanbul has an average of 6.5 hours of sun in each day of a year as shown in Figure 4. In addition, the geographical location of Istanbul is an advantage for producing efficient solar electricity.

PV Array SWA-350 XL manufactured by SolarWorld is used in this building. The PV electrical energy is rated as 0.35 kW per hour, and as mentioned earlier the cost of electricity is 0.125 USD/kWh. Therefore, the amount of energy and cost of each module annually is as follow,

$$0.35\frac{\text{Wh}}{\text{hour}}\times 6.5\frac{\text{hour}}{\text{day}}\times 365\frac{\text{day}}{\text{year}}=830,375\frac{\text{Wh}}{\text{year}}=830.375\frac{\text{kWh}}{\text{year}} \tag{13}$$

$$830.375\frac{\text{kWh}}{\text{year}}\times 0.125\frac{\text{USD}}{\text{kWh}}=103.80\frac{\text{USD}}{\text{year}} \tag{14}$$

Each module can save 103.80 USD/year and the capital cost of each module is 400 USD. In order to have 7% return on investment, 7% of the capital cost of solar panel is deducted from electricity produced in each year. As a result, the total saving from each panel would be 75.8 USD/year.

The total capital cost of HVAC system is 285,758 USD. The solar panels should be able to accommodate for 20,000 USD annually. Thus, the total of 264 panels are required. The capital cost of solar system would be 105,600 USD with occupying total rooftop area of 5,667 ft^2.

The annual amount of energy and cost savings for the solar system are calculated below.

$$830.375 \frac{\text{kWh}}{\text{year}} \times 264 \text{ panels} = 219,219 \frac{\text{kWh}}{\text{year}} \tag{15}$$

$$219,219 \frac{\text{kWh}}{\text{year}} \times 0.125 \frac{\text{USD}}{\text{kWh}} = 27,402.38 \frac{\text{USD}}{\text{year}} \tag{16}$$

The total energy produced by each solar panel is expected to reduce by 3% in the first year and not more than 0.7% over each year afterwards due to efficiency degradation. On the other hand, the cost of electricity is increasing by 3.5% each year. These factors were considered during the life cycle analysis of the entire building over 50 years life of the project.

11.4 Total Cost

The total cost of the energy (gas and electricity) required for this project by considering all aforementioned needs and analysis found to be 11,000,664 USD over the life of the project. By considering the total energy produced by the PV array, the cost of the energy is reduced to 9,509,511 USD. With consideration of the maintenance and capital cost, the total cost of the building adds to 12,912,288 USD. This amount equates to 184.46 USD/ft^2 which is less than 200 USD/ft^2 budget stated by the owner.

12. Conclusion

The HVAC system for a 70,000 ft^2, four story mixed use building in north of Istanbul, Turkey was analyzed in this report. The building contains retail spaces, a restaurant, office spaces, as well as hotel area. The design of the system is based on OPR, complied with latest ASHRAE Standards 55, 62.1, 90.1, 189.1, and ASHRAE Handbooks with consideration of Turkey Building Codes.

Ventilation, heating, and cooling loads were determined using HAP software. There are four VAV Air Handling Units used in this building to provide thermal comfort along with minimum air quality requirements. The main heating and cooling requirements are provided by air source heat pump located on the main building roof. In addition, a gas fired condensing boiler is selected for back up purpose with capability of providing the total peak building heating load. The total heat and cool capacity required for this facility is 385,000 BTU and 113.9 tons respectively.

The anticipated electric consumption found to be 1,167,300 kWh per annum. The total amount of natural gas required for equipment and heating is estimated to be 3,577.2 Million BTU annually. With accordance to OPR for utility cost and escalation rate, the total utility cost over 50 year totals to 11,000,664 USD.

13. Acknowledgements

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15. Appendix

Appendix A: Building Envelope

Table 16. Assemblies of Building Envelope

Building Envelope	Assembly	U-Value (Btu/h-ft^2-F)
Roof	0.034” Flat Metal Deck + 9.5” Rigid Closed-Cell Polyisocyanurate Foam Core Insulation + 0.38” Light Colored Membrane Roofing	0.031
Spandrel Wall	1/2” Spandrel Bronze-Tinted Glass, Opaque, Backed With Air Space + 1” Rigid Mineral Fiber Insulation + 4” Mineral Fiber Batt Insulation + 5/8” Gypsum Wall Board	0.043
Masonry Mass Wall	4” Light-Brown-Colored Face Brick + 4” Lightweight Concrete Block + 3-1/2” Mineral Fiber Batt Insulation + 5/8” Gypsum Wall Board	0.068
Window	Aluminum Frame With Thermal Breaks + 1/8” Bronze-Tinted Outdoor Pane, Fixed Windows + 1/2” Air Space + 1/8” Clear Indoor Pane With Light-Translucent Roller Shades, Low Emissivity Coating	0.42
Floor (Ground)	4” Heavyweight Concrete Slab On Grade	0.074

Building envelope includes walls, roofs, floors, etc. Thermal resistances of these building components are used to establish the overall thermal conductance (U-value) for the heat loss and the heat gain calculations as $U = 1/R_T$, where R_T is the total thermal resistance. Wall, window, roof, and floor assemblies can have many variations, and assemblies of the walls, the windows, the roof, and the floor in the building are shown in Table 16.

However, a masonry mass wall used in the building have a specific geometry as shown in Figure 11, so its thermal conductance is significant different from the same type of the masonry mass walls but with a flat geometry. To enhance an accuracy of load calculations, the thermal conductance of the specific masonry mass wall can be determined as shown in Figure 11.

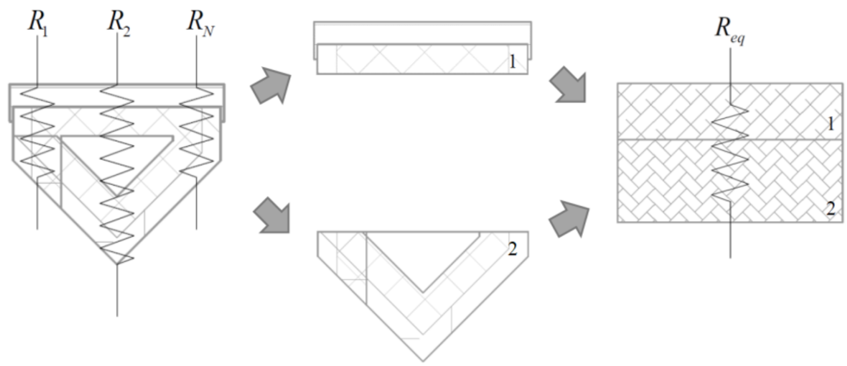


Figure 11. Illustration of Determining U-Value of Specific Masonry Mass Wall

In Figure 11, the equivalent thermal resistance, R_{eq}, is the average of the total thermal resistance as,

$$R_{eq} = \frac{\sum R_N}{N} = \frac{R_1 + R_2 + ... + R_N}{N} \tag{17}$$

where,

R_{eq} = Equivalent thermal resistance of the specific masonry wall
R_N = Resistances at different points along the wall
N = Total number of resistances

In addition, by dividing a part into two sections and based on the theory of heat transfer, the conduction and convection resistances of each section can be expressed as,

$$R_i = \frac{1}{h_i A_i} + \frac{L_i}{k_i A_i} \tag{18}$$

where,

h = Heat transfer coefficient
A = Area of the wall normal to the direction of heat transfer
L = Length of the wall parallel to the direction of heat transfer
k = Thermal conductivity

Because the resistances are in series and can be summed as,

$$R_{eq} = R_1 + R_2 \tag{19}$$

In Table 17, the result shows the thermal resistance of the masonry wall with the specific geometry is approximately 1.7 times greater than the thermal resistance of the masonry wall with the flat geometry, which mean the specific masonry wall is more efficient to maintain indoor temperature.

