Ice Storage System Design

Rendering Courtesy Moody·Nolan, Inc.

George W. Hays PK-8

Cincinnati Public School

Cincinnati, OH

Prepared For

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By

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Architectural Engineering

Mechanical Option

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George W. Hays PK-12 Public School

Cincinnati, OH

CINCINNATI PUBLIC SCHOOLS

Project Overview

TOTAL AREA
66,338 ft^2

BUILDING FOOTPRINT
35288 ft^2

ABOVE GRADE STORIES
Three

TOTAL BUILDING HEIGHT
75 ft

CONSTRUCTION COSTS
$11,149,342

Project Design Team

Moody-Nolan, Inc.
Architect & Civil Engineer

ThermalTech Engineering, Inc.
MEP Engineer

GOP Limited
Structural Engineer

Turner/DAG/TYS
Construction Manager

Structural System

5 INCH SLAB ON GRADE WITH MESH AND POLYPROPYLENE FIBERS

ELEVATED FLOORING SYSTEM CONSISTS OF CONCRETE SLABS ON METAL DECKING

OUTSIDE WALLS ARE COMPOSED OF A BRICK VENEER WITH CEMENT MASONRY BLOCK BACK UP

EPDM MEMBRANE ROOF SYSTEM WITH RIGID INSULATION AND METAL ROOF DECK

Electrical System

MAIN SWITCH BOARD: 2000A, 480Y/ 277V, 3P, 4W, 65000 A/C

PRIMARY SERVICE: 480Y/ 277V, 3P, 4 WIRE
SECONDARY SERVICE: 208Y/ 120V, 3P, 4 WIRE

60 kW NATURAL GAS DRIVEN EMERGENCY GENERATOR

Lighting System

MAIN LIGHTING SYSTEM IS 2' X 4' GRID MOUNTED FLUORESCENT TROFFERS

GYMNASIUM LIGHTING WITH 22" DIAMETER LOW BAY FLUORESCENT FIXTURES

Mechanical System

THREE VAV AHU's WITH HEATING AND COOLING WATER COILS WITH AIR FLOW CAPACITIES OF 22,000; 18,000; & 12,000 CFM

EACH AHU HAS A TOTAL ENERGY WHEEL AND VFD

78 SINGLE DUCT OR SERIES FAN POWERED TERMINAL DEVICE WITH LOCAL HOT WATER RE-HEAT AND PLENUM RETURN

TWO RADIANT PANELS WITH CAPACITIES OF 853 & 1280 MBH

ONE 170 TON AIR COOLED CHILLER

TWO 1500 MBTU/HR NATURAL GAS BOILERS

RODRICK A CROUSEY

WWW.ARCHE.PSU.EDU/THESIS/EPORTFOLIO/2007/PORTFOLIOS/RAC254/

MECHANICAL OPTION

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Jamie White	LLI Engineering

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Executive Summary

This report is an educational document that examines the design and redesign alternatives of the George W. Hays PK-8 School in Cincinnati, OH. All of the redesign ideas are based upon the proposal of implementing an ice storage system. The report analyzes different ice storage types and strategies after which a complete annual simulation and analysis of three scenarios was done. The first calls for a reduction in chiller size from 170 tons to 100 tons. This case requires an ice storage capacity of 358 ton-hr. The second case involved a 90 ton chiller with an ice storage system of 486 ton-hr. Finally, the third system was an 85 ton chiller with an ice storage system of 600 ton-hr.

Each of these systems saw an increased first cost due to the introduction of an ice storage tank, slab on grade, and ice storage components including a glycol solution, glycol monitoring equipment, and glycol mixing equipment. This increase in cost exceeded cost reductions from a reduced chiller size, reduced electrical equipment and reduced piping. These increases in costs ranged from $7,876 to $25,046.

The annual electric bill decreased in each of the three scenarios. Despite an increase in overall electric use, the electrical demand limitations reduced the annual electric bill by $1,575 to $3,979.

The final cost analysis showed that the 90 ton chiller and 486 ton-hr ice storage tank was the most economical decision with a payback period of 3 years. This report used this payback along with other advantages of an ice storage system to conclude that the implementation of an ice storage system in this building would be beneficial.

1.0 System and Building Summary

The mechanical system for the building was designed with the goal of maintaining thermal comfort with minimum energy usage. The components of the system include a single centrifugal chiller, two hot water boilers, and three air handling units.

All of they systems work together to achieve the mechanical goals of the system. To help ensure a proper monitoring and coordination of this system, a direct digital control system was called for that allows the owner to monitor and record all of the major system components from locations away from the site. The system components work together to manage four daily timeframes: Unoccupied, Startup, Occupied, Coast-Down. Since no occupants are expected to be in the space, the Unoccupied timeframe has no requirements for ventilation or thermal comfort. To save energy the system is turned off during the night hours. Towards the end of the first Unoccupied period the Startup is activated, where the system activates prior to occupancy. This Startup period is necessary because of the lag systems naturally have due to thermal mass and unconditioned air in the space overnight. Because of unknown factors regarding the response of a system prior to construction, the building controls system has a memory that continuously adjusts the Startup time based upon previously recorded data. During the Occupied hours the system is run in a way to achieve thermal comfort and required ventilation to the space. Towards the end of the Occupied timeframe is the Coast-Down. During the Coast-Down period, the thermal components of the system begin to turn off with the anticipation of the thermal lag of the building maintaining thermal comfort conditions until the Occupied period is over and the second Unoccupied period of the day begins. Like the Startup, the Coast-Down period changes based on previously recorded data. By implementing this system, thermal comfort is ensured in the early hours of the day and energy is saved in the afternoon by taking advantage of the natural lag of the building.

1.1 Cooling Systems

The only active cooling system for the building is a single 170 ton centrifugal chiller, CHLR-1. This system is only responsible for serving the cooling coils in the three AHU's. The chiller is designed to run at a set supply water temperature of 43°F. The controls logic calls for the chiller to be activated any time the Outdoor Air (OA) temperature is greater then 55°F degrees and at least one AHU is in occupied mode. When the OA temperature falls bellow 50°F, the chiller is disabled and the AHU's are put into full economizer mode,

which is further discussed in the description of the AHU's. The chilled water bypass valve is staged according to the Differential Pressure (DP) of the evaporator to ensure the minimum recommended flow rate stated by the chiller manufacturer.

A single 300 gpm pump provides the required pressure drop for the circuit. To help prevent cavitation, a suction diffuser is incorporated at the inlet of the pump. Flow conditions can be verified with a DP gage across the pump and suction diffuser. A single gage is used in this application connected to pipes from three locations: prior to the diffuser, in between the pump and diffuser, and after the pump. This gage reads absolute pressures at the different locations at different points in time. The absolute pressures are then subtracted to find the differential pressure across the desired component. Having a single gage instead of multiple gages ensures an accurate DP even if the gage is not reading the proper absolute pressure. Details such as the single gage are implemented to ensure the future maintenance team will have access to adequate knowledge about the operating conditions of the system.

1.2 Heating Systems

The central heating system for the building is served by a hot water system containing two identical 1,500-MBH non-condensing boilers. In accordance with initial design goals the boilers have a high efficiency, each with two variable frequency drive secondary pump motors. The boilers are designed for a supply water temperature of 180°F. Hot water supply temperatures vary based on OA temperature. In addition to the central heating system, the boilers serve several cabinet unit heaters and local reheat coils at the Variable Air Volume (VAV) boxes.

Previous experiences by the mechanical engineer had shown that school reception areas are more likely to receive complaints about not falling within the bounds of the occupants desired thermal comfort region. For this reason the design called for a 1280 MBH electric radiant panel in the reception area that included a thermostat that could be controlled by the occupant.

1.3 OA Summary and Findings

The building is broken up into three main zones. Each of the three air handling units is responsible for supplying an appropriate amount of OA to its respective zone. Each zone is mainly limited to a particular type of space. This helps to keep the critical space representative of all the spaces in the zone, because the minimum E_{vz} can be expected to be somewhat similar for

spaces serving a similar function with similar OA requirements, thus limiting the amount of OA brought to unnecessary spaces. Each zone is served by one AHU. Each AHU is an indoor modular Air Handling Unit located in a mechanical room or mezzanine. Each AHU has an integral heat recovery wheel, a return or relief fan, an economizer section, heating and cooling coil, and a supply fan. The zone breakups according to AHU are shown in Figure 1-1.

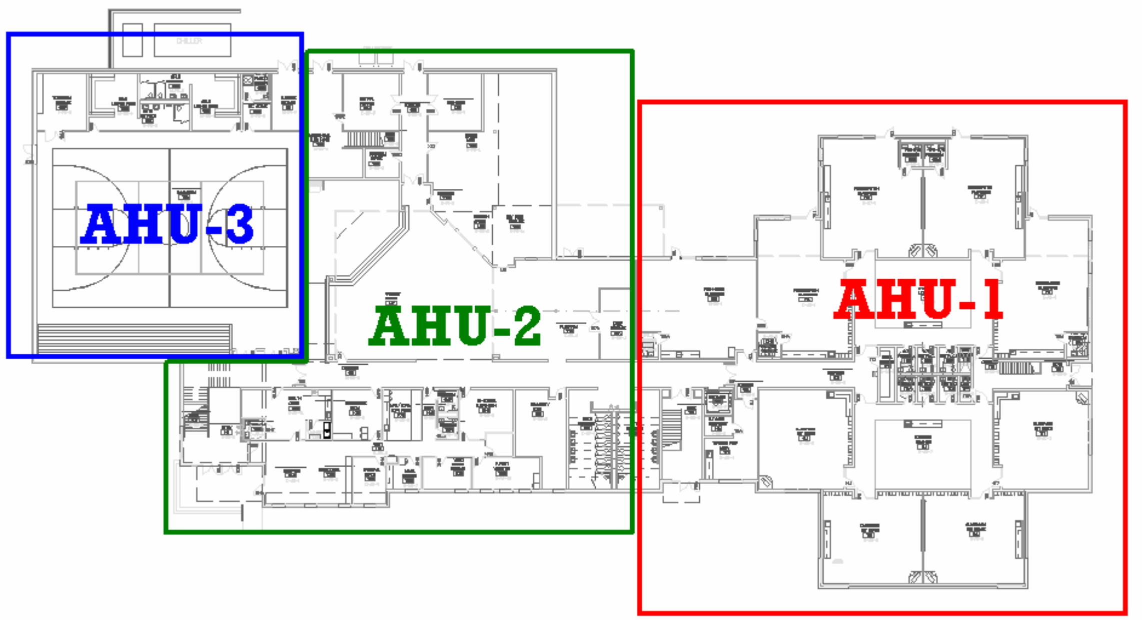

Figure 1.1
Air Handling Unit Zone Distribution

Table 1.1 highlights the basic comparisons between the designed OA flow and the ASHRAE 62.1 calculated OA flow rates according to ASHRAE Standard 62.1.

The sum of the V_{oz} values in all zones served by AHU-1, AHU-2, and AHU-3 was 18,890 cfm, or 65% of the some of the V_{ot} values of 29,142 cfm. The reason for this increase is the critical zone requiring a higher fraction of OA then some of the other zones. In order to supply a sufficient amount of OA to the critical space, the system is forced to supply excess OA to non-critical spaces.

TABLE 1.1

Unit	Type	Zone Served	Net Area Served [Sq. ft]	Calculated OA [CFM]	Calculated OA Intake Percentage	Designed OA [CFM]
AHU-1	VAV	Three story classroom wing	24,700	13,529	50%	11,066
AHU-2	VAV	1st and 2nd floor classrooms and auxiliary areas	19,100	9,078	40%	8,296
AHU-3	VAV	Gymnasium and the gymnasium support areas	6,900	6,535	51%	8,632

Air Handling Unit Summary

1.4 AHU-1

Air Handling Unit 1 supplies the three-story classroom wing of the building. The net area served by AHU-1 is 24,700 ft^2. This gross area (including walls and spaces not in the Breathing Zone or not in spaces requiring OA) is 31662 ft^2. The unit has a single supply VAV fan that moves 22,000 cfm with a static pressure drop of 6 in wg. This fan is responsible for the pressure drop from the OA intake to each of the VAV boxes. From there, the individual VAV boxes supply an adequate pressure to supply the air to the individual spaces. The return fan has a capacity of 19,000 cfm with a design static pressure drop of 3 in wg. This fan draws the air from a plenum return to a short length of duct where it is then either blown into the mixed air supply or blown out of the building as exhaust air.

The design mixed air temperature for the chilled water coil is 81.1°F DBT and 66.1°F WBT. 936 MBH of cooling is required to bring the supply air conditions to 52.4°F DBT and 51.8°F WBT. A heating coil of 741 MBH of heating is required to bring the heating design entering coil temperature of 33.8°F to the winter supply temperature of 65°F. The supply air volume is determined by adjusting flow based upon the static pressure in the ductwork with a minimum flow volume above 11,066 cfm to ensure the required minimum value of OA is always supplied. The pressure in the ductwork changes as the local VAV boxes adjust airflow volumes based on space temperature.

To help reduce loads on the building and to achieve design goals, a total enthalpy wheel is used to pre-condition the OA. This design is effective because of a high percentage of OA. The high OA percentage directly correlates to a high Exhaust Air (EA) volume. Though necessary for ventilation, the high EA rate results in a rejection of Return Air (RA). Since the

RA is many times closer to the SA conditions then the OA is, the high OA percentage increases the load across the heating and cooling coils. A total energy wheel transfers both latent and sensible energy between the RA and OA. Dumping energy into the OA stream in winter and extracting energy from the OA stream in summer reduces the load on both of the coils.

For transition seasons, AHU-1 enters economizer mode when the OA temperature is closer to the desired SA temperature then the RA temperature is. In economizer mode the system brings in above-design OA to save on energy use. Since the OA is closer to supply conditions then the RA is, the load on the coil is reduced.

AHU-1 complies with Section 5 of ASHRAE Std. 62.1. The OA intake is on an elevated vertical wall in a location free from the potential contaminant sources detailed in Table 5-1 of 62.1.

The sum of the V_{oz} values in the zones served by AHU-1 was 9,746, or 72% of the V_{ot} value of 13,529 cfm. The reason this percentage is higher then the percentage for the entire building is due to two major components; the diversity factor applied to this space and critical Z_d value in this space being somewhat representative of the other spaces served by AHU-1.

The OA fraction, Z_d (the equivalent to Z_p but for Appendix A from ASHRAE Std. 62.1) for the critical space was 0.7. Because calculations were done using Appendix A, the minimum E_{vz} was the value that determined the critical space, not the maximum Z_p. For AHU-1 the minimum value for E_{vz} was 0.59 from the Extended Learning Area rooms: 113, 120, 211, 218, and 306. This value represents a dense population and low envelope, resulting in a high OA%.

1.5 AHU-2

Air Handling Unit 2 supplies a two story office and auxiliary classrooms wing of the building serving a net area of 19,100 sq ft. The total area of this zone is 21,451 sq ft. AHU-2 has a design airflow of 18,000 cfm with a supply fan designed for 6 in wg. It is equipped with a total enthalpy wheel and space heating is done by hot water in each zone.

AHU-2 complies with section 5 of ASHRAE Std. 62.1. The OA intake is on a roof, but elevated more then 1 ft above the roof, meeting the requirements of Table 5-1 of 62.1.

The sum of the V_{oz} values in the zones served by AHU-2 was 4,678, or 52% of the V_{ot} value of 9,078 cfm. The reason for this large difference in values is due to no diversity factor being applied for the space, and the critical zone not being representative of the other zones served by AHU-2.

The OA fraction, Z_d, for the critical space was 0.87.

The design OA was 46% of the total flow rate or 8,296 cfm. Similar to AHU-1, this value is only 91% of the 9,078 cfm calculated. The reasons for this discrepancy are like those mapped out for AHU-1 and because of differing OA requirements at the critical space. The design assumed 20 cfm/per for a workshop, Room 133, resulting in 40 cfm. From ASHRAE 62.1 it was assumed 10 cfm/per and .18 cfm/sq ft for a workshop resulting in 73 cfm of OA. This OA requirement caused the E_{vs} to become 0.31, making Room 133 the critical zone. The next lowest E_{vs} value was 0.41, Room 224. If the workshop, Room 133, was supplied OA by another unit or by other means, the E_v value for the system would be 0.41. The resulting OA would then be 6872 cfm reducing the required OA by 76%, allowing the current design to meet with ASHRAE 62.1.

The percentage OA calculated was 40%, which is comparable to the 46% designed. The lower OA percentages in this wing of the building are because of more offices or other low density occupancies.

1.6 AHU-3

Air Handling Unit 3 supplies the gymnasium and gymnasium support areas. The net area of these spaces is 6,900 ft^2 with a gross area of 8,844 ft^2. Because of the high OA percentages and the availability of implementing a total energy wheel, AHU-3 is a 100% OA system. Because of the uniqueness of this system, the controls are determined directly by space temperature, not duct pressure.

AHU-3 complies with Section 5 of ASHRAE Std. 62.1. The OA intake is on an elevated vertical wall in a location free from the potential contaminant sources detailed in Table 5-1 of 62.1.

The sum of the V_{oz} values in the zones served by AHU-3 was 4,466, or 68% of the V_{ot} value of 6,535 cfm. This value falls in line with the average of all three AHU's.

The critical space served by AHU-3 was the gymnasium, Room 105, with a E_{vs} value of 0.68. As shown in the assumptions, the gymnasium was assumed to

be entirely a spectator area and not a play area. This is because of the potential of large gatherings using the gymnasium as a seating area. This assumption resulted in a dense population, increasing the percent OA required for the zone making a 100% OA system logical for these zones.

2.0 Design Load Estimation

Carrier's Hourly Analysis Program (HAP) was used to model George W. Hays facility to find a design load estimation. This estimation is based off of data taken from design documents and heating and cooling outdoor air conditions from the ASHRAE Handbook of Fundamentals, 2005. The summer design conditions are based off of the temperature that weather data has shown to exceed 0.4% of the time. Conditions beyond these values are not part of the summer design conditions because the building has a thermal mass that will be able to absorb energy as long as the OA conditions do not exceed the 0.4% range for an extended period of days. A dry bulb temperature of 93°F and a wet bulb temperature of 74°F are the conditions that meet the 0.4% condition. For cooling design the 99.6% condition of 4°F was used, meaning that 99.6% of the days in Cincinnati, Ohio are shown to exceed 4°F. The exact values for lighting were used in all spaces by looking at the electrical schedules and drawings for each space. The space occupancies and square footages were determined from building drawing documents.

The design simulation resulted in a total cooling load close to what the drawings suggested. The scheduled chiller has a nominal capacity of 170 tons (165 actual tons according to design documents) and the HAP analysis called for 158 tons of refrigeration. Discrepancies between the modeled systems and the way the two different programs interoperate the systems may be a reason for error. The HAP analysis finding that the building required 158 tons of peak cooling capacity was used as the basis of the analysis of the building.

Despite the HAP analysis calling for the proper nighttime shutdown of the system, the HAP output was still showing nighttime cooling requirements. For future analysis of the building, this output was used as a base comparison and foundation for the redesign ideas.

3.0 Design Goals

According to the mechanical engineer, the main mechanical focus for all projects for the Cincinnati Public Schools is energy conservation. From previous experience, the design team anticipated a high percentage of OA in each of the three zones. Zone 1, consisting of mainly classrooms and Zone 2, consisting of a mixture of classrooms, offices, and general spaces were expected to have an OA percentage around 50%. The zone for the gymnasium and the gymnasium support area, Zone 3, was expected to have an even higher OA percentage around 70%. Due to the high percentage of required outdoor air, complete enthalpy wheels were implemented as an early design objective. The design team was also focused on implementing high efficiency boilers to supply the decoupled heating and domestic hot water systems.

Thermal comfort is a goal that is incorporated into every design by the mechanical designer. Thermal comfort means creating an atmosphere at which the occupants are expected to be comfortable in terms of both Dry Bulb Temperature (DBT) and Relative Humidity. This goal is achieved by combining design experience and advice from professional journals such as ASHRAE to combine the components of work level and clothing level to determine a desired DBT and Relative Humidity along with providing a reasonable level of occupant control.

Mechanical designers are also restrained by space limitations. Though there were no initial specific floor area limits on the mechanical system, an initial goal by a mechanical designer is to place the equipment within an area that is agreeable by the architect and owner. Excess mechanical space can result in lost rentable space, or even the possibility of affecting the overall aesthetics of the building.

4.0 Design Conditions

Design conditions include the desired Indoor Air (IA) temperature in addition to the various determined design OA temperatures. The specific values for these various conditions can be found on Table 4.1.

