Energy Cost Reduction for Medical Facilities

by

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Abstract

An intensive energy audit was performed on Russell Medical Center located in Alexander City, AL. It began in 2009 and was completed in 2012. The purpose of this audit is to find effective strategies to reduce energy costs for large medical inpatient facilities and also to research energy auditing techniques. After completion, the audit resulted in an accurate baseline energy model with numerous energy cost savings strategies to implement. Upon implementation of some suggested non capital intensive energy reduction strategies, Russell Medical Center has saved two percent in energy costs. Many other strategies remain on the table to be implemented at RMC. These additional measures could result in an additional twenty five percent reduction in energy costs. This thesis will discuss in detail, the energy audit technique and structure successfully implemented at Russell Medical Center as well as the energy cost reduction measures that were suggested.

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Chapter 1: Introduction

1.1 Background

In the today's society, energy is a primary concern. The production of energy, its use, and the effects are debated heavily. Ever increasing energy costs and public energy use awareness has created interest in reducing energy consumption. This concern has produced new more energy efficient standards implemented by organizations such as the United States Department of Energy (DOE) and the American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE). Consequently, many existing facilities do not meet new energy efficiency requirements. It is in the best interest of existing facility owners to invest and upgrade their systems thus reducing their energy demand. Investing to improve the energy efficiency of buildings provides an immediate and relatively predictable positive cash flow resulting from lower energy bills [1]. Also, owners can better insure the value of other areas of their facility such as indoor air quality, water quality and usage, and overall building safety. A way for owners to invest to meet new specifications and reduce their future energy consumption is by commissioning an energy audit. An energy audit, if conducted thoroughly, brings to light any building energy use deficiencies. The energy audit provides a categorized breakdown of building energy use in order to establish a baseline. From that baseline, improvement goals can be set allowing for energy and cost reduction strategies to be recommended and implemented. Once the energy and cost reduction strategies have been implemented a subsequent, follow-up energy audit can be performed to measure and ensure the success of the strategies that were put into action. This thesis describes energy audits that have been undertaken at Russell Medical Center in Alexander City, Alabama over the last three years. It represents a good example as to how an audit is conducted, the steps required to implement the audit, and the results, both positive and negative, that accrue from an audit.

1.2 Facility Background

Russell Medical Center (RMC) is a multi-building rural non-profit medical facility. As one of the largest energy consumers in the area, RMC acknowledged their own energy inefficiency and decided to commission an energy audit to improve their campus's performance. In 2008, RMC commissioned an energy audit to be performed by Auburn University students under the supervision of Dr. David Dyer. This project continues today and will continue into the future.

1.3 Motivation for Research

The pressing call for America to reduce energy consumption and the need of facilities to reduce their energy cost is the primary motivation for this research and thesis. Russell Medical Center is an excellent platform to research, develop, and implement effective energy audit strategies that can not only reduce their energy consumption and cost but also be applied in the future to other similar facilities. Another motivation is to test current energy audit and building cost reduction strategies to gauge their effectiveness. Finally, RMC is willing to grant full access to a facility and is open to suggestion to try new energy reducing strategies. This attitude is critical to the success of an energy audit and is not generally found.

1.4 Objective of Research Project

The primary goals of this project set by Russell Medical Center and the Auburn Engineering Team is to develop an energy audit plan, perform the audit, analyze the data collected, establish a baseline, make recommendations for improvement, and measure results from the implementation of recommendations. Generally, recommendations requiring any capital cost should have a simple initial payback period of three years. This constraint was set initially but modified over the course of the project. The reason for changing the payback period is the fact that some inefficient equipment had reached its useful life so that a differential payback period was used based on the difference between standard and high efficiency equipment. The scope of this thesis is to document the entire Russell Medical Center energy audit and draw conclusions from the data and analysis in order to establish good energy audit and energy reduction strategies for Russell Medical Center and for future implementation at other similar facilities. This thesis covers the energy audit process from planning to implementation. It gives detailed explanations and analysis of all energy consuming systems audited at Russell Medical Center. It will discuss the data collected from the systems and the analysis completed in order to make recommendations. Finally, this thesis will discuss all conclusions and future recommendations for RMC.

Chapter 2: Literature Review

2.1 Energy Audit Techniques

Rajen Tibrewala [2] generally describes an energy audit as a critical examination of an energy consuming facility. The energy audit is the beginning for an energy conservation program [2]. An energy audit should typically be performed by an individual or team of individuals that has expertise in the types of systems contained in the facility to be audited [2]. In general, there are four types of energy audit [1]. These four types are the walk-through audit, the utility cost analysis audit, the standard energy audit, and the detailed energy audit [1]. Krarti [1] describes the walk through audit as a short on-site visit of the facility to identify simple and inexpensive actions that can be implemented to provide immediate energy usage reduction and cost savings. Examples of typical recommendations of walk-through audits are maintenance practice improvements, replacement of broken equipment, insulation of exposed piping, better monitoring of lighting, etc. The utility cost analysis audit basically requires the auditor to look over utility bills, double check bill calculations for errors, determine dominate charges for utility bills, and establish trends for usage and demand in order to make recommendations to reduce utility power usage [1]. The standard energy audit includes all the tasks mentioned in the previous two audits plus investigating the facility farther. The standard energy audit includes the development of a baseline for energy use, an evaluation of possible energy savings, and an evaluation of the economics associated with the areas of possible improvement [1].

The detailed energy audit includes all items mentioned in the previous three audits but takes them into great detail. When performing a detailed energy audit, a full analysis is done on all energy consuming systems within the facility. The detailed energy audit is the most comprehensive and time consuming energy audit [1]. It includes detailed measurement of building conditions and building model simulations. The detailed energy audit should produce detailed energy reduction recommendations and economic evaluations [1]. Krarti [1] describes a detailed energy audit with four main steps while Tibrewala [2] describes it with nine steps. Krarti has basically combined many of Tibrewala's nine steps and condensed the general procedure into four steps. Krarti's [1] four steps are building and utility data analysis, walk-through survey, establishment of a baseline for building energy use, and evaluation of energy savings measures. He further divides the buildings equipment into two categories. The categories are thermal systems and electric systems. According to Krarti [1], each of the four steps is to be performed for the thermal and electrical systems.

2.2 Detailed Energy Audit Planning and Procedure

Because of its complexity, a detailed energy audit requires significant planning in order to produce satisfactory results. First and foremost, the objectives of the audit must be clearly defined [2]. Planning can then proceed around the overall objectives and a general procedure can be created. Krarti [1] notes that often the energy audit is not a linear process but can be iterative. Different parts of the audit tend to overlap and are repeated. Also, Krarti [1] notes that once deep into the audit, tasks may be reduced in scope or even eliminated based on the findings of other areas. With all of that in mind, the auditor must plan to be flexible and innovative. Once the objectives are defined, Tibrewala [2] emphasizes that the facility must be divided into energy cost centers. An energy cost center is the smallest segment of the facility for which actual energy consumption can be measured and held accountable for its energy use [2]. Tibrewala [2] states that generally a facility may be divided into energy cost centers by department, process, equipment, building, and type of service [2]. One can then proceed with preliminary building and utility data analysis. Its main purpose is to evaluate the characteristics of the energy cost centers and establish patterns of energy use [1]. Past utility and weather data for the region are examples of items that are collected and analyzed in this step. Once preliminary building and utility data analysis is complete, Krarti's [1] procedure moves to the facility survey phase. Examples of task performed in this step are evaluating and collecting data on major energy use equipment and maintenance procedures. This is where the bulk of the building systems data is gathered. All of the facility's energy cost centers are visited and analyzed. Typical energy cost centers are discussed in the next section. After the facility survey is complete, a baseline for building energy use is developed and established to represent the existing energy use and operating conditions for the building [1]. This baseline is calibrated and established from the data collected in the preliminary building and utility data analysis phase and the information collected in the facility survey. Establishing a good baseline for building energy use allows the auditor to continue to the next step which is identification and evaluation of energy savings measures. Cost effective energy conservation measures are determined in this step using energy savings and economic analysis techniques for all of the facility's energy cost centers [1]. Finally these recommendations and conclusions are then summarized in a report given to the facility [2].

2.3 Typical Energy Conservation Areas

When studying methods to perform detailed energy audits, one will find that there are some typical areas in all facilities that have energy reduction possibilities. Typically, in the planning phase of the detailed audit, the facility is divided into these common energy cost centers [2]. Generally, there is overlap in most literature on where these typical energy conservation areas exist. Krarti [1] summarizes that the typical areas for energy conservation measures are the building envelope, electrical systems, HVAC systems, compressed air systems, energy management controls, and water management. The building envelope is basically the structure of the building such as the walls, roofs, floors, windows, and insulation. Typically the building envelope if inadequate, can account for a major portion on energy loss [1]. However, replacement or modification of the building envelope can be expensive. For most large commercial buildings, electrical systems dominate the utility bill [1]. Lighting, office equipment, and electrical motors, all combine to consume large amounts of electricity. Electricity rate schedules, demand charges, ratcheting clauses, and low power factors all combine to form a significant electricity cost. Demand charges alone can account for up to thirty to seventy percent of total electricity charges [3]. One way to lower electric costs is to make improvements to another common energy cost center, the HVAC system. HVAC energy use can account for up to forty percent of the total energy demand of a facility [1]. Often times a large emphasis is put on the HVAC systems because there is often great potential for energy savings. Large heating and cooling systems are expensive to operate and can be very costly if operated inefficiently. Standards such as ASHRAE 90.1 exist for defining energy efficient systems in new facilities, but no standards exist for existing buildings [4]. Many low to no cost improvements can be made in the HVAC energy cost center [4]. Compressed air systems can also account for significant energy costs. Krarti [1] estimates that only twenty to twenty five percent of input electricity is delivered as actual compressed air energy. Building management controls are also a common area where energy can be managed more efficiently. These building management systems (BMS) are the brains behind most of the mechanical energy consuming equipment. If working properly they can create a very energy efficient system. Poorly maintained and tuned systems can increase heating and cooling loads and electrical usage [1]. Krarti [1] also points to water management as a common energy conservation area of interest. Efficient water management can produce up to fifty percent in water cost savings. These typical energy cost centers are found in most large commercial sites and should all be analyzed in a detailed energy audit.

2.4 Techniques to Increase Efficiency

There are several documented techniques to increase the efficiency of the systems that make up the energy cost centers discussed earlier. There is ample opportunity to reduce the energy costs that pertain to the building envelope. However, one drawback to these techniques is that many of them are often very expensive. Krarti [1] emphasizes that the replacement of standard windows with more energy efficient low-emissivity windows can lead to energy savings. When solar radiation strikes an unshaded window, typically eighty four percent of the total solar heat is admitted into the space [5]. This measure will reduce cooling loads in the summer, but conversely will increase heating loads in the winter in climates where there is a significant seasonal temperature difference. He goes further to say that the addition of thermal insulation can be a cost effective energy conservation measure [1]. Thermal insulation can have a great impact on both the cooling and heating loads which in turn lead to reduced electrical and fuel costs. Another area to attack pertaining to the building envelope is infiltration. Infiltration is the leakage of unconditioned air into the building envelope. Infiltration, if unchecked, can account for a large portion of the heating and cooling load [5]. Infiltration can be quantified using the crack method [5]. This method accounts for cracks that could leak air and the building pressure difference that drives the leakage [5].

The electrical system is oftentimes the primary energy cost center in a facility. Many techniques have been developed in order to help reduce costs in this sector. The low hanging fruit in the electrical area is frequently found in the facility's lighting. Approximately twenty percent of all electricity generated in the United States today is used for lighting [4]. Replacing inefficient incandescent lamps with higher efficiency fluorescent or light emitting diode (LED) lamps can reduce power consumption. New LED lamps can be up to eighty percent more efficient than incandescent lamps [6]. Lighting control strategies can also help reduce lighting energy costs. Thumann [4] recommends sensored compensators to turn off lights when they are not needed. He states that advertised paybacks for these systems range from two to four years [4]. The peak electricity demand of a facility is a noteworthy area where energy saving techniques have been documented. Thumann [4] claims that the user will get the most electrical energy per dollar if the load is kept constant, thereby minimizing the demand charge. He states the objective of demand control is to even out the peaks and valleys of consumption by rescheduling the use of energy during peak power demand periods [4]. Dyer and Maples [7] suggest using computer based load shedding programs to shut off loads on a prioritized basis and adding local or global capacitance to adjust power factor to avoid low power factor penalties. Almost all sources recommend direct digital control of demand to monitor and control the facility's usage.

Significant electrical and fuel savings can be enjoyed by implementing HVAC energy efficiency techniques documented in multiple sources. As stated earlier HVAC equipment can account for up to forty percent of the facilities energy consumption [1]. Many low to no cost energy saving measures exist in this area. Krarti [1] suggests HVAC energy savings measures such as temperature reset strategies to reduce cooling and heating loads, retrofitting variable volume air and water distribution systems to reduce fan and pump power, and installing heat recovery systems to recover waste heat energy. Dyer and Maples [7] suggest chilled water plant optimization by raising chilled water temperatures and reducing chiller condenser water temperatures. Dyer and Maples [7] claim that a five degree increase in chill water supply temperature can result in as much as an eight percent savings for centrifugal chillers. These are only a few HVAC energy conservation techniques. Thumann [4] discusses optimizing boiler efficiency through proper combustion tuning and maintenance. Dyer and Maples [7] discuss lowering steam generation costs by suggesting waste heat recovery economizers, boiler blow down optimization, reducing steam line pressures, optimization of boiler loads, preheating combustion air, and properly maintaining steam trap systems. Krarti [1] mentions the importance of building management control systems. These building management control systems manage the HVAC and electrical systems. Krarti [1] suggests that building management control system verification is a very important technique for keeping energy costs low. This includes tasks such as verifying sensors that feed data into the control system and validating sequences of operation. In summary, there are many documented energy conservation techniques available. Many of the techniques listed above are applied and discussed in greater detail later in this thesis.

2.5 Standards

The primary authority in the United States on HVAC standards is ASHRAE. ASHRAE was founded in 1894 and has several publications such as ASHRAE 62.1 and ASHRAE 90.1 that outline engineering standards for ventilation and energy efficient design. As mentioned earlier in this document many existing buildings do not meet new standards for energy efficiency [1]. Many aging buildings do not even meet older standards. When performing an energy audit, it is important that any recommendations for energy conservation meet the specifications and guidelines given by these standards.

ASHRAE 90.1 [8] states that its purpose is to establish the minimum energy requirements of buildings, other than low rise residential buildings. It provides criteria for determining compliance with these energy efficiency standards. These standards can and should be applied to the typical energy cost centers discussed earlier.

First, there are numerous standards pertaining to the building envelope. Earlier, some typical measures to improve the building envelope were discussed (e.g. adding insulation and high efficiency fenestration). ASHRAE 90.1 [8] section 5.1.3 states that “alterations to the building envelope shall comply with the requirements of Section 5 for insulation, air leakage, and fenestration applicable to those specific portions of the building that are being altered”. Some examples of the requirements in ASHRAE 90.1 Section 5 are standard 5.5.3.2, which states that any above grade walls shall comply with the insulation values specified in the table given and standards 5.8.2.5, which maintains that the solar heat gain coefficient for the overall fenestration area shall be determined in accordance with National Fenestration Rating Council (NFRC) 200 [8].

ASHRAE 90.1 has multiple standards for electric equipment. It has an entire section on lighting. Lighting is one of the typical areas of availability for energy conservation. ASHRAE 90.1 [8] standard 9.1.2 asserts that “alteration of lighting systems in any building space or exterior shall comply with the lighting power density requirements of section nine applicable to that space or area and the automatic shut-off requirements of section 9.4.1.1”. Section 10.4.1 pertains to electric motors. It says that “electric motor manufactured alone or as a component of another piece of equipment on or after December 19, 2010 shall comply with the requirements of the Energy Independence and Security Act of 2007” [8].

A large section of ASHRAE 90.1 [8] is dedicated HVAC. ASHRAE 90.1 [8] standard 6.1.1.3 which pertains to alterations to HVAC in existing buildings is pertinent to energy audits. Standard 6.1.1.3 states that new HVAC equipment as a direct replacement of existing HVAC equipment shall comply with the specific minimum efficiency requirements applicable to that equipment [8]. One of the techniques to reduce HVAC energy consumption that was recommended earlier was zone temperature reset and automatic shutdown of equipment when situations allowed. Some examples of HVAC standards that pertain to these specific areas are ASHRAE 90.1 standard 6.4.3.3.1 and standard 6.4.3.3.2. Standard 6.4.3.3.1 asserts that "systems shall be equipped with at least one of the following measures: controls that can start and stop the system under different time schedules, an occupant sensor that can shut the system down when no occupant is sensed, or a manually operated timer" [8]. Standard 6.4.3.3.2 says that heating and cooling systems should have controls with the capability to automatically restart and temporarily operate the system to maintain an adjustable set point [8]. There are standards and load calculation methods located in ASHRAE 90.1 for nearly all types of HVAC mechanical equipment. When considering recommendations for implementation, it is important to keep these standards in mind so that the building is in compliance with code. A few examples are discussed in this section to point out the relevance of these standards. The primary purpose of the energy audit is to reduce energy cost and create an overall improved facility. These standards are in place to instill guidelines to insure that the job is done properly.

2.6 Results and Recommendations of Similar Energy Audits

Energy audits have been carried out on many facilities similar to Russell Medical Center. In this section, the results and recommendations of a two examples are briefly discussed to serve as a comparison to the audit performed at Russell Medical Center. The first such audit was carried out at Hoboken University Medical Center in Hoboken, New Jersey. The audit was performed by Dome Tech Inc. located in Edison New Jersey in 2010. Hoboken University Medical Center is 350,000 square feet facility with a current annual electricity expenditure of just over one million dollars and a natural gas expenditure of around a half million dollars [9]. This facility is close in size and energy expenditure to Russell Medical Center located in Alexander City, Alabama, but electricity rates at Hoboken are higher than at Russell. The HVAC mechanical equipment at Hoboken University Medical Center is similar to the equipment currently at Russell Medical Center. Heating is accomplished with steam generating boilers with a capacity of 1,400 horsepower that produce steam to heat hot water to be distributed for use [9]. Chilled water for cooling is produced with two centrifugal water cooled chillers with a total capacity of one thousand tons [9]. Dome Tech Inc. only listed the total baseline energy costs. They were not broken up into individual energy consumption areas. The baseline was established using Energy Star Portfolio Manager, a computer software program used to analyze building energy consumption. The software was validated using utility bills. From this baseline data, Dome Tech Inc. proposed several energy conservation measures. Nearly all involved some sort of capital expenditure. Some of these expenditures are discussed below.

The first energy conservation measure proposed at Hoboken University Medical Center was a steam trap repair and preventative maintenance [9]. It was suggested by Dome Tech Inc. that many of Hoboken's steam traps were leaking based on its data. Leaking steam traps cost the facility in lost thermal energy and in excess make-up water usage. Dome Tech Inc. maintains that a proper steam trap maintenance program will cost approximately $12,500 and save around $7,000 annually [9]. The second energy conservation measure for Hoboken is a proposed lighting upgrade. Dome Tech Inc. proposed replacing 32 watt T-8 fluorescent lamps with 25 watt T-8 fluorescent lamps and adding occupancy sensors. Dome Tech Inc. asserts that lighting occupancy sensors can reduce lighting costs by thirty percent [9]. Occupancy sensors turn lights off in zones that are unoccupied. This measure is projected to cost $137,600 and save $42,000 annually [9]. A few of the larger more expensive energy conservation measures were to replace the building management control system at a cost of $405,000 and a projected savings of $62,000, and to replace the current aging 800 horsepower steam water tube boiler with a higher efficiency model at a cost of approximately two million dollars with a payback around $118,000 [9]. All energy conservation measures recommended at Hoboken University Medical Center are listed in the Table. Table 2.1 cites an overall capital expenditure of $3,729,460 in order to save 1,007340 kilowatt hours or 14% in electrical power and 196,930 therms or 54% in natural gas [9]. This entire project is projected to have an estimated ten year payback [9].

Table 2.1: Hoboken Audit Results [9]

Energy Conervation Measures at Hoboken University Medical Center					
Energy Conervation Measures	Energy Savings in KWh	Energy Savings in Therms	Capital Cost $	Annual Savings $	Simple Payback- yrs
Steam Trap Maintenace/Repair	0	5,745	12,420	6840	1.8
Lighting Upgrade	312,500	0	128,240	42,470	3.0
Convert 100% Outside Air AHU's	207,900	12,360	134,900	41,740	3.2
Upgrade Building Management System	316,150	17,480	405,000	61,900	6.5
Add Boiler Flue Gas Economizer	0	21,230	175,950	25,260	7.0
Upgrade Boiler Combustion Controls	0	9,770	136,390	15,250	8.9
Upgrade Chilled Water Distribution	51,260	0	79,540	6,660	11.9
Install Small Boiler for Summer Loads	0	22,630	314,280	26,930	11.7
Replace 800Hp Water Tube Boiler	0	99,915	1,646,670	118,900	13.8
Boiler Blowdown Heat Recovery	0	1,180	22,500	1,405	16.0
Free Cooling Waterside Economizer	26,250	0	63,000	3,410	18.5
Water Source Heat Pump Replacement	77,060	0	207,260	10,000	20.7
Upgrade Windows	16,220	6,620	403,310	9,990	40.4
TOTAL:	1007340	196930	3729460	370755	10.0

The recommendations for energy savings are similar to those made for Russell Medical Center. However, all projects had significant capital expenditure costs, which was not the case for RMC.

Hudson County Meadowview Psychiatric Hospital located in Secaucus, New Jersey is another facility that was audited in 2010 by the Concord Engineering Group located in Voorhees, New Jersey. After conducting a detailed energy audit, The Concord Engineering Group states that Meadowview Hospital is a 63,000 square foot facility that has an annual electrical cost estimated at $690,991 and an annual natural gas cost estimated at $675,114 [10]. As with the previous audit discussed, this report did not contain an itemized baseline energy breakup. Steam is generated to provide heating and chilled water is generated to provide cooling. Five energy conservation measures were recommended by The Concord Engineering Group.

The first energy conservation measure recommended by The Concord Engineering Group was to upgrade the existing lighting [10]. Many existing lighting fixtures at Meadowview Hospital are T-12 fluorescent lamps. The Concord Engineering Group recommends replacing these fixtures with more efficient T-8 fluorescent lamps and replacing fluorescent lamp exit signs with LED lamp exit signs [10]. This project is estimated to cost $11,946 and save approximately $4,382 annually with a simple payback of 2.7 years [10]. The next energy conservation measure was to install lighting controls to better manage lighting throughout the facility [10]. Lighting controls will utilize occupancy sensors to turn off lights in unoccupied spaces [10]. This project is expected to cost $2,940 and will save $230 per year [10]. The third energy conservation measure was to retrofit the air handlers with new variable frequency drive fans [10]. Variable frequency drive fans have the ability to throttle back and use less fan horsepower during periods of reduced demand. It is estimated that this measure will cost $58,625 to install and save an annual $25,295 [10]. The fourth energy conservation measure was to upgrade the kitchen exhaust hood controls [10]. This measure entailed adding a new variable frequency drive fan to the exhaust hood to allow it to ramp up and down with demand and a new controller to manage fan run time [10]. This project was estimated to cost $46,044 to install and will save $4,855 dollars [10]. The fifth energy conservation measure was to install variable frequency drive pumps to the chilled water loop system [10]. This will allow for the pumps to throttle back during periods of low demand similar to the variable speed drive fans. It will also entail replacing the current three-way valves with new two way valves. This energy conservation measure is estimated to cost $51,850 and will save $4,172 per year [10]. All energy conservation measures are summarized in the Table 2.2. The total capital investment for all energy conservation measures is $171,405 and it is projected to save $38,934 with an overall simple payback of 4.39 years [10]. If all projects are implemented then it could theoretically reduce electricity costs by five percent.

Table 2.2: Meadowview Audit Results [10]

Energy Conervation Measures at Meadowview Psychiatric Hospital			
Energy Conservation Measures	**Capital Cost -$**	**Annual Savings -$**	**Simple Payback**
ECM #1 Lighting Equipment Upgrade	$11,946	$4,382	2.7
ECM #2 Lighting Controls Upgrade	$2,940	$230	12.8
ECM #3 AHU Supply & Return VFD Retrofit	$58,625	$25,295	2.3
ECM #4 Kitchen Exhaust Hood	$46,044	$4,855	9.5
ECM #5 Variable Primary Chilled Water Flow	$51,850	$4,172	12.4
TOTAL:	**$171,405**	**$38,934**	**4.39**

This audit did not provide an in depth analysis on the mechanical systems to find low to no cost items. If a more thorough audit were performed, more energy savings opportunities could possibly be found.

Chapter 3: The Russell Medical Center Energy Audit

3.1 Russell Medical Center Energy Audit Type

A detailed energy audit was carried out at Russell Medical Center. This audit covered the entire Alexander City Campus which consists of a main hospital, three professional office buildings, and a cancer center occupying a total of 289,041 square feet. As previously discussed, the detailed energy audit is an in depth analysis of all energy consuming systems at a facility. The RMC audit covered the building envelope, electrical system, HVAC system, and even electricity billing and rates. The facility was divided up into energy cost centers and data was collected and analyzed at each energy cost center. The detained energy audit at RMC required substantial planning and consists of multiple phases. The audit process was completed and repeated multiple times over the course of three years.

3.2 RMC Energy Audit Planning and Structure

As discussed previously in the literature review, energy audit planning is a crucial phase. Proper planning must be completed in order to have an orderly and productive audit. RMC energy audit planning resulted in a nine step general procedure for executing the audit. Those nine steps were as follows:

1. Define Audit Objectives
2. Divide the Facility into Energy Cost Centers
3. Preliminary Data Collection and Organization
4. Execute a Building Survey
5. Organize and Analyze Data from Building Survey
6. Establish a Baseline Energy Profile
7. Formulate Recommendations
8. Evaluate Economics and Feasibility of Recommendations
9. Draw Final Conclusions and Document Entire Audit in Report

The first task in the planning phase for the RMC energy audit was to clearly define the project objectives and guidelines. Clearly the general goal of the project was to find energy saving measures complete with a clear path to achieve them, but there was also many sub goals on the path to achieving the main goal. The first of which dealt with the execution of the audit. A standard was set that the audit should be executed in a way that did not interfere with the day to day operation of the facility. RMC is a fully functional medical facility with several critical care departments. It was important that the audit be carried out in a fashion that did not put any patients at risk. Also it was noted that a good professional relationship should be established with RMC personnel. All RMC rules and regulations would be followed. Secondly, a high standard was set for data collection. It was mandated that accurate useable data must be collected in order to achieve success. No data collection short cuts would be permitted. A guideline was set that any new recommendations would comply with the engineering standards discussed previously. Goals for task completion were set, so that the audit progressed in a timely fashion. RMC administrators initially declared that any capital investment should be reclaimed within three years. With that in mind, a goal was set to find low to no cost recommendations for implementation to help offset any capital investment recommendations.

With audit objectives clearly defined, the facility was then divided up into energy cost centers. The campus at RMC consists of a main hospital for primary patient care, three professional office buildings (POB) for private practices and administration, and a cancer treatment center. The primary energy cost centers were set as the three facilities mentioned above. Each primary energy cost center was then divided into the various energy cost centers that existed at each facility. For example, the main hospital was divided into its sub energy cost centers such as building envelope, electrical systems, HVAC, building management system, and water systems. Those sub energy cost centers were further divided into smaller categories. Energy cost centers interact with other energy cost centers throughout the facility. Making changes to one energy cost center usually affects other energy cost centers. Figure 3.1 illustrates the division and interaction of RMC’s energy cost centers.

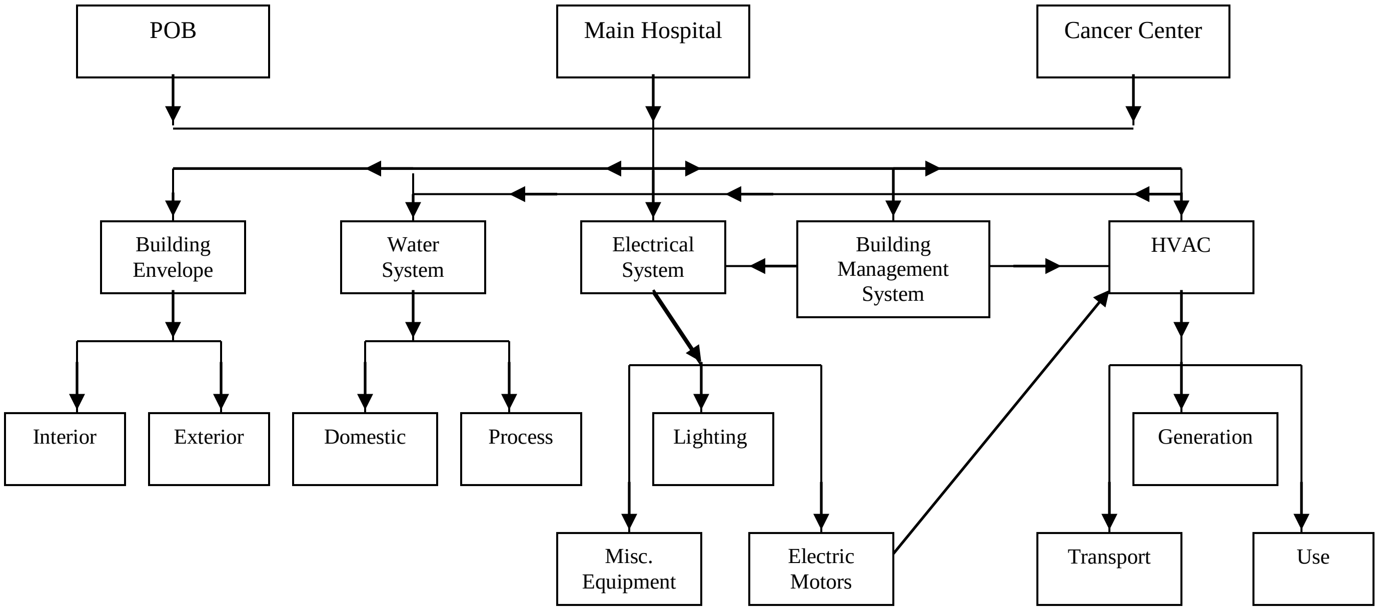


Figure 3.1: Energy Cost Center Interaction

With RMC's energy cost center's properly divided, preliminary data collection began. This step primary consisted of gathering utility bills, building plans, and equipment schedules. Utility bills were gathered from the previous two years and organized into an understandable form. Building plans and equipment schedules were reviewed in order to become familiar with RMC's building layout and equipment types. RMC's climate zone was established and weather data was collected. A general engineering assessment was formed from the preliminary data in order to prepare for the building survey phase.

The building survey phase was a crucial segment of the RMC energy audit. The building survey consisted of multiple onsite visits and was by far the lengthiest process. All energy cost centers were visited in order to gather data. Great emphasis was placed on data collection during the building survey. Many energy cost centers were visited multiple times to ensure accurate useable data was collected. The majority of time was spent on the HVAC system. Data collection for this energy cost center can be troublesome due to the difficulties with measuring air flow. Care must be taken to get an accurate result. Data collection will be discussed in greater detail later in this thesis.

After the building survey was completed, large amounts of data had to be organized into a useable form. Raw data was categorized and placed in Excel spreadsheets and other computer programs to make it easier to spot trends. The data's accuracy was evaluated and faulty data was discarded. Decisions were made to determine if any data needed to be retaken. Once all information was structured, analysis began and systems within energy cost centers were characterized. Multiple analysis methods were utilized for each energy cost center to develop building energy models. Engineering principles and computer software was utilized when applicable in order to work towards establishing a baseline energy profile.

A baseline energy profile was established from all preliminary and site survey data gathered from all energy cost centers. Compiled and analyzed site survey data was used to characterize the mechanical systems. Energy usage from energy cost centers such as electrical systems and HVAC systems was formulated from the site survey data and compared to utility bill data to verify accuracy. Once acceptable accuracy was verified, a baseline energy profile was established. This profile consisted of an itemized energy cost center breakdown at multiple levels. Creating an accurate baseline energy profile was a crucial step for the success of the audit. Recommendations cannot be accurately formulated without it.

Utilizing the building survey data analysis and baseline energy profile, energy reduction engineering principles were applied in order to formulate recommendations to improve RMC's energy efficiency. Energy reducing recommendations were formulated in most all energy cost centers with the majority of them belonging to the HVAC energy cost center. Each recommendation was thoroughly analyzed for how it might affect other recommendations and the building as a whole. These Recommendations will be discussed in detail in later chapters. After possible recommendations were identified, they were evaluated for economic feasibility. Each possible energy saving recommendation was brought back to the dollar because it was understood that cost would ultimately be the deciding factor. Some recommendations were thrown out at this stage because they did not make economic sense. Many low to no cost and capital intensive recommendations were found as a result of the audit. Once these recommendations were evaluated for building and economic feasibility, the final step of drawing final conclusions and documenting the process was carried out. Final recommendations were submitted to RMC management for a decision about implementation.

Chapter 4: Russell Medical Center Energy Cost Center Overview

4.1 Introduction

As shown in the previous chapter, RMC was broken down into three primary energy cost centers and those primary energy cost centers were broken down into five sub energy cost centers. This chapter consists of an overview of the existing systems and equipment that were visited during the RMC audit. The existing electrical system, HVAC system, building envelope, water distribution system, and building management system is discussed. The building envelope is not discussed in great detail primarily because the current building envelope is sufficient in most places, and also due to the fact that modifications to the building envelope are expensive and mostly outside the range of the expected investment return for this audit. The building envelope overview will be limited to the roof section of the main hospital because this was the only part of the building envelope that was analyzed in detail. The majority of the focus will be on the HVAC system due its size and complexity. All three primary facilities at RMC had significant energy costs stemming from HVAC systems. In order to better organize these large systems, the HVAC energy cost center was further divided into three categories. All HVAC equipment fell into one of these categories. The three categories were generation, transport, and end use. This chapter will describe the HVAC systems and equipment in detail at RMC. It is important to understand that many of these systems work together and overlap. For example, the water distribution system and HVAC system are linked.

Process and domestic hot water is heated by the HVAC system. Also, many components of the HVAC systems are driven by electrical motors from the electrical system. As a result, some electrical systems such as pump and fan motors may be discussed as part of the HVAC system.

4.2 RMC Campus Layout

Russell Medical Center is a multi-building campus consisting of a main hospital, three professional office buildings and a cancer treatment center. These facilities contained within the RMC campus were deemed primary energy cost centers. Figure 4.1 shows an aerial view and map of the entire campus.

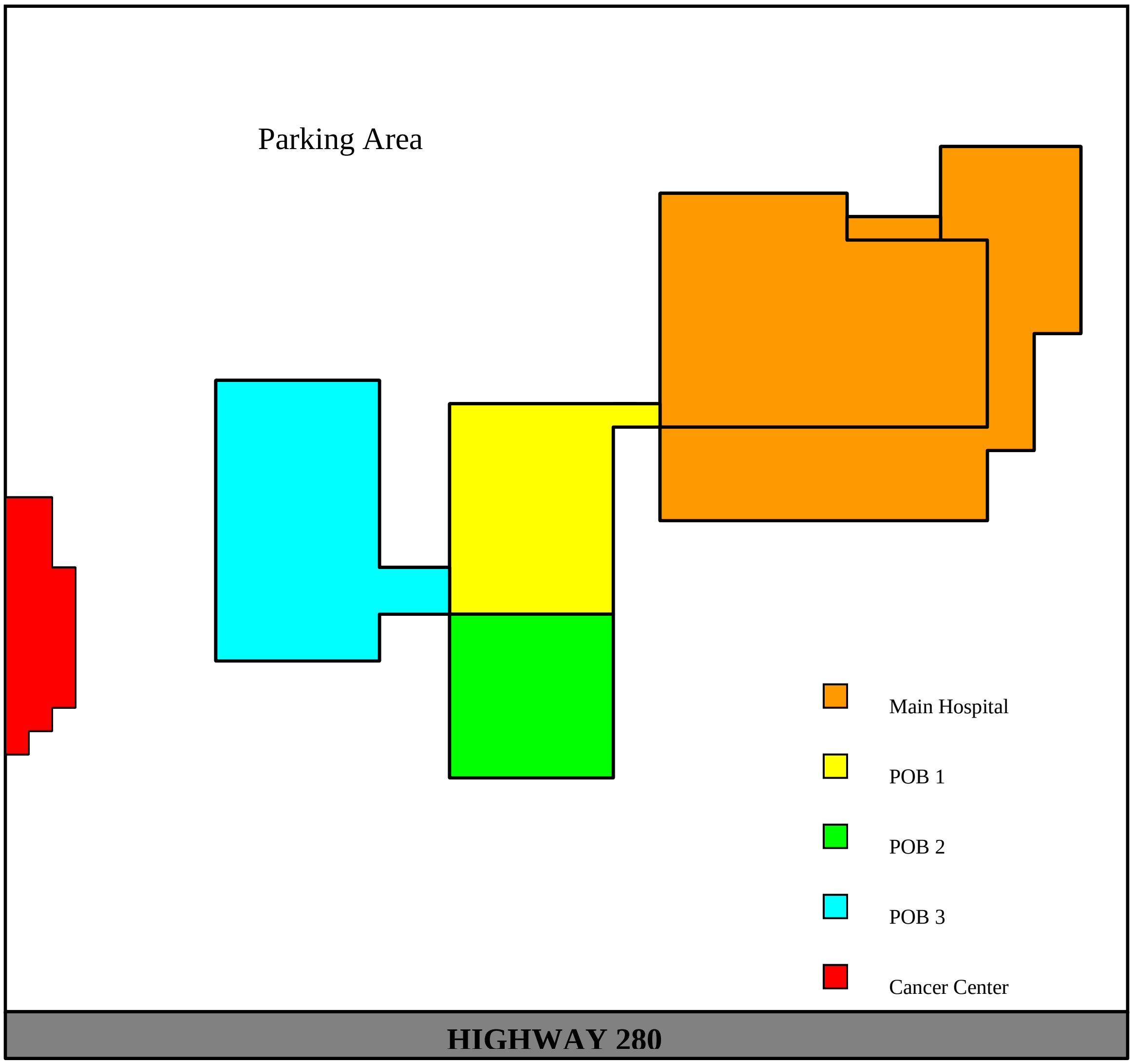


Figure 4.1: RMC Campus Layout

4.3 Building Envelope Overview

As previously stated in the introduction to this chapter, the building envelope overview is limited to the roof of the main hospital because that was the only area that was analyzed over the course of the audit. Other areas such as walls and fenestration were considered adequate. The roof of the main hospital covers an area of 46,900 square feet. It consists mostly of a built up roof system. Built up roofing systems consists of multiple layers of insulation, ply sheets, and asphalt. The top layer is generally a reflective coating or membrane that sometimes is covered with gravel or some other type of ballast. The main hospital roof is partially covered with gravel ballast. Large areas of the main hospital roof do not have gravel ballast. Those areas have a black top membrane.

4.4 HVAC Overview

The purpose of an HVAC system is to provide the required environmental conditions in each zone within the facility. In a hospital like RMC, it is necessary to control many aspects of the environment such as temperature, relative humidity, and ventilation. The HVAC system at RMC was designed to control the aforementioned parameters all during the worst case scenario. The system consists of a complex network of equipment working together to achieve the desired environmental conditions. In order to more accurately describe the RMC HVAC system, it was divided into three sub categories. Those categories are generation, transport, and end use. Generation refers to the systems that generate heating and cooling. The three primary products generated by RMC's HVAC system are steam, hot water, and chilled water. Transport refers to the systems that transport and distribute the generated product for use. Some systems can have characteristics of both generation and transport systems. For this thesis, these multipurpose systems will be classified by their primary function, but it all functions are noted. End use refers to how and where the product is used. It refers to systems that use a generated product and also it refers to energy recovery systems that extract remaining available energy in the product to increase efficiency.

4.4.1 Main Hospital Steam Generation

Building heat for RMC is currently provided by a steam system. In steam systems, water is alternately boiled and then condensed as it progresses around the heating loop, thus the latent heat of the phase change is the primary energy carrier [7]. The boiler system adds heat which is given up through heat exchangers throughout the facility. The condensate is then recycled. Steam for use in the main hospital is generated by two 150 horsepower three pass Kewanee wetback fire tube boilers. The specifications of each boiler are listed in Table 4.1.

Table 4.1: Boiler Specifications

Boiler Specifications for Boiler Numbers 1 & 2	
Manufacturer	American Standard
Type	Kewanee Scotch Firetube Generator
Model Number	M5653
Boiler Horsepower	150
Max Working Pressure psig	150
Heating Surface Area (ft^2)	750
Capacity (lbs steam/hour)	6000
Max Firing Rate (Mbh)	6278

These boilers are located on the ground floor inside the boiler room which contains both boilers and their support equipment. The two fire tube boilers are cylindrical in shape and are sixty five inches in diameter and one hundred sixty four inches long.

Figure 4.2: RMC Firetube Boiler

In fire tube boilers, heat is generated by combustion of air and fuel mixed at the proper ratio. Air is pushed into the combustion chamber by a forced draft fan where it mixes with a fuel brought in through a pressurized fuel line. RMC's boilers are scotch marine meaning they consist of a large horizontal tube that houses the combustion flame. The combustion gases then pass through numerous small tubes above the main tube and out through the exhaust stack. The tubes are submerged in water within the boiler shell. RMC's boilers are three pass meaning the combustion gases make three passes through the water in order to maximize heat transfer. Heat generated by combustion is transferred to the water from the tubes containing the hot combustion products via thermal conduction and radiation. The heat transferred to the water causes it to boil into steam which accumulates in the shell and builds pressure. The steam is then distributed to systems throughout the facility for use. Each boiler is independently controlled by its own burner management system. This system initiates and controls the startup procedure and then controls the firing rate by modulating the air and fuel intake. The primary boiler fuel at RMC is natural gas. Both boiler one and two can be fired on natural gas. Boiler one is dual fuel capable. It can be fired on both natural gas number two fuel oil. This ensures fuel redundancy. RMC's boilers are tuned for two firing rates, high fire and low fire. The low fire position is for lower demand situations while the high fire position is the boilers maximum capacity. The air and fuel ratio is controlled by a jack shaft that links the fuel flow control valve with the air intake damper. Several safety devices are incorporated into the system in order to prevent catastrophic failure. These safeties include devices like steam pressure relief valves, low water level cutoffs, and automatic fuel shutoff valves.

Fire tube boiler systems also have support equipment in order to sustain the process. At RMC, condensate is captured at steam traps throughout the facility and is pumped back to a condensate tank located in the boiler room. The primary purpose of a steam trap is to prevent the loss of live steam and discharge the condensate back to the condensate tank. There are several different steam trap designs. RMC utilizes mechanical steam traps. Two examples are float type and inverted bucket designs, which are both used by RMC. Fundamentally, these designs work by a mechanical float that rises and falls with condensate level to open or close a valve to release condensate or trap steam. At RMC, condensate from the traps is accumulated in a condensate tank which is held at atmospheric pressure. Make up water and water treatment chemicals are added at the condensate tank. Make up water is generally too chemically basic or "hard" to be placed directly into a boiler system. Hard water refers to water that contains a high mineral content. Hard water is undesirable in boiler systems because the high mineral content creates scale buildup on surfaces which reduces heat transfer. The water at RMC is softened by a zeolite water softening system. The zeolites act as an ion exchange bed softening the make-up water. The softened water and other chemicals are fed into condensate tank as needed. At RMC sulfites are injected into the condensate tank to reduce dissolved oxygen levels in order to prevent pitting damage. The treated condensate is then pumped from the tank back into the boiler by feed water pumps into order to be reheated to make more steam. Figure 4.3 on the next page contains a basic line diagram outlining RMC's steam system.

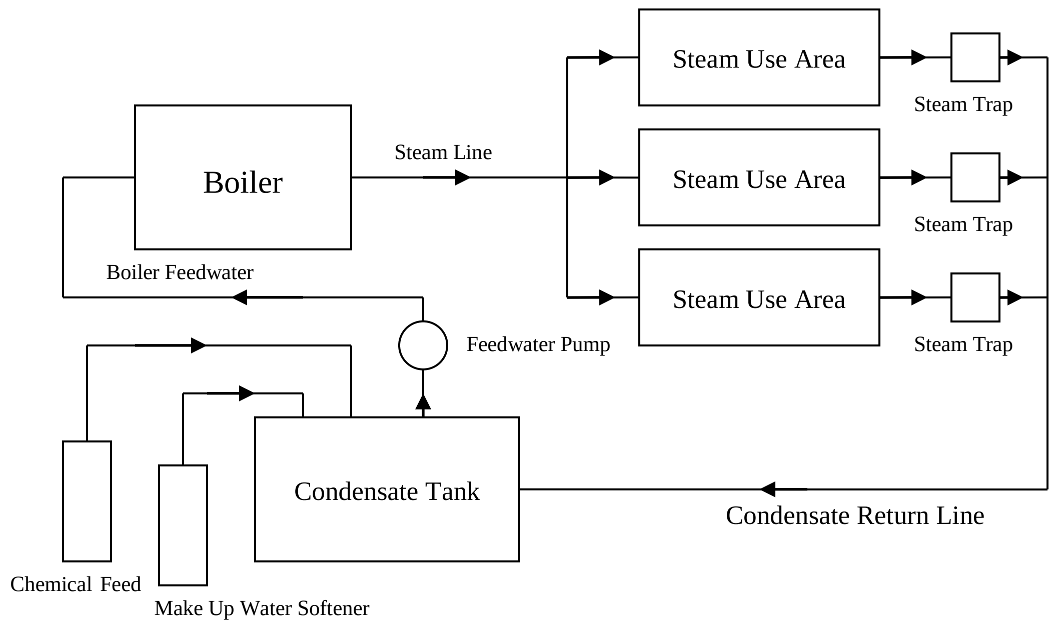


Figure 4.3 RMC Steam System Diagram

4.4.2 Main Hospital Process Hot Water Generation

The main hospital at RMC uses process hot water to provide building heating. Most of this process hot water is made by shell and tube heat exchangers which will be discussed in detail later in the end use section of this chapter. Process hot water is also generated by two Teledyne LAARS Mighty Therm hot water generators located outside the building. The specifications of these two boilers are listed in Table 4.2.

Table 4.2: Main Hospital Secondary Boiler Specs.

Model No.	HH2200EN18JCACXX
Serial No.	C99J05132
National Board No.	92491
Altitude	0-2000 ft
Fuel Type	Natural Gas
Minimum Input	665,100 BTU/hr
Maximum Input	2,200,000 BTU/hr
Output	1,782,000 BTU/hr
Gas Manifold Pressure	4 in. Water Column
Minimum Gas Supply Pressure	7 in. Water Column
Maximum Gas Supply Pressure	9 in. Water Column
Maximum Gas Orifice	21 DMS
Minimum Relief Valve Capacity	2200 lb/hr
Maximum Working Pressure	160 psi
Maximum Water Temperature	240 °F

These boilers are fired on natural gas and are connected into the process hot water piping network. They are used primarily in an auxiliary support role. They provide extra capacity to support the primary system during cold temperatures when heating demand is high. These hot water generators provide hot water at a temperature of one hundred eighty degrees Fahrenheit to be used to heat the building. This system works using combustion of air and fuel to produce heat to warm the process water. These boilers are natural draft meaning no forced draft fan is used to provide combustion air. Water is pumped through a finned heat exchanger where heat is transferred from the combustion gases to the water via conduction, convection, and radiation. The water is sent out to the hospital for use and is returned back to the hot water generator. It is a closed loop system and works in conjunction with the primary system to meet demand.

4.4.3 Main Hospital Chilled Water Generation

Chilled water is the primary source for cooling at RMC. Chilled water systems remove heat from the building by blowing air across chilled water coils so that the heat is transferred from the air to the chilled water. The heated water is circulated to the chiller where the water is cooled, thus the chilled water serves as the medium for moving heat from various parts of the facility to the chiller plant where the chiller and associated equipment remove the heat from the circulated water and transfer it to the atmosphere [7]. At RMC's main hospital, chilled water is generated by two Trane Centravac water cooled centrifugal chillers with a combined capacity of 750 tons located on the ground floor in the chiller plant mechanical room. The specifications for each chiller are listed in Table 4.3.

Table 4.3: Liquid Cooled Chiller Specs.

Chiller Specifications		
	Chiller 1	Chiller 2
Model	CVHE450	CVHE500
Capacity (tons of cooling)	350	400
Compressor Horsepower	20	25
Working Fluid	R-123	R-123
Voltage	480	480
RLA (amps)	263	306
Design Evaporator Pressure Drop (psig)	10	10
Design Entering Evaporator Water Temp (degrees F)	50	55
Design Leaving Evaporator Water Temp (degrees F)	40	45
Desgin Evaporator Water Temp Differential (degrees F)	10	10
Design Evaporator Flowrate (gpm)	630	864
Design Condenser Pressure Drop (psig)	10	10
Design Entering Condenser Water Temp (degrees F)	85	95
Design Leaving Condenser Water Temp (degrees F)	85	95
Desgin Condenser Water Temp Differential (degrees F)	10	10
Design Condenser Flowrate (gpm)	900	1080

RMC's two centrifugal, water cooled chillers utilize a vapor compression cycle with R-123 as the refrigerant working fluid. An ideal vapor compression cycle is a four step process. The process begins with a refrigerant working fluid in a saturated vapor state. Work is done by a compressor to isentropically compress the working fluid to state two, a superheated vapor. The high pressure, high temperature, superheated vapor then flows into the condenser where heat is rejected, causing the working fluid to condense into a saturated liquid at state three. The saturated liquid at high pressure is then throttled through an expansion orifice reducing its temperature and pressure. Some of the liquid flashes to vapor producing a saturated mix at state four. The cool saturated mix then flows through the evaporator where heat is absorbed causing the working fluid to evaporate back into a saturated vapor. The ideal vapor compression cycle is illustrated in Figure 4.4.

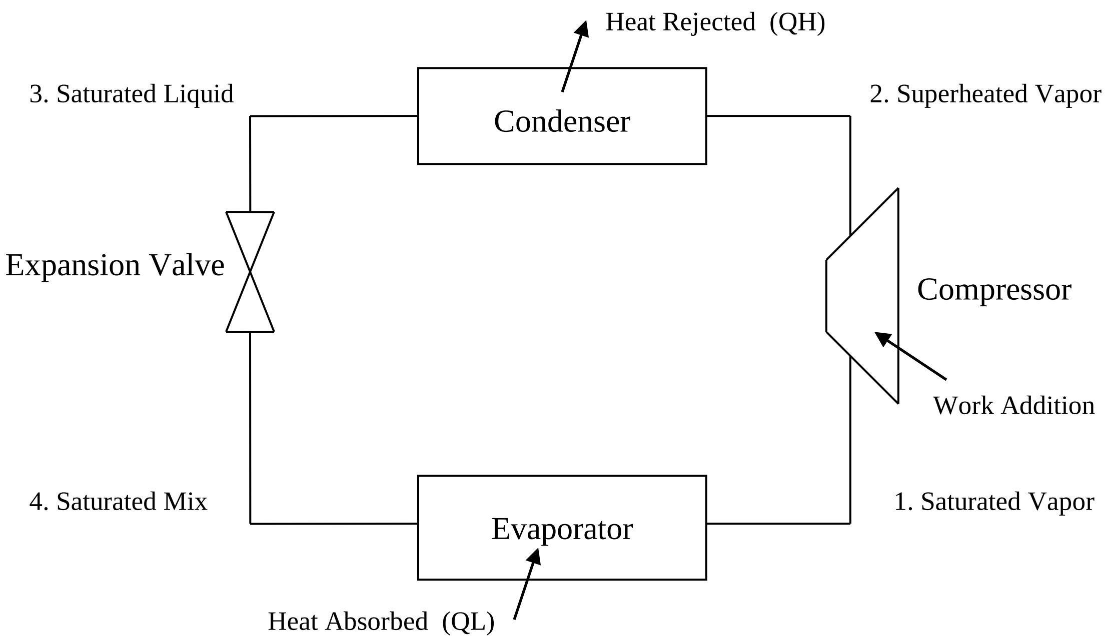


Figure 4.4: Ideal Vapor Compression Cycle

RMC's chiller design expands this basic vapor compression cycle process. The Trane Centravac chillers at RMC use a three stage cascade vapor compression design. A multistage compressor, two expansion valves and a two stage flash chamber is added to the basic system. This configuration increases efficiency by using multistage compression and by throttling the refrigerant in stages to an intermediate pressure. This action preflashes some of the refrigerant into a colder vapor to be separated by the flash chamber. The flash chamber separates the preflashed vapor from the liquid refrigerant and mixes it with warmer vaporized refrigerant entering the compressor to lower the entering refrigerant temperature. Lowering the temperature of the refrigerant in between compression stages reduces the specific volume of the refrigerant thereby reducing the total amount of work needed to create the desired temperature lift. Figure 4.5 on the next page illustrates the RMC Trane Centravac® chiller design with a schematic of the components.

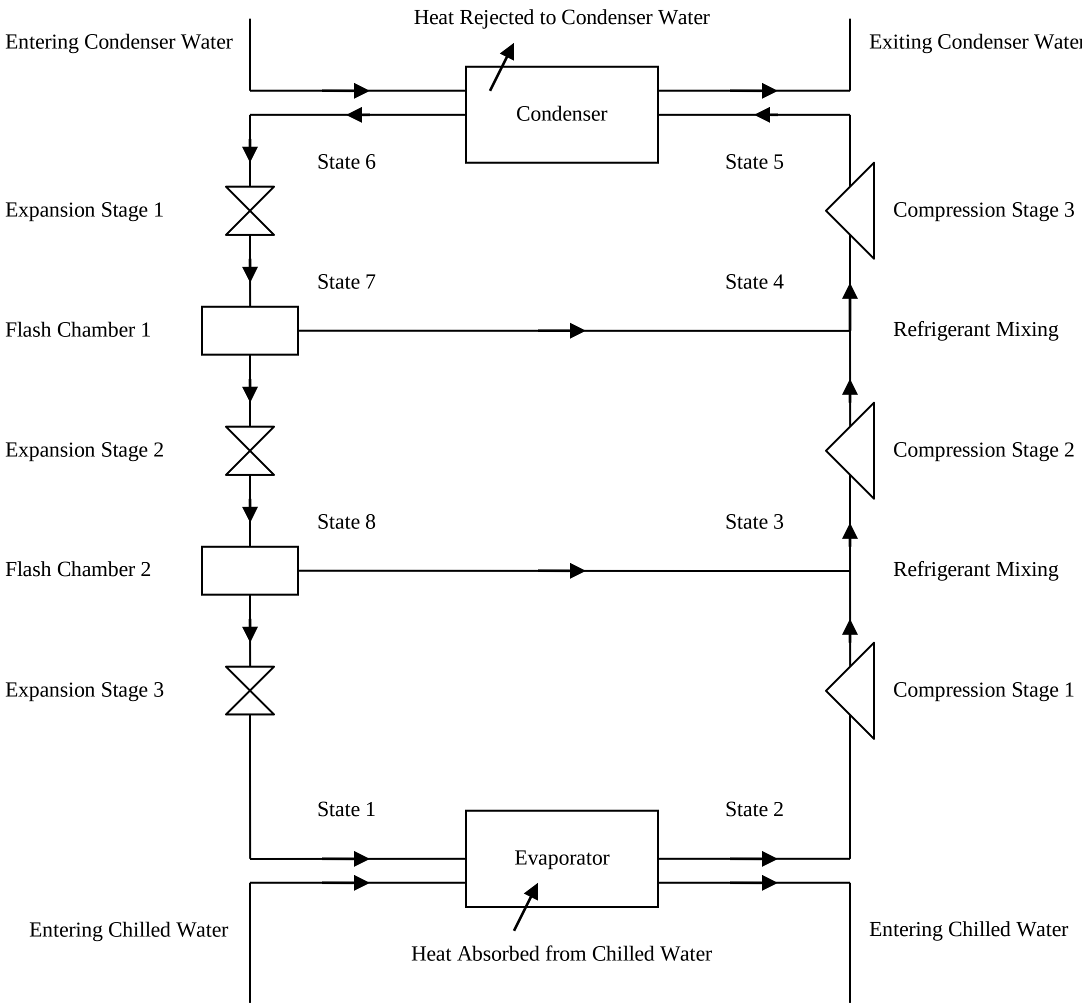


Figure 4.5 RMC Chiller Schematic

The heat gained from compressing the refrigerant and from the building must be given up in order for the refrigerant to condense. RMC’s chillers are water cooled. In the condenser heat exchanger, heat is rejected from the warm refrigerant to the cooler condenser water being pumped in from the cooling tower. The heated condenser water is then pumped outdoors to a cooling tower to be cooled. The cooling tower will be discussed in the next section. The chiller produces chilled water by pumping the warmer water returning from the hospital piping network through the evaporator heat exchanger. Heat is transferred by convection from the chilled water into the cold refrigerant. Regenerated chilled water is then pumped throughout the facility for use.

Figure 4.6 RMC Chiller Photo

4.4.4 Main Hospital Cooling Tower

The primary purpose of the cooling tower is to reject heat gained from the building and from compression of the refrigerant. This heat must be rejected to a place outside the building. Water cooled chiller condensers absorb heat from the refrigerant and transfer it to condenser water being pumped through the heat exchanger. This heat absorbed by the cool water is then transferred outside to the cooling tower where it is rejected to the atmosphere. RMC uses an Evapco two zone, mechanical draft, counter flow cooling tower to reject unwanted heat. The specifications of this cooling tower are listed in Table 4.4.

