Final Report

Park Place Corporate Center One

Connor Blood

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1.0 Executive Summary

Park Place Corporate Center One (Park Place 1) provides a unique opportunity to study an old office building with poor energy performance. With today's economy and rising energy rates, a growing number of building owners are looking to renovate older buildings in place of building new, more expensive ones. The study of Park Place 1 offers the chance to understand the effects of central plant modification.

The purpose of this report was to understand and analyze the existing building mechanical systems and then redesign them in an effort to save money and improve performance. This study focuses more on cooling than on heating due to the opportunity for energy savings. The redesign of existing systems does not include a redesign of the heating methods. By extension, because the building is existing, façade redesigns and other building related energy efficiency opportunities were neglected due to cost limitations. As an offshoot of mechanical system redesign, two other building systems—structural and electrical—were also analyzed to understand the possible impacts of changing the mechanical system.

Before addressing the designed systems, the building heating and cooling load was determined using Trane Trace 700. In summary, the total cooling load on the building was around 225 tons. Given the square footage of approximately 100,000 ft^2, a value of 445 ft^2/ton indicated that the building is relatively inefficient by modern construction standards, and thus plant improvements could render substantial cost savings.

To offset the 225 ton cooling load, a design solution came in the form of two packaged DX rooftop units. Each unit was sized at 120 tons with a 20°F air side ΔT and 45,000 cfm's of supply air. While this was adequate, it was simple and offered a chance to redesign a system that could potentially save the owner money.

As an office building, Park Place 1 had a low load factor of around 34%. In doing research, it was determined that thermal energy storage could potentially yield a viable improvement to the existing system. Ice storage became the primary consideration in moving forward from the original design.

To truly understand the impact and potential cost savings of thermal energy storage, several alternatives had to be considered. Since the original design was a completely air dependent system, a chilled water system had to be studied. From that chilled water system, ice storage could be added but in what quantities? An optimization study had to be conducted that would range from including two ice tanks to six. A final study was done to see what the impacts would be of reducing the supply air temperature of the air handling units from 55°F to 50°F. This would save fan energy at potentially little cost to the water side of the system.

The results of the study indicated three primary conclusions pertaining to the different alternatives and a single final conclusion about which system would perform the best over the life of the building. First, the chilled water system was not practical. It had the highest first cost, highest operating cost, and highest energy expenditure of any system. Second, the optimization

study determined that two ice storage tanks would yield the shortest payback period. Third, reducing the cfm's in the final alternative justified making the chiller work slightly harder to produce cold water.

The main conclusion and final recommendation was that the system combining two ice storage tanks with a reduced sized air handling unit that provided 50°F supply air would be the best system to install. The evidence to support such conclusions is that it would cost the owner the least amount of money over the life of the building and would have the smallest first cost.

The results of the electrical and structural study indicate that no major changes to either system would be required to implement the proposed mechanical systems. The electrical panelboard that services the mechanical equipment could be reduced from 800 A to 700 A but changes in conductor, conduit, and main distribution panel sizes would be nonexistent. For the structural system, the roof deck would need to be increased in gage by one size but no changes to structural members (beams, girders, and columns) would be required.

2.0 Project Information

2.1 Design Goals

Park Place 1 was constructed in 1982 under the title of Park Ridge Building One. Since 1982, the building has changed ownership and tenant hands several times with the most recent transfer occurring in December of 2009. DiCicco Development, the current owner of Park Place 1, purchased the property to house an expanding list of clients that desired modern office space. To meet the needs of their clients, DiCicco Development renovated the building to provide a comfortable, environmentally friendly office type work space.

2.2 Location

Park Place 1 is located in RIDC Park West in Findlay Township, Pennsylvania. **Figure 1** shows the building designated by the letter "A" just west of Pittsburgh. Satellite images can be seen below in **Figure 2** and **Figure 3**. North is towards the top of the page. Park Place 1 is highlighted in red. An almost identical building, Park Place 2, is located just to the northwest of Park Place 1 and can be seen in both images.

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2.3 Project Team

- **Owner** *DiCicco Development*
- **Architect** *Williams/Trebilcock/Whitehead Architects*
- **MEP/FP Engineer** *CJL Engineering*
- **Structural Engineer** *Williams/Trebilcock/Whitehead Architects*
- **Landscape Architect** *Williams/Trebilcock/Whitehead Architects*

3.0 Building Overview & Existing Conditions

3.1 Architecture

Designed to reflect the up and coming modern architectural office buildings of the time, Park Place Corporate Center One from the exterior appears to be completely glass. The façade of the building, seen below in **Figure 4**, is comprised of large glass panes separated by black aluminum mullions. Completely reflective, the glass allows for no view into the building from the exterior. This ensures privacy for the building tenants. The reflective glass ties the building and site together as it is impossible to look at the building without seeing its surroundings.

Figure 4 – Park Place 1 Exterior

3.2 Sustainability Features

Because the building was constructed in 1982, no substantial effort was made to incorporate sustainable ideas into the design—thus the building is being renovated.

3.3 Building Façade

Park Place 1 has a curtain wall system of steel, aluminum, and glass. Steel serves as the structural support for the system with aluminum mullions separating large panes of dark, reflective glass.

On the interior, gypsum wall board conceals a single layer of insulation and a vapor barrier.

3.4 Electrical System

Building power enters from the northwest corner on grade from a utility provided transformer. The power is bused to an electrical room located in the building core where it is wired through a main switch board rated at 1000 amps at 277/480 Volts 3-phase. From this main switch board, a fraction of the power input to the building is passed through a second transformer also located in the electrical room. This second transformer is sized at 150 KVA and is intended to deliver 120/208 Volt 3-phase power. 120 Volt power is then provided to a panel board rated at 400 amps of 3 phase 120/208 Volts. Finally, power is provided to a panelboard intended for use by HVAC equipment located on the roof. This panel board is rated at 800 amps of 480 Volt 3 phase power.

As a backup power supply, an emergency generator located in the building rooftop penthouse is rated to provide 40 kW of 120/208 Volt 3 phase power in the event of system failure.

3.5 Lighting System

Park Place 1 utilizes, almost exclusively, linear T8 lamps in recessed 2 feet by 4 feet luminaires. These luminaires are located on a general grid pattern in all tenant occupied spaces. The few exceptions are in public entry spaces such as the main lobby and the rear lobby. These locations have ceiling hung, indirect incandescent lamps located in pendent type luminaires. All lighting is controlled by a building automation system capable of utilizing time of day schedules to ensure maximum energy performance.

3.6 Structural System

The structure begins with 3000 psi concrete foundation piers which lie just beneath a four inch thick slab on grade. Upon those piers lie base plates which distribute loads from ASTM

A36 steel columns. Those columns support loads that range from 150 to 450 kips depending on the location within the building. Five bays of twenty-four feet in the north-south direction and seven bays of twenty-four feet in the east-west direction form the basis of the steel structure. Floor point loads are supported by 4.5 inch topping on top of twenty gauge metal decking. The decking is supported by beams, usually W 16 x 26, which distribute loads to girders, usually W 21 x 50. The steel structure is a simply supported frame designed to absorb all lateral loads from wind that is collected by the building's curtain wall.

3.7 Fire Protection System

While the building is not sprinkled, smoke detectors, audio alerts, and strobe lighting alerts have been provided on all floors. The RTU's are also equipped with both supply and return duct smoke detectors. All fire protection systems are located on an emergency panel board to ensure that in the event of a fire or power failure, all emergency systems will be provided with power from the generator first.

3.8 Transportation

The building has two primary points of entry located on the north and south sides of the building. The main lobby area, located on the south side entrance is in the center of the building where the majority of the parking is located on the site. From the main entry lobby, the two elevators located in the building core can be entered. From the north entrance, to enter the elevators one must move through a corridor that contains the restrooms and pass into the main entry lobby. The elevators move between the first floor and the fifth floor, leaving the rooftop penthouse accessible by stairs only. Two staircases are located on the east and west sides of the building core, the elevators passing between the two of them.

3.9 Telecommunications

All telecommunication lines are supplied to the building from the southwest corner. A telecommunications room located on the first floor adjacent to the electrical room houses all equipment required for distribution to the rest of the building. 4 inch PVC conduit supplies telephone and internet to Intermediate Distribution Frame (IDF) rooms located on each floor. The systems are scheduled to be upgraded as part of the renovation but no details have been released to date.

3.10 Utility Rates

3.10.2 Electrical Rates

Electrical rates for Park Place 1 are based on Duquesne Light Company's *Schedule of Rates for Electric Service in Allegheny and Beaver Counties*. The building is considered General Service Large as its demand is not less than 300 kilowatts. The corresponding rate structure is shown below in **Figure 5**.

Duquesne Light Electricity Rates	
Demand Charges	
First 300 kW or less of Demand	$2,121.00
Additional kW of Demand	6.45 per kW
Energy Charges	
All kilowatt-hours	0.1236 cents per kWh
Transmission Service Charges	
Demand Charge	$3.05 per kW

Figure 5 – Electricity Rates

3.10.2 Natural Gas Rates

Natural Gas is provided to Park Place 1 by Columbia Gas of Pennsylvania. Columbia Gas of Pennsylvania considers the building Large General Sales Service. The gas tariff information is provided below in **Figure 6**.

Pennsylvania Natural Gas Rate in $ per Mcf				
	Distribution Charge	Gas Supply Charge	Gas Cost Adjustment	Total Effective Rate
First 1,000 Mcf per month	2.9708	6.1909	0.5988	9.7605
Next 4,000 Mcf per month	2.9094	6.1909	0.5988	9.6691
Next 5,000 Mcf per month	2.8766	6.1909	0.5988	9.6663
All Mcf per Month Over 10,000	2.6047	6.1909	0.5988	9.3944

Figure 6 – Natural Gas Rates

4.0 Existing Mechanical System Summary

4.1 Introduction

Park Place 1 has a central building mechanical system that serves 100% of the building to satisfy all heating, cooling, ventilation, and exhaust requirements. The building spaces are currently served by variable air volume (VAV) valves that allow for full mechanical modulation during part load occupancy. The base supply duct system is intended to suit future expansion with the assumption of VAV terminal boxes being used to supply air to individual spaces. Air will be supplied to these boxes through one of two vertical shafts that house both supply and return/exhaust air ducts. Two packaged rooftop air handling units (RTU) equipped with variable speed drives will split the building loads equally. Two existing gas fired boilers will meet the majority of the perimeter heating load by supplying hot water to perimeter duct reheat coils with the RTU's serving as the air handling unit and primary source of heating.

4.2 Design Criteria & Objectives

In the design of any system, several factors need to be weighed. The ultimate goal of a mechanical system is to provide air that both legally and practically meets the needs of the building occupants within the boundaries of cost. This is a relationship that involves three major parties: the owner, the occupant, and the government. Each party has needs that need to be addressed in the design process by the engineer. For the engineer to accomplish such a task, he or she must look at each party individually and then weigh the considerations. For the owner, system cost becomes the major focal point. This entails first cost, operating cost, and maintenance cost. For the occupant, air cleanliness, temperature, and humidity are the primary concerns all while maintaining a certain level of ventilation. For the government, compliance with modern codes is mandatory and therefore can be one of the basic starting points for the design engineer.

The mechanical system of Park Place 1 is intended to meet all of the requirements of ASHRAE Standard 55 – 2004 Thermal Environmental Conditions for Human Occupancy, ASHRAE Standard 62.1 – 2007 Ventilation for Acceptable Indoor Air Quality, and ASHRAE Standard 52.2 which pertains to particle removal from the supply air stream. The new mechanical equipment that was installed during the building renovation was intended to meet the requirements of ASHRAE Standard 90.1 – 2007 Energy Standard for Buildings Except Low-Rise Residential Buildings. For further study on compliance with ASHRAE Standards 62.1 and 90.1, please see Technical Report 1. Such topics address the needs of compliance with government standards.

With respect to building occupancy, Park Place 1 is exclusively an office building. As examples, there are no laboratories requiring very specific air quality conditions, no garages that need special exhaust system considerations, and no gyms that need a very precise temperature set point. To design a successful space, typical office building assumptions in accordance with the ASHRAE standards were made. This implies that occupants would be relatively sedentary and wearing normal clothing, internal loads would be predominantly driven by lighting, people, and receptacle loads, and that construction would be of medium to low quality because of age.

The owner, DiCicco Development, has cost in mind. Park Place 1 is a building that was designed for profit. DiCicco Development wants their occupants to be happy with their experience of renting one of their spaces. With that said, the building owner made it clear to the design team that it was their goal to provide an environmentally responsible building that at the same time satisfied the occupants who would be exposed to the systems. Because of the buildings age and the consideration that the building is to be rented for profit, DiCicco Development wanted to find the best solution that weighed first cost, operating cost, system efficiency, and maintainability. While certain modern systems could potentially have been more viable in the long term, DiCicco Development did not want a system with a lengthy payback period. Their goal was a system with a reasonably low initial cost and a consideration for operating cost. They wanted to find an economic balance. It was also made clear to the design engineer that the personnel in charge of maintenance, while experienced, was not sophisticated enough to handle an extremely complicated system. Also, because the building was not LEED rated previously, it did not become a major priority for the design team.

4.3 Outdoor and Indoor Design Conditions

In determining equipment capacity, both outdoor and indoor design conditions must be determined. Indoor design conditions are chosen in accordance with ASHRAE Standard 55 and are subject to personal preference amongst the building occupancy. In other words, not everyone agrees on what is comfortable and therefore a range of temperature control must be provided. Outdoor design conditions are based on TMY2 weather data which is collected over years of recorded weather data and trends.

Park Place 1 is located in Findlay Township, Pennsylvania, a suburb of Pittsburgh, Pennsylvania. Because Pittsburgh is the closest major city that has weather data accumulated and documented, it was used as the basis of location for design. Pittsburgh is known for having relatively cold winters and warm, humid summers as seen below in **Figure 7**. The 0.4% and 99.6% design days were chosen to be used for equipment selection for Park Place 1. Together, both outdoor and indoor conditions must be considered to appropriately size mechanical equipment.

Outdoor Design Conditions		
	Summer (0.4 %)	Winter (99.6 %)
Dry Bulb (°F)	89.1	1.8
Wet Bulb (°F)	72.5	-
Dew Point (°F)	65.6	-
Clearness	0.97	0.97
Ground Reflectance	0.2	0.2
Wind Velocity	11.7	15

Figure 7 - Outdoor Design Conditions

In **Figure 8** below, the indoor design conditions can be seen. They are the typical set points for an office building.

Thermostat Settings	
Cooling Dry Bulb (°F)	75
Heating Dry Bulb (°F)	70
Relative Humidity (Cooling Only) (%)	50
Cooling Drift point (°F)	81
Heating Drift point (°F)	64

Figure 8 - Indoor Design Conditions

4.4 Ventilation Requirements

An in depth ventilation requirement study was performed on the building in Technical Report 1 for Park Place 1. The results of that report can be seen below in **Figure 9**.

Outdoor Air Requirement		
E_v		0.9
Max Z_p		0.16
CFM of OA required- 1st Floor		3154
CFM of OA required- 2nd through 5th Floor		11411
Total Building OA Requirement (CFM)		14565
Available OA (CFM)	Minimum	9,000
	Maximum	up to 90,000

Figure 9 – Outdoor Air Requirement

An individual space analysis was done for all occupied spaces. Only the totals are shown in **Figure 9** above. The important thing to note is that only the base building systems are truly being analyzed due to the fact that the building is a tenant fit out. The base building systems are more than capable of delivering the proper amount of outdoor air.

4.5 Air Supply System

Packaged Rooftop Units

The building air supply is handled entirely but two identical rooftop units (RTU-1, RTU-2) for the heating and cooling seasons.

For cooling, the RTU's are direct transfer (DX) type, meaning the air stream is cooled by a cooling coil that has liquid refrigerant circulating through it. The RTU's have air-cooled condensers with accompanying fans that increase the heat transfer rejection rate to the ambient surroundings to turn the refrigerant from a compressed gas back to a liquid. The compressors are direct drive scroll type with hermetic motors. The supply air temperature from the RTU's can be modulated along with the supply cfm's, but for capacity was sized for a leaving air temperature of 55°F (the desired supply air temperature to the occupied space).

For heating, the RTU's have forced draft gas burners that are capable of providing 85°F air. When speaking with the design engineer, it was determined that the existing gas-fired boilers were capable of handling the entire heating load for the building and that the RTU's would be used as the main heating source following the renovation. During the heating season, the boilers' heating capacity will be used as a redundant back up, the primary purpose being to supply 180°F water to reheat coils around the building perimeter.

RTU-1 and RTU-2 are responsible for cooling air during the summer, warming air during the winter, and also moving air throughout the building during both seasons. There are no other air handling units in the building. To meet the air handling requirement, both RTU's have a supply (airfoil type) and return (forward curved type) fan equipped with variable speed drives that allow for full modulation of supply and return air quantities. The RTU's are also equipped with a 100% outdoor air economizer which allows for each RTU to serve as a dedicated outdoor air system should the opportunity present itself. The economizers, in combination with the unit controls, are also capable of demand control ventilation based on CO_2 measurements taken in the occupied spaces. The RTU's are capable of providing 45,000 cfm's of supply air each. For air quality purposes, MERV 7 prefilters and MERV 13 final filters have been installed into the units to remove potential air contaminants. For safety, the units have been equipped with smoke detectors in both the supply and return ducts that are wired directly to the units' control systems.

Because controls are an ever increasing priority in the HVAC industry, the RTU's have been equipped with microprocessor controls. This system consists of temperature and pressure (thermistor and transducer) sensors and a human interface panel that are capable of tying into the building automation system (BAS) that is included as part of the building renovation.

The RTU's performance characteristics can be seen below in **Figure 10** – Rooftop Unit Schedule.

Rooftop Unit Schedule			
Name	RTU-1	RTU-2	Units
Air Quantity	45,000	45,000	cfm
Minimum Outdoor Air	4,500	4,500	cfm
Heat Output	1,100,000	1,100,000	Btu/Hr
Gas Input	1,380,000	1,380,000	Btu/Hr
Entering Air Temperature- Heating	63	63	°F
Leaving Air Temperature- Heating	85	85	°F
Cooling Capacity	115	115	Tons
Entering Air Temperature DB- Cooling	77.5	77.5	°F
Entering Air Temperature WB- Cooling	64.3	64.3	°F
Ambient Temperature- Cooling	92.0	92.0	°F

Figure 10 – Rooftop Unit Schedule

Distribution

An essential part of any mechanical system is the delivery of air from the air handling unit to the occupied space. In this case, outdoor air and return air mix in the packaged rooftop units' mixing boxes where a portion of that air is exhausted, the rest of be recycled and sent through the system. Once that mixed air is re-filtered and re-conditioned in one of the two RTU's, it is pushed through one of two central shafts that run vertically through the center of the building. From these shafts, main branch ducts at every floor deliver supply air to individual terminal boxes where the air is then supplied to the space. If, in some cases, the run of duct work is over an extended length of duct, hot water duct reheat coils have been installed to increase the supply air temperature during the heating season. This is especially applicable to perimeter spaces. If a future designer desires the use of a fan-coil unit or a reheat coil in a VAV box, the ability to tap off of the main hot water supply line is feasible.