Table 4.1

	SUMMER†			WINTER††
	DBT [F]	WBT [F]	% RH	DBT [F]
OUTDOOR 1*	88	73	-	5
OUTDOOR 2**	-	75	100	-
INDOOR	74	-	50	70
UTILITY SPACES	65	-	-	-

*Design condition based off of DBT
**Design condition based off of WBT
†Summer OA conditions based off of 2% ASHRAE Fundamentals 2001
††Winter OA conditions based off of 99.6% ASHRAE Fundamentals 2001

Design Conditions

All of the OA conditions were determined by the mechanical engineer using ASHRAE Fundamentals 2001. Two different possible summer design OA conditions were of interest to the designer. The first condition is the 2% DBT condition. This value is the DBT that is surpassed 2% of the hours in a year (175 hours per year). A 2% design condition is acceptable because of the thermal mass of a building allowing the building to maintain indoor air conditions when the OA conditions exceed design for a limited span of time. This span of time is not expected to be exceeded when using a 2% design condition. Associated with the design DBT is a Wet Bulb Temperature (WBT). This value gives the designer a point on the psychrometric chart to base the design of the system on. The other potential design condition is the 2% WBT condition. Similar to the 2% DBT condition, the 2% WBT condition is the WBT that is surpassed 2% of the hours in a year. This design WBT has the potential of accumulating a latent load large enough to require more tons of cooling then would be required if looking at the design DBT alone.

The IA conditions were decided by analyzing the conditions of the respective spaces keeping in mind energy usage and thermal comfort. Specific variables taken into account include the amount of clothing occupants are expected to wear, the expected level of activity for the occupants, and the OA conditions. Clothing has an effect on the amount of heat the occupants' bodies are able to reject due to varying thermal resistance. Activity level has been proven to have a direct effect on the occupant's metabolic rate, or the energy that the person is creating. This metabolic variance changes the occupant's perception of what defines comfortable conditions. Finally, the OA temperature has two major influences on the decided space temperature. First, cooling summer air requires more energy then cooling winter air and the converse is also true. Therefore to be energy conscious, summer IA conditions can be decided to be warmer then the winter IA conditions. The second effect deals with acclimation. A human body adjusts over time to different temperatures. Therefore, in the winter months the occupant will define thermal comfort as being cooler then the defined thermal comfort in the summer.

5.0 Proposed Redesign Ideas

All proposed redesign ideas were analyzed in comparison to the current building design with respect to cost, the effect on the occupants of the building, the building's effect on the community around it, and the educational value.

5.1 Mechanical Components

The redesign idea for the mechanical components of the building includes the introduction of an ice storage system. Three main components of ice storage were inspected: proper equipment selection, proper controls methodology, and proper simulation. The ice storage system lends itself to the Hays School because of a summer load profile indicating several large load peaks (Figure 5.1). Distributing cooling energy into the nighttime hours should reduce the

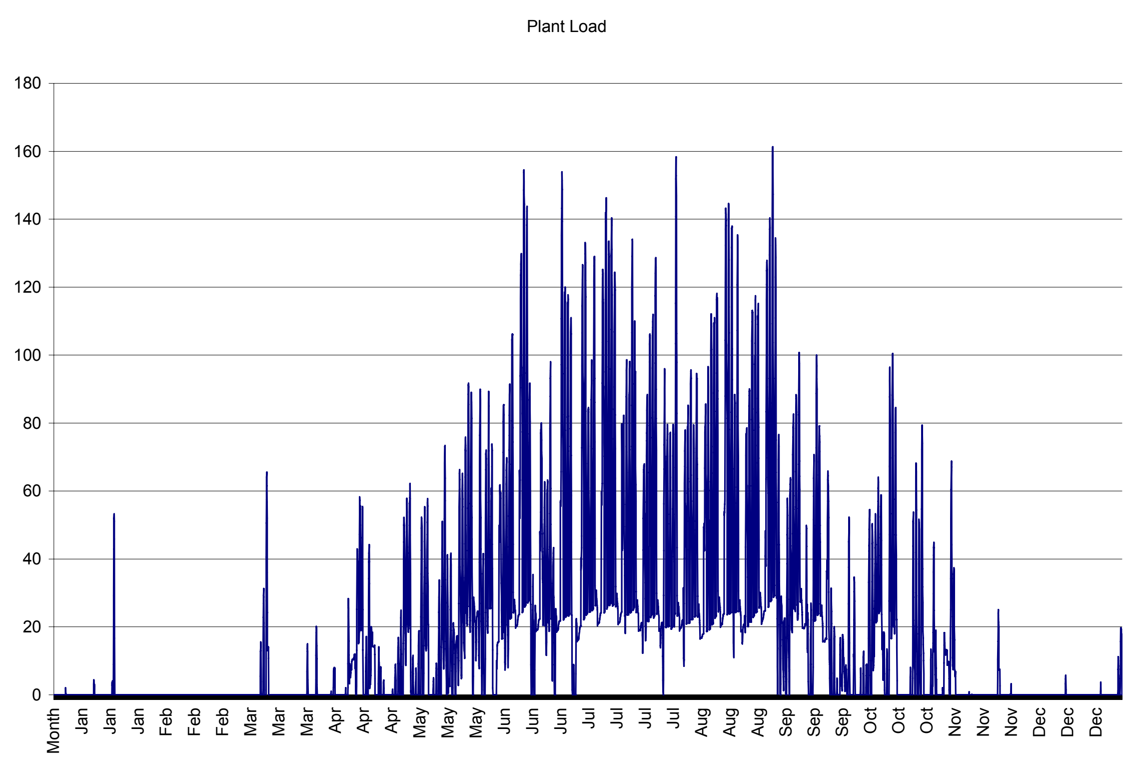

Figure 5.1
Annual Thermal Load Profile

electric bill by lowering the peak electric demand. In addition to lowering costs, this will also lower the demand required for the city power, providing a service that will benefit the community as a whole. The ice storage system will have two expected drawbacks: increased mechanical space and complications with low supply temperatures from the chiller. The design day data was gathered from the Trane Hourly Analysis Program (HAP) file used to analyze the building. This data was then used to look at designing the system based off of full storage, load leveling partial storage, or demand limiting. The method used to select the size of the ice storage system has a direct influence on the method of controls chosen. Using the yearly data from the HAP file and the controls method chosen, an Excel worksheet was created to run a yearly analysis of the system. A lifecycle cost comparison between ice storage and a system not using ice storage was used to determine the practicality of the proposal.

5.2 Breadth Components

The ice storage system will have direct impacts on both the structural and electrical components of the building. Four different scenarios will be analyzed. First is the current scenario where the ice storage is not implemented and the electrical and structural components remain unchanged. The next three scenarios use different sized ice storage tanks. An additional grade level component will serve as support for the physical ice storage tanks. The manufacturer of the ice storage system was contacted for recommendations on the type of support system that would be best. In addition to a slab on grade, several electrical components were downsized as a result of the ice storage system. The redesigns of these components were evaluated for a more in-depth look at the cost comparison of an ice storage system.

6.0 Controls Methodology

The controls help to define a basis for sizing the equipment and analyzing energy savings. These early decisions must be made based off of professional recommendation and considering how the system will respond. The original building design already called for a detailed controls system that would be important for ensuring the ice storage system is working properly. The four main components of the controls methodology are: chiller operation, charging cycle, operation sequencing, and chiller placement in the system.

6.1 Chiller Operation

The two main chiller operation strategies considered were full storage and partial storage, load leveling. Demand limiting strategies were not considered because of the complexities of predicting the load. A mixture between full storage and load leveling partial storage was applied to the Hays

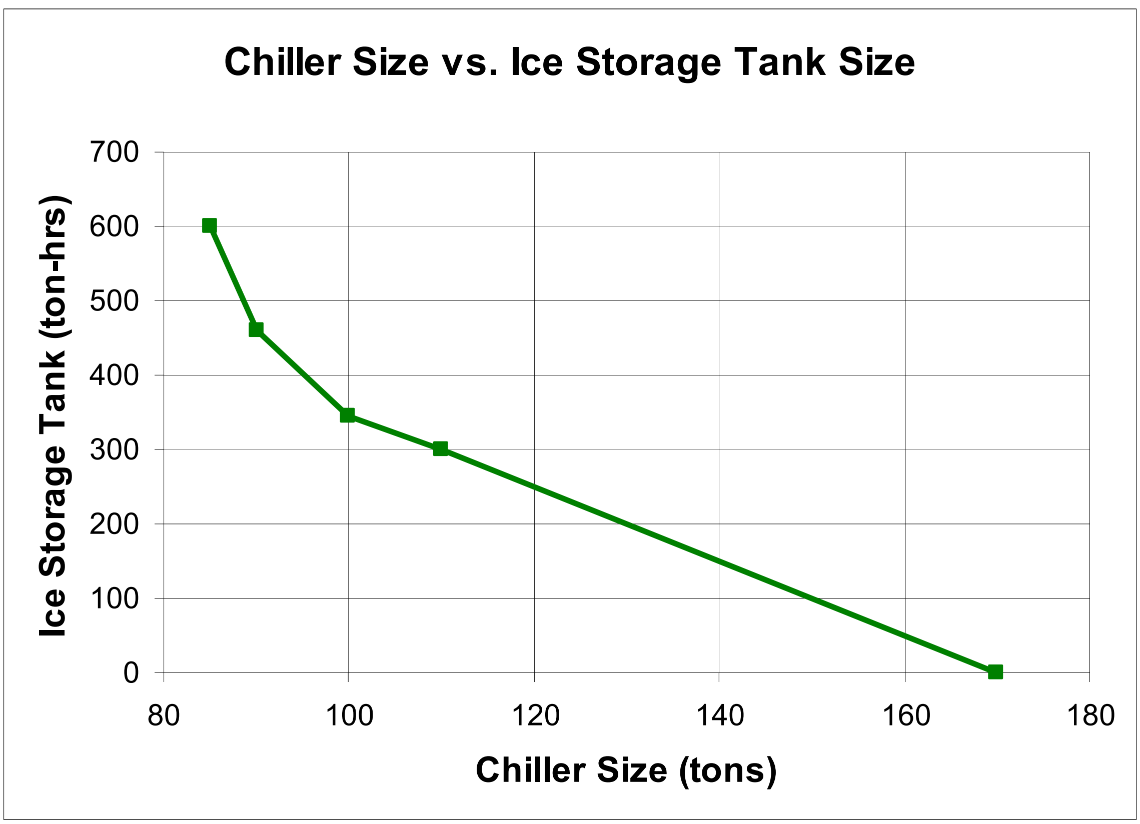

Figure 6.1
Relationship Between Required Ice Storage Size and Required Chiller Size

School. Some of the factors that influenced the design were, peak monthly chiller kW, chiller ice storage size denominations, and a cost analysis of three different chiller size and ice storage tank size combinations. It was determined early on that a full storage system would most likely not be cost effective. Since Cincinnati does not have a reduced off-peak kilowatt-hour (kW-hr) charge, the increased energy usage required for only running the chiller at night in ice-making conditions could result in an overall more expensive electric bill. Another factor as to determining the chiller size for a partial storage system was the relationship between the reduced chiller size and the increased required ton-hr of storage capacity, as shown in Figure 6.1. As this figure shows, smaller chiller sizes results in a nonlinear increase in required ice storage capacity. This figure implies that chillers under 85 tons will result in too large of an increase in the ice storage system to be considered economical. The chiller size was also determined in part by the analysis of the annual thermal load profile shown in Figure 5.1. This figure shows large spikes throughout the summer months. To reduce these spikes and level the thermal load profile again, a chiller size between 80 and 100 tons would be ideal. In accordance with these two comparisons, an 85, 90, and 100 ton chiller were all analyzed to ensure the most cost effective system selection.

6.2 Charging Cycle

A daily charging cycle was considered to be the most economical based off of professional advice and other similar projects. There would not be enough space for an ice storage system of that size and there is nothing in particular about the load profile of the Hays School that would suggest that a week-long load profile would be beneficial. However, with the three analyses that were performed, it was acknowledged that for the 85 ton chiller system it was not possible to achieve as a daily charging cycle. At around 85 tons it becomes necessary for the system to build up ice over multiple days to be able to handle the design day. This raises immediate concerns about the reliability of this system because it was not intended to be analyzed as an extended charging cycle system. If there are multiple high demand days in a row that the HAP file had not prepared for, this ice storage system would lose control. To the best of my understanding, the HAP program was not designed with the intention of preparing the designer for a design week and it is possible that this data is not useable for a week analysis. However, this analysis was completed in its entirety similarly to the other two systems for educational reasons and interest in whether or not a chiller of this size would be economical if there was not a concern about losing control of the system.

6.3 Operation Sequencing

The two main strategies of the system operation are chiller priority and storage priority. In a storage priority system, at the point in the month when the demand charge is being met, it is likely that the ice storage system is only meeting a small portion of the load. This is unlikely to make up for the extra costs involved in the large storage system or the increased off-peak electric usage while the ice is being made. Since ice priority systems increase the off-peak electric consumption while decreasing the on-peak electric consumption, ice priority systems are the most ideal in a scenario where the off-peak electric utilization charge is less than the on-peak charge. A chiller priority system will limit the amount of ice created at inefficient temperatures by only making ice on days that exceed the chiller capacity load. For these reasons a chiller priority system was favored over an ice priority system.

6.4 Chiller Placement

Through literary review and recommendations made by CALMAC, it was determined that the most effective place for the chiller is upstream of the ice storage device. With the chiller upstream of the ice storage system, the temperature of the glycol solution entering the chiller will be warmer then it would be in a scenario where the chiller is downstream of the ice storage system. This will allow the chiller to run at a higher efficiency; however, this also means that the temperature of the solution entering the thermal storage tanks is already slightly cooled. Since the temperature of the ice cannot be changed, there is a decreased delta T in the ice storage tanks. This smaller temperature drop will mean that the ice storage discharge rate will be slower than it would be in a chiller down stream system. The chiller upstream system will naturally lend itself well to a chiller priority system. However, if it was feared that the ice storage charging or discharging rate was a potential problem, this component of the system may be modified.

6.5 Ice Storage Tank Type

The two considered types of ice storage tanks were internal freeze- internal melt and internal freeze- external melt. With an internal melt system, the ice gathered on the coils will begin to melt from the inside which will create a layer of water insulation between the glycol solution and the ice. Internal melt systems will have a greater range of discharge temperature and discharge rate than an external melt system. An external melt system has complications with introducing a new flow cycle of water. In an external melt system, a glycol system flows through the pipes to freeze the ice, but then water is run directly over the ice in the discharge cycle. Despite the more steady temperatures and increased discharge rate, the extra complications of adding in a new flow cycle are not beneficial. For this analysis only internal freeze-internal melt systems will be analyzed.

7.0 Mechanical Equipment Simulation

There is limited software available with quality ice storage simulation ability. To ensure that I was aware of the calculations I was performing, I only used data from the mechanical analysis software up until the point of normal chiller and electric analysis data. To extend that data into analyzing an ice storage system I created my own program that would do a complete hour by hour analysis of the ice storage system.

7.1 Chiller Load Simulation

The mechanical equipment simulation was performed using HAP and the Excel sheet made for designing the equipment. Initially, all of the data for the system was entered up to the system level. This data was then extracted and used to determine the hour by hour load on the building. By implementing a system for charging an ice storage system I was able to divert daytime chiller use to the nighttime. The system assumed a 98% thermal storage efficiency and adjusted load capacities of the chiller based upon whether or not ice was being created. From this I was able to determine the chiller load for each hour of the day, which led to the selection of the chiller. As shown in Appendix A, a 30% glycol solution sees a 97% reduction in chiller capacity. To convert this into energy usage I had to then multiply the chiller load by the proper kW/ton. This value for kW/ton took into account the leaving water temperature (changing depending on if ice was being formed) and the part load efficiency. For tonnage values within 5% of the total chiller capacity I used the design kW/ton. For other tonnage at a supply temperature of 44°F, I used the (Integrated Part Load Value) IPLV. I could not use the IPLV for the upper 5% because this would have a direct affect on my electric demand value. Since the rest of the values were primarily for simulation purposes I decided that the IPLV would be a reasonable estimate. I was not able to account for change in efficiencies due to outdoor air dry bulb temperature. Though this does have a significant affect on an air-cooled chillers load capacity and efficiency, I was not able to include it into my calculations. I consider this to be safe because it would aid the ice storage system which is using nighttime air over the base case which is drawing in more summertime hot air. Table 8.1 shows the kilowatts per ton used in each situation for the overall system design. Values that could be found in Appendix A were, but calculations were performed on values not directly found in Appendix A.

To determine the values in Table 8.1 not found in Appendix A, calculations and estimations were performed. It was assumed that the Carnot efficiency

(η_{carnot}) would remain constant in all conditions as long as the same chiller was being used. η_{carnot} is simply defined as the actual Coefficient Of Performance (COP) divided by the Carnot COP (COP_{carnot}). η_{carnot} is initially found by using some reference location of data, where the value for COP can trivially be derived from Appendix A. The values for COP_{carnot} are then calculated using Equation 7.1.

$$COP_{carnot} = T_{low}/(T_{high}-T_{low}) \qquad \text{Eqn. 7.1}$$

In the reference condition the COP_{carnot} is used to find η_{carnot}, in the charging condition, COP_{carnot} is used with η_{carnot} to find the actual COP. This actual COP was then used to estimate the energy usage of the chiller at those conditions. IPLV values were found on the cut sheets in Appendix A and factored by the ratios of peak kW/ton to obtain an estimated energy consumption of the chiller in all modes.

There was not Trane chiller data available for 85 ton chillers. As will be discussed in the next section, the 85 ton chiller was shown to be a dangerous choice and is only being done for educational reasons. Because this is being done for educational reasons, assuming the COP of the 85 ton chiller was equal to that of the shown 80 ton chiller was acceptable.

Table 7.1

Chiller Conditions 80 Tons

	Low T	High T	Tons	kW	COP	COPcarnot	η Carnot	kw/Ton	IPLV kw/Ton
Reference	499.7	544.7	78.8	75.6	3.7	11.1	0.3	1.1	
Charging	484.7	544.7	44.4	58.5	2.7	8.1	0.3	1.3	1.2
Discharging	509.7	554.7	79.8	85.4	3.3			1.1	1.0
As Designed	503.7	554.7						1.1	0.8

Chiller Conditions 90 Tons

	Low T	High T	Tons	kW	COP	COPcarnot	η Carnot	kw/Ton	IPLV kw/Ton
Reference	499.7	544.7	89.9	88.9	3.6	11.1	0.3	1.1	
Charging	484.7	544.7	60.0	81.5	2.6	8.1	0.3	1.4	1.3
Discharging	503.7	554.7	90.8	99.8	3.2			1.1	0.9
As Designed	509.7	554.7						1.1	0.8

Chiller Conditions 100 Tons

	Low T	High T	Tons	kW	COP	COPcarnot	η Carnot	kw/Ton	IPLV kw/Ton
Reference	499.7	544.7	99.9	101.7	3.5	11.1	0.3	1.1	
Charging	484.7	544.7	55.7	78.0	2.5	8.1	0.3	1.4	1.2
Discharging	509.7	554.7	100.6	113.6	3.1			1.1	1.0
As Designed	503.7	554.7						1.1	0.8

Chiller properties at different loading conditions

In addition to these values, special calculations were performed within the Excel program to account for changes in capacity and COP that are dependant on how the chiller responds when its set point is exceeded. Chiller priority ice storage systems only discharge ice when the chillers load is exceeded. The system knows when the chiller is exceeded because of an increase in temperature of the supply temperature. A valve can then be modulated to allow for some of the chiller water to be sent through the ice storage system to maintain a constant supply temperature. When the compressor's capacity is exceeded, both the motor amps and the capacity of the chiller increase as a result of the system attempting to achieve a supply temperature that it cannot achieve, and the supply temperature of the chiller increasing. At this point, instead of the chiller controlling the system, the chiller is being controlled by the system. It was estimated that for a screw chiller, COP increases of 3 %/°F can be expected because of the disproportionate high increase in capacity over the increase in amps after the chiller exceeds the designed supply temperature. The temperature exiting the chiller was estimated in the program by doing a direct interpolation between the maximum chiller temperature (calculation shown in Section 8) and the supply temperature in relation to the respective chiller load. As a best estimate for the new capacity, the designed capacity of the chiller was then increased by a factor half as much as the COP was raised.

7.2 Ice Storage System Simulation

The ice storage simulation began with the building thermal load data from the HAP file. From this a "charging potential" was determined for each hour. This potential was found by determining if the chiller was either in a potential charging mode or in a potential discharging mode by comparing the building loads with the respective charging and non-charging chiller capacities. Negative values would indicate that the chiller had the potential for charging. These values were then broken up into either positive charging or discharging. For each hour, it was determined if charging or discharging was possible. This was a result of the load and the previously charged amount of the ice storage system. If the ice storage system could discharge, then the load on the chiller that exceeded the chiller capacity was subtracted from the capacity of the ice storage system. Once the load of the building drops to be lower then the tonnage capacity of the chiller reduced for ice making capacity, the chiller switches onto ice making mode and the ice storage system begins to charge. A check in the system was put in to ensure that the ice storage system was not dropping bellow zero ton-hrs.