Table 4.4 RMC Chiller Specs

Cooling Tower Specifications	
Manufacturer	Evapco
Type	Mechanical Draft, Counterflow, Two Zone
Design Entering Water Temp (degrees F)	95
Design Exiting Water Temp (degrees F)	85
Design Flow for each Zone (gpm)	990
Fan Horsepower	20
Design Ambient Wetbulb Temp (degrees F)	78

Mechanical draft, counter flow cooling towers like the tower at RMC work primarily by the principle of evaporative cooling. When water evaporates, its latent heat or the amount of heat needed to evaporate water is absorbed from the air, thus producing a cooling effect. In a mechanical draft cooling tower, warm water is sprayed down through the fill. The fill is a series of grated partitions that slows the water down to increase heat transfer. At the same time, air is pulled up through the cooling tower by induced draft fans. This places the upward moving air in counter flow with the falling water. During this process water is partly cooled by the transfer of sensible heat raising the dry bulb temperature of the moving air but mostly by the exchange of latent heat resulting from the evaporation of a small portion of the water [7]. The cooled water falls to the cooling tower basin where it is collected and recirculated back to the chiller condenser heat exchanger. RMC’s cooling tower has two zones each with its own fan. Each zone serves a chiller. These fans are controlled by the building management system and have two operational speeds. The lower speed is for times of reduced demand, while the higher speed is used in times of maximum demand. The cooling tower at RMC also serves another purpose. When wet bulb temperatures permit, the cooling tower provides chilled water for RMC. At a theoretical one hundred percent efficiency, a cooling tower could cool water to the wet bulb temperature. In actuality, a correctly designed and functioning cooling tower can cool the water to within a few degrees of the wet bulb temperature. Besides the energy needed to run the fans and pumps, it can essentially provide free water cooling. This allows for the cooling tower to work as a water side economizer. During certain times of year, wet bulb temperatures are often well below the needed chilled water supply temperature at RMC. When this occurs, RMC’s cooling tower works in tandem with a plate and frame heat exchanger to produce chilled water for the hospital. The plate and frame heat exchanger will be discussed in greater detail later in this thesis.

4.4.5 POB Hot Water Generation Systems

Hot water provides building heat for the POB. Unlike the hospital, no steam is generated to make hot water. Hot water is produced by two natural gas fired hot water generators with a total capacity of just over two million BTU's per hour. The specifications for these two hot water generators are listed in Table 4.5.

Table 4.5: POB Hot Water Generator Specs

POB Hot Water Generator Specifications		
	Boiler 1	Boiler 2
Type	Natural Draft / Copper Tube	Fan Assisted / Copper tube
Output (MBH- thousand BTU/hr)	890	1285
Design Pressure Drop (FT of head)	6	10
Design Flowrate (gpm)	60	120
Design Exiting Water Temp (degrees F)	180	180

The POB's hot water generators work similar to the ones previously described in the hospital hot water generation section. Heat is generated by combustion of natural gas and air. The heat from the flame and combustion gases is then transferred to water being pumped through a heat exchanger. The generators produce hot water at 180 degrees Fahrenheit to be distributed throughout the building. These hot water generators are connected in a closed hot water loop similar to a chilled water loop. Hot water leaves the generator and is circulated to heating coils throughout the facility where heat is removed by air being blown across the coil. The water then recirculates back to the boiler for regeneration.

4.4.6 POB Chilled Water Generation Systems

Chilled water is also the primary source of cooling for the POB at Russell Medical Center. The POB has a chilled water loop similar to the one found in the main hospital with the exception being the chiller design type. The POB utilizes four chillers with a combined capacity of 346 tons to supply its cooling demand. The specifications for these four chillers are listed in Table 4.6.

Table 4.6 POB Air-Cooled Chiller Specs

POB Chiller Specifications				
	Chiller 1	Chiller 2	Chiller 3	Chiller 4
Make	Trane	Trane	Carrier	Carrier
Type	Screw Air Cooled	Screw Air Cooled	Reciprocating Air Cooled	Reciprocating Air Cooled
Capacity (tons of cooling)	120	120	53	53
Refrigerant	R-22	R-22	R-22	R-22
Voltage	460/3	460/3	208/3	208/3
Evaporator Pressure Drop (ft of head)	15	15	20	20
Exiting Water Temp (degrees F)	44	44	45	45
Design Ambient Temp (degrees F)	95	95	95	95
Design Evaporator Flowrate (gpm)	294	294	126	126

The POB chillers also utilize a vapor compression cycle to cool the refrigerant; however their design is more similar to the basic vapor compression cycle. These chillers do not utilize multistage compression or expansion. The primary chiller design differences are in the compressor type and condenser cooling medium. All four chillers are air cooled. Instead of rejecting condenser heat to flowing water, these chillers reject condenser heat to the atmosphere. They do not require a cooling tower. Fans circulate air over the condenser in order to cool the condenser. Chillers one and two have helical rotary screw type compressors driven by electric motors. Screw type compressors are positive displacement and have a nearly constant flow performance characteristic [7]. They basically consist of two helically grooved rotors mated in stationary housing. As the rotors rotate, the refrigerant gas is compressed by direct volume reduction between the rotors [7]. Chillers three and four have reciprocating compressors. The reciprocating compressor is also a positive displacement machine that maintains a mostly constant volumetric flow over a wide range of pressure ratios [7]. It utilizes a piston inside a cylinder to compress the refrigerant vapor. Other than these differences, the process is the same. Warm water from the hospital is pumped through the evaporator where its heat is rejected to the refrigerant. The regenerated chilled water is then recirculated back to the hospital for use. Chillers three and four are used mainly as auxiliary chillers when demand is high. The bulk of the cooling load is carried by chillers one and two.

4.4.7 Cancer Center Hot Water Generation

Hot water is the primary source of heat for the cancer center. Hot water is generated by a natural gas fired hot water generator very similar in design to the ones found in the POB. The hot water is distributed through the building for use in heating coils found in air handlers and air terminal boxes. The hot water is then recirculated back to the boiler where it is regenerated. Hot water for the cancer center is produced at 200 degrees Fahrenheit. The specifications for this hot water generator are listed in Table 4.7.

Table 4.7: Cancer Center Hot Water Generator Specs

Cancer Center Hot Water Generator Specifications	
	Boiler 1
Type	Natural Draft / Copper Tube
Output (MBH- thousand BTU/hr)	800
Design Pressure Drop (FT of head)	6
Design Flowrate (gpm)	120
Design Exiting Water Temp (degrees F)	200

4.4.8 Cancer Center Chilled Water Generation

Chilled water for the cancer center is produced by two screw type, air cooled chillers with a combined capacity of 120 tons of cooling. The chilled water produced is distributed by a chilled water piping loop for use by chilled water coils in the cancer center. The chiller design is similar to the screw type chillers previously discussed in the POB generation section. The specifications for these chillers are listed in Table 4.8.

Table 4.8 Cancer Center Chiller Specs

Cancer Center Chiller Specifications		
	Chiller 1	Chiller 2
Type	Screw Air Cooled	Screw Air Cooled
Capacity (tons of cooling)	60	60
Refrigerant	R-22	R-22
Voltage	460/3	460/3
Evaporator Pressure Drop (ft of head)	20	20
Exiting Water Temp (degrees F)	45	45
Design Ambient Temp (degrees F)	95	95
Design Evaporator Flowrate (gpm)	138	138

4.4.9 Main Hospital Steam Transport

The heat contained in steam generated by RMC's boilers must be distributed throughout the facility for use. As steam is generated by the boilers it accumulates and builds pressure in the system. This creates a gradient causing steam to flow through a piping network to equipment where the heat contained in the steam is extracted. The steam piping network at RMC consists of different sizes of schedule 40 steel pipes. The largest pipe is at the main steam header. The piping reduces in size as it braches off throughout the network. Piping is sized based on flow capacity. The steam at RMC flows to shell and tube heat exchangers, sterilizers, and humidifiers. These systems either consume the steam or extract the heat from it; they will be discussed in detail later in the end use section of this thesis.

Some steam using equipment at RMC requires lower steam pressures than what is needed to properly distribute the steam throughout the hospital. Steam pressure must be reduced before it enters this equipment. At RMC, steam pressure reducing stations can be found throughout the facility. Steam pressure reduction is accomplished by a steam pressure reducing valve or PRV. RMC primarily utilizes pilot operated pressure reducing valves. The valves function by balancing downstream pressure against a control spring. A downstream sensing line senses pressure downstream and feeds pressure back to the pilot diaphragm. This pressure throttles the pilot valve to provide a constant flow to the main diaphragm which controls the main valve. When low pressure is sensed downstream the spring opens the pilot valve which allows upstream pressure to open the main valve. When downstream pressure rises it is balanced against a control spring which throttles the pilot valve. The pilot valve reduces the pressure to the main diaphragm which closes the main valve to control flow. The reduced pressure steam then flows through the equipment for use. An upstream water separator filters water to ensure dry steam is fed into the equipment. The downstream safety valve is there to protect the equipment from a PRV failure. The safety valve will lift and relieve pressure at a point before the protected equipment’s max pressure rating. When the heat is extracted from the steam, it condenses back into water. As discussed earlier in the steam boiler section, steam traps trap the useable steam in the system so all thermal energy can be extracted. Steam traps must also allow condensate to recirculate back to the condensate tank. RMC has steam lines that extend a large distance from the condensate tank which make it is necessary to pump condensate back to the tank. RMC has a total of five condensate pumps to help boost the condensate back to the tanks. These pumps have a combined flow rate of 34 gallons per minute at max capacity. They modulate on and off as needed. The specs for these pumps are listed Table 4.9.

Table 4.9 Condensate Pump Specs

Condensate Pump Specs					
	CP-1	**CP-2**	**CP-1E**	**CP-G1**	**CP-2E**
Type	Duplex	Simplex	Simplex	Simplex	Simplex
SQ.FT. EDR	30,000	1000	1000	1000	1000
Flow (gpm)	30	1	1	1	1
Discharge Pressure (psig)	30	20	20	20	20
Motor HP	1	0.25	0.25	0.25	0.25

4.4.10 Main Hospital Hot Water Transport

Process hot water is the medium that carries most of the energy used to heat occupied building zones. After this hot water is generated, it must be distributed by transport systems. The process hot water distribution system contained in the main hospital is a closed loop constant volume system with a piping network that extends hospital wide. Process hot water is pumped to equipment such as air handlers, air terminal boxes, and fan coil units. After its available heat is removed by this equipment, it is then recirculated back to the generation systems. Process hot water is transported through schedule 40 steel pipe wrapped in thermal insulation. The piping diameter decreases with the hot water flow capacity as the system branches off from the main piping. Process hot water flow is driven by constant speed, electric motor driven, centrifugal pumps plumbed in parallel. The specifications for these pumps are listed in Table 4.10.

Table 4.10: Main Hospital Hot Water Pump Specs

Process Hot Water Pump Specifications					
Mark	**Type**	**GPM**	**Head (Ft)**	**Motor HP**	**Motor V/Ø**
HWP - 1	Base Mounted	300	80	15	460/3
HWP - 2	Base Mounted	300	80	15	460/3
HWP - 3	In - Line	17	45	1	460/3
HWP - 4	Base Mounted	175	65	5	460/3
HWP - 5	Base Mounted	175	65	5	460/3

In centrifugal pumps, an electric motor driving a rotating impeller inside the pump housing accelerates the fluid outward creating fluid pressure to drive the flow. The primary process hot water pump is HWP-1. The pumps were sized based on the flow capacity of the heat exchanger converters. HWP-2 is used as a backup pump for system redundancy. HWP 4 is also a primary hot water loop pump with HWP-5 used as its backup. HWP-3 is used as an inline booster pump to boost flow to distant equipment. It is located in the pent house located on the third floor which is a significant distance away from the primary pump.

Water flows away from the pumps through the piping network where control valves modulate water flow through devices like heating coils. RMC utilizes pneumatically actuated two way valves to modulate the flow through heating coils positioned in air handlers and air terminal boxes. Pneumatically actuated valves use compressed air to drive valves open and closed. The control valves for process hot water at RMC are either gate valves or globe valves depending on the characteristics of the flow. Balancing valves and check valves are also found in the piping network where needed. Balancing valves are used to maintain water flow to specific areas within design parameters. Check valves prevent backflow in the system.

Another important piece of equipment found in the RMC process hot water distribution network is the expansion tank. An expansion tank is a pressure vessel used to absorb excess pressure caused by the thermal expansion of hot water. As water is heated, its volume expands increasing the pressure on the piping. In order to compensate, expansion volume is needed. An expansion tank contains a dry side and a wet side separated by a rubber diaphragm. The dry side is filled with compressed air and the wet side is connected to the hot water piping network bringing it in contact with the water contained in the system. As heat is added and thermal expansion occurs, the dry side air is compressed decreasing the dry side volume thereby effectively absorbing the excess pressure. All equipment mentioned above works together to distribute process hot water at RMC. Figure 4.7 is not an exact representation of RMC’s hot water distribution system; it is intended to illustrate how RMC's hot water distribution equipment is interconnected.

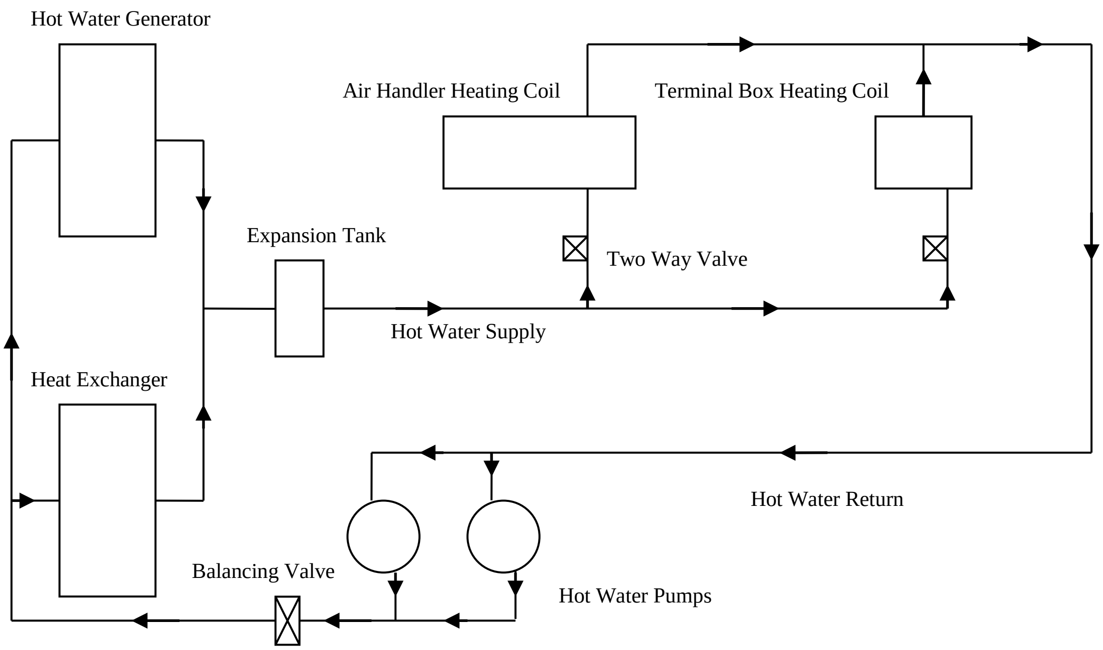


Figure 4.7: Hot Water Distribution Schematic

4.4.11 Main Hospital Chilled Water Transport

Heat gained from the building must be removed in order to accomplish building cooling requirements. Chilled water is the primary transport medium for this unwanted building heat. Once chilled water is generated, it must be transported throughout the facility in order to absorb heat. This is accomplished by RMC's chilled water distribution system. There are multiple configurations for chilled water distribution systems. RMC distributes chilled water with a circulating, constant volume, three way valve system. A constant volume system indicates that constant chilled water flow is maintained around the loop at all times. Chilled water flow is induced by two constant volume centrifugal pumps. The specifications for these pumps are listed in Table 4.11.

Table 4.11: Main Hospital Chilled Water Pump Specs

Chilled Water Pump Specifications					
Mark	Type	GPM	Head (Ft)	Motor HP	Motor V/Ø
CHP - 1	Base Mounted	630	90	20	460/3
CHP - 1	Base Mounted	864	90	25	460/3

These pumps were specified based on chiller flow capacity. RMC's chilled water pumps have a total flow capacity of 1,494 gallons per minute, and are plumbed in a parallel configuration. A general rule is that pumps are plumbed in parallel in order to achieve optimal flow or in series to achieve optimal pumping head. RMC's chilled water piping is made of schedule 40 steel piping at different diameters based on flow capacity requirements. The piping is wrapped in thermal insulation to minimize heat gain or loss. The chilled water distribution system utilizes chiller isolation valves in order to achieve chiller isolation. Chiller isolation implies the ability of a specific chiller to serve particular zones. Engineers designed RMC's network for chiller one to serve critical areas such as operating rooms and medical laboratories that require lower chilled water temperatures, while chiller two serves the remainder of the lower demand areas with higher chilled water temperatures. The network was designed with isolation valves placed in the chilled water supply piping, so that flow is diverted to the proper area. If a chiller is tripped offline due a mechanical failure or brought offline for maintenance, these isolation valves can be opened allowing either chiller to serve any area. This configuration also places each chiller with a designated pump. The system is not plumbed in a header configuration where any pump can serve any chiller. Flow can also bypass the chillers into a plate and frame heat exchanger when environmental conditions permit in order to utilize the cooling tower for free cooling.

Once chilled water passes through the chillers, it flows to air handler chilled water coils for use. As discussed previously in this section, RMC employs a constant volume three way valve system. In order for constant volume chilled water pumps to maintain constant flow throughout the network, total system pumping head cannot change. If a chilled water coil control valve modulates closed, flow must be directed through a bypass line of equal resistance so that pumping head does not significantly change. Three way chilled water control valves and air handler bypass lines with balancing valves allow a constant volume system to work properly.

Three way control valves modulate chilled water flow through air handler cooling coils based on cooling demand. As these valves modulate closed, flow is diverted into a cooling coil bypass line to the chilled water return piping. Water balancing valves tuned to create the same flow resistance as the cooling coil are placed in the bypass lines to maintain the proper system head. Numerous balancing and check valves are located throughout the network to provide proper water flow balance and to prevent backflow.

The pumps, chiller isolation valves, three way control valves, bypass lines, and balancing valves all work together to keep a steady balanced chilled water flow circulating throughout the hospital to remove building heat. Figure 4.8, while not an exact representation of RMC's system, fundamentally illustrates the three way valve constant volume chilled water distribution system used.

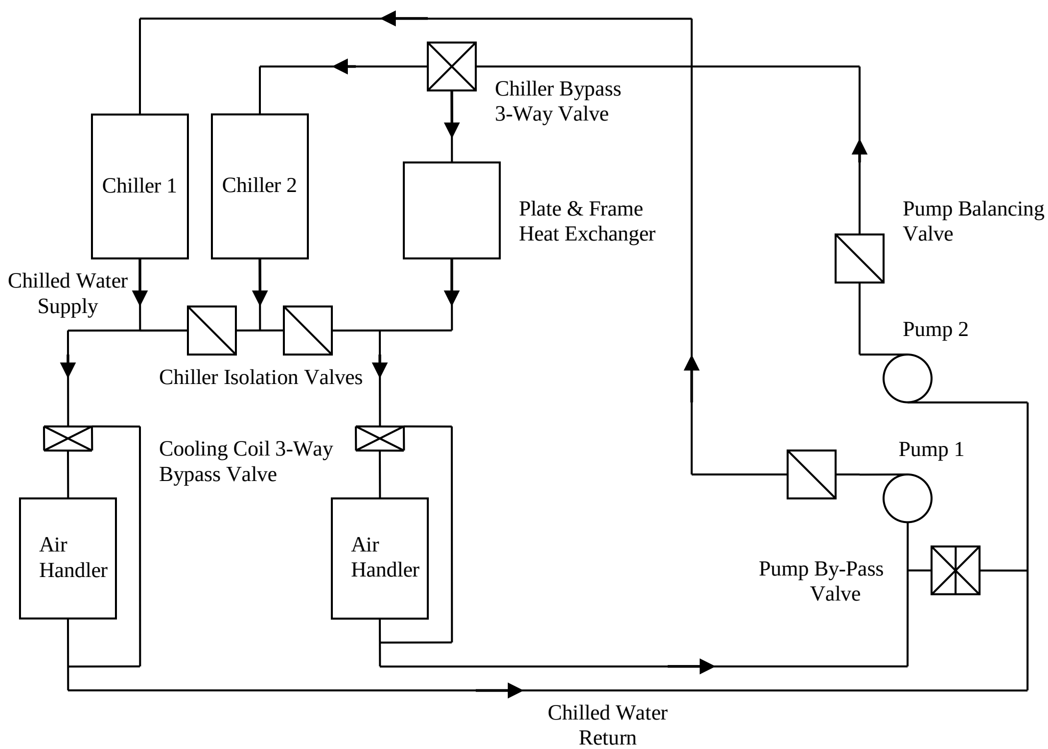


Figure 4.8: Constant Volume Chilled Water Distribution Schematic

4.4.12 Main Hospital Ventilation and Conditioned Air Transport Systems

Specific indoor environmental conditions must be met for a medical facility like RMC to operate properly. Proper building ventilation must exist. Building ventilation refers to the amount of fresh outdoor air that is circulated through the facility. ASHRAE has strict standards for proper building ventilation. It requires a certain amount of fresh air to be brought into the facility based on occupancy load. This is intended to provide proper indoor air quality for the building's occupants. Humans give off carbon dioxide as a byproduct when breathing. High carbon dioxide levels are dangerous for building occupants. ASHRAE states that the maximum carbon dioxide concentration that can exist anywhere in an occupied building is 1000 parts per million. In order to keep this level at a minimum, contaminated air must be exhausted, and fresh make up air must take its place. Along with keeping carbon dioxide levels down, proper ventilation also keeps other dangerous airborne contaminates such as radon gas and carbon monoxide to a minimum. In addition to proper building ventilation, indoor temperature and relative humidity levels must be maintained. Different parts of RMC require different temperature and relative humidity levels. Zone temperature levels are set to provide occupant comfort, and to keep certain equipment within its operating temperature standards. Relative humidity level is important in hospitals to provide occupant comfort, prevent mold growth, and to act as a spark arrester in areas where pure oxygen is used for medical purposes. At RMC, all of the aforementioned indoor environmental requirements are met with the ventilation and conditioned air transport system.

The primary ventilation and conditioned air transport system at RMC is the air handler and air terminal box system. Air handlers and air terminal boxes can have many configurations but they all accomplish the same goal. An air handler basically performs three transport functions. First, it transports building heat from air to water or vice versa. Secondly, it transports moisture to or from the air. Thirdly, it transports fresh conditioned air to zones throughout the facility. As discussed earlier, chilled and heated water is pumped out to air handlers where it flows through cooling and heating coils contained within the air handler units. These coils are finned tube, cross flow heat exchangers. In an air handler like at RMC, air flow is induced by electric motor driven fans also contained within the air handler. The fans force air across the heat exchangers where heat is transported. When building heat is required, sensible heat is transferred from the hot water flowing through the heating coil to the cool air flowing across the coil. When cooling is required, heat is absorbed from the warm air flowing across the cooling coil by the chilled water flowing through the cooling coil. The cooling coil also performs dehumidification. RMC’s geographical location places it in a climate zone with relative humidity levels above hospital requirements during most of the year. During these times, the cooling coil cools the air below the dew point which causes the moisture in the air to condense thereby removing it from the supply air. The condensation of water also generates latent heat. The cooling coil effectively absorbs both sensible and latent heat. Humidifiers located downstream of the coils can add moisture to the air when needed. A humidifier like at RMC sprays steam into the supply air thereby adding moisture to the air increasing relative humidity. The air handler fans are the prime mover for air which becomes the new heat transport medium. Air is transported through ducting of varying sizes and configurations in order to balance and provide the correct airflow. All supply and return air circulated through RMC is transported through ducting. Heat and moisture absorbed by the conditioned air in the space is transported through the ducting network back to the air handler to be reconditioned, or heat and moisture is carried out to the space through the ducting by the flowing air. Also, fresh ventilation air is transported to the required zones to maintain indoor air quality.

A typical air handler at RMC is basically a large box shaped plenum through which air is forced across several components. The basic components of the air handlers are air filters, heating coils, cooling coils, fans, humidifiers, and air flow dampers. Outdoor air enters the air handler through ductwork routed from the exterior of the building in a proper location. Outdoor air flow is modulated by a damper placed in the ductwork or at the entrance of the air handler. This damper can be either be preset to a permanent position or it can modulate open and closed to control the amount of fresh air that is brought in to the air handler. Both cases are found at RMC. It is discharged into a mixing chamber where it is mixed with recirculated return air that is returned through ducting from the zones the air handler serves. Return air is also modulated by a damper placed in the ductwork or at the entrance to the air handler. Outdoor and return air is then mixed in a mixing chamber. The mixing chamber acts like an energy recovery unit to raise or lower the enthalpy of the air based on the need for heating or cooling. The mixed air then passes through a filter used to remove particulate contaminants found in the air. After moving across the filter, air then flows across a preheat heating coil. It is primarily used in extremely cold outdoor temperatures when the mixing chamber cannot raise the temperature above the freezing point of water. Air then flows over a cooling coil which is used to remove the sensible heat and latent heat load from the air. Once air is across the cooling coil it enters the fan where it is accelerated to be sent out for use. The air then flows through another filter and across the humidifier where the relative humidity of the air can be raised if needed. The conditioned air then flows through supply ducts to the sir conditioned space. Numerous balancing dampers are located throughout the duct network in order to properly divert and balance the airflow to each zone. Figure 4.9 illustrates a typical RMC air handler layout.

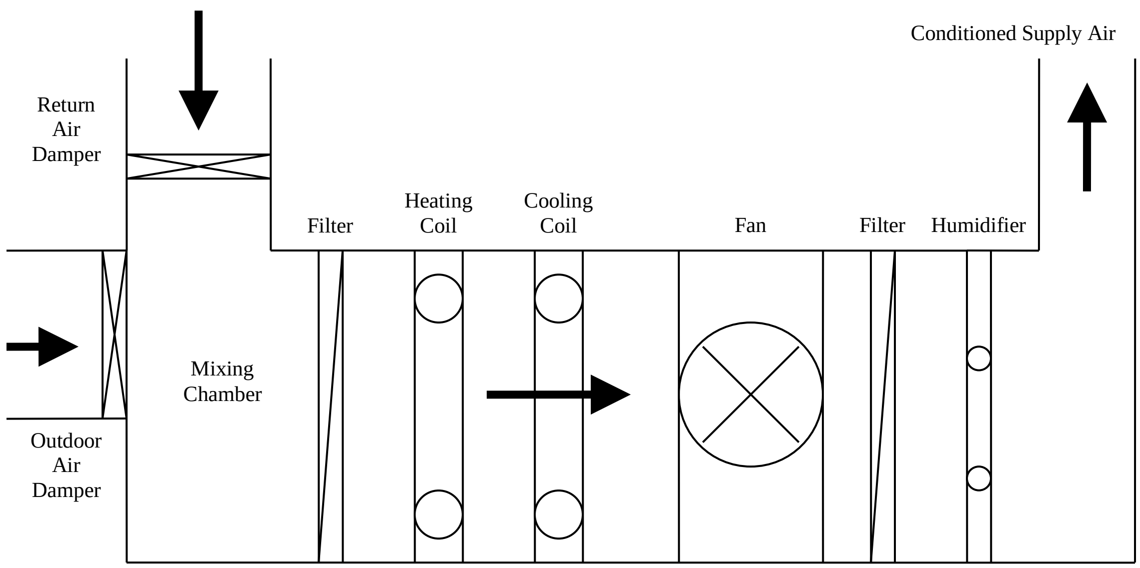


Figure 4.9: RMC Air Handler Layout

Air handlers and air terminal boxes at RMC can be classified into three main categories. The categories are constant volume with terminal reheat, variable volume with terminal reheat, or constant volume single zone. These categories basically refer to how the air handlers work in conjunction with their respective air terminal boxes.

At RMC, air terminal boxes serve two general transport functions. They transport the heat from hot water into the air with air terminal box mounted reheat coils and distribute fresh conditioned air into the space. In order to dehumidify, the air temperature is dropped below the dew point temperature by the air handler cooling coil. The dew point temperature is often well below zone temperature comfort levels. For example, oftentimes air is cooled in the air handler to a temperature of fifty-five degrees Fahrenheit, but the zone temperature requirement is seventy degrees Fahrenheit. The supply air must be warmed in order to satisfy zone requirements. This is accomplished with heating coils mounted in the air terminal box. These heating coils are basically smaller versions of air handler heating coils. Air terminal boxes at RMC are either constant volume or variable volume. Constant volume air terminal boxes receive and distribute a constant volume of air continuously. A constant volume air terminal box is basically a box containing a balancing damper and a mounted heating coil connected to the supply ductwork. At RMC, constant volume boxes are primarily used in critical areas like operating rooms with high fresh air change requirements and low temperature set points. Variable volume air terminal boxes are more complex. They consist of an electronically controlled and actuated airflow damper that modulates the supply airflow based on the zone heating load. The damper opens to allow more conditioned airflow into the space in order to satisfy the zone heating or cooling load. Once the heating or cooling load is satisfied the damper modulates to a minimum outdoor airflow position predetermined by the zone occupancy requirements. With the air flow damper at a minimum position, reheat coils are used to maintain the space discharge temperature at a level to satisfy space requirements.

The constant volume systems with terminal reheat at RMC consist of an air handler with a constant speed fan. The fan operates at a constant speed and generates enough duct pressure to deliver the required airflow to each air terminal box. Each box is sized and balanced to deliver the design airflow to a space. If required, terminal reheat is used to raise the supply temperature air from the air handler before being discharged into the space. Outside and return air dampers are typically fixed in a permanent position based on outdoor air flow requirements. Table 4.12 contains a list of the constant volume air handlers at RMC along with available design specs. Not all data was available. Constant volume air handlers serve 26 constant volume air terminal boxes throughout the main hospital at RMC.

Table 4.12: Constant Volume Air Handler Specs

Constant Volume Multizone Airhandler Units - Main Hospital									
	AHU - OR1	AHU - OR2	AHU - 3R	AHU - 2C	AHU - 3B	AHU - 2	AHU - 4	AHU - G2	AHU - 3A
Design Airflow (cfm)	-	5580	5,065	4,950	4,070	18,125	3,550	2,860	2,380
Design Outside Air (cfm)	-	-	1,025	885	-	-	-	970	-
Design Cooling Coil Water Flow (gpm)	-	67	38	35	40	130	19	21	18
Design Cooling Load (Tons of Cooling)	-	27	15	14.5	16.6	54	7.9	8.7	7.5
Design Heating Coil Water Flow (gpm)	-	-	11	9.4	-	-	-	4	-
Design Heating Coil Load MBH	-	-	220	187	-	-	-	71	-
Fan Horsepower	15	10	7.5	7.5	7.5	30	7.5	3	7.5

Variable volume systems with terminal reheat consist of an air handler equipped with a variable air volume fan design connected to variable volume air terminal boxes. The fan is equipped with inlet guide vanes to modulate the air flow across the fan. The inlet guide vanes on the fan and the variable volume air terminal boxes work together to modulate air flow to zones. When downstream air terminal box air flow dampers are open, the inlet guide vanes on the fan are also wide open loading the fan with the maximum amount of air. When air terminal box dampers satisfy zone conditions and close to a minimum position, there is an increase in static air pressure in the ducting. The increase in static pressure is sensed by sensors located in the ductwork which causes the inlet guide vanes on the fan to modulate closed reducing air flow until the static pressure set point is achieved. When the air terminal boxes are in minimum position, reheat coils modulate to maintain space temperature. Outdoor air and return air dampers also modulate in order to compensate for the changing air volume. As the total volumetric flow rate of air decreases, the outside air damper must modulate open in order to maintain a correct fresh air ratio. The variable volume air handlers at RMC are listed in Table 4.13 along with available specifications for each.

Table 4.13: Variable Volume Air Handler Specs

Variable Volume Multizone Airhandler Units - Main Hospital									
	AHU-OR3	AHU-1E	AHU-G1	AHU-2E	AHU-MRI	AHU-ER	AHU-ICU	ER-Isolation	L-1
Design Airflow (cfm)	22,400	22,000	8,380	14,000	-	-	-	-	-
Design Outside Air (cfm)	5,610	5,000	3,400	4,600	-	-	-	-	-
Design Cooling Coil Water Flow (gpm)	175	169	85	110	-	-	-	-	-
Design Cooling Load (Tons of Cooling)	73	70	35	45	-	-	-	-	-
Design Heating Coil Water Flow (gpm)	42	38	15	24	-	-	-	-	-
Design Heating Coil Load MBH	847	760	290	484	-	-	-	-	-
Fan Horsepower	50	50	15	30	-	-	-	-	-

Variable volume air handler units serve 110 variable volume air terminal boxes at RMC. Figure 4.10 illustrates how air handlers and air terminal boxes are interconnected.

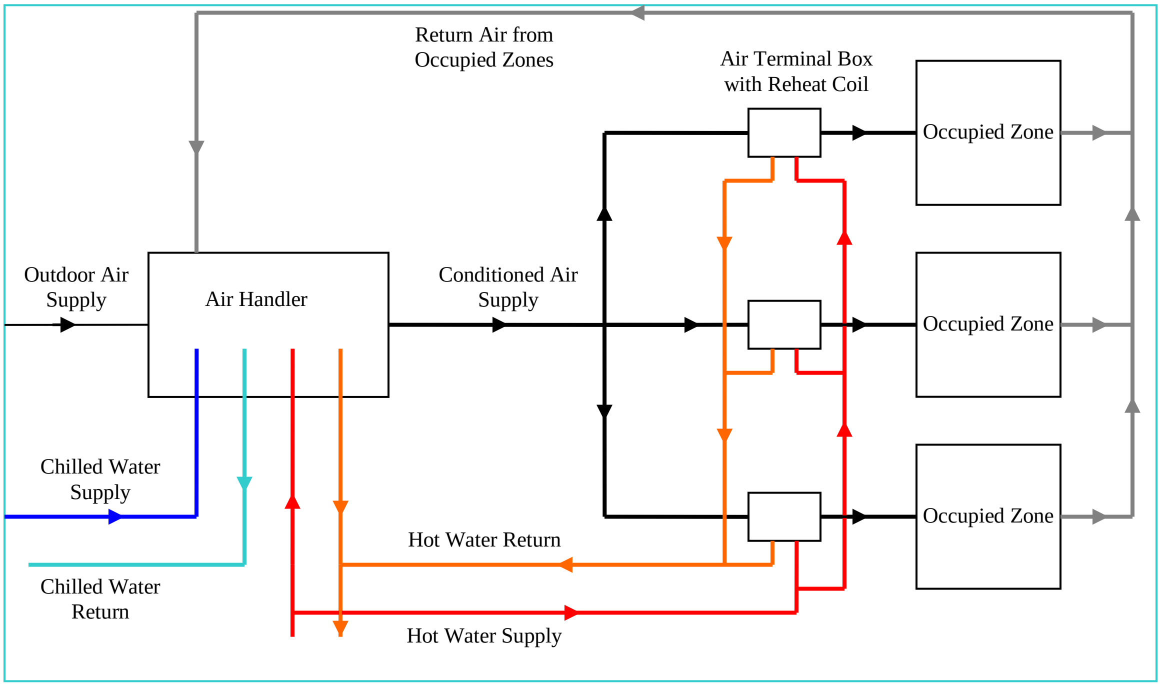


Figure 4.10: Air Transport System Schematic

It is important to note that AHU- OR3 serves hospital operating rooms and functions as a constant volume system during the daily operating schedule in order to ensure positive room pressure and sufficient zone air changes.

There are two constant volume single zone units in the main hospital at RMC. Constant volume single zone air handlers utilize a constant volume fan to transport air to a single zone. This configuration has no air terminal boxes. All air conditioning is done inside the air handler and is discharged from the air handler in the state that the occupied zone requires. Table 4.14 contains the available specifications for the two constant volume single zone air handlers at RMC.

Table 4.14: Constant Volume Single Zone AHU Specs

Constant Volume Single Zone Airhandlers - Main Hospital		
	AHU - M	AHU - K
Design Airflow (cfm)	1,200	8,100
Design Outside Air (cfm)	150	-
Design Cooling Coil Water Flow (gpm)	7	100
Design Cooling Load (Tons of Cooling)	3	41
Design Heating Coil Water Flow (gpm)	2	-
Design Heating Coil Load MBH	22	-
Fan Horsepower	1	10

As discussed previously in this section, air handlers bring in a specified amount of fresh outdoor air. In order to draw in this continuous amount of fresh air, used air must be exhausted back to the outside. Failure to exhaust used air will result in over pressurization of the building and air ducting. Typically medical facility operators want to maintain building pressure slightly positive or neutral to help prevent infiltration. In order to accomplish this, facilities generally exhaust only the amount of fresh air that is brought in or slightly less. RMC relieves unwanted air with a network of 26 constant volume, electric motor driven exhaust fans located on the roof. Ducting networks connect the fans to the zones the air is exhausted from. The exhaust fans at RMC are interlocked with their respective air handlers so that if an air handler fan trips offline, so does the exhaust fan that is interlocked with it. This prevents the building pressure from going negative.

4.4.13 POB Hot and Chilled Water Transport Systems

Like in the main hospital, hot water is the medium that carries building heat throughout the facility. Hot water travels through a closed loop distribution network that distributes and recirculates hot water primarily to air handlers and air terminal box reheat coils. Hot water is also delivered to a shell and tube heat exchanger, unit heaters, and powered induction units. Hot water is pumped by three centrifugal electric motor driven pumps through two piping loops. Two of these pumps are plumbed in parallel and serve a hot water loop plumbed through POB three, while the remaining pump serves a separate hot water loop in POB 2. The specifications for these pumps are listed in Table 4.15.

Table 4.15: POB Hot Water Pump Specs

POB Hot Water Pumps			
	HWP-1 (POB2)	HWP-1 (POB3)	HWP-2 (POB3)
Discharge Flow (gpm)	60	114	114
Pumping Head (FT)	50	70	70
Pump RPM	1750	1750	1750
Motor Horsepower	2	5	5

Hot water piping in the POB is schedule 40 steel pipe of various sizes wrapped in thermal insulation. Numerous balancing valves are placed throughout the network to balance hot water flow. Once hot water reaches air handler heating coils and terminal box reheat coils, the flow rate is modulated with two way valves. Expansion tanks are also included in each hot water loop to compensate for the thermal expansion of water.

The POB employs chilled water as the medium to absorb heat gained in the building and transport it away. Chilled water is circulated through the POB in a thermally insulated piping network. Like the main hospital, the POB also uses a constant volume three-way valve system to distribute chilled water. The system is designed with four constant volume centrifugal pumps. Two pumps are plumbed in parallel and located in the basement mechanical room in POB three to serve the lead screw type chillers. Two more also plumbed in parallel are located in the penthouse of POB two to serve the lag chillers when they are cycled on. The specifications for these four pumps are listed in Table 4.16.

Table 4.16: POB Chilled Water Pump Specs

POB Chilled Water Pumps				
	CHP-1 (POB2)	CHP-2 (POB2)	CHP-1 (POB3)	CHP-2 (POB3)
Discharge Flow (gpm)	126	126	294	294
Pumping Head (FT)	60	60	80	80
Pump RPM	1750	1750	1750	1750
Motor Horsepower	5	5	7.5	7.5

Once the chilled water is pumped through the chillers and regenerated, it flows out to POB air handler chilled water coils where three-way valves modulate flow through or around the coil based on demand. Chiller isolation valves are present, but are not used in the same manner as in the main hospital. They are used simply to valve off a chiller when it is offline. All chilled water regenerated by chillers flows into a main header and is distributed to all air handlers. No POB chiller has a designated pump.

4.4.14 POB Ventilation and Conditioned Air Transport Systems

The POB, like the main hospital, has indoor environmental conditions that must be maintained in order to provide occupant comfort. Even though the environmental specifications are not as tight as those for the main hospital, temperature, humidity, and ventilation must be controlled. The POB primarily consists of office space, private medical practices, and physical fitness facilities. This generally makes the POB a fairly densely populated area most of the day. Adequate ventilation must be provided to keep carbon dioxide levels within regulation. Fresh conditioned air is brought in with air handler systems similar to those in the main hospital, but some have different configurations. These air handlers serve air terminal boxes throughout the facility. The POB's provides conditioned air with nine variable volume air handler systems. The nine air handler units are listed in Table 4.17 along with available specifications.

Table 4.17: POB Variable Volume Airhandlers

Variable Volume Multizone Air Handler Units - POB									
	AC-GE	AC-1E	AC-2E	AC-G	AC-1	AC-2	AHU-1	AHU-2	AHU-3
Design Airflow (cfm)	13,000	13,000	16,000	17,500	12,000	12,000	-	-	-
Design Outside Air (cfm)	1700	1700	1800	3500	1800	1800	-	-	-
Design Cooling Coil Water Flow (gpm)	78	80	94	134	74	86	-	-	-
Design Cooling Load (Tons of Cooling)	32	33	39	55	31	36	-	-	-
Design Heating Coil Water Flow (gpm)	8	8	13	48	33	33	-	-	-
Design Heating Coil Load MBH	115	115	160	481	329	329	-	-	-
Fan Horsepower	22.5	22.5	27.5	40	20	20	-	-	-

Some of these air handlers are designed with different configurations than what is found in the main hospital. AC-GE, 1E, and 2E have a return fan in series with a supply fan. They also have built in return air relief. The built in return air relief damper is designed to relieve return air in order to take on fresh make up air. As discussed earlier, some air must be exhausted from the building in order to make room for incoming fresh air, control building static pressure, and to provide continuous ventilation. Instead of the air handler being interlocked with an exhaust fan that removes secondary air from the occupied zone. The unwanted secondary air is removed at the air handler through the relief damper. Figure 4.11 illustrates this air handler configuration.

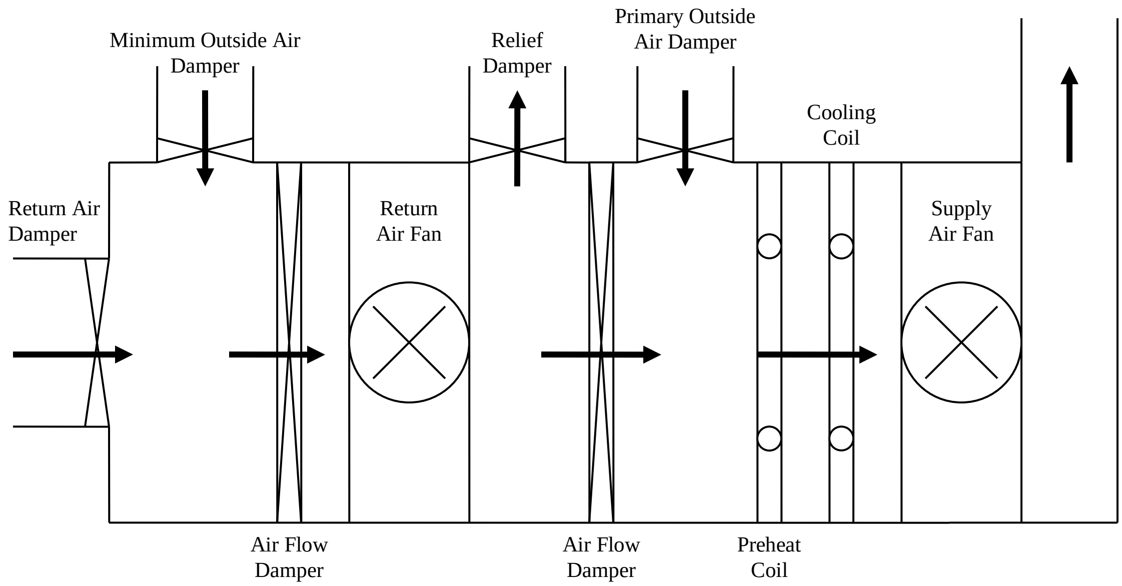


Figure 4.11 POB Airhandler Configuration

AHU-3 also has a slightly different configuration. This unit is located deep inside POB one. This location creates a long distance for outside air to travel to reach the air handler. To compensate, AHU-3 was designed with an inline outside air booster fan mounted inside the ductwork to boost make up air flow. AHU-3 also does not have a preheat coil. It can only provide building cooling. AHU-1 and AHU-2 are set up to use plenum return. These two units are located on the roof of POB one inside the penthouse. They are positioned side by side inside the penthouse where return air is discharged directly into the open space. The air handlers draw their return air from the open space within the penthouse where it is mixed with fresh outside air drawn through the outside air ducts. The diagram below illustrates this configuration.

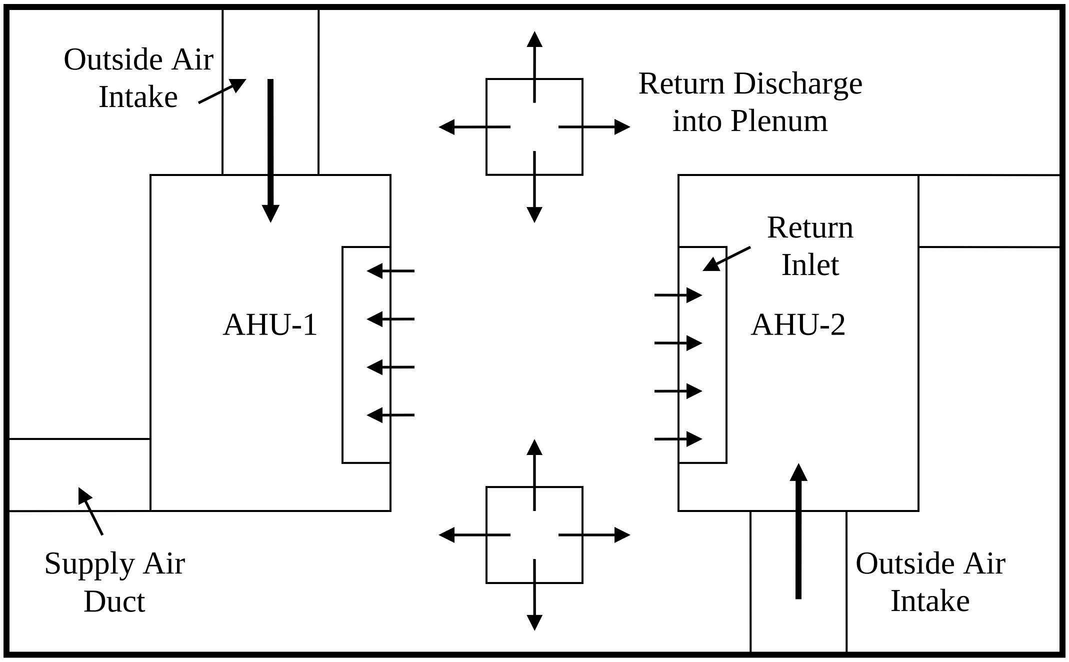


Figure 4.12 Alternate POB Airhandler Configuration

The nine POB air handlers serve three different types of variable air volume terminal boxes. The POB contains sixty two variable volume boxes with reheat coils. They function like the ones discussed previously in the main hospital section on air terminal boxes. There are seventeen air terminal boxes that do not contain reheat coils. These air terminal boxes modulate the air flow damper only to control zone temperature. The damper modulates open in cooling mode to flow more conditioned air to cool the occupied space. If zone temperature requirements are satisfied, the damper will fully close. The POB also uses a device known as a powered induction unit. A powered induction unit is a device that mixes primary conditioned air with secondary or recirculated air, from the space. It employs a fan to pull secondary air out of the zone and mix it with the primary air being delivered through ducting from the air handler. It also contains a heating coil to raise the temperature of the mixed air if needed.

Another significant difference between the main hospital and POB air distribution system is the type of air handler fan. Instead of modulating air flow with inlet guide vanes, the POB air handlers are equipped with variable frequency drive (VFD) fans. Variable frequency drive fans have the capability to throttle back fan rotational velocity as demand decreases thereby decreasing the air flow rate. As terminal box air flow dampers satisfy, static pressure increases beyond the set point in the supply air ductwork. This pressure is sensed and the VFD fans compensate by decreasing fan rpm. As dampers modulate open and duct static pressure drops, the VFD increases its fan speed to raise pressure and increase flow.

4.4.15 Cancer Center Hot and Chilled Water Transport Systems

Hot water is distributed to the cancer center air handlers and air terminal boxes through a circulating hot water piping network. Flow is provided by two constant volume centrifugal pumps plumbed in parallel. The specifications for these pumps are found in Table 4.18.

Table 4.18 Cancer Center HW Pump Specs

Cancer Center Hot Water Pump Specifications		
	HWP-1	HWP-2
Design Discharge Flowrate (gpm)	43	43
Design Pumping Head (FT)	33	33
Motor Horsepower	2	2

These pumps feed a piping network equipped with an expansion tank to compensate for thermal expansion. Once hot water reaches the air handlers and air terminal boxes, flow is modulated by two way hot water valves. Balancing valves are found throughout the network to balance water flow.

The chilled water distribution system for the cancer center is also a constant volume three-way valve system. Chilled water flow in induced by a single constant volume centrifugal pump. The specifications for this pump are listed in Table 4.19.

Table 4.19: Cancer Center Chilled Water Pump Specs

Cancer Center Chilled Water Pump Specifications	
	CHP-1
Design Discharge Flowrate (gpm)	138
Design Pumping Head (FT)	57
Motor Horsepower	5

The chilled water travels through insulated schedule 40 steel pipes to air handlers where three-way valves modulate flow through the cooling coil or through the bypass. Chilled water then returns to the chillers where it is regenerated. Balancing valves are found throughout the network to balance flow.

4.4.16 Cancer Center Ventilation and Conditioned Air Transport Systems

Ventilation and conditioned air distribution in the cancer center is carried out with two variable volume air handlers that serve 37 variable volume air terminal boxes with reheat. Both cancer center air handlers have the same configuration found in POB air handlers AC-GE, AC-1E, and AC-2E. They have built in return air relief along with a return air fan and a supply air fan. The specifications for the cancer center air handlers are listed in Table 4.20.

Table 4.20: Cancer Center Airhandler Specs

Variable Volume Mutizone Air Handler Units - Cancer Center		
	AC-1	AC-2
Design Airflow (cfm)	15,315	10,910
Design Outside Air (cfm)	2,300	1,010
Design Cooling Coil Water Flow (gpm)	105	70
Design Cooling Load (Tons of Cooling)	44	29
Design Heating Coil Water Flow (gpm)	10	10
Design Heating Coil Load MBH	100	100
Fan Horsepower	40	30

The air handler fans are equipped with variable frequency drive electric motors. All air terminal boxes in the cancer center are variable volume with terminal reheat. Five exhaust fans pulling air from bathrooms and medical labs also exhaust unwanted air from the building.

4.4.17 Main Hospital End Use - Hot Water Converters / Steam Coils / Humidifiers

The first steam end use system in the main hospital to discuss is the hot water convertors. Hot water convertors refer to non-storage calorifier shell and tube heat exchangers that use steam to heat hot water. They consist of a bundle of tubes fitted with tube sheets that separate the two fluid mediums enclosed in an outer cylindrical shell. Steam is the heat transferring medium which flows through the U-shaped tubes surrounded by water flowing through the outer shell. The water absorbs the heat from the steam primarily by convection as it passes around the tubes. As the heat is absorbed by the water, the steam condenses and flows out of the tubes as condensate. The convertors at RMC use one shell pass and two tube passes to produce process hot water at 180 degrees Fahrenheit. The main hospital contains five hot water convertors with different heating capacities located throughout the facility.

The heating coils in seven air handlers at RMC use steam instead of hot water as the heat transfer medium. Air handler steam coils are cross flow, finned, cooper tube, heat exchangers that transfer heat to moving air to heat the building. The design is very similar to hot and chilled water coils. Basically, steam is discharged into a header which feeds the individual tubes. Smaller diameter tubes make multiple passes through the heat exchanger. The tubes are finned to facilitate better heat transfer. Steam flows through these U-shaped tubes and into a return header where it flows out as condensate. Air blown across the tubes absorbs the heat from the steam contained in the tubes. Steam traps hold steam inside the coils until all available heat is absorbed.

Steam Humidifiers are located on some air handlers in the main hospital where strict humidity control is warranted. During the winter months, cold air can become very dry due to building heating as the air is circulated through the facility. The cold air contains little moisture and when it is heated the relative humidity decreases due to the expansion of the air. Low humidity levels can facilitate static electricity sparks and cause breathing problems for patients in the hospital. This makes it necessary to occasionally humidify the air. Humidification is accomplished with direct vaporizer humidifiers located at the air handlers. Steam is directly injected into the airstream by a header which extends across the air handler. The dry air absorbs the moisture contained in the saturated vapor and the relative humidity level is increased. This steam is lost to the system and must be made up with make-up water.

4.4.18 Main Hospital End Use - Medical Sterilizers / Washer / Hot Water Storage

To sterilize medical utensils RMC employs four AMSCO autoclave sterilizers. Steam sterilizes much quicker than hot air. High quality steam must be used. Basically, steam is released into the sterilization chamber where it builds pressure. Steam is less dense than air so it floats above the air. As steam pressure builds the air is pushed out of the chamber through a vent by the steam. These sterilizers have an operating temp of 315 degrees Fahrenheit. They can each consume up to 200 pounds of steam per hour. The washer works similarly to the sterilizers and can consume up to 160 pounds of steam per hour.

The hot water storage tanks at RMC use steam to heat domestic and some process hot water. This water is not recirculated and must be made up. The tanks consist of a larger outer shell containing water with a primary steam coil running down through the tank. Heat is transferred from the steam to hot water. RMC has two 1,000 gallon storage tanks. One tank contains hot water maintained at 180 degrees Fahrenheit while the other is maintained at 140 degrees Fahrenheit. The lower temperature tank serves general domestic hot water needs for showers and sinks. The higher temperature tank is primarily used by kitchen dishwashers and for cooking purposes.

4.4.19 Main Hospital End Use – Fan Coils / Unit Heaters

A Fan coil unit is essentially a small, floor mounted air handler unit not connected to any ductwork that serves the space where it is installed. They contain both a heating and cooling coil supplied by hot and chilled water. A built in fan pulls air directly from the space and blows it across the coils to provide heating or cooling. The fan coil units at RMC do not provide any ventilation. They only condition the air. The units are usually controlled manually or with a thermostat. The main hospital at RMC has four different fan coil unit designs. They have the same configuration but have different heating and cooling capacities. The main hospital upper levels contain several fan coil units mostly located in patient rooms. The unit heaters found in the main hospital consist of a heating coil and a fan. The fan pulls air from the space and blows it across a heating coil with hot water flowing through it. Heat is transferred from the hot water to the air. The main hospital contains five units mostly used to provide heating in mechanical and storage rooms.

4.4.20 Main Hospital End Use - Plate and Frame Heat Exchanger

As discussed earlier in this thesis, the cooling tower can be used to provide chilled water to the main hospital when outdoor environmental conditions permit. The cooling tower condenser water loop is separate from the hospital chilled water loop. The condenser water cooling tower loop must be coupled with the chilled water loop in order to absorb the heat gained in the hospital. The plate and frame heat exchanger at RMC serves this purpose. A plate and frame heat exchanger is basically a series of thin corrugated metal plates compressed together in a rigid frame, used to transfer heat between a primary and secondary fluid. The plates are usually separated by gaskets or welded. The two fluids flow through alternated hot and cold parallel flow channels. The large surface area of the metal plates creates efficient heat transfer. At RMC, the cold primary fluid is the water supplied by the cooling tower and the secondary fluid is the warm water returning from the hospital chilled water loop. Heat is absorbed by the cold primary fluid thereby cooling the warm secondary fluid. The cooled secondary fluid is pumped out to the hospital for use in air handler cooling coils. Table 4.21 lists the specifications of the plate and frame heat exchanger at RMC.

Table 4.21: Plate and Frame Specs

Plate and Frame Heat Exchanger Specifications	
	HX-1
Capacity (MBH)	4300
Cold Side Flow (gpm)	1080
Hot Side Flow (gpm)	864
Cold Side Entering Fluid Temp (degrees F)	50
Hot Side Leaving Fluid Temp (degrees F)	52
Max Pressure Drop Across Heat Exchanger (FT)	20

4.4.21 POB and Cancer Center Hot and Chilled Water End Use Systems

The POB has many of the same end use systems contained in the hospital. One system is a shell and tube heat exchanger to heat water for the indoor swimming pool. This shell and tube heat exchanger utilizes a single pass design for both fluids. Circulated swimming pool water flows through the outer shell while hot water produced by the hot water generator flows through the tube. Heat is transferred from the process hot water to the swimming pool water. The POB also contains numerous fan coil units and unit heaters with the same configuration as described for the hospital units. The cancer center contains unit heaters to provide space heating where needed.

4.5 Building Management System Overview

The large HVAC system at RMC requires an overall building management system to monitor and control indoor environmental conditions. RMC's HVAC systems are managed by two separate automatic direct digital control (DDC) systems. A Siemens DDC system monitors and controls most of the main hospital systems along with some of the POB's systems. A separate Landis and Staefa system controls most of the POB and some main hospital systems. These two systems control the chilled and hot water generation and distribution systems, as well as the ventilation and conditioned air distribution systems. These building automation systems execute process logic based sequences of operation written for the HVAC systems. Both systems contain many microprocessor based distributed control panels (DCP) connected to a central computer. The DCP controllers perform proportional, proportional integral, and proportional integral derivative control loops to maintain different set points based on the application. Two position controllers are also used. The DCPs are connected to the central computer, but function independently of the central computer. The controllers modulate numerous control valves, dampers, and fans based on feedback given from sensors measuring the conditions found at the equipment and in the occupied space. The central computer contains the graphic user interface (GUI). The system can be monitored and adjusted from the GUI. Any system failures and errors are reported on the GUI. Actuators used to modulate dampers and control valves use a 0-10 volt DC signal to control valve movement and position. Sensors are located at BMS controlled mechanical equipment, and in the occupied space. They sense and monitor environmental conditions and equipment status to feedback to the DCP. At RMC, different types of sensors measure things like temperature, humidity, flow rate, static pressure, and electric current or voltage.

