ABSTRACT

JOSHI, ANVAY. Modeling and Analysis of the Chilled Water System at a Manufacturing and Warehousing Facility. (Under the direction of Dr. Stephen Terry).

A plant built in the 1960s recently shifted its manufacturing operations to a different location leaving this plant to serve as a warehousing, packaging, and shipping facility. The plant had installed three chillers with a constant speed pumping system to serve its cooling needs. Some sections of the plant were expanded in the 1980s and rooftop units were added to better deal with the increase in cooling demand. Since then the plant has operated without any significant upgrades to its HVAC system. Mechanical equipment has gotten more energy efficient over the years. This thesis looks to improve the current pumping and air distribution system. However, there was a lack of information on the kind of air distribution system the plant uses. To deduce this, an energy model was created in Carrier HAP using the utility bills obtained from the facility. It pointed to the plant using a Constant Air Volume system with electric reheat in some zones. An additional benefit of the energy model is that it helps to examine the effects of recommendations on the current chiller system and better understand the costs of operation. Four different recommendations were made which consider various water-side and air-side HVAC arrangements to find the most energy efficient option. Implementation costs were looked at along with simple payback periods to determine the most practical solution for the plant. A primary secondary loop along with VFDs on the cooling tower loop proves to be an optimal strategy for the plant to adopt. Changing the air side system will reduce the energy bills by a huge margin but would prove to be extremely uneconomical to implement.

Modeling and Analysis of the Chilled Water System at a
Manufacturing and Warehousing Facility

by
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DEDICATION

To my family, all my friends, professors and to my future self to serve as a reminder to put enough work in because it will all be worth it. It always works out in the end.

BIOGRAPHY

Anvay was raised in Mumbai, a city he holds dear to his to heart for all the memories and friends made. He finished his bachelor's in mechanical engineering from a college near Palm Beach in Mumbai. In the final year of his undergraduate course, he came across ISHRAE (the Indian counterpart of ASHRAE) and was introduced to HVAC. Feeling the need to learn more about this field at a higher level and backed by his parents, he decided to apply to different universities in the US for his Master's. After finishing one year at North Carolina State University, Anvay was given the opportunity to join the Industrial Assessment Center where he accompanied the center director and other IAC members to conduct energy audits at various facilities. After graduation, Anvay looks to enter the energy engineering industry and desires to tell his grandkids about the time he survived a pandemic and got a job.

When Anvay is not working, he enjoys playing tennis, brewing a cup of flavorful coffee, and feeling hopeful for Manchester United FC.

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This was a long, sometimes stressful but majorly a very enjoyable learning experience. As I wrap up my thesis, I would like to thank quite a few people.

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CHAPTER 1: INTRODUCTION

1.1: Energy Conservation in Manufacturing Plants

As any energy efficiency advocate would agree, one must be on a constant lookout for ways to implement energy saving measures at a place where the need is evident. Such was the case for a manufacturing plant which was subject to an energy assessment by the Industrial Assessment Center at North Carolina State University. During the energy assessment, the team came across some significant energy saving opportunities mostly relating to the chiller operation which will be discussed further.

The industrial sector consumes about one-third of the total energy consumed in the United States, of which about two thirds is attributed to the manufacturing plants. According to the US Department of Energy, the current energy consumption can be reduced by about 20 percent by incorporating an energy efficiency-oriented mindset into the decision-making process [1] [2].

The aim of this thesis is to provide engineering solutions for the said plant. This facility underwent a significant change in terms of its manufacturing goals which will be discussed later. The solutions include new technologies and control strategies that were not available to the facility when it was last updated. Successful implementation of these solutions will help save significant amount of energy resulting in cost savings. Any kind of practical recommendation is incomplete without a cost analysis of the suggestion provided. Therefore, at the end of each recommendation, there will be a discussion on the cost effectiveness of each solution.

The thesis will be structured as follows: Chapter 1 is the introduction to the plant, their manufacturing processes and how it has changed over the years. It will also identify the key areas which need to improve in terms of energy efficiency. Chapter 2 will cover the basics of mechanical systems like chillers and pumping systems to better understand the recommendations provided later. Chapter 3 will review the existing literature to grasp the mathematical side of thermal systems which will be used in calculating energy savings. Chapter 4 will deal with the solutions suggested to improve energy efficiency of the plant and will look at the feasibility of each one of them. Finally, Chapter 5 will provide a summary of the results obtained through Chapter 4.

1.2: Facility Overview

1.2.1: Current Plant Layout and Process Description

A general understanding of the plant's background, operation and the products produced would prove beneficial before diving into the recommendations for improving energy efficiency.

This manufacturing plant is located in the southeastern region of the United States. This plant is a part of a larger corporation with over 30 subsidiary brands under its belt. Among many of its facilities around the world, the one discussed here is in the US.

The plant has a footprint of about 550,000 ft^2. It used to be heavily involved in the production of tools but over the years it has shifted its focus to being, for the most part, a warehouse for the manufactured products. The facility stores and ships tools that they receive from their manufacturing facilities. The products are stored, packaged and then shipped to the customers. The plant also has a few labs for testing their products. Figure 1 shows a general layout of the facility.

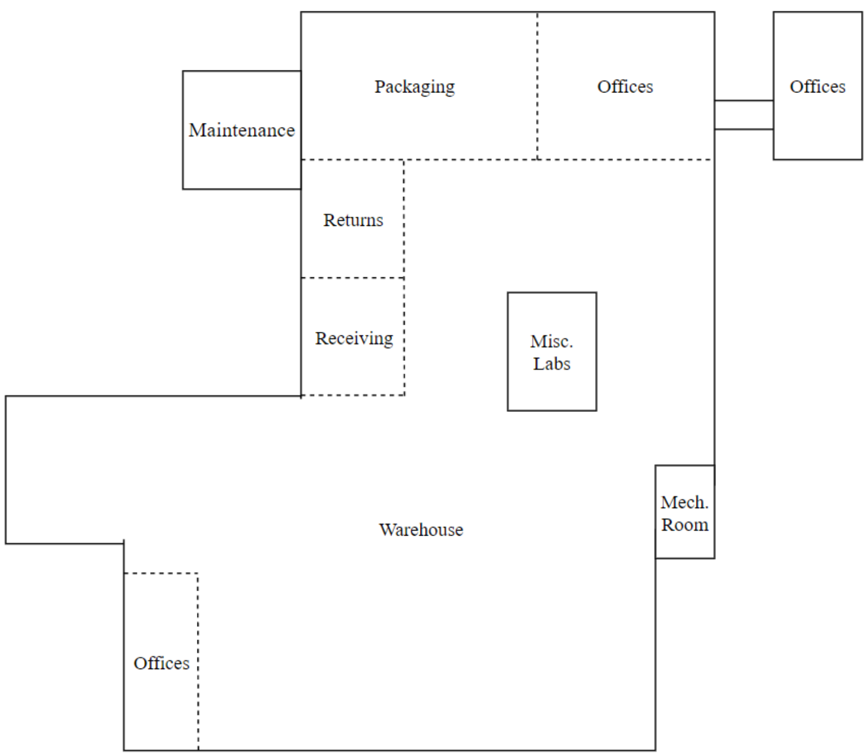


Figure 1: Facility layout

Figure 2 below shows the process flow diagram.

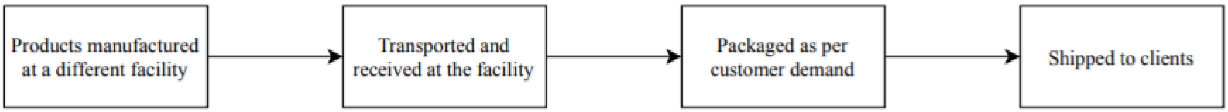


Figure 2: Process flow diagram

1.2.2: Old Plant Layout and Process Description

To gain a better understanding of how the facility has shifted its operations over the years, the original facility layout and process description have been discussed below.

The old facility was a 300,000 ft^2 building that was constructed in 1960s. The site was expanded by 250,000 ft^2 about 20 years later when it became the main corporate headquarters for the division which added a large distribution center to the facility.

The building was heated by hot water provided by a hot water generator. Steam for process heating was produced by a steam boiler. Air conditioning was provided by a combination of chilled water and roof top units (RTUs). Most of the main production space and part of the distribution center was cooled with chilled water produced by a 750-ton chiller. The remainder of the distribution center, the offices, and a few sections of production were cooled using RTUs. Lighting throughout the facility consisted of fluorescents and high-pressure sodium fixtures.

The old facility layout is given in Figure 3 below.

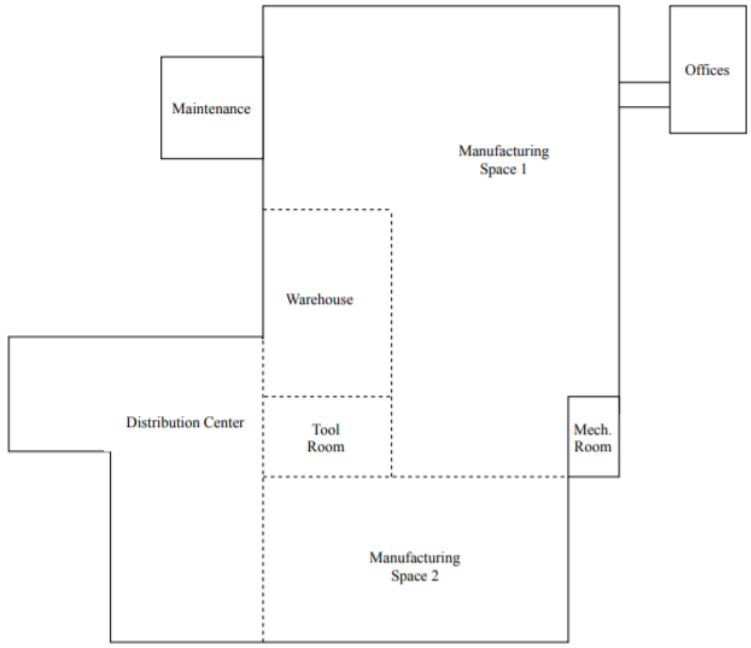


Figure 3: Old facility layout

The plant received the metal raw materials which were cut according to the dimensions of the product. The raw material was then cleaned in an electro-cleaner solution, rinsed, and dried. Paint was applied and cured using an infrared oven. After some metal working, the product was assembled on-site and packaged for shipment.

The plant also manufactured wooden products. The process is similar to how the metal products were made. The wood was cut, shaped, painted and then cured in a steam heated oven. Packaging of the final product was done and stored in the distribution center ready to be shipped.

Figure 4 below shows the process flow diagram of the old facility.

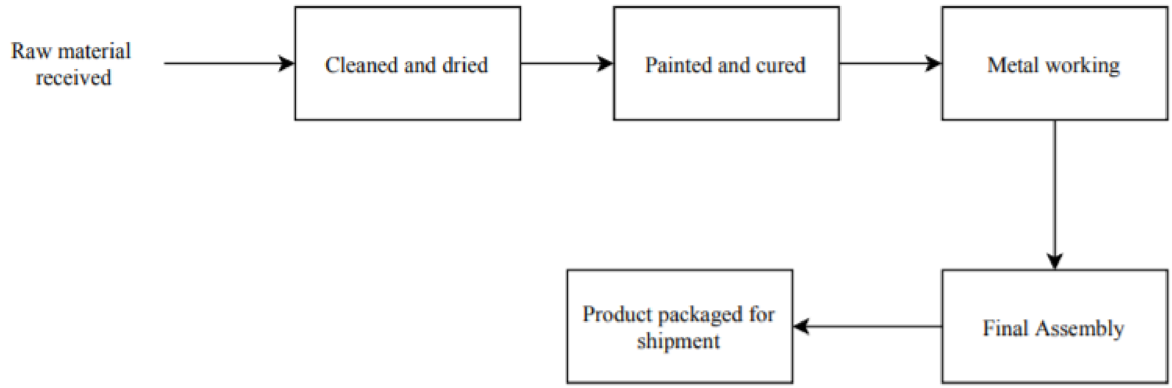


Figure 4: Old process flow diagram

1.3: Energy Use

This section dives into the annual energy consumption of the old as well as the current facility. The usage has been divided into two parts, electricity and natural gas consumption. The following three sections will calculate the energy, demand and natural gas rates which will be later used for the calculation of cost savings obtained by the implementation of different recommendations.

1.3.1: Electricity Usage

The facility uses electricity provided by the local utility under the large general service rate schedule. Table 1 shows the energy usage for 2018-19. Figure 5 and Figure 6 show the graphs for the energy and demand costs and consumption values on a monthly basis.

Table 1: Electrical Energy Use and Cost

Month	Energy (kWh)	Demand (kW)	Energy Charge	Demand Charge	Total
Jul-18	976,000	1,675	$47,439	$21,708	$69,147
Aug-18	988,800	1,609	$48,055	$20,853	$68,907
Sep-18	963,200	1,534	$47,084	$19,881	$66,964
Oct-18	803,200	1,471	$39,493	$19,064	$58,557
Nov-18	678,400	1,237	$34,719	$16,032	$50,750
Dec-18	758,400	1,168	$40,009	$15,137	$55,146
Jan-19	742,400	1,176	$39,215	$15,241	$54,456
Feb-19	790,400	1,432	$39,470	$18,559	$58,029
Mar-19	668,800	1,146	$35,925	$14,852	$50,777
Apr-19	716,800	1,441	$35,794	$18,675	$54,469
May-19	752,000	1,490	$37,550	$19,310	$56,861
Jun-19	921,600	1,389	$46,016	$18,001	$64,018
Total	**9,760,000**	**16,768**	**$490,768**	**$217,313**	**$708,081**

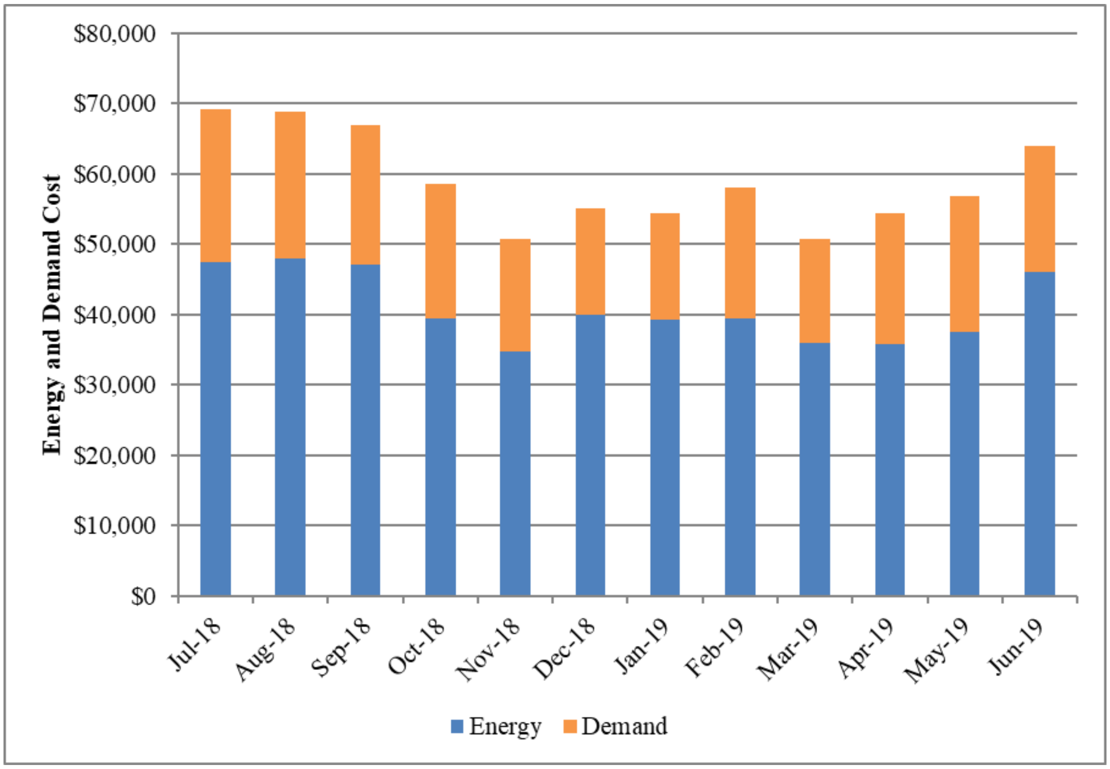


Figure 6: Electrical energy cost

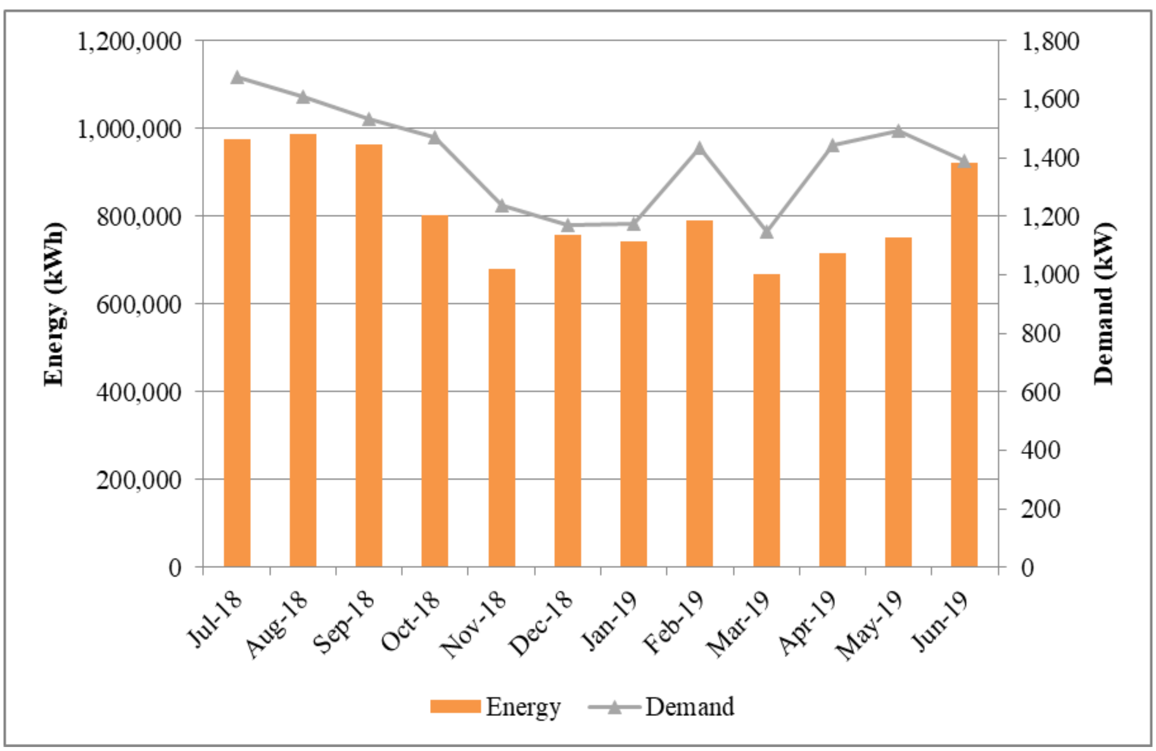


Figure 5: Electrical energy usage and demand

1.3.2: Natural Gas Energy Usage

The facility uses natural gas provided by the local gas utility. Table 2 shows the natural gas usage for 2018-19. Figure 7 shows the graph for the natural gas usage and cost values on a monthly basis.

Table 2: Natural Gas Use and Cost

Month	Fuel (MMBtu)	Gas Cost
Jul-18	62	$547
Aug-18	813	$5,719
Sep-18	1,072	$7,545
Oct-18	1,656	$12,967
Nov-18	1,600	$11,733
Dec-18	1,111	$7,278
Jan-19	100	$893
Feb-19	100	$135
Mar-19	1	$114
Apr-19	0	$107
May-19	2	$121
Jun-19	2	$121
Total	**6,519**	**$47,281**

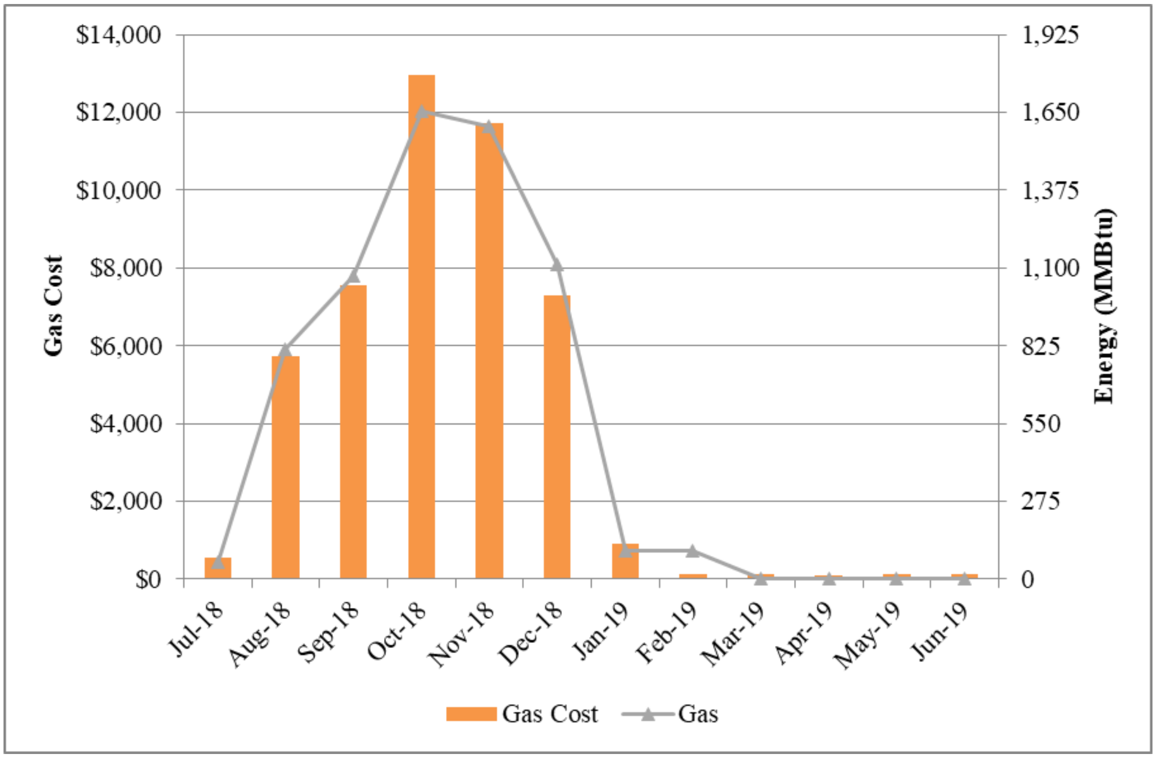


Figure 7: Natural gas energy usage and cost

1.3.3: Total Energy Usage

The total energy, demand and natural gas consumption and cost values have been used to calculate the rates as shown below.

Average Energy Rate	=	$490,768/yr. / 9,760,000 kWh/yr.
	=	$0.0503/kWh
Average Demand Rate	=	$217,313/yr. / 16,768 kW/yr.
	=	$12.96/kW

The average cost of natural gas per MMBtu is:

Average Natural Gas Rate	=	$47,281/yr. / 6,519 MMBtu/yr.
	=	$7.25/MMBtu

Figure 8 shows the total energy (Electricity and Natural Gas) cost values on a monthly basis.

