University of British Columbia

2016 ASHRAE Student Design Competition

Design Calculations

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1 EXECUTIVE SUMMARY

This report has been prepared by students at the University of British Columbia for submission to the 2016 ASHRAE Student Design Competition (Design Calculations). This year's competition requires accurate load calculation and sizing of a Variable Air Volume Air Handling Unit (VAV AHU) HVAC system for a two-story mixed-use office building with approximately 32,000 ft^2 of floor area located in Beijing, China. The Introduction section of this report presents the key parameters that are relevant to the design calculation analysis presented. These key parameters include requirements in the Owner's Project Requirements (OPR) document, climate conditions, building envelope and zoning. The Load Calculation section of the report describes the use of the TRACE 700 software package by Trane Inc. to perform heating and cooling calculations, and presents the load calculation results. The System Selection section provides a detailed description of the system selected based on the load calculation results. The ASHRAE Standards section of the report summarizes compliance of the project outcomes with ASHRAE 55-2013, ASHRAE 62.1-2013 and ASHRAE 90.1-2013 standards. The Plant Selection section of the report outlines the key considerations in defining plant alternatives for providing the selected system with heating and cooling energy. The Life Cycle Cost Analysis section of the report presents the economic analysis that was used to make plant decisions and assess the 50-year life cycle cost against the project budget of 200 USD/ft^2. Lastly, the Energy Conservation Measures section describes selection of photovoltaic array and solar hot water systems in accordance with the OPR. A set of mechanical drawings are appended to this report to serve as a preliminary schematic of the project design.

Analysis of the Beijing weather data revealed that Beijing climate is moist and is classified by Climate Zone 4A. The OPR and building drawings were then used to define building envelope properties for Climate Zone 4A that were in compliance with Standard 189.1. Space by space analysis of the building drawings in conjunction with requirements in the OPR revealed spaces with similar occupancy, lighting, plug loads and temperature requirements could be grouped into thermal zones. A carefully selected set of zones served as the basis of the subsequent load calculation and system selection.

After initial verification of TRACE 700 load calculation results against hand calculations, the thermal zones defined by the design team were input into TRACE 700 for load calculation with the Beijing weather data. The total system peak loads, based on building block loads calculated using TRACE 700 were calculated to be 1,095 MBh (91.3 ton) for cooling and 398.9 MBh for heating. The calculated total loads were satisfied by creating 4 zone groups, each of which provided with its own air handing unit. Two of these zone groups divided the most of the building into a perimeter zone group and a core zone group. Solar heat gain and envelope heat transfer were considerably higher for the perimeter zone group, while the core zone group was dominated by internal heat gains. In addition to the perimeter and core zone groups, the other two zone groups were comprised of the "east wing" zone group and server room zone group. The east wing zone group consists of dispatch center zones with 24/7 operation and occupancy schedules. The server room zone group was also isolated from the rest of the building due to its unique temperature requirements and 24/7 operation schedule. Based on life cycle cost analysis, a 60 ton air-cooled chiller with 486 ton-hours of ice storage was selected as the primary cooling plant for two VAV air handling units serving the perimeter zone group and core zone group respectively. The VAV air handling unit serving the east wing group and the computer room AC unit serving the server room zone group have dedicated DX cooling due to much smaller cooler loads compared to the other two zone groups. The heating plant for all systems consists of three high efficiency natural gas condensing boilers with a total capacity of 400 MBh. The heating plant serves heating coils in zone-level terminals as well as outdoor air pre-heat coils in the air handling units.The ambulance garage air is kept emission free by a directly connected tailpipe exhaust system.

The total final cost for the project is estimated at 4,233,277 USD. This value includes capital, maintenance, and utility costs and incorporates a constant inflation of 3% over the 50-year project life.

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4 INTRODUCTION

4.1 OWNER'S PROJECT REQUIREMENTS

The Owner's Project Requirements (OPR) outline the following general design goals:

> *"Every effort should be made to provide a sustainable design taking into account, energy efficiency, health and safety, occupant comfort, functionality, longevity, flexibility, and serviceability/maintainability. Systems shall be selected based on the lowest possible life cycle cost that includes first cost of materials and long term operating costs"*

Some highlights from the OPR are:

1. Thermal comfort and air quality will be maintained by Variable Air Volume (VAV) Air Handling Units (AHUs).
2. Special consideration must be paid to a 24/7-dispatch center for Emergency Medical Technicians (EMTs).
3. Accommodations must be made for a vehicle garage as part of the dispatch center. A V-8 diesel engine will be run twice a week for 45 minutes.
4. Building must comply with ASHRAE Standards: 55, 62.1, and 90.1.
5. Energy Conservation Measures (ECMs) should be considered.
6. Superior acoustic control in the offices and conference rooms.
7. Design a photovoltaic (PV) array that can support 5% of the total building energy.
8. The interior conditions shown in *Table 1* must be maintained for spaces other than the server room. The server room must be maintained at 73.4°F (23°C) DB (50% RH) all year round.

Summer	73.4°F (23°C) DB (55% RH)
Winter	70°F (21°C) DB

Table 1 – Design Conditions

4.2 CLIMATE CONDITIONS

4.2.1 Climate Zone

ASHRAE climate zones are defined in Standard 189.1 and are based on the number of heating degree days under 50°F (HDD50) and cooling degree days over 65°F (CDD65). Chapter 14 of the ASHRAE Fundamentals Handbook defines climate information for the classification of locations into climate zones. Beijing is located in climate zone 4A based on the criteria for climate zones in ASHRAE 189.1. The number of heating degree days and cooling degree days are summarized in *Table 2* below.

HDD65Beijing	5252
CDD50Beijing	4115
Climate Zone	4A

Table 2 – Climate Zone Classification

4.2.2 Weather

An ASHRAE IWEC[1] weather file was analyzed to better understand the climate in Beijing. The results are shown in *Figure 1 and 2*. Because of the high average absolute humidity during summer months (*Figure 2*), high cooling loads due to dehumidification were anticipated. Additionally, there are large seasonal variations in temperature, with a long cooling season, shown in *Figure 1*. The load calculation methodology, described in Section 5, uses the 99.6% heating design temperature of 12.4°F as well as the 0.4% percent cooling design temperatures of 95°F DB (72°F MCWB) and 80.4°F WB.

[1] International Weather for Energy Calculations

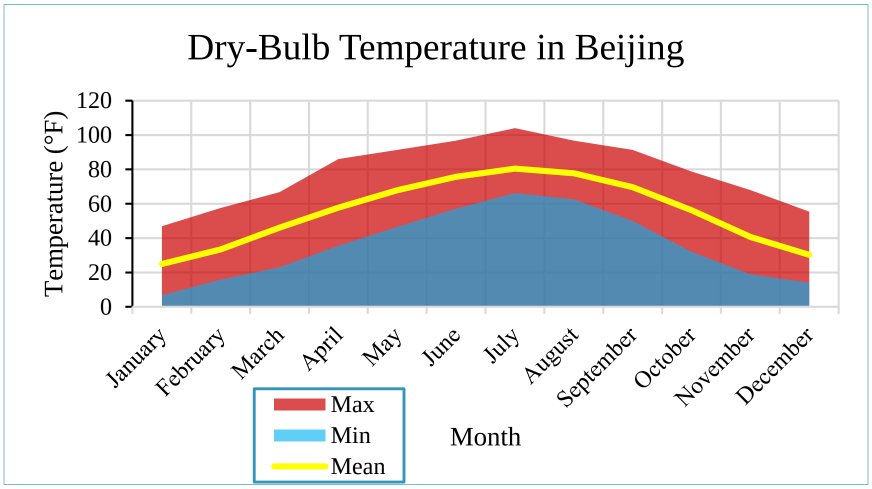


Figure 1 – Monthly Dry-Bulb Temperature Trend for Beijing

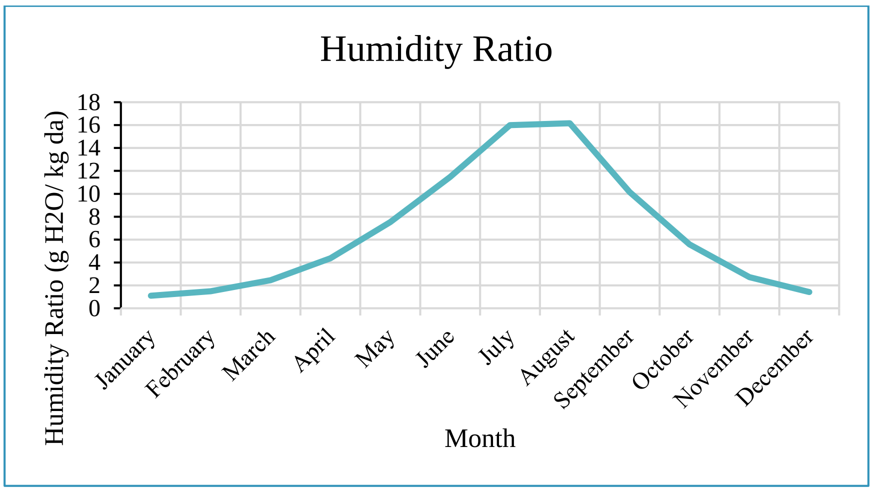


Figure 2 – Monthly Humidity Ratio Trend for Beijing

4.3 Envelope

The maximum envelope U-values for the various ASHRAE climate zones are defined in Appendix E of ASHRAE Standard 189.1 (summarized in *Table 3*). For this competition, it was assumed that the building envelope U-values met the requirements of this standard. However, physically realizable construction assemblies that met these U-values had to be created TRACE 700 and eQuest.

Assembly	**Standard 189.1 U-Value (Btu/hr-ft^2-°F)**	**Assembly Makeup (Exterior → Interior)**	**Calculated U-Value**
Walls	≤ 0.094	Face Brick - 8" Light Weight Concrete – Filled Concrete block	0.0920
Roof	≤ 0.029	Water Proof Membrane – 12" Heavy Weight Concrete – 10" Insulation	0.0280
Windows	≤ 0.380	1/8" Bronze Glass – 1/2" air – 1/8" glass low-e coating	0.350
Doors	≤ 0.450	TRACE 700: 90.1-10 Nonresidential (Zone 1-6) eQuest: Generic door	0.450

Table 3 – Building Envelope Values

4.4 ZONING - OCCUPANCY, PLUG LOADS, AND SCHEDULES

Primary zoning was performed by combining rooms that were assumed to have similar occupancy, plug loads, schedules and solar heat gains. The end result of initial zoning is shown in *Figure 3*. The key implication of creating zone boundaries was to ensure that regions with high perimeter losses could be separated from regions with high internal gains for efficient zone group allocations. The zone group allocations are described in Section 5.2.

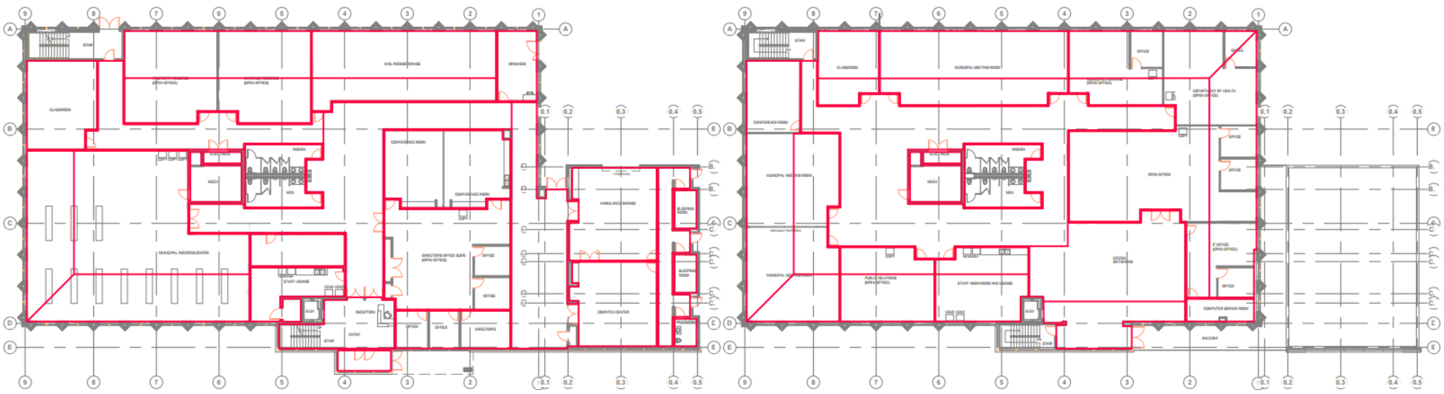


Figure 3 – Zoning – Floor 1 (left) – Floor 2 (right)

5 LOAD CALCULATION

5.1 INTRODUCTION AND VERIFICATION

The design and analysis software TRACE 700 by Trane Inc. was used to calculate heating and cooling loads. Before proceeding with load calculations, the accuracy of any software results needed to be verified. In order to accomplish this, a single room example from ASHRAE Fundamentals 2013 (Chapter 18) was modeled in TRACE 700 and compared with the solutions provided. The RTS methodology (ASHRAE Tables) was selected for cooling load calculations and the UATD methodology was selected for heating load calculations in TRACE 700. A comparison of the solutions and TRACE 700 is shown in *Table 4*. The small differences in results are expected based on the large number of input variables in even a small example.

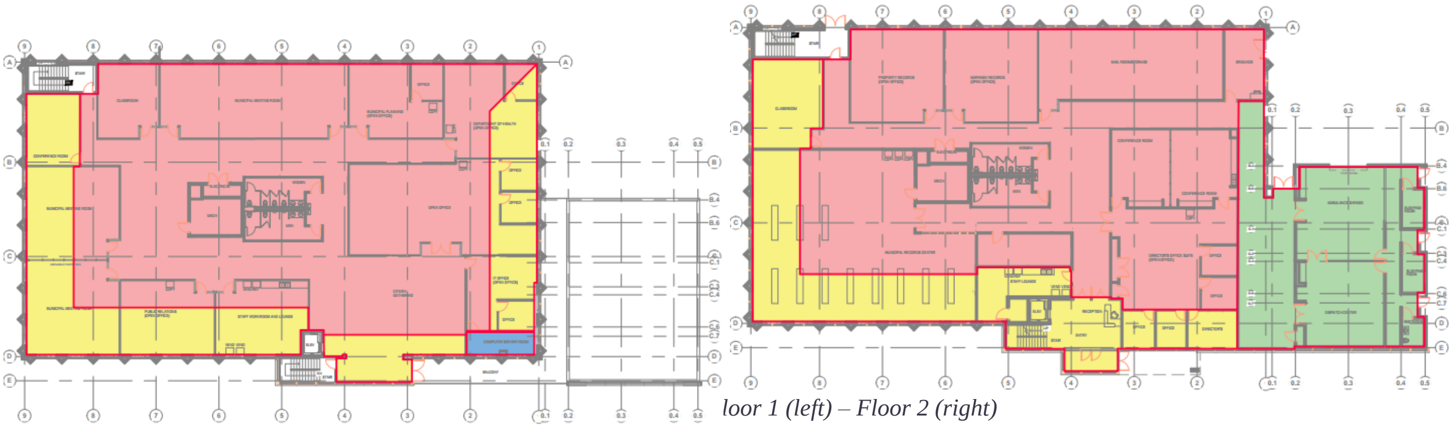
Load	***Example (Btu/h)***	***TRACE 700 (Btu/h)***	***Difference (%)***
Cooling	3675	3812	+3.73%
Heating	4426	4370	-1.27%

Table 4 – Verification Results

5.2 RESULTS

Prior to calculating building loads, the building was divided into 41 thermal zones as described in the (*see* Section 4.4). These zones were then consolidated into 4 different zone groups (*Figure 4*). The perimeter (yellow), core (red), east wing (green) and server room (blue) are served by one AHU each. The perimeter and core classification for zone groups were created to avoid simultaneous heating and cooling demand from one AHU. Inefficient zone groups could cause simultaneous heating and cooling demand from one AHU, forcing the system to use excess energy. The east wing is served by one system since its occupancy schedule is different from the rest of the building. The east wing includes the ambulance dispatch center, which operates 24 hours a day, 7 days a week. A dedicated system for this area allows that system to be smaller and run near its nominal capacity, resulting in energy savings during 24h operation. An independent system for the computer server room was one of the requirements set out by the owner.