Table 17. U-Values of Original and Specific Masonry Mass Wall

	Masonry Mass Wall	
	Flat Geometry	**Specific Geometry**
R-Value ($h\text{-}ft^2\text{-}F/\ Btu$)	14.7	25.6
U-Value ($Btu/h\text{-}ft^2\text{-}F$)	0.068	0.039

Appendix B: Heating Load Calculation

The heating load of a building is the amount of heat required for a building heating system to maintain acceptable indoor temperatures during heating season. The heating system is designed with heating equipment that can generate sufficient heat to balance the maximum probable heat loss through the building. Therefore, the calculation of heating load is based on estimates of maximum heat losses, Q_{Total}, as following,

$$(Q_H)_{Total} = \sum Q_H \tag{20}$$

where, Q_H is the heat loss escaped through an exterior surface above ground, a slab-on-grade floor, and air infiltration.

The definition of the heat loss through the exterior surfaces above ground (such as roofs, walls, windows, etc.) is,

$$Q_H = U \times A \times \Delta T \tag{21}$$

where,

U = Overall thermal conductance (BTU/h-ft^2-F)
A = Area of a building component (ft^2)
ΔT = Difference between indoor (design) and outdoor temperatures

The definition of the heat loss through the slab on grade floor is,

$$Q_H = \frac{L \times \Delta T}{1.21 + 0.124 \times R_{edge} + 0.0103 \times R_{edge}^2} \tag{22}$$

where,

L = Length of exposed edge (ft)
R_{edge} = R-value of edge insulation (h-ft^2-F/BTU)

The definition of the heat loss through the air infiltration is,

$$Q_H = 1.08 \times Q_{air} \times \Delta T \tag{23}$$

$$Q_H = 4840 \times Q_{air} \times \Delta W \tag{24}$$

where,

Q_{air} = Air leakage (CFM), which is $Q_{air} = V_{room} / 60 \times ACH$
V_{room} = Room volume (ft^3)
ACH = Air change method, 0.2 ACH for newly energy efficient buildings, and 0.5 ACH for most residential and commercial buildings
ΔW = Difference in moisture content between indoor and outdoor air

Appendix C: Cooling Load Calculation

The cooling load of a building is the amount of cooling required for a building cooling system to remove the effects of heat gains in spaces in a building. Sources of the heat gains include solar radiation through glasses, heat conduction through exterior surfaces, sensible heat from lightings and equipment, sensible and latent heat from occupants, sensible and latent heat from miscellaneous equipment, and infiltration air and ventilation.

Therefore, the calculation of cooling load is based on estimates of maximum heat gains, Q_{Total}, as following,

$$(Q_C)_{Total} = \sum Q_C \tag{25}$$

where, Q_C is the heat entered into the building from the sources.

Heat is conducted through whenever there is a temperature difference between outdoor and indoor, so the definition of the cooling load due to exterior opaque surfaces (roofs and walls) is,

$$Q_C = A \times U \times CLTD_{corr} \tag{26}$$

$$CLTD_{corr} = \left[(CLTD + LM) \times k + (78 - T_i) + (T_o - 85)\right] \times f \tag{27}$$

where,

$CLTD_{corr}$ = Corrected CLTD (°F)
CLTD = Cooling load temperature difference (1989 ASHRAE Fundamental Handbook – Chapter 26, Tables 28-29 for roofs, and Tables 30-31 for walls)
LM = Latitude-month (1989 ASHRAE Fundamental Handbook – Chapter 26, Table 32)
k = Color adjustment factor
f = Ventilation adjustment factor
T_i = Indoor (design) temperature (°F)
T_o = Outdoor temperature (°F)

Heat transmission through glazing consists of heat conduction and solar radiation, so the definition of the cooling load due to exterior transparent surfaces (windows and skylights) is,

$$Q_C = Q_{radiation} + Q_{conduction} \tag{28}$$

$$Q_{radiation} = A \times SC \times SHGF \times CLF \tag{29}$$

$$Q_{conduction} = A \times U \times CLTD_{corr} \tag{30}$$

$$CLTD_{corr} = CLTD + (78 - T_i) + (T_o - 85) \tag{31}$$

where,

CLTD = Cooling load temperature difference (1989 ASHRAE Fundamental Handbook – Chapter 26, Table 33)
SC = Transparent surface shading coefficient
SHGF = Solar radiation (1989 ASHRAE Fundamental Handbook – Chapter 26, Tables 34-35)
CLF = Cooling load factor with no interior shade or with shade (1989 ASHRAE Fundamental Handbook Tables – Chapter 26, 36-39)

Body can release sensible and latent heat, and it is one of the internal heat gains in the building, so the definition of the cooling load due to occupants is,

$$Q_C = Q_{sensible} + Q_{latent} \tag{32}$$

$$Q_{sensible} = n \times Q_S \times CLF \tag{33}$$

$$Q_{latent} = n \times Q_L \tag{34}$$

where,

n = Number of occupants
Q_S = Body sensible heat (2017 ASHRAE Fundamental Handbook – Chapter 18, Table 1)
CLF = Cooling load factor (1989 ASHRAE Fundamental Handbook – Chapter 26, Table 40)
Q_L = Body latent heat (2017 ASHRAE Fundamental Handbook – Chapter 18, Table 1)

Lightings are the internal heat gain in the building, so the definition of the cooling load due to lightings is,

$$Q_C = 3.412 \times W \times UF \times BF \times CLF \tag{35}$$

where,

W = Light wattage (2017 ASHRAE Fundamental Handbook – Chapter 18, Table 2)
UF = Use factor, 1 for commercial
BF = Ballast factor, 1.2 for fluorescent lights, and 1 for others
CLF = Cooling load factor (1989 ASHRAE Fundamental Handbook – Chapter 26, Tables 43-47)