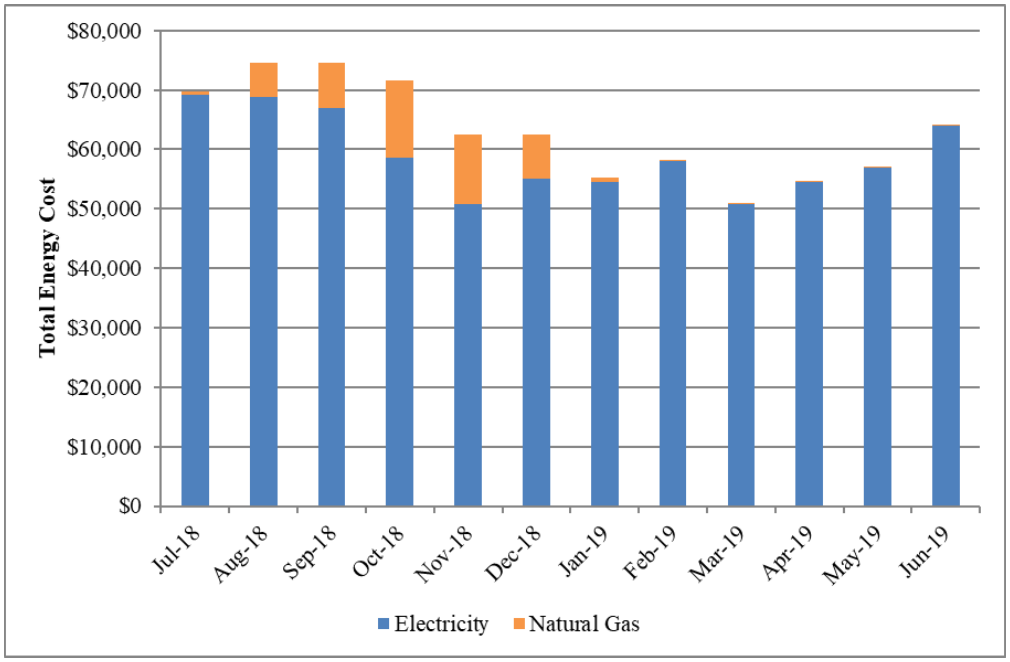


Figure 8: Total energy cost

1.4: Comparison of the current with the old facility's energy consumption values

Figure 9-Figure 11 give a perspective of how the energy and natural gas consumption values have changed as the plant shifted its focus from being a predominantly manufacturing facility to mostly a packaging and shipping facility.

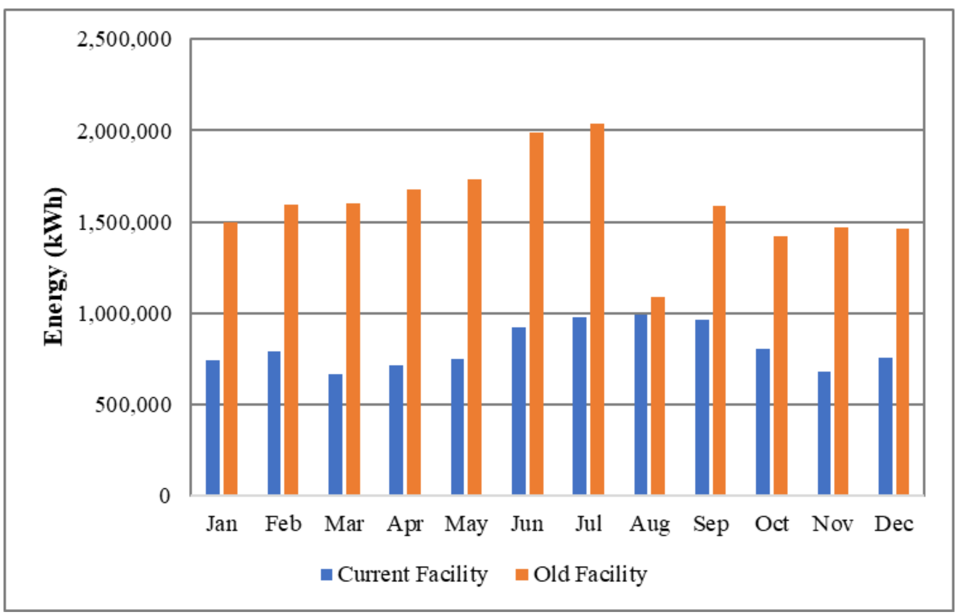


Figure 9: Comparison of energy values

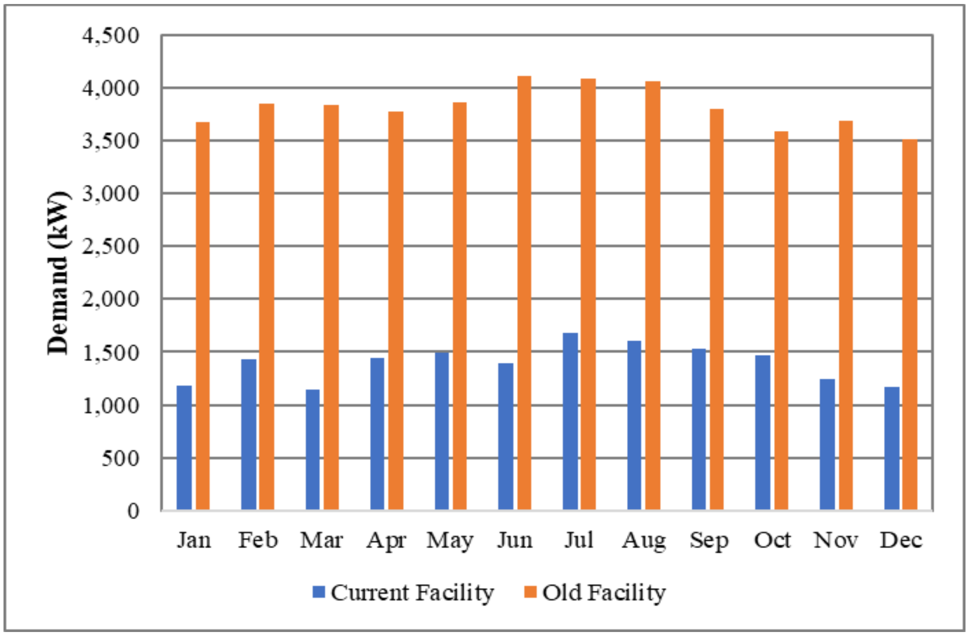


Figure 10: Comparison of demand values

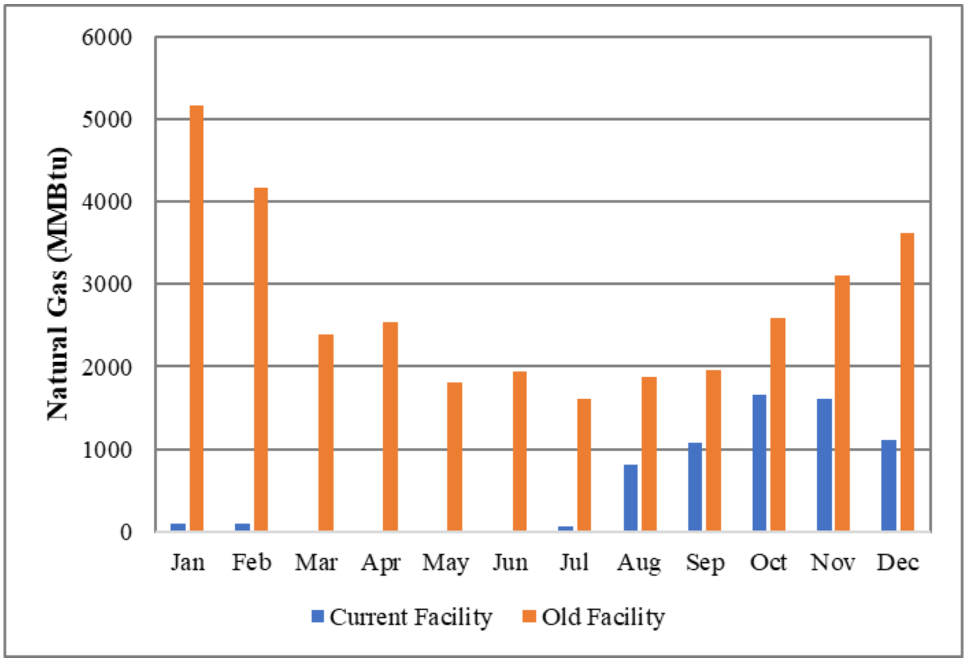


Figure 11: Comparison of natural gas values

The total energy, demand and natural gas consumption values have been shown in Table 3. This table clearly shows the stark difference between the values. They have significantly gone down over the years. When the plant was manufacturing, much of the gas energy was used to make process steam. This steam was used to heat vats and other process uses. This has essentially been eliminated, making fuel use only for space heating.

Electricity was used to drive many process machines, as well as some ovens and drying tunnels. Lighting power use was also higher because lighting technologies were not as advanced. Metal halide / high pressure sodium, as well as magnetic ballasted T12 fluorescent fixtures have been replaced with electronic ballasted T8 lamps – with a reduction in lighting power of 10% or more.

Something to take into consideration is how the change in process load affects the heating and cooling loads. Higher energy used in the process – both electricity and steam dissipate large quantities of heat into the production space. This increased the demand for cooling in the summer and decrease the need for heating in the winter. In many parts of the plant space heating may not even be needed on the coldest of days and space cooling required.

With the changeover to warehousing space, this process load is essentially eliminated, leaving only lighting. This changes the dynamics of the HVAC system, decreasing HVAC load (and making the chiller and pumping systems operate inefficiently as a result) and increase the energy used for heating. This helps in understanding the basis of suggesting the solutions that will be presented later.

Table 3: Comparing Annual Energy Values

	Old Facility	**Current Facility**	**% Reduction**
Energy	1,465,200 kWh	758,400 kWh	48%
Demand	3,514 kW	1,168 kW	66%
Natural Gas	3,612 MMBtu	1,111 MMBtu	69%

1.5: Energy Saving Opportunities

During the energy assessment of the plant, the team inspected the chiller system as part of a routine walk-around of the facility. The entire plant was being served by a chiller system which consisted of three chillers, one rated at 750 tons and two at 500 tons. Keeping in mind the energy values from Table 3, it is interesting to note that the chillers were sized according to the demand that the manufacturing facility experienced. Even then, the only chiller that was kept running was the 750-ton unit. It is assumed that the two 500-ton units were kept on standby. The 750-ton chiller was adequate for the facility’s demand when they manufactured products. The heat load from the machinery, people and lighting must have been enough to guarantee efficient loading of the chiller.

The facility since then, has seen some significant changes in their operation. They have converted into a warehouse for the most part. The load on the chillers today, includes the shell load and some lights. This is far off from what the load used to be for the chillers. The plant keeps the 750-ton chiller running to serve the current load. The two 500-ton units are not operable and might need overhaul with a refrigerant replacement (from R-11 to a new non-ozone depleting refrigerant). The following observations were made during the initial visit to the plant.

1. For a chiller to perform at its maximum efficiency, that temperature difference is usually 10-12°F. This is obtained when the chiller is loaded at about 60-70%. Chillers can account for up to 20% of the total energy consumption of a facility [3]. Improving the efficiency of the chiller and pumping system will help significantly reduce energy and the associated energy costs.
2. The next area of improvement in terms of energy efficiency would be the rooftop direct expansion (DX) units. These serve a significant area of the plant. This area could be served by the chiller which would help increase the loading on the chiller. The DX units can be fitted with chilled water coils being served by the chiller. Increasing the loading on the chiller would help in increasing the efficiency thus resulting in energy savings.
3. The chiller system utilizes multiple pumps for the pumping chilled water throughout the plant. This system was designed for three chillers instead of one. The entire plant’s needs can be met while keeping just one pump running as will be shown later.
4. Utilizing a primary-secondary or a primary-only variable flow chilled water pumping system.

CHAPTER 2: BASICS

2.1: Heating, Ventilating and Air Conditioning Systems

An HVAC system is a collection of equipment which is responsible for the air conditioning of a building by utilizing a distribution network which supplies filtered air. The complexity of any HVAC system will depend on the need of the space. While every manufacturing facility has different needs, the fundamentals of operation of HVAC systems will remain the same. For instance, some plants may be oriented towards pharmaceutical products. Such facilities will have much more stringent requirements in terms of quality of air, temperature setpoints and ventilation rates. There will be some changes in terms of how the plant's system is designed but the basic components do not change. These systems are usually designed with the ASHRAE guidelines in mind.

Depending on the scale, manufacturing facilities can easily consume millions of kilo-Watt hours of energy. HVAC systems are responsible for more than 30% of the total energy consumed by a manufacturing facility [4]. The facility engineers are on a constant lookout for ways to save the plant some energy and turn their recommendations into cost savings.

The following few sections will look at some of the components which make up the HVAC system of the plant being discussed for this thesis.

2.2: Central HVAC System

A centralized HVAC system is responsible for the cooling and heating of a large facility or building with multiple zones while having a base location [5]. The equipment in this base location utilize fuel and electricity to meet the cooling and heating demands of the facility. This is achieved with

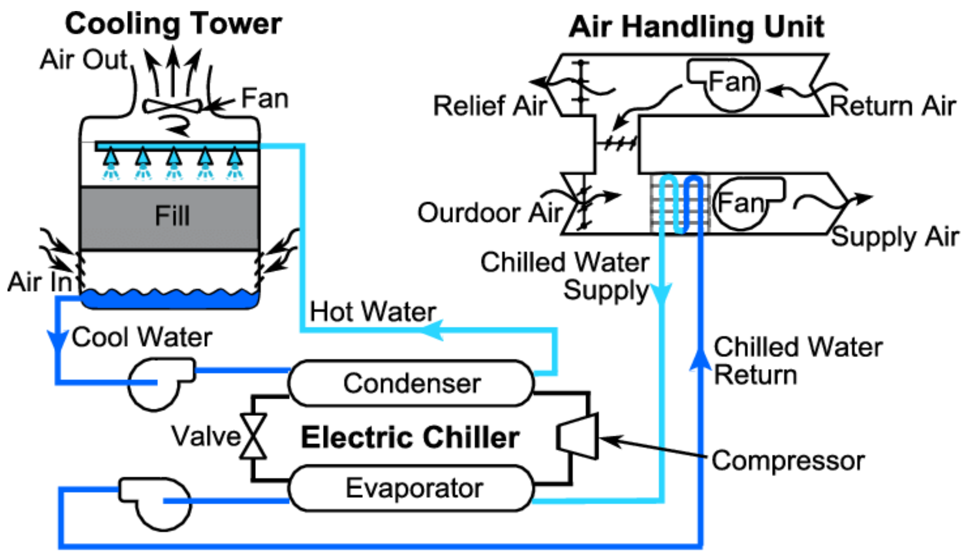


Figure 12: Chilled water system of the facility [7]

the help of chilled water, hot water and air delivery systems.

Figure 12 illustrates a chilled water loop of the facility. It consists of a chiller, air handling unit (AHU), cooling tower, refrigerant and the working fluid responsible for cooling the plant which in this case is water.

Chillers are machines responsible for removing heat from the working fluid that flows through the plant and has picked up the heat from the spaces. In this case, the working fluid is water. Therefore, chillers produce chilled water which is used for the air conditioning of a facility.

Figure 13 below shows the types of chillers.

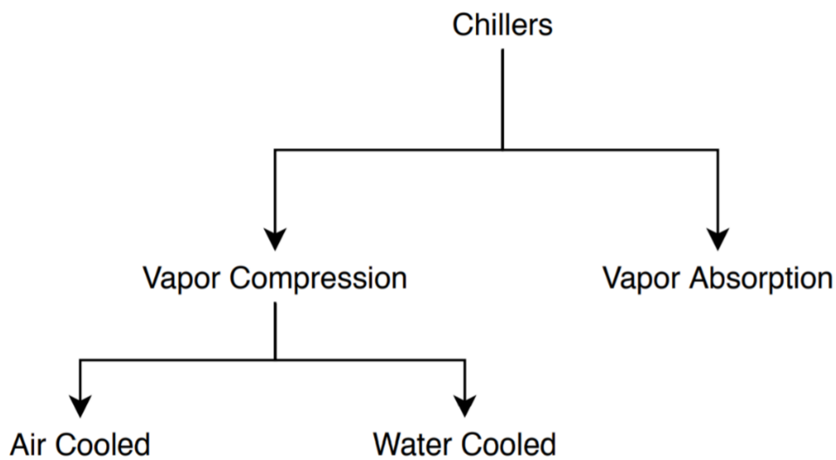


Figure 13: Types of Chillers

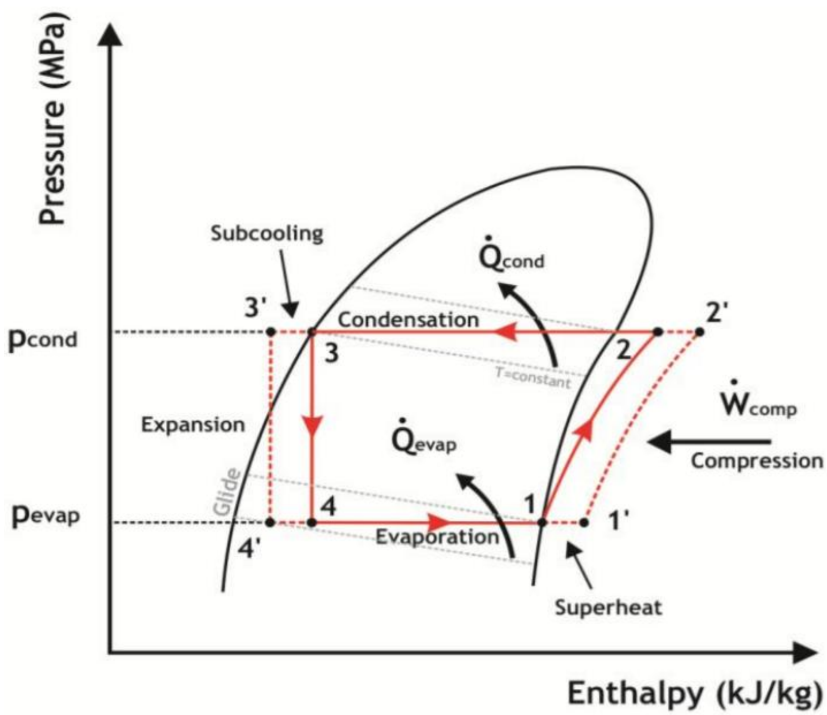


Figure 14: P-h diagram of the vapor compression cycle [8]

Figure 14 illustrates a vapor compression cycle. Process 1-2 compresses the refrigerant from low-temperature low-pressure saturated vapor to superheated vapor. This then passes through the condenser (Process 2-3) where it rejects heat to air or water depending on the condenser type. The temperature reduces and it reaches state 3. Process 3-4 is where the saturated liquid passes through an expansion valve and flashes into a liquid-vapor mixture. This mixture passes through the evaporator, where it absorbs heat and turns into saturated vapor and the cycle begins again.

Chillers can also be classified based on the compressor technology. There are 4 main types, centrifugal, reciprocating, scroll and screw chillers. With increasing complexity, the classification of chillers can be as varied as one would want it to be. For the sake of simplicity, the water-cooled centrifugal chiller will be discussed since the facility uses that type of chiller.

Stating that a chiller is centrifugal is equivalent to saying that the chiller's compressor is centrifugal. It is the driving force of the chiller and is necessary for the compression of the refrigerant to a higher pressure and temperature. Figure 15 below shows the basic components of a centrifugal compressor. The cool vapor refrigerant enters the compressor after having passed through the evaporator. The impeller, which is spun by a motor, rotates at a high speed which increases the kinetic energy of the refrigerant. There is also a compression effect which increases the pressure of the vapor. The high velocity refrigerant vapors enter the diffuser. Its primary goal is to increase the pressure and slow the gas down. The velocity gets converted to pressure. Thus, the compression process is complete. Finally, it passes through the volute and moves on to the condenser.

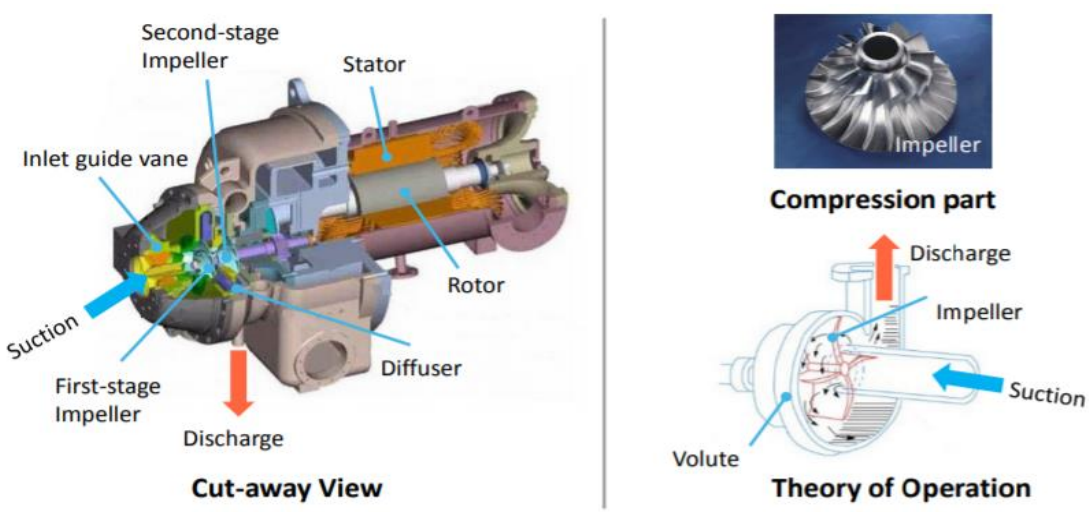


Figure 15: Centrifugal compressor [9]

Condensers are heat exchangers responsible for extracting heat from the high temperature refrigerant. There are a few types of condensers which include air cooled, water cooled and evaporative cooled condensers. For this thesis, a water-cooled shell and tube condenser will be considered.

As seen in Figure 16, the hot refrigerant vapors enter the condenser (shell side) and are cooled by water (tube side). This exchange of heat between the refrigerant and water follows the principles of any shell and tube heat exchanger. The tubes of the condenser have cold water flowing through them, while the tubes are surrounded by the refrigerant. Usually baffles are installed to distribute the gas across the tubes and increase the heat transfer coefficient. The refrigerant loses its heat to the cold water and condenses. The water leaves the condenser and goes to the cooling tower to lose its heat.

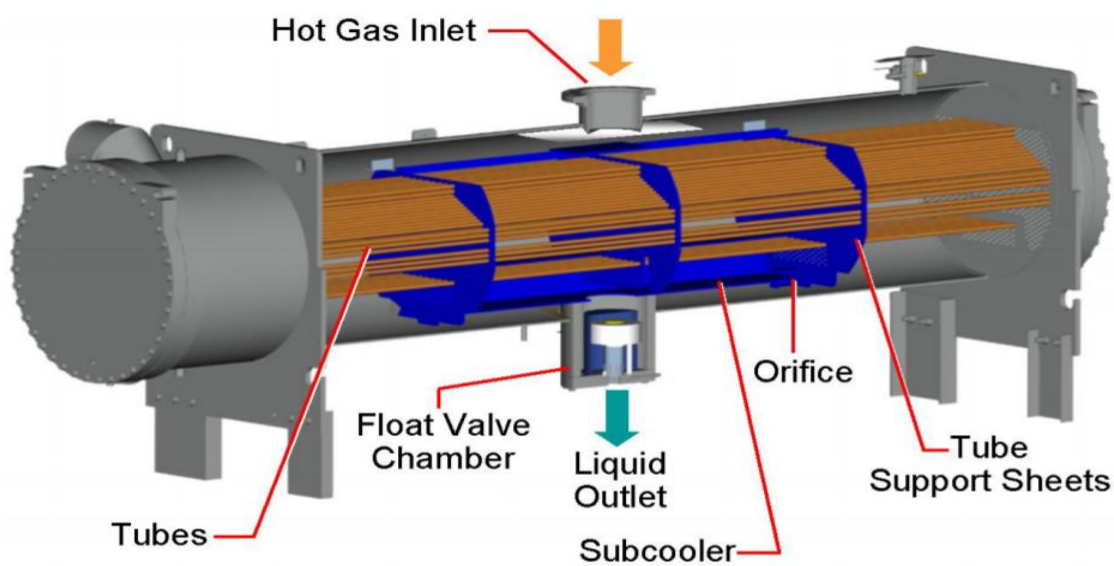


Figure 16: Water cooled condenser [10]

From the condenser the refrigerant passes through an expansion valve to get to the evaporator. It acts as a metering device. It will control the amount of refrigerant flowing into the evaporator based on the cooling loads. If more than desired refrigerant flows into the evaporator, then not all of it gets converted into vapor. This will cause a liquid-vapor mixture to flow into the compressor which will cause damage as they are not designed to handle liquids. After passing through the expansion valve, the refrigerant loses pressure and will turn into a liquid-vapor mixture.