Supply Fans				
Unit	Type	hp	CFM	Service
RTU-1	AHU	75	45000	Whole Building
RTU-2	AHU	75	45000	Whole Building
FPCV-A	Terminal	1/6	200	Office Space
FPCV-B	Terminal	1/6	350	Office Space
FPCV-C	Terminal	1/4	750	Office Space
FPCV-D	Terminal	1/2	1000	Office Space
FPCV-E	Terminal	3/4	1400	Office Space
FPCV-F	Terminal	1	1800	Office Space
FPCV-G	Terminal	1	2300	Office Space
FPVV-A	Terminal	1/6	200	Office Space
FPVV-B	Terminal	1/6	350	Office Space
FPVV-C	Terminal	1/4	750	Office Space
FPVV-D	Terminal	1/2	1000	Office Space
FPVV-E	Terminal	3/4	1400	Office Space
FPVV-F	Terminal	1	1800	Office Space
FPVV-G	Terminal	1	2400	Office Space

Figure 11 – Supply Fan Data

Return/Exhaust Fan Compliance				
Unit	Type	hp	CFM	Service
EF-1	Exhaust	1	3500	Restrooms
RTU-1	Return	40	40500	Whole Building
RTU-2	Return	40	40500	Whole Building

Figure 12 – Return/Exhaust Fan Data

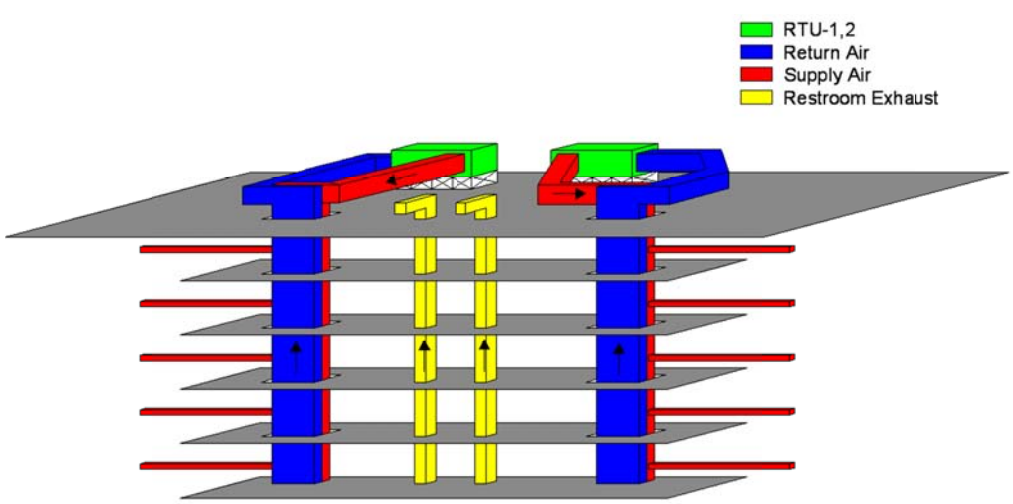

Figure 13 – Air Circulation Schematic

Seen above in **Figure 13**, the air circulation for Park Place 1 begins and ends on the roof. In **Figure 13**, the RTU's shown in green are located just outside of the rooftop penthouse. They supply fresh air through the ducts shown in red to all occupied spaces on floors one through five. The air leaves from the unit, enters into the rooftop penthouse, is pushed down through one of two vertical shafts, and then is distributed through branch ducts to terminal boxes. The blue ducts shown are similar except that in place of branch ducts, a pressurized ceiling plenum returns air to the RTU's. The two ducts shown in yellow are restroom exhaust shafts that are completely separate of the rest of the mechanical system. These two shafts are powered by to exhaust fans located within the rooftop penthouse. **Figure 14** below depicts a more technical schematic of the same system.

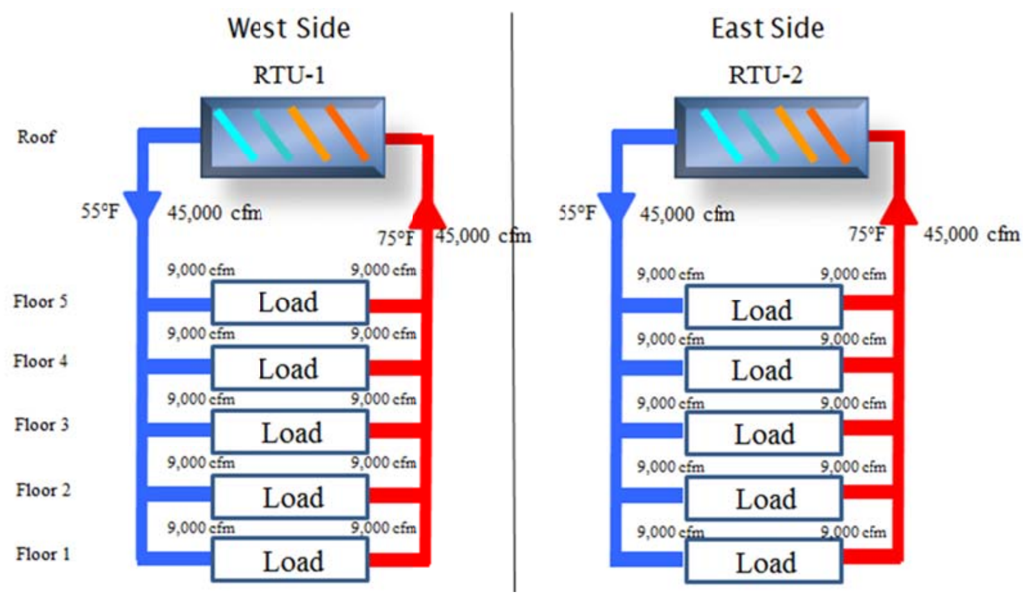

Figure 14 – Air Volume Flow Rate Schematic

Terminal Units

Because the building is a tenant fit out, there is a variety of potential system designs that could be implemented in conjunction with the base building mechanical systems. According to the design engineer, VAV boxes, fan-powered boxes, or a number of other terminal units can be used to supply air to individual spaces. In the event that supplemental heat is needed, hot water from the boiler can be made available. It is the intention of the owner and designer however, that the boiler be used as little as possible.

4.6 Air Exhaust/Return System

Once the air has passed through the occupied space, it is returned through a pressurized ceiling plenum where it is drawn by a return fan in the RTU's. A certain percentage of that air is exhausted, the remainder to go through the process again. Restroom exhaust is handled by a separate duct that runs through a separate vertical shaft (one additional shaft for each side of the building). Any additional exhaust requirements, such as kitchen hoods can be connected to the restroom exhaust shafts. 100% of this restroom air is exhausted; none of it is recirculated.

4.7 Evaluation of Mechanical System

When Park Place 1 was originally constructed, it was done so with the intention that an interior air handler would use a split system for cooling and a gas fired boiler for heating. Because the two systems were separate, the controls were complicated and the system did not perform well. This caused inefficiencies and unhappy tenants. When DiCicco Development purchased the building and decided to improve the mechanical systems, they made an excellent choice to consolidate the two systems into one.

While some buildings have a hot/chilled water system, the design engineers saw an opportunity to economically use an air only system. Park Place 1 does have a hot water system but no chilled water system. To combine the two systems into one with a changeover would have been costly and required the further purchase of cooling equipment. The option of using modern rooftop units capable of providing both heating and cooling would eliminate the need for the boiler, pumps, and water system as a whole. Also, because the RTU's are new, they function more efficiently, are more reliable, easier to control, and require one service contract as opposed to several. While they are probably not the most energy efficient or cheapest choice over the life of the equipment, they come with minimal first cost and reliability. When the design considerations were taken into account, modern rooftop units were probably one best choices to meet the needs of the owner.

There is still some room for investigation however. Park Place 1 offers many opportunities for systems with higher first cost but lower operating cost. These systems will be explored throughout the rest of this study.

5.0 Existing Mechanical System Performance

5.1 Design Loads

Trane Trace 700 was used to perform an energy analysis on Park Place 1. The results of the energy analysis can be seen in **Figure 15** below.

Design Load Summary		
Cooling	Peak Cooling Airflow:	84,700 cfm
	Peak Ventilation Load:	28.3 Tons
	Peak Internal Load:	67.2 Tons
	Peak Envelope Load:	113.6 Tons
	Peak Pump/Fan Load:	15.9 Tons
	Total Peak Load:	225.0 Tons
Heating	Peak Heating Airflow:	25,000 cfm
	Peak Ventilation Load:	-1,041,317 Btu/h
	Peak Internal Load:	16,628 Btu/h
	Peak Envelope Load:	-907,421 Btu/h
	Total Peak Load:	-1,932,110 Btu/h

Figure 15 – Design Load Summary

5.2 Design Energy Consumption

The annual energy consumption was calculated by Trace 700 as a part of the energy model for the entire building. The annual energy consumption of the building is divided into electricity which is used by the RTU's, fans, pumps, receptacles, and lights. The other source of energy is gas which is consumed entirely for heating purposes. Shown below in **Figure 16** and **Figures 17** and **18** are the output reports from Trace 700 providing numerical outputs and their corresponding graphs respectively.

	Monthly Energy Consumption													
	Month	January	February	March	April	May	June	July	August	September	October	November	December	Total
Electric	Consumption (kWh)	41,722	37,629	47,068	47,267	67,150	78,524	81,488	78,084	59,684	47,764	43,337	40,047	669,764
	Demand (kW)	282	291	331	368	421	465	488	472	455	371	318	273	488
Gas	Consumption (therms)	7,887	7,389	5,870	1,914	153	0	0	0	268	2,487	3,739	6,650	36,357

Figure 16 – Monthly Energy Consumption

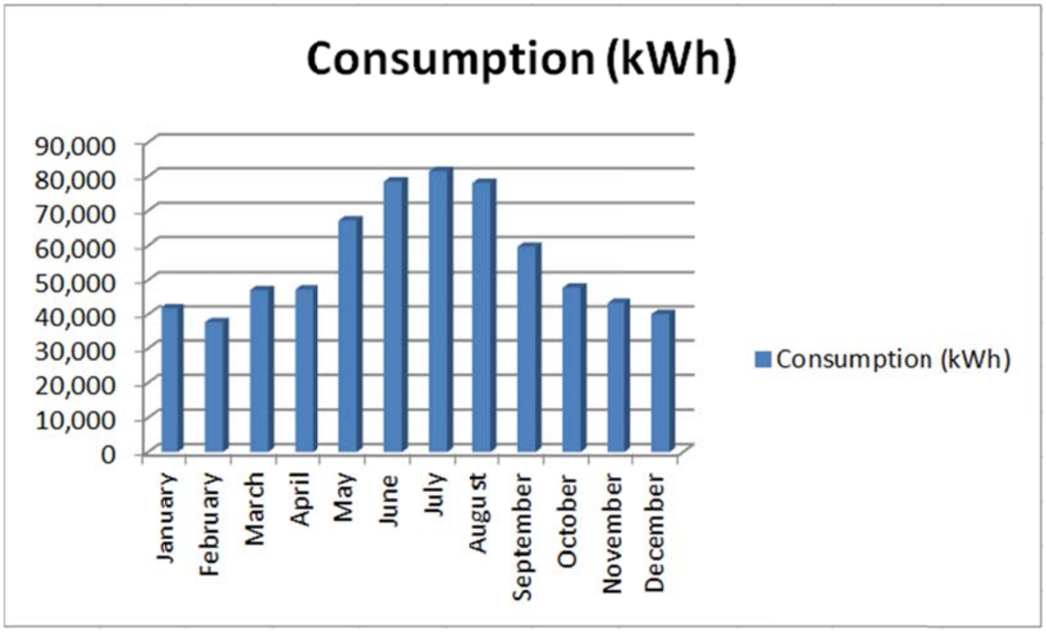

Figure 17 – Electric Consumption

Shown above in **Figure 17** is the electricity consumption expressed in kWh's. The graph demonstrates that electricity will be used throughout the entire year, with the peak occurring during the summer months when the cooling load is at its maximum. The electricity demand shown below in **Figure 18** demonstrates a similar concept of maximum load occurring during July, usually the warmest month.

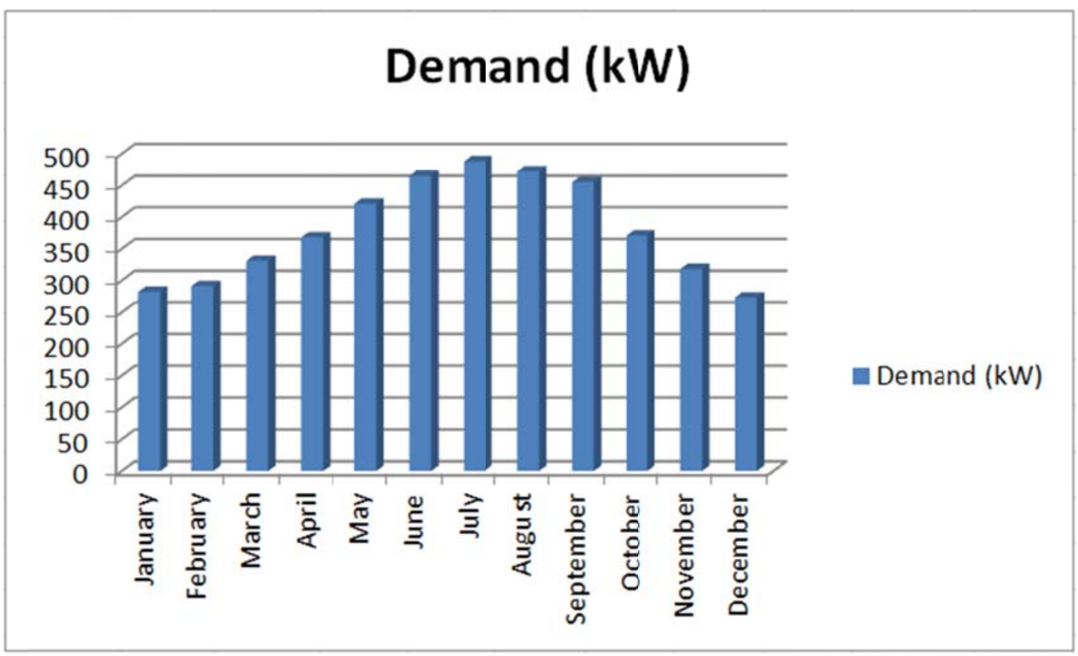

Figure 18 – Electric Demand

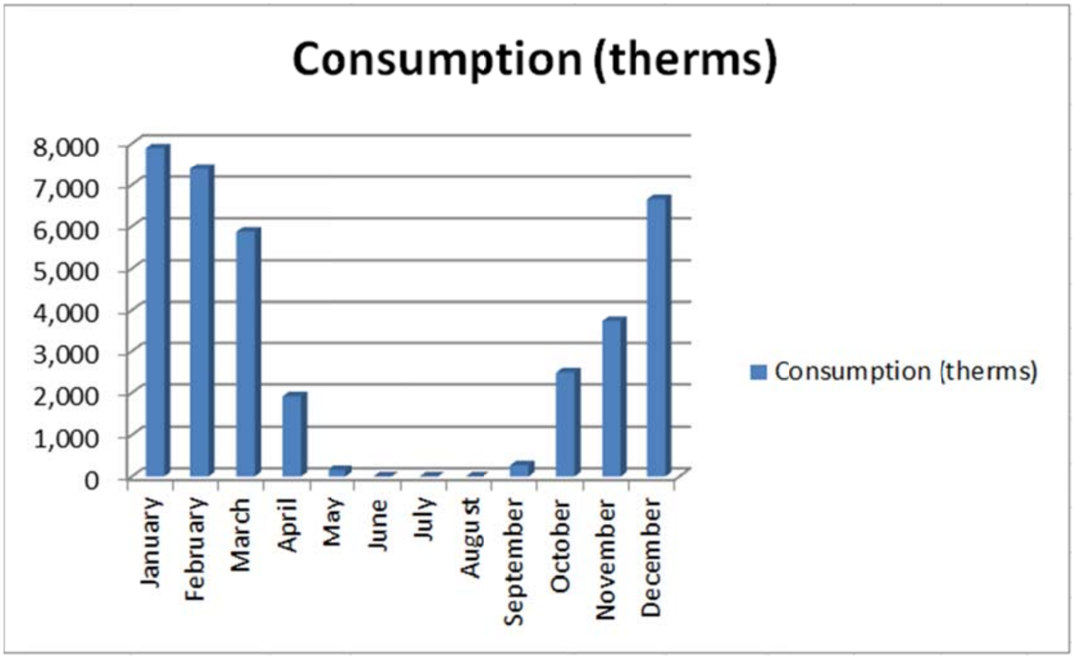

Figure 19 – Gas Consumptions

It is evident that from the figures above, the predominant consumption of energy will be in the form of electricity. Electricity is used in the building throughout the entire year as compared to gas which is used just during the heating months. **Figure 18** above shows that a peak shaving strategy during the summer months could be an excellent way to reduce cost by reducing the electricity demand on the building.

6.0 Proposed Redesign

6.1 Overview & Reasoning

The proposed redesign consists of four sections—the original design and three alternatives. For simplicity, the original design will be referred to as Alternative 1 with the subsequent alternatives as 2, 3, and 4. Each alternative introduces a single new variable to the proceeding alternative (indicated in red in **Figure 20**). Alternative 2 introduces a chilled water system that includes an air-cooled rotary liquid chiller and two air handling units (AHU) to replace the packaged rooftop units. Alternative 3 adds thermal energy storage (TES) to the system presented in Alternative 2. Finally, Alternative **4** drops the supply air temperature of Alternative 3 by 5°F. The alternatives are displayed below in **Figure 20**.

Alterative Name	Cooling Equipment	Heating Equipment	Air Handling Equipment	Air Distribution Temperature
Alternative 1	Packaged RTU	Packaged RTU	Packaged RTU	55°F
Alternative 2	Single Air-Cooled Chiller	Gas-Fired Boiler	Independent AHU	55°F
Alternative 3	Single Air-Cooled Chiller + TES	Gas-Fired Boiler	Independent AHU	55°F
Alternative 4	Single Air-Cooled Chiller + TES	Gas-Fired Boiler	Independent AHU	50°F

Figure 20 – Alternatives Summary

Alternative 2 provides a control from which to measure the effectiveness of Alternatives 3 and 4. At the end of that analysis, a comparison to the original system designed, Alternative 1, will be discussed.

As an office building, Park Place 1 has a low load factor of around 34%. As a result, an opportunity for TES could save the building owner money in up front equipment cost as well as building operating cost. This study seeks to find an energy efficient and economically viable alternative to the original design, and TES could be the answer.

Thermal energy storage offers several advantages. First, the initial capital cost of a TES system can be lower than a comparable system without. This results from the opportunity for a reduced sized chiller to operate from 20 to 24 hours a day at maximum capacity as opposed to a larger chiller operating for a shorter period of time. Second, a smaller chiller would demand less energy which would reduce the peak demand charge from the utility provider. Third, TES systems operate during the night when ambient temperatures and utility rates are lower. This implies that the coefficient of performance (COP) would be higher and the cost of energy would be cheaper. Act 129, passed in 2008, imposed new requirements on electric distribution companies (EDC) with the goal of reducing energy consumption and demand. As an offshoot of this act and with increasing demand for power, most EDC's will implement a time-of-day usage charge which will reduce the cost of using energy during the night and will penalize heavy users for consumption during peak hours. Should such measures be taken, TES will become even more profitable for Park Place 1. The combination of such strategies could save the owner money and will be analyzed in the following sections.

6.2 Alternative 2 – RTU Substitution with Chilled Water System

6.2.1 System Description and Components

The two packaged rooftop air handling units from Alternative 1will be replaced with a chilled water system and two identical air handling units. The chilled water is generated by a single air-cooled helical rotary chiller located on the roof of Park Place 1. This chilled water serves cooling coils contained in both air handling units also located on the roof of the building. The system requires pumps, actuators, and balancing valves to ensure proper control. Both the pumps and air handling units are equipped with variable speed drives which implies variable refrigerant flow through the chiller and variable air volume supplied by the air handling units.

Heating will be handled by the existing boiler and pumping system. Hot water produced by this system will be piped to the air handling unit.

The preliminary system schematic is shown below in **Figure 21**.