7.3 Building Electric Load Simulation

A further HAP analysis was done on the building to combine the cooling system with the heating system in a complete building analysis. This data was used to create an output of the building's total electric consumption. From this data and in corroboration with the Duke Energy electric rate structure (shown in Appendix C), the annual electric bill for the base case building conditions was determined. To find the electric consumption for any given hour of the ice storage building I subtracted the chiller kW that I calculated for the base case from the total building kW, and then added the ice storage system chiller kW. Special attention had to be paid to the kW consumption of the chiller. Each hour of the year was analyzed to determine which kW/ton category from Table 7.1 it fell into. This was done for each hour of the year so that a monthly electric bill could be developed. Because of a requirement that the minimum demand charge for a month is no less then 85% of the highest demand charge in the summer months, there is a variation in required demand and billed demand. The month by month breakups of these demands are shown in Figures 7.1 & 7.2.

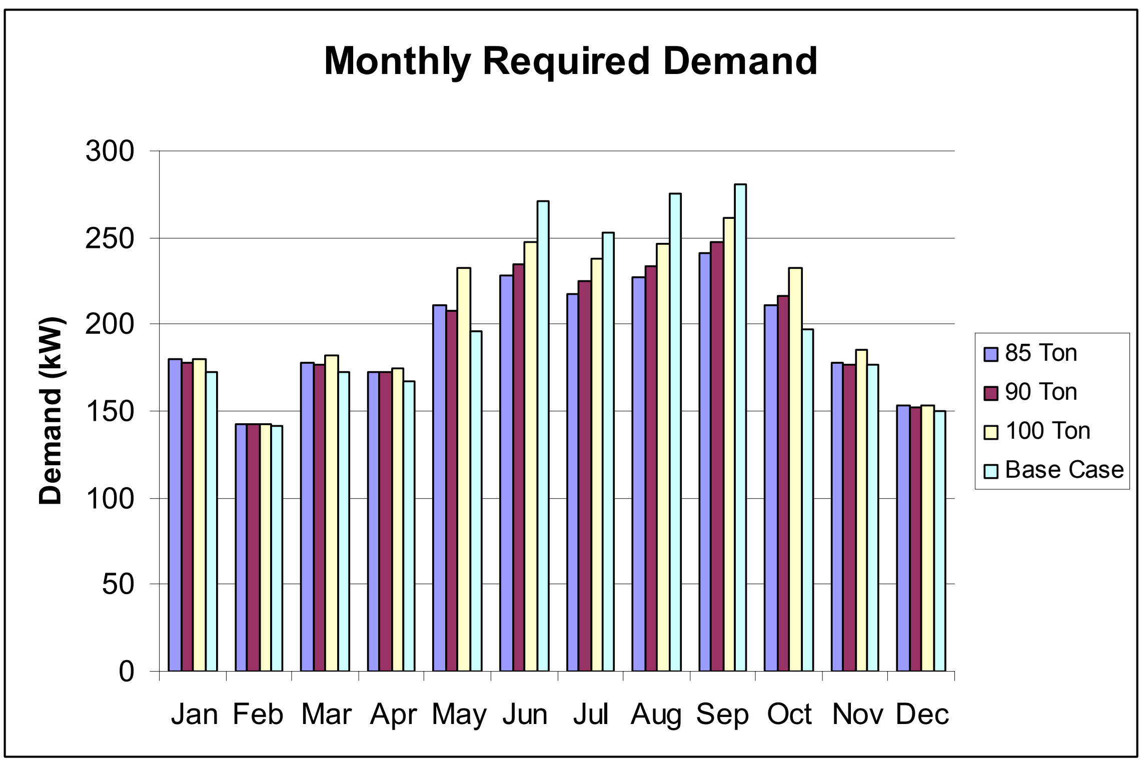

Figure 7.1
The Monthly Required Demand for Each Scenario

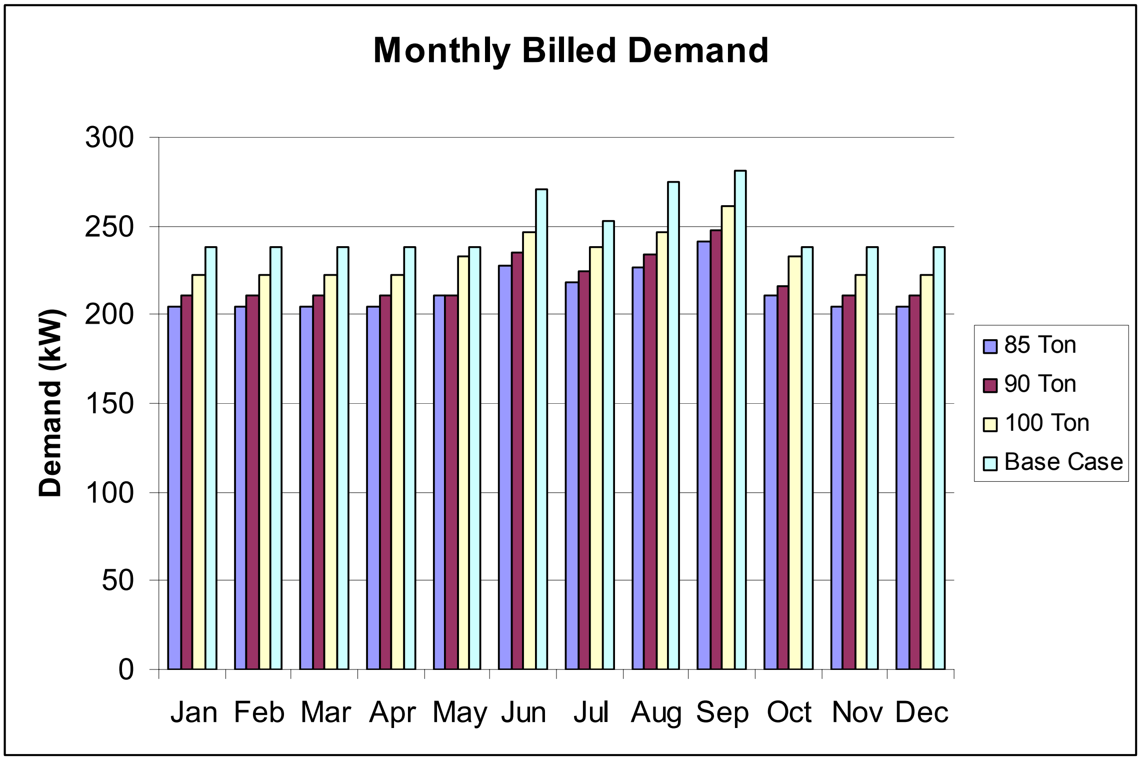

Figure 7.2
The Monthly Billed Demand for Each Scenario

The minimum demand charge requirement lends a strong advantage to an ice storage system. In Figure 7.1 during the winter months of the year, the ice storage system scenarios show an increased demand need. However, for those same months in Figure 7.2, the billed demand for the ice storage system scenarios is lower then in the base case.

Table 7.2 gives a summary of the estimated annual electric bill. It shows an inverse relationship between electric consumption and chiller size. This is because the smaller chillers tend to have a lower COP and because these systems rely more heavily on the ice storage system. As Table 7.1 displayed, each chiller has a drastically decreased COP in the ice making stage as a result of a low supply temperature. Therefore, for each ton of cooling done using the ice storage system there is a greater amount of energy required than there would be had the chiller directly cooled the space. Despite the increase in kW-hr, each of the ice storage systems showed an annual savings in the electric bill due to a decreased demand charge.

Table 7.2

Anual kW-h			
85	90	100	Base Case
535795	530902	539144	513111

Demand Peak kW			
85	90	100	Base Case
241	248	261	281

Annual Electric Bill				
85	90	100	Base Case	
35507	36162	37911	39486	Annual Bill ($)
10%	8%	4%	0%	% Annual Savings
0.066	0.068	0.070	0.077	$/kW

Annual Electric Bill Summary

8.0 Mechanical Equipment Selection

8.1 Chiller and Ice Storage Tank

According to the CALMAC representative, a chiller under an ice storage system is expected to be 55% smaller than in an equal system without an ice storage system. This brings a rough estimate of the size of the chiller for the Hays School to go from 170 tons to around a 100 ton chiller. The mechanical drawings called for either a screw or a scroll chiller. According to the ASHRAE Design Guide for Cool Thermal Storage, "Reciprocating and rotary screw chillers are adaptable to a wide range of leaving temperatures and can generally be applied to ice storage systems with little difficulty." A centrifugal chiller could also be applied to the system but there are further complications involving the specifics of the operating conditions and the compression ratio. To help keep parallelism between the base condition and the proposed idea, a screw chiller was decided upon.

The condenser for the base case was designed to be air-cooled. On the CALMAC website there is an article showing an elementary school with a load of 190 tons where ice storage was implemented. In this case an air-cooled chiller was also used. It was perceived early on that air-cooled would be the most economical for a small chiller and to keep similarities with the base case, an air-cooled system was chosen.

To determine the exact sizes of the equipment, the hourly excel program was used. The program used the hour-by-hour analysis of the building, user defined ice storage system information, and user defined chiller information to determine the minimum amount of ton-hr of capacity left in the system for an entire year.

The first goal was to establish the flow rate required by the chillers. To maintain the same delta T called for in the base case, the ice storage system needs to drop a 58°F return solution to a 43°F supply. The specific heat of this solution is 0.89 btu/(lb-°F). By comparing these with a design day of 158 tons, Equation 8.1 can be used to solve for the required mass flow rate of 146000 lb/hr.

$$\dot{Q}=\dot{m}c_p\Delta T. \qquad \text{Eqn. 8.1}$$

The specific gravity of the solution is 1.057, giving it a density of 8.77 lb/gal, and a total required flow rate of 277 gpm.

8.1.1 Small 85 Ton Chiller Scenario

The 85 ton chiller was based of a pseudo extended charging cycle as opposed to a daily charging cycle as the other chillers are. This is because the small 85 ton chiller does not have the capabilities of charging enough ice in a single night to overcome the design day. For this reason, the 85 ton chiller scenario will not be selected, but is still being analyzed for educational reasons. Figure 8.1 shows this extended cycle over one entire week, beginning with Monday morning. For these days, the ice storage system is not able to recharge each night. However, over the course of one week, the system does recharge itself to maximum capacity. Another aspect of the program is revealed by the level portions of the ice storage system on the weekends. This is a display of a safety in the program to ensure that the chiller does not attempt to make ice during the daytime hours. By implementing this, the system will only charge ice during the nighttime hours when the outdoor air dry bulb temperature is the coolest, maximizing the chiller efficiency.

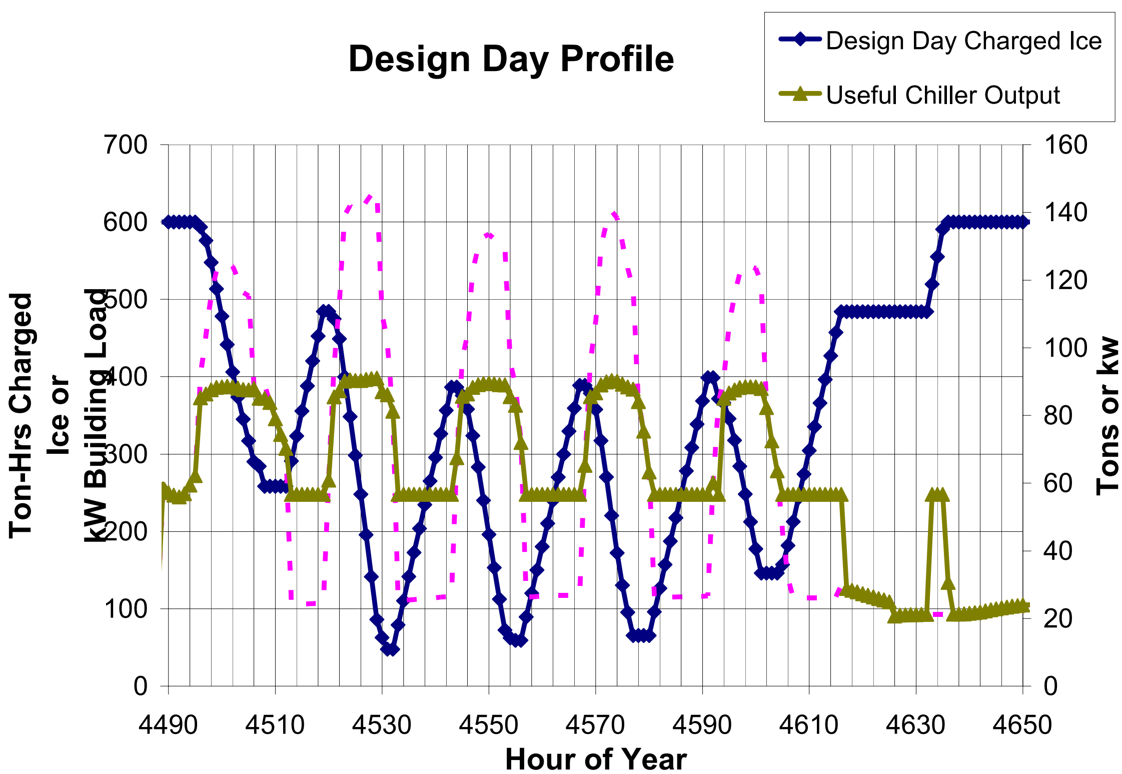

Figure 8.1
90 Ton Chiller Design Week Chiller, Charged Ice, and Thermal Load Profiles

8.1.2 Medium 90 Ton Chiller Scenario

The 90 ton chiller had a total of 1,910 ton-hrs of required cooling on the design day and works with a normal daily charging cycle. The analysis in the program resulted in an ice storage system with a useable capacity of 486 ton-hrs. This demand could be met with three CALMAC 190A, 162 ton-hr ice storage tanks.

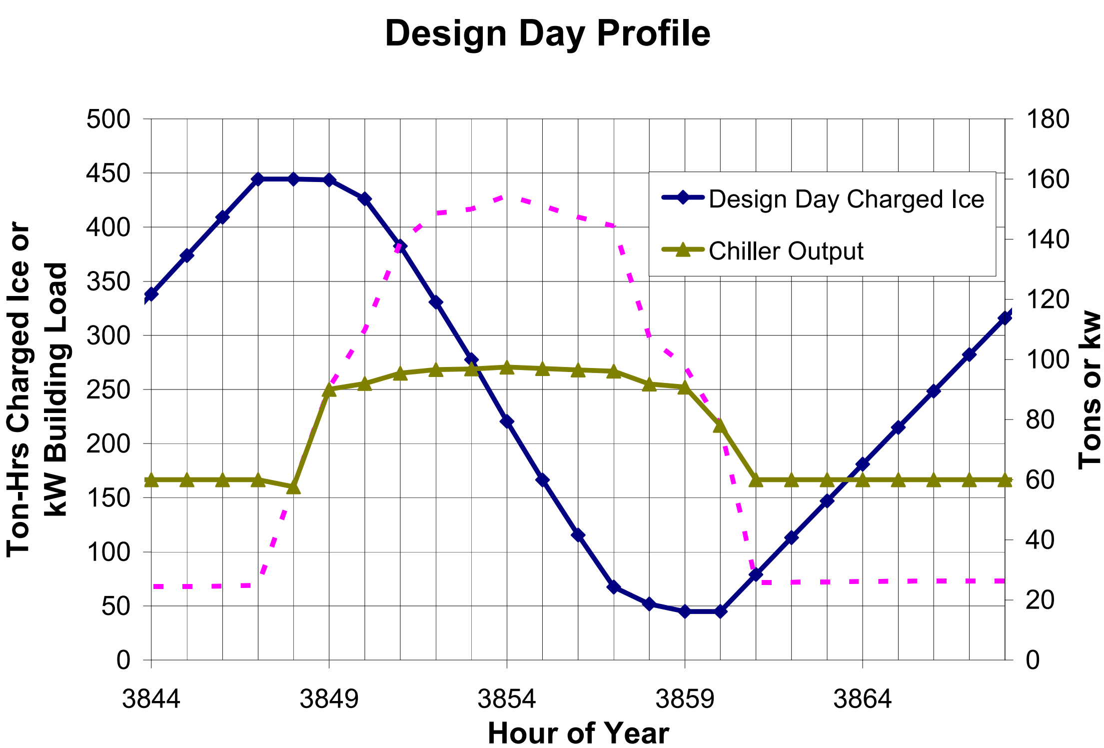

Figure 8.2
90 Ton Chiller Design Day Chiller, Charged Ice, and Thermal Load Profiles

Figure 8.2 shows how this system responds on the design day. The ice storage capacity in ton-hrs is shown to charge while the building load is smaller then the building capacity and discharge during the hours that the cooling load exceeds the capacity of the chiller. It is also shown that the chiller capacity varies depending upon whether ice is being formed or discharged, and by how much the chiller's nominal capacity is being exceeded.

On an off-design day, a similar effect is seen but to a lesser extent. Figure 8.3 shows how the system responds on a non-design summer day with a considerable amount of required cooling (140 out of 158 tons). This figure displays how the chiller output follows the building load, until the chiller capacity is exceeded. At this point, the ice storage system begins to discharge to bring the solution to supply temperature. Another interesting component of this figure is how the ice storage system still has an available 225 ton-hrs of cooling available despite this day still being a reasonably warm day.

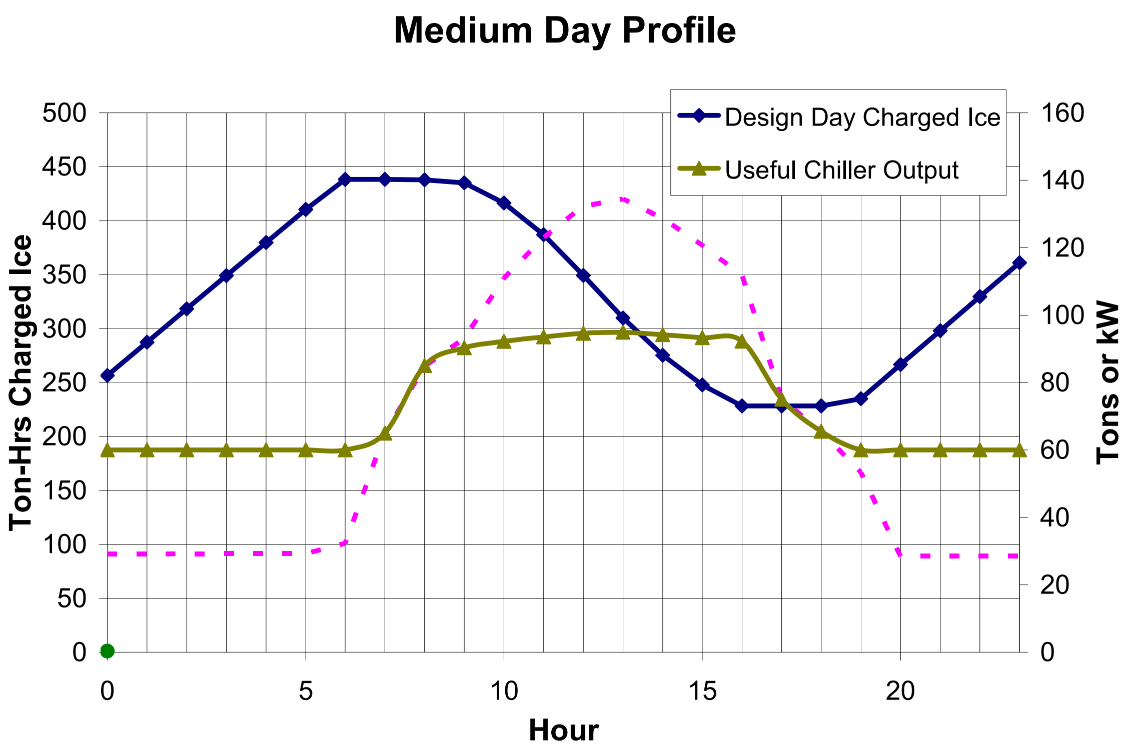

Figure 8.3
90 Ton Chiller Medium Day Chiller, Charged Ice, and Thermal Load Profiles

Finally, Figure 8.4 shows how the building responds on a spring day when the building load is only around 55 tons. ARI IPLV ratings for chillers presume the buildings load seen by the chiller system 50% of the design for 57% of the time that the chiller system is running. This implies that Figure 8.4 is typical of a large portion of the days that the chiller system is running. On these days the chiller and the load profile are perfectly in sync. This means that the ice storage system is not used. The benefit of this is that the electrical demand is limited on the peak days, but the decreased chiller efficiency from making ice is not a disadvantage on most days. This profile is perfectly representative of every day in which the building load does not exceed 90 tons. This figure also portrays another advantage of an ice storage system. The total ton hours of cooling required on this day is 617. The ice storage system has the ability to do 486 ton-hrs of cooling. This presents a redundancy in cooling that is not available with typical one chiller systems. This means that if there is a chiller malfunction that the ice storage system can take care of the entire building load for more then half of the day. This could be very important for a school that may have plays or sporting events in the evening. With the base case system, if there is a chiller failure before the event, then there are no provisions to ensure that cooling can be done. This can result in canceled sporting events or performances. With the ice storage system, if there is an unexpected chiller failure before an event, then the ice storage system will be able to provide several hours of cooling without the chiller.