Appliances are the internal heat gain in the building, and some of appliances release both sensible and latent heat, but some of appliances release only sensible heat, so the definition of the cooling load due to appliances is,

$$Q_C = Q_{sensible} + Q_{latent} \tag{36}$$

$$Q_{sensible} = Q_S \times CLF \tag{37}$$

$$Q_{latent} = Q_L \tag{38}$$

where,

n = Number of appliances
Q_S = Appliance sensible heat (2017 ASHRAE Fundamental Handbook – Chapter 18, Tables 4-12)
CLF = Cooling load factor (1989 ASHRAE Fundamental Handbook – Chapter 26, Tables 48-49)
Q_L = Appliance latent heat (2017 ASHRAE Fundamental Handbook – Chapter 18, Tables 4-12)

Infiltration is the amount of air leakage into the building, so the definition of the cooling load due to air leakage is,

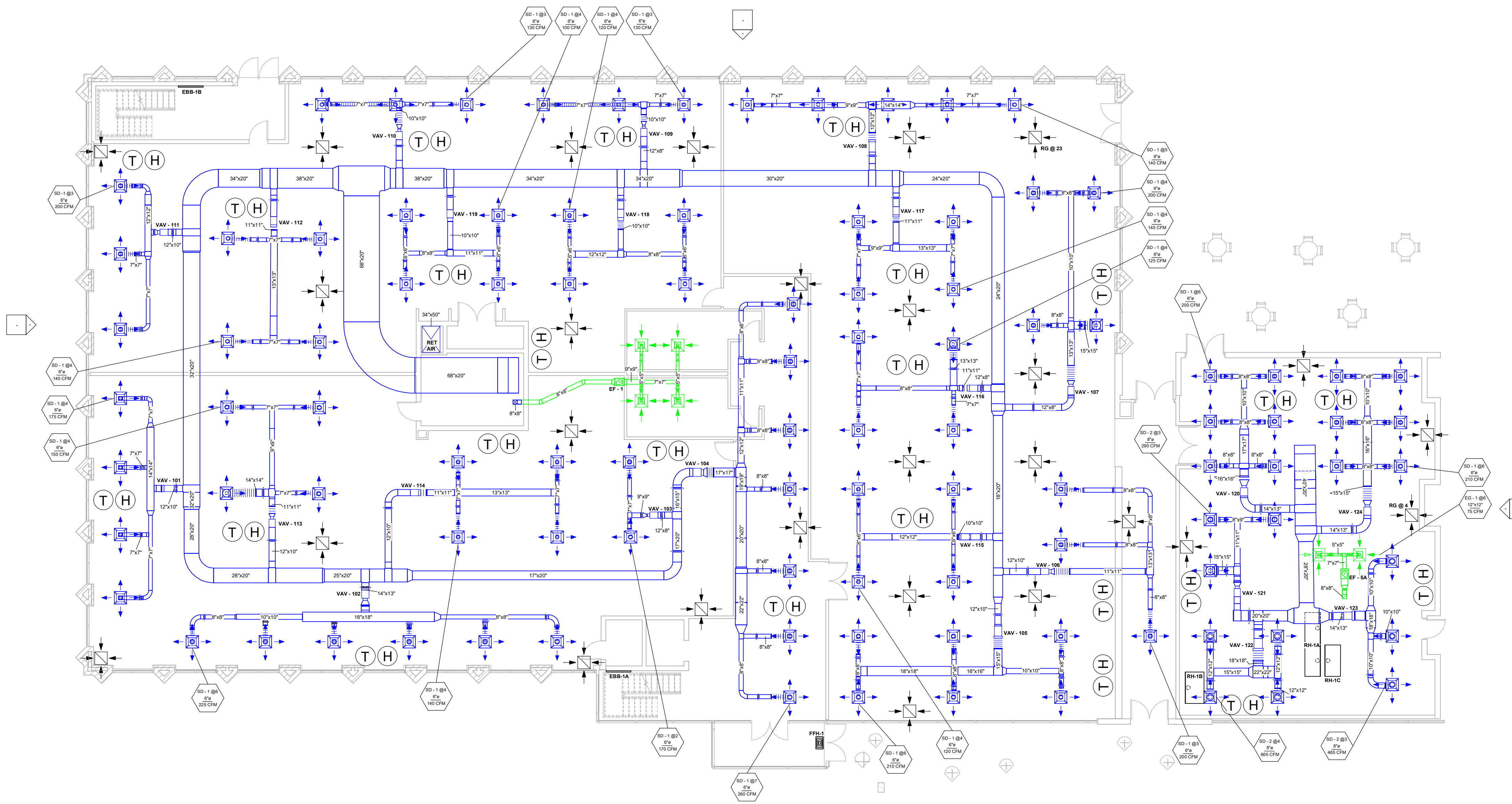
$$Q_C = Q_{sensible} + Q_{latent} \tag{39}$$

$$Q_{sensible} = 1.08 \times Q_{air} \times \Delta T \tag{40}$$

$$Q_{latent} = 4840 \times Q_{air} \times \Delta W \tag{41}$$

where,

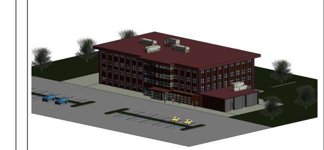
Q_{air} = Air leakage (CFM), which is $Q_{air} = V_{room} / 60 \times ACH$
V_{room} = Room volume (ft^3)
ACH = Air change method, 0.2 ACH for newly energy efficient buildings, and 0.5 ACH for most residential and commercial buildings
ΔW = Difference in moisture content between indoor and outdoor air