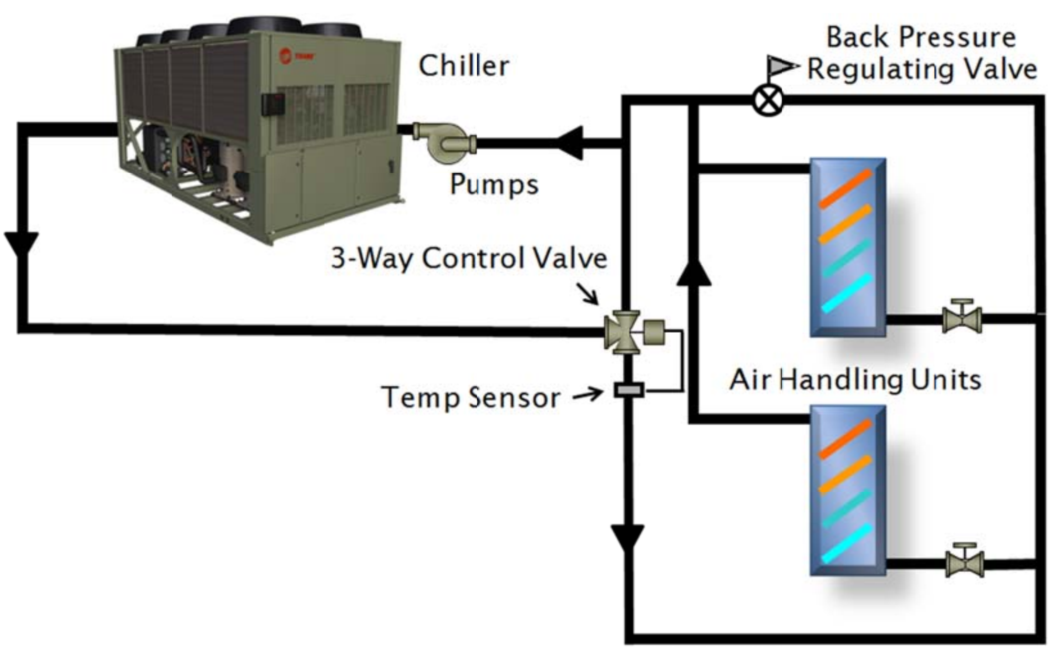

Figure 21 – Preliminary Chilled Water System Schematic

6.2.2 Assumptions

For assumptions used in **Section 6.2.3** – Sizing Calculations, please see **Figure 22** – Alternative 2 Assumptions below.

Assumptions							
Equipment	Property						
Chiller	Condenser	Heat Rejection	Ambient Air Temp				
		Air Cooled	92°F				
		Fluid Properties			Load		
	Evaporator	Entering Temp	Leaving Temp	Fluid	Total	Sensible	
		55°F	45°F	Water	230 Tons	172 Tons	
	Electrical	Phase	Voltage	Frequency			
		3 Ph	460 V	60 Hz			
Individual Air Handling Unit		Water Side			Air Side		
		Flow	Entering Water Temp	Leaving Water Temp	Entering DB	Leaving DB	Airflow
	Cooling Coil	276 GPM	45°F	55°F	77.5°F	55°F	45,000 CFM
	Heating Coil	140 GPM	180°F	160°F	63°F	85°F	45,000 CFM
		Outdoor Air		Return Air		Mixed Air	
	Return Fan	Temp	Airflow	Temp	Airflow	Temp	Airflow
		92°F	6,750 CFM	75°F	38, 250 CFM	77.5°F	45,000 CFM
		Supply Air					
	Supply Fan	Temp	Airflow				
		55°F	45,000 CFM				

Figure 22 – Alternative 2 Assumptions

6.2.3 Sizing Calculations

6.2.3.1 Chiller

To select a chiller, the flow rate, temperature drop across the evaporator, ambient temperature, and nominal tonnage are required for a preliminary selection. Once this selection has been made, the performance of the chiller must be compared with the requirements of the system to ensure the chiller is capable of handling the loads associated with the air handling unit. Of course many other variables and decisions are required, but these selections are not calculation based and will be summarized in Section 6.2.4. The only sizing calculation that needed to be performed was to determine the maximum flow rate through the chiller during the design condition given the assumed variables. The calculation is shown below.

$$Q_{TOTAL} = 500 * \text{Volume Flow Rate [GPM]} * (T_{EVAP,IN} - T_{EVAP,OUT})$$

$$230 \text{ [Tons]} * 12{,}000 \text{ [Btu/h/Ton]} = 500 * \text{Volume Flow Rate [GPM]} * (55°F - 45°F)$$

$$\text{Volume Flow Rate} = 552 \text{ GPM}$$

$$\text{Volume Flow Rate per AHU} = GPM_{TOTAL} / 2 \text{ AHU's}$$

$$\text{Volume Flow Rate per AHU} = 276 \text{ GPM}$$

Q_{TOTAL} = *Total Peak Load on Cooling Coil in Air Handling Unit [Btu/h] – As taken from energy model*

Once the design flow rate was determined, a chiller selection was made based on 230 nominal tons, 552 GPM, a 10°F ΔT across the evaporator, and 92°F ambient air.

6.2.3.2 Air Handling Unit

With the total building sensible load known from the energy model, the required air supply volume was calculated given an assumed temperature difference between supply air and room air. The calculation for needed capacity is shown below.

Airside

$$Q_{SENSIBLE} = 1.08 * \text{Volume Flow Rate [cfm]} * (T_{BEFORE\ COIL} - T_{AFTER\ COIL})$$

$$2058.2 \text{ MBh} = 1.08 * \text{Volume Flow Rate [cfm]} * (77.5°F - 55°F)$$

$$\text{Volume Flow Rate}_{REQUIRED} = 84{,}700 \text{ cfm}$$

As a measure of safety, the design supply air volume used will be 90,000 cfm, which agrees with the original design as done by the engineer.

Waterside

Because the chiller is equipped with a variable frequency drive, the chiller can be modulated to meet the load on the cooling coil in the air handling unit. The waterside calculation for the cooling coil is identical to the chiller calculation assuming a 10 degree temperature rise across the coil.

6.2.3.3 Piping

Pipe sizing will be done in accordance with the ASHRAE Handbook of Fundamentals. An assumed pressure drop of 4 feet of head loss per 100 feet of distance will be assumed. At 552 GPM, an associated pipe diameter of 8 inches was selected. Smaller pipe size selections can be seen in the system schematic in Section 6.2.6.

6.2.4 Equipment Selection and Performance

All equipment has been selected using Trane Official Product Selection System (TOPSS). Therefore, all selected equipment is manufactured by Trane and has performance characteristics and prices as determined by Trane.

6.2.4.1 Chiller

An image of the selected chiller can be see in **Figure 23**. Performance information of the chiller can be seen below in **Figure 24**.

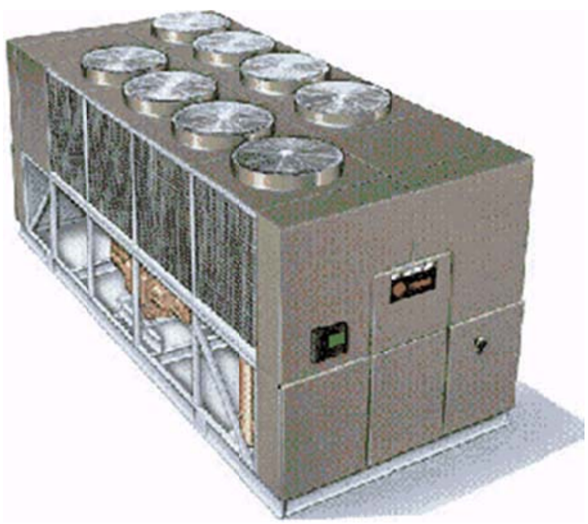

Figure 23 – Trane Air-Cooled Chiller: Model RTAC

Chiller Selection		
General	Unit Name:	CH-2
	Basis of Selection:	Trane
	Trane Model:	Air-Cooled Series R(TM) (RTAC)
	Unit Type:	Standard Efficiency
	Capacity:	246.5 tons
	Efficiency:	10.1 EER
	COP:	2.97
	IPLV:	13.6 EER
	NPLV:	14.0 EER
	Refrigerant:	HFC-134a
	Shipping Weight:	14,507.0 lb
	Dimensions (LxWxH):	268" x 89" x 93"
	Rated Capacity (AHRI):	237.20 tons
	Rated Efficiency (AHRI):	9.6 EER
Evaporator	Leaving Temp:	45.00°F
	Entering Temp:	55.00°F
	Flow Rate:	589.10 GPM
	Pressure Drop:	18.30 ft H2O
	Configuration:	2 Pass
	Fluid Type:	Water
Electrical	Unit Voltage:	460V/60Hz/3Ph
	Unit Power:	292.30 kW
	Compressor Power:	271.30 kW
	Fan Motor Power:	20.30 kW
	Number of Condenser Fans:	14
Control Accessories	VFD:	Yes
	Ice Making:	No

Figure 24 – Chiller Performance

6.2.4.2 Air Handling Unit

The selected air handling unit is a Trane T-Series Climate Changer Exterior Air Handler. It consists, in order, of a return fan with a VFD, dry bulb 100% outdoor air economizer, short-bag type filter section, horizontal hot water heating coil, access section, horizontal chilled water cooling coil, supply fan with a VFD, and a discharge plenum. Performance and other information of each section can be seen below in **Figure 25**. **Figures 26** and **27** depict fan curves for the return and supply fans respectively.

Air Handling Unit Selection		
General	Unit Name:	AHU-1, AHU-2
	Basis of Selection:	Trane
	Trane Model:	Outdoor T-Series Climate Changer
	Unit Type:	Standard Efficiency
	Shipping Weight:	24,628.6 lb
	Dimensions (LxWxH):	473" x 154" x 120"
Return Fan	Fan Size:	40"
	Fan Type:	FC
	Motor HP:	40 hp
	Break HP:	43.5 hp
	Fan Airflow:	45,000 CFM
	External Static Pressure:	1.5" H2O
	Total Static Pressure:	2.842" H2O
	Speed:	434 rpm
	Motor Class:	ODP NEMA Premium Efficiency
	Motor Voltage:	460V/60Hz/3Ph
	VFD:	Yes
	Fan Module Pressure Drop:	1.849" H2O
Economizer	OA Capability:	0-100%
	Total OA Pressure Drop:	0.428" H2O
	Return Damper Pressure Drop:	0.551" H2O
	Exhaust Damper Pressure Drop:	0.551" H2O
	Supply Fan Pressure Drop:	0.428" H2O
Filter Section	Prefilter Type:	Pleated media coated - MERV 7
	Prefilter Pressure Drop:	0.620" H2O
	Primary Filter Type:	Short Bag 85% - MERV 13
	Primary Filter Pressure Drop:	0.813" H2O
	Total Pressure Drop:	1.433" H2O

Continues to next page.

Heating Coil	Rows:	2
	Fin Spacing:	80 per foot
	Fin Material:	Aluminum
	Tube Material:	Copper
	Airflow:	45,000 CFM
	Face Velocity:	589 ft/min
	Entering Dry Bulb	63°F
	Leaving Dry Bulb:	90°F
	Air Pressure Drop:	0.161" H2O
	Entering Water Temperature:	180°F
	Leaving Water Temperature:	160°F
	Fluid Pressure Drop:	6.73' H2O
	Fluid Flow Rate:	132 GPM
	Total Capacity:	1321.75 MBH
Cooling Coil	Rows:	4
	Fin Spacing:	123 per foot
	Fin Material:	Aluminum
	Tube Material:	Copper
	Airflow:	45,000 CFM
	Face Velocity:	451 ft/min
	Entering Dry Bulb	77.5°F
	Leaving Dry Bulb:	55°F
	Entering Wet Bulb:	64.30°F
	Leaving Wet Bulb:	54.12°F
	Air Pressure Drop:	0.402" H2O
	Entering Water Temperature:	45°F
	Leaving Water Temperature:	55°F
	Fluid Pressure Drop:	13.14' H2O
	Fluid Flow Rate:	272.32 GPM
	Sensible Capacity:	1,113.4 MBH
	Total Capacity:	1,366.32 MBH

Supply Fan	Fan Size:	40"
	Fan Type:	AF
	Motor HP:	75 hp
	Break HP:	75.336 hp
	Fan Airflow:	45,000 CFM
	External Static Pressure:	3.5" H2O
	Total Static Pressure:	6.511" H2O
	Speed:	1,112 rpm
	Motor Class:	ODP NEMA Premium Efficiency
	Motor Voltage:	460V/60Hz/3Ph
	VFD:	Yes
	Fan Module Pressure Drop:	4.067" H2O

Figure 25 – Air Handling Unit Selection

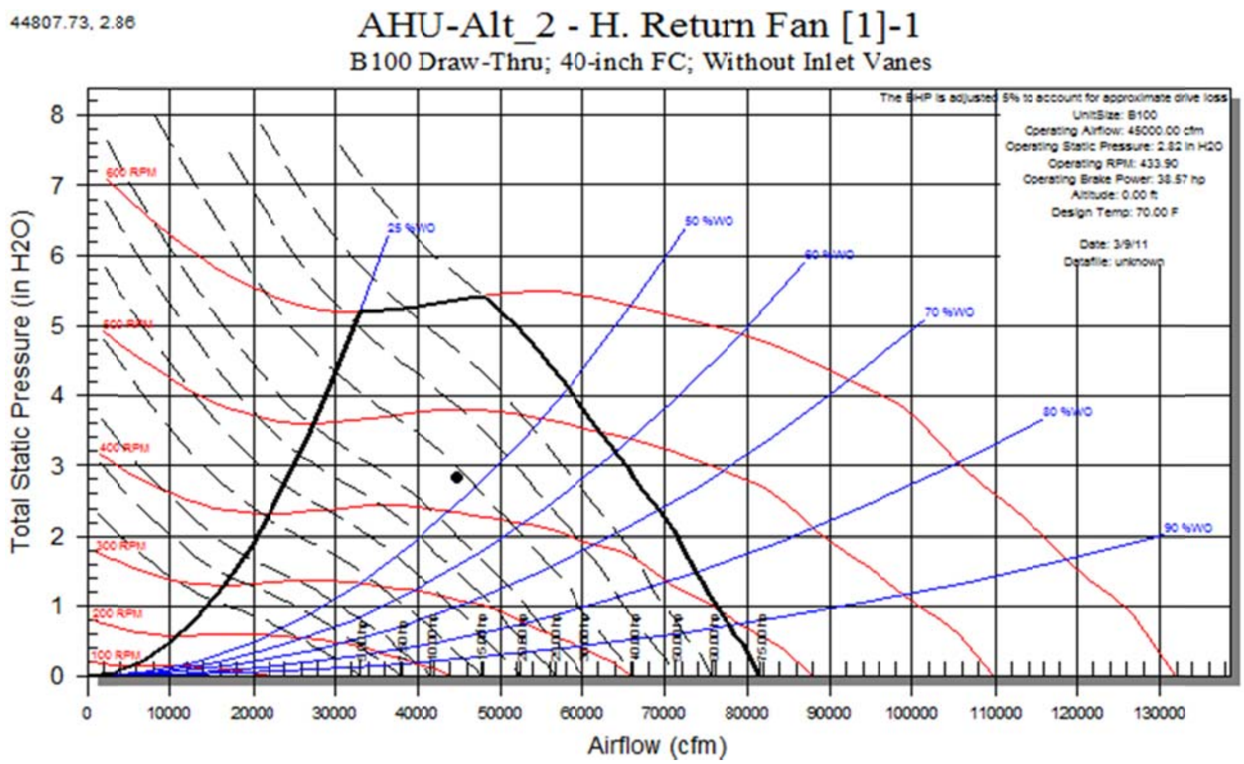

Figure 26 – Return Fan Performance Graph

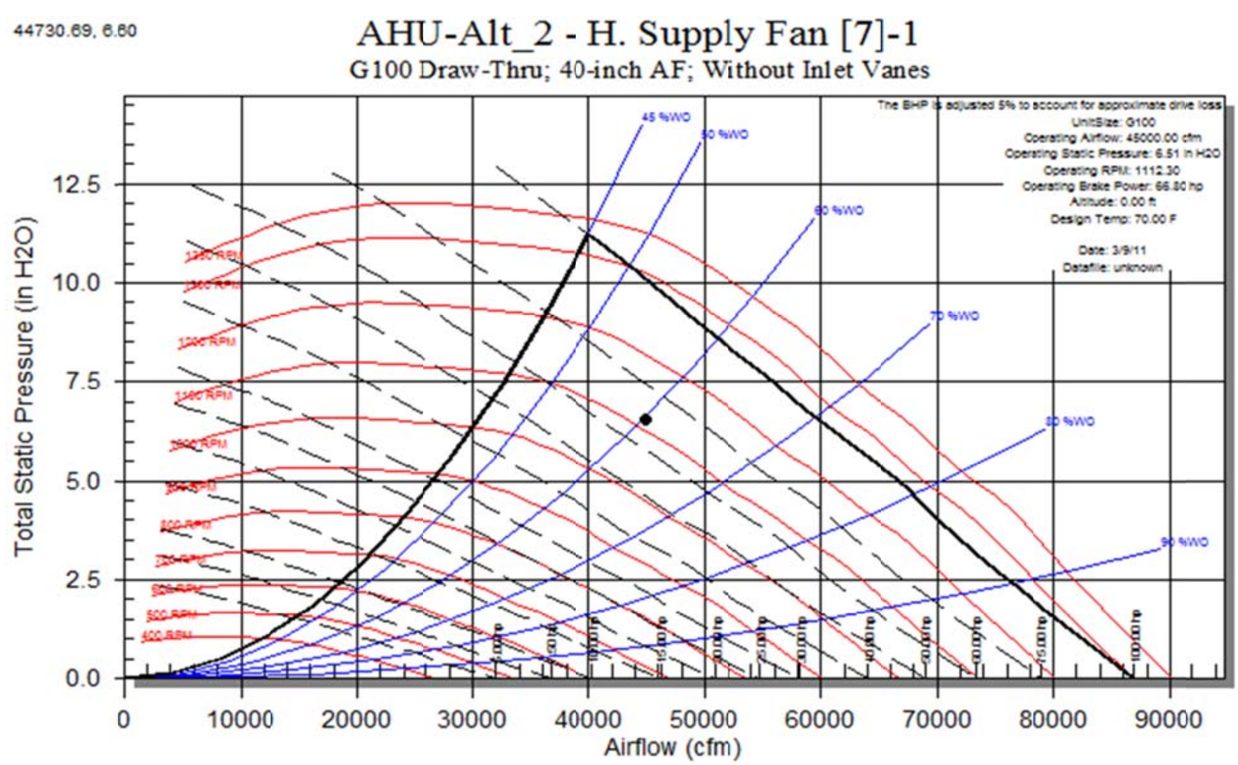

Figure 27 – Supply Fan Performance Graph

6.2.5 Analysis

With any chilled water system, it is important to note that the chiller and air handling unit will need to function in unison with each other. The performance and control of one will affect the other. The design began with the load to the space, was translated to the air handling unit supply air volume, which in turn dictated the cooling coil capacity requirement, which then finally determined the chiller size.

A comparison of design and performance points for both the air handling unit and chiller can be seen below in **Figure 28**. For the chiller selection, the fluid flow rate and rated capacity are slightly oversized. This is due to the fact that no manufacturer fabricates custom equipment at such small sizes. The nominal chiller size selected was 230 tons, however, at the given design parameters, the capacity was slightly larger than desired. The choice to make the system variable primary flow will allow the chiller to operate right at the design condition which will improve efficiency and performance.