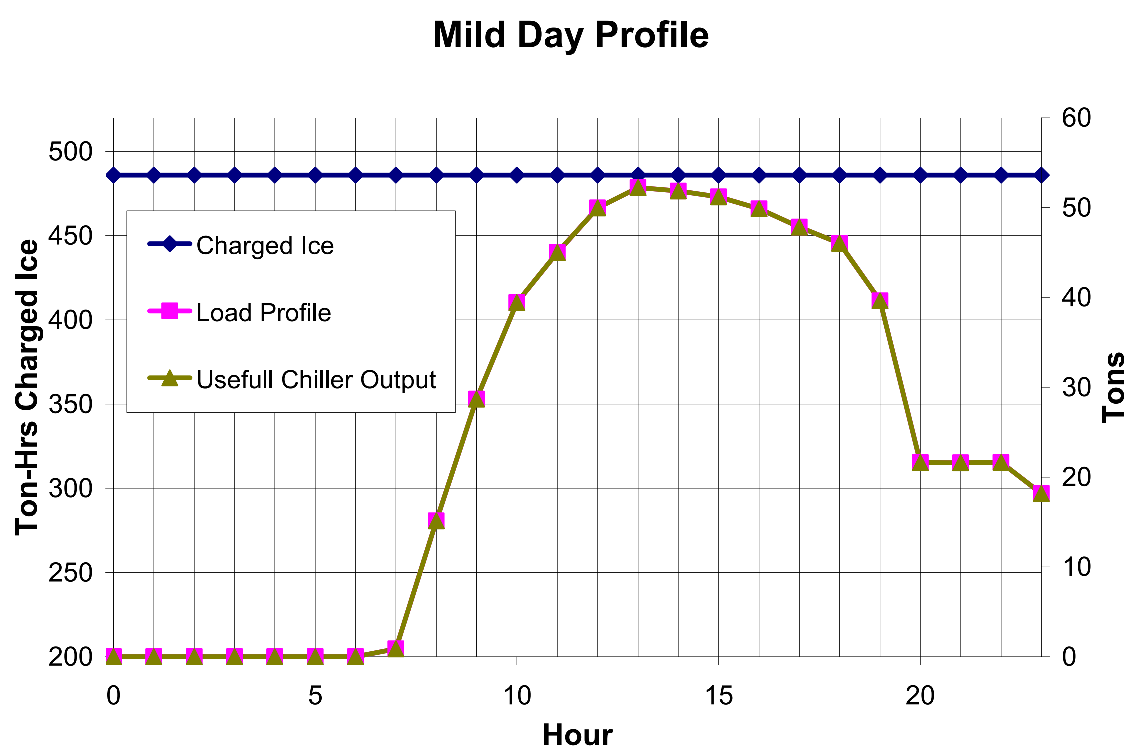

Figure 8.4
90 Ton Chiller Average Day Chiller, Charged Ice, and Thermal Load Profiles

Figure 8.5 shows how this system helps to decrease the electrical demand charge and why the overall electrical usage increases. During the peak portion of the day, the ice storage building uses less electricity; however, during the night, from the poor COP's, the electrical consumption for the ice storage building goes up. It is obvious that the overall effect is a decreased demand for the ice storage system with an increased overall daily electrical usage.

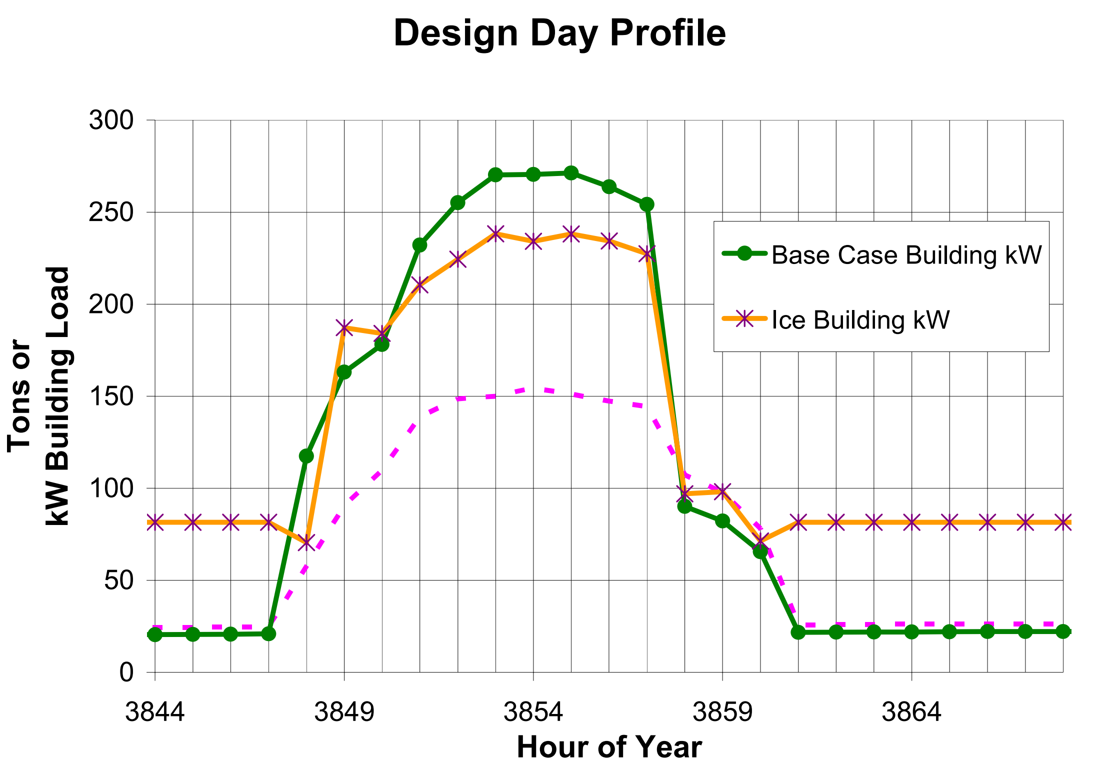

Figure 8.5
Design Day Electrical Comparison – 90 Ton Chiller and Base Case

Appendix B outlines specific information about the selected ice storage tanks. For charging and discharging rates the CALMAC Model 1190 was used. This is the only model that CALMAC released charge and discharge information for. Since the 90-ton chiller scenario is the only scenario that exclusively uses this ice storage tank, the charging and discharging analysis will not be performed for the other scenarios. As shown in Figure 8.2, the ice storage system must be capable of charging at a rate of 36 tons/hr. Distributed evenly among the three tanks in parallel, this results in a requirement of 12 tons/hr/tank. As shown in Figure 8.6, an average charging brine temperature of 25°F suggests a minimum flow of around 50 GPM/tank. In four tanks, this would total 200 GPM, significantly less then the 277 GPM that the proposal calls for. Though there is no specific data about the discharge rates, the CALMAC representative did agree that the system as setup is within the capabilities of the ice storage system.

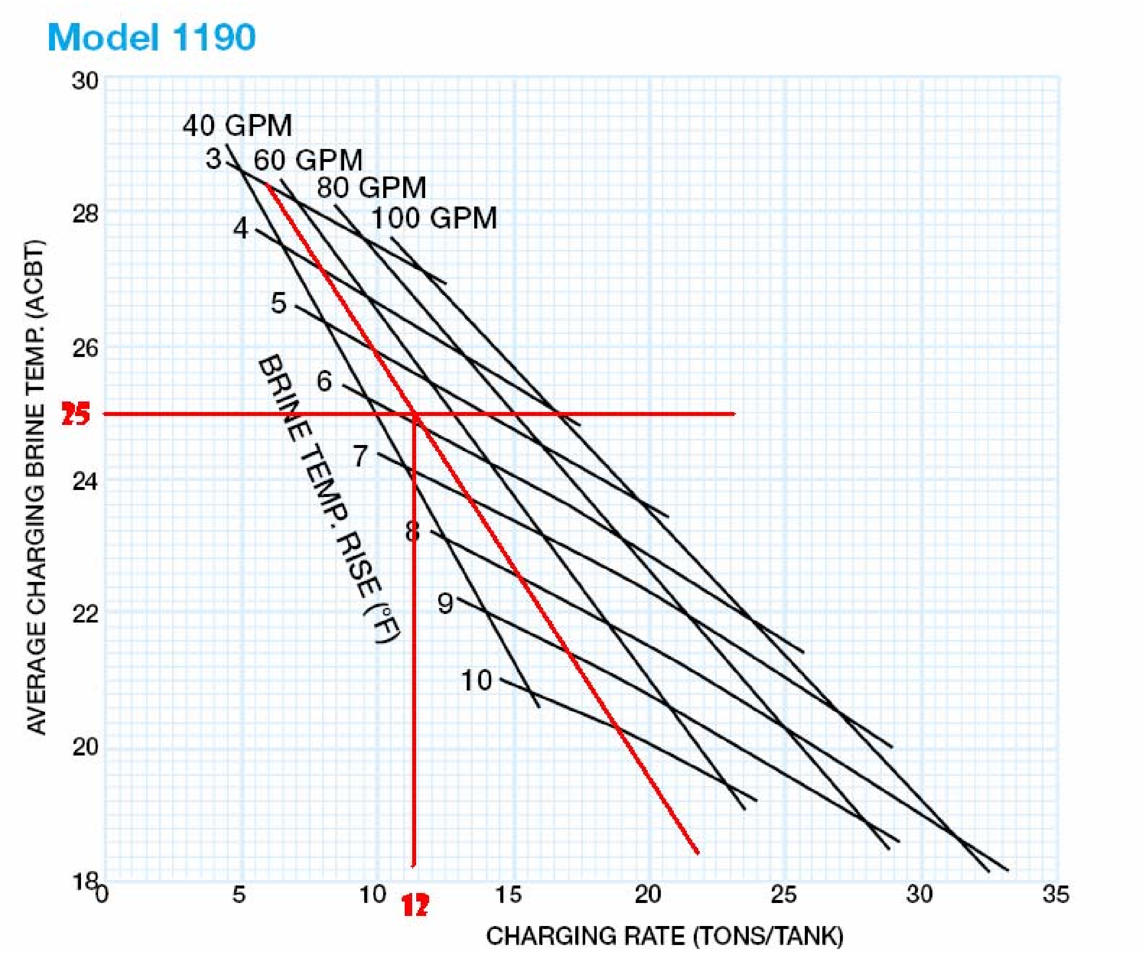

Figure 8.6
Charging Rates for CALMAC Ice Storage Tank

8.1.3 Large 100 Ton Chiller Scenario

The large 100 ton chiller system runs very similar to the 90 ton system, as shown in Figure 8.7. The larger chiller will naturally result in an increased demand charge, as already stated in Table 7.2. This is counterbalanced by lower electricity consumption then the 90 ton case. The 100 ton system also only requires a 358 ton-hr ice storage system which will result in a lower first cost.

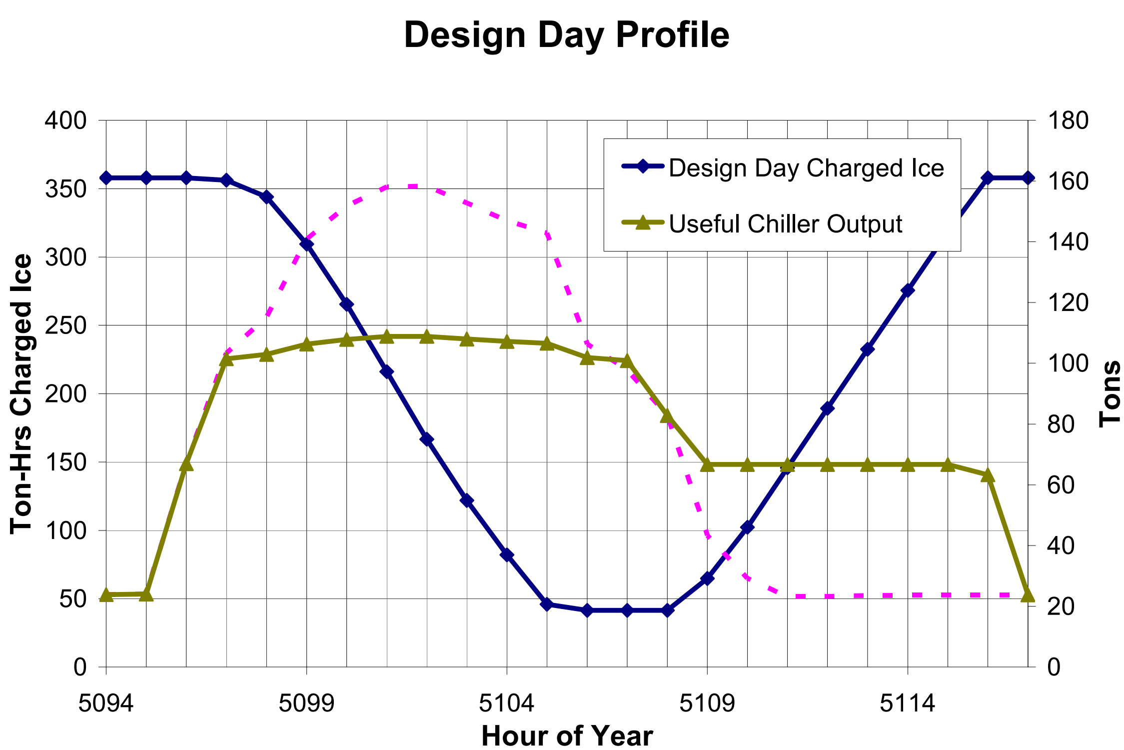

Figure 8.7
100 Ton Chiller Design Day Chiller, Charged Ice, and Thermal Load Profiles

8.2 Piping and Cooling Coils

The original drawings called for a coil Entering Water Temperature (EWT) of 43°F and a Leaving Water Temperature (LWT) of 58°F. This high delta T limits the possibilities of an ice storage system saving money on piping. CALMAC recommends a constant water supply temperature of 43°F and a returning temperature of 60°F, or a 17°F delta T. The flow rate of the base case system called for 300 gpm, the proposed redesign system was designed for 277 gpm. Typically there would be a greater reduction in flow with an ice storage system. The low temperature ability of the chiller presents an opportunity to obtain a large delta T across the cooling coils. However, in this scenario both the base case and the proposed case have large delta T's so there is only a small reduction in supply flow. Most of the advantages to this small supply flow were offset by the higher viscosity of the 30% glycol solution. This means that the piping and cooling coil will be close to the same size as called out in the base case drawings. The main reduction in size between the piping and coils came from a reduction of 43' of main 6" piping to 5". This conclusion was drawn by analyzing the pipe loss equation.

$$\Delta P=\lambda*(1/d_n)*(\rho V^2/2) \qquad \text{Eqn. 8.1}$$

From this equation the only direct reference to fluid properties is the density. The Specific Gravity (SG) can be used as a multiplier to the equation, yielding the new equation,

$$\Delta P=\lambda*(1/d_n)*(\rho V^2/2)*SG \qquad \text{Eqn. 8.2}$$

The λ term, D'Arcy-Weisbach friction coefficient, also includes data referring to the specific fluid properties.

$$1 / \lambda^{1/2} = -2 \log ((2.51 / (Re\ \lambda^{1/2})) + ((k / d_h) / 3.72)) \qquad \text{Eqn. 8.3}$$

The Reynolds Number, Re, in this equation is dependant directly on fluid properties. Since Re is a function of the ratio of density over viscosity, the Reynolds Number may be adjusted by multiplying it by a factor of the specific gravity (1. 057) over the ratio of viscosities, (34.03/31.5 = 1.080) to equal a Re factored by 0.98. Because of the log relationship, and because both scenarios are well within the turbulent, more level portion of the relationship, the 0.98 multiplier on the Reynolds Number will not significantly affect the value of the D'Arcy-Weisbach friction coefficient.

The specific gravity of a 30% glycol solution is 1.057. To determine if a pipe could be downsized, the original pressure drop in the pipe was found. The pipe for the new glycol solution flow of 277 gpm was then sized for the original pressure drop in the system divided by the specific gravity. In all cases the piping came out to require the same nominal pipe size with the exception of the 43' of main piping that was downsized from 6" to 5".

However, more piping will be required to connect the ice storage tanks to the chiller. Including the bypass around the ice storage tanks, an additional 70 ft of 5" copper pipe is required.

8.3 Refrigerant and Mixing Equipment

To prevent freezing in the chiller system a 30% glycol solution was chosen. This is in part due to the recommendations of the CALMAC representative. This decision was checked with the freezing point of the solution to make sure that the selection was correct. As shown in Table 8.1, a 30% glycol solution will freeze at 2°F, well below the expected temperatures of 25°F. The volumes of the piping in the building were summed up to be almost 1500 gallons and the volume of the four tanks combined is 400 gallons. This combines for a total requirement of 1900 gallons of the 30% glycol solution. In addition to a different refrigerant, the ice storage system requires a lifting bar, a makeup system, and an inventory meter required for maintaining and instigating the glycol solution. The system also requires an annual monitoring of the system and a semiannual addition of biocide.

Table 8.1

Freezing Point								
Ethylene Glycol Solution (% by volume)		0	10	20	30	40	50	60
Temperature	(°F)	32	23	14	2	-13	-36	-70
	(°C)	0	-3	-8	-16	-25	-37	-55

Freezing temperatures for Ethylene Glycol Solutions

The glycol solution will limit the capacity of a chiller with respect to a water system. According data from Appendix A, these chillers will have a capacity reduction of 97%. For this reason it is important to note that all of the required chiller capacities shown are assumed to be after the 97% reduction. This will be taken into account in the cost analysis, because this will result in a slightly larger chiller. This is not expected to have a significant effect on the energy consumption of the chiller, according to Appendix A the energy data is in the vicinity of 1%.

8.4 System Design

To determine the required change in temperature across the chiller during peak demand, a quick analysis on the system must be done. At peak load, a 277 GPM of a water solution with a density of 8.3 lb/gal would have a calculated flow rate of 2299 lb/min, or 137,946 lb/hr. To translate this value into the glycol solution, it must be multiplied by the specific gravity of a 30%

glycol solution, 1.057, giving a value of 145,809 lb/hr of the glycol solution. To interpret this value as a thermal load Equation 8.4 was used,

$$\dot{Q}=\dot{m}c_p\Delta T. \qquad \text{Eqn. 8.4}$$

Using the flow rate calculated, a c_p of 0.89, and the maximum discharge of the ice storage system equal to 60 tons, a ΔT of 5.5°F was found. By adding this value to the desired supply temperature of 43°F, a maximum supply temperature of 48.5 is required from the chiller. The ice storage system is then responsible for cooling the water to the supply temperature of 43°F. This temperature was the temperature used in determining the COP and capacity of the increase of the chiller.

The system is setup in a manner typical of CALMAC's recommendations. In the charging stage, a 25°F chiller discharge temperature at 277 gpm is required. Figure 8.9 shows the expected charging cycle of the ice storage system. If the building does have a thermal load while charging is being done, Valve V2, will open to supply a mixed 44°F to the air handlers.