4.5.1 Chiller Plant Control

The Chiller Plant control for the main hospital is managed by the Siemens DDC system. Each individual chiller has built in manufacturer controls that locally monitor operation and modulate chiller functions. The Siemens system communicates with the built in manufacturer control to manage the overall system. It manages the system by working together with the built in chiller control to execute a sequence of operations. The Siemens system monitors and modulates chiller operational status, chilled water supply and return temperature, condenser water supply and return temperature, chiller cooling load, pump status, and isolation valve position, cooling tower fan speed, cooling tower temperature, and plate and frame heat exchanger status. The controller executes the sequence of operations to handle chiller start-up and sequencing and uses PID control to maintain set points for each parameter. Safety interlocks are built into the system to prevent dangerous situations like current over load or a chiller starting up with no water flow. The system reports errors and failures to the central computer where they can be seen on the GUI. The Siemens system is programmed to execute the sequence of operations shown on the next page.

Chiller Plant Control Sequence of Operation

a. Chiller 1 starts and runs continuously as the lead chiller, with a duel set-point based on the daily OR schedule (Set-points: 6 AM to 3 PM set-point = 40 degrees Fahrenheit; 3 PM to 6 AM set-point = 44 degrees Fahrenheit). Isolation valves V-1 and V-2 remain open and chiller 2 remains off until 40 degree Fahrenheit chilled water cannot be maintained during 6 AM to 3 PM, or differential temperature on chiller 1 exceeds 12 degrees Fahrenheit, at which time valves V1 and V2 close and chiller 2 runs at a set-point of 44 degrees Fahrenheit.

b. When outdoor temperature permits, chiller 1 remains off, and chiller 2 becomes the lead chiller. If outdoor temperature falls to 45 degrees dry bulb, chiller 2 is commanded off, and valves 3 and 4 open to the plate/frame heat exchanger HX-1.

c. During economizer operation, cooling tower fans cycle to maintain 45 degree Fahrenheit water.

d. During economizer operation, and during the OR schedule 6 AM to 3 PM Monday through Friday chiller 1 runs to maintain 40 degree Fahrenheit leaving chilled water, with valves V1 and V2 closed. Valve V5 modulates to maintain 65 degree Fahrenheit minimum condenser water to chiller 1.

e. If chilled water to the main hospital from the economizer exceeds 55 degrees Fahrenheit, valves V3 and V4 open to chiller 2 and chiller 2 starts.

f. During start up, chiller 2 head pressure is controlled by first opening valve V6 to the by-pass, and then controlling flow to the condenser by modulating valve V7 by chiller head pressure. When condenser water temperature to chiller 2 rises to 65 degrees Fahrenheit minimum, valve V7 modulates fully open, and valve V6 modulates open to maintain 65 degree Fahrenheit minimum.

The POB chillers are managed by the Landis and Staefa system. Chiller sequencing is executed using a lead lag system. The two screw type chillers are the lead chillers carrying the base load. The two reciprocating chillers function as lag chillers and cycle on to boost capacity when demand is high. Individual manufacturer built in PID controllers monitor and maintain chiller operation and set-points at each chiller. Chilled water pumps are also control by the Staefa system and are interlocked with the chillers to ensure flow.

4.5.2 Hot Water Heating System Control

As discussed earlier, hot water is produced with hot water convertors and then distributed in the hot water distribution system. The Siemens system controls and monitors the hot water distribution system by executing the sequence of operations given below.

a. Pump HW-1 operates when commanded by the BMS. When flow is detected, steam valve A and Steam Valve B modulate in sequence to maintain a predetermined reset schedule. The schedule is as follows:

   i. If Outside Air Temp = 10 degrees Fahrenheit, then set-point = 200 degrees Fahrenheit.

   ii. If Outside Air Temp = 60 degrees Fahrenheit, then set-point = 140 degrees Fahrenheit.

b. If hot water flow fails, pump HWP-2 will start and an alarm will be initiated at the central computer.

4.5.3 Air Handler Control

Air handlers located in the main hospital are controlled by the Siemens DDC system. Each air handler has a local DCP controller to execute the control loop to individually and automatically control air handler functions. Each air handler is outfitted with sensors used to monitor control points. Outside air and relative humidity are monitored by global sensors located in specific locations on the outside perimeter of the building. Typically each air handler is equipped with a mixed air temperature, preheat temperature, and supply air temperature sensor placed in the proper location. Static pressure is monitored by pitot tube type differential pressure sensors. Air flow is also monitored by pitot tube differential pressure sensors that measure the total pressure and static pressure differential. Return air relative humidity is monitored by duct mounted thin film polymer sensing elements. Inlet guide vane, damper, and valve position is monitored by potentiometers. All of this information is fed back to the controller and can be viewed on the GUI. Set-Points are adjusted at the GUI. Figure 4.13 shows the GUI for air handler AHU-OR3.

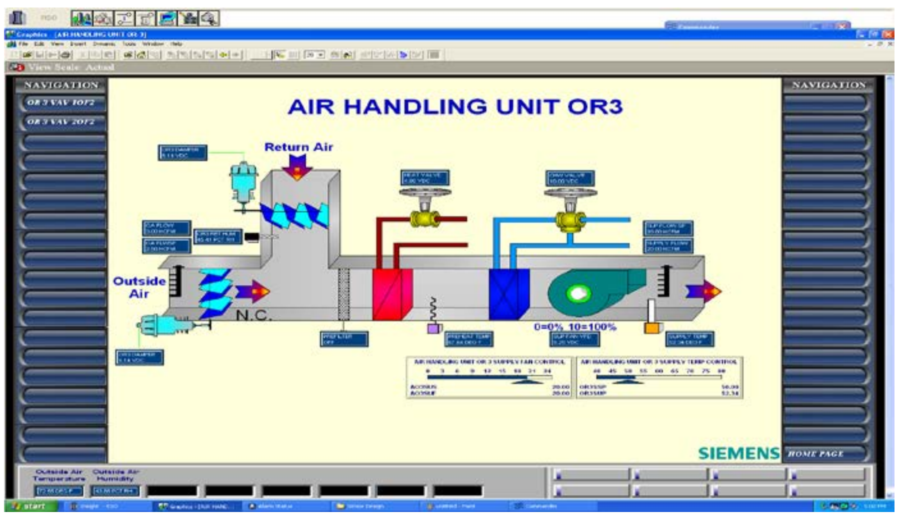


Figure 4.13: AHU OR-3 GUI

Control directives for each type of air handler are contained in a specific sequence of operations. The control system executes these directives to maintain quality environmental conditions. Control sequences for each type of air handler are as follows:

Typical Variable Volume Air Handling Unit

a. Supply fan shall be energized through the BMS as programmed subject to the fire alarm system, and the low temperature cutout.

b. The inlet guide vanes shall modulate to maintain required air flow against static of system as sensed by the static pressure sensor located 2/3 down the supply duct.

c. Outside air quality shall be maintained at a constant value by modulating the outside air and return air dampers to maintain the minimum outside air set-point as measured at the outside air flow monitor. The outside air CFM shall be indicated on graphic user interface.

d. When the supply air temperature falls below set-point, the chilled water valve shall modulate closed and the hot water valve shall modulate open as required to maintain supply air temperature at set-point. When the supply air temperature rises above the set-point, the hot water valve shall modulate closed and the chilled water valve shall modulate open to maintain the supply air temperature at set-point.

e. The return humidity shall be maintained at set-point by modulating the humidifier steam valve, based on a sensor located in the return air duct.

f. Status points will be provided for pre and final filters, and fan status.

Typical Constant Volume Air Handling Unit

a. Supply fan shall be energized through the BMS as programmed subject to the fire alarm system, and the low temperature cutout.

b. When the fan runs, the outside air damper will open.

c. Discharge air temperature set-point will be reset based on a room temperature sensor. The discharge temperature will be maintained by modulating the chilled water valve and the hot water valve in sequence.

d. Pneumatic room thermostats will modulate hot water valves for duct mounted hot water booster coils.

The Staefa system for the POB executes similar control sequences to control its air handlers. The cancer center air handlers are controlled by the Siemens system which utilizes the sequences outlined above.

4.5.4 Air Terminal Box Control

The main hospital air terminal boxes are controlled by DDC controllers managed by the Siemens system. Each box is also outfitted with sensors to monitor and feedback conditions to the controllers. Each air terminal box is outfitted with sensors to measure air flow, room temperature, room humidity, damper position, and hot water valve position. The information from all sensors is visible on the GUI. Each DDC controller executes the sequence of operations below.

Control Sequence for Variable Volume Air Terminal Boxes with Reheat

a. When the space temperature as measured by the space temperature sensor falls below set-point, the controller will modulate the primary air damper to the minimum position, and modulate the hot water valve open to maintain room temperature. When the space temperature rises above set-point, the hot water valve will modulate closed and the damper will modulate open as necessary to maintain space temperature. The damper will modulate open only as far as necessary to maintain the maximum air flow set-point.

b. AHU-OR3 will be programmed for constant volume operation during the OR schedule, or when commanded by the BMS.

The POB Staefa system and the cancer center Siemens system uses the same sequence to control it variable volume air terminal boxes.

4.6 Electric System Overview

4.6.1 Lighting

The lighting at RMC consists mostly of T-8 type fluorescent lamps with electronic ballast ranging from 32 -100 watts. The ballast controls the current that flows into the light to ensure it is at the proper frequency, to avoid a strobing effect. The T-8 style lamps are found throughout RMC in many sizes and fixture combinations. The lighting is controlled by wall mounted manual switches. No occupancy sensors are currently in use at RMC. RMC contains over 8,000 total lamps with an overall lighting power demand of greater than 310,000 watts.

4.6.2 Miscellaneous Electrical Systems

For the purposes of this audit, miscellaneous electrical systems refers to all electrical equipment found in the main hospital with the exception of lighting and HVAC related electrical components. Miscellaneous electrical equipment includes electric motors, remote refrigeration equipment, personal computers, electronic medical analysis equipment, and electronic lab equipment. RMC contains a large amount of miscellaneous electrical equipment with an estimated power demand of over 300 kw.

4.7 Water System Overview

Water usage can be divided into two categories at RMC. The first is process water usage and the second is domestic water usage. Water is provided to RMC by the Alexander City Water Authority. Some systems at RMC recirculate water and require very little make up while others just consume water and dispose it into the sewer system. Water quality is a huge factor for the maintenance and operation of some systems at RMC. Factors such as hardness and alkalinity must be monitored and controlled in order to prevent scale build up and corrosion in equipment and piping.

4.7.1 Process / Domestic Water

Process water is any water that is used to facilitate a process. It is not consumed by building occupants and is generally not potable. Process water generally refers to the water used in the chilled water, hot water, and steam systems. It is often recirculated and treated with chemicals to maintain sustainable operating conditions. Process hot water refers to systems that use water at a temperature of 180 degrees Fahrenheit or more. The chilled and hot water systems are completely closed loop systems and require little make up. However, the cooling tower water loop and steam system requires a significant amount of make-up water due to losses that occur during operation. Cooling towers lose water to evaporation and drift. Steam systems need make up water to replace the water lost from defective steam traps and humidification. Not all process water at RMC is recirculated. Equipment like industrial dish and linen washers use high temperature process hot water and dispose of it into the sewer system.

Domestic water is potable water that is consumed by RMC's occupants. It comes into direct contact with RMC occupants through drinking, showering etc. Domestic hot water refers to hot water generated at a temperature of no more than 140 degrees Fahrenheit. In medical facilities, process hot water must be stored at least at a temperature of 140 degrees Fahrenheit to prevent legionella.

Chapter 5: Data Collection

5.1 Introduction

A successful energy audit depends on accurate useable data. Steps three and four of the RMC energy audit plan are dedicated to collecting energy usage data. Step three calls for preliminary data collection while step four consists of collecting operational data at each energy cost center during a walk through building survey. The main purpose of the data collected at RMC is to expose the current operating conditions and to provide the information needed to perform engineering analysis. Certain data analysis questions must be answered in order to know what data to collect. Random data are not very useful. The auditor must know what kinds of analysis he or she plans to perform and on what systems in order to collect the correct data.

Data was collected from each energy cost center mentioned in the RMC plan multiple times over the course of three years. Past energy bills were gathered along with operational specifications and parameters from HVAC and electrical equipment. Relevant instrumentation was used to measure equipment operational parameters. This chapter will discuss data collection methods and equipment used for the RMC energy audit.

5.2 Preliminary / Building Survey Data Collection

Preliminary data collection primarily entails gathering historical records and facility design documents. For RMC, it consisted of collecting energy bills, weather data, building plans, and mechanical equipment specifications. RMC's energy bills for electricity, natural gas, and water usage show the overall building energy consumption and cost. Energy bills were gathered from two subsequent years in order to see any variation. The energy bills at RMC are very complicated. Multiple rate schedules and adjustment factors are applied facility wide. Care is taken in order to properly classify and organize each energy bill. Historical and current climate data were collected in order to establish general weather patterns for the geographical location of RMC. An area's climate greatly influences HVAC energy consumption. Building plans and mechanical equipment specifications show building design parameters. RMC's building plans provide the existing facility layout and mechanical equipment schedules for each energy cost center. Specifications for the energy consuming equipment located in each energy cost center were found and organized into tables. All preliminary data were collected, organized, and combined for analysis to provide a general impression of the building's design vs. actual energy use. This information also aided in the planning for step four, the building survey. Step four of the RMC audit is to perform a building survey. The building survey includes a walkthrough of all energy cost centers at RMC. Each energy cost center contains systems from which to gather operational data. Step four is the most important and time consuming undertaking of the entire audit. The accuracy of the data collected by the building survey influences all subsequent analysis.

5.3 Instrumentation

In order to collect operational data from mechanical systems, proper measurement instrumentation must be used. The measurement instrumentation used for the RMC audit consists of portable hand held devices. These devices are used to obtain direct measurements of operational conditions primarily from HVAC mechanical and electrical systems and in occupied zones of RMC. HVAC parameters such as air temperature, water temperature, relative humidity, air flow, duct static pressure, water pressure, carbon dioxide concentration, and combustion products are measured. Most all temperature measurements are taken by a standard K-type thermocouple and thermocouple reader. Relative humidity is measured by a psychrometer. Air flow is directly measured by a hot-wire anemometer, and by a pitot tube connected to a monometer. A monometer is also used to measure static pressure and pressure differential across two zones. Carbon dioxide concentration is measured with an air quality monitor. Boiler combustion parameters are measured with a combustion analyzer. High pressures are measured with a standard electronic pressure transducer. All specifications and brands of this equipment can be found in Appendix A. Control system sensors and instrumentation is also used to measure building environmental conditions.

5.4 Data Collection Strategies

Data collection strategies were developed and implemented in order to ensure smooth and organized collection of data that could be effectively used for analysis. Different strategies were applied to each energy cost center depending on its systems. In order to properly perform engineering analysis on each system contained in the energy cost centers, certain questions must be answered and certain parameters must be defined. Questions like "What systems primarily influence current operational conditions?", "What kind of analysis to perform?", and "What information is needed to perform the required analysis?" must be addressed. The answers point to the correct data to collect. Data collection at RMC was centered on defining the missing information needed to complete a thorough analysis and establish current operating conditions. Later in follow up data collection surveys, data were collected to test analysis results or verify new conditions. Still even in follow up surveys, data were collected to provide the information needed to do further analysis and establish new operating conditions. Consistency was emphasized during data collection. Data collection procedures for each system were standardized in an effort to minimize error due to how the data were acquired. Measurements were taken with multiple instruments to verify information and minimize error due to instrumentation. Any variation for any reason was noted. Sometimes areas could not be accessed to collect accurate data. Oftentimes this occurred with the HVAC system. Proper test ports were not in place or the system was closed off by walls or other obstructions. Decisions were made about how to deal with these situations as they occurred. Any abnormalities such as odd sounds or vibrations with any equipment were also noted during the building survey data collection phase. Data sheets were tailor made for each system so that data could be recorded in an orderly manner. Most of the data were organized electronically with spread sheet software. Since RMC's climate zone includes seasonal weather changes, seasonal data must be collected in order to properly perform engineering analysis. Due to time constraints and importance factor some energy cost centers received more attention than others. The sections below each contain descriptions of the methods used for each energy cost center.

5.5 Building Envelope Data Collection.

The roofing system was the only section of the building envelope that was considered for the RMC energy audit. First, preliminary data from the plans were gathered to define the system design and materials. Things like insulation thermal resistance coefficients and top layer emissivity were found in the preliminary data. Once this information was gathered, the missing information that remained was roof surface temperature. In order to provide an accurate description of the energy lost or gained during each season, roof surface temperature was measured on selected hot, cold, and mild temperature days. Roof surface temperature was measured with a contact k type thermocouple and recorded for each selected day.

5.6 HVAC Data Collection

HVAC data collection during the building survey was a multifaceted laborious process. Operational data were collected on multiple HVAC systems and air conditioned spaces throughout RMC. The main focus was to collect operational data at each HVAC system and air conditioned space that could be used to verify design conditions and expose excess energy consumption. Specific data collection procedures were followed for each system in order to gather accurate, useable data. Seasonal weather changes for RMC's climate zone made it necessary to collect HVAC data multiple times on selected days. In order for the data to show an accurate representation of the whole system, data were collected on multiple days of similar weather conditions for each season. Peak cooling conditions would occur during the hottest summer months as peak heating occurs during the colder months. Off peak times fill in the gaps. HVAC operational data for multiple days on both peak and off peak conditions were collected in order to represent how each system responded to each weather condition.

The HVAC generation systems at RMC account for a large proportion of energy expenditure. In order to produce an accurate energy model, high quality data must be recorded from these systems. As discussed earlier, the HVAC generation systems at RMC consist of chillers, cooling towers, fire tube boilers, and hot water generators. Operational data were taken at each generation system. The main goal of generation system analysis is to establish overall energy efficiency for each system. Therefore, the main goal for generation system data collection was to gather the missing operational information needed to formulate and complete engineering analysis.

The standard calculation of efficiency for chiller systems found at RMC is coefficient of performance (COP). COP can be calculated directly or indirectly. Both methods require operational system data. At RMC, data were collected at each chiller in order to be able to calculate a COP. The data collection procedure for RMC's water cooled chillers can be found in appendix B. The procedure found in appendix B includes measuring evaporator and condenser water inlet and exit temperatures and pressures, refrigerant temperatures and pressures, and current loads. Data collection procedures were written for all HVAC equipment in a similar manner. Care was taken to directly measure parameters consistently. For example, water was drained from the pipes in order to directly measure water temperature as opposed to measuring the temperature of the skin of the pipe. All of these parameters were measured at peak and off peak weather conditions. This information also fills in the missing parameters in other equations needed to perform a complete analysis of the chilled water generation systems at RMC.

Cooling tower effectiveness is the standard efficiency measurement of a cooling tower system. Cooling tower inlet and exit temperatures and pressures, outdoor dry bulb and wet bulb temperature, and water flow rate was measured in order to be able to calculate cooling tower effectiveness. Cooling tower operational information was gathered at peak and off peak seasons. Fan speed and set-points were also recorded to aid in later analysis.

A commonly found efficiency measurement for steam boiler systems is combustion efficiency. The information required to find combustion efficiency can be collected from the combustion products exiting the exhaust stack and the entering combustion air temperature. A combustion gas analyzer was used to measure stack gas temperature, oxygen levels, and carbon monoxide levels. Other information like steam pressures, feed water temperatures, and water chemical composition were measured with the correct instrumentation for use in other analysis. Data were collected on the boiler support equipment. Condensate tank water temperatures and water chemical levels were measured and recorded. Entering condensate temperature was measured as well as entering make up water temperature. The chemical composition of the make-up water was measured in order to look at its water chemistry and help define how much make up water was being used. Data were taken on the boiler stack gas economizer in order to analyze its energy recovery. Entering and exiting economizer feed water and stack gas temperatures were measured to aid in that process. Information was collected on the hot water generators in a similar manner. Entering and exiting temperatures were measured along with pressures to determine flow rates.

Several procedures were developed to collect data on RMC’s HVAC transport systems. Transport systems move large amounts of energy around the facility and proper data from these systems is needed to know how efficiently the transportation occurs. Operational data were collected on the hot water, chilled water, conditioned air, and ventilation systems that make up the large distribution network throughout RMC.

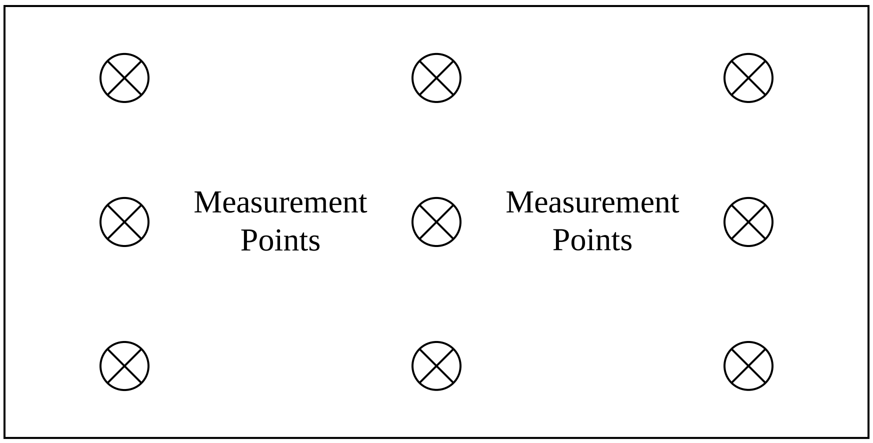
Centrifugal pumps are the prime movers for the chilled and hot water distribution system. Discharge and suction pressures were recorded for each pump so that each pump could be evaluated. Thermal insulation placement and valve and damper positions were also recorded to better understand where thermal energy is lost and gained and how flow is balanced throughout the transport networks. The most intricate data collection procedures were for air handlers. There are many points of interest at each air handler that require data collection. When evaluating an air handler, heating and cooling coil load and air flow are the main areas of emphasis. Data collection centered on those areas to provide the necessary information to allow for later evaluation. One important factor to note is that air moving through air handlers and ductwork at RMC is mostly under turbulent flow conditions. Turbulent flow is difficult to measure accurately unless it is fully developed. Air flow measurement was a very tedious and demanding part of the data collection phase of the RMC energy audit.

Data were collected at all air handlers at RMC. In general, twenty seven data points were measured at thirty one air handlers at RMC. A total of 837 data points were measured multiple times during peak and off peak seasons. A general data collection procedure for each air handler can be found in appendix B. The data points taken at each air handler are shown in Table 5.1.

Table 5.1: Air Handler Data Collection Points

Air Handler Data Points						
Air Flow	**Air Temperature**	**Relative Humidity**	**Static Pressure**	**Water Temperature**	**Fan**	**Water Pressure**
Outside Air Flow	Outside Air Temp	Outside Air RH	Return SP	Chilled Water Supply	Fan Speed	CW Supply Press.
Return Air Flow	Return Air Temp	Return Air RH	SP across Filters	Chilled Water Return	Guide Vane Position	CW Return Press.
Supply Air Flow	Mixed Air Temp	Mixed Air RH	SP across Fan	Hot Water Supply		HW Supply Press.
	Preheat Air Temp	Preheat Air RH	Discharge SP	Hot Water Return		HW Return Press.
	Supply Air Temp	Supply Air RH				

Air flow was measured with two instruments, a hot wire anemometer and a pitot tube connected to a monometer. A hot wire anemometer directly measures air velocity by passing a voltage through a small wire. As air flows over the wire, the cooling effect changes the voltage which corresponds to an air velocity. A pitot tube, monometer setup works by placing a tube into the flow to measure total pressure and a tube normal to the flow to measure static pressure. The monometer then senses the difference between total pressure and static pressure giving an effective velocity pressure, which can be converted into an air velocity. As stated earlier, most air flow in duct work at RMC falls in the turbulent regime. The most accurate measurements can be taken when the turbulent flow is fully developed. Fully developed flow occurs in long straight sections of duct. Air flow was measured in the straightest sections of duct connected to each air handler. Air flow velocity can vary within a duct depending on the geometry and orientation of the ductwork. Typically in horizontal duct, air velocity is less at the edges due to friction and greater in the center. Bends in the duct can force air to one side or cause eddies. To get the most accurate results, a traverse method was utilized to collect air flow information in multiple locations for each duct. An example of an air flow measurement traverse is shown in Figure 5.1.



Cross-Sectional View
of Square Duct

Figure 5.1: Duct Measurement Traverse

Holes were drilled into the ductwork to access these measurement points. A probe from the hot wire anemometer was inserted into the duct normal to the flow placing the sensing head into the flow at each measurement location. Multiple readings were taken at each point. The outliers were discarded and the remaining values were averaged to give an overall average air velocity. Pitot tube results were averaged in a similar fashion. Outside air velocity, was measured in the outside air duct and return velocity was measured in the return duct. They were then added together to get the total incoming air flow. Oftentimes, air handlers had multiple return ducts. Flow velocity was measured in each individual return duct and added together to get a total return air flow. Supply air flow velocity was also measured in the outgoing supply air ducts in the same fashion. Duct dimensions were also measured and recorded in order to later calculate the volumetric flow rate.

Temperature measurements were all taken with K-type thermocouples. As shown previously in the HVAC overview, each air handler has multiple zones within to condition the air flowing through. Air temperature data were recorded at each zone. First, the incoming outside air temperature was measured by inserting the temperature probe into the outside air duct. Air temperature can vary within a duct or zone much like air flow. To compensate, a traverse method was used again to take multiple temperature readings within each duct. The values were averaged to obtain an overall average outside air temperature. Return air temperature was measured in the same fashion. Mixed air temperature was measured within the mixing chamber as far downstream as possible to get the best mixed air results. Air handler access doors allowed access to the mixing chamber. Once inside, the mixed air temperature was measured and recorded at multiple points on the traverse. The values were averaged to produce an overall average mixed air temperature. Preheat temperature measurements were done downstream of the preheat coil in the same manner. The supply temperature was measured downstream of the cooling and heating coil. Data was taken at multiple points within the zone and averaged.

Relative humidity was measured with a hand held psychrometer. Like temperature, it was measured in each zone of the air handler. Due to its size, it was often impossible to get the psychrometer into the outside and return air ducts. To counteract that problem, outside air relative humidity was measured at the location of the outside air intake. Return air relative humidity was measured in the zones that the return air was coming from. Some larger return ducts could be accessed and return air was directly measured in the return air duct. Mixed air relative humidity was taken multiple locations within the mixing chamber and averaged. Preheat was taken in the same manner downstream of the preheat coil. Supply relative humidity was measured and averaged downstream of the cooling coil.

Static pressure was measured with a pitot tube and monometer setup. Access holes were drilled in each zone of the air handler to measure static pressure. Static pressure was first measured at the mixing chamber. The static pressure difference was measured across the air filters both before and after the fan by placing one tube on the high pressure side and one tube on the low pressure side after the filter. In a single fan setup, static pressure before the fan is negative and positive after the fan. Static pressure difference was then measured across the heating and cooling coils to obtain pressure drop information. Next, the static pressure difference across the fan was measured. Lastly, discharge static pressure was measured in the supply duct.

Water temperatures were taken at both preheat and cooling coils with a K-type thermocouple. Chilled water supply and return temperature were measured by draining chilled water from a test port into a container and directly measuring its temperature. Hot water supply and return temperature were measured in the same manner. Chilled and hot water pressure drops across each coil were measured by inserting a digital pressure gauge into water pressure test ports on both the supply and return sides of the coil. Fan speed was recorded on air handler units with VFD fans by reading the fan status reports found at the fan control module. Inlet guide vane positions were read directly from the BMS GUI.

Air terminal box data collection was also a large time consuming part of the building survey. Each air terminal box in the main hospital was visited. The main data points collected at each air terminal box was air flow and air temperature before and after the reheat coil. Both constant volume and variable volume air terminal boxes were visited. Data collection methods similar to the air handler air flow and temperature measurement procedures were used. Temperature and air flow data was collected at each terminal box to expose how air temperatures were affected during transport and also to show if proper air flow was present.

Data were collected for the larger end use systems like the plate and frame heat exchanger, and hot water convertors. Water temperature and pressure drop were recorded on both the hot and cold sides of the plate and frame to be able to perform analysis to show if the heat exchanger was within design operating conditions. For the hot water convertors, steam entering temperature was recorded along with the condensate temperature exiting. Also hot water entering and exiting temperatures and pressures were recorded. This information was used to determine the heat transfer characteristics and efficiency of each convertor.

5.7 Occupied Zone Data

The purpose of the HVAC system is to provide acceptable environmental conditions in the occupied zones. To verify that the HVAC system was fulfilling its duties, data was collected throughout RMC in the occupied zones of all buildings. Zone temperature and relative humidity were measured and recorded in a room by room walk through. Carbon dioxide levels were also recorded. Indoor air quality and proper ventilation is a major concern for densely populated healthcare facilities. Carbon dioxide levels were measured by a handheld indoor air quality monitor in the room by room walk through. Hospital operating rooms are required to be held at a positive room pressure in relation to its surrounding areas. This was verified at each operating room. Multiple walk throughs were carried out in order to verify whether or not the HVAC system was consistently maintaining acceptable indoor environmental conditions during significant changes in outdoor environmental conditions.

5.8 Building Management System Data Collection

The Building Management System (BMS) at RMC automatically controls the operation of the HVAC system. Many sensors located throughout RMC feedback equipment conditions and environmental information to the BMS. The information feedback from those sensors dictates how the control system responds. The information measured by the sensors can be read on the graphical user interface. In order for the BMS to maintain proper control, this information must be accurate. To verify its accuracy, it was recorded simultaneously with the data being collected at the mechanical equipment by the BMS controls. The bulk of this information comes from the chiller plant and air handlers. All the manually collected data at each system was cross referenced with the sensor that was measuring the same data to be sent to the BMS. Typically, each air handler was outfitted with sensors to measure data at the same points collected by personnel during the building survey. For example, AHU-OR3 is outfitted with the following sensors:

1. Outside Air Temperature
2. Outside Air Relative Humidity
3. Return Air Temperature
4. Return Air Relative Humidity
5. Preheat Temperature
6. Supply Temperature
7. Outside Air Flow
8. Supply Air Flow
9. Static Pressure Monitors Across Filters

10. Discharge Static Pressure

The data collected manually were measured from the same location as the sensor to get the most realistic reading. Over 500 points were recorded from the BMS sensors. The BMS controls the modulation of chilled water and hot water valves along with air handler dampers and fans. On the Graphic User Interface (GUI), a reading is given to indicate the position of control valves and dampers. The readings were cross referenced with the actual position of the valves and dampers in the field to verify their operation. If discrepancies were found, they were recorded.

5.9 Electrical System Data Collection

Electrical system data collection consisted mainly of collecting lighting system data. First, the building plan lighting equipment schedules were gathered. From the lighting schedules, the lighting fixture type, total quantity, and location were determined. The information obtained from the lighting schedules was then verified with a complete walk-through building lighting survey. Data on lighting operational hours for each RMC facility was also obtained during the lighting survey.

5.10 Initial Data Analysis

Initial data analysis was performed to help verify the validity of air flow data recorded from air handlers and air terminal boxes. Due to the difficulty associated with measuring turbulent air flow, data can be inaccurate and misleading if left unchecked. A way to establish air flow data accuracy is to compare results to design specifications. The instrumentation used to measure air flow at RMC measured air flow velocity. The design specification for air flow is represented by a volumetric flow rate. In order to compare the actual data to design parameters, the actual data was converted from flow velocity to volumetric flow rate. Converting the velocity data to volumetric flow rate offers a better idea about how much air is moving through an air handler. Volumetric flow rate is computed with the simple formula shown below:

$$\dot{V} = V_A * A_C \qquad (1.)$$

Where: $\dot{V}$ is the volumetric flow rate in $\frac{ft^3}{Min}$.

V_A is the air flow velocity in $\frac{ft}{Min}$.

A_C is the cross sectional area of the duct.

Once volumetric flow rates were computed from actual velocity data from all air streams entering and leaving the air handler, the measured flow rates could be compared to design values to gauge their accuracy. Another indication of valid air flow data is the difference between inflowing air volume and out flowing air volume. Incoming air volume cannot greatly exceed out flowing air volume or vice versa. Percent difference between inflowing and out flowing air was calculated with the formula below.

$$\%Difference = \frac{IC - OG}{\left(\frac{IC + OG}{2}\right)} * 100 \qquad (2.)$$

Where: IC is the Incoming Volumetric Flow rate.

OG is the Outgoing Volumetric Flow Rate.

The acceptable percent difference between volumetric flow rates was set at twenty percent. Any data that exceeded a twenty percent difference between incoming and outgoing volumetric flow rate was thrown out and retaken.

5.11 Data Organization

The large amount of data at RMC was initially recorded on data sheets tailor made for each system. This data sheet format allows the auditor to see the general location to record data when collecting it and to visualize where and how the data were taken later on when reviewing the data for analysis. The data sheet in the figure below is an example of a data sheet created for air handler AHU-OR3.

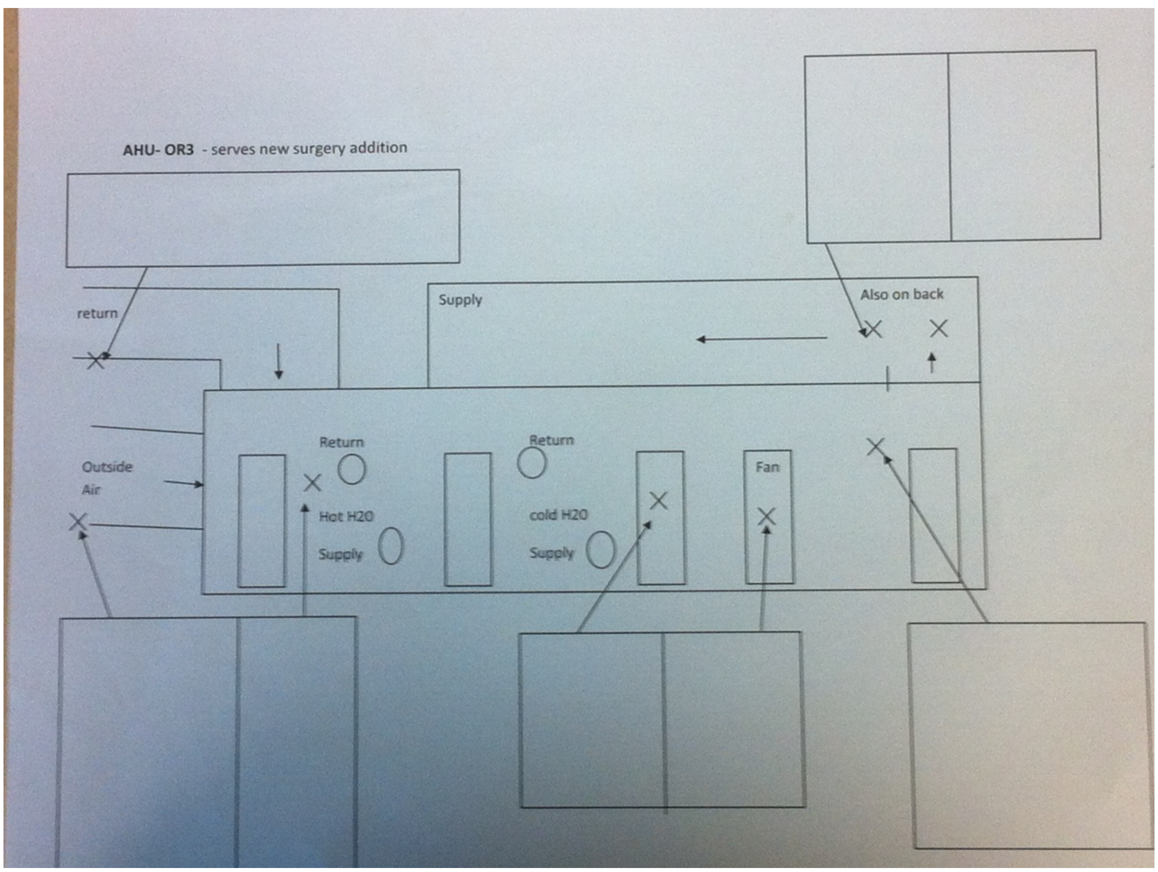


Figure 5.2: AHU Data Sheet

The data sheet shows above shows the system configuration and the general location where the data were recorded. It has space for the auditor to pencil in data as it is recorded. After the data were recorded on the data sheets, the information was transferred to an electronic format, usually spreadsheet software, to better organize and store the data. Spreadsheet software like Microsoft Excel is a powerful data organization and analysis tool. It provided the valuable capability of being able to quickly perform the initial analysis discussed earlier for large amounts of data. Data were organized by energy cost center and then further broken down into the individual systems within the energy cost center. For example, data form the air handler systems were organized under the HVAC energy cost center and then further broken down into sub categories like Main Hospital Transport Systems and Conditioned Air and Ventilation systems. Any abnormalities or difficulties were noted in the data sheets for each system.

5.12 Sources of Error

Operational data was taken on numerous energy consuming systems throughout RMC. Although most of the data recorded were deemed acceptable, some error still remained. Several factors contributed to error in the data collected at RMC. The first source to discuss is instrumentation. While the instrumentation used was calibrated, accurate, and repeatable, there is still some error. For example, the hot wire anemometer used to measure flow velocity has a very sensitive probe that is susceptible to dirt and other air borne contamination. A small amount of contamination can skew readings. Another source of error for the hot wire anemometer is operator error. The anemometer probe must be correctly oriented in the flow to get accurate readings. Ductwork that is hard to access can make it difficult for the operator to keep the instrument correctly oriented. Another contributing factor is the nature of what is being measured by the instrumentation. Most of the flow measurements taken by the anemometer at RMC were turbulent flow. Turbulent flow creates eddies within the flow that spin at a different velocity than the bulk motion. The anemometer is sensitive to these eddies; which causes error in the air velocity measurement. The instrument condition, operator, and nature of the sampled medium can all three combine to generate significant error. Measurements were taken within acceptable limits; however they were not one hundred percent accurate. Human error also contributed in other areas other than air flow measurement. Even with emphasis placed on measurement consistency, variance still occurred in how measurements were taken. On occasion, test ports were not present or the ideal location could not be accessed, so a less desirable measurement had to be taken.

When recording data on a large HVAC system, it is best when all data is taken under the same weather conditions so that the system's response for that particular condition is reflected by the data. Recording data on the whole system under the same weather conditions was very difficult to do with the limited manpower and equipment used for the RMC audit. Due to size of the HVAC system at RMC, data had to be taken on different parts of the system on different days. In performing the RMC building survey, data were recorded on multiple days with very similar weather conditions, but the conditions varied slightly from day to day. This variation, while slight, still produced some error. RMC's building management system lacks data trending capability, which is very useful in continuously being able to record and monitor data on the HVAC system.

To compensate, data were collected as often as possible, but it was impossible to monitor the entire HVAC system continuously under all conditions.

5.13 Data Conclusions

Both preliminary and building survey data collection was a tedious, time consuming process. When collecting data for an energy audit, it is important to collect all the data needed to perform valuable engineering analysis. Data collection for the RMC energy audit was based on that concept. The focal point of data collection at RMC was to record data that would be needed to perform engineering analysis and later to verify applied recommendations. Some aspects of data collection were straightforward. Preliminary data collection primarily consisted of gathering documented specifications and information from plans and energy bills. Building survey data collection was more laborious. Certain aspects of the building survey data collection were straightforward, while other parts where more complex. Temperature and relative humidity can be measured with great accuracy and simplicity. On the other hand, turbulent air flow is very difficult to measure accurately and requires some innovation to produce quality data. Functional data sheets tailor made for individual systems like the one shown earlier are important for data organization. They allow the auditor to have a good mental picture of where and how the data were collected. While some data had significant sources of error and could not be trusted, most of the data collected at RMC were instrumental in characterizing each energy cost center and exposing opportunities for improvement. The data showed valuable insights into where systems were under and over performing. The data pointed to the overall condition of RMC's energy cost centers and provided a sufficient foundation from which to build a baseline energy profile and perform engineering analysis. In conclusion, the correct data must be successfully recorded and organized with great care in order to have a successful audit. All raw data can be found in Appendix C.

Chapter 6: Energy Cost Center Analysis

6.1 Main Hospital Building Envelope Analysis

As stated earlier the building envelope analysis will be limited to roofing. Other areas were found to be adequate. Also, building envelope changes can be very costly and most changes would fall outside of RMC's investment return standards. The roof of the main hospital covers an area of 46,900 square feet. It consists mostly of a built up roof system. Built up roofing systems consists of multiple layers of insulation, ply sheets, and asphalt. The top layer is generally a reflective coating or membrane that sometimes is covered with gravel or some other type of ballast. The main hospital roof is partially covered with gravel ballast. Large areas of the main hospital roof do not have gravel ballast. Those areas have a black top membrane. A disadvantage of a black top membrane is that high roof surface temperatures are created due the dark surface color which absorbs thermal radiation. Data collected from RMC shows that the roof surface temperature is typically 50 to 70 degrees Fahrenheit above ambient air temperature. On hot summer days this makes the roof surface temperature very high. This heat is conducted through the roof and into the building increasing the HVAC load. The United States Department of Energy roof heat load calculator was used to estimate the solar heat gain through the main hospital roof with the black top membrane. The total solar load for the Alexander City area is estimated at 1470.6 BTU/ ft^2 per day or 68,971,140 BTU per day. The current black top roof on the main hospital covers an area of 28,859 square feet.

Table 6.1 shows the results obtained from the DOE roof heat load calculator for the current flat, black membrane built up roof based on a solar reflectance of five percent and an infrared emittance of ninety percent.

Equation (3.) for roof cooling energy usage and equation (4.) for heating energy usage are shown below:

$$\frac{CL(\frac{\frac{BTU}{ft^2}}{yr}) * RA(ft^2) * CE(\frac{kW}{Ton}) * 24(\frac{hr}{day}) * CDD(\frac{days}{yr})}{8760(\frac{hr}{yr}) * 12000(\frac{BTU}{\frac{hr}{Ton}})} = \frac{Kwh}{yr} \qquad (3.)$$

Where: CL is Roof Cooling Load.

RA is Roof Area in square feet.

CDD is Cooling Degree Days in a year.

$$\frac{HL \frac{\frac{BTU}{ft^2}}{yr} * RA(ft^2) * 24(\frac{hr}{day}) * HDD(\frac{days}{yr})}{8760(\frac{hr}{yr}) * BE} = \frac{BTU}{yr} \qquad (4.)$$

Where: HL is Roof Heating Load.

RA is Roof Area.

BE is Boiler Efficiency (84%).

HDD is Heating Degree Days in a year.

Table 6.1: Main Hospital Black Roof Analysis Results

Black Roof Analysis Results for Main Hospital - 28859 ft^2	
Solar Load BTU/ft^2/day	1470.6
Black Roof Cooling Load BTU/ft^2/year	5203
Black Roof Heating Load BTU/ft^2/year	3760
Black Roof Cooling Energy Usage Kwh/yr	50496
Black Roof Heating Energy Usage BTU/yr	741802428
Black Roof Cooling Demand in Kw	1
Black Roof Heating Demand in BTU	353913
Black Roof Cooling Cost in $/yr ($.065 per Kwh)	3282
Black Roof Heating Cost in $/yr ($6 per million BTU)	4450
Total Black Roof Cost in $/yr	7732

As can be seen from the results, the current main hospital roofing type contributes relatively significantly to HVAC costs. Improvements could possibly be made to this energy cost center.

6.2 POB Building Envelope Analysis

The building envelope analysis for the POB at RMC was also limited to roofing for the same reasons as before. POB three is equipped with a flat, black membrane, built up roofing system. It covers an area of 11,091 square feet. This area will be analyzed with the same method used before for the main hospital. Equations (3.) and (4.) from above where used again to determine roof related energy usage. Table 6.2 shows the results for the black roof found on top of POB 3.

Table 6.2: POB Black Roof Analysis Results

Black Roof Analysis Results for POB - 11091 ft^2	
Solar Load BTU/ft^2/day	1470.6
Black Roof Cooling Load BTU/ft^2/year	5203
Black Roof Heating Load BTU/ft^2/year	3760
Black Roof Cooling Energy Usage Kwh/yr	19404
Black Roof Heating Energy Usage BTU/yr	285087173
Black Roof Cooling Demand in Kw	0.384
Black Roof Heating Demand in BTU	136014
Black Roof Cooling Cost in $/yr ($.065 per Kwh)	1261
Black Roof Heating Cost in $/yr ($6 per million BTU)	1710
Total Black Roof Cost in $/yr	2971

6.3 Main Hospital Miscellaneous Electrical Usage

Miscellaneous electrical equipment encompasses all electricity using equipment contained in the facility not including lighting. This covers office equipment, medical examination equipment, cooking equipment, electrical motors used to drive elevators, air compressors, and self-contained refrigeration equipment. Due to the size of RMC, it would be very difficult and time consuming to individually account for and quantify each piece of electrical equipment. For this analysis, it was all lumped together. EQUEST building energy simulation software was the primary tool used to quantify miscellaneous electrical use. Basically, energy use from other areas was analyzed and quantified using EQUEST and other methods. The energy use from the other areas was then subtracted from the total building energy use obtained from RMC electricity bills. The remainder was considered miscellaneous electrical usage. EQUEST is equipped with built in miscellaneous equipment profiles based on time of day and building population estimates. EQUEST also allows for custom profiles to be uploaded. Information was gathered about miscellaneous electrical usage from other audits on similar facilities. Electrical usage and demand data were gathered on some larger RMC electrical equipment such as large electrical motors. All of this information was used to create a miscellaneous electrical usage profile for RMC. The miscellaneous usage was quantified and entered into EQUEST on wattage per square foot basis for each area. For example, electric medical equipment was simulated in EQUEST by allocating 1 watt per square foot in medical and clinical care areas for the electric equipment. The usage profile created for all areas of the facility and uploaded into EQUEST and simulated over a year's time. The results of the EQUEST simulation for the years 2008-2010 can be found in table 6.3 below.

Table 6.3: Main Hospital Misc. Electrical Equipment Analysis Results

Miscellaneous Electrical Equipment Usage/Demand for Main Hospital in KWH / KW			
	2008	2009	2010
Jan	134500	131300	128200
Feb	121500	121500	121500
Mar	134500	137700	140800
Apr	135400	135400	135300
May	137500	131300	131400
Jun	129200	135400	135400
Jul	137600	137500	134400
Aug	137500	134500	137700
Sep	129200	132200	132200
Oct	137600	134400	131300
Nov	125900	126000	129000
Dec	134500	137600	134500
Total Usage	1594900	1594800	1591700
Peak Demand	280	280	280
Watt per ft^2	1.8	1.8	1.8
Total Cost -$	120730.5	120672	120460.5

The table above shows that miscellaneous electrical usage and demand remains consistent over the span of three years. Collectively, all the miscellaneous equipment in the main hospital calls for a peak power demand of 280 kilowatts or about 1.8 watts per square foot. Miscellaneous electrical usage accounts for a significant power expense at the RMC main hospital facility.

6.4 Main Hospital Lighting Analysis

Multiple methods were used to analyze the lighting contained in the RMC main hospital. Data gathered in a complete lighting audit was analyzed and used to make estimations of the main hospital's lighting related electricity usage and cost. The lighting audit data was also fed into EQUEST building energy simulation software as another method to quantify the main hospitals lighting energy usage and cost. The lighting audit showed that there are a total of 5,165 lamps in the main hospital with a power demand of 160,107 watts. Two assumptions were made about the usage of lighting at the main hospital. Eighty percent of the lamps are positioned in areas of the hospital that are occupied 24 hours a day. For this analysis, the main hospital lighting was considered operational for the full twenty four hours in a day for eighty percent of the hospital. Secondly, the remaining twenty percent was considered operational for twelve hours a day. Lighting has two energy costs associated with it. First, there is a cost to power the lights; second there is a cost to remove the heat generated by the lighting. The CLF method or cooling load factor method was used to calculate the lighting cooling load. The CLF method is a hand calculation method which utilizes constants from tabulated data gathered by ASHRAE over several case studies. Equation (5.) was used to calculate the electricity consumed by the lighting. Equation (6.) was used to estimate the lighting cooling load. Equation (7.) is used to estimate the energy consumed in removing the heat generated by the lighting.

$$\frac{LP * BF * hr}{1000} = \frac{Kwh}{day} \qquad (5.)$$

Where: LP is the Total Lighting Power in watts.

BF is the lighting fixture ballast factor.

hr is the number of operational hours in a day.

$$\frac{LP * 3.41 * BF * CLDon * hrs(on) + LP * 3.41 * BF * CLD(off) * hrs(off)}{Totalhours} = CoolingLoad\frac{BTU}{hr} \qquad (6.)$$

Where: LP is the Total Lighting Power in Watts.

BF is the Ballast Factor.

CLDon is the operational cooling load factor. (0.816 to 1)

CLD(off) is the residual cooling load factor used to estimate the delayed radiation related cooling load after the lighting is turned off. (0.1321)

hrs(on) is the number of hours in operation.

Hrs(off) is the number of hours not in operation.

$$\frac{CL * CE * 24}{12000} = CoolingLoadUsage\frac{Kwh}{day} \qquad (7.)$$

Where: CL is Cooling Load in BTU/hr.

CE is Chiller Efficiency in KW/Ton.

Table 6.4 shows the results of the manual main hospital lighting analysis. Table 6.4 contains a breakdown of the cost to power and to cool the lights. Roughly 83 percent of the energy is used to power the lights. Cooling energy accounts for 17 percent of the total lighting energy usage. An electricity usage rate of $0.065 per kWh and a demand rate of $5.25 per KW were used to find the electricity usage and demand cost.

Table 6.4 Main Hospital lighting Audit Results

Main Hospital Lighting Audit Results	
Lamps	5165
Wattage	160107
Total Cooling Load	411629
Total Cooling Usage/day	576
Total Cooling Usage /Year	210342
Total Power Usage/day	2875
Total Power Usage/year	1049339
Combined Total/ day	3451
Combined Total/ Year	1259681
Demand Lighting	159
Demand Total	186
Usage Cost to Cool	$ 13,672.23
usage Cost to Power	$ 68,207.04
Total Usage Cost	$ 81,879.27
Total Demand Cost	$ 11,718.00
Total Cost/year	$ 93,597.27

EQUEST was also used to simulate hospital lighting usage. EQUEST allows the lighting to be dynamically simulated over an entire year. Data was entered into EQUEST on wattage per square foot per zone basis. For example, medical exam space was assigned a value of 1.5 watts per square foot. Each zone was assigned different values based on data gathered during the audit. Operational hours were specified in EQUEST for each area in the main hospital. EQUEST also has built in occupancy profiles tuned to medical facilities to simulate human occupancy over a given period of time. EQUEST also accounts for the cooling load generated by the lighting. The EQUEST simulation resulted in a total lighting related energy usage of 1,235,900 kilowatt-hours. The monthly lighting related electricity usage results given by EQUEST are shown in Figure 6.1.

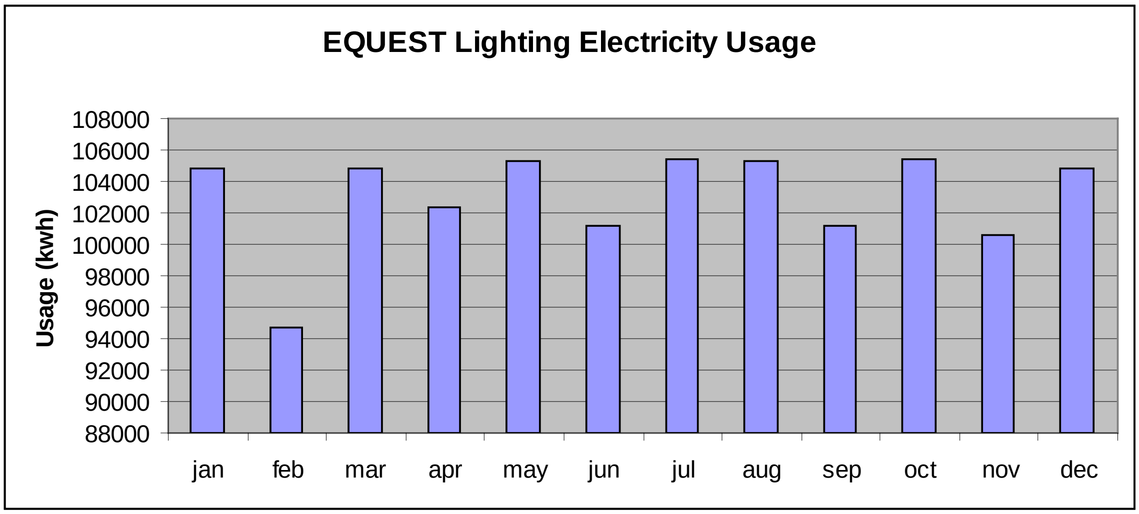


Figure 6.1: EQUEST Lighting Analysis Results

The manual calculation resulted in a total lighting related energy usage of 1,259,681 kilowatt-hours. That results in a percent difference between the two methods of 1.8%. Obtaining similar results using both methods further confirms that lighting energy is correctly estimated

6.5 POB Miscellaneous Energy Use

Miscellaneous Electricity Usage was simulated over the course of a year with EQUEST exactly as it was with the main hospital. All miscellaneous equipment was lumped together to get a total miscellaneous usage. The results of the simulation are shown in Table 6.5.

Table 6.5: POB Misc. Electrical Usage Results

Miscellaneous Electrical Equipment Usage/Demand for POBs in KWH / KW			
	2008	2009	2010
Jan	69400	67200	64600
Feb	62900	62900	62900
Mar	69800	72000	74600
Apr	71600	71600	71600
May	72000	67200	67200
Jun	66800	71600	71600
Jul	72000	72000	69800
Aug	72400	69800	72000
Sep	66400	69000	69000
Oct	72000	69800	67200
Nov	64200	63800	66400
Dec	69400	72000	69400
Total Usage	828900	828900	826300
Peak Demand	230	230	230
Watt per ft^2	1.8	1.8	1.8

The POB also has a miscellaneous energy demand of 1.8 watts per square foot similar to the main hospital. The POB's total miscellaneous energy use is a significant energy cost and is relatively constant year to year.

6.6 POB Lighting Analysis

The POB lighting analysis was also conducted in the same fashion as the main hospital. Data gathered from the POB lighting audit was fed into the EQUEST model and also used to manually estimate lighting related energy costs. Equations (5.), (6.), and (7.) were used in the manual method. All three buildings were lumped together as one for both the manual and EQUEST simulations. The same electricity billing rates were used in this analysis. The POB's are not occupied all hours of the day like the main hospital. A ten hour day was assumed for all the lighting in the POB. It was also assumed that most of the lighting in the POB is turned off during unoccupied times. The POBs contain a total of 3,515 lamps which produce a demand of 120 kilowatts. Table 6.6 shows the results of the manual estimation method.

Table 6.6: POB Lighting Analysis Results

Results of POB Lighting Audit		
Lamps		3515
Total Wattage		120396
Cooling Load (BTU/hr)		154101
Cooling Usage (kwh)/day		462
Cooling Demand (kw)		46
Lighting Usage (kwh/day)		1204
Lighting Usage (kwh/ yr)		439445
Lighting Demand (kw)		120
Total Usage (kwh/day)		1666
Total/yr		608186
Total Demand		167
Usage Cost /day	$	108.31
Demand Cost/month	$	874.79
Total Cost/day	$	137.47
Total Cost usage /yr	$	39,532.10
Total Cost	$	50,053.10

The EQUEST simulation resulted in a total yearly energy consumption of 615,600 kilowatt-hours of electricity. The EQUEST monthly lighting use breakdown is shown in Figure 6.2.

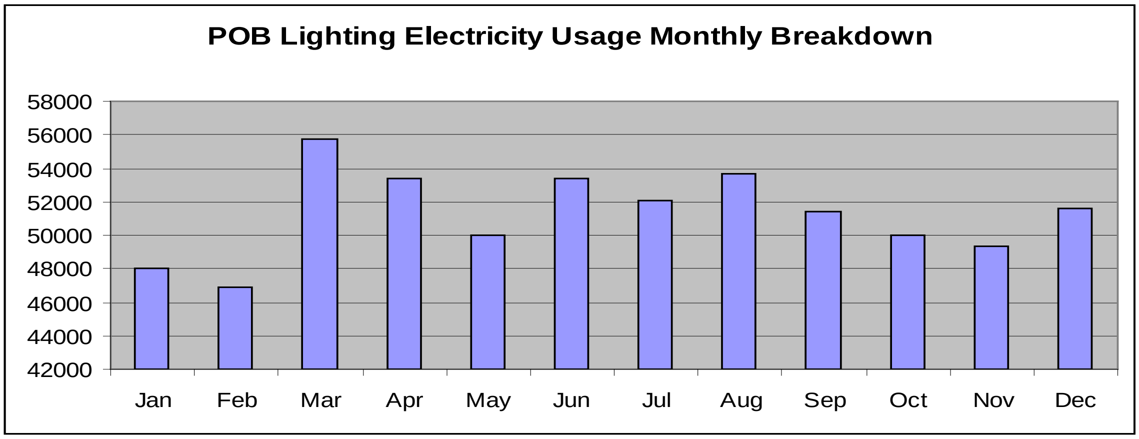


Figure 6.2: POB EQUEST Lighting Results

The manual method estimated a total lighting related energy usage of 608,186 kilowatt-hours per year. There is a 1.2 % difference between the two methods. The POB uses a bit more energy to handle the lighting related cooling load than the main hospital because of the lower efficiency of the air cooled chillers that serve the POBs.

6.7 Cancer Center Miscellaneous Electricity Usage

The Cancer Center miscellaneous electricity usage was analyzed and simulated with EQUEST. The cancer center contains a CT scanner and linear accelerator which mostly contributes to its large miscellaneous usage. The results of the EQUEST simulation can be seen in Table 6.7.

