THERMAL ENERGY STORAGE DESIGN FOR EMERGENCY COOLING

by

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Abstract

Emergency cooling systems are applied to any application where the loss of cooling results in damage to the product, loss of data, or equipment failure. Facilities using chilled water for cooling that experience an electrical power outage, even a small one, would cause the chiller to shut down for 20 minutes or more. If emergency cooling is not available, temperatures would continue to increase to dangerous levels, potentially damaging the facility. Examples of facilities that could be protected by having emergency cooling systems are data centers, hospitals, banks, control rooms, laboratories, clean rooms, and emergency shelters among others.

This project addresses the current lack of information and methods needed to correctly design emergency cooling systems. Three application uses were investigated for the possible benefits of having emergency cooling systems. The software TRNSYS was used to simulate five typical emergency cooling systems for each of the three applications. The characteristics and differences of the systems developed from the simulations were then analyzed and documented.

The five systems simulated include a pressurized chilled water tank (parallel), atmospheric chilled water tank (parallel and series), low temperature chilled water tank (parallel), and ice storage tank (series). Simulations showed that low temperature chilled water tanks were less stratified than regular chilled water tanks by approximately 10%. Simulations also showed that the differences between atmospheric and pressurized tanks were negligible. Each tank discharged energy in the same manner and managed to replenish itself in the same amount of time. Examination of the different system configurations showed that tanks in series with the thermal load have issues with recharging due to its inability to isolate itself from the thermal load. It was also observed that while low temperature chilled water and ice storage tanks had the potential of reducing the storage tank volume, the amount of time ragged cooling will last is decreased by at least a factor of two.

The examination of the five systems produced the desired design methodologies needed to address the lack of information on emergency cooling systems. With the reported information designers can effectively engineer systems to meet their needs.

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I would also like to express my gratitude to the ASHRAE Technical Committee 6.9 for the aid and support supplied throughout the project.

Furthermore, I would also like to thank my fellow colleague, Walter Thomas Bembry IV, for the encouragement and company provided while working long hours together. Additionally, I would like to acknowledge my family for their steadfast support.

Lastly, I offer my regards to all of those who supported me in any respect during my research.

Preface

This thesis is submitted to partially fulfill the requirements of the thesis option of my master's degree at the Department of Mechanical Engineering, College of Engineering, Kansas State University. It represents a documentation of my research which was conducted from August 2008 until May 2010. The project "Thermal Energy Storage Design for Emergency Cooling, 1387-rp" was sponsored by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE).

The research conducted during this project will be incorporated into the ASHRAE handbook as seen fit by the sponsoring ASHRAE Technical Committee 6.9. The information then can be used to assist companies in the installation of thermal energy storage systems. The hope is to expand on the information currently available for thermal energy storage design and costs.

CHAPTER 1 - Introduction

Emergency cooling systems have taken on a more pertinent role in industry over the past decade. Several companies now rely on computer equipment to store large quantities of data that require tightly controlled environments. Ambient temperatures in data centers must be controlled to within specifications so that important information is not lost. The cost associated with such a loss in data is many times too much for a company to recover from.

A loss of power, even one of short duration, would cause a disruption in the cooling system. An interruption in cooling could lead directly to a substantial increase in ambient temperatures within a data center. The failure of equipment due to high temperatures would soon follow. An emergency cooling system would be able to provide the necessary cooling until the main chiller can be restarted or the data center is shut down.

Currently emergency cooling is usually provided by vapor compression refrigeration systems. This type of system has a major weakness in that electrically driven compressors must be operated which can be very problematic during a power loss. A backup power supply could be provided but the compressors consume a large amount of energy. A backup supply would have to be increased to an ungainly size and would be very expensive.

A fast growing method for emergency cooling is thermal energy storage (TES) tanks. Several companies have begun to store energy in a medium (water, ice, air, etc.) until needed. The main advantage to this method is that the power requirement is rather small during emergency conditions. The backup power supply is significantly reduced in size when compared to vapor compression refrigeration systems. TES tanks also offer flexible operating characteristics as both the temperature and delivery flow rate may be changed easily. The study of thermal energy storage tanks during emergency conditions is the main focus of this thesis.

The importance of having a TES tank was shown when Intel (2006) experienced a power outage. A 48,000 gallon TES tank storing chilled water was used to provide cooling to 20,000 servers at a major regional data center. After the power disruption occurred, the emergency cooling system immediately started pulling water from the TES tank. The emergency cooling system continued to remove heat from the servers over the next 15 minutes. The main chiller system was restarted by this time and able to retake the thermal load. This was a great example of an emergency cooling system protecting servers from damage and loss of data. The potential cost of losing the data center to high temperatures far exceeded the price of installing an emergency cooling system using a TES tank (Intel 2007).

ASHRAE's HVAC Handbook (2007) does not provide engineers with the proper amount of information needed to correctly design emergency cooling systems. The goal of ASHRAE in assigning this project was to bridge the current gap in information. The methods and design methodology for emergency cooling created will be used to help designers in the field.

The objective of this research was to develop practical design criteria for the design of emergency cooling systems. TES systems would then provide a cost effective way to supply emergency cooling. As a result of this research, emergency cooling systems will be designed better and operate more efficiently.

The first task was to research applications where emergency cooling is needed. From the research, three applications were selected (data centers, clean rooms, laboratories). A profile of the thermal load produced over a normal 24 hour day was built for each application and used for testing in the simulations. The three applications selected should be noticeably different to ensure the results are accurate over a wide assortment of applications.

Several styles of cooling were researched to locate the best available type in providing emergency cooling. New and established technologies were investigated on the potential use for emergency cooling. The approaches that were the most reliable, cost effective and practical for emergency cooling were chosen for further testing (Chilled Water, Ice). All other technologies either could not handle the large thermal loads or were not yet fully developed to be considered.

Research was then conducted into the different emergency cooling systems. Five diverse cooling systems that utilize a TES tank were then designed to simulate real life examples. In the five systems modeled, two of them use atmospheric chilled water tanks while a third makes use of a pressurized chilled water tank. The relationship between atmospheric and pressurized tanks when discharging was a closely monitored design methodology. The other two systems use a high energy density medium in a low temperature chilled water tank and an ice storage tank. The difference in how energy discharges from high energy density tanks and chilled water tanks was also closely followed. The five systems were either modeled in a parallel or series configuration.

The five emergency cooling systems were then modeled in a computer program (TRNSYS) to conduct simulations. The simulations were conducted using the three cooling load profiles over a normal 24 hour time period. Thirty minutes was picked for the amount of time emergency cooling must be provided by the TES tank because the main chiller should be restored to full operating capacity at the end of a half in hour. In the event power is still off after 30 minutes and the main chiller can not be restarted, the residual heat produced by the equipment should have dissipated enough to no longer be a threat.

Each of five emergency cooling systems was simulated over three different thermal loads (1, 5, 15 MW), chiller set point temperatures (40, 45, 50°F), and temperature differences between the supply and return lines (Δ 10, 15, 20°F). Each simulation was conducted three times using a different cooling load profile from the applications.

The results from the simulations were compiled and analyzed to come up with important design criteria that could be used in the industry. The effects the configuration of the system, changing system parameters, and types of TES tank had on the results were documented. These results formed the beginning of the design criteria.

The design criteria could only be formed after careful examination of all of the results. The goal is to hopefully develop enough design criteria to aid any engineer in designing/examining TES systems.

Research Plan

The following outline was used to fulfill the objectives of this thesis:

I. Identify Emergency Cooling Applications.

   a. Research cooling applications and design characteristics.

   b. Determine three emergency cooling applications including data centers for further study.

II. Identify available types of emergency cooling when power supply is disrupted.

   a. Compare and evaluate potential methods.

   b. Select most practical methods for emergency cooling.

III. Design emergency cooling systems based on methods selected.

IV. Develop design guidelines for each method when applied to emergency cooling.

   a. Simulate the five selected systems over a wide range of system parameters.

   b. Develop design methodology for emergency cooling using thermal energy storage for the three cooling load applications.

CHAPTER 2 - Emergency Cooling Applications and Types

This chapter deals with the emergency cooling applications and the types of cooling used for emergency cooling systems. For accurate results, comparable to real life situations, the thermal load can not be set at a constant value. The thermal load must be modeled after practical applications that have the need for emergency cooling. The variations in thermal loads of practical applications will provide the accuracy wanted.

There are currently several different types of systems that could be used to provide emergency cooling. Fully developed technologies along with new methods were considered and examined in full detail. The benefits and shortcomings of each type are listed farther below in the chapter.

Emergency Cooling Applications

Several applications that could potentially use emergency cooling were researched and studied. Hospitals, clean rooms, control rooms, emergency shelters, banks, data centers and others were analyzed and the three most pertinent applications were selected for a more detailed study. The three applications selected are data centers, clean rooms, and laboratories. The temperature guidelines and cooling load profiles were very different for each application. These differences will provide a wide array of results.

The other main characteristic studied in each application was the differences in thermal load during normal and emergency conditions. Some applications only need to provide cooling to critical areas when power is lost, so a drop in the thermal load may occur. If the cooling profile changes during emergency conditions for the application then it is explicitly stated. It will also be stated if there is no change in the profile during emergency conditions.

Graphs of the three cooling load profiles are shown below in each section. The 30 minute loss of cooling is assumed to take place at hour eight. So the emergency cooling system can be sized correctly, the maximum thermal load for each profile is assumed to happen at the same time. If the thermal load drops during emergency conditions for the application then another graph is given showing the change.

High Density Server Cabinet Data Centers

A data center is a facility used to house computer systems and associated components. The thermal load for data centers have progressively risen over the past couple years as shown by Figure 2-1. Today's businesses have become increasingly reliant on the health of their data centers. Modern data centers produce a very concentrated thermal load that requires a large cooling system to keep ambient temperatures from reaching high levels. A power outage, even one of short duration, could cause ambient temperatures to spiral out of control in a small amount of time. Temperature sensitive equipment could quickly be destroyed and valuable digital information lost. After a power outage that causes damage to a data center, only 10 percent businesses are still operating after two years and 50 percent never fully recover, according to a recent study (Opengate Data Systems 2009). An emergency cooling system that is correctly designed and operated ensures against loss of data.

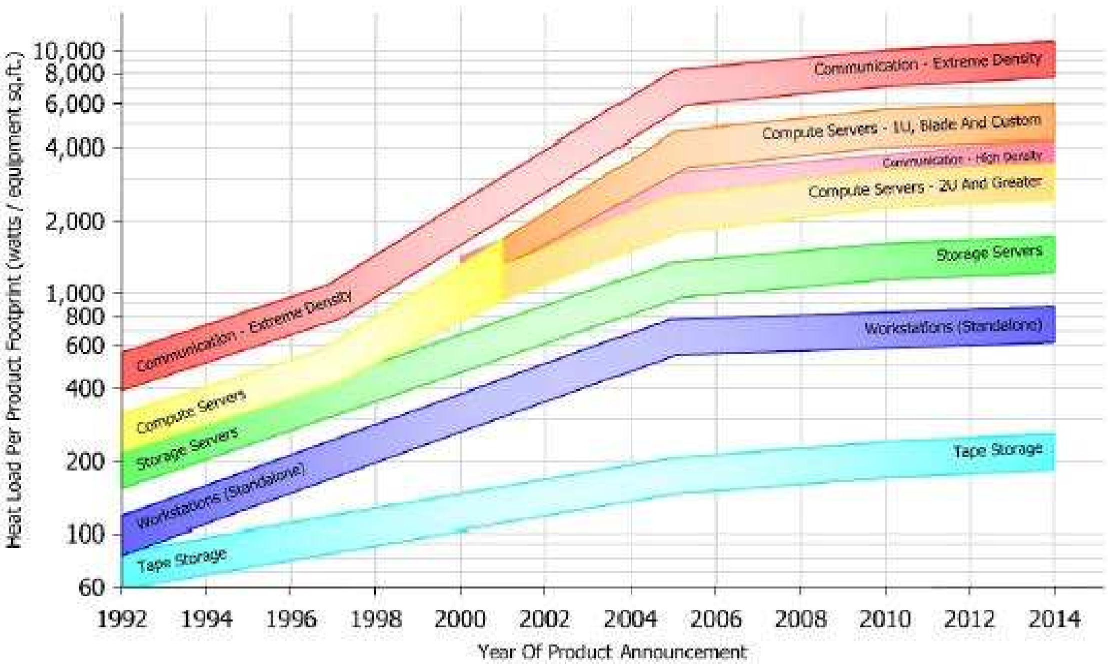

Figure 2-1 Heat Density Trend Projections (ASHRAE Datacom 2009)

Data centers are classified into different environmental classes. The main difference in the classes is how strict the environment is controlled. The strictest controlled class (Class 1) was chosen for this application because the tightly controlled environment (dew point, temperature, relative humidity) will be the hardest to maintain. Class one data centers typically support mission critical operations; enterprise servers, and storage products. The recommended temperature can be set with the class type chosen. The recommended temperature range for the class one data center is 18°C (64.4°F) to 27°C (80.6°F). The temperature range can be expanded to 15°C (59°F) through 32°C (89.6°F) if the allowable range is used. Data processing equipment cannot tolerate a large change in temperature so the ambient temperature must stay within ±2°C (±3.6°F) of the set point (ASHRAE Datacom 2009).

The effectiveness at which equipment racks are cooled is measured (cooling index) by ASHRAE. This study is on the overall system response so the equipment racks are assumed to be operating at 100% effectiveness for this report (ASHRAE Datacom 2009). Through research, an example of a hypothetical cooling load profile for data centers is shown below. The profile was based off a data center in Austin, TX using a thermal load of one megawatt as the maximum. Most data centers only run at 70-80% of the maximum thermal load (Fournier 2008).

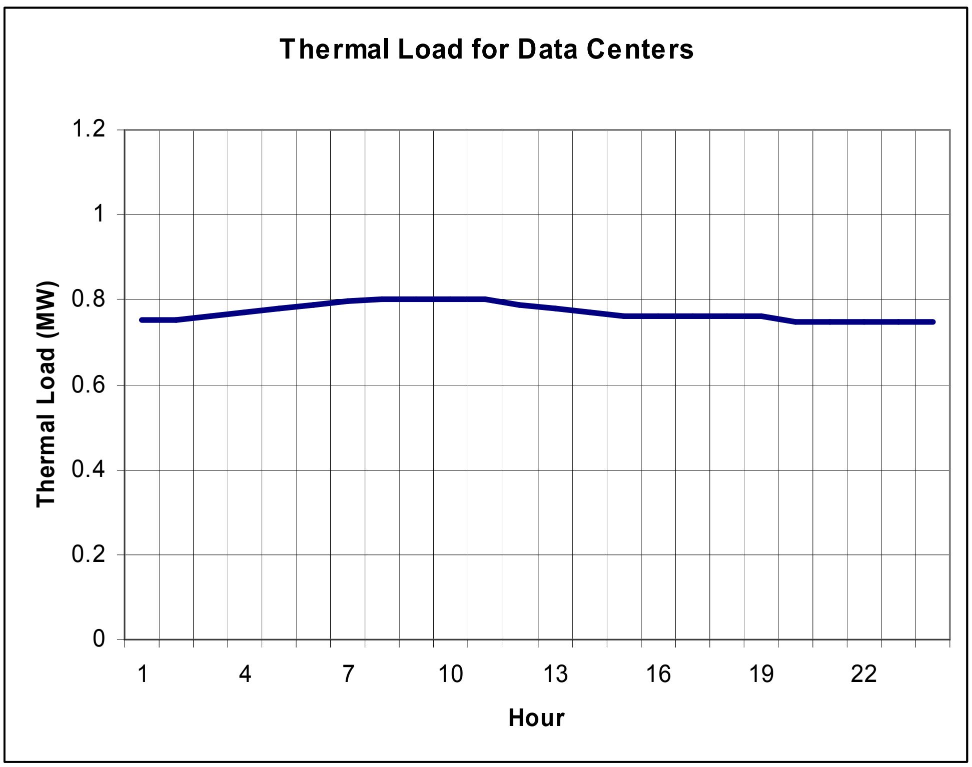

Figure 2-2 Cooling Load Profile for Data Centers

The thermal load was almost entirely flat for the 24 hour time period. The variation in the thermal load is less than 5% and only changes slightly in relation to the exterior environment. Based on research, there would be no drop in thermal load during a power outage (ASHRAE Datacom 2009). The profile would remain constant whether operating during normal or emergency conditions.

Manufacturing Clean Rooms

The call for more clean rooms is rising as technology advances and the need for clean work environments increases. Clean space environments are mainly used in manufacturing, packaging, and research for the following industries: biotechnology, microelectronics, aerospace, and miscellaneous applications (food processing, manufacture of artificial limbs, automotive paint booths, etc.). The environment needs to be strictly controlled for many of these industries or the products within the clean room could fail to meet the standards. A considerable amount of revenue could be lost if this happens.

The ambient temperature required is different for each of the manufacturing industries. Aerospace clean rooms require a temperature in the range of 22.8 ± 2.8°C (73± 5°F). A lower temperature may be used in the winter with a higher one in the summer. The microelectronics industry requires an ambient temperature to be at a constant 22.2 ± 0.28°C (72± 0.5°F). The biotechnology industry limits pharmaceutical products to a temperature range between 15°C (59°F) and 30°C (86°F) using the United States Pharmacopoeia guidelines (ASHRAE HVAC Applications 2007).

The process equipment and fans are the major internal thermal load components in clean room facilities. Almost all clean rooms are found entirely within conditioned space, so typical heat sources of infiltration, fenestration, and heat conductance from adjoining spaces are normally 2 to 3% of the total load (ASHRAE HVAC Applications 2007). The thermal load is assumed to be at a fairly fixed rate for the 24 hours since the majority of thermal energy is coming from the process equipment and fans. There would be a small amount of variation in the thermal load for the time period due to human occupancy throughout the day. There would be no reduction in cooling needs during emergency conditions because the majority of the thermal load comes from the equipment.

The biotechnology industry temperature guidelines were picked arbitrarily for this report. The recommended temperature range for this industry is 15°C (59°F) to 30°C (86°F). Most pharmaceutical products also do not tolerate a large change in temperature so the ambient temperature must stay within ±1°C (±1.8°F) of the set point (ASHRAE HVAC Applications 2007). The cooling load profile in Figure 2-3 was modeled from a pharmaceutical clean room for one megawatt.

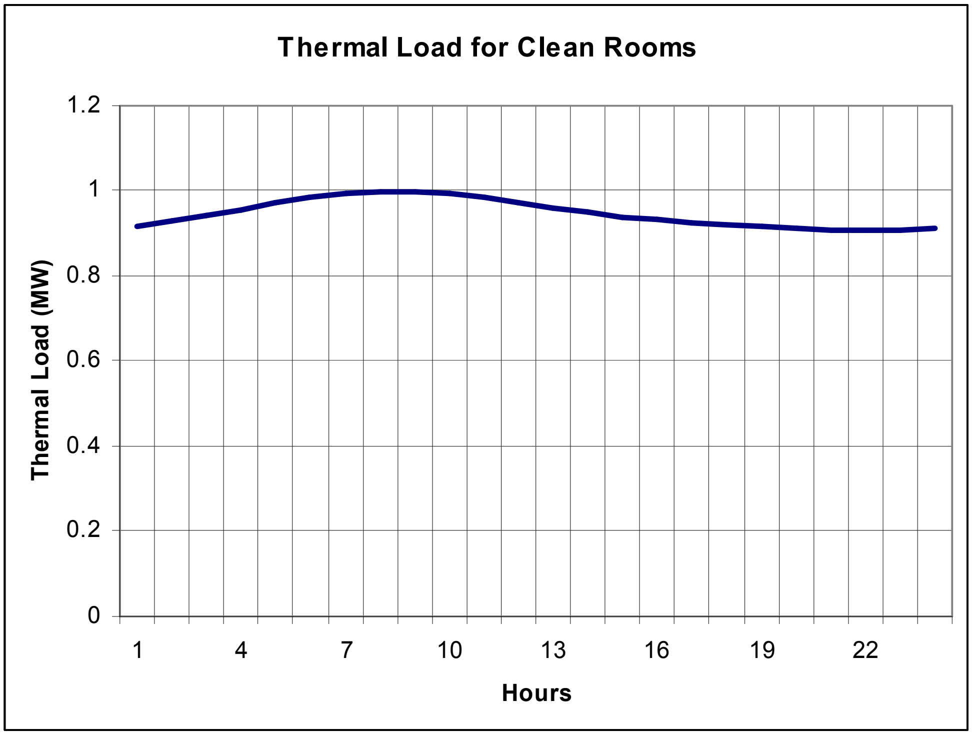

Figure 2-3 Cooling Load Profile for Clean Rooms

The above graph shows there is little change in the profile during the time period. The thermal load varies by less than 10% overall. The thermal load would be at its maximum during the middle of the day when occupancy is the highest. The load would decrease slightly as the room shuts down for the night. The profile would remain constant during normal and emergency conditions.

Laboratory and Research Facilities

Modern laboratories require regulated temperature, static pressure, humidity, and air cleanliness. Laboratories can be divided into four major categories: biological, chemical, animal, and physical. Each category requires its own set of environmental parameters (ASHRAE HVAC Applications 2007). A temperature increase due to a loss in power could cost millions of dollars in lost research. An example of this is shown in the next paragraph.

Seattle's Fred Hutchinson Cancer Research Center lost one of its three cooling systems to its freezer farm in 2008. The research center deals with the treatment of cancer, HIV/AIDS, and other life threatening diseases. The temperature in the freezer farm immediately began to rise and millions of dollars worth of research was put at risk. A chiller was rented from a nearby company and was installed in time before the freezer farm temperatures reached critical. The rented chiller lasted until main cooling could be restored (Engineered System 2008). Emergency cooling in this situation would have been very useful.

The temperature guideline for animal laboratories was picked arbitrarily to be used in this report. The temperature range for laboratories is from 17.8°C (64°F) to 29.5°C (85.1°F). Very strict control is required with a temperature of ±1.1°C (±2°F) of the set point. The cost to maintain a facility within such a strict requirement is extremely high and can be lessened by designing the building for selected species and their specific requirements. The cooling load profile for laboratories was created using ASHRAE's Laboratory Modeling Guide (ASHRAE HVAC Applications 2007) and a facility in Duluth, MN called Great Research Laboratory as a reference (Byrne 1999).

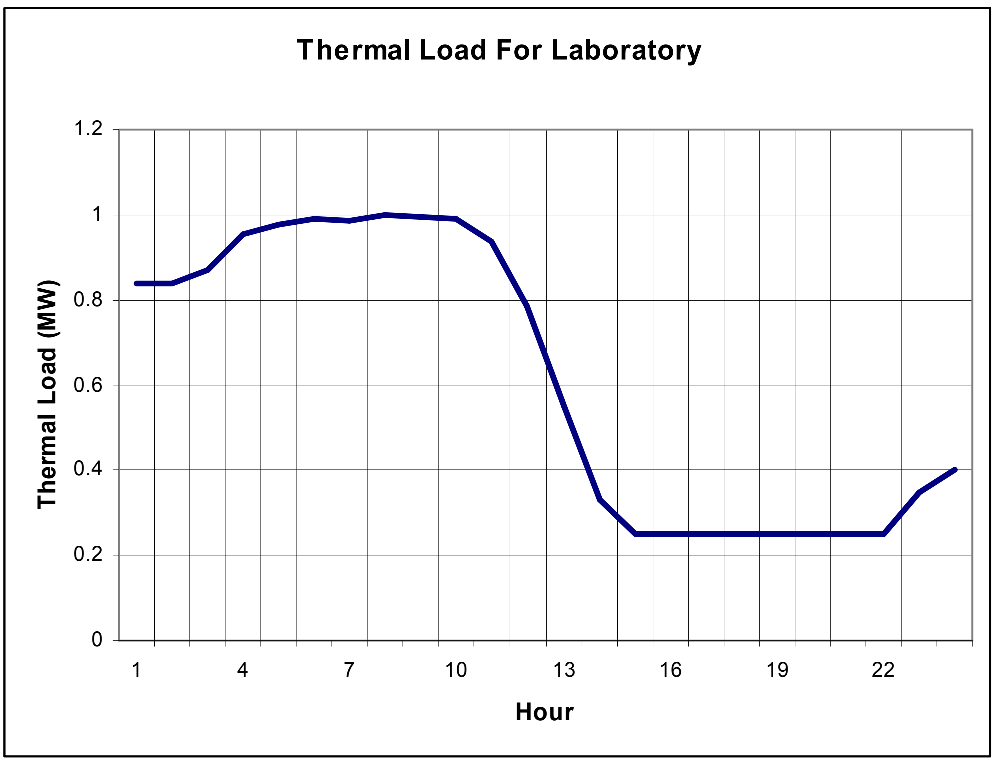

Figure 2-4 Cooling Load Profile for Laboratories

The above graph shows the large amount of variation in the thermal load over the time period. The thermal load reached its maximum during the middle of the day when laboratory occupancy is the fullest and the majority of equipment is running. The thermal load rapidly decreases as the facility shuts down for the night.

In emergency situations, only critical areas would receive the initial emergency cooling. This would initially result in about a 20% reduction in the thermal load during emergency conditions. Temperatures in other areas would begin to rise and reach a critical level. At this point, a portion of the emergency cooling would have to be diverted to the other areas. The longer emergency conditions last, the larger the thermal load becomes. The reaction of the cooling load profile during emergency conditions is shown in Figure 2-5. It shows that at hour eight when emergency conditions start how the thermal load decreases by about 20% and slowly rises during the ½ hour. The thermal load then rises to normal after the emergency situation is over (Byrne 1999).

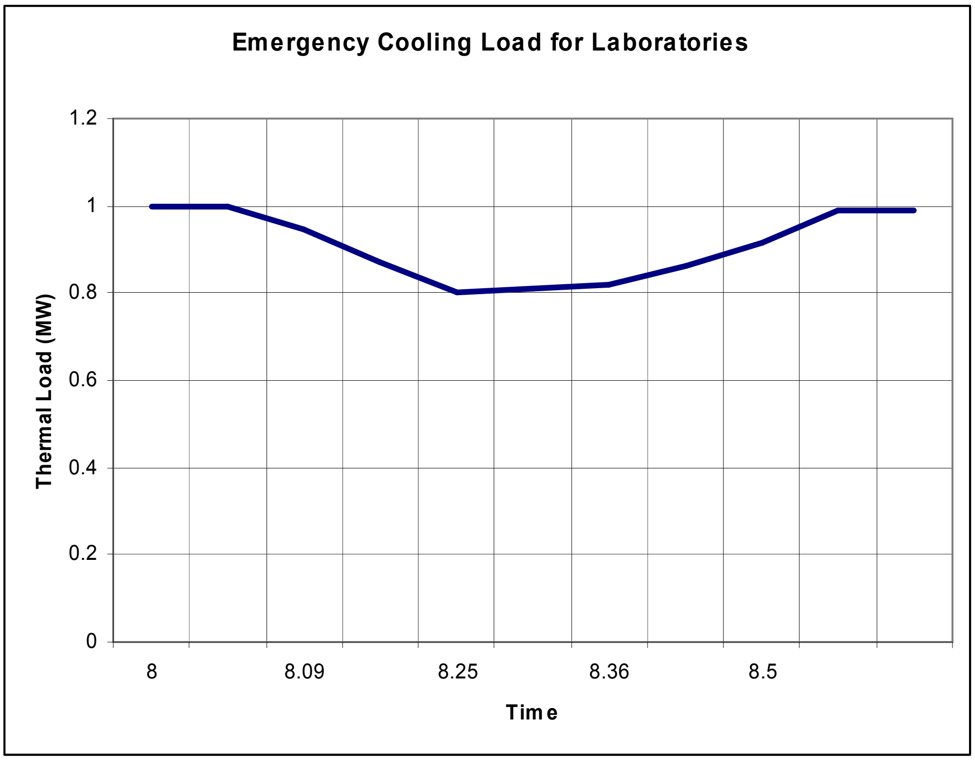

Figure 2-5 Emergency Load Profile for Laboratories

Emergency Cooling Types

Multiple types of systems were examined for emergency cooling. The types that were considered are listed below. Several of the types have been used for many years while others are just getting started and may not be fully developed yet. The type of cooling must be able to handle a wide range of thermal loads and be cost effective to install. The reliability and feasibility of the type to provide emergency cooling must also be very high.

Compressed Air

Compressed air stored in standard cylinders maintained at full capacity by an onsite compressor was considered. A company (Active Power) has recently been making huge strides in this area. This option has several drawbacks though. A high cost per unit storage and the amount of cooling storage is limited to 100 kW (Active Power 2008). This option was not seriously considered because of these drawbacks.

Chilled Water

Chilled water stored in thermal energy storage (TES) tanks is widely used in companies today because of the low cost and high availability. The size of the tank can vary greatly depending on the amount of cooling needed and the difference between the chilled water supply and return temperatures. A temperature difference of 5.56°C (10°F) between the supply and return lines is consider normal. A temperature difference as high as 11.11°C (20°F) can be used but is considered very aggressive. Almost all chilled water storage tanks are stratified at a temperature of 4.45 to 7.22°C (40 to 45°F). Typically chilled water storage is the most economical when the storage is more than 7000 kWh (2000 ton-hrs) (Dorgan & Elleson 1993).

Chilled water temperatures below 4.45ºC (40ºF) are referred to as low temperature chilled water. The lower limit on regular chilled water temperature, which is approximately 4.1ºC (39.4°F), is dictated by the temperature at which the maximum density of pure water occurs. A special additive of consisting of sodium nitrate and sodium nitrite must be added to the chilled water to maintain stratification effects due to density issues (Andrepont 1999).

Ice

Ice storage is also a commonly used emergency cooling system due to its high energy density storage capacity. The latent heat of fusion from ice to water is 334.4 kJ/kg (143.5 Btu/lb). There are several different methods for ice creation within storage tanks but this paper only covered the two most common processes. The two processes studied were internal and external ice melt on coil systems (Moran & Shapiro 2004). The two figures below show how each system works.

Ice storage tanks that are external melt are non-pressurized and have exit temperatures of 1.1-2.2°C (34-36°F) when discharging. Internal melt systems use a coolant loop to charge and discharge the heat transfer fluid. The discharge temperature is usually between 2.2-3.3°C (36-38°F) (Dorgan & Elleson 1993). The cost of ice storage is also relatively low.