Level 1
1 : 75

APPENDIX D-1

REV.	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	5/4/2018



2018 ASHRAE DESIGN COMPETITION

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL FLOOR PLAN - LEVEL 1

Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

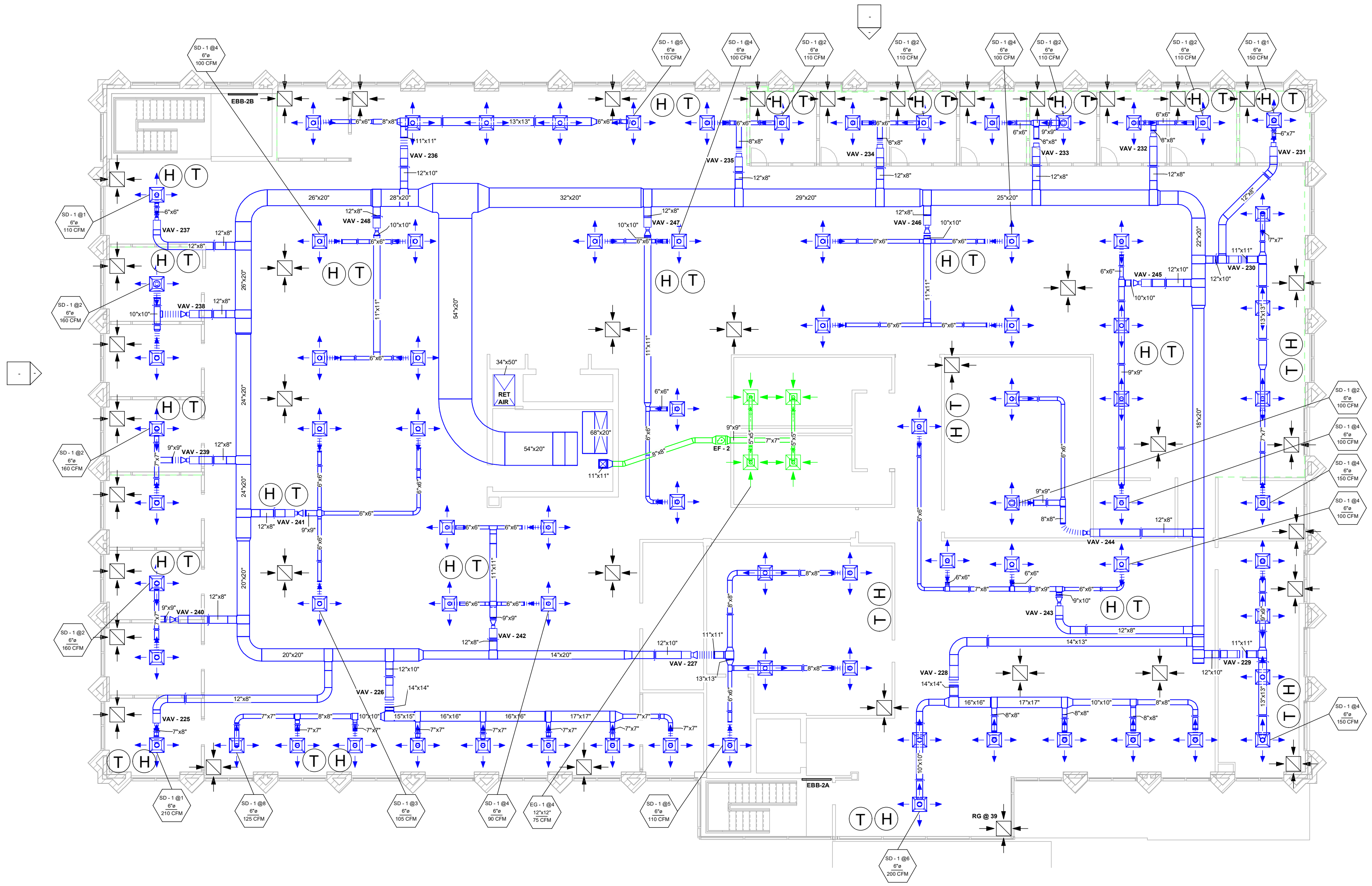
M - 1001

Scale	1 : 75

2018-05-03 11:03:24 AM

APPENDIX D-2

REV.	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	5/4/2018



Level 2
1 : 75

2018 ASHRAE DESIGN COMPETITION

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL FLOOR PLAN - LEVEL 2

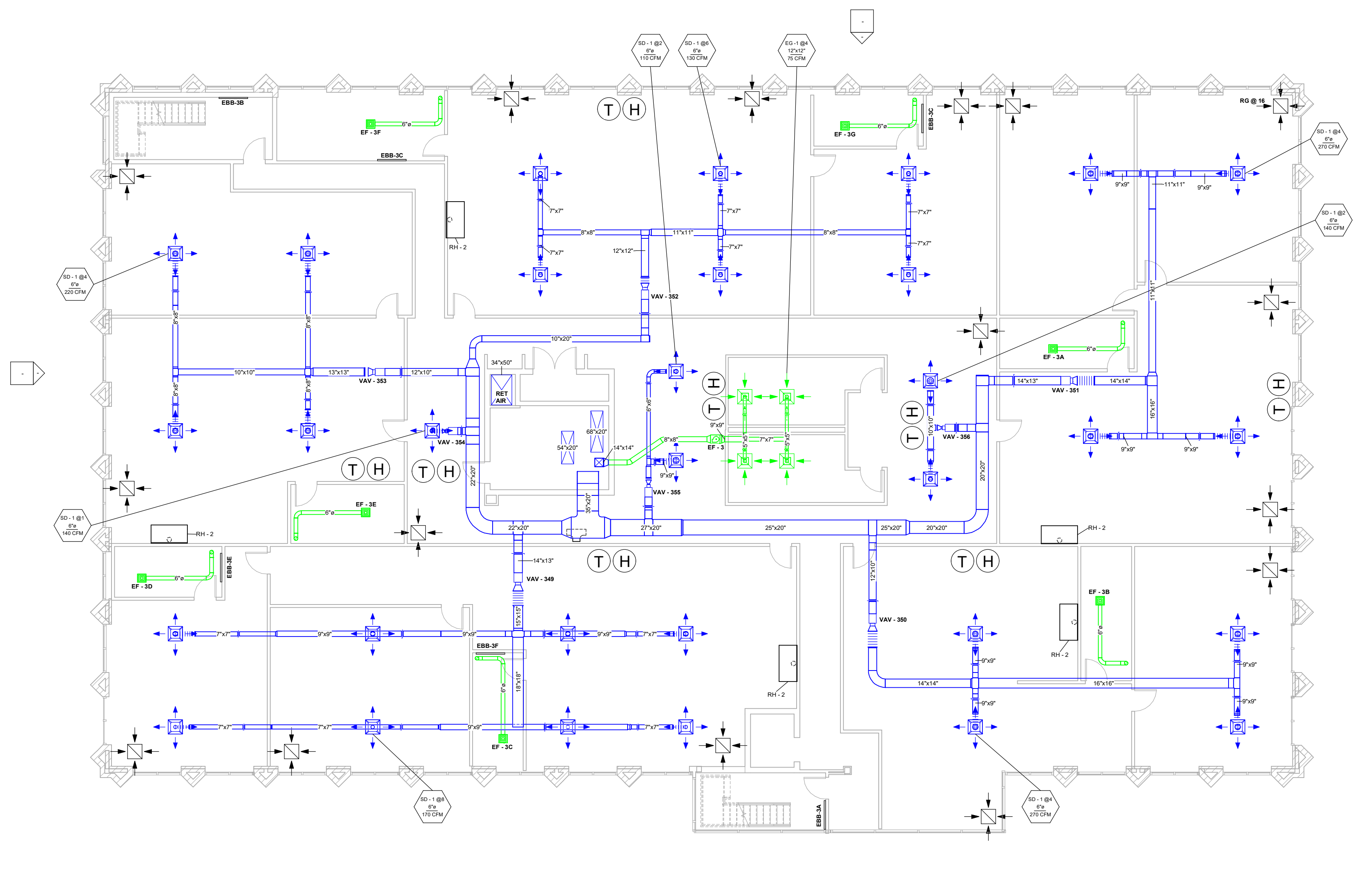
Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