Table 6.7: Cancer Center Misc. Energy Usage

Miscellaneous Electrical Equipment Usage/Demand for POBs in KWH / KW			
	2008	2009	2010
Jan	24540	22750	22750
Feb	22280	22280	22280
Mar	24750	26540	26640
Apr	25440	25440	25440
May	25530	23750	23750
Jun	23660	25440	25440
Jul	25530	24740	24740
Aug	25740	25540	25540
Sep	23460	24450	24450
Oct	25530	23750	23750
Nov	22660	23440	23440
Dec	24540	24540	24540
Total Usage	293660	292660	292760
Peak Demand	86	86	86
Watt per ft^2	5.5	5.5	5.5

6.8 Cancer Center Lighting Analysis

The cancer center lighting analysis was limited to EQUEST baseline analysis. Adequate lighting data for the Cancer Center could not be located.

6.9 HVAC Analysis

The HVAC analysis for the three facilities was broken down into generation, transport, and end use systems. Not all mechanical systems were analyzed in depth, but all major energy using equipment was considered in this analysis. Multiple tools and methods were utilized to estimate the energy usage of each piece of HVAC equipment. EQUEST was the primary HVAC analysis tool because of its ability to dynamically simulate off peak conditions. Other models were created and used for certain pieces of equipment and will be discussed in greater detail in each individual section.

6.9.1 Main Hospital Steam / Hot Water Generation Systems Analysis

In order to quantify the energy usage of the steam boilers at RMC, first combustion efficiency was measured and calculated by a Testo combustion analyzer for each fire tube steam boiler. The test simply consisted of measuring the stack flue gas temperature, the combustion air temperature, the oxygen percentage in the exhaust gas, and the carbon monoxide percentage in the exhaust gas. The tests were conducted at both low and high fire burner settings. The results of the combustion analysis are shown in Table 6.8.

Table 6.8: Boiler Combustion Analysis Results

Steam Boiler Combustion Analysis		
	Boiler 1	Boiler 2
Steam Pressure	81	90
Percent O2 Low Fire	7.5	6.5
Percent O2 High Fire	3.6	5
Carbon Monoxide Low Fire	1	0
Carbon Monoxide High Fire	0	0
Stack Temp Low Fire (F)	355	343
Stack Temp High Fire (F)	370	375
Combustion Efficiency Low Fire	83.7	84
Combustion Efficiency High Fire	84	84.4

The results of the test show that the RMC steam boilers are tuned fairly well. Some efficiency could be gained by fine tuning the oxygen level, but overall the burners are tuned to a satisfactory level. Energy bill data were used to establish the natural gas use of the two fire tube boilers. The natural gas flow into the boiler room is metered by RMC. Figure 6.3 shows the actual natural gas usage by the boilers for years 2008-2011 plotted monthly.

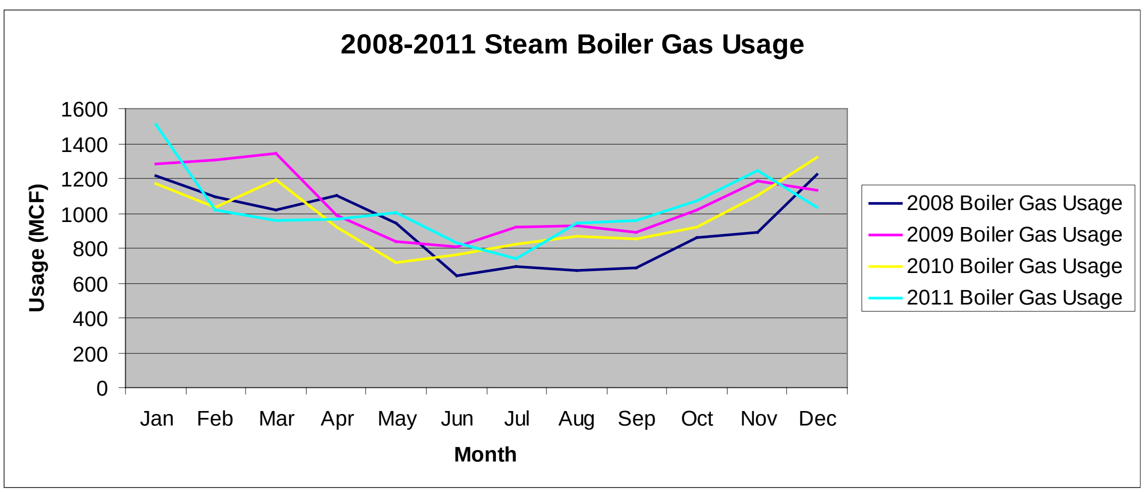


Figure 6.3: Steam Boiler Natural Gas Usage

The average total gas usage over the four years is 11,908 MCF (thousand cubic feet) per year or approximately 12.5 billion BTU per year. That reduces to an average daily use of about 34.25 million BTU per day. EQUEST was also used to simulate boiler gas usage. It resulted in a yearly usage of approximately 11 billion BTU. The actual results are somewhat higher than EQUEST results possibly because EQUEST does not simulate steam boiler systems well. The hot water generators are not individually metered and are in place primarily for redundancy. The maximum heating demand for the four years of this analysis never exceeded the capacity of the two steam boilers. It was concluded that these auxiliary boilers were not used during this analysis.

6.9.2 Main Hospital Chilled Water Generation Analysis

Chiller efficiency must be established in order to quantify energy use of the chilled water generation systems. The coefficient of performance or COP of the chiller is a measure of the chiller's efficiency. COP can be measured and calculated with a direct method. Some data are required to calculate COP for any given chiller. Chilled water flow through the evaporator, chilled water temperature difference across the evaporator, and compressor input power must be measured or known. The data were collected for each chiller at RMC during the building survey. Because chiller efficiency changes with load, it is important to collect data for multiple chiller loads in order to know the COP at all possible conditions. Equation (8.) below is used to calculate COP once all parameters were defined. Equation (9.) is used to convert COP into a kilowatt per ton of cooling ratio.

$$COP = \frac{CWF * E\Delta T * 0.147}{PI} \qquad (8).$$

Where: CWF is chilled water flow in gpm (gallons per minute).

EΔT is Temperature Difference across the Evaporator.

PI is compressor power input.

$$\frac{Kw}{ton} = \frac{12}{COP * 3.412} \qquad (9).$$

Where: COP is Coefficient of Performance

Table 6.9 shows the data collected at RMC and the results of the COP calculations for both chillers one and two. The conditions shown in Table 6.9 were measured and calculated for a condenser water entering temperature of 85 F.

Table 6.9: COP Calculation Results

Chiller 1 COP Table					**Chiller 2 COP Table**			
	Load					Load		
	100	75	50			100	75	50
Power Input	205	166	106		Power Input	278	201	145
Flow (gpm)	688	688	688		Flow (gpm)	990	990	990
Load Tons	300	225	150		Load Tons	360	270	180
E Delta T	10	8	5		E Delta T	10	7	5
COP	4.93	4.87	4.77		COP	5.23	5.07	5.02
Kw/Ton	0.713	0.722	0.737		Kw/Ton	0.672	0.694	0.701

The data and calculations show that chiller one has a COP of just under five, and chiller two has a COP of just over five. The water cooled chillers at RMC average about 0.7 kilowatts per ton over all conditions.

Now that chiller efficiency has been established, total chiller energy usage can be calculated if the building load is known for all weather conditions. Chillers are only at or near design capacity for a small amount of time in a given season. Over ninety percent of the chillers lifespan is spent at part load. Therefore in order to properly estimate chiller energy usage, one has to know off peak building loads. Interior building cooling loads and lighting cooling loads remain relatively constant throughout the year and can be directly quantified and simulated. The major contributing factors to building cooling load and differences in it are solar heat gain and outdoor air temperature. The building cooling load changes significantly with outdoor air temperature. In order to correctly estimate chiller energy usage during part load conditions, a relationship between outdoor air temperature and chiller load must be established. To establish this relationship, RMC chiller load data are recorded for multiple weather conditions. Chiller load data are then plotted against outdoor air temperature. The plot reveals a strong linear relationship between chiller load and outdoor air temperature. Figure 6.4 illustrates this linear relationship between cooling load and outdoor air temperature.

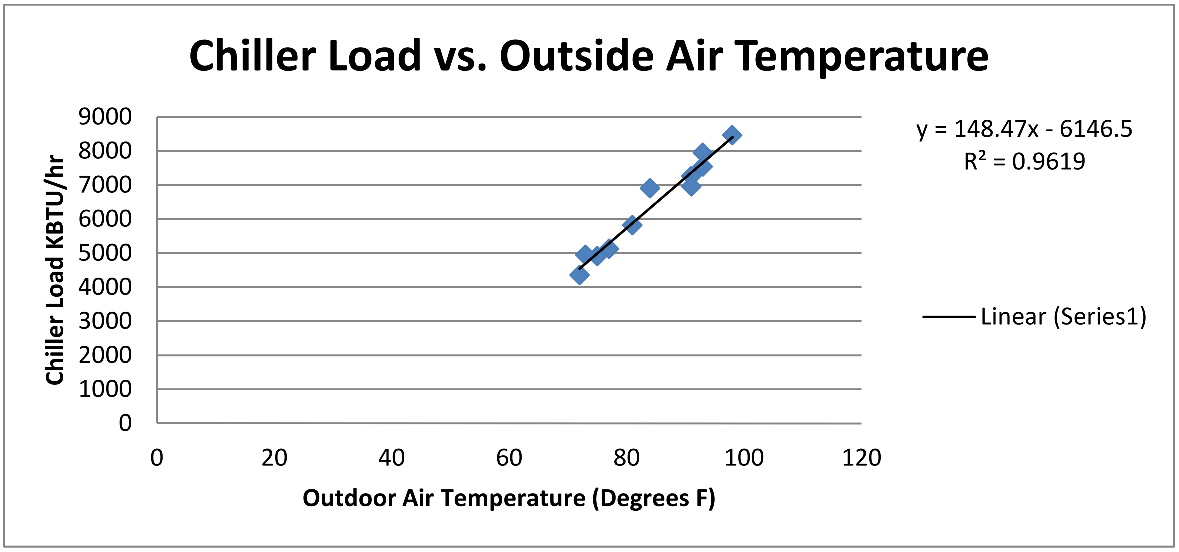


Figure 6.4: Plot of Chiller Load vs. Outside Air Temperature

The equation shown in the figure is then used to estimate chiller load for any given outdoor air temperature. The equation for chiller load and weather data are uploaded into MATLAB and simulated over the course of a year. The MATLAB simulation utilizes the chiller COP found previously to estimate energy usage. Flow through the chillers on both the evaporator and condenser side remains constant. The complete MATLAB code for this analysis can be found in Appendix C. The results of the simulation for years 2008-2011 are shown in the Table 6.11 and Figure 6.5 below. The chillers are simulated at the operating conditions found at the time of the simulation. Condenser water set point temperatures ranged from 85 to 72 degrees Fahrenheit. Chill water supply temperature is held as 42 degrees Fahrenheit for all years simulated. Table 6.10 and Figures 6.5 and 6.6 show the main hospital chiller electricity usage and demand plotted monthly over the entire year. The average chiller electricity usage for the MATLAB simulation over the four years analyzed is found to be 1,877,084 kilowatt hours per year. The average yearly demand is found to be 4,767 kilowatts. The average chiller operational cost is found to be an estimated $147,000 per year.

Table 6.10: Main Hospital Chiller Energy Usage 2008-2011

2008-2011 Main Hospital Chiller Energy Usage/Demand								
	2008		**2009**		**2010**		**2011**	
	Demand (KW)	**Usage (KWH)**	**Demand (KW)**	**Usage (KWH)**	**Demand (KW)**	**Usage (KWH)**	**Demand (KW)**	**Usage (KWH)**
Jan	387	127632	404	131332	338	120457	349	82920
Feb	324	91815	288	85045	239	71036	285	59602
Mar	362	115008	345	118883	324	93856	334	99813
Apr	370	142804	426	136647	430	145555	428	133315
May	428	177190	427	192614	451	203703	480	188888
Jun	490	219336	475	236951	487	239013	503	251229
Jul	511	232402	474	234375	512	265298	501	267888
Aug	493	233544	474	235695	507	243653	494	252511
Sep	427	203759	427	205866	492	215919	441	163475
Oct	362	124490	425	144126	492	185070	364	110189
Nov	305	95820	309	96696	324	103892	296	88246
Dec	288	104714	260	83644	259	81925	261	70496
Total	**4745**	**1868513**	**4733**	**1901875**	**4856**	**1969376**	**4737**	**1768572**
Cost-$	$24,912	$121,453	$24,847	$123,622	$25,492	$128,009	$24,868	$114,957

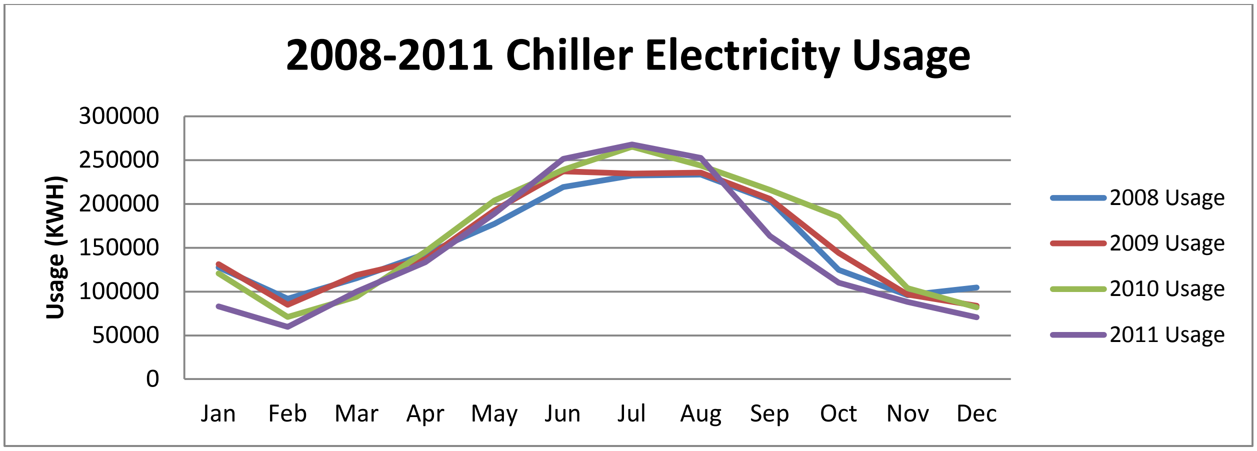


Figure 6.5: Monthly Chiller Electricity Usage

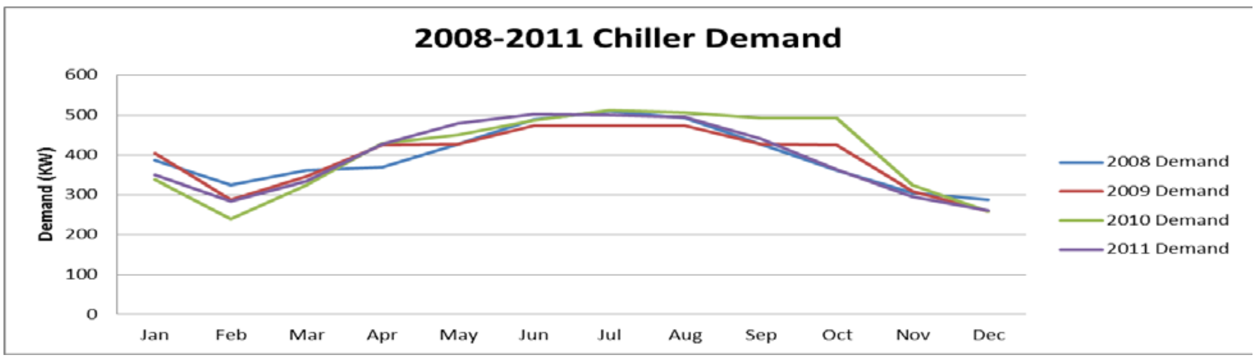


Figure 6.6: Monthly Chiller Electricity Demand

EQUEST is also used to simulate chiller energy usage. Chiller efficiency is uploaded directly into the EQUEST simulation wizard. EQUEST calculates chiller loads based on weather data and the building envelope model that is created by the user. All information related to the building envelope, HVAC equipment, electrical equipment, and building occupancy hours are uploaded into EQUEST. EQUEST dynamically calculates building loads and chiller usage based the information input by the user. The results of the EQUEST chiller usage simulations for years 2008-2011 are shown in Table 6.11. The results are similar to the MATLAB simulation but there are some differences in the total usage. The two models average a 4% difference.

Table 6.11 EQUEST Chiller Usage Results

EQUEST Main Hospital Chiller Simulation Results 2008-2011								
	2008		2009		2010		2011	
	Demand	Usage	Demand	Usage	Demand	Usage	Demand	Usage
Jan	250	94800	250	93100	240	88900	210	82500
Feb	230	84800	250	84800	2401	81100	200	75300
Mar	250	113500	250	113900	230	108100	200	97300
Apr	330	137100	340	136300	330	128200	250	112100
May	420	187100	400	186600	380	178000	370	152400
Jun	470	234000	480	235000	480	231100	450	199200
Jul	460	262800	470	262900	470	260400	420	224800
Aug	450	260800	430	260700	450	259900	400	224500
Sep	400	211300	400	211700	410	206500	370	174200
Oct	390	129800	390	129500	400	122700	310	108300
Nov	250	100300	250	100000	220	95100	190	86900
Dec	230	88900	210	89000	210	85200	190	79900
Total	4130	1905200	4120	1903500	6221	1845200	3560	1617400

6.9.3 Main Hospital Cooling Tower Analysis

The cooling tower is primarily analyzed using a cooling tower model based on the counter flow airside effectiveness model developed by James Braun [11]. The reader should refer to Braun [11] for a complete development of the basic model equations. Braun's [11] model makes the following basic assumptions:

1. Heat and mass transfer in the normal direction to flows only.
2. Heat and mass transfer through tower walls are neglected.
3. Heat transfer from the tower fans is neglected.
4. Constant water and dry air specific heats.
5. The mass fraction of water vapor in the air/vapor mixture is approximately equal to the humidity ratio.
6. Uniform temperature throughout the water stream at any cross section.
7. Uniform tower cross sectional area.

The cooling tower model is created and used to find the cooling tower outlet water temperature given inlet conditions of both the air and water. Braun [11] and Weber [12] developed the following differential equations from steady state energy and mass balances on an incremental volume to describe the cooling tower.

$$\frac{d\omega_a}{dV} = -\frac{NTU}{V_T}(\omega_a - \omega_{s,w}) \tag{10.}$$

Where: ω_a is the specific humidity of dry air.

NTU is the number of transfer units.

$\omega_{s,w}$ is the specific humidity of the saturated air.

V_T is the Volume of the Tower.

$$\frac{dh_a}{dV} = -\frac{Le*NTU}{V_T}\left[\left(h_a - h_{s,w}\right) + (\omega_a - \omega_{s,w})(\frac{1}{Le} - 1)h_{g,w}\right] \tag{11.}$$

Where: Le is the Lewis Number.

h_a is the enthalpy of moist air.

$h_{s,w}$ is the enthalpy of saturated air.

$h_{g,w}$ is the enthalpy of water above reference state for liquid water at T(ref).

$$\frac{dT_w}{dV} = \frac{\frac{dh_a}{dV} - C_{pw}(T_w - T_{ref})\frac{d\omega_a}{dV}}{\left[\frac{\dot{m}_{w,i}}{\dot{m}_a} - (\omega_{a,o} - \omega_a)\right]C_{pw}} \tag{12.}$$

Where: C_{pw} is the specific heat of water.

T_w is temperature of the water.

T_{ref} is reference temperature (32 degrees)

$\dot{m}_{w,i}$ is mass flow rate of the water.

$\dot{m}_a$ is mass flow rate of the air.

If the inlet conditions, the number of transfer units, and the Lewis number are known, the exit conditions of both the air and water streams can be calculated. Braun [11] applied Merkel's assumptions to simplify the analysis in order to avoid solving complex differential equations. The Merkel assumption neglects the effect of water loss due to evaporation and sets the Lewis number to 1. When applied, these assumptions reduce the differential equations to:

$$\frac{dh_a}{dV} = -\frac{NTU}{V_T}(h_a - h_{s,w}) \tag{13.}$$

$$\frac{dT_w}{dV} = \frac{\dot{m}_a(\frac{dh_a}{dV})}{\dot{m}_w C_{pw}} \tag{14.}$$

Braun [11] introduces the C_s (saturation specific heat) term, which is the derivative of the saturated air enthalpy with respect to temperature evaluated at the water temperature.

$$C_s = \left[\frac{dh_s}{dT}\right]_{T=T_w} \qquad (15.)$$

Where: h_s is the saturated air enthalpy.

The introduction of C_s allows equation (14.) to be written in terms of air enthalpy.

$$\frac{dh_{s,w}}{dV} = \frac{\dot{m}_a C_s(\frac{dh_a}{dV})}{\dot{m}_w C_{pw}} \qquad (16.)$$

Braun [11] then suggests that if the saturation air enthalpy were linear with respect to temperature then the exit conditions could be solved for analytically. Saturation air enthalpy does not vary linearly with temperature, but Braun [11] states that if an appropriate average slope between inlet and outlet conditions is chosen then an air side effectiveness can be developed. Braun [11] defines the air side effectiveness as the ratio of the actual heat transfer to the maximum possible heat transfer if the exiting air were saturated at the temperature of the incoming water. Braun [11] writes the heat transfer in terms of the air side effectiveness as follows:

$$\dot{Q} = \dot{\varepsilon}_a \dot{m}_a \left(h_{s,w,i} - h_{a,i}\right) \qquad (17.)$$

Where: $\dot{\varepsilon}_a$ is the airside effectiveness.

$h_{s,w,i}$ is the enthalpy of the entering water.

$h_{a,i}$ is the enthalpy of the entering air.

The air side effectiveness $\dot{\varepsilon}_a$ is defined as follows:

$$\dot{\varepsilon}_a = \frac{1-e^{-NTU(1-m^*)}}{1-m^* e^{-NTU(1-m^*)}} \qquad (18.)$$

Where: m^* is a dimensionless variable.

$$m^* = \frac{\dot{m}_a}{\dot{m}_w(\frac{C_{pw}}{C_s})} \qquad (19.)$$

C_s is estimated as the average slope between the inlet and outlet water conditions.

$$C_s = \frac{h_{s,w,i} - h_{s,w,o}}{T_{w,i} - T_{w,o}} \qquad (20.)$$

Where: $h_{s,w,o}$ is the enthalpy of the exiting water.

$h_{s,w,i}$ is the enthalpy of the entering water.

$T_{w,i}$ is the temperature of the entering water.

$T_{w,o}$ is the temperature of exiting water.

Braun [11] states that the outlet air enthalpy and water temperature can then be calculated using the following equations:

$$h_{a,o} = h_{a,i} + \varepsilon_a(h_{s,w,i} - h_{a,i}) \qquad (21.)$$

$$T_{w,o} = \frac{\dot{m}_{w,i}(T_{w,i} - T_{ref})C_{pw} - \dot{m}_a(h_{a,o} - h_{a,i})}{\dot{m}_{w,o}C_{pw}} + T_{ref} \qquad (22.)$$

Water loss due to evaporation is neglected for this analysis, so the mass flow rate entering is set equal to the mass flow rate exiting. That allowed equation (22.) to be reduced to the following:

$$T_{w,o} = \frac{(T_{w,i}) - \dot{m}_a(h_{a,o} - h_{a,i})}{\dot{m}_w C_{pw}} \qquad (23.)$$

In order to obtain the outlet water temperature, one must first find the design conditions of the cooling tower. The main hospital cooling tower design specifications can be found in Section 4.4.1.4. The RMC design cooling tower conditions are a max inlet temperature of 95 degrees Fahrenheit with an exit temperature of 85 degrees

Fahrenheit at a 78 degree Fahrenheit wet bulb temperature. It is designed for a ten degree range and a seven degree approach. The design water flow rate is 1980 gallons per minute or roughly 3 gallons per minute per ton of cooling. The design airflow rate is 181,500 cubic feet per minute max with a turn down ratio of one half. The liquid to gas ratio (L/G) ratio is 1.2. To run the model, the water and air volumetric flow rate are converted to a mass flow rate. This is accomplished simply by multiplying by the respective density of the fluid. MATLAB is used to run the model to find the exiting water temperature of the cooling tower. The model is arranged to accept weather conditions and chiller load to predict an outlet water temperature. The curve for chiller load is established previously in the chiller analysis section. The model accepts outdoor air temperature, relative humidity, and wet bulb temperature as inputs. From those inputs, entering water and air enthalpy are calculated using equations defined from curve fits of air and water enthalpy at different weather conditions. Tower NTU must be defined in order to correctly predict outlet water temperatures. NTU is a dimensionless constant that defines the heat transfer capability of the tower relative to its size. It remains constant with a constant air and water flow rate. If air flow rate or water flow rate changes, the NTU value changes. RMC‘s cooling tower is equipped with two speed fans. It has two possible air flow rates and one possible water flow rate. Based on these facts, three separate NTU values are defined. One value for maximum air flow, one value for half speed air flow, and one value for natural convective airflow are found. Calculating NTU requires an iterative process. Equations (18, 19, 20, 21, and 23) are iterated in order to find the appropriate value for NTU. In order to start the process an initial exiting water temperature is assumed. A good first assumption is wet bulb temperature because that is the theoretical limit of a cooling tower's performance. Entering water temperature is also assumed. Any manually collected or published data on the cooling tower's performance can and should be used to provide an initial inlet and outlet water temperature. After initial inlet and outlet water temperature assumptions are made, an initial NTU value is also assumed. A very small value is first assigned, and it is increased with every iteration. Each iteration produces a new outlet water temperature. The expected outlet water temperature converges at the correct NTU value for the RMC cooling tower. This process is automated in MATLAB and is repeated for each of the possible air flow rates. The correct NTU value for the RMC cooling tower at design conditions is found to be 6.72. This value decreases with the smaller air flow rates. Half flow produces an NTU value of 4, and the fan off NTU is found to be 0.8. The MATLAB program used to find NTU is found in Appendix D.

Once the NTU value is correctly established, the model is modified to be able to dynamically predict outlet water temperatures over time given different chiller loads and weather conditions. With NTU properly identified, it no longer requires iteration. Cooling tower air side effectiveness can be directly calculated which allows for the outlet water temperature to be calculated for any given weather condition. The model's accuracy is shown by Figure 6.7.

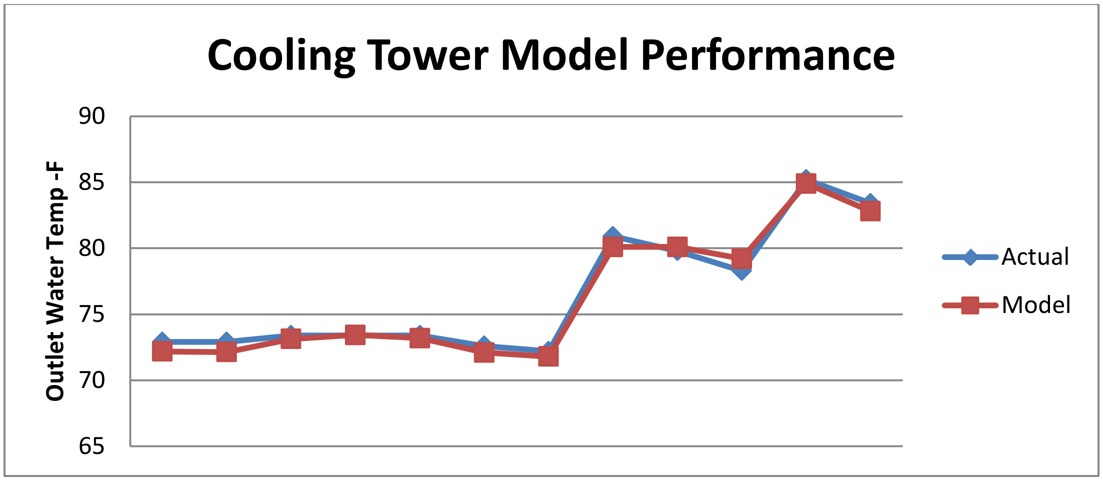


Figure 6.7: Cooling Tower Model Accuracy

The purpose of this model is to establish the energy use of the cooling tower over time. In order to quantify cooling tower energy use, a two speed fan controller is modeled in MATLAB to simulate the cooling tower fans ramping up and down to meet the desired condenser water set point. Many changes in fan speed can damage two speed fan motors, so a broad control range is generally placed around the set point in order to minimize the number of times the fan ramps up and down. At RMC this range is 10 degrees, or set point plus or minus five degrees. Half speed engages once the outlet water temp is less than two degrees below set point. The model is programed to track fan energy use as the fan ramps up and down. The basin of the tower holds a significant amount of water and cannot be neglected. The first order differential equation below was written in order to account for the water temperature difference in the basin of the tower.

$$\frac{dT_{basin}}{dt} = \frac{\dot{m}_w(T_{w,o} - T_{basin})}{m_{basin}} \qquad (24.)$$

Where: T_{basin} is the temperature of the water exiting the cooling tower basin.

m_{basin} is the mass of the water contained in the cooling tower basin.

Chiller load was simulated with the curve established previously based on outside air temperature. Cooling tower entering water temperature is calculated using the First Law of Thermodynamics to solve for the new entering water temperature after all building heat and the heat of compression are added to the condenser inlet temperature.

$$T_{w,i} = \frac{\dot{Q}_{chiller}*12000*1.25}{60*\dot{m}_w} + T_{basin} \qquad (25.)$$

Where: $\dot{Q}_{chiller}$ is the heat rejected to the condenser from the building.

Weather data containing dry bulb outdoor air temperature, relative humidity, and wet bulb temperature are uploaded into the model. In the model, the weather data are updated every twenty minutes. To ensure proper model resolution, the model time step was simulated at every 30 seconds for an entire year's data. An example model simulation output figure of the cooling tower water outlet temperature plotted over time is shown in figure 6.8 below.

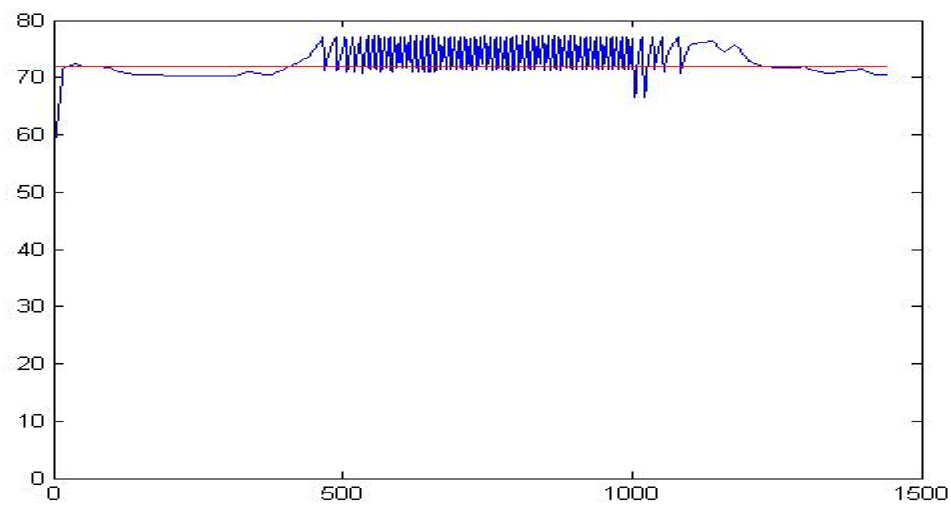


Figure 6.8: Two-Speed Fan Cooling Tower Simulation Plot

The model calculates cooling tower outlet water temperature, condenser water entering temperature, condenser water exiting temperature, cooling tower fan run time, cooling tower fan cycles, cooling tower fan electricity demand, and cooling tower fan energy usage. The MATLAB program for the cooling tower energy simulation is found in Appendix D. Cooling tower fan energy usage and cost can is shown in Table 6.12 below.

Table 6.12: Cooling Tower Energy Usage

Cooling Tower Model Energy Usage Simulation Results				
	2008	**2009**	**2010**	**2011**
Usage-KWH	65335	65190	74254	172855
Cost-$	$4,247	$4,237	$4,827	$11,236

6.9.4 Main Hospital Transport System -Steam Trap Analysis

Steam trap systems frequently leak and therefore were analyzed to check if any leakage was occurring at RMC. A good indicator of a leaking steam trap system is the make-up water percentage to the boiler. In order to find the make-up water percentage, the typical dissolved solids (TDS) levels are measured at three locations in the system. TDS levels are measured by water conductivity instrumentation. The boiler feed water, softened make-up water, and the returning condensate TDS levels were measured. The results of those measurements were fed into the mass balance equation below to calculate the percentage of make-up boiler water used by RMC.

$$\%\, make\ up = \frac{TDS_{Feedwater} - TDS_{condensate}}{TDS_{Makeupwater} - TDS_{condensate}} * 100 \qquad (26.)$$

It is found that the main hospital boiler system had a 30% make up water rate. This is a fairly large make up water level which means that there are significant losses of steam or condensate in the system. Those losses contribute to a significant energy loss and cost.

For this analysis, it is assumed that half the losses were due to steam venting and half the losses were due to condensate loss. The boilers at RMC produce approximately 2500 pounds of steam per hour on average.

$$Steam\ Energy\ Loss = \frac{\%loss_{Steam} * SteamFlow \frac{lb_{steam}}{hr} * 1{,}150 \frac{BTU}{lb_{steam}} * 8760 \frac{hr}{yr}}{Boiler\ Efficiency} \qquad (27.)$$

$$\frac{0.15 * 2500 \frac{lb_{steam}}{hr} * 1{,}150 \frac{BTU}{lb_{steam}} * 8760 \frac{hr}{yr}}{.8} = 4{,}722{,}187{,}500 \text{ BTU}$$

$$\frac{4{,}722{,}187{,}500 \text{ BTU} * \$6}{1{,}000{,}000\ BTU} = \$28{,}333 \text{ per year}$$

$$Cond.\ Energy\ Loss = \frac{\%loss_{Cond.} * SteamFlow \frac{lb_{steam}}{hr} * 140 \frac{BTU}{lb_{condensate}} * 8760 \frac{hr}{yr}}{Boiler\ Efficiency} \qquad (28.)$$

$$Cond.\,Energy\ Loss = \frac{0.15 * 2500\,\frac{lb_{steam}}{hr} * 140\,\frac{BTU}{lb_{steam}} * 8760\,\frac{hr}{yr}}{0.8}$$

$$= 574{,}875{,}000 BTU$$

$$\frac{574{,}875{,}000\ \text{BTU} * \$6}{1{,}000{,}000\ BTU} = \$3{,}449\ \text{per year}$$

$$Total\ Cost = \$28{,}333 + \$3{,}449 = \$31782\ per\ year$$

6.9.5 Main Hospital Transport System - Air Handler Analysis

The main hospital air handler analysis first consists of making coil load calculations from raw data collected during the building survey. Data were taken at each air handler in order to verify design and to be able to quantify energy use. Cooling coil loads were calculated using the First Law of Thermodynamics. Due to the warm and humid summers, the air handlers typically have a significant sensible and latent cooling load. In order to dehumidify, the air must be cooled below the dew point to condense unwanted moisture in the air to properly control building climate. Volumetric flow rate of the air, temperature, and humidity were measured at each air handler. Cooling coil sensible and latent load are estimated using the following equations:

$$Sensible\ Cooling\ Load\ (Tons) = \frac{\dot{V}_{air} * \rho_{air} * C_{p,air}(T_{mixed\ air} - T_{supply})}{200\frac{Btu}{min/ton}} \qquad (29.)$$

Where: $\dot{V}_{air}$ is the volumetric flow rate of air in cubic feet per min(cfm).

ρ_{air} is the density of air in lbm/ft^3.

$C_{p,air}$ is the specific heat of air in BTU/lbm-F.

$T_{mixed\ air}$ is the temperature of the mixed air in degrees Fahrenheit.

T_{supply} is the temperature of the supply air in degrees Fahrenheit.

$$Latent\ Cooling\ Load(Tons) = \frac{\dot{V}_{air} * \rho_{air} * 1000\frac{BTU}{lb}(\omega_{mix} - \omega_{supply})}{200\frac{BTU}{min/ton}} \qquad (30.)$$

Where: ω_{mix} is the specific humidity of the mixed air stream.

ω_{supply} is the specific humidity of the supply air stream.

$$Total\ Cooling\ Load(Tons) = Sensible + Latent \qquad (31.)$$

Water flow rate through the coil is calculated by the following equation:

$$Coil\ Flow\ rate(gpm) = \frac{\dot{Q}_{Total\ Coil\ Load}\frac{BTU}{min}}{\rho_{water}(\frac{lbm}{gal}) * C_{pwater}(\Delta T_{water Across\ Coil})} \tag{32.}$$

The chilled water distribution system at the main hospital is plumbed so that chiller one serves the critical area air handlers and chiller two serves the remainder. The typical supply water temperature for chiller one air handlers was 42 degrees Fahrenheit. Chiller two air handlers are typically supplied with 42 to 43 degree chilled water. Tables 6.13 and 6.14 list the findings of the initial air handler analysis.

Table 6.13: Main Hospital Chiller 1 AHU Analysis Results

Chiller 1 Units	ahu-2 bot	ahu-4 top	ahu-OR3	ahu-Or2	ahu-ER	ahu OR-1
% OA	26	25	30	81	23	78
Actual GPM	129	21	188	93	93	82
Part Load GPM	113	18	141	46	81	40
AirFlow	17500	2866	23000	7956	11859	6998
Outside Air Temp Full Load	98	98	98	98	98	98
Outside Air 50% Load	70	70	70	70	70	70
Return Air Temp	71	71	68	62	73	63
Full Load OA RH	60	60	60	60	60	60
50% Load OA RH	40	40	40	40	40	40
RA RH	50	50	55	55	55	60
Mix RH Full Load	53	53	57	59	56	60
Mix RH Part Load	43	43	45	52	43	56
Mix SH Full Load	0.01180	0.01180	0.01230	0.01550	0.01180	0.01550
Mix SH Part Load	0.0068	0.0068	0.0066	0.0074	0.0068	0.0074
Full Load Mix Temp (F)	81	81	80	86	80	86
50% Load Mix Temp (F)	71	71	69	68	71	68
Supply Temp (F)	53	54	52	52	52	52
Supply RH	100	100	100	100	100	100
Supply SH	0.0087	0.00874	0.0081	0.0081	0.0081	0.0081
Full Load Coil Load (Tons)	64	10	94	46	46	41
50% Load Coil Load (Tons)	28	4	35	11	20	10
Total Full Load (tons)	302					
Total Part Load (tons)	110					

Table 6.14: Main Hospital Chiller 2 Airhandler Analysis Results

Chiller 2 Units	ahu-3A	ahu-3b	ahu-1e	ahu-g1	ahu-G2	ahu - L1	ahu - MRI	ahu - 2C	ahu - K	ahu - 2E	ahu - 3R
% OA	27	47	50	8	17	0	0	35	51	29	10
Actual GPM	30	21	273	49	34	80	64	53	116	122	69
Part Load GPM	13	9	119	31	21	50	41	27	21	61	46
AirFlow	2500	1400	18525	5200	3324	8360	6318	3900	6000	9800	7500
Outside Air Temp Full Load	98	98	98	98	98	98	98	98	98	98	98
Outside Air 50% Load	70	70	70	70	70	70	70	70	70	70	70
Return Air Temp	62	70	70	69	73	71	70	72	70	72	72
Full Load OA RH	60	60	60	60	60	60	60	60	60	60	60
50% Load OA RH	40	40	40	40	40	40	40	40	40	40	40
RA RH	55	50	50	50	50	55	55	50	50	55	50
Mix RH Full Load	56	55	55	51	52	55	55	54	55	56	51
Mix RH Part Load	44	45	45	41	42	40	40	44	45	44	41
Mix SH Full Load	0.00920	0.01310	0.01350	0.00840	0.01020	0.00876	0.0084	0.0121	0.0135	0.0121	0.0089
Mix SH Part Load											
Full Load Mix Temp (F)	72	83	84	72	77	71	70	81	84	80	74
50% Load Mix Temp (F)	64	70	70	69	73	71	70	71	70	72	72
Supply Temp (F)	50	51	52	53	55	54	52	52	60	54	55
Supply RH	100	100	100	100	100	100	100	100	100	100	100
Supply SH	0.0075	0.0078	0.0081	0.0087	0.0091	0.00874	0.0081	0.0081	0.001	0.00874	0.0091
Full Load Coil Load (Tons)	12	9	114	21	14	33	27	22	48	51	29
50% Load Coil Load (Tons)	3	2	30	8	5	13	10	7	5	15	11
Total Full Load (Tons)	380										
Total 50% Load (Tons)	110										

The analysis yields a combined full load demand of approximately 682 tons of cooling. That is just over the capacity of the chillers, but it is calculated at the most extreme conditions. The part load analysis yields an estimated 220 tons of cooling demand.

The largest energy cost at the air handler is from the air handler fans. Due to the fact that many of the air handler systems at RMC are variable volume and are at part load most of the time, the analysis must be dynamic and take into account part load situations. In order to do that, one must have trended data on the air terminal boxes and the air handler fans. Trended data is not available at RMC; however EQUEST can simulate variable building loads and can be uploaded with all the air handler specifications. EQUEST software is used to account for the variable loading on the air handler fans because it can simulate building load at all points in the building during the entire day and for all weather conditions. EQUEST was programed with space temperature settings and all air handler and air terminal box design specifications. The air handler systems were then simulated over the course of a year. The results of the simulation are shown in Table 6.15. Table 6.16 shows total air handler fan electricity use for all air handlers and ventilation fans combined on a monthly basis.

Table 6.15: Ventilation Fan Energy Consumption

Air Handler and Ventilation Fan Energy Use 2008-2011				
	2008	**2009**	**2010**	**2011**
Jan	100900	100100	99900	93700
Feb	90400	90300	90000	83300
Mar	105400	105600	106000	100700
Apr	106000	106100	106200	100900
May	112000	112000	112000	107300
Jun	109800	110100	110000	106900
Jul	113800	114000	113800	110300
Aug	113900	113800	113800	110900
Sep	108800	109000	108900	104900
Oct	107900	108200	108200	102800
Nov	99300	99200	99400	94200
Dec	99200	99900	100000	93600
Total:	1267400	1268300	1268200	1209500

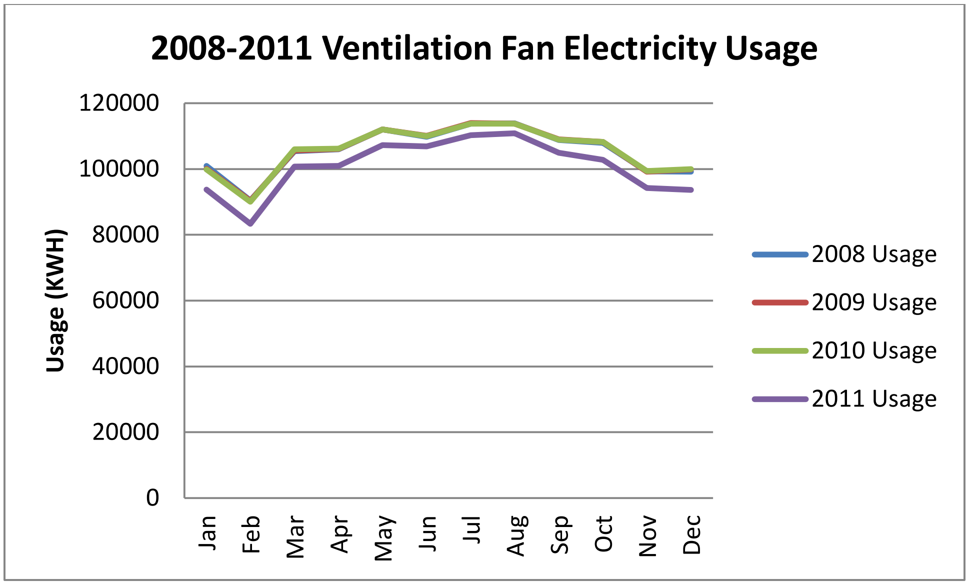


Figure 6.9: Ventilation Fan Energy Usage 2008-2011

6.9.6 Main Hospital Transport System- Chilled / Condenser / Hot Water Pumps

The transport pumps at the main hospital consist of constant volume chilled and condenser water pumps dedicated to their respective chillers. The combined pumping power is 90 horsepower. This analysis also includes hot water pumps which have a combined 30 horsepower. A total pump efficiency of 80 percent is assumed for each pump. When in operation, the pumps do not vary flow but are staged on and off depending on which chiller is operating. EQUEST is used once more to determine the total electricity consumption of all the transport pumps over the course of one year. The results of the EQUEST simulation can be seen in Table 6.16. Figure 6.10 shows pump energy usage plotted monthly.

Table 6.16: EQUEST Transport Pump Analysis Results

Main Hospital Transport Pumps 2008-2011 in KWH				
	2008	**2009**	**2010**	**2011**
Jan	40100	38700	38700	38800
Feb	36700	36600	36700	36800
Mar	40600	40600	40600	40700
Apr	41000	40500	40000	39400
May	47300	47300	46200	42600
Jun	53500	53600	52900	49300
Jul	58400	58200	57400	53400
Aug	58600	58700	58500	53900
Sep	51400	51300	50800	45500
Oct	42200	42100	41900	41300
Nov	39300	39300	39300	39400
Dec	40600	40600	40600	40700
Total:	**549700**	**547500**	**543600**	**521800**

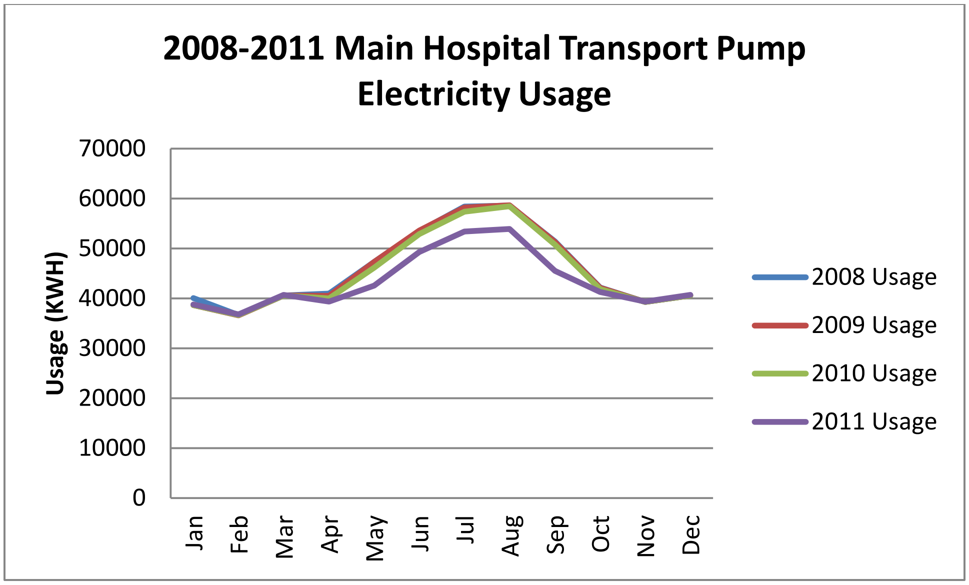


Figure 6.10: Main Hospital Transport Pump Energy Consumption 2008-2011

6.9.7 Main Hospital End Use System- Fan Coil Units

Fan Coil Units are placed in some patient rooms and offices around the main hospital. Their use depends on the occupancy of the room. If in cooling mode operation they will consume approximately 4300 BTU's per hour per unit based on a ten degree temperature difference across the coil. There are a total of 16 units throughout the hospital. The maximum demand of all the units is an estimated 69,000 BTU's per hour or 6 tons of cooling energy. At a chiller efficiency of 0 .7 kw/ton the fan coil units account for an estimated 4 kilowatts maximum cooling electricity demand. They are equipped with ¼ horsepower electric fan motors. If maximum demand is needed for 24 hours the units would consume 100 kwh of electricity costing approximately $6.50 per day of use at maximum demand. If used for heating, they will consume an estimated 8,600 BTU's per hour per unit based on a 20 degree temperature difference across the coil for a total of 137,600 BTU's per hour for all units at maximum demand. At an overall boiler efficiency of 80%, the total input energy needed would be an estimated 172,000 BTU's per hour. At an average natural gas price of $6 per million BTU, 24 hours of operation would cost an estimated $25 per day at maximum demand.

6.9.8 Main Hospital End Use Systems - Plate and Frame Heat Exchanger Analysis

The plate and frame heat exchanger that serves the main hospital has a total design cooling capacity of 4,300,000 BTU's per hour or approximately 360 tons. Data recorded during the building survey shows water entering the cold side at 46 degrees Fahrenheit and leaving the hot side at 53 degrees Fahrenheit. The heat exchanger approach as seen from the data is 7 degrees Fahrenheit. The design approach for the heat exchanger is 2 degrees Fahrenheit. The heat transfer for a plate and frame heat exchanger can be described using the following equation.

$$\dot{Q}_{HX} = U_{overall} A_{HX} * LMTD \qquad (33.)$$

Where: $$LMTD(\log\ mean\ Temp\ Differece) = \frac{(T_{1hot} - T_{2cold}) - (T_{2hot} - T_{1cold})}{\ln\frac{(T_{1hot} - T_{2cold})}{(T_{2hot} - T_{1cold})}}$$

$U_{overall}$ is the overall heat transfer coefficient in BTU/hr-ft^2-F.

A_{HX} is the heat exchanger surface area.

The design log mean temperature difference for the heat exchanger at RMC is 2.466. Assuming the design load and a typical overall heat transfer coefficient of 1000 BTU/hr-ft^2-F to solve for the required area yields an estimated heat exchanger area of 1,744 square feet. Actual data at RMC yields an LMTD of 5.36. Solving for the new overall heat transfer coefficient yields a heat transfer coefficient of 461 BTU/hr-ft^2-F. That is below typical design standards and could explain the poor performance of the heat exchanger. The heat exchanger is more than likely fouled on the cold side.

The purpose of the plate and frame heat exchanger at RMC is to allow free cooling when weather conditions permit. The cooling tower model described in the cooling tower analysis section is used to predict the possible amount of operational hours for free cooling. The program checks weather conditions and the cooling tower outlet water temperature. A time counter is programmed to count the number of hours the cooling tower outlet water temperature is at or below 50 degrees Fahrenheit which is the design entering water temperature for the plate and frame heat exchanger. The simulation resulted in an average total of an estimated 1000 to 1200 hours of free cooling in a typical year.

6.9.9 Sterilizer / Washer Analysis

It was estimated by the Director of Surgical Services, Robert Ribolini that on average the hospital used two sterilizers for four to six hours per day, five days a week. The equivalence of the average of this was taken as each sterilizer operating for 2.5 hour per day. A large industrial washer was also included in this analysis. Tables 6.17 and 6.18 show the total usage of each sterilizer.

Table 6.17: Sterilizer Energy Use

Sterilizers			Supply Pressure	Assumed Supply	Supply	Vapor	Peak Flow	Peak Flow	Peak load	Dynamic Flow	Enthalpy			
Model	Type	Size (in)	Range (psig)	Pressure (psia)	Temperature (F)	Quality (%)	Rate (lbs/hr)	Rate (lbs/hr)	(Btu/hr)	Rate (lbs/hr)	(Btu/lbm)	Btu/hr	Btu/hr	KW
3033	Vacamatic	24x36x36	50-80	80	312	97+	190		526435	112	1183	132496	307580	90
3043	Vacamatic	24x36x48	50-80	80	312	97+	255	445		148	1183	175084		
G-116	Gravity	16x16x26	50-80	80	312	97+	83		235417	35	1183	41405	157339	46
2023	Vacamatic	20x20x38	50-80	80	312	97+	116	199		98	1183	115934		
			Supply Pressure	Assumed Supply	Supply	Vapor	Steam Use	Cycles per	Steam use		Enthalpy			
Model	Type	Size (in)	Range (psig)	Pressure (psia)	Temperature (F)	Quality (%)	(lbs/cycle)	Hour	(lbs/hr)		(Btu/lbm)	Btu/hr		
444	Washer	24x26x24	30-80	80	312	97+	40.3	4	161.2		1183	190700		
											Total	655619		

Table 6.18: Sterilizer Energy Use Min/Max/Average

Operation: 2 Sterilizers 4 to 6 hr per day, both 5 days a week		
	per day	per month
Minimum usage(BTU)	1657326.667	36007394.84
Maximum usage(BTU)	3559140	77326553.57
Average(BTU)	2608233.333	56666974.21

The results of the analysis indicate that the sterilizers consume up to 680 million BTU per year at an estimated operating cost of around $4,000 per year at an average natural gas rate of $6.00 per million BTU.

6.9.10 POB Hot Water Generation Systems

The POB is equipped with three natural draft hot water generators. The design average efficiency of these generators is 80 percent. EQUEST is used to simulate the boiler use over the course of a year. All design specifications pertaining to the hot water generators are uploaded into EQUEST. The POB hot water generators handle all space heating and miscellaneous needs for the POB. The only miscellaneous usage comes from a heat exchanger that is used to heat an indoor swimming pool. The overall natural gas usage quantified from the EQUEST simulations from years 2008-2011 is seen in Table 6.19 where the total gas usage is represented in BTU per year.

Table 6.19: POB Hot Water Boiler Natural Gas Usage

POB Hot Water Generator Gas Usage.				
	2008	**2009**	**2010**	**2011**
Jan	1400000000	1620000000	1420000000	1410000000
Feb	1230000000	1400000000	1230000000	1230000000
Mar	910000000	1030000000	900000000	900000000
Apr	680000000	770000000	680000000	690000000
May	580000000	680000000	600000000	600000000
Jun	500000000	550000000	490000000	490000000
Jul	490000000	550000000	500000000	510000000
Aug	490000000	560000000	490000000	500000000
Sep	460000000	610000000	550000000	550000000
Oct	790000000	900000000	800000000	800000000
Nov	1070000000	1230000000	1070000000	1070000000
Dec	1350000000	1510000000	1340000000	1350000000
Total:	**9950000000**	**11410000000**	**10070000000**	**10100000000**
Cost:	**$59,700.00**	**$68,460.00**	**$60,420.00**	**$60,600.00**

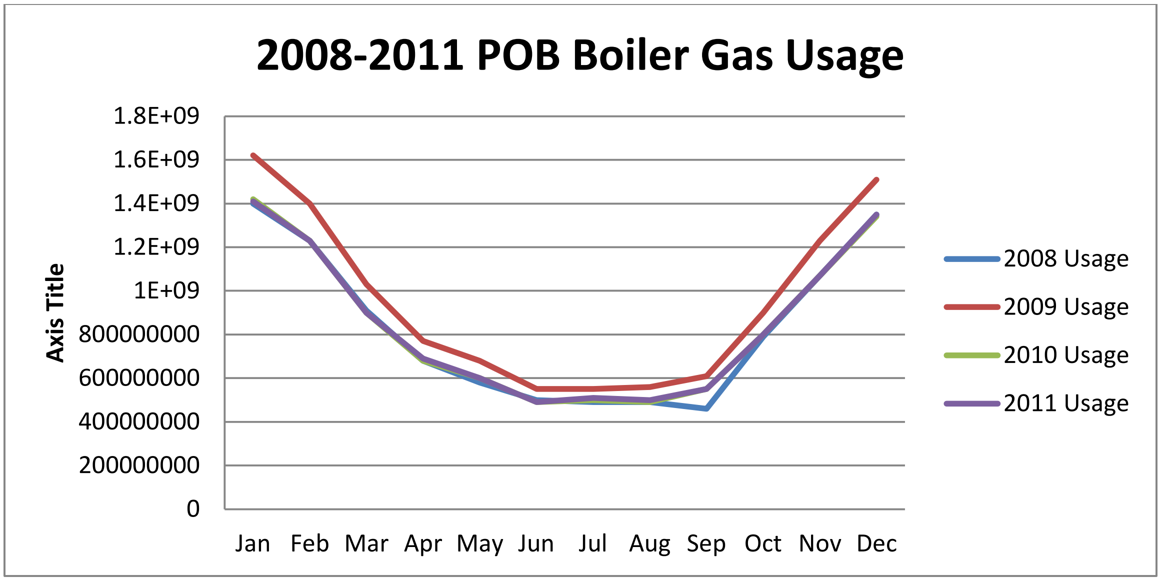


Figure 6.11: POB Monthly Boiler Gas Usage 2008-2011

6.9.11 POB Chilled Water Generation Systems

There are four chillers that serve the POB. The lead chillers are two screw type air cooled units, and the remaining two lag chillers are reciprocating, air-cooled models. Equations (8.) and (9.) are used to calculate COP and a Kw/ton ratio. The results yield an estimated 1.35 kw per ton for the screw type units and 1.8 kw per ton for the reciprocating units. The reciprocating units only engage when the load exceeds 240 tons because of their inefficient compressors. The specifications for the chillers were uploaded into EQUEST. The typical chilled water supply temperature for the POB is 44 degrees Fahrenheit. The results of the EQUEST simulations are shown in Table 6.20 and Figure 6.12.