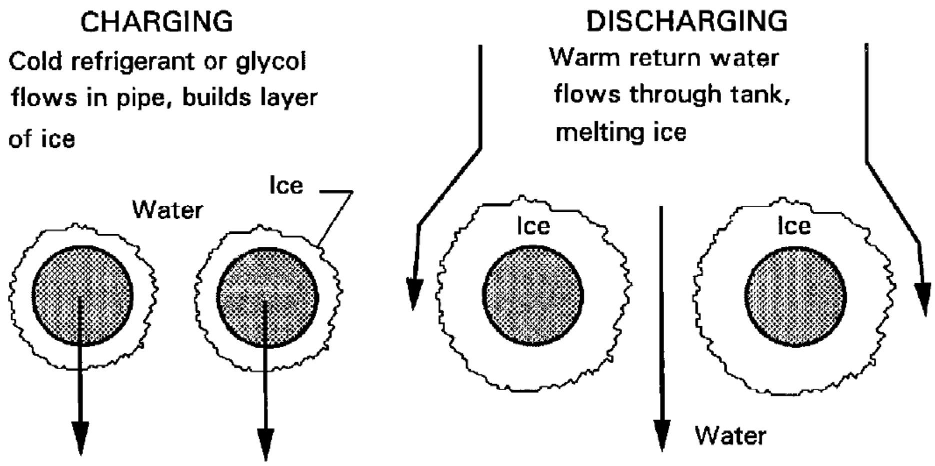

Figure 2-6 External Ice Melt

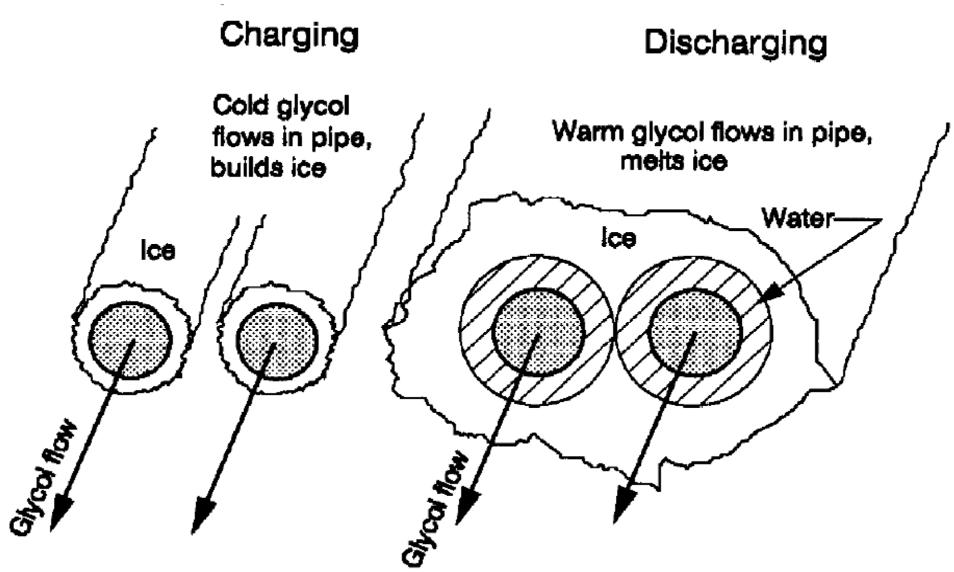

Figure 2-7 Internal Ice Melt

Eutectics

Eutectics are a mixture of inorganic salts, water, and nucleating agents used to stabilize the melting point of the fluid medium. Plastic storage containers (Figure 2-8) are stacked on top of each other where the inorganic salt material is encapsulated. Inorganic salts have a freezing point of 8.2°C (47°F). The tank discharge temperature range is from 8.9-10°C (48-50°F) which is above normal for HVAC applications. Eutectics can also be corrosive and are mostly used in heat storage applications (Dorgan & Elleson 1993).

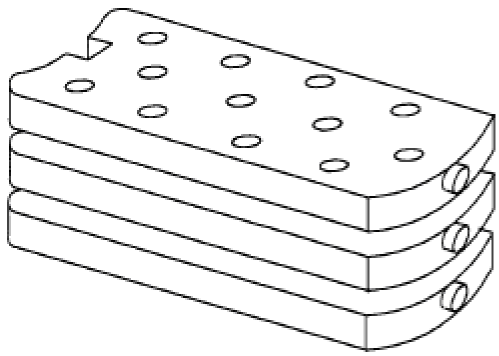

Figure 2-8 Eutectic Salt Storage Containers

Selected Emergency Cooling Systems

Using the information above, chilled water and ice storage were selected to be used for thermal storage for the systems designed in chapter three. These two systems cost the least and are the most commonly used systems based on the research above and industrial contacts. Several examples of these systems are found in companies today. The other two systems (Eutectics & compressed air) were either not cost effective or not practical for large thermal loads.

CHAPTER 3 - Emergency Cooling Water Systems

The five emergency cooling systems designed for further study are explained in great detail in this chapter. Several systems are found in industry with similar setups, as the examples below show. Simulations will then be conducted using these five systems and the three cooling applications discussed in chapter two.

The configuration, tank type, and setup of each system were carefully chosen and designed so a comparison could easily be made between the systems. The differences in the systems were closely monitored during simulations to locate effects they may have. How each system operates during normal, emergency, and recharging operations is explained in detail below.

Parallel Pressurized Chilled Water Tank

The first system uses a pressurized chilled water storage tank in parallel with the thermal load. Pressurized tanks are not cost effective or practical for large systems so this system was only tested over the low thermal loads.

The system shown in Figure 3-1 uses a primary and secondary loop to control the flow to the thermal load. The main advantage of having the TES tank in parallel is that the tank can be isolated from the thermal load and replenished easily.

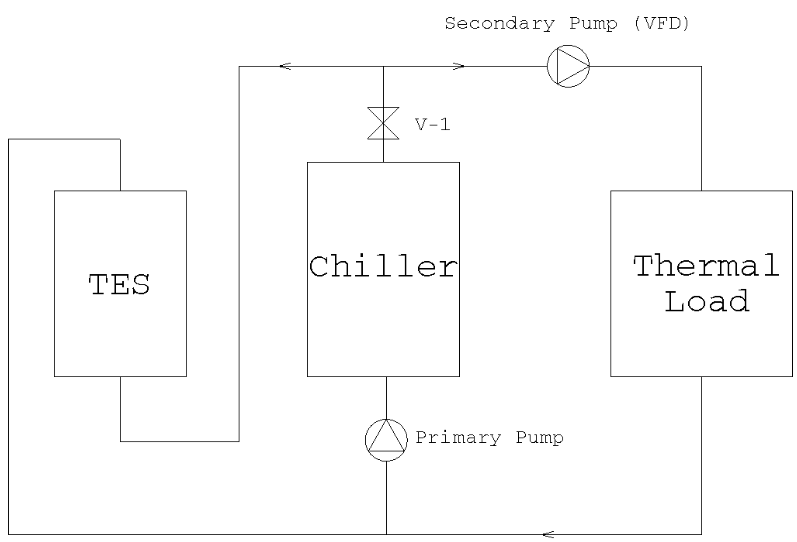

Figure 3-1 Parallel Pressurized Chilled Water Tank

<u>Normal Operation:</u> The TES tank is fully charged while V-1 is open. The chiller operates at capacity that may or may not meet the thermal load. The secondary pump provides flow that

gives the desired temperature change across the thermal load. The primary pump operates at a constant flow.

Emergency Cooling: The valve V-1 is closed during an emergency cooling situation. The secondary pump provides sufficient flow to meet the temperature difference across the thermal load by drawing chilled water from the TES tank. The secondary pump is powered by a backup power system. The primary pump would remain off until the chiller is restarted.

Recharge TES: When the chiller capacity exceeds the thermal load, the excess chilled water will flow into the TES tank. When total volume flowed equals the tank volume, the TES has been recharged. The time to recharge will be a function of the excess chiller capacity and TES volume.

Parallel Atmospheric Chilled Water Tank

The second system uses an atmospheric chilled water storage tank in parallel with the thermal load. Atmospheric tanks are the most widely seen TES tank in the industry (compared to pressurized tanks). The setup shown in Figure 3-2 is commonly seen in the field today with the TES tank in parallel with the thermal load.

A primary and secondary loop is utilized to control the flow to the thermal load just like the system above with the pressurized TES tank. The only difference between the first system and this one is that this TES tank is atmospheric and the previous one is pressurized. A comparison between pressurized and atmospheric TES tanks using results from these two systems can be made.

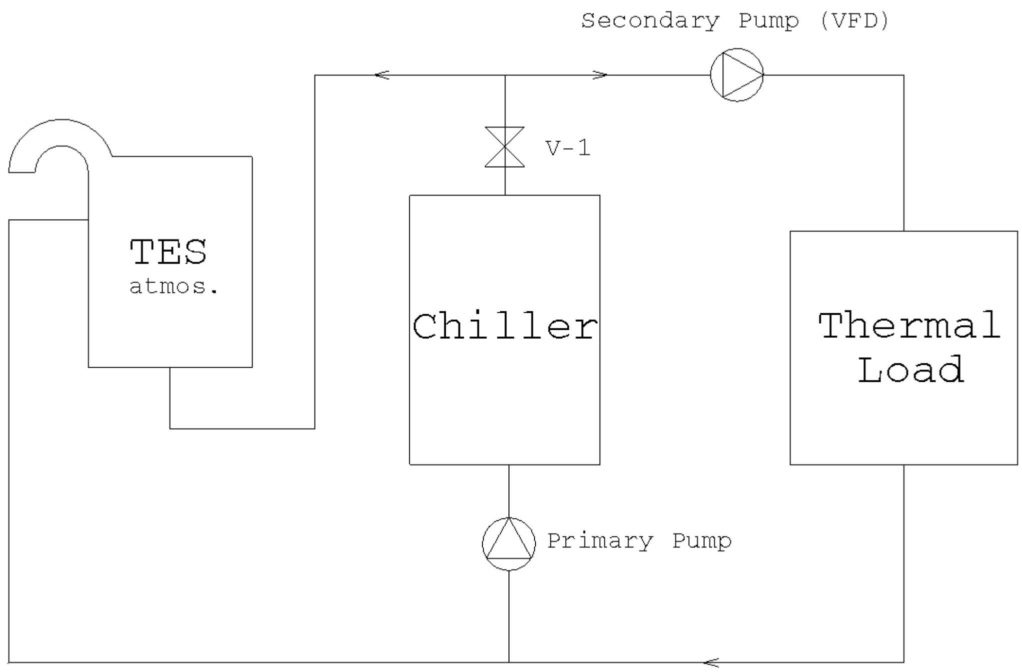

Figure 3-2 Parallel Atmospheric Chilled Water Tank

Normal Operation: The TES tank is fully charged while V-1 is open. The chiller operates at capacity that may or may not meet the thermal load. The secondary pump provides flow that gives the desired temperature change across the thermal load. The primary pump operates at a constant flow.

Emergency Cooling: Valve V-1 is closed during an emergency cooling situation with the primary pump and chiller shut off. The secondary pump provides sufficient flow to meet the temperature difference across the thermal load by drawing chilled water from the TES tank. The secondary pump is powered by a backup power system.

Recharge TES: When the chiller capacity exceeds the thermal load, the excess chilled water will flow into the TES tank. When total volume flowed equals the tank volume, the TES has been recharged. The time to recharge will be a function of the excess chiller capacity and TES volume.

Series Atmospheric Chilled Water Tank

The third system contains an atmospheric chilled tank in series with the thermal load. The chilled water tank in series was not a very common setup seen in industry today. The differences in the design methodologies of parallel and series configuration will provide a great deal of insight to designers.

The system in Figure 3-3 uses one variable speed pump to control the flow to the thermal load and no bypass line is present. The primary pump (VFD) in this configuration can not drop below the minimum flow requirements for the chiller. One main disadvantage of this setup is that the TES tank, chiller, and thermal load can not be isolated from each other. This will lead to problems with recharging the TES tank which will be demonstrated in chapter four.

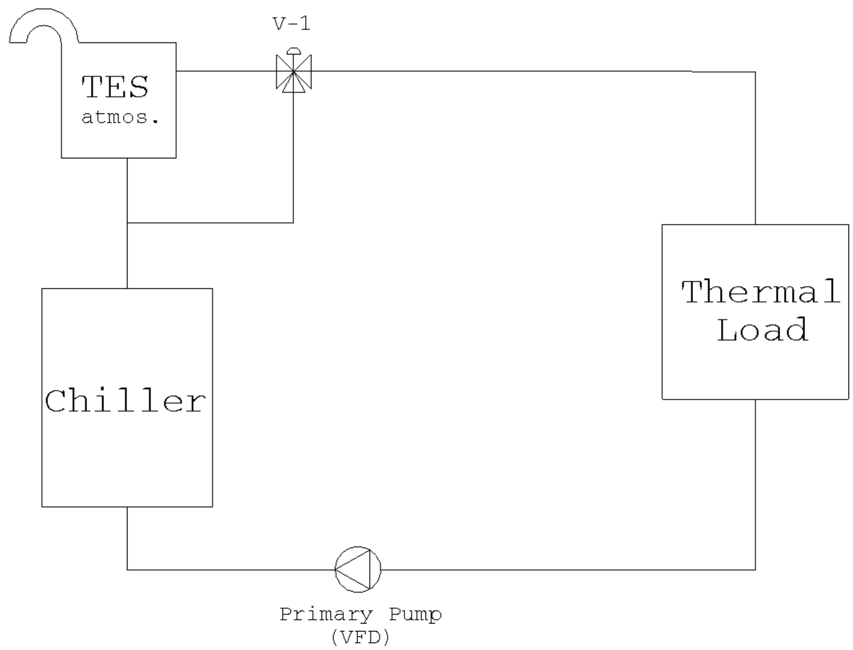

Figure 3-3 Series Atmospheric Chilled Water Tank

Normal Operation: The primary pump regulates the flow to maintain the desired change in temperature across the thermal load. The primary pump reduces flow to sustain the desired change in temperature as the load decreases. V-1 is allowing all of the flow to bypass the TES tank during normal operation.

Emergency Cooling: The primary pump operates at a flow such that the desired change in temperature is maintained at the load and V-1 is diverting all flow through the TES tank. Chilled water flows from the atmospheric TES tank to the thermal load. The chiller is powered down during this operation.

Recharge TES: When the chiller is operating at a capacity greater than the thermal load, V-1 modulates the flow to partially allow the tank to recharge. Once the entire flow equals the tank volume then it is considered fully charge.

Parallel Low Temperature Chilled Water Tank

Several systems currently employ a TES tank of low temperature chilled water (<40°F) used for emergency conditions with regular chilled water (>40°F) utilized for normal conditions. The main advantages of such systems are that the TES tank is considerably smaller and an alternative cooling source is attached to the tank. The system in Figure 3-4 utilizes a low temperature chilled water tank in parallel configuration. An additive has been added to the tank to ensure the proper stratification levels occur.

If the additive was not added to the low temperature chilled water tank, the water with the highest density would fall to the bottom of the tank. While discharging, the incoming hot water would mix with the stored chilled water. When the mixed water reached the temperature of highest density (39.4°F), it would fall to the bottom of the tank. The chilled water exiting the tank would then be at 4.1°C (39.4°F). The modulating valve in the low temperature chilled water system (V-2 in Figure 3-4) would be able to account for this and allow for more flow through the TES tank in emergency conditions.

A primary and secondary loop is used to control the flow just like in the previous parallel configurations. The refrigeration unit attached to this system will be able to cool the water in the TES tank regardless of the main chiller. The differences in design methodology between regular and low temperature chilled water TES systems learned here will provide engineers with the information necessary to make the correct decisions.

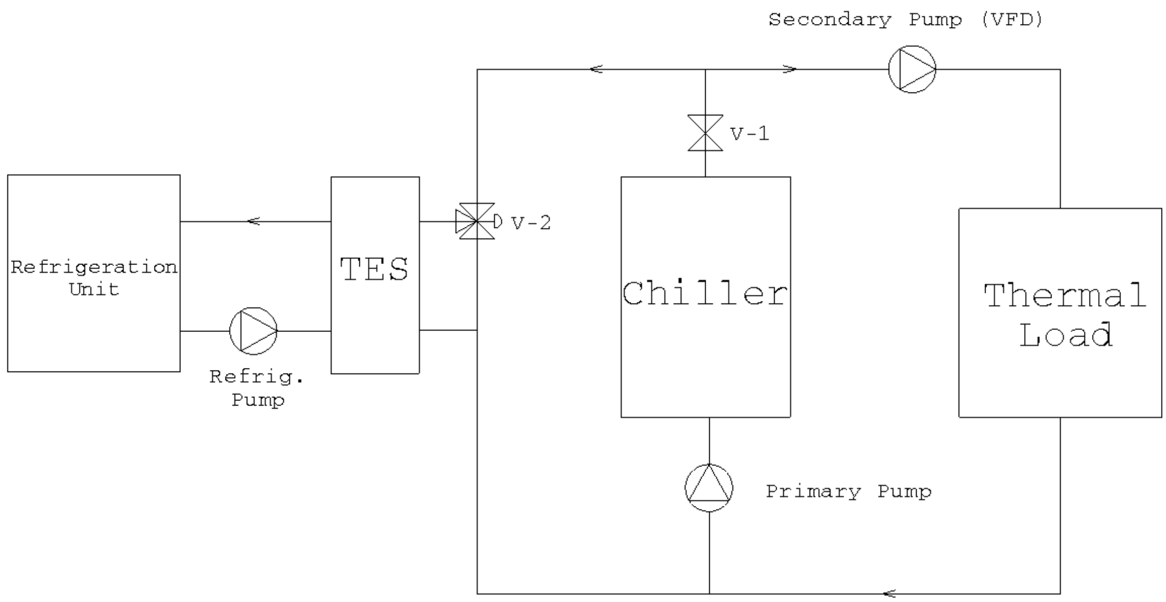

Figure 3-4 Parallel Low Temperature Chilled Water Tank

Normal Operation: During this operation V-1 remains open to allow for chilled water to reach the thermal load. The chiller operates at full capacity while the primary pump operates at a constant flow and the secondary pump (VFD) provides the appropriate amount to flow to maintain the desired temperature difference across the thermal load. As the load reduces, excess flow from the chiller goes through V-2 returning to the chiller return line.

Emergency Cooling: For this operation V-1 will remain closed and the primary pump is shut off. The secondary pump provides sufficient flow to maintain the temperature difference across the thermal load. The chilled water will be drawn from the TES tank by modulating V-2.

Recharge TES: Normal operation will apply for main chiller system in that V-2 allows the excess chilled water to bypass the TES tank. The refrigeration unit would operate to lower the TES temperature to 1.7°C (35°F). The refrigeration unit contains a DX HX to lower the TES temperature. The time to recharge is a function of the excess chiller capacity, refrigeration capacity, and TES volume.

Series Ice Storage Tank

The final storage system in Figure 3-5 uses the high energy density of an ice storage tank in series. Several different ice storage tanks are found in the field today. The main advantages to ice storage are the smaller size of the tank and the alternative cooling source attached to the system. The alternative cooling source (Glycol Refrigeration Unit) allows for the ice storage tank to be recharged at all times. External ice melt coils were chosen instead of internal ice melt coils system. Research found that internal ice systems were unable to discharge the energy held in the ice fast enough for emergency cooling. External ice melt coils did not have this problem due to the water coming into direct contact with the ice on the coil.

The flow to the thermal load is controlled by a variable speed pump just like in the previous series configuration. The primary pump (VFD) in this configuration can not drop below the minimum flow requirements for the chiller. A glycol refrigeration unit is attached to the ice builder tank to replenish the ice tank. The disparity between ice storage and chilled water TES systems will be useful in helping designers choose between the systems.

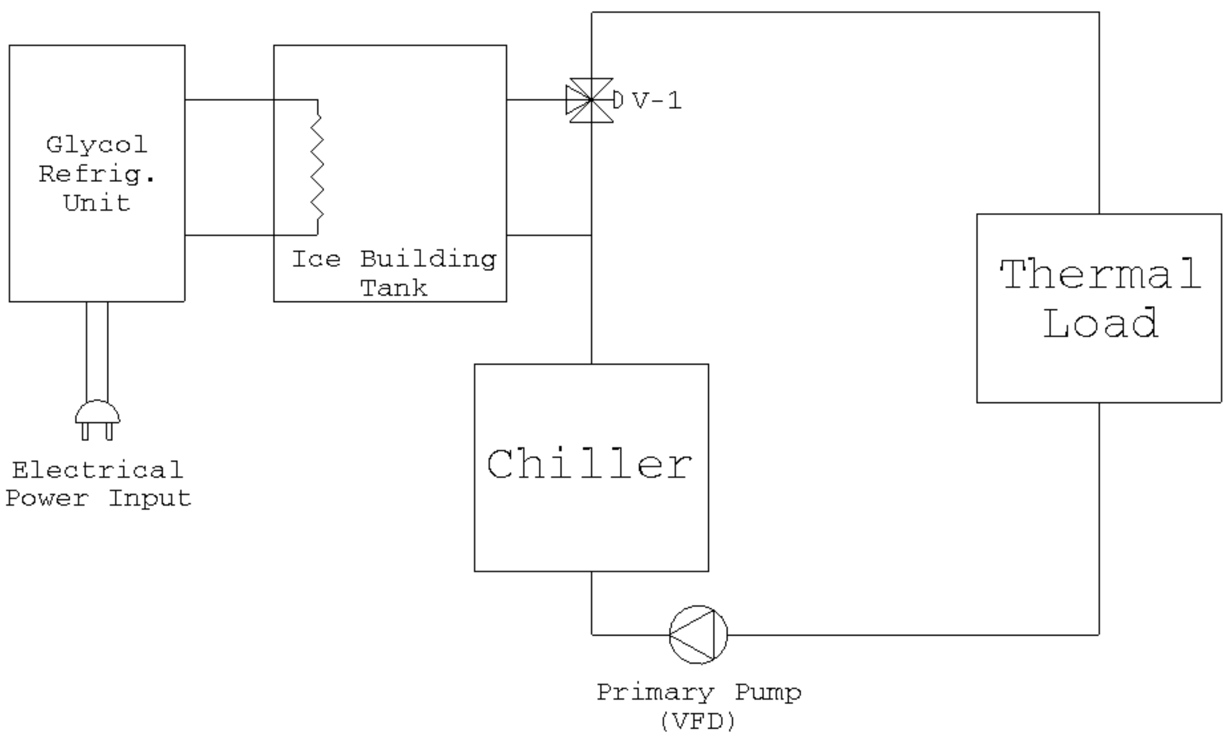

Figure 3-5 Series Ice Storage Tank

Normal Operation: During this mode of operation the ice builder tank is fully charged while, V-1 diverts the flow to bypass the ice tank. The primary VFD pump provides the flow to meet the desired temperature difference across the thermal load. As the load decreases, the secondary pump reduces flow to match the change in temperatures across the thermal load.

Emergency Cooling: For this mode of operation, V-1 modulates the flow so a portion enters the ice tank while the rest bypasses the ice tank. The primary pump provides flow to the load using backup power, to maintain the desired temperature difference across the thermal load. The chiller remains off in this mode of operation.

Recharge TES: The ice tank will recharge using the glycol refrigeration unit. The recharge time is dependent upon the size of the glycol refrigeration unit and storage tank effectiveness.

CHAPTER 4 - Testing and Data Collection

The five different storage systems described in chapter three were simulated at three different thermal loads (1, 5, & 15MW) at a chiller set point temperature of 7.22°C (45°F) and at three different temperature differences (Δ 10, 15, & 20°F) across the cooling coils. Then without changing any of the component sizes the systems were ran at chiller set point temperatures of 4.45°C (40°F) and 10°C (50°F) to see if this had any effect. The only exception was the system that used a pressurized chilled water tank was simulated at only one and five megawatts. A thermal load of fifteen megawatts was too large to realistically simulate a pressurized tank. The matrix bellows shows how many different simulations were done for one of the storage systems.

	1 MW		
Storage 8			
	Chilled Temp.		
Delta T	40F	45F	50F
10F			
15F			
20F			

	5 MW		
Storage 8			
	Chilled Temp.		
Delta T	40F	45F	50F
10F			
15F			
20F			

	15 MW		
Storage 8			
	Chilled Temp.		
Delta T	40F	45F	50F
10F			
15F			
20F			

Figure 4-1 Simulations

TRNSYS Simulation Program

The simulations for this thesis were carried out by a Fortran based program called Transient System Simulation program (TRNSYS). TRNSYS is a total and extensible simulation environment used for the transient simulation of energy systems. It is used by researchers and engineers throughout industry to authenticate energy concepts. Some of the many applications studied with TRNSYS are solar systems, low energy buildings, HVAC systems, renewable energy system, and cogeneration.

One of main features of TRNSYS is its ability to quickly and easily alter component models (chillers, pumps, etc.) within the program. An existing model can simply be changed to fit the user's specific needs. In addition, other applications (e.g. Microsoft Excel, Matlab, etc.) can be connected to TRNSYS for pre-processing and post-processing of the simulation.

Figure 4-2 is a simplified system shown in TRNSYS Simulation Studio that will be used to better explain how the program operates and conducts simulations. It is not representative of any storage system. The core of TRNSYS is in its ability to link the inputs and outputs of different components together in the Simulation Studio thus creating a system. The Simulation

Studio is the main visual interface where systems can be created by putting components into the workspace, connecting them together, and setting the parameters for the system.

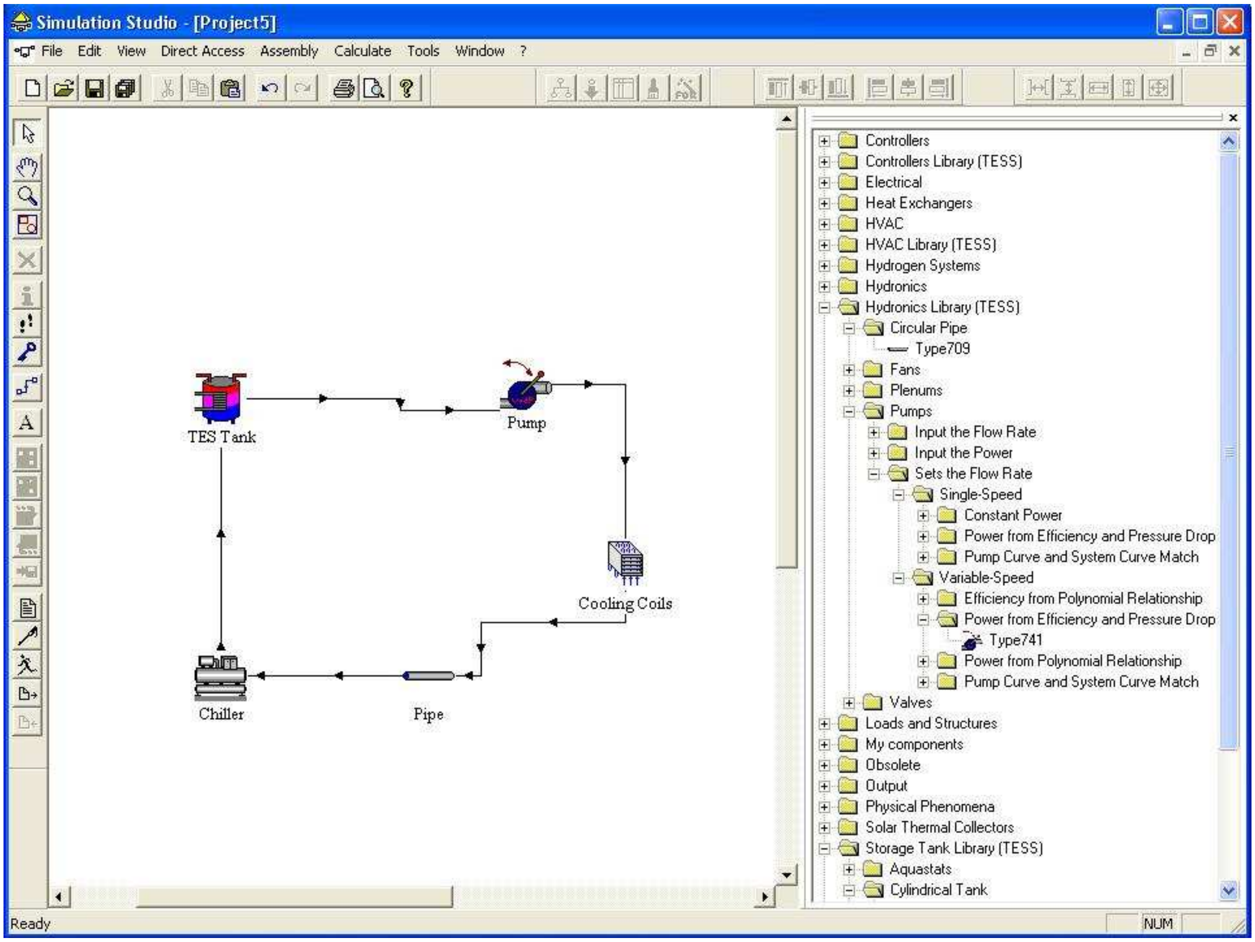

Figure 4-2 TRNSYS Simulation Studio

Each component is described by a mathematical model in the TRNSYS simulation engine and has a set of matching Proforma's in the Simulation. The proforma is basically a black-box description of the component: inputs, outputs, and parameters. Figure 4-3 shows an example of a proforma opened up for a pipe. Every component contains its own special proforma that is connected to its mathematical model. The links between the components (black lines with arrows) can be opened up and a window appears where the outputs (flow rate, temperature, etc.) of one component can be graphically linked into the inputs of the next desired component.

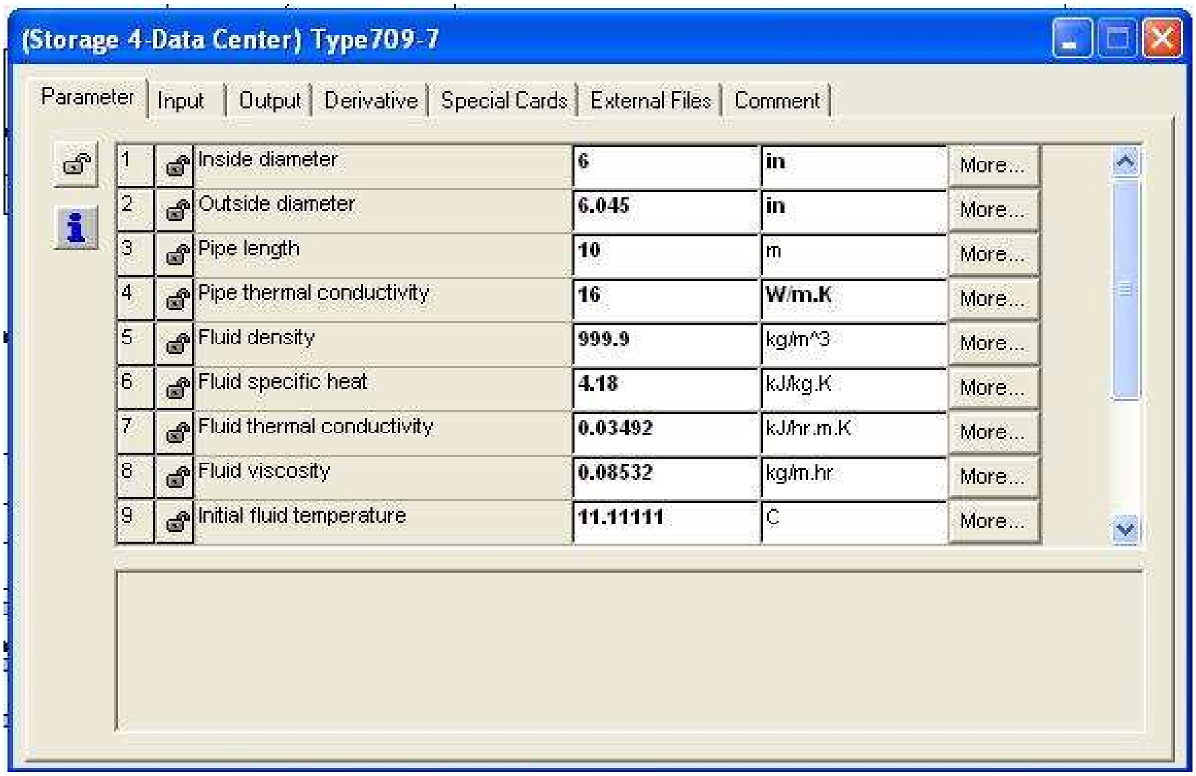

Figure 4-3 Proforma for a Pipe component

The simulation engine in TRNSYS is programmed in Fortran. The engine will read all of the information from the Simulation Studio (components, parameters, etc.) and take the Fortran based mathematical model of each component to run a simulation. The simulation is divided up into equal time steps. At the end of each time step, the engine will take the outputs from the components and pass them onto the inputs of next link components. The process is then repeated again and again. The user sets how big the time step will be for the simulation. This is a major decision. If the time step is too small, the engine may not have enough time to perform necessary calculations thus causing the simulation to move sluggishly. If the time step is too big, the simulation will not accurately represent real life systems because information is not moving between components fast enough. After careful consideration and some experimentation, the time step was set to 15 seconds for all simulations. This gave the engine enough time to make the necessary calculations but was fast enough to accurately model real systems.