Zone area, minimum outdoor air requirements (see Standard 62.1), occupant schedule, wall and window U-Values (Envelope) were added to the model. The results of the TRACE model are presented in *Table 5* and ***Error! Reference source not found.***.



loor 1 (left) – Floor 2 (right)

Zone	Total	O/A	Total	O/A	Cooling Load (MBh)	Heating Load (MBh)
	Heating (CFM)		Cooling (CFM)			
Perimeter	16,340	4,272	4,272	4,272	418.4	156.9
Core	16,569	5,863	5,863	8,446	667.9	229.5
Server Room	4,053	0	4,053	0	79.7	5.8
East Wing	2,437	747	761	747	87.0	33.6
Total	**39,400**	**10,882**	**14,949**	**13,465**	***see comments below**	

Table 5 – Trace Airflow and Load Results

The load results summarized in *Table 5* are the peak block load for the heating and cooling coil serving that system (*the total is omitted because these peaks occur at different times and are not the same as the block load, which was used to size the system). These loads account for system effects such as heat exchanger losses and heat gain fans while the system is in cooling mode. Additionally, plug loads, lighting and fan energy heat gains are excluded from heating loads, resulting in a conservative design.

5.3 Assessment of Conservativeness

To assess the design conservativeness for unexpected demands and small levels of facility expansion, adding a safety factor to coil capacities was initially considered. At the same time, it was kept in mind that grossly oversizing system coils would result in the system operating at lower capacity and possibly lower efficiency all year round. Consequently, a safety factor larger than 10% was determined to negatively affect the humidity control capability of cooling coils. After applying a 10% safety factor to coil capacities, a histogram analysis was performed on the heating and cooling load profiles for each of the four systems in order to assess the benefits of oversizing equipment and coils. TRACE 700's built-in "oversize" functionality was used to increase the coil capacity by a maximum of 10% without affecting the airflows. Then, a year of system simulation was performed, where the system faced all loads including heat gain from lighting and equipment. *Figure 5 and Figure 6* show histograms of the system's heating and cooling load profiles from the simulation. It is evident that after applying an "oversize" factor, there are less than 50 hours in a year where the system heating or cooling load is between 75-100% of design capacity. The exception to this discovery is the server room, where all the cooling hours are within 75-100% of the coil capacity. This is because the primary loads in the server room are constant heat rejection from data equipment. Based on these findings, it was determined that oversizing is an unnecessary measure for all systems due to the inherently conservative design calculation methodologies. These inherently conservative design calculation methodologies neglect many load reducing factors such as equipment heat rejection during heating mode. While there might be a small advantage to oversizing the cooling capacity of the server room system in terms of providing extra capacity for server room upgrades, the relatively small size of the system makes it feasible to consider and analyze the economics of replacing the cooling system to meet any future server room demands.

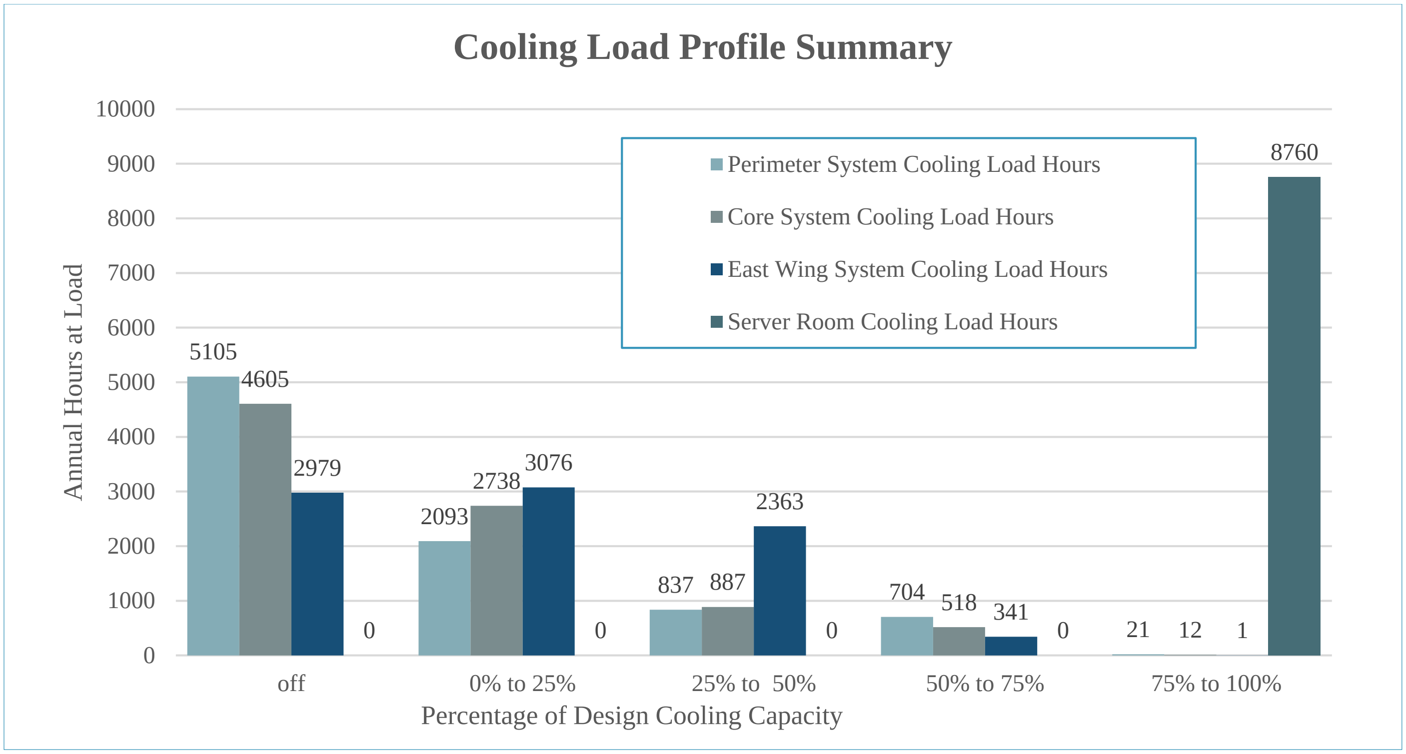


Figure 5 – Cooling Load Profile Summary

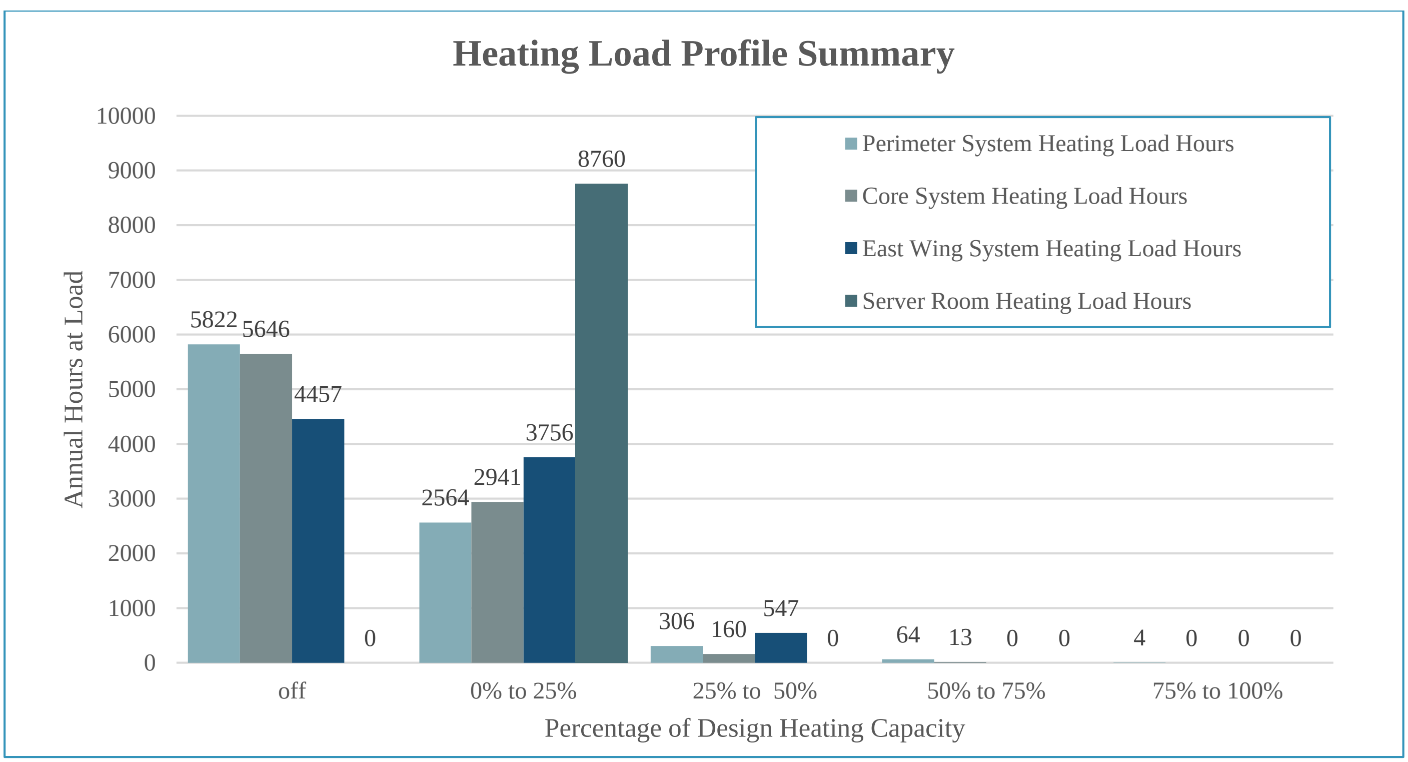


Figure 6 – Heating Load Profile Summary

6 System Selection

6.1 Introduction and Overview

For this project, the Owner required the use of a variable air volume (VAV) air handling unit system. VAV systems come in a number of different configurations to derive energy and control benefits in a large multi-zone system. For example, one VAV system configuration that significantly reduces overall ventilation loads involves the use of a variable air volume dedicated outdoor air system (VAV-DOAS), which delivers the exact amount of required outdoor air to each zone. It is evident from the calculation procedure, summarized in *Section 6.2.3*, that the system ventilation efficiency for a dedicated outdoor air system is 100%, which eliminates the need to over-ventilate certain zones to ensure adequate ventilation in the critical zone. However, this form of a VAV system would require supplemental zone-level heating and cooling equipment such as four-pipe fan-coil units, terminal heat pumps, or radiant panels. The VAV-DOAS is simply a decentralized system for providing conditioned outdoor air to meet space ventilation requirements, rather than space load. In a strict sense, heating and cooling for all spaces is not primarily done by the VAV-DOAS in this configuration. Since the owner has strictly specified the use of VAV air handling units for the building, the primary system configuration choice for this project includes VAV air handling units with central outdoor air pre-heat, central cooling and zone-level heating coils in the VAV terminal units as illustrated in *Figure 7* below.

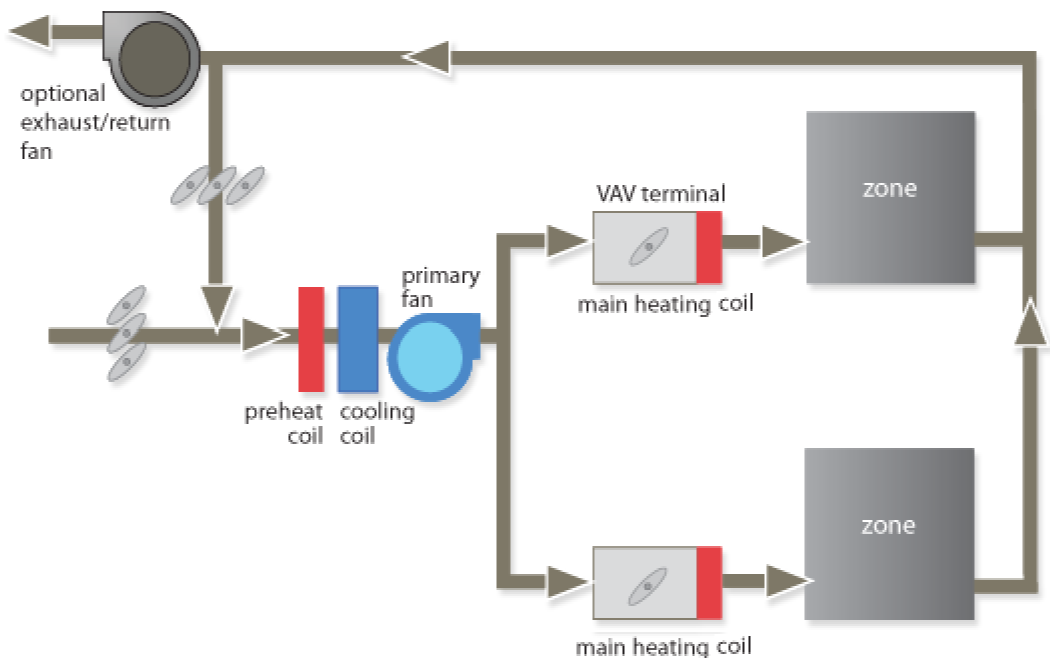


Figure 7 – Schematic of Selected VAV system with zone level reheat

The proposed systems in this building are based on the zoning results summarized in *Section 5.1.2*. Consequently, the selected system configuration includes three main VAV air-handling systems, in addition to a computer room AC-unit and dedicated exhaust system for the ambulance garage. The three VAV air-handling units are described in *Section 7.2* and the computer room AC unit and the ambulance garage exhaust system are described in *Section 7.3*.

6.2 VARIABLE AIR VOLUME AIR HANDLING SYSTEMS

With the exception of spaces such as the server room and ambulance garage that require dedicated systems, all zones presented in *Section 4.4* are served by one of three multi-zone VAV air handling units located on the roof of the building. AHU-1, the perimeter system, serves 13 first and second floor zones with west, south and east exterior wall exposures. AHU-2, the core system, serves 28 first and second floor zones that have either no exterior wall exposure, or a north wall exposure. AHU-3, the east wing system, serves 6 dispatch center zones on the first floor that are expected to have 24/7 occupancy all year. While detailed quantitative performance information for AHU-1, AHU-2, AHU-3 and their sub-systems can be found in the equipment schedules, key features of the system are summarized in the following sections.

6.2.1 Central Cooling

The three VAV-AHUs have central cooling coils that are sized to satisfy the total block cooling loads for their respective zone groups. The total cooling coil capacities for AHU-1, AHU-2 and AHU-3 are 401.5 MBh (33.5 ton), 643.8 MBh (53.7 ton) and 92.6 MBh (7.7 ton) respectively. The high cooling coil capacities are mostly due to the high latent loads from the moist climate. The sensible capacities of the three cooling coils are 270.6 MBh, 375.4 MBh, and 35.1 MBh respectively. To ensure that these cooling coil processes are physically realizable with a reasonable apparatus dew point (ADP) temperature, the cooling coil entering and leaving conditions were plotted on a psychrometric chart. The lines connecting the psychrometric state points were then extended in order to check for intersection with the saturation curve at the approximate ADP temperature. Consequently, the cooling coil process for AHU-1, AHU-2 and AHU-3 were found to be valid with approximate ADP temperatures of 54°F, 51.2°F and 53.6°F respectively.

6.2.2 Outdoor-Air Conditioning and Energy Recovery

While primary heating for the supply air is done in VAV-terminals at the zone-level, AHU-1, 2 and 3 have their own pre-heat coil to condition the required quantity of system-level outdoor air calculated in *Section 7.2.3*. Using the prescriptive path of ASHRAE 90.1-2013 as a basis, exhaust air energy recovery is implemented in each of the three systems using an enthalpy wheel with an effectiveness of at least 76%. While this form of energy recovery satisfies the minimum 50% effectiveness requirement in Section 6.5.6.1 of ASHRAE 90.1-2013, the enthalpy wheel also permits efficient summer dehumidification and winter humidification of the outdoor air stream. Consequently, the capacities of the central outdoor air pre-heat coils can be smaller to include the impact of exhaust air recovery. The capacities are 25.1 MBh, 44.8 MBh and 7.3 MBh for AHU-1, AHU-2 and AHU-3 respectively.

6.2.3 Economizer Controls

Complying with the economizer requirement in Section 6.5.1 of ASHRAE 90.1-2013, all three air-handling units include an air-side economizer with a high set-point of 65°F for fixed dry-bulb temperature control.

6.2.4 Filters

AHU-1, AHU-2 and AHU-3 include a MERV-11[2] filter section upstream of the coils in accordance with particulate matter analysis presented in *Section 6.2.2*.