Table 6.20: POB Chiller Consumption 2008-2011

POB Chiller Usage 2008-2011				
	2008	**2009**	**2010**	**2011**
Jan	64100	69700	70000	70500
Feb	91500	64100	64200	64600
Mar	91900	91500	92800	92400
Apr	115700	114800	115300	115500
May	170000	169100	168500	168600
Jun	216700	218700	219800	219800
Jul	241800	240800	240800	239100
Aug	238600	235700	237500	239700
Sep	185900	188700	188500	189200
Oct	105600	105700	104900	104500
Nov	76000	75100	75900	75200
Dec	65200	64400	64800	65300
Total:	**1663000**	**1638300**	**1643000**	**1644400**
Cost:	**108095**	**106489.5**	**106795**	**106886**

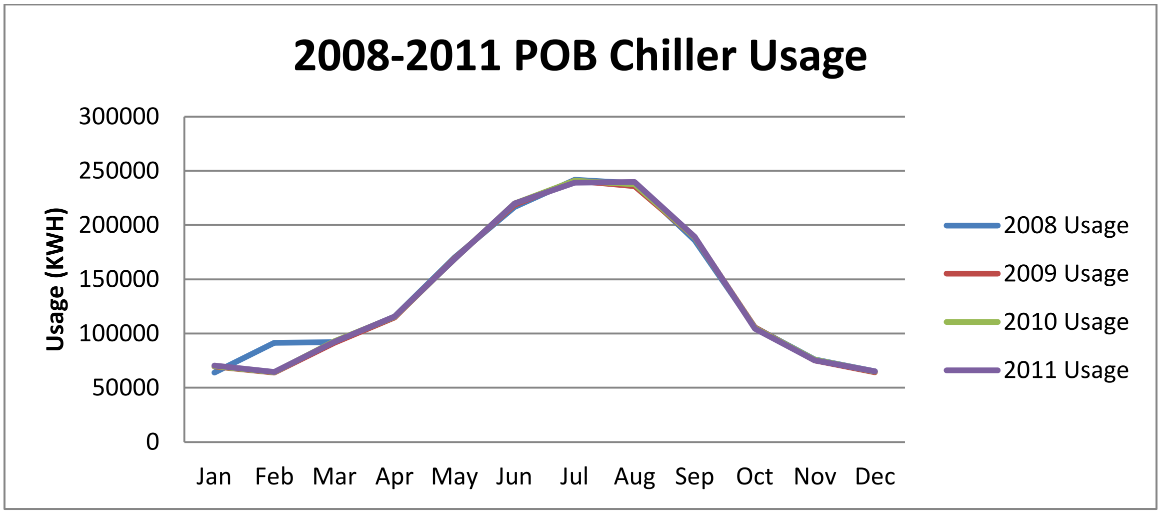


Figure 6.12: POB Monthly Chiller Consumption 2008-2011

The average total chiller electricity usage for the POB is an estimated 1,647,175 kwh per year.

6.9.12 POB Transport Systems - Air Handler Analysis

The air handler analysis for the POB is carried out exactly as it was for the main hospital. Data are gathered at every air handler in order to be able to calculate cooling coil load at design and part load conditions. Tables 6.21 and 6.22 shows the results of the POB air handler thermal analysis.

Table 6.21: POB AHU Analysis 100% Load

POB Air Handler Analysis 100% Load	ahu-GE	ahu-E1	ahu-E2	ahu ac1	ahu ac2	ahu-1	ac-2	ac-1	ac-g
RA Temp(degrees F)	72	70	73	73	73	70	72	71	72
RA Relative Humidity	39	38	45	40	40	35	35	35	36
RA CO2	654	825	902	741	741	814	744	707	707
OA Temp(degrees F)	81	81	81	70	70	75	none	none	none
OA Relative Humidity	23	23	23	42	42	14	none	none	none
Oa CO2	555	555	555	545	545	550	none	none	none
Mixed Temp (F)	80	80	80	80	80	80	80	80	80
Mixed RH	26	38	45	40	40	39	33	36	36
Mixed Enthalpy	36	45	42	42	42	38	37	36	38
mixed ω	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010	0.010
Supply Temp (F)	55	52	57	60	56	52	52	52	52
Supply RH	69	92	86	72	76	81	81	84	84
Supply Enthalpy	29	22.830	35.210	35.500	31.580	25.600	23.330	21.650	21.650
Supply ω	0.006	0.005	0.008	0.007	0.007	0.006	0.005	0.005	0.005
RA velocity(avg) ft/min	502	1293	1100	1050	1050	na			
RA CFM	4016	10995	13200	8932	8932	4576			
OA velocity(avg) ft/min	1029	125	150	1500	1500	na			
OA CFM	8544	237	1322	2844	2844	3976			
Supply velocity(avg) ft/min	1400	2150	1550	1000	3100	2200	1800	2000	1500
Supply CFM	6998	10750	13484	7750	20688	8785	9000	10000	10563
CFM RA + OA	5644	11232	14522	11776	11776	8552	9000	10000	10563
Chill water supply temp (degrees F)	49	44	44	44	44	42	42	42	43
Chill water return temp (degrees F)	60	50	60	50	50	50	50	50	48
Chill water difference (supply vs. return) degrees F	11	6	16	10	10	8	8	8	5
Percent OA(by cfm)	20	2	9	24	24	46	none	none	none
Tons of cooling in btu	311650	566955	462124	274536	787972	423775	474660	527400	557066
actual gpm	57	189	58	55	158	106	119	132	223
Total Cooling Load	366								

Table 6.22: POB AHU Analysis 50% Load

POB Air Handler Analysis 50% Load	ahu-GE	ahu-E1	ahu-E2	ahu ac1	ahu ac2	ahu-1	ac-2	ac-1	ac-g
RA Temp(degrees F)	72	70	73	73	73	70	72	71	72
RA Relative Humidity	39	38	45	40	40	35	35	35	36
RA CO2	654	825	902	741	741	814	744	707	707
OA Temp(degrees F)	71	71	71	71	71	71	71	71	71
OA Relative Humidity	23	23	23	42	42	14	none	none	none
Oa CO2	555	555	555	545	545	550	none	none	none
Mixed Temp (F)	71	71	71	71	71	71	71	71	71
Mixed RH	26	38	45	40	40	39	33	36	36
Mixed Enthalpy	36	45	42	42	42	38	37	36	38
mixed ω	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008	0.008
Supply Temp (F)	55	52	57	60	56	52	52	52	52
Supply RH	69	92	86	72	76	81	81	84	84
Supply Enthalpy	29	22.830	35.210	35.500	31.580	25.600	23.330	21.650	21.650
Supply ω	0.006	0.005	0.008	0.007	0.007	0.006	0.005	0.005	0.005
RA velocity(avg) ft/min	502	1293	1100	1050	1050	na			
RA CFM	4016	10995	13200	8932	8932	4576			
OA velocity(avg) ft/min	1029	125	150	1500	1500	na			
OA CFM	8544	237	1322	2844	2844	3976			
Supply velocity(avg) ft/min	1400	2150	1550	1000	3100	2200	1800	2000	1500
Supply CFM	6998	10750	13484	7750	20688	8785	9000	10000	10563
CFM RA + OA	5644	11232	14522	11776	11776	8552	9000	10000	10563
Chill water supply temp (degrees F)	49	44	44	44	44	42	42	42	43
Chill water return temp (degrees F)	60	50	60	50	50	50	50	50	48
Chill water difference (supply vs. return) degrees F	11	6	16	10	10	8	8	8	5
Percent OA(by cfm)	20	2	9	24	24	46	none	none	none
Tons of cooling in btu	180641	365715	209703	129456	400689	259325	306180	340200	359336
actual gpm	33	122	26	26	80	65	77	85	144
Total Cooling Load	**213**								

The analysis resulted in a full load air handler cooling coil load of 366 tons which is 6 tons above capacity and a part load of 213 tons.

The ventilation fan analysis is performed with EQUEST in order to properly simulate part load conditions. All of the air handlers in the POB's are equipped with variable speed drive fans allowing them to ramp up and down depending on demand.

The results of the EQUEST analysis are shown in Table 6.23 where the totals given are in kilowatt-hours.

Table 6.23: POB Ventilation Fan Energy Consumption 2008-2011

POB Air Handler and Ventilation Fan Energy Use 2008-2011				
	2008	**2009**	**2010**	**2011**
Jan	29300	29200	29200	29000
Feb	26100	26100	26000	26000
Mar	23200	23400	23400	23300
Apr	23200	23100	23100	23000
May	26600	26400	26200	26200
Jun	28400	29200	29200	29300
Jul	29900	30300	30100	29700
Aug	29900	29500	29600	30100
Sep	25500	26100	26100	25900
Oct	23100	23000	22700	22500
Nov	23200	23000	23000	22900
Dec	26900	26600	26900	27100
Total:	**315300**	**315900**	**315500**	**315000**
Estimated Cost:	**$20,494.50**	**$20,533.50**	**$20,507.50**	**$20,475.00**

6.9.13 POB Transport Systems - HVAC Transport Pumps

The POB HVAC transport pumps include four chilled water pumps and three hot water pumps with a combined total 42 horsepower. EQUEST is used to model their energy usage. The results are given in Table 6.24 where the totals are shown in kilowatt-hours.

Table 6.24: POB Transport Pump Energy Consumption

POB Transport Pumps 2008-2011 in KWH				
	2008	**2009**	**2010**	**2011**
Jan	3900	3900	3900	3800
Feb	3400	3400	3400	3400
Mar	3900	4100	4000	4000
Apr	4500	4500	4400	4400
May	5700	5700	5700	5700
Jun	6700	6700	6700	6700
Jul	7400	7400	7400	7500
Aug	7400	7400	7400	7400
Sep	6200	6200	6200	6200
Oct	4300	4300	4200	4200
Nov	3600	3700	3600	3600
Dec	3600	3600	3600	3600
Total:	**60600**	**60900**	**60500**	**60500**
Cost:	**$3,939.00**	**$3,958.50**	**$3,932.50**	**$3,932.50**

6.9.14 POB End Use Systems – Pool Heat Exchanger

The only end use system that will be analyzed for the POB is the hot water convertor for the indoor swimming pool located in POB3. The heat exchanger uses water from the industrial hot water system to heat the pool water to around 100 Fahrenheit. Assuming a thirty degree temperature difference is maintained at maximum demand and an eighty percent efficient boiler system, it consumes approximately 6,250 BTUs per min or 375,000 BTUs per hour to keep the pool warm. If maximum demand is needed for 24 hours, the unit consumes 9,000,000 BTUs per day costing an estimated $54 dollars per day at maximum demand to keep the pool warm.

6.9.15 Cancer Center Generation / Transport Systems

The Cancer center is equipped with one natural draft hot water generator and one screw type air cooled chiller. The chiller was found to have a 1.35 kw/ton efficiency, and the hot water generator is eighty percent efficient. EQUEST was used to model both the chilled and hot water systems. The results in kilowatt-hours can be seen in Tables 6.25 and 6.26.

Table 6.25: Cancer Center Chiller Usage

Cancer Center Chiller Usage				
	2008	**2009**	**2010**	**2011**
Jan	8230	9300	8010	8050
Feb	7110	7920	7060	7040
Mar	12180	12460	12200	12200
Apr	17590	17690	17600	17580
May	28150	28040	28170	28130
Jun	38580	38370	38730	38730
Jul	43540	43270	43650	43580
Aug	43110	42720	43120	43230
Sep	33090	33060	33330	33260
Oct	15810	15920	15800	15740
Nov	9250	9850	9260	9270
Dec	6430	7630	6520	6610
Total:	**263070**	**266230**	**263450**	**263420**
Cost:	**$17,099.55**	**$17,304.95**	**$17,124.25**	**$17,122.30**

Table 6.26: Cancer Center Boiler Usage

Cancer Center Hot Water Generator Usage				
	2008	**2009**	**2010**	**2011**
Jan	211900000	304300000	214200000	212700000
Feb	183600000	260200000	183600000	183300000
Mar	137800000	200200000	134700000	134300000
Apr	107600000	164300000	107200000	108600000
May	89400000	147200000	93500000	92100000
Jun	73400000	117000000	71100000	71000000
Jul	69600000	119300000	71800000	73600000
Aug	68500000	116700000	69500000	68000000
Sep	80000000	128500000	78700000	78400000
Oct	115100000	179100000	118500000	118500000
Nov	155100000	224500000	154100000	153900000
Dec	194200000	278900000	194300000	194400000
Total:	**1486200000**	**2240200000**	**1491200000**	**1488800000**
Cost:	**$8,917.20**	**$13,441.20**	**$8,947.20**	**$8,932.80**

The cancer center has two chilled water pumps and two hot water pumps to supply chilled and hot water to the building. EQUEST models produce an average pumping usage of 26,612 kilowatt-hours per year costing an estimated $1,700 dollars per year. There are only two air handlers in the Cancer Center. EQUEST estimates the average ventilation fan energy usage at 61,332 kilowatt-hours per year costing approximately $4,000 dollars per year.

6.10 Building Management System Analysis

The building management system was analyzed primarily for its functionality. The main focus of this section is on the HVAC control system contained in the Main Hospital. One problem with the control system pertains to the sequence of operations for the main hospital air handlers. The sequence of operations for the chilled water and preheat control valves are as follows:

> "When the supply air temperature falls below set-point, the chilled water valve shall modulate closed and the hot water valve shall modulate open as required to maintain supply air temperature at set-point. When the supply air temperature rises above the set-point, the hot water valve shall modulate closed and the chilled water valve shall modulate open to maintain the supply air temperature at set-point."

The problem with this sequence is that it calls on the hot water valve to modulate open to raise the discharge air temperature when it falls below set point. This sequence, if carried out correctly, will open the hot water valve even for small changes in the discharge air temperature. This is a very wasteful method to control discharge air temperature because the air handlers at RMC are almost always in cooling mode. Discharge air temperature could be more efficiently controlled by modulating the chilled water coil alone to maintain discharge air temperature. For example, if the discharge air temperature falls below set point, the chilled water coil should modulate closed appropriately to reduce flow through the chill water coil thereby raising the discharge air temperature. This method prevents the hot water valve from being opened unnecessarily. The hot water valves and coils placed in the air handlers are primarily there for cooling coil freeze protection. Almost all heating needs are handled by the reheat coils located in the air terminal boxes.

Upon further inspection it is found that the air handler preheat hot water control valves and the chilled water control valves are set up on the same PID control loop with no dead band between them. This setup allows for both the chilled and hot water coils to be open at the same time causing simultaneous heating and cooling within the air handler. Simultaneous heating and cooling within the air handler is a very wasteful practice and it occurs regularly at RMC. Figure 6.13 on the next page is a screen shot taken from the GUI at RMC. It illustrates simultaneous heating and cooling.

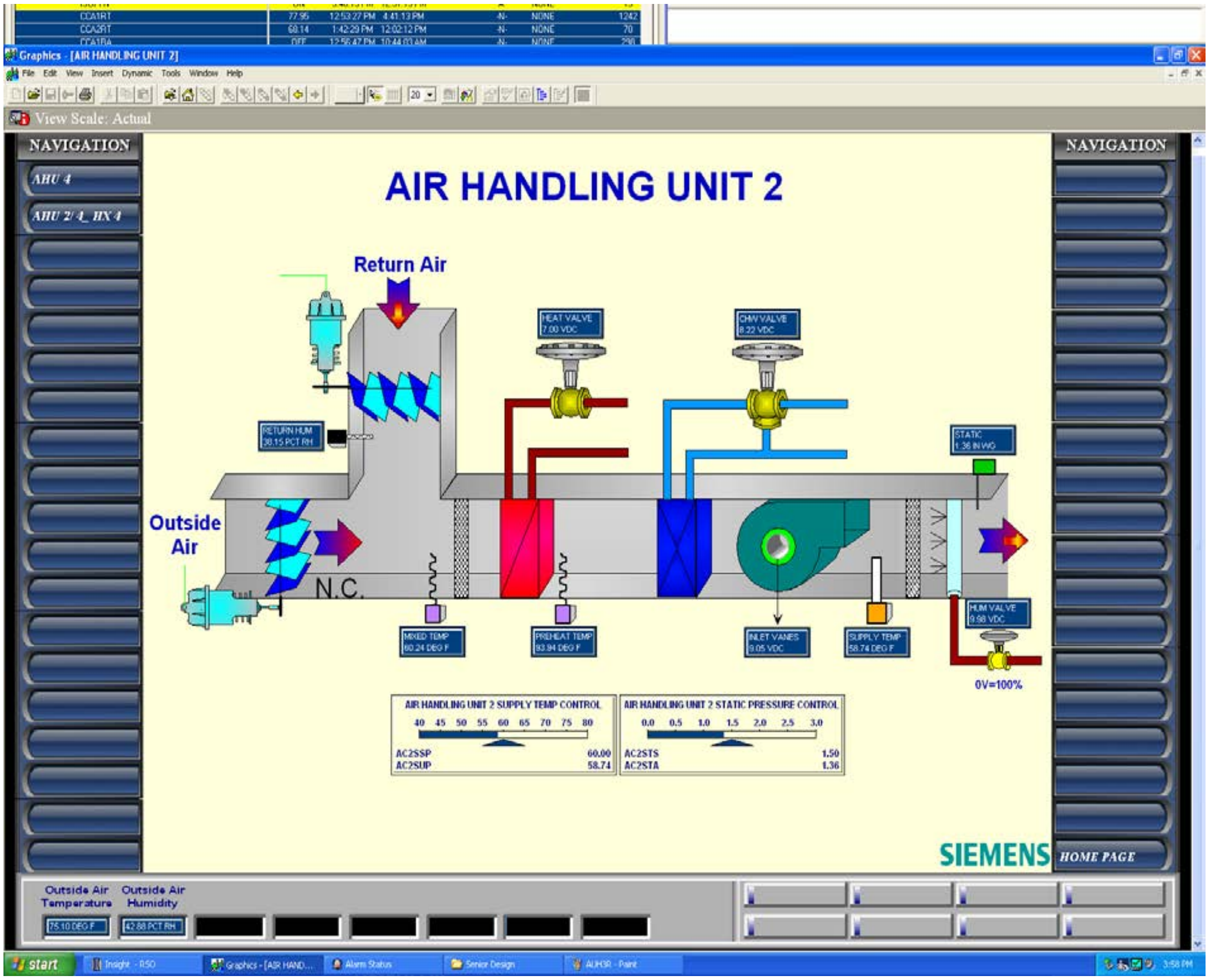


Figure 6.13: Simultaneous Heating and Cooling Example

As can be seen in Figure 6.13 the mixed air temperature is 60 degrees Fahrenheit. The temperature after the preheat coil is approximately 94 degrees Fahrenheit. The discharge air temperature is 58 degrees Fahrenheit with a set point of 60 degrees Fahrenheit. The system heats the air up 34 degrees only to cool it back to 58 degrees Fahrenheit. If this unit were operating properly, the cooling coil and heating coil would be closed because the mixed air temperature will satisfy set point.

All control valves and actuators were checked during the building survey. It was found that the outside air dampers were not functioning correctly on many of the air handler units with outside air control. The main reason for the malfunctioning dampers is the systems controls the damper setting based on the air flow reading it receives from the air flow sensors. The pitot tube type air flow sensors in place at RMC are inaccurate and in most cases not functioning at all. There is practically no outside air control at RMC. Also many other sensors like temperature and static pressure sensors placed within the air handlers to feedback information to the control system are malfunctioning. A complete list of these malfunctioning sensors can be found in Appendix E. Overall, the control system for the main hospital at RMC does maintain satisfactory indoor air quality, but it does so with a brute force tactics. Changes made to the sequences of operation and updates to some hardware and software would make a more energy efficient system.

6.11 Water Usage Analysis

Water usage was tracked for the year 2010. Table 6.27 shows 2010 monthly water consumption at RMC. This includes all water consumption industrial and domestic.

Table 6.27: 2010 Water Consumption

2010 Water Usage	
	Gallons Used
Jan	905000
Feb	810000
Mar	715000
Apr	815000
May	1105000
Jun	1640000
Jul	1175000
Aug	1325000
Sep	1495000
Oct	980000
Nov	885000
Dec	775000
Total:	**12625000**

Chapter 7: Baseline Energy Models

7.1 Introduction

With analysis complete on all energy using facilities and equipment at RMC, baseline energy models were constructed. The baseline energy models encompass the whole facility and portray a breakdown of the energy use from all the major contributors. The first baseline energy models represent energy usage from the year 2008 through the year 2010. No significant energy reduction strategies discussed in this thesis were applied during those years. All three years are compared to ensure a consistent baseline energy model. The models serves as the baseline from which all applied energy cost reduction strategies and future recommendations are measured. The RMC campus consumes a yearly average of 10.5 million kilowatt hours of electricity and 61.7 billion BTU's of natural gas at a total average utilities cost of 1.1 million dollars per year.

7.2 Weather Effects

When comparing multiple yearly energy models, it is important to look at the weather conditions for those years because the largest contributor to energy consumption in many facilities is the HVAC system. Figure 7.1 below is an outdoor temperature plot for years 2008-2011 for Alexander City Alabama.

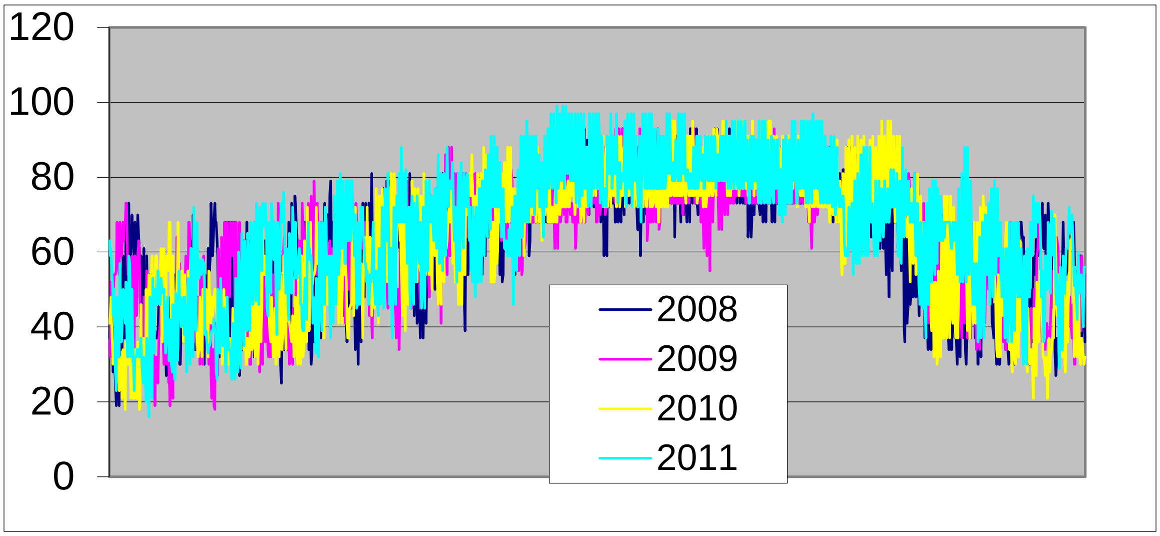


Figure 7.1: Alexander City, AL Air Temperature 2008-2011

It can be seen from the figure that the weather is fairly consistent over the four years of analysis.

Table 7.1: Weather Stats

2008-2011 Weather Statsfor Alexander City Alabama				
	2008	2009	2010	2011
Average Temperature	63	63.1	62	65
90 degree Days	32	36	42	77
80 degree Days	106	104	145	145
Freezing Days	46	41	49	31

Weather analysis shows that the average temperature is slightly hotter for 2011. 2011 has a total of 77 days at or exceeding 90 degrees Fahrenheit. That is 35 more days at peak load than in previous years. Prolonged heat waves and cold spells translate into greater

HVAC energy costs. 2010 was the coldest year with a total of 49 days at or below freezing.

7.3 Actual Energy Usage

The actual electricity and gas usage was calculated from RMC energy bills. The actual energy usage from the bills represents all the electricity and gas that was paid for over the four year analysis. The actual electricity usage for all facilities for each year plotted monthly is represented in Figures 7.2 through 7.5.

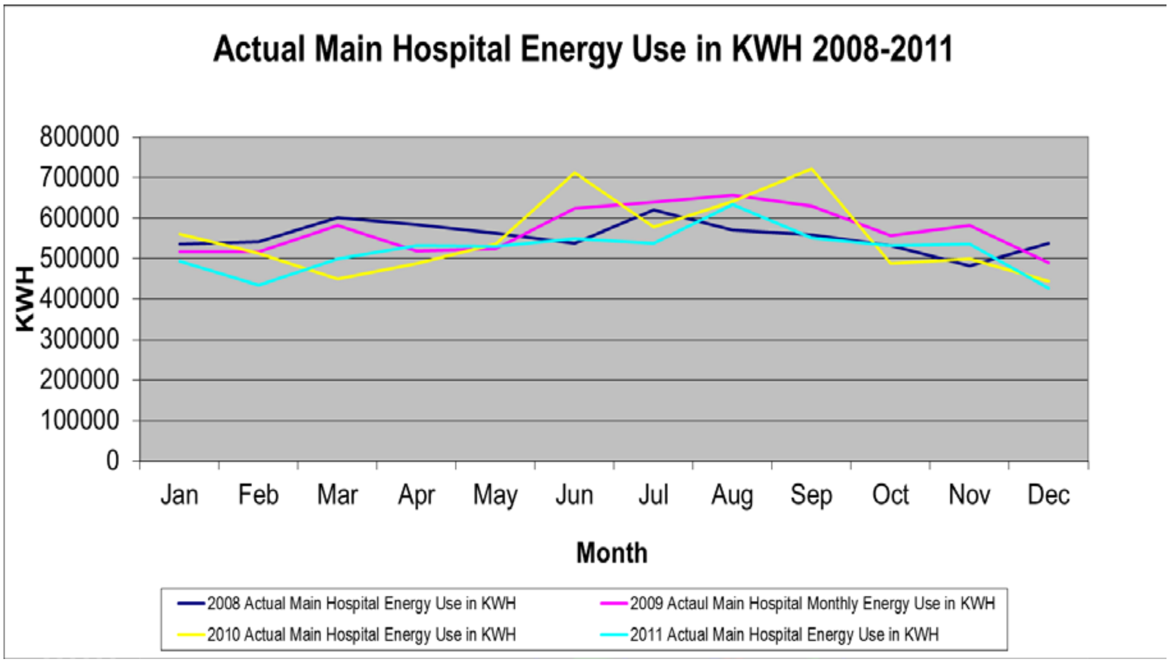


Figure 7.2: Main Hospital Electricity Consumption 2008-2011

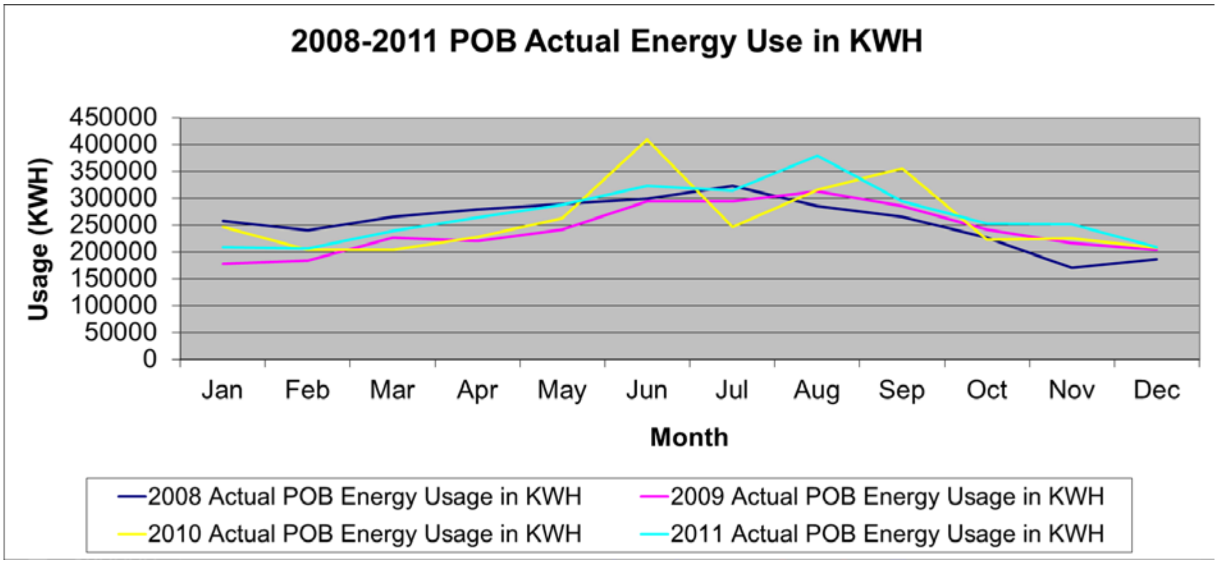


Figure 7.3: POB Electricity Consumption 2008-2011

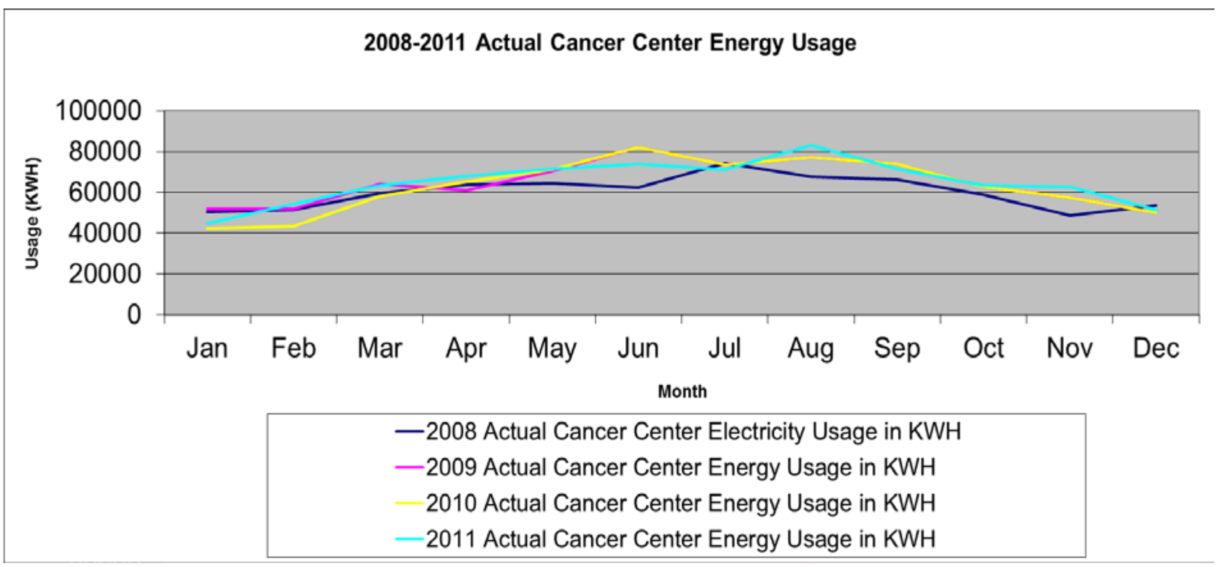


Figure 7.4: Cancer Center Electricity Consumption 2008-2011

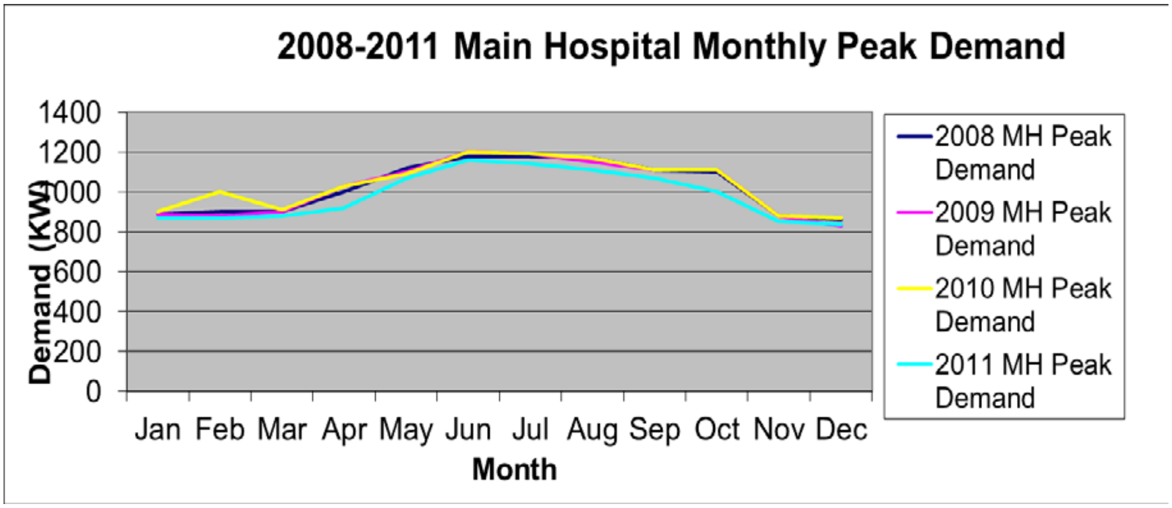


Figure 7.5: Main Hospital Peak Demand 2008-2011

Actual natural gas usage is represented in Figures 7.6 through 7.8. The POB and Cancer Center are lumped together because the data came from a combination meter.

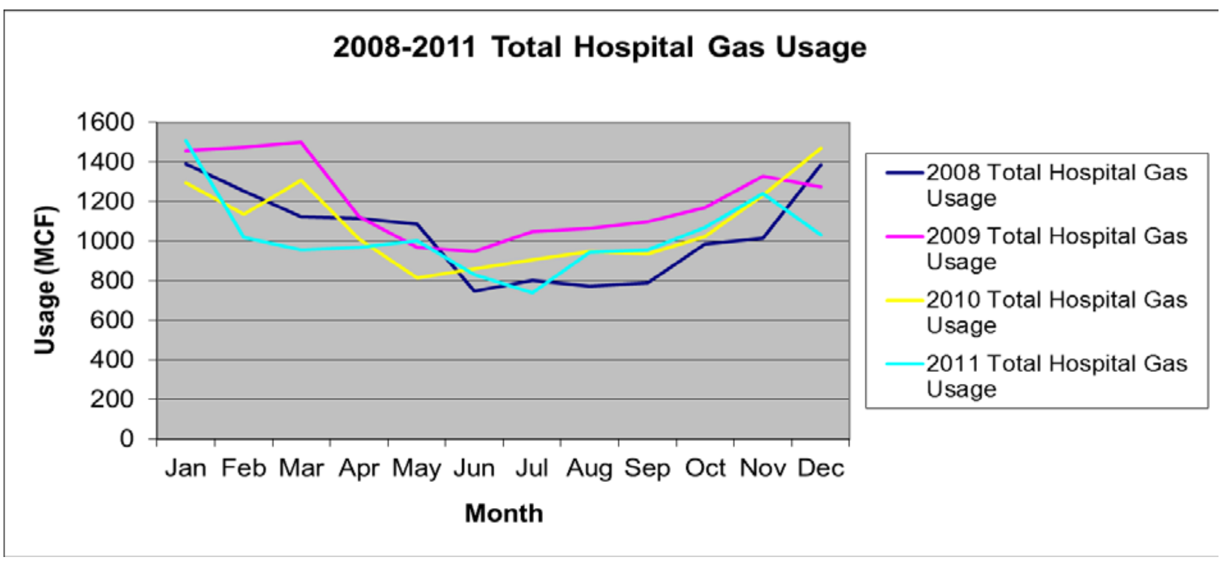


Figure 7.6: Main Hospital Gas Consumption 2008-2011

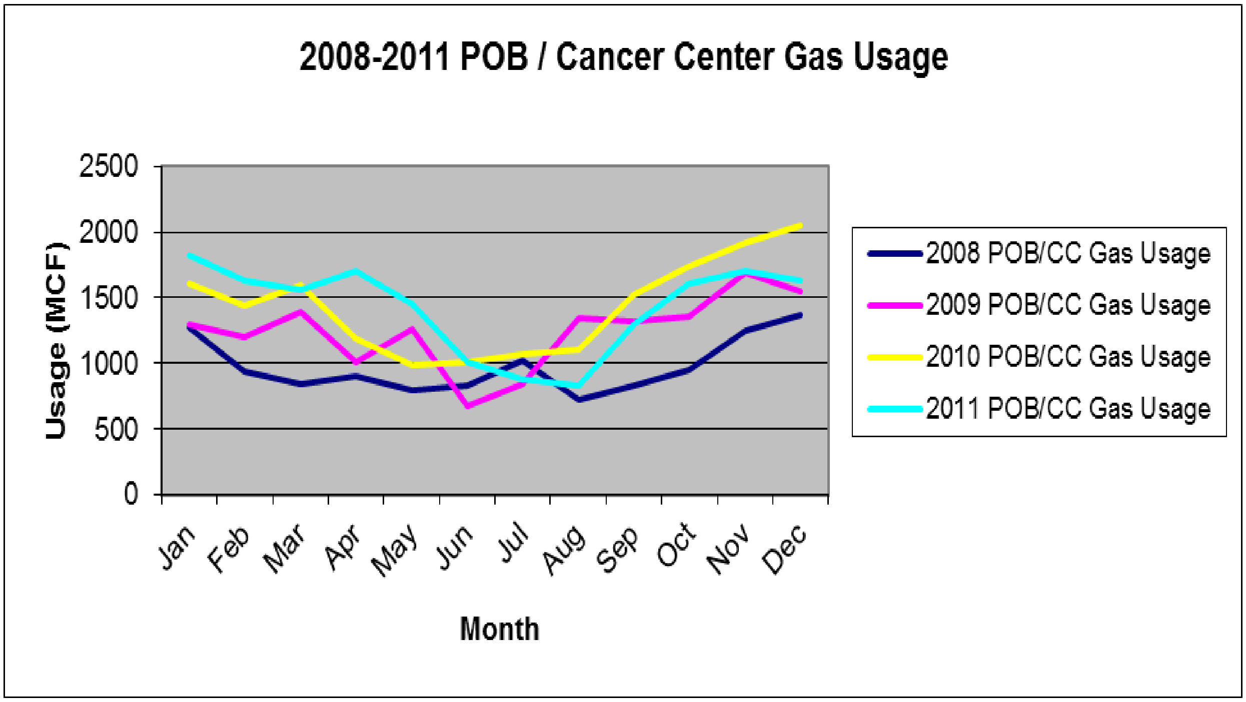


Figure 7.7 POB/Cancer Center Gas Consumption 2008-2011

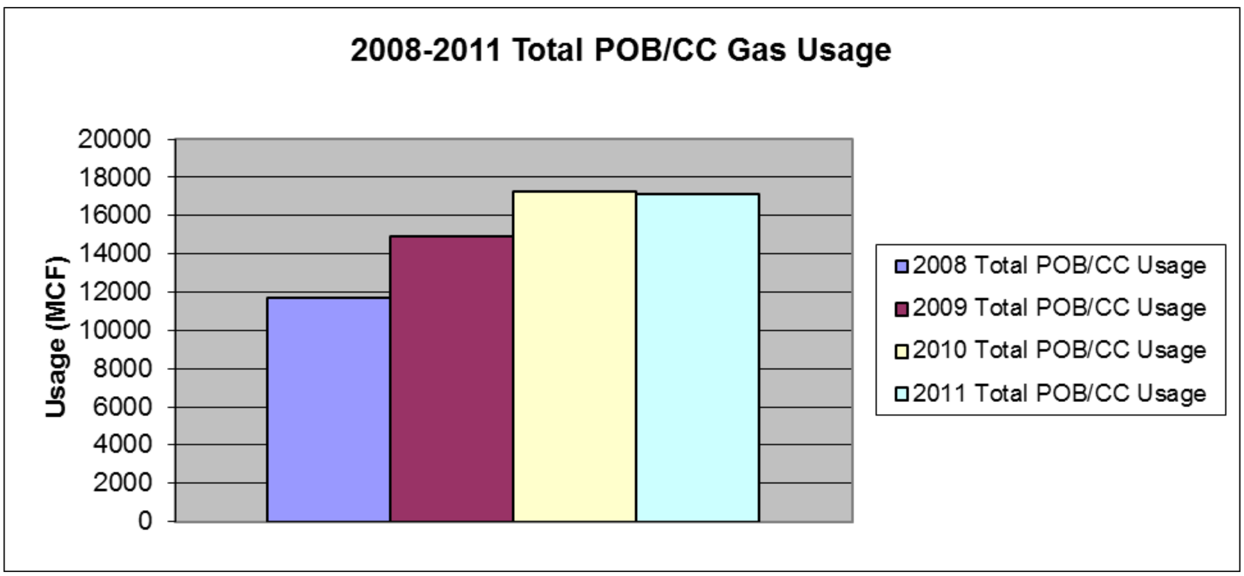


Figure 7.8 POB/Cancer Center Total Yearly Gas Consumption 2008-2011

7.4 Actual vs. Model Comparison

EQUEST models are one of the main energy analysis tools used to establish the baseline energy model for RMC. In this section the EQUEST models are compared to the actual usage to verify the model's accuracy. Figures 7.9 through 7.12 represent a comparison of the main hospital EQUEST electricity model to the actual electricity usage for the years 2008-2011.

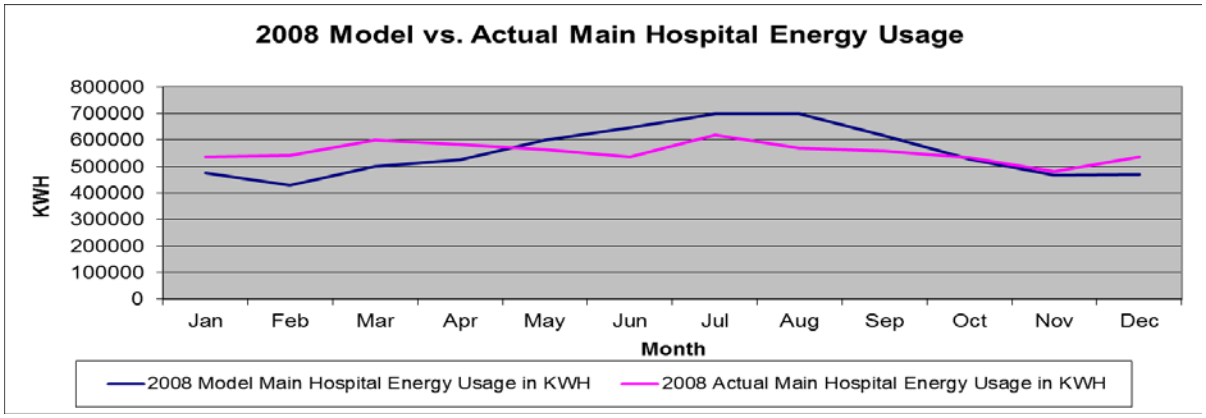


Figure 7.9: 2008 Main Hospital Model vs. Actual Electricity Usage

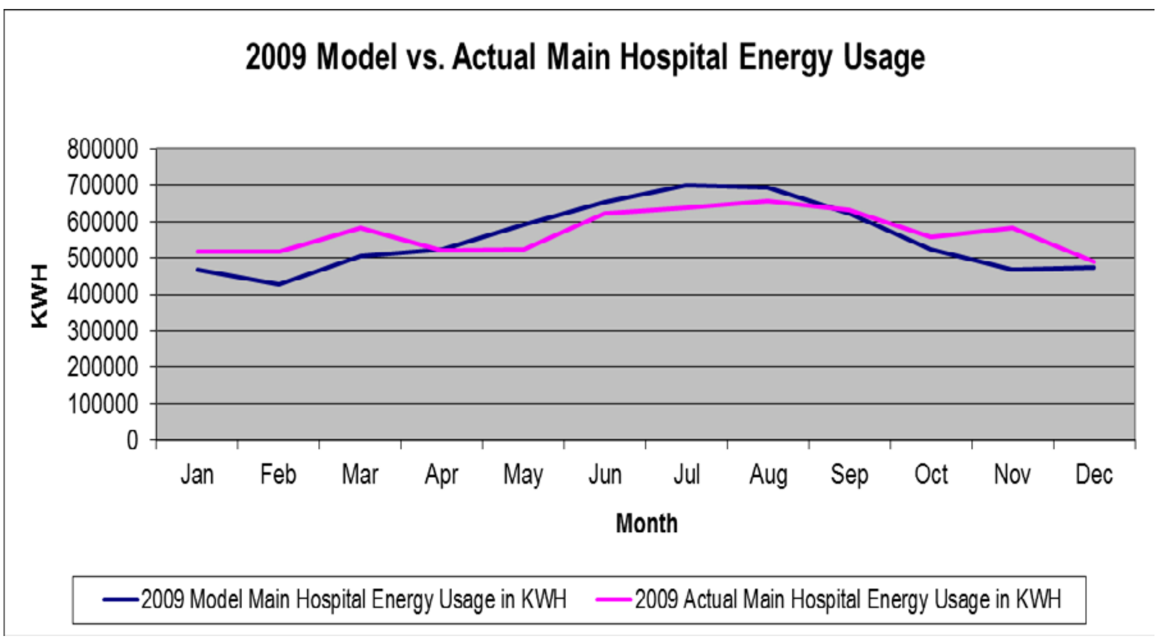


Figure 7.10: 2009 Main Hospital Model vs. Actual Electricity Usage

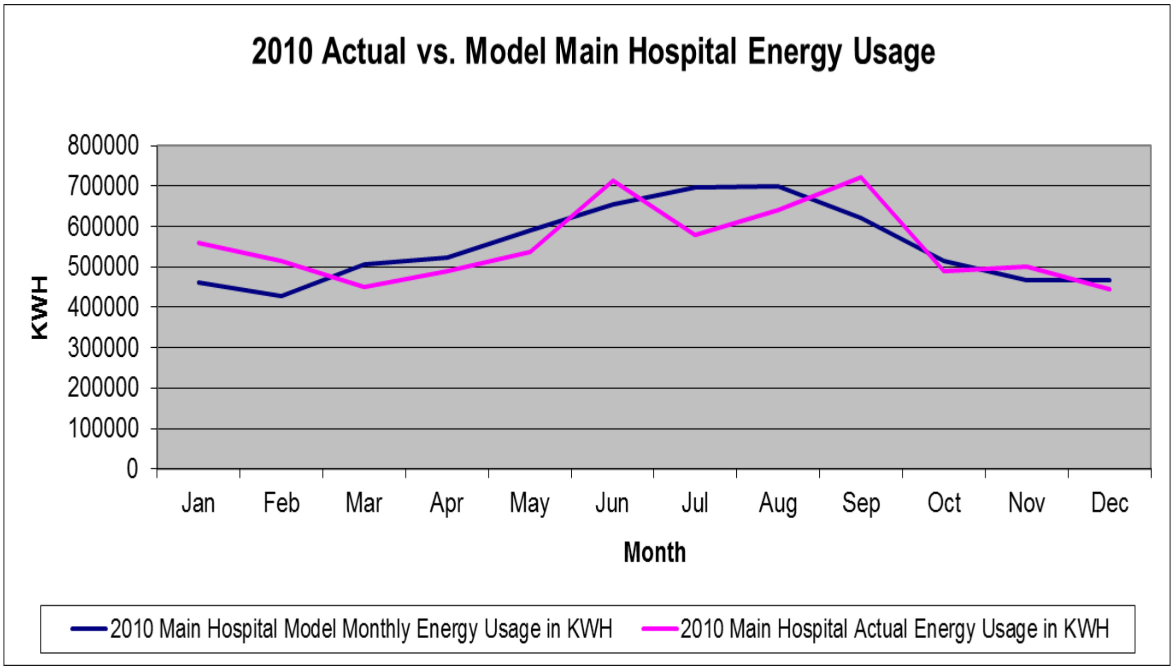


Figure 7.11: 2010 Main Hospital Model vs. Actual Electricity Usage

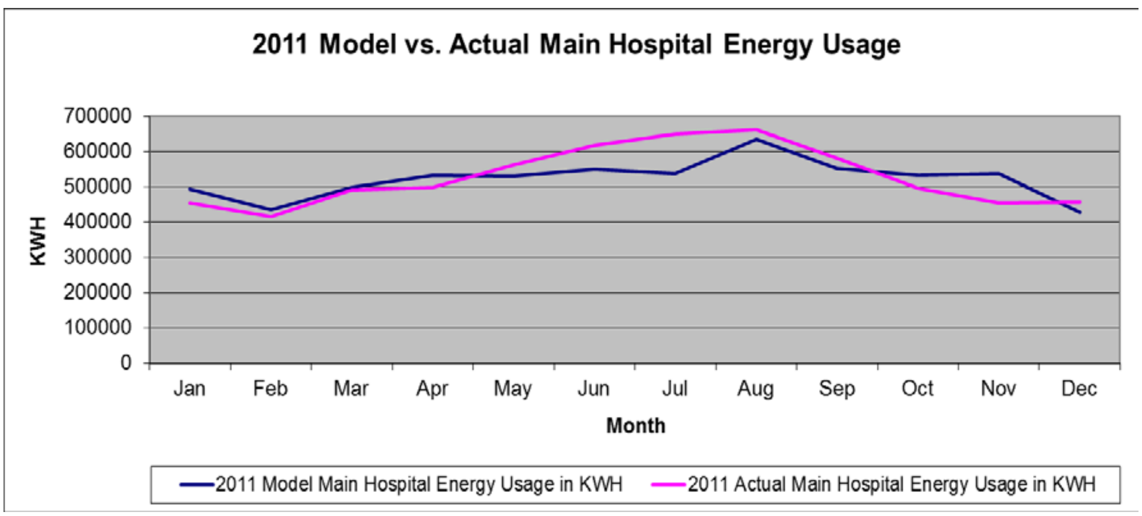


Figure 7.12 2011 Main Hospital Model vs. Actual Electricity Usage

The electricity usage comparisons show that the EQUEST model tends in most cases to over predict summer peak electricity usage and under predict shoulder and winter month electricity usage. The average standard deviation of the model from the actual average electricity usage over the years 2008-2011 was found to be 54,520 kilowatt hours per month. The average standard error of the model electricity usage from the actual electricity usage over years 2008-2011 was found to be 4,543 kilowatt-hours per month.

The results vs. actual results are compared for the POB and Cancer Center and are represented in Figures 7.13 and 7.14.

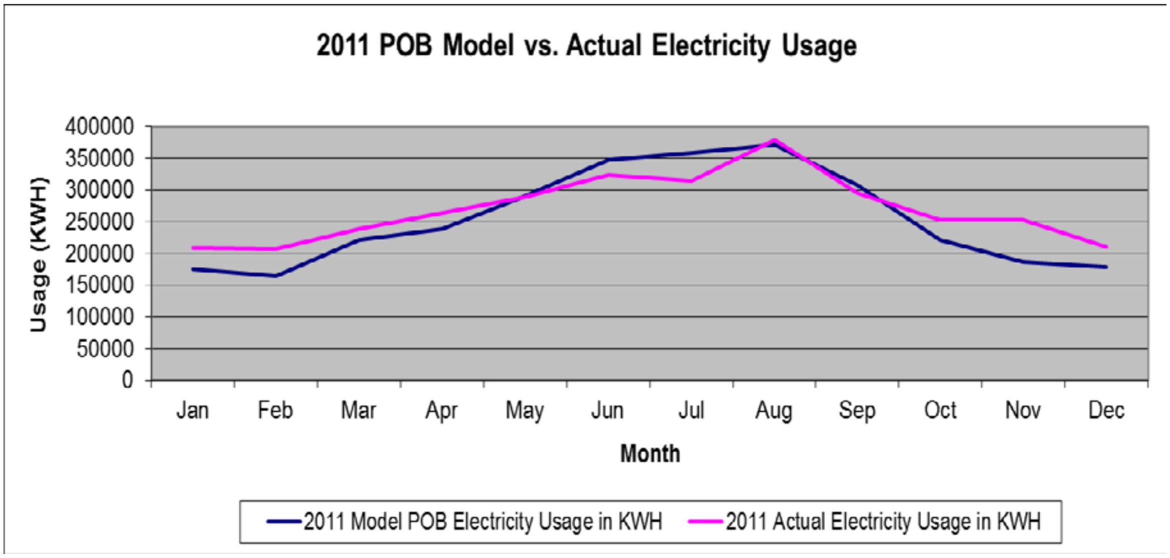


Figure 7.13: 2011 POB Model vs. Actual Electricity Usage

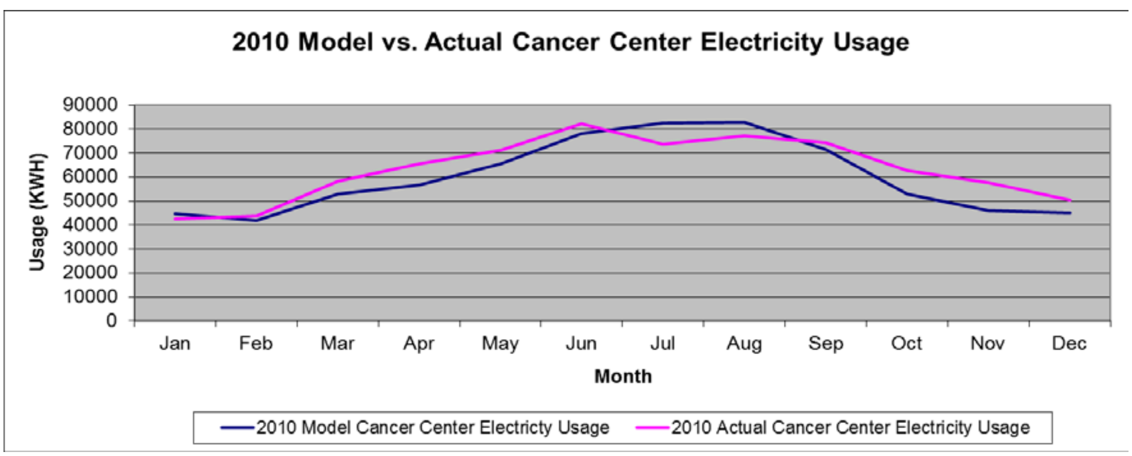


Figure 7.14: 2010 POB Model vs. Actual Electricity Usage

EQUEST models for electricity use in the POB and Cancer Center tend to over predict the peak months and under predict the shoulder months as is noted above for the main hospital. The average standard deviation of the model from the average actual usage for the POB was found to be 35,637 kilowatt hours per month and the average standard error was found to be 2,969 kilowatt-hours per month. The average standard error for the Cancer Center model was found to be 623 kilowatt-hours per month.

The natural gas EQUEST vs. actual usage models are represented by Figures 7.15 and 7.16.

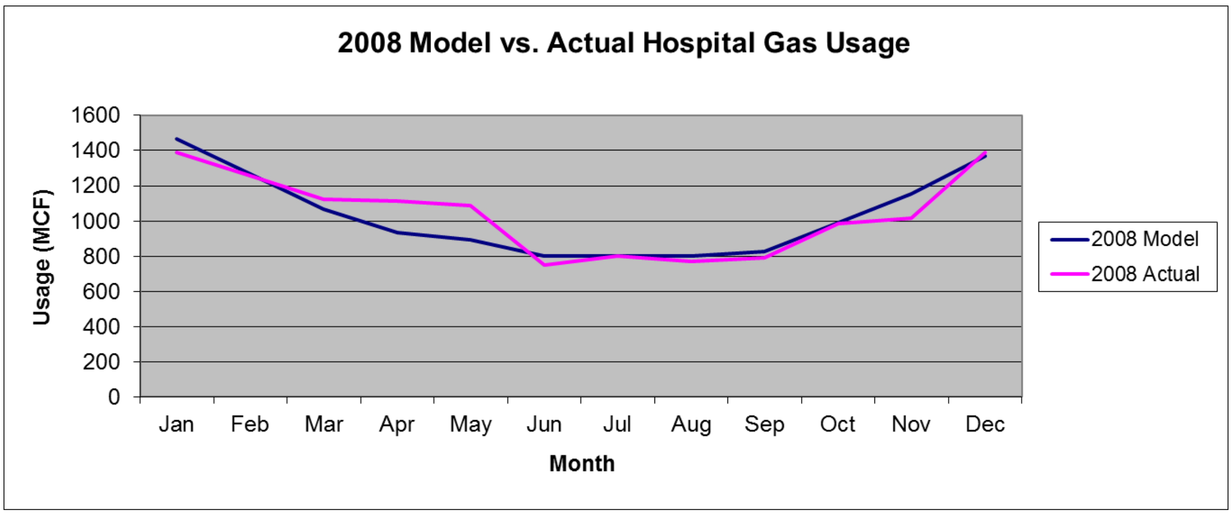


Figure 7.15: 2008 Main Hospital Model vs. Actual Natural Gas Usage

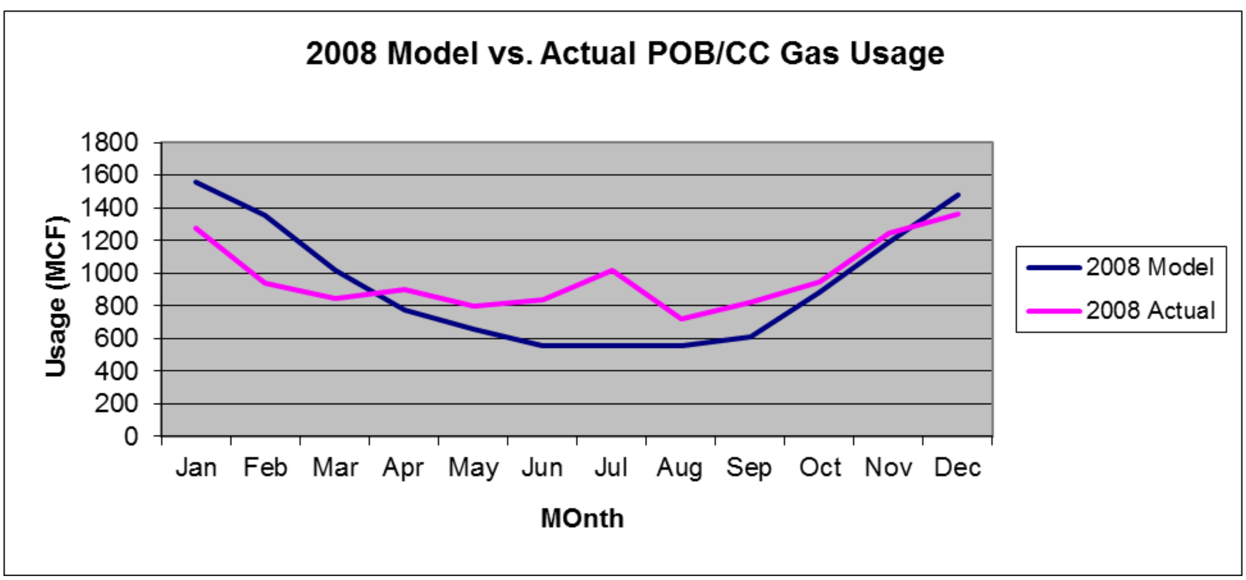


Figure 7.16: 2008 POB/ Cancer Center Model vs. Actual Natural Gas Usage

The EQUEST natural gas usage model tends to over predict peak usage and under predict off peak usage. The average standard deviation of the model from the average usage for the main hospital was found to be 114 MCF month with an average standard error of 9.5 MCF per month. The average standard deviation of the model from the average POB/Cancer Center usage was found to be 363 MCF per month with an average standard error of 30 MCF per month.