The facility has Trane CenTraVac centrifugal water-cooled chillers. These chillers use multiple orifice expansion devices [10]. This is one of the more inexpensive and simpler forms of achieving the expansion of the refrigerant. When the liquid refrigerant passes through the device, some of it will flash to gas and this can be sent to an economizer. The gas is then redirected to a lower pressure stage of a multi-stage compressor. Thus, the low-pressure liquid refrigerant flows to the evaporator with more room for heat absorption. The liquid refrigerant in the economizer can be passed through a second expansion device to be sent over to the evaporator. This type of expansion valve, although relatively cheap and simple to manufacture, can lead to a dip in performance when the chiller is made to operate at the edge of its operating limits. There are two conditions where the said operating limits of a chiller could occur. One would be the high head, low flow condition. This could occur when the chiller is undersized for the application it is meant to be used for. Another reason could be an ineffective condenser operation where it is, for some reason, not able to transfer heat from the hot compressed gas to condenser water. The expansion valve will not be able to create a liquid seal at the orifice and hot gas enters the compressor. It is also referred to as hot gas bypass, which is when high pressure gas is circulated from the compressor directly to the inlet of the evaporator to keep the flow rate up.

The other condition would be low head, high flow which could occur when the chiller is oversized. Here, the chiller will be operated at a lower load which implies achieving less cooling at the expense of using more energy. Both scenarios lead to inefficient operation of the chiller. Therefore, it is important to ensure that the chiller is sufficiently loaded to obtain efficient performance.

After the expansion valve, the refrigerant enters the evaporator ready to soak up some heat and as a result cool water flowing throughout the facility. Just like the condenser, the evaporator is a heat exchanger as well. The Trane chiller at the facility uses a flooded shell and tube type evaporator. The shell side will be filled with refrigerant whereas the tube side will have water flowing through it as shown in Figure 17.

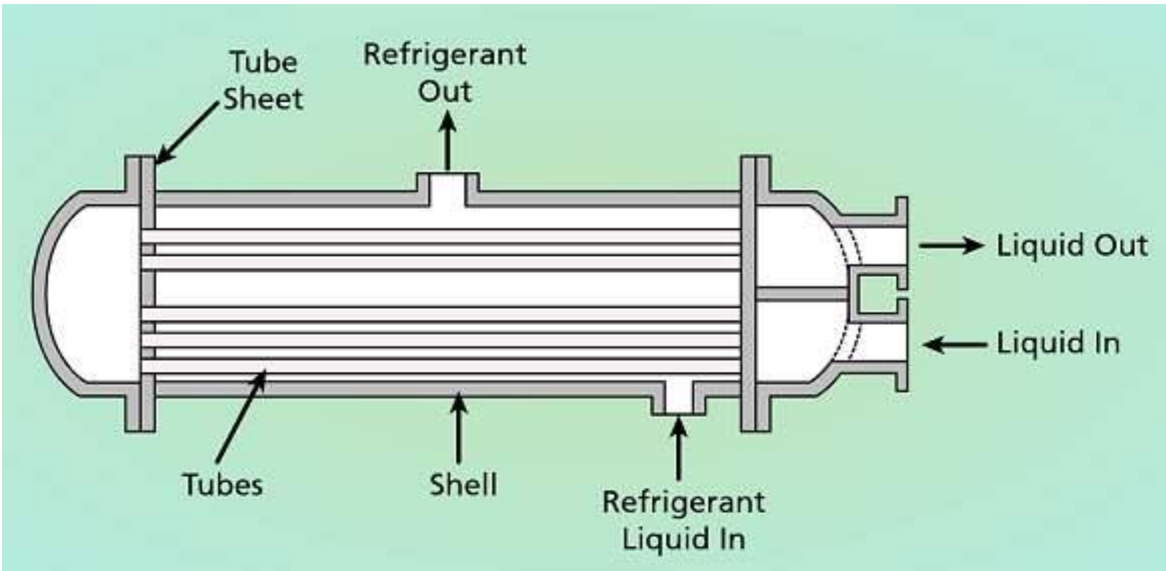


Figure 17: Shell and tube evaporator [12]

The shell side refrigerant turns into vapor state after absorbing heat from the water. The suction side connects to the compressor. Measures must be implemented to prevent droplets of refrigerant from entering the compressor because that could cause some damage. The evaporator can be designed such that there is sufficient space for the vapor to collect in a region above the tubes to reduce its velocity as shown in Figure 18. It would ensure that the vapor does not carry away droplets along with it and the droplets being heavier will stay near the tubes. Eliminators like wire meshes can be employed which can trap the droplets from reaching the compressor exhaust as shown in Figure 19.

The refrigerant vapor enters the compressor, and the refrigeration cycle resumes. The chilled water output from the evaporator is sent to different air handling units in the facility to be used for cooling the air. The warm air rejects its heat to the cold water which is then sent back to the evaporator to be cooled by the refrigerant.

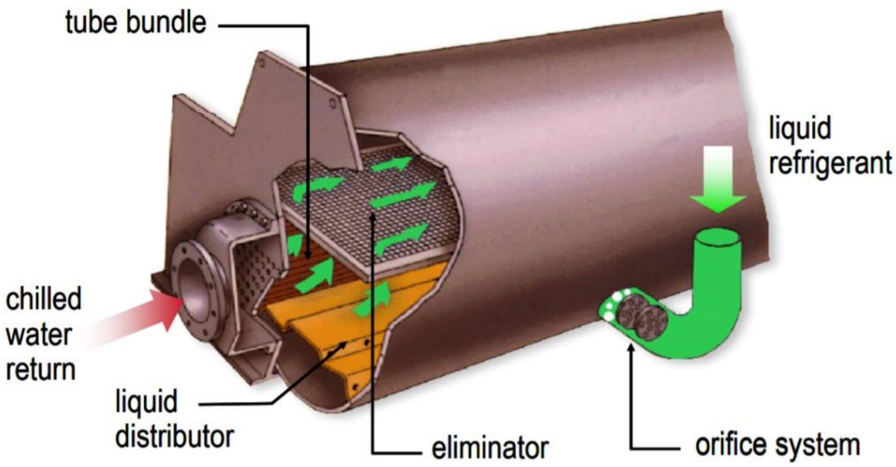


Figure 19: Mesh eliminator [13]

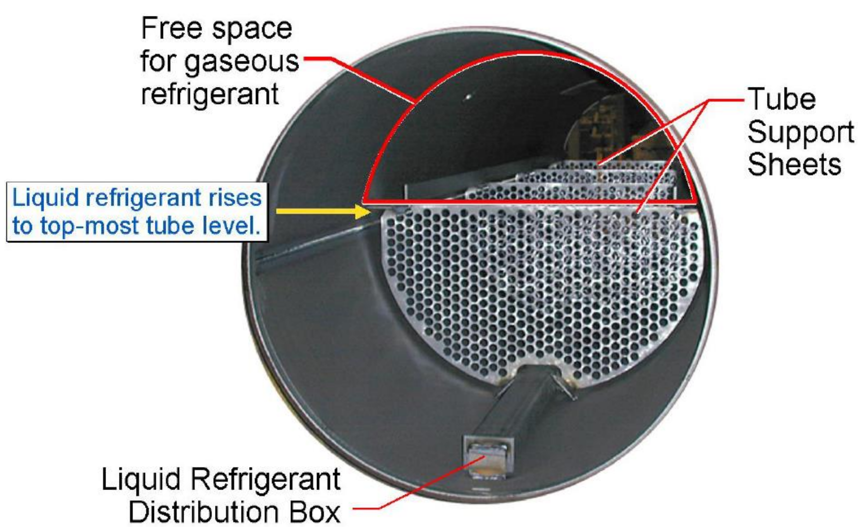


Figure 18: Space elimination of refrigerant droplets [10]

Air handling units are responsible for the distribution of conditioned air throughout the facility. Air-Conditioning, Heating and Refrigeration Institute sets standards and guidelines for manufacturers to enable fair comparisons and ratings which help customers make better purchase decisions. One of many standards is the AHRI Standard 430 which deals with AHUs. It defines the AHU as "A factory-made encased assembly consisting of a supply fan or fans in parallel which may also include other necessary equipment to perform one or more of the functions of circulating, cleaning, heating, cooling, humidifying, dehumidifying and mixing of air. It shall not contain a source of mechanical cooling. This assembly is capable of use with duct work having a total static resistance of at least 0.5 inches of H_2O." [13]The two main types of AHUs would be packaged and central air handlers. Packaged AHUs are generally used when the tonnage requirement is fairly low (5-30 ton range) [14].These units are typically used with a corresponding split AC unit; however, they can be fitted with chilled water piping.

Central station AHUs are used when the cfm requirement is much greater. The range could be anywhere from 1,500 cfm to over 100,000 cfm. Their design is dependent on the application they have to serve. Clean rooms in pharmaceutical plants would need high grade filters and strict temperature control as opposed to a manufacturing plant where temperature can vary between a larger range and the air does not have to be as clean. Therefore, as shown in Figure 12, the AHU sends cool air to specific zones in the plant with the help of fans and ducts. Return air from spaces is mixed with fresh air which is conditioned and sent back throughout the facility.

As discussed earlier, the chiller condenser is responsible for cooling the refrigerant. This can be done by using either an air-cooled or a water-cooled condenser. The chiller at the facility has a water-cooled system which makes use of a cooling tower, as shown in Figure 12. The major benefit of using cooling towers over air-cooled condensers would be that significant energy savings are obtained. In air cooled heat exchangers, the water rejects its heat to the air that blows over the pipes. For setting up the air flow, fans will be required which would demand a lot of space. Moreover, the cost of running fans would result in higher energy bills. Cooling towers are more efficient in terms of bringing the temperature of water down. They can cool water to within 4 to 5°F of the ambient wet-bulb temperature as opposed to air cooled heat exchangers which cool water to within 20°F of the ambient dry-bulb temperature [15].

A cooling tower works on the basic principle of mass and heat transfer. The water to be cooled is pumped to the tower which is sprayed onto a fill. It is used to increase the surface area and the time of exposure of water to improve the heat transfer rate between atmospheric air and water. The air which flows across the fill absorbs some heat from the warm water and the heat transfer is dependent on the relative humidity. Lower relative humidity value will be better for the cooling tower because the air will be able to absorb more water thus resulting in better cooling. As the water evaporates it will reduce the temperature of the water that is left behind. The cool water falls to the bottom of the tower where it is collected and sent to the condenser. Some droplets of water that usually get carried away with the vapors can be trapped by using drift eliminators. These structures reduce the velocity of the water droplets and ensure that they fall back down into the tower. Make up water is introduced to replace the water that is lost due to evaporation or blowdown [16].

The facility uses a direct contact cooling tower which means that the water circulating in the condenser comes in contact with the air that is used to cool it. As shown in Figure 20, the fan is at the top of the tower which makes it an induced draft type as opposed to the fan being on the side (forced draft). The condenser uses the cool water from the tower for exchanging heat with the hot refrigerant vapor.

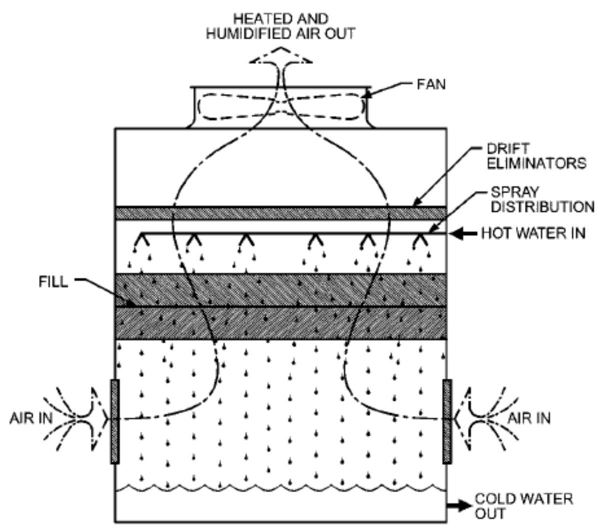


Figure 20: Induced draft counterflow cooling tower [16]

2.3: Packaged Rooftop System

In the facility, part of the production area and the warehouse were not being served by the main chiller loop. They were being served by direct expansion rooftop units. These units are all-in-one HVAC systems that contain all components necessary to condition the air as desired. Figure 21 shows such a system.

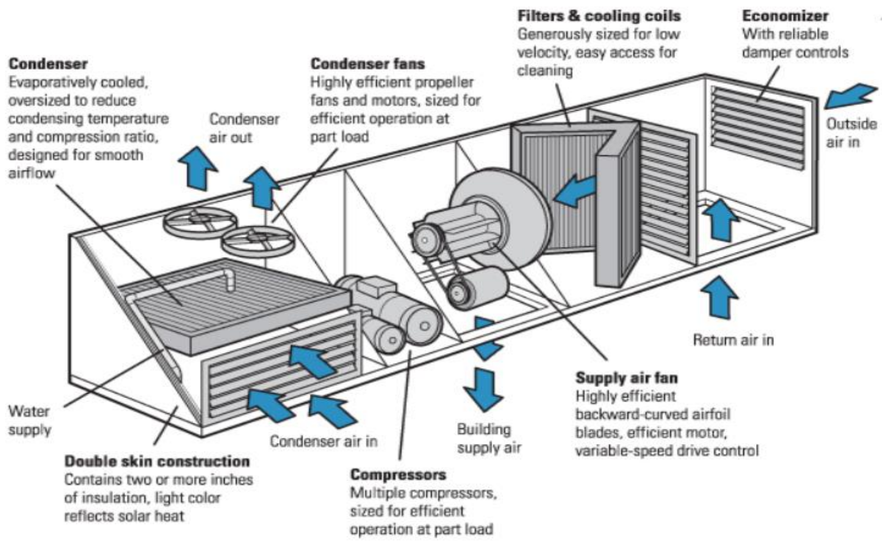


Figure 21: DX Rooftop Unit [18]

The outside air is sucked in with the help of fans and it passes through filters which help keep the air relatively clean. The air passes over a cooling coil which is where heat exchange between the cool refrigerant and the warm air takes place. The warm refrigerant goes over to the compressor, followed by the condenser and the expansion valve and follows the vapor compression cycle. The rooftop unit at the facility is a constant air volume system (CAV) with terminal reheat which uses hot water. A CAV system will supply a fixed volume of air while varying the temperature as per the demand of the facility. Though inefficient as compared to the variable air volume systems, it is relatively inexpensive and for a space with a fairly constant cooling load, the CAV systems can be a viable option.

CHAPTER 3: LITERATURE REVIEW

3.1: Chilled water pumps

Pumps are an essential part of any chiller system. They are used to transfer fluids to provide cooling services. Commonly a pumping system will have 5 main components: pumps, piping, valves, prime movers and the equipment (AHUs, fan coil units or CAV/VAV boxes) [18].

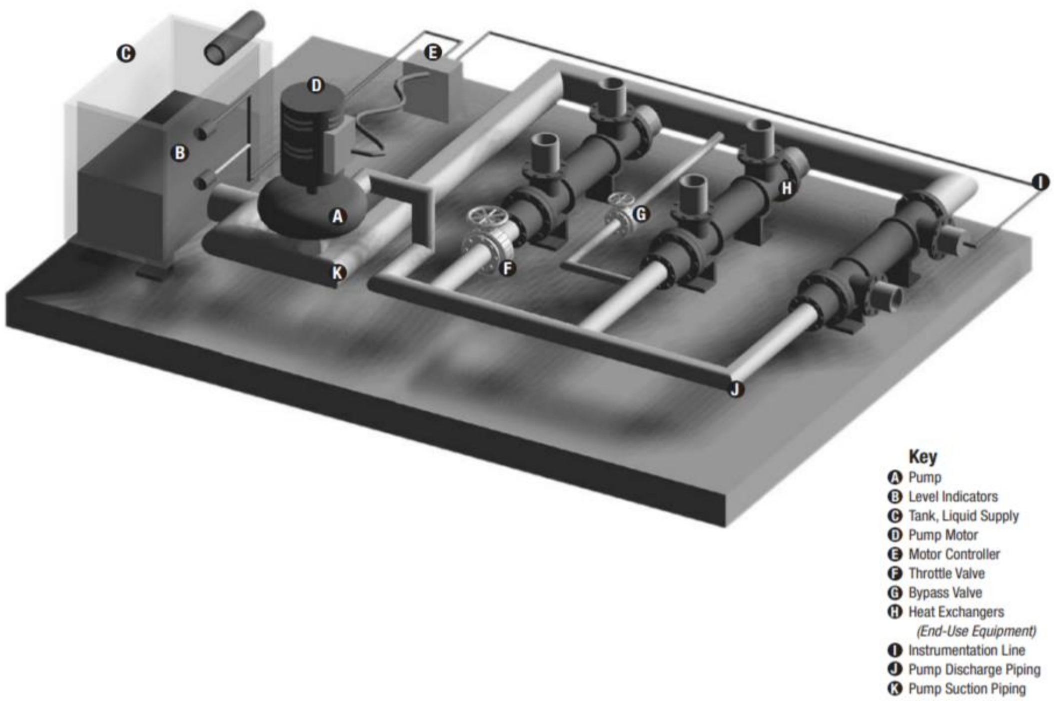


Figure 22: Key components of a pumping system [18]

A chilled water system makes use of piping and pumps to ensure chilled water is transported to different zones of the building. The efficient design and selection of equipment for the system becomes of utmost importance since pumps consume around 6 to 12% of the total annual energy consumption of the facility [19]. An undersized pump will provide lower flow and the HVAC system will struggle to provide appropriate amount of cooling. On the opposite end of the spectrum, an oversized pump will move large amounts of water which will result in over cooling and energy being wasted [20]. Chiller plants usually make use of centrifugal pumps, which is the case for this plant as well, for several reasons which include lower operating costs, ease of maintenance, varying range of flow rate and transport of low viscosity liquid (water) [21].

3.1.1: Pump sizing

There are two key parameters to consider when sizing a pump – head and system flow rate. The final amount of flow and head that one can expect to obtain from a pump will depend on the piping layout. Friction in pipes, velocity of fluid and elevation difference are some of the factors that affect the final output of the pump.

3.1.2: Head and types of head

The term head is used to obtain a measure of the kinetic energy that a pump can create or it can be described as the height of the liquid column produced if all of the kinetic energy is transferred from the pump to the liquid [21].

Different types of head that may come into consideration while sizing a pump –

Static head – The elevation from the source of water to the point where the water is delivered. It is independent of the flow rate of the system.

Static suction head – The vertical distance measured from the pump's center line to the source of water.

Static discharge head – The vertical distance measured from the pump's center line to the point where the water is delivered.

Friction head – This is essentially the resistance to flow offered by the pipes and fittings. With varying velocity of fluid, material, size and length of pipes, types and number of valves, and fluid properties, the value of friction head will vary.

The head loss due to friction in pipes is termed as major head loss and is calculated by Darcy's formula.

$$h_{f,\mathrm{major}} = \frac{f * L * \upsilon^2}{2 * g * D}$$

Where, $h_{f,major}$ = head loss due to friction in pipes (ft)

f = friction factor, can be determined with the use of the Moody chart which makes use of the roughness values of the material of pipe and the Reynolds number

L = length of the pipe (ft)

υ = mean velocity of flow (ft/sec)

g = acceleration due to gravity (ft/sec^2)

D = internal diameter of the pipe (ft)

The loss in head due to fittings or valves is termed as minor head loss and can be calculated by –

$$h_{f,\min or} = K * \frac{\upsilon^2}{2 * g}$$

Where, $h_{f,minor}$ = head loss due to resistance from valves/fittings (ft)

K = resistance coefficient, which is a constant depending on the fitting or valve used. Crane Technical Paper 410 has data on 'K' values for different valves and fittings

υ = mean velocity of flow (ft/sec)

g = acceleration due to gravity (ft/sec^2)

Pressure head – It refers to the difference in head between the inlet and outlet. In a case where the pump receives water from a pressurized tank and discharges to a tank open to water, a pressure head will exist between the inlet and outlet.

Velocity head – A change in velocity of the fluid at the inlet and outlet will create a difference in the kinetic energy and thus produce a head.

Depending on the kind of the system it is, the values of head can be zero. The piping layouts can be classified as closed loop or open loop.

A **closed loop** consists of a liquid being pumped in a continuous loop with the inlet and outlet being indistinguishable. In such systems the change in pressure, velocity and static head values can be considered to be zero. An example of a closed loop system will be the HVAC chilled water loop which moves from the chiller to all the AHUs and then back to the chiller. It might seem counter-intuitive to consider the static head to be zero since there clearly is an elevation which the liquid must climb to reach different parts of the building. This can be explained by the fact that for every foot the liquid must rise, it is being pulled down by gravity at a different location since it is a closed loop. Thus, the static head can be neglected. In a closed loop system, a pump essentially just needs to overcome the friction head drop to sufficiently serve the area.

In an **open loop system**, fluid is not moved around in a loop and instead is pumped from one location to a different one without the liquid being transported back. The diagram below shows an open loop system. Clearly, the pressure, velocity and static head values may not necessarily be zero. For example, when the tanks are open to atmosphere, the pressure head will be zero. If a tank is pressurized, then a pressure head will exist.

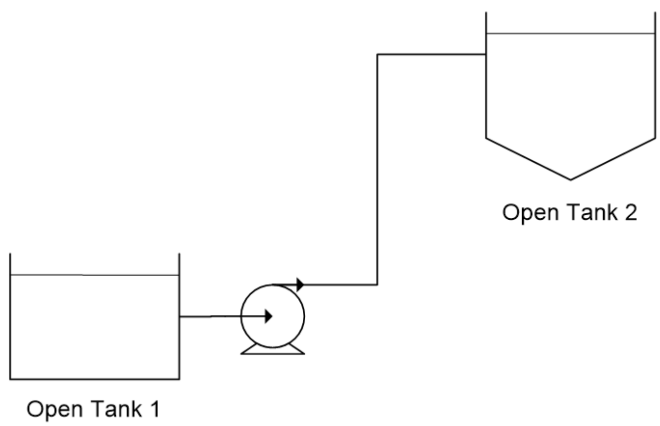


Figure 23: Open loop system

3.1.3: System curve

The system curve describes the relationship between the head and flow rate for the piping system. As discussed above, in a closed loop system, only the friction head needs to be considered in the calculations since other head values will be zero. Therefore, a pump needs to be sized such that it can overcome the friction losses provided by the system. The sum of $h_{f,major}$ and $h_{f,minor}$ will be the head requirement of the pump.

$$\Delta h_{pump} = h_{f,\text{major}} + h_{f,\text{minor}}$$

$$\Delta h_{pump} = \frac{f * L * \upsilon^2}{2 * g * D} + K * \frac{\upsilon^2}{2 * g}$$

The only variables in the above equation are the velocity and friction factor terms. Velocity is equal to the flow rate divided by the area of the pipe which means velocity and flow rate are directly corelated. The friction factor does not show a huge variation with changing flow rate [22]. Thus, the equation can be simplified to –

$$\Delta h_{pump} = C * Q^2$$

This equation can be plotted on a graph of the pump curves which is provided by the manufacturer to find the operating point as shown in the figure below. For a closed loop, the system curve will pass through (0,0) on the graph whereas in case of an open loop, the curve will be shifted upwards along the y-axis by a value equal to the static head.