At the start of a simulation, the engine will first pull all of the component proformas from Simulation Studio and put the values into the mathematical model. The mathematical model will then process the values from the components to produce the outputs using the equations present. Many of components contain simple equations which directly produce an output from the input but several of components require iterative solutions. The TES tank for example contains a differential equation that must be processed and several iterations have to be done before an output can be produced. At the end of the time step, the outputs from one component are passed along to the inputs of the next linked component. For example, the TES tank outlet mass flow rate is passed into the pump inlet mass flow rate. This process is repeated again and again until the end of the simulation.

If the values for a component do not converge before the end of the time step, then TRNSYS will produce an error and stop the simulation. This usually means that a component was setup wrong and needs to be fixed before proceeding. This sort of error stops users from designing systems that are unreasonable.

For all of the simulations, data was taken at every time step and recorded in a excel file for further examination. A quick analysis of the data could then be conducted and the pertinent graphs created.

A screenshot of the five systems from chapter three created in the TRNSYS Simulation Studio can be seen in Appendix A. The level of complexity taken into account by the program is immense. For example, the outside environmental conditions had to be specified for components to be correctly simulated. TRNSYS allowed us to change the outside environmental conditions as the day progressed. Outside air temperatures, dew point, and pressure could all change within the 24 hour time period (A normal spring day was used for our simulations). This was just one of the many extra things that TRNSYS did to more accurately model a system.

The following sections deal with how the different components were modeled in TRNSYS. Only the main components that deal with the thermal energy transfer will be discussed (chiller, cooling coils, TES tank, etc.). Several additional components (Tee, Valves, etc.) were needed to correctly model the systems but are not talked about due to their simple nature. These components are used to divert flow and make use of basic mass balance equations. The relevant equations that the mathematical model of the component utilizes are shown and discussed.

Chiller

The chiller in the simulations was modeled after a vapor compression water cooled centrifugal chiller. This device cools a fluid stream on the evaporator side while rejecting heat to another fluid stream on the condenser side. This component model relies on a catalog data lookup method to correctly model the chiller. The user must provide performance data for the chiller to lookup, the rated capacity, and the rated COP. For the simulations, performance data was collected from a chiller produced by the TRANE company along with the rated capacity and COP.

The chilled load must first be calculated,

$$\dot{Q}_{Load} = \dot{m}_{chw} \times Cp_{chw}(T_{chw,in} - T_{chw,set})$$

The load met (Q_{met}) by the chiller is automatically limited by the capacity of the machine specified in the parameters. The part load ratio is therefore,

$$PLR = \frac{\dot{Q}_{Load}}{Capacity}$$

The Fraction of Full Load Power (FFLP) can now be looked up in the catalog data by the program and used the chiller's power draw calculation. The COP is specified by the user in the parameters.

$$P = \frac{Capacity}{COP} FFLP$$

The amount of heat energy rejected to the cooling tower fluid stream is therefore,

$$\dot{Q}_{REJECTED} = \dot{Q}_{Met} + P$$

The outlet chilled water temperature and cooling tower water temperature can now be calculated.

$$T_{chw,out} = T_{chw,in} - \frac{\dot{Q}_{met}}{\dot{m}_{chw} Cp_{chw}}$$

$$T_{cw,out} = T_{cw,in} - \frac{\dot{Q}_{rejected}}{\dot{m}_{cw} Cp_{cw}}$$

Pipe

This component models the thermal behavior of fluid flow in a circular pipe. The pipe will have no effect on the flow rate. The only thing really of interest is how much thermal energy is lost to the environment from the pipe. The user must provide the pipe's diameter, length, and thermal conductivity along with the insulation thickness and thermal conductivity. The method the mathematical model used for calculations is shown below.

The user was asked to supply the physical parameters of the pipe in proforma. These will be used to calculate the energy loss. The first equation that is needed to be calculated is the overall loss coefficient per unit area for the pipe,

$$UA = \frac{1}{R_{inside} + R_{pipe} + R_{insul} + R_{outside}}$$

The resistance equations are,

$$R_{inside} = \frac{1}{h_{inside} A_{inside}} \qquad R_{pipe} = \frac{\ln \frac{d_{pipe,o}}{d_{pipe,i}}}{2\pi\ k_{pipe}\ L_{pipe}} \qquad R_{pipe} = \frac{\ln \frac{d_{insul,o}}{d_{pipe,o}}}{2\pi\ k_{insul}\ L_{pipe}} \qquad R_{outside} = \frac{1}{h_{outside} A_{outside}}$$

The total energy loss rate to the environment is then and the final outlet temperature of the water,

$$\dot{Q}_{env} = -(UA)(T_{chw,in} - T_{env})$$

$$T_{chw,out} = T_{chw,in} - \frac{\dot{Q}_{env}}{\dot{m}_{chw}\, Cp_{chw}}$$

TES Tank for Chilled Water

The TES tank used for chilled water in the simulations was a fluid-filled, constant volume cylindrical tank with a vertical configuration like in Figure 4-4. The tank is divided into 90 (n = 90) temperature nodes to model stratification. Research found that the TES tank will be approximately 90% stratified with the thermocline layer occupying 10% of the tank volume when discharging. Through experimentation and a few calculations, it was found that the TES tank with 90 nodes would correctly model this. The first node is considered at the top and node 90 at the bottom of the tank. Each constant volume node is assumed to interact thermally with the surrounding nodes by either fluid conduction or fluid movement. The user must provide the tank volume, height, and loss coefficients.

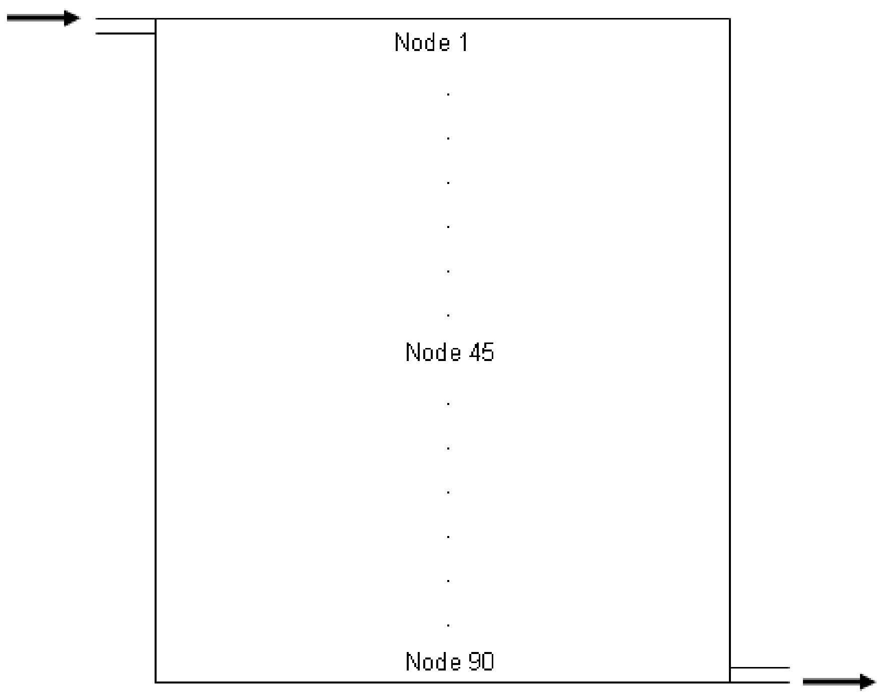

Figure 4-4 TES Tank

This storage tank model interacts thermally with the environment through the heat losses from the top, bottom, and side areas of the storage tank. The equations for the heat transfer from the top, bottom and side for tank node *n* are:

$$\dot{Q}_{loss,top} = (UA)_{top}(T_{n=1} - T_{env,top})$$

$$\dot{Q}_{loss,bottom} = (UA)_{bottom}(T_{n=90} - T_{env,bottom})$$

$$\dot{Q}_{loss,side,n} = (UA)_{side,n}(T_n - T_{env,side})$$

The tank calculations break down into one differential equation,

$$dT_{Tank} / dt = (Q_{in,Tank} - Q_{out,Tank}) / C_{Tank}$$

Where $Q_{in,Tank}$ and $Q_{out,Tank}$ are functions of the ambient temperature, the inlet fluid conditions, and flow rates.

TES tank for Ice Storage

The ice storage tank used by TRNSYS was an external ice-on-coil tank. The storage tank is modeled as a heat exchanger where the charge/discharge rate is a function of whether or not the system is charging or discharging and the log mean temperature difference between the brine and the storage freezing temperature (T_{fr}). The user must specify the storage capacity of the tank, and the energy capacity of the tank. The equations used by the mathematical model are shown below (Potter & King 1998).

The overall loss coefficient must first be calculated whether the tank is discharging or charging. The L_i are the coefficients of fifth order polynomial fit and are list in Table 4-1. The coefficient *y* is the fraction of ice discharged or charged depending on the current operation of the component.

$$UA_{ice} = (L_1 + L_2 \times y + L_3 \times y^2 + L_4 \times y^3 + L_5 \times y^4 + L_6 \times y^5) \times Q_{stored} \times \Delta t / \Delta T_{lm,nom}$$

Table 4-1 Ice Storage Coefficients

	IOC External	
Coefficient	***Charge***	***Discharge***
L_1	1.3879	1.1756
L_2	-7.633	-5.3689
L_3	26.3423	17.3602
L_4	-47.6084	-30.1077
L_5	41.8498	25.6387
L_6	-14.2948	-8.5102

For discharging, *y* is the fraction discharged and

$$\Delta T_{lm,ice} = \frac{(T_{ice,in} - T_{fr}) - (T_{ice,out} - T_{fr})}{\ln \dfrac{(T_{ice,in} - T_{fr})}{(T_{ice,out} - T_{fr})}}$$

For charging, *y* is the fraction of storage charged and

$$\Delta T_{lm,ice} = \frac{(T_{fr} - T_{ice,out}) - (T_{fr} - T_{ice,in})}{\ln \dfrac{T_{fr} - T_{ice,out}}{T_{fr} - T_{ice,in}}}$$

For ice-on-coil external storage, melting occurs on the outer surface of the ice on the coils during discharging. For that reason, convection at the ice/water interface is the only resistance to heat transfer. The heat transfer equation becomes:

$$\dot{Q}_{ice} = (UA)_{ice} \Delta T_{lm,ice}$$

Cooling Coils

This component models the performance of a dehumidifying cooling coil using the effectiveness model outlines by Braun (Braun 2002). The geometry of the cooling coil and air duct must be specified by the user. A coil selection program produced by the company MultTherm (MultiTherm 2010) was used to select the correct cooling coil for our heat transfer needs. The geometry of the cooling coil produced by this program was used in the proforma of the cooling coil in TRNSYS.

The Braun method has shown the air-side heat transfer effectiveness can be determined by using the relationships for sensible heat exchangers. This component models the performance of cooling coils utilizing this effectiveness model for counterflow geometries. In order to calculate heat transfer coefficients between the air stream and coil, fin efficiencies are required (Braun 2002).

The components mathematical model followed the Braun method in making the initial calculations necessary to get to the coil performance equations.

The three heat transfer rates are calculated from energy balances on the water and air streams. The total energy transferred across the coil is,

$$\dot{Q}_{Coil} = \dot{m}_w \times Cp_w (T_{w,o} - T_{w,i})$$

with the heat transfer caused by the condensing the moisture in the air is calculated as

$$\dot{Q}_{lat} = \dot{m}_a (\omega_{a,i} - \omega_{a,o}) h_{fg}$$

The heat of vaporization for water (h_{fg}) is assumed to be held constant at standard conditions. The sensible heat transfer is then simply,

$$\dot{Q}_{sens} = \dot{Q}_{coil} - \dot{Q}_{lat}$$

Pumps

A program by Bell & Gossett (Bell & Gossett 2009) was used to select the pumps for the systems. The specifications for the pumps were then taken from the information the company made available. The method used here to calculate the heat transfer to the fluid from the pump is the same for constant and variable speed pumps. The user must specify the rated flow capacity, overall pump efficiency, and the pressure drop the pump must overcome.

This pump, along with almost all pumps in TRNSYS, takes the mass flow rate as an input but ignores the value except to perform mass balance checks. The outlet flow rate is based on the rated flow rate parameter and the current value of its control signal input. The equations the mathematical model uses to calculate the outlet water temperature are shown below.

The ideal work done in pumping the fluid is:

$$\dot{W}_{pumping} = \frac{\Delta p \times \dot{m}}{\rho_{fluid}}$$

The inefficiency of the pumping process is taken into account in the work done at the pump's shaft.

$$\dot{P}_{shaft} = \frac{\dot{W}_{pumping}}{\eta_{pumping}}$$

The energy transferred from the pump motor to the fluid stream is calculated by:

$$\dot{Q}_{fluid} = \dot{P}_{shaft}(1 - \eta_{pumping})$$

The temperature exiting the pump can now be calculated.

$$T_{w,out} = T_{w,in} + \frac{\dot{Q}_{fluid}}{\dot{m}_w Cp_w}$$

Controller

The controller used to control the flow was a PI controller. The PI controller calculates the control signal (c) required to maintain the controlled variable (y) at the setpoint (ySet). The tracking error is proportional to the control signal, as well as to the integral of that tracking error. It is based on state-of-the-art discrete algorithms for PI controllers and implements anti windup for the integrator. The user must specify the set point temperature (yset), gain constant, and integral time.

The algorithm used by this controller was presented by (Åstrøm and Hagglund 1995).

$$v(t) = K\left[e(t) + \frac{1}{T_i}\int_0^t e(\tau)d\tau\right] = P + I$$

The two terms in the equation can easily be identified as the P-term (proportional to the error), and the I-term (proportional to the integral of the error). This equation is sometimes referred to a series PI algorithm.

For cooling applications, the control signal must increase when the tracking error (setpoint-controlled temperature) decreases. Effectively more cooling should be provided when the temperature is above the setpoint. A negative gain constant is then required and integral time should be set to a low value to provide a fast response.

System Criteria

The controller in the system was set to maintain an inlet air temperature to the thermal load at 24°C (75.2°F) if possible by controlling the flow rate of the chilled water to the cooling coil. The temperature criteria for each load profile are listed in the Table 4-1. Under normal conditions the inlet air temperature to the thermal load must stay within the stability range. However during emergency conditions the inlet air temperature can exceed the stability range limits but must still be under the upper limit of the allowable range though.

Table 4-2 Temperature Criteria

	Allowable Range	Stabilty Range
Data Centers	18°C to 27°C	±2°C
Clean Rooms	15°C to 30°C	±1°C
Laboratories	18°C to 29.5°C	±1.1°C

The sizing of all the components in the system is discussed in Appendix B but TES tanks will be explained to some extent here. Research found that chilled water stratified TES tanks are 90% stratified along with 1%-2% thermal losses due to environmental conditions (Dorgan & Elleson 1993). Almost all TES tanks are stratified and not well-mixed so all simulations were for stratified tanks. The TES tanks also changed in size for the different temperature differences across the cooling coils (thermal load) because of the amount of chilled water needed is slightly smaller for the higher temperature differences. The low temperature chilled water storage tanks were also considerably smaller than the TES tanks for regular chilled water systems.

Example Simulation

The substantial amount of simulations completed for the five systems from chapter three obviously does not allow for each result to be shown here. The data collected was compressed into several smaller files and analyzed. The results from the examination of the data are explained in chapter five. An example of one of the systems simulated is shown below at certain parameters. A walk through will be performed so the simulation may be better understood.

The example simulation shown is for the series atmospheric chilled water tank system at a thermal load of five megawatts. The chiller set point temperature was set at 7.22°C (45°F) with a temperature difference across the cooling coils of 5.56°C (10°F). With these conditions set, the components could then be sized according to Appendix B. The cooling load profile used in this example was for clean rooms.

As stated above, the inlet air temperature to the thermal load was maintained at a temperature of 24°C (75.2°F). This was done by controlling the chilled water flow rate to the cooling coil. The tank was sized to provide enough cooling to hold the ambient air temperature at 24°C (75.2°F) for 30 minutes. The tank holds 431.6 m^3 (114,000 gallons). The clean rooms load profile has the highest thermal load of all three cooling load profiles and will use all of thermal energy available in the TES tank. The other two profiles (Data centers & Laboratories) would leave a portion of the thermal energy in the tank.

Emergency cooling was needed at the eighth hour for thirty minutes before the chiller would be operating at full capacity again. The figure below shows the inlet air temperature to the thermal load and the temperatures of the chilled water in the TES tank at different locations. As seen below the inlet air temperature was held at 24°C (75.2°F) while the chilled water temperatures in the TES tank rose during emergency conditions but eventually recharged. A closer look will be taken at the chilled water temperatures farther below. This was considered a successful simulation as the inlet air temperature held steady during emergency conditions and the tank eventually recharged.

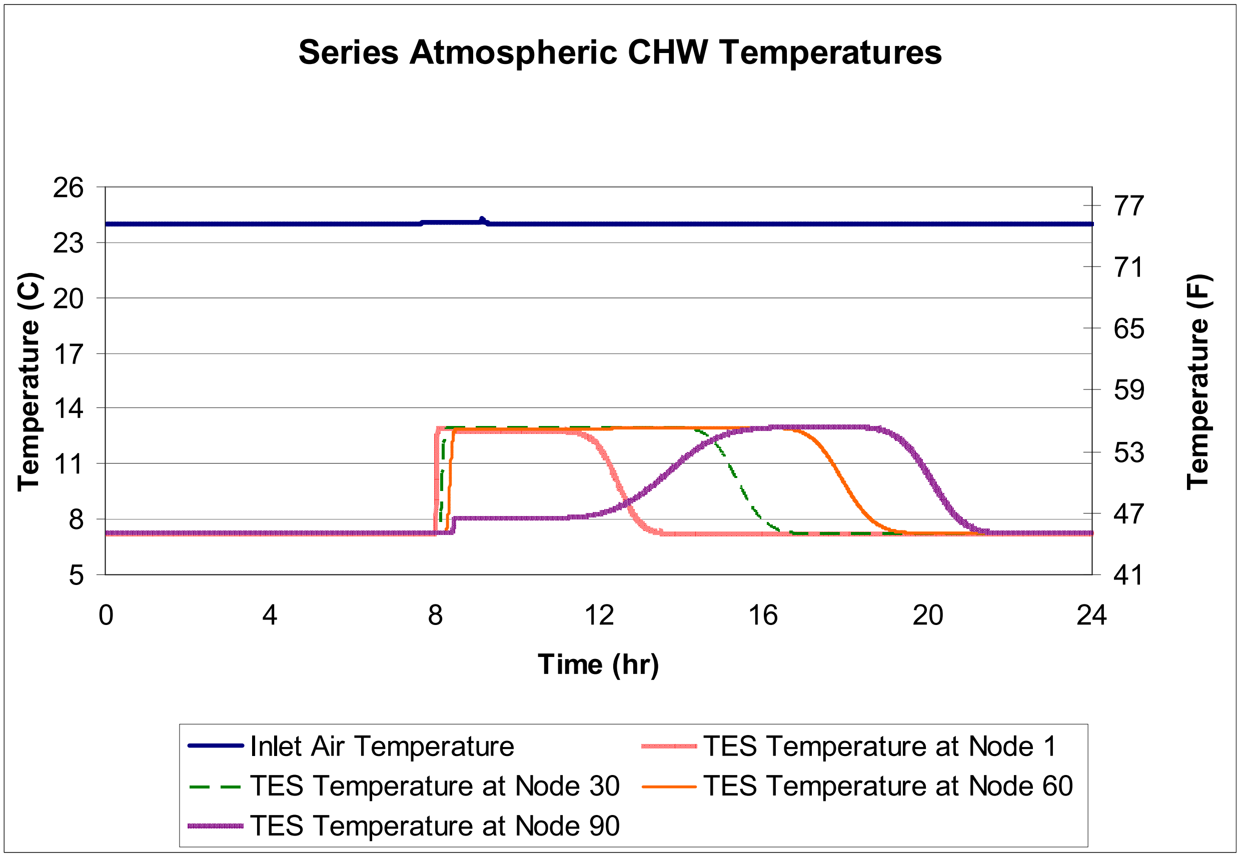

Figure 4-5 Series Atmospheric CHW Tank - Temperatures

Figure 4-6 shows the chilled water temperatures at different locations in TES tank during emergency conditions. This is a much closer view of the TES temperatures during discharge. This graph gives an idea of where the thermocline layer is in the tank. If the slope of each line is examined, it can be seen that the slope is decreasing ever slightly as the node location increases. This can be contributed to the thermocline layer increasing as it moves through the tank which would be expected.

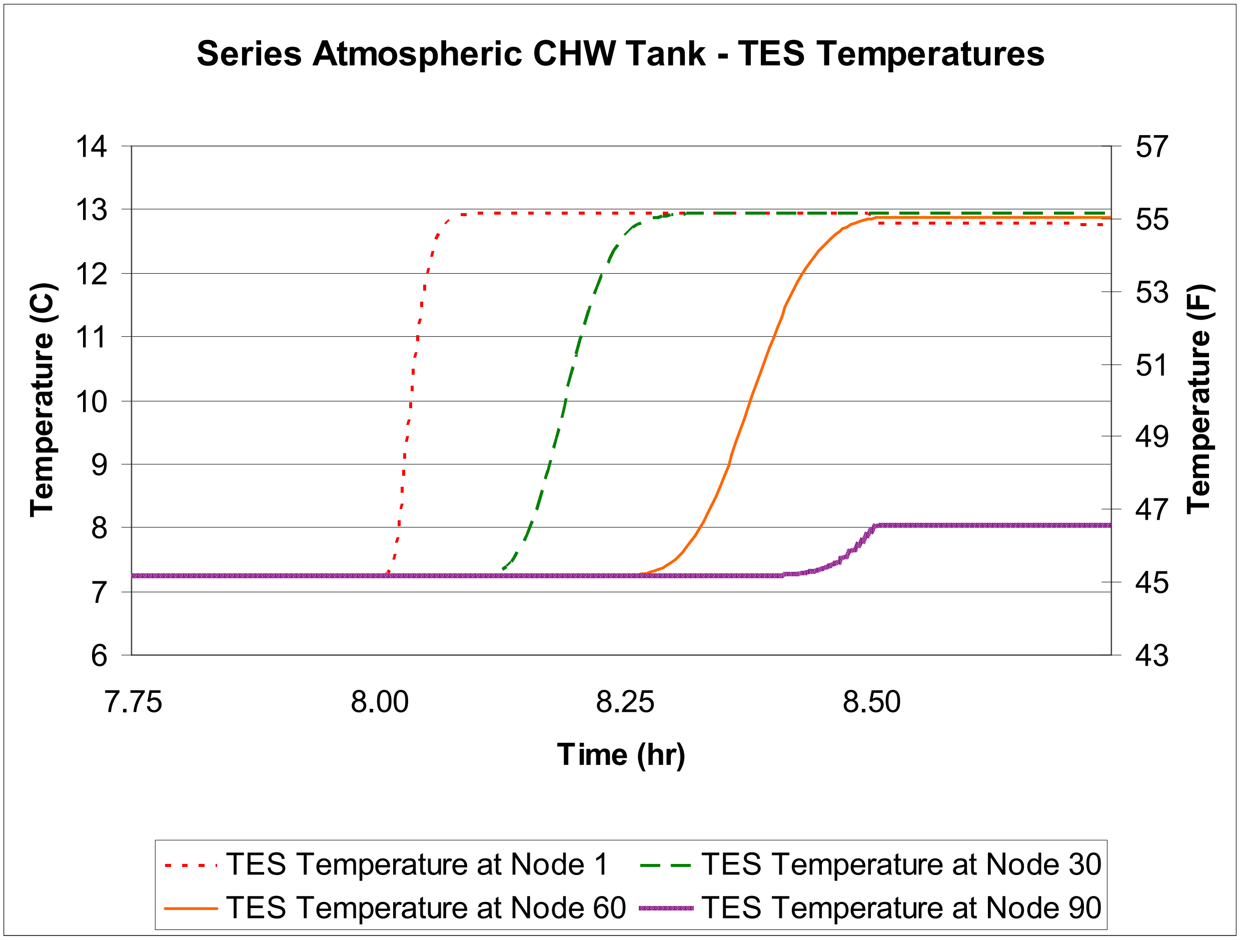

Figure 4-6 Series Atmospheric CHW Tank – TES Tank Temperatures

Figure 4-7 shows the flow rate for the system over the full simulation. It shows how the variable speed pump in the system reacts to the changes in thermal load by altering the flow to the cooling coils.

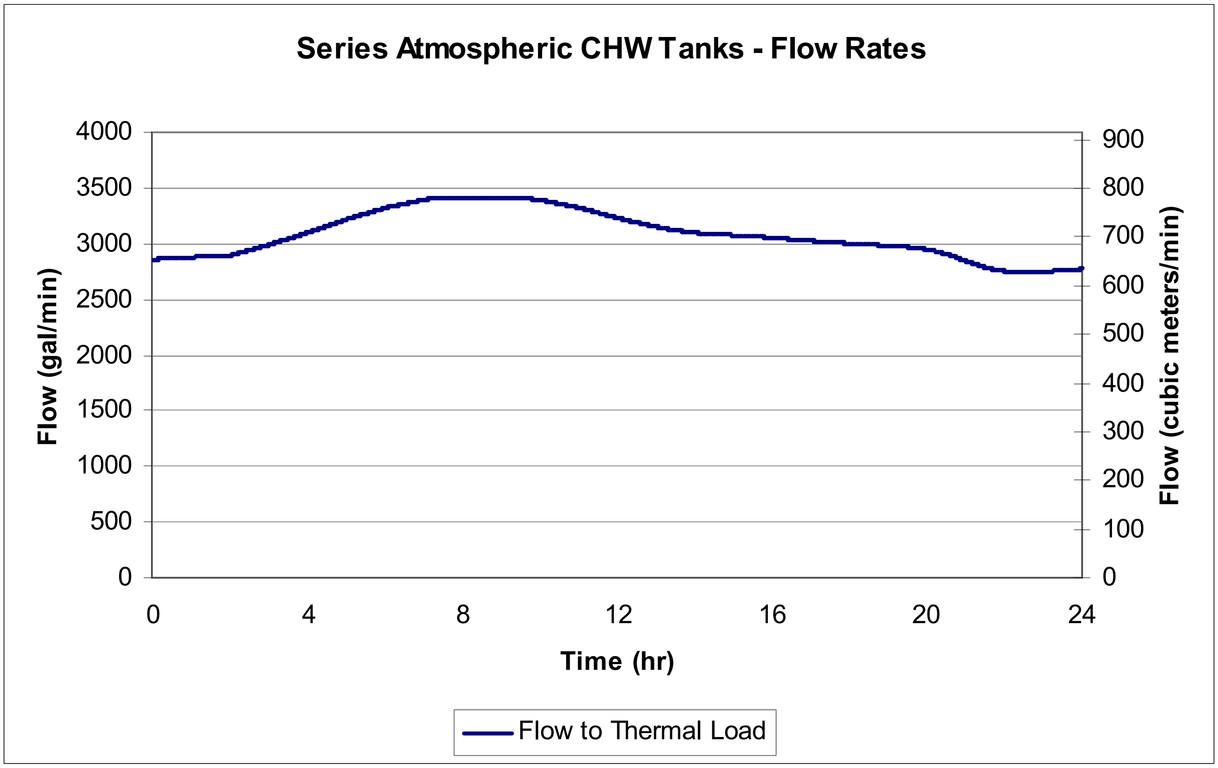

Figure 4-7 Series Atmospheric CHW Tanks - Flow Rates

The above explanation showed what was looked for in an individual simulation. After all simulations have been run for a particular storage system and the results analyzed, trends start to appear. The tendencies of systems are then analyzed further and this forms the beginning of many of the design criteria. The design criteria of different systems are then compared to each other to locate the characteristics of each system type.

CHAPTER 5 - Design Analysis and Criteria

This chapter deals with the results from the analysis of the multitude of simulations conducted. The tank energy ratio, stratification levels, recharging conditions, flow rates, ragged cooling conditions, temperature differences across the cooling coils, chiller set point temperatures, and tank sizes are all examined for regular chilled water tanks, low temperature chilled water tanks, and ice storage tanks. The differences between the systems can be seen further below.

Many of design criteria listed below were only formed after combining the data from many different simulations. Some of the graphs below are a specific example of a certain simulation; these graphs are specifically stated if so, while other charts are a combination of all simulations for a certain storage system (ex. Energy ratio & tank size). The trends produced over many simulations are also acknowledged.

Design Analysis

This section deals with analysis of the results of the simulations conducted. The design criteria listed farther below are directly taken from this section. Graphs and tables are provided to support statements made.

Energy Ratio

This energy calculation is based on defining a lowest possible temperature (T_{ref}) that could provide cooling and computing the cooling capacity (internal energy) in the tank according to

$$U = \sum_{i=1}^{n} M_i c_v (T_i - T_{ref})$$

where U is the stored cooling capacity, M_i is the mass associated with water increment i, c_v is the constant volume heat capacity, T_i is the water temperature for water increment i, and T_{ref} is the minimum temperature that provides cooling.

The energy ratio is the energy left in the tank over the maximum energy storage capacity in the tank. The energy storage in the tank refers to the tank's cooling capacity. This is represented on the y-axis in the graphs below. The label on the x-axis depends on whether the simulation is representing chilled water or ice storage. The x-axis for the chilled water graphs is the location of the thermocline in tank. The x-axis for ice storage graphs is the fraction of ice burned in the tank. This graph allows for the user to determine what fraction of cooling capacity (energy) is left in the tank. The charts also work for any thermal load, chiller set point temperature (40°F, 45°F, and 50°F), and all three cooling load profiles (Data Centers, Clean Rooms, and Laboratories).

Chilled Water Tanks

The graph in Figure 5-1 is the energy ratio versus the location of the thermocline in the TES tank for the three different temperature differences across the cooling coil for chilled water. The location of the thermocline is the beginning of the thermocline layer in the TES tank. It is defined as the percentage through the tank the thermocline layer is with 0.00 being at the top of the tank and 1.00 at the bottom of the tank. This chart was made from the simulations conducted of the three storage systems using chilled water. An equation fit was performed on each of the three data streams and the equations of the lines were displayed on the graph.