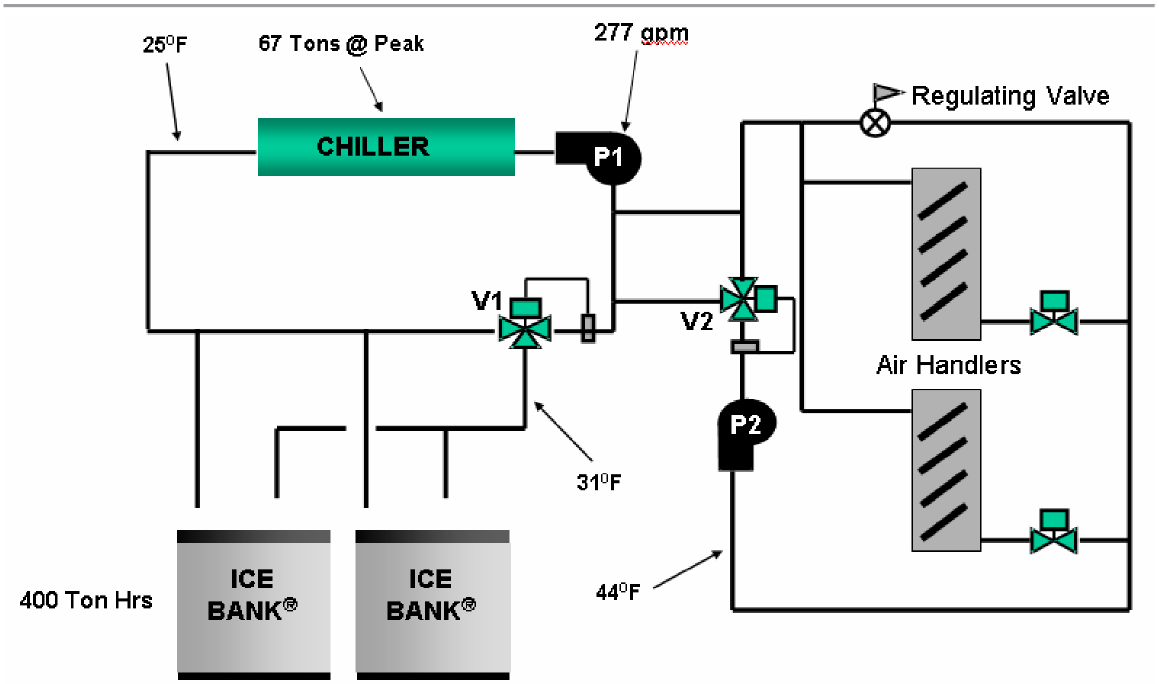

Figure 8.9
Ice Storage System Charging Cycle

Figure 8.10 shows the conditions of the system during peak unloading. According to the CALMAC representative, the ice storage system discharges the solution at temperatures ranging from 28°F to 34°F. Valve V1 adjusts to ensure that the mixed temperature between the 49°F solution leaving the chiller and the solution leaving the ice storage tanks maintains a 43°F supply temperature. This solution is then sent to valve V2 which adjusts to maintain the desired flow through pump P2 and maintain a final supply temperature of 44°F. The solution is then expected to return to the chiller at 58°F where it is cooled to 25°F and repeats the cycle. Because this is a chiller priority system, the chiller will always attempt to handle the entire load of the building. Since the building load is larger then the capacity of the chiller, the chiller will not maintain the designed supply temperature of 43°F. As mentioned, this will result in an increased chiller capacity. The warmer solution will then be sensed by Valve V1 which will modulate to send a portion of the water through the ice storage tanks. The 49°F solution shown in Figure 8.10 is the maximum water temperature that will leave the chiller. This corresponds to hour 3,854 on Figure 8.2.

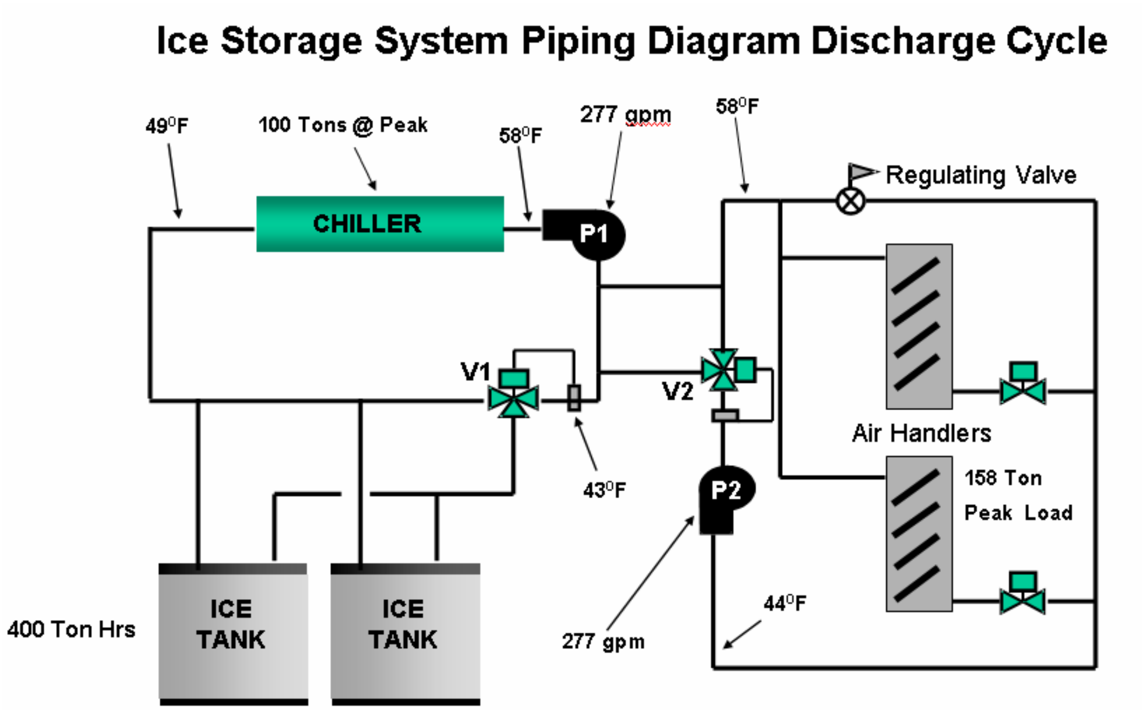

Figure 8.10
Ice Storage System Peak Discharging Cycle

The systems with the ice storage tank should not require much more maintenance then a typical chiller system.

9.0 Electrical Equipment Selection

The buildings Main Distribution Panel (MDP) was designed for a connected load of 1014.92 kVA and 1223 A. After demand factors and a 15% spare capacity for expansion the demand load was 1087.05 kVA and 1310 A. The final engineer's selection was for a three pull, four wire 480Y/277V 2000 A MDP. The chiller had two panels of equal size responsible for a load of 333 Minimum Current Ampacity (MCA). The wires to the chiller were originally designed to be three #350 and one #1 ground requiring 2-1/2" conduit, resulting in a voltage drop of 1.09 volts or 0.39 % from the MDP.

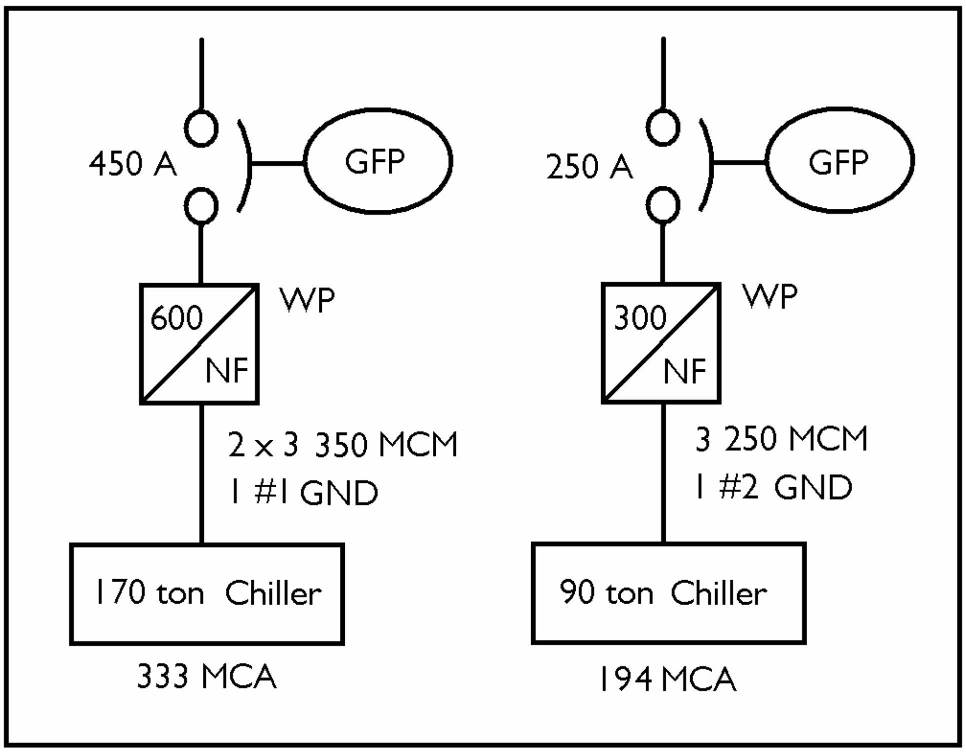

Figure 9.1
Designed and proposed electrical equipment

The electrical data for the chiller, shown in Appendix A, shows the MCA for the wire to be 194. The manufacture decided this value by adding 125% of the largest compressor plus 100% of the second compressor and the sum of all of the condenser fans. Because this load is going to motors, it was multiplied by 125% for selecting a wire size. 250 MCM wire rated at 255 A was used. The voltage drop table in Appendix D showed that in magnetic conduit, a 3/0 AWG wire sees 0.054 voltage drop for every 1000 ampere-feet. With a total wire length of 83' a 0.86 change in voltage was calculated or 0.5%. The ground fault protection device was designed to 80% smaller then the wire capacity which was 204 A resulting in a 250 A breaker, which is smaller in capacity then the wiring and larger then the expected amperage. The non-fused switch was reduced from 600 A to 300 A. According the chart in Appendix D outlying the ground wire size, a 250 A wire requires a #2 AWG ground and a 250 amp MOP requires a # 4 AWG ground, a #2 AWG was chosen. The conduit for the system was sized at 2-1/2" according the NEC table in Appendix D. This reduction in electrical equipment should bring a reduction in the upfront cost of the proposed ice storage design.

The other two scenarios were similarly analyzed and the results are shown in Table 10.1.

Table 9.1

Chiller	MCA	MOP	Time Delay	Qty	wire gauge	ground	conduit
80	164	200	225	1	4/0	#2	2"
90	194	250	250	1	250	#2	2-1/2"
100	218	250	300	1	300	#2	2-1/2"
170	333	450		2	#350	#1	2-1/2"

Electrical System Downgrade Summary

10.0 Ice Storage Tanks Placement

In order to implement the ice storage system a separate concrete slab on grade will be required. The ice storage system requires the implementation of two 89" diameter tanks and two 74" diameter tanks. The largest tanks are 16,765 lb (P). The soil was stated on the drawing documents to have an allowable bearing, q_a, of 2000 psf. To determine the minimum area needed to support the tank on the soil, Equation 10.1 is solved for the footing area, A_{FTG}.

$$q_a \geq P/A_{FTG} \qquad \text{Eqn. 10.1}$$

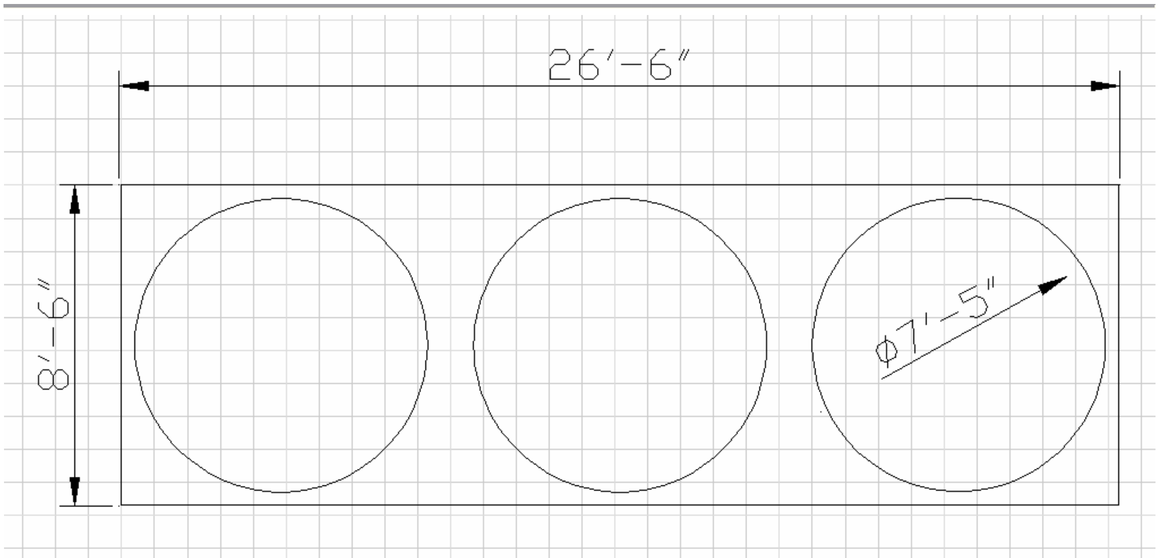

Figure 10.1
Ice Storage System Layout

The minimum area is solved to be 8.38 ft^2, significantly smaller then the area of a single tank, 43 ft^2. It is therefore safe to design the area of the slab based upon tank size and maintenance space. Figure 10.1 shows the proposed layout of the four tanks. Each tank is a minimum of 4 inches from the edges and 14 inches from other tanks. This will allow enough space for any possible cleaning that must be done. Other then cleaning, the tanks are not expected to have any other maintenance issues that would require workers to enter between the tanks. The total slab width is 318" by 102", yielding a slab area of 225 ft^2. To determine the amount of steel reinforcement required, a quick pressure analysis was done. Each tank is designed to sit on a 6" flange that surrounds the tank. On the larger tank, this flange has an area of 1,564 in^2.

With a weight of 16,765 lb distributed across the flange, the pressure on the concrete is equal to 10.7 psi.

This pressure is small enough to ensure that reinforcement will only be necessary to prevent thermal breakage. Building codes require that in order to prevent thermal breakage, a minimum of 0.0018% of the face of the slab must be reinforcement. Reinforcement bars must also be a minimum of 3" from the bottom of the slab and 2.5" from the top of the slab to prevent cracking. After including the probable width of the reinforcement, the slab is found to be a minimum of 6" deep. This defines the concrete slab to be a total of 4.17 C.Y. The face that is 318" has a minimum required reinforcement area of 3.4344 in^2. With 12" spacing and ensuring that the reinforcement is not within 3 inches of any side, there needs to be 26 bars (this also allows for a minimum of 3" between a bar and the edge of the concrete). Dividing the total area by the number of bars gives that each bar must be at least 0.132 in^2. The smallest bar that meets this requirement is a #4 bar. A #4 bar is 0.668 lb/ft, bars running perpendicular to the 318" face have a total summed length of 96" long times 26 bars, equaling 2,496" (allowing 3" at either end). The bars running perpendicular to the 102" face must have a minimum area of 1.1016 in^2. Eight bars at 12" requires a bar area of 0.13377 in^2. This again requires a #4 bar. The total length of the bars running perpendicular to the 102" face is 312" long times 8 bars, equaling 2,496". The total length of the rebar is 416 ft which is equal to 0.139 tons.

The 100 ton chiller system required three ice storage tanks of an equal diameter to those called out in the 90 ton system. Since none of these tanks are heavier then for the 90 ton system, the concrete slab will be the exact same size for the 100 ton system as it was in the 90 ton system.

The 85 ton chiller system requires six ice storage tanks. Two have a diameter of 89" and four have a diameter of 74". Like the 90 ton system, each tank is designed to sit on a 6" flange that surrounds the tank. On the heaviest, 74" tank, this flange has an area of 1281 in^2. With a weight of 10,760 lb distributed across the flange, the pressure on the concrete is equal to 8.4 psi. This pressure is small enough to ensure that reinforcement will only be necessary to prevent thermal breakage.

The total dimensions on the slab are 204" by 273" and 6" deep. The bars running perpendicular to the 204" side require 2.2032 in^2 and 16 bars meaning that again, this side uses 4272" of #4 bars. The bars perpendicular to the 273 side need 2.9484 in^2 of reinforcement which again is 22 #4 bars totaling 4356". The total 719' of #4 reinforcement bars weighs 0.24 tons.

Figure 10.2 shows the proposed location of the ice storage tanks. There is plain grass space behind the building and away from any areas such as playgrounds and basketball courts. This space also shares a wall with the chiller room. The only restraint on this location is the possibility of adding a 6" curb along one edge of the slab to account for a 6' rise in the grade. However, because of the small size of the slab, this portion could be easily excavated to be a uniform height. The tanks are not complicated pieces of equipment and do not involve any extra consideration concerning a level, dry, or exposed surface. All of the critical maintenance and hookup locations are at the top of the 8'-5" tanks.

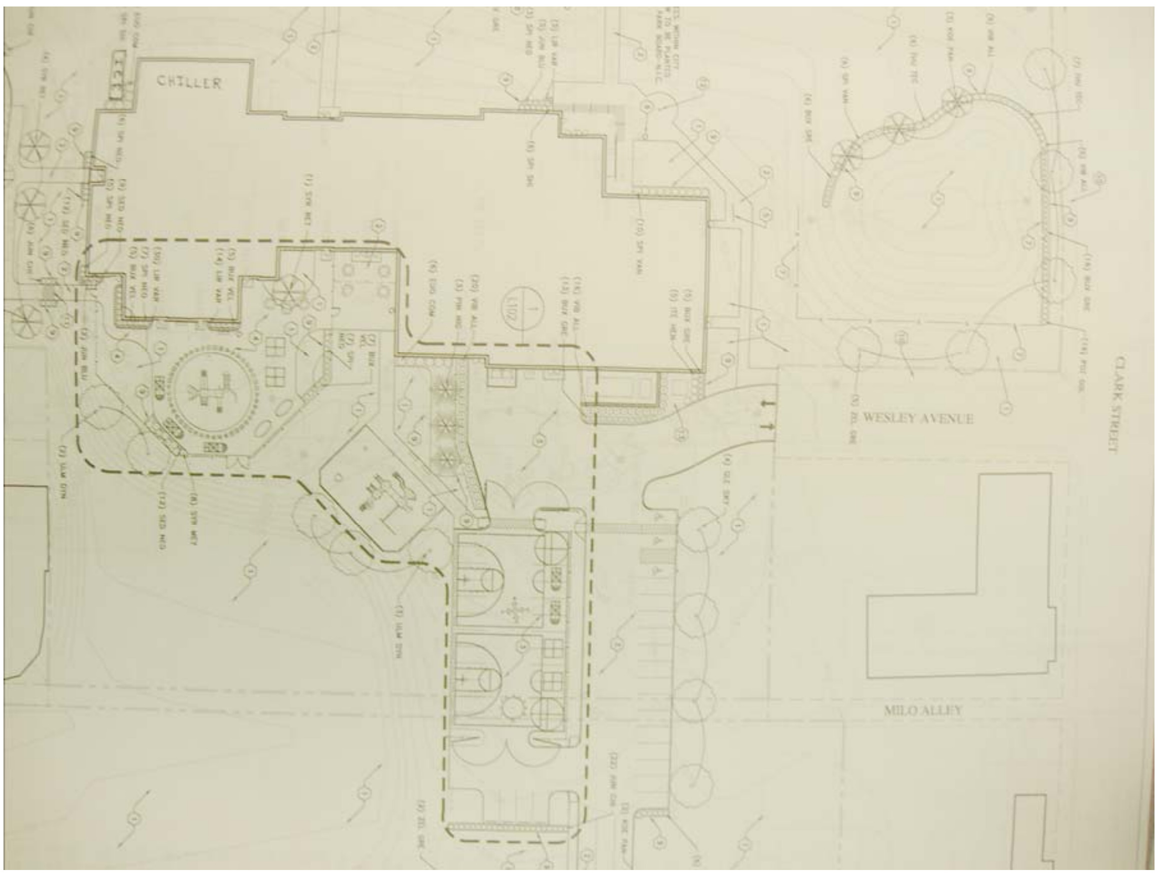

Figure 10.2
Site Plan

11.0 Cost Analysis

The major cost reductions of the proposed chiller systems was a result of electrical demand savings, pipe size reductions, electrical equipment reductions and chiller size reductions. These were balanced against extra costs involving ice storage tanks, electrical consumption increases, and a glycol solution and the neccisary equipment for managing the system. Each of the three scenarios simulated resulted in a payback period less then 10 years resulting in a favorable lifecycle cost.

Values for the overall cost analysis were found from a mixture of R.S. Means and manufacturer price estimates with an estimated instillation charge. The final cost summary is shown in Appendix E, a summary of these results are shown in Table 11.1.

Table 11.1

	Case		
	100 ton	90 ton	85 ton
Extra First Cost	7876	8633	25046
Annual Savings	1575	3324	3979
i	0.060	0.060	0.060
n	6.12	2.91	8.14
PV	7876	8633	25046

Total Cost Comparison and Summary

As shown, the case involving a 90 ton chiller and 486 ton-hrs of ice storage tanks is the most cost effective scenario with a payback of 2.91 years. As the summary shows, the annual electrical savings are not very significant, only around $3,000. However, the extra first costs of the system are also very small at only around $8,633. Despite the quick three year payback period, because of the low order of magnitude in money, it is reasonable to state that the 90 ton case is approximately the same cost as the original chiller system and that it does not result in a significantly reduced energy bill, nor does it result in a significant increase in first cost.

The first cost in the 85 ton chiller scenario is much larger then the other two because of the extended charging cycle that requires much larger ice tanks. This was predicted in Table 6.1 as to not bringing a favorable life cycle cost. Though it does bring an increased annual savings, this savings is not large enough to justify the increase in first cost and storage tank size.

The 100 ton chiller cost was similar to that of the 90 ton, but to a lesser degree. The annual savings were half as much as in the 90 ton case, but the extra first cost was $1000 less resulting in a six year payback. This analysis shows that there are no direct economic advantages of the 100 ton chiller over the 90 ton chiller.