Once the supply air volume was determined based on the load requirement of the occupied space, overall AHU size could be selected. This would dictate module size for fans, economizer, filters, and coils. The primary analysis took place in the design of the cooling coil. **Figure 28** shows that, like the chiller, the flow rate is slightly larger than designed. For both pieces of equipment, this is probably due to fouling and other realistic losses that are not accounted for in design calculations. What is most important however, is that the rated capacity of the coil is very close to the design requirement. For sensible capacity, the coil performs slightly better than needed. In an office building, an accurate balance of sensible and latent cooling is generally not vital. A slightly lower dry bulb temperature and slightly elevated wet bulb temperature can be acceptable. For the total capacity, the coil is undersized by around one ton of cooling at the given flow rate and temperature drop. This does not imply that the coil will underperform though. Once installed, the coil can modulate both the flow rate and temperature drop so that the required capacity could be met. This increase in required capacity would be seen by the chiller which is, as previously stated, oversized.

Design vs. Equipment Performance				
Unit	Subunit	Property	Design	Equipment Performance
Chiller	Evaporator	ΔT Fluid	10°F	10°F
		Flow Rate	552 GPM	589.10 GPM
		Capacity	230 Tons	237.2 Tons
AHU	Return Fan	Air Flow	45,000 CFM	45,000 CFM
		ESP	1.5" H2O	1.5" H2O
	Cooling Coil	Air Flow Rate	45,000 CFM	45,000 CFM
		ΔT Air	22.5°F	22.5°F
		Water Flow Rate	276 GPM	294.55 GPM
		ΔT Fluid	10°F	10°F
		Sensible Capacity	1029.1 MBH	1,113.4 MBH
		Total Capacity	1389.4 MBH	1,366.32 MBH
	Supply Fan	Air Flow	45,000 CFM	45,000 CFM
		ESP	3.5" H2O	3.5" H2O

Figure 28 – Design vs. Equipment Performance

6.2.6 Schematics

The chilled water system schematic can be seen in **Figure 29** below. The air distribution system schematic can be seen in **Figure 30**.

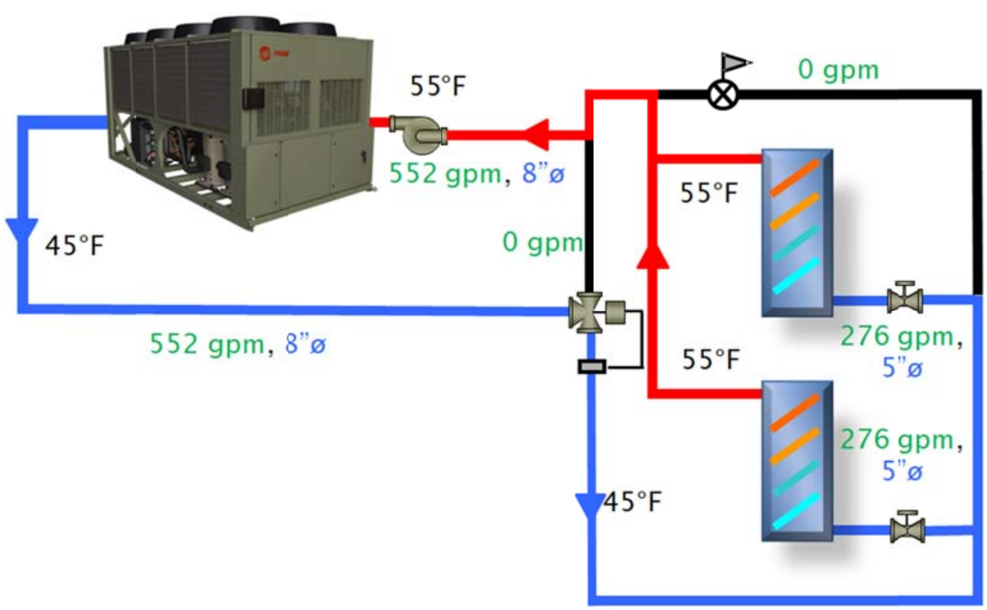

Figure 29 – Final Chilled Water System Schematic

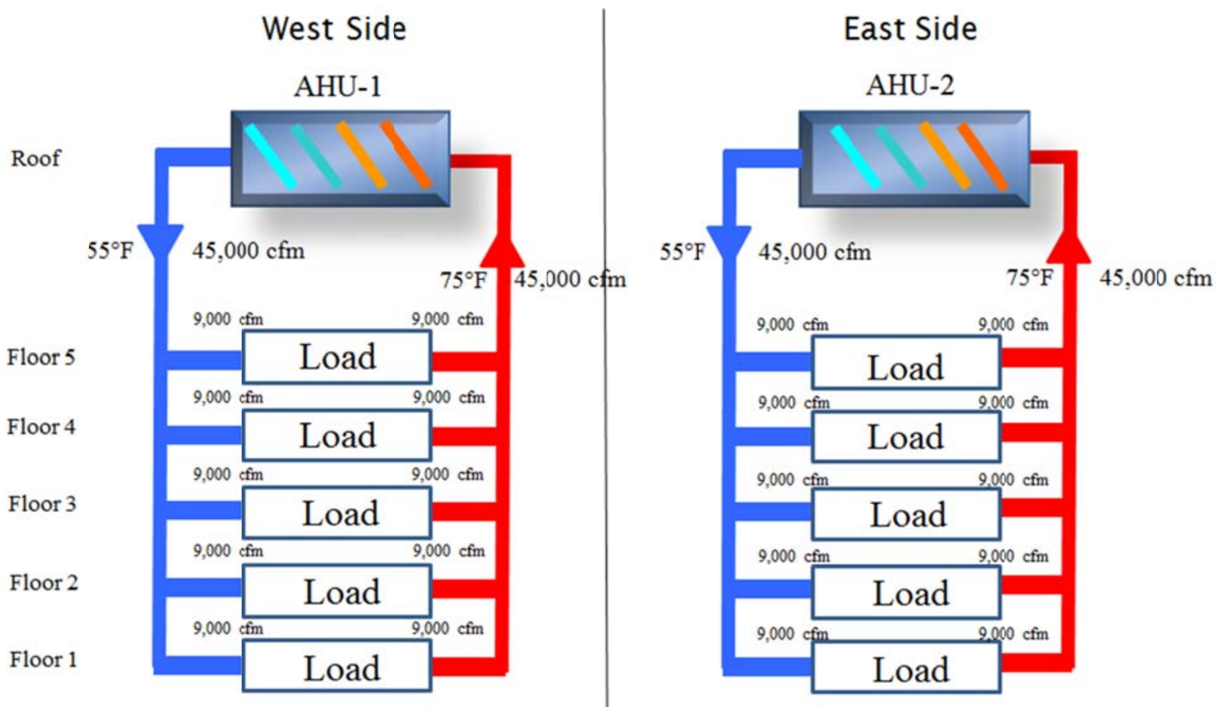

Figure 30 – Final Air Distribution System Schematic

6.3 Alternative 3 - Thermal Energy Storage

6.3.1 System Description and Components

Alternative 3 will assume all components of Alternative 2 with the addition of ice storage tanks. The chiller in Alternative 3 will be smaller, have less capacity, and will be cheaper than that of Alternative 2, but all other equipment will remain and perform the same. The ice storage tanks will use Calmac as the basis of design.

Below in **Figure 31** is the preliminary schematic to provide an initial understanding.

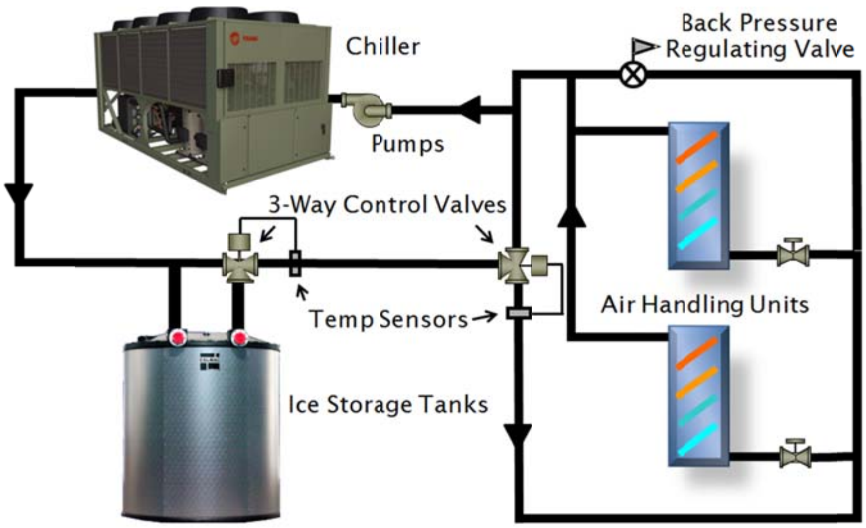

Figure 31 – Preliminary Ice System Schematic

6.3.2 Research and Assumptions

The following research and assumptions in **Figures 32 and 33** were collected from the Calmac website, Paul Valenta (The Calmac National Sales Representative), and the Pittsburgh Trane Engineering Sales Team. **Figure 32** presents the chiller assumptions for both the ice making and not ice making conditions. **Figure 33** shows the assumptions for the ice storage tanks at the same instances.

Assumptions				
Equipment	Property			
Chiller Ice Making	Condenser	Heat Rejection	Ambient Air Temp	
		Air-Cooled	77°F	
		Fluid Properties		
	Evaporator	Entering Temp	Leaving Temp	Fluid
		34°F	26°F	25 % Ethlene Glycol
	Electrical	Phase	Voltage	Frequency
		3 Ph	460 V	60 Hz
	Miscellaneous	% Rated Capacity		
		65%		
Chiller Not Ice Making	Condenser	Heat Rejection	Ambient Air Temp	
		Air-Cooled	92°F	
		Fluid Properties		
	Evaporator	Entering Temp	Leaving Temp	Fluid
		60°F	52°F	25 % Ethlene Glycol
	Electrical	Phase	Voltage	Frequency
		3 Ph	460 V	60 Hz

Figure 32 – Chiller Assumptions

Assumptions			
Equipment	Property		
Ice Storage Tanks- Charging	Tank Capacity	145 Ton-hrs/tank	
	Tank Charge Time	14 Hours	
	Leaving Fluid Temperature	Beginning	Ending
		34°F	26°F
	Entering Fluid Temperature	Beginning	Ending
		26°F	20°F
Ice Storage Tanks- Discharging	Tank Capacity	145 Ton-hrs/tank	
	Tank Discharge Time	10 Hours	
	Leaving Fluid Temperature	Beginning	Ending
		34°F	36°F
	Entering Fluid Temperature	Beginning	Ending
		52°F	52°F

Figure 33 – Ice Storage Tank Assumptions

6.3.3 Sizing Calculations

6.3.3.1 Chiller

Selecting the appropriately reduced size chiller depends on the total requirement of ton-hours on the design day. Calmac recommends using the following equation in selecting a chiller. The calculation for Park Place 1 is shown below using that equation.

$$NCC = ton\text{-}hours_{\ Design\ Day} / \Sigma\ (\%\ RCC_{\ NIM}\ x\ Time_{\ NIM} + RCC_{\ IM}\ x\ Time_{\ IM})$$

Where,

NCC	=	*Nominal Chiller Capacity [Tons]*
RCC_{NIM}	=	*Rated Chiller Capacity during Not Ice Making Hours [%]*
$Time_{NIM}$	=	*Total Discharge Time [Hours]*
RCC_{IM}	=	*Rated Chiller Capacity during Ice Making Hours [%]*
$Time_{IM}$	=	*Total Charge Time [Hours]*

For Park Place 1,

$Ton\text{-}hours_{DD}$	=	*2332 ton-hours*
NCC_{NIM}	=	*100 %*
$Time_{NIM}$	=	*10 Hours*
NCC_{IM}	=	*65 %*
$Time_{IM}$	=	*14 Hours*

Therefore,

$$NCC\ [Tons] = 2332\ [ton\text{-}hours] / \Sigma\ (100\ \%\ x\ 10\ [Hours] + 65\ \%\ \ x\ 14\ [Hours])$$

$$NCC\ = 116.86\ [Tons]$$

As a measure of safety, a 25% increase adjustment factor brings the total chiller nominal tonnage to:

$$116.86\ [Tons]\ x\ 1.25 = 146.07\ [Tons]$$

The resulting chiller selection will then be sized for 150 Tons of cooling capacity during ice making.

Once a storage capacity is selected in Section 6.3.3.3, it must be confirmed that the chiller is capable of creating that amount of storage in the hours designated as ice making. This calculation will be shown in Section 6.3.3.3.

A second calculation that must be done is a flow rate balance for the system. When both the ice and chiller are operating, the supply water temperature to the coil is 45°F with a leaving temperature of 60°F. This 15°F ΔT reduces the water flow rate from 552 gpm to 368 gpm—a reduction that will save pumping power. Because water will be mixing at the three-way valve, a water mixing calculation must be done. The calculation is shown below.

$$MFT\ x\ MFVFR = FT_1\ x\ FVFR_1 + FT_2\ x\ FVFR_2$$

Given,

MFT	=	*Mixed Fluid Temperature [°F]*
$MFVFR$	=	*Mixed Fluid Volume Flow Rate [gpm]*
FT_1	=	*Fluid Temperature 1 [°F]*
$FVFR_1$	=	*Fluid Volume Flow Rate 1 [gpm]*
FT_2	=	*Fluid Temperature 2 [°F]*
$FVFR_2$	=	*Fluid Volume Flow Rate 2 [gpm]*

Therefore,

$$60[°F] \times FVFR_1 + 34[°F] (368 - FVFR_1) = 45 [°F] \times 368 [gpm]$$

$$FVFR_1 = 156 \text{ gpm}$$
$$FVFR_2 = 212 \text{ gpm}$$

Both $FVFR_1$ and $FVFR_2$ can be seen in **Figure 41**.

6.3.3.2 Air Handling Unit

All air handling unit calculations are the same as Alternative 2. To see the calculations, go to Section 6.2.3.2.

6.3.3.3 Storage Tanks

Assuming that a chiller selection has already been made based on Section 6.3.3.1 above, storage tank calculations can be performed. The primary consideration is the assumptions that each tank can store about 145 ton-hours of cooling when fully charged. Therefore, the important study is to determine the optimal number of ton-hours, and thus tanks, desired in comparison to the cost of the equipment. To do so, an optimization study will be performed in Section 6.3.4.

It is also important to understand what the storage capacity is for the chiller. The calculation is shown below.

$$NCC \times Time_{IM} = \text{Total Ton-hours possible}$$

$$150 \text{ [Tons]} \times 14 \text{ Hours} = 2100 \text{ Ton-hours}$$

$$\text{Number of Tanks Possible} = \text{Total Ton-hours possible} / \text{Tank Storage Capacity}$$

$$\text{Number of Tanks Possible} = 2100 \text{ Ton-hours} / 145 \text{ Ton-hours per Tank}$$

$$\text{Number of Tanks Possible} = 14$$

Based on the calculations above, the number of tanks can range anywhere from one to fourteen tanks. An economic and energy analysis must be done to determine the optimum number.

6.3.3.4 Pumps

The pumping requirement for Alternatives 2, 3, and 4 is 552 gpm's. This requirement will be handled by two identical, parallel pumps that are both equipped with variable frequency drives. The head loss throughout the system is approximately 80 ft with the summary shown below in **Figure 34**. Based on **Figure 34** the pump size required can be calculated using a pump characteristic curve. From that curve, the horsepower requirement can be determined.

Fluid Pressure Drop Summary	
Element	PD (ft of head)
Chiller	39.6
AHU's	20.0
Piping	10.0
Valves, Fittings, Other	10.0
Total	79.6

Figure 34 – Fluid Pressure Drop Summary

Shown below is the pump characteristic for Alternatives 2, 3, and 4. The given parameters are a flow of 276 gpm and a total head loss of 80 ft. Based on this information and **Figure 35** below, a 10 hp pump was selected.

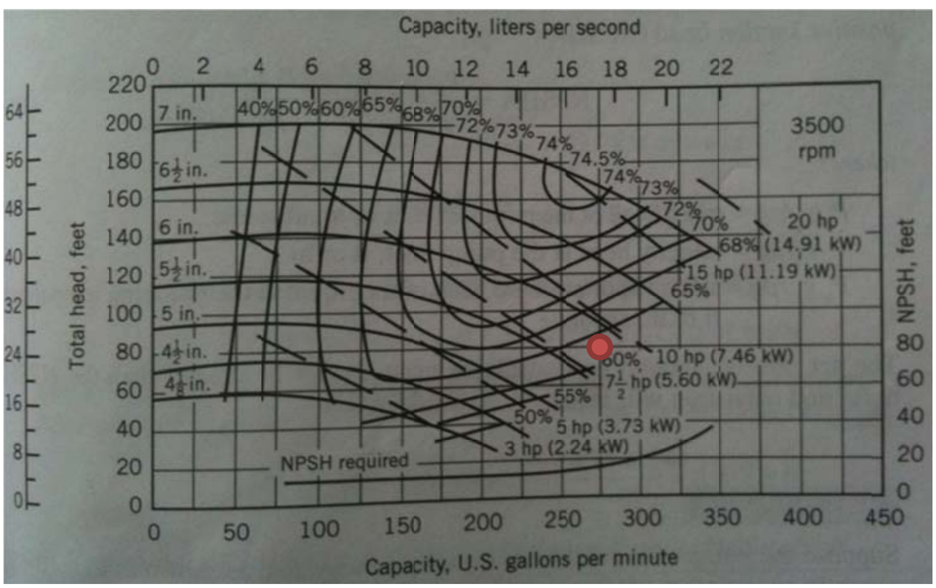

Figure 35 – Pump Characteristic Curve

6.3.4 Optimization Study

Five tank quantities were proposed as an initial study—2, 3, 4, 5, and 6 tanks. To do the study, each tank storage capacity was incorporated into the previously generated Trane Trace 700 energy model. Using Alternative 2 as the starting point, the reduced size chiller was substituted into the program and selected to run with optimal performance. This means that the program selected when to use the ice storage to best save energy use in the building as a whole.

Once the model was complete and simulations were run, a cost analysis was completed to determine which quantity of tanks would yield the best results. To do this, the electricity requirement in both consumption and demand was priced according to Duquesne Light's Energy Rate Tariff. This information would provide an estimate for the total cost per year of electricity. The total electricity cost for each quantity of tanks was then compared with the total electricity cost from Alternative 1. The difference provided the savings per year. The savings, in terms of dollars, was then divided into the total cost of the ice storage system to find which quantity of tanks had the shortest payback period. The quantity with the shortest payback period was then selected and used for Alternative 4.

Cost information was provided by Calmac and was assumed to be $13,000 per tank for the equipment and $26,000 per tank for installation for a total of $39,000 per tank. The summary calculation is shown below in **Figure 36**. The total cost is based on information about electricity use collected from the energy model. Energy consumption information can be found later in this report in Section 7.1.

Tank Quantity Optimization							
Cost Information				Cost of Annual Energy Consumption		Savings per Year	Simple Payback (years)
Number of Tanks	Cost per Tank (Equipment)	Cost per Tank (Install)	Total	With Tanks	Without Tanks		
2	$13,000	$26,000	$78,000	$116,499.15	$122,162.39	$5,663.24	**13.77**
3	$13,000	$26,000	$117,000	$115,209.81	$122,162.39	$6,952.58	16.83
4	$13,000	$26,000	$156,000	$114,086.52	$122,162.39	$8,075.87	19.32
5	$13,000	$26,000	$195,000	$113,244.23	$122,162.39	$8,918.16	21.87
6	$13,000	$26,000	$234,000	$112,791.03	$122,162.39	$9,371.36	24.97

Figure 36 – Tank Quantity Optimization

The optimization study indicates that the ideal number of tanks is 2. Increasing the number of tanks also increased the payback period making further study unnecessary. To summarize, the total cost of the system is $78,000 with an annual savings of $5,663.24. When the total cost is divided by the annual savings, the simple payback period can be calculated to be 13.77, or about 14 years.