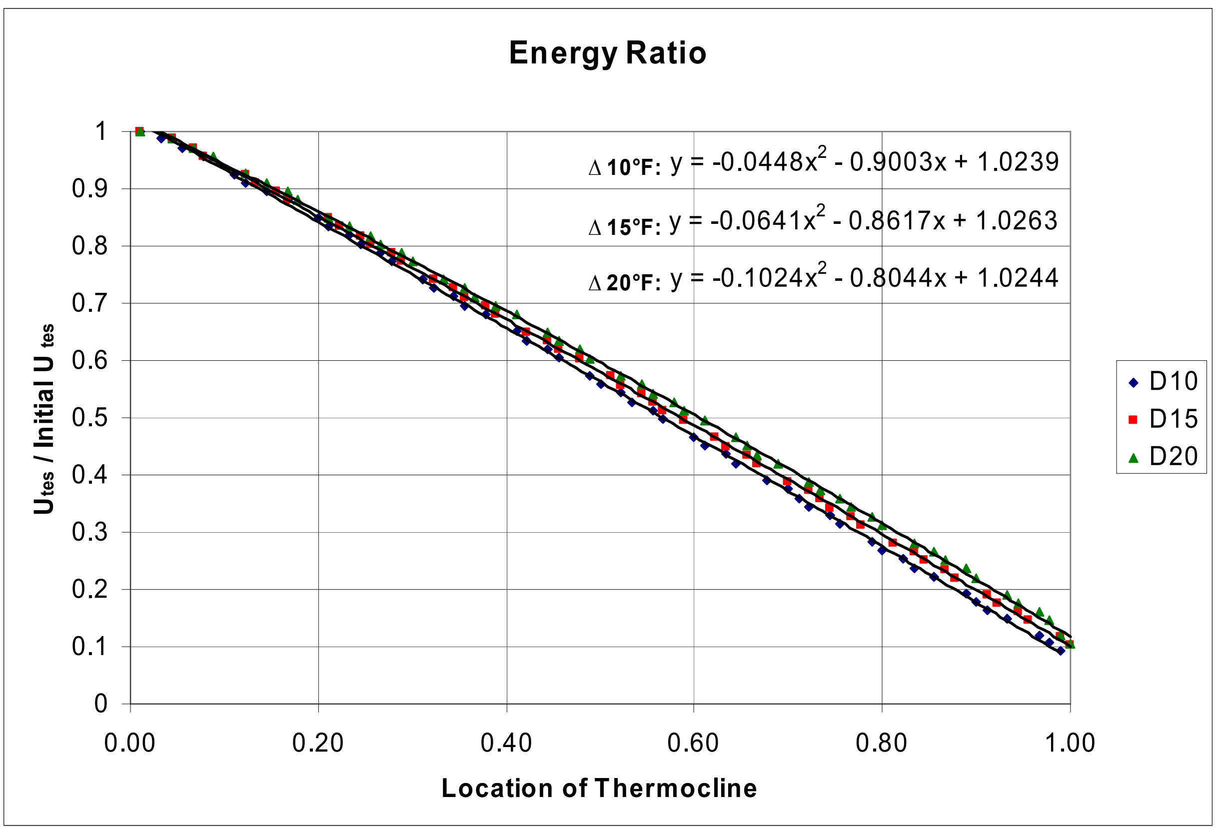

Figure 5-1 Energy Ratio for Chilled Water

The above graph is true for all chilled water TES tanks in series and parallel and whether the tank is atmospheric or pressurized. The discharging rate of energy from the tank is the same no matter what the system configuration is and the chiller set point temperatures also made no difference in terms of energy ratio tends. The temperature difference across the coil made very little difference in terms of the energy ratio. The change in the data from the temperature differences across the coil is less than two percent, so it can be concluded that this has very little effect on the energy ratio.

The thermocline layer is at the bottom of the tank while the tank still holds 10% of its total energy as shown on the graph. This is due to the fact that the tank is 90% stratified. The final 10% is not at a constant temperature and has been partially mixed with the incoming hot water. This final portion of energy will only support ragged cooling for a certain amount of time. This is more thoroughly discussed farther below in the ragged cooling section.

Low Temperature Chilled Water Tanks

The chart below in Figure 5-2 is the energy ratio versus location of the thermocline for low temperature chilled water. This chart was prepared from the simulations conducted from the storage system using a low temperature chilled water tank. An equation fit was also performed on the three data streams as well.

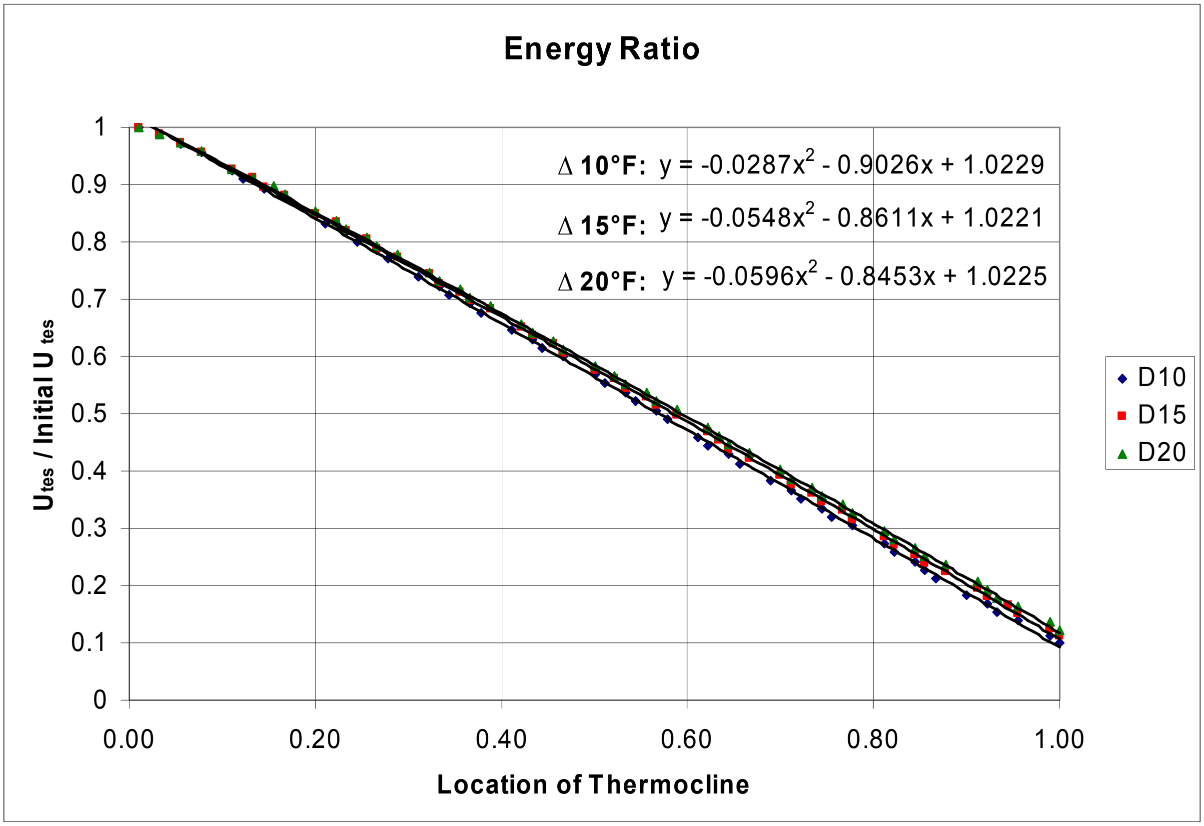

Figure 5-2 Energy Ratio for Low Temperature Chilled Water

This graph was very similar to regular chilled water but the data streams decay at different rates compared to regular chilled water as shown by the equations. The end point for the energy ratio is same for both graphs but the low temperature chilled water decay rate is slightly more linear when compared to regular chilled water. The change between the data streams for the different temperature differences across the coils is also negligible for low temperature chilled water.

The thermocline layer is also at the bottom of tank while the tank still holds 10% of its total energy just like the chilled water graph. The final 10% of the energy will only support ragged cooling.

Ice Storage Tanks

Figure 5-3 shows the energy ratio versus the fraction of ice burned in the ice storage tank for the three temperature differences across the coil. The data from this graph was made from the simulations of storage systems using ice storage. A single equation fit was performed on all three of the data streams and is shown on the graph.

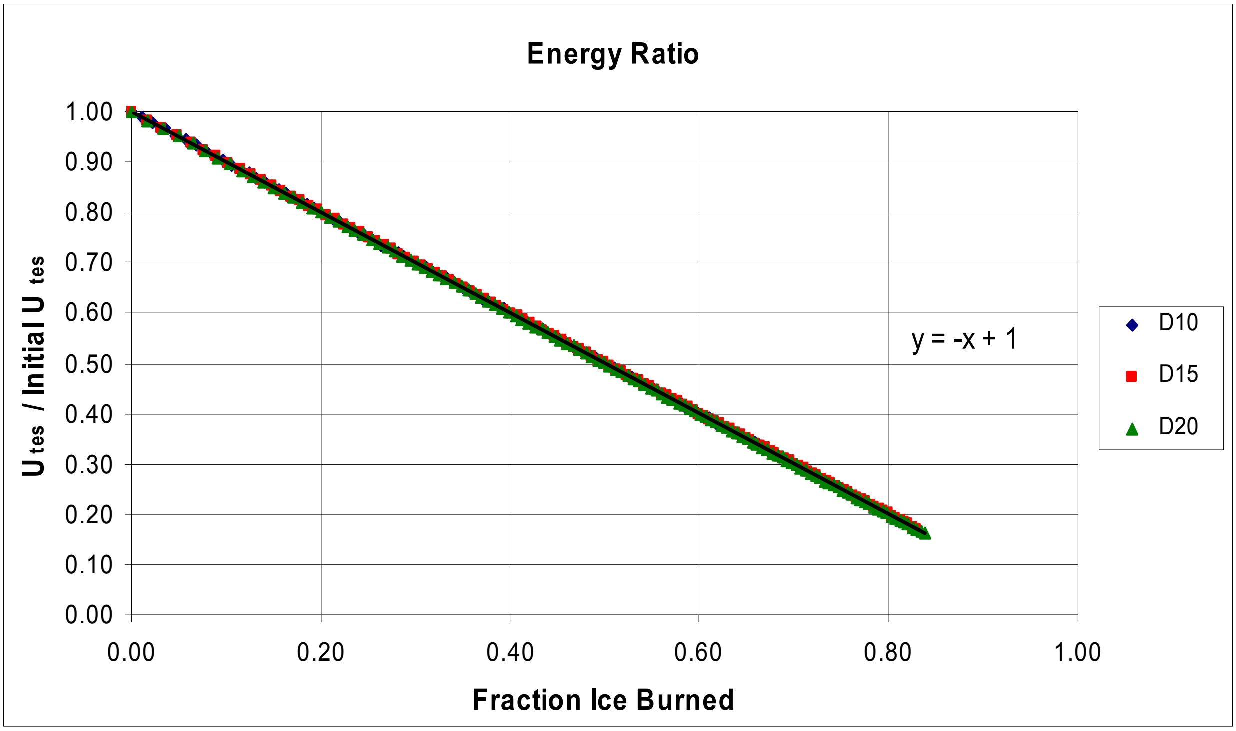

Figure 5-3 Energy Ratio for Ice Storage Tanks

Figure 5-3 shows that ice has the same declining energy rate no matter what the temperature difference across the coil or the chiller set point temperature is. It may be hard to see on this graph but the higher the chiller set point temperature the more ice that is melted. The temperature difference across the coil makes no difference in the amount of ice used.

One main difference between ice storage and the chilled water storage is that there is no ragged cooling with any energy that is left in the tank. If all of the ice is used up in the tank, no additional thermal energy will be available and temperatures will immediately begin to rise at a substantial rate. This will be better discussed in the ragged cooling section.

Tank Sizes

Figures 5-4 through 5-6 were prepared for a designer that wished to determine the size of a TES tank based on the amount of time desired for emergency cooling for the three different temperature differences across the cooling coil. An equation fit was performed for each of the three data streams. These equations could be used by an engineer to easily size a system for any amount of time. The graphs for chilled water also account for the fact that the tank will be 90% stratified. The left and right y-axis represents the thermal load over the volume of the tank in metric and English units. An example of how to use the graphs is shown below.

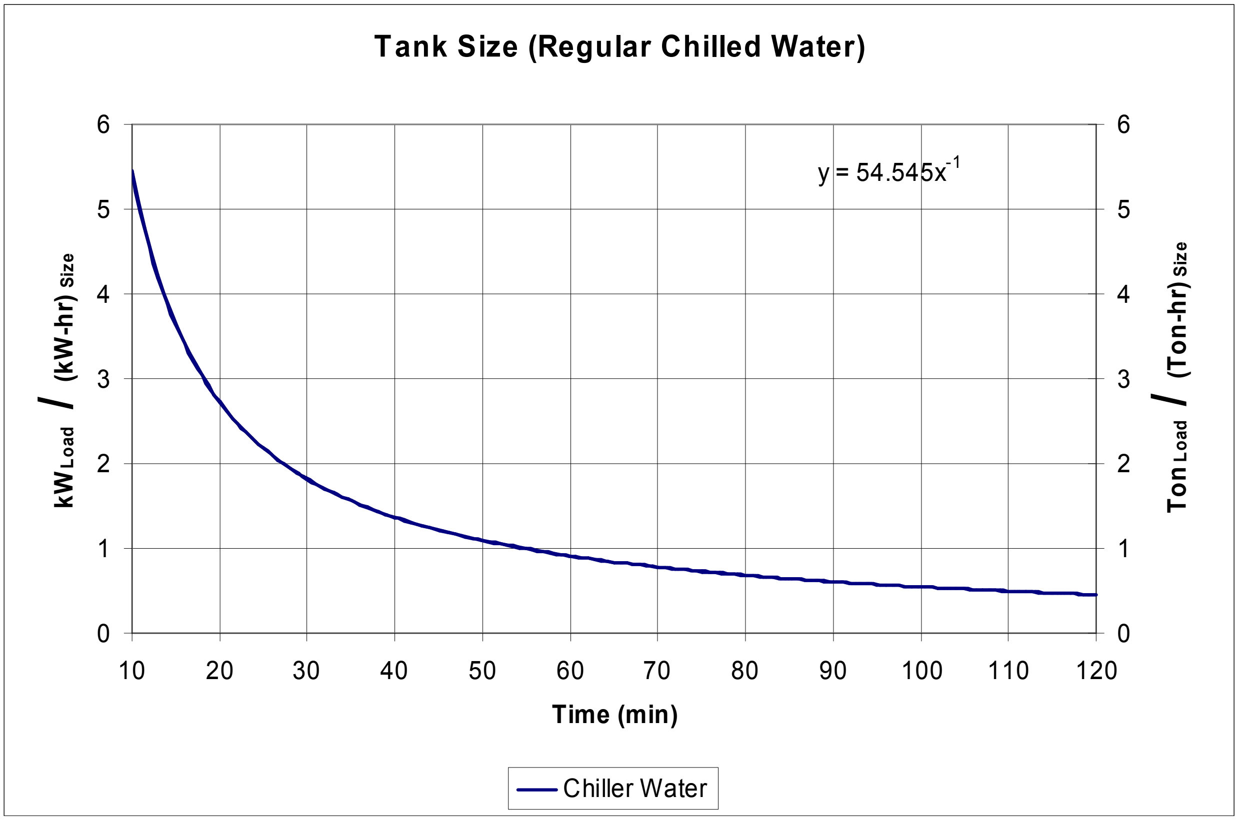

Figure 5-4 Tank Size – Regular Chilled Water

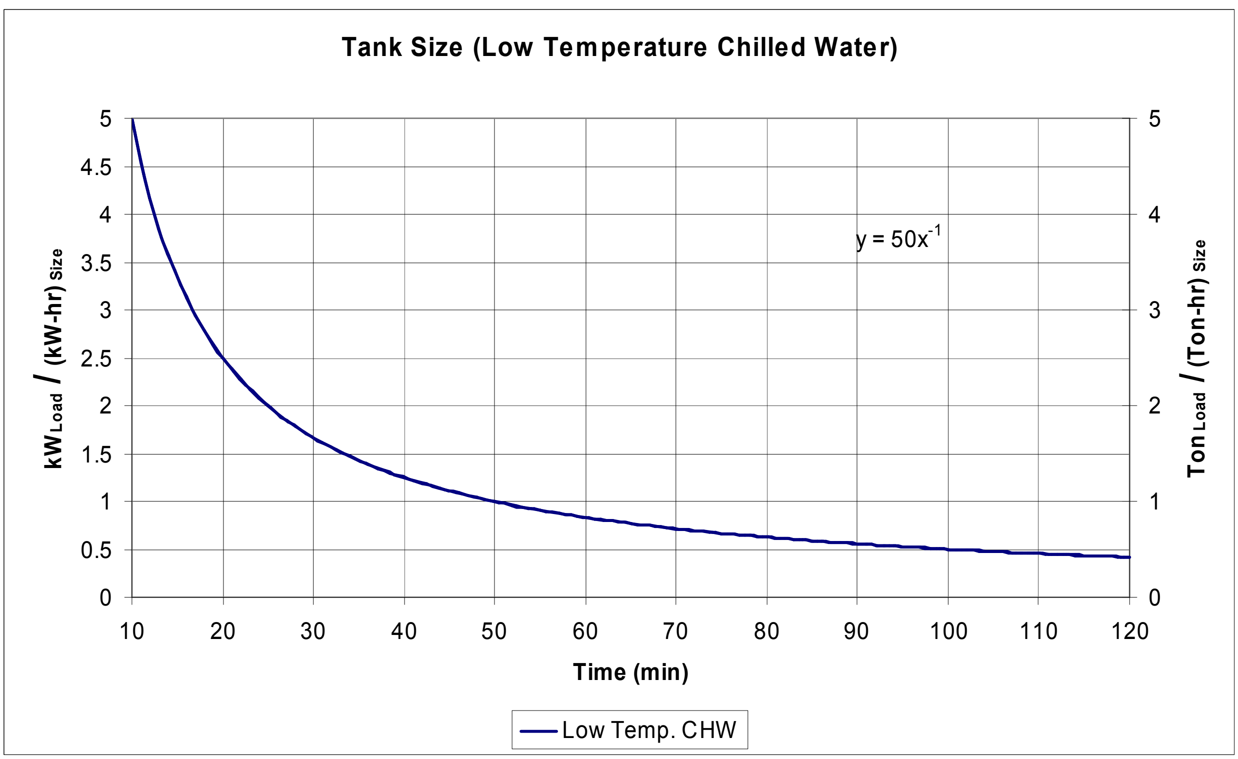

Figure 5-5 Tank Size - Low Temperature Chilled Water

The graph in Figure 5-6 accounts for the fact that ice storage will have two separate storage units of the same size. As this storage component of the system will sit idle for extended periods, thermal losses (typically <= 1% / day) will require periodic recharging of the ice storage tank. This will take place after 15% to 25% of the thermal storage has been lost depending on system requirements. A redundant storage tank is included in the system to accommodate for this. One ice tank will always be at full capacity each day. The other unit will typically be used to augment the cooling system during the day, and then be rebuilt to full capacity overnight. The units are used in an alternate, lead/lag pattern each day.

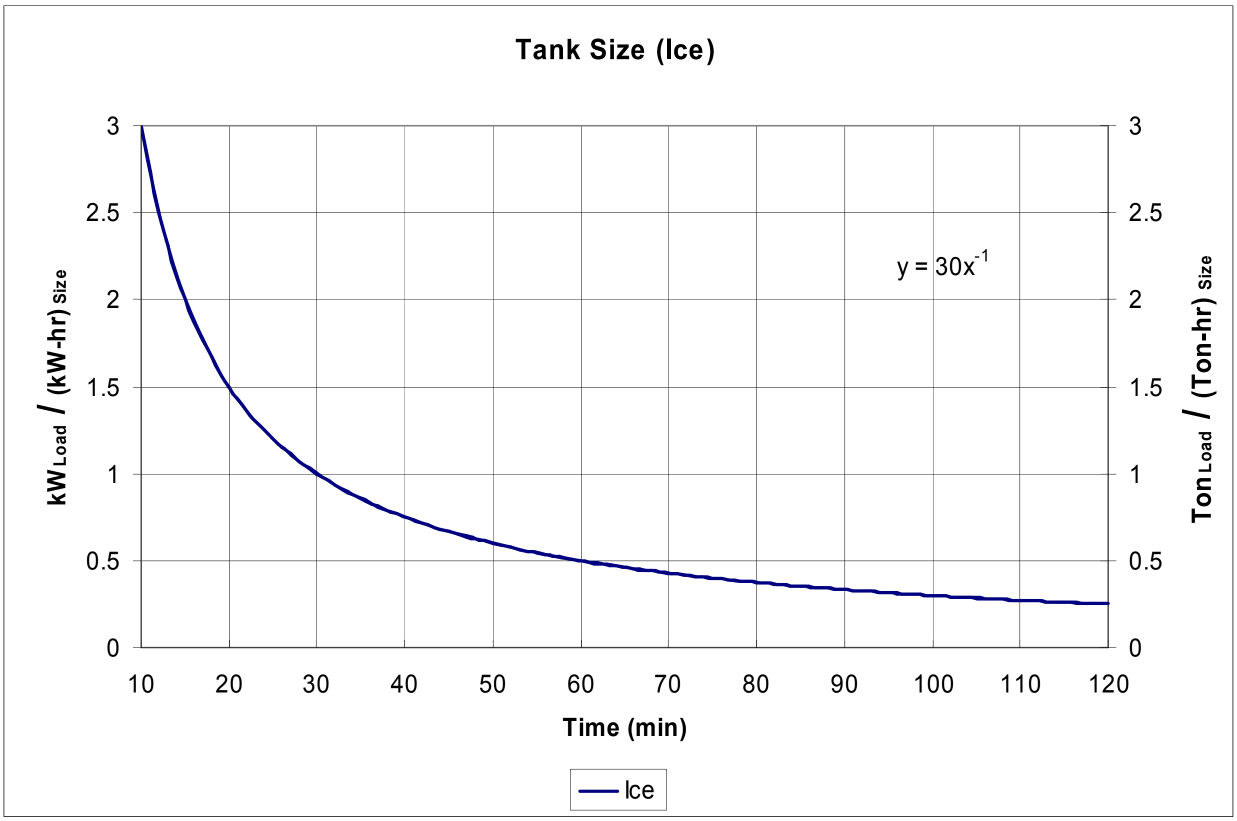

Figure 5-6 Tank Size - Ice Storage

The above graphs can work one of two ways. A designer must either know the amount of time emergency cooling needs to last or the size of the TES tank. For the example calculation, it is assumed the TES tank must cool a thermal load of five megawatts for 30 minutes.

Example:

Known: Thermal Load = 5 MW Cooling Time = 30 minutes

Using the amount of time cooling needs to last in emergency conditions (30 minutes), the thermal load over the volume of the tank can be found in Figures 5-4 through 5-6.

Regular CHW: 1.82 kW_{Load}/(kW-hr) Low Temp. CHW: 1.67 kW_{Load}/(kW-hr)

Ice: 1 kW_{Load}/(kW-hr)

With the thermal load that is needed to be cooled known, the size of the TES tank can be determined.

Regular Chilled Water:

$5\ MW = 5000\ kW = 1.82\ \dfrac{kW}{kW-hr}(Size\ of\ TES\ Tank)$ → Size of TES Tank = 2747.3 kW-hr

Low Temperature Chilled Water:

$5\ MW = 5000\ kW = 1.67\ \dfrac{kW}{kW-hr}(Size\ of\ TES\ Tank)$ → Size of TES Tank = 2994 kW-hr

Ice Storage:

$5\ MW = 5000\ kW = 1\ \dfrac{kW}{kW-hr}(Size\ of\ TES\ Tank)$ → Size of TES Tank = 5000 kW-hr

The calculations can also be reversed if the size of the TES tank is known and the amount of time emergency cooling will last wishes to be found.

Example:

Known: Thermal Load = 5 MW Size of Tank = 4000 kW-hr

The thermal load over the size needs to be calculated first,

$$\frac{5000\ kW}{4000\ kW-hr} = 1.25\ \frac{kW}{kW-hr}$$

Using the Figures 5-4 through 5-6, the amount of time emergency cooling will last for each type of cooling can be found.

Regular Chilled Water: 44 minutes

Low Temperature Chilled Water: 40 minutes

Ice Storage: 24 minutes

Energy Losses

The energy losses of TES tanks are an important feature to study and even necessary to know when initially sizing. Ideally during discharging, chilled water tanks would be perfectly stratified and energy losses negligible within the tank. Analysis of the simulations showed that the energy losses for each type of TES tank were different and must be accounted for when sizing the tanks. The three figures below show how each type of TES tank differs from an ideal discharge.

The energy ratio is the energy left in the tank over the maximum energy storage capacity in the tank. This is represented on the y-axis in the graphs below. The label on the x-axis is the mass ratio. The mass ratio is the mass of thermal energy of the fluid left in the tank over the maximum amount of mass the tank can contain. Each of the graphs contains a black linear line that represents how energy would ideally discharge from the tank. The charts also work for any thermal load (1 MW, 5MW, and 15MW), chiller set point temperature (40°F, 45°F, and 50°F), and all three cooling load profiles (Data Centers, Clean Rooms, and Laboratories).

Regular Chilled Water Tanks

The graph in Figure 5-7 represents a regular chilled water tank. This graph holds true no matter what the system configuration is, parameters are, or whether the TES tank is atmospheric or pressurized. Energy discharges from the tank very close to the ideal rate but not exactly. This can be contributed to the fact that the tank is not perfectly stratified and the thermal losses to the environment.

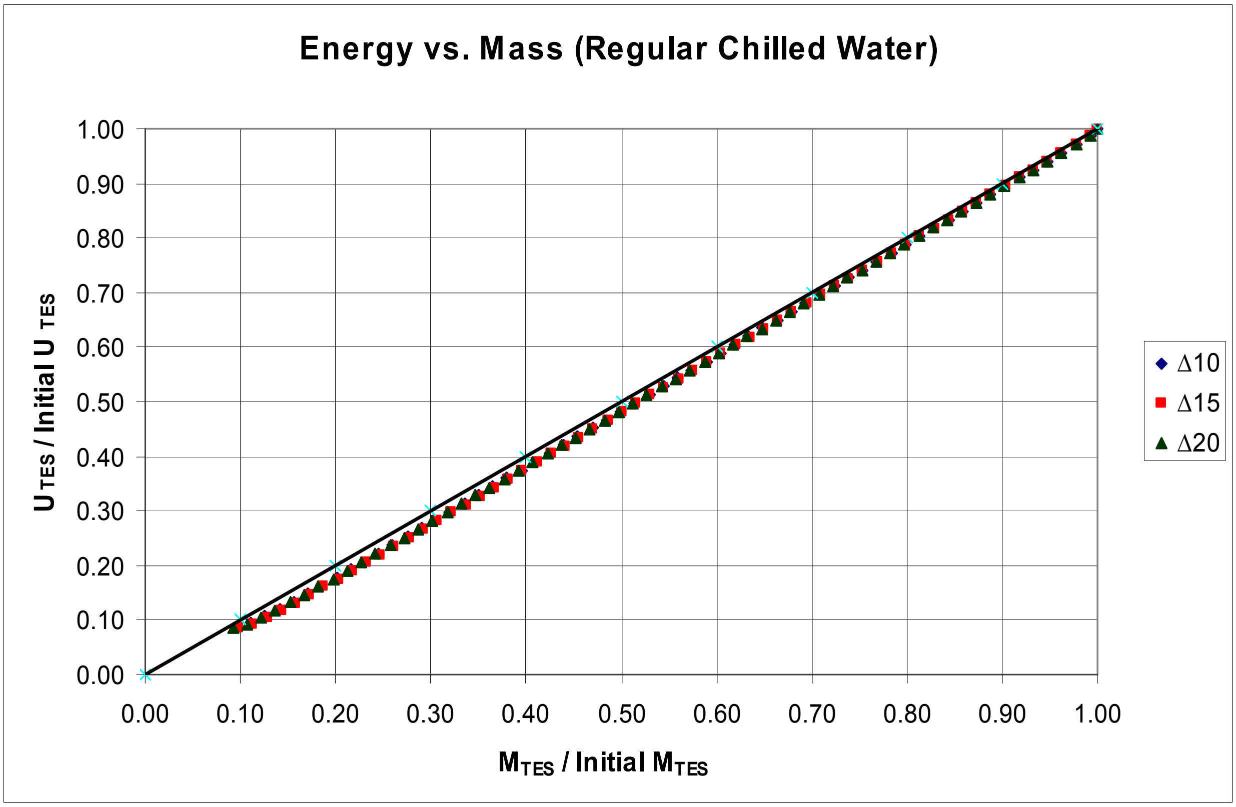

Figure 5-7 Energy vs. Mass (Regular Chilled Water)

The tank still holds approximately 10% of its initial mass at the end of discharging. This is due to the fact that tank is 90% stratified. The final 10% of the mass is not at a constant temperature and has been partially mixed with the incoming hot water. This final portion will only support ragged cooling for a certain amount of time.

Low Temperature Chilled Water Tanks

Figure 5-8 shows the energy losses for a low temperature chilled water tank. Energy discharges from low temperature chilled water tanks very differently from regular chilled water. Ideally the tank would hold 10% of its mass at the end of discharging due to the 90% stratification in the tank. The same stratification effects were applied to low temperature chilled water tank as to regular chilled water tanks. However due to the lower temperature of the chilled water in the tank, there was additional mixing between the incoming hot water and the chilled water. This led to an additional part of the mass ratio to be in the thermocline layer. This is why the data for low temperature chilled water tanks diverges so far from the ideal line.

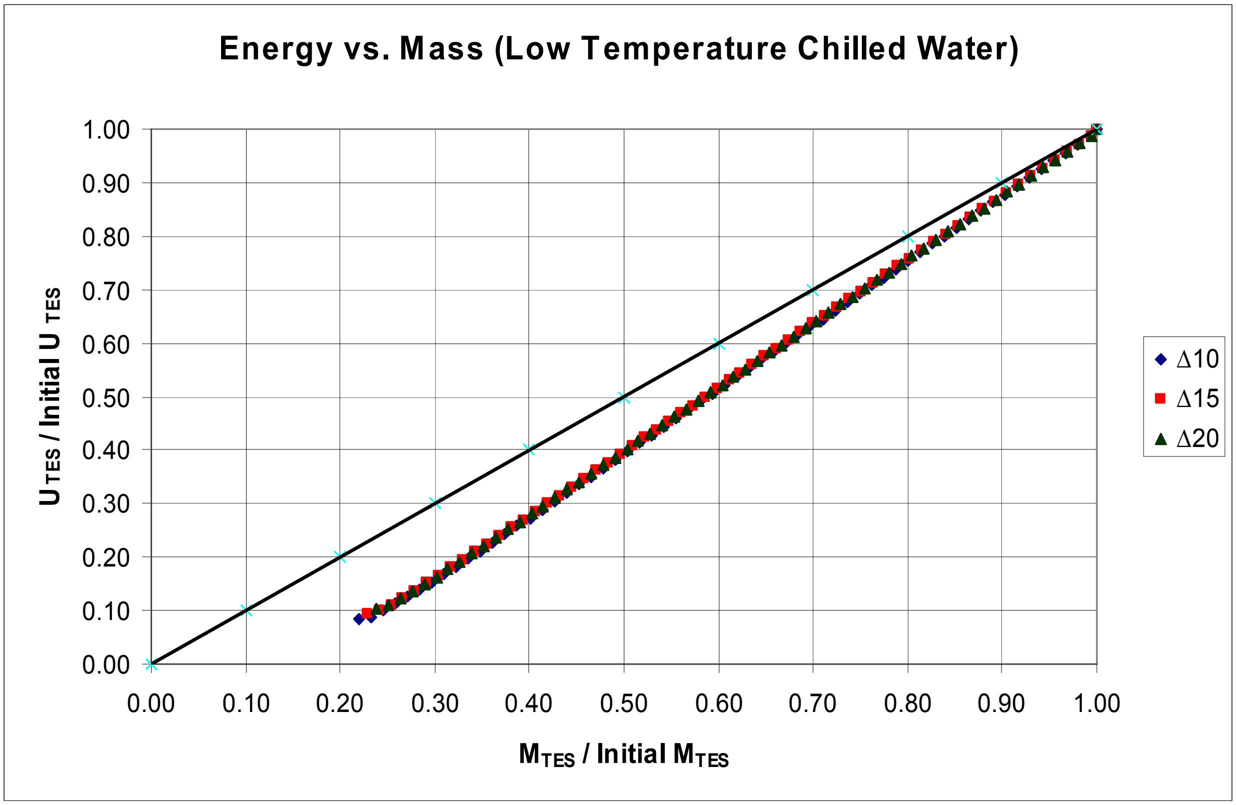

Figure 5-8 Energy vs. Mass (Low Temperature Chilled Water)

Ice Storage Tanks

Figure 5-9 symbolizes energy losses for ice storage tanks. Energy discharges from the tank ideally for ice. This is due to ice not having the stratification effects that plagued chilled water.