6.2.5 Fans

The primary supply and return fans include variable frequency drives (VFD) for speed control in response to static pressure measurements at the VAV-terminal zones. Based on pressure drop analysis downstream of the AHUs, the required external static pressures (E.S.P.) for the supply fans in AHU-1, AHU-2 and AHU-3 were determined to be 2.5 in. wg, 2.5 in. wg, and 1.5 in. wg respectively. Similarly, the required E.S.P. for the return fans in AHU-1, AHU-2 and AHU-3 were determined to be 1.25 in. wg, 1.25 in. wg, and 0.75 in. wg respectively. Both the supply and return fans are forward curved housed centrifugal fans that are selected for operating efficiencies greater than 25% to ensure that the fan motor name plate horsepower ratings comply with the limitations in Section 6.5.3.1 of ASHRAE 90.1-2013.

[2] Minimum Efficiency Reporting Value (MERV) is a scale to rate the effectiveness of air filters.

6.2.6 Zone-level Conditioning and Control

The primary supply air heating is done by zone level coils in VAV terminal units sized for the zone's peak load. The controller of the VAV-terminal units and static pressure sensors, as well as zone-level thermostats and humidistats communicate with the central direct digital control (DDC) panel to allow automatic supply air temperature set-point reset, static pressure set-point reset and adjustment of ventilation rates in response to zone-level changes.

6.2.7 Distribution Design and Noise Control

Noise control is an important criterion in designing the distribution system to meet the Owner's requirement of maintaining noise levels below NC 35 level in office spaces and NC 30 in dispatch center spaces. Even though the rooftop outdoor air handling units are located outdoors, the discharge sound pressure levels from fans needed to be considered in the distribution zones close to the system fans. The distribution zone closest to an AHU supply or return fan is separated by 40 feet of ductwork. For all aspect ratios of rectangular ductwork, the attenuation due to duct friction at frequencies above 250 Hz is 0.1 dB/ft. For some aspect ratios, the attenuation at lower frequencies is better than 0.1dB/ft. Consequently, the air handling units have been specified to have discharge sound pressure levels that are no more than 4 dB above the NC 35 curve for AHU-1 and AHU-2, and no more than 4 dB above the NC 30 curve sound pressure levels for AHU-3.

Since the VAV-terminal units introduce noise from air damper turbulence and are located in the ceiling space, both the radiated and discharge noise levels are important. The VAV terminal units were selected using Price All-In-One Engineering software which calculates the radiated and discharge NC levels for the desired terminal size, coils, and range of airflow modulation. The discharge ductwork downstream of all VAV terminal units is specified to have at least 5 ft. of duct lined with standard acoustical liner. Additionally, based on results from the selection software, silencer models with additional attenuation were chosen where the NC 30 and NC 35 levels were exceeded 5 ft. downstream of lined ductwork. Lastly, the square cone supply diffusers, liner slot plenum diffusers and egg-crate return/exhaust grilles used in all zones were selected from the manufacturer's catalog to ensure that the noise performance was better than NC 20 for the maximum airflow, while optimizing the diffuser throw for optimal thermal comfort and zone air distribution. Selecting VAV terminal units, diffusers and grilles that have worst-case noise ratings at or below the desired NC level ensures that the noise criteria is met after duct friction attenuation and decibel addition of all noise sources.

6.3 Computer Server and Data Rooms

The first step in designing the HVAC system for the server room in the building was to determine the loads from the servers. The relevant chapter of the ASHRAE handbook, chapter 19 of *Applications*, suggests using online tools from server manufacturers to estimate the cooling loads from IT equipment. This was the approach taken in this project.

The Dell data center capacity planner was selected since it provides an uncomplicated user interface and cooling load reports. The OPR states that the IT center contains 4 racks of blade servers and two racks of networking equipment. Each blade rack can house 16 servers, for a total of 64 servers. Servers that were mid-range in terms of performance were selected, in order to make the arrangement representative of the median capacity in the marketplace. The blade servers chosen were Dell PowerEdge M630 machines. The total cooling load from the racks is slightly higher than the sum of the individual servers due to the power supply equipment for the rack. The team also assembled two 24-unit racks of network switches, as per the instructions of the OPR. The total cooling load from the IT equipment is 72,600 Btu/h based on information in *Figure 8, Figure 9 and Figure 10.*

These equipment loads were input into TRACE 700. The program then output a total load for the data center that included loads from solar gain, wall and window transmission, ventilation and other sources. The loads are summarized in (*Table 6*).

Loads	Btu/h	Tons
Per Server	771	0.064
Per Server Rack	15,136	1.260
Per Network Rack	6000	0.500
Total Equipment	72,600	6.050
Other Sources	8530	0.710
Grand Total	**81,130**	**6.800**

Table 6 - Summary of Data Center Cooling Loads

PowerEdge M1000E Blade
(Qty:1)

PowerEdge M1000E Blade		
Input Power:	4436watts	15136.3btu/h *
Input Current:		41.2amps
Weight:	159.3kg	351.3lbs

Configuration	
M1000E Enclosure	PowerEdge M1000E Blade Server Chassis

Figure 8 - Output of Dell Data Center Capacity [1 Blade Rack]

PowerEdge M630
(Qty:16)

PowerEdge M630		
Input Power:	226watts	771.1btu/h *
Maximum Potential Power:*	297.8watts	1016.2btu/h *
Input Current:		2.1amps
Weight:	5.4kg	11.9lbs

Figure 9 - Output of Dell Data Center Capacity [Server]

Rack Power

Name	Location (RU)	Weight (lbs)	Thermal (watts)	Amps @ 208 volts	Flow Rate (CFM)
PowerConnectÂ 3424 @ 24	24	7.10	40.00	0.19	
PowerConnectÂ 3424 @ 23	23	7.10	40.00	0.19	
PowerConnectÂ 3424 @ 22	22	7.10	40.00	0.19	
PowerConnectÂ 3424 @ 21	21	7.10	40.00	0.19	

Figure 10 - Output of Dell Data Center Capacity [4 Network Switches]

After the loads had been determined, a system to provide the necessary cooling needed to be selected, while at the same time providing opportunities for energy conservation measures (ECMs). The team settled on the Emerson PET series of computer room air conditioners (CRAC). This system is designed for small data centers and has the option for a waterside economizer to provide free cooling. The device comes in sizes ranging from 5.5kW (1.56 tons) to 19.4kW (5.5 tons). To provide the cooling needed, the team selected one 10.4kW (2.95 ton) machine and one 15.4kW (4.37 ton) machine. This combination meets the loads with a small safety factor.

Table 6.8.1-11 of ASHRAE 90.1-2013 has minimum efficiency requirements for sensible coefficient of performance (SCOP) of cooling units that are placed in server rooms. This unit is the ratio of sensible cooling energy supplied to the space to the electrical input power to the compressor. The relevant values, for air-cooled air conditioning units are presented in *Table 7*.

Net sensible cooling capacity	Minimum SCOP Downflow / Upflow
≤65, 000 Btu/h	2.20 / 1.99
≥65, 000 Btu/h and <240, 000 Btu/h	2.10 / 1.99
≥240,000 Btu/h	1.90 / 1.79

Table 7 - Minimum SCOP from ASHRAE 90.1 (Adapted from Table 6.8.1-11)

Using manufacturer performance data, the team found the SCOPs to be 4.08 and 3.84 for the 2.95 and 4.37 ton machines respectively. Sample calculations can be found in *Appendix A*.

The selected machines have a glycol-cooled condenser. The glycol-cooled rejects heat to an outdoor fluid cooler. The advantage of this configuration is that the glycol cooling loop can also feed the free cooling coil in the CRAC unit for implementation of a water-side economizer.

6.4 Ambulance Garage Vehicle Exhaust System

6.4.1 Overview of Design

This section outlines design and selection of the vehicle exhaust system for the ambulance garage based on the Owner's requirements. Based on the floor plan and expected use information in the OPR, the vehicle exhaust system has been designed to serve one V-8 diesel ambulance that is expected to run indoors for about 45 minutes twice a week. Table 6.5 in ASHRAE 62.1-2013 demands that, "stands where engines are run shall have exhaust systems that directly connect to the engine exhaust and prevent escape of fumes." Accordingly, the Nederman MagnaTrack HS system was selected to provide a versatile track mounted exhaust system that can serve the single ambulance in multiple configurations. This exhaust system is designed for emergency vehicle stations and features a track-mounted hose that can attach to either high-level or low-level tailpipes as seen *in Figure 11*.

Figure 11 - High-level and Low-level Tailpipe Connection

The track ensures that the exhaust system can be connected to the engine exhaust whether the ambulance is driven in forward, or reversed into the ambulance garage. Based on the dimensions of the ambulance garage, a 22.9ft long track was determined to provide adequate coverage for the ambulance parked in multiple orientations. Based on findings that a wide range of ambulance vehicles have low level exhaust, a hose length of 13.1ft and hose diameter of 6.3" was specified to strike an optimal balance between pressure drop and hose coverage. The manufacturer recommends exhaust airflow of 590-750 cfm for heavy vehicles. Based on this airflow and pressure drop data from the manufacturer, the maximum system pressure drop was calculated to be 9 in. w.c. Based on these recommendations, the exhaust fan was sized for 750cfm and

10 in. w.c. of negative pressure. The specified system also includes an available automatic start-stop system that starts the source ventilation as soon as the engine is turned on.

6.4.2 Adequacy and Safety of the System

High concentrations of particulate matter from diesel engine emissions can severely impact health and safety of occupants. Consequently, this section provides a summary of the design review and engineering checks performed on the exhaust system to ensure safety and adequacy.

The first engineering check was to ensure adequacy of the exhaust airflow rate. The owner has specified a test of a standard V8 diesel engine. Upon research, it was found that the primary ambulance fleet in Beijing is operated by Beijing Emergency Medical Center with the newest additions being Mercedes-Benz Vito and Mercedes-Benz Sprinter van models[3]. However, according to Mercedes-Benz China website, these models are not available with a V8 diesel engine and are instead available with smaller naturally aspirated and turbo-charged gasoline and diesel engines[4]. It can be safely assumed that these models will not be the largest engines run within the facility and will be equipped with current technology in emissions reduction. An older model heavy-duty vehicle that is advertised as a “first-class ambulance” by Beijing EMS alongside Mercedes-Benz is the Louis-Chevrolet Ambulance[5] as seen in *Figure 12*.

Figure 12 - Louis-Chevrolet Ambulance in Beijing EMS fleet

This model shares the same platform as the 1996-2002 Chevrolet Express/Savanna model vans in North America. *Figure 13* shows a cut-out from the vehicle feature sheet that lists the available engines.

Engine Stats

Engine	Over/Under 8500 lbs. GVWR	Horsepower @ rpm	Torque (lbs.-ft.) @ rpm
4300 V6 (L35)	All	200 @ 4400	250 @ 2800
5000 V8 (L30)	Under	220 @ 4600	280 @ 2800
5700 V8 (L31)	Under	245 @ 4600	325 @ 2800
5700 V8 HD (L31)	Over	250 @ 4600	330 @ 2800
7400 V8 (L29)	Over	290 @ 4000	410 @ 3200
6.5L Turbo Diesel V8 (L65)	Over	200 @ 3400	385 @ 1800

Chevy Van/Chevy Express ▶ New Feature for 1996.

Figure 13 - 1996 Chevrolet Express Feature Sheet with Engine Stats

[3] http://www.beijing120.com/inf_05_en.asp
[4] http://www.mercedes-benz.com.cn/content/china/mpc/mpc_china_website/enng/home_mpc/van/home/new_vans/showroom.html
[5] http://www.beijing120.com/gui_04_en.asp

The only V8 diesel engine available is the 6.5L (397 cu. in.) Turbo Diesel V8 (L65) Engine. Based on this information and typical parameters for a turbo-diesel engine, the exhaust air flow rate can be determined using the following derived expression[6]:

$$Exhaust\ CFM = \frac{Engine\ Size\ (cu.in.) * RPM}{3456} * \eta_v * \frac{[Exhaust\ Temp.(^oF) + 460]}{540} \qquad \textit{(Equation 15)}$$

The engine is known to have maximum engine speed of 3400 rpm[7]. However, since the engine is run indoors, the indoor idling engine speed is not expected to exceed 1600 rpm, even if a high-idle kit is installed for increased alternator output and/or diagnostic tests[8]. Based on the type of engine, conservative values for volumetric efficiency[9] and exhaust temperature can be taken to be 1.5 and 900^{o}F respectively. These values result in a calculated exhaust flow rate of 694 cfm at 1600 rpm which is below the vehicle exhaust system flow rate. The calculation can be further validated by extrapolating exhaust cfm data for 4 cycle turbo diesel engines published in Ventaire's Engineering Handbook for Vehicle and Welding Fume Removal[10]. As seen in *Figure 14*, the extrapolated data for a 400 cu. in turbo diesel engine running at 1500 rpm appears to be very close to a value of 650 cfm obtained by performing the above calculation at 1500 rpm.

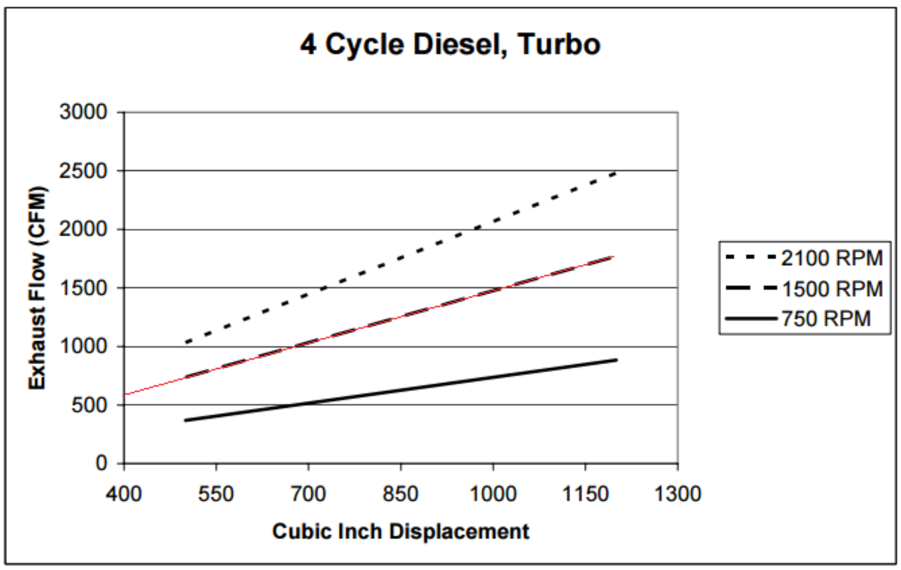


Figure 14 - 4 Cycle Turbo Diesel Engine Exhaust Flow Data (Ventaire, 2006) Extrapolated for Validation

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[6] http://www.asia.donaldson.com/en/exhaust/support/datalibrary/1053747.pdf
[7] http://www.duramaxhub.com/65-diesel.html
[8] http://www.kussmaul.com/pdfs/091-84-gm1.pdf
[9] Ratio of volume of air-fuel mixture drawn into cylinder at atmospheric pressure to volume of the cylinder. This ratio is usually less than 1 for naturally aspirated engines and greater than 1 for engines with forced induction (eg. turbocharger)
[10] http://www.ventaire.com/installs/Yellow%20Engineering%20Handbook.pdf

7 ASHRAE STANDARDS

7.1 STANDARD 55-2013 THERMAL ENVIRONMENTAL CONDITIONS FOR HUMAN OCCUPANCY

The goal of this standard is to, "determine thermal environmental conditions (temperature, humidity, air speed, and radiant effects) in buildings and other spaces that a significant proportion of the occupants will find acceptable at a certain metabolic rate and clothing level." Thermal comfort can be divided into two categories: global and local. Global thermal comfort refers to the occupants' comfort over their entire body and is affected by clothing insulation, air speed and temperature, humidity, radiant temperature and metabolic rate. Local thermal comfort refers to small areas of skin that may be thermally uncomfortable and is mostly affected by radiant asymmetry and drafts in the space.