6.3.5 Equipment Selection and Performance

6.3.5.1 Chiller

The selected chiller performance characteristics are shown below in **Figure 37** for not ice making and **Figure 38** for ice making modes.

Chiller Selection- Not Ice Making		
General	Unit Name:	CH-3, 4
	Basis of Selection:	Trane
	Trane Model:	Air-Cooled Series R(TM) (RTAC)
	Unit Type:	Standard Efficiency
	Unit Nominal Tonnage	200
	Capacity:	229.30 tons
	Efficiency:	10.6 EER
	COP:	3.11
	Refrigerant:	HFC-134a
	Shipping Weight:	12,885.0 lbs.
	Dimensions (LxWxH):	232" x 89" x 93"
	Rated Capacity (AHRI):	198.70 tons
	Rated Efficiency (AHRI):	9.7 EER
Evaporator	Leaving Temp:	52.00°F
	Entering Temp:	60.00°F
	Flow Rate:	735.10 GPM
	Pressure Drop:	29.20 ft H2O
	Configuration:	2 Pass
	Fluid Type:	25% Ethylene Glycol
Electrical	Unit Voltage:	460V/60Hz/3Ph
	Unit Power:	259.20 kW
	Compressor Power:	241.10 kW
	Fan Motor Power:	17.30 kW
	Number of Condenser Fans:	12
Control Accessories	VFD:	Yes
	Ice Making:	Yes

Figure 37 – Ice Making Chiller Operating During Discharge Conditions

Chiller Selection- Ice Making		
General	Unit Name:	CH-3, 4
	Basis of Selection:	Trane
	Trane Model:	Air-Cooled Series R(TM) (RTAC)
	Unit Type:	Standard Efficiency
	Unit Nominal Tonnage	200
	Capacity:	153.70 tons
	Efficiency:	10.3 EER
	COP:	3.02
	Refrigerant:	HFC-134a
	Shipping Weight:	12,885.0 lbs.
	Dimensions (LxWxH):	232" x 89" x 93"
	Rated Capacity (AHRI):	198.70 tons
	Rated Efficiency (AHRI):	9.7 EER
Evaporator	Leaving Temp:	26.00°F
	Entering Temp:	34.00°F
	Flow Rate:	495.50 GPM
	Pressure Drop:	14.30 ft H2O
	Configuration:	2 Pass
	Fluid Type:	25% Ethylene Glycol
Electrical	Unit Voltage:	460V/60Hz/3Ph
	Unit Power:	179.00 kW
	Compressor Power:	160.00 kW
	Fan Motor Power:	18.10 kW
	Number of Condenser Fans:	12
Control Accessories	VFD:	Yes
	Ice Making:	Yes

Figure 38 – Ice Making Chiller Operating During Charging Conditions

6.3.5.2 Ice Storage Tanks

Two Calmac Model 1190 tanks were selected according to the direction of Calmac and its representatives. The major consideration was tank capacity and performance. Each Calmac Model 1190 tank was assumed to produce 145 ton-hours of cooling capacity when fully charged and was capable of meeting the required system glycol temperatures.

Figures 39 and **40** depict the system characteristics and a cutaway of the tank design respectively.

Ice Tank Selection		
General	Unit Name:	IT-1, 2
	Basis of Selection:	Calmac
	Calmac Model:	1190A
	Max Capacity:	162 Ton-Hr
	Net Capacity:	145 Ton-Hr
	Max Operating Temp:	100°F
	Factory Tested Pressure:	250 psi
	Max Operating Pressure:	90 psi
	Dimensions (ODxH):	89" x 101"
	Weight Filled:	16,765 lbs.
	Floor Loading:	388 lbs/ft^2
	Volume of Water/Ice:	1,655 gallons
	Volume of Solution in HX	148 gallons
	Inlet/Outlet Flange Connections	2

Figure 39 – Ice Tank Performance Characteristics

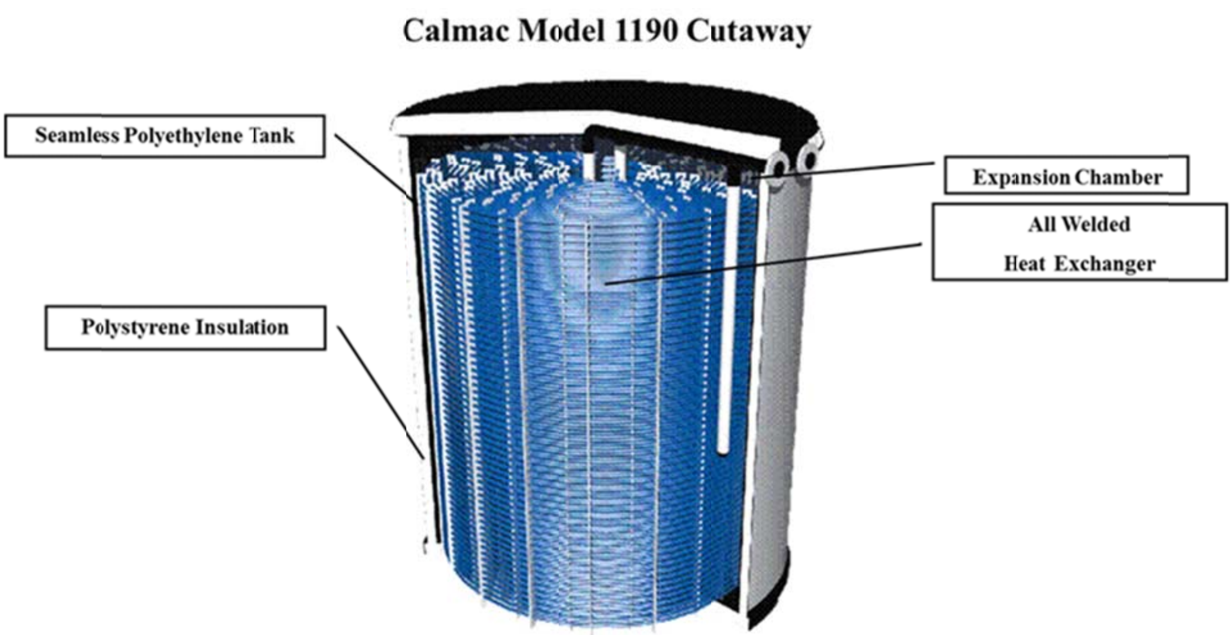

Figure 40 – Calmac Model 1190 Cutaway

6.3.6 Schematic

Three schematics are displayed below with corresponding capacities, temperatures, and flow rates. **Figure 41** shows the system charging during night time, ice making operation. The black line indicates no flow.

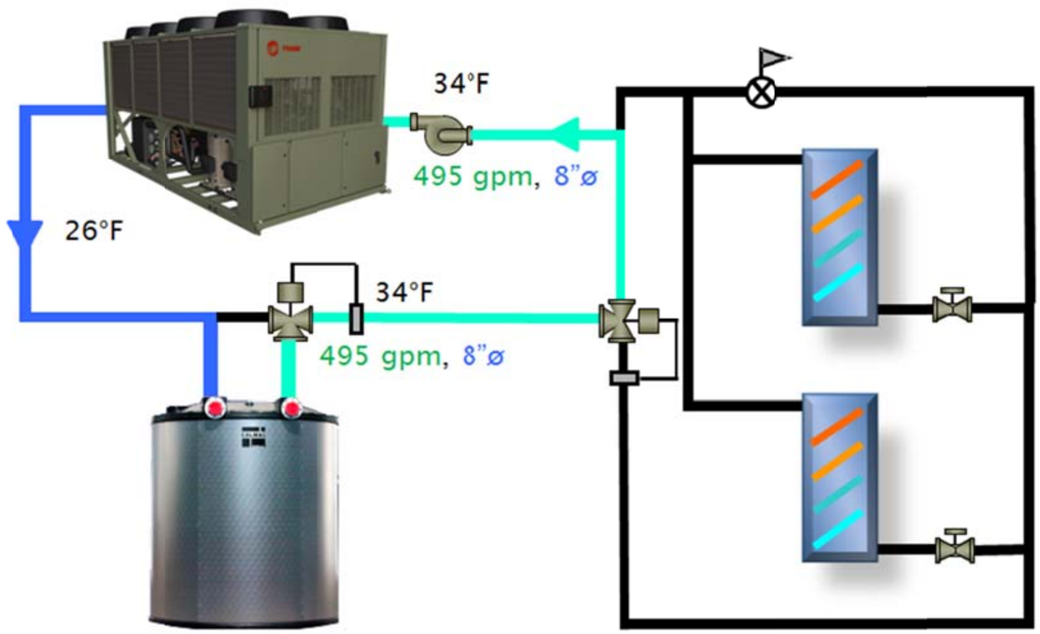

Figure 41 – Ice Storage Charging

Figure 42 depicts the system functioning with the chiller handling the entire building cooling load. The ice storage is bypassed. **Figure 43** shows the system operation while the ice storage tanks are being utilized.

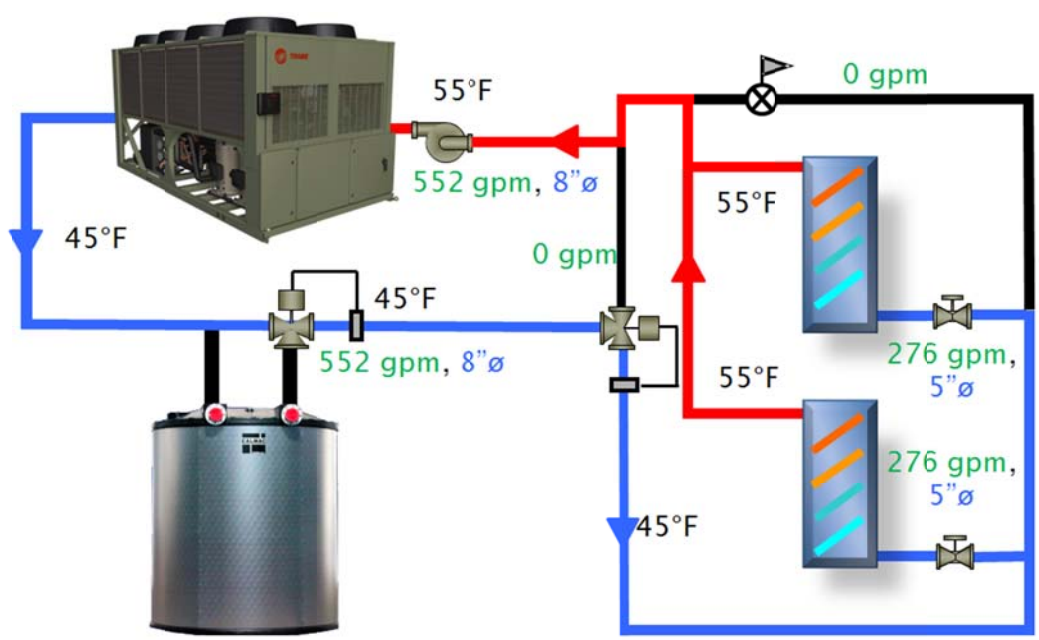

Figure 42 – Chiller Day Time Operation

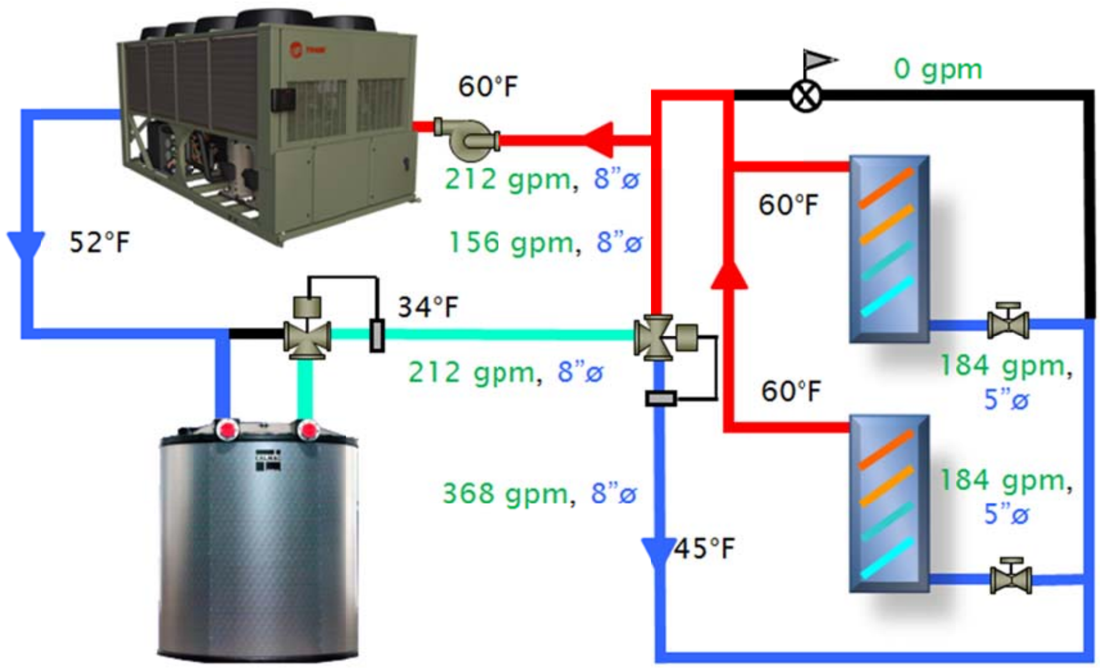

Figure 43 – Ice Discharge Schematic

6.4 Alternative 4 - Reduced Supply Air Temperature

6.4.1 System Description and Assumptions

Alternative 4 is identical to Alternative 3 except the supply air temperature was dropped from 55°F to 50°F. It was assumed that the optimal tank selection (2 tanks) from Alternative 3 would be used in Alternative 4. The goal was to see if dropping the supply air temperature would increase energy savings.

Dropping the supply air temperature forces the cooling plant to work a little bit harder to produce colder ethylene glycol or a larger volume flow rate. However, the reduction in supply air temperature will imply a larger temperature drop over the cooling coil which allows for a reduction in the volume flow rate of the supply air. This reduction in supply air quantity enables fan energy savings.

6.4.2 Sizing Calculations

6.4.2.1 Air Handling Unit

Assuming no change in the ethylene glycol flow rate or temperature drop and a predetermined ΔT over the cooling coil of 27.5°F, the new air flow rate must be determined. The calculation is shown below.

$$Q_{SENSIBLE} = 1.08 * Volume\ Flow\ Rate\ [cfm] * (T_{BEFORE\ COIL} - T_{AFTER\ COIL})$$

$$2058.2\ MBh = 1.08 * Volume\ Flow\ Rate\ [cfm] * (77.5°F - 50°F)$$

$$Volume\ Flow\ Rate_{REQUIRED} = 69{,}300\ cfm$$

An assumed total flow rate of 70,000 cfm's (35,000 cfm's per AHU) was used for energy model simulation.

The selected AHU for Alternative 4 is also identical to the AHU's selected for the previous Alternatives with the exception that it has been reduced in overall size and capacity. This translates into first cost savings and annual energy consumption savings. The selection summary can be seen below in **Figure 44**.

Air Handling Unit Selection		
General	Unit Name:	AHU-1, AHU-2
	Basis of Selection:	Trane
	Trane Model:	Outdoor T-Series Climate Changer
	Unit Type:	Standard Efficiency
	Shipping Weight:	23,804 lbs
	Dimensions (LxWxH):	473" x 154" x 120"
Return Fan	Fan Size:	40"
	Fan Type:	FC
	Motor HP:	20 hp
	Break HP:	19.769 hp
	Fan Airflow:	35,000 CFM
	External Static Pressure:	1.5" H2O
	Total Static Pressure:	2.169" H2O
	Speed:	375 rpm
	Motor Class:	ODP NEMA Premium Efficiency
	Motor Voltage:	460V/60Hz/3Ph
	VFD:	Yes
	Fan Module Pressure Drop:	1.677" H2O
Economizer	OA Capability:	0-100%
	Total OA Pressure Drop:	0.216" H2O
	Return Damper Pressure Drop:	0.279" H2O
	Exhaust Damper Pressure Drop:	0.279" H2O
	Supply Fan Pressure Drop:	0.216" H2O
Filter Section	Prefilter Type:	Pleated media coated - MERV 7
	Prefilter Pressure Drop:	0.573" H2O
	Primary Filter Type:	Short Bag 85% - MERV 13
	Primary Filter Pressure Drop:	0.752" H2O
	Total Pressure Drop:	1.325" H2O

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Heating Coil	Rows:	2
	Fin Spacing:	80 per foot
	Fin Material:	Aluminum
	Tube Material:	Copper
	Airflow:	35,000 CFM
	Face Velocity:	589 ft/min
	Entering Dry Bulb	63°F
	Leaving Dry Bulb:	90°F
	Air Pressure Drop:	0.161" H2O
	Entering Water Temperature:	180°F
	Leaving Water Temperature:	160°F
	Fluid Pressure Drop:	6.73' H2O
	Fluid Flow Rate:	132 GPM
	Total Capacity:	1321.75 MBH
Cooling Coil	Rows:	6
	Fin Spacing:	123 per foot
	Fin Material:	Aluminum
	Tube Material:	Copper
	Airflow:	35,000 CFM
	Face Velocity:	451 ft/min
	Entering Dry Bulb	77.5°F
	Leaving Dry Bulb:	50°F
	Entering Wet Bulb:	66.00°F
	Leaving Wet Bulb:	49.90°F
	Air Pressure Drop:	0.384" H2O
	Entering Water Temperature:	45°F
	Leaving Water Temperature:	55°F
	Fluid Pressure Drop:	11.34' H2O
	Fluid Flow Rate:	272.32 GPM
	Sensible Capacity:	1,022.04 MBH
	Total Capacity:	1,507.88 MBH

Supply Fan	Fan Size:	40"
	Fan Type:	AF
	Motor HP:	50 hp
	Break HP:	41.523 hp
	Fan Airflow:	35,000 CFM
	External Static Pressure:	3.5" H2O
	Total Static Pressure:	5.883" H2O
	Speed:	973 rpm
	Motor Class:	ODP NEMA Premium Efficiency
	Motor Voltage:	460V/60Hz/3Ph
	VFD:	Yes
	Fan Module Pressure Drop:	3.787" H2O

Figure 44 – Alternative 4 AHU Performance Characteristics

7.0 Energy and Cost Evaluation

As brief recap, there were four systems modeled. Alternative 1 had two packaged rooftop units, Alternative 2 was a basic chilled water system, Alternative 3 added two ice storage tanks to the chilled water system of Alternative 2, and finally Alternative 4 reduced the size of the air handling units in Alternative 3 by reducing the supply air temperature and thus the air quantity required to cool the space.

This section aims to address the two major considerations of any mechanical system design: energy performance and cost. Energy performance was modeled using Trane Trace 700 and cost considerations were taken from a number of different sources.

Finally, because cooling was the focus of the redesign and heating was assumed to be the same in each alternative, only cooling and its equipment are presented in this section.

7.1 System Performance

The system performance was evaluated according to total monthly electricity consumption and demand. The original and proposed equipment have no alternative energy supply and thus looking at monthly energy consumption is adequate in determining which system will be the most efficient and therefore save the most money.

Figures 45-47 depict the monthly energy consumption and demand for each Alternative. **Figure 45** gives the numerical outputs from Trace 700 while **Figures 46** and **47** show the same information in graphical form.