M - 1002

Scale 1 : 75

2018-05-03 11:11:14 AM

APPENDIX D-3



① Level 3
1 : 75

REV.	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	5/4/2018

2018 ASHRAE DESIGN COMPETITION

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL FLOOR PLAN - LEVEL 3

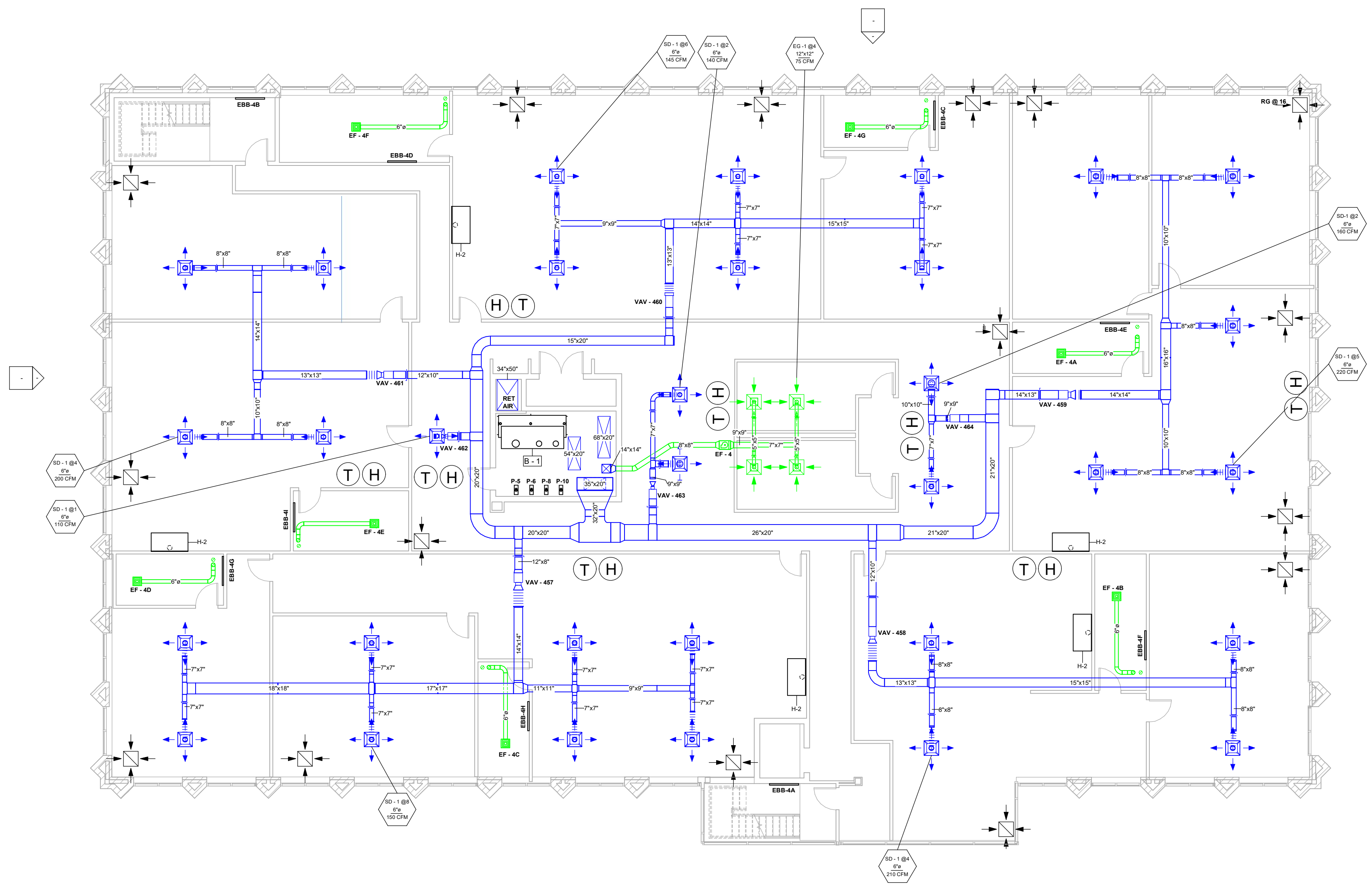
Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

M - 1003

Scale	1 : 75

2018-05-03 11:04:22 AM

APPENDIX D-4



1 Level 4
1 : 75

REV.	DESCRIPTION	DATE
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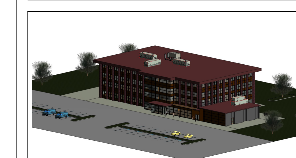


2018 ASHRAE DESIGN COMPETITION

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL FLOOR PLAN - LEVEL 4

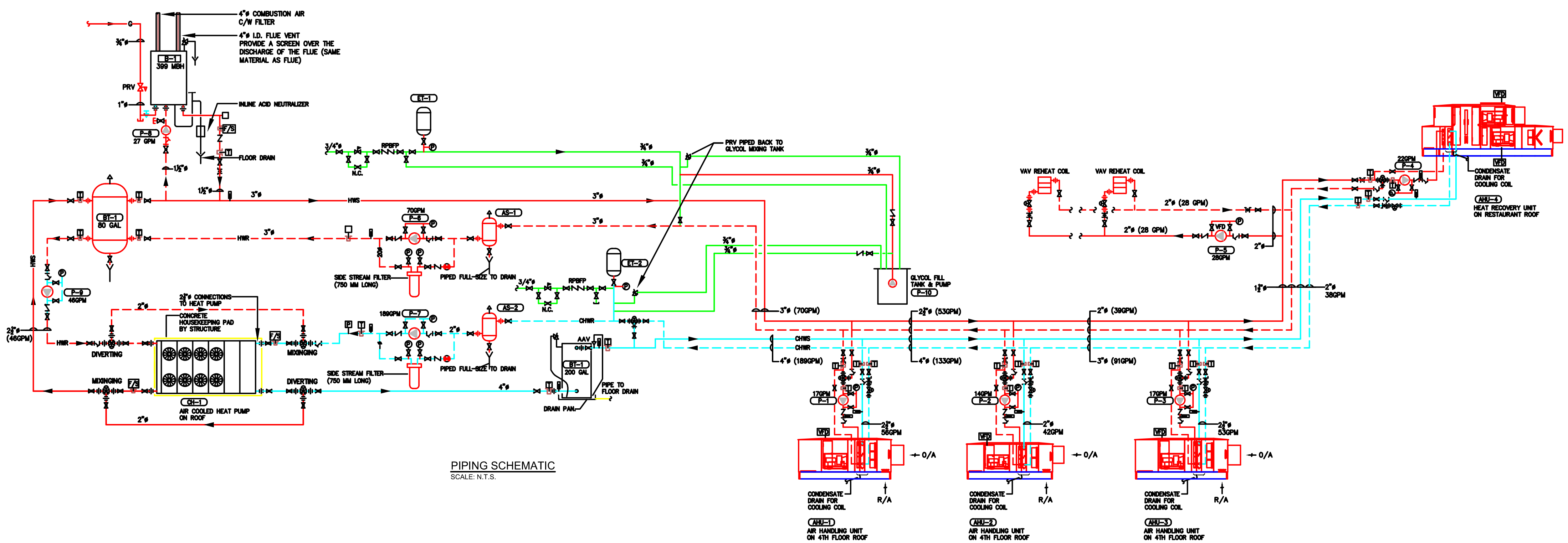
Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

M - 1004

Scale 1 : 75

2018-05-03 11:04:45 AM

APPENDIX E



PIPING SCHEMATIC
SCALE: N.T.S.