12.0 Final Discussion

The 85 ton chiller scenario will require the school to sacrifice a significant portion of space with very little payback. As expected, the economics of an extended charging cycle do not turn out to be favorable in this scenario.

The 90 ton and 100 ton chiller scenarios both resulted in a small annual savings and a small first cost. Since these values are small, the decision as to whether these are proposals that should be implemented into the design must be based on the pros and cons that do not deal directly with economics. A major concern with a one chiller system, like the one in the Hays School is a lack of redundancy. The failure of the chiller means no cooling for the building. With the ice storage system, on non-design days there is the availability of some redundancy. As the analysis in Section 8.1.2 displayed, even with chiller failure on a warm day, the ice storage system can account for cooling the building for half of the day. On a typical day (under 486 ton-hrs of cooling), a charged ice storage system can handle the entire day of cooling. This results in a higher system reliability and the opportunity to work on a chiller for a couple of hours on the design day, without losing control of the system. This benefit is far more apparent in the 90 ton system then it is with the 100 ton system, making the implementation of the 90 ton chiller with 486 ton-hrs of cooling the most reasonable choice.

Cincinnati does not currently have any time of use electrical charge reductions. If in the future Cincinnati was to implement a reduction in electrical costs at night, the annual energy savings would increase to more significant values. Despite relatively small energy savings with the current electric bill, there is the potential that in the future the proposed ice storage system could see very significant energy savings. By using less on-peak electricity, the Hays School will be doing a service to the community by decreasing the likelihood of brownouts. Though one school will not have a significant impact on the electrical grid, if more of the Cincinnati schools were run with a similar system it could have a significant positive influence on the community as a whole.

Implementing a 90 ton chiller with a 486 ton-hr ice storage system would benefit the George W. Hays by granting a favorable lifecycle cost, increased redundancy and reliability, along with the potential of benefiting the community as a whole.

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Appendix A – Selected Chiller Data

TRANE

Air-Cooled Series R™ Rotary Liquid Chiller

Model RTAA
70 to 125 Tons

Built for Industrial and Commercial Markets

November 2006 RLC-PRC016-EN

TRANE

General Data

Table G-1 — General Data RTAA — 70-125 Ton

Size		70	80	90	100	110	125
Compressor							
Quantity		2	2	2	2	2	2
Nominal Size (1)	(Tons)	35/35	40/40	50/40	50/50	60/50	60/60
Evaporator							
Water Storage	(Gallons)	39.8	37.3	34.4	32.1	53.4	45.8
	(Liters)	150.6	143.1	130.2	121.5	202.11	173.4
Min. Flow	(GPM)	84	96	108	120	132	150
	(L/Sec)	5.3	6.1	6.8	7.6	8.3	9.5
Max. Flow	(GPM)	252	288	324	360	396	450
	(L/Sec)	15.9	18.2	20.4	22.7	25.0	28.4
Condenser							
Qty of Coils		4	4	4	4	4	4
Coil Length	(In)	156/156	156/156	168/156	168/168	204/168	204/204
	(mm)	3962/3962	3962/3962	4267/3962	4267/4267	5182/4267	5182/5182
Coil Height	(In)	42	42	42	42	42	42
	(mm)	1067	1067	1067	1067	1067	1067
Fins/Ft.		192	192	192	192	192	192
Number of Rows		2	2	2	2	2	2
Condenser Fans (60 Hz)							
Quantity (1)		4/4	4/4	5/4	5/5	5/5	5/5
Diameter	(In)	30	30	30	30	30	30
	(mm)	762	762	762	762	762	762
Total Airflow	(CFM)	71750	71750	77640	83530	87505	91480
Nominal RPM	60 Hz	1140	1140	1140	1140	1140	1140
Tip Speed	(Ft/Min)	8954	8954	8954	8954	8954	8954
Motor HP (Ea)		1.25	1.25	1.25	1.25	1.25	1.25
Condenser Fans (50 Hz)							
Quantity (1)		4/4	4/4	5/4	5/5	5/5	5/5
Diameter	(In)	30	30	30	30	30	30
	(mm)	762	762	762	762	762	762
Total Airflow	(CFM)	59172	59200	63963	68724	72104	75492
Nominal RPM	50 Hz	970	970	970	970	970	970
Tip Speed	(Ft/Min)	7618	7618	7618	7618	7618	7618
Motor HP (Ea)		1.25	1.25	1.25	1.25	1.25	1.25
Min Starting/Oper Ambient (2)							
Std Unit	(Deg F)	25	25	25	25	25	25
	(Deg C)	-3.9	-3.9	-3.9	-3.9	-3.9	-3.9
Low Ambient	(Deg F)	-10	-10	-10	-10	-10	-10
	(Deg C)	-23.3	-23.3	-23.3	-23.3	-23.3	-23.3
General Unit							
Refrigerant		HCFC-22	HCFC-22	HCFC-22	HCFC-22	HCFC-22	HCFC-22
No. of Independent Refrigerant Circuits		2	2	2	2	2	2
Refrigerant Charge (1)	(Lb)	58/58	61/61	73/61	73/73	98/73	98/98
	(Kg)	26/26	28/28	34/28	34/34	44/34	44/44
Oil Charge (1)	(Gallons)	2.5/2.5	2.5/2.5	3/2.5	3/3	3/3	3/3
	(Liters)	10.6/10.6	10.6/10.6	12.7/10.6	12.7/12.7	12.7/12.7	12.7/12.7

1. Data containing information on two circuits shown as follows: ckt 1/ckt2.
2. Minimum start-up/operating ambient based on a 5 mph wind across the condenser.

Table G-2 — General Data Pump Package

Pump Package Size		C2	D3	D5	E2	E3	F5	F7	G3	G5
Pump										
Quantity		2	2	2	2	2	2	2	2	2
Motor HP	(each)	2	3	5	2	3	5	7.5	3	5
Water Storage										
4" connection	(Gallons)	13.64	13.54	16.18	16.25	16.25	23.54	23.54	23.62	23.62
	(Liters)	51.63	51.25	61.25	61.51	61.51	89.11	89.11	89.41	89.41
6" connection	(Gallons)	--	16.8	19.41	--	19.59	26.09	26.09	26.56	26.56
	(Liters)	--	64.28	73.47	--	74.16	98.76	98.76	100.54	100.54

1. Information given for 460/60/3 units only.

Selection Procedure

The chiller capacity tables, P-1 through P-4, cover the most frequently encountered leaving water temperatures. The tables reflect a 10°F (6°C) temperature drop through the evaporator. For temperature drops other than 10°F (6°C), refer to Table F-1, and apply the appropriate Performance Data Adjustment Factors. For chilled brine selections, refer to Figures F-2 and 3 for Ethylene and Propylene Glycol Adjustment Factors.

To select a Trane air-cooled Series R™ chiller, the following information is required:

1. Design load in tons of refrigeration
2. Design chilled water temperature drop
3. Design leaving chilled water temperature
4. Design ambient temperature

The following formulas can be used to determine approximate evaporator flow rates:

$$\text{GPM} = \frac{\text{Tons} \times 24}{\text{Temperature Drop (Degrees F)}}$$

OR

$$\text{L/S} = \frac{\text{kW (Capacity)} \times .239}{\text{Temperature Drop (Degrees C)}}$$

NOTE: Flow rates must fall within the limits specified in Table G-1 (for GPM or for l/s). Formulas listed above are useful tools to estimate evaporator flow rates.

For specific chiller performance, contact a local Trane sales engineer.

Selection Example

Given:

Required System Load = 115 Tons
Leaving Chilled Water Temperature (LCWT) = 44°F Chilled Water Temperature Drop = 10°F Design Ambient Temperature = 95°F
Evaporator Fouling Factor = 0.0001

1. To calculate the required chilled water flow rate we use the formula given below:

   $$\text{GPM} = \frac{115 \text{ Tons} \times 24}{10°\text{F}} = 276 \text{ GPM}$$

2. From Table P-1 (RTAA Performance Data), an RTAA 125 at the given conditions will produce 120.0 tons with a compressor power input of 136.2 kW and a unit EER of 9.8.
3. To determine the evaporator pressure drop we use the flow rate (GPM) and the evaporator water pressure drop curves, Figure F-1. Entering the curve at 276 GPM, the pressure drop for a nominal 125 ton evaporator is 18 feet.
4. For selection of chilled brine units or applications where the altitude is significantly greater than sea level or the temperature drop is different than 10°F, the performance adjustment factors from Tables F-1, F-2, and/or F-3 should be applied at this point.

For example:

Corrected Capacity = Capacity (unadjusted) x Glycol Flow Rate Adjustment Factor

5. The final unit selection is:
- QTY (1) RTAA 125
- Cooling Capacity = 120.0 tons
- Entering/Leaving Chilled Water Temperatures = 54/44°F
- Chilled Water Flow Rate = 276 GPM
- Evaporator Water Pressure Drop = 18 feet
- Compressor Power Input = 136.2 kW
- Unit EER = 9.8

Minimum Leaving Chilled Water Temperature Setpoint
The minimum leaving chilled water temperature setpoint for water is 40°F. For those applications requiring lower setpoints, a brine solution must be used. Contact the local Trane sales engineer for additional information.

Chilled Brine Solutions
Series R chillers can utilize a wide variety of chilled fluids other than water in the evaporator, including ethylene glycol and propylene glycol. Chillers using media other than water are excluded from the ARI 550/590 Certification Program, but are rated in accordance with the ARI Standard. Trane factory performance tests are only performed with water as the chilled fluid. When considering selection of media other than water, contact the local Trane sales office for chiller selections and factory performance testing information.

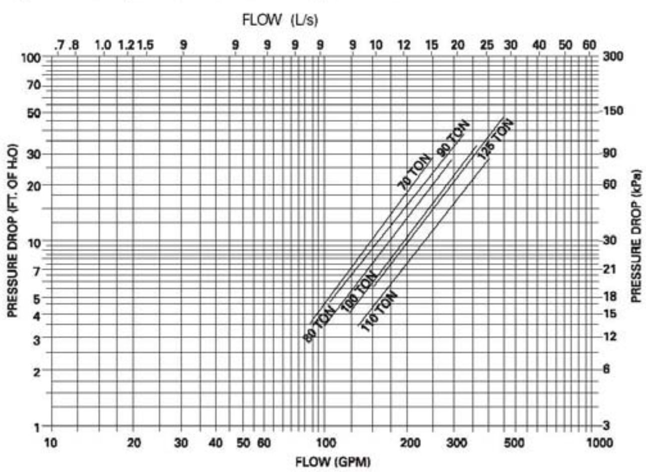
TRANE

Performance Adjustment Factors

Table F-1 — Performance Data Adjustment Factors

Fouling Factor	Chilled Water Temp. Drop	Altitude: Sea Level CAP	Sea Level GPM	Sea Level KW	2000 Feet CAP	2000 Feet GPM	2000 Feet KW	4000 Feet CAP	4000 Feet GPM	4000 Feet KW	6000 Feet CAP	6000 Feet GPM	6000 Feet KW
0.00010	8	1.000	1.249	1.000	0.996	1.245	1.004	0.991	1.240	1.007	0.987	1.234	1.014
	10	1.000	1.000	1.000	0.997	0.996	1.004	0.993	0.992	1.007	0.988	0.988	1.015
	12	1.001	0.835	1.001	0.997	0.832	1.004	0.993	0.828	1.009	0.988	0.824	1.015
	14	1.003	0.716	1.001	0.999	0.714	1.004	0.994	0.711	1.009	0.990	0.708	1.015
	16	1.004	0.628	1.001	1.000	0.626	1.005	0.997	0.623	1.009	0.991	0.620	1.016
0.00025	8	0.988	1.235	0.996	0.984	1.230	1.000	0.980	1.225	1.004	0.975	1.220	1.010
	10	0.988	0.989	0.998	0.986	0.985	1.000	0.981	0.981	1.004	0.977	0.976	1.011
	12	0.990	0.825	0.998	0.987	0.822	1.000	0.983	0.819	1.005	0.978	0.815	1.011
	14	0.991	0.708	0.998	0.988	0.706	1.001	0.984	0.703	1.005	0.980	0.700	1.011
	16	0.993	0.621	0.999	0.990	0.619	1.001	0.986	0.617	1.006	0.981	0.614	1.012

Figure F-1 — Evaporator Water Pressure Drops, 70-125 Ton Units



TRANE

Performance Adjustment Factors

Figure F-2 — Ethylene Glycol Performance Factors

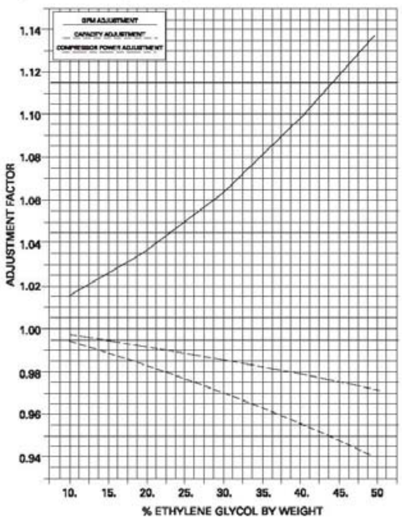


Figure F-3 — Propylene Glycol Performance Factors

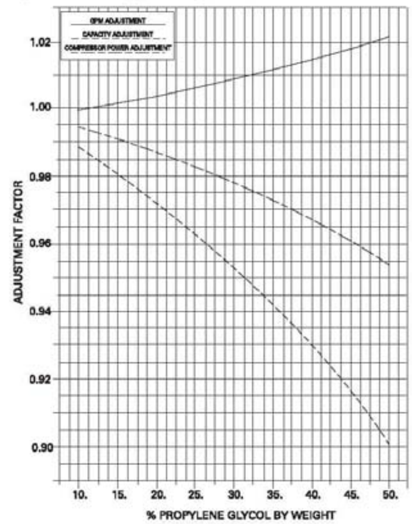
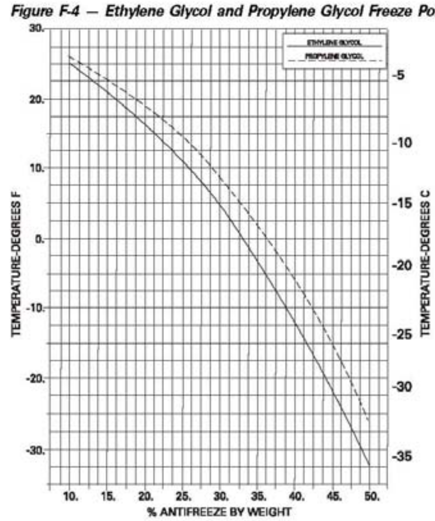


Figure F-4 — Ethylene Glycol and Propylene Glycol Freeze Point



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TRANE

Performance Data

Full Load

Table P-1. 60 Hz chillers in English units

Evaporator Leaving Water Temperature (F)	Unit Size	85 Tons	85 kW input	85 EER	95 Tons	95 kW input	95 EER	105 Tons	105 kW input	105 EER	115 Tons	115 kW input	115 EER
40	70	68.7	64.3	11.2	64.6	70.8	9.7	60.4	77.9	8.4	55.5	84.8	7.1
	80	78.8	75.6	11.2	74.4	83.1	9.7	69.8	91.3	8.4	65.0	100.4	7.2
	90	89.9	88.9	10.9	84.8	97.0	9.5	79.5	106.1	8.2	73.9	116.4	7.0
	100	99.9	101.7	10.6	94.2	110.5	9.3	88.2	120.5	8.1	81.9	131.9	6.9
	110	107.7	110.7	10.6	101.6	120.3	9.3	95.2	131.2	8.1	88.4	143.6	6.9
	125	119.3	122.0	10.8	112.4	132.3	9.4	105.2	144.1	8.2	97.5	157.5	7.0
42	70	71.1	65.1	11.5	66.9	71.6	10.0	62.6	78.7	8.6	57.1	84.8	7.3
	80	81.6	76.8	11.4	77.1	84.2	9.9	72.3	92.5	8.6	67.4	101.6	7.3
	90	93.0	90.3	11.1	87.8	98.4	9.7	82.3	107.5	8.4	76.5	117.8	7.2
	100	103.2	103.3	10.8	97.4	112.0	9.5	91.2	122.1	8.2	84.7	133.5	7.1
	110	111.3	112.4	10.8	105.0	121.9	9.5	98.4	132.9	8.2	91.5	145.3	7.0
	125	123.2	124.0	11.0	116.2	134.2	9.6	108.7	146.1	8.3	100.9	159.5	7.1
44	70	73.5	65.9	11.8	69.3	72.4	10.2	64.9	79.6	8.8	58.6	84.8	7.5
	80	84.5	78.0	11.7	79.8	85.4	10.2	74.9	93.7	8.8	69.9	102.8	7.5
	90	96.1	91.7	11.3	90.8	99.8	9.9	85.1	108.9	8.6	79.2	119.2	7.4
	100	106.6	104.9	11.0	100.6	113.6	9.7	94.3	123.7	8.4	87.6	135.1	7.2
	110	114.9	114.2	11.0	108.5	123.7	9.7	101.7	134.6	8.4	94.6	147.1	7.2
	125	127.3	126.0	11.1	120.0	136.2	9.8	112.4	148.1	8.5	104.3	161.5	7.3
46	70	76.0	66.8	12.0	71.7	73.2	10.5	67.2	80.5	9.0	60.2	84.8	7.7
	80	87.4	79.2	11.9	82.6	86.6	10.4	77.6	94.9	9.0	72.4	104.1	7.7
	90	99.4	93.1	11.6	93.8	101.2	10.1	88.0	110.4	8.8	81.7	120.4	7.5
	100	110.1	106.6	11.2	103.9	115.3	9.9	97.4	125.3	8.6	90.6	136.7	7.4
	110	118.6	116.0	11.2	112.0	125.5	9.9	105.1	136.4	8.6	97.7	148.9	7.4
	125	131.4	128.0	11.3	123.9	138.3	10.0	116.0	150.1	8.7	106.7	162.1	7.4
48	70	78.5	67.6	12.3	74.1	74.1	10.7	69.5	81.4	9.3	61.8	84.8	7.9
	80	90.4	80.4	12.1	85.4	87.9	10.6	80.3	96.2	9.2	74.4	104.7	7.9
	90	102.6	94.6	11.8	96.9	102.7	10.3	90.9	111.8	9.0	82.9	120.1	7.7
	100	113.6	108.3	11.4	107.3	117.0	10.1	100.6	127.0	8.8	92.0	136.7	7.5
	110	122.4	117.8	11.4	115.6	127.3	10.0	108.4	138.3	8.7	99.4	148.9	7.5
	125	135.6	130.2	11.5	127.9	140.4	10.1	119.8	152.2	8.8	107.2	160.2	7.5
50	70	81.1	68.5	12.6	76.5	75.0	11.0	71.8	82.3	9.5	63.4	84.8	8.1
	80	93.4	81.7	12.4	88.3	89.1	10.8	83.0	97.4	9.4	75.9	104.8	8.0
	90	106.0	96.2	12.0	100.1	104.2	10.5	93.9	113.3	9.2	84.3	120.0	7.8
	100	117.2	110.1	11.6	110.7	118.7	10.3	103.8	128.7	8.9	93.4	136.5	7.6
	110	126.2	119.7	11.6	119.2	129.2	10.2	111.9	140.1	8.9	101.0	148.7	7.6
	125	139.9	132.3	11.7	131.9	142.6	10.3	123.6	154.3	9.0	108.0	158.5	7.7

Condenser Entering Air Temperature (F): 85, 95, 105, 115

Notes:
1. Ratings based on sea level altitude and evaporator fouling factor of 0.00010 hr-sq. ft.-°F/Btu.
2. Consult Trane representative for performance at temperatures outside of the ranges shown.
3. kW input is for compressors only.
4. EER = Energy Efficiency Ratio (Btu/watt-hour). Power inputs include compressors, condenser fans and control power.
5. Ratings are based on an evaporator temperature drop of 10°F.
6. Ambient temperatures 115°F and greater reflect the high ambient condenser option.
7. Interpolation between points is permissible. Extrapolation is not permitted.
8. Rated in accordance with ARI Standard 550/590.