Monthly Energy Consumption

Alternative	Electricity	January	February	March	April	May	June	July	August	September	October	November	December	Total
1	Consumption (kWh)	41,722	37,629	47,068	47,267	67,150	78,524	81,488	78,084	59,684	47,764	43,337	40,047	**669,764**
	Demand (kW)	282	291	331	368	421	465	488	472	455	371	318	273	**488**
2	Consumption (kWh)	44,407	40,057	47,560	47,657	69,541	83,851	88,478	82,812	61,822	48,325	43,639	42,287	**700,436**
	Demand (kW)	278	288	330	383	439	476	500	476	460	385	318	272	**500**
3 (2 Tanks)	Consumption (kWh)	44,912	40,326	48,540	46,948	68,540	80,619	85,139	79,788	60,724	48,156	44,757	43,138	**691,587**
	Demand (kW)	222	232	276	305	372	426	450	424	404	305	262	218	**450**
3 (3 Tanks)	Consumption (kWh)	44,912	40,326	48,540	47,429	69,182	80,282	83,934	79,809	61,350	48,795	44,757	43,138	**692,454**
	Demand (kW)	221	228	256	291	343	400	428	398	370	293	241	216	**428**
3 (4 Tanks)	Consumption (kWh)	44,912	40,326	48,540	48,173	69,100	80,537	83,489	80,415	60,484	48,795	44,757	43,138	**692,666**
	Demand (kW)	221	228	243	282	325	370	404	369	350	279	233	216	**404**
3 (5 Tanks)	Consumption (kWh)	44,912	40,326	48,540	48,173	67,751	81,297	83,603	81,508	59,892	48,795	44,757	43,138	**692,692**
	Demand (kW)	221	228	241	259	312	351	379	351	336	260	231	216	**379**
3 (6 Tanks)	Consumption (kWh)	44,912	40,326	48,540	49,324	67,282	81,801	83,784	80,278	60,014	50,019	44,757	43,138	**694,175**
	Demand (kW)	221	228	241	248	303	336	362	337	324	251	231	216	**362**
4	Consumption (kWh)	41,722	37,629	47,068	47,276	67,150	78,524	81,488	78,048	59,684	47,764	43,337	40,047	**669,737**
	Demand (kW)	282	291	331	368	421	465	488	472	455	371	318	273	**488**

Figure 45 – Monthly Energy Consumption

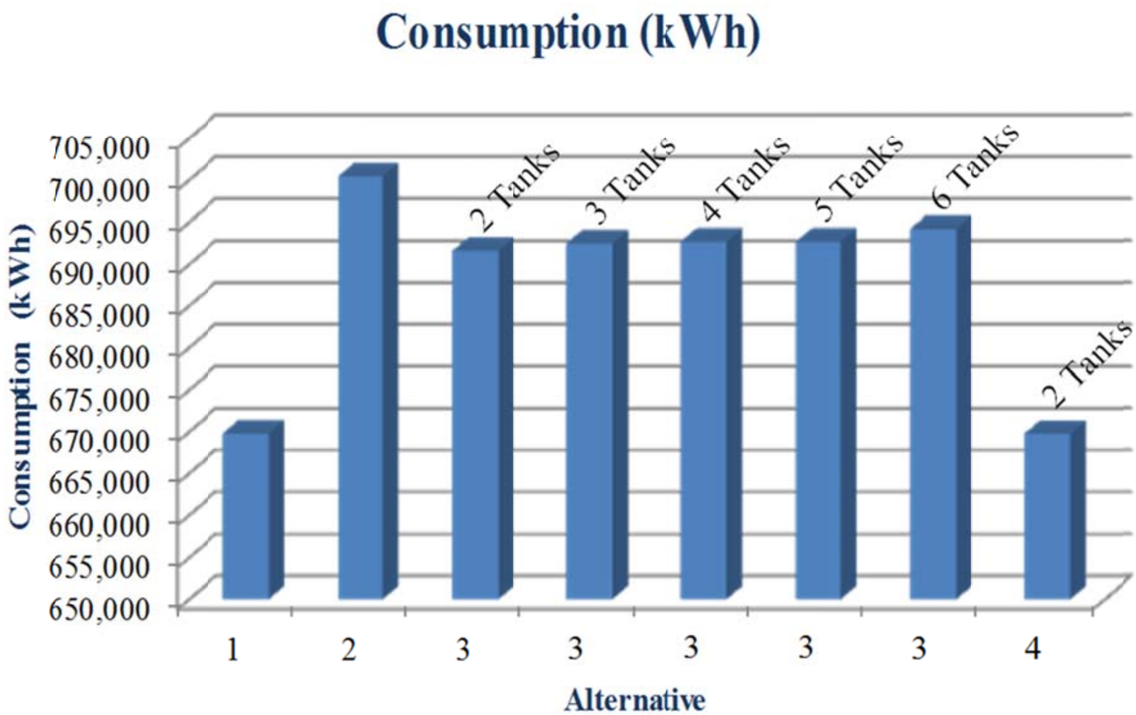

Figure 46 – Annual Energy Consumption

Figure 46 suggests four significant observations. First, in considering the total annual consumption of electricity, the smallest consumer is Alternative 1, the packaged rooftop units. This makes sense because producing chilled water almost always requires more power than a packaged DX rooftop unit for producing cool air in a small scale building. Along those same lines, the chilled water system consumes the most electricity of any alternative. This also makes sense because this alternative has the largest sized chiller. Alternatives 3 and 4 have a smaller chiller which consumes less electricity. Third, adding ice storage tanks increases the total annual consumption. Finally fourth, by reducing the supply and return fan energy, as demonstrated by Alternative 4, the total consumption of the chilled water plus ice system is comparable to the packaged RTU alternative (Alternative 1).

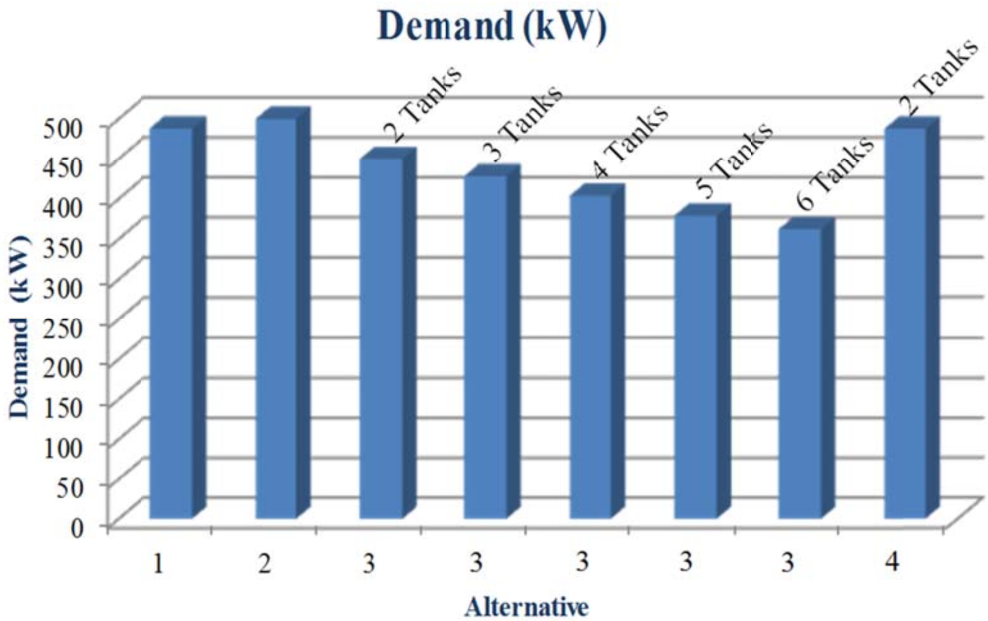

Figure 47 – Annual Peak Demand

Figure 47 demonstrates the true advantages of using a thermal storage system. It is clear from the graph that by adding storage tanks, the overall demand is decreased significantly. Also important to note is that Alternative 4 offers no real advantage in demand savings even with two ice storage tanks.

System control is also a very important part of maximizing the benefit of TES. During the night the chiller must be set to try and achieve an impossibly low leaving evaporator target temperature. This ensures that the chiller is functioning at full capacity all the time to ensure a full charge in the tanks during the charging hours.

Discharging of the tanks is where money and energy are saved in a TES system. Deciding when to use the available stored ice capacity will limit the building electrical demand. In this study, when to use the stored ice capacity was determined by Trace 700. This is one of the most beneficial parts of the energy modeling program and running simulations allows for a preliminary basis from which to control the system once it is installed. In a real system, the true performance would need to be mapped and adjusted to determine when best to discharge the ice tanks.

The design day discharge is shown below in **Figures 48** with **Figures 49** and **50** showing the same information graphically.

Design Day Ice Analysis							
Late July	Typical Weather		Load (Tons)	Chiller		Ice Storage	
Hour	OADB	ODWB	Cooling	Load (Tons)	Demand (kW)	Capacity (Ton-hrs)	Mode
1	68.3	62.9	2.6	2.6	2.3	290	-
2	67.2	62.3	2.4	2.4	2.2	290	-
3	66.4	62.0	2.3	2.3	2.1	290	-
4	65.8	61.6	2.2	2.2	2.0	290	-
5	65.6	61.8	2.5	2.5	2.3	290	-
6	66.0	62.5	2.8	2.8	2.5	290	-
7	67.0	63.7	20.6	20.6	18.5	290	-
8	68.9	64.6	144.0	144.0	113.9	290	-
9	71.5	65.1	130.1	130.1	103.0	290	-
10	74.6	66.3	150.8	150.8	131.9	290	Discharging
11	78.0	67.4	171.2	161.0	149.9	280	Discharging
12	81.3	68.8	181.7	161.0	160.6	259	Discharging
13	83.6	70.0	193.8	161.0	168.5	226	Discharging
14	85.4	70.7	207.3	161.0	173.3	180	Discharging
15	86.0	71.0	219.0	161.0	175.1	122	Discharging
16	85.4	70.2	220.3	161.0	173.3	63	Discharging
17	84.0	69.4	214.4	161.0	169.1	9	Discharging
18	81.7	68.3	170.1	161.0	161.8	0	Discharging
19	79.1	66.8	2.6	156.8	127.4	0	Charging
20	76.4	65.7	2.8	138.5	107.4	154	Charging
21	74.2	65.1	2.5	119.6	94.6	290	Charging
22	72.1	64.7	2.3	2.3	2.1	290	-
23	70.5	63.7	2.4	2.4	2.2	290	-
24	69.3	63.4	2.4	2.4	2.2	290	-

Figure 48 – Design Day System Performance

Figures 48 and **49** show that the total peak load has been reduced from 220.3 tons to 161.0 tons—a savings of 59.3 tons. In **Figure 49** below, the area designated by the letter "A" shows the peak shaving strategy employed. By integrating the area between the curves, the total number of ton-hours shaved can be determined. "B" represents the time during which the ice tanks are charging.

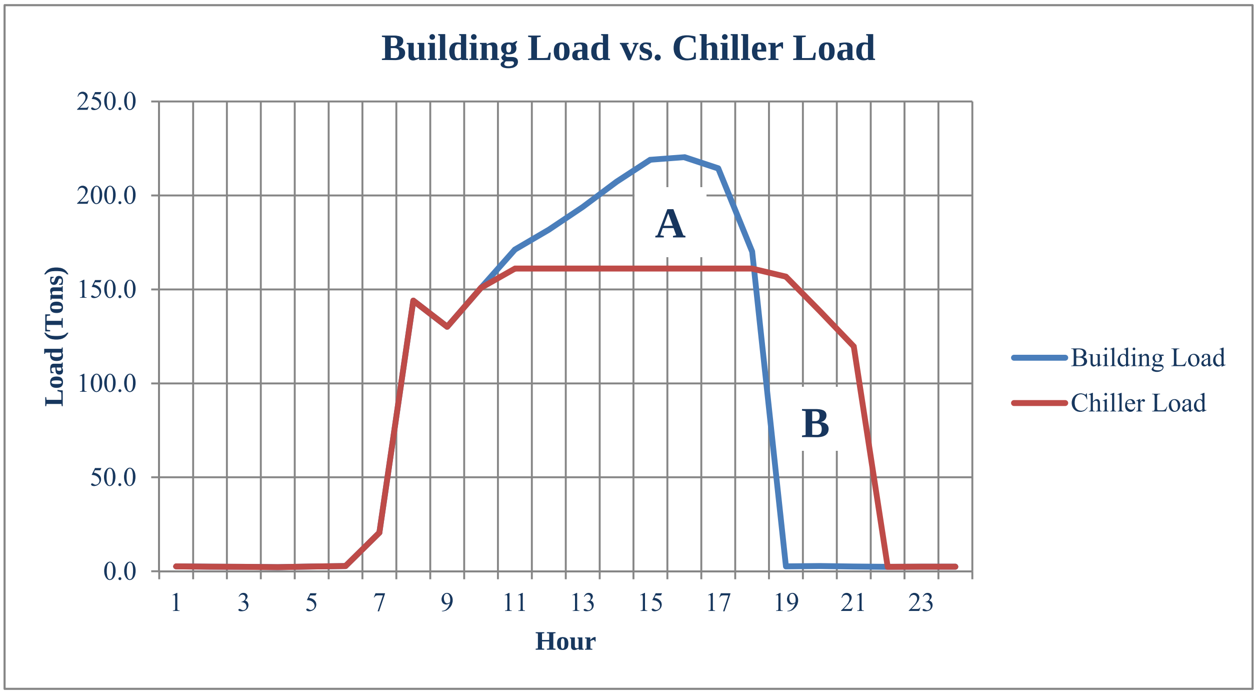

Figure 49 – Building Load vs. Chiller Load

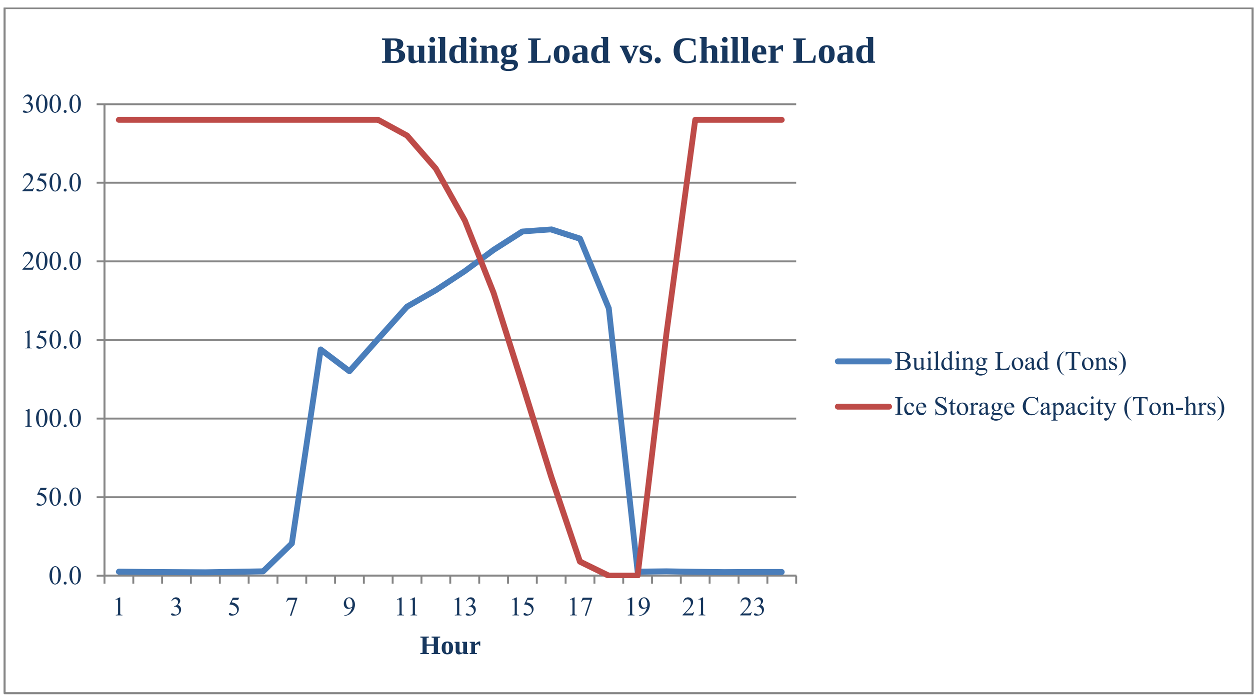

Figure 50 – Building Load vs. Ice Storage Capacity

While understanding the energy consumption is crucial, it is not helpful unless it can be put into terms of dollars and savings. The initial cost and payback period are what will determine which system is best suited to serve Park Place 1. Energy savings and cost analysis will be discussed in sections 7.2 and 7.3.

7.2 Energy Savings

The energy savings were calculated by subtracting the operating energy use of the original system (Alternative 1) from that of each proposed alternative (Alternatives 2, 3, 4). **Figure 51** below shows energy savings in green. The negative sign (shown in green) implies that total usage of that alternative was less than that of the original design. For example, in Alternative 2, annual energy consumption was 700,436 kWh while for the original design it was 669,764 kWh. The difference then in 30,672 kWh more of energy used in Alternative 2 than Alternative 1—an undesirable result. **Figure 52** shows the same percent difference graphically.

Energy Usage Comparison			
Alternative	Energy Type	Difference	% Difference
1	Consumption (kWh)	-	-
	Demand (kW)	-	-
2	Consumption (kWh)	30,672	4.38%
	Demand (kW)	12	2.40%
3 (2 Tanks)	Consumption (kWh)	21,823	3.16%
	Demand (kW)	-38	-8.44%
3 (3 Tanks)	Consumption (kWh)	22,690	3.28%
	Demand (kW)	-60	-14.02%
3 (4 Tanks)	Consumption (kWh)	22,902	3.31%
	Demand (kW)	-84	-20.79%
3 (5 Tanks)	Consumption (kWh)	22,928	3.31%
	Demand (kW)	-109	-28.76%
3 (6 Tanks)	Consumption (kWh)	24,411	3.52%
	Demand (kW)	-126	-34.81%
4	Consumption (kWh)	-27	0.00%
	Demand (kW)	0	0.00%

Figure 51 – Energy Usage Comparison

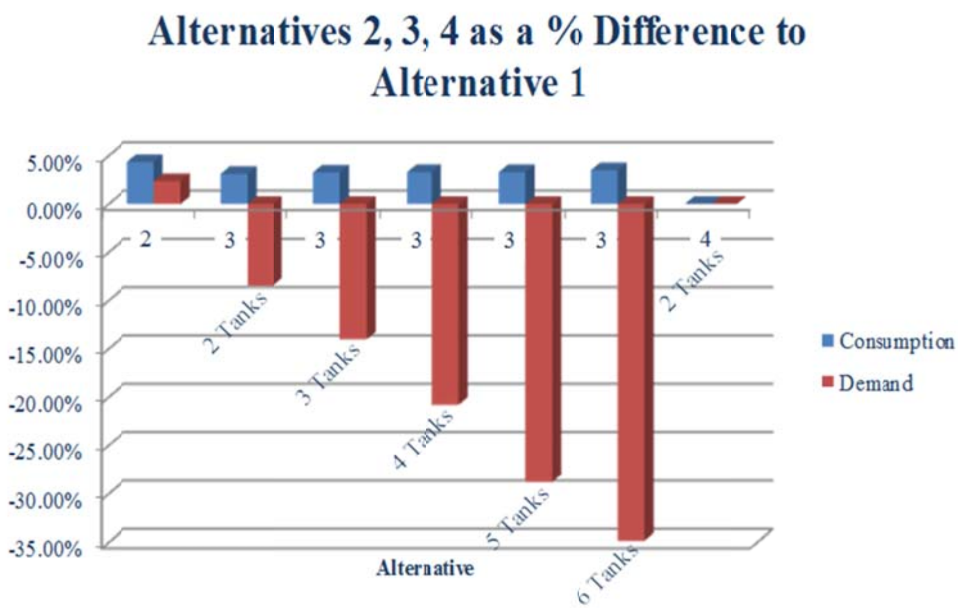

Figure 52 – Alternatives 2, 3, 4 as a % Difference to Alternative 1

7.3 System Cost

To calculate the operating cost of each system, the electricity consumption and demand was multiplied by its corresponding cost as provided by the Schedule of Rates from Duquesne Light. A summary of the calculation is shown below in **Figure 53**. To avoid redundancy, only the calculation for Alternative 1 is shown in **Figure 53**. The results for each alternative are shown in **Figure 54**. The same summary information from **Figure 54** is presented graphically in **Figure 55**.