REV.	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	5/4/2018

2018 ASHRAE DESIGN COMPETITION

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL PIPING SCHEMATIC

Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

M - 2001

Scale	N.T.S.

AIR HANDLING UNITS

TAG	LOCATION	SERVICE	MAKE	MODEL	SUPPLY FAN: AIR FLOW (CFM)	SUPPLY FAN: ESP (IN W.C.)	SUPPLY FAN: FRPM	SUPPLY FAN: BHP (HP)	SUPPLY FAN: MOTOR (HP)	SUPPLY FAN: MOTOR (RPM)	SUPPLY FAN: ELECTRICAL (V/PH/HZ)
AHU - 1	4th FLOOR ROOF	CRU_L1	TRANE	CSAA-25	12500	5.604	2091	16.894	20	1800	460/3/50
AHU - 2	4th FLOOR ROOF	OFFICE_L2	TRANE	CSAA-21	10100	4.972	2249	12.923	15	1800	460/3/50
AHU - 3	4th FLOOR ROOF	HOTEL_L3_L4	TRANE	CSAA-25	12300	5.535	2068	16.393	20	1800	460/3/50
AHU - 4	RESTAURANT ROOF	RESTAURANT	TRANE	CSAA-14	7683	6.271	2330	11.559	15	1800	460/3/50

TAG	HEATING COIL: TOTAL CAPACITY (MBH)	AIR: EDB (F)	AIR: LDB (F)	AIR: PD (W.C.)	WATER : AHRI 410: FLOW (GPM)	WATER : AHRI 410: ET (F)	WATER : AHRI 410: LT (F)	WATER : AHRI 410: WPD (W.C.)
AHU - 1	253.9	75	93.73	0.072	16.91	180	150	0.6
AHU - 2	207.06	75	93.9	0.07	13.79	180	150	0.48
AHU - 3	251.81	75	93.88	0.07	16.77	180	150	0.59
AHU - 4	322.9	53.06	95	0.186	21.51	180	150	1.52

TAG	COOLING COIL: TOTAL CAPACITY (MBH)	SENSIBLE CAPACITY (MBH)	AIR: EDB (F)	AIR: EWB (F)	AIR: LDB (F)	AIR: LWB (F)	AIR: PD (W.C)	WATER: AHRI 410: FLOW (GPM)	WATER: AHRI 410: ET (F)	WATER: AHRI 410: LT (F)	WATER: AHRI 410: WPD (W.C)
AHU - 1	415	317	80.8	66.7	57.76	56.07	0.42	55.14	41	56	6.45
AHU - 2	314.4	243.42	80.8	66.7	58.92	56.81	0.38	41.77	41	56	12.45
AHU - 3	392	302.13	80.8	66.7	58.5	56.55	0.397	52.08	41	56	5.85
AHU - 4	282	207.65	80.8	66.7	54.06	53.59	0.665	37.47	41	56	6.94

TAG	EXHAUST FAN: AIR FLOW (CFM)	ESP (IN W.C.)	FRPM	BHP (HP)	MOTOR (HP)	MOTOR (RPM)	ELECTRICAL (V/PH/HZ)
AHU - 1	-	-	-	-	-	-	-
AHU - 2	-	-	-	-	-	-	-
AHU - 3	-	-	-	-	-	-	-
AHU - 4	4133	3.84	1629	3.732	5	1800	460/3/50