7.5 Energy Usage Break Up

With the models validated, a baseline electricity and natural gas usage break up was established. The usage form each energy cost center for the years 2008-2010 are averaged together to establish a baseline energy profile for RMC. The RMC campus wide energy use break up is shown in Figure 7.17. The RMC campus consumed an average of 61.7 billion BTUs per year form 2008-2011. That is about 214,000 BTU per square foot per year. The campus average electricity usage is 10,507,081 kwh per year or 36.35 kwh per square foot per year with a maximum demand of 8.5 watts per square foot. The main hospital consumes 64 percent of the total electricity usage with the POB accounting for 29 percent and the cancer center accounting for the remaining 7 percent. The average natural gas usage is 25.5 billion BTUs per year.

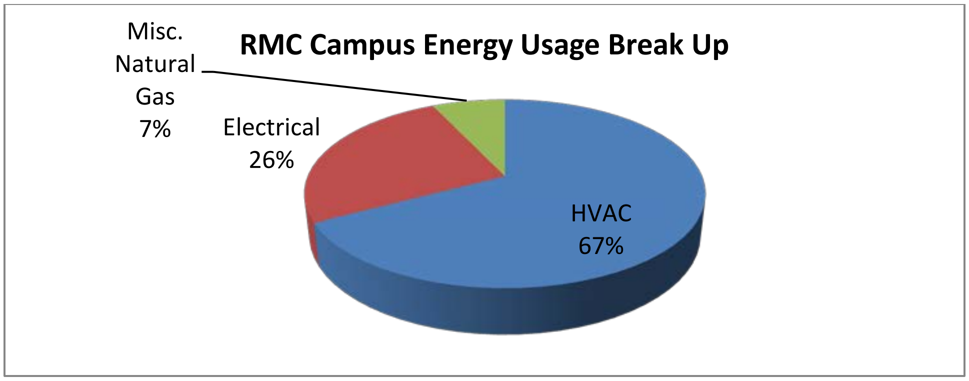


Figure 7.17: RMC Campus Energy Usage Break Up

The energy usage break up for the main hospital is shown in Figure 7.18.

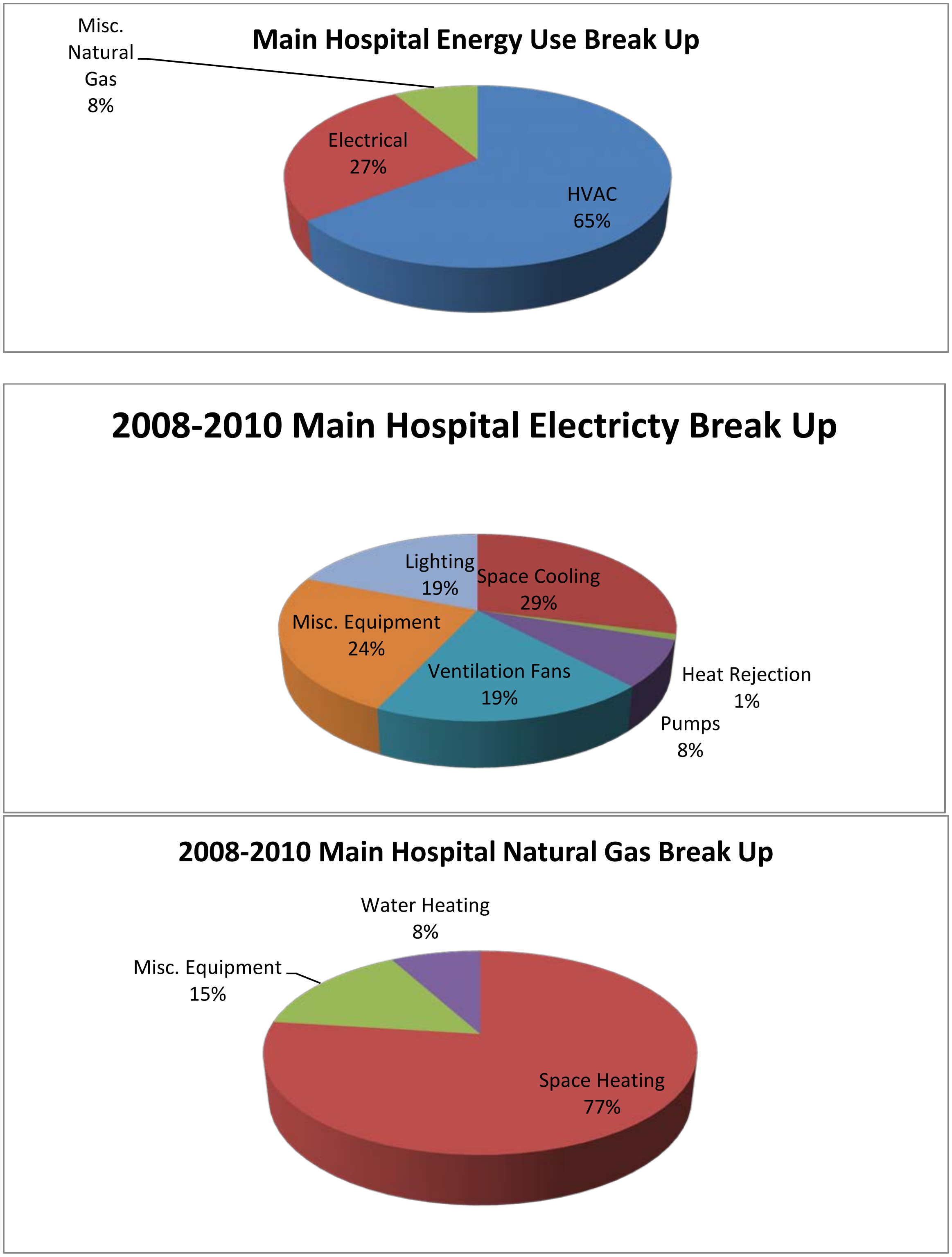
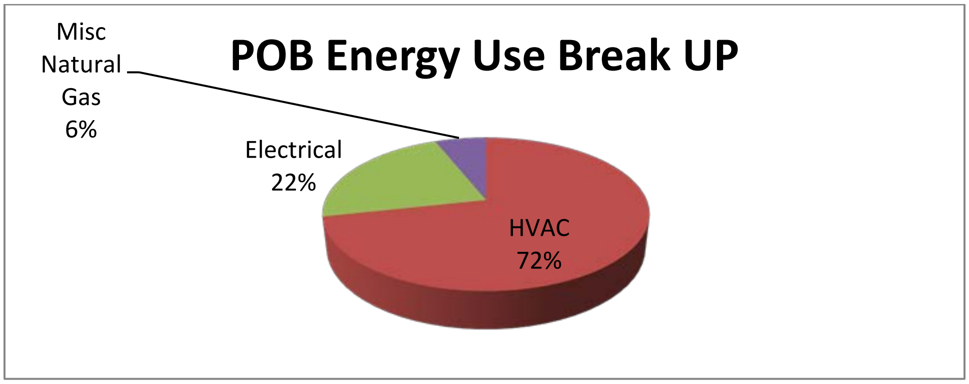
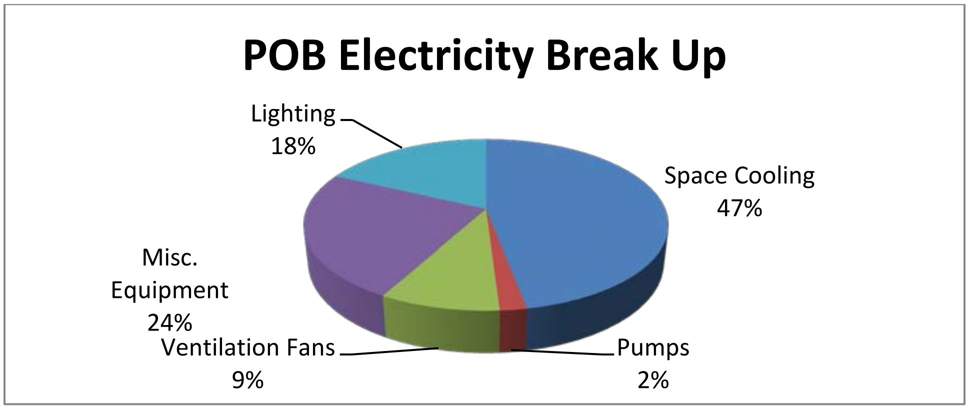


Figure 7.18 Main Hospital Energy Break Up

The main hospital consumed an average of 6,713,485 kilowatt-hours per year from 2008-2011. That is approximately 45 kwh per square foot per year with a maximum demand of 1200 kilowatts or 8 watts per square foot. It also consumed an average of 13.9 billion BTU's per year of natural gas. The POB energy break up is shown in Figure 7.19.





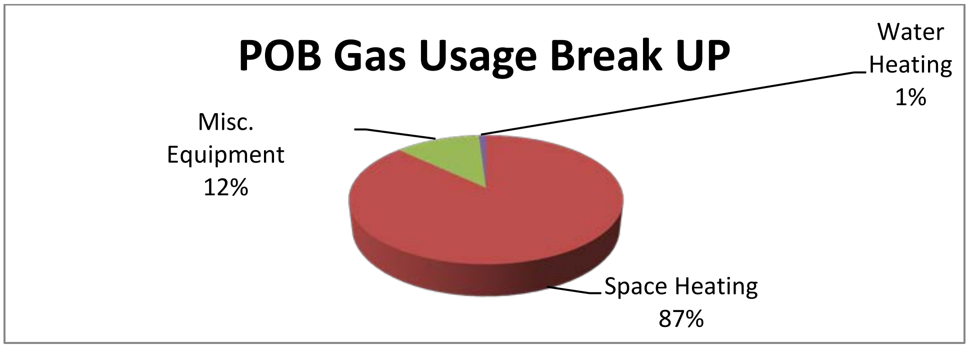
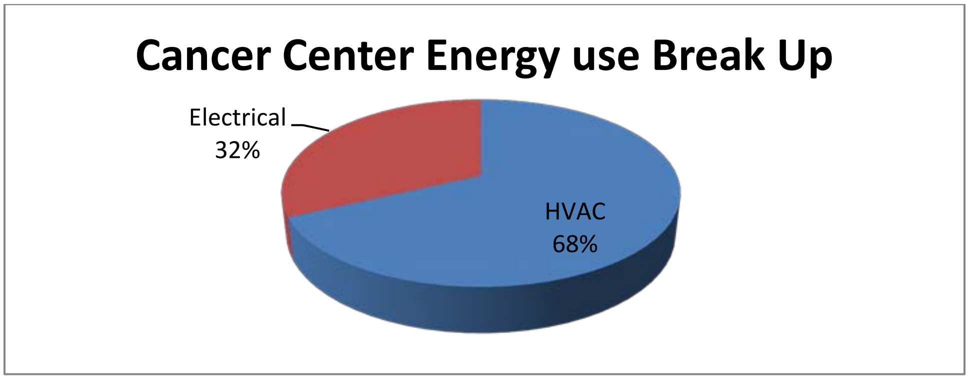
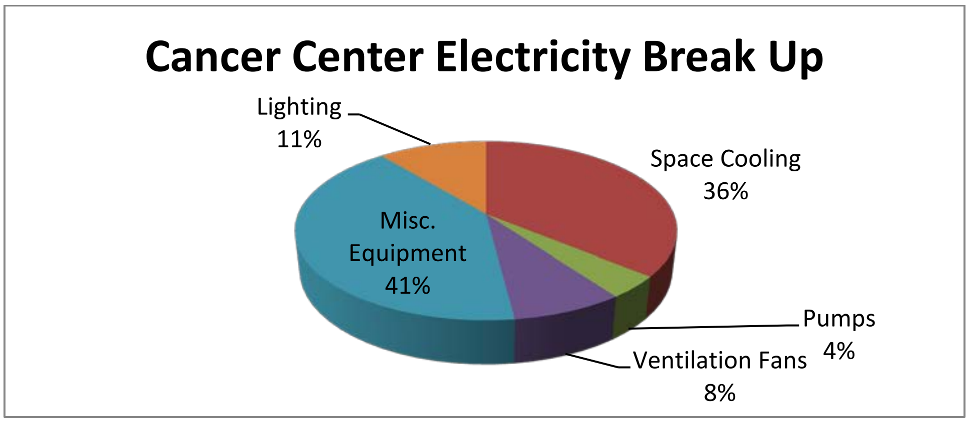


Figure 7.19 POB Energy Break Up

The POB consumed an average of 3,041,001 kwh of electricity per year for 2008-2011. The maximum demand is roughly 1080 kilowatts or 8.6 watts per square foot. The POBs consumed an average of 10.6 billion BTUs per year of natural gas. The cancer center energy break up is shown in Figure 7.20.





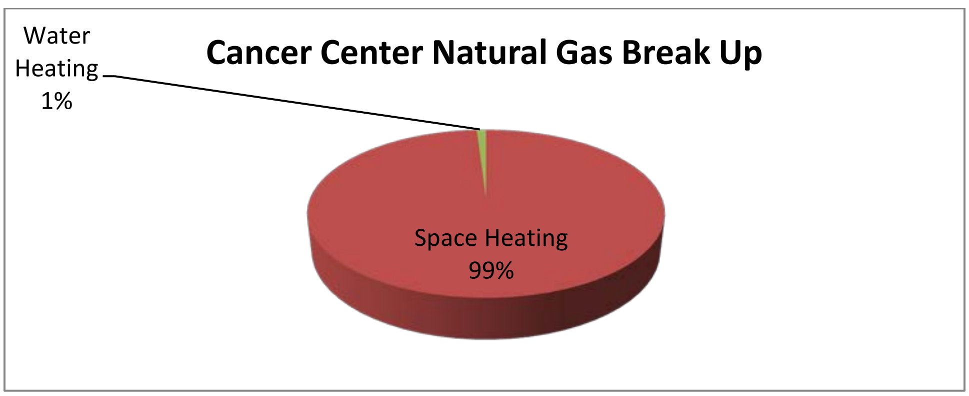


Figure 7.20: Cancer Center Energy Break Up

The Cancer Center consumed an average of 752,595 kilowatt-hours of electricity per year with a maximum demand of 224 kilowatts or 14 watts per square feet. The Cancer Center also consumed an average of 1.5 billion BTUs per year of natural gas.

Chapter 8: Energy Cost Reduction Strategies

8.1 Introduction

In this chapter, energy cost reduction strategies for each RMC energy cost center will be categorized into two categories. The two categories are non-capital intensive cost reduction strategies and capital intensive strategies. Each recommendation will include both engineering and economic analysis.

8.2 Non Capital Intensive Energy Cost Reduction Strategies

Non capital intensive energy cost reduction refers to strategies that can be implemented for under $5,000 dollars. There are several cost saving strategies highlighted in this section for both the electrical and HVAC energy cost centers for each facility contained within the RMC campus.

8.2.1 Electrical System Cost Reduction – Combined Electricity Metering

RMC currently purchases electricity from the Alexander City Utilities System (ACUS) which is a municipally operated utilities provider. ACUS has several classifications for electricity billing rates. RMC currently pays the standard industrial rate which has a tiered structure for electricity billing that will be described later. Significant energy cost savings could be achieved if RMC could change to the large industrial rate.

8.2.1.1 Combined Electricity Metering Analysis

RMC currently pays the standard industrial rate for all electricity consumed on campus. This rate is assigned to facilities that have a maximum fifteen minute interval integrated electric demand of 50 kilowatts or more. The standard industrial rate is applied as follows:

Electrical Demand Rate:

$5.25 per kilowatt

Electrical Usage Rate:

$0.0680 per kilowatt-hour for first 20,000 kilowatt-hours plus

$0.0630 per kilowatt-hour for next 80,000 kilowatt-hours plus

$0.0590 per kilowatt-hour for kilowatt-hours consumed over 100,000 kilowatt-hours

Each facility has its own individual electricity meter for a total of five electricity meters monitoring usage on five separate buildings. The standard industrial rate is applied to each facility and RMC is billed for each separate meter. The large industrial rate offered by ACUS is significantly cheaper than the standard industrial rate. In order to qualify for the industrial rate, the facility has to have a maximum integrated demand of 1200 kilovolt-amperes or larger. The large industrial rate would be applied as follows:

Electrical Demand Rate:

$5.50 per KVA

Electrical Usage Rate:

$0.0620 per kilowatt hour for first 250 kilowatt hours per kilovolt-ampere

$0.0470 per kilowatt hour for all over 250 kilowatt hours per kilovolt-ampere

To qualify for the large industrial rate, it is suggested that RMC negotiate an agreement with ACUS to combine all the individual electricity meters into one meter that would monitor the campus as a whole. All individual electricity meters are contained with a contiguous campus operated by the same organization and should be combined into one electricity meter monitoring power consumption to the entire campus. If all individual meters were combined, it would push the minimum RMC electrical demand to an estimated 1600 kilowatts which is well above the large industrial requirement. Implementing the one meter concept would cut electricity usage costs because of the lower effective rate. Demand costs would also theoretically be reduced. Currently the maximum demand at each meter serving each facility is recorded monthly. The maximum demand is billed to RMC for each individual meter which means the maximum demand charges are added together. That method doesn't effectively capture the true maximum demand because more than likely the peak at each facility occurs at a different time during the month. One meter would reflect the true monthly peak demand because individual facility demand peaks would offset the other facilities not being at peak at the same instant. Hence, the true peak is smaller than the individual facility peak sum currently used. Combining meters could be accomplished either manually of electronically. A new meter could physically be installed at the substation that serves RMC, or the one meter concept could be electronically applied to ACUS's billing software for RMC. The capital cost to implement should be minimal compared to the savings.

8.2.1.2 Combine Electricity Metering Savings

To calculate the theoretical savings of combining all electricity meters into one, all RMC electricity usage from each individual meter was added according to past electricity consumption. The cost was then calculated with the large industrial rate applied. The savings is simply the difference between the standard and large industrial costs. Table 8.1 contains the results of a calculation performed for the year 2010. Demand savings could not be exactly accounted for because of the lack of hourly demand data. For this analysis it is simply assumed that demand is reduced by ten percent.

Table 8.1: Utility Savings

2010 RMC Monthly Electricity Usage Industrial Rate						
	Main Hospital Usage	**MH Demand**	**Combined POB Usage**	**Combined POB Demand**	**Cancer Center Usage**	**Cancer Center Demand**
January	$33,575	$4,725	$10,583	$3,623	$2,770	$992
February	$30,796	$5,250	$10,176	$5,250	$2,845	$992
March	$27,084	$4,778	$13,592	$4,095	$3,763	$851
April	$29,310	$5,408	$14,813	$4,253	$4,221	$903
May	$32,109	$5,723	$17,480	$5,040	$4,577	$1,019
June	$42,561	$6,300	$21,056	$5,565	$5,280	$1,155
July	$34,663	$6,248	$22,094	$5,355	$4,741	$1,113
August	$38,341	$6,143	$22,053	$5,250	$4,958	$1,108
September	$43,041	$5,828	$18,749	$5,250	$4,770	$1,076
October	$29,296	$5,828	$13,598	$5,040	$4,053	$1,029
November	$30,002	$4,620	$11,456	$3,570	$3,720	$803
December	$26,724	$4,568	$10,931	$3,098	$3,269	$772
Year Usage Total:	$397,502		$186,581		$48,967	
Year Demand Total		$65,419		$55,389		$11,813
Year Total:	$462,921		$241,970		$60,780	
Campus Usage Total	**$765,671**					
Campus Demand Total	**$132,621**					
Campus Total	**$898,292**					
2010 RMC Monthly Electricity Usage Cost Large Industrial Rate						
	Campus Usage	**Campus Demand**				
January	$48,300	$12,231				
February	$46,018	$15,049				
March	$42,198	$12,733				
April	$46,152	$13,833				
May	$51,423	$15,428				
June	$68,219	$17,050				
July	$53,611	$16,651				
August	$59,775	$16,369				
September	$64,928	$15,916				
October	$47,022	$15,579				
November	$44,847	$11,777				
December	$40,555	$11,048				
Year Usage Total:	**$613,048**					
Year Demand Total		**$156,298**				
Year Total:	**$769,346**					
Savings	**$128,946**					

The savings analysis resulted in a possible savings of $128,946 if the one meter concept were applied at RMC in 2010. The average annual savings for three past years of data is $131,527. It is difficult to determine capital cost for this project because it is unknown what stipulations ACUS may require to implement the one meter plan, however as stated earlier the capital cost should be significantly lower than the possible savings. For this analysis it is assumed the capital cost will be limited to a new electronic meter installed at the proper location at a cost of the maximum $5,000 to fall under non capital intensive cost reduction.

8.2.2 Main Hospital HVAC Cost Reduction

8.2.2.1 Generation System Cost Reduction

8.2.2.1.1 Chilled Water Supply Temperature Reset

8.2.2.1.2 Chilled Water Supply Temperature Reset Analysis

The design chilled water temperature for RMC is 42 degrees Fahrenheit. It remains constant year round despite different weather conditions and chiller heat loads. The main purpose for this low chilled water temperature is to meet peak demand requirements, and to provide cold enough chilled water to meet dehumidification standards. RMC requires that the indoor relative humidity is to not exceed 60% due to medical indoor air quality regulations set by ASHRAE. The indoor temperature is maintained within the comfort zone of 65 to 75 degrees Fahrenheit year round. The chilled water temperature can only be raised if the indoor relative humidity and temperature requirement can still be met. Because of the climate where RMC is located, oftentimes dehumidification is needed. Dehumidification requires that all incoming air be cooled below the dew point to condense any unwanted moisture. Generally the air must be cooled to at least 55 degrees Fahrenheit in order to properly dehumidify. The low discharge air temperature can only be met with a low chilled water temperature. However, if outdoor weather conditions are within a certain range then dehumidification is not required. Data collected at RMC shows if outdoor wet bulb temperature is at or below 65 degrees Fahrenheit then the building heat load is at or below sixty percent of the design chiller capacity. Any dehumidification needed could be accomplished with a higher chilled water supply temperature. It can be seen on a psychrometric chart that if the wet bulb temperature is at or below 50 degrees Fahrenheit then no dehumidification is required. If no dehumidification is required, then chilled water temperature can be raised substantially.

A study on Alexander City weather data revealed that there is an average total of 2,700 hours per year when the wet bulb temperature is below 50 degrees Fahrenheit and an average of 5,300 hours per year that the wet bulb temperature is at or below 65 degrees Fahrenheit. Chilled water could be raised to 50 degrees Fahrenheit for an average of 2,700 hours per year and could be raised to 45 degrees Fahrenheit for another 2,600 hours. Dyer and Maples [7] published that a 1.6% savings in compressor horsepower per degree increase could be recognized if chilled water temperature reset is implemented correctly. This savings is possible because raising the chilled water temperature reduces the temperature lift that must be supplied by the chiller. A possible savings of 12.8% could be recognized for 2,700 hours per year and a 4.8% savings could be enjoyed for the remaining 2,600 hours. Chilled water reset could be handled manually, or could be easily automated by integrating it into the current chiller control sequence. One issue to be addressed is that chiller one serves all critical air handler units where discharge temperature is maintained below 55 degrees Fahrenheit at all times due to surgeon requirements. This issue could be handled by only raising chilled water temperature for chiller one outside of surgery hours which are from 6 am to 3pm daily.

8.2.2.1.3 Savings Due to Chilled Water Supply Temperature Reset

In order to correctly estimate the possible savings due to implementing a chilled water reset strategy, a program was written in MATLAB to check weather data for the correct conditions and then correctly apply the improved chiller efficiency due to raising the chilled water temperature. The program takes into account the current chiller sequence of operations and stages the chillers appropriately based on demand. It simulates chiller operation on a month by month basis. Chiller loading is based on the chiller load curve previously established in this thesis. The MATLAB program can be found in Appendix D. The results of the MATLAB simulation can be found in Table 8.2.

Table 8.2: Savings From Chilled Water Temp. Reset

Energy Savings From Raising Chilled Water Temperature				
	Chiller 1 Usage	**Chiller 2 Usage**	**Usage Savings**	**Demand Savings**
Jan	687	77412	9441	13
Feb	1515	82531	8948	15
Mar	8443	97939	9214	0
Apr	21320	113581	6646	8
May	42254	128042	4170	0
Jun	75171	138936	832	0
Jul	84489	143112	505	0
Aug	85357	143201	350	0
Sep	61451	136566	2143	0
Oct	16414	102088	5380	0
Nov	3126	85369	8447	14
Dec	1351	96349	7742	0
Total	**401578**	**1345126**	**63818**	**50**
Savings	$4,411			

The MATLAB simulation predicted an estimated chiller electricity usage savings of 63,818 kwh per year and a demand savings of 50 kw. If implemented correctly, chilled water reset could reduce RMC's chiller energy cost by $4,400 per year. Manual implementation would not incur any capital cost, but it would require a substantial amount of attention to recognize the possible energy savings. An automated chilled water reset strategy would be more practical and would better accomplish the task of appropriately raising chilled water temperatures. An automated chilled water reset strategy would only require small changes to the current chiller plant sequence of operation. An estimated maximum capital cost of $5,000 will be used for this 5 year economic analysis. The results of the economic analysis can be found in Table 8.3.

$$Simple\ Payback = \frac{Capital\ Investment}{Yearly\ Savings} \qquad (34.\ \&\ 35.)$$

$$Rate\ of\ Return = -Capital\ Investment + \sum_{1}^{Number\ of\ Years} Yearly\ Savings * (1+d)^{-Number\ Of\ Years}$$

Table 8.3 Chilled Water Reset Economic Analysis Results

Economic Analysis for Chilled Water Reset	
Length of Analysis (Years)	5
Capital Cost	$5,000.00
Yearly Savings	$4,400.00
Simple Payback (Years)	1.1
Rate of Return	84%

8.2.2.2 Decrease Condenser Water Temperature Set Point

8.2.2.2.1 Decrease Condenser Water Temperature Set Point Analysis

The design condenser water inlet temperature for the water cooled, centrifugal chillers found at RMC is 85 degrees Fahrenheit. Chiller energy savings can be recognized if that condenser water temperature is lowered. However, one must be careful when lowering the condenser water temperature. Due to the fixed expansion orifice found in the RMC chillers, lowering condenser water too much can cause refrigerant stack which can create nusance trips and less efficient operation. The water cooled centrifugal chillers found at RMC can safely handle condenser water inlet temperatures as low as 65 degrees Fahrenheit without causing any significant problems. Lowering condenser water inlet temperature can increase chiller efficiency because like raising the chilled water supply temperature, it also reduces the temperature lift that must be supplied by the chiller. Dyer and Maples [7] suggest that for every degree condenser water temperature is reduced, 1.1 percent points can be gained in chiller efficiency. However there is an energy tradeoff that occurs between the cooling tower and the chiller. More cooling tower fan energy is needed to produce lower condenser water temperatures. Results from the simulation models used in this analysis suggest that the actual net gain is approximately one half percent point increase in efficiency per degree that condenser water temperature is reduced. The cooling tower model discussed earlier in this thesis was used in conjunction with a simple chiller model to simulate lowering the condenser water temperature to find the condenser water temperature set point that produces the optimal energy savings for RMC. The model assumes a one percent point increase in efficiency per degree condenser water temperature drop and also quantifies how much extra fan energy is needed to accomplish that one degree drop in set point. The model was used to simulate both fixed and variable condenser water set points. It was found that a variable set point did produce slightly better results but not enough to warrant the extra cost that would be required for implementation. The program used to simulate lowering condenser water temperature can be found in Appendix D. The results of the simulations can be found in Table 8.4. Usage values are in kilowatt hours. Savings are in dollars.

Table 8.4: Savings Results For Decreasing Condenser Water Set Point

2008 Condenser Water Results	85	80	75	72	70	68	65
Total Pump Usage	515027.3	515027.3	515027.3	515027.3	515027.3	515027.3	515027.3
Total CT Fan Usage	65335.2	97942.8	127802.2	143565.5	151957.2	159685.5	170142.5
Total Chiller Usage	1826611.49	1747116.72	1683659.64	1656335.12	1642545.42	1631451.84	1618229.4
Total Plant Usage	2406975.19	2360088.02	2326490.39	2314929.09	2309531.09	2306165.86	2303400.42
Usage Cost	$156,453.39	$153,405.72	$151,221.88	$150,470.39	$150,119.52	$149,900.78	$149,721.03
Demand Cost	$30,512.16	$29,174.99	$28,594.34	$28,109.03	$28,031.96	$27,965.81	$27,942.29
Total Cost	$186,965.55	$182,580.71	$179,816.21	$178,579.42	$178,151.48	$177,866.59	$177,663.31
Savings From Design	$ -	$ 4,384.79	$ 7,149.29	$ 8,386.08	$ 8,814.02	$ 9,098.91	$ 9,302.19
Percent Savings From Design	0	2.3	3.8	4.5	4.7	4.9	5.0

2009 Results	85	80	75	72	70	68	65
Total Pump Usage	509132	509132	509132	509132	509132	509132	509132
Total CT Fan Usage	65190	98559	133063	148282	155552	163104	175143
Total Chiller Usage	1861188	1778588	1709677	1683294	1670852	1659502	1644676
Total Plant Usage	2435511	2386279	2351873	2340708	2335537	2331738	2328952
Usage Cost	$ 158,308.20	$ 155,108.14	$152,871.72	$ 152,145.99	$ 151,809.89	$ 151,562.94	$151,381.88
Demand Cost	$ 30,447.69	$ 29,216.78	$ 28,636.44	$ 28,349.42	$ 28,198.64	$ 28,114.33	$ 28,075.69
Total Cost	$ 188,755.89	$ 184,324.92	$181,508.16	$ 180,495.42	$ 180,008.53	$ 179,677.27	$179,457.57
Savings From Design	0	4431	7248	8260	8747	9079	9298
Percent Savings From Design	0.0	1.4	2.9	3.5	3.7	3.9	4.0

2010 Results	85	72	65
Total Pump Usage	514832.34	514832.34	514832.34
Total CT Fan Usage	74254.53	145872.535	165810.43
Total Chiller Usage	1929101.8	1768535.5	1733009.7
Total Plant Usage	2518188.7	2429240.3	2413652.5
Usage Cost	163682.3	157900.6	156887.4
Demand Cost	30955.8	29054.3	28599.5
Total Cost	$ 194,638.10	$ 186,954.91	$ 185,486.95
Savings From Design	$ -	$ 7,683.19	$ 9,151.15
Percent Savings From Design	0	3.9	4.7

8.2.2.2.2 Savings Due to Reducing Condenser Water Temperature Set Point

A fixed year round condenser water set point of 65 degrees Fahrenheit yielded an estimated average savings of 4.5 % in net chiller plant energy savings over the three years of analysis. If the condenser water set point is lowered to 65 degrees Fahrenheit then an estimated $9,250 per year could be saved. A set point of 65 degrees Fahrenheit would be very close to the chiller's design lower limit. Although it produces the best energy savings, it may not be the best operating point considering RMC's chillers are nearing their expected service lives. A fixed set point of 72 degrees Fahrenheit produced an estimated average savings of 4.1 percent and would create a safer operating point for the aging chillers at RMC. If the condenser water set point at RMC is set to 72 degrees Fahrenheit, RMC could expect an estimated $8,000 in annual energy savings at no capital cost. The simulation was also performed in EQUEST. EQUEST resulted in an estimated 118,000 kwh per year net savings worth approximately $7,600 with a fixed set point at 72 degrees Fahrenheit.

8.2.2.3 Increase Free Cooling Time

8.2.2.3.1 Increase Free Cooling Time Analysis

The current operating condition of the plate and frame heat exchanger doesn't allow for the maximum available free cooling operating hours. The plate frame design approach is 2 degrees Fahrenheit. In other words, the plate and frame hot side output temperature should be within two degrees of the cold side inlet temperature. Data taken on RMC's plate and frame heat exchanger currently shows a 7 degree approach. The unsatisfactory performance of the heat exchanger is most likely due to scale build up over time on the heat exchanger surface. If the plate and frame heat exchanger were taken apart and cleaned the design approach could possibly be restored. In the current operating condition the free cooling is limited to an estimated 800 hours. If the plate and frame heat exchanger design approach is restored it would allow for an additional 400 hours of free cooling.

8.2.2.3.2 Savings Due to Increasing Free Cooling Time

The average main hospital cooling load during free cooling operation is approximately 180 tons. The current overall average chiller efficiency is 0.7 kw/ton. If free cooling is operational, the chillers are turned off.

$$Current\ Free\ Cooling\ Savings = 180\ ton * .7\frac{kw}{ton} * 800hrs * 0.065\frac{\$}{kwh} = \$6{,}552$$

$$Max\ Free\ Cooling\ Savings = 180ton * .7\frac{kw}{ton} * 1200hrs * 0.065\frac{\$}{kwh} = \$9{,}828$$

$$Estimated\ Savings = \$9{,}828 - \$6{,}552 = \$3{,}276$$

Cleaning the plate and frame heat exchanger would require no capital investment. It could be easily accomplished by current maintenance personnel in a single day.

8.2.2.4 HVAC Transport System Cost Reduction

8.2.2.4.1 Reduce Outdoor Air Intake

8.2.2.4.2 Reduce Outdoor Air Intake Analysis

Outdoor air is drawn into RMC at the air handlers in order to provide adequate ventilation and indoor air quality. Excess outdoor air intake can cause excess energy usage because of the extra energy needed to cool and heat the outdoor air. Data taken during the building survey indicates that some units bring in an excess amount of outdoor air. Reducing outdoor air by too much will improve energy costs but will sacrifice indoor air quality. Care must be taken to find the correct outdoor air intake level. Also, it is important to be sure that the outside air dampers on variable volume systems properly adjust to maintain minimum outside air flow as the fans and dampers throttle back to minimum flow positions. Currently at RMC, there are five air handler units found to have excess outdoor air intake. Those units are listed in Table 8.5 .

Table 8.5: Candidate AHU's for OA Reduction

Units With Excess Outdoor Air Intake				
	Total Air Flow(cfm)	OA %	Current OA Flow (cfm)	Min Required Airflow (cfm)
OR1	7000	78	5460	3000
OR2	8000	81	6480	3000
1-E	20000	50	10000	5000
2-C	4000	35	1400	885
K	6000	51	3060	1000

The outside air intake on these units should be reduced to the minimum required levels. Reducing outside air intake saves both cooling and heating energy.

8.2.2.4.3 Savings From Reducing Outdoor Air Intake

The total excess outdoor air intake for these five units is 13,515 cfm. Equations (36.), (37.), and (38.) below are used to estimate the possible savings from reducing outdoor air intake. The equations were programmed into a MATLAB simulation which brought in a full year of weather data for Alexander City, Alabama. The weather data was fed into the equations to estimate energy savings over time. An electricity usage rate of \$0.065 per Kwh and a natural gas rate of \$6 per million BTU were applied to quantify cost savings. The MATLAB program used to simulate outdoor air reduction can be found in Appendix D. Table 8.6 shows the results of the MATLAB simulation.

$$Sensible\ Load\ Savings\ OA\ Reduction\ (Tons) = \frac{\dot{V}_{air} * \rho_{air} * C_{p,air}(T_{OA} - T_{supply})}{200 \frac{Btu}{min/ton}} \quad (36.)$$

Where: $\dot{V}_{air}$ is the volumetric flow rate of the excess outside air. (13, 515 cfm)

ρ_{air} is the density of air in lbm/ft^3. (0.075)

$C_{p,air}$ is the specific heat of air in BTU/lbm-F. (0.24)

T_{OA} is the temperature of the outdoor air in degrees Fahrenheit.

T_{supply} is the temperature of the supply air in degrees Fahrenheit.

$$Latent\ Cooling\ Load\ Savings\ OA\ Reduction (Tons) = \frac{\dot{V}_{air} * \rho_{air} * 1000 \frac{BTU}{lb} (\omega_{OA} - \omega_{supply})}{200 \frac{BTU}{min/ton}} \quad (37.)$$

Where: ω_{OA} is the specific humidity of the outside air stream.

ω_{supply} is the specific humidity of the supply air stream.

$$Heating\ Load\ Savings\ OA\ Reduction\ \left(\frac{BTU}{hr}\right) = \dot{V}_{air} * \rho_{air} * C_{p,air}\left(T_{supply} - T_{OA}\right) * 60 \frac{min}{hr} \quad (38.)$$

Table 8.6 Potential OA Reduction Savings

Potential Savings From Reducing Outdoor Air Intake				
	Cooling Energy Savings (kwh)	**$ Savings**	**Heating Energy Savings BTU**	**$ Savings**
Jan	1112	$ 72.00	130353796	$ 782.00
Feb	2116	$ 137.00	82254452	$ 493.00
Mar	3703	$ 240.00	49505445	$ 297.00
Apr	7562	$ 491.00	16138531	$ 96.00
May	14277	$ 928.00	1595851	$ 9.00
Jun	23948	$ 1,556.00	0	$ -
Jul	28578	$ 1,875.00	0	$ -
Aug	28495	$ 1,852.00	0	$ -
Sep	21021	$ 1,366.00	428155	$ 2.00
Oct	6883	$ 447.00	22667898	$ 136.00
Nov	2399	$ 155.00	71336494	$ 428.00
Dec	4171	$ 271.00	77812342	$ 466.00
Total	144265	$ 9,390.00	452092964	$ 2,709.00
YearlySavings	**$ 12,099.00**			

EQUEST was also used to simulate reducing outdoor air intake. The results of the EQUEST analysis were similar to MATLAB results. EQUEST predicts a cooling energy savings of 176,000 kwh and a heating energy savings of 400 million BTU for a total cost savings of $13,840.

Because the majority of the excess outside air intake comes from constant volume air handler units with fixed but adjustable outside air intake dampers, a considerable amount of the predicted savings are gained with only a manual adjustment of the outdoor air intake dampers. This adjustment could be handled by in house maintenance personnel and requires no capital investment. However to fully maximize the savings potential, some control system upgrades would have to be implemented. This requires a substantial capital cost that will be discussed later in the section on control system upgrades.

8.2.2.5 Increase Discharge Air Temperature Set Point

8.2.2.5.1 Increase Discharge Air Temperature Set Point Analysis

The discharge air temperature must be kept low enough to properly dehumidify in order to maintain the strict indoor relative humidity requirements of RMC. Data gathered during the building survey shows that the average discharge air temperature for the main hospital is 52 degrees Fahrenheit. The average mixed air temperature at peak load is 79 degrees Fahrenheit and 72 degrees Fahrenheit at part load. The average space temperature throughout RMC is 70 degrees Fahrenheit. Analysis with a psychrometric chart shows that the relative humidity is 52 percent when air is cooled to 52 degrees Fahrenheit and reheated to 70 degrees Fahrenheit. If the air were cooled to only 55 degrees Fahrenheit and reheated to 70 degrees then the relative humidity would be 56 percent. A relative humidity of 56 percent is within the relative humidity specification and would create savings for both reheating and cooling. Several zones in RMC require the lower discharge air temperature because of the space temperature requirements, but the discharge air temperature in the majority of the main hospital could be raised to 55 degrees Fahrenheit and still achieve adequate indoor air quality. The discharge air temperature could be raised on a total of nine air handlers.

8.2.2.5.2 Savings for Increasing Discharge Air Temperature

A simple savings analysis based on the First Law of Thermodynamics is shown below. The main savings estimation is performed with EQUEST software because of its ability to accurately simulate variable air volume systems with reheat over time.

The following data were used in performing the simple analysis:

Total number of hours per year = 8,760

Total Airflow of Nine Units = 71,500 cfm

Density of Air = 0.075 lbm/ft^3

Specific Heat of Air = 0.24 BTU/lbm-F

Heat Release due to Condensation = 1000 BTU/lbmw

Average Mixed Temperature at Peak Load = 79 degrees F

Average Mixed Temperature at Part Load = 72 degrees F

Average Current Discharge Air Temperature = 52 degrees F

New Discharge Air Temperature = 55 degrees F

Average Specific Humidity of Mixed Air at Peak Load =0 .012 lbmw/lbma

Average Specific Humidity of Mixed Air at Part Load = 0.0095 lbmw/lbma

Specific Humidity of Current Discharge Air = 0.008 lbmw/lbma

Specific Humidity of New Discharge Air =0.009

Current Reheat Coil Average Discharge Air Temperature 75 degrees F

Overall Boiler Efficiency = 80 %

Peak Load Energy Savings

In this section, a description of peak load energy savings is estimated.

Current Cooling Load at 52 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (79 - 52) + 71{,}500 * 0.075 * 1000 * (0.012 - 0.008)}{200} = 280\,Tons$$

New Cooling Load at 55 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (79 - 55) + 71{,}500 * 0.075 * 1000 * (0.012 - 0.009)}{200} = 234\,Tons$$

Energy Savings = 280-234 = 46 Tons

Current Reheat Load at 52 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (75 - 52)}{0.8} = 2{,}220{,}075\frac{BTU}{hr}$$

New Reheat Load at 55 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (75 - 55)}{0.8} = 1{,}930{,}500\frac{BTU}{hr}$$

Savings = 2,220,075-1,930,500 = 289,575 BTU/hr

Part Load Savings Estimation:

In this section, the energy savings at part load conditions is estimated.

Current Cooling Load at 52 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (72 - 52) + 71{,}500 * 0.075 * 1000 * (0.095 - 0.008)}{200} = 168\,Tons$$

New Cooling Load at 55 Degree Discharge Air Temperature:

$$\frac{71{,}500 * 0.075 * .24 * (72 - 55) + 71{,}500 * 0.075 * 1000 * (0.095 - 0.009)}{200} = 122\,Tons$$

Savings = 168-122 = 46 Tons

Heating Load Energy Savings Part Load = Peak Load Energy Savings

Cooling Load Cost Savings Estimation:

The following is an estimation of cooling load cost savings:

$$46\,Tons * .7\,\frac{kw}{Ton} * 8{,}760hrs * 0.065\,\frac{\$}{kwh} = \$18{,}334\;Per\;Year$$

Heating Load Cost Savings Estimation:

The heating load cost savings are estimated as follows assuming reheat coils are active for 8 hours total out of day.

$$289{,}575\,\frac{BTU}{hr} * 8\,\frac{hr}{day} * 365\,\frac{day}{year} * 6\,\frac{\$}{10^6 BTU} = \$5{,}073\;Per\;Year$$

Total Simple Analysis Savings for Increasing Discharge Air Temperature

The total cost savings are obtained by adding the cooling and heating load cost savings obtained above.

Total Savings = $18,344 + $5,073 = $23,407 Per Year

Note: Simple analysis does not include ventilation fan costs.

EQUEST software was used to simulate raising the discharge air temperature by three degree Fahrenheit on the nine candidate air handler units. EQUEST accounts for both the thermal savings and the ventilation fan penalty. Raising the discharge air temperature will cause the ventilation fans in variable volume air handler units to run at higher power levels for longer periods of time in order to satisfy zone cooling requirements. The higher discharge air temperature requires a larger volume of air to accomplish the same heat removal as the lower supply temperature. EQUEST predicted a total cooling energy reduction of 208,000 kwh with ventilation fan penalty of 67,300 kwh hours creating a net cooling load savings of 140,700 kwh which is worth $9,145 per year. EQUEST also predicted a reheat savings of 980 million BTU per year which is worth an estimated $5,880 per year at six dollars per million BTU. The total EQUEST predicted savings is an estimated $15,025 per year. If the $4,400 ventilation fan penalty is subtracted from the simple analysis total, it brings the two methods within twenty percent of each other. The EQUEST simulation is by far the more accurate energy savings prediction because of its ability to simulate the entire system as a whole over time using accurate weather data. Increasing the supply temperature by three degrees is a no capital cost recommendation that can be implemented as easily as changing the discharge air temperature set point on the control system interface.

8.3 POB Non Capital Intensive Energy Cost Reduction

8.3.1 POB Electrical System Cost Reduction

No non capital intensive electrical system cost reduction strategies were found to be viable for the POB.

8.3.2 POB HVAC Cost Reduction

8.3.2.1 POB Generation System Energy Cost Reduction

8.3.2.1.1 Increase POB Chilled Water Supply Temperature Set Point

8.3.2.1.2 Increase POB Chilled Water Supply Temperature Set Point Analysis

The POB currently operates with a chilled water supply temperature of 44 degrees Fahrenheit. Like with the main hospital, the chilled water supply temperature can be raised during off peak loading conditions. As mentioned earlier, when the wet bulb temperature is at or below 50 degrees Fahrenheit, no dehumidification is required and the chilled water temperature can be raised. It can also be raised by a smaller amount when the wet bulb temperature is at or below 65 degrees Fahrenheit. There is an estimated 2,700 hours per year when the chilled water temperature can be raised substantially and another 2,600 hours that is can be raised by a smaller amount. It is recommended that RMC raise the POB chilled water supply temperature to 50 degrees Fahrenheit when the wet bulb temperature is at or below 50 degrees Fahrenheit. It is also recommended that RMC raise the chilled water supply temperature to 46 degrees Fahrenheit when the outdoor wet bulb temperature is at or below 65 degrees Fahrenheit.

8.3.2.1.3 Savings Due to Increased POB Chilled Water Temperature Set Point

The savings estimation was performed with EQUEST software due to its ability to dynamically simulate all HVAC mechanical systems at once. The chilled water reset strategy described earlier was simulated in EQUEST. The results are shown in Table 8.7. Usage savings are in kilowatt hours.

Table 8.7: Potential Savings Due to POB Chilled Water Reset

Savings Due to POB Chilled Water Reset	
Current Chiller Energy Usage	1643000
Predicted Energy Usage	1613900
Usage Savings	29100
Cost Savings	**$ 1,891.50**

EQUEST predicted an estimated savings of two percent on POB chiller electricity cost over the course of one year. As mentioned previously, chilled water reset can be manually implemented, but maximum savings is best enjoyed with an automated chilled water reset strategy. The POB chilled water reset strategy must be added to POB chilled water plant sequence of operations in order to automate the process. Sensors are currently in place to read outdoor wet bulb temperature. The computer code used to execute the sequence must be altered with the correct reset parameters. It is estimated that this will cost a maximum of $5,000 to implement. The results of an economic analysis on this project can be found in Table 8.8.

Table 8.8: Chilled Water Reset Economic Analysis Results

Chilled Water Reset Economic Analysis	
Analysis Length (Years)	5
Simple Payback (Years)	2.6
Rate of Return %	25.8

8.3.2.2 Increase POB Discharge Air Temperature

8.3.2.2.1 Increase POB Discharge Air Temperature Analysis

The current average operating discharge air temperature is 52 degrees Fahrenheit similar to the main hospital. A discharge air temperature of 55 degrees Fahrenheit will accomplish the dehumidification needs of the facility. Increasing discharge air temperature will create both cooling and heating energy savings. Once again, EQUEST software was used to simulate raising the discharge air temperature because of its ability to accurately simulate variable air volume systems with reheat.

8.3.2.2.2 Savings Due to Increasing Discharge Air Temperature

The results of the EQUEST simulation can be found in Table 8.9.

Table 8.9 Potential Savings Due to Increasing Discharge Air Temp. Set Point

Savings Due to Increasing POB Discharge air Temperaure						
	Current Cooling Usage(kwh)	Current Fan Usage (kwh)	New Cooling Usage (kwh)	New Fan Usage (kwh)	Current Reheat Usage (BTU)	New Reheat Usage (BTU)
Jan	70000	29200	56100	26400	1330000000	1150000000
Feb	64200	26000	51300	23900	1140000000	980000000
Mar	92800	23400	78400	24000	790000000	660000000
Apr	115300	23100	102900	24800	580000000	480000000
May	168500	26200	154000	28900	500000000	410000000
Jun	219800	29200	207200	32300	790000000	310000000
Jul	240800	30100	229200	33300	400000000	320000000
Aug	237500	29600	225900	32900	390000000	310000000
Sep	188500	26100	177000	28800	450000000	360000000
Oct	104900	22700	91900	23800	700000000	580000000
Nov	75900	23000	61400	22200	970000000	820000000
Dec	64800	26900	50600	24600	1240000000	1060000000
Total Usage	1643000	315500	1485900	325900	9280000000	7440000000
Net Usage Savings (kwh)	**146700**					
Reheat Savings (BTU)	**1840000000**					
Net Cost Savings	**$ 20,575.50**					

A total cost savings of $20,575 dollars per year is predicted using EQUEST simply by raising the discharge supply temperature by three degrees to 55 degrees Fahrenheit.

This recommendation requires no capital cost. It simply requires the discharge air temperature set point be raised on the control system interface for each air handler unit contained in the POB.

8.4 Cancer Center Non Capital Intensive Cost Energy Cost Reduction

8.4.1 Cancer Center HVAC Cost Reduction

8.4.2.1Cancer Center Generation System Energy Cost Reduction

8.4.2.1.1 Increase Cancer Center Chilled Water Supply Temperature

8.4.2.1.2 Increase Cancer Center Chilled Water Supply Temperature Analysis

The Cancer Center at RMC currently has a chilled water supply temperature of 42 degrees Fahrenheit. When the weather conditions are satisfactory, the chill water temperature can be increased to reduce the temperature lift seen by the chiller and therefore reduce chiller plant energy costs. It is recommended that the chill water supply temperature set point be raised to 50 degrees Fahrenheit when the wet bulb temperature is at or below 50 degrees Fahrenheit. It can also be increased to 45 degrees Fahrenheit when the outdoor wet bulb temperature is at or below 65 degrees Fahrenheit.

8.4.2.1.3 Savings Due to Increasing Cancer Center Chilled Water Temperature

An EQUEST simulation was used to predict the possible energy savings due to a chill water reset strategy applied to the Cancer Center. The results of the simulation are shown in Table 8.10.

Table 8.10: Potential Savings Due to Cancer Center Chilled Water Temp. Reset

Savings Due to Cancer Center Chilled Water Reset	
Current Chiller Energy Usage	264040
Predicted Energy Usage	258740
Usage Savings	5300
Cost Savings	**$ 344.50**

Chill water reset savings were estimated by EQUEST to be approximately two percent. Due to the small size of the cancer center the savings are relatively insignificant when compared to other recommendations for other parts of RMC. With an estimated $5,000 implementation cost, the simple payback on this recommendation alone is 14.5 years. It could still be economical if RMC could convince the contractor to bundle chilled water reset for the Cancer Center with the upgrades suggested for the main hospital and POB.

8.5 Summary and Evaluation of Non Capital Intensive Cost Reduction

A total of nine non capital intensive cost reductions strategies were discussed in this section. All nine suggestions and the economics associated with each are listed in Table 8.11.

Table 8.11: Non Capital Intensive Cost Reduction Recommendations

Non Capital Intensive Energy Cost Reduction Recommendations				
Recommendation	**Potential Savings(per yr)**	**Associated Capital Cost**	**Simple Payback yr.**	**Rate of Return**
Chilled Water Supply Temp. Reset	$4,400	$5,000	1.1	84% over 5 yr
Decrease Condenser Water Temp Set Point	$8,000	$0	N/A	N/A
Increase Free Cooling Time	$3,276	$0	N/A	N/A
Reduce OA Intake	$12,099	$0	N/A	N/A
Increase AHU Discharge Air Temp.	$15,025	$0	N/A	N/A
POB Chilled Water Temp. Reset	$1,891	$5,000	2.6	25.8 % over 5 yr
Increase POB AHU Discharge Ait Temp.	$20,575	$0	N/A	N/A
Cancer Center Chilled Water Supply Temp Reset	$344	$5,000	N/A	N/A
Combine Electricity Meters	$131,527	$5,000	0.04	N/A
Totals:	**$197,137**	**$20,000**	**0.07**	**985% over 5 yr**

If all recommendations are successfully implemented, it would save an estimated $197,137 in energy annual costs. All recommendations have strong economics with the exception of a Cancer Center chilled water supply temperature reset strategy. The Cancer Center has a relatively small HVAC energy cost compared to the other facilities. A

Cancer Center chilled water reset strategy could be neglected without sacrificing any significant energy savings. Five out the nine recommendations require no capital cost to implement and produce significant energy savings. Obviously the potential savings of combined metering are very large, however this project will require extensive negotiation with ACUS possibly taking months to reach an agreement satisfactory to both RMC and ACUS.

8.6 Capital Intensive Energy Cost Reduction Strategies

Capital intensive energy cost reduction refers to strategies that can be implemented for over $5,000. There are several cost saving strategies highlighted in this section for the building envelope, electrical, and HVAC energy cost centers for each facility contained within the RMC campus.

8.6.1 Main Hospital Capital Intensive Energy Cost Reduction Strategies

8.6.2 Main Hospital Building Envelope Capital Intensive Cost Reduction

8.6.2.1 TPO Roofing

Thermoplastic Polyolefin (TPO) roofing is a thermally adhered, light colored roofing system that is used to replace traditional built-up roofing systems. Because of its light color, high solar reflectance, and infrared emittance it can greatly reduce roof surface temperature during the summer months. A reduced roof surface temperature translates into less heat transfer into the building thereby reducing building cooling loads.

However, a heating penalty is paid during the winter months due to the reduced solar heat gain.

8.6.2.1.1 TPO Roofing Analysis

The proposed TPO roofing system for RMC is to replace 28,859 square feet of black built up roof. The proposed roofing system also includes an additional 1 inch layer of isocyanurate insulation board. The proposed roof has a manufacturer's rated R-value of 20, a solar reflectance value of 0.77, and an infrared emittance value of 0.87. The DOE roof savings calculator developed by Oak Ridge National Laboratories was used to help quantify the possible savings. The cooling load for the proposed roof was found to be 1,555 $\frac{BTU}{ft^2}$ per year as opposed to the cooling load for the current roof of 5,203 $\frac{BTU}{ft^2}$ per year. The heating load for the proposed roof was calculated to be 4,317 $\frac{BTU}{ft^2}$per year as opposed to 3,760 $\frac{BTU}{ft^2}$ per year for the current built up roof.

8.6.2.1.2 TPO Roofing Savings

The data used for calculating roofing energy savings follow:

Chiller Efficiency = 0.7 kw/ton

Cooling Hours Per Year 5,280 hrs

Electricity Usage Rate= $0.065 per kwh

Total Roof Area = 28,859 ft^2

Natural Gas Price = $6 per million BTU

$$Cooling\ Load\ Energy\ Savings = 5{,}203 - 1{,}155 = 4{,}048\,\frac{BTU}{ft^2/year}$$

$$Heating\ Energy\ Penalty = 4{,}317 - 3760 = 557\,\frac{BTU}{ft^2/year}$$

The cost savings based on these data are:

$$Cooling\ Cost\ Savings = \frac{4{,}048\,\frac{BTU}{ft^2/yr}}{2{,}195.5\,\frac{\frac{BTU}{ft^2/yr}}{Ton}} * \frac{.7\,\frac{kw}{Ton} * 5{,}280\,\frac{hrs}{yr} * 0.065\,\frac{\$}{kwh}}{28{,}859 ft^2} = 0.015\,\frac{\$}{ft^2/yr}$$

$$Heating\ Penalty\ Cost = 557\,\frac{BTU}{ft^2/yr} * \frac{\$6}{10^6 BTU} = 0.003\,\frac{\$}{ft^2/yr}$$

$$Net\ Savings = 0.015 - .003 = 0.012\,\frac{\$}{ft^2/yr}$$

$$Total\ Net\ Savings = 0.012\,\frac{\$}{\frac{ft^2}{yr}} * 28{,}859 ft^2 = \$346\ per\ year$$

Economic Analysis:

The capital cost of installing the proposed TPO roofing system is $111,648 per year which equals $3.87 per square foot. The simple payback period is beyond a century. This project purely as a retrofit is not economically sound. If the current roof has exceeded it life expectancy and must be replaced, a TPO roofing system replacement would only offer a small energy cost savings.

8.6.3 Main Hospital Electrical System Capital Intensive Cost Reduction

8.6.3.1 LED Lighting

Currently 88 percent of the main hospital is illuminated with T-8 fluorescent lamps. Light emitting diode (LED) technology has been applied to general lighting fixtures and many companies currently offer LED lamp drop in replacements for T-8 fluorescent lamps. The main advantage of LED lighting is that it offers equivalent illumination with greater energy efficiency. Typically LED lamps consume 40 percent less power than their fluorescent counterparts and have double the life expectancy. LED lighting can reduce the energy needed to power the lights and can also reduce building cooling loads.

8.6.3.1.1 LED Lighting Analysis

This section will look at replacing the T-8 fluorescent lamps in the main hospital with new, more energy efficient, LED replacement lamps. The main hospital contains a total of 4,574 T-8 fluorescent lamps. These lamps create an energy demand of 140,894 watts. The proposed LED replacement lamp is made to fit inside the current T-8 fixture. Ballast are not needed for LED lighting, therefore the fixture must be rewired so that the ballast is bypassed. The cost of the LED replacement lamps ranges from fifty to seventy dollars per lamp which is a great deal more expensive than current fluorescent T-8 lamps. The life expectancy of the LED lamps is 60,000 hours or approximately seven years.