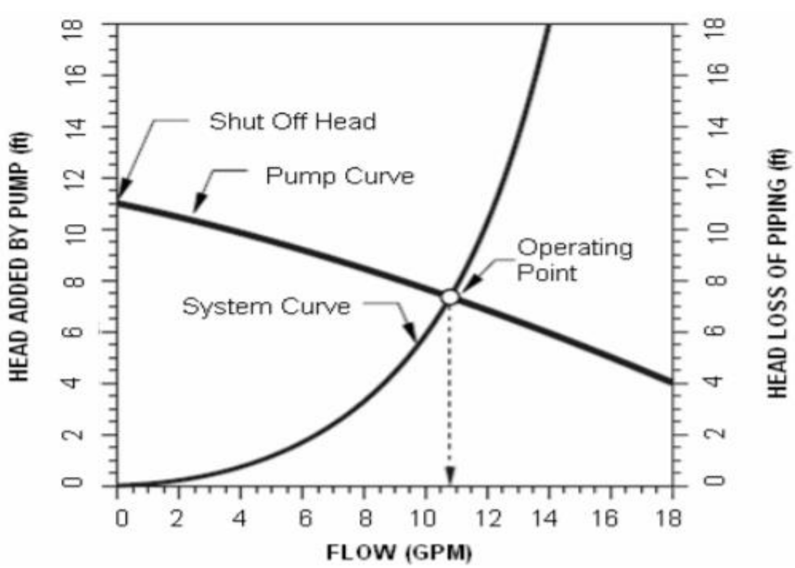


Figure 24: An arbitrary system curve [21]

In a pump curve sheet, there is more information provided, in the form of curves, like efficiency, NPSHr, speed and required input power. The figure below shows a generic pump curve plotted for an arbitrary RPM and impeller size. The pump selection is done such that the operating point lies in the region where maximum efficiency can be obtained.

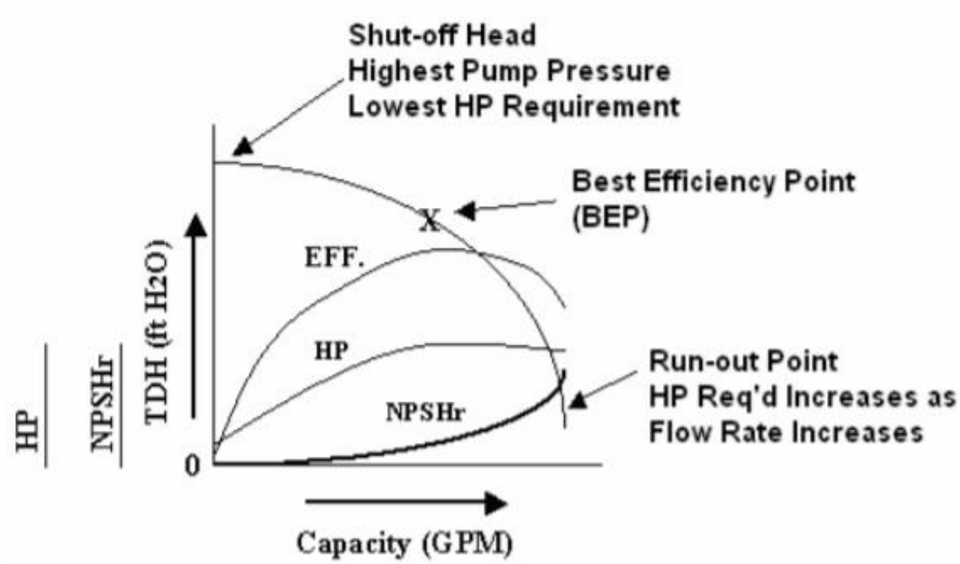


Figure 25: Pump curve details [21]

3.2: Control valves

3-way and 2-way valves are two of the most common types of control valves that are used in a chilled water piping system. 3-way valves are generally used for constant volume flow applications. They have been traditionally used in systems where the chiller is not capable of handling flow rate changes.

Improper balancing is one key issue with using 3-way valves. At part loads, these valves have been found to increase flow rate as compared to design values. Most 3-way valves used have two linear characteristic control ports. A linear characteristic port will allow flow to increase proportionally with the valve opening as shown in the figure below. A better-chosen valve would have the bypass port with linear characteristics and the flow control port with equal percentage characteristic. This configuration will help in keeping the flow rate within design values when at part load conditions [23].

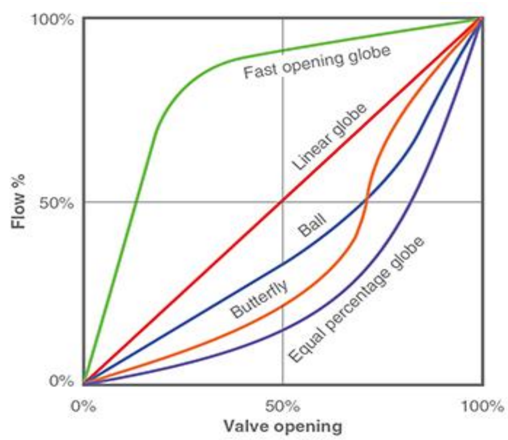


Figure 26: Flow characteristics of a typical globe valve [24]

The reason for increased flow rates when the system is at part load can be explained as follows. When the system is at say, 50% load, the bypass valve opens partly, and chilled water has 2 paths to flow through. Pressure drop varies with the square of the flow rate which leads to increased total flow rate through the valve. An example discussed in 'ASHRAE - Fundamentals of Design and Control of Central Chilled-Water Plants' clearly demonstrates the issue poorly chosen 3-way valves generate for the chiller system. The screenshot of the table in Figure 27, below depicts the result of the example from the book. At 50% flow to coil position, the flow rate rises to 132 GPM when it should be at 100 GPM. To avoid such a scenario, the valves need to be designed such that the flow control port has equal percentage characteristic [25].

Item	Pressure Drop (ft) @ 100 gpm		
	100% to Coil	50% to Coil	0% to Coil
Pipe/valves	2	2	2
Coil and/or bypass	8	2	6
Globe control valve	10	7.5	12
Total	20	11.5	20
gpm @ 20 ft ΔP*	100	132	100

Actual ΔP available may change.

Figure 27: Flow variation as a function of valve position [25]

To make the control systems simpler to operate, the heat transfer at coil side and the valve lift need to have an approximate linear relationship. The heat transfer varies non-linearly with respect to flow rate which is where the valve's equal percentage characteristic will come into play. The figure below shows the ideal combination of the equal percentage valve and coil heat transfer against varying flow rate to create a linear relationship.

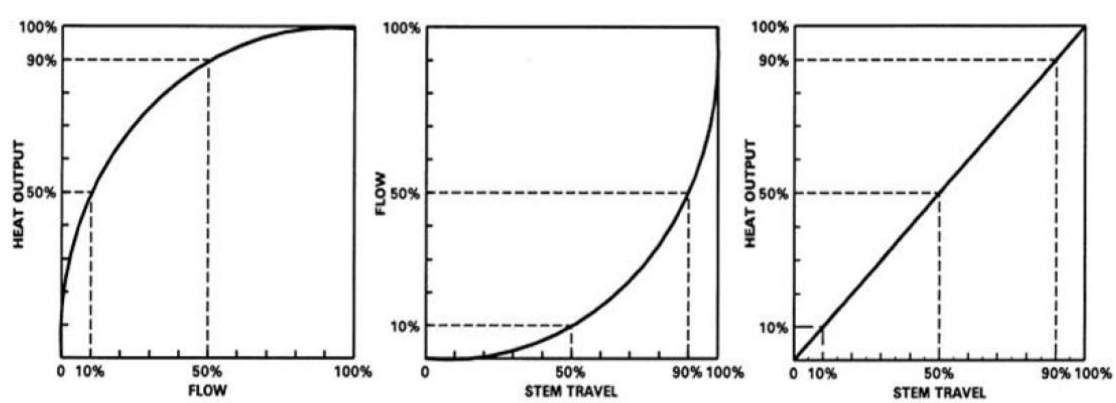


Figure 28: Ideal combination of equal percentage valve curve with water coil emission curve [23]

2-way valves, on the other hand, are used for variable flow. The valve can throttle flow when the operation is part load and there is no requirement of bypassing the flow as is done in 3-way valves.

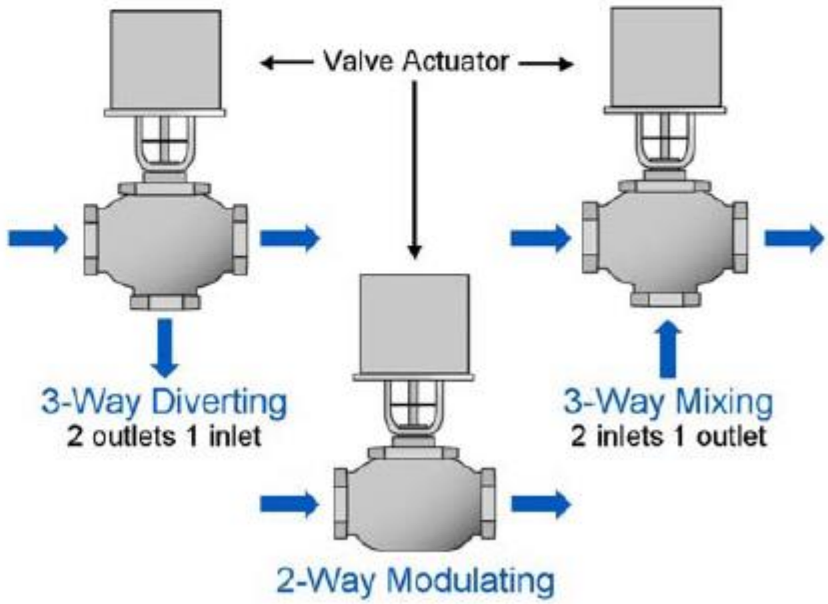


Figure 29: 3-way and 2-way valves [26]

3.3: Degrading ΔT for a chiller plant

The low ΔT syndrome refers to the degradation in the difference between the incoming and outgoing chilled water temperatures. There can be a lot of different causes which contribute to lower ΔT but the effect is the same and can be understood with the chiller capacity equation as will be derived in Section 4.6.1.

$$\frac{23.97}{\Delta T(^{o}F)} = \frac{\dot{V}(GPM)}{Q(tons)}$$

Lowering ΔT limits the chiller capacity and the amount of heat it can remove from the space. Where one chiller could probably be sufficient to remove the heat from the space, the facility is forced to turn on an additional chiller along with its pumps resulting in more energy used.

Some causes of low ΔT include but are not limited to [27],

1. Use of 3-way valves – Theoretically, these valves maintain constant flow and ΔT will change with load. However, at part load conditions, the flow in 3 way valves increases thus lowering ΔT [28]. This is an issue for variable flow systems. The pumps will never get a chance to slow down and there will be no energy savings.
2. Improper setpoint calibration and coil selection – The lowering of setpoints below design values will lead to coils opening the valves completely. This would lead to an increase in flow rate and simultaneous reduction in ΔT. Low ΔT can also be caused by undersized coils where the terminal boxes are trying to remove more heat than it was designed for.
3. Cooling coils piped in parallel – It is not a rare occurrence to find the chilled water coils piped such that the entering air and chilled water flow are on the same side. It leads to lower heat transfer as opposed to the coils being piped in a counterflow arrangement. With lower heat transfer, more amount of water will be needed to bring the air to setpoint temperature.
4. Oversizing control valves – An oversized valve will cause the system to over- and under-shoot the setpoints. It will cause the system to use more chilled water than necessary.
5. Reduction of velocity in pipes – If the flow in the pipes becomes laminar, the heat transfer rate will reduce causing coils to call for more water, thus degrading ΔT.

6. Cooling coil inefficiencies – The coil effectiveness reduces over time due to fouling on water or air side pipes. The setpoint is not reached due to ineffective heat transfer and thus flow rate required increases.

The issue of low ΔT due to bad design or improper operation can be looked at by the facility and be rectified to a certain degree, however, there are some causes which are bound to creep in as the system ages. To improve the chiller plant's operation at part load the facility can look to a lot of different solutions. Opting for a variable speed chiller in place of a constant speed unit will help save energy in the long run. It is a fact that a chiller will operate at design load for very few days throughout the year. Most of its time is spent at part load and improving part load efficiency will greatly benefit the plant. The figure below shows a clear distinction between fixed and variable speed chillers in terms of % kW consumed. A variable speed chiller outperforms the constant speed chiller by a fair margin at part load conditions.

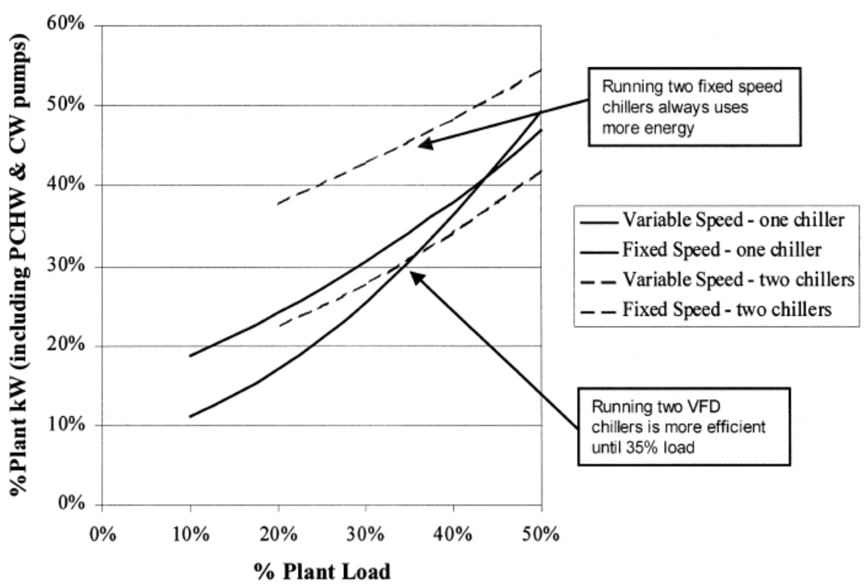


Figure 30: Plant kW consumption vs Plant load [27]

A different approach would be to change the pumping system to variable speed. A constant primary variable secondary system or a variable primary only system would decrease the amount of energy consumed by a large extent and will be discussed in the next section.

Lowering the supply chilled water temperature is a solution which seems like a natural course to take. It will surely increase the ΔT and reduce pump energy consumption but it will come at the cost of decreased chiller efficiency and result in greater energy consumption overall. Therefore, this is a quick solution for plant managers but not something to be applied for many operating hours.

The common leg in the primary secondary system can be fitted with a check valve to control the flow direction through the pipe. As discussed, when the secondary flow is greater than the primary flow, ΔT starts degrading and the system will spiral out of control. With the check valve, this scenario can be avoided. It will essentially put the primary and secondary pumps in series. The flow rate could increase by 30% in the primary loop. The extra flow of water will inevitably lead to a rise in pumping energy. As a result, a greater amount of chilled water is produced and could lead to overcooling of spaces if the secondary pumps are not VFD [27].

When the team visited the facility, they could hear some flow through an inactive chiller. A possible explanation can be formulated as follows. The team noticed that the valve on the common leg was in a closed position. This could cause some of the return chilled water to go through the primary pumps of the inactive chillers. The bypassed water mixes with the chilled water from other chillers raising the supply chilled water temperature, resulting in lower ΔT values. It can be avoided with the use of a headered layout for the primary pumps.

From the above discussion and understanding of low ΔT syndrome, one could conclude that the facility suffers from it. The facility at the moment uses just the 750-ton chiller to meet its demand. It will be discussed in Chapter 4 that the flow rate through the chiller is much higher than the usual 2.4 GPM/ton. This means the chiller is being over pumped to meet the cooling demand of the plant. When the Carrier HAP simulations were run for the plant, the maximum tonnage required came out to be 600 tons. The chiller is being over pumped without being fully loaded is a classic case of low ΔT.

3.4: Chilled water distributions systems

There are different approaches an engineer could take to design a chilled water distribution system. The designer would need to make sure that the system is energy efficient by considering how the components react to changing loads. It is of great importance to look at different arrangements especially for a facility which has seen a drastic change in its cooling load and operation. Some of the more common arrangements have been discussed below.

3.4.1: Constant-flow chilled water systems

This is the simplest type of any water distribution system. It consists of constant speed pumps which distribute chilled water at constant flow rate throughout the facility. The flow rate is set as per the peak loads, which is why the amount of water flowing through the chiller does not change. Temperature control at the cooling coils is done with the help of 3-way valves. At part load, a certain amount of chilled water will flow through the 3-way bypass line.

There is not a lot of room to save pumping energy in such a system. Assuming a parallel chiller constant flow system, like the one in Figure 31, exists in some facility and the chillers and loads are reasonably equal to each other in terms of tonnage and cooling requirements respectively. It will function well when the loading on cooling coils is high. If the load drops and load 3 requires no further cooling, theoretically, chiller 1 and 2 should be able to satisfy the cooling demand but with the use of 3-way valve there is no flow variation in the distribution load side. The chilled water simply bypasses the cooling coils when the load is inactive. If chillers 1 and 2 are the only chillers running, the flow rate drops and cooling coils 1 and 2 are starved. This facility would look to turn the third chiller on along with its associated pump to meet the cooling requirement.

Therefore, the plant will have to operate with all chillers running throughout the year. This is a specific case of operation but a common one, nonetheless. Having the flexibility of varying flow rates will help in great amounts of energy and cost savings.

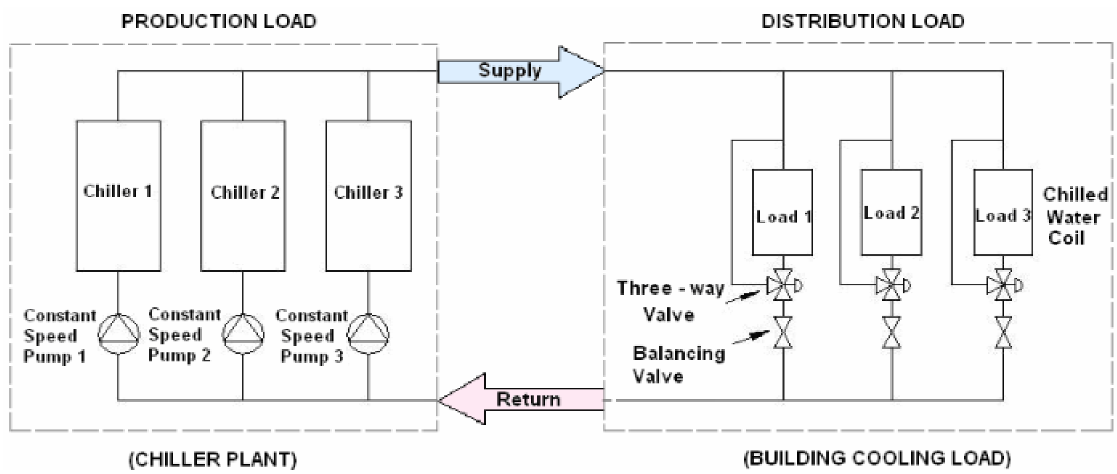


Figure 31: Constant flow chilled water system [19]

3.4.2: Primary - Secondary chilled water system

The primary loop is responsible for the production of chilled water and the secondary loop supplies chilled water to serve the cooling loads of the building as shown below. The primary loop is generally served by constant flow pumps which keep water running through the chillers. The secondary loop can make use of constant speed or VFD pumps. A bypass line is installed which allows excess chilled water to circle back to the chillers instead of having it flow through the entire

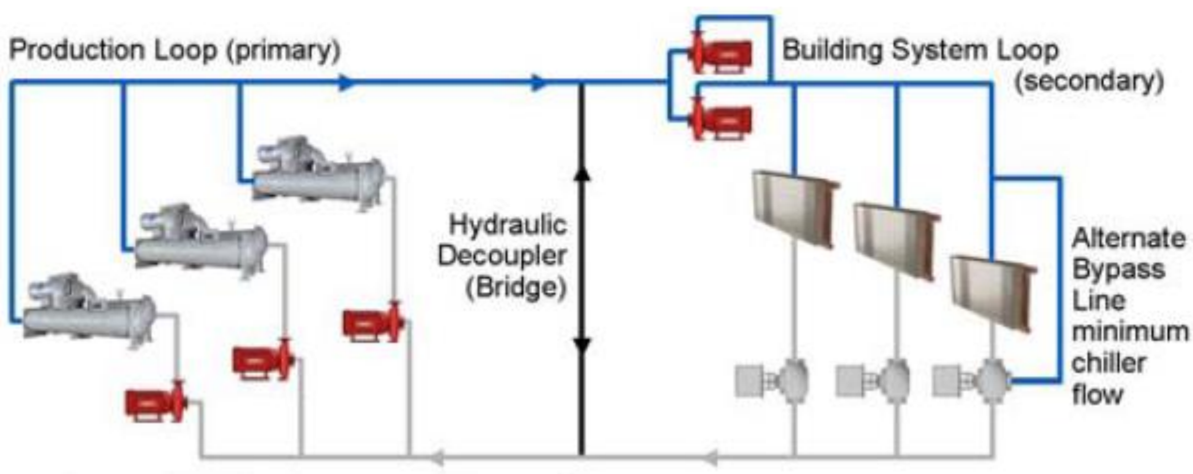


Figure 32: Primary - secondary chilled water system

plant. This helps in saving pump energy. The bypass line is made such that its run is very short to have a low pressure drop. It is usually contained within the mechanical room.

If the secondary loop pumps are constant speed, then it will have 3-way valves to vary the amount of chilled water according to the cooling load. Although better than primary-only constant speed pumping, this system has its own disadvantages. At part loads if a chiller and its pump are shut down, the flow rate in the primary loop will reduce. This could create a scenario where there is more water flowing through the secondary loop. Excess water will then flow through the bridge and mix with chilled supply water which results in degradation of supply temperature. The cooling coils will then open their valves to ask for more chilled water since the demand is not being met and this could result in zones never reaching setpoint temperatures. This would prompt the engineers to start another chiller to meet the cooling requirements. Since both pumping loops are constant volume, pumping energy savings will be low. For this to work, the loads and chiller system would have to be very similar in cooling requirement and tonnage.

For improving the energy efficiency of this system, the secondary pumps need to be driven by VFD motors which is how primary secondary systems have been traditionally made. 2-way control valves open according to the demand of the coils and VFDs can indirectly vary the amount of chilled water flowing through the coils. The speed of the pumps is set by using the differential pressure across the most critical coil in the system. When the valves close, the VFDs lower pump speed and thus lower the flow. The excess water will flow through the bypass line, thus ensuring constant flow through the primary loop.

As opposed to constant speed pumps, this system will save energy by running the secondary pumps as per the demand of the plant. According to the affinity law for a pump, the power consumed is directly proportional to the cube of the speed of the pump. If the speed is reduced to half the original speed, the power consumed will drop by about 85%, thus resulting in significant energy and cost savings.

This system can suffer from the issue of low ΔT unless the problem is mitigated. Generally, a chiller will use one of two control strategies – flow based or load based.

In a flow-based control strategy, the entire goal of the system would be to keep the primary loop flow rate greater than the rate in the secondary loop. This would ensure optimal loading of chillers and prevent excessive use of energy. In case of low ΔT, the temperature difference will drop from an ideal value of say 10 °F to 7 °F. The cooling coils will ask for more chilled water since they are not able to meet the cooling loads. This would cause an increase in the secondary loop and the flow rate becomes greater than the primary loop. This leads to further degradation in supply chilled water temperature. To correct this an additional chiller and its pump is brought on-line to keep the flow rate up. Thus, the plant is forced to run a couple of chillers with both being partially loaded, resulting in unnecessary energy consumption rise.

In a load-based control strategy, a new chiller is not started until the chillers running are fully loaded. In case of low ΔT, the flow rate in the secondary loop increases. The chillers will remain lowly loaded. The supply water temperature degrades, and no extra chillers are started. There is no way of controlling the temperature setpoints for the zones.

3.4.3: Primary pump arrangements

There are 2 main types of arrangements that can be looked at for piping the primary pumps to the chillers as shown in Figure 33.

Dedicated pumping is having a separate pump for every chiller in the plant. This provides ease in managing controls of the chiller system. The pumps can be sized as per each chiller's requirement which means reduction in pumping energy as opposed to headered pumps. Headered pumps are sized as per the largest pressure drop that exists in any of the chillers. This layout is cheaper than headered pumps due to lack of extra piping, valves and simple controls.

Headered pumps are preferred for primary only variable flow systems since the headered design can slowly ramp up the flow rate through the chiller being brought on. This ensures that there is no sudden change in the flow rate of other operating chillers and would prevent them from tripping. Headered pump layout also allows for easy addition of pumps for redundancy. The facility can have more or lesser number of pumps than there are chillers which also means two pumps could be used to serve a single chiller. The most important application of headered pumps would be in the case of low ΔT syndrome. This arrangement would help the chiller to be over pumped and be loaded fully before bringing on the next chiller.