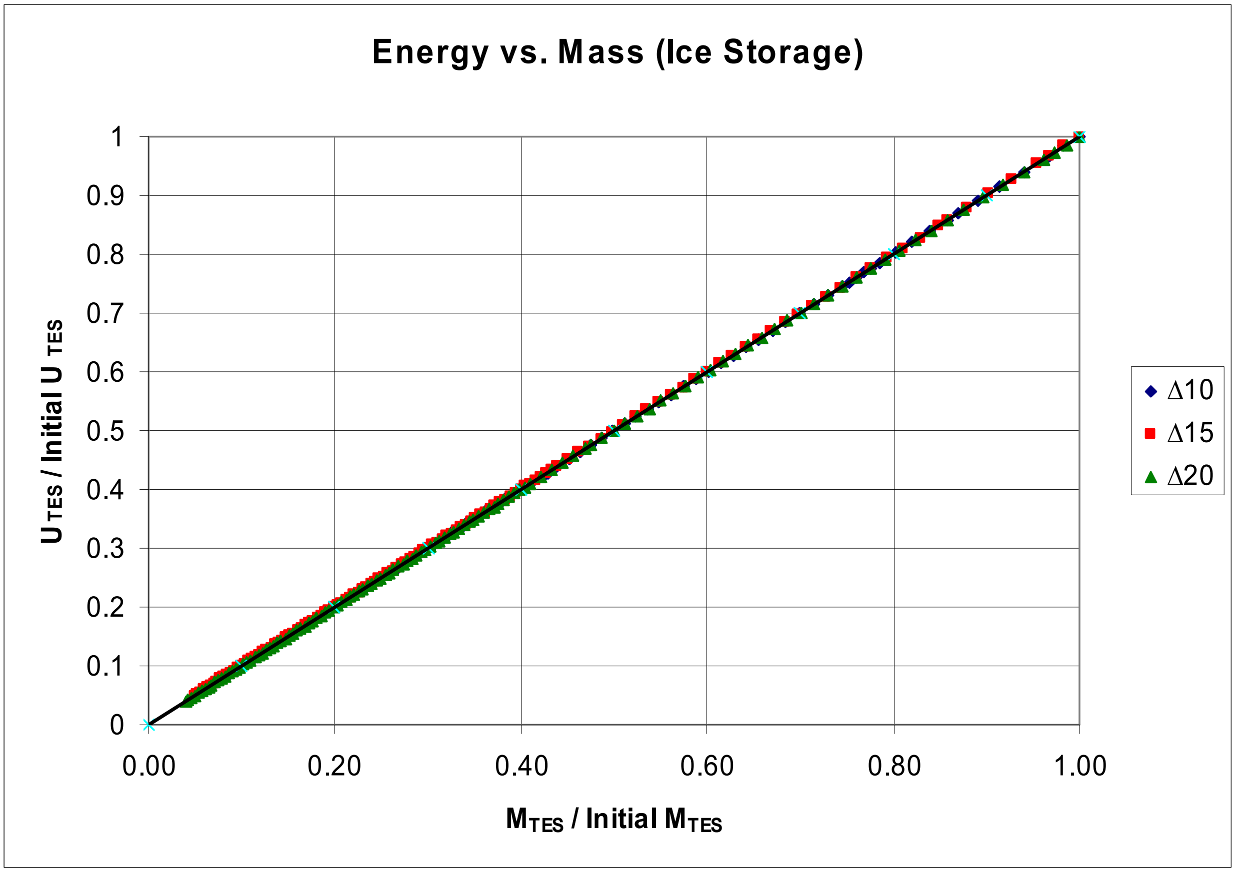

Figure 5-9 Energy vs. Mass (Ice Storage)

Stratification Levels

This part of the report deals with the stratification level throughout the tank dependent on time. It gives a good representation of where the thermocline layer is in the tank during discharging conditions and how the thermocline layer changes as it moves down the tank. The location of different nodes throughout the TES tank is explained above in chapter four.

The left y-axis in the graphs below is the storage per node in ton-hrs/node and the right y-axis is the average tank temperature for a stratified TES tank. While the graphs below are for a particular thermal load and temperature difference across the coil, the trends are the same for the different thermal loads, temperature differences across the coil, and chiller set point temperatures.

Figure 5-10 represents the stratification levels for the parallel atmospheric chilled water tank system at five megawatts and a temperature difference of 15°F, while Figure 5-11 is for the series atmospheric chilled water tank system at one megawatt and a temperature difference of 10°F. The scales between the graphs may be different but the trends are basically identical. As the tank begins discharging during emergency conditions, node one immediately decreases as the thermocline layer works its way down the tank. The preceding nodes also decrease when the thermocline moves down the tank as time increases.

One particular feature of note is how the thermocline layer changes as the tank discharges. Looking closely at the graphs it can be seen that as time increases, the nodes discharge the amount of energy at a slower rate. The tank was sized to be 90% stratified but this is not achieved immediately during discharging conditions. The thermocline layer increases in size as time increases which can be seen by the slower rate at which energy discharges from the nodes. The location of the nodes in the TES tank were described in Figure 4-4 on page 28.

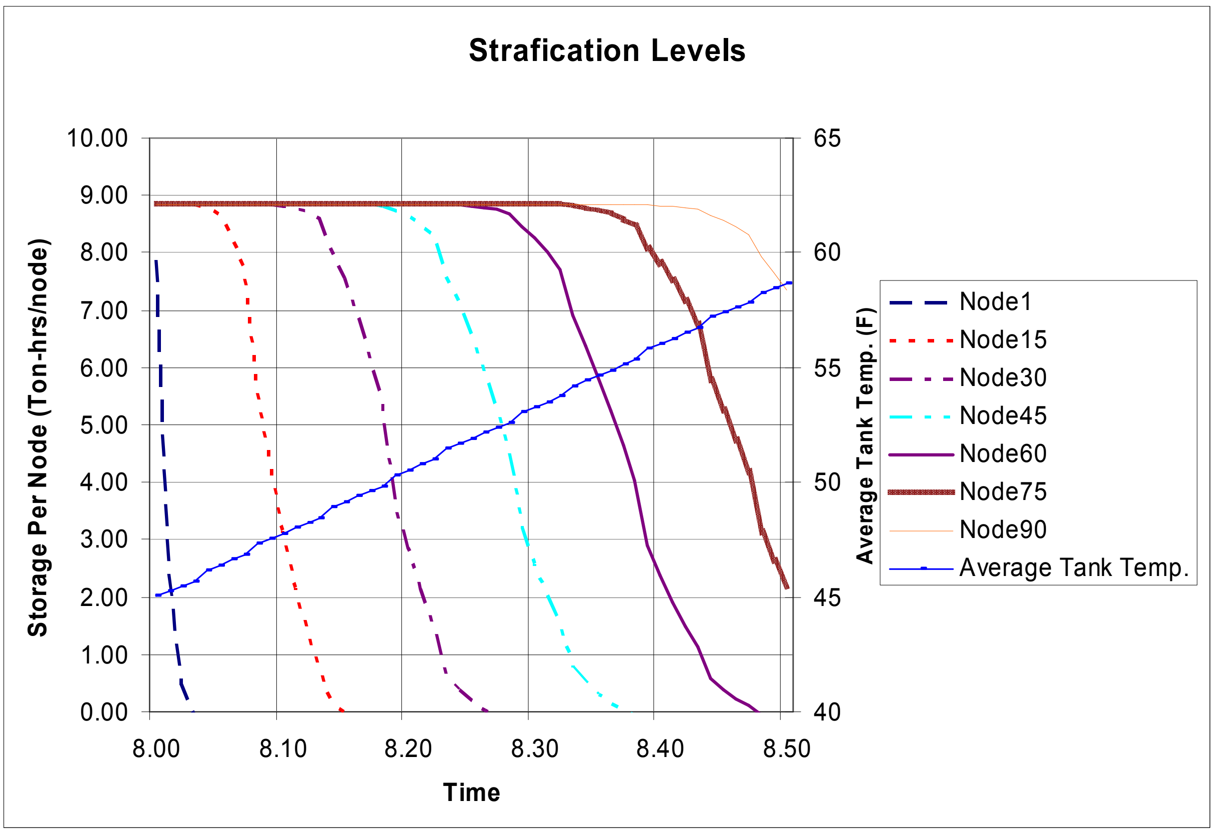

Figure 5-10 Stratification for Parallel Atmospheric CHW Tank at 5 MW and Δ of 15°F

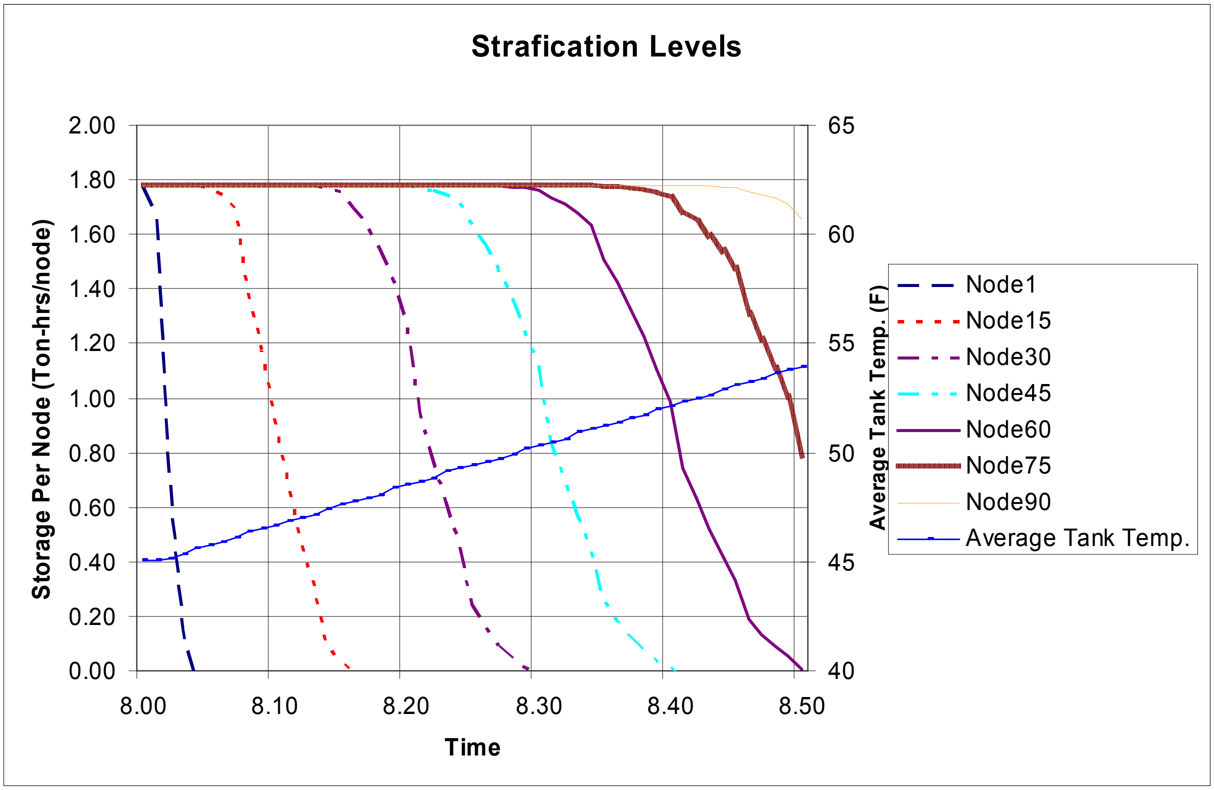

Figure 5-11 Stratification Levels for Series Atmospheric CHW Tank at 1 MW and Δ of 10°F

System Parameters

As discussed in chapter seven, the chilled water set point temperatures were changed from 7.22°C (45°F) to 4.45°C (40°F) and 10°C (50°F) without changing any of the component sizes. Then each chiller set point temperature simulation was simulated at three different temperature differences (Δ 10, 15, & 20°F) across the cooling coils. The effects of these changes on the systems are shown in this section. Five tables are at the end of this section showing a summary for each system of the average inlet air temperature to the thermal load and whether not the system recharges for all the simulations. The reactions for changing the system parameters were the same for all three thermal loads (1, 5, & 15MW).

Chiller Set Point Temperatures

Changing the chiller set point temperature from the original temperature of 7.22°C (45°F) had major effects on the systems. This is shown in Figure 5-12 in the flow rates at different chiller set point temperatures (40°F, 45°F, & 50°F). These flow rates represent a thermal load of one megawatt and a temperature difference of 5.56°C (10°F) across the cooling coil. The trends are the same for all thermal loads and temperature differences across the cooling coil.

Increasing the chiller set point temperature to 10°C (50°F) caused insufficient heat transfer in the cooling coils in many applications. A larger flow rate was required to meet the thermal demands as is seen by the graph below in which the flow rate for the chiller set point temperature of 10°C (50°F) is much higher than the temperature of 7.22°C (45°F). Decreasing the chiller set point temperature to 4.45°C (40°F) led to a smaller flow rate because the cooling coils are designed to utilize an incoming chilled water temperature of 7.22°C (45°F). The blips that occur at hour ten in Figure 5-12 are due to the tank recharging at this point and the extra flow being used for this.

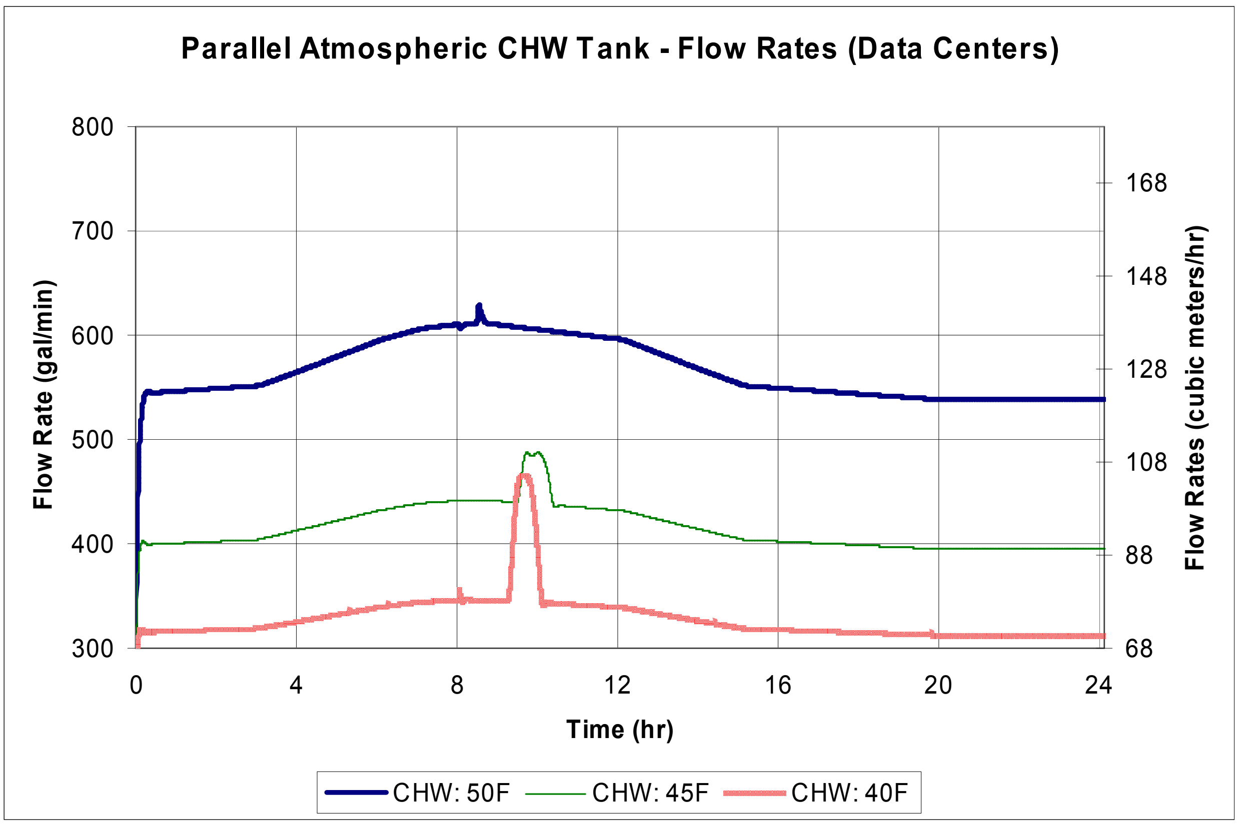

Figure 5-12 Parallel Atmospheric Chilled Water Tank - Flow Rates (Δ10F-Data Centers)

For the data centers cooling load profile, the thermal load was not high enough to cause the flow to be at its maximum as the clean rooms and labs load profiles would. The flow rates for the clean rooms profile are shown in Figure 5-13. Having a chiller set point temperature of 10°C (50°F) caused the flow rate to be maximized for the whole time and caused the inlet air temperature to the thermal load to rise above the original 24°C (75.2°F). This is shown in the

Figure 5-14 with inlet air temperatures to the thermal load. The inlet air temperatures to the thermal load is raised above the 24°C (75.2°F) for the clean room and laboratory load profiles because of the high thermal loads. This led to multiple problems in which the systems were not able to recharge or hold the air temperature stability. The blip that occurs on the chilled water line of 40°F (red line) at hour ten is because of the extra flow being pulled to recharge the TES tank.

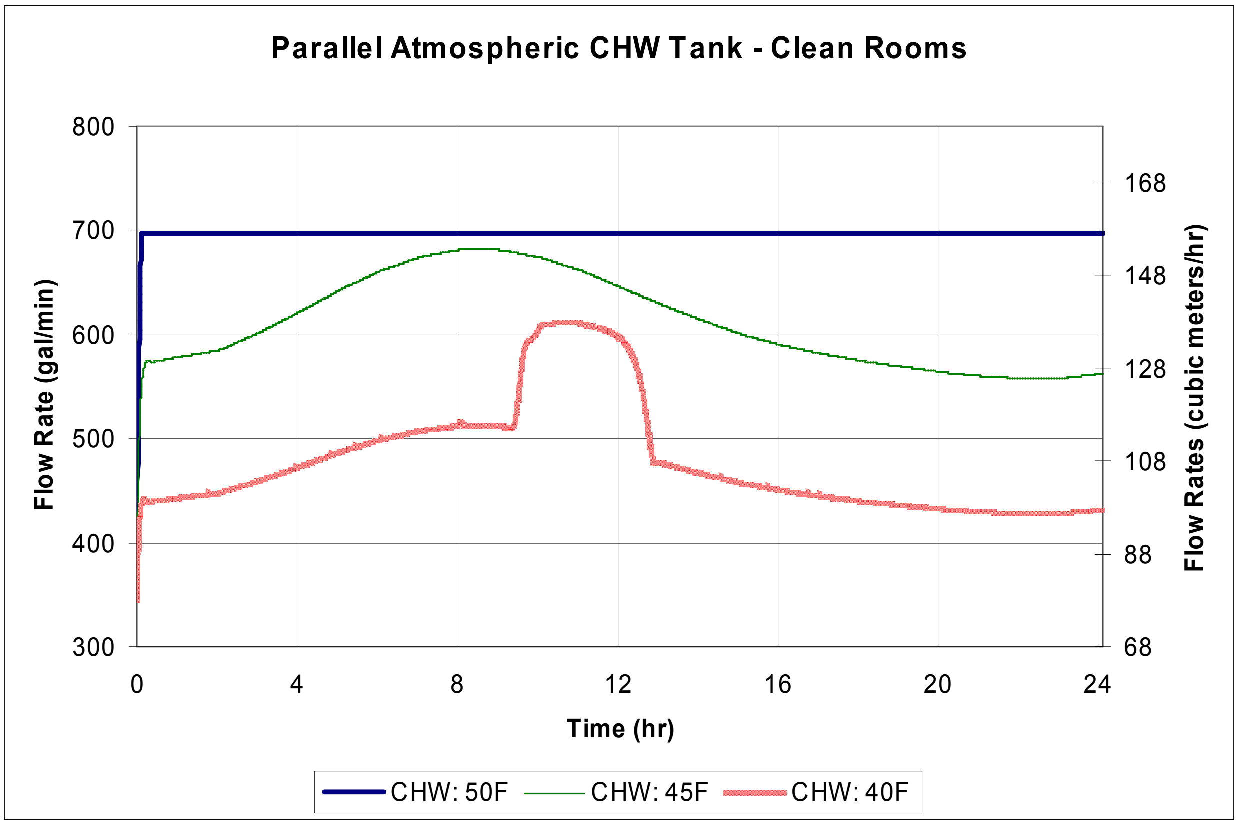

Figure 5-13 Parallel Atmospheric Chilled Water Tank - Flow Rates (Δ10F-Clean Rooms)

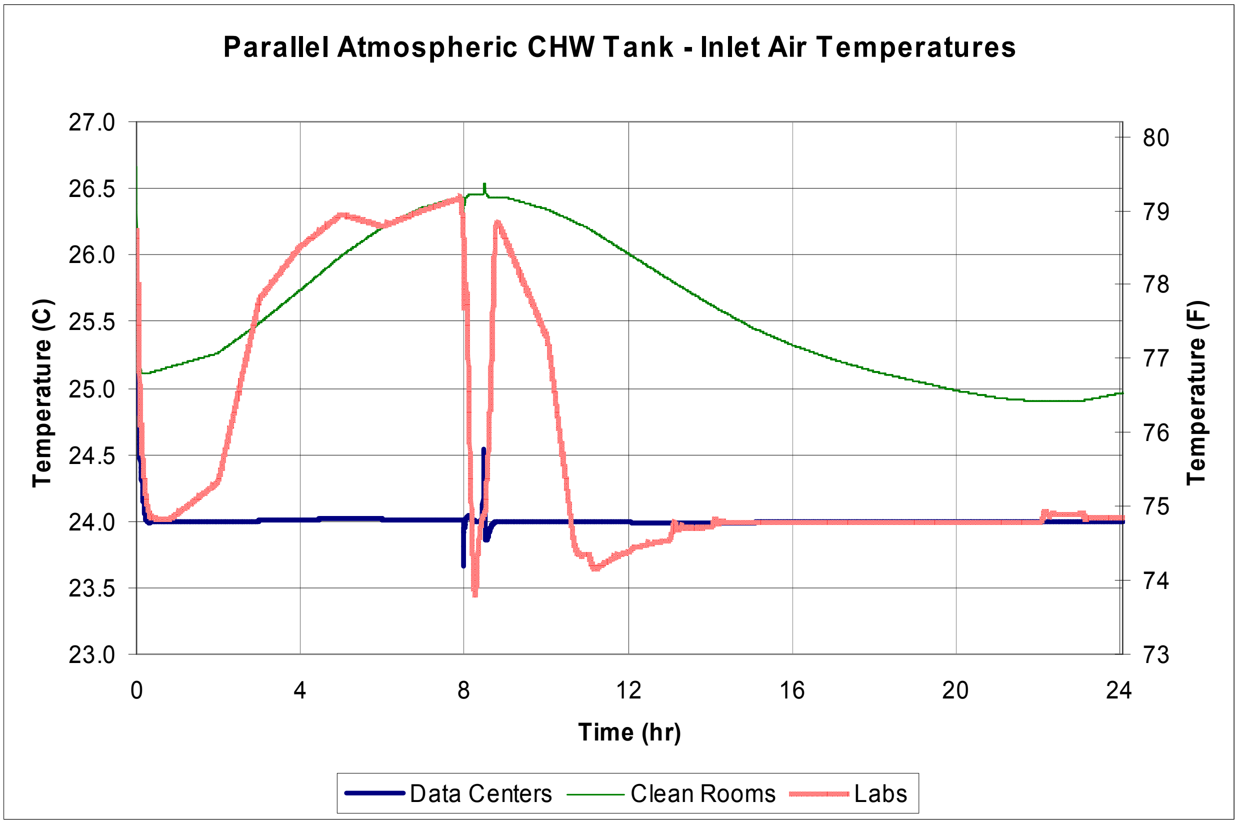

Figure 5-14 Parallel Atmospheric Chilled Water Tank – Inlet Air Temp. (Δ10F-Clean Rooms)

With the chiller set point temperature at 10°C (50°F), the systems that rely on the main chiller to recharge the TES tank either take longer to replenish the thermal energy in the tank or do not recharge at all (clean rooms profile). The systems also had issues with maintaining the air temperature stability at the higher set point temperature. The laboratory profile even failed all together for all simulations at a set point temperature of 10°C (50°F).

At a chiller set point temperature of 4.45°C (40°F), the system was able to meet all of the design conditions but did not operate in the way expected. A lower chiller set point temperature caused additional heat transfer in the cooling coils which led to less flow rate needed by thermal load. The spikes in the flow rates at a chiller set point temperature of 4.45°C (40°F) are caused by the additional flow that is available to recharge the system.

The results from changing the chiller set point temperature are the same for all thermal loads and all temperature differences across the thermal load. All systems also displayed the same trends when changing set point temperatures.

Temperature Differences across the Cooling Coils

Increasing the temperature across the cooling coils from the original difference of 5.56°C (10°F) to 8.33°C (15°F) and 11.11°C (20°F) has a minor effect on the systems. Generally the temperature difference between the supply and return lines had no real effects on the systems except for the recharging time of tank. This can be seen in Figure 5-15 as the overall tank temperature does not completely reach the original starting point within the twenty-four hour simulation for the higher temperature differences. The tank will eventually recharge if given enough additional time (4 to 8 hours). The temperature of the chilled water within the TES tank rises at hour eight when emergency conditions happen. The system tries to recharge the TES tank by returning the tank temperatures to the original starting point. Simulations that will eventually recharge but not within the twenty-four hour time period will be represented by a “Yes+” on the tables below. In storage systems that use alternative cooling (Low Temp. CHW & Ice), this was not an issue due to these systems having other cooling options than the main chiller. All other storage systems display the same trend in having slower recharging times as the temperature difference across the cooling coil increases.

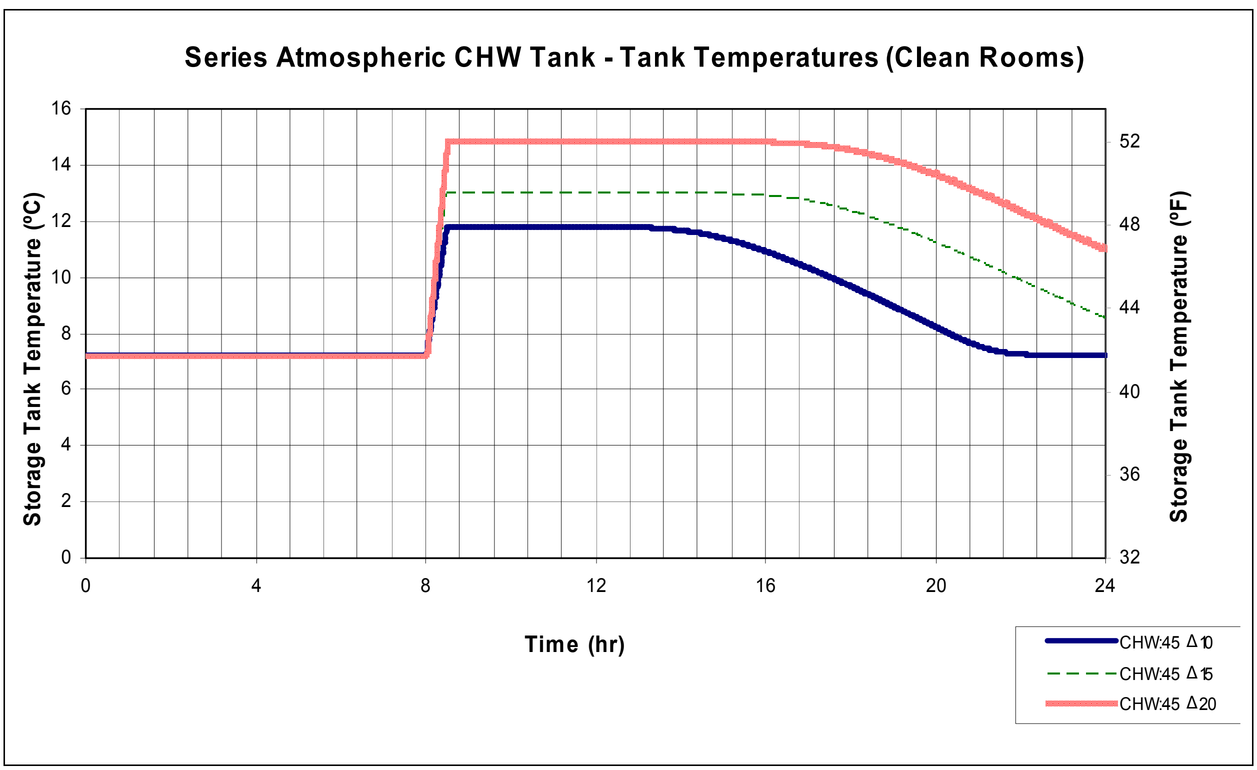

Figure 5-15 Series Atmospheric Chilled Water Tank – Tank Temperatures (Clean Rooms)

Summary Tables

This section shows a summary of the simulations for the storage systems. It shows whether or not the simulation was able to hold the design criteria during the normal operating conditions and states whether or not the TES tank recharges. The values that did not meet the preset standards are in bold.