To test compliance with Standard 55, the team selected a room in the north east corner of the second floor. The test room is representative of most spaces in the building. Hence, results from this space can be reasonably extrapolated to the rest of the building. The following assumptions were made and are summarized in *Table 8*:

- The occupants will wear trousers, a long-sleeve shirt and suit jacket.
- The majority of occupants will be seated while reading, writing and typing (typical office work).
- The HVAC system will maintain design humidity and temperature.
- The graphical method outlined in Standard 55 can be employed to determine occupant comfort.
- Local thermal comfort does not need to be analyzed based on the assumed met and clo levels.
- Operative temperature range for our assumed clothing level can be calculated from *Equation 7.1 and 7.2*.

$$t_{min,Icl} = \frac{\left[(I_{cl} - 0.5clo)t_{min,1.0\ clo} + (1.0clo - I_{cl})t_{min,0.5clo}\right]}{0.5\ clo} \qquad \text{(Equation 7.1)}$$

$$t_{max,Icl} = \frac{\left[(I_{cl} - 0.5clo)t_{max,1.0\ clo} + (1.0clo - I_{cl})t_{max,0.5clo}\right]}{0.5\ clo} \qquad \text{(Equation 7.2)}$$

I_{cl} = clothing value [clo]

$t_{min/max}$ = minimum/maximum temperature [°F]

- Maximum acceptable air velocity in the occupied zone can be calculated from *Equation 7.3*.

$$V = 31375.7 - 857.295t_a + 5.86288\ t_a^2 \qquad \text{(Equation 7.3)}$$

t_a = design temperature [°F]

Model Input	***Value***
Clothing Insulation	0.96 clo
Metabolic Rate	1.0 – 1.1 met
Design Relative Humidity	55%
Indoor Design Temperature	73.4°F
Operative Temperature Range	69.0°F – 75.9°F
Outdoor Heating Design Temperature	12.2°F (99.6%)
Outdoor Cooling Design Temperature	94.9°F (0.4%)

Table 8 – Standard 55 Assumptions

The temperature used to evaluate thermal comfort is called the operative temperature and is calculated using *Equation 7.4*. It is an average of the mean radiant temperature of the surfaces in the space, and the air temperature. The coefficient A corrects for different air speeds within the space. For air speed less than 40ft/min, A=0.5, meaning radiant and convective heat transfer are equally important to thermal comfort. As the air speed climbs, A gets larger, reflecting the growing importance of convective heat transfer.

$$t_o = At_a + (1 - A)t_r \qquad \text{(Equation 7.4)}$$

t_r = mean radiant temperature [°F]

t_a = average air temperature [°F]

A = coefficient representing the ratio of heat transfer due to convection/radiation

The mean radiant temperature of the space's interior surfaces is calculated using *Equation 7.5*. The mean radiant temperature is a single temperature used to represent a, "uniform, black enclosure that exchanges the same amount of heat by radiation with the occupant as the actual enclosure." It is a model used to simplify temperature asymmetry in a room.

$$T_r^4 = T_1^4 F_{p-1} + T_2^4 F_{p-2} + \cdots + T_N^4 F_{P-N} \qquad \text{(Equation 7.5)}$$

T_N = surface temperature of surface N [°F]

F_{p-N} = angle factor between an occupant and surface N [°F]

The results of our analysis are shown in *Table 9 and Figure 15*. In *Figure 15*, the red line represents the relative humidity requirement, the orange line represents the allowable operative temperature range based on our assumed clothing level, and the blue line represents our calculated extreme operative temperatures. If all assumptions are valid, majority of the occupants in this space will find the environment thermally comfortable. The main constraint the assumptions place on the system design is the average air velocity calculated here. This will be used when selecting diffuser location and air distribution performance.

Parameter	***Value***
Operative Temperature (Winter)	71.5 °F
Operative Temperature (Summer)	73.7 °F
Maximum Average Air Velocity	36.87 ft/min

Table 9 – Standard 55 Results

(a)

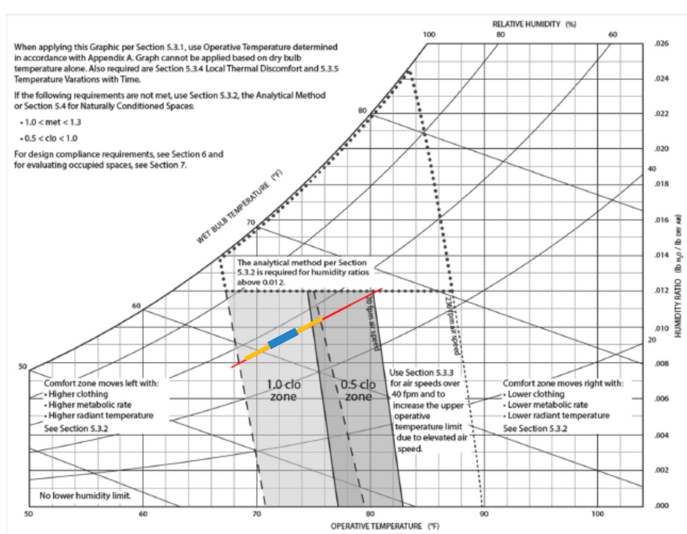

Figure 15 – Standard 55 Results

7.2 STANDARD 62.1-2013 VENTILATION FOR ACCEPTABLE INDOOR AIR QUALITY

The purpose of this standard is to "*specify minimum ventilation rates and other measures intended to provide indoor air quality that is acceptable to human occupants and that minimizes adverse health effects*". The standard is careful to note that acceptable indoor air quality may not be achieved even though the requirements are fulfilled. This could be due to inadequate filtration or poor local outdoor air quality.

7.2.1 Recirculation and Outdoor Air Intake Locations

Standard 62.1 divides air into different classes based on the purpose of the space. Most air in the building is Class 1, and can be recirculated. The exceptions are air in washrooms and garage, which belong to Class 2. Air from these locations cannot be recirculated into Class 1 spaces. The standard also sets out minimum distances between Class 2 exhausts and air intakes. (Table 5.5.1, ASHRAE 62.1-2013).

7.2.2 Particulate Matter

China's 2.5PM standard is 35 $\mu g/m^3$, meaning there are 35µg of particles with a diameter of 2.5µm or less in each m^3 of outdoor air. However, the yearly average reading is much higher than the local government standard[11]. Based on Standard 62.1-2013, a MERV 11 filter or higher is required to maintain clean indoor air. These filters are placed upstream of all cooling coils and other wetted surfaces, through which air is supplied to occupied spaces.

7.2.3 Outdoor Air Requirement

To calculate the minimum outdoor air (OA) requirement, the team used the Ventilation Rate Procedure (VRP). The outdoor airflow rates are determined based on space type, occupancy level and floor area. The procedure is summarized in *Figure 16* and explained here:

1) The minimum breathing zone OA is calculated using *Equation 7.6*. Where the outdoor airflow rate required per person and outdoor airflow rate required per unit area are determined from Table 6.2.2.1 of ASHRAE 62.1-2013. The team used the standard occupant densities for both load calculations and the airflow rate required.
2) The minimum OA per zone is calculated using *Equation 7.7,* based on the minimum breathing zone OA and distribution effectiveness. For heating, when, "the ceiling supply of warm air is 15°F above space temperature and uses a ceiling return," (Table 6.2.2.2, ASHRAE 62.1-2013) the zone air distribution effectiveness is taken to be 0.8. For cooling, zone air distribution effectiveness is taken to be 1.0.
3) The design uses a multiple-zone recirculating system. Hence, the primary OA fraction is calculated using *Equation 7.8*.
4) The uncorrected outdoor air intake is calculated using *Equation 7.9*. Because the "real" occupant diversity is unknown at this time, a value of 1 is used to represent the assumption that the system will always be operating with the default level of occupant density from *Step 1*.
5) Next the system ventilation efficiency needs to be determined. Because the maximum primary OA fraction is larger than 0.55 (Table 6.2.5.2, ASHRAE 62.1-2013) the procedure from Standard 62.1 Appendix A is needed to calculate the system efficiency.
   a) First the average outdoor air fraction is calculated using *Equation 7.10*.
   b) Then the zone level ventilation efficiency is calculated using *Equation 7.11*. Finally, the system ventilation efficiency is determined using the minimum zone level ventilation efficiency served by one system.
6) The total outdoor required air intake is determined calculated using *Equation 7.12*.

[11] http://www.livefrombeijing.com/

Step 1 — Minimum Breathing Zone OA : $V_{bz} = R_p \times P_Z + R_a \times A_Z$

R_p = outdoor airflow rate required per person	P_z = number of people in the ventilation zone	R_a = outdoor airflow rate required per unit area	A_z = zone floor area	**(Equation 7.6)**

Step 2 — Minimum Zone OA: $V_{oz} = V_{bz}/E_z$

E_z = zone air distribution effectiveness				**(Equation 7.7)**

Step 3 — Primary OA Fraction: $Z_{pz} = V_{oz}/V_{pz}$

V_{pz} = zone primary airflow (including outdoor air and recirculated air)				**(Equation 7.8)**

Step 4 — Uncorrected Intake OA : $V_{ou} = D\Sigma_{\text{all zones}}(R_P \times P_Z) + \Sigma_{\text{all zones}}(R_a \times A_z)$

$D = P_s/\Sigma_{all\ zones}\ P_z$ (occupant diversity)	P_s = system population	P_z = zone population		**(Equation 7.9)**

Step 5a — Average OA Fraction: $X_s = V_{ou}/V_{ps}$

V_{ps} = system primary airflow				**(Equation 7.10)**

Step 5b — Zone Ventilation Efficiency: $E_{vz} = 1 + X_s - Z_{pz}$

				(Equation 7.11)

Step 6 — Corrected Intake OA: $V_{ot} = V_{ou}/E_v$

E_v = system ventilation efficiency				**(Equation 7.12)**

Figure 16 – 62.1 VRP Method Flowchart

7.3 STANDARD 90.1-2013 *ENERGY STANDARD FOR BUILDINGS EXCEPT LOW-RISE RESIDENTIAL BUILDINGS*

The purpose of ASHRAE Standard 90.1-2013 is to provide minimum energy-efficiency requirements for the design and construction of new buildings. For a new building, the design must comply with the provisions of Section 5-10 or Section 11 (energy cost budget method) within ASHRAE 90.1-2013. For the purposes of this project, Sections 5, 6, and 9 (Building Envelope, HVAC System and Lighting) within ASHRAE 90.1-2013 are most applicable. Service water heating, power and other equipment are beyond the scope of the analysis performed for the Design Calculations competition.

7.3.1 Building Envelope

Minimum building envelope values are required by ASHRAE Standard 189.1-2014, which supersedes Standard 90.1-2013. Based on the cooling and heating degree days, Beijing is located in climate zone 4. The values listed in Table E-4 (ASHRAE 189.1-2014), under the non-residential column, are used and listed in *Table 3* – Section 4.3.

7.3.2 HVAC System

The cooling for the server room is from a glycol-cooled computer room AC unit (CRAC) as described in Section 6.3. This equipment meets the minimum sensible COP specified in Table 6.8.1-11 of ASHRAE 90.1-2013.

Power Usage Effectiveness (PUE) is the ratio of the data center power usage with cooling power, to the IT equipment energy. ASHRAE-90.1 defines maximum values for this ratio for the various climate zones. ASHRAE-90.1 requires that the PUE be calculated at 100% IT equipment power and 50% IT equipment power. Detailed calculation outlining compliance for the chosen system is shown in Appendix A.

Climate Zone	*PUE Maximum*
4A	*1.36*

Table 10 – 90.1 PUE Requirements

Economizers for comfort cooling are required since the building's cooling load is > 54000 btu/h and is located in climate zone 4A, based on data in Table 6.5.1-1 of ASHRAE 90.1-2013. The high limit shutoff of the air-side economizers is $T_{OA} > 65°\text{F}$, as prescribed in Table 6.5.1.1.3 of ASHRAE 90.1-2013. The server room does not require an economizer (based on table 6.5.1-2), although the unit chosen has a water-side economizer. The economizer only runs when the outdoor temperature is 55°F and below to ensure that the relative humidity does not exceed 50%.

Energy recovery is also required based on the airflow rate and climate zone of the building (Table 6.5.6.1-1, ASHRAE 90.1-2013). The energy recovery systems need at least 50% energy recovery effectiveness (Equation 7.13). Further details of the selected energy recovery device (enthalpy wheel) can be found in Section 6.2.2.

$$h_{OA,1} - h_{OA,2} = 0.5(h_{OA} - h_{RA}) \quad \textit{(Equation 7.13)}$$

$$h_{OA,1} = outdoor\ air\ enthalpy\ entering\ energy\ recovery\ system$$

$$h_{OA,2} = outdoor\ air\ enthalpy\ leaving\ energy\ recovery\ system$$

$$h_{RA} = \ return\ air\ enthalpy$$

Based on manufacturer data, the enthalpy wheels used in the VAV air handling units have a total effectiveness value greater than 76%.

7.3.3 Lighting

Lighting power densities are taken from Table 9.6.1 of ASHRAE 90.1-2013. The space-by-space method was followed and the appropriate lighting power densities for different space types were used to calculate the lighting load.

8 PLANT SELECTION

Once the block loads for each of the four main systems were determined using TRACE 700, the plant selection process started by seeking answers to two questions:

- What is the most desirable energy source and working fluid required for the central heating and cooling plant?
- How many central heating and plants are required for the building?

For the first question, the two main contenders were electricity and natural gas purchased from the utility provider at the rate structure provided by the competition. The most common utility source for cooling around the world is electricity to power the vapour-compression refrigeration cycle in a direct expansion (DX) plant or chiller. Consequently, economies of scale have led to a reduction in the relative first cost of electrically powered cooling equipment. Additionally, performance requirements for electrically powered equipment and standard AHRI tests are readily available from Table 6.8.1-1 to Table 6.8.1-13 in ASHRAE 90.1-2013. These factors established electricity as the desired utility for cooling energy.

Selecting the desired utility for heating energy was not as straight-forward as cooling energy. Using electricity as an energy input, the approximate coefficients of performance (COP) range from 1 for an electric resistance heater, to 3 for an air-source heat pump, and 5 for a ground-source heat pump, assuming air and ground temperatures of 32°F and 50°F respectively. While electric heat pumps are energy efficient, the utility rate structures provided by the competition included significantly lower rates for natural gas. The team compared the cost for one kWh of energy from gas and electricity. The gas consumption rate at the beginning of the project is 0.0263 USD/kWh while electricity has an off-peak and on-peak consumption rate of 0.0850 USD/kWh and 0.1615 USD/kWh respectively. Furthermore, the peak electric demand during a billing period is subject to an additional charge of 9.75USD/kW. The on-peak period is from 9am-7pm, Monday through Saturday. Additionally, the electricity rates will inflate by 3.5%/yr, while natural gas rise at 3%/yr. Using the energy modelling results from TRACE 700, the annual building heating energy demand was determined to be 40,583 kWh, with 21.7% of the demand during on-peak hours. Due to the uneven rate structures, preliminary analysis involved fixing the efficiency of the natural gas plant at 90% and varying the COP of electric heating affected the incremental demand (in kW) and electrical energy consumption (in kWh).

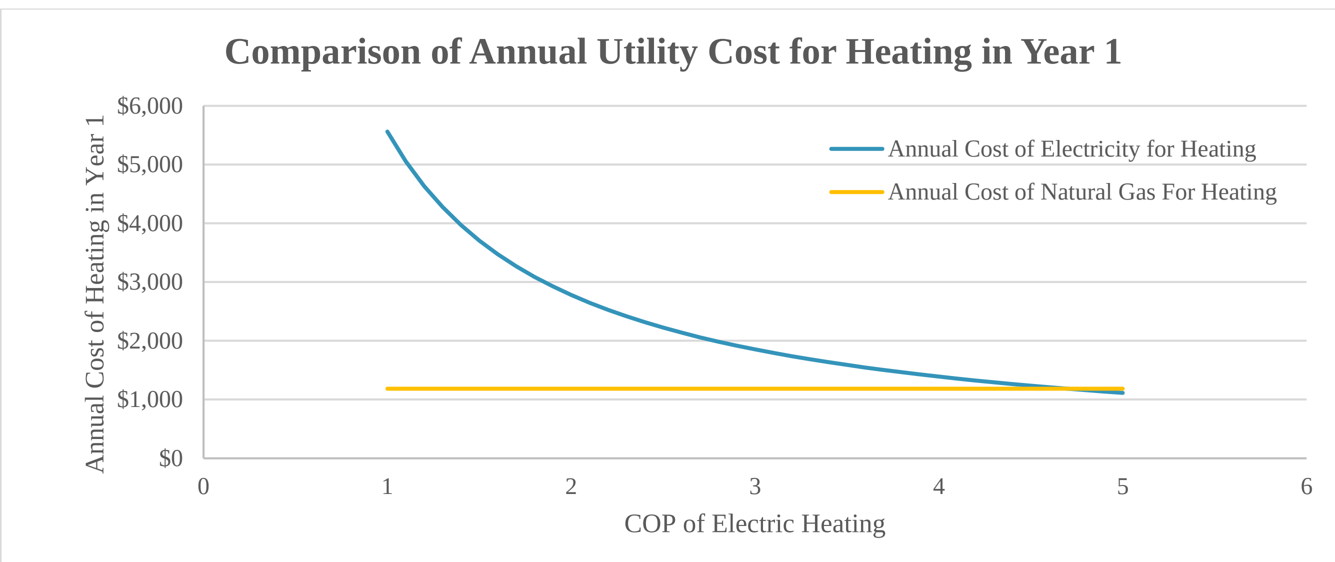

Figure 17 – Comparison of Annual Utility Cost for Heating in Year 1

As seen in *Figure 17*, results from the preliminary analysis indicate that the benefits from higher energy efficiency are not realized with electric-source heating when the COP is below 4.5. Hence, natural gas heating with high efficiency boiler was selected as the heating plant in the primary plant alternative. The high energy efficiency of electrically powered air-source heat pumps, electric-source heating as the heating plant in the secondary plant alternative.