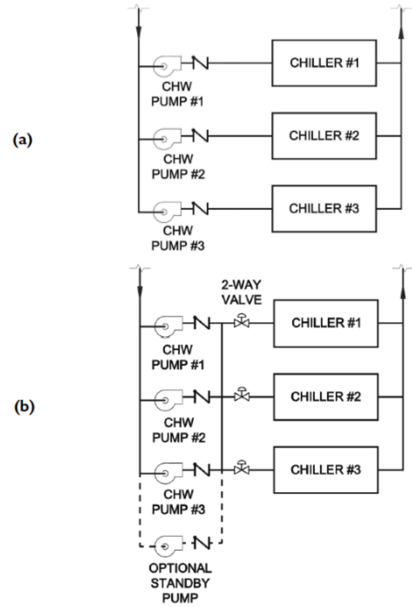


Figure 33: (a) Dedicated pumping
(b) Headered pumping [25]

3.4.4: Primary-only, variable flow chilled water system

The system uses just one set of VFD pumps instead of primary and secondary pumps as seen above. This results in savings in terms of pumping energy and lower land space is required in the facility. The pumps being driven by VFDs, will vary the amount of water through the chiller based on the cooling demand of the facility. The designers will especially need to look at the chiller's minimum required flow rate. If the flow through the chiller becomes too low the chilled water may leave at temperatures near freezing, thus shutting the chiller down.

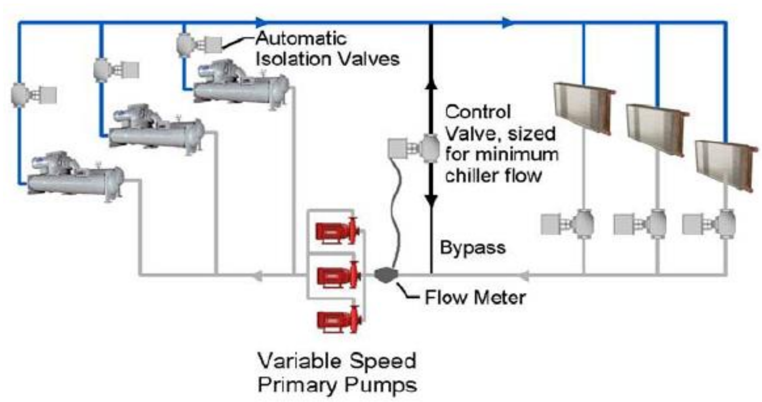


Figure 34: Primary- only, variable flow chilled water system

The main advantage of this system over the primary-secondary system is the lower first cost required. With the secondary pumps eliminated, the system would save the initial costs associated with the secondary pumping setup (secondary pumps, valves and fittings associated with the pumps and controls setup). Space in the facility is conserved since the primary pumps are the only ones taking up square footage instead of the usual two sets of pumps. With lower valves and fittings in the system, the overall head requirement reduces when compared to primary secondary systems since pumps. The overall savings obtained would be greater since the constant speed pumps are eliminated and the flow rate through the entire system can be varied.

This system is however difficult to implement from the controls standpoint. The chiller staging needs to be carefully looked at. When a chiller must be switched on, the flow rate through all chillers must gradually change. A sudden drop in the flow rate through a chiller could cause the chiller to trip. To avoid this issue, the operating chiller needs to be unloaded briefly as the other chiller establishes flow through its evaporator [29].

There are other options for chilled water distribution systems like Distributed pumping system or Tertiary pumping system which are generally used for huge plants or campuses and will not be a practical application for the facility being discussed for this thesis.

CHAPTER 4: ENERGY MODEL AND RECOMMENDATIONS

To understand how the energy consumption of the facility will be affected after implementing the energy conservation measures, an energy model was created in Carrier HAP. The model will be referred to as the base model and energy conservation measures will be applied to the model to study the effect on the overall energy consumption of the plant. The monthly demand and energy consumption numbers were provided by the facility. The model tries to represent the numbers as closely as possible. Due to lack of mechanical drawings of the plant, the model is built on a lot of assumptions. Several models were created in an attempt to match the numbers with the actual consumption of plant. The current model achieves a reasonable level of accuracy.

4.1: Weather data

The weather information used to model the facility in HAP can have a significant amount of effect on the energy values. Carrier HAP uses weather data compiled by the National Renewable Energy Laboratory. Figure 35 shows the weather data of the city where the facility is located, and it was used for simulating the energy model made in HAP.

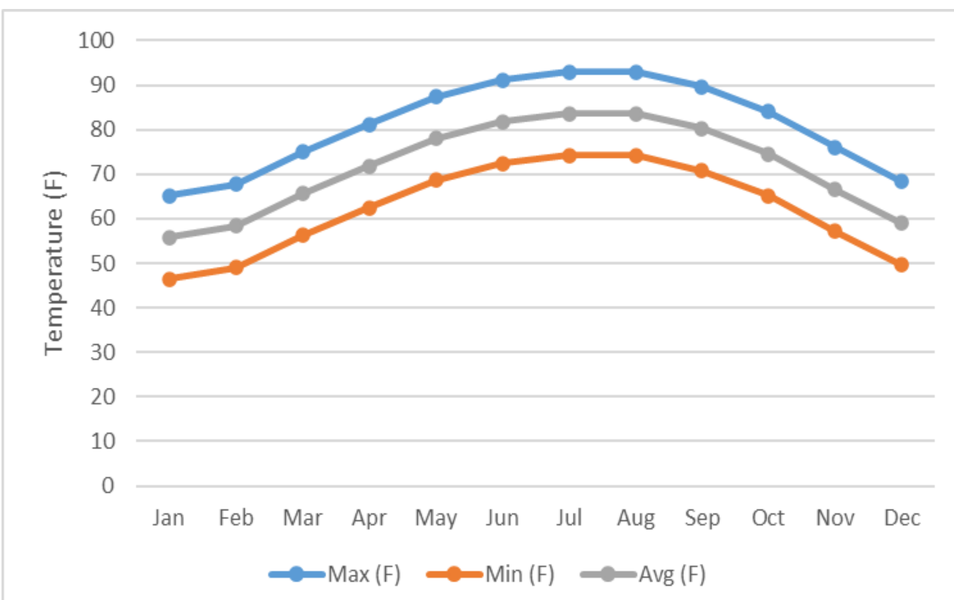


Figure 35: Weather data used in HAP

4.2: Spaces

The switch from manufacturing to being a warehouse meant that the spaces were re-purposed to meet the new needs. Considering Figure 1, the spaces were entered into HAP with their areas, internal heat gains, schedules, and outside air requirements. The following table shows a summary of the spaces that were created.

Table 4: Classification of spaces

Space	Area (ft^2)
Offices & Reception	86,000
Misc. Labs	20,000
Warehouse	323,000
Packaging & Shipping	73,000
Maintenance	18,000
Mechanical Room	7,500

These spaces were modeled into HAP while making certain assumptions which helped in creating loads which resembled closely with the energy bills. The facility was split into 13 spaces as shown in Figure 36.

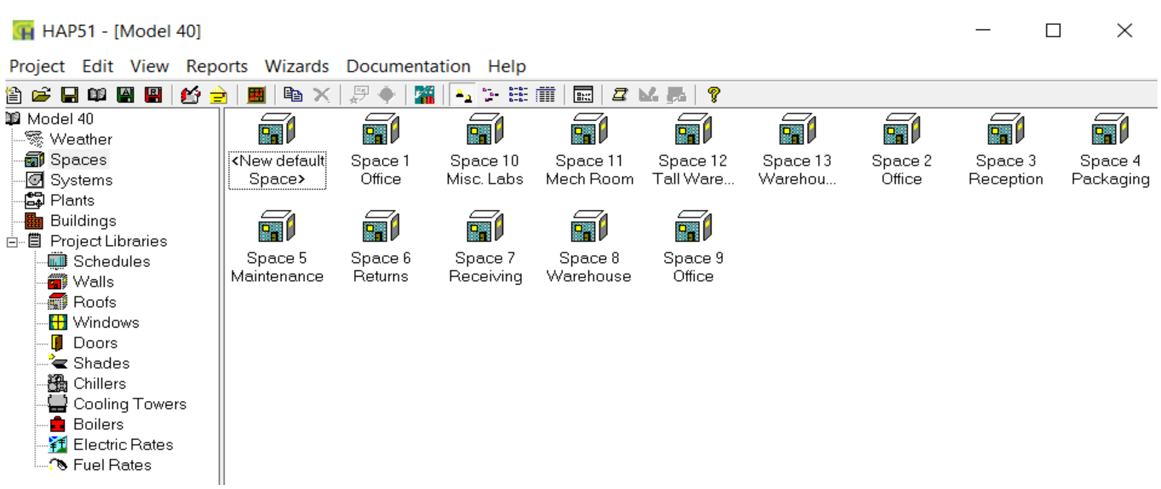


Figure 36: Spaces in the facility

The warehouse, packaging, returns and receiving areas were manufacturing spaces before the plant shifted operations. The outside air requirement for these spaces has been assumed to be the same as when the plant was into manufacturing. For the ventilation rates, HAP uses ASHRAE standards 62.1 and 90.1 – 2013. The lighting density for the facility was taken to be 1 W/ft^2, taking reference from the ASHRAE handbook. For the equipment load it was assumed that the density is 0.25 W/ft^2.

Schedules were made for equipment, lighting and people. The facility operates 6 days per week and 18 hours a day. This brings the operational hours to 5,400 hours in one year. The offices, however, are on the more traditional schedule of 8 hours per day. Figure 37 and Figure 38 show the hourly profile and its assignment used for the packaging space. It represents the schedule followed by people as they come and go according to the working times of the facility. Similar schedules have been made for lighting and equipment for different spaces. For all the wall and roof constructions, HAP's default values were used. The overall U value for the wall came out to be 0.225 Btu/(hr-ft^2-°F) and the roof's U value is 0.121 Btu/(hr-ft^2-°F).

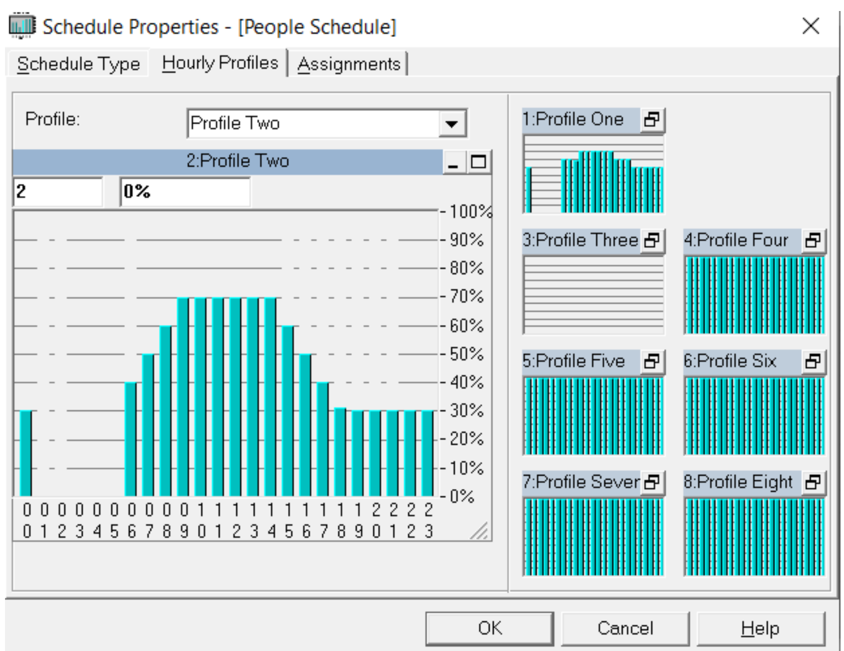


Figure 37: Sample hourly profile

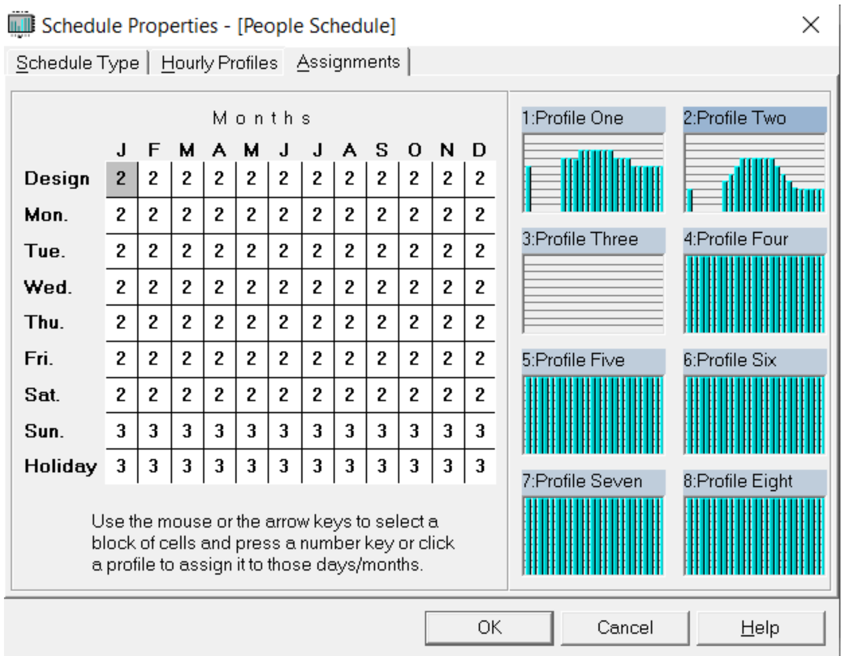


Figure 38: Schedule assignment for people

4.3: Systems

Carrier HAP provides a few different systems that can be used in terms of the equipment type and the air systems. As discussed in Chapter 1, a chiller system is in place to meet the demands of the plant along with some RTUs. The office, reception and maintenance areas have RTUs for meeting the air conditioning demand.

Based on the results obtained after running a few simulations, all air systems were chosen to be Constant Air Volume systems with terminal reheat. The exact type of system is not known because the facility could not provide mechanical drawings of the HVAC system. The results of simulations with VAV will be discussed later in this chapter. Some areas of the facility which have been assumed to be served by the chiller may have been served by an RTU but without the drawings or confirmation from the facility there is no way of knowing. Upon visual inspection of the roof of the facility using Google Earth, it has been assumed that the office spaces, reception area and the maintenance room are served by multiple RTUs ranging from 5-25 tons whereas a fraction of the warehouse area is served by four 75 ton RTU units.

Figure 39: Google Earth image of the facility

The CAV system used in the model supplies a constant volume of air with a central fan to multiple zones in the facility. The supply air temperature is maintained as per the user's needs. When a zone calls for cooling, it receives a constant volume of air which is at the supply temperature and the terminal reheat coil remains off. When the zone calls for heating, the reheat coil switches on and heats the supply temperature air according to the zone's thermostat.

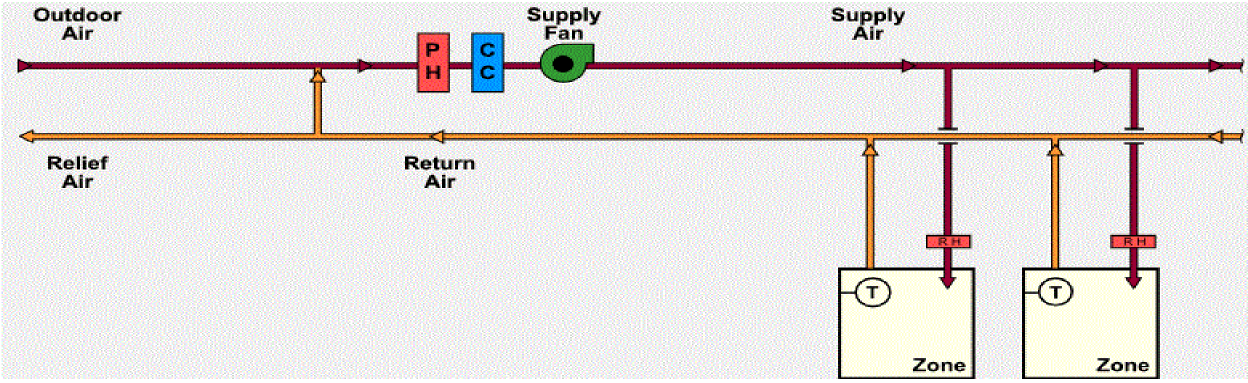


Figure 40: CAV with terminal reheat

Depending on the type of space the thermostat setpoints were set. For occupied conditions, the cooling setpoints lie in between 72-75 °F whereas for the unoccupied condition, the setpoints range from 78-80 °F. For heating, the occupied condition setpoint was 70 °F and 62-65 °F for unoccupied.

4.4: Simulations

To create the base model which resembles the current energy consumption of the facility, multiple simulations with different combinations of air systems, thermostat set points, outside air ventilation rates, schedules and internal load were tested. The approach was to match the demand values first and then look at energy values.

Initially, it was assumed that the chiller would have a VAV system and the RTUs will be served by CAV systems with hot water terminal reheat. When the model was simulated, the monthly energy values were significantly lower as compared to the actual consumption as seen in the figures below. Using a VAV system and keeping other parameters unchanged, consistently provided lower energy values while the demand values fairly matched up.

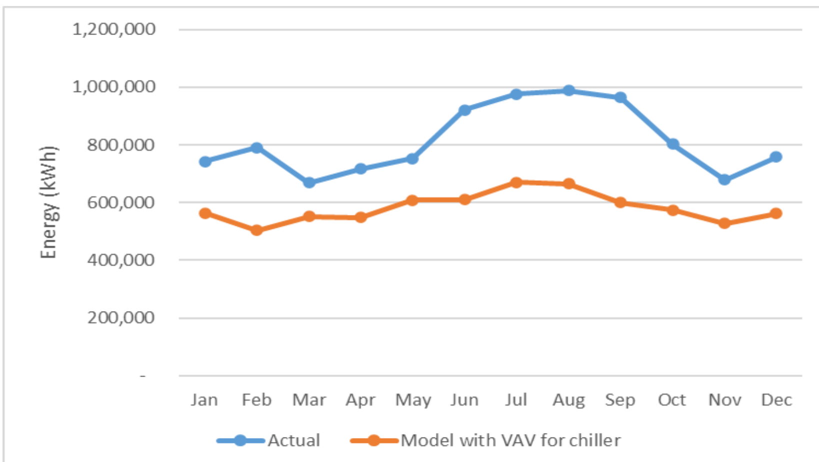


Figure 41: Energy consumption comparison for a VAV

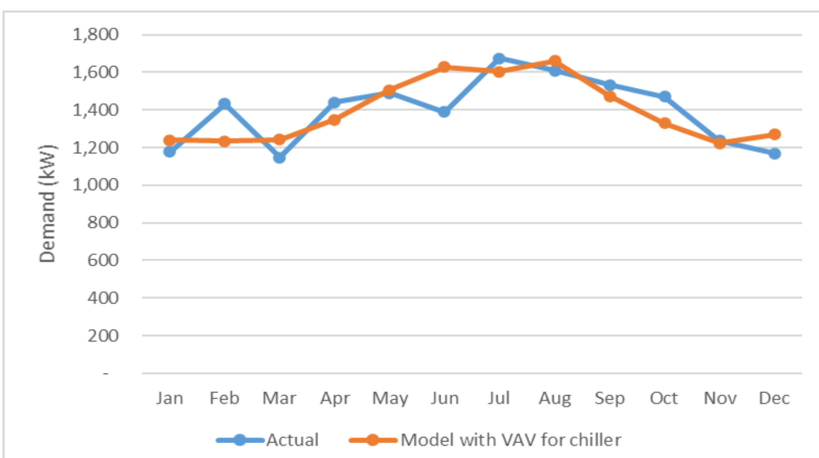


Figure 42: Comparing demand values for a VAV model

In order to drive the energy numbers up while maintaining monthly demand, a plant wide CAV system was implemented. This resulted in a significant rise in energy consumption while keeping demand within acceptable margins. The reason CAV produces favorable results is because it runs the fans constantly when the unit is in occupied mode. The demand is set when the fans are at full load. This explains the increase in energy consumption while demand remains fairly unchanged. In case of the VAV system, the fans are throttled back when the unit is not at full load, reducing the energy use significantly. The following three figures and tables compare the monthly energy and natural gas consumption values with the actual numbers obtained from the bills that were provided.

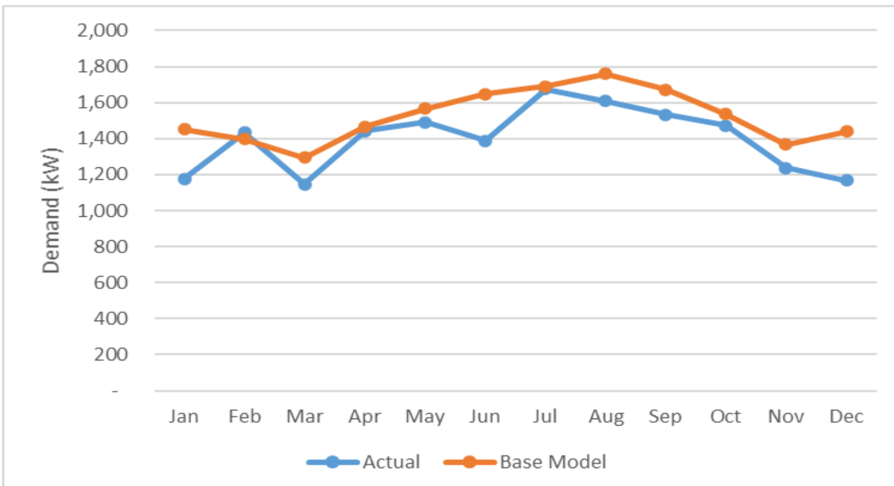


Figure 43: Demand values of facility vs Base model

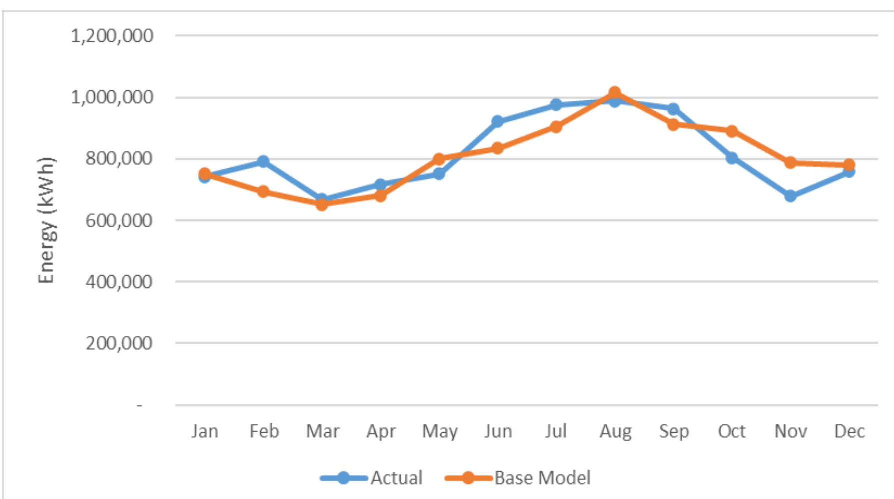


Figure 44: Energy values of facility vs Base model

Table 5: Actual energy and demand values compared with the base model

Month	Actual demand (kW)	Base model demand (kW)	% Difference
Jan	1,176	1,452	(23.43)
Feb	1,432	1,397	2.42
Mar	1,146	1,294	(12.89)
Apr	1,441	1,466	(1.71)
May	1,490	1,566	(5.11)
Jun	1,389	1,647	(18.57)
Jul	1,675	1,689	(0.84)
Aug	1,609	1,759	(9.35)
Sep	1,534	1,671	(8.94)
Oct	1,471	1,537	(4.49)
Nov	1,237	1,366	(10.40)
Dec	1,168	1,438	(23.10)

Month	Actual energy consumed (kWh)	Base model energy (kWh)	% Difference
Jan	742,400	750,490	(1.09)
Feb	790,400	692,853	12.34
Mar	668,800	652,314	2.47
Apr	716,800	681,494	4.93
May	752,000	799,512	(6.32)
Jun	921,600	834,377	9.46
Jul	976,000	904,597	7.32
Aug	988,800	1,015,629	(2.71)
Sep	963,200	913,345	5.18
Oct	803,200	890,053	(10.81)
Nov	678,400	786,772	(15.97)
Dec	758,400	780,044	(2.85)

Table 6: Actual gas consumption values compared with the base model

Month	Actual gas consumption (MMBtu)	Base model gas consumption (MMBtu)	Difference (MMBtu)
Jan	100	479	(379)
Feb	100	473	(373)
Mar	1	22	(21)
Apr	0	12	(12)
May	2	0	2
Jun	2	0	2
Jul	62	0	62
Aug	813	725	10.82
Sep	1,072	889	17.07
Oct	1,656	1,259	23.97
Nov	1,600	1,571	1.81
Dec	1,111	2,025	(82.27)

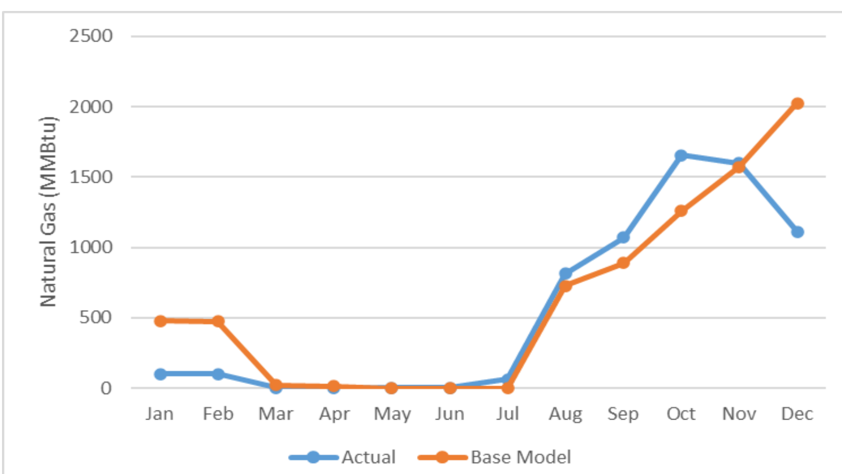


Figure 45: Natural gas consumption of facility vs Base model

Table 7 and Table 8 show a summary of the energy and natural gas consumption and costs associated for the base model simulation. To arrive at the desired results, one key assumption that had to made was to set some terminal reheating sources associated with RTUs to electric resistance. Whereas, all chiller CAV systems have the reheating source as hot water.