Table 5-1 Parallel Pressurized Chilled Water Tank

	Parallel Pressurized Chilled Water Tank				
			Average Air Inlet (°C)	Air Stability (±°C)	Recharges
CHW: 40F	Delta T:10F	Data Center	24.00	0.79	Yes
		Clean Rooms	24.00	0.44	Yes
		Labs	23.99	0.77	Yes
	Delta T:15F	Data Center	24.00	0.81	Yes
		Clean Rooms	24.00	0.42	Yes
		Labs	23.99	0.70	Yes
	Delta T:20F	Data Center	24.00	0.74	Yes
		Clean Rooms	24.00	0.75	Yes
		Labs	23.99	0.87	Yes
CHW: 45F	Delta T:10F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.14	Yes
		Labs	23.99	0.94	Yes
	Delta T:15F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.59	Yes
		Labs	23.99	0.80	Yes
	Delta T:20F	Data Center	24.00	0.26	Yes
		Clean Rooms	24.00	0.41	Yes
		Labs	23.99	0.70	Yes
CHW: 50F	Delta T:10F	Data Center	24.00	0.78	Yes
		Clean Rooms	25.58	0.82	**No**
		Labs	24.62	**1.49**	Yes
	Delta T:15F	Data Center	24.00	0.84	Yes
		Clean Rooms	25.67	0.78	**No**
		Labs	24.66	**1.50**	Yes
	Delta T:20F	Data Center	24.00	1.06	Yes
		Clean Rooms	25.58	0.78	**No**
		Labs	24.62	**1.44**	Yes

Table 5-2 Parallel Atmospheric Chilled Water Tank

	Parallel Atmospheric Chilled Water Tank				
			Average Air Inlet (°C)	Air Stability (± °C)	Recharges
CHW: 40F	Delta T:10F	Data Center	24.00	0.79	Yes
		Clean Rooms	24.00	0.44	Yes
		Labs	23.99	0.77	Yes
	Delta T:15F	Data Center	24.00	0.81	Yes
		Clean Rooms	24.00	0.42	Yes
		Labs	23.99	0.70	Yes
	Delta T:20F	Data Center	24.00	0.74	Yes
		Clean Rooms	24.00	0.75	Yes
		Labs	23.99	0.87	Yes
CHW: 45F	Delta T:10F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.14	Yes
		Labs	23.99	0.94	Yes
	Delta T:15F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.59	Yes
		Labs	23.99	0.80	Yes
	Delta T:20F	Data Center	24.00	0.26	Yes
		Clean Rooms	24.00	0.41	Yes
		Labs	23.99	0.70	Yes
CHW: 50F	Delta T:10F	Data Center	24.00	0.78	Yes
		Clean Rooms	25.58	0.82	**No**
		Labs	24.62	**1.49**	Yes
	Delta T:15F	Data Center	24.00	0.84	Yes
		Clean Rooms	25.67	0.78	**No**
		Labs	24.66	**1.50**	Yes
	Delta T:20F	Data Center	24.00	1.06	Yes
		Clean Rooms	25.58	0.78	**No**
		Labs	24.62	**1.44**	Yes

Table 5-3 Series Atmospheric Chilled Water Tank

		Series Atmospheric Chilled Water Tank			
			Average Air Inlet (°C)	Air Stability (± °C)	Recharges
CHW: 40F	Delta T:10F	Data Center	24.00	0.10	Yes
		Clean Rooms	24.00	0.28	Yes
		Labs	23.99	0.77	Yes
	Delta T:15F	Data Center	24.00	0.13	Yes
		Clean Rooms	24.00	0.22	Yes
		Labs	23.99	0.70	Yes
	Delta T:20F	Data Center	24.00	0.10	Yes
		Clean Rooms	24.00	0.14	Yes
		Labs	23.99	0.64	Yes
CHW: 45F	Delta T:10F	Data Center	24.00	0.17	Yes
		Clean Rooms	24.00	0.14	Yes
		Labs	23.99	0.94	Yes
	Delta T:15F	Data Center	24.00	0.32	Yes
		Clean Rooms	24.00	0.58	Yes+
		Labs	23.99	0.80	Yes
	Delta T:20F	Data Center	24.00	0.26	Yes
		Clean Rooms	24.00	0.42	Yes+
		Labs	23.99	0.74	Yes
CHW: 50F	Delta T:10F	Data Center	24.00	0.57	Yes
		Clean Rooms	25.74	0.81	**No**
		Labs	24.69	**1.54**	Yes
	Delta T:15F	Data Center	24.00	0.67	Yes
		Clean Rooms	25.67	0.78	**No**
		Labs	24.66	**1.50**	Yes
	Delta T:20F	Data Center	24.00	0.68	Yes
		Clean Rooms	25.58	0.78	**No**
		Labs	24.62	**1.44**	Yes

Table 5-4 Parallel Low Temperature Chilled Water Tank

	Parallel Low Temp. Chilled Water Tank				
			Average Air Inlet (°C)	Air Stability (± °C)	Recharges
CHW: 40F	Delta T:10F	Data Center	24.00	0.79	Yes
		Clean Rooms	24.00	0.44	Yes
		Labs	23.99	0.77	Yes
	Delta T:15F	Data Center	24.00	0.81	Yes
		Clean Rooms	24.00	0.42	Yes
		Labs	23.99	0.70	Yes
	Delta T:20F	Data Center	24.00	0.74	Yes
		Clean Rooms	24.00	0.75	Yes
		Labs	23.99	0.87	Yes
CHW: 45F	Delta T:10F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.14	Yes
		Labs	23.99	0.94	Yes
	Delta T:15F	Data Center	24.00	0.31	Yes
		Clean Rooms	24.00	0.59	Yes
		Labs	23.99	0.80	Yes
	Delta T:20F	Data Center	24.00	0.26	Yes
		Clean Rooms	24.00	0.41	Yes
		Labs	23.99	0.70	Yes
CHW: 50F	Delta T:10F	Data Center	24.00	0.78	Yes
		Clean Rooms	25.61	0.82	Yes
		Labs	24.59	**1.51**	Yes
	Delta T:15F	Data Center	24.00	0.84	Yes
		Clean Rooms	25.64	0.78	Yes
		Labs	24.66	**1.43**	Yes
	Delta T:20F	Data Center	24.00	1.06	Yes
		Clean Rooms	25.54	0.78	Yes
		Labs	24.61	**1.46**	Yes

Table 5-5 Series Ice Storage Tank

	Series Ice Storage Tank				
			Average Air Inlet (°C)	Air Stability (± °C)	Recharges
CHW: 40F	Delta T:10F	Data Center	24.00	0.25	Yes
		Clean Rooms	24.00	0.34	Yes
		Labs	23.99	0.84	Yes
	Delta T:15F	Data Center	24.00	0.19	Yes
		Clean Rooms	24.00	0.27	Yes
		Labs	23.99	0.85	Yes
	Delta T:20F	Data Center	24.00	0.18	Yes
		Clean Rooms	24.02	0.24	Yes
		Labs	23.99	0.71	Yes
CHW: 45F	Delta T:10F	Data Center	24.00	0.22	Yes
		Clean Rooms	24.00	0.19	Yes
		Labs	23.99	0.95	Yes
	Delta T:15F	Data Center	24.00	0.35	Yes
		Clean Rooms	24.00	0.64	Yes
		Labs	23.99	0.84	Yes
	Delta T:20F	Data Center	24.00	0.32	Yes
		Clean Rooms	24.00	0.38	Yes
		Labs	23.99	0.78	Yes
CHW: 50F	Delta T:10F	Data Center	24.00	0.63	Yes
		Clean Rooms	25.69	0.88	Yes
		Labs	24.68	**1.57**	Yes
	Delta T:15F	Data Center	24.00	0.69	Yes
		Clean Rooms	25.61	0.82	Yes
		Labs	24.65	**1.46**	Yes
	Delta T:20F	Data Center	24.00	0.76	Yes
		Clean Rooms	25.53	0.84	Yes
		Labs	24.61	**1.54**	Yes

Storage Tank Recharging

This section deals with how each system recharges the TES tank. The system with a chilled water TES tank in series has problems with recharging because the tank can not be isolated from the thermal load. This means that the hot water stored in the TES tank after emergency conditions must also pass through the thermal load before reaching the chiller. This causes recharging times of the TES tanks in series to increase. The TES tanks in parallel can recharge without passing the hot water in the tank through the thermal load. This means the TES tanks in parallel can recharge much faster.

A TES tank with alternative cooling (Ice & Low Temperature CHW) is not affected by the system's configuration in recharging. This is a major advantage to having a tank with alternative cooling that does not have to be recharged by the main chiller.

Ragged Cooling Conditions

A significant amount of additional simulations were also carried out to observe the amount of time the system can last past the original 30 minutes before reaching the upper limits of the recommended temperature ranges. The upper temperature limit for each cooling load profile was discussed in chapter two. The ragged cooling effects as the air inlet temperature approaches the upper limit of each system type are discussed below.

Figure 5-16 shows what ragged cooling looks like for the system with a parallel atmospheric chilled water tank using the laboratory load profile. The inlet air temperature was allowed to rise to the upper limit of 29.5°C (86.1°F) while the temperatures in the TES tank also rose. The TES tank also became less stratified during ragged cooling.

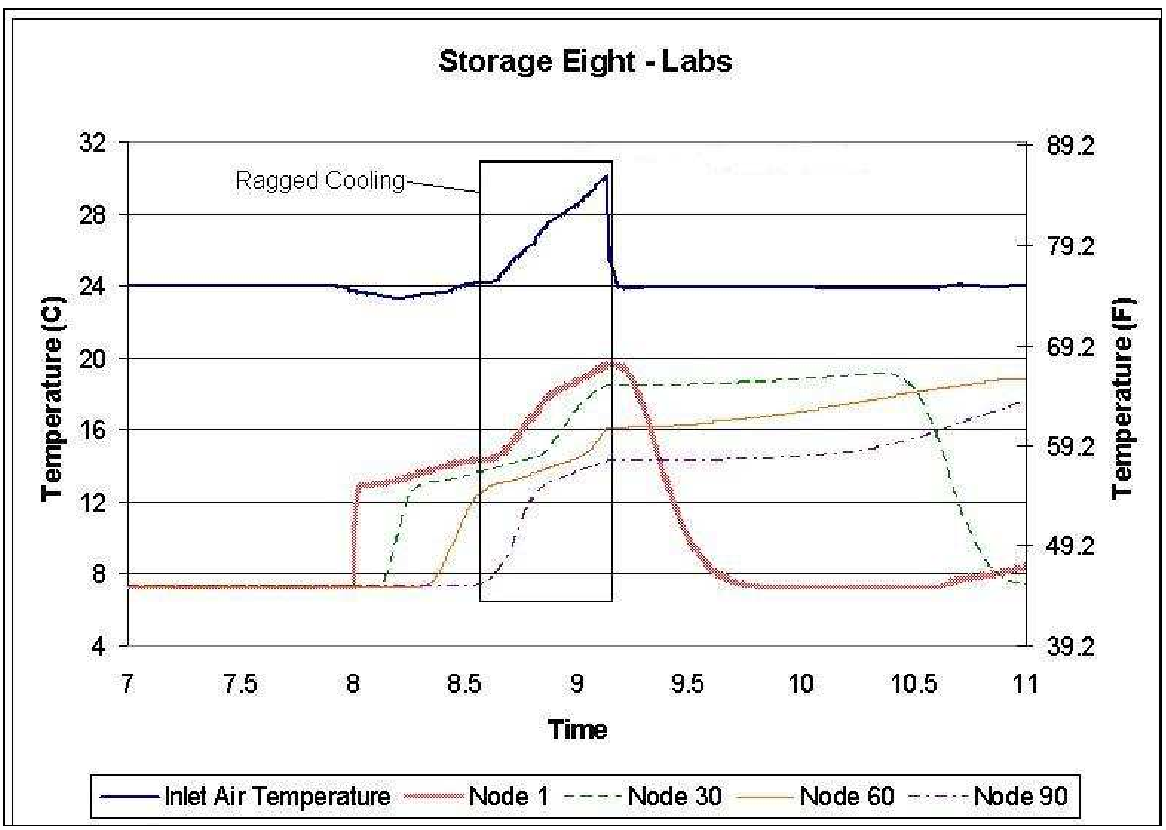

Figure 5-16 Ragged Cooling

Figures 5-17 through 5-19 show the amount of time the ragged cooling will last for the three different storage types. The amount of extra time is dependent upon the temperature difference across the cooling coils and the cooling load profile used. Ice and low temperature chilled water both have a high energy density and thus a smaller tank. While this may work for the original 30 minutes of cooling needed, the amount of time ragged cooling last is greatly reduced. After the tank has been emptied of the original high energy density fluid in it, the smaller tank temperatures increase more rapidly compared to regular chilled water. A regular chilled water tank size will be quite large and more able to support ragged cooling longer.

The y-axis Figures 5-17 through 5-19 gives the extra time (in minutes beyond the original 30 minutes of cooling) before the inlet air temperature to the thermal load reaches the upper temperature limit. The x-axis is the temperature difference (10°F, 15°F, & 20°F) across the cooling coils of the chilled water.

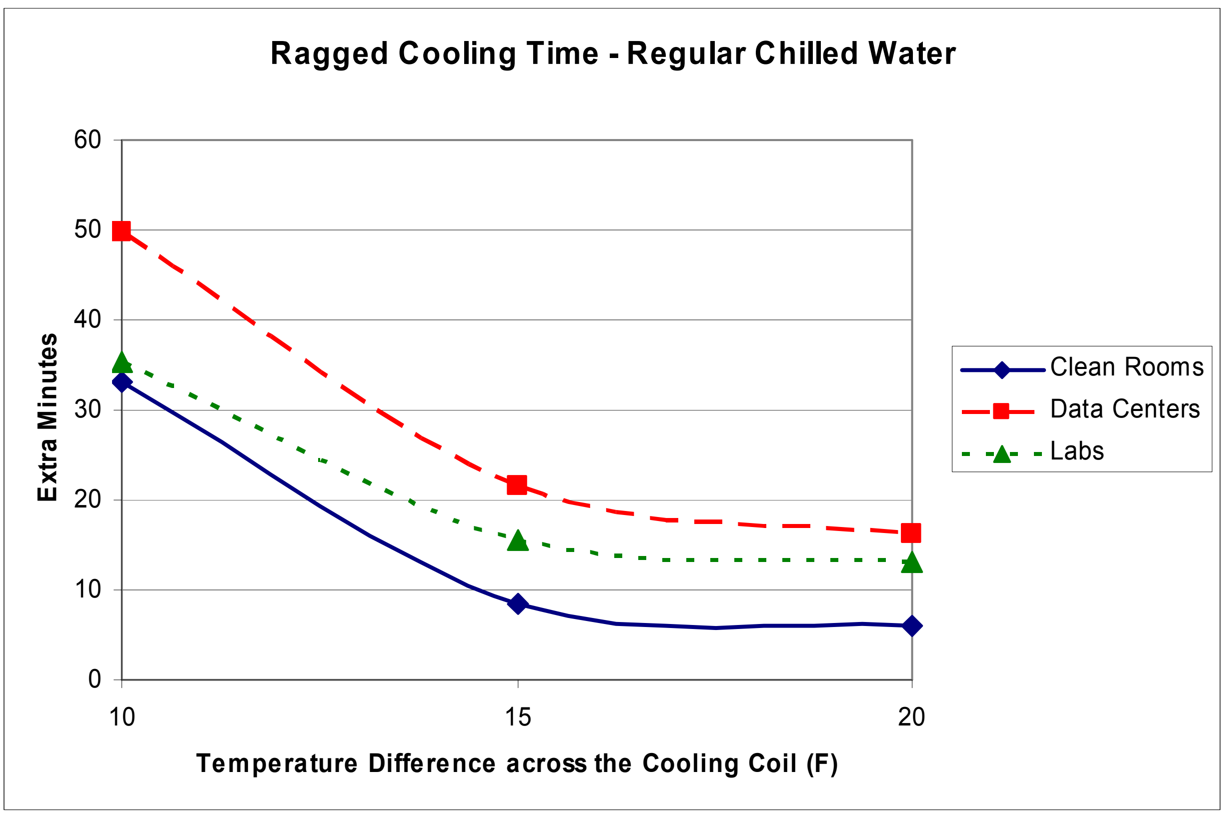

Figure 5-17 Ragged Cooling Time - Regular Chilled Water

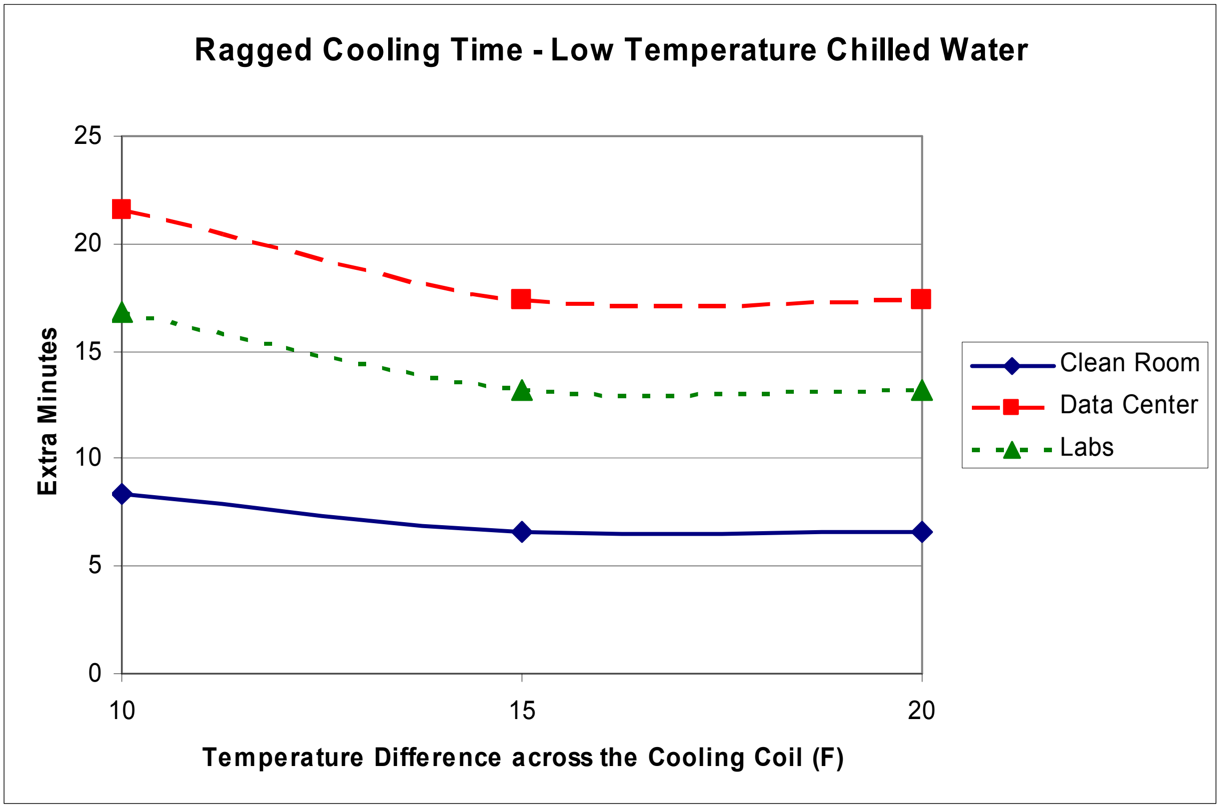

Figure 5-18 Ragged Cooling Time - Low Temperature Chilled Water

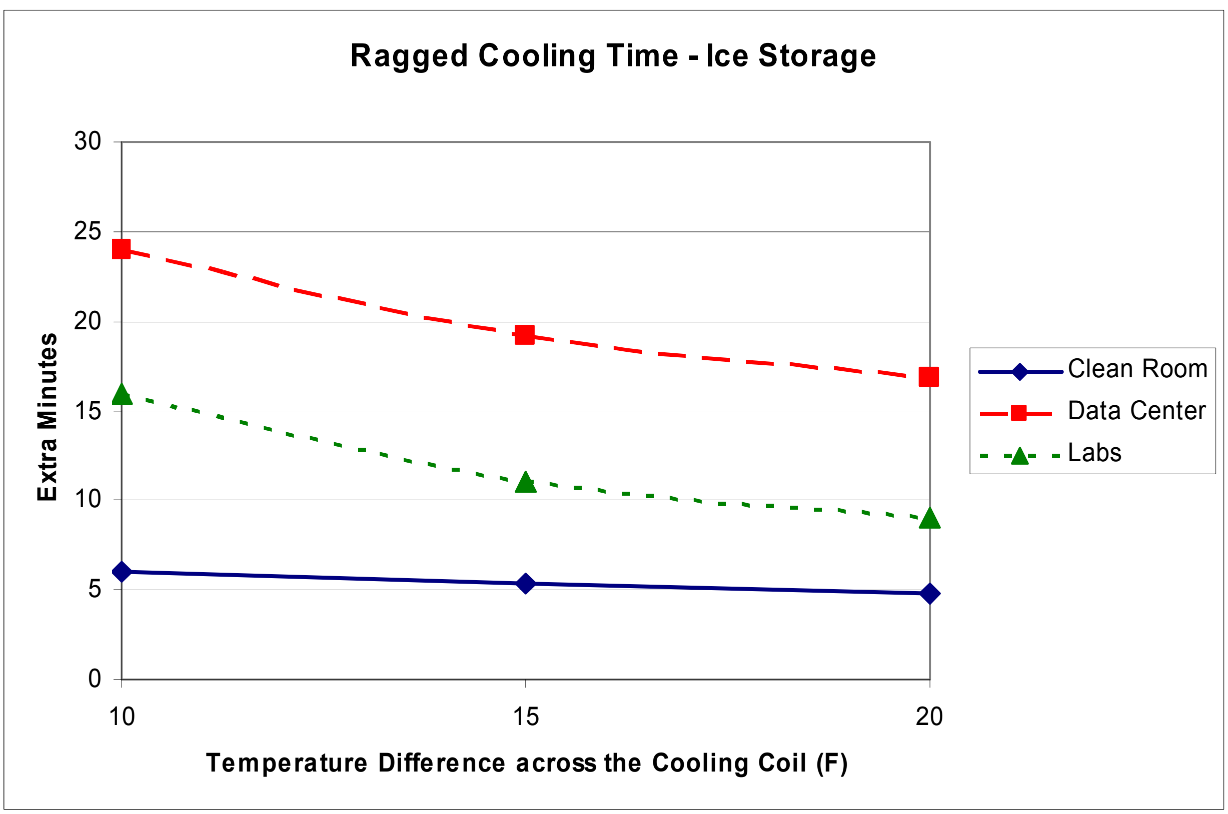

Figure 5-19 Ragged Cooling Time - Ice Storage

Table 5-6 Ragged Cooling-Temperature Rise

Regular Chilled Water			Low Temp. Chilled Water			Ice Storage		
Clean Rooms			**Clean Rooms**			**Clean Rooms**		
Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)
10	5.50	3.06	10	1.40	0.78	10	1.00	0.56
15	1.40	0.78	15	1.10	0.61	15	0.90	0.50
20	1.00	0.56	20	1.10	0.61	20	0.80	0.44
Data Centers			**Data Centers**			**Data Centers**		
Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)
10	14.23	7.90	10	6.17	3.43	10	6.86	3.81
15	6.17	3.43	15	4.97	2.76	15	5.49	3.05
20	4.63	2.57	20	4.97	2.76	20	4.80	2.67
Laboratories			**Laboratories**			**Laboratories**		
Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)	Δ Temp. (F)	Minutes/Degree (C)	Minutes/Degree (F)
10	6.44	3.58	10	3.05	1.70	10	2.91	1.62
15	2.84	1.58	15	2.40	1.33	15	2.00	1.11
20	2.40	1.33	20	2.40	1.33	20	1.64	0.91

The table above gives the amount of time ragged cooling will last per degree the inlet air temperature rises. The table is divided into the three different storage types and each temperature difference has a different rate. A designer of a storage system could use this table to determine how long ragged cooling will last for his/her system.

Design Criteria

During the analysis of the simulation results of the five systems, features were discovered that are important to emergency cooling design. These features, called design criteria, are listed below. Many of them were developed by studying the differences between system configurations, tank types, and changing system parameters.

- TES tanks are needed to provide the necessary cooling in the event of main chiller loss.

This criterion was intuitive after conducting research and simulations. No method was found where in the event the main chiller losses power can the system effectively control the ambient air temperatures. Figure 5-20 shows that without a TES tank, how fast the ambient air temperature will rise during emergency conditions at hour eight. The ambient air temperature is over 38°C (100.4°F) in less than five minutes and reaches 60°C (140°F) within nine minutes. All equipment within room would be destroyed by the high temperatures at this point.

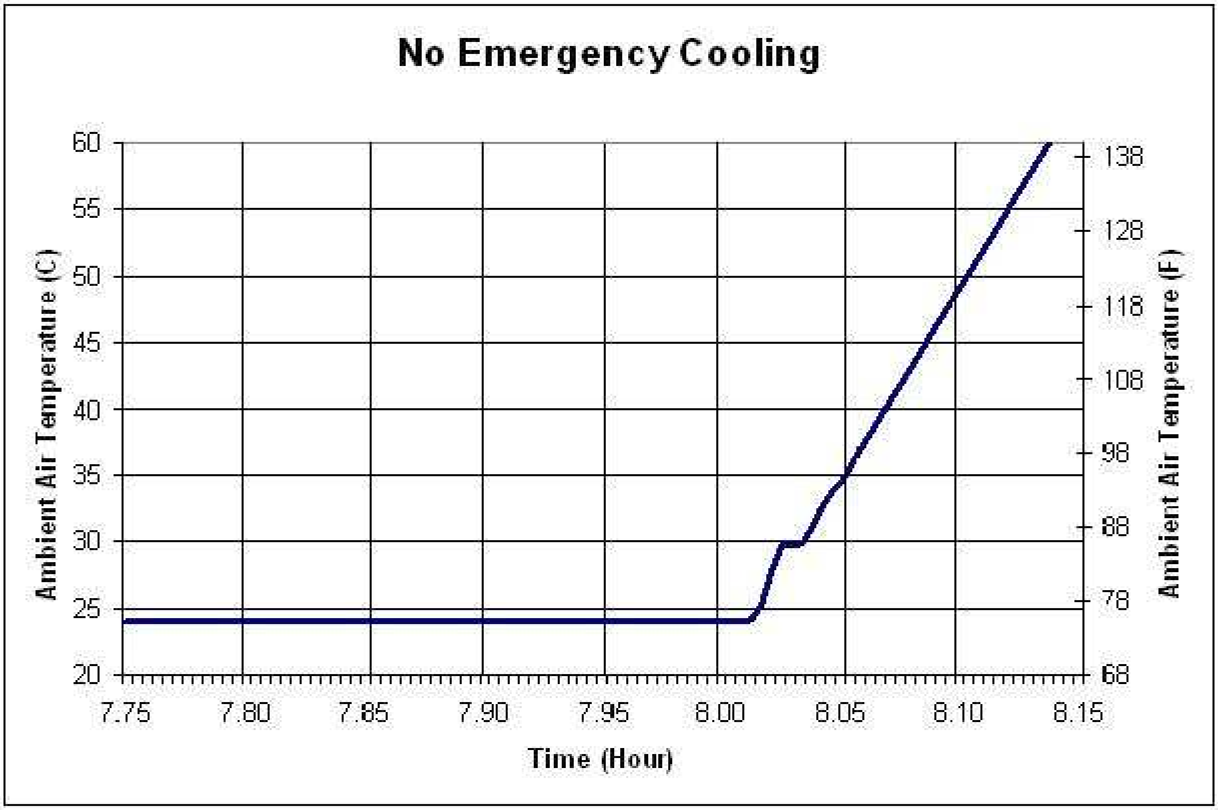

Figure 5-20 No Emergency Cooling

- System parameters have a minor effect on how energy discharges from TES tanks.

This is shown in the energy ratio graphs for the different tank types. Changing the chiller set point temperature had absolutely no effect on the way energy discharged from the tank. The energy ratio graphs were exactly the same when comparing the two. Increasing the temperature difference between the chilled water supply and return lines caused a small variation in the energy ratio of about three percent. Figure 5-21 shows how small the change was when the temperature difference was increased.

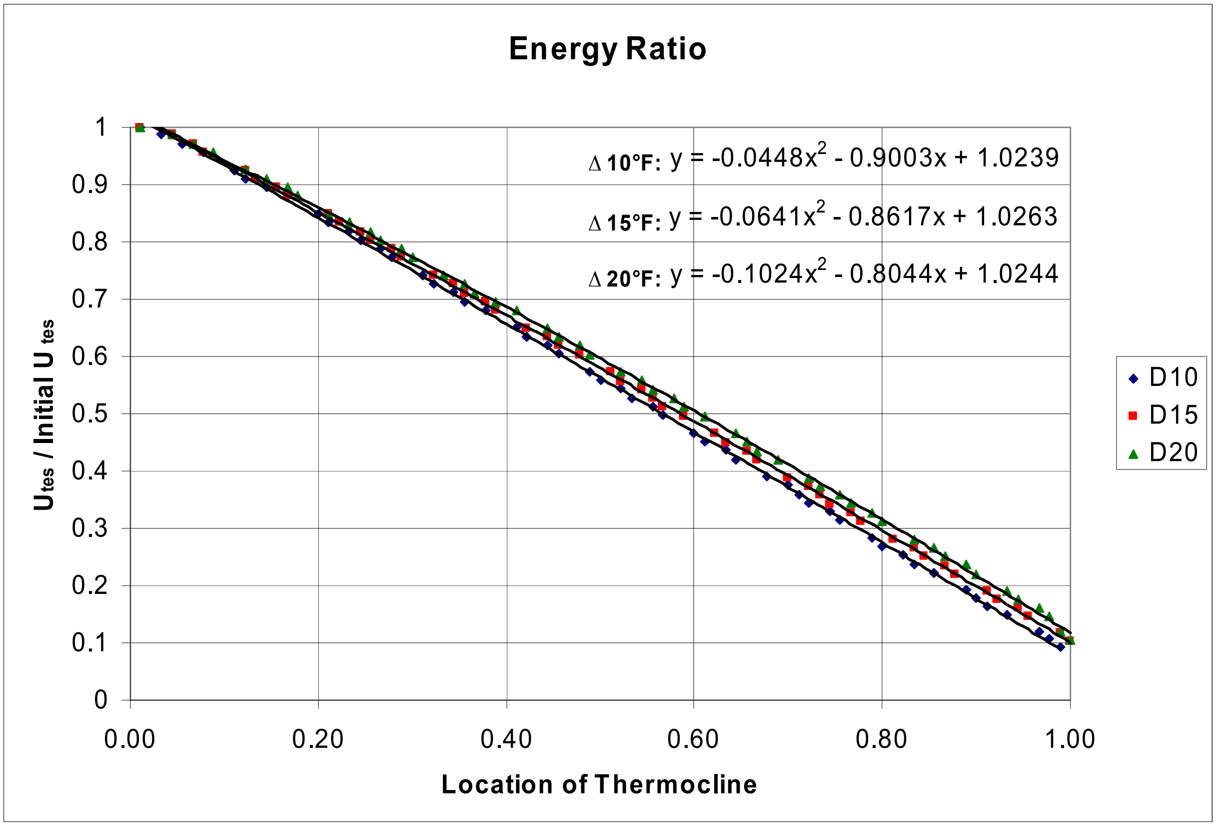

Figure 5-21 Energy Ratio for Chilled Water

- The longer emergency cooling lasts, the larger the thermocline layer becomes.

With all of the TES tanks stratified, a thermocline layer forms in the TES tank during discharging. This layer will increase in size and take up a larger percentage of the tank as time progresses. This can be seen in Figure 5-22. This graph is the chilled water temperatures at different locations in a TES tank during emergency conditions. Looking closely at the slope of each line, it can be seen that the slope is decreasing ever slightly as the node location increases. This can be contributed to the thermocline layer taking longer to pass through each node because of its increase in size.