Annual Operating Cost							
Alternative	Electricity	January	February	March	April	May	June
1	Consumption (kWh)	41,722	37,629	47,068	47,267	67,150	78,524
	Cost	$5,156.84	$4,650.94	$5,817.60	$5,842.20	$8,299.74	$9,705.57
	Demand (kW)	282	291	331	368	421	465
	Cost	$2,121.00	$2,121.00	$2,415.50	$2,767.00	$3,270.50	$3,688.50
	Total	$7,277.84	$6,771.94	$8,233.10	$8,609.20	$11,570.24	$13,394.07

Annual Operating Cost Continued						
July	August	September	October	November	December	Total
81,488	78,084	59,684	47,764	43,337	40,047	669,764
$10,071.92	$9,651.18	$7,376.94	$5,903.63	$5,356.45	$4,949.81	$82,782.83
488	472	455	371	318	273	488
$3,907.00	$3,755.00	$3,593.50	$2,795.50	$2,292.00	$2,121.00	$34,847.50
$13,978.92	$13,406.18	$10,970.44	$8,699.13	$7,648.45	$7,070.81	$117,630.33

Figure 53 – Annual Operating Cost

Annual Operating Cost Summary	
Alternative	Total Operating Cost
1	$117,630.33
2	$122,162.39
3 (2 Tanks)	$116,499.15
3 (3 Tanks)	$115,209.81
3 (4 Tanks)	$114,086.52
3 (5 Tanks)	$113,244.23
3 (6 Tanks)	$112,791.03
4	$117,626.99

Figure 54 – Annual Operating Cost Summary

Notice that the total operating cost from **Figure 53** corresponds to that of **Figure 54** for Alternative 1.

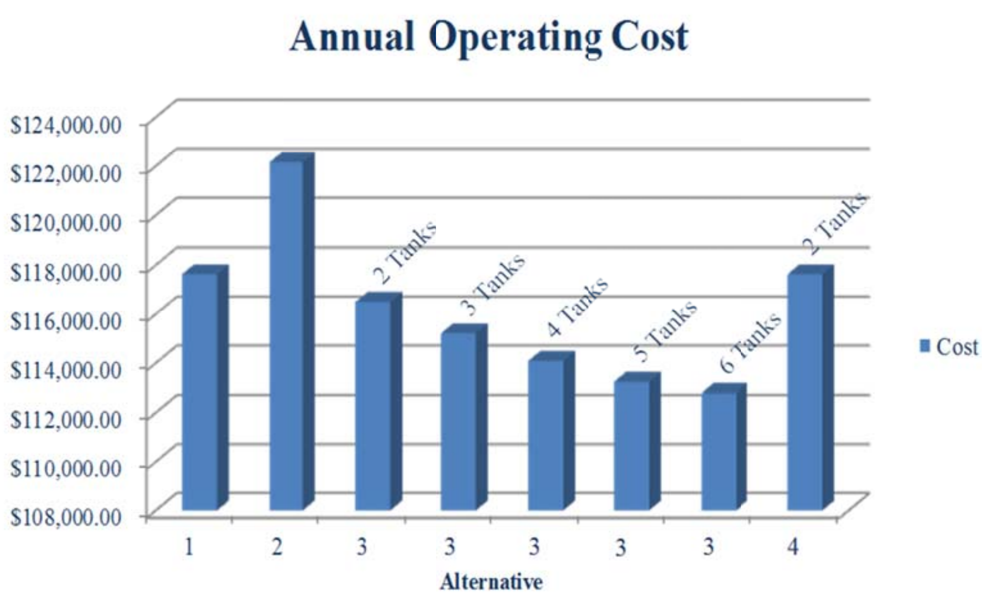

Figure 55 – Annual Operating Cost Summary

To determine which system will be the best to use, a system cost analysis was done. For each alternative, the cost of the entire system was estimated. This initial cost combined with the expected life of the equipment—twenty years—times the cost of operation per year gave a total cost for the owner. Other financial rates such, inflation, interest, and rising energy rates were assumed to be the same for each study and could therefore be disregarded in calculating the present value cost of the system. Ignoring rising energy costs does create some deal of inaccuracy however, making accurate predictions is nearly impossible. If rates continue to rise, the study will tend to favor the more expensive first cost systems. If energy becomes cheaper, then the systems with less initial investment will become more desirable.

Figure 56 presents the total cost to the owner over a twenty year span. The pricing of piping includes pipe, valves, fittings, hangers, labor and insulation. **Figure 57** presents the same information graphically.

System Cost Analysis							
Alternative	Equipment	Quantity	Cost Per Unit	Total Cost	Energy Cost Per Year	Life Cycle (years)	Total System Cost
1	120 Ton Packaged RTU	2	$138,737.00	$277,474.00			
	Installation	-		$554,948.00			
			Total	$832,422.00	$117,630.33	20	$3,185,028.60
2	250 Ton Air-Cooled RTAC Chiller	1	$93,494.00	$93,494.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Installation	-	-	$636,884.00			
			Total	$955,326.00	$122,162.39	20	$3,398,573.79
3 (2 Tanks)	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	2	$13,000.00	$26,000.00			
	Installation	-	-	$501,896.00			
			Total	$837,709.00	$116,499.15	20	$3,167,692.06
3 (3 Tanks)	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	3	$13,000.00	$39,000.00			
	Installation	-	-	$527,896.00			
			Total	$876,709.00	$115,209.81	20	$3,180,905.29

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3 (4 Tanks)	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	4	$13,000.00	$52,000.00			
	Installation	-	-	$553,896.00			
			Total	$915,709.00	$114,086.52	20	$3,197,439.35
3 (5 Tanks)	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	5	$13,000.00	$65,000.00			
	Installation	-	-	$579,896.00			
			Total	$954,709.00	$113,244.23	20	$3,219,593.62
3 (6 Tanks)	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$98,063.50	$196,127.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	6	$13,000.00	$78,000.00			
	Installation	-	-	$605,896.00			
			Total	$993,709.00	$112,791.03	20	$3,249,529.60
4	200 Ton Air-Cooled RTAC Chiller	1	$84,865.00	$84,865.00			
	T-Series Air Handling Unit	2	$92,281.00	$184,562.00			
	Variable Speed Pump	2	$8,000.00	$16,000.00			
	Piping (lft)	100	$128.21	$12,821.00			
	Ice Storage Tanks	2	$13,000.00	$26,000.00			
	Installation	-	-	$478,766.00			
			Total	$803,014.00	$117,626.99	20	$3,155,553.80

Figure 56 – System Cost Analysis

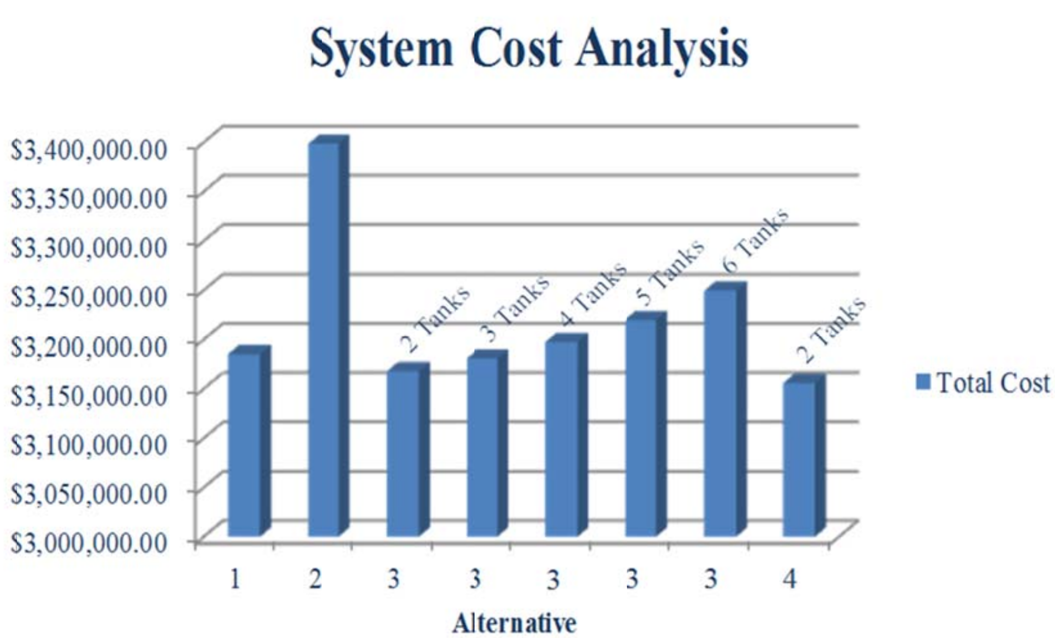


Figure 57 – System Cost Analysis

7.4 Conclusions

The systems that would best serve Park Place 1 would be Alternatives 1, 3 or 4. Each offers several benefits and drawbacks that will be discussed in this section. A decision as to which one should be implemented must be determined by the owner of the building.

Alternative 1

Advantages

· Packaged System
  · One piece of equipment
  · Controls
    · Included
    · Factory mounted
    · Factory commissioned
· Lowest commissioning cost
· Low installation cost
· Reduced maintenance cost
· Ease of operation
· High Reliability

Disadvantages

· High operating cost
· Limited flexibility
· Limited customization

Alternative 3

Advantages

· Lowest electrical demand
· Reduced energy consumption
· High potential payback in case of future energy rate increases
· Reduced first cost of equipment
· Long term savings

Disadvantages

· Complicated controls
· Multiple pieces of equipment
· High commissioning cost
· Complex system operation
· High installation cost

Alternative 4

Advantages	Disadvantages
· Lowest electrical consumption	· Complicated controls
· High potential payback in case of future energy rate increases	· Multiple pieces of equipment
· Lowest first cost of equipment	· High commissioning cost
· Long term savings	· Complex system operation
	· High installation cost

Based on the advantages and disadvantages the best system to install is probably Alternative 4 which incorporates the two ice storage tanks and the reduced supply air temperature to the occupied space. The major benefits of this system are that the initial cost of equipment is the cheapest and the cost to operate the system is the cheapest. The major draw backs are that the system will be difficult to control and the cost of installation will be increased significantly with respect to Alternative 1. Also, the system operator will need a relatively advanced level of knowledge to ensure the system operates ideally. The margin for error in Alternative 4 is the greatest.

8.0 Electrical Breadth Analysis

8.1 Objectives

Ice storage offers the benefit of reducing the peak demand load required by the building it services. In the case of Park Place 1, a source of financial savings could take place in the reduced cost of using smaller electrical equipment in place of larger equipment that must service a greater demand. The purpose of this study is to determine what adjustments need to be made to the existing electrical system to support the new mechanical equipment proposed in the redesign alternatives.

8.2 Existing Conditions

When the building was constructed, all mechanical equipment was wired from a motor control center (MCC) rated at 600 A. The renovation of the building removed that 600 A MCC and replaced it with a 600 A panelboard. An additional 800 A panelboard was also added to the rooftop penthouse during the renovation giving a total electrical capacity of 1400 A. This 1400 A capacity would have to serve the several different pieces of equipment and circuits shown below in **Figure 58**.

Rooftop Panelboard Service	
Element	Design Requirement
RTU-1	400 A-3P
RTU-2	400 A-3P
Elevator 1	100 A-3P
Elevator 2	100 A-3P
Exhaust Fan (Restrooms)	15 A-3P
Lifesafety Circuitry	70 A-3P
Capacitor Circuit	100 A-3P
5th Floor Circuitry	150 A-3P
Total	1335 A

Figure 58 – Rooftop Panelboard Service

Based on **Figure 58** above, the elevators, exhaust fan, life safety circuitry, capacitor circuit, and 5^{th} floor circuitry have a combined requirement of 535 A. These elements are serviced by the 600 A panelboard which leaves the remaining 800 A from the RTU's to be served by the 800 A panelboard.

Both the 600 A and 800 A panelboards located in the rooftop penthouse are fed directly from the main distribution panel (MDP) that services the entire building. That MDP, rated for 1800 A (277/480V-3ϕ-4W), is located on the first floor of Park Place 1 in an electrical closet.

The wire that connects the panelboards in the rooftop penthouse and the MDP is currently sized as four sets of 700 MCM copper wire with four 3/0 ground wires. Each set of wires is contained in a raceway sized as 4" PVC.

8.3 Calculations

8.3.1 New Equipment Electrical Loads

The electrical requirements of the proposed redesign are shown below in **Figure 59**.

New Equipment Electrical Loads	
Element	Design Requirement
CH-3,4	414 A-3P
AHU-1	100 A-3P
AHU-2	100 A-3P
P-1	30 A-3P
P-2	30 A-3P
Total	674 A

Figure 59 – New Equipment Electrical Loads

Figure 59 above demonstrates a reduction in electrical demand of 800 A – 674 A, or 126 A, from replacing the mechanical equipment of Alternative 1 by that of Alternatives 2, 3, and 4.

8.3.2 Over Current Protection Device

Section 240.6 of the *National Electric Code 2008* was used to determine possible reductions in fixed-trip circuit breakers. This was done by comparing the maximum required ampacity from the previous section with the possible circuit breaker sizes.

In the case of Park Place 1, the previous required circuit breaker size for mechanical equipment (RTU-1 plus RTU-2) was 800 A as determined in **Figure 58**. That load was reduced to 674 A in Alternatives 2 through 4 which permits a reduction in circuit breaker size from 800 A to 700 A.

8.3.3 Connected Load

The connected load was determined by using the equation:

Power [Watts] = Full Load Current x 1.73 x Voltage x Power Factor

The assumed power factor for motors greater than 5 hp is equal to 0.90.

Connected Load Summary					
Element	FLC (Amps)	PF	Voltage (V)	Watts	
				Total	Phase
CH-3,4	414	0.9	480	309,407	103,136
AHU-1	100	0.9	480	74,736	24,912
AHU-2	100	0.9	480	74,736	24,912
P-1	30	0.9	480	22,421	7,474
P-2	30	0.9	480	22,421	7,474

Figure 60 – Connected Load Summary

8.3.4 Feeder Sizing

Section 310.16 of the *NEC 2008* was used to determine feeder and conduit sizes for the new mechanical equipment. The difference between the original design of Alternative 1 and Alternatives 2, 3, and 4 was small and therefore no changes to the feeder or conduit sizes were made.

8.3.5 Panelboard Schedule

The panelboard schedule for the modified panelboard is shown below in **Figure 61**. The panelboard will serve all new mechanical equipment which includes the chiller, two AHU's, and two pumps.

PANELBOARD SCHEDULE

VOLTAGE:	480Y/277V,3PH,4W	PANEL TAG:	RTP-1	MIN. C/B AIC:	10K
SIZE/TYPE BUS:	700A	PANEL LOCATION:	Rooftop Penthouse	OPTIONS:	PROVIDE FEED THROUGH LUGS
SIZE/TYPE MAIN:	700A/3P C/B	PANEL MOUNTING:	SURFACE		FOR PANELBOARD 1L1B

DESCRIPTION	LOCATION	LOAD (WATTS)	C/B SIZE	POS. NO.	A	B	C	POS. NO.	C/B SIZE	LOAD (WATTS)	LOCATION	DESCRIPTION
CH-1	Roof	103136	450/3P	1	*			2		0		0
CH-1	Roof	103136		3		*		4		0		
CH-1	Roof	103136		5			*	6		0		
AHU-1	Roof	24912	100/3P	7	*			8		0		
AHU-1	Roof	24912		9		*		10		0		
AHU-1	Roof	24912		11			*	12		0		
AHU-2	Roof	24912	100/3P	13	*			14		0		
AHU-2	Roof	24912		15		*		16		0		
AHU-2	Roof	24912		17			*	18		0		
P-1	Roof	7474	30/3P	19	*			20		0		
P-1	Roof	7474		21		*		22		0		
P-1	Roof	7474		23			*	24		0		
P-2	Roof	7474	30/3P	25	*			26		0		
P-2	Roof	7474		27		*		28		0		
P-2	Roof	7474		29			*	30		0		
		0		31	*			32		0		
		0		33		*		34		0		
		0		35			*	36		0		
		0		37	*			38		0		
		0		39		*		40		0		
		0		41			*	42		0		

CONNECTED LOAD (KW) - A Ph.	167.91	TOTAL DESIGN LOAD (KW)	503.72
CONNECTED LOAD (KW) - B Ph.	167.91	POWER FACTOR	0.90
CONNECTED LOAD (KW) - C Ph.	167.91	TOTAL DESIGN LOAD (AMPS)	674

Figure 61 - Panelboard Schedule

8.4 Conclusions

The difference in the electrical system requirements between the original design and the proposed redesign is negligible. There was a reduction in connected mechanical equipment load which was translated into a reduction in the panelboard sizing and the main distribution panel sizing, however, the cost savings would be small to negligible as well. Considering that this building is existing, it would be more expensive to remove the existing electrical equipment and replace it with equipment rated for a smaller capacity. The final recommendation is therefore, that the existing electrical system will work well for Alternative 1, and will be slightly oversized, though still practical, for Alternatives 2 through 4.

9.0 Structural Breadth Analysis

9.1 Objectives

The primary purpose for doing a structural study is to determine whether or not the existing structure is capable of supporting the added weight of the purposed mechanical equipment. In the case where the structure is not adequate, a redesign will be done to ensure that all mechanical equipment can safely reside on the roof of Park Place 1.

Figure 62 shows a summary of the weight that will be added to the roof of Park Place 1 under the most intensive alternative (Alternative 3) by weight.

Summary of Weight Added			
Unit Name	Unit Type	Quantity	Unit Weight
CH-3	Chiller	1	13 Kips
AHU- 1,2	Air Handling Unit	2	25 Kips
IT-1,2	Ice Storage Tank	2	17 Kips

Figure 62 – Summary of Weight Added to the Roof Structure

9.2 Overview

The existing roof structure was examined to determine where the most structurally sound areas were located. It was determined that the north side of the building offered the largest columns combined with the shortest spans and therefore the best opportunity to add additional equipment. This part of the roof also currently supports the existing packaged rooftop units of the building therefore some consideration was taken by the structural engineer to increase the structural strength on that side of the building. The question is whether or not it is enough for the vastly larger equipment proposed in Alternatives 2, 3, and 4. Alternative 3 specifically has the largest total combined weight of all equipment and will therefore be used for analysis. Should Alternative 3 comply with the existing design, then it will be assumed that so too can Alternatives 2 and 4.

Figure 63 below shows the relevant parts of the roof structure as well as where the proposed mechanical equipment will be placed. It also provides names for the beams, girders, and columns throughout the rest of this study.