When simulations were run with hot water as the reheating source for the entire plant, the energy consumption was significantly lower than what is seen in Figure 44 whereas the natural gas consumption was a lot higher. This is opposite of what we see in Figure 45, where the actual natural gas consumption is low, which is what prompted the reheat source to be chosen as electric.

The motivation for the designers for using CAV for the entire facility could be because of a couple of reasons. First, VAV systems did not become mainstream up until the 1980s. The designers might not have had the option of using VAV systems. This plant was built in 1960s and VAV had started to become widely adopted because of the 1970s energy crisis [30].

Terminal electric reheating was prevalent around that time and the industry started to move towards hot water heating much later [31]. Another reason could be that even though hot water coils provide energy savings over a period of time, the initial cost of installation of electric heaters is much lesser than installing hot water piping for the entire facility [32].

As seen in previous figures, the numbers differ from the actual consumption values to a certain degree. There can be different causes of inaccuracy in the energy model's results.

1. The weather data used by HAP represents the temperature profile of the weather station nearest to the location of the facility. The actual location could have a different profile as compared to the HAP weather data. Although the variation is expected to not be drastic, the inaccuracies could add up to produce variations.
2. Without accurate information on the internal loads, lights and people the variation can be significant. The error in results because of scheduling differences could add up as well.
3. The base model uses a CAV system for the entire facility which may not be the case. Without the availability of MEP drawings, the model is based off several engineering guesses. Considering the amount of different mechanical systems that HAP provides, it would be difficult to reach a configuration that describes the current HVAC system of the facility.

Table 7: Summary of energy and demand costs for base model

Billing Period	Energy (kWh)	Energy charge	Demand (kW)	Demand Charge	Total Charges
Jan	750,490	$37,750	1,451	$18,811	$56,561
Feb	692,853	$34,851	1,397	$18,109	$52,960
Mar	652,314	$32,811	1,293	$16,766	$49,578
Apr	681,494	$34,279	1,465	$18,995	$53,275
May	799,512	$40,215	1,566	$20,298	$60,513
Jun	834,377	$41,969	1,647	$21,345	$63,315
Jul	904,597	$45,501	1,689	$21,889	$67,390
Aug	1,015,629	$51,086	1,759	$22,802	$73,888
Sep	913,345	$45,941	1,671	$21,657	$67,598
Oct	890,053	$44,770	1,537	$19,920	$64,689
Nov	786,772	$39,575	1,365	$17,699	$57,275
Dec	780,044	$39,236	1,437	$18,634	$57,870
Totals	**9,701,480**	**$487,984**	**18,281**	**$236,927**	**$724,912**

Table 8: Base model natural gas consumption and costs

Month	Fuel (MMBtu)	Gas Cost
Jan	479	$3,469
Feb	473	$3,427
Mar	22	$161
Apr	12	$90
May	0	-
Jun	0	-
Jul	0	-
Aug	725	$5,254
Sep	889	$6,442
Oct	1,259	$9,129
Nov	1,571	$11,391
Dec	2,025	$14,679
Total	**7,454**	**$47,281**

Table 9 shows a summary of results obtained after simulating the base model. The cost of running the system fans is almost as high as cost of cooling. This could be attributed to the entire system being CAV which utilizes constant air flow regardless of the cooling/heating requirement. The cost of heating is lower than the cooling costs but is still relatively higher for the location which is considered to be a cooling dominant place. This could be explained by the fact that the terminal heating is mostly electric and although it is an efficient method of transferring heat, it is expensive nonetheless.

Table 9: Annual component costs of the base model

Component	**Annual Cost**	**Percent of Total (%)**
Air System Fans	$167,521	21.5
Cooling	$159,559	20.5
Heating	$125,502	16.1
Pumps	$62,921	8.1
Heat Rejection Fans	$17,673	2.3
HVAC Sub-Total	**$533,177**	**68.4**
Lights	$192,149	24.7
Electric Equipment	$53,635	6.9
Non-HVAC Sub-Total	**$245,784**	**31.6**
Grand Total	**$778,960**	**100**

4.5: Recommendations

This section will involve descriptions of different recommendations identified at the subject facility. Each recommendation will go through a brief explanation of the problem, the suggested action followed by cost and energy savings. At the end of each recommendation, simple payback period will be calculated.

4.6: Chilled water pump upgrade

The facility uses centrifugal pumps to distribute chilled water at a constant flow rate regardless of the cooling demand of the zones. The 750-ton chiller has been set for a cooling load when manufacturing was still predominant in the facility. Since then there has been no change made to the pumping system to ensure that it meets the actual cooling demand. This results in excess chilled water being pumped throughout the facility when the actual cooling requirement is low. The amount of energy wasted in moving large quantities of chilled water is high.

Consider a zone in the facility, which has been repurposed to be a storage area. The CAV unit responsible for maintaining temperature receives excess amount of chilled water since the current pumping system ensures that the CAV unit receives chilled water at a constant flow rate. Since the HVAC system is CAV, it constantly blows a fixed volume of air over the cooling coils thus achieving more cooling than necessary. To maintain the setpoint temperature, the unit compensates by heating the air up to setpoint via its electrical terminal reheat. Through overcooling and reheating the system experiences huge energy losses and a rise in energy bills.

The diagram below shows the current chiller and pump system at the facility. Some pressure readings shown (gauges B, C, D) were taken when the team visited the plant. The 750-ton chiller was active whereas the other 500-ton units were turned off. At a time when manufacturing was the norm, the plant must have operated the 750 and 500-ton units with the other 500-ton chiller on standby. Each chiller has its own 15 hp booster pump and the chilled water is supplied to the facility with the help of a 75 hp and two 50 hp pumps.

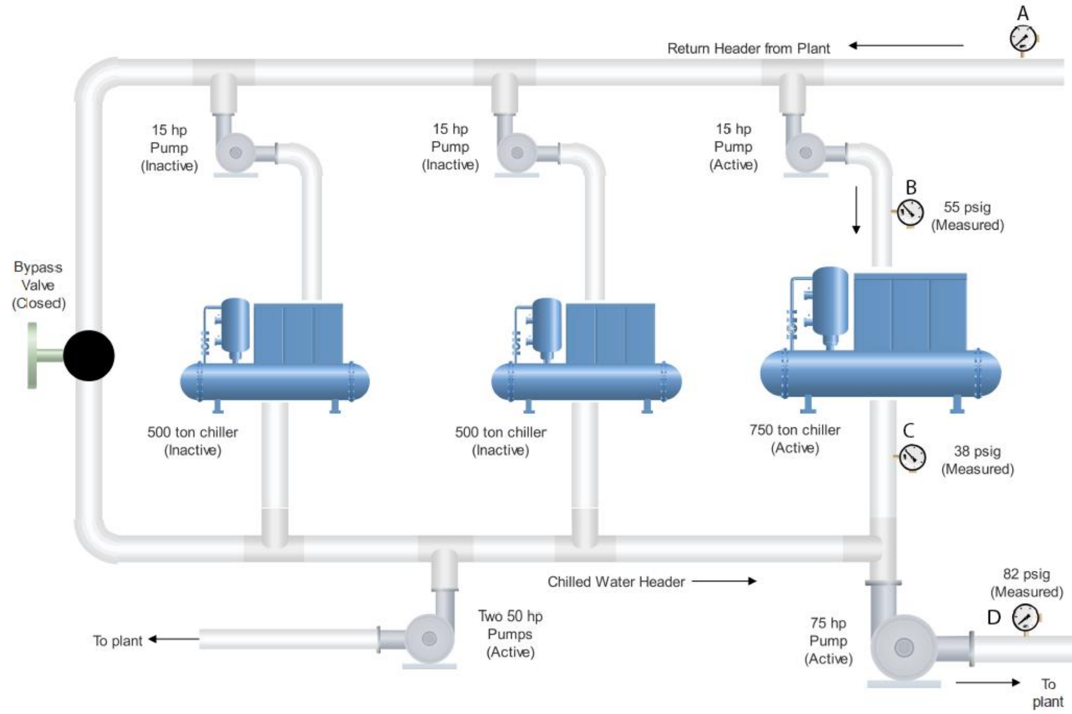


Figure 46: Pumping layout of the plant

To calculate exact savings of this recommendation, more information about the pumps would have been needed. With the help of mechanical and plumbing drawings along with pump curves and additional pressure measurements, the recommendation could have shown the complete picture in terms of cost savings. Pressure and temperature readings could not be taken at multiple locations due to asbestos issues. Therefore, multiple assumptions have been made which will be discussed when needed.

4.6.1: Calculating flow rates

The motivation behind sizing a pump, even though a pumping setup exists, is to gain an understanding of the demand of the facility and figure out if the pumps are oversized or undersized.

The pump which is selected needs to deliver the desired gallons per minute of water through the entire facility to ensure cooling needs are met. This can be calculated as follows –

$$Q = \dot{m} * C_p * \Delta T$$

$$Q(tons)(\frac{12{,}000Btu}{hr}) = \dot{V}(\frac{gal}{\min})(\frac{1ft^3}{7.48gal})(\frac{60\min}{1hr}) * \rho(\frac{62.4lbs}{ft^3}) * c_p(\frac{1Btu}{lbs * F}) * \Delta T(^0F)$$

$$\frac{23.97}{\Delta T(^oF)} = \frac{\dot{V}(GPM)}{Q(tons)}$$

A thumb rule for a chiller is that it is usually designed for a ΔT of 10 °F with a 10 psig drop [33]. A 10 °F ΔT will result in the above equation simplifying to give a value of 2.4 GPM/ton. This can be used as a thumb rule to find the approximate flow rate for any chiller operating at maximum efficiency.

Therefore, the ideal flow rate through the 750 ton chiller will be equal to 1,800 GPM. This flow rate is for when the chiller is functioning at peak efficiency. The current pressure drop across the chiller can be calculated since we have pressure measurements at points B and C as shown and it is equal to 17 psig. If a 10 psig drop corresponds to 1,800 GPM through a 750 ton chiller, then a 17 psig drop will correspond to about 2,350 GPM of water flow.

The flow through the chiller is much higher than what it was designed for. A possible explanation for the high GPM setpoint can be made as follows. At some point during the facility's operation the plant technicians must have turned off the 500 ton chiller. This could have caused the water, meant for the 500 ton chiller, to flow through the 750 ton unit. The reason behind this assumption could be explained by the flow rates. If both chillers were operating at one time, the total tonnage would have been 1,250 tons. With the 2.4 GPM/ton thumb rule, the total flow would have amounted to about 3,000 GPM which is reasonably close to the calculated 2,350 GPM water flow through the chiller.

4.6.2: Calculating head losses

As discussed in Chapter 3, in a closed loop system, a pump would need to overcome just the friction losses. Ideally, one would use the friction loss equations to calculate the losses, but it is not possible without plumbing drawings. From the pump layout it is known that the pressure at point B is 55 psig. The flow rate through the pipe as calculated is 2,350 GPM. Therefore, we can calculate the pressure drop knowing the horsepower of the pump.

$$P = \frac{\rho g H Q}{\eta}$$

Where, P = horsepower of the pump (hp) – Multiply by 550 to convert to ft-lb/sec

ρg = 62.4 lb/ft^2-sec^2 (for water)

H = head drop (ft of water)

Q = Flow rate (ft^3/sec) – Divide by 449 to convert to from GPM

η = Efficiency assumed to be 75%

$$15 * 550 = \frac{62.4 * H * \frac{2350}{449}}{0.75}$$

$$H = 19 \text{ ft of water}$$

19 ft of head in pressure terms would be 8.2 psig. Pressure reading at B was 55 psig. Thus, pressure at A would be 46.8 psig. This is helpful for calculating the head loss due to friction. The pressure differential between points D and A is 35.2 psig which converts to a head drop of about 85 feet.

4.6.3: Energy and cost savings

With the current pumping layout, some flow occurs through the 50 hp pumps and some through the 75 hp pump. The pressure at the return header is indicative of friction losses in a loop which could be served by any of the three pumps. So, if it is proven that the 75 hp pump can supply the entirety of the 2,350 GPM flow with a head of about 85 ft then it can be said that the facility can safely turn off all the other pumps. The piping layout can be modified to ensure that the zones being served by the 50 hp pumps are served by the 75 hp pump instead.

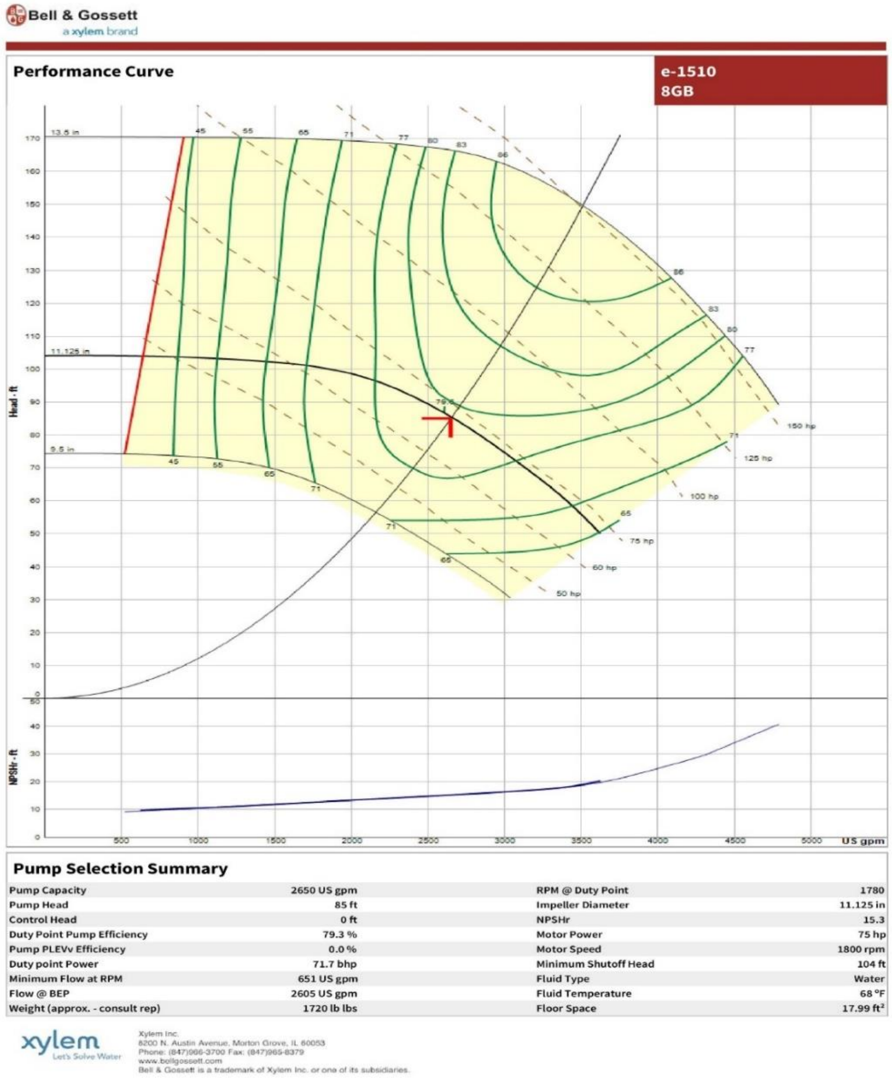
$$P = \frac{\rho g H Q}{\eta}$$

$$75 * 550 = \frac{62.4 * 85 * \frac{Q}{449}}{0.75}$$

$$Q = 2{,}610 \text{ GPM}$$

Based on the calculation, the 75 hp might be able to serve the entire facility's chilled water needs. This assumption is made considering that the current flow through the chiller is enough to serve the needs of the facility. A different way of sizing a pump would be to simply use the head and flow values in a pump sizing tool. The screenshot on the next page has been taken from the 'Bell & Gossett ESP-PLUS' pump sizer. The pump has been sized for a head of 85 feet and 2,610 GPM. The efficiency is about 80% which is close to the assumption made during the manual calculation above. The NPSH-r is 15.3 ft (~7 psig) for the Bell & Gossett pump. The suction pressure measured at the inlet of the pump at the facility was 38 psig which is enough to meet the required NPSH. This pump is also capable of supplying 3,000 GPM of water.

The Bell & Gossett pump shown is from the e-1510 family which is the end suction type of pump. This type of pump is generally used for HVAC applications because of a lower initial cost. If the facility is using an in-line or a double suction pump then there are options available, in the range of 75-90 hp for the flow and head required by the plant. This would mean that the 75 hp pump, no matter the type, is sufficiently sized for supplying chilled water to the entire facility. Hence, all the other active pumps can be turned off which will not only save energy but reduce demand as well.



Pump Selection Summary

Pump Capacity	2650 US gpm	RPM @ Duty Point	1780
Pump Head	85 ft	Impeller Diameter	11.125 in
Control Head	0 ft	NPSHr	15.3
Duty Point Pump Efficiency	79.3 %	Motor Power	75 hp
Pump PLEVv Efficiency	0.0 %	Motor Speed	1800 rpm
Duty point Power	71.7 bhp	Minimum Shutoff Head	104 ft
Minimum Flow at RPM	651 US gpm	Fluid Type	Water
Flow @ BEP	2605 US gpm	Fluid Temperature	68 °F
Weight (approx. - consult rep)	1720 lb lbs	Floor Space	17.99 ft²

Figure 47: Pump sized using Bell & Gossett ESP-PLUS

Demand Reduction	=	(15 hp + 2 x 50 hp) x 0.746 kW/hp
	=	85.79 kW
Energy Savings	=	85.79 kW x 5,400 hrs/yr
	=	463,266 kWh/yr
Cost Savings	=	85.79 kW x 12 months x $12.96/kW
		+ 463,266 kWh x $0.0503/kWh
	=	$36,644/yr

4.6.4: Implementation cost

New piping additions will have to be made for the 75 hp pump to supply water through the loops currently being served by the 50 hp pumps. Asbestos was present on some pipes which will have to be removed to go ahead with the installation of new piping. The abatement cost is difficult to predict and has not been considered.

According to the Carrier - System Design Manual for piping, the recommended velocity range for the header of the chilled water system is 4-15 ft/sec but to keep erosion due to water flow to a minimum, the velocity of water should not exceed 8 ft/sec if the plant is operating for 6,000 hours/yr. Thus, it is assumed that the velocity of water is 6 ft/sec. With this velocity and a flow rate of 2,610 GPM, the pipe diameter required is about 12 inches. Therefore, a 60 ft 12-inch steel header is recommended along with three gate valves and three Tee joints. The valves and joints help to ensure that the plant can choose to operate with all or any of the three pumps. The table below shows the implementation costs of the recommended solution. The total cost comes out to be $65,575 and with $36,644 as cost savings, the simple payback period will be 2 years.

Table 10: Implementation costs for piping

Item	Unit Price (materials and labor)[1]	Cost
60 feet of 12" Steel Pipe	$230 / foot	**$13,800**
Three 12" Gate Valves	$7,450 each	**$22,350**
Three 12" Tee Joints	$2,475 each	**$7,425**
Engineering	50% of material and labor costs	**$22,000**
Total Cost of Project (excluding asbestos abatement)		**$65,575**

Simple Payback Period = Implementation cost / Cost savings

= $65,575 / $36,644/yr

= 2 years

[1] Costs from Means' Mechanical Cost Data, 2019 for Schedule 40 steel pipe and fitting

4.7: Re-design the chilled water distribution system

As discussed in Chapter 1, the facility was built in the 1960s and then underwent expansion in the 1980s. With the expansion, two 500-ton chillers were added with one of them kept on standby. The facility added more constant flow pumps to the layout. The CAV boxes must have utilized 3-way valves to control the flow of water through them. On basis of first costs, the plant must have opted for the primary only, constant flow pumping system.

The team noted that the mechanical room had a bypass line as shown in the pumping layout Figure 46. The facility at some point must have considered to shift to a primary secondary loop for chilled water distribution as it was a commonly implemented system during those days [34]. Traditionally, when a primary secondary system is implemented, the primary loop has constant flow while the secondary loop has variable frequency drive pumps. Initial cost could be one of the reasons as to why the facility did not go ahead with installing VFDs for secondary pumps.

As the manufacturing in the facility reduced, so did the cooling demand and the plant could get by with running just the 750-ton chiller and the RTU units. The 500-ton chiller was turned off but all pumps were kept running. The water flow rate was not adjusted per the cooling load of the plant. Today, the plant still uses a constant flow primary loop with a decoupler whose bypass valve is closed. The closing of the valve can be explained by considering that the plant is suffering from low ΔT syndrome, and closing the valve ensures all of the chilled water flows to the plant and none of it is diverted through the decoupler.

This recommendation will explore the primary secondary and variable primary pumping systems along with installation and effect of using a Variable Air Volume system instead of the CAV system which currently exists in the plant.

4.8: Simulation setup

Carrier HAP was used for simulating the different cases to obtain energy, demand and gas consumption values along with the associated costs to run the entire plant.

The following systems have simulated, analyzed and compared with the base model from Section 4.4:.

1. Convert the current system to a primary secondary pumping layout while retaining the RTUs.
2. Convert the current system to a primary secondary pumping layout, refurbish the 500-ton chiller and convert all the DX RTUs to be run by chilled water.
3. Convert the current system to a primary-only variable pumping layout while retaining the RTUs.
4. Convert the current chiller CAV system to a VAV system. Change pumping to a primary secondary pumping layout, leaving the RTUs untouched.

The figure below shows the areas of the plant being served by RTUs and the chiller.