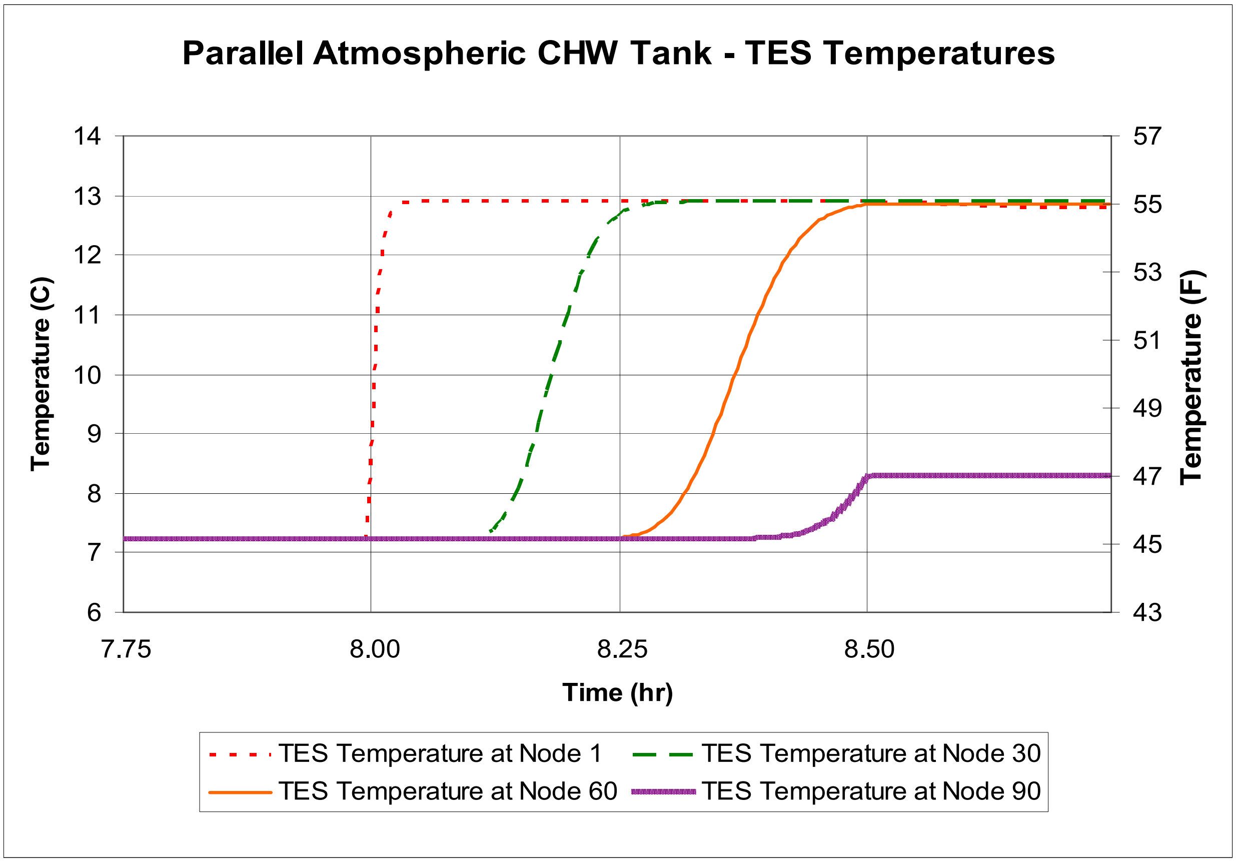

Figure 5-22 Parallel Atmospheric CHW Tank - TES Temperatures (CHW: 45°F Δ 10°F)

- Increasing the chiller set point temperature from original system settings, will require additional flow to cool the thermal load.

Changing the chiller set point temperature without changing any additional components can have repercussions throughout the system. Increasing the chiller set point temperature to 10°C (50°F) causes insufficient heat transfer in the cooling coils. A larger flow rate was required to meet the thermal demands. Figure 5-23 is the flow rates for the laboratory cooling load profile at different chilled water set point temperatures. It shows how the flow rate for the chiller set point temperature of 10°C (50°F) is much higher than the flow rate for the temperature of 7.22°C (45°F) and even reaches the maximum amount of flow that can provided during hours zero to ten.

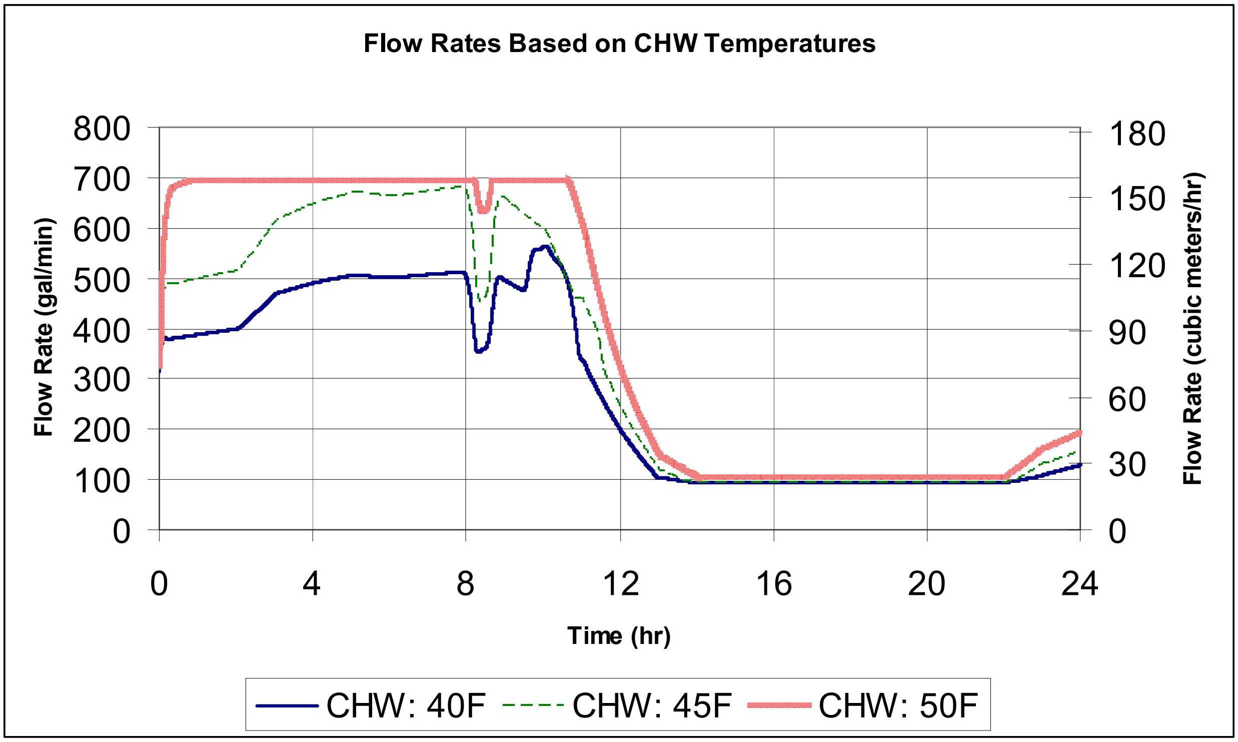

Figure 5-23 Flow Rates based on different CHW Temperature (Labs)

- Decreasing the chiller set point temperature from original system settings, will reduce the amount of flow needed to cool the thermal load.

Changing the chiller set point temperature without changing any additional components can have a significant impact throughout the system. Decreasing the chiller set point temperature to 4.45°C (40°F) led to a smaller flow rate because the cooling coils are designed to utilize an incoming chilled water temperature of 7.22°C (45°F). This can be seen above in Figure 5-23 where the flow rate at 4.45°C (40°F) is the lowest.

- Systems have a difficult time maintaining the ambient air (±) temperature at the preset conditions when the chiller set point temperature is increased from its original settings.

This design criterion goes with the previous two in that with the flow rate being affected as much as it is, the ambient air temperatures will have also changed. Looking at Figure 5-23, it can been seen that the flow rate is at the maximum for a chilled water set point temperature of 10°C (50°F) for the first 10 hours of the simulation. With the flow rate so high, it is doubtful that the ambient air temperature is being maintained at the desired 24°C (75.2°F). This can be seen in Figure 5-24 as the inlet air temperature to the thermal load could not be maintained at 24°C

(75.2°F) when the flow rate was at the maximum. This led to large variations in the inlet air temperature, which cannot be tolerated by a laboratory.

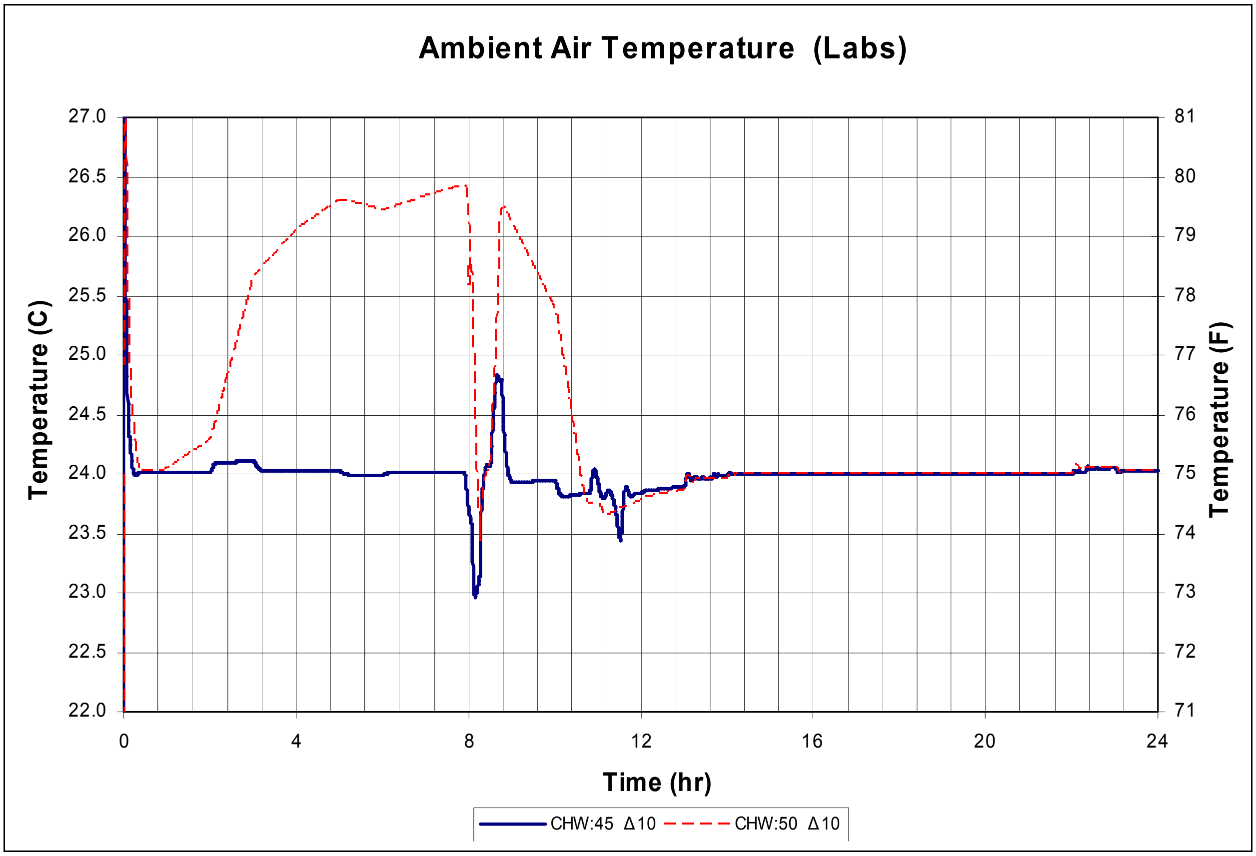

Figure 5-24 Ambient Air Temperatures for different Chiller Set Point Temperatures (Labs)

- TES tanks in parallel configuration with no alternative cooling source can recharge faster than tanks in series configuration.

Only one major difference was found between TES tanks in parallel and series configuration with no alternative cooling source. TES tanks in parallel can effectively be isolated from the thermal load when recharging while the series configuration can not do this. This means that when recharging, a TES tank in series must pass all of the hot water stored in the tank after emergency conditions through the cooling coils, effectively limiting the rate at which it can replenish itself. This is shown in Figure 5-25 with an example of the TES tank temperature of the two different chilled water system configurations. The system in parallel will recharge its tank about four hours faster than the tank in series.

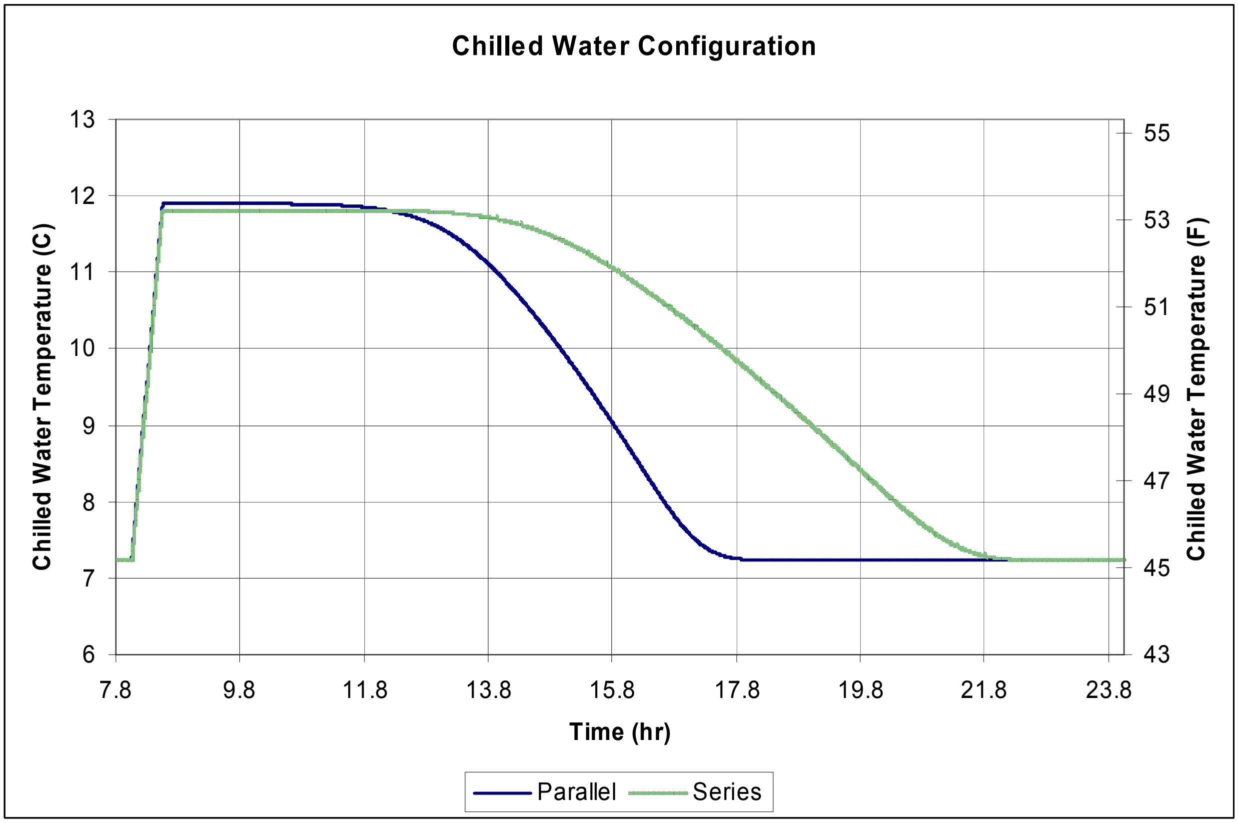

Figure 5-25 Chilled Water Configuration (CHW: 45°F Δ10°F)

- Raising the temperature difference across the cooling coils will increase the amount of time it takes to recharge the TES tanks.

Increasing the temperature difference between the chilled water supply and return lines adds to the time it takes for the TES tanks to recharge. This is true regardless of the system configuration. Figure 5-26 is the TES tank temperatures at different temperature differences across the cooling coils. It gives an example of how the larger temperature differences take longer to recharge the TES tank after emergency conditions

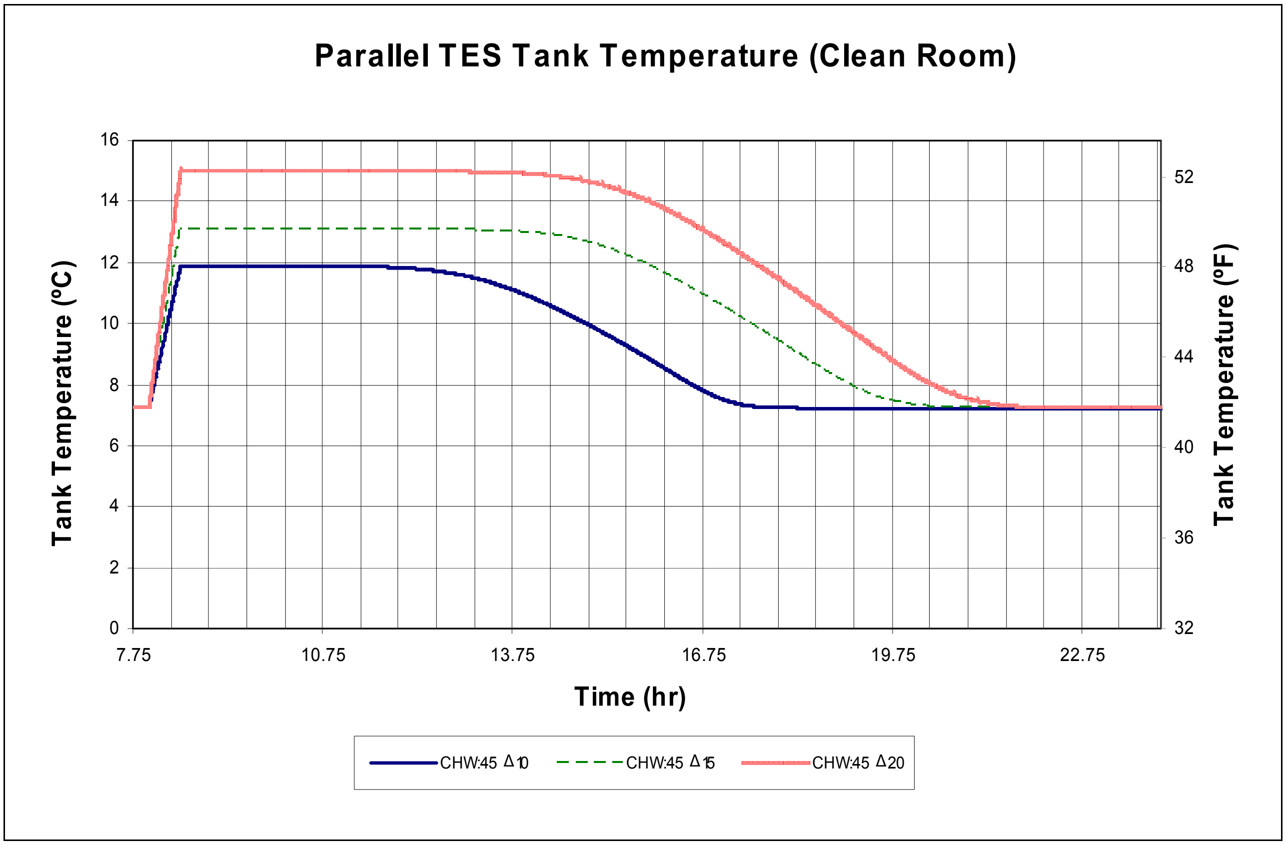

Figure 5-26 Parallel TES Temperatures

- The amount of time ragged cooling lasts depends on which application of emergency was used and the temperature difference across the cooling coil.

Regular chilled water TES storage will provide the longest ragged cooling time when compared to the high energy density medium (e.g. Ice, Low Temp CHW). The large size of a regular chilled water tank makes it better able to handle ragged cooling after the original thermal energy has been discharged. The small size of high energy density tanks does not allow for much ragged cooling after the original energy is gone. This is shown in Figure 5-27 in the time ragged cooling will last using the clean room thermal load profile for the different system types.

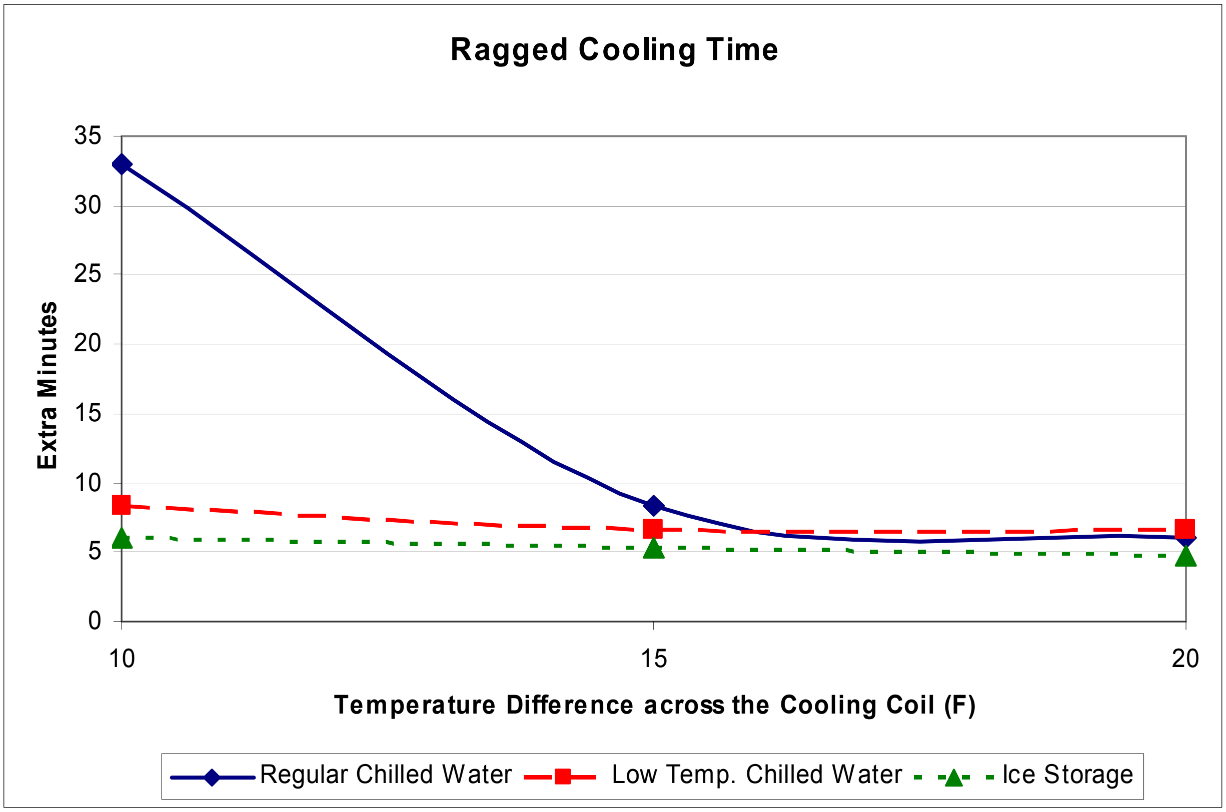

Figure 5-27 Ragged Cooling (Clean Rooms)

- The discharging and recharging characteristics are the same for atmospheric and pressurized TES tanks.

Pressurized and atmospheric TES tanks have the same characteristics when discharging and recharging. There was absolutely no difference in how the tanks react. Both of the TES tanks fit onto the equation lines developed in Figure 5-1 when discharging. The time it takes to recharge the tank is also the same for the two different tanks types.

- Low temperature chilled water tanks are less stratified than regular chilled water tanks.

As the temperature of the TES tank approaches the freezing point of water, stratification of the tank reacts differently. An additive market by the Cool Solutions Company called 'socool' must be added to the water to stabilize the stratification within the tank. The thermocline layer will take up a larger percentage of tank's volume and will have to be accounted for when sizing low temperature TES tanks. The graph below signifies the difference between regular chilled water and low temperature chilled water. The regular chilled water has close to 10% of its mass left in tank when only 10% of the tank's energy is still left. This would be expected of a 90%

stratified TES tank. Low temperature chilled water tanks have over 20% of their mass still left when 90% of their energy has been expended. This is because the thermocline layer has taken up a larger percentage of the tank's overall volume.

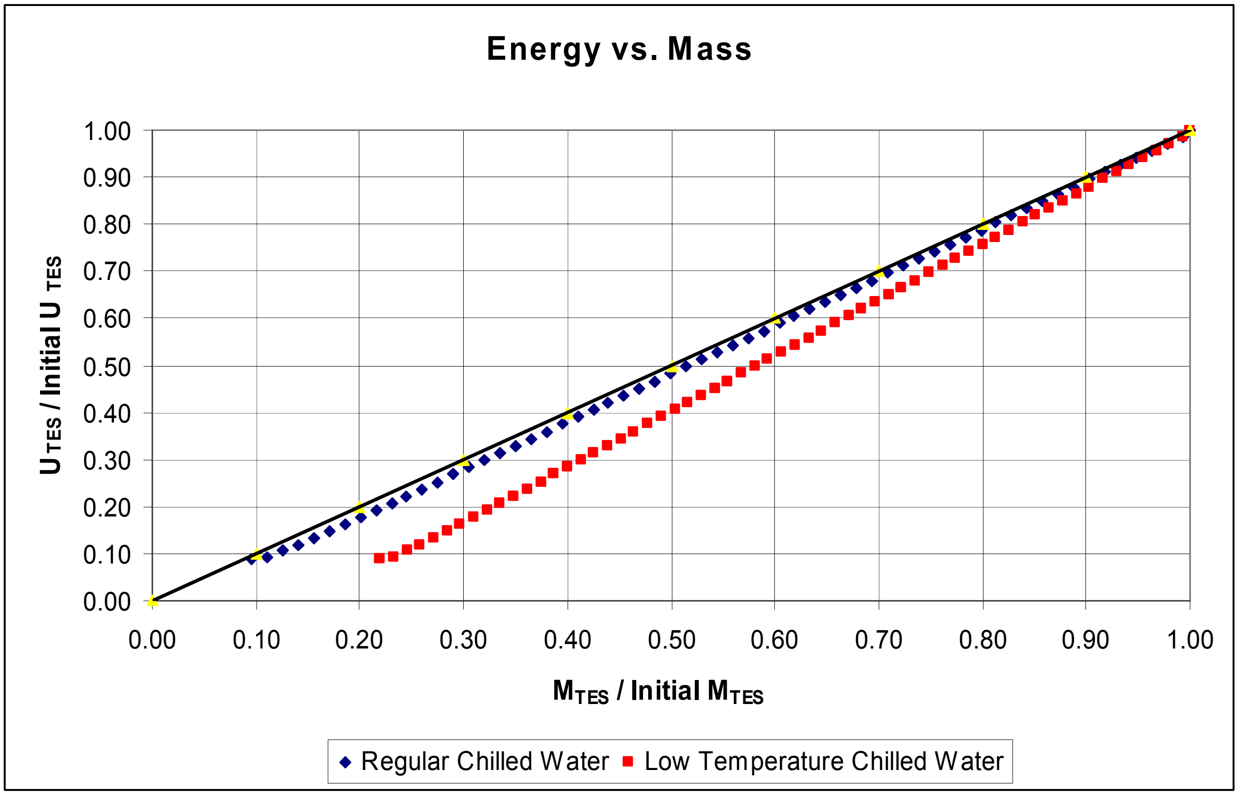

Figure 5-28 Mass vs. Energy (Chilled Water)

- TES tanks with alternative cooling sources will recharge faster than tanks that rely on the main chiller to recharge.

Systems (Ice, Low Temp CHW) that use an alternative cooling source, (e.g. additional chiller, ice builder, refrigeration unit) will replenish the thermal energy in the TES tank the fastest. Systems that rely on the main chiller to recharge the TES tank must wait until the thermal load of the application decreases enough to allow for substantial flow to be diverted to the TES tank. This usually does not happen until nightfall. This is shown in Figure 5-29. The systems that utilize an alternative cooling source were able to recharge two to six hours faster than those that did not. This graph used the clean room thermal cooling load profile which has the highest thermal load demand. If one of the other cooling load profiles (e.g. Data Centers) was used, this characteristic would not be as obvious due to the decrease in thermal load demand. Additional flow could be diverted from the thermal load thus leading to faster recharge times. One other important point to note is that, systems with alternative cooling sources will always recharge the TES tank.

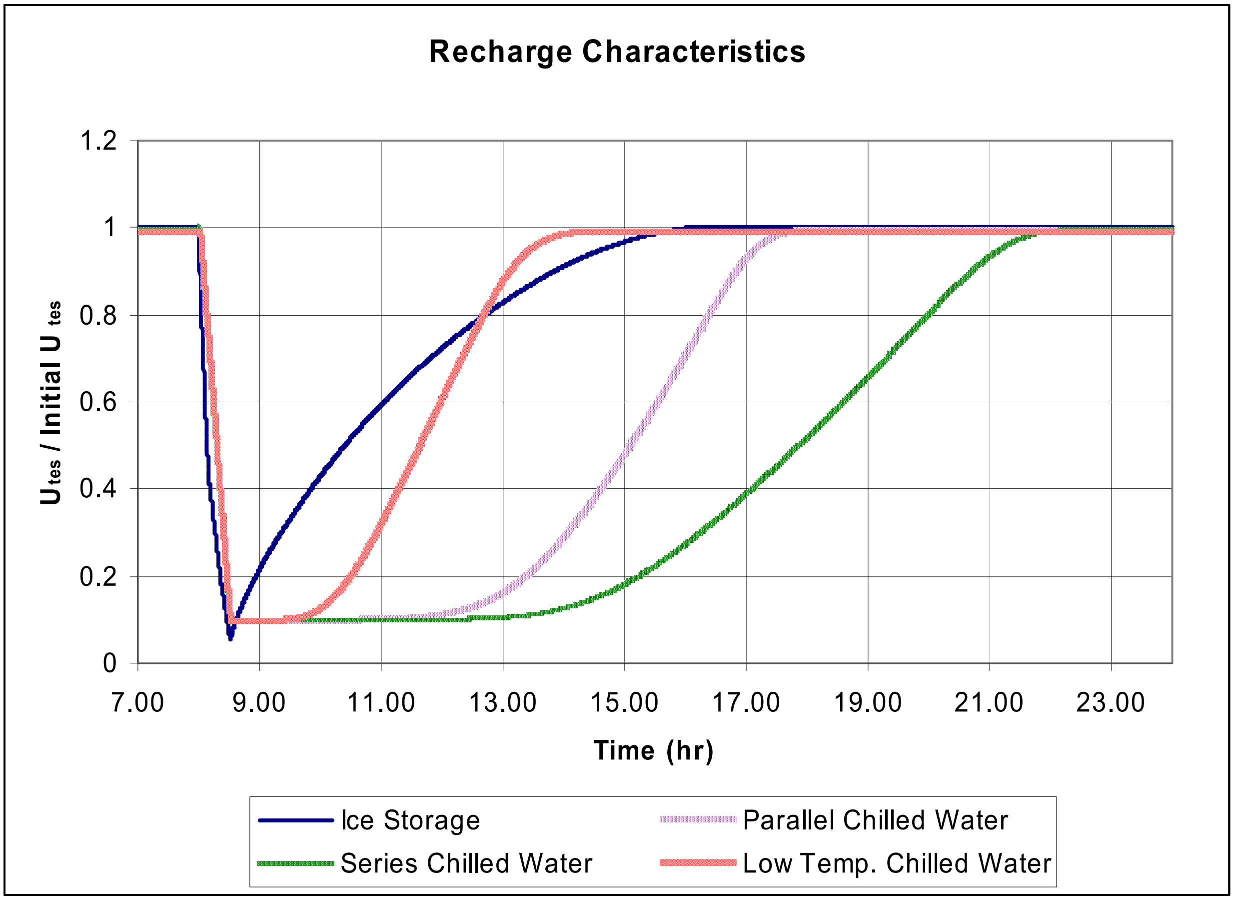

Figure 5-29 Recharge Characteristics-Clean Room (CHW: 45°F Δ10°F)

Summary of Design Criteria

- TES tanks are needed to provide the necessary cooling in the event of main chiller loss.
- System parameters have a minor effect on how energy discharges from TES tanks.
- The longer emergency cooling lasts, the larger the thermocline layer becomes.
- Increasing the chiller set point temperature from original system settings, will require additional flow to cool the thermal load.
- Decreasing the chiller set point temperature from original system settings, will reduce the amount of flow needed to cool the thermal load.
- Systems have a difficult time maintaining the air stability (±) temperature when the chiller set point temperature is increased from original settings.