8.6.3.1.2 LED Lighting Savings

The data used in the lighting cost savings follow:

Current T-8 Demand = 140,894 watts

LED Demand = 84,537 watts

Operational Cooling Load Factor = 0.816

Residual Cooling Load Factor = 0.1321

Number of Hours in Year = 8,760 hours

Electricity Usage Rate = $0.065 per kwh

Electricity Demand Rate =$5.25 per kw

Chiller Efficiency = 0.7 kw/ton

Current Price of LED Lamps = $60 per lamp

Total # of Lamps = 4,574

Savings Calculations:

Savings calculations are based on 24 hour operation.

$$LED\ Demand\ Savings = .4 * 140{,}894\ watts = 56{,}357\ watts$$

$$Cooling\ Load\ Savings = \frac{56{,}357 watts * 24hr * 3.41 \frac{BTU}{W\ hr} * 0.816}{24hr} = 156{,}816 \frac{BTU}{hr}$$

$$Cooling\ Load\ Usage\ Savings = \frac{156{,}816 \frac{BTU}{hr} * 0.7 \frac{KW}{Ton} * 8760 \frac{hrs}{yr}}{12{,}000 \frac{BTU}{hr/Ton}} = 80{,}132\ \frac{kwh}{yr}$$

$$Cooling\ Demand\ Savings = \frac{156{,}816 \frac{BTU}{hr} * 0.7 \frac{KW}{Ton}}{12{,}000 \frac{BTU}{hr/Ton}} = 9\ KW$$

$$Power\ Savings = \frac{56{,}357 W * 8760 \frac{hr}{yr}}{1{,}000 \frac{W}{KW}} = 493{,}687 \frac{kwh}{yr}$$

$$Total\ Power\ Usage\ Savings = 493{,}687 + 80{,}132 = 573{,}819\,\frac{kwh}{yr}$$

$$Total\ Demand\ Savings = 56 + 9 = 65KW$$

$$Operation\ Cost\ Savings = 573{,}819\,\frac{kwh}{yr} * 0.065\,\frac{\$}{Kwh} + 65KW * 5.25 * 12\,\frac{months}{yr}$$

$$= \$41{,}393\ per\ year$$

Economic Analysis:

Based on the savings determined above, an economic analysis follows:

$$LED\ Capital\ Cost = 4{,}574\ lamps * 60\,\frac{\$}{Lamp} = \$274{,}440$$

The savings of $41,393 per year calculated above allow for simple payback and minimum attractive rate of return to be calculated. The results are shown in in Table 8.12.

Table 8.12 LED Lighting Economic Analysis Results

LED Economic Analysis	
Simple Payback	6.6 years
7 year MARR	1.38%
10 year MARR	-10.73%

A complete LED retrofit would require a substantial capital cost. If implemented, the project would save $41,393 dollars per year in operational costs, which equates to a seven year MARR of 1.38 percent. After the first seven years the project would create a net surplus of $3,787. However, the life expectancy of the lamps is only seven years. Due to the current cost of LED lighting, maintenance and replacement costs erase any gains due to energy savings. The 10 year MARR on this project is negative making LED lighting not practical in the long run if the cost of LED lighting doesn't decrease over time and energy costs do not increase significantly. Over a ten year period RMC currently would spend roughly $96,341 per year to own, operate, and maintain their current lighting fixtures. If a LED retrofit were implemented, the cost would increase to $107,000 per year to own, operate, and maintain LED lighting. The aforementioned costs were estimated from simple analysis not including interest.

8.6.4 Main Hospital Capital Intensive HVAC Cost Reduction

8.6.4.1 Main Hospital Generation Systems Capital Intensive Cost Reduction

8.6.4.1.1 Boiler Plant Upgrade

The current fire tube steam boilers serving the main hospital have been in operation for fifty years. They have greatly exceeded their expected service life and are becoming dangerous to operate. Over time, new technologies have been developed to replace the current system and offer significant energy savings. The proposed upgrade will eliminate the hospitals need for central steam production and transition to a state of the art, safer and more efficient hot water heating system. This project will require substantial engineering planning and capital costs. This section will not present the proposed system and its implementation plan in great detail. It will only summerize the energy savings opportunity that would be created by the new system. The major changes include replacing the old steam boilers with new energy efficient condensing hot water boilers, adding electric sterilizers and humidifiers, changing steam coils to hot water coils, and some piping changes.

8.6.4.1.2 Boiler Plant Upgrade Analysis

Currently the main hospital boiler plant consumes an average of 13,000 MCF per year of natural gas. That equates to 13.6 billion BTU’s per year. It produces an average of 2,500 pounds of steam per hour. The new proposed condensing hot water generators can achieve a 90 percent overall efficiency with proper control of the hot water return temperature. With a current overall boiler efficiency of 80 percent, the proposed boilers would create a 10 percent reduction in fuel usage. Steam boiler systems require a network of steam traps, condensate return lines, and condensate pumps. These systems can be very costly to maintain and often contribute to unnecessary energy waste. At present the facility must use amines for corrosion control, sodium sulfite to control oxygen, and polymers to control scaling. These chemicals require substantial cost that would be eliminated by the new system.

8.6.4.1.3 Boiler Plant Upgrade Savings

Fuel Savings:

The data and analysis for determining the boiler plant fuel savings follow:

Current Overall Efficiency = 80%

New Efficiency = 90%

Current Fuel Consumption = 13.6 billion BTU year

Natural Gas Price = Average $6 per million BTU

$$Current\ Boiler\ Fuel\ Cost = 13{,}650{,}000{,}000 \frac{BTU}{yr} * 6 \frac{\$}{10^6 BTU} = \$81{,}900$$

$$Fuel\ Savings = .10 * \$81{,}900 = \$8{,}190\ per\ year$$

Savings Due To Eliminating Steam Trap Losses:

Note: In the calculations below, it is assumed that makeup water is at 60 F, condensate is lost at 200 F, steam is produced at 85 psig.

The data and analysis for determining the steam trap savings follow:

Current make-up water percentage = 30 %

Steam Flow = 2,500 lbs/hr

Enthalpy of Steam = 1,150 BTU/lb

Enthalpy of Condensate = 140 BTU/lb

$$Steam\ Energy\ Loss = \frac{\%loss_{Steam} * SteamFlow \frac{lb_{steam}}{hr} * 1{,}150 \frac{BTU}{lb_{steam}} * 8760 \frac{hr}{yr}}{Boiler\ Efficiency} \qquad (27.)$$

$$\frac{0.15 * 2500 \frac{lb_{steam}}{hr} * 1{,}150 \frac{BTU}{lb_{steam}} * 8760 \frac{hr}{yr}}{.8} = 4{,}722{,}187{,}500 \text{ BTU}$$

$$Steam\ Loss\ Cost = \frac{4{,}722{,}187{,}500 \text{ BTU} * \$6}{1{,}000{,}000\ BTU} = \$28{,}333 \text{ per year}$$

$$Cond.\ Energy\ Loss = \frac{\%loss_{Cond.} * SteamFlow \frac{lb_{steam}}{hr} * 140 \frac{BTU}{lb_{condensate}} * 8760 \frac{hr}{yr}}{Boiler\ Efficiency} \qquad (28.)$$

$$\frac{0.15 * 2500 \frac{lb_{steam}}{hr} * 140 \frac{BTU}{lb_{steam}} * 8760 \frac{hr}{yr}}{0.8} = 574{,}875{,}000 BTU$$

$$Condensate\ Loss\ Cost = \frac{574{,}875{,}000 \text{ BTU} * \$6}{1{,}000{,}000\ BTU} = \$3{,}449 \text{ per year}$$

$$Total\ Savings\ Due\ to\ Steam\ Trap\ Elimination = \$31{,}782\ per\ year$$

Chemical Cost Savings:

RMC currently spends about $10,000 annually on chemical treatment which can be eliminated with the proposed new system.

Total Cost Savings:

The total cost savings for the new boiler plant/conversion to hot water are the sum of the fuel, maintenance, and chemical savings.

$$Total\ Estimated\ Cost\ Savings = Fuel\ Savings + Maintenace\ Savings + Chemical\ Savings$$
$$= \$50{,}000\ per\ year$$

Economic Analysis:

Table 8.13 contains the capital cost of equipment needed to complete the new boiler plant hot water conversion.

Table 8.13: Boiler Plant Upgrade Capital Costs

Equipment/Labor	Cost
New Hot Water Boilers	$209,800
Pumps	$10,000
VFD Drives	$4,472
Expansion Tank	$9,830
Water Treatment	$940
New Piping	$550
Industrial Water Heater	$950
Domestic Hot Water Tank	$28,631
Construction/Labor	$29,440
Steam Coil Replacement	$75,000
Sterilizer and Humidification	$435,600
Total	**$805,213**

The total capital cost of the boiler plant upgrade would be an estimated $805,213.

The simple payback and MARR were calculated using the capital cost and savings results obtained above. Table 8.14 contains the results of the simple economic analysis.

Table 8.14: Boiler Plant Upgrade Economic Analysis Results

Boiler Plant Upgrade Economic Analysis	
Simple Payback	16 years
25 year MARR	3.71%

The proposed system would have a life expectancy of twenty five years. The MARR would be an estimated 3.71 percent over that twenty five year lifetime. This investment would not be attractive for a retrofit project, but as mentioned earlier, the current boiler system has greatly exceeded its expected service life and must be replaced.

8.6.4.1.4 VFD Cooling Tower Fan Upgrade

The cooling tower at RMC is currently equipped with two speed fans. Two speed fans are controlled by a simple on/off controller. The fans cycle between the two speeds in order to maintain the condenser water set point. It can be damaging to the fans to cycle between speeds too often. Because RMC is limited by two fan speeds and cycle frequency, the fans have to operate with a broad control range. In other words if the condenser water temperature set point is 72 degrees, the fans operate with a control range of 72 degrees plus or minus 5 degrees. The two setting limit makes the cooling tower frequently over and under shoot the condenser water set point. This lack of tight control causes cooling tower fan energy waste, and can affect chiller efficiency because of the broad condenser water temperature range they operate with. Variable Frequency Drive (VFD) fans can minimize the over shoot and correctly maintain the correct fan speed to maintain the condenser water set point. PID control and variable speed electric motors allow reduction in energy costs because the fans can efficiently operate at the correct fan speed minimizing fan energy usage and holding the condenser water temperature at a constant set point which allows the chiller to operate at its optimal efficiency range at all times.

8.6.4.1.5 VFD Cooling Tower Fan Analysis

The cooling tower model used to estimate cooling tower energy cost was reprogrammed to simulate a variable frequency drive cooling tower fan. As stated previously in the cooling tower analysis section, if the air flow rate through the cooling tower changes the NTU value of the cooling tower also changes. A VFD fan is capable of many air flow rates. In order to properly simulate a VFD, the cooling tower model is used to find an NTU value for the cooling tower over a broad range of air flow rates. A plot of the NTU value at each air flow rate is generated to find a relationship between NTU value and air flow. The NTU vs. air flow plot is shown in Figure 8.1.

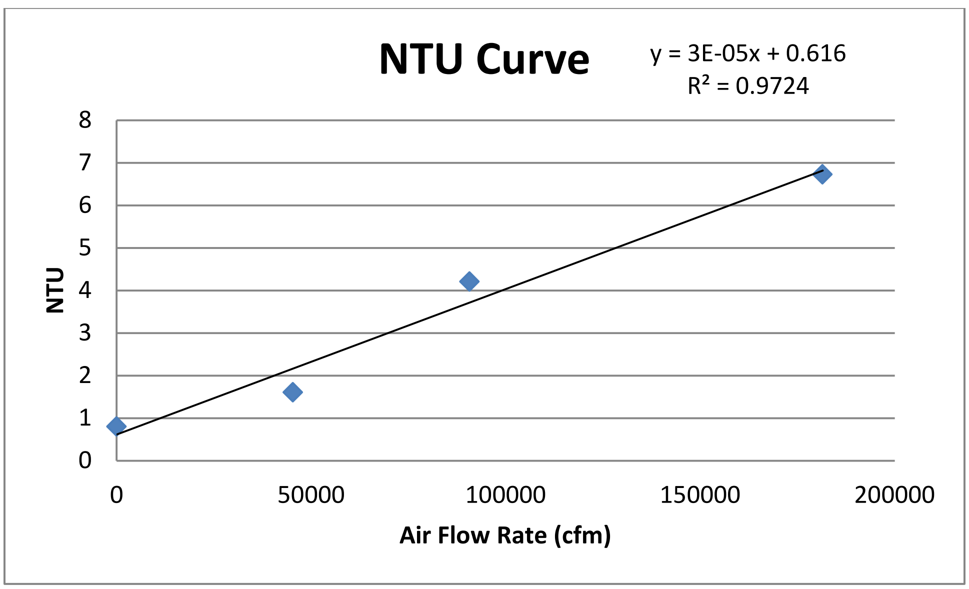


Figure 8.1: Plot of NTU vs. Air Flow Rate

The curve fit shows a strong linear correlation between air flow and NTU value. The equation of the line shown in Figure 8.1 is used in the new VFD cooling tower model to assign a cooling tower NTU value for a given flow rate. Next a fan curve for a variable frequency drive fan is generated using fan affinity laws (Equation 38.) for fan horsepower and flow. For this model, the VFD fan is specified to operate in the same air flow range as the previous two speed fans.

$$Fan\ Power\ Affinity\ Law \qquad (38.)$$

$$\frac{P_1}{P_2} = \left(\frac{\dot{V}_1}{\dot{V}_2}\right)^3$$

Where: P_1 is the Old Fan Power.

P_2 is the New Fan Power.

$\dot{V}_1$ is the Old Volumetric Flow Rate.

$\dot{V}_2$ is the New Volumetric Flow Rate.

The fan curve is shown in Figure 8.2.

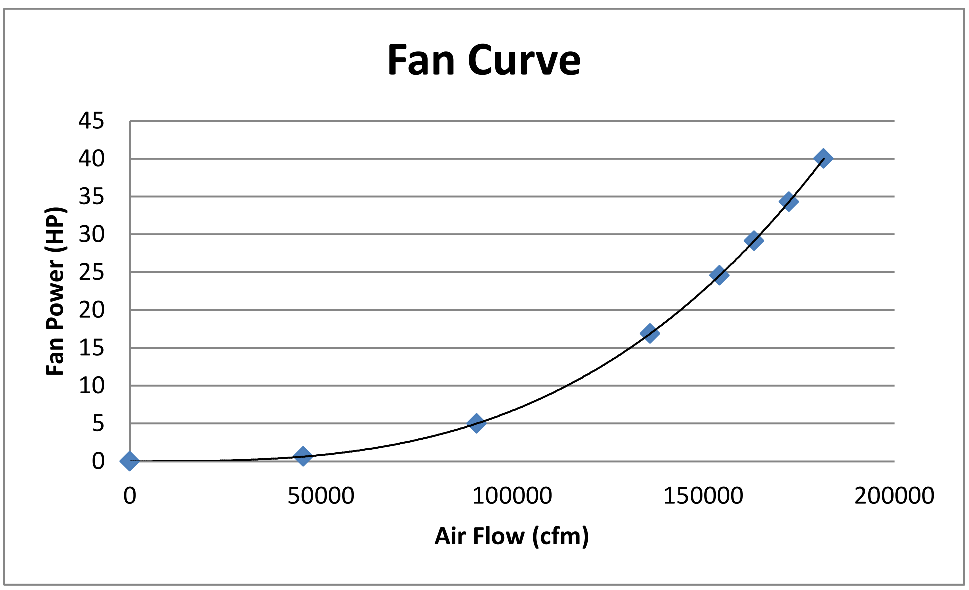


Figure 8.2: VFD Fan Curve

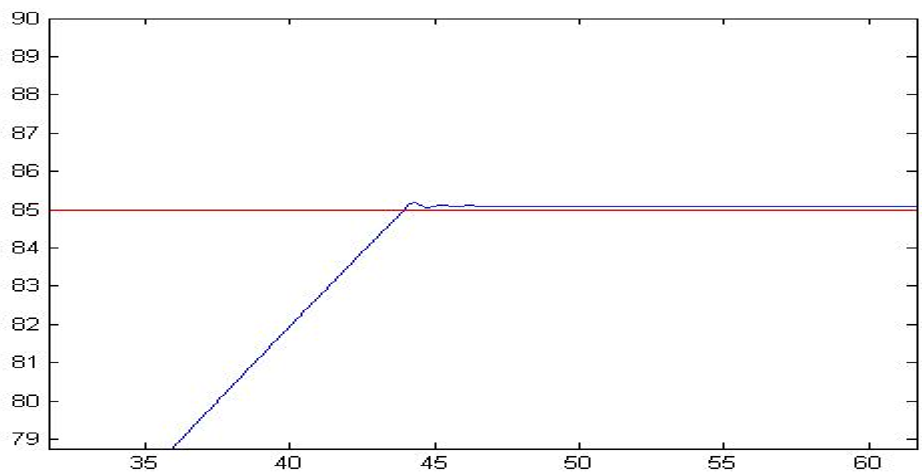
After the both the fan curve and NTU curve are created. A PID controller is added to the model to control the fan speed. The PID controller is based off a simple temperature controller and is set up to control fan speed based on the temperature difference between the inlet condenser water temperature to the condenser and the set point. The controller is shown in the MATLAB code below.

```
error(n+1)=((SetPoint(n))-(Tbasin(n)));
error_Ki(n+1)=error_Ki(n)+.5*dt*(error(n+1)+error(n));
error_Kd(n+1)=(error(n+1)-error(n))/dt;

Power(n+1)=abs((P*error(n+1))+(I*error_Ki(n+1))+(D*error_Kd(n+1)));
```

The integral error was numerically integrated with MATLAB and the derivative error was numerically differentiated in MATLAB. The proportional, integral, and derivative constants were tuned to give the best control. With the above changes made, the variable speed drive fan is simulated using the new variable speed drive cooling tower model. The MATLAB program can be found in Appendix D. Figure 8.3 on the next page shows the difference in control between a variable speed drive cooling tower fan and a two-speed system.

VFD Control



2 Speed Control

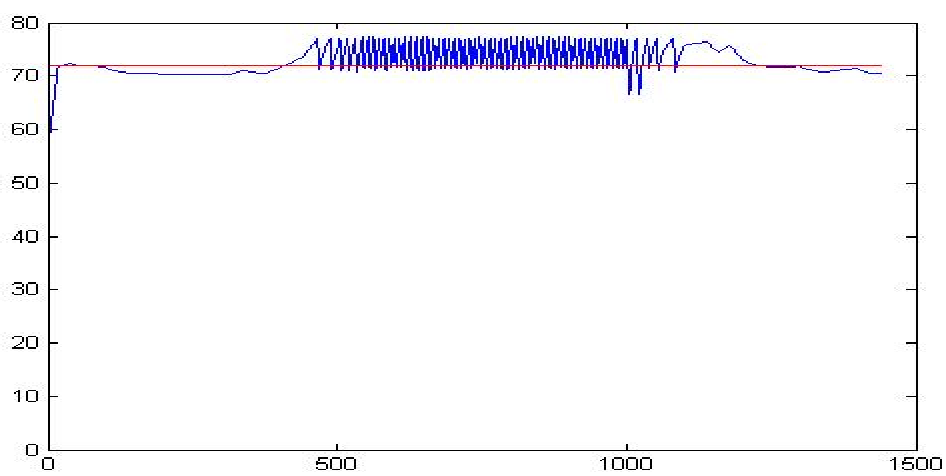


Figure 8.3: Two-Speed / VFD Control Comparison

Figure 8.4 shows the performance of the VFD fan versus a two speed system over the course of one month.

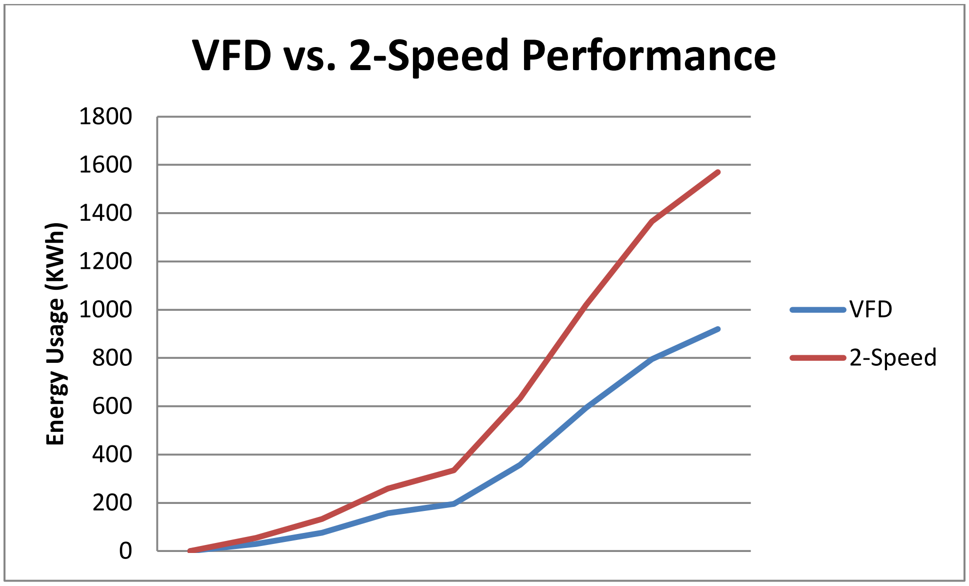


Figure 8.4: VFD vs. Two Speed Fan Energy Consumption

8.6.4.1.6 VFD Cooling Tower Fan Savings

Placing a VFD drive on the cooling tower offers both a fan electricity savings and chiller electricity savings if the condenser water set point is lowered. For this savings analysis the condenser water set point at the previously recommendation of 72 degrees Fahrenheit. The VFD cooling tower model MATLAB program found in Appendix D was used to estimate possible savings of installing VFD drives on the cooling tower. Table 8.15 shows only the fan energy savings for a condenser water set point of 85 and 72 degrees Fahrenheit. Usage savings are in kilowatt hours.

Table 8.15 Potential Energy Savings of VFD on Cooling Tower

Total CT Fan Usage Savings	**85**	**72**
2008	9778	8056.0
2009	8477	7224.0
2010	9451	7135.0
Average	9235	7471.0
Cost Savings Per Year	**$600.28**	**$485.62**

Table 8.16 lists the possible savings of installing VFD drives in tandem with lowering the condenser water set point.

Table 8.16: Potential Energy Savings of VFD plus Lower Condenser Water Set Point

2008 VFD on CT Savings	**85**	**80**	**75**	**72**	**70**	**65**
Total Pump Usage	515027.3	515027.3	515027.3	515027.3	515027.3	515027.3
Total CT Fan Usage	55556.8	83973.9	120093.3	135509.1	146223.0	158098.8
Total Chiller Usage	1773814.7	1696292.6	1631152.3	1601471.8	1585703.8	1572198.7
Total Plant Usage	2344398.2	2295293.3	2266272.3	2252007.7	2246953.5	2245324.2
Usage Cost	152385.9	149194.1	147307.7	146380.5	146052.0	145946.1
Demand Cost	28739.7	27868.5	27450.0	27230.0	27180.7	27221.2
Total Cost	181125.5	177062.5	174757.7	173610.5	173232.6	173167.3
Savings From Design	5840.0	9903.0	12207.8	13355.0	13732.9	13798.2
Percent From Design	3.1	5.3	6.5	7.1	7.3	7.4

2009 VFD on CT	**85**	**80**	**75**	**72**	**70**	**65**
Total Pump Usage	509132.3	509132.3	509132.3	509132.3	509132.3	509132.3
Total CT Fan Usage	56713.7	88072.7	125139.3	141057.7	152698.2	162940.2
Total Chiller Usage	1806265.9	1727930.1	1655935.6	1625991.1	1610315.0	1597394.3
Total Plant Usage	2376572.9	2325142.1	2296662.7	2279995.6	2282408.5	2273281.3
Usage Cost	154477.2	151134.2	149283.1	148199.7	148356.5	147763.3
Demand Cost	29141.8	28189.3	27715.9	27456.7	27355.8	27301.7
Total Cost	183619.0	179323.5	176999.0	175656.4	175712.4	175065.0
Savings From Design	5136.8	9432.4	11756.9	13099.5	13043.5	13690.9
Percent From Design	1.8	4.1	5.3	6.0	6.0	6.4
Max Best Case Savings	14577.8654					
Percent Savings	7					

2010 VFD on CT	**85**	**72**	**65**
Total Pump Usage	514832.34	514832.34	514832.34
Total CT Fan Usage	64803.24	142737.76	165496.29
Total Chiller Usage	1879291.4	1707308.915	1662958.43
Total Plant Usage	2453233.98	2359186.015	2337594.06
Usage Cost	159460.2087	153347.091	151943.614
Demand Cost	29481.165	27942.6	27609.3825
Total Cost	188941.3737	181289.691	179552.996
Savings From Design	5696.7263	13348.40903	15085.1036
Percent From Design	2.926829999	6.858065828	7.75033439
Max Best Case Savings	15085		
Percent Savings	7.75		

If the condenser water set point were set at the recommended 72 degree Fahrenheit, then an average savings of 6.65% percent can be expected. That equates to an average savings of about $13,250 per year. The majority of that savings comes from the improved chiller efficiency made possible by better condenser water control of VFD cooling tower fans.

Economic Analysis:

The capital cost to install VFD drives on the RMC cooling tower is an estimated $20,000 dollars. VFD drives can save an estimated $13,250 dollars per year in electricity costs if implemented correctly. Table 8.17 shows the result of a simple economic analysis for this proposed project.

Table 8.17 VFD on Cooling Tower Economic Analysis Results

Economic Analysis	
Simple Payback	1.5 years
5 year MARR	59%
10 year MARR	65%

A simple payback under two years and a high MARR makes this recommendation very economical.

8.6.4.2 Main Hospital Transport System Capital Intensive Cost Reduction

8.6.4.2.1 Variable Volume Chilled Water Loop

The current main hospital chilled water transport system is a constant volume, three-way valve system. The system is designed to have enough capacity to meet peak demand. During off peak conditions, the system cannot compensate to reduce energy costs. A variable volume 2-way valve system would allow the chilled water pumps to throttle back with decreasing demand, creating energy savings. The proposed system would be a primary-secondary chilled water loop. Figure 8.5 illustrates a primary secondary system.

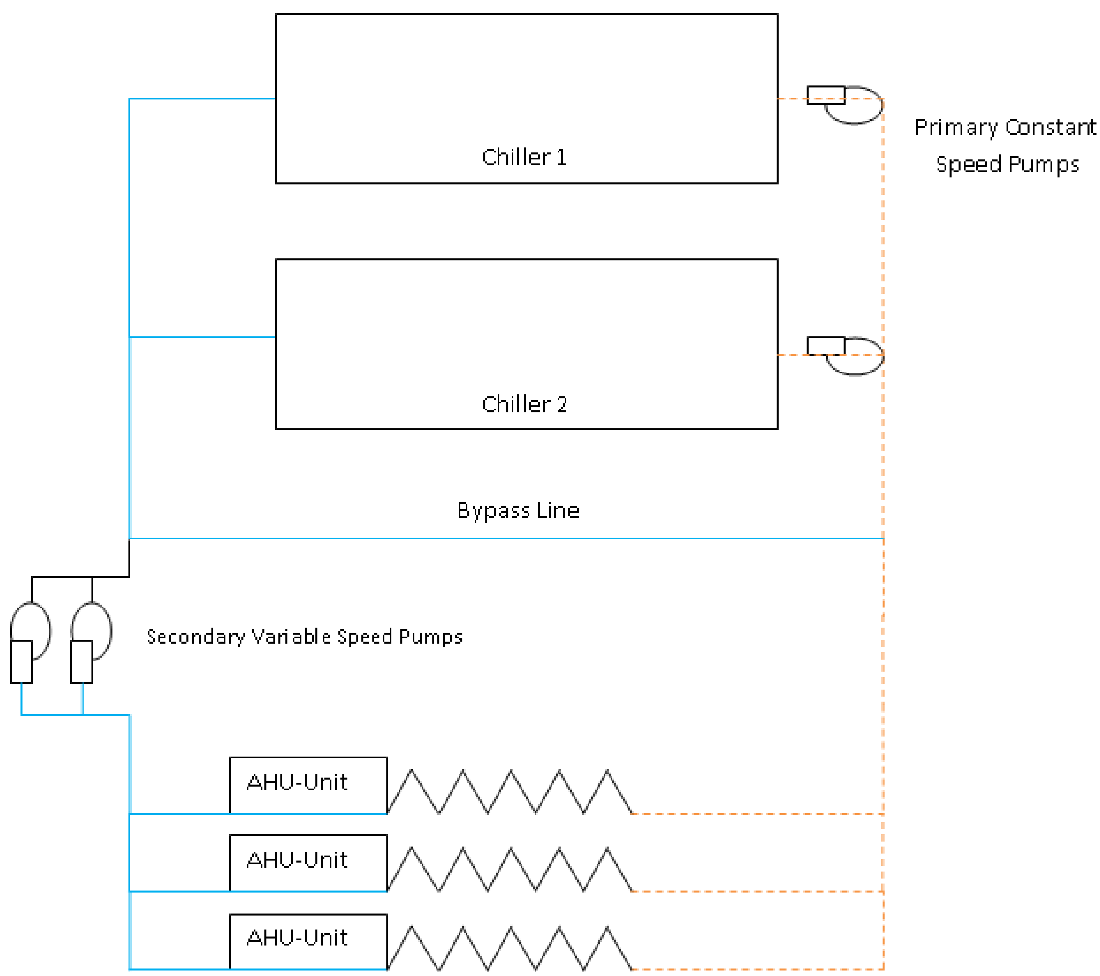


Figure 8.5: Primary / Secondary Chilled Water Loop Schematic

The proposed primary-secondary system utilizes two way valves to control water flow through each air handler unit instead of three way valves. It includes a primary constant volume loop which continuously circulates water through the chillers with constant volume pumps. A secondary supply loop would use VFD pumps to pull water from the primary loop and pump it out to the air handler cooling coils. At peak demand, the air handler two-way valves are completely open requiring maximum flow. As demand decreases, the two way valves begin to close increasing the water pressure in the system. As the pressure rises, the VFD pumps compensate by throttling back reducing the chilled water flow to maintain a constant system pressure. The reduced flow creates energy savings during off peak times. The proposed system would entail replacing the three – way valves with two valves, removing the constant volume chilled water pumps, and replacing them with two constant volume primary pumps and two variable volume VFD secondary pumps.

8.6.4.2.2 Variable Volume Chilled Water Loop Analysis

In order to implement this suggestion, new variable volume pumps must be specified to replace the current constant volume pumps. The two primary loop pumps will remain constant volume because the current main hospital chillers require constant flow. The secondary loop pumps will be variable volume. The primary and secondary pumps should be plumbed in a parallel configuration with the correct isolation valves in place. A simple pump specification and analysis is given below.

New Pump Specification:

Chiller 1 mass flow rate of water = 314,874 lb/hr

Chiller 1 Design Pressure Drop = 15.2 psig

Chiller 2 mass flow rate of water = 431,827 lb/hr

Chiller 2 Design Pressure Drop = 12.7 lb/hr

Equation (39.) is used to find the Required Ideal Pump Horsepower.

$$HP_{Required\ ideal} = \frac{\dot{m}_{water}\frac{lbm}{hr} * \Delta P_{Across\ Chiller}\frac{lbf}{in^2} * 144\frac{in^2}{ft^2}}{\rho_{water}\frac{lbm}{ft^3} * 778\frac{lbf*ft}{BTU} * 2545\frac{BTU}{hr*HP}} \qquad (39.)$$

Primary Loop Pump Horsepower:

$$Chiller\ 1\ HP_{Required\ Ideal} = \frac{314874\,\frac{lbm}{hr} * 15.2\,\frac{lbf}{in^2} * 144\,\frac{in^2}{ft^2}}{62.2\,\frac{lbm}{ft^3} * 778\,\frac{lbf*ft}{BTU} * 2545\,\frac{BTU}{hr*HP}} = 5.57HP$$

$$Chiller\ 2\ HP_{Required\ Ideal} = \frac{431827\,\frac{lbm}{hr} * 12.7\,\frac{lbf}{in^2} * 144\,\frac{in^2}{ft^2}}{62.2\,\frac{lbm}{ft^3} * 778\,\frac{lbf*ft}{BTU} * 2545\,\frac{BTU}{hr*HP}} = 6.39HP$$

Assume 80% Pump Efficiency.

$$Chiller\ 1\ HP_{Actual} = \frac{5.57}{0.8} = 7HP$$

$$Chiller\ 2\ HP_{Actual} = \frac{6.39}{0.8} = 8HP$$

Note: This analysis neglected the length of chilled water piping contained in the primary loop.

Secondary Loop Pump Power:

$$Secondary\ Loop\ HP = Old\ Pump\ HP - Primary\ Loop\ Hp = 45 - 16 = 37HP$$

The primary loop will require two eight horsepower constant volume pumps. The secondary loop will require two variable speed pumps of at least seventeen horsepower each. To find the possible savings from reducing the chilled water flow, an average cooling load must be established. Table 8.18 contains the average main hospital cooling loads for each month.

Table 8.18 Average Main Hospital Cooling Load

Average Main Hospital Monthly Cooling Load	
	Cooling Load (Tons)
Jan	159
Feb	206
Mar	228
Apr	313
May	392
Jun	521
Jul	521
Aug	506
Sep	360
Oct	254
Nov	225
Dec	190
Overall Average	**320**

The overall average cooling load is 320 tons. From the information contained in table above the secondary pump power can be estimated for different loads. At the average load of 320 tons only one chiller and one secondary pump would be online.

$$Secondary\ Pump\ HP\ at\ Average\ Load = \frac{Average\ Load}{Maximum\ Load} * HP_{Secondary\ Pump\ @Full\ Load} \quad (40.)$$

$$Secondary\ Pump\ HP\ at\ Average\ Load = \frac{320\ Tons}{660\ Tons} * 17HP = 8.2HP$$

It can be seen from Table 8.17 that the minimum cooling load is around 200 tons and the maximum average load is around 500 tons. Both secondary pumps would be online above average cooling load.

$$Secondary\ Pump\ HP\ at\ Below\ Average\ Load = \frac{200\ Tons}{660\ Tons} * 17HP = 5\ HP$$

$$Secondary\ Pump\ HP\ at\ Above\ Average\ Load = \frac{500\ Tons}{660\ Tons} * 34HP = 25\ HP$$

For this simple analysis, it was estimated that there are approximately 4,446 hours at average load, 2,208 hours near maximum load, and 2,088 hours near minimum load.

8.6.4.2.3 Variable Volume Chilled Water Loop Savings

The savings analysis uses the information calculated above to make savings estimates for this recommendation. The current total pump power for the main hospital chilled water plant is forty five horsepower. Each chiller has a dedicated chilled water pump that cycles on and off with its respective chiller. It was found that under the current operational conditions, both chilled water pumps are operational for a total of 3,738 hours and only one pump is operational for the remaining 5,022 hours.

Current Estimated Operating Costs:

$$Usage\ Cost = \left(\left(45HP * 0.746\frac{kw}{HP} * 3738hrs\right) + \left(25HP * 0.746\frac{kw}{HP} * 5022hrs\right)\right) * 0.065\frac{\$}{kwh} = \$14{,}244\ per\ year$$

$$Demand\ Cost = [(34kw * 5months) + (18kw * 6months)] * 5.25\frac{\$}{kwh} = \$1{,}459\ /yr$$

$$Total\ Current\ Operating\ Cost = \$14{,}244 + \$1{,}459 = \$15{,}703$$

New Estimated Operating Costs:

Primary Pump Costs:

$$Primary\ Usage\ Cost = \left(\left(16HP * 0.746\frac{kw}{HP} * 3738hrs\right) + \left(8HP * 0.746\frac{kw}{HP} * 5022hrs\right)\right) * 0.065\frac{\$}{kwh} = \$4{,}777\ per\ year$$

Secondary Pump Costs:

$$Average\ Load\ Cost = 8.2HP * 0.746\frac{kw}{HP} * 4464hrs * 0.065\frac{\$}{kwh} = \$1{,}774\ per\ year$$

$$Above\ Average\ Load\ Cost = 25HP * 0.746\frac{kw}{HP} * 4464hrs * 0.065\frac{\$}{kwh} = \$2{,}676\ per\ year$$

$$Below\ Average\ Load\ Cost = 5HP * 0.746\frac{kw}{HP} * 2088hrs * 0.065\frac{\$}{kwh} = \$506\ per\ year$$

$$Demand\ Cost = (\$252\ + \$378 + \$204) = \$834$$

$$New\ Total\ Operational\ Cost = Primary\ Pump\ Cost + Secondary\ Pump\ Cost$$

$$= \$4{,}777 + \$5{,}790 = 10{,}567\ per\ year$$

$$Savings = Current\ Cost - New\ Cost = \$5{,}136\ per\ year$$

Converting to a variable volume chilled water system will save RMC an estimated 76,274 kwh per year which equates to a chiller plant cost savings of $5,136 per year. This project recommendation was also simulated with EQUEST software. EQUEST predicted a 76,000 kwh savings with demand savings of 90 kw worth an estimated $5,412 per year.

Economic Analysis:

This project will require a retrofit of multiple components and a substantial capital cost. Table 8.19 below contains estimated costs of each component.

Table 8.19: Capital Costs of Variable Volume Loop

Cost of Two 8 HP Primary Constant Volume Pumps …………………………. $ 2,000

Cost of Two 20 HP Secondary Variable Speed Pumps including VFD drive… $ 8,000

Cost of replacing three way valves with two way valves……………………... $20,000

Installation and miscellaneous cost 60 hrs/$100 per hr ……………………….. $ 6,000

Total Cost……………………………………………………………………$36,000

Using the costs from Table 8.19 and the savings calculated above gives the return values that are listed in Table 8.20.

Table 8.20: Variable Volume Loop Economic Analysis Results

Economic Analysis	
Simple Payback	7 years
10 year MARR	7%

This project is outside of RMC's generally accepted payback period in its current form. However, three way valves can be made to function like two way valves in certain cases. It only requires that the manual bypass valve be completely closed. If the three way valves at RMC can function like two way valves, then that reduces the project's capital cost by $20,000 dollars. That reduces the simple payback to three years which makes this investment more attractive.

8.6.4.2.4 VFD Fan Air Handler Upgrade

The variable volume air handlers found in the main hospital are equipped with constant speed fans. They modulate air flow with inlet guide vanes. This set up does allow for some energy conservation during off peak periods because the inlet guide vanes do effectively unload the fan. Savings are limited because the fan rpm does not dramatically change. However, if VFD drives were retrofitted even more energy savings could be enjoyed because of the VFD’s ability to reduce fan rpm and therefore save even more fan power.

8.6.4.2.5 VFD Fan Air Handler Fan Upgrade Analysis

Accurately analyzing variable volume fan systems is difficult because one has to know how all variable volume terminal boxes are behaving at once in order to determine if the fan can be throttled back. If the variable volume air handler serves a large zone with many terminal boxes, many different conditions can exist within the zone. EQUEST software is capable of looking at all variables at once and is ideal for analyzing the effects of variable air volume systems. EQUEST was used to estimate the possible savings of a VFD retrofit on nine variable air volume air handlers located in the main hospital. These nine air handlers serve the largest sections of RMC and account for a large portion of the main hospitals fan energy use. Baseline EQUEST simulations have shown that ventilation fans account for 19 percent or roughly 1.2 million kwh per year of the facilities electricity use.

8.6.4.2.6 VFD Fan Air Handler Fan Upgrade Savings

The average ventilation fan energy usage for the main hospital is 1,268,000 kwh per year. The demand ranges from 150 to 170 kw per month for a total of 1,920 kw per year.

$$Current\ Fan\ Usage\ Cost = 1{,}268{,}000\ kwh * 0.065\frac{\$}{kwh} = \$82{,}420\ per\ year$$

$$Current\ Fan\ Demand\ Cost = 1{,}920kw * 5.25\frac{\$}{kw} = \$10{,}080\ per\ year$$

$$Total\ Fan\ Cost = \$92{,}500\ per\ year$$

The results of the EQUEST simulation suggest that variable speed drive fans can reduce ventilation fan usage by 102,000 kwh per year and reduce demand by 12 kw.

$$VFD\ Fan\ Usage\ Savings = 102{,}000\ kwh * 0.065\frac{\$}{kwh} = \$6{,}630\ per\ year$$

$$VFD\ Fan\ Demand\ Savings = 12kw * 5.25\frac{\$}{kw} = \$63\ per\ year$$

$$Total\ Estimated\ Savings = \$6{,}693\ per\ year$$

Economic Analysis:

The capital cost of retrofitting VFD drive fans is roughly $3,000 per small horsepower application and $6,000 per large horsepower application. The total capital cost would be an estimated $36,000.

Using the savings calculated above and the equipment described above yields the simple economic return shown in Table 8.21.

Table 8.21: VFD AHU Retrofit Economic Analysis Results

Economic Analysis	
Simple Payback	5.4 years
7 year MARR	6.8%

The simple payback period is just outside the ideal RMC payback period but the project does offer an attractive rate of return of 6.8%.

8.6.5 Main Hospital Control System Capital Intensive Cost Reduction

8.6.5.1 Main Hospital Control System Upgrade

The existing HVAC control system consists of an antiquated Siemens PLC/PID control front end with pneumatic actuators as well as a second energy management system controlling some processes. The Auburn Engineering Team has found several deficiencies with the current control system's sequence of operations that has led to unnecessary energy costs. This system allows for simultaneous heating and cooling to occur sporadically throughout the facility. Simultaneous heating and cooling within the air handler greatly increases energy costs. Another disadvantage of the system is its inability to be updated with current energy efficient control sequences and strategies that have since been developed. The new proposed system will eliminate simultaneous heating and cooling, improve costly outdoor air control, and allow for temperature and static pressure reset. With this new proposed HVAC controls changes, RMC will realize reduced energy consumption and improved indoor air quality.

8.6.5.1.1 Main Hospital Control System Upgrade Analysis

The new proposed control system will operate with a new sequence of operations. The new sequence requires several upgrades to be implemented. The new sequence can be found in Appendix F. To better control indoor air quality, it is recommended that RMC abandon the current outside air management strategy in favor of a system that manages outside air intake based on indoor carbon dioxide levels. Such a system would control the outside air dampers based on carbon dioxide levels measured by sensors placed in the return ductwork. The current strategy involves measuring the incoming airflow. Air flow measurement is inherently inaccurate making it difficult to maintain the correct outside air intake. An advantage of the new strategy is that the system would better manage outside air flow when zones are unoccupied. If zones were unoccupied, the system would shut down the outside air intake because it is not needed allowing for energy savings. It would also better manage periods of high occupancy because the outdoor intake could be properly adjusted to compensate for the extra occupancy of the zone.

Secondly, the new system will have a better zone thermostat management strategy. Some zones in the main hospital are unoccupied overnight. The proposed system will be equipped with zone temperature reset. During unoccupied times the zone thermostat set point is increased or decreased based on the season. Temperature reset can save energy in several ways. During the cooling season, the system would increase the room set point and disengage reheat. The increased set point decreases the cooling load and allow for the ventilation fans to throttle back to minimum amounts. Outside air intake is also reduced because the zone is unoccupied. No reheat saves natural gas usage. During the heating season, the zone set point is reduced thereby reducing the heating load and also allowing the ventilation fans and outdoor air levels to be reduced to minimum levels.

The new proposed system also includes discharge air temperature reset. If outdoor conditions permit and no dehumidification is required, the system increases the discharge air temperature effectively reducing reheat and cooling loads. This system works in tandem with the chiller plant controls and also raises the chilled water temperature when conditions permit.

Most importantly, the new system eliminates the sporadic simultaneous heating and cooling within the air handler that currently exists. This action reduces both cooling and heating energy requirements. The new sequence eliminates this issue by separating the control of the pre-heat and cooling coil valves. The valves are currently controlled in tandem by the same PID loop reading the discharge air sensor. The new sequence controls discharge air temperature by modulating only the cooling coil valve. The new system controls the cooling coil based on the discharge air sensor. The heating valve is controlled based on a mixed air sensor placed before the cooling coil. It only allows the preheat valve to open if the mixed air temperature falls below the discharge air set point. The main hospital generally always maintains a cooling load, so it practically eliminates the use of the preheat coil.

8.6.5.1.2 Main Hospital Control System Upgrade Savings

EQUEST is the primary tool used in this savings analysis. The savings analysis for the control system upgrade first assumed that the minimum air handler discharge supply temperature is raised to 55 degrees Fahrenheit on all possible units and that the discharge air temperature reset maximum is 60 degrees Fahrenheit. The discharge air temperature reset is based on outside air temperature and humidity. If outside air temperature and relative humidity are less than 60 degrees Fahrenheit and 60 percent relative humidity, the discharge air temperature is reset to 60 degrees Fahrenheit.

Secondly, it assumes that the chilled water supply temperature is reset a maximum of ten degrees Fahrenheit if the outside air temperature and relative humidity fall below 60 degrees Fahrenheit and 60 percent relative humidity. Third, it assumes that the unoccupied zone thermostat reset temperature for cooling is 78 degrees Fahrenheit and 65 degrees Fahrenheit for heating. Zone thermostat is applied to one zone on the first floor. The model assumes an unoccupied period of 12 hours. Lastly, it assumes that minimum outdoor air is controlled via carbon dioxide sensors placed in the return ductwork. The minimum carbon dioxide level is set at 800 ppm which is below the ASHRAE accepted minimum of 1000 ppm. The new control strategy is simulated for the main hospital for one year. The EQUEST simulation results in a 377,000 kwh reduction in space cooling. The increased supply temperature caused by the discharge air temperature reset creates a 77,000 kwh ventilation fan usage penalty. The net electricity savings is an estimated 300,000 kwh. Also a net demand savings of 220 KW is predicted. EQUEST predicts an estimated reheat savings due to discharge air temperature reset of 3.5 billion BTU's. Savings due improved outside air control, discharge temperature reset, unoccupied mode, and chilled water reset are shown below:

Space Cooling Energy Savings = 377,000 kwh

Ventilation Fan Penalty = 77,000kwh

Net Savings = 300,000 kwh

$$Net\ Electricity\ Savings = 300{,}000\ kwh * 0.065\frac{\$}{kwh} = \$19{,}500\ per\ year$$

$$Net\ Demand\ Savings = 220KW * 5.25\frac{\$}{kwh} = \$1{,}155\ per\ year$$

Savings Due to Simultaneous Heating and Cooling Elimination:

At any given time there is an average of 10,000 cfm of air that is being over heated and over cooled by 20 degrees Fahrenheit. That results in 216,000 BTU/hr of waste for both heating and cooling. Below is a simple First Law savings calculation.

$$Heating\ Savings = 10{,}000\frac{ft^3}{min} * 0.075\frac{lbm}{ft^3} * 0.24\frac{BTU}{lbm°F}(20°F) * 60\frac{min}{hr} * 8760\frac{hr}{yr} * 6\frac{\$}{10^6 BTU}$$

$$= \$11{,}350\ per\ year$$

$$Cooling\ Savings = \frac{10{,}000\frac{ft^3}{min} * 0.075\frac{lbm}{ft^3} * 0.24\frac{BTU}{lbm°F}(20°F) * 60\frac{min}{hr} * 8760\frac{hr}{yr} * .7\frac{kw}{Ton} * 0.065\frac{\$}{kwh}}{12{,}000\frac{BTU/hr}{Ton}}$$

$$= \$7{,}174\ per\ year$$

$$Cooling\ Demand\ Savings = 12.6kw * 5.25\frac{\$}{kw} * 12\ months = \$794\ per\ year$$

$$Total\ Simultaneous\ Heating\ \&\ Cooling\ Savings = \$19{,}318\ per\ year$$

$$Total\ Control\ System\ Estimated\ Savings = \$39{,}973\ per\ year$$

Economic Analysis:

The proposed control system upgrade requires a substantial retrofit of new sensors, actuators, control hardware, and software. The estimated capital costs are shown in Table 8.22.

Table 8.22: Control System Upgrade Capital Costs

Cost of energy management system/sensor replacement/upgrades: ……..$ 100,000

Cost of miscellaneous hardware improvements, consultants, etc.……….. $100,000

Total Estimated Capital Cost: ……………………………………………..$200,000

Based on these data, simple economic analysis results are shown in Table 8.23.

Table 8.23: Control System Upgrade Economic Analysis Results

Economic Analysis	
Capital Cost	$200,000.00
Yearly Savings	$40,000
Simple Payback	5 years
7 year MARR	9.20%

This project is just outside the accepted RMC payback standards, but the control system at RMC is outdated and needs upgrade. This project is very economical and will allow for other future energy saving upgrades in the future that the current system cannot.

8.6.6 POB Capital Intensive Energy Cost Reduction Strategies

8.6.7 POB HVAC Capital Intensive Cost Reduction

8.6.7.1 POB Generation System Capital Intensive Cost Reduction

8.6.7.1.1 POB Chilled Water Link

Air cooled chillers are less efficient than the liquid cooled chillers used for the hospital. Air cooled chillers typically have a COP of two to three while liquid cooled chillers have a COP of four to five. At present, the facility chillers are located in two buildings and are not connected. The main hospital building is cooled with two water-cooled chillers while the Professional Office Building (POB) is cooled with air-cooled chillers. If the entire facility were put on a consolidated chilled water loop, the less efficient air cooled chillers could be completely or partially replaced by the more efficient water cooled chillers. Also the hospital plate and frame heat exchanger could be utilized by the POB in winter months. It is important to note that in order to achieve the desired savings, the hospital and POB supply water temperatures should be set at the proper levels. Another benefit of a consolidated chill water loop is redundancy. If there was a problem with the liquid cooled chillers, the air-cooled chillers could be used. Adequate cooling could be maintained while repairs were being made.

8.6.7.1.2 POB Chilled Water Link Analysis

The main hospital chiller plant has a total capacity of 660 tons with an overall chiller efficiency of 0.7 kilowatts per ton. Its average chiller load is 320 tons. The POB has a total chiller capacity of 360 tons with a chiller efficiency of 1.3 to 1.5 kilowatts per ton based on which type of chiller is online. POB average load is around 200 tons. During most of the year, the main hospital has enough capacity to support both facilities. For this analysis, curves for chiller load based on outside air temperature are programmed into a MATLAB simulation which looks at weather data and calculates an overall cooling load for both facilities. If that overall cooling load is within the hospitals cooling capacity, it simulates shifting the load to the main hospital chiller plant. It then calculates the net savings that could be achieved. The MATLAB program can be found in Appendix D.

8.6.7.1.3 POB Chilled Water Link Savings

Table 8.24 below contains the net savings that resulted from the MATLAB simulation.

Table 8.24 Potential Savings for POB Chilled Water Link

Savings Potential for Central Chilled Water Plant		
	Usage Savings in KWH	**Demand Savings in KW**
Jan	39796	96
Feb	43095	102
Mar	51930	54
Apr	57147	54
May	58630	59
Jun	53693	64
Jul	56243	65
Aug	60603	64
Sep	60362	59
Oct	51362	54
Nov	45810	99
Dec	48057	96
Total Savings	**626728**	**866**
Cost Savings -$	**$40,737**	**$4,547**
Total Cost Savings	**$45,284**	

The simulation resulted in a total savings of $45,284 per year. A new 10 horsepower chilled water booster pump will need to be added to overcome the added resistance of the new chilled water piping that would link the two facilities.

$$Cost\ to\ Operate\ Booster\ Pump = 10HP * 0.746\frac{kw}{HP} * 8760hrs * 0.065\frac{\$}{kwh}$$

$$= \$4{,}247\ per\ year$$

Taking into the account the booster pump makes a net overall savings of $41,037 dollars per year.

Economic Analysis:

The capital cost of implementing this project is an estimated $114,000 for installation of new chilled water pipe to link the facilities and also for the new pump. Table 8.25 shows the results of the economic analysis.

Table 8.25 POB Chilled Water Link economic Analysis Results

Economic Analysis	
Capital Cost	$114,000.00
Yearly Savings	$41,037
Simple Payback	2.7 Years
5 year MARR	23.43%

8.6.8 POB End Use System Capital Intensive Cost Reduction

8.6.8.1 Pool Cover

RMC operates two indoor rehabilitation pools that currently are uncovered even when not in use. Since the pools are operated at elevated temperature, significant heat loss and evaporation occurs. A pool cover can significantly reduce swimming pool heating costs associated with heat loss and evaporation of water into conditioned air. The cover provides and insulating, thermal and vapor barrier.

8.6.8.2 Pool Cover Analysis/Savings

The tables below give the energy savings per year due to the new cover. These savings were calculated using a "Heating & Dehumidification Costs due to Evaporation from Swimming Pools" spreadsheet from www.energysavers.gov.

Table 8.26 Large Pool Energy Savings

Item	**Baseline**	**With Cover**
Water temp (^oF)	88	88
Room air temp (^oF)	80	80
Room relative humidity (%)	30%	30%
Hours per year uncovered	8760	2920
Efficiency or COP of water heater	0.8	0.8
Savings per year		$6,036

Table 8.27 Small Pool Energy Savings

Item	Baseline	With Cover
Water temp (^oF)	94	94
Room air temp (^oF)	80	80
Room relative humidity (%)	30%	30%
Hours per year uncovered	8760	2920
Efficiency or COP of water heater	0.8	0.8
Savings per year		$972

It was calculated that the evaporation rate for the two pools is about 18.79 gallons per hour. At 5,840 covered hours per year results in saving approximately $300.00 dollars per year in sewage charges and approximately $150 dollars per year in makeup water. The total savings potential of the two pool covers is $7,458 per year.

Economic Analysis:

Table 8.28 contains the result of a simple economic analysis for the pool cover project.

Table 8.28 Pool Cover Economic Analysis Results

Economic Analysis	
Capital Cost	$13,641.00
Yearly Savings	$7,458
Simple Payback	1.8 years
3 year MARR	29.50%

8.7 Summary and Evaluation of Capital Intensive Cost Reduction Strategies

A total of nine capital intensive cost reductions strategies are discussed in this section. All nine suggestions and the economics associated with each are listed in Table 8.29.

Table 8.29: Non Capital Intensive Energy Cost Reduction Recommendations

Capital Intensive Energy Cost Reduction Recommendations				
Recommendation	**Potential Savings(per yr)**	**Associated Capital Cost**	**Simple Payback yr.**	**Rate of Return**
TPO Roofing Retrofit	$346	$111,648	100+	N/A
LED Lighting Retrofit	$41,393	$274,440	6.6	1.38% over 7 yr
Boiler Plant Upgrade	$50,000	$805,213	16	3.71% over 25 yr
VFD Fan Cooling Tower Retrofit	$13,250	$20,000	1.5	59% over 5 yr
Variable Volume Chilled Water Loop Retrofit	$5,136	$36,000	7	7% at 10 yr
VFD Airhandler Fan Retrofit	$6,693	$36,000	5.4	6.8% at 7 yr
Control System Upgrade	$40,000	$200,000	5	9.2% at 7 yr
Chilled Water Link	$41,037	$114,000	2.7	23% at 5 yr
Pool Cover	$7,458	$13,641	1.8	29.5% at 3 yr
Totals:	**$205,313**	**$1,610,942**	**7.84**	**4.67% at 10 yr**

The TPO roofing retrofit project doesn't make economic sense purely as a retrofit. If the current roof is at the end of its useful life, then the TPO roofing would be a good candidate for replacement. It still would not create substantial energy savings over the current built-up roofing design. LED lighting offers substantial energy savings potential, but it comes at a very expensive capital cost. The simple payback period and rate of return can be misleading because it doesn't account for the service life of the LED lighting. The service life of LED lighting is seven years. As soon as the investment begins to become profitable, the lighting reaches the end of its useful life and must be replaced at great cost. It is more economical to continue to operate the current fluorescent lamps even though they consume more energy and do not last as long. T-8 fluorescent lamps are 1/12 the cost of LED lamps. It is predicted that the cost of LED lighting will drastically decrease in the near future, but currently LED lighting is not economical. The boiler plant upgrade comes at a very substantial capital cost, but it is the most urgent equipment replacement needed at RMC. The current steam boiler system has dangerously surpassed its service life and should be replaced as soon as funding becomes available. The control system upgrade also requires substantial capital cost. It is important that this upgrade be implemented because the control system is the key to efficient operation for most of the HVAC systems. It will allow all HVAC related energy reduction strategies to make the maximum impact. Without tight reliable control, the current energy waste at RMC will most likely continue and suggested improvements will not perform as well as they should. The POB chilled water link also comes with a substantial cost, but the individual economics for this project are within the RMC standard investment return requirement of three years. The project will create a unified chilled water system at RMC and offer critical redundancy for both facilities if mechanical issues arise with any of the chillers. The variable volume chilled water loop retrofit has a lengthy payback by itself, but it would resolve the water balance issues currently impeding efficient chilled water loop operation at RMC. If the current three way valves in place at RMC, can be modified to act as two way valves as discussed earlier, the capital cost of this project will be reduced bringing the economics of the retrofit within RMC’s standard payback requirement of three years. The VFD fan retrofit for the cooling tower has strong economics and falls within RMC’s payback requirements. The VFD fan retrofit for the variable volume air handlers falls outside the three year payback requirement, but if implemented it would work well with a new control system upgrade to increase energy efficiency at RMC. The VFD fan air handler retrofit becomes profitable after five years. The lifetime for the fans is much larger than five years. The pool cover is a simple project with strong economics. It will save heating energy as well as make up water usage. If

TPO roofing and LED lighting are eliminated because the benefit cost ratio is unsatisfactory and the boiler plant replacement is not considered because it is a required capital expenditure because the current system has exceeded it service life, then the total economics of the suggested recommendations look much better. Table 8.29 shows the adjusted economics with the aforementioned projects neglected or eliminated.

Table 8.30: Adjusted Capital Intensive Energy Cost Reduction Recommendations

Adjusted Capital Intensive Energy Cost Reduction Recommendations				
Recommendation	**Potential Savings(per yr)**	**Associated Capital Cost**	**Simple Payback yr.**	**Rate of Return**
VFD Fan Cooling Tower Retrofit	$13,250	$20,000	1.5	59% over 5 yr
Variable Volume Chilled Water Loop Retrofit	$5,136	$36,000	7	7% at 10 yr
VFD Airhandler Fan Retrofit	$6,693	$36,000	5.4	6.8% at 7 yr
Control System Upgrade	$40,000	$200,000	5	9.2% at 7 yr
Chilled Water Link	$41,037	$114,000	2.7	23% at 5 yr
Pool Cover	$7,458	$13,641	1.8	29.5% at 3 yr
Totals:	**$113,574**	**$419,641**	**3.69**	**11% at 5 yr**

The simple payback would be 3.69 years which is just outside RMC's requirements. If the three valves can be made to function as two way valves then the simple payback will move within the RMC requirements. If the savings from the non-capital intensive projects are also considered with the above adjusted recommendations, then the simple payback becomes 1.4 years. The economics of all the combined projects are shown in Table 8.31.