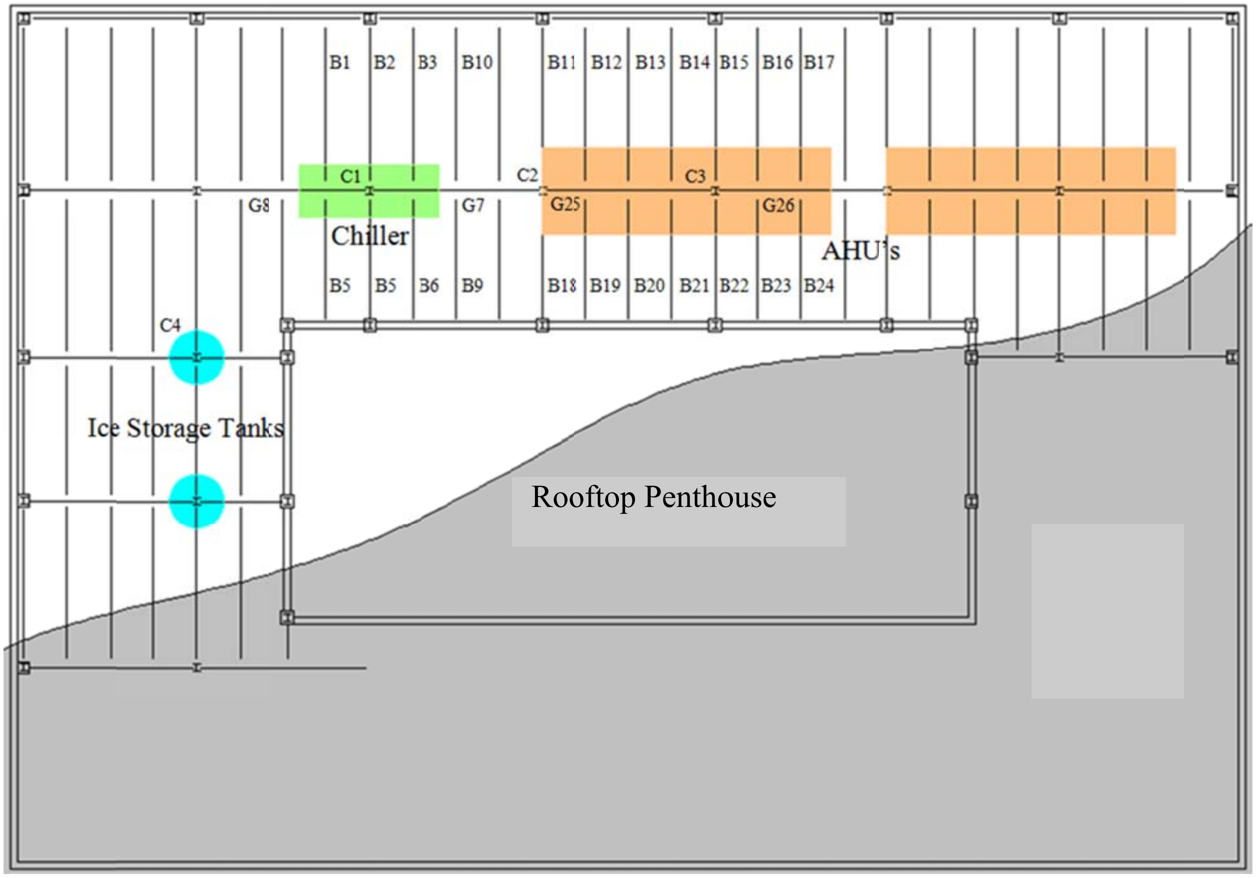


Figure 63 - Rooftop Structure and Mechanical Equipment Layout

A summary of the beams, girders, and columns can be seen below in **Figure 64**. The corresponding calculations and graphs can be seen in Section 9.3 and the Appendix of this report. All calculations were done using RISA, a structural system modeling program. Compliance is designated as either passing or failing in the final column of **Figure 64** below.

Beam, Girder, Column Summary									
Name	Category	Size	Span (ft)	Tributary Span (ft)	Reactions (kips)		Capacity (kip-ft)		Compliance?
					South Node	North Node	Required (M_u)	Designed (Φ*M_n)	
B1	Beam	W 12x19	24	6	8.218	5.041	31.99	92.6	Yes
B2	Beam	W 14x22	24	6	8.218	5.041	31.99	125.0	Yes
B3	Beam	W 12x19	24	6	8.218	5.041	31.99	92.6	Yes
B4	Beam	W 12x14	17	6	3.775	6.713	17.78	65.2	Yes
B5	Beam	W 24x68	17	6	3.775	6.713	17.78	664.0	Yes
B6	Beam	W 12x14	17	6	3.775	6.713	17.78	65.2	Yes
B9	Beam	W 12x14	17	6	3.366	3.366	14.306	65.2	Yes
B10	Beam	W 12x19	24	6	4.752	4.752	28.512	92.6	Yes
B11	Beam	W 14x22	24	6	5.679	3.477	27.468	125.0	Yes
B12	Beam	W 12x19	24	6	11.358	6.954	54.936	92.6	Yes
B13	Beam	W 12x19	24	6	11.358	6.954	54.936	92.6	Yes
B14	Beam	W 12x19	24	6	11.358	6.954	54.936	92.6	Yes
B15	Beam	W 14x22	24	6	11.358	6.954	54.936	125.0	Yes
B16	Beam	W 12x19	24	6	11.358	6.954	54.936	92.6	Yes
B17	Beam	W 12x19	24	6	11.358	6.954	54.936	92.6	Yes
B18	Beam	W 24x68	17	6	3.237	4.533	18.117	664.0	Yes
B19	Beam	W 12x14	17	6	6.475	9.065	36.234	65.2	Yes
B20	Beam	W 12x14	17	6	6.475	9.065	36.234	65.2	Yes
B21	Beam	W 12x16	17	6	6.475	9.065	36.234	75.4	Yes
B22	Beam	W 24x68	17	6	6.475	9.065	36.234	664.0	Yes
B23	Beam	W 12x19	17	6	6.475	9.065	36.234	92.6	Yes
B24	Beam	W 12x19	17	6	6.475	9.065	36.234	92.6	Yes

Name	Category	Size	Reactions (kips)		Capacity (kip-ft)		Compliance
			West Node	East Node	Required	Designed	
G7	Girder	W 18x35	17.287	13.88	117.855	249.0	Yes
G8	Girder	W 18x50	10.488	15.652	98.237	379.0	Yes
G25	Girder	W 18x35	30.634	30.634	245.076	249.0	Yes
G26	Girder	W 18x35	26.717	18.881	198.063	249.0	Yes

Name	Category	Size	Height	Capacity (kips)		Compliance
				Required (P_u)	Designed (Φ * P_n)	
C1	Column	W 10x49	12'	47.87	513	Yes
C2	Column	W 10x49	12'	54.726	513	Yes
C3	Column	W 10x49	12'	77.774	513	Yes
C4	Column	W 10x49	12'	37.94	513	Yes

Figure 64 – Beam, Girder, Column Summary

From **Figure 64** above, it is evident that all structural members are capable of supporting the newly added mechanical equipment of the proposed alternatives. Therefore, no beams, girders, or columns need to be redesigned.

Apart from the steel structural members, the roof deck was also analyzed for compliance and is summarized in the following section.

9.3 Calculations and Analysis

Example calculations for a beam, girder, and column are shown in the proceeding section. The calculation begins with the loads that are existing, continues with the loads created by the new equipment, then finally summarizes inputs and subsequently outputs from RISA. Outputs for the other beams not shown in the following example can be found in the Appendix of this report.

The assumptions made throughout the calculations can be seen below in **Figure 65**.

Assumptions			
Element		Loads (psf)	
		Dead	Live
Existing	Snow	-	30
	Superimposed*	15	-
	Dead Load	-	-
	Total	15	30
New	AHU	52	-
	Chiller	91	-
	Ice Storage	388	-
	Concrete Slab	50	-

Misc.	1. All connections are simply connected (pinned). 2. Concrete is assumed to carry 150 pcf of dead load. 3. Concrete slab will be 4" thick 4. Wind loads will be assumed negligible relative to orginal desig

*Note: Superimposed Dead Load includes (MEP, Ceiling, Lighting, Electrical Conduit, etc.)

Figure 65 – Structural Assumptions

For Beam B1

Equations:

$$\text{Dead Load x Safety Factor}_{\text{Dead Load}} + \text{Live Load x Safety Factor}_{\text{Live Load}} = \text{Factored Load}$$

$$\text{Pounds per ft}^2 \text{ x Tributary Length [ft] / 1000 [lbs/kip] = kips per linear ft}$$

Calculations:

Existing Load:

$$\left(15\ [psf]_{\text{Total DL}} \times 1.2\right) + \left(30\ [psf]_{\text{Total LL}} \times 1.6\right) = 66\ [psf]$$

$$66\ [psf] \times 6\ [ft] / 1000\ [lbs/kip] = 0.369\ [klf]$$

New Load:

$$(50\ [psf]_{\ Slab} + 91\ [psf]_{\ Chiller}) = 141\ [psf]$$

$$141\ [psf] \times 6\ [ft] / 1000\ [lbs/kip] = 1.015\ [klf]$$

Therefore, the distributed loads on beam B1 are 0.369 klf for the entire span of the beam (24 ft) and 1.015 klf for the part of the beam that the chiller and slab rest on which equates to half of the width of the chiller—3.7 ft. The length of 3.7 ft is used because half of the chiller width rests on beam B1 and the other half rests on beam B5.

The beam was modeled accordingly and can be seen below in **Figure 66**. The chiller load on the right side of **Figure 66** spans 3.7 ft while the existing load spans the entire length.

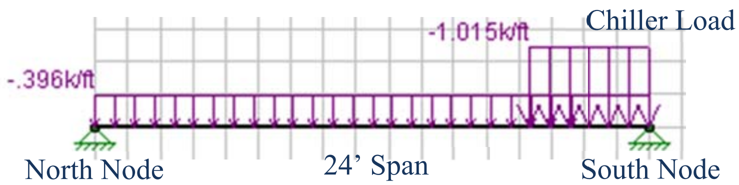


Figure 66 – RISA Model of Beam B1

Three values were collected from the output of RISA. The first and second were the reactions at each of the two nodes—this information was useful in analyzing the girders. The third piece of useful information was the maximum moment on the beam which would dictate whether or not the beam was capable of supporting the required load.

These three pieces of information are presented above in **Figure 64** and dictate compliance with the available capacity in the beam.

For Girder G7

For analyzing girder G7, the reactions that the beams create on the girder must be collected and added at each point of contact. This means that the beams were analyzed first and the resulting reactions were complied to study the girder. The resulting point loads are shown below in **Figure 67**.

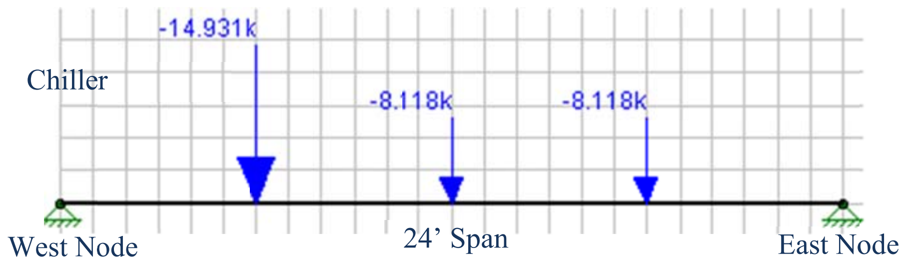


Figure 67 – RISA Model of Beam G7

In **Figure 67** above, the resulting point load of -14.931 k is equal to the summation of the south reaction of beam B3 and the north reaction of beam B6. When looking at the summary table in **Figure 64** the corresponding values just mentioned are -8.218 k and -6.713 k which, when added together, equals the mentioned -14.931 k.

For Column C1

$$\text{Total Load on Column} = \Sigma\left(\text{Total Load of Supported Girders}\right)$$

$$\text{Total Load on C1} = \text{South Reaction of B2} + \text{North Reaction of B5} + \text{West Reaction of G7} + \text{East Reaction of G8}$$

$$\text{Total Load on C1} = 8.218\ [\text{kips}] + 6.713\ [\text{kips}] + 17.287\ [\text{kips}] + 15.652\ [\text{kips}]$$

$$\text{Total Load on C1} = 47.87\ [\text{kips}]$$

The total load on column C1 is therefore 47.87 kips which is less than the maximum capacity that column C1 is capable of handling—513 kips. Thus, the column complies and can be marked in **Figure 64** as complying.

For Roof Deck

The process for calculating the compliance of the roof deck is relatively straight forward. All that needs to be done is the total load on the roof deck in psf, the span of the deck, and the type of deck must be determined. Once those values have been established, they must be compared to a roof deck catalog that contains rated capacity.

Accordingly, for Park Place 1, the calculation is shown below.

$$\text{Max Roof Deck Load} = \text{Total}_{LL} + \text{Total}_{DL}$$

$$\text{Max Roof Deck Load} = \text{Snow Load} + \text{Chiller Load} + \text{Chiller Slab Load} + \text{Existing Dead Load}$$

$$\text{Max Roof Deck Load} = 30\ [\text{psf}] + 141\ [\text{psf}] + 15\ [\text{psf}]$$

$$\text{Max Roof Deck Load} = 186\ [\text{psf}]$$

The three selection criteria are:

$$\text{Max Roof Deck Load} = 186\ [\text{psf}]$$

$$\text{Roof Deck Span (Tributary Width)} = 6\ [\text{ft}]$$

$$\text{Roof Deck Span (Bays)} = 3\ [\text{Bays}]$$

Required Roof Deck Type = 1.5 B16

Existing Roof Deck Type = 1.5 B18

What the above information means is that the existing roof deck at Park Place 1 is not adequate to support the proposed equipment of Alternatives 2, 3, 4. If Alternative 2, 3, or 4 were to be implemented, the roof deck would need to be replaced.

9.4 Conclusions

The structural study done on the roof structure of Park Place 1 indicates that the steel members of the building (beams, girders, and columns) are capable of supporting the new mechanical equipment of the proposed alternatives.

It also finds that the roof deck is not capable of supporting the distributed loads created by the new mechanical equipment. If the new mechanical equipment were to be used, a thicker gage of metal deck must also be installed. This would come at considerable cost to the redesign and would need to be studied further to determine economic impacts.

10.0 Credits and Acknowledgements

This thesis is dedicated to my parents, Jeff and Lorraine Blood.

I would also like to thank all who have helped me get to this point, the AE department, my friends, and my fellow AE's.

A special thanks to:

Sam DiCicco - DiCicco Development
Craig Duda - CJL Engineering
Harry Hoover - CJL Engineering
Chris Carroll - CJL Engineering
Tim White - Pennsylvania Trane
Jim Kusick - Pennsylvania Trane
Carl Caruso - Pennsylvania Trane
Greg Basinger - Pennsylvania Trane
Paul Valenta - Calmac
William Bahnfleth - Thesis Advisor

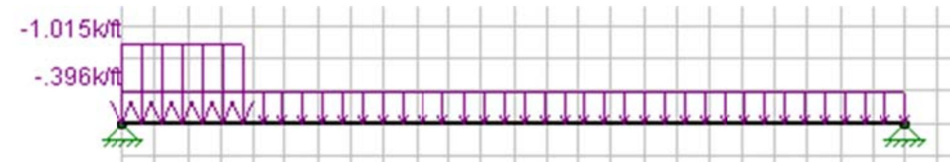
11.0 List of Figures

APPENDIX

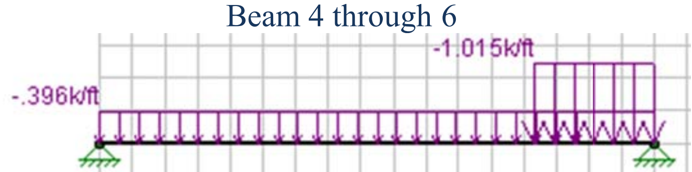
RISA Outputs

*Note: For beams, the right side of the page is considered the south node, the left side the north. The shorter spans are 17 ft while the longer spans are 24 ft.

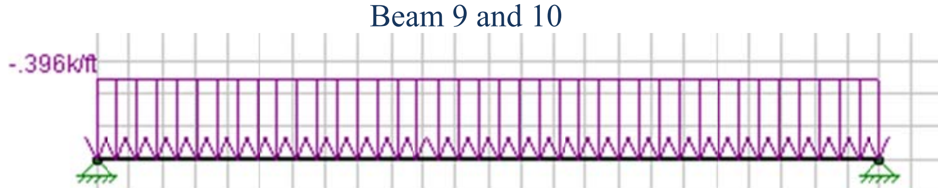
Beam 1 through 3



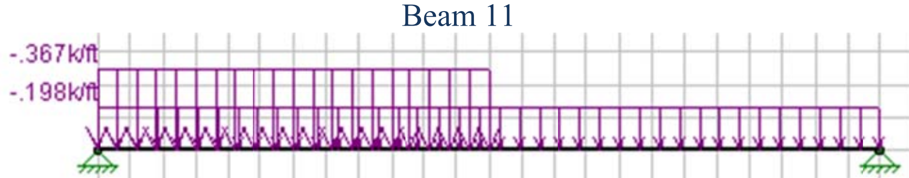
Beam 4 through 6



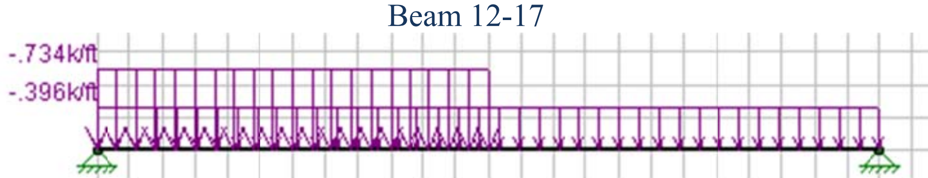
Beam 9 and 10



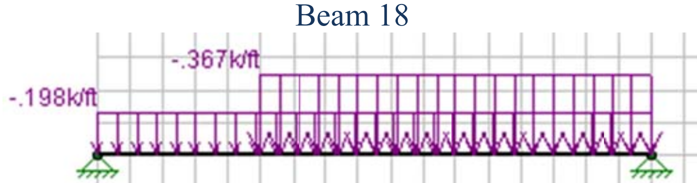
Beam 11



Beam 12-17



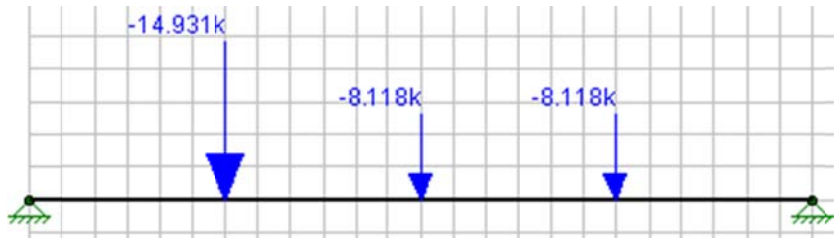
Beam 18



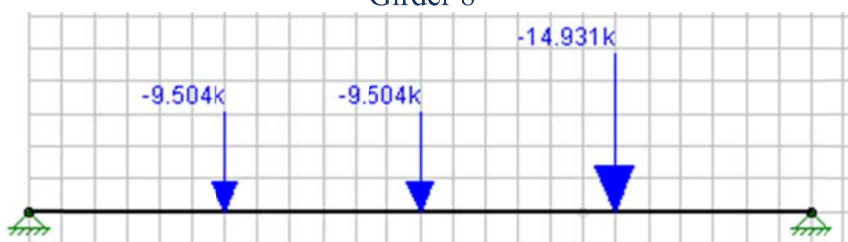
Beam 19-24

*Note: For girders, the left side of the page is the west node while the right side of the page is the east side. All spans are 24 ft.

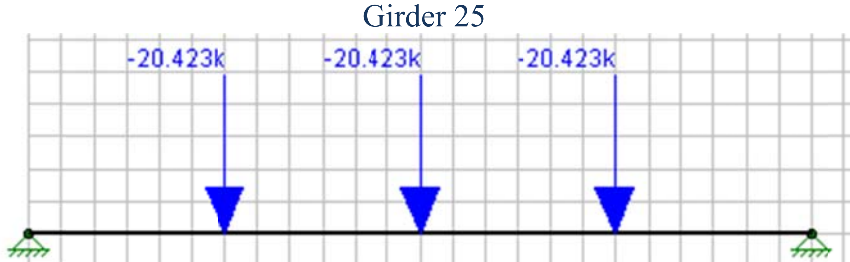
Girder 7



Girder 8



Girder 25



Girder 26