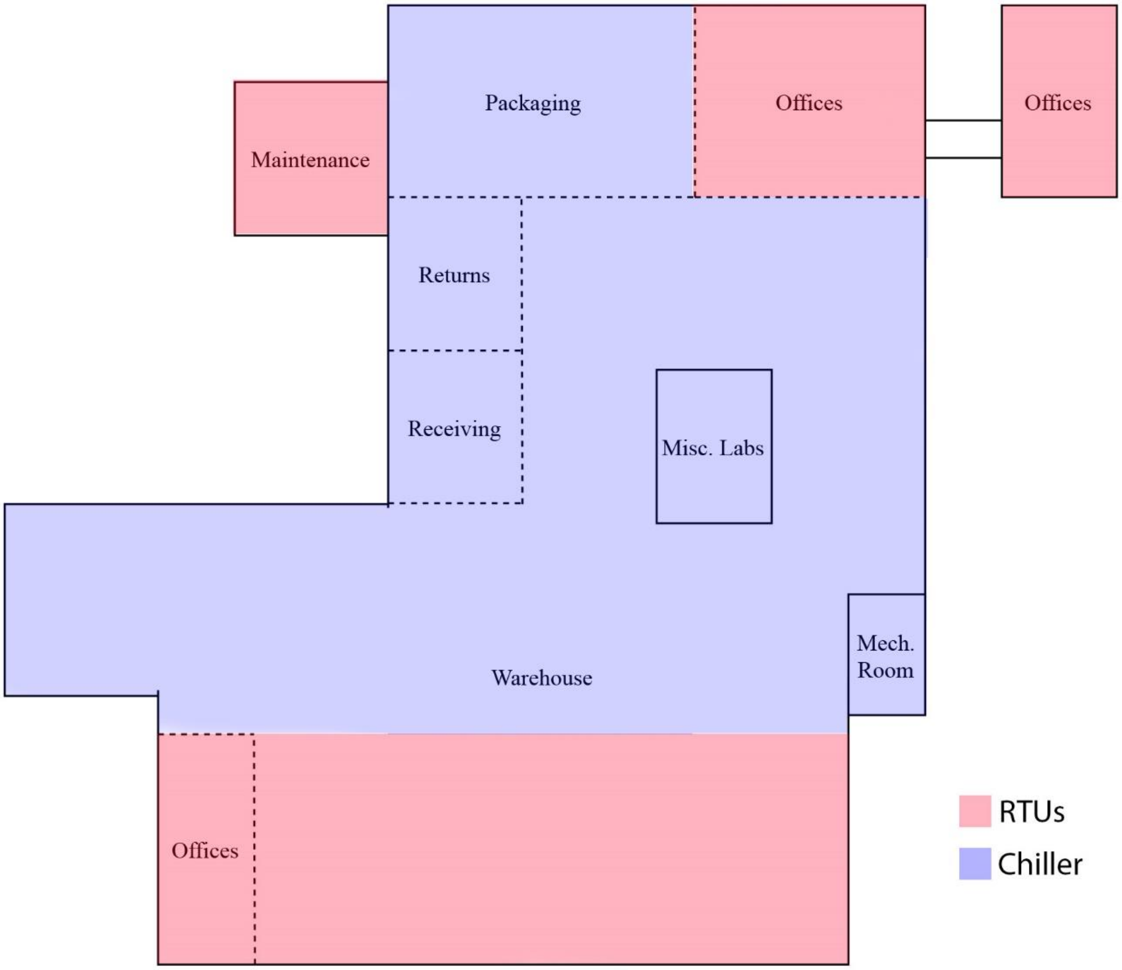


Figure 48: Areas being served by chillers and RTUs

4.9: Primary secondary loop

Converting the constant primary to a primary secondary system would need the addition of VFDs to the 75 hp pump that the plant has. It will be assumed that the motor is compatible for the addition of a VFD.

The current pumping layout has three 15 hp pumps. If we consider a standard pressure drop of 10 psig through the chiller and a flow rate of 2.4 GPM/ton then according to the 'Bell & Gossett ESP-PLUS' pump sizer, the recommended horsepower for the pump was shown to be 15 hp. Therefore, it could be said that the sole purpose of these pumps is to push water through the respective chillers.

In order to create a primary loop of water distribution, the approximate head to be overcome by a pump can be calculated as follows,

Total head requirement for the primary pump	=	Pressure drop in chiller + Pressure drop due to piping and valves in the primary loop
	=	10 psig + 10 psig
	=	45 ft of head

The flow rate for the pump will be considered to be equal to 2.4 GPM/ton, which for the combined setup of the 750-ton and the 500-ton chiller will come out to be 3,000 GPM. Sizing a pump for the head and flow calculated, the pump sizer recommends a 50 hp pump. The plant will have to invest in a new primary pump to ensure primary pumping occurs smoothly.

To save costs, the plant can instead, make use of the existing three 15 hp pumps. The chillers use dedicated pumping to meet their flow rate requirements. If the plant shifts to the headered arrangement for pumping, having three 15 hp pumps in parallel will meet the primary loop's head and flow requirements as shown in the figure on the next page.

Shifting to the headered arrangement allows the plant to use all three of its pumps and save money by not buying a newer pump. The facility can also effectively tackle the issue of low ΔT by over pumping the chiller and increasing loading before having to bring a different chiller online.

While setting up the energy model for the primary secondary system, the base model's water distribution tab was modified. As calculated in the previous recommendation, the secondary pumping head was taken to be 85 ft and the head for the primary pumps was taken to be 45 feet. The minimum flow for the variable speed secondary pumps was set at 20%. A control head setpoint of 30% has been assumed for all simulations. Control head for a pump is defined as the minimum amount of head that is required to be maintained in the loop to ensure that the pump is able to quickly establish flow through the most critical coil. ASHRAE recommends a control head setpoint to be in the range of 30%-40% of design head [35].

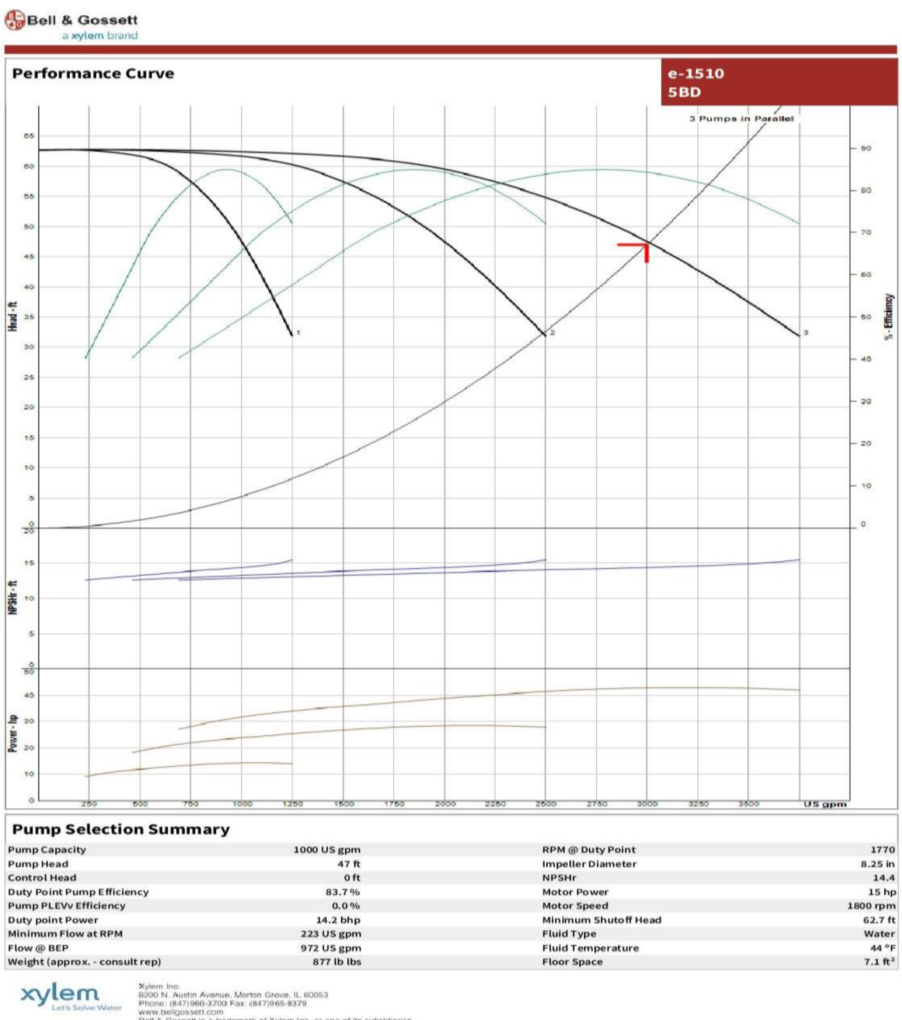


Figure 49: Three 15 hp pumps in parallel (Bell & Gossett ESP-PLUS)

4.9.1: Energy savings

The following figures and tables will illustrate the results obtained after running the simulation and the results have been compared with the base model energy, demand and natural gas consumption values.

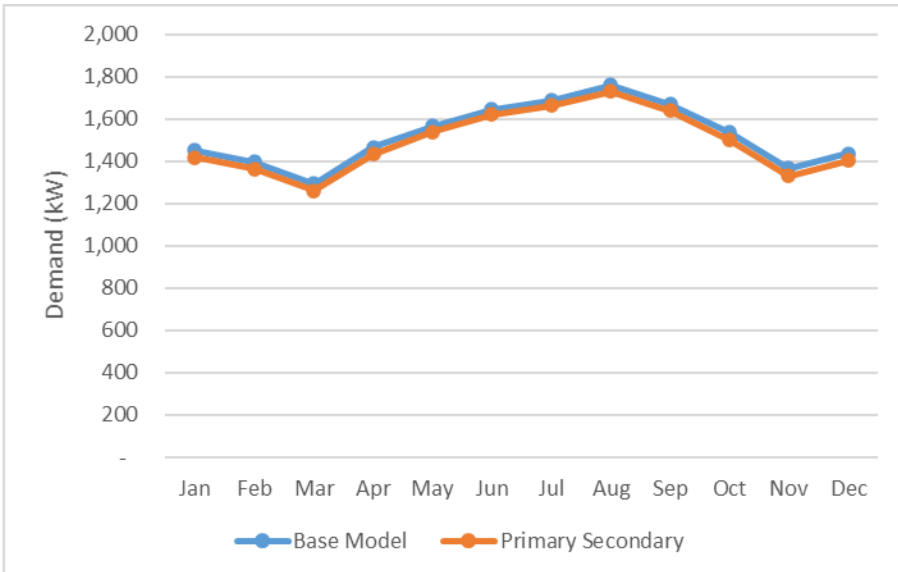


Figure 50: Demand values of base model vs primary secondary

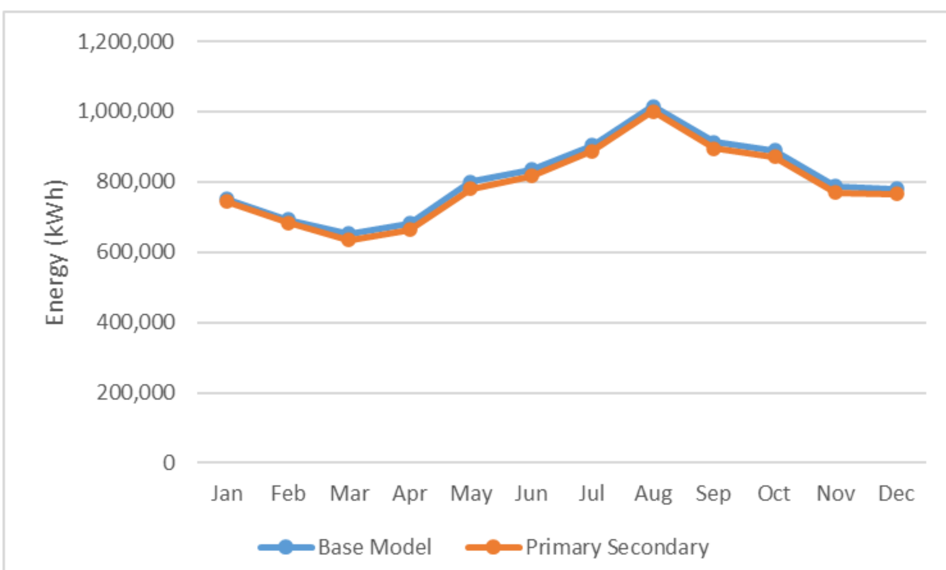


Figure 51: Energy values of base model vs primary secondary

Table 11: Energy and demand comparison of base model vs primary secondary

Month	Base model demand (kW)	Primary secondary demand (kW)	% Drop
Jan	1,452	1,417	2.36
Feb	1,397	1,364	2.41
Mar	1,294	1,260	2.60
Apr	1,466	1,434	2.14
May	1,566	1,538	1.78
Jun	1,647	1,622	1.50
Jul	1,689	1,665	1.40
Aug	1,759	1,730	1.67
Sep	1,671	1,639	1.94
Oct	1,537	1,501	2.33
Nov	1,366	1,330	2.64
Dec	1,438	1,402	2.47

Month	Base model energy (kWh)	Primary secondary energy (kWh)	% Drop
Jan	750,490	743,603	0.92
Feb	692,853	684,643	1.18
Mar	652,314	634,193	2.78
Apr	681,494	664,052	2.56
May	799,512	780,564	2.37
Jun	834,377	817,088	2.07
Jul	904,597	887,455	1.89
Aug	1,015,629	1,000,288	1.51
Sep	913,345	895,384	1.97
Oct	890,053	870,680	2.18
Nov	786,772	768,676	2.30
Dec	780,044	765,460	1.87

Table 12: Natural gas comparison of base model vs primary secondary

Month	Base model gas consumption (MMBtu)	Primary secondary gas consumption (MMBtu)	Reduction (MMBtu)
Jan	479	483	(4)
Feb	473	475	(2)
Mar	22	23	(1)
Apr	12	13	(1)
May	0	0	0
Jun	0	0	0
Jul	0	0	0
Aug	725	590	135
Sep	889	758	131
Oct	1,259	1,125	134
Nov	1,571	1,447	124
Dec	2,025	1,924	101

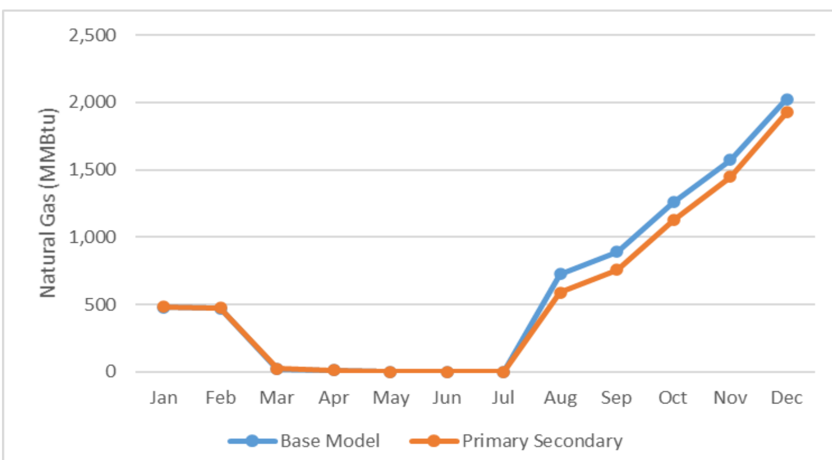


Figure 52: Natural gas consumption of base model vs primary

Table 13: Annual cost comparison of base model vs primary secondary

Component	Base model	Primary secondary	% Drop
Air System Fans	$167,521	$162,900	2.8
Cooling	$159,559	$156,089	2.2
Heating	$125,502	$121,136	3.5
Pumps	$62,921	$51,578	18.0
Heat Rejection Fans	$17,673	$17,415	1.5
HVAC Sub-Total	**$533,177**	**$509,117**	**4.5**
Lights	$192,149	$192,667	(0.3)
Electric Equipment	$53,635	$53,775	(0.3)
Non-HVAC Sub-Total	**$245,784**	**$246,442**	**(0.3)**
Grand Total	**$778,960**	**$755,559**	**3.0**

The main reason the plant should consider shifting to a primary secondary system is to obtain pump energy savings. From the table above it is clear that the cost of running the pumps has gone down 18% of what it was in the base model. With the possibility of varying flow through the secondary loop, the plant can save energy pumping only what is needed through the entire plant. The VFDs can ramp the speed up and down depending on demand of chilled water in the facility.

The plant should also see a marginal decrease in the cost of running air system fans. Since Carrier HAP sizes the system fans for the users, a lower energy usage denotes that the primary secondary system can make do with a slightly smaller fan or run it at a lower speed. The fan for the chiller AHUs was rated at 158 kW as opposed to the 150 kW fan for the primary secondary loop.

The sizing of fans also affects the heating energy as it is all dependent on how the zones are set up in the facility. If one single AHU serves many zones, it may have to run the fan depending on the cooling requirement of the most critical zone. This could create a scenario where the AHU runs its fans for just one zone while other zones are forced to constantly reheat the air according to setpoint of the spaces. In such a case there will be no savings obtained.

The second area of savings is the cooling costs. The change in cooling costs is marginal and the change can be explained as follows. The default chiller setup in Carrier HAP for simulations is such that the chiller sees its maximum efficiency at 75% loading. When the simulation was carried out, the loading of the chiller decreased from 79% for the base model to 77% in the primary secondary model. This loading can change based on how the zoning is done and what the scheduling of people, equipment and lighting of the plant is. Therefore, as the loading reduced and approached loading of 75%, which results in the highest efficiency, the energy use decreased and that resulted in the cooling cost savings as seen in the table.

The third area of savings is in the heating costs. With a varying chilled water flow through the cooling coils, the amount of reheating required will go down. This again depends on the critical zone since this is a CAV system. If the critical zone has a cooling demand far greater than the other zones, there will always be a need for greater reheating in other zones.

The plant should consider looking at supply air temperature resets to save energy. Identifying the critical zone and using aggressive resets will help the plant save significant amount of energy. Since the plant is largely a warehouse the AHU controls could be set up such that the supply temperatures do not stray far away from each other. This will reduce the amount of reheating required for spaces. This will require rezoning of spaces throughout the facility and changing controls of the AHUs. This option will not be discussed in the implementation section.

Assume there is a zone being served by an AHU which is smaller in terms of square footage when compared to other zones. The space is also the critical zone and thus dictates the supply air temperature of all zones being served by the AHU. With a CAV system the AHUs will need to cool down certain cubic feet of air regardless of the cooling demand of different spaces. The smaller space has a requirement of say, 30% of the total supply air. This means that the remaining 70% of cool air needs to be reheated to meet the needs of the other zones. This leads to huge energy losses through reheating.

For a facility which is, for the most part, just a warehouse, the tonnage required as shown by Carrier HAP, is higher than what is expected. Proper zoning of spaces and resetting thermostats could lead to the required tonnage of the chiller plant to go down. The airflow rates of the spaces need to be re-assessed as well. The purpose for which the spaces were being used, has changed significantly over the years. The controls have been set up according to the flow rate requirement of the manufacturing era of the plant. With greater air flow rate requirement, the overall tonnage and energy consumption values go up. Thus, checking the flow rates and setting them according to the current ASHRAE standards is a simple and a cheap option for the plant.

4.9.2: Cost savings and implementation

Cost savings can be calculated from Table 13,

Cost Savings (Table 13) = \$778,960 - \$755,559

= \$23,401/yr

The implementation of this recommendation will need the plant to change the dedicated pumping arrangement of the three 15 hp pumps. The plant needs to add piping to create a headered arrangement for the chiller pumps. From Section 4.6.4, the calculated 12-inch header will be used to join the pumps in parallel. Three 2-way valves will need to be installed on the pump's supply lines as shown in Figure 33. This will ensure that there is no mixing of flows if a pump is not running.

In addition to this, the plant needs to open the valve on the bypass line that currently exists. This completes the primary loop.

On the other hand, the secondary loop would need significant modifications,

1. The issue of low ΔT must be solved before going ahead with any recommendations. Issues discussed in Section 3.3 need to be looked at and rectified. With a higher-than-normal flow rate through the chiller, the most probable cause is that the cooling coils are ineffective in transferring heat efficiently. Simply cleaning the coils will improve the heat transfer by a huge margin and the flow rates can then be reduced.
2. The plant needs to add a VFD to the 75 hp pump which is currently a constant speed pump. It is assumed that the motor of the pump is compatible with a VFD.

3. The plant needs to consider shifting to 2-way valves or close the bypass line of the 3-way valve to convert it to a 2-way valve. The cheaper option is to just close the bypass line of the 3-way valve. This could prove inefficient because of the valve characteristics as discussed in Section 3.2:. Lower % of valve opening could lead to more flow and thus more heat transfer which then results in more reheat. Therefore, the valves should be tested and balanced before the bypass line is closed.

The 15 hp pumps should remain operational and the energy consumption has been accounted for through the simulations. The plant could turn the two 50 hp pumps off and add piping such that the 75 hp pump can supply the entire plant's chilled water similar to Section 4.6.3,

Demand Reduction	=	(2 x 50 hp) x 0.746 kW/hp
	=	74.6 kW
Energy Savings	=	74.6 kW x 5,400 hrs/yr
	=	402,840 kWh/yr
Cost Savings	=	74.6 kW x 12 months x $12.96/kW
		+ 402,840 kWh x $0.0503/kWh
	=	$31,864/yr
Total cost savings	=	$31,864/yr + $23,401/yr
	=	$55,265/yr

This recommendation borrows implementation costs from Section 4.6.4 and adds newer components,

Table 14: Implementation costs

Item	Unit Price (materials and labor)[2]	Cost
70 feet of 12" Steel Pipe (Includes Headered Pumping)	$230 / foot	**$16,800**
Three 12" Gate Valves	$7,450 each	**$22,350**
Three 12" Tee Joints	$2,475 each	**$7,425**
VFD for 75 hp Pump	$7,710	**$7,710**
Addition of Three 2-way Valves (Globe)	$11,630 each	**$34,890**
Inspection and cleaning of cooling coils	$2,000 each	**$10,000**
Engineering	50% of material and labor costs	**$50,000**
Total Cost of Project (excluding asbestos abatement)		**$149,175**

Simple Payback Period = Implementation cost / Cost savings

= $149,175 / $55,265/yr

= 2.7 years

Therefore, the total cost comes out to be $149,175 and with $55,265 as cost savings, the simple payback period will be 2.7 years.

[2] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

4.9.3: Additional recommendation

The plant could consider closing the bypass line of the 3-way valves and retain constant speed pumps. This does convert the system to a pseudo-variable flow system but comes at a price. With constant speed pumps and 2-way valves, the pressures across the valves will rise. With a rise in pressure differences, a small variation in the lift will result in a great change in the flow rate and this will lead to loss of control and eventually temperature control could be lost. An additional drawback is the hit in performance the pump takes. With higher pressures, the loads acting on the pumps will be higher thus reducing its life quickly. This solution should be considered just to test out if a shift to variable speed pumps would benefit the plant in question. Operating the constant speed pumps with 2-way valves for two or three months could provide the facility with a decent amount of data that they could use to present a case to the upper management for investing in variable speed drives [36].

4.10: Primary secondary loop with RTUs converted to chilled water

This recommendation builds on the previous one which added a primary secondary loop to the 750-ton chiller and included a headered pumping arrangement. In addition to that, the plant could refurbish the 500-ton chiller and convert all the rooftop units to chilled water.

The motivation comes from the possibility of increasing the overall efficiency of the chiller plant. Rooftop units lose out to chillers in efficiency terms as the years progress mainly due to lack of maintenance on the RTUs. The efficiency of a RTU is understood by energy efficiency ratios (EER). It is the ratio of the cooling capacity of the air conditioner (Btu/h) to the power input (W). Thus, higher the EER, better the efficiency of the unit.

In the Carrier HAP simulations, an EER rating of 11 has been assumed for all RTUs as per ASHRAE 90.1-2004 guidelines. The assumed efficiency of the RTUs is most probably higher than the actual efficiency of the units but is still less efficient when compared with the 0.6 kW/ton for chillers. If the RTUs are very old, they could very well be operating at an EER of less than 5 whether proper maintenance was carried out or not. Thus, an important fact to understand would be that converting RTUs to CHW could boost the energy savings by two times of what will be calculated below depending on how old the units are.

The ease of the conversion process depends on different factors as is the case with most engineering solutions. A more informed decision can be made based on the type and condition of the RTUs. It is possible to replace the DX coils but CHW pipes will need a greater number of rows to achieve the same amount of cooling. The engineers would need to look at the space available in the RTUs or make modifications to the unit.