Table 8.31: Energy Cost Reduction Recommendations Combined

Energy Cost Reduction Recommendations Combined				
Recommendation	**Potential Savings(per yr)**	**Associated Capital Cost**	**Simple Payback yr.**	**Rate of Return**
Chilled Water Supply Temp. Reset	$4,400	$5,000	1.1	84% over 5 yr
Decrease Condenser Water Temp Set Point	$8,000	$0	N/A	N/A
Increase Free Cooling Time	$3,276	$0	N/A	N/A
Reduce OA Intake	$12,099	$0	N/A	N/A
Increase AHU Discharge Air Temp.	$15,025	$0	N/A	N/A
POB Chilled Water Temp. Reset	$1,891	$5,000	2.6	25.8 % over 5 yr
Increase POB AHU Discharge Ait Temp.	$20,575	$0	N/A	N/A
Cancer Center Chilled Water Supply Temp Reset	$344	$5,000	N/A	N/A
Combine Electricity Meters	$131,527	$5,000	0.04	N/A
VFD Fan Cooling Tower Retrofit	$13,250	$20,000	1.5	59% over 5 yr
Variable Volume Chilled Water Loop Retrofit	$5,136	$36,000	7	7% at 10 yr
VFD Airhandler Fan Retrofit	$6,693	$36,000	5.4	6.8% at 7 yr
Control System Upgrade	$40,000	$200,000	5	9.2% at 7 yr
Chilled Water Link	$41,037	$114,000	2.7	23% at 5 yr
Pool Cover	$7,458	$13,641	1.8	29.5% at 3 yr
Totals:	**$310,711**	**$439,641**	**1.4**	**49.5% over 3 yr**

Chapter 9: Current Implementation / Actual Results / Final Conclusions

9.1 Current Implementation

Currently four energy reduction strategies have been implemented at RMC. The strategies are all implemented at the main hospital facility and were completed by the end of 2010. No suggested energy reduction strategies have been applied to the POB or the Cancer Center. The four implemented energy reduction strategies that have been applied are as follows:

1. Increase Discharge Air Supply Temperature
2. Reduce Condenser Water Temperature Set Point
3. Increase Free Cooling
4. VFD Ventilation Fan Retrofit

It is suggested to increase the minimum discharge air supply temperature to 55 degrees Fahrenheit on the air handler units that did not serve critical zones. It is predicted that this energy reduction strategy will save an estimated 140,700 kwh of electricity worth $9,145 per year and save an estimated 900 million BTU's of natural gas usage worth $5,880. This recommendation comes at no capital cost. RMC also decreased their condenser water temperature set point to the suggested 72 degrees Fahrenheit. Analysis predicts that this energy reduction strategy will save 118,000 kwh of electricity worth $8,000. This suggestion is implemented at no capital cost. The plate and frame heat exchanger was cleaned and fill media in the cooling tower was replaced to increase cooling tower efficiency and allow for increased free cooling use. Savings Analysis predicts that increasing free cooling time will save an estimated 50,400 kwh of electricity worth $3,200. This suggestion is implemented at a $10,000 capital cost. RMC retrofitted VFD drives onto the nine suggested variable volume air handlers. Analysis projects that this suggestion will save 102,000 kwh per year worth $6,630 at a capital cost of $36,000. All implemented projects combine for a total projected energy savings of 411,100 kwh and 900 million BTU's of natural gas worth a total of $26,721 in electricity usage savings and $5,400 dollars in natural gas savings or around 6 percent. The total capital expenditure for these projects was $46,000 dollars with a projected simple payback of 1.43 years.

9.2 Actual Savings Results

The four energy reduction strategies were completed by the end of 2010 making 2011 the first full year to recognize any energy savings. The baseline from which 2011 energy usage is measured is established by averaging the energy usage from years 2008-2010. The average electricity usage for years 2008-2010 is found to be 6,713,485 kwh after averaging the total electricity usage for years 2008-2010 as found on RMC's energy logs. The average gas usage is found to be 12.3 billion BTU's for years 2008-2010. The energy usage for 2011 is found to be 6,255,475 kwh of electricity and 12.8 billion BTU's of natural gas.

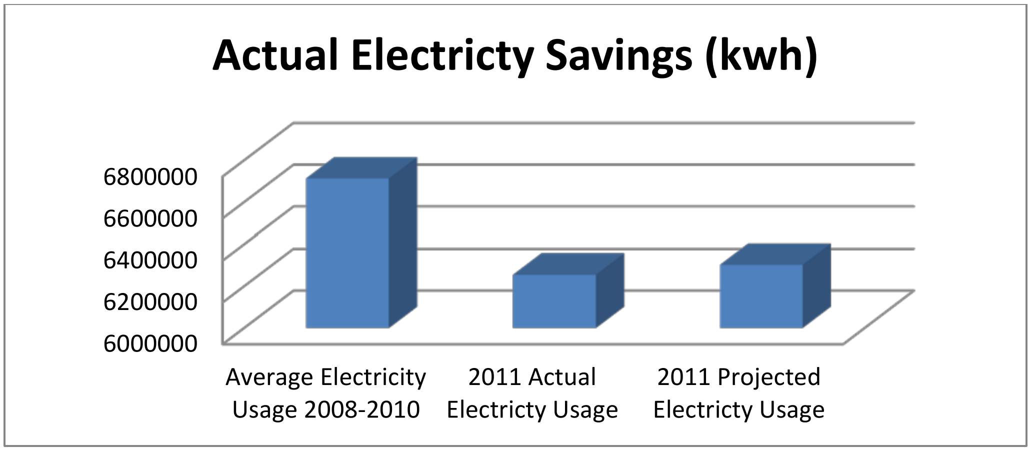


Figure 9.1: Actual Electricity Savings

The actual electricity savings seen at the main hospital is 458,010 kwh hours or about seven percent of the main hospital's energy cost. The actual results exceed the projected the savings by 46,910 kwh. The savings projections are conservative and could explain the difference. An alternative explanation is that over the process of the entire energy audit, awareness for energy waste was increased, so this extra savings is probably best attributed to the RMC staff. The projected electricity savings targets were met, however natural gas usage increased and the projected 900 billion BTU savings did not actually occur. The main hospital used 500 million more BTU's than the average. The most probable explanation for the increase is that RMC's boilers have greatly surpassed their recommended useful life causing their performance to decrease each year erasing any possible savings. Also, steam system waste could have increased beyond the levels they were at during the original analysis which would further contribute to the increase in gas usage. The actual electricity savings of 458,010 kwh is worth a total of $29,770. The increased gas usage cost RMC an extra $3,000 based on a natural gas price of six dollars per million BTU. Accounting for the gas penalty brings the total net savings to $26,770. RMC recovered 58 percent of their initial investment in the first year. If electricity savings continue at this level in the future and gas usage does not increase dramatically, RMC will have recouped their investment by the end of 2012 and will enjoy a projected rate of return of ten percent by the end of 2012.

9.3 Final Conclusions

9.3.1 Energy Audit Conclusions

The RMC energy audit was carried out with very little diversion from the original plan. The preliminary planning stage is very important for establishing the order and structure of the audit as well as the timing for data collection. The RMC audit experience showed the importance of acquiring up to date and accurate preliminary data before beginning the building survey because setbacks occurred due to inaccurate preliminary information. Ample time should be set aside during the preliminary data collection phase to ensure that all plans are current and all the equipment in each energy cost center is completely understood. The practice of dividing the facility into energy cost centers proved to be a good organizational structure for the energy audit. It allows the auditor to effectively organize and attack each section simultaneously because the equipment in each energy cost center is defined and it is understood how each energy cost center interacts with others. Also it makes the documentation phase flow smoothly. A complete building survey cannot be accomplished in a short period of time especially when dealing with the HVAC system. The data collected during the building survey serves as the basis of real time information about the facility being audited. The RMC audit practice of developing step by step data collection procedures proved to be very productive because it allows for extra man power to be utilized without a having to undergo a large amount of training. HVAC related equipment can be greatly affected by different weather conditions and it is important that energy auditors collect data for multiple weather conditions as in the RMC audit. If trended data is available, data collection for multiple conditions is easy to accrue, but in the case of RMC where trended data is not available, the data must be physically collected from the system. It is also good practice to proceed with caution when referencing equipment data logs kept by facility personnel. Its accuracy should be verified before using it in important calculations. When establishing a baseline energy model it is important to look at the facility as a whole. Most energy cost centers affect others and it is important to understand these interactions. Throughout the RMC audit, EQUEST proved to be good software to create baseline energy models and analyze recommendations because of its ability to analyze the facility as a whole based on weather conditions. When analyzing HVAC equipment it is crucial to have access to accurate weather data. The weather data should be fed into all models used to simulate HVAC equipment. Proper utilization of accurate weather data takes the guess work out HVAC equipment off peak analysis and provides the most accurate conclusions. Also when performing energy savings analysis on HVAC equipment, it is essential that the auditor fully understand the effects of the changes on the whole HVAC system. Oftentimes recommendations that produce savings in one area create a penalty in another. The auditor should make sure there is a net savings gain before recommending a energy reduction strategy. When making recommendations, ensure that the economics of each recommendation is solid. If the project isn't economical it should have some other benefit that justifies its need before it is recommended. Finally ensure accurate documentation off all phases during the entire audit. Information and the time spent to gather it is wasted without proper documentation.

9.3.2 Energy Cost Recommendation Conclusions

Currently RMC has implemented four of the energy cost recommendations. Analysis predicted a six percent reduction in main hospital energy costs due to those recommendations. RMC has enjoyed an actual seven percent reduction in energy costs in the main hospital. That is about a 2.4 percent reduction for the entire RMC campus. Table 9.1 below contains the remainder of the energy cost saving recommendations to be implemented.

Table 9.1: Additional Energy Cost Reduction Recommendations

Additional Energy Cost Reduction Recommendations to be Implemented				
Recommendation	**Potential Savings(per yr)**	**Associated Capital Cost**	**Simple Payback yr.**	**Rate of Return**
Boiler Plant Upgrade	$50,000	$805,213	16	3.71% over 25 yr
POB Chilled Water Temp. Reset	$1,891	$5,000	2.6	25.8 % over 5 yr
Increase POB AHU Discharge Ait Temp.	$20,575	$0	N/A	N/A
Cancer Center Chilled Water Supply Temp Reset	$344	$5,000	N/A	N/A
Combine Electricity Meters	$131,527	$5,000	0.04	N/A
VFD Fan Cooling Tower Retrofit	$13,250	$20,000	1.5	59% over 5 yr
Variable Volume Chilled Water Loop Retrofit	$5,136	$36,000	7	7% at 10 yr
Control System Upgrade	$40,000	$200,000	5	9.2% at 7 yr
Chilled Water Link	$41,037	$114,000	2.7	23% at 5 yr
Pool Cover	$7,458	$13,641	1.8	29.5% at 3 yr
Totals:	**$311,218**	**$1,203,854**	**3.86**	**9.2% over 5 yr**

If the above recommendations are successfully implemented, RMC could potentially achieve an additional 25 percent energy cost savings at a capital cost of 1.2 million dollars. The additional recommendations are projected to bring a 9.2 percent rate of return over the next five years after implantation if all projects are implemented at the same time. The projected results of the RMC recommendations compare well with other energy saving recommendations made for energy audits on similar facilities. The Meadowview Hospital energy audit discussed in chapter two of this thesis projected a three percent savings with a simple payback period of 4.3 years.

9.3.3 Possible Impediments to Energy Savings

The boiler plant upgrade is the most important recommendation of the entire audit. The steam boiler system and its support equipment have greatly exceeded their service life and as a consequence become less efficient each year. This conclusion is evident by the continuing increase in natural gas usage each year despite implementation of energy savings recommendations. Without a boiler plant upgrade natural gas usage will continue to increase each year and will render any related energy cost recommendations ineffective form a total usage standpoint. The savings recommendations will save natural gas usage but will not be evident because of the continuing decrease steam boiler total system efficiency. The second most critical recommendation is the control system upgrade. If the control system does not function efficiently, then the entire HVAC system will function inefficiently. The control system manages set points and adjusts the system during off peak periods. The majority of the HVAC related savings recommendations are based on the assumption that full advantage is taken for off peak loading situations. If the control system does not effectively adjust the HVAC system to take full advantage of off peak loading situations then most of the energy cost reduction strategies will be ineffective. Also simultaneous heating and cooling occurs as a direct result of an inefficient control system. If this upgrade is not implemented it will continue to reduce possible energy savings potential. Proper HVAC system control is essential to HVAC efficiency.

9.3.4 Final Conclusions

The RMC energy audit was successfully carried out over the last three years. The structure and strategy of the audit did successfully produce positive energy savings results for RMC. RMC stands to gain further energy cost savings with successful implementation of all recommended energy cost saving strategies. When performing an audit on a medical facility, it is important that all planning and action is made and carried out with a functioning medical facility in mind, and that no recommendation or audit procedure will place any patient in danger. This research produced an effective energy audit strategy evident by the success in its implementation at RMC. Many of the energy saving recommendations discussed in this thesis are not limited to RMC and would be successful if properly applied to other medical facilities. With a similar energy audit strategy and analysis techniques significant energy savings could possibly be found at any medical facility.

9.4.5 Areas for Further Research

Further research should be carried out to develop a complete chiller plant optimization strategy. This thesis touched on chiller optimization by lowering the condenser water set point and implementing a chilled water temperature reset strategy. The next step would be in varying both evaporator and condenser water flow to further reduce chiller plant related energy consumption. A complete chiller model and pumping model to work in tandem with the cooling tower model developed for the RMC audit should be produced to find the optimum operating point that produces the most efficient chiller plant operation.

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Appendix A

Audit Equipment Specifications:

Testo 335 Manometer:

Instrument	Measurement	Accuracy
Testo 335	O_2	±0.2 Vol. %
	CO	±10 ppm or ±10% of mv (0 to 200 ppm) ±20 ppm or ±5% of mv (201 to 2000 ppm) ±10% of mv (2001 to 10000 ppm)
	Temperature	±0.9 °F (+32 to +210 °F) ± 0.5 % of rdg. (remaining range)
Testo 510	Differential Pressure	±0.01" H20 (0 to 0.12" H20) / ±0.02 H20 (0.13 to 0.4" H20)
HHF42	Hot-Wire Anemometer	±(5% +1 d) reading or ±(1% +1 d) full scale the greater thereof
HH12B	Digital Thermometer	± (0.1% rdg +1°C) on -60 to 1372°C ± (0.1% rdg +2°C) on -60 to -200°C ± (0.1% rdg +2°F) on -76 to 1999°F ± (0.1% rdg +4°F) on -76 to -328°F
Hygrotest 6200	Humidity Gage	±2%

Appendix B

Data Collection Procedures:

Data Collection Procedure for Centrifugal Water Cooled Chillers

1. Select a proper location or test port to measure evaporator entering water temperature.

   a. If no test port is available, place thermocouple probe in direct contact with outer surface of evaporator water piping. (Note: All measurements should be recorded consistently)

   b. If test port is available, slowly open test port valve to allow water to flow into a container.

   c. Immediately use thermocouple to measure entering evaporator water temperature. Allow thermocouple measurement to properly settle.

   d. Repeat the process to ensure consistency. Record the findings on the data sheet.

2. Repeat the procedure in step one to measure evaporator exiting water temperature.

3. Find the test port to measure evaporator entering water pressure.

   a. If no test port is available, data cannot be taken.

   b. If test port is available and pressure gauge is present, close test port isolation valve to isolate pressure gauge. Remove existing pressure gauge and replace with digital pressure gauge.

c. Open test port to pressurize the digital gauge.

d. Record the reading on the data sheet in the proper location.

e. Close isolation valve and replace digital gauge with RMC gauge. Open test port to pressurize RMC gauge. Record the RMC gauge reading.

4. Find test port for exiting evaporator water pressure and repeat procedure in step three.

5. Select a proper location or test port to measure condenser entering water temperature.

a. If no test port is available, place thermocouple probe in direct contact with outer surface of condenser water piping. (Note: All measurements should be recorded consistently)

b. If test port is available, slowly open test port valve to allow water to flow into a container.

c. Immediately use thermocouple to measure entering condenser water temperature. Allow thermocouple measurement to properly settle.

d. Repeat the process to ensure consistency. Record the findings on the data sheet.

6. Repeat procedure in step five to measure condenser exiting water temperature.

7. Find the test port to measure condenser entering water pressure.

a. If no test port is available, data cannot be taken.

b. If test port is available and pressure gauge is present, close test port isolation valve to isolate pressure gauge. Remove existing pressure gauge and replace with digital pressure gauge.

c. Open test port to pressurize the digital gauge.

d. Record the reading on the data sheet in the proper location.

e. Close isolation valve and replace digital gauge with RMC gauge. Open test port to pressurize RMC gauge. Record the RMC gauge reading.

7. Find test port for exiting condenser water pressure and repeat procedure in step seven.

8. Go to chiller control terminal. Scroll through options on the interface to select the chiller report heading. Record the refrigerant temperature and pressure data for the evaporator and condenser found in the chiller report. Also record compressor current loads and total chiller cooling load.

Air Handler Procedure.

1. Record Air handler Name and location. Note air handler configuration and zones of service.
2. Locate return air duct or ducts. Find the straightest accessible section of duct with the least amount of obstructions. (i.e. bends, instrumentation, etc.)
3. If no test port holes are present, drill small holes in the proper location of the ductwork to create a measurement traverse.
4. Record shape of ductwork and measure the dimensions. If insulation is present on the outside or inside of the duct, subtract insulation thickness from the duct dimensions.
5. Zero hot wire anemometer and insert anemometer probe into the test port. Take care to ensure the test probe is oriented correctly into the air flow.

6. Move the probe to each location on the measurement grid and record the most stable value seen on the anemometer display. Average the recorded values to get an average duct air flow. Discard any significantly outlying recordings.
7. Locate the outside air duct or ducts. Find the straightest accessible section of duct with the least amount of obstructions. (i.e. bends, instrumentation, etc.)
8. Repeat steps three through six to measure outside air flow velocity.
9. Locate supply air duct or ducts. Find the straightest accessible section of duct with the least amount of obstructions. (i.e. bends, instrumentation, etc.)
10. Repeat steps three through six to measure supply air flow velocity.

Appendix C

Raw RMC Utility Data: 2008-2011

Electricity Usage Summary										
2008 RMC Monthly Electricity Usage in KWH and Average Peak Demand in KW										
	Main Hospital Usage	**MH Demand**	**POB1 Usage**	**POB1 Demand**	**POB2 Usage**	**POB2 Demand**	**POB3 Usage**	**POB3 Demand**	**Cancer Center Usage**	**CC Demand**
January	535762	857	89447	156	36787	70	131716	221	50309	88
February	541626	812	78229	143	30261	68	131922	241	51642	100
March	600699	860	87875	141	35900	80	142058	251	59567	102
April	584565	878	95423	165	48171	77	136130	232	63887	100
May	563541	902	93872	148	49159	86	145809	255	64615	105
June	537496	1000	96497	170	50383	88	152280	288	62539	111
July	619551	1000	110233	200	31014	50	181546	300	74358	108
August	570727	1000	90351	166	31100	50	164097	290	67890	104
September	559675	1000	69239	120	32242	54	163913	290	66185	108
October	532859	800	57996	95	32965	54	136294	241	59070	95
November	481581	746	50799	83	25023	45	94863	170	48810	95
December	537568	833	53882	91	26938	50	105859	200	53584	95
Year Total:	6665650		973843		429943		1686487		722456	
Daily Average:	18262		2668		1178		4621		1979	
Peak Demand:	916		200		88		300		111	
Average Demand:	891		140		64		248		101	
Campus Total	10478379									
Campus Daily Average	28708									
Campus Peak Demand	2474									
2008 RMC Model Monthly Electricity Usage in KWH and Average Peak Demand in KW										
	Main Hospital Usage	**MH Demand**	**POB1 Usage**	**POB1 Demand**	**POB2 Usage**	**POB2 Demand**	**POB3 Usage**	**POB3 Demand**	**Cancer Center Usage**	**CC Demand**
January	476500	890					180000	670	47210	156
February	429300	900					165000	1000	42000	209
March	501000	900					212300	780	50510	162
April	526200	1000					242100	800	56480	170
May	599200	1120					298400	1000	67610	206
June	645700	1180					336500	1070	75640	224
July	699700	1180					368800	1000	83160	210
August	698500	1170					367600	1000	83080	211
September	617100	1110					301800	990	69760	203
October	526900	1100					232600	960	55160	196
November	467100	880					182600	710	44830	155
December	469100	860					177100	680	45020	148
Year Total:	6656300		0		0		3064800		720460	
Daily Average:	18236		0		0		8397		1974	
Peak Demand:										
Average Demand:										
Campus Total	10441560									
Campus Daily Average	28607									

2009 RMC Monthly Electricity Usage in KWH and Average Peak Demand in KW

	Main Hospital Usage	MH Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	Cancer Center Usage	CC Demand	Combined POB Monthly
January	517412	824	50713	85	26917	45	100371	190	51926	128	178001
February	516687	833	52149	83	29004	50	103595	179	51814	95	184748
March	581414	833	67607	95	30465	50	128155	208	64210	95	226227
April	519554	775	61996	104	28795	50	129670	225	60983	108	220461
May	525139	860	64865	108	28201	58	148346	333	70508	125	241412
June	623313	1000	70694	125	38417	79	185859	300	82229	137	294970
July	639314	1000	78154	116	32572	54	183784	304	73670	112	294510
August	656444	1000	80175	125	32184	50	201222	314	77117	125	313581
September	630780	1000	72933	116	29455	50	183137	291	74131	125	285525
October	556499	875	63748	109	29787	51	148746	263	62745	112	242281
November	582165	833	58844	91	32989	50	124726	204	57466	105	216559
December	490440	750	57919	98	40537	70	105589	176	50302	106	204045
Year Total:	6839161		779797		379323		1743200		777101		2902320
Daily Average:	18737		2136		1039		4776		2129		7952
Peak Demand:	1000		125		79		333		137		
Average Demand:	882		105		55		249		114		
Campus Total	10518582										
Campus Daily Average	28818										
Campus Peak Demand	2494										

2009 RMC Model Monthly Electricity Usage in KWH and Average Peak Demand in KW

	Main Hospital Usage	MH Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	Cancer Center Usage	CC Demand	Combined POB Monthly
January	468800	890					175600	690	46030	189	175600
February	429200	880					164300	700	42350	188	164300
March	505300	900					216900	790	51550	163	216900
April	524800	1030					242200	810	56580	171	242200
May	591500	1100					288100	950	65430	194	288100
June	654600	1200					347900	1070	78000	224	347900
July	699700	1190					370400	1020	83320	212	370400
August	694800	1150					360200	1000	81750	211	360200
September	621200	1110					308700	1000	71260	205	308700
October	523000	1110					227900	960	54200	196	227900
November	466800	880					181500	660	44560	151	181500
December	473500	830					181300	630	46290	146	181300
Year Total:	6653200		0		0		3065000		721320		3065000
Daily Average:	18228		0		0		8397		1976		8397
Peak Demand:											
Average Demand:											
Campus Total	10439520										
Campus Daily Average	28601										

2010 RMC Monthly Electricity Usage and Peak Demand in KWH and KW											
	MH Usage	MH Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	CC Usage	CC Demand	Combined POB Monthly
January	560600	907	56127	104	37794	112	153339	311	42378	80	247260
February	513499	861	31433	148	33676	83	138605	317	43575	91	203714
March	450579	886	54225	147	28154	81	122023	280	58136	95	204402
April	488302	848	58885	154	29264	81	139422	432	65417	106	227571
May	535743	937	65517	163	31923	81	166058	450	71063	116	263498
June	712903	1000	86057	189	39627	124	283313	495	82229	137	408997
July	579033	1000	67216	191	28571	124	150612	495	73670	112	246399
August	641378	1000	75930	191	31156	80	208766	495	77117	125	315852
September	721029	1000	90595	191	37882	119	226916	495	74131	125	355393
October	488071	999	52836	190	27317	105	143501	439	62745	112	223654
November	500036	937	47007	103	32223	109	146608	439	57466	105	225838
December	444471	875	46133	105	31440	112	130260	439	50302	106	207833
Year Total:	6635644		731961		389027		2009423		758229		3130411
Daily Average:	18180		2005		1066		5505		2077		8576
Peak Demand:	1000		191		124		495		137		
Average Demand:	938		156		101		424		109		
Campus Total	10524284										
Campus Daily Average	28834										
Campus Peak Demand	2480										

2010 RMC Model Monthly Electricity Usage and Peak Demand in KWH and KW											
	MH Usage	MH Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	CC Usage	CC Demand	Combined POB Monthly
January	462300	900					170900	690	44640	189	170900
February	427000	1000					164000	1000	42000	189	164000
March	506800	910					221900	780	52750	162	221900
April	522100	1030					242600	810	56590	172	242600
May	589700	1090					287800	960	65420	194	287800
June	654600	1200					348400	1060	78040	220	348400
July	695700	1190					366000	1020	82340	212	366000
August	700300	1170					365300	1000	82830	211	365300
September	621700	1110					309300	1000	71320	205	309300
October	515900	1110					222000	960	52920	196	222000
November	467700	880					185700	680	45830	153	185700
December	467100	870					176800	590	45140	147	176800
Year Total:	6630900		0		0		3060700		719820		3060700
Daily Average:	18167		0		0		8385		1972		8385
Peak Demand:	1200		0		0		1060		220		
Average Demand:	1038		#DIV/0!		#DIV/0!		879		188		
Campus Total	10411420										
Campus Daily Average	28524										

2011 RMC Electricity												
	Hospital Usage	Hospital Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	CC Usage	CC Demand	Combined POB Monthly	
Jan	492913	750	47913	75	32247	73	128560	194	44889	75	208720	
Feb	434755	787	43655	73	29561	54	133388	238	54329	98	206604	Main Hospital
Mar	498713	770	46223	73	31188	53	161594	285	63559	104	239005	POB
Apr	532149	800	47810	75	32867	56	183335	302	68183	108	264012	Cancer Center
May	530763	800	51270	79	35573	62	201574	333	71638	108	288417	
Jun	549454	1000	49416	80	42214	80	231941	375	73885	116	323571	
Jul	537691	1000	46841	80	54127	130	213912	362	71161	116	314880	
Aug	633378	1000	58602	83	70830	145	250343	376	83043	116	379775	
Sep	550493	1000	51499	80	41286	81	201933	354	71408	112	294718	
Oct	532223	900	51782	80	37651	75	163962	266	63641	100	253395	
Nov	536793	900	54468	80	33814	63	164097	270	62922	145	252379	
Dec	426150	785	49236	70	28683	50	131856	200	51657	80	209775	
Total Usage	6255475		598715		470041		2166495		780315		3235251	
Daily Average	17138		1640		1288		5936		2138		8864	
Peak Demand		1000		83		145		376		145		
Campus Total	10271041											
Campus Daily Average	28140											
Campus Peak Demand	2443											

2011 Model RMC Electricity												
	Hospital Usage	Hospital Demand	POB1 Usage	POB1 Demand	POB2 Usage	POB2 Demand	POB3 Usage	POB3 Demand	CC Usage	CC Demand	Combined POB Monthly	Total Peak Demand
Jan	453700	870					175400	690	45970	190	175400	1750
Feb	414700	870					164700	710	42210	188	164700	1768
Mar	491400	880					221400	790	52760	162	221400	1832
Apr	497400	920					238200	810	55580	172	238200	1902
May	560400	1070					291200	1000	66350	204	291200	2274
Jun	616300	1160					347900	1060	78050	223	347900	2443
Jul	649000	1140					358800	1020	81020	212	358800	2372
Aug	661000	1110					372000	1000	84200	209	372000	2319
Sep	580600	1070					308900	1000	71220	202	308900	2272
Oct	496200	1000					220700	750	52850	166	220700	1916
Nov	454800	850					185800	690	45830	154	185800	1694
Dec	455700	840					178100	640	45540	148	178100	1628
Total Usage	6331200		0		0		3063100		721580		3063100	
Daily Average	17346		0		0		8392		1977		8392	
Peak Demand												
Campus Total	10115880											
Campus Daily Average	27715											

Appendix D

Matlab Programs:

Cooling Tower Model To Find NTU Value:

This program uses an iterative procedure to find the design NTU value for the RMC cooling tower.

```
% Initial Design Weather Parameters
RH=41;
RHSAT=100;
TempFair=98;
TempFEnteringWater=95;
Wetbulb=78;


% Degree F to K

TempK=((TempFair-32)*(5/9))+273.15;
TempKWater=((TempFEnteringWater-32)*(5/9))+273.15;

%Calculation of Saturated Vapor Pressure of Moist Air and Saturated Air

Satvapp=exp(77.345+(0.0057*TempK)-
(7235/TempK))/(TempK^8.2)*(1.450377*10^-4);
SatvappWater=exp(77.345+(0.0057*TempKWater)-
(7235/TempKWater))/(TempKWater^8.2)*(1.450377*10^-4);

%Calculation of Actual Vapor Pressure of Moist Air and Saturated Air

Actvapp=(RH*Satvapp)/100;
ActvappSat=(RHSAT*SatvappWater)/100;

%Calculation of Specific Humidity lbmw/lbma of Moist Air and Sat Air

SpecificHumidity=(.622*Actvapp)/(14.7-Actvapp);
SpecificHumiditySAT=(.622*ActvappSat)/(14.7-ActvappSat);


%Enthalpy Calculations

%Enthaplpy of moist air and Sat Water
```

```
Henteringair=(.240*TempFair)+SpecificHumidity*(.444*TempFair+1075);
HEnteringwater=(.240*TempFEnteringWater)+SpecificHumiditySAT*(.444*Temp
FEnteringWater+1075);

%Specfication of design water and air flow through the cooling tower
gpmperton=[3]; %n
cfmperton=[275];%m

% outer loop allows for changes in air and water flow rate
x=1;
for n= 1:1
for m=1:1

% calculation of air and water mass flow rate and liguid to gas ratio
for cooling tower
massflowwater(n)=(gpmperton(n)*8.33);
massflowair(n)=(cfmperton(m)*.075);
LGratio(n)=massflowwater(n)/massflowair(n);

% Initial Conditions to Start NTU Iterative Calculation
Error(x)=100;
 NTU(x)=.1;
TempFExitingWater(x)=Wetbulb;
HExitingwater(x)=enthalpywater(100,Wetbulb);

% Iterative Loop that calculates the Cooling Tower Design NTU by
iterating NTU until cooling tower exiting water converges to the
desired value.
while Error>.0000001

 % Calculation of Cs value - slope of the water saturation line between
 % entering and exiting values
Cs(x)=(HEnteringwater-HExitingwater(x))/(TempFEnteringWater-
TempFExitingWater(x));

% Calculation of M star a dimensionless parapmeter needed to calculate
airside effectiveness
mstar(x)=(massflowair(n)*Cs(x))/(massflowwater(n));

% Calculation of model air side effectiveness value at a given NTU
modeleffectiveness(x)=(1-exp((-1)*NTU(x)*(1-mstar(x))))/(-
mstar(x)*exp(-NTU(x)*(1-mstar(x))))

% Calculation of Enthalpy of the air exiting the cooling tower
Haout(x)=Henteringair+modeleffectiveness(x)*(HEnteringwater-
Henteringair);

% Calculation of Exiting Water Temperature
TempFExitingWater(x+1)=TempFEnteringWater-((massflowair(n)*(Haout(x)-
Henteringair))/(massflowwater(n)));

% Calcuation of Error between teo iterations of exiting water
temperature to stop the loop when temperature of the exiting
```

```
% water converges
Error(x+1)=abs(TempFExitingWater(x+1)-TempFExitingWater(x));

% NTU iterative step to produce a new NTU value
 NTU(x+1)=NTU(x)+.001;

% calculation of a new exiting water enthalpy to start the next
iteration
HExitingwater(x+1)=enthalpywater(100,TempFExitingWater(x+1));

% index step
x=x+1;
end
```

RMC Cooling Tower Model:

This model is used to simulate the RMC Cooling Tower over time. The RMC cooling tower is equipped with two speeds fans. It also calculates the theoretical savings due to reducing condenser water set point.

```
clear all
clc


 %Importation of raw weather data

[TempFair,Wetbulb,dewpoint,RH] =
textread('december2011weather.txt','%d%d%d%d');

% Initial Conditions for model
plateandframehours=0;
RHSAT=100;
massflowair=181500*.075;
massflowwater=1980*8.33;
LGRatio=massflowwater/massflowair;
Range=10;
SetPoint=[85];

% outer for loop to change exiting water set point
for e=1:1
% initial conditions to start simulation
SetPoint(e)=SetPoint(e);
TempFExitingWater(1)=Wetbulb(1)+7;
TempFEnteringWater(1)=TempFExitingWater(1)+10;
TempFEnteringWater2(1)=TempFEnteringWater(1);
Airflowhigh=181500*.075;
Airflowlow=90750*.075;
naturaldraftflow(1)=10000;
NTU(1)=6.72;
FanHP(1)=40;
n=1;
massbasin=4000*8.33;
Tbasin(1)=TempFExitingWater(1);
ts=.5;
t=0;
Fancyclecounter=0;
FanRunTime=0;
x=2;
% for loop to update weather conditions
for s=1:length(TempFair)

 % calculation of cooling tower and chiller cooling load
 qchiller(s)=(148.98*TempFair(s)-6200.1)/12;
 if qchiller(s)<150
     qchiller(s)=150;
 end
```

```
 % inner loop to simulate cooling tower at given weather condition
 % inner loop simulates a given weather condition for 30 second
intervals
for n=n:n+40
   %Resets cooling tower set point for plate and frame operation if
needed
    if Wetbulb(s)<40
        SetPoint(e)=40;

    else
        SetPoint(e)=85;
    end
     SetPointGraph(n+1)=SetPoint(e);


Airflowoff=naturaldraftflow(n)*.075;

% calculation of air and water entalpy using function files

Henteringair(n)=enthalpyair(RH(s),TempFair(s));
HEnteringwater(n)=enthalpywater(RHSAT,TempFEnteringWater(n));
HExitingwater(n)=enthalpywater(100,TempFExitingWater(n));

% calculation of Cs value- slope between two points on saturation curve
Cs(n)=(HEnteringwater(n)-HExitingwater(n))/(TempFEnteringWater(n)-
TempFExitingWater(n));

% limits for Cs value set to keep model within functional parameters
if Cs(n)<1.3
    Cs(n)=1.35;
end
%Calculation of mstar-dimensionless value needed for air side
effectiveness calculation
mstar(n)=(massflowair*Cs(n))/(massflowwater);

% mstar limits
if mstar(n)<1;
    mstar(n)=1.1;
end
%sets NTU value during fan ramp up
if FanHP==40
 NTU=.2025*x+.66;
end
x=x+14;
% NTU limits
if NTU>6.72
    NTU=6.72;
end

% Calculation of model air side effectiveness
modeleffectiveness(n)=(1-exp((-1)*NTU*(1-mstar(n))))/(-mstar(n)*exp(-
NTU*(1-mstar(n))));
```

```
% Calculation of Exiting air enthalpy
Haout(n)=Henteringair(n)+modeleffectiveness(n)*(HEnteringwater(n)-
Henteringair(n));

% Calculation of exiting water temperature
TempFExitingWater(n+1)=TempFEnteringWater(n)-((massflowair*(Haout(n)-
Henteringair(n)))/(massflowwater));

% Numerical intergration to find cooling tower basin water temperature
Tbasindot(n+1)=(massflowwater*(TempFExitingWater(n+1)-
Tbasin(n)))/massbasin;
Tbasin(n+1)=Tbasin(n)+.5*ts*(Tbasindot(n+1)+Tbasindot(n));

% calculation of heat transfer to the air and water as well as new
cooling tower entering
% water temperature and cooling tower range and approach
 qair(n)=(modeleffectiveness(n)*massflowair*(HEnteringwater(n)-
Henteringair(n))*60)/12000;
 qwater(n)=(massflowwater*(TempFEnteringWater(n)-
TempFExitingWater(n))*60)/12000;
 Approach(n)=TempFExitingWater(n)-Wetbulb(s);

TempFEnteringWater(n+1)=(((qchiller(s)*12000)/60)/massflowwater)+Tbasin
(n+1);
 Range(n)=TempFEnteringWater2(n+1)-TempFExitingWater(n+1);

 % Calculation of Fan Energy demand and consumption
  FanPower(n)=(FanHP*.746);
  FanEnergyUsage(n)=(FanPower(n)*1)/120;
  FanEnergyUsagesum(n)=sum(FanEnergyUsage);

  % Counter to count number of fan cycles
  if n>2 && FanPower(n)>FanPower(n-1)
      Fancyclecounter=Fancyclecounter+1;
  end
  if n>2 && FanPower(n-1)>FanPower(n) && FanPower(n)~=0
      Fancyclecounter=Fancyclecounter+1;
  end

  % Calculation of exiting water temperature enthalpy
 HExitingwater(n)=enthalpywater(100,TempFExitingWater(n+1));

 % calculation of natural convective flow when fan is off
 deltaho(n)=HEnteringwater(n)-Haout(n);
 deltahi(n)=HExitingwater(n)-Henteringair(n);
 deltah(n)=(deltaho(n)-deltahi(n))/(log(deltaho(n)/deltahi(n)));
 naturaldraftflow(n+1)=.056*181500*(deltah(n)/9.07)^.2;

 %Calculation of chiller efficiency at given entering condenser
temperature
 Tempdiffcompresser=85-Tbasin(n);
 if Tempdiffcompresser==0;
     kwperton(n)=.7;
```

```
 end
 if Tempdiffcompresser~=0
     kwperton(n)=.7-(Tempdiffcompresser*.01*.7);
 end

 % Conditional statement that disengages the chiller power calculator
when plate and frame heat exchanger is in operation
 if Tbasin(n+1)<=50.5
     ChillerPower(n)=0;
 ChillerEnergyUsage(n)=ChillerPower(n)/120;
 ChillerEnergyUsagesum(n)=sum(ChillerEnergyUsage);
 plateandframehours=plateandframehours+1;
 else
     % Chiller energy demand and consuption calculator when chiller is
in operation.
     ChillerPower(n)=kwperton(n)*qchiller(s);
 ChillerEnergyUsage(n)=ChillerPower(n)/120;
 ChillerEnergyUsagesum(n)=sum(ChillerEnergyUsage);
 end

% Calculation of chiller plant water pump energy demand and consuption
  TotalPumpPower(n)=.746*90;
 PumpEnergyUsage(n)=(TotalPumpPower(n))/120;
 PumpEnergyUsagesum(n)=sum(PumpEnergyUsage);

 %Calculation of total chiller plant power demand and consumption

TotalPlantPowerDemand(n)=ChillerPower(n)+FanPower(n)+TotalPumpPower(n);

TotalPlantPowerUsage(n)=ChillerEnergyUsage(n)+FanEnergyUsage(n)+PumpEne
rgyUsage(n);

TotalPlantPowersum(n)=FanEnergyUsagesum(n)+ChillerEnergyUsagesum(n)+Pum
pEnergyUsagesum(n);

 % Fan controller - Resets Fan speed based on Set point

     if Tbasin(n+1)>SetPoint(e)+5
    massflowair=Airflowhigh;
%     NTU=6.72;
if n>2 & FanHP==0
       x=1;
end
    FanHP=40;

    end

if Tbasin(n+1)>SetPoint(e)-1 && Tbasin(n+1)<SetPoint(e)
    massflowair=Airflowlow;
    NTU=3;
    FanHP=5;
end

     if Tbasin(n+1)<SetPoint(e)-5
    massflowair=Airflowoff;
```

```
    NTU=.8;
    FanHP=0;
     end
 % calculation of fan run time
 if FanHP==40 || FanHP==5
     FanRunTime=FanRunTime+.5;
 end
% time step
 t(n+1)=t(n)+.5;
end

end
```

%function files used by program calculate enthalpy of air and water

```
 function out=enthalpywater(RHSATf,TempFEnteringWaterf)

TempKWaterf=((TempFEnteringWaterf-32)*(5/9))+273.15;

SatvappWaterf=exp(77.345+(0.0057*TempKWaterf)-
(7235/TempKWaterf))/(TempKWaterf^8.2)*(1.450377*10^-4);

ActvappSatf=(RHSATf*SatvappWaterf)/100;

SpecificHumiditySATf=(.622*ActvappSatf)/(14.7-ActvappSatf);

HEnteringwaterf=(.240*TempFEnteringWaterf)+SpecificHumiditySATf*(.444*T
empFEnteringWaterf+1075);

out=HEnteringwaterf;

function out =enthalpyair(RHf,TempFairf);


TempKf=((TempFairf-32)*(5/9))+273.15;

Satvappf=exp(77.345+(0.0057*TempKf)-
(7235/TempKf))/(TempKf^8.2)*(1.450377*10^-4);

Actvappf=(RHf*Satvappf)/100;

SpecificHumidityf=(.622*Actvappf)/(14.7-Actvappf);

Henteringairf=(.240*TempFairf)+SpecificHumidityf*(.444*TempFairf+1075);

out=Henteringairf;
```

RMC Cooling Tower Model (VFD Fans):

This model was used to simulate RMC cooling tower with VFD fans equipped.

```
% Command used to import raw weather data
 [TempFair,Wetbulb,dewpoint,RH] =
textread('december2010weather.txt','%d%d%d%d');

 % Program initial conditions
RHSAT=100;
Range=10;
SetPoint=85;
TempFExitingWater(1)=Wetbulb(1)+12;
TempFEnteringWater(1)=TempFExitingWater(1)+10;
TempFEnteringWater2(1)=TempFEnteringWater(1);
naturaldraftflow(1)=10000;
NTU(1)=6.72;
FanHP(1)=40;
n=1;
massbasin=4000*8.33;
Tbasin(1)=TempFExitingWater(1);
ts=.1;
t=0;
G=2;
P=.5*G;
D=.001
I=.0001;
error(1)=0;
error_Ki(1)=0;
dt=.1;
massflowair(1)=10000*.075;
AirFlow(1)=1;

% Outer loop to update weather conditions and calculte cooling load
for s=1:length(TempFair)
 qchiller(s)=(148.98*TempFair(s)-6200.1)/12;
 if qchiller(s)<150
     qchiller(s)=150;
 end

 % inner loop simulates cooling tower operation with a six second time
step
for n=n:n+200
    SetPoint(n+1)=SetPoint(n);

% PID controller parameter reset
if Tbasin(n)>SetPoint(n)
    S=30;
    P=.5*S;
end
if Tbasin(n)<SetPoint(n)
    P=.5*G;
end
massflowwater=1980*8.33;
```

```
% PID controler used to operate VFD Fans
error(n+1)=((SetPoint(n))-(Tbasin(n)));
error_Ki(n+1)=error_Ki(n)+.5*dt*(error(n+1)+error(n));
error_Kd(n+1)=(error(n+1)-error(n))/dt;

if error(n+1)<0
Power(n+1)=abs((P*error(n+1))+(I*error_Ki(n+1))+(D*error_Kd(n+1)));
if Power(n+1)>40
    Power(n+1)=40;
end

% Calculates Cooling tower air flow at given fan power
AirFlow(n+1)=6.7801*Power(n+1)^3-
528.65*Power(n+1)^2+14420*Power(n+1)+32000;
if AirFlow(n+1)>195000
    AirFlow(n+1)=195000;
end
end

% Limits to help control set point overshoot

if error(n+1)>0
    Power(n+1)=0;
    AirFlow(n+1)=naturaldraftflow(n);
end
massflowair(n+1)=AirFlow(n+1)*.075;

%Calculation of air and water enthaply using function files
Henteringair=enthalpyair(RH(s),TempFair(s));
HEnteringwater=enthalpywater(RHSAT,TempFEnteringWater(n));
HExitingwater=enthalpywater(100,TempFExitingWater(n));

%calculation of Cs Value
Cs=(HEnteringwater-HExitingwater)/(TempFEnteringWater(n)-
TempFExitingWater(n));
% Cs value limits
if Cs<1.3
    Cs=1.35;
end
%Calculation of mstar and mstar limits
mstar=(massflowair(n)*Cs)/(massflowwater);
if mstar<1;
    mstar=1.1;
end

%Calculation of NTU value for given air flow rate
NTU=3*10^-5*(massflowair(n)/.075)+1;
%NTU limits
if NTU>7
    NTU=7;
End
```

```
%Calculation of model air side effectivness
modeleffectiveness=(1-exp((-1)*NTU*(1-mstar)))/(-mstar*exp(-NTU*(1-
mstar)));

%model effectiveness lower limit
if modeleffectiveness<.1
    modeleffectiveness=.1;
end

 % calculation of exiting air and water enthalpy, exiting water
temperature, basin water temperature, and new entering water
temperature

Haout=Henteringair+modeleffectiveness*(HEnteringwater-Henteringair);
TempFExitingWater(n+1)=TempFEnteringWater(n)-((massflowair(n)*(Haout-
Henteringair))/(massflowwater));
Tbasindot(n+1)=(massflowwater*(TempFExitingWater(n+1)-
Tbasin(n)))/massbasin;
Tbasin(n+1)=Tbasin(n)+.5*ts*(Tbasindot(n+1)+Tbasindot(n));
TempFEnteringWater(n+1)=(((qchiller(s)*12000)/60)/massflowwater)+Tbasin
(n+1);
HExitingwater=enthalpywater(100,TempFExitingWater(n+1));

%natural convetive flow calculation
 deltaho=HEnteringwater-Haout;
 deltahi=HExitingwater-Henteringair;
 deltah=(deltaho-deltahi)/(log(deltaho/deltahi));
 naturaldraftflow(n+1)=.056*181500*(deltah/9.07)^.2;

 %Chiller efficiency calculation for given condenser water set point

  Tempdiffcompresser=85-Tbasin(n);
 if Tempdiffcompresser==0;
     kwperton(n)=.7;
 end
 if Tempdiffcompresser~=0
     kwperton(n)=.7-(Tempdiffcompresser*.01*.7);
 end

 %cooling tower fan, chiller, and total plant power demand and
consumption
FanPower(n)=(Power(n+1)*.746);
FanEnergyUsage(n)=(FanPower(n)*1)/600;
FanEnergyUsagesum(n)=sum(FanEnergyUsage);

 ChillerPower(n)=kwperton(n)*qchiller(s);
 ChillerEnergyUsage(n)=ChillerPower(n)/600;
 ChillerEnergyUsagesum(n)=sum(ChillerEnergyUsage);

 TotalPumpPower(n)=.746*90;
 PumpEnergyUsage(n)=(TotalPumpPower(n))/600;
 PumpEnergyUsagesum(n)=sum(PumpEnergyUsage);


TotalPlantPowerDemand(n)=ChillerPower(n)+FanPower(n)+TotalPumpPower(n);
```

```
TotalPlantPowerUsage(n)=ChillerEnergyUsage(n)+FanEnergyUsage(n)+PumpEne
rgyUsage(n);

TotalPlantPowersum(n)=FanEnergyUsagesum(n)+ChillerEnergyUsagesum(n)+Pum
pEnergyUsagesum(n);

 %Time step
 t(n+1)=t(n)+.1;
end

end
```

Matlab Program used to calculate number of available free cooling hours and number of hours to raise chilled water supply temperature:

```
[TempFair,Wetbulb,dewpoint,RH] =
textread('december2011weather.txt','%d%d%d%d');

 min=0;
 min2=0;
 for n=1:length(TempFair)
     qchiller(n)=(148.98*TempFair(n)-6200.1)/12;
     if qchiller(n)<150;
         qchiller(n)=150;
     end
     if qchiller(n)>=360
         min=min+1;
     end
     if qchiller(n)<360
         min2=min2+1;
     end
     if Wetbulb(n)<=50

         min=min+1;
     end
     if Wetbulb(n)<=35
         min2=min2+1;
     end
 end
     hours=(min*20)/60
     hours2=(min2*20)/60
```

Savings due to Chilled Water Reset Program:
This program was used to calculate the savings due to chilled water reset.

```
 % Command used to import weather data
 [TempFair,Wetbulb,dewpoint,RH] =
textread('december2011weather.txt','%d%d%d%d');

% outer loop updates weather data and calculates cooling load
% It also determines which chiller is online
  n=1;
     for s=1:length(TempFair)
 qchiller(s)=(148.98*TempFair(s)-6200.1)/12;
 if qchiller(s)<150
     qchiller(s)=150;
 end

 if qchiller(s)>360
     qchiller1(s)=.4*qchiller(s);
     qchiller2(s)=.6*qchiller(s);
 end
 if qchiller(s)<360;
     qchiller1(s)=0;
     qchiller2(s)=qchiller(s);
 end

 %inner loop sets chiller efficiency at different chilled water supply
 %temperatures
    for n=n:n+40
     kwperton(n)=.7;
     kwperton1(n)=.7;
     kwperton2(n)=.7;

     %chiller effieciency with 5 degree rise
     if Wetbulb(s)<=65
         kwperton1(n)=.6664;
         kwperton2(n)=.6664;
     end

     %chiller efficiency with 8 degree rise
     if Wetbulb(s)<=50
         kwperton1(n)=.6104;
         kwperton2(n)=.6104;
     end

     % Calcualtion of Chiller plant power demand and consumption while
     % varying supply temperature
     %Also Energy Savings is Calculated
  ChillerPower(n)=kwperton(n)*qchiller(s);
 ChillerEnergyUsage(n)=ChillerPower(n)/120;
 ChillerEnergyUsagesum(n)=sum(ChillerEnergyUsage);

 ChillerPower1(n)=kwperton1(n)*qchiller1(s);
 ChillerEnergyUsage1(n)=ChillerPower1(n)/120;
```

```
 ChillerEnergyUsagesum1(n)=sum(ChillerEnergyUsage1);

 ChillerPower2(n)=kwperton2(n)*qchiller2(s);
 ChillerEnergyUsage2(n)=ChillerPower2(n)/120;
 ChillerEnergyUsagesum2(n)=sum(ChillerEnergyUsage2);
 Savings(n)=ChillerEnergyUsagesum(n)-
(ChillerEnergyUsagesum2(n)+ChillerEnergyUsagesum1(n));
 maxdemand(n)=(ChillerPower1(n)+ChillerPower2(n));
    end
     end
```

Outdoor Air Intake Savings Program:
This program calculates the savings due to reducing outdoor air intake.

```
%Command used to import weather data
 [TempFair,Wetbulb,dewpoint,RH] =
textread('december2009weather.txt','%d%d%d%d');


s=1;
for n=1:length(TempFair)

TempK(n)=((TempFair(n)-32)*(5/9))+273.15;

%Calculation of Saturated Vapor Pressure of Moist Air and Saturated Air

Satvapp(n)=exp(77.345+(0.0057*TempK(n))-
(7235/TempK(n)))/(TempK(n)^8.2)*(1.450377*10^-4);

%Calculation of Actual Vapor Pressure of Moist Air and Saturated Air

Actvapp(n)=(RH(n)*Satvapp(n))/100;

%Calculation of Specific Humidity lbmw/lbma of Moist Air and Sat Air

SpecificHumidity(n)=(.622*Actvapp(n))/(14.7-Actvapp(n));

%calculates additional outside air sensible and latent cooling load if
the outside air is greater than building discharge air set point

if TempFair(n)>=55

    CoolingSavingsSensible(n)=(((13515*.075*.24*(TempFair(n)-
55))*60)/12000);


    CoolingSavingsLatent(n)=(((13515*.075*1000*(SpecificHumidity(n)-
.010))*60)/12000);
    HeatingSavings(s)=0;
    TotalEnergyHeatingSavings(s)=HeatingSavings(s)*(20/60);
TotalHeatingUsageSavingssum(s)=sum(TotalEnergyHeatingSavings);
s=s+1;
end

%calculates additional heating load if the outdoor air temperature is
below discharge air temperauture setpoint.
if TempFair(n)<55
    CoolingSavingsSensible(n)=0;
    CoolingSavingsLatent(n)=0;
    HeatingSavings(s)=(13515*.075*.24*(55-TempFair(n))*60);
    TotalEnergyHeatingSavings(s)=HeatingSavings(s)*(20/60);
TotalHeatingUsageSavingssum(s)=sum(TotalEnergyHeatingSavings);
s=s+1;
```

```
end
if CoolingSavingsLatent(n)<0
    CoolingSavingsLatent(n)=0;
end
%calculate energy savings due to reducing outdoor air intake
TotalCoolingEnergySavings(n)=CoolingSavingsSensible(n)+CoolingSavingsLa
tent(n);
TotalCoolingUsageSavings(n)=TotalCoolingEnergySavings(n)*.7*(20/60);
TotalCoolingUsageSavingssum(n)=sum(TotalCoolingUsageSavings);

end

%Calculates total $ savings due to reducing outside air intake
TotalCoolingDollars=TotalCoolingUsageSavingssum(n-1)*.065;
TotalHeatingDollars=(TotalHeatingUsageSavingssum(s-1)*6)/1000000;
```

Chilled Water Link Savings:
This program calculates the theoretical savings of shifting the POB cooling load to the Main Hospital which contains more efficient liquid cooled chillers.

```
clear all
clc


%command used to import weather data

 [TempFair,Wetbulb,dewpoint,RH] =
textread('december2008weather.txt','%d%d%d%d');

 %outer loop update weather data
for s=1:length(TempFair)
%calcualtion of main hospital cooling load
MHqchiller(s)=(148.98*TempFair(s)-6200.1)/12;
 if MHqchiller(s)<150
     MHqchiller(s)=150;
 end
 %calculation of POB cooling load
 POBqchiller(s)=(28.651*TempFair(s)-175.8)/12;
 if POBqchiller(s)<20
     POBqchiller(s)=20;
 end
 %chiiler effiencies for both air and liquid cooled chillers
 kwpertonMH=.7;
 kwpertonPOB=1.3;
 kwpertonPOB2=1;
 if POBqchiller(s)>240
     kwpertonPOB=1.575;
 end

 %calcualtion of liquid cooled chiller power demand and consumption
 ChillerPower(s)=kwpertonMH*MHqchiller(s);
 ChillerEnergyUsage(s)=ChillerPower(s)*(20/60);
 ChillerEnergyUsagesum(s)=sum(ChillerEnergyUsage);

 %calcualtion of air cooled chiller power demand and consumption
 ChillerPower2(s)=kwpertonPOB*POBqchiller(s);
 ChillerEnergyUsage2(s)=ChillerPower2(s)*(20/60);
 ChillerEnergyUsagesum2(s)=sum(ChillerEnergyUsage2);

 %calcualtion of available capacity for liquid cooled chillers
 if MHqchiller(s)<660
   Availablecapacity(s)=660-MHqchiller(s);
 end
  if MHqchiller(s)>660
      Availablecapacity(s)=0;
  End
```

```
  %calculation of new total chiller power if the load is switched to
liquid cooled chillers
  %also accounts for plate and frame heat exchanger usage
 if Availablecapacity(s)>POBqchiller(s)
   ChillerPower3(s)=kwpertonMH*POBqchiller(s);
 ChillerEnergyUsage3(s)=ChillerPower3(s)*(20/60);
 ChillerEnergyUsagesum3(s)=sum(ChillerEnergyUsage3);
 else
    ChillerPower3(s)=kwpertonPOB2*POBqchiller(s);
 ChillerEnergyUsage3(s)=ChillerPower3(s)*(20/60);
 ChillerEnergyUsagesum3(s)=sum(ChillerEnergyUsage3);
 if Wetbulb(s)<40 && (MHqchiller(s)+Availablecapacity(s))<360
    ChillerPower3(s)=0;
 ChillerEnergyUsage3(s)=ChillerPower3(s)*(20/60);
 ChillerEnergyUsagesum3(s)=sum(ChillerEnergyUsage3);
 end
 end
end

%calcualtion of total savings due to shifting POB chiller cooling load
to Main Hospital chiller plant
SavingsDemand=max(ChillerPower2)-max(ChillerPower3);
 SavingsUsage=ChillerEnergyUsagesum2(s)-ChillerEnergyUsagesum3(s);
 fprintf('%8.0f\n%4.0f\n',SavingsUsage,SavingsDemand)
```

Appendix E

Air Handler Sensor Information:

AHU 3A

- Mixed Temperature Sensor-Working
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Cannot be adjusted on the graphics**

AHU 3B

- Mixed Temperature Sensor-Working
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-No Dampers on graphics**

AHU 3R

- **Mixed Temperature Sensor-Not on Graphics, so no way of seeing if it works**
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Do not work**

AHU 1E

- **Mixed Temperature Sensor-Not on Graphics, so no way of seeing if it works**
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Do not work**

AHU 2E

- **Mixed Temperature Sensor-No sensor**
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Cannot be adjusted on the graphics**

AHU G1

- **Mixed Temperature Sensor- Not on Graphics, so no way of seeing if it works**
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Do not work**

AHU G2

- **Mixed Temperature Sensor-Not on Graphics, so no way of seeing if it works**
- **Preheating Temperature Sensor-Does not work**
- Supply Temperature Sensor-Working
- Dampers-Working

AHU 2

- **Mixed Temperature Sensor-Does not work**
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Cannot be adjusted from graphics**

AHU 4

- Mixed Temperature Sensor-Working
- Preheating Temperature Sensor-Working
- Supply Temperature Sensor-Working
- **Dampers-Cannot be adjusted from graphics**

AHU-K

- **Mixed Temperature Sensor- No sensor**
- **Preheating Temperature Sensor-Not working**
- **Supply Temperature Sensor-Not working**
- **Dampers-Cannot be adjusted from graphics**

AHU EDI

- Mixed Temperature Sensor-Working
- Preheating Temperature Sensor-Working
- Supply Temperature Sensors-Working
- Dampers-Working

AHU ER-ICU

- **Mixed Temperature Sensor-Not working**
- **Preheating Temperature Sensor-Not working**
- **Supply Temperature Sensor-Not working**
- **Dampers- Cannot be adjusted from graphics**

AHU MRI

- **Mixed Temperature Sensor-Not on graphics, so no way of seeing if it works**
- **Preheating Temperature Sensor-Not working**
- **Supply Temperature Sensor-Not working**
- **Dampers-Cannot be adjusted from graphics**

AHU OR2

- Mixed Temperature Sensor-Working
- Preheating Temperature Sensor- Working
- Supply Temperature Sensor-Working
- **Dampers-Cannot be adjusted from graphics**