VAV TERMINAL UNIT SCHEDULE

EQ TAG	MAKE	MODEL	LOCATION	BOX INLET	MAX FLOW (CFM)	MIN FLOW (CFM)	REHEAT COIL	NUMBER OF ROWS	MBH	GPM	WPD (W.C.)
VAV - 101	E.H.PRICE	SDVQ - 8	CRU_S_Z1	8	700	129	YES	1	6.9	0.48	0.2
VAV - 102	E.H.PRICE	SDVLP - 12	CRU_S_Z2	12	1350	252	YES	1	12.6	0.84	0.2
VAV - 103	E.H.PRICE	SDVQ - 8	CRU_S_Z3	8	340	131	YES	1	2	0.13	0.2
VAV - 104	E.H.PRICE	SDVLP - 10	CORRIDOR_Z4	12	1820	201	YES	1	14.9	1	0.17
VAV - 105	E.H.PRICE	SDVLP - 9	CRU_E_Z5	9	1260	157	YES	1	10.2	0.88	0.17
VAV - 106	E.H.PRICE	SDVLP - 10	CRU_E_Z6	10	600	218	YES	1	3.3	0.22	0.17
VAV - 107	E.H.PRICE	SDVLP - 9	CRU_E_Z7	9	800	145	YES	1	7.1	0.48	0.17
VAV - 108	E.H.PRICE	SDVLP - 10	CRU_E_Z8	10	700	189	YES	1	8.5	0.63	0.17
VAV - 109	E.H.PRICE	SDVQ - 8	CRU_N_Z9	8	390	114	YES	1	5.9	0.4	0.2
VAV - 110	E.H.PRICE	SDVQ - 8	CRU_N_Z10	8	390	114	YES	1	5.9	0.4	0.2
VAV - 111	E.H.PRICE	SDVQ - 8	CRU_N_Z11	8	600	128	YES	1	6.1	0.4	0.2
VAV - 112	E.H.PRICE	SDVLP - 12	CRU_N_Z12	12	560	222	YES	1	3.3	0.22	0.2
VAV - 113	E.H.PRICE	SDVLP - 12	CRU_S_Z13	12	600	231	YES	1	3.5	0.23	0.2
VAV - 114	E.H.PRICE	SDVLP - 10	CRU_S_Z14	10	560	217	YES	1	3.3	0.22	0.17
VAV - 115	E.H.PRICE	SDVLP - 10	CRU_E_Z15	10	480	187	YES	1	2.8	0.19	0.17
VAV - 116	E.H.PRICE	SDVLP - 10	CRU_E_Z16	10	500	191	YES	1	2.8	0.19	0.17
VAV - 117	E.H.PRICE	SDVLP - 12	CRU_E_Z17	12	580	222	YES	1	3.3	0.22	0.17
VAV - 118	E.H.PRICE	SDVLP - 10	CRU_N_Z18	10	480	178	YES	1	2.7	0.18	0.17
VAV - 119	E.H.PRICE	SDVLP - 9	CRU_N_Z19	9	400	145	YES	1	2.2	0.15	0.17
VAV - 120	E.H.PRICE	SDVLP - 12	RESTAURANT_Z20	12	1200	303	YES	1	9.1	0.61	0.2
VAV - 121	E.H.PRICE	SDVLP - 10	RESTAURANT_Z21	10	870	213	YES	1	9.7	0.64	0.17
VAV - 122	E.H.PRICE	SDVLP - 14	RESTAURANT_Z22	14	2660	101	YES	1	6.5	0.44	0.27
VAV - 123	E.H.PRICE	SDVLP - 12	RESTAURANT_Z23	12	1395	122	YES	1	8	0.6	0.2
VAV - 124	E.H.PRICE	SDVLP - 14	RESTAURANT_Z24	14	1260	323	YES	1	10.3	0.68	0.27
VAV - 225	E.H.PRICE	SDVQ - 4	2ND_O1_Z1	4	210	12	YES	1	1	0.12	0.15
VAV - 226	E.H.PRICE	SDVQ - 8	2ND_O_Z2	8	1000	89	YES	1	8	0.53	0.2
VAV - 227	E.H.PRICE	SDVLP - 9	2ND_CORRIDOR_Z3	9	550	154	YES	1	2.5	0.17	0.17
VAV - 228	E.H.PRICE	SDVLP - 10	2ND_LOB_S_Z4	9	1200	102	YES	1	8	0.58	0.17
VAV - 229	E.H.PRICE	SDVQ - 8	2ND_BRK_Z5	8	600	112	YES	1	5.1	0.34	0.17
VAV - 230	E.H.PRICE	SDVQ - 7	2ND_OE_Z6	7	600	61	YES	1	5.8	0.37	0.2
VAV - 231	E.H.PRICE	SDVQ - 4	2ND_O1B_Z7	4	150	10	YES	1	1.8	0.11	0.15
VAV - 232	E.H.PRICE	SDVQ - 4	2ND_O10/7_Z8	4	220	21	YES	1	1.9	0.13	0.15
VAV - 233	E.H.PRICE	SDVQ - 4	2ND_O14/6_Z9	4	220	21	YES	1	1.9	0.13	0.15
VAV - 234	E.H.PRICE	SDVQ - 4	2ND_O12/3_Z10	4	220	21	YES	1	1.9	0.13	0.15
VAV - 235	E.H.PRICE	SDVQ - 4	2ND_O9/11_Z11	4	220	21	YES	1	1.9	0.13	0.15
VAV - 236	E.H.PRICE	SDVQ - 7	2ND_O_N_Z12	7	550	78	YES	1	6.5	0.44	0.2
VAV - 237	E.H.PRICE	SDVQ - 4	2ND_O_Z13	4	110	27	YES	1	1.6	0.12	0.15
VAV - 238	E.H.PRICE	SDVQ - 4	2ND_O8/7_Z14	4	320	25	YES	1	2.1	0.14	0.15
VAV - 239	E.H.PRICE	SDVQ - 4	2ND_O4/5_Z15	4	320	25	YES	1	2.1	0.14	0.15
VAV - 240	E.H.PRICE	SDVQ - 4	2ND_O2/3_Z16	4	320	25	YES	1	2.1	0.14	0.1
VAV - 241	E.H.PRICE	SDVQ - 7	2ND_O_Z17	7	315	80	YES	1	1.3	0.1	0.2
VAV - 242	E.H.PRICE	SDVQ - 7	2ND_O_Z18	7	360	76	YES	1	1.3	0.1	0.2
VAV - 243	E.H.PRICE	SDVQ - 7	2ND_O_Z19	7	400	77	YES	1	1.2	0.1	0.2
VAV - 244	E.H.PRICE	SDVQ - 4	2ND_O_Z20	4	200	39	YES	1	0.8	0.1	0.15
VAV - 245	E.H.PRICE	SDVQ - 7	2ND_O_Z21	7	400	77	YES	1	1.2	0.1	0.2
VAV - 246	E.H.PRICE	SDVQ - 7	2ND_O_Z22	7	100	80	YES	1	1.3	0.1	0.2
VAV - 247	E.H.PRICE	SDVQ - 7	2ND_O_Z23	7	400	76	YES	1	1.3	0.1	0.2
VAV - 248	E.H.PRICE	SDVQ - 7	2ND_O_Z24	7	400	80	YES	1	1.3	0.1	0.2
VAV - 349	E.H.PRICE	SDVLP - 12	3RD_SW_Z1	12	1360	193	YES	1	17.5	1.17	0.28
VAV - 350	E.H.PRICE	SDVLP - 10	3RD_SE_Z2	10	1080	130	YES	1	14.4	0.86	0.17
VAV - 351	E.H.PRICE	SDVLP - 10	3RD_NE_Z3	10	1080	179	YES	1	17	1.13	0.21
VAV - 352	E.H.PRICE	SDVQ - 8	3RD_NE_Z4	8	780	150	YES	1	13	0.87	0.68
VAV - 353	E.H.PRICE	SDVLP - 9	3RD_W_Z5	9	880	133	YES	1	11.2	0.75	0.17
VAV - 354	E.H.PRICE	SDVQ - 4	3RD_CORRIDOR_Z6	4	140	30	YES	1	0.8	0.1	0.15
VAV - 355	E.H.PRICE	SDVQ - 4	3RD_CORRIDOR_Z7	4	220	49	YES	1	0.8	0.1	0.15
VAV - 356	E.H.PRICE	SDVQ - 6	3RD_CORRIDOR_Z8	5	280	64	YES	1	3.9	0.26	0.16
VAV - 457	E.H.PRICE	SDVLP - 10	4TH_SW_Z1	10	1200	193	YES	1	18.2	1.22	0.3
VAV - 458	E.H.PRICE	SDVLP - 9	4TH_SE_Z2	9	840	130	YES	1	13.9	0.92	0.17
VAV - 459	E.H.PRICE	SDVLP - 10	4TH_NE_Z3	10	1100	179	YES	1	18.3	1.22	0.3
VAV - 460	E.H.PRICE	SDVLP - 9	4TH_N_Z4	9	870	150	YES	1	13	0.9	0.17
VAV - 461	E.H.PRICE	SDVLP - 9	4TH_W_Z5	9	800	133	YES	1	12.4	0.83	0.17
VAV - 462	E.H.PRICE	SDVQ - 4	4TH_CORRIDOR_Z6	4	110	30	YES	1	1.3	0.1	0.15
VAV - 463	E.H.PRICE	SDVQ - 4	4TH_CORRIDOR_Z7	4	280	49	YES	1	2.1	0.14	0.15
VAV - 464	E.H.PRICE	SDVQ - 5	4TH_CORRIDOR_Z8	5	320	54	YES	1	4.5	0.3	0.15

STORAGE TANK

TAG	SERVICE	LOCATION	MAKE	MODEL	GALLONS
STH-1	HTG WATER LOOP	4TH FLOOR ROOF	A.O. SMITH	T-80 STD	80
STC-1	CLG WATER LOOP	4TH FLOOR ROOF	A.O. SMITH	T-200 STD	200