- TES tanks in parallel configuration with no alternative cooling source can recharge faster than tanks in series configuration.
- Raising the temperature difference across the cooling coils will increase the amount of time it takes to recharge the TES tanks.
- The amount of time ragged cooling lasts depends on which application of emergency is used and the temperature difference across the cooling coil.
- Low temperature chilled water tanks are less stratified than regular chilled water tanks.
- The discharging and recharging characteristics are the same for atmospheric and pressurized TES tanks.
- TES tanks with alternative cooling sources will recharge faster than tanks that rely on the main chiller to recharge.

CHAPTER 6 - Conclusions

The research project my thesis was based on, achieved its finals aims in developing design methodologies for TES systems. Important thermal applications and the design of emergency cooling systems were analyzed in great detail. The design criteria obtained from the simulation results will provide a designer with additional information on how to best maximize the efficiency of their emergency cooling system.

The simulation program (TRNSYS) utilized to generate the results needed for the design methodologies succeeded in accurately simulating the system. The only system in question was the TES tanks that use low temperature chilled water. The additional additives necessary in low temperature chilled water tanks to provide correct stratification required close scrutiny when simulating to make sure TRNSYS accounted for this. All simulations for low temperature chilled water tanks showed that TRNSYS was successful but further research could be conducted into this area.

The research project was also very successful in identifying five emergency cooling systems that are commonly used by companies and providing useful information. While a designer's current system maybe setup slightly different (valves, piping, etc.), this investigation showed that energy discharging from the tank is independent of the setup and configuration (parallel or series). However, the manner in which the tank will recharge after emergency conditions is very dependent on the setup and configuration.

Several design criteria were developed from the simulation results for TES systems. The main points are:

- TES tanks containing chilled water or ice are needed to provide the necessary cooling in the event of main chiller loss. No method was found where, in the event the main chiller loses power, the system can effectively control the air temperature supplied to the the thermal load. Air temperatures may soar to over 53.3°C (128°F) within six minutes at which point the equipment within the room would be destroyed.

- System parameters have a minor effect on how energy discharges from TES tanks. Changing the chiller setpoint temperature or the temperature difference between the chilled water supply and return lines had little to no effect on how energy discharged from the TES tank.

- Chilled water TES tanks in parallel configuration with no alternative cooling source can recharge faster than TES tanks in series configuration. TES tanks in parallel can effectively be isolated from the thermal load when recharging while the series configuration cannot do this. A TES tank in series must pass all of the hot water stored in the tank after emergency conditions through the cooling coils before reaching the chiller when recharging. In effect, limiting the rate at which it can replenish the TES tank.

- Raising the temperature difference between the chilled water supply and return line will increase the amount of time it takes to recharge the TES tanks. Increasing the temperature difference will decrease the pump size thus causing longer recharging times.

- The amount of time ragged cooling will last depends on which application of emergency cooling is used and what type of medium is used for emergency cooling. Low temperature chilled water and ice storage use smaller high energy density tanks that do not allow for much ragged cooling after the original energy (cooling capacity) is gone. Regular chilled water uses large TES tanks that are more able to provide ragged cooling after the original energy is used up.

- The discharging and recharging characteristics are the same for atmospheric and pressurized TES tanks. The results from the simulations showed no difference in the characteristics between atmospheric and pressurized TES tanks.

Research conducted for this project also showed other uses for emergency cooling. Several applications were discovered where TES tanks were utilized for load shifting during peak hours. These TES tanks also double as emergency cooling when needed. While this was not studied in this project, additional research could be conducted into this area to further the information available on TES tanks. Also, high energy density methods (i.e. compressed air) may become a viable option to provide emergency cooling if the applications become modular where then thermal load for each module is relatively small.

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Appendix A - TRNSYS Screenshots

The screenshots from TRNSYS of the five systems designed from chapter three are shown here. The level of complexity taken into account for each system can be seen in the figures.

TRNSYS Screenshot of Parallel Pressurized TES Tank

The figure below is a screenshot of what a storage system that uses a pressurized TES tank in parallel looks like in TRNSYS at a thermal load of one megawatt with all of its components.

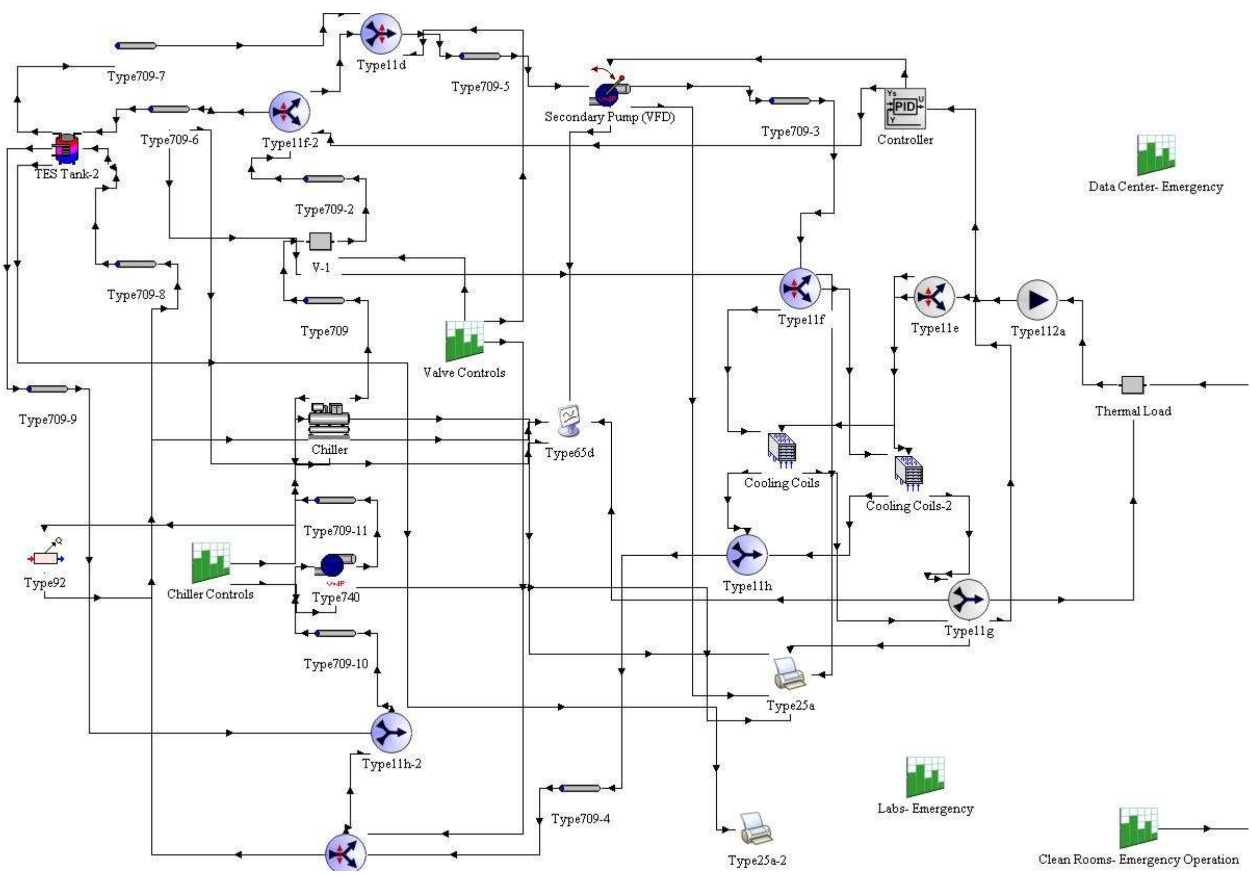

Figure A-1 Parallel Pressurized TES Tank (1 MW)

TRNSYS Screenshot of Parallel Atmospheric TES Tank

The figure below is a screenshot of a storage system that uses an atmospheric TES tank in parallel looks like in TRNSYS at a thermal load of one megawatt with all of its components.

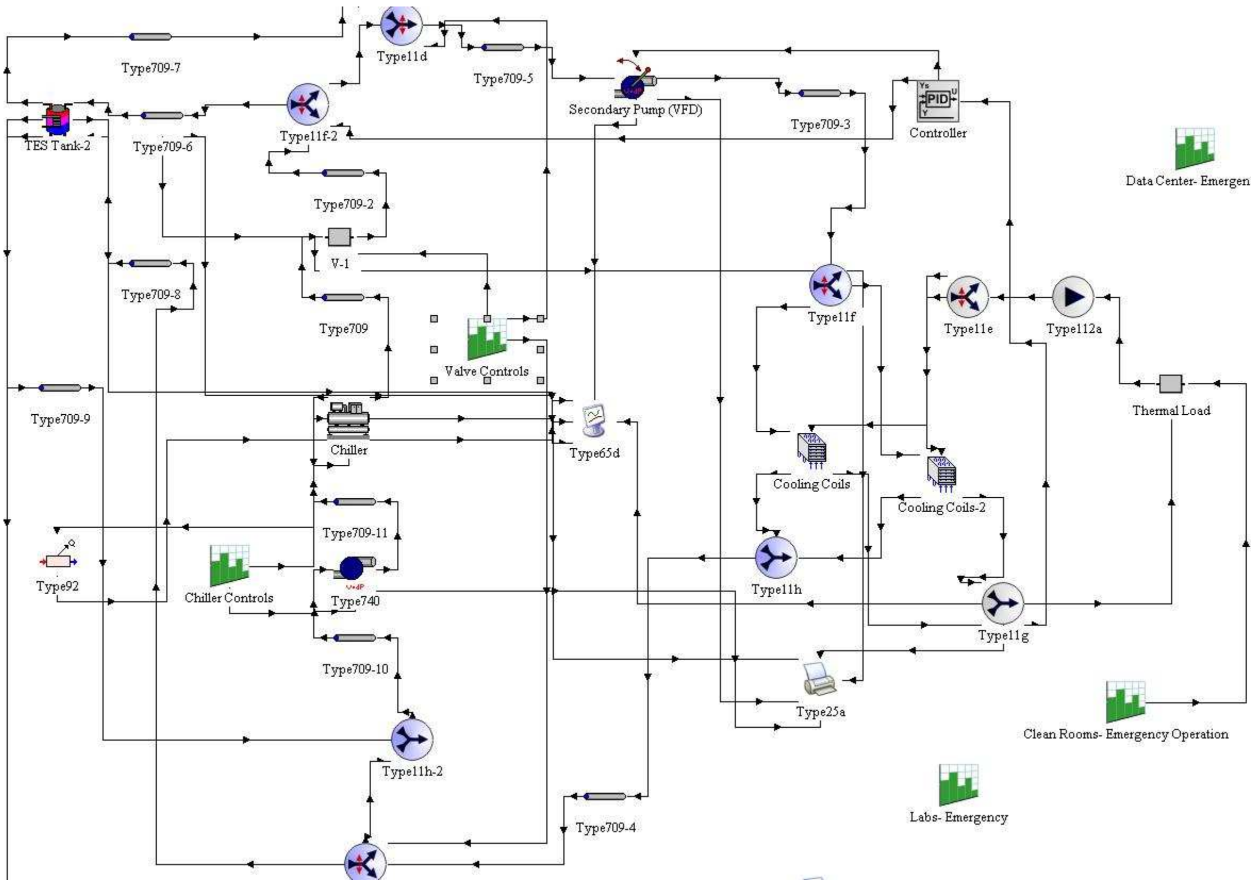

Figure A-2 Parallel Atmospheric Chilled Water Tank (1 MW)

TRNSYS Screenshot of Series Atmospheric TES Tank

The figure below is a screenshot of what a storage system that uses an atmospheric TES tank in series looks like in TRNSYS at a thermal load of five megawatts with all of its components.

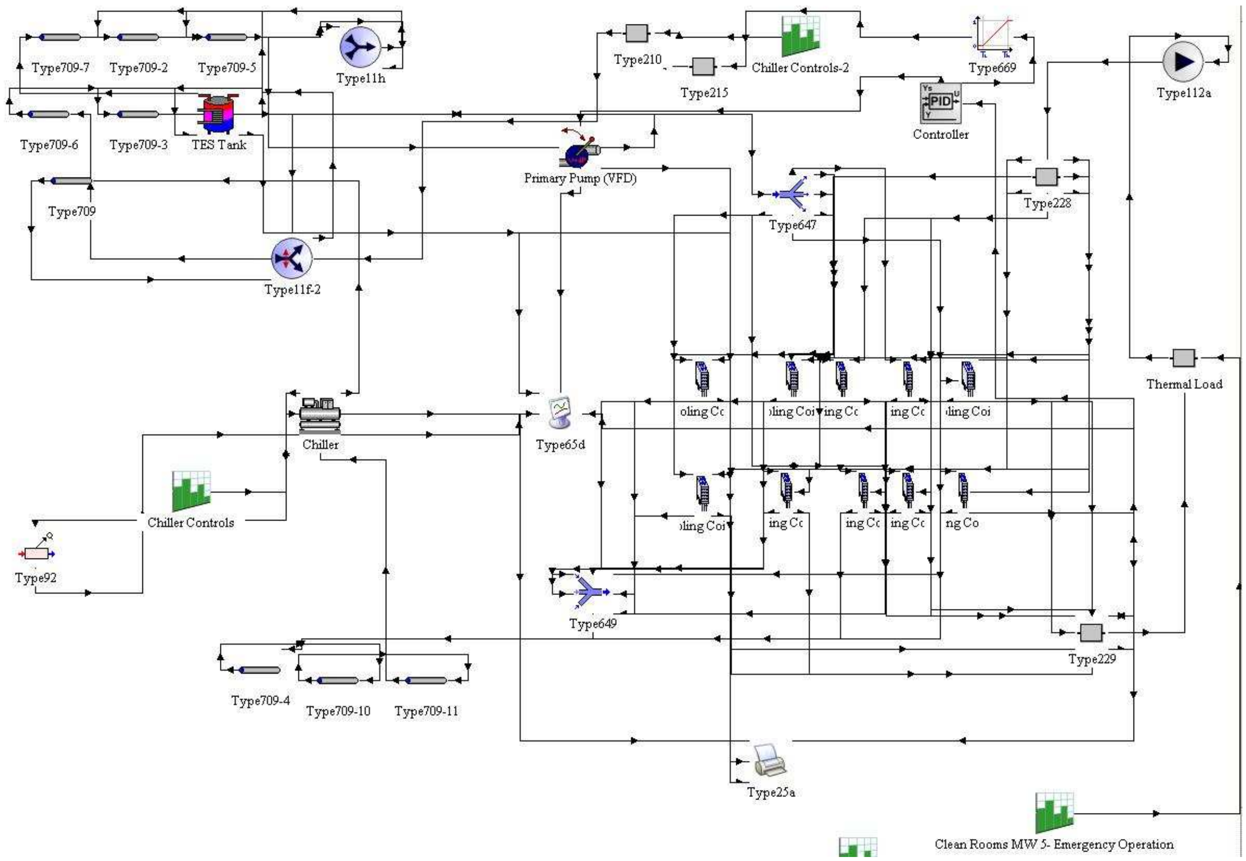

Figure A-3 Series Atmospheric Chilled Water Tank (5 MW)

TRNSYS Screenshot of Parallel Low Temperature TES Tank

The figure below is a screenshot of what a storage system that uses a low temperature TES tank in parallel looks like in TRNSYS at a thermal load of fifteen megawatts with all of its components.

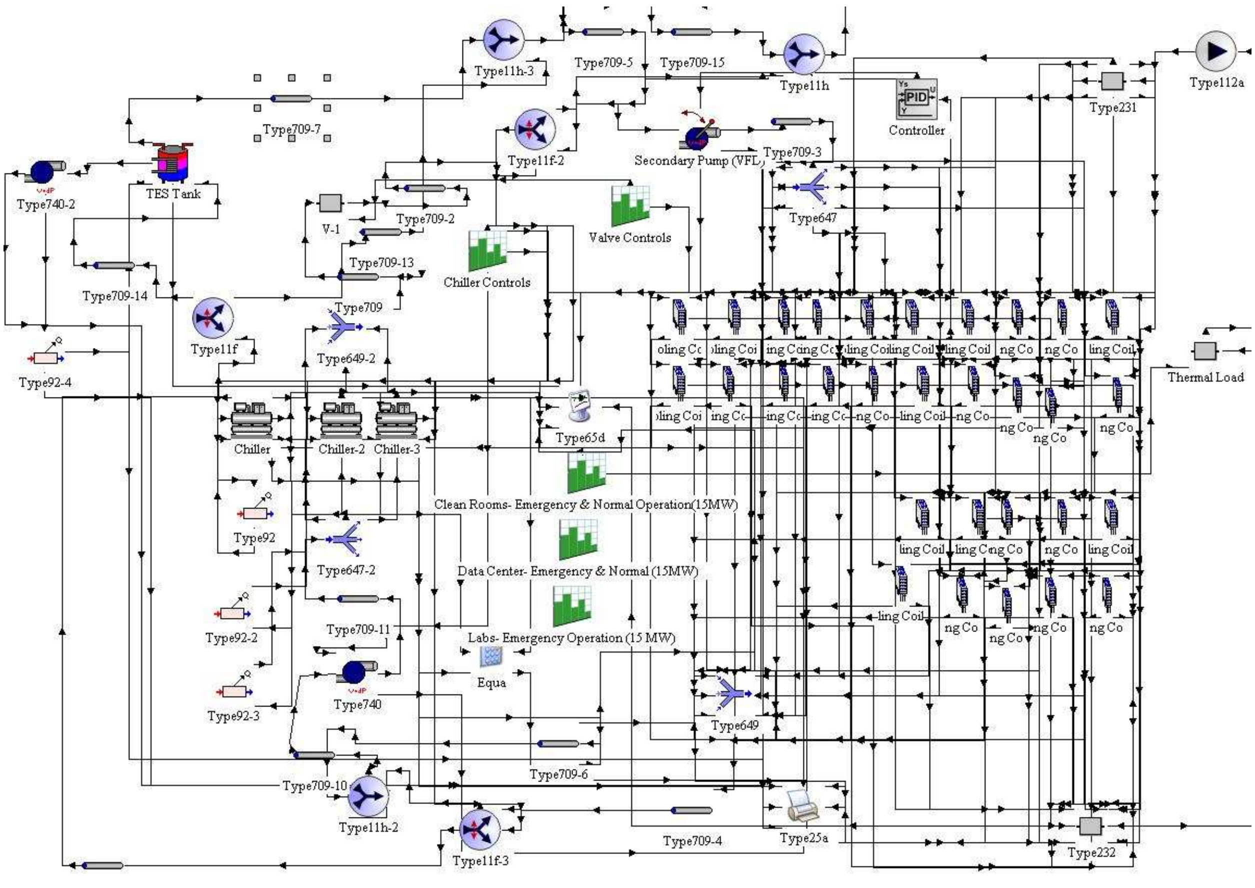

Figure A-4 Parallel Low Temperature Chilled Water Tank (15 MW)

TRNSYS Screenshot of Series Ice Storage TES Tank

The figure below is a screenshot of what a storage system that uses an ice storage tank in series looks like in TRNSYS at a thermal load of five megawatts with all of its components.

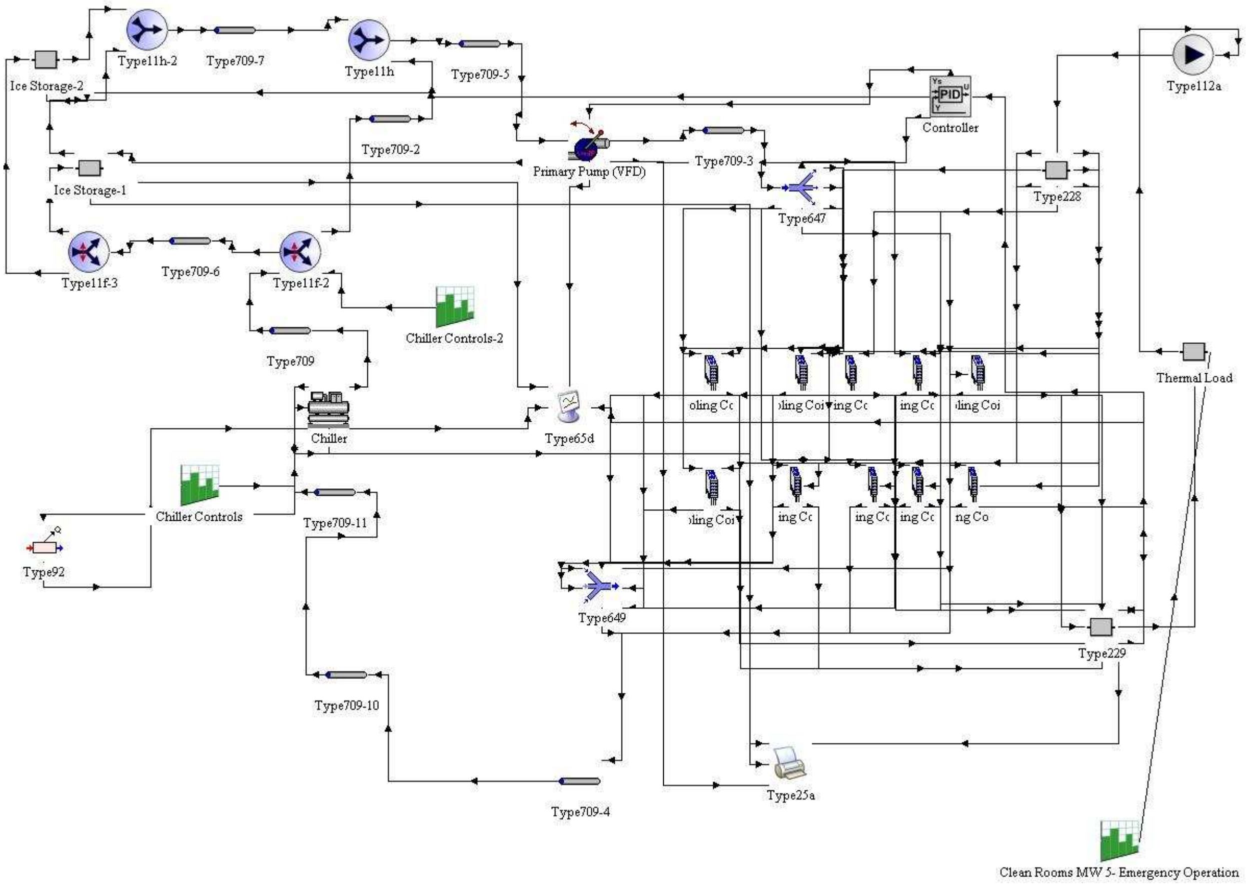

Figure A-5 Series Ice Storage Tank (5 MW)

Appendix B - Component Sizing

The following section deals with the sizing of the components for the systems. So that the performance of different emergency cooling systems can be compared on a "level" field several system characteristics and operating parameters are assumed to be same for all systems. Consequently, performance results of the system simulations can then be directly compared to one another. Multiple thermal loads will be used in the simulations but the components were sized to cool a thermal load of 1 MW (3.413 MBtu/hr) in this section. For different thermal loads (ex. 10 MW or 15 MW) the equations and method of sizing the components will the same. The reference conditions are:

- The TES tank must be able to provide at least 30 minutes of cooling without assistance from the chillers.
- The TES tank must also recharge sometime during the simulation.

Using these conditions, the sizes of certain components in the system are determined.

Chiller

The chiller was designed to meet a load of one megawatt. The chillers also have maximum load of 10% higher than what they were designed for redundancy. The chiller usually has a set point temperature between 4.45°C (40°F) to 12.78°C (55°F) for chilled water. Any set point temperature below 4.45°C (40°F) is considered low temperature chilled water while all temperatures above will be considered regular chilled water. The set point temperature had to be set low enough to provide adequate cooling but high enough above the freezing point to prevent freezing conditions. The temperature of the chilled water returning to the chiller is usually between 10°C (50°F) to 21.11°C (70°F). The difference between the supply and return temperatures of the chilled water is typically 5.56-6.67°C (10-12°F). Temperature differences as high as 11.11°C (20°F) can be done but this is considered very aggressive cooling. The cooling tower was then sized to meet chillers cooling needs. Chiller starting and stopping features were also modeled in the component. The table below summarizes the conditions for one megawatt.

Chiller Parameters	
Design Chiller Load	1 MW (3.413 MBtu/hr)
Minimum Chiller Load	1000 W (3413 Btu/hr)
Maximum Chiller Load	1.1 MW (3.753 MBtu/hr)

Fluid Properties

With the temperatures of the chilled water decided, the properties of the fluid could then be found in the tables located in various tables.

Chilled Water Properties					
	Temperatures				
Properties	1.67 C (35 F)	4.45 C (40 F)	7.22 C (45 F)	10 C (50 F)	12.78 C (55 F)
Density kg/m^3 (lbm/ft^3)	999.93 (62.42)	999.99 (62.43)	999.86 (62.42)	999.7 (62.41)	999.28 (62.39)
Viscosity kg/m-hr (lbm/ft-hr)	6.11 (4.11)	5.57 (3.75)	5.13 (3.45)	4.71 (3.16)	4.09 (2.75)
Specific Heat kJ/kg-K (BTU/lbm-F)	4.21 (1.005)	4.20 (01.003)	4.20 (1.003)	4.19 (1.001)	4.19 (1.001)
Conductivity W/m-K (BTU/hr-ft-F)	0.57 (0.33)	0.58 (0.33)	0.58 (0.34)	0.59 (0.34)	0.59 (0.34)
Thermal Expansion Coefficient 1/K (1/R)	0.00000038 (6.87e-7)	0.0000074 (1.3e-5	0.000049 (8.1e-5)	0.000088 (1.6e-4)	0.000124 (2.2e-4)

Pumps

There are two different types of pumps used for the simulations; constant speed pump and variable speed pump. The flow rate of the pumps first had to be calculated before any of them could be accurately sized. The heat transfer equation below was used to calculate the flow rate.

$$Q = \dot{m} c_p \Delta T$$

Q = 1 MW (3.413 MBtu/hr)

C_p = 4.20 kJ/kg-K (1.0 Btu/lb-F)

The pump speed will mostly be determined by the difference in the supply and return temperatures. The difference in the supply and return temperatures will be between a standard temperature difference such as 5.56 to 6.67°C (10-12°F) and an aggressive temperature difference of 11.11°C (20°F).

Δ T = 5.56 to 11.11°C (10 to 20°F)

So,

Mass flow rate = 42.9 to 21.4 kg/s (94.6 to 47.2 lb/s)

The pumps can now be selected depending on temperature difference. The variable speed pump and constant speed pump are the same size. The pumps were selected using a Bell & Gossett pump selection program.

Pump Conditions	
Variable and Constant Speed Pumps	2.6 to 1.3 m^3/min (680 to 340 GPM)

TES Tank

The TES tank has to be large enough to provide cooling for 30 minutes. This is about the amount of time needed to restart a chiller after a short power failure or shut down the facility. Depending on the application the time needed for cooling may change. Knowing the flow rate required to meet the thermal load, the amount of storage needed can be found. The size of the tank will then depend on the temperature difference in the supply and return temperatures of the chilled water.

Amount required for 30 minutes,

$$(2.6 \text{ to } 1.3\ m^3/min)(30\ min) = 78 \text{ to } 39\ m^3\ (20{,}400 \text{ to } 10{,}200 \text{ gallons})$$

The tank must be able to hold between 78 to 39 m^3 (20,400 to 10,200 gallons) depending on the temperature difference. The TES tank is about 90% stratified so the tank must also be oversized by 10% to accommodate for this. The tank size must then 86.7 to 43.3 m^3 (22,667 to 11,333 gallons). The size must then be rounded up to a common tank size. The table below summarizes all of the conditions. The insulation used around the tank was set to allow a heat gain of two percent of the thermal energy stored in the tank over twenty-four hours. Low temperature chilled water and ice storage tanks were also sized accordingly.

TES Tank Conditions	
Tank Capacity (Chilled Water)	87.1 to 45.5 m^3 (23,000 to 12,000 gallons)
Tank Capacity (Low Temp. Chilled Water)	54 to 37 m^3 (14,300 to 9,800 gallons)
Tank Capacity (Ice Storage)	5500 kg (12,125.4 lbs)

Pipes

There are several different pipe sections in the simulations. All of the pipe sections are similar so the different simulations can be compared. Each pipe segment is 10 meters long. A stainless steel pipe was used with one inch calcium silicate insulation.

Pipe Conditions	
Inside Diameter	0.2027 m (7.981 in)
Outside Diameter	0.2032 m (8 in)
Pipe Length	10 m (32.8 ft)
Pipe Thermal Conductivity	16 W/m-K (8.1 Btu/(hr-ft-F))
Insulation Thickness	0.0254 m (1 in)
Insulation Thermal Conductivity	0.045 W/m-K (0.32 Btu/hr-ft-F)

Cooling Coils

The cooling coils were sized to cool one megawatt. A MultiTherm program was used to choose a coil from the Diversified Heat Transfer Company. The size and setup were taken directly from an actual cooling coil advertised on the Diversified Heat Transfer website.

The tubes are round seamless copper staggered in the direction of the airflow. The tubes have a 5/8" O.D. x 0.20" wall thickness. The tubes come with a 0.008" thick aluminum fins.

Cooling Coil Conditions	
Outside Tube Diameter	0.15875 m (0.625 in)
Inside Tube Diameter	0.1537 m (0.605 in)
Tube Spacing	0.033 m (1.299 in)
Center to Center Distance	0.0381 m (1.5 in)
Number of Rows	2
Tube Thermal Conductivity	400 W/m-K (231.1 Btu/hr-ft)
Fin Thickness	0.000203 m (0.00799 in)
Fin Spacing	0.002337 m (0.092 in)
Fin Thermal Conductivity	250 W/m-K (144.5 Btu/hr-ft)