Once the energy source was selected, the number of central heating and cooling plants required for the building needed to be determined. TRACE calculated a building block load of 1,095 MBh (91.3 ton) for cooling and 398.9 MBh for heating. Given the different occupancy and usage schedule for the dispatch center compared to the remaining office spaces, the plants for the dispatch center were isolated from the rest of the system. With a chilled water plant, it was found to be more feasible to have a single cooling plant for AHU-1 and 2. However, with a DX plant, limits of maximum refrigerant piping lengths made it more feasible to have separate condensing units for AHU-1 and 2 that could be combined with the air handling unit.

With the above considerations, the two plant alternatives were prepared for energy modelling and life cycle cost analysis as described in further detail in *Section 8.1 and 8.2*.

8.1 Alternative 1 - Air-cooled Chiller w/ Ice storage and Natural Gas Boilers

In the first alternative, a single air-cooled chiller was included to meet the block load for AHU-1 and AHU-2. From the rate structures discussed, it was evident that thermal ice storage, as shown in *Figure 18*, is a lucrative option to reduce utility costs. By generating ice during off peak hours, much of the cooling load during on-peak hours can be met by melting the ice. The thermal ice storage system was modelled to store 486 ton-hours of energy, which is between a third and quarter of the cooling energy required on the peak cooling design day. During on-peak time, the chiller was modelled to supplement any additional demands that could not be met by the ice-melt. This modelling configuration allowed the chiller to be downsized by 75% from 79.7 tons to 60 tons. Meanwhile, AHU-3 and the server room unit were provided with their own dedicated unitary DX cooling plant due to their smaller block loads and 24/7 occupancy schedules. A single 296 MBh heating plant with two high efficiency condensing natural gas boilers was selected for the zones served by AHU-1 and AHU-2.. A secondary 31 MBh heating plant with a single high efficiency condensing natural gas boiler was included to serve the dispatch center zones with 24/7 occupancy and to provide backup heating capacity in the server room. In this alternative, the condensing unit for the server room AC unit is glycol cooled. This configuration allows the condenser heat to be used for preheating the outdoor air in AHU-1 and AHU-2 before rejecting the heat via a fluid cooler. This configuration also allows the glycol loop to serve as a water-side economizer with a secondary free cooling coil in the computer room AC unit. Since an air-side economizer impacts the relative humidity, a water-side economizer is desirable for the computer room.

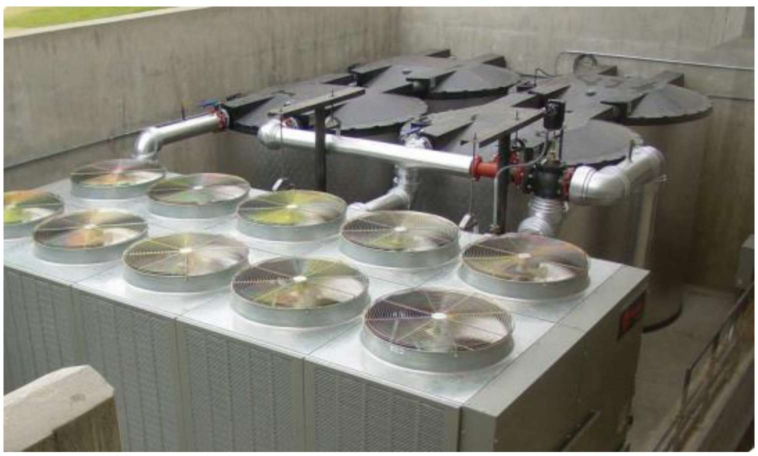

Figure 18 – Air-cooled chillers with thermal ice storage tanks

8.2 Alternative 2 - Central DX-Cooling and Zone-Level Air-Source Heat Pump

In the second alternative, DX cooling plants were grouped with the electric-source heating option due to the similarity of the heating and cooling plants. In this alternative, each system had its own condensing unit to serve the respective block load. For the heating plant, each zone was equipped with an air source heat pump. In this alternative, water-side economizers were implemented in the same way as Alternative 1, since air-side economizer operation is not desirable for a computer room with narrow relative humidity requirements.

Life cycle costing can be performed over the specified project life of 50 years, with inputs from the utility rates and energy model. The plant selection decision in this project is largely dependent on the life cycle cost analysis criteria to provide the owner with the most cost effective design. Life cycle cost analysis is discussed in detail in *Section 9*.

9 Life Cycle Cost Analysis

In general, a life cycle cost analysis is based on the capital cost of the equipment, the utilities and maintenance cost over the life of the equipment, and the system replacement cost, should the equipment fail over the lifespan. For this project, the analysis included the cost of the initial system, utilities, and maintenance. The life cycle cost compared the two alternatives discussed in *Section 8*.

The budget for this project is 200 USD/ft^2. With a building area of 32,312 ft^2 (3001.88 m^2) the total budget is 6,462,400 USD. The capital cost is an estimate based on guidance from an industry liaison. The initial cost of is 999,500 USD for alternative 1, and 669,000 USD for alternative 2. Alternative 1's capital cost is 15.45% of the total budget, and alternative 2 is 10.35%. *Table 1314 in Appendix A* shows the detailed cost breakdown for both alternatives. *Figure 19* shows the annual cost comparison for the two alternatives.

Project Information

Location
Project Name
User
Company
Comments

Study Life: 50 years
Cost of Capital: 4 %
Alternative 1: Chiller+Boiler
Alternative 2: Air Source Heat Pump

Economic Comparison of Alternatives

	Yearly Savings ($)	First Cost Difference ($)	Cumulative Cash Flow Difference ($)	Simple Payback (yrs.)	Net Present Value ($)	Life Cycle Payback (yrs.)	Internal Rate of Return (%)	Life Cycle Cost Difference
Alt 1 vs Alt 2	11,341	330,500	1,408,849	29.1	218,659	30.0	6.3	218,658.50

Annual Operating Costs

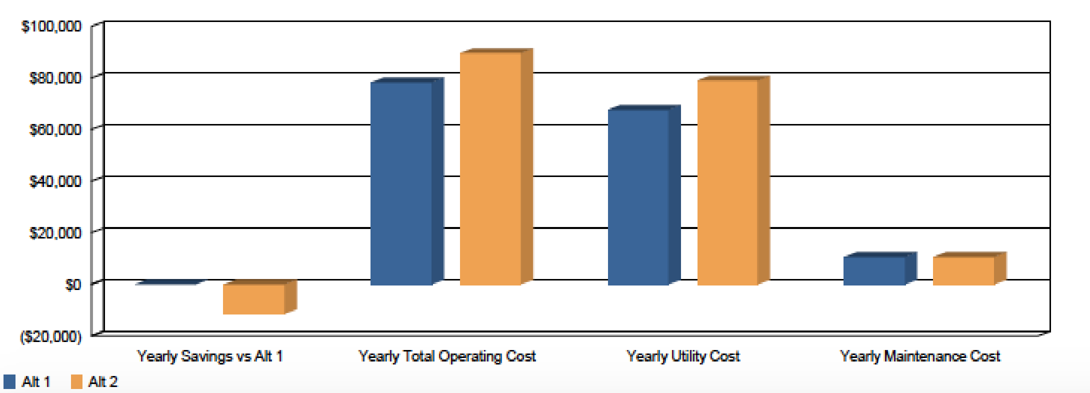

Figure 19 – System Cost Comparison – Annual

The utility escalation rate is based on the ten-year average increase for utilities in the United States, which is given in the OPR. Fluctuations in the utility price during the day are also accounted for in the analysis. The total utility cost for alternative 1 is 780,712 USD/yr, and 842,366 USD/yr for alternative 2. *Figure 20* shows the cost difference for both alternatives month by month based on hourly energy modelling simulations in TRACE 700.

	Yearly Savings vs Alt 1	Yearly Total Operating Cost ($)	Yearly Utility Cost ($)	Yearly Maintenance Cost ($)	Plant kWh/ton-hr
Alt 1	0	78,322	67,659	10,663	0.600
Alt 2	-11,341	89,663	79,000	10,663	0.337

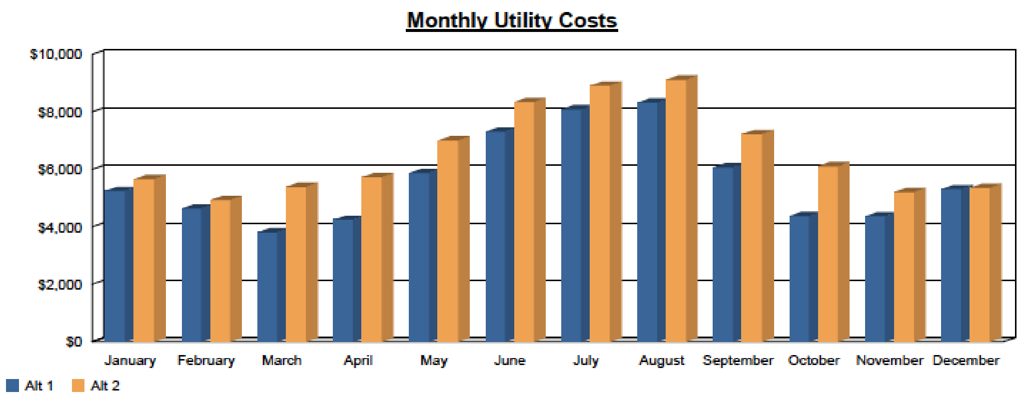

Figure 20 – System Cost Comparison – Monthly

The maintenance costs for both alternatives are the same, 10,663 USD/yr based on mean $/ft^2 maintenance cost for similar 2-story buildings in the ASHRAE Service Life and Maintenance Cost Database[12]. The total utility and maintenance cost for alternative 1 is 78,322 USD/yr, and 89,663 USD/yr respectively. Although the initial cost for alternative 1 is higher, the operating cost is much lower compared with alternative 2. Additionally, alternative 1 is less energy efficient, as can be seen from the kWh/ton-hr metric in *Figure 20*. A large source of savings in alternative 1 is the thermal storage. This allows the system to create ice at night, when electricity is cheaper, and then use this ice for cooling during the day. Additional savings come from the low cost of natural gas. The annual costs of alternative 1 are 11,341 USD less than those of alternative 2. Based on the economic analysis, alternative 1 is the more attractive plant.

To complete the economic analysis, the net present values for both alternatives over the 50-year life of the system was compared, with a 3% inflation rate. The net present value of alternative 1 is 218,658 USD more than alternative 2 over the required service life. The comparison of operation cost and net present value of the two alternatives both indicate that alternative 1 will cost less than alternative 2 over a 50 year project life.

10 Energy Conservation Measures

10.1 Photovoltaic Array

The OPR requests that a photovoltaic (PV) panel array be sized in order to cover 5% of the building's total electrical demand. Based on energy modelling results, 5% of the total electricity requirement of the building, is 26170 kWh annually. The Suniva[13] OPT 285-60-4-100 Silver Mono Solar Panel rated at 285W was selected based on its relatively high module efficiency of 17.34%. To provide 5% of the total electricity requirement of the building, 61 modules would be required as per the detailed calculations in *Section 14.1.2*. The total cost of 61 modules of the selected model is 19,520 USD. The photovoltaic array panels are located on the roof, covering an area of 1073 ft^2.

[12] http://xp20.ashrae.org/publicdatabase
[13] http://www.wholesalesolar.com/1524436/suniva/solar-panels/suniva-opt285-60-4-100-silver-mono-solar-panel

10.2 SOLAR HOT WATER

The viability of a solar hot water system was considered on a life cycle cost basis using RETScreen. The solar collector used for calculation is a Thermo Dynamics Ltd. G32-P. Full technical data can be found in Appendix B.

Hot water consumption was assumed to be 1 gal/person-day based on data from ASHRAE *Applications*. It is difficult to predict the exact occupancy for this building. Therefore, a range of values was used in this analysis. The TRACE 700 and eQuest models assumed peak occupancy of approximately 600. In reality, the average occupancy would be lower. A value of 300 occupants was used as the upper boundary for this analysis. This assumes the building operates at 50% peak capacity on average. This is a conservative estimate when one considers a substantial contribution of the building's peak occupancy comes from gathering and meeting spaces. All of these spaces being at capacity every day of the month is unlikely. A value of 100 occupants was used as the lower boundary. When the real domestic hot water usage can be determined, a more accurate analysis can be performed. *Table 1112* summarizes assumptions and *Table 1213* summarizes results.

ASSUMPTIONS	***HIGH OCCUPANCY***		***LOW OCCUPANCY***	
Case	**Base**	**Proposed**	**Base**	**Proposed**
Occupancy	300		100	
Hot Water Use	1.0 gal/person-day			
Energy Source	Natural Gas			
Energy Prices	7.70 USD/mmBtu			
Annual Fuel Requirements [mmBtu]	**113.9**	**1.5**	**38.0**	**5.3**

Table 11 – Solar Hot Water Analysis Assumptions

For the high occupancy case, nearly all of the building's domestic hot water needs can be met and a simple payback period of 5.8 years can be achieved with an array of 7 collectors (31.1 ft^2 ea.). The ecological benefit of the system is a reduction of greenhouse gas emissions of approximately 5.9 tCO_2. The lower occupancy limit is an attempt to bracket the real value. If the real value is within the boundaries, the investment will still have a payback period of 8.4 – 8.5 years, a rate of return of 15.1 – 15.3% and a reduction in greenhouse gas emissions of 1.7 – 5.9 tCO_2.

RESULT	***HIGH***	***LOW***
Number of Panels Required	7	2
Collector Area	31.1 ft^2	
Total Collector Area	217.7 ft^2	62.22 ft^2
Inflation Rate	3.0 %	
Project Life	50 yr	
Initial Cost	**7,364 USD**	**2,104 USD**
Annual Energy Savings	**877 USD/yr**	**292 USD/yr**
Simple Payback Period	**8.5 yr**	**8.4 yr**
Internal Rate of Return	**15.1%**	**15.3%**
GHG Emission Reduction	**5.9 tCO_2**	**1.7 tCO_2**

Table 12 – Solar Hot Water Analysis Results

Table 1112 shows that both proposed cases have some annual fuel requirements for heating energy. This is because the solar panel capacity is undersized. An additional panel would result in unused heating power. In a more detailed design, this heat could be recovered elsewhere in the system and wouldn't be wasted, but for the purposes of this project, the solar hot water system is decoupled from the HVAC system. Additionally, this model assumes a panel life of 50 years with installation cost included. If the panels needed to be replaced or repaired, it would make the project less attractive. The team recommends domestic hot water use data be gathered (in the real building) before calculating the cost-benefit of a solar hot water system.

11 Conclusion

The total system peak loads, based on building block loads calculated using TRACE 700, are 1,095 MBh (91.3 ton) for cooling and 398.9 MBh for heating. The total system airflow capacity is 39,400 cfm. Air-cooled chillers with ice storage and natural gas boilers serve 3 AHUs that feed 41 VAV terminal boxes. An independent computer room air conditioner maintains server room environmental requirements. The ambulance garage air is kept emission free by a directly connected tailpipe exhaust system. Photovoltaic and solar thermal panels have been sized and selected for the project as energy conservation measures in accordance with the OPR.

The air-cooled chillers with ice storage and natural gas boilers were chosen based on life cycle cost analysis. Compared to a central DX cooling plant, and an air source heat pump, the system is 11,341 USD cheaper considering only capital, utility and maintenance cost. By adding inflation and depreciation over a 50 year life, the chosen alternative's net present value is 218,658 USD higher than the competing alternatives. A major difference is the incorporation of the highly cost effective ice storage system. This system allows the cooling plant to be downsized while taking advantage of cheaper off-peak energy costs. The life cycle cost of the project is estimated at 4,233,277 USD over a 50 year project life. This value includes capital, maintenance, and utility costs and incorporates a constant inflation of 3%. The estimated life cycle cost translates to approximately 131 USD/ft^2, which is well below the project budget of 200 USD/ft^2.