RETURN/EXHAUST GRILLS

TAG	MAKE	MODEL	CFM	SIZE
EG - 1	E.H.PRICE	80 - Egg Crate Grille	-	12"x12"
RG - 1	E.H.PRICE	80 - Egg Crate Grille	-	12"x12"

SUPPLY DIFFUSERS

TAG	MAKE	MODEL	CFM	SIZE
SD -1	E.H.PRICE	SCD	-	6" Dia
SD -2	E.H.PRICE	SCD	-	8" Dia

WATER PUMPS

EQ TAG	SERVICE	LOCATION	MAKE	MODEL	ELECTRICAL V/PH/HZ	HP	WATER FLOW (GPM)	TOTAL HEAD (ft)	CONNECTION (in)
P-1	AHU-CRU	HOTEL ROOF	GRUNDFOS	MAGNA3 25-60	230/1/50	57W	17	10	1.5"
P-2	AHU-OFF	HOTEL ROOF	GRUNDFOS	MAGNA3 25-60	230/1/50	48W	14	10	1.25"
P-3	AHU-HTL	HOTEL ROOF	GRUNDFOS	MAGNA3 25-60	230/1/50	57W	17	10	1.5"
P-4	HRV-REST	RESTAURANT ROOF	GRUNDFOS	MAGNA3 25-60	230/1/50	74W	22	10	1.5"
P-5	VAV BOXES	4TH MECH. ROOM	GRUNDFOS	CR 5-4	230/3/50	494W	28	40	2"
P-6	HOT WATER LOOP	4TH MECH. ROOM	GRUNDFOS	CR 32-6	460/3/50	2.95 HP	70	70	2.5"
P-7	HILLED WATER LOC	HOTEL ROOF	GRUNDFOS	TPE 100-160/2-S	460/3/50	2.95 HP	189	15	4"
P-8	BOILER PUMP	4TH MECH. ROOM	GRUNDFOS	MAGNA3 25-80	230/1/50	124W	27	10	2"
P-9	CHILLER PUMP	HOTEL ROOF	GRUNDFOS	MAGNA3 40-60F	230/1/50	150W	46	10	2.5"
P-10	GLYCOL FILL	4TH MECH. ROOM	GRUNDFOS	ALPHA2 L 25-40	230/1/50	16W	7	4	1"

BOILER

TAG	SERVICE	LOCATION	MAKE	MODEL	CAS INPUT (MBH)	CSA OUTPUT (MBH)	ELECTRICAL V/PH/HZ	ELECTRICAL POWER (W)
B-1	ENTIRE BUILDING	4TH MECH ROOM	IBC	SL 80-399	399	382	120/1/60	345

EXHUAST FANS

EQ TAG	SERVICE	LOCATION	MAKE	MODEL	ELECTRICAL V/PH/HZ	HP	AIR FLOW (CFM)	STATIC PRESSURE (wg)
EF - 1	WASHROOMS	RETAIL_L1	GREENNHECK	CSP-A410	230/1/50	120.5W	300	0.375
EF - 2	WASHROOMS	OFFICE_L2	GREENNHECK	CSP-A410	230/1/50	120.5W	300	0.375
EF - 3	WASHROOMS	HOTEL_L3	GREENNHECK	CSP-A410	230/1/50	120.5W	300	0.375
EF - 3A	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3B	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3C	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3D	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3E	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3F	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 3G	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4	WASHROOMS	HOTEL_L4	GREENNHECK	CSP-A410	230/1/50	120.5W	300	0.375
EF - 4A	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4B	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4C	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4D	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4E	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4F	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 4G	Suite WC	HOTEL_L3	GREENNHECK	SP-A110	230/1/50	17.7W	75	0.25
EF - 5A	WASHROOMS	RESTAURANT	GREENNHECK	CSP-A410	230/1/50	120.5W	300	0.375

EXHUAST HOOD

TAG	MAKE	SERVICE	LOCATION	MODEL	HOOD TYPE	CFM	SIZE	ELECTRICAL V/PH/HZ
RH - 1A	GREENHECK	GAS BURNER & FRYER	RESTAURANT	GO/GD1	1	1900	84"X30"	230/1/50
RH - 1B	GREENHECK	STEAM TABLE	RESTAURANT	GO/GD3	2	800	52"X32"	230/1/50
RH - 1C	GREENHECK	GAS GRIDDLES	RESTAURANT	GO/GD4	1	1435	76"X26"	230/1/51
RH - 2	BROAN	RESIDENTIAL KITCHEN	SUITE	BU224WH	-	160	24"X12"	230/1/50

ELECTRICAL BASE BOARD

EQ TAG	SERVICE	LOCATION	MAKE	MODEL	ELECTRICAL V/PH/HZ	POWER
EBB-1A	1st South Stair	1st South Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-1B	1st North Stair	1st North Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-2A	2nd South Stair	2nd South Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-2B	2nd North Stair	2nd North Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-3A	3rd South Stair	3rd South Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-3B	3rd North Stair	3rd North Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-3C	3rd N_WC	3rd N Suite WC	OUELLET	OFM0500	230/1/50	500W
EBB-3D	3rd N_WC_Living	3rd N Suite Living WC	OUELLET	OFM1000	230/1/50	1000W
EBB-3E	3rd_SW_WC_Bed	3rd SW Suite Bed WC	OUELLET	OFM0500	230/1/50	500W
EBB-3F	3rd_SW_WC_Living	3rd SW Suite Living WC	OUELLET	OFM0500	230/1/50	500W
EBB-4A	4th South Stair	4th South Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-4B	4th North Stair	4th North Stair	OUELLET	OFM1000	230/1/50	1000W
EBB-4C	4th N_WC	4th N_WC	OUELLET	OFM0500	230/1/50	500W
EBB-4D	4th N_WC_Living	4th N_WC_Living	OUELLET	OFM1000	230/1/50	1000W
EBB-4E	4th NE_WC	4th NE_WC	OUELLET	OFM0300	230/1/50	300W
EBB-4F	4th SE_WC	4th SE_WC	OUELLET	OFM0300	230/1/50	300W
EBB-4G	4th_SW_WC_Bed	4th_SW_WC_Bed	OUELLET	OFM0500	230/1/50	500W
EBB-4H	4th_SW_WC_Living	4th_SW_WC_Living	OUELLET	OFM0300	230/1/50	300W
EBB-4I	4th_W_WC	4th_W_WC	OUELLET	OFM0300	230/1/50	300W
FFH-1	1st S Vestibute	1st S Vestibute	OUELLET	OACP1500	230/1/50	1000W

APPENDIX F

REV.	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	5/4/2018

ASHRAE®

2018 ASHRAE DESIGN COMPETITION

BCIT

BRITISH COLUMBIA INSTITUTE OF TECHNOLOGY

MECHANICAL SECHEDULES

Project number	BCIT-2018W-1718-08
Date	5/4/2018
Drawn by	S. ALMASYABI
Checked by	A. WANG

M - 3001

Scale	N.T.S.