Performance Data

Part Load

Table P-5 — ARI Part-Load Values (60 Hz)

Unit	% Load	Tons	EER	IPLV
RTAA 70	100	69.3	10.2	**13.7**
	75	52.0	12.0	
	50	34.6	14.6	
	25	17.3	16.3	
RTAA 80	100	79.8	10.2	**13.3**
	75	59.8	11.7	
	50	39.9	14.9	
	25	20.0	12.7	
RTAA 90	100	90.8	9.9	**13.1**
	75	68.1	11.3	
	50	45.4	14.7	
	25	22.7	13.8	
RTAA 100	100	100.6	9.7	**12.6**
	75	75.5	11.0	
	50	50.3	13.5	
	25	25.2	15.3	
RTAA 110	100	108.5	9.7	**12.5**
	75	81.4	11.0	
	50	54.3	13.3	
	25	27.1	15.2	
RTAA125	100	120.0	9.8	**12.7**
	75	90.0	11.2	
	50	60.0	13.8	
	25	30.0	13.8	

Table P-6 — ARI Part-Load Values (50 Hz)

Unit	% Load	kW cooling	COP	IPLV
RTAA 70	100	211.1	3.2	**4.4**
	75	158.3	3.9	
	50	105.6	4.7	
	25	52.8	5.1	
RTAA 80	100	243.8	3.2	**4.2**
	75	182.8	3.7	
	50	121.9	4.8	
	25	60.9	4.0	
RTAA 90	100	276.9	3.1	**4.2**
	75	207.7	3.6	
	50	138.5	4.7	
	25	69.2	4.3	
RTAA 100	100	306.9	3.0	**4.0**
	75	230.2	3.5	
	50	153.5	4.3	
	25	76.7	4.8	
RTAA 110	100	331.4	3.0	**4.0**
	75	248.5	3.5	
	50	165.7	4.2	
	25	82.8	4.8	
RTAA125	100	365.4	3.1	**4.0**
	75	274.0	3.6	
	50	182.7	4.3	
	25	91.3	4.3	

TRANE

Electrical Data — Packaged Unit

Table E-1 — Electrical Data (50 & 60 Hz, 3 Phase)

	Unit Wiring					Motor Data						
Unit Size	Rated Voltage (9)	# of Power Connections (1)	MCA (3)	Max. Fuse, HACR Breaker or MOP (2,11)	Rec. Time Delay or RDE (4)	Qty	Compressor (Each) RLA (5)	Compressor (Each) LRA (8)	Qty.	Fans (Each) kW	FLA	Control kW (7, 10)
RTAA 70	200/60	1	300	400	350	2	115 - 115	800 - 800	8	1.0	5.1	0.75
	230/60	1	265	350	300	2	100 - 100	690 - 690	8	1.0	5.0	0.75
	380/60	1	163	200	200	2	61 - 61	400 - 400	8	1.0	3.2	0.75
	460/60	1	133	175	150	2	50 - 50	330 - 330	8	1.0	2.5	0.75
	575/60	1	108	125	125	2	40 - 40	270 - 270	8	1.0	2.2	0.75
	400/50	1	133	175	150	2	50 - 50	325 - 325	8	0.7	2.5	0.75
RTAA 80	200/60	1	361	500	400	2	142 - 142	800 - 800	8	1.0	5.1	0.75
	230/60	1	319	400	350	2	124 - 124	760 - 760	8	1.0	5.0	0.75
	380/60	1	194	250	225	2	75 - 75	465 - 465	8	1.0	3.2	0.75
	460/60	1	160	200	175	2	62 - 62	380 - 380	8	1.0	2.5	0.75
	575/60	1	131	175	150	2	50 - 50	304 - 304	8	1.0	2.2	0.75
	400/50	1	160	200	175	2	62 - 62	375 - 375	8	0.7	2.5	0.75
RTAA 90	200/60	1	428	600	500	2	192 - 142	990 - 800	9	1.0	5.1	0.75
	230/60	1	378	500	450	2	167 - 124	820 - 760	9	1.0	5.0	0.75
	380/60	1	230	300	300	2	101 - 75	497 - 465	9	1.0	3.2	0.75
	460/60	1	190	250	225	2	84 - 62	410 - 380	9	1.0	2.5	0.75
	575/60	1	154	200	175	2	67 - 50	328 - 304	9	1.0	2.2	0.75
	400/50	1	190	250	225	2	84 - 62	402 - 375	9	0.7	2.5	0.75
RTAA 100	200/60	1	483	600	600	2	192 - 192	990 - 990	10	1.0	5.1	0.75
	230/60	1	426	500	500	2	167 - 167	820 - 820	10	1.0	5.0	0.75
	380/60	1	259	350	300	2	101 - 101	497 - 497	10	1.0	3.2	0.75
	460/60	1	214	250	250	2	84 - 84	410 - 410	10	1.0	2.5	0.75
	575/60	1	173	225	200	2	67 - 67	328 - 328	10	1.0	2.2	0.75
	400/50	1	214	250	250	2	84 - 84	402 - 402	10	0.7	2.5	0.75
RTAA 110	200/60	1	535	700	600	2	233 - 192	1190 - 990	10	1.0	5.1	0.75
	230/60	1	471	600	600	2	203 - 167	1044 - 820	10	1.0	5.0	0.75
	380/60	1	287	400	350	2	123 - 101	632 - 497	10	1.0	3.2	0.75
	460/60	1	235	300	300	2	101 - 84	522 - 410	10	1.0	2.5	0.75
	575/60	1	191	250	225	2	81 - 67	420 - 328	10	1.0	2.2	0.75
	400/50	1	236	300	300	2	101 - 84	512 - 402	10	0.7	2.5	0.75
RTAA 125	200/60	1	576	800	700	2	233 - 233	1190 - 1190	10	1.0	5.1	0.75
	230/60	1	507	700	600	2	203 - 203	1044 - 1044	10	1.0	5.0	0.75
	380/60	1	309	400	350	2	123 - 123	632 - 632	10	1.0	3.2	0.75
	460/60	1	253	350	300	2	101 - 101	522 - 522	10	1.0	2.5	0.75
	575/60	1	205	250	225	2	81 - 81	420 - 420	10	1.0	2.2	0.75
	400/50	1	253	350	300	2	101 - 101	512 - 512	10	0.7	2.5	0.75

Notes:

1. As standard, all 70-215 ton units require a single point power connection.
2. Max Fuse or HACR type breaker = 225 percent of the largest compressor RLA plus 100 percent of the second compressor RLA, plus the sum of the condenser fan FLA per NEC 440-22. Use FLA per circuit, NOT FLA for the entire unit).
3. MCA - Minimum Circuit Ampacity - 125 percent of largest compressor RLA plus 100 percent of the second compressor RLA plus the sum of the condenser fans FLAs per NEC 440-33.
4. RECOMMENDEDTIME DELAY OR DUAL ELEMENT (RDE) FUSE SIZE: 150 percent of the largest compressor RLA plus 100 percent of the second compressor RLA and the sum of the condenser fan FLAs.
5. RLA - Rated Load Amps - rated in accordance with UL Standard 1995.
6. Local codes may take precedence.
7. Control kW includes operational controls only. Does not include evaporator heat tape.
8. LRA - Locked Rotor Amps - based on full winding (x-line) start units. LRA for wye-delta starters is 1/3 of LRA of x-line units.
9. VOLTAGE UTILIZATION RANGE:

Rated Voltage	Utilization Range
200	180-220
230	208-254
380	342-418
460	414-506
575	516-633

10. A separate 115/60/1 or 220/50/1, 15 amp customer provided power connection is also needed to power the evaporator heat tape (420 watts @ 120 volts or 420 watts @ 220 volts).
11. If factory circuit breakers are supplied with the chiller, then these values represent Maximum Overcurrent Protection (MOP).

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TRANE

Electrical Data

Pump Package Unit

Table E-2 — Electrical Data (60 Hz, 3 Phase)

	Unit Wiring					Motor Data						
Unit Size	Rated Voltage (9)	Pump HP, FLA	MCA (3)	Max. Fuse, HACR Breaker or MOP (2,11)	Rec. Time Delay or RDE (4)	Qty	Compressor (Each) RLA (5)	LRA (8)	Qty.	Fans (Each) kW	FLA	Control kW (7, 10)
RTAA 70	460/60	2 , 3.1	136	175	150	2	50 - 50	330 - 330	8	1.0	2.5	0.75
	460/60	3 , 4.1	137	175	150	2	50 - 50	330 - 330	8	1.0	2.5	0.75
	460/60	5 , 6.6	139	175	175	2	50 - 50	330 - 330	8	1.0	2.5	0.75
RTAA 80	460/60	2 , 3.1	163	200	200	2	62 - 62	380 - 380	8	1.0	2.5	0.75
	460/60	3 , 4.1	164	225	200	2	62 - 62	380 - 380	8	1.0	2.5	0.75
	460/60	5 , 6.6	166	225	200	2	62 - 62	380 - 380	8	1.0	2.5	0.75
	460/60	7.5 , 10.3	170	225	200	2	62 - 62	380 - 380	8	1.0	2.5	0.75
RTAA 90	460/60	3 , 4.1	194	250	225	2	84 - 62	410 - 380	9	1.0	2.5	0.75
	460/60	5 , 6.6	196	250	225	2	84 - 62	410 - 380	9	1.0	2.5	0.75
	460/60	7.5 , 10.3	200	250	225	2	84 - 62	410 - 380	9	1.0	2.5	0.75
RTAA 100	460/60	3 , 4.1	218	300	250	2	84 - 84	410 - 410	10	1.0	2.5	0.75
	460/60	5 , 6.6	221	300	250	2	84 - 84	410 - 410	10	1.0	2.5	0.75
	460/60	7.5 , 10.3	224	3000	250	2	84 - 84	410 - 410	10	1.0	2.5	0.75
RTAA 110	460/60	3 , 4.1	239	300	300	2	101 - 84	522 - 410	10	1.0	2.5	0.75
	460/60	5 , 6.6	242	300	300	2	101 - 84	522 - 410	10	1.0	2.5	0.75
	460/60	7.5 , 10.3	246	300	300	2	101 - 84	522 - 410	10	1.0	2.5	0.75
RTAA 125	460/60	3 , 4.1	256	350	300	2	101 - 101	522 - 522	10	1.0	2.5	0.75
	460/60	5 , 6.6	259	350	300	2	101 - 101	522 - 522	10	1.0	2.5	0.75
	460/60	7.5 , 10.3	263	350	300	2	101 - 101	522 - 522	10	0.7	2.5	0.75

Notes:

1. As standard, all 70-215 ton units require a single point power connection.
2. Max Fuse or HACR type breaker = 225 percent of the largest compressor RLA plus 100 percent of the second compressor RLA, plus the sum of the condenser fan FLA per NEC 440-22. Use FLA per circuit, NOT FLA for the entire unit).
3. MCA - Minimum Circuit Ampacity - 125 percent of largest compressor RLA plus 100 percent of the second compressor RLA plus the sum of the condenser fans FLAs per NEC 440-33.
4. RECOMMENDEDTIME DELAY OR DUAL ELEMENT (RDE) FUSE SIZE: 150 percent of the largest compressor RLA plus 100 percent of the second compressor RLA and the sum of the condenser fan FLAs.
5. RLA - Rated Load Amps - rated in accordance with UL Standard 1995.
6. Local codes may take precedence.
7. Control kW includes operational controls only. Does not include evaporator heat tape.
8. LRA - Locked Rotor Amps - based on full winding (x-line) start units. LRA for wye-delta starters is 1/3 of LRA of x-line units.
9. VOLTAGE UTILIZATION RANGE:

Rated Voltage	Utilization Range
460	414-506

10. A separate 115/60/1, 15 amp customer provided power connection is also needed to power the evaporator heat tape (420 watts @ 120 volts).
11. If factory circuit breakers are supplied with the chiller, then these values represent Maximum Overcurrent Protection (MOP).

TRANE

Electrical Data — Wire Size

Table E-3 – Customer Wire Selection

		Wire Selection Size to Main Terminal Block		Wire Selection Size to Disconnect (1)		Wire Selection Size to Circuit Breaker (1)	
Unit Size	Rated Voltage	Terminal Size Ckt 1	Connector Wire Range Ckt 1	Disconnect Size Ckt 1	Connector Wire Range Ckt 1	Factory Mounted Internal Circuit Breaker Size (3) Ckt 1	Connector Wire Range Ckt 1
RTAA 70	200/60	760 Amp	Lug Size D	400 Amp	Lug Size B	350 Amp	Lug Size B
	230/60	760 Amp	Lug Size D	400 Amp	Lug Size B	300 Amp	Lug Size B
	380/60	335 Amp	Lug Size E	250 Amp	Lug Size A	200 Amp	Lug Size A
	460/60	335 Amp	Lug Size E	250 Amp	Lug Size A	150 Amp	Lug Size A
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	125 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	250 Amp	Lug Size A	150 Amp	Lug Size A
RTAA 80	200/60	760 Amp	Lug Size D	400 Amp	Lug Size B	400 Amp	Lug Size B
	230/60	760 Amp	Lug Size D	400 Amp	Lug Size B	350 Amp	Lug Size B
	380/60	335 Amp	Lug Size E	250 Amp	Lug Size A	225 Amp	Lug Size A
	460/60	335 Amp	Lug Size E	250 Amp	Lug Size A	175 Amp	Lug Size A
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	150 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	250 Amp	Lug Size A	175 Amp	Lug Size A
RTAA 90	200/60	760 Amp	Lug Size D	600 Amp	Lug Size C	500 Amp	Lug Size C
	230/60	760 Amp	Lug Size D	400 Amp	Lug Size B	450 Amp	Lug Size C
	380/60	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B
	460/60	335 Amp	Lug Size E	250 Amp	Lug Size A	225 Amp	Lug Size A
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	175 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	250 Amp	Lug Size A	225 Amp	Lug Size A
RTAA 100	200/60	760 Amp	Lug Size D	600 Amp	Lug Size C	600 Amp	Lug Size C
	230/60	760 Amp	Lug Size D	600 Amp	Lug Size C	500 Amp	Lug Size C
	380/60	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B
	460/60	335 Amp	Lug Size E	250 Amp	Lug Size A	250 Amp	Lug Size A
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	200 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	250 Amp	Lug Size A	250 Amp	Lug Size A
RTAA 110	200/60	760 Amp	Lug Size D	600 Amp	Lug Size C	600 Amp	Lug Size C
	230/60	760 Amp	Lug Size D	600 Amp	Lug Size C	600 Amp	Lug Size C
	380/60	335 Amp	Lug Size E	400 Amp	Lug Size B	350 Amp	Lug Size B
	460/60	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	225 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B
RTAA 125	200/60	760 Amp	Lug Size D	600 Amp	Lug Size C	N/A	N/A
	230/60	760 Amp	Lug Size D	600 Amp	Lug Size C	600 Amp	Lug Size C
	380/60	335 Amp	Lug Size E	400 Amp	Lug Size B	350 Amp	Lug Size B
	460/60	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B
	575/60	335 Amp	Lug Size E	250 Amp	Lug Size A	225 Amp	Lug Size A
	400/50	335 Amp	Lug Size E	400 Amp	Lug Size B	300 Amp	Lug Size B

Lug Size A = #4 to 350 MCM per phase
Lug Size B = 2/0 to 250 MCM & 2/0 to 500 MCM per phase
Lug Size C = (2) 400 MCM to 500 MCM per phase
Lug Size D = (2) #4 to 500 MCM per phase
Lug Size E = #6 to 400 MCM per phase
Lug Size F = (2) #2 to 600 MCM per phase
Lug Size G = (2) #1 to 500 MCM per phase
Lug Size H = (4) #2 to 600 MCM per phase

Notes
1. Non-fused unit disconnect and circuit breaker are optional.
2. Copper wire only, sized per N.E.C., based on nameplate minimum circuit ampacity (MCA).
3. Circuit Breaker sizes are for factory mounted only. Field installed circuit breakers need to be sized using HACR breaker recommendations from Table E-1.

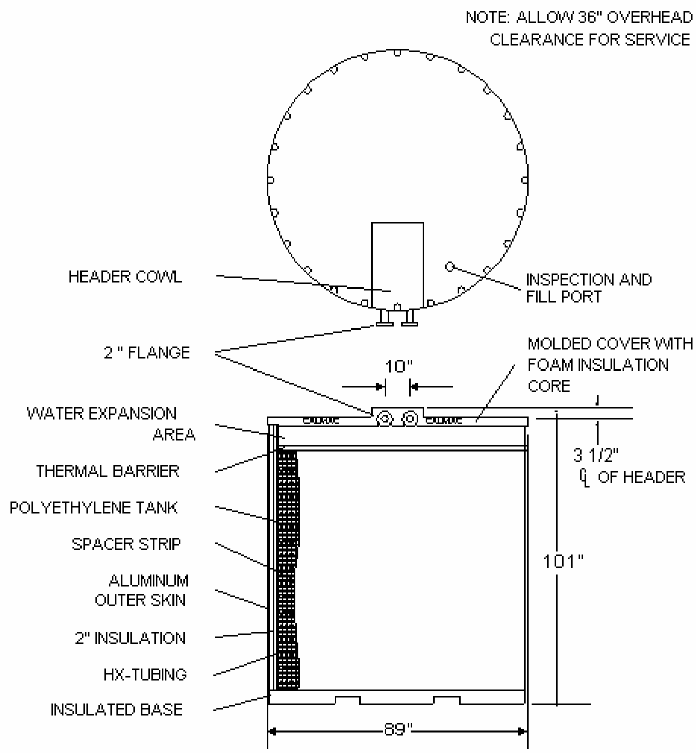
Appendix B – CALMAC Ice Storage Data

TANK MODELS	1082A	1098A	1105A	1190A
Net-Usable Capacity Ton-Hr	82	98	105	162
With Mix Air	79	92	--	156
Max. Operating Temp.,ºF	100	100	100	100
Factory Tested Pres., PSI	250	250	250	250
Max. Operating Pres., PSI	90	90	90	90
Dimensions (OD×H), in.	74×82	89×68	74x101	89x101
Dimensions (W×L×H), in..	--	--	--	--
Shipping Weight, Lbs	1,025	1,225	1,275	1,950
Weight, Filled, lb	8,455	10,100	10,760	16,765
Floor Loading, lb/Sq Ft	283	234	360	388
Volume Of Water/Ice, Gal	820	980	1,045	1,655
Vol. Solution in HX, Gal	78	90	99	148
With Mix Air	75	87	--	143
Type of connection	2" Flange	2" Flange	2" Flange	2" Flange

i. Typical value, actual varies with conditions
ii. Consult factory for higher ratings
iii. Shipping weight may vary slightly because of differences in volumes of residual water from hydrostatic test

CALMAC ICE BANK
Thermal Energy Storage

Technical Bulletin
Ice Bank Detail
Model 1190A
November 1999
CS-5



Calmac Manufacturing Corp. 101W. Sheffield Ave. Englewood, NJ 07631 / Ph 201.569.0420 / Fax 201.569.7593

Appendix C – Duke Energy Electric Rates

P.U.C.O. Electric No. 19
Sheet No. 40.10
Cancels and Supersedes
Sheet No. 40.9
Page 1 of 4

Duke Energy Ohio
139 East Fourth Street
Cincinnati, Ohio 45202

RATE DS

SERVICE AT SECONDARY DISTRIBUTION VOLTAGE

APPLICABILITY

Applicable to electric service for usual customer load requirements is available to a customer only where the Company specifies service at the standard secondary system voltage and where the Company determines that facilities of adequate capacity are available and adjacent to the premises to be served and the Company determines that the customers average monthly demand is greater than 15 kilowatts. Electric service must be supplied at one point of delivery.

For customers taking service under any or all of the provisions of this tariff schedule, this same schedule shall constitute the Company's Standard Service Offer.

TYPE OF SERVICE

Alternating current 60 Hz, single phase or three phase, at Company's standard distribution voltage.

NET MONTHLY BILL

Computed in accordance with the following charges provided, however, that the minimum monthly load factor, expressed as hours-use per month, shall not be less than 71 kWh per kW. When applicable, the minimum monthly load factor shall be achieved by calculating the billing demand as the monthly kWh usage divided by 71 (kilowatt of demand is abbreviated as kW and kilowatt-hours are abbreviated as kWh):

1. Distribution Charges
   (a) Customer Charge per month

Single Phase Service	$ 7.50
Single and/or Three Phase Service	$15.00

   (b) Demand Charge

All kilowatts	$ 3.7908 per kW

2. Applicable Riders
   The following riders are applicable pursuant to the specific terms contained within each rider:
   Sheet No. 51, Rider AAC, Annually Adjusted Component Rider
   Sheet No. 52, Rider DSMR, Demand Side Management Cost Recovery Rider
   Sheet No. 53, Rider FPP, Fuel and Economy Purchased Power Rider
   Sheet No. 54, Rider IMF, Infrastructure Maintenance Fund Rider
   Sheet No. 55, Rider RSC, Rate Stabilization Charge Rider
   Sheet No. 56, Rider SRT, System Reliability Tracker
   Sheet No. 57, Rider TCR, Transmission Cost Recovery Rider
   Sheet No. 58, Rider DRI, Distribution Reliability Investment Rider
   Sheet No. 59, Rider RSS, Rate Stabilization Surcredit Rider
   Sheet No. 81, Rider EER, Energy Efficiency Revolving Loan Program Rider
   Sheet No. 83, Rider OET, Ohio Excise Tax Rider
   Sheet No. 84, Rider RTC, Regulatory Transition Charge Rider
   Sheet No. 85, Rider SC, Shopping Credit Rider
   Sheet No. 86, Rider USR, Universal Service Fund Rider
   Sheet No. 103, Rider MSR-E, Merger Savings Credit Rider-Electric

Filed pursuant to an Order dated March 29, 2006 in Case No. 06-407-GE-ATA before the Public Utilities Commission of Ohio.