Throughout the load calculations and system design, the team complied with ASHRAE Standards 55-2013, 62.1-2013, and 90.1-2013. Thermal comfort was tested by modeling a single occupant in a room with wall temperature asymmetry. Using conservative assumptions and estimates, it was determined the majority of occupants should be thermally comfortable. ASHRAE-62.1-2013 fresh air requirements were included in TRACE 700 while calculating building loads. Airflow requirements were then manually calculated in order to confirm the TRACE 700 outputs, which showed that software and hand calculation results were within 8% of each other. The prescriptive path method from Standard 90.1 was used for this project. *Section 6* has references to system selection choices that were made in accordance with the standard.

12 Acknowledgements

The design team would like to thank Dr. Nima Atabaki (Senior Instructor - UBC), Dr. Steven Rogak (Professor - UBC), and Alireza Nazari (Principal – Integral Group) for providing invaluable guidance, technical reviews and recommendations at various stages in the project. The design team would also like to thank Branislav Cvijetinovic (Mechanical EIT – Prism Engineering) for providing design feedback and industry knowledge.

13 References

ASHRAE. (2013). *ASHRAE Handbook: Applications.* Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc.

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14 Appendix A

14.1 Calculations

14.1.1 Server Room 90.1 Compliance

$$SCOP = \frac{Q_{sens}}{W_{input}}$$

$$SCOP_{2.95ton} = \frac{9.8kW}{2.4kW}$$

$$\boldsymbol{SCOP_{2.95ton} = 4.08}$$

$$SCOP_{4.37ton} = \frac{12.3kW}{3.2kW}$$

$$\boldsymbol{SCOP_{4.37ton} = 3.84}$$

PET series								
A-air cooling W-water cooling G-Glycol cooled								
Rated Unit Performance R-22 and R-407C R-410A								
PET		6	8	10	12	16	18	20
Cooling capacity	kw	5.5	7.8	10.3	12.5	15.4	17.5	19.4
Sensible capacity	kw	5.3	7.6	9.8	12.3	14.9	17.1	18.3
Sound pressure level	Db(A)	49	49	48	50	51	53	53
Scroll compressor	n	1						
Input power	kw	1.5	1.9	2.4	3.2	4.1	4.7	5.8

Figure 21 - Emerson PET series performance data

$$PUE_{0,100\%} = \frac{Computer\ Room\ Power\ (CRP)_{100\%}}{IT\ Equipment\ Power\ (ITP)_{100\%}}$$

$$CRP_{100\%} = \frac{q_{cool,100\%}}{COP_{avg}} + ITP_{100\%}$$

$$CRP_{100\%} = \frac{25.7kW}{\left(\frac{2.95ton}{7.32ton}\right)4.08 + \left(\frac{4.32}{7.32}\right)3.84} + 21.28kW$$

$$CRP_{100\%} = 27.85kW$$

$$\boldsymbol{PUE_{0,100\%}} = \frac{27.85}{21.28} = \mathbf{1.31}$$

$$CRP_{50\%} = \frac{q_{cool,50\%}}{COP_{avg}} + ITP_{50\%}$$

$$CRP_{50\%} = \frac{15.06kW}{3.91} + 10.64kW$$

$$CRP_{50\%} = 14.49kW$$

$$\boldsymbol{PUE_{0,50\%}} = \frac{14.49}{10.64} = \mathbf{1.36}$$

14.1.2 Photovoltaic Array

$$\text{Overall Annual electrical Load} = 523411 \text{ kWh}$$

$$5\% \ of \ the \ overall \ load = 523411kWh \ \cdot 5\% = 26170.55kWh$$

$$Total \ annual \ irradiance \ incident \ on \ horizontal \ surface = 140.65 \frac{kWh}{ft^2}$$

PV selection: Suniva OPT 285-60-4-100 Silver Mono Solar Panel with 285W output

Output Watts: 285W

Price: $320

Module Efficiency = 17.34%

$$PV \ array \ horizontal \ area \ required = \frac{26170.55 \ kWh}{0.1734 * 140.65 \ kWh/ft^2} = 1073 \ ft^2$$

$$60 - cell \ module \ area = 17.69 \ ft^2$$

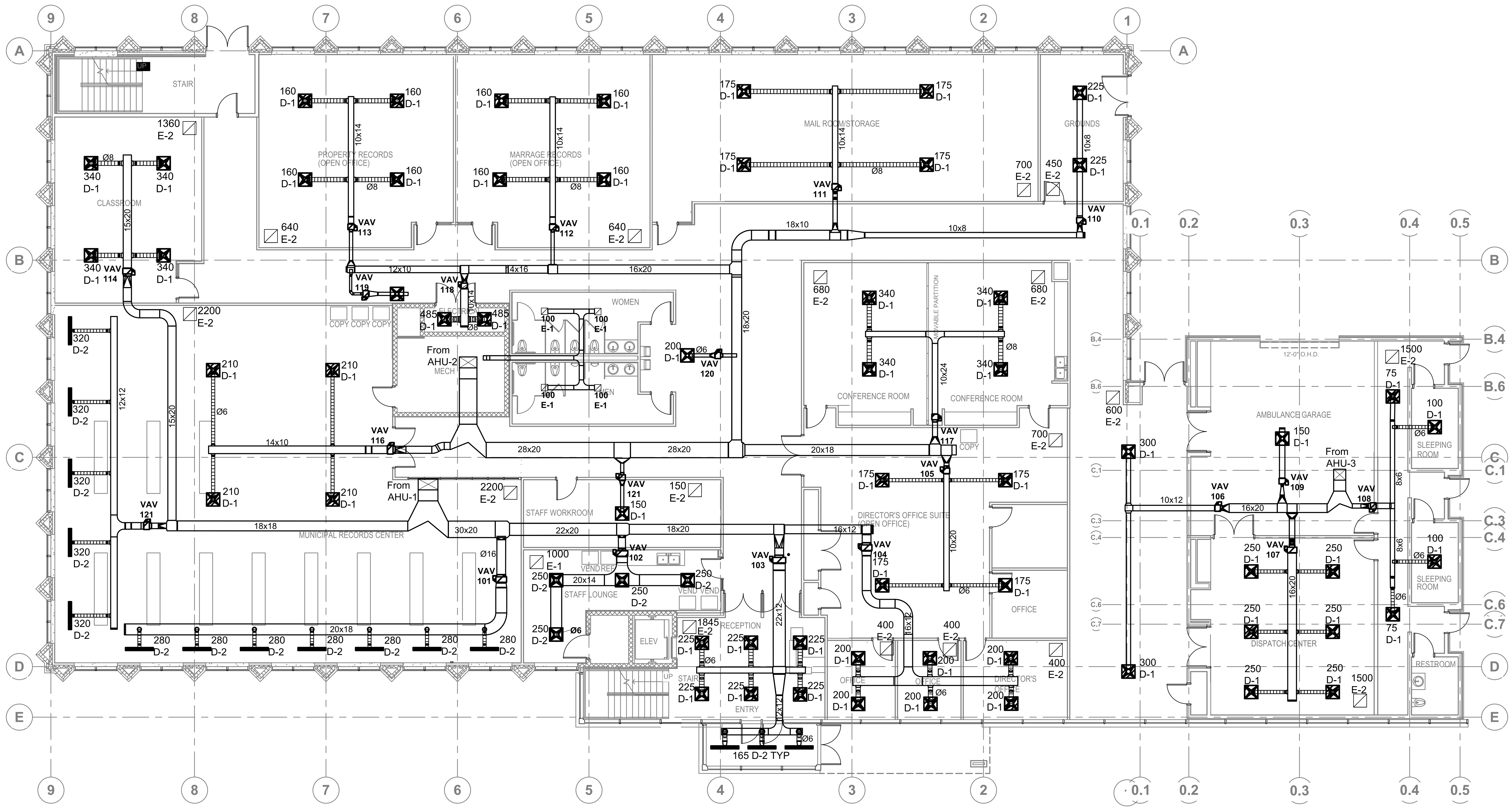
$$Number \ of \ 60 - cell \ PV \ array \ modules \ = \frac{1073ft^2}{17.69 \ ft^2} \approx 61modules$$

$$Total \ Cost = 61 \ modules * \$320 = \$19,520$$

14.2 Tables

Equipment Type		Alternative 1	Alternative 2
AHU	AHU-1	90,000	90,000
	AHU-2	90,000	90,000
	AHU-3	25,000	25,000
	AHU-4	25,000	25,000
Air Cooled HR Chiller w/ Ice Making	CH-1 and CH-2 (~88.8 ton total, outdoor)	300,000	
Air Cooled Condensing Unit/ Air Source Heat Pump	HP-1 (35.1 tons)		35,000
	HP-2 (61.2 tons)		60,000
	HP-3/CU-3 (8 tons)	7,000	12,000
	HP-4 /CU-4 (8 tons)	7,000	12,000
Boiler	B-1 (248.6 MBh)	12,000	
	B-2 (248.6 MBh)	12,000	
	B-3 (40 MBh)	2,500	
Duct Work	Take off from drawings	115,000	115,000
Piping		110,000	55,000
Thermal Ice Storage	486 ton-hrs	100,000	
VAV Boxes & Diffusers	41 VAV boxes @ $800ea 200 grilles & diffusers @ $100ea	60,000	60,000
Pumps		19,000	9,500
Controls, Misc. Equipment, Exh Fans, DHW Tanks, Exp Tank		25,000	25,000
Backup Heating Plant, Heat Pump Implementation (Electric Resistance)			55,500
Total Cost		**999,500**	**669,000**

Table 13 – System Alternative Cost Summary



1 MECHANICAL FLOOR PLAN - LEVEL 1
SCALE: 1/8" = 1'-0"

REV	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	2016-05-04

ASHRAE
SENIOR DESIGN COMPETITION

ASHRAE University of British Columbia Student Branch

UBC

UNIVERSITY OF BRITISH COLUMBIA
DEPARTMENT OF MECHANICAL ENGINEERING
2054-6250 APPLIED SCIENCE LANE
VANCOUVER, BC CANADA V6T 1ZR

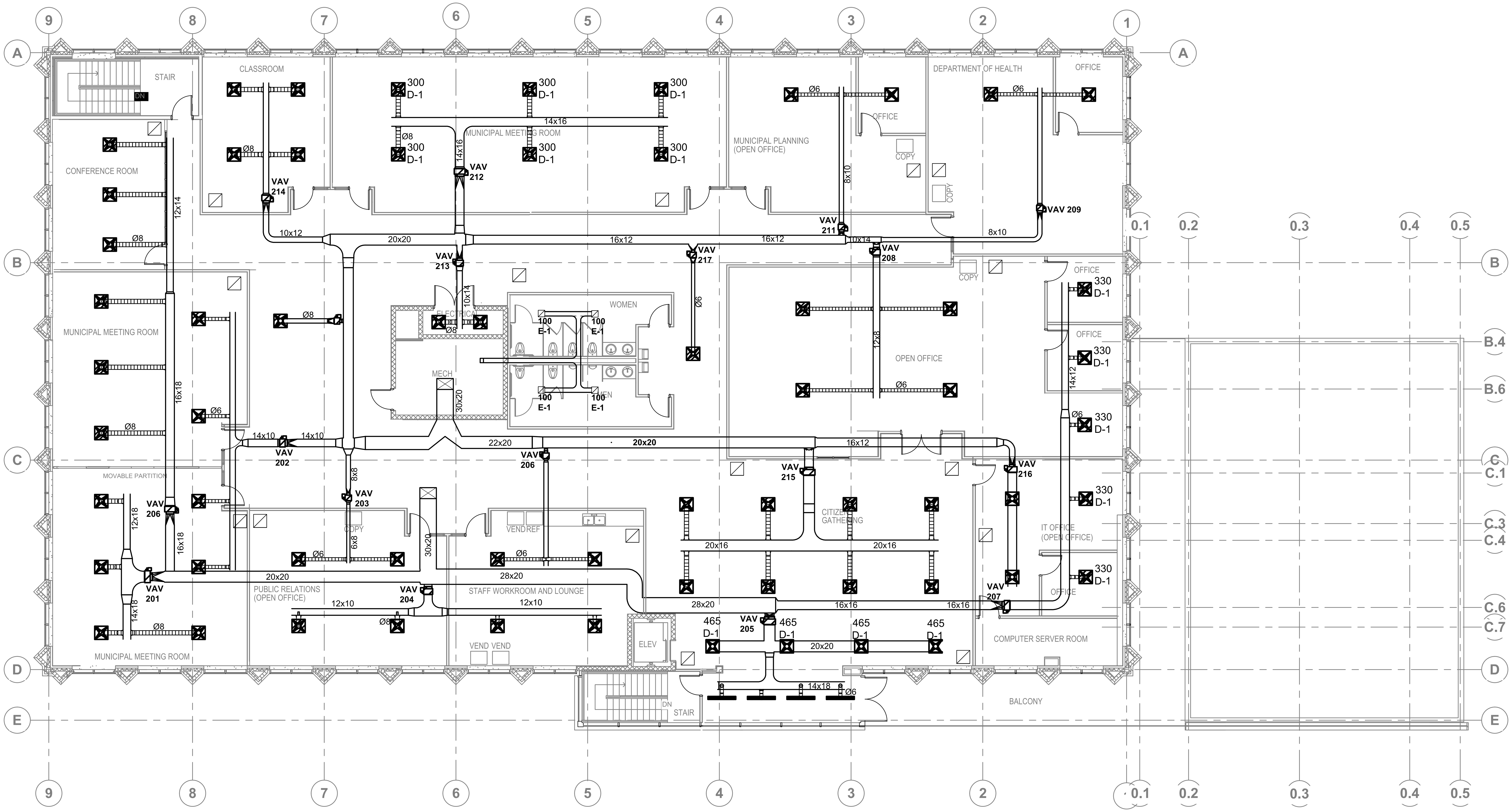
MECHANICAL FLOOR PLAN - LEVEL 1

PROJECT NUMBER	UBC-2015W-ME45X
DATE	2015-05-04
DRAWN BY	Z.B.
CHECKED BY	S.J.

M-1011

SCALE	1/8" = 1'-0"

2016-04-29 2:46:07 PM



1 MECHANICAL FLOOR PLAN - LEVEL 2
SCALE: 1/8" = 1'-0"

REV	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	2016-05-04

ASHRAE
SENIOR DESIGN COMPETITION

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DEPARTMENT OF MECHANICAL ENGINEERING
2054-6250 APPLIED SCIENCE LANE
VANCOUVER, BC CANADA V6T 1ZR

MECHANICAL FLOOR PLAN - LEVEL 2

PROJECT NUMBER	UBC-2015W-ME45X
DATE	2015-05-04
DRAWN BY	Z.B.
CHECKED BY	S.J.
	M-1012
SCALE	1/8" = 1'-0"

2016-04-29 2:46:07 PM

System Checksums

By UBC

Solar Perimeter — **Variable Volume Reheat (30% Min Flow Default)**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 8 / 16; Outside Air: OADB/WB/HR: 91 / 82 / 152

CLG Space Peak — Mo/Hr: 9 / 16; OADB: 85

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	3,097	3,097	1	0	0
Glass Solar	60,668	0	60,668	15	85,147	41
Glass/Door Cond	16,530	0	16,530	4	9,982	5
Wall Cond	3,704	2,750	6,454	2	2,319	1
Partition/Door	6,777		6,777	2	6,777	3
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	26,282		26,282	7	4,207	2
Sub Total ==>	113,960	5,847	119,808	30	108,432	52
Internal Loads						
Lights	14,849	5,217	20,066	5	14,849	7
People	101,057	0	101,057	25	54,273	26
Misc	29,983	0	29,983	7	29,983	14
Sub Total ==>	145,888	5,217	151,105	38	99,104	48
Ceiling Load	1,469	-1,469	0	0	1,065	1
Ventilation Load	0	0	85,395	21	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-8,756	-8,756	-2		
Sup. Fan Heat			33,625	8		
Ret. Fan Heat		20,274	20,274	5		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	261,318	21,113	401,451	100.00	208,601	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 12

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	-3,540	2.44
Glass Solar	0	0	0.00
Glass/Door Cond	-39,483	-39,483	27.22
Wall Cond	-8,307	-14,014	9.66
Partition/Door	-13,044	-13,044	8.99
Floor	-7,305	-7,305	5.04
Adjacent Floor	0	0	0
Infiltration	-25,977	-25,977	17.91
Sub Total ==>	-94,117	-103,364	71.25
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-5,501	0	0.00
Ventilation Load	0	-74,942	51.66
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	28,972	28,972	-19.97
Exhaust Heat		9,543	-6.58
OA Preheat Diff.		-5,277	3.64
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-70,645	-145,069	100.00

TEMPERATURES

	Cooling	Heating
SADB	60.5	86.5
Ra Plenum	74.1	67.9
Return	75.4	67.9
Ret/OA	76.3	52.4
Fn MtrTD	0.2	0.0
Fn BldTD	0.5	0.0
Fn Frict	1.5	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	14,299	3,805
Terminal	14,299	3,805
Main Fan	14,299	3,805
Sec Fan	0	0
Nom Vent	3,805	3,805
AHU Vent	3,805	3,805
Infil	334	400
MinStop/Rh	5,043	3,805
Return	14,632	4,205
Exhaust	4,139	4,205
Rm Exh	0	0
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	26.6	100.0
cfm/ft²	1.93	0.51
cfm/ton	427.41	
ft²/ton	221.60	
Btu/hr·ft²	54.15	-19.64
No. People	235	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	33.5	401.5	270.6	13,351	76.3	65.1	74.5	58.3	55.4	60.6
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	33.5	401.5								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	7,414		
Part	1,749		
Int Door	0		
ExFlr	250		
Roof	4,161	0	0
Wall	7,083	2,960	42
Ext Door	0	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-120.5	3,805	58.3	86.5
Aux Htg	0.0	0	0.0	0.0
Preheat	-25.1	3,805	52.4	58.3
Reheat	-49.8	3,805	58.3	70.0
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-145.6			

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
Alternative - 1 System Checksums Report Page 5 of 10

System Checksums

By UBC

Core — **Variable Volume Reheat (30% Min Flow Default)**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 8 / 15; Outside Air: OADB/WB/HR: 91 / 82 / 152

CLG Space Peak — Mo/Hr: 7 / 16; OADB: 94

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	7,244	7,244	1	0	0
Glass Solar	8,709	0	8,709	1	10,596	4
Glass/Door Cond	5,584	0	5,584	1	6,587	2
Wall Cond	1,708	785	2,493	0	2,235	1
Partition/Door	35,663		35,663	6	35,663	12
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	72,765		72,765	11	20,787	7
Sub Total ==>	124,429	8,028	132,458	21	75,869	26
Internal Loads						
Lights	38,047	13,368	51,415	8	38,228	13
People	168,953	0	168,953	26	91,110	31
Misc	79,508	0	79,508	12	79,917	27
Sub Total ==>	286,508	13,368	299,876	47	209,255	72
Ceiling Load	6,411	-6,411	0	0	6,888	2
Ventilation Load	0	0	184,514	29	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-29,160	-29,160	-5		
Sup. Fan Heat			34,577	5		
Ret. Fan Heat		21,532	21,532	3		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	417,348	7,357	643,797	100.00	292,013	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 12

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	-14,025	6.70
Glass Solar	0	0	0.00
Glass/Door Cond	-19,229	-19,229	9.18
Wall Cond	-7,096	-10,850	5.18
Partition/Door	-35,259	-35,259	16.84
Floor	-4,624	-4,624	2.21
Adjacent Floor	0	0	0
Infiltration	-58,520	-58,520	27.94
Sub Total ==>	-124,728	-142,507	68.04
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-8,478	0	0.00
Ventilation Load	0	-118,964	56.80
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	50,132	50,132	-23.94
Exhaust Heat		8,803	-4.20
OA Preheat Diff.		-6,899	3.29
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-83,074	-209,436	100.00

TEMPERATURES

	Cooling	Heating
SADB	55.9	81.6
Ra Plenum	76.0	66.9
Return	77.3	66.9
Ret/OA	78.8	48.4
Fn MtrTD	0.2	0.0
Fn BldTD	0.5	0.0
Fn Frict	1.5	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	14,141	5,432
Terminal	14,141	5,432
Main Fan	14,141	5,432
Sec Fan	0	0
Nom Vent	7,766	5,432
AHU Vent	7,766	5,432
Infil	955	928
MinStop/Rh	5,432	5,432
Return	14,946	5,825
Exhaust	8,571	5,825
Rm Exh	150	535
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	54.9	100.0
cfm/ft²	0.67	0.26
cfm/ton	263.59	
ft²/ton	395.43	
Btu/hr·ft²	30.35	-10.14
No. People	396	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	53.7	643.8	375.4	13,729	78.8	68.0	84.7	53.6	53.0	58.7
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	53.7	643.8								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	21,215		
Part	17,629		
Int Door	0		
ExFlr	153		
Roof	10,449	0	0
Wall	3,583	947	26
Ext Door	0	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-170.3	5,432	53.6	81.6
Aux Htg	0.0	0	0.0	0.0
Preheat	-44.8	7,766	48.5	53.6
Reheat	-87.2	5,432	53.6	67.9
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-215.0			

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
Alternative - 1 System Checksums Report Page 1 of 10

System Checksums

By UBC

East Wing — **Variable Volume Reheat (30% Min Flow Default)**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 8 / 24; Outside Air: OADB/WB/HR: 78 / 77 / 140

CLG Space Peak — Mo/Hr: 7 / 10; OADB: 85

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	2,040	2,040	2	0	0
Glass Solar	885	0	885	1	9,565	23
Glass/Door Cond	-147	0	-147	0	764	2
Wall Cond	2,899	943	3,841	4	2,834	7
Partition/Door	7,110		7,110	8	7,110	17
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	8,649		8,649	9	1,574	4
Sub Total ==>	19,395	2,983	22,378	24	21,847	54
Internal Loads						
Lights	3,570	1,254	4,824	5	4,655	11
People	18,024	0	18,024	19	11,040	27
Misc	1,848	0	1,848	2	1,848	5
Sub Total ==>	23,443	1,254	24,697	27	17,544	43
Ceiling Load	1,453	-1,453	0	0	1,362	3
Ventilation Load	0	0	45,079	49	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-4,435	-4,435	-5		
Sup. Fan Heat			3,108	3		
Ret. Fan Heat		1,722	1,722	2		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	44,290	72	92,549	100.00	40,753	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 12

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	-3,557	11.34
Glass Solar	0	0	0.00
Glass/Door Cond	-7,631	-7,631	24.32
Wall Cond	-7,565	-10,092	32.16
Partition/Door	-6,488	-6,488	20.68
Floor	-5,564	-5,564	17.73
Adjacent Floor	0	0	0
Infiltration	-10,896	-10,896	34.72
Sub Total ==>	-38,144	-44,229	140.95
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-3,204	0	0.00
Ventilation Load	0	-14,199	45.25
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	25,036	25,036	-79.78
Exhaust Heat		4,090	-13.03
OA Preheat Diff.		-2,077	6.62
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-16,312	-31,380	100.00

TEMPERATURES

	Cooling	Heating
SADB	58.3	90.0
Ra Plenum	78.5	63.2
Return	79.4	63.2
Ret/OA	78.8	48.7
Fn MtrTD	0.2	0.0
Fn BldTD	0.4	0.0
Fn Frict	1.3	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	2,109	658
Terminal	2,109	658
Main Fan	2,109	658
Sec Fan	0	0
Nom Vent	899	658
AHU Vent	899	658
Infil	178	178
MinStop/Rh	658	658
Return	2,271	824
Exhaust	1,061	824
Rm Exh	16	13
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	42.6	100.0
cfm/ft²	0.65	0.20
cfm/ton	273.44	
ft²/ton	423.51	
Btu/hr·ft²	28.33	-9.82
No. People	45	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	7.7	92.6	35.1	1,498	78.8	73.3	114.3	56.4	55.3	63.3
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	7.7	92.6								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	3,266		
Part	4,009		
Int Door	0		
ExFlr	197		
Roof	2,478	0	0
Wall	2,707	279	10
Ext Door	135	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-24.8	658	56.4	90.0
Aux Htg	0.0	0	0.0	0.0
Preheat	-7.3	899	49.2	56.4
Reheat	-8.5	658	56.4	67.9
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	-32.1			

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
Alternative - 1 System Checksums Report Page 2 of 10

System Checksums

By UBC

Serverroom — **Computer Room Unit**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 7 / 16; Outside Air: OADB/WB/HR: 94 / 82 / 145

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

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	147	147	0	0	0
Glass Solar	671	0	671	1	2,909	4
Glass/Door Cond	671	0	671	1	-622	-1
Wall Cond	303	174	476	1	-155	0
Partition/Door	1,094		1,094	1	1,094	1
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	0		0	0	0	0
Sub Total ==>	2,739	321	3,059	4	3,226	4
Internal Loads						
Lights	498	175	673	1	494	1
People	0	0	0	0	0	0
Misc	72,600	0	72,600	91	72,600	95
Sub Total ==>	73,098	175	73,273	92	73,094	96
Ceiling Load	6	-6	0	0	0	0
Ventilation Load	0	0	960	1	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		-2	-2	0		
Sup. Fan Heat			2,190	3		
Ret. Fan Heat		0	0	0		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	75,842	488	79,481	100.00	76,320	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 12

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	-265	4.59
Glass Solar	0	0	0.00
Glass/Door Cond	-2,072	-2,072	35.93
Wall Cond	-789	-1,242	21.53
Partition/Door	-1,393	-1,393	24.15
Floor	0	0	0.00
Adjacent Floor	0	0	0
Infiltration	0	0	0.00
Sub Total ==>	-4,254	-4,972	86.21
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-8	0	0.00
Ventilation Load	0	-798	13.83
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	0	0	0.00
Exhaust Heat		2	-0.04
OA Preheat Diff.		0	0.00
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-4,262	-5,767	100.00

TEMPERATURES

	Cooling	Heating
SADB	55.0	74.4
Ra Plenum	73.5	73.2
Return	73.5	73.2
Ret/OA	73.6	73.0
Fn MtrTD	0.1	0.0
Fn BldTD	0.1	0.0
Fn Frict	0.4	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	3,696	3,696
Terminal	3,696	3,696
Main Fan	3,696	3,696
Sec Fan	0	0
Nom Vent	12	12
AHU Vent	12	12
Infil	0	0
MinStop/Rh	0	0
Return	3,696	3,696
Exhaust	12	12
Rm Exh	0	0
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	0.3	0.3
cfm/ft²	23.94	23.94
cfm/ton	557.99	
ft²/ton	23.31	
Btu/hr·ft²	514.91	-40.49
No. People	0	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	6.6	79.5	78.8	3,696	73.6	61.4	61.4	54.5	54.0	61.2
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	6.6	79.5								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	154		
Part	427		
Int Door	0		
ExFlr	0		
Roof	154	0	0
Wall	352	96	27
Ext Door	0	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	-5.8	3,696	73.0	74.4
Aux Htg	0.0	0	0.0	0.0
Preheat	0.0	0	0.0	0.0
Reheat	0.0	0	0.0	0.0
Humidif	-0.5	12	2.0	61.1
Opt Vent	0.0	0	0.0	0.0
Total	-6.3			

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
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System Checksums

By UBC

Egress Stairwell (Unconditioned) — **Unit Heaters**

COOLING COIL PEAK / CLG SPACE PEAK

Cooling Coil Peak — Peaked at Time: Mo/Hr: 0 / 0; Outside Air: OADB/WB/HR: 0 / 0 / 0

CLG Space Peak — Mo/Hr: 0 / 0; OADB: 0

	Space Sens. + Lat. Btu/h	Plenum Sens. + Lat Btu/h	Net Total Btu/h	Percent Of Total (%)	Space Sensible Btu/h	Percent Of Total (%)
Envelope Loads						
Skylite Solar	0	0	0	0	0	0
Skylite Cond	0	0	0	0	0	0
Roof Cond	0	0	0	0	0	0
Glass Solar	0	0	0	0	0	0
Glass/Door Cond	0	0	0	0	0	0
Wall Cond	0	0	0	0	0	0
Partition/Door	0		0	0	0	0
Floor	0		0	0	0	0
Adjacent Floor	0	0	0	0	0	0
Infiltration	0		0	0	0	0
Sub Total ==>	0	0	0	0	0	0
Internal Loads						
Lights	0	0	0	0	0	0
People	0	0	0	0	0	0
Misc	0	0	0	0	0	0
Sub Total ==>	0	0	0	0	0	0
Ceiling Load	0	0	0	0	0	0
Ventilation Load	0	0	0	0	0	0
Adj Air Trans Heat	0		0	0	0	0
Dehumid. Ov Sizing			0	0		
Ov/Undr Sizing	0		0	0	0	0
Exhaust Heat		0	0	0		
Sup. Fan Heat			0	0		
Ret. Fan Heat		0	0	0		
Duct Heat Pkup		0	0	0		
Underflr Sup Ht Pkup			0	0		
Supply Air Leakage		0	0	0		
Grand Total ==>	0	0	0	100.00	0	100.00

HEATING COIL PEAK

Mo/Hr: Heating Design; OADB: 12

	Space Peak Space Sens Btu/h	Coil Peak Tot Sens Btu/h	Percent Of Total (%)
Envelope Loads			
Skylite Solar	0	0	0.00
Skylite Cond	0	0	0.00
Roof Cond	0	-328	3.42
Glass Solar	0	0	0.00
Glass/Door Cond	-4,152	-4,152	43.22
Wall Cond	-2,044	-2,430	25.29
Partition/Door	-309	-309	3.22
Floor	-881	-881	9.17
Adjacent Floor	0	0	0
Infiltration	-1,508	-1,508	15.69
Sub Total ==>	-8,894	-9,607	100.00
Internal Loads			
Lights	0	0	0.00
People	0	0	0.00
Misc	0	0	0.00
Sub Total ==>	0	0	0.00
Ceiling Load	-714	0	0.00
Ventilation Load	0	0	0.00
Adj Air Trans Heat	0	0	0
Ov/Undr Sizing	0	0	0.00
Exhaust Heat		0	0.00
OA Preheat Diff.		0	0.00
RA Preheat Diff.		0	0.00
Additional Reheat		0	0.00
Underflr Sup Ht Pkup		0	0.00
Supply Air Leakage		0	0.00
Grand Total ==>	-9,607	-9,607	100.00

TEMPERATURES

	Cooling	Heating
SADB	0.0	60.0
Ra Plenum	0.0	51.5
Return	0.0	60.0
Ret/OA	0.0	60.0
Fn MtrTD	0.0	0.0
Fn BldTD	0.0	0.0
Fn Frict	0.0	0.0

AIRFLOWS

	Cooling	Heating
Diffuser	0	0
Terminal	0	0
Main Fan	0	0
Sec Fan	0	0
Nom Vent	0	0
AHU Vent	0	0
Infil	0	28
MinStop/Rh	0	0
Return	0	28
Exhaust	0	28
Rm Exh	0	0
Auxiliary	0	0
Leakage Dwn	0	0
Leakage Ups	0	0

ENGINEERING CKS

	Cooling	Heating
% OA	0.0	0.0
cfm/ft²	0.00	0.00
cfm/ton	0.00	
ft²/ton	0.00	
Btu/hr·ft²	0.00	0.00
No. People	0	

COOLING COIL SELECTION

	Total Capacity ton	Total Capacity MBh	Sens Cap. MBh	Coil Airflow cfm	Enter DB °F	Enter WB °F	Enter HR gr/lb	Leave DB °F	Leave WB °F	Leave HR gr/lb
Main Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Aux Clg	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Opt Vent	0.0	0.0	0.0	0	0.0	0.0	0.0	0.0	0.0	0.0
Total	0.0	0.0								

AREAS

	Gross Total	Glass ft²	Glass (%)
Floor	263		
Part	757		
Int Door	0		
ExFlr	32		
Roof	263	0	0
Wall	838	190	23
Ext Door	59	0	0

HEATING COIL SELECTION

	Capacity MBh	Coil Airflow cfm	Ent °F	Lvg °F
Main Htg	0.0	0	60.0	60.0
Aux Htg	0.0	0	0.0	0.0
Preheat	0.0	0	0.0	0.0
Humidif	0.0	0	0.0	0.0
Opt Vent	0.0	0	0.0	0.0
Total	0.0			

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
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SYSTEM SUMMARY

DESIGN COOLING CAPACITIES

By UBC

Alternative 1

Building Airside Systems and Plant Capacities

Plant	System	Peak Plant Loads: Main Coil ton	Aux Coil ton	Opt Vent Coil ton	Misc Load ton	Stg 1 Desic Cond ton	Stg 2 Desic Cond ton	Base Utility ton	Peak Total ton	Block Plant Loads: Time Of Peak mo/hr	Main Coil ton	Aux Coil ton	Opt Vent Coil ton	Misc Load ton	Stg 1 Desic Cond ton	Stg 2 Desic Cond ton	Base Utility ton	Block Total ton
Office - Ice Chiller		87.1	0.0	0.0	0.0	0.0	0.0	0.0	87.1	8/13	79.7	0.0	0.0	0.0	0.0	0.0	0.0	79.7
	Solar Perimeter	33.5	0.0	0.0	0.0	0.0	0.0	0.0	33.5	8/13	26.2	0.0	0.0	0.0	0.0	0.0	0.0	26.2
	Core	53.7	0.0	0.0	0.0	0.0	0.0	0.0	53.7	8/13	53.5	0.0	0.0	0.0	0.0	0.0	0.0	53.5
Server Room DX		6.6	0.0	0.0	0.0	0.0	0.0	0.0	6.6	1/1	6.6	0.0	0.0	0.0	0.0	0.0	0.0	6.6
	Serverroom	6.6	0.0	0.0	0.0	0.0	0.0	0.0	6.6	1/1	6.6	0.0	0.0	0.0	0.0	0.0	0.0	6.6
East Wing DX		7.7	0.0	0.0	0.0	0.0	0.0	0.0	7.7	8/5	6.3	0.0	0.0	0.0	0.0	0.0	0.0	6.3
	East Wing	7.7	0.0	0.0	0.0	0.0	0.0	0.0	7.7	8/5	6.3	0.0	0.0	0.0	0.0	0.0	0.0	6.3
Building totals		101.4	0.0	0.0	0.0	0.0	0.0	0.0	101.4		92.6	0.0	0.0	0.0	0.0	0.0	0.0	92.6

Building peak load is 101.7 tons.