Issued: March 31, 2006 Effective: April 3, 2006

Issued by Sandra P. Meyer, President

P.U.C.O. Electric No. 19
Sheet No. 40.10
Cancels and Supersedes
Sheet No. 40.9
Page 2 of 4

Duke Energy Ohio
139 East Fourth Street
Cincinnati, Ohio 45202

NET MONTHLY BILL (Contd.)

3. Market Price Generation Charges – Market Based Standard Service Offer

(a) Demand Charge

First 1,000 kilowatts	$6.5088 per kW
Additional kilowatts	$5.1488 per kW

(b) Energy Charge

Billing Demand times 300	$0.016640 per kWh
Additional kWh	$0.013826 per kWh

The Generation Charges listed above are applicable to all customers except those customers that switch to a Certified Supplier for their generation service. For customers who are procuring their energy supply from a Certified Supplier and receiving a shopping credit on December 31, 2004, the Generation Charges shown below will continue to apply until December 31, 2005.
Customers who return to the Company's energy supply after January 2, 2005, will be billed for generation service for each hour at the higher of the following:

1. The demand-related component of the Market Price Generation Charge, plus the energy-related component of the Market Price Generation Charge, plus Rider FPP, or
2. The demand-related component of the Market Price Generation Charge, plus the incremental dispatch cost of the highest cost generation unit/purchased power to serve Duke Energy Ohio load.

The following Generation Charges apply to customers receiving a Shopping Credit during 2005:

(a) Demand Charge

First 1,000 kilowatts	$7.6574 per kW
Additional kilowatts	$6.0574 per kW

(b) Energy Charge

Billing Demand times 300	$0.028568 per kWh
Additional kWh	$0.016366 per kWh

When both single and three phase secondary voltage services are required by a Distribution customer, the monthly kilowatt-hour usage and kilowatt demands shall be the respective arithmetical sums of both services.

MINIMUM BILL PROVISION

The minimum bill shall be 85% of the highest monthly kilowatt demand as established in the summer period and effective for the next succeeding eleven (11) months plus the Customer Charge.

In no case, however, shall the minimum bill be less than the Customer Charge.

METERING

The Company may meter at secondary or primary voltage as circumstances warrant. If the Company elects to meter at primary voltage, the kilowatt-hours registered on the Company's meter will be reduced one and one-half (1.5) percent for billing purposes.

Filed pursuant to an Order dated March 29, 2006 in Case No. 06-407-GE-ATA before the Public Utilities Commission of Ohio.

Issued: March 31, 2006 Effective: April 3, 2006

Issued by Sandra P. Meyer, President

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Duke Energy Ohio
139 East Fourth Street
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DEMAND

The demand shall be the kilowatts derived from the Company's demand meter for the fifteen-minute period of customer's greatest use during the billing period, as determined by the Company, adjusted for power factor, as provided herein. At the Company's option, a demand meter may not be installed if the nature of the load clearly indicates the load will have a constant demand, in which case the demand will be the calculated demand.

In no event will the billing demand be taken as less than the higher of the following:

a) 85% of the highest monthly kilowatt demand as established in the summer period and effective for the next succeeding eleven (11) months; or

b) One (1) kilowatt for each single phase meter and five (5) kilowatts for each three phase meter.

The Company may re-determine customer's billing demand at any time in recognition of a permanent change in load due to such occurrences as the installation of load control equipment or a temporary change due to malfunctions of such equipment.

If a customer requests reconnection of an account within twelve (12) months of a disconnection order, the customer's demand record for the period of disconnection will be re-established for purposes of billing and administration of the preceding clause.

For purposes of administration of the above clause, the summer period is defined as that period represented by the Company's billing for the four (4) revenue months of June through September. The winter period is defined as that period represented by the Company's billing for the eight (8) revenue months of January through May and October through December.

POWER FACTOR ADJUSTMENT

The power factor to be maintained shall be not less than 90% lagging. If the Company determines customer's power factor to be less than 90%, the billing demand will be the number of kilowatts equal to the kilovolt amperes multiplied by 0.90.

Power factor may be determined by the following methods, at the Company's option:

a) Continuous measurement
   - the power factor, as determined during the interval in which the maximum kW demand is established, will be used for billing purposes; or

b) Testing
   - the power factor, as determined during a period in which the customer's measured kW demand is not less than 90% of the measured maximum kW demand of the preceding billing period, will be used for billing purposes until superseded by a power factor determined by a subsequent test made at the direction of Company or request of customer.

LATE PAYMENT CHARGE

Payment of the total amount due must be received in the Company's office by the due date shown on the bill. When not so paid, an additional amount equal to one and one-half percent (1.5%) of the unpaid balance is due and payable. The late payment charge is not applicable to unpaid account balances for services received from a Certified Supplier.

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Cincinnati, Ohio 45202

TERMS AND CONDITIONS

The initial term of contract shall be for a minimum period of three (3) years terminable thereafter by a minimum notice of either the customer or the Company as prescribed by the Company's Service Regulations.

The Company is not obligated to extend, expand or rearrange its transmission system voltage if it determines that existing distribution and/or transmission facilities are of adequate capacity to serve the customer's load.

If the Company offers to provide the necessary facilities for transmission service, in accordance with its Service Regulations, an annual facilities charge, applicable to such additional facilities, is established at twenty (20) percent of actual cost. The annual facilities charge shall be billed in twelve monthly installments to be added to the demand charge.

SERVICE REGULATIONS

The supplying and billing for service and all conditions applying thereto, are subject to the jurisdiction of the Public Utilities Commission of Ohio, and to the Company's Service Regulations currently in effect, as filed with the Public Utilities Commission of Ohio.

Filed pursuant to an Order dated March 29, 2006 in Case No. 06-407-GE-ATA before the Public Utilities Commission of Ohio.

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Appendix D – NEC Tables

Table D.1

Ampacities of Not More Than Three Single Insulated Conductors, Rated 0 through 2000 Volts, in Raceway in Free Air (from *NEC* Table 310-16)

Based on Ambient Air Temperature of 30°C (86°F).

Size	Temperature Rating of Conductor. See Table 310-13.								Size
	60°C (140°F)	75°C (167°F)	85°C (185°F)	90°C (194°F)	60°C (140°F)	75°C (167°F)	85°C (185°F)	90°C (194°F)	
AWG MCM	TYPES †TW, †UF	TYPES †FEPW, †RH, †RHW, †THW, †THWN, †XHHW, †USE, †ZW	TYPE V	TYPES TA, TBS, SA, AVB, SIS, †FEP, †FEPB, †RHH, †THHN, †XHHW'	TYPES †TW, †UF	TYPES †RH, †RHW, †THW, †THWN, †XHHW †USE	TYPE V	TYPES TA, TBS, SA, AVB, SIS, †RHH, †THHN, †XHHW*	AWG MCM
	COPPER				ALUMINUM OR COPPER-CLAD ALUMINUM				
18	....	....	....	14	....	....	....	....	....
16	....	....	18	18	....	....	....	....	....
14	20†	20†	25	25†	....	....	....	....	....
12	25†	25†	30	30†	20†	20†	25	25†	12
10	30	35†	40	40†	25	30†	30	35†	10
8	40	50	55	55	30	40	40	45	8
6	55	65	70	75	40	50	55	60	6
4	70	85	95	95	55	65	75	75	4
3	85	100	110	110	65	75	85	85	3
2	95	115	125	130	75	90	100	100	2
1	110	130	145	150	85	100	110	115	1
1/0	125	150	165	170	100	120	130	135	1/0
2/0	145	175	190	195	115	135	145	150	2/0
3/0	165	200	215	225	130	155	170	175	3/0
4/0	195	230	250	260	150	180	195	205	4/0
250	215	255	275	290	170	205	220	230	250
300	240	285	310	320	190	230	250	255	300
350	260	310	340	350	210	250	270	280	350
400	280	335	365	380	225	270	295	305	400
500	320	380	415	430	260	310	335	350	500
600	355	420	460	475	285	340	370	385	600
700	385	460	500	520	310	375	405	420	700
750	400	475	515	535	320	385	420	435	750
800	410	490	535	555	330	395	430	450	800
900	435	520	565	585	355	425	465	480	900
1000	455	545	590	615	375	445	485	500	1000
1250	495	590	640	665	405	485	525	545	1250
1500	520	625	680	705	435	520	565	585	1500
1750	545	650	705	735	455	545	595	615	1750
2000	560	665	725	750	470	560	610	630	2000
	AMPACITY CORRECTION FACTORS								
Ambient Temp. °C	For ambient temperatures other than 30°C (86°F), multiply the ampacities shown above by the appropriate factor shown below.								Ambient Temp. °F
21-25	1.08	1.05	1.04	1.04	1.08	1.05	1.04	1.04	70-77
26-30	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00	79-86
31-35	.91	.94	.95	.96	.91	.94	.95	.96	88-95
36-40	.82	.88	.90	.91	.82	.88	.90	.91	97-104
41-45	.71	.82	.85	.87	.71	.82	.85	.87	106-113
46-50	.58	.75	.80	.82	.58	.75	.80	.82	115-122
51-55	.41	.67	.74	.76	.41	.67	74	.76	124-131
56-60	....	.58	.67	.71	....	.58	.67	.71	133-140
61-70	....	.33	.52	.58	....	.33	.52	.58	142-158
71-80	....	.... *	.30	.41	....	....	.30	.41	160-176

† Unless otherwise specifically permitted elsewhere in this Code, the overcurrent protection for conductor types marked with an obelisk (†) shall not exceed 15 amperes for 14 AWG, 20 amperes for 12 AWG, and 30 amperes for 10 AWG copper; or 15 amperes for 12 AWG and 25 amperes for 10 AWG aluminum and copper-clad aluminum after any correction factors for ambient temperature and number of conductors have been applied.

* For dry and damp locations only. See 75°C column for wet locations.

Table D.2

Minimum Size of Equipment Grounding Conductors for Grounding Raceways and Equipment (from *NEC* Table 250-95)

Rating or Setting of Automatic Overcurrent Device in Circuit Ahead of Equipment, Conduit, etc., Not Exceeding (Amperes)	Size: Copper Wire No.	Size: Aluminum or Copper-Clad Aluminum Wire No.[a]
15	14	12
20	12	10
30	10	8
40	10	8
60	10	8
100	8	6
200	6	4
300	4	2
400	3	1
500	2	1/0
600	1	2/0
800	0	3/0
1000	2/0	4/0
1200	3/0	250 MCM
1600	4/0	350 MCM
2000	250 MCM	400 MCM
2500	350 MCM	600 MCM
3000	400 MCM	600 MCM
4000	500 MCM	800 MCM
5000	700 MCM	1200 MCM
6000	800 MCM	1200 MCM

[a] See installation restrictions in Section 250-92(a).

Table D.3

Minimum Size of Conductors for Grounding AC Systems
(from *NEC* Table 250-94)

Size of Largest Service-Entrance Conductor or Equivalent Area for Parallel Conductors		Size of Grounding Electrode Conductor	
Copper	**Aluminum or Copper-Clad Aluminum**	**Copper**	**[a]Aluminum or Copper-Clad Aluminum**
2 or smaller	0 or smaller	8	6
1 or 0	2/0 or 3/0	6	4
2/0 or 3/0	4/0 or 250 MCM	4	2
Over 3/0 thru 350 MCM	Over 250 MCM thru 500 MCM	2	0
Over 350 MCM thru 600 MCM	Over 500 MCM thru 900 MCM	0	3/0
Over 600 MCM thru 1100 MCM	Over 900 MCM thru 1750 MCM	2/0	4/0
Over 1100 MCM	Over 1750 MCM	3/0	250 MCM

[a] See installation restrictions in Section 250-92(a).

See Section 250-23(b).

Where there are no service-entrance conductors, the grounding electrode conductor size shall be determined by the equivalent size of the largest service-entrance conductor required for the load to be served.

Table D.4

Voltage Drops for 60 Hz Systems*

Size AWG or MCM	COPPER: Magnetic Conduit or Armour 80% P.F.	90% P.F.	100% P.F.	COPPER: Non-Mag. Conduit or Armour 80% P.F.	90% P.F.	100% P.F.	ALUMINUM: Magnetic Conduit or Armour 80% P.F.	90% P.F.	100% P.F.	ALUMINUM: Non-Mag. Conduit or Armour 80% P.F.	90% P.F.	100% P.F.
14	2.540	2.790	3.067	2.535	2.780	3.060						
12	1.570	1.749	1.917	1.565	1.749	1.923	2.460	2.748	3.020	2.448	2.743	3.020
10	.993	1.103	1.200	.987	1.103	1.201	1.553	1.732	1.900	1.547	1.726	1.900
8	.635	.699	.750	.629	.693	.751	.993	1.103	1.195	.981	1.091	1.195
6	.421	.462	.485	.461	.456	.485	.647	.710	.762	.641	.710	.768
4	.277	.300	.306	.271	.294	.306	.421	.456	.491	.410	.450	.479
2	.185	.196	.196	.179	.191	.191	.271	.294	.300	.266	.289	.306
1	.150	.162	.150	.150	.156	.150	.225	.237	.242	.219	.231	.242
1/0	.127	.133	.121	.121	.127	.121	.185	.196	.191	.179	.191	.191
2/0	.109	.110	.098	.098	.104	.092	.150	.156	.150	.144	.150	.150
3/0	.092	.092	.081	.081	.087	.075	.127	.133	.121	.121	.127	.121
4/0	.081	.075	.064	.069	.069	.057	.104	.104	.098	.098	.104	.098
250	.070	.070	.054	.064	.064	.051	.092	.092	.081	.086	.087	.081
300	.064	.064	.045	.056	.055	.042	.081	.081	.069	.075	.075	.069
350	.058	.055	.039	.051	.049	.036	.075	.075	.058	.069	.069	.057
400	.055	.051	.035	.047	.044	.032	.069	.069	.053	.064	.064	.050
500	.049	.045	.029	.042	.039	.026	.058	.057	.043	.053	.051	.040
600	.046	.041	.024	.038	.034	.022	.055	.051	.036	.048	.046	.034
700	.043	.038	.021	.036	.032	.019	.051	.047	.032	.044	.041	.029
750	.042	.037	.020	.034	.031	.017	.049	.046	.030	.042	.039	.028
1000	.038	.032	.016	.029	.025	.013	.044	.040	.024	.039	.036	.024

* Values are per 1000 ampere-feet for three single conductors in conduit.

1. Values are based on three-phase, line-to-neutral voltages. For line-to-line voltage drops, multiply by a factor of 1.73. For single-phase circuits, multiply by a factor of 2.0.

2. Values are for conductor operating temperatures up to 75°C. For conductors operating at 90°C, multiply by a factor of 1.1.

Source: Courtesy of Canada Wire and Cable Limited

Table D.5

Maximum Number of Conductors in Trade Size of C[illegible]uit or Tubing (from *NEC* Chapter 9, Tables 3A and 3B)

Conduit Trade Size (inches)		½	¾	1	1¼	1½	2	2½	3
Type Letters	**Conductor Size**								
THW	14	6	10	16	29	40	65	93	143
	12	4	8	13	24	32	53	76	117
	10	4	6	11	19	26	43	61	95
	8	1	3	5	10	13	22	32	49
	6	1	2	4	7	10	16	23	36
	4	1	1	3	5	7	12	17	27
	3	1	1	2	4	6	10	15	23
	2	1	1	2	4	5	9	13	20
	1		1	1	3	4	6	9	14
	1/0		1	1	2	3	5	8	12
	2/0		1	1	1	3	5	7	10
	3/0		1	1	1	2	4	6	9
	4/0			1	1	1	3	5	7
	250 MCM			1	1	1	2	4	6
	300 MCM			1	1	1	2	3	5
	350 MCM				1	1	1	3	4
	400 MCM				1	1	1	2	4
	500 MCM				1	1	1	1	3
XHHW	14	9	15	25	44	60	99	142	
	12	7	12	19	35	47	78	111	171
	10	5	9	15	26	36	60	85	131
	8	2	4	7	12	17	28	40	62
	6	1	3	5	9	13	21	30	47
	4	1	2	4	7	9	16	22	35
	3	1	1	3	6	8	13	19	29
	2	1	1	3	5	7	11	16	25
	1		1	1	3	5	8	12	18
	1/0		1	1	3	4	7	10	15
	2/0		1	1	2	3	6	8	13
	3/0		1	1	1	3	5	7	11
	4/0		1	1	1	2	4	6	9
	250 MCM			1	1	1	3	4	7
	300 MCM			1	1	1	3	4	6
	350 MCM			1	1	1	2	3	5
	400 MCM				1	1	1	3	5
	500 MCM				1	1	1	2	4

These values apply only when all conductors in the conduit run are the same type and size.

For a complete listing of all types of cables, for conductor sizes above 500 MCM, and for conduit trade sizes above 3 inches, see *NEC* Chapter 9, Tables 1 to 7.

Appendix E – Detailed Cost Data

Proposed Systems Equipment			100 ton	
	per item	units	items	Price
Glycol	9	Gal	283.44	3051
98A with Lifting bar, makeup system and invitory meter	7664	item	2.00	17728
105A with Lifting bar, makeup system and invitory meter	8105	item	0.00	0
1190A with Lifting bar, makeup system and invitory meter	11700	item	1.00	12900
Primary Pump	7050	item	1.00	7050
Seccondary Pump	8050	item	1.00	8050
Screw Chiller	-	item	1.00	62000
5" Piping	92	ft	157.00	14444
3-1/2" Piping	58	ft	904.00	52432
3" Piping	48	ft	180.00	8640
Concrete	105	CY	4.17	438
#4 bar	2550	ton	1.67	4259
4/0	840	CLF	0.00	0
250 MCM	900	CLF	0.00	0
300 MCM	1050	CLF	2.49	2615
2" Conduit	17.7	CLF	0	0
2-1/2" Conduit	24	CLF	1.64	39
#2 GND	229	CLF	1.66	380
300 Amp Non-Fused Switch	-	item	1.00	1300
250 A GFP	735	item	1.00	735
			TOTAL	**196060**
			90 ton	
	per item	units	items	Price
Glycol	9	Gal	289.89	3109
98A with Lifting bar, makeup system and invitory meter	7664	item	0.00	0
105A with Lifting bar, makeup system and invitory meter	8105	item	0.00	0
1190A with Lifting bar, makeup system and invitory meter	11700	item	3.00	38700
Primary Pump	7050	item	1.00	7050
Seccondary Pump	8050	item	1.00	8050
Screw Chiller	-	item	1.00	55000
5" Piping	92	ft	157.00	14444
3-1/2" Piping	58	ft	904.00	52432
3" Piping	48	ft	180.00	8640
Concrete	105	CY	4.17	438
#4 bar	2550	ton	1.67	4259
4/0	840	CLF	0.00	0
250 MCM	900	CLF	2.49	2241
300 MCM	1050	CLF	0.00	0
2" Conduit	17.7	CLF	0	0
2-1/2" Conduit	24	CLF	1.64	39
#2 GND	229	CLF	1.66	380
300 Amp Non-Fused Switch	-	item	1.00	1300
250 A GFP	735	item	1.00	735
			TOTAL	**196817**

			85 ton	
	per item	units	items	Price
Glycol	9	Gal	392.7709	4034.938
98A with Lifting bar, makeup system and invitory meter	7664	item	2	17728
105A with Lifting bar, makeup system and invitory meter	8105	item	4	37220
1190A with Lifting bar, makeup system and invitory meter	11700	item	0	0
Primary Pump	7050	item	1.00	7050
Seccondary Pump	8050	item	1.00	8050
Screw Chiller	-	item	1.00	51000
5" Piping	92	ft	157.00	14444
3-1/2" Piping	58	ft	904.00	52432
3" Piping	48	ft	180.00	8640
Concrete	105	CY	7.16	751.8
#4 bar	2550	ton	2.88	7344
4/0	840	CLF	2.49	2091.6
250 MCM	900	CLF	0.00	0
300 MCM	1050	CLF	0.00	0
2" Conduit	17.7	CLF	1.64	29.028
2-1/2" Conduit	24	CLF	0.00	0
#2 GND	229	CLF	1.66	380.14
300 Amp Non-Fused Switch	-	item	1	1300
250 A GFP	735	item	1.00	735
			TOTAL	**213231**

Base Case System Equipment	**market price**			**Total**
	per item	units	items	Price
CHW Pump	8600	item	1.00	8600
Chiller	77000	item	1.00	82000
6" Piping	225	ft	87.00	19575
4" Piping	65	ft	904.00	58760
3" Piping	48	ft	180.00	8640
350 MCM	1150	CLF	4.98	5727
2-1/2" Conduit	24	CLF	0.83	20
#1 GND	390	CLF	1.66	647
600 Amp Non-Fused Switch	3425	item	1.00	3425
450 A GFP	790	item	1.00	790
			TOTAL	**188184**