Building maximum block load of 91.3 tons occurs in August at hour 13 based on system simulation.

SYSTEM SUMMARY

DESIGN HEATING CAPACITIES

By UBC

Alternative 1

System Coil Capacities

System Description	System Type	Main System Btu/h	Aux System Btu/h	Preheat Btu/h	Reheat Btu/h	Humid. Btu/h	Optional Vent Btu/h	Stg 1 Desic Regen Btu/h	Stg 2 Desic Regen Btu/h	Stg 1 Frost Prevention Btu/h	Stg 2 Frost Prevention Btu/h	Heating Totals Btu/h
Solar Perimeter	Variable Volume Reheat (30% Min Flow Default)	-120,460	0	-25,127	-49,816	0	0	0	0	0	0	-145,587
Core	Variable Volume Reheat (30% Min Flow Default)	-170,271	0	-44,751	-87,198	0	0	0	0	0	0	-215,022
Egress Stairwell (Unconditioned)	Unit Heaters	0	0	0	0	0	0	0	0	0	0	0
Serverroom	Computer Room Unit	-5,767	0	0	0	-483	0	0	0	0	0	-6,250
East Wing	Variable Volume Reheat (30% Min Flow Default)	-24,822	0	-7,245	-8,509	0	0	0	0	0	0	-32,067
Totals		**-321,321**	**0**	**-77,123**	**-145,523**	**-483**	**0**	**0**	**0**	**0**	**0**	**-398,926**

Building Plant Capacities

Plant	System	Peak Loads: Main Coil MBh	Preheat Coil MBh	Reheat Coil MBh	Humid. Coil MBh	Aux Coil MBh	Opt Vent Coil MBh	Misc Load MBh	Stg 1 Desic. Regen. MBh	Stg 2 Desic. Regen. MBh	Stg 1 Frost Prev. MBh	Stg 2 Frost Prev. MBh	Base Utility MBh	Absorption Load MBh
Heating plant - All Boilers		321	77	0	0	0	0	0	0	0	0	0	0	0
	Solar Perimeter	120	25	0	0	0	0	0	0	0	0	0	0	0
	Core	170	45	0	0	0	0	0	0	0	0	0	0	0
	Egress Stairwell (Unconditioned)	0	0	0	0	0	0	0	0	0	0	0	0	0
	Serverroom	6	0	0	0	0	0	0	0	0	0	0	0	0
	East Wing	25	7	0	0	0	0	0	0	0	0	0	0	0

Building peak load is 398.9 MBh.

Project Name:
Dataset Name: T2CAPSTN.TRC

TRACE® 700 v6.3.2 calculated at 02:35 PM on 03/29/2016
Design Capacity Quantities report Page 1 of 2

1 TRACE 700 SYTEM CHECKSUM AND PLANT CAPACITY REPORTS
SCALE: N.T.S.

REV	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	2016-05-04

SENIOR DESIGN COMPETITION

University of British Columbia Student Branch

UNIVERSITY OF BRITISH COLUMBIA
DEPARTMENT OF MECHANICAL ENGINEERING
2054-6250 APPLIED SCIENCE LANE
VANCOUVER, BC CANADA V6T 1ZR

SYSTEM CHECKSUM REPORTS

PROJECT NUMBER	UBC-2015W-ME45X
DATE	2015-05-04
DRAWN BY	S.J.
CHECKED BY	Z.B.

M-2001

SCALE N.T.S.

2016-05-04 8:34:18 PM

AIR HANDLING UNIT SCHEDULE

TAG	LOCATION	MAKE	MODEL	SUPPLY FAN							RETURN/EXHAUST FAN						
				AIR FLOW	ESP	FRPM	BHP	MOTOR		ELECTRIC	AIR FLOW	ESP	FRPM	BHP	MOTOR		ELECTRIC
				CFM	IN. WC		HP	HP	RPM	V/PH/HZ	CFM	IN. WC		HP	HP	RPM	V/PH/HZ
AHU-1	ROOF	DAIKIN	CAH035GDGM	14300	2.5	1225	15	20	1160	480/3/60	14300	1.25	1975	11.17	15	1750	480/3/60
AHU-2	ROOF	DAIKIN	CAH035GDGM	14130	2.5	1225	15	20	1160	480/3/60	14130	1.25	1975	11.17	15	1750	480/3/60
AHU-3	DISPATCH CENTER ROOF	DAIKIN	DPS007A	2110	1.5	1550	2.75	8	1550	480/3/60	2110	0.75	1325	0.9	4	1325	480/3/60

TAG	COOLING COIL										
	TOTAL CAPACITY	SENSIBLE CAPACITY	EDB	EWB	LDB	LWB	EWT	LWT	FLUID FLOW	WDP	ROW
	MBH	MBH	°F	°F	°F	°F	°F	°F	GPM	FT	
AHU-1	401.5	270.6	76.3	65.1	58.3	55.4	45	55.8	71.4	69.9	6
AHU-2	643.8	375.4	78.8	68	53.6	53	45	56.2	114.5	109.4	6
AHU-3	92.6	35.1	78.8	73.3	12.8	56.4	DX- R410A				3

TAG	ENTHALPY WHEEL				PRE-HEAT COIL								REMARKS
	HEATING AIRFLOW	COOLING AIRFLOW	HEATING EFFECTIVENESS	COOLING EFFECTIVENESS	TOTAL CAPACITY	EDB	LDB	EWT	LWT	FLUID FLOW	WDP	ROW	
	CFM	CFM	SENSIBLE/TOTAL	SENSIBLE/TOTAL	MBH	°F	°F	°F	°F	GPM	FT		
AHU-1	3805	3805	79.8/78.4	79.1/76.8	25.1	52.4	58.3	160	143.2	78.2	2.1	2	1,2
AHU-2	5432	7766	79.8/78.4	79.1/76.8	44.8	48.5	53.6	160	140	139.6	5.2	2	1,2
AHU-3	658	899	86/84	86/83	7.3	49.2	56.4	160	141	22.7	1.9	2	1,2,3

NOTES:

1. VFD FOR SUPPLY FAN, RETURN FAN AND ENTHALPY WHEEL MOTOR
2. MERV 11 FILTER
3. INTEGRAL AIR-COOLED CONDENSING UNIT WITH PACKAGED IEER OF 19.9 (ASHRAE 90.1-2013 COMPLIANT)

VAV TERMINAL UNIT SCHEDULE

TAG	MAKE	MODEL	ZONE SERVED	UNIT SIZE	FLOOR	MAX AIR FLOW	MIN AIR FLOW	INLET SIZE	EAT	LAT	LAT	CAPACITY	ROWS	FLUID TYPE	FLUID FLOW	MAX AIR PD	EWT	LWT	FLUID PD	WEIGHT	REMARKS
						CFM	CFM	INΦ/IN x IN	°F	°F	°F	MBH			GPM	IN WG	°F	°F	FT	LBS	
VAV-101	E.H. PRICE	SDV8	1-MRC-S	10	L1	1719	516	10	58.30	78.70	25.94	11.60	1	WTR	0.60	0.62	180.00	140.60	0.08	52	1,2
VAV-102	E.H. PRICE	SDV8	1-STFLING	8	L1	945	283	8	58.30	80	26.78	6.90	1	WTR	0.30	0.42	180.00	133.10	0.09	44	1,2
VAV-103	E.H. PRICE	SDV8	1-RECEP	9	L1	1194	358	9	58.30	84.40	29.11	10.30	1	WTR	0.60	0.32	180.00	145.10	0.08	44	1,2
VAV-104	E.H. PRICE	SDV8	1-DIROFF-S	8	L1	1033	310	8	58.30	90.50	32.50	11.00	1	WTR	1.37	0.49	180.00	163.60	0.98	37	1,2
VAV-105	E.H. PRICE	SDV8	1-DIROFF-INT	7	L1	567	170	7	53.60	90.10	32.28	6.90	1	WTR	0.47	0.17	180.00	149.80	0.18	37	1,2
VAV-106	E.H. PRICE	SDV8	1-HW3	7	L1	601	180	7	56.40	90.50	32.50	6.80	1	WTR	0.44	0.19	180.00	148.60	0.17	37	1,2
VAV-107	E.H. PRICE	SDV8	1-DSP	8	L1	1094	328	8	56.40	90.50	32.50	12.40	1	WTR	2.42	0.55	180.00	169.50	2.40	37.00	1,2
VAV-108	E.H. PRICE	SDV8	1-SLP	4	L1	190	57	4	56.40	106.30	41.28	3.20	1	WTR	0.30	0.04	180.00	158.40	0.07	44	1,2
VAV-109	E.H. PRICE	SDV8	1-GRG	4	L1	151	45	4	56.40	112.60	44.78	2.80	1	WTR	0.33	0.03	180.00	162.60	0.08	36.00	1,2
VAV-110	E.H. PRICE	SDV8	1-GRND-N	5	L1	376	113	5	53.60	90.20	32.33	4.60	1	WTR	0.32	0.14	180.00	150.30	0.07	44.00	1,2
VAV-111	E.H. PRICE	SDV8	1-MAIL-N	5	L1	418	125	5	53.60	90.30	32.39	5.10	1	WTR	0.39	0	180.00	153.30	0.10	37	1,2
VAV-112	E.H. PRICE	SDV8	1-PR/MR-INT	6	L1	514	154	6	53.60	90.40	32.44	6.30	1	WTR	0.64	0.24	180.00	159.70	0.22	37	1,2
VAV-113	E.H. PRICE	SDV8	1-PR/MR-N	7	L1	557	167	7	53.60	90.20	32.33	6.80	1	WTR	0.46	0.16	180.00	149.50	0.18	61	1,2
VAV-114	E.H. PRICE	SDV8	1-CLASS	9	L1	1178	353	9	58.30	87.10	30.61	11.20	1	WTR	0.79	0.31	180.00	150.80	0.12	44	1,2
VAV-116	E.H. PRICE	SDV8	1-MRC-INT	7	L1	694	208	7	53.60	90.30	32.39	8.50	1	WTR	0.76	0.24	180.00	157.00	0.39	37	1,2
VAV-117	E.H. PRICE	SDV8	1-CONF	9	L1	1148	366	9	53.60	84.30	29.06	12.50	1	WTR	0.94	0.31	180.00	152.90	0.17	44	1,2
VAV-118	E.H. PRICE	SDV8	1-ELEC	8	L1	838	251	8	53.60	78.30	25.72	6.90	1	WTR	0.30	0.34	180.00	133.20	0.09	44	1,2
VAV-119	E.H. PRICE	SDV8	1-HW1	4	L1	108	58	4	53.60	104.70	40.39	3.30	1	WTR	0.32	0.01	180.00	158.70	0.07	44	1,2
VAV-120	E.H. PRICE	SDV8	1-HW2	4	L1	176	134	4	53.60	90.50	32.50	5.50	1	WTR	0.47	0.03	180.00	155.80	0.13	44	1,2
VAV-121	E.H. PRICE	SDV8	1-STFWORK	4	L1	102	33	4	53.60	117.10	47.28	2.30	1	WTR	0.30	0.01	180.00	164.00	0.07	37	1,2
VAV-121	E.H. PRICE	SDV8	1-MRC-W	9	L1	1367	410	9	58.30	91	32.56	14.60	1	WTR	1.79	0.41	180.00	163.20	0.50	37	1,2
VAV-201	E.H. PRICE	SDV8	2-MT/CNF-S	7	L2	715	214	7	58.30	85	29.44	6.30	1	WTR	0.30	0.25	180.00	137.60	0.09	52	1,2
VAV-202	E.H. PRICE	SDV8	2-MT/CNF-INT	7	L2	689	232	7	53.60	86.90	30.50	8.60	1	WTR	0.64	0.24	180.00	152.50	0.30	44	1,2
VAV-203	E.H. PRICE	SDV8	2-PR-INT	4	L2	180	54	4	53.60	105.70	40.94	3.10	1	WTR	0.30	0.04	180.00	158.40	0.07	52	1,2
VAV-204	E.H. PRICE	SDV8	2-STF-S	6	L2	526	158	6	58.30	87.00	30.56	5	1	WTR	0.30	0.24	180.00	146.40	0.07	52	1,2
VAV-205	E.H. PRICE	SDV8	2-CTZGTH-S	14	L2	3241	972	14	58.30	85.10	29.50	28.70	1	WTR	1.76	0.56	180.00	146.60	0.72	44	1,2
VAV-206	E.H. PRICE	SDV8	2-STF-INT	4	L2	215	65	4	53.60	102.00	38.89	3.50	1	WTR	0.31	0.05	180.00	157.00	0.07	76	1,2
VAV-206	E.H. PRICE	SDV8	2-MT/CNF-W	12	L2	2590	777	12	58.30	86.10	30.06	23.80	1	WTR	2.40	0.74	180.00	159.70	0.99	44	1,2
VAV-207	E.H. PRICE	SDV8	2-OFF-E	9	L2	1335	401	9	58.30	82.50	28.06	10.70	1	WTR	0.60	0.39	180.00	143.60	0.08	61	1,2
VAV-209	E.H. PRICE	SDV8	2-OFF-N	7	L2	585	176	7	53.60	90.30	32.39	7.20	1	WTR	0.52	0.18	180.00	151.60	0.21	52	1,2
VAV-211	E.H. PRICE	SDV8	2-OFF-INT	8	L2	1091	327	8	53.60	90.40	32.44	13.40	1	WTR	4.70	0.55	180.00	174.10	6.92	36	1,2
VAV-212	E.H. PRICE	SDV8	2-MT-N	8	L2	1013	350	8	53.60	90.30	32.39	14.30	1	WTR	1.94	0.54	180.00	164.90	1.69	52	1,2
VAV-213	E.H. PRICE	SDV8	2-ELEC	7	L2	774	232	7	53.60	79.50	26.39	6.70	1	WTR	0.30	0.30	180.00	134.20	0.09	44	1,2
VAV-214	E.H. PRICE	SDV8	2-CLASS-N	5	L2	478	282	5	53.60	82.00	27.78	8.90	1	WTR	0.90	0.21	180.00	159.70	0.38	37	1,2
VAV-215	E.H. PRICE	SDV8	2-CTZGTH-INT	12	L2	2283	1208	12	53.60	67.10	19.50	18.10	1	WTR	0.65	0.60	180.00	123.60	0.11	44.00	1,2
VAV-215	E.H. PRICE	SDV8	2-HW1	5	L2	345	231	5	53.60	90.20	32.33	9.40	1	WTR	2.57	0.12	180.00	172.50	2.02	37	1,2

NOTES:

1. LINER: AHRI 885-2008 MINERAL FIBER
2. DISCHARGE ATTENUATION: ATT3

REV	DESCRIPTION	DATE
A	ISSUED FOR COMPETITION REPORT	2016-05-04

ASHRAE

SENIOR DESIGN COMPETITION

ASHRAE University of British Columbia Student Branch

UBC

UNIVERSITY OF BRITISH COLUMBIA
DEPARTMENT OF MECHANICAL ENGINEERING
2054-6250 APPLIED SCIENCE LANE
VANCOUVER, BC CANADA V6T 1ZR

EQUIPMENT SCHEDULES

PROJECT NUMBER	UBC-2015W-ME45X
DATE	2015-05-04
DRAWN BY	S.J.
CHECKED BY	Z.B.

M-3001

SCALE	N.T.S.

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