The controls of the RTUs would need to be changed. The fan speed is set for cooling certain cubic feet of air for a certain number of DX cooling coils. With an increase in the number of rows for CHW pipes, the fan speed would need to be increased to overcome a greater pressure drop.

With a constant volume system, the fans run continuously without considering the occupancy and this results in high energy consumption. The plant could look to change the controls to have a demand-controlled ventilation (DCV) system in place with variable speed fans. Variable air volume system with DCV will be discussed later.

If the RTUs are added to the chilled water loop, the secondary pumps will need to be selected with a slightly greater head requirement. As calculated in previous recommendations, the pump was sized for a head of 85 feet. The required head of the pump will increase since the pump will have to supply water over a greater distance. Referencing Figure 48, it is assumed that the maintenance room is the most hydraulically remote point. An assumption commonly made in the engineering industry is that for every 100 ft of chilled water piping, there will be head loss of 10 ft. This includes friction losses, cooling coil head drop, loss due to valves and pipe bends. Google Earth was used to calculate the approximate increase in the distance that the chilled water would have to travel to reach the most hydraulically remote coil.

For the simulation, a head value of 110 ft was used based on the extra distance and a safety factor. The current setup of the plant has two 50 hp pumps and one 75 hp pump. These pumps, arranged in parallel, are more than capable of supplying the chilled water demand of the plant. The exact arrangement can be decided with more information on pumps and the piping layout of the plant. Bell & Gossett's pump sizer was used to generate the pump curve as shown in Figure 53.

The inputs for sizing the pump were 3,000 GPM with a head of 140 ft. The motivation behind choosing 140 ft as opposed to the 110 ft was to future-proof the system. If at all the plant decides to move to a primary-only variable flow system, then a head of 140 ft would be required to pump through the chillers and to the entire plant, including the RTUs.

The sizing was done for three VFD pumps in parallel. According to the sizer, three 50 hp VFD pumps will be needed to meet the head and flow requirements. A parallel arrangement was chosen because it allows the plant to easily do maintenance work on one of the pumps if needed without stopping the flow of chilled water through the facility.

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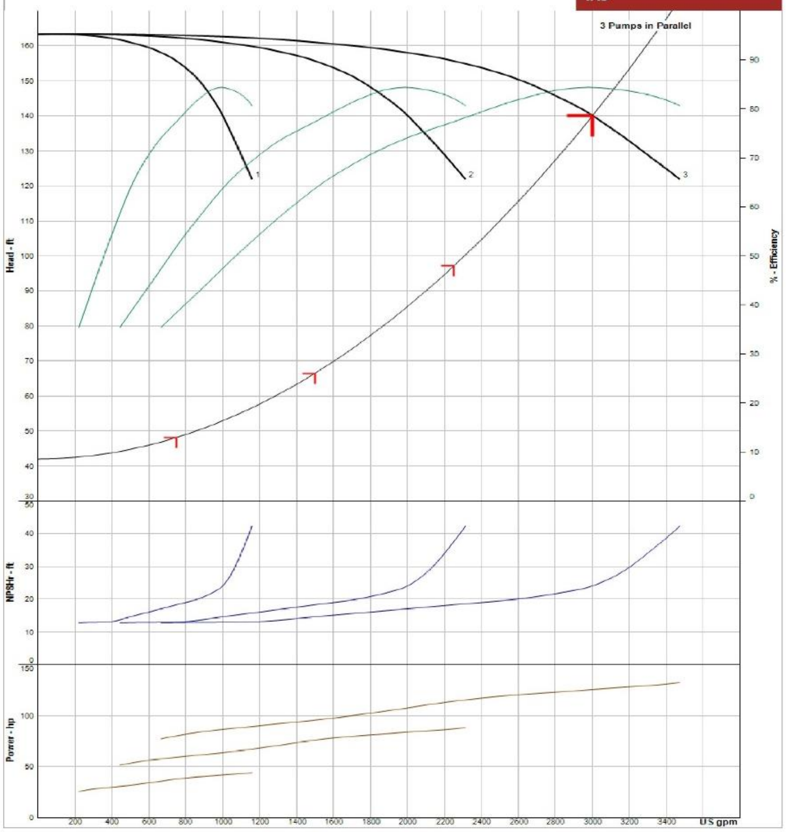
Performance Curve e-1510 4AD



Pump Selection Summary

Pump Capacity	1000 US gpm	RPM @ Duty Point	3436
Pump Head	140 ft	Impeller Diameter	6.875 in
Control Head	42 ft	NPSHr	24.2
Duty Point Pump Efficiency	84.3 %	Motor Power	50 hp
Pump PLEVv Efficiency	77.3 %	Motor Speed	3600 rpm
Duty point Power	41.9 bhp	Minimum Shutoff Head	163 ft
Minimum Flow at RPM	228 US gpm	Fluid Type	Water
Flow @ BEP	990 US gpm	Fluid Temperature	68 °F
Weight (approx. - consult rep)	864 lbs	Floor Space	6.65 ft^2

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Figure 53: Pump sizing for the secondary loop

The HVAC zoning in the facility was modified for the simulation. The problems that arise with not redefining zones have been discussed in the previous recommendation. The warehouse previously being served by RTUs has been combined with the chiller-served warehouse area to form one single zone being served by the chiller. Thus, the supply air temperature for both spaces will be the same. The plant can save significant reheating energy by combining zones and with variable flow secondary pumps.

The facility also needs to look at refurbishing one 500-ton chiller to ensure the cooling demand is met. Since more spaces are added to the chiller system the overall tonnage goes up and the plant will need to use its 500-ton chiller along with the 750-ton chiller.

To change operation to primary secondary pumping, the plant manager needs to check the position of the common bridge. It should be located such that it lies in between the primary chiller loop and secondary loop (front loaded). Having the common bridge positioned as shown in the figure below (back loaded), will lead to unequal loading of chillers because of laws of hydraulics (mixing of flows at the T-section) [23].

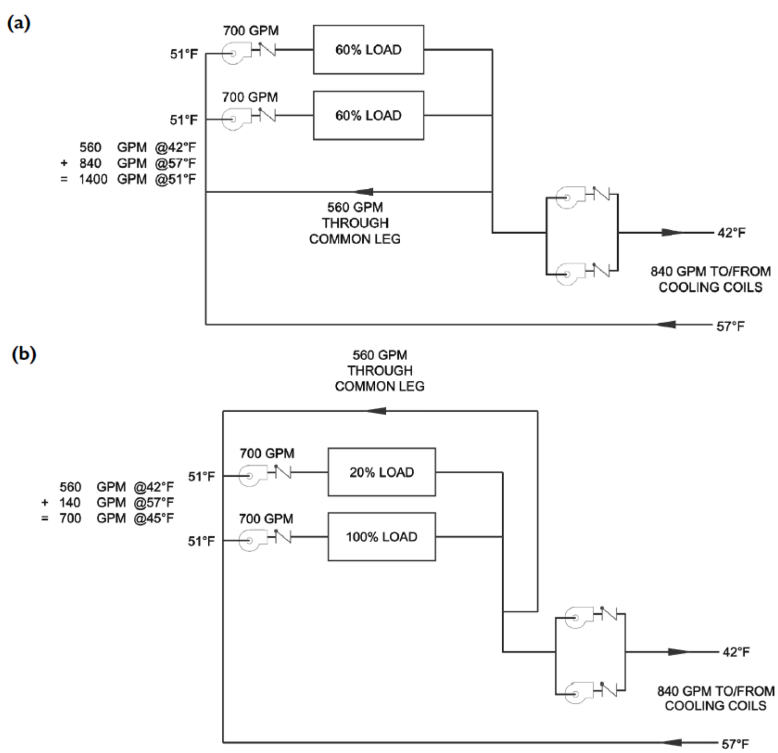


Figure 54: (a) Front vs (b) back-loaded common leg [25]

4.10.1: Energy savings

The following figures and tables will illustrate the results obtained after running the simulation and the results have been compared with the base model energy, demand and natural gas consumption values. For ease of understanding the model will be referred to as P-S RTU in the figures and respective tables.

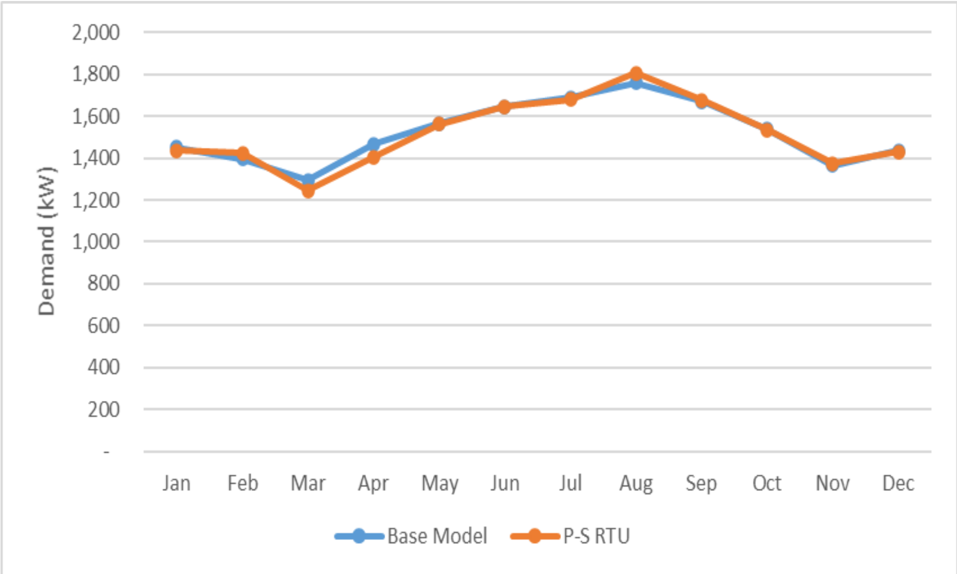


Figure 55: Demand values of base model vs P-S RTU

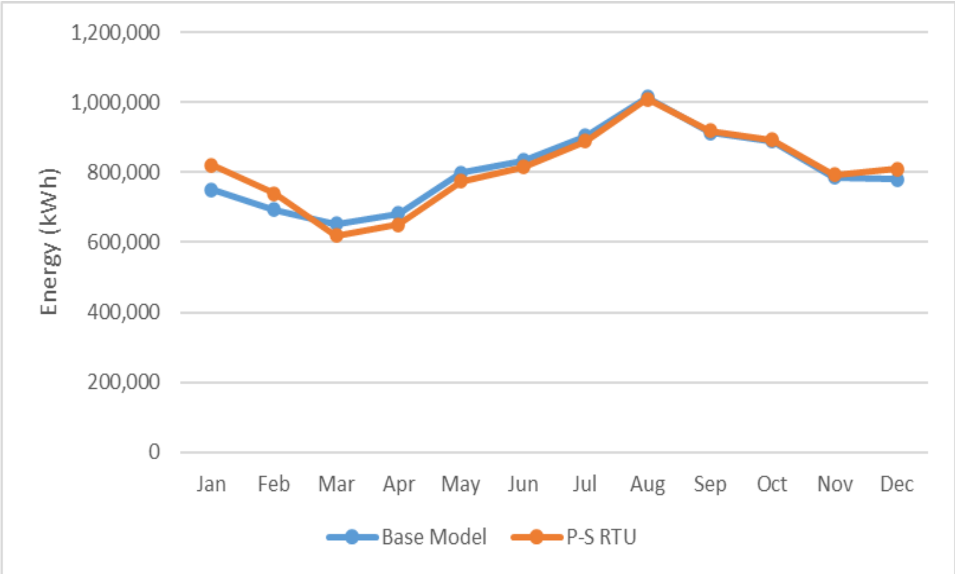


Figure 56: Energy values of base model vs P-S RTU

Table 15: Energy and demand comparison of base model vs P-S RTU

Month	Base model demand (kW)	P-S RTU demand (kW)	% Drop
Jan	1,452	1,437	1.01
Feb	1,397	1,423	(1.80)
Mar	1,294	1,246	3.66
Apr	1,466	1,404	4.18
May	1,566	1,563	0.18
Jun	1,647	1,647	0.00
Jul	1,689	1,681	0.47
Aug	1,759	1,805	(2.60)
Sep	1,671	1,677	(0.34)
Oct	1,537	1,536	0.07
Nov	1,366	1,374	(0.59)
Dec	1,438	1,431	0.45

Month	Base model energy (kWh)	P-S RTU energy (kWh)	% Drop
Jan	750,490	820,326	(9.31)
Feb	692,853	740,291	(6.85)
Mar	652,314	618,653	5.16
Apr	681,494	650,542	4.54
May	799,512	774,782	3.09
Jun	834,377	815,088	2.31
Jul	904,597	890,071	1.61
Aug	1,015,629	1,009,757	0.58
Sep	913,345	919,569	(0.68)
Oct	890,053	892,771	(0.31)
Nov	786,772	793,699	(0.88)
Dec	780,044	809,272	(3.75)

Table 16: Natural gas comparison of base model vs P-S RTU

Month	Base model gas consumption (MMBtu)	P-S RTU gas consumption (MMBtu)	Reduction (MMBtu)
Jan	479	475	4
Feb	473	481	(8)
Mar	22	20	2
Apr	12	12	0
May	0	0	0
Jun	0	0	0
Jul	0	0	0
Aug	725	0	725
Sep	889	310	579
Oct	1,259	458	801
Nov	1,571	584	987
Dec	2,025	962	1,063

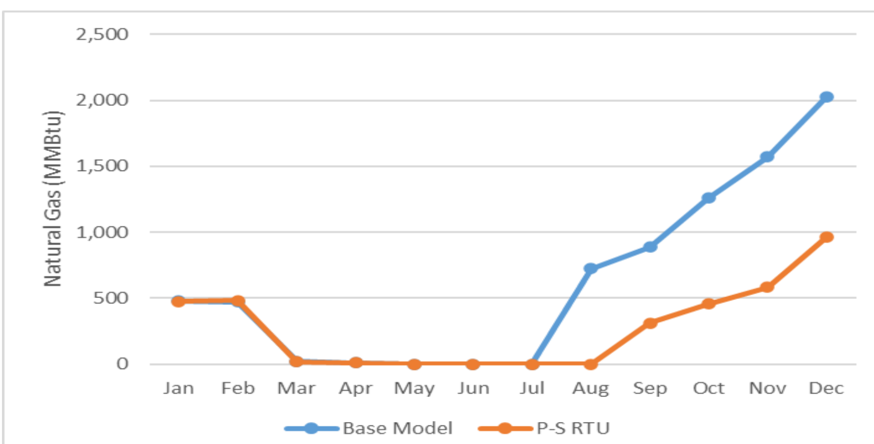


Figure 57: Natural gas consumption of base model vs P-S RTU

Table 17: Annual cost comparison of base model vs P-S RTU

Component	Base model	P-S RTU	% Drop
Air System Fans	$167,521	$169,935	(1.4)
Cooling	$159,559	$138,772	13.0
Heating	$125,502	$109,202	13.0
Pumps	$62,921	$52,521	16.5
Heat Rejection Fans	$17,673	$30,724	(73.8)
HVAC Sub-Total	**$533,177**	**$501,155**	**6.0**
Lights	$192,149	$192,586	(0.2)
Electric Equipment	$53,635	$56,046	(4.5)
Non-HVAC Sub-Total	**$245,784**	**$248,632**	**(1.2)**
Grand Total	**$778,960**	**$749,787**	**3.7**

The plant's overall fan energy increases. With more chilled water flowing because of RTU conversion, the cooling tower fans need to work much harder to cool the condenser water. Therefore, the energy cost of heat rejection fans jumps up in percentage. The slight increase in cost of running the air system fans can be attributed to RTUs running the fans harder to overcome the pressure drop due to CHW coils. This actual cost could be higher because Carrier HAP assumes the system to originally be a chilled water system and not a retrofit for RTUs. To reduce cost of operating the cooling tower fans, they could be fit with a VFD which would immensely boost savings, or the plant could look for a cheaper option in the form of a 2-speed motor.

Cooling energy cost reduction can be attributed to the boost in efficiency of chillers when compared with RTUs. 0.6 kW/ton for chillers is better than the 11 EER for RTUs in efficiency terms. A direct consequence of reduction in cooling costs is the reduction in heating costs. The combination of VFD secondary pumps along with rezoning the plant helps reduce the terminal reheat required.

The cost of pumping reduces because of the variable flow pumps in the secondary loop. This cost is a little higher when compared to the precious model's pumping cost which is because of the increased head requirement this time around. Just like the previous recommendation the cost reduction for pumping accounts for the greatest percentage change in terms of savings.

4.10.2: Cost savings and implementation

Cost savings can be calculated from Table 17,

Cost Savings (Table 17)	=	\$778,960/yr - \$749,787/yr
	=	\$23,401/yr

To restate a previously mentioned aspect of cost savings – the cost reduction of \$23,401/yr will be greater if the energy efficiency ratings of the RTUs is lesser than what was assumed for the energy model. However, the current cost savings will be used to calculate the simple payback period.

For implementing this recommendation, the following considerations need to be made –

1. The upgrades required to create the primary secondary chilled water distribution system will be carried over to this recommendation. This includes the opening of the valve on the bypass line, converting the dedicated chiller pumping to a headered arrangement, elimination of the low ΔT issue on the air side, adding VFD to the 75 hp pump and modifying the 3-way valves.
2. VFDs need to be added to the two 50 hp pumps. The final arrangement of the three secondary pumps can be decided only if complete information on piping layout is available. For this recommendation it is assumed that the plant's existing pumps if arranged in parallel and with VFDs can handle the demand of the plant. In the event that the head requirement can not be met with the existing setup, a smaller 15 hp pump can be installed near the most remote coil.
3. The RTUs need to be switched to CHW. This would need CHW piping runs to all the RTUs in the area. Google Earth was used to calculate the length of extra piping needed and it

came out to be approximately 1,500 ft. Using the flow rate, velocity and area relationship, the pipe diameter can be approximated to be around 6 inches. Reheating methods have been assumed to stay the same. Piping hot water will lead to significant rise in costs and thus is not considered and has not been simulated for this recommendation.

4. Controls would need to be modified for all RTUs and AHUs. The fans of RTUs would need to run at higher speeds. For the implementation cost, a section on maintaining duct static pressure using updated controls was looked up in RSMeans. It is also assumed that the total number of RTUs is 15 which will be used for the cost calculation.
5. The 500-ton chiller would need to be refurbished and started up.

This recommendation borrows implementation costs from the previous recommendation and adds newer components,

Table 18: Implementation costs

Item	Unit Price (materials and labor)[3]	Cost
80 feet of 12" Steel Pipe (Includes Headered Pumping and Parallel Secondary Pumping)	$230 / foot	**$18,400**
Addition of Three 2-way Globe Valves	$11,630 each	**$34,890**
VFD for 75 hp Pump	$7,710	**$7,710**
Two VFDs for 50 hp Pumps	$5,420 each	**$10,840**
1,500 ft of 6-inch Chilled Water Piping (along with hangers)	$60 / foot	**$90,000**
Setting up controls for RTUs	$1,500 each	**$22,500**
Overhaul the 500-ton chiller	~10% of the cost of a new 500-ton chiller	**$25,000**
Inspection and cleaning of cooling coils	$2,000 each	**$20,000**
Engineering	50% of material and labor costs	**$114,670**
Total Cost of Project (excluding asbestos abatement)		**$344,010**

[3] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

Simple Payback Period = Implementation cost / Cost savings

= \$344,010 / \$23,401/yr

= 15 years

Therefore, the total cost comes out to be \$344,010 and with \$23,401 as cost savings, the simple payback period will be 15 years.

4.10.3: Additional recommendation

With a payback of 12.3 years, the previous recommendation is not practical to be implemented. To avoid the hassle of conversion, the plant could consider replacing the units entirely. ASHRAE codes have become far more stringent over the years which has led to manufacturers creating more efficient rooftop units. Newer units can reach EER ratings greater than 15 which would lead to a significant gain in energy savings.

An energy model was made and simulated in Carrier HAP where the EER rating for all the units was improved from 11 to 16. An EER of 16 was taken based on claims made my multiple RTU manufacturers on their respective websites.

With an EER of 16, the annual cost of running the plant came out to be \$726,321. The cost of cooling saw an 18% decline as compared to the base model. This model is a modification of the recommendation discussed in section 4.9, where the 750-ton chiller runs using the primary secondary water distribution system and the RTUs are not switched to chilled water.

The calculated tonnage of the 5 zones being served by the RTUs was about 400 tons. Three zones had a tonnage requirement of 30 tons each and the other 2 zones had a requirement of 210 tons and 100 tons respectively. Using RSMeans, the costs of buying and installing a new single zone RTU with electric heating for different tons of cooling were looked at.

Table 19: Implementation costs

Item	Unit Price (materials and labor)[4]	Cost
30-ton RTU	$26,225 each	**$78,675**
50-ton RTU	$41,750 each	**$83,500**
75-ton RTU	$62,550 each	**$187,650**
Total Cost of RTUs		**$187,650**

The total cost of implementation would be the addition of costs from the table above and from Table 14. The total cost savings would be the difference in operation costs of the base model and the cost obtained for this recommendation.

Implementation cost = $187,650 + $149,175

= $336,825

Cost savings = $778,960/yr - $726,321/yr

= $52,639/yr

Simple Payback Period = Implementation cost / Cost savings

= $336,825 / $52,639/yr

= 6.5 years

Therefore, the total cost comes out to be $336,825 and with $52,639 as cost savings, the simple payback period will be 6.5 years. This is a more reasonable payback period, but the implementation cost is very high. It could be tough to recommend either of the two solutions, but if the facility is looking to resume manufacturing then this option should be strongly considered.

[4] Costs from Means' Mechanical Cost Data, 2019

4.11: Primary-Only variable flow systems

This recommendation is similar to the one discussed in Section 4.9. The water distribution system is converted to a primary-only variable flow system. The RTUs are not converted to chilled water because of the high capital investment required. The office and warehouse area located at the bottom in Figure 48 could be converted to chilled water since they are much closer to the mechanical room as compared to the maintenance room and the office spaces. However, the tonnage required to run the entire space is greater than 750 tons. This would require the 500-ton chiller to be refurbished and started up which would add to the implementation costs and the payback period would remain high. Thus, the RTUs running on chilled water will not be considered for the primary secondary or the primary-only variable flow systems.

4.11.1: Energy savings

The following figures and tables will illustrate the results obtained after running the simulation and the results have been compared with the base model energy, demand and natural gas consumption values.

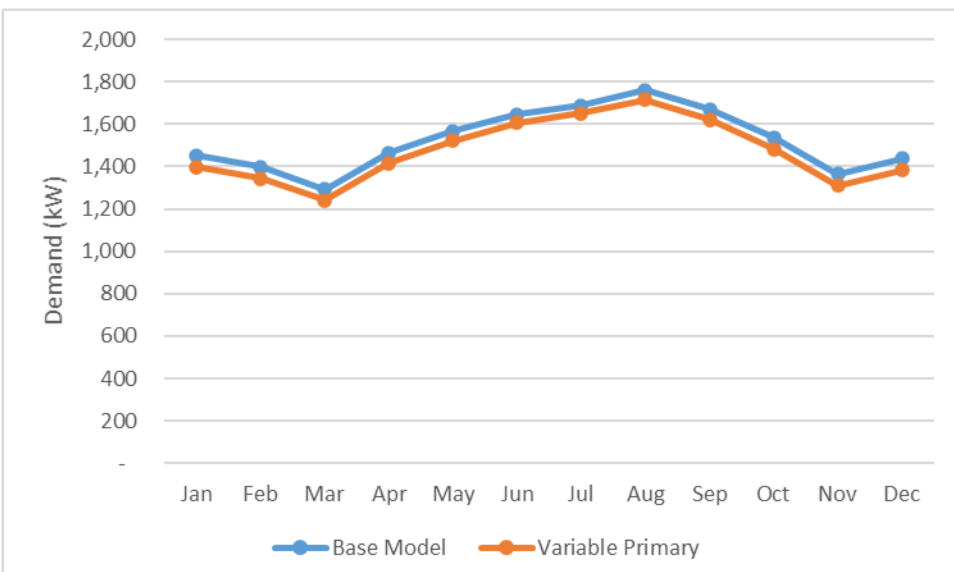


Figure 58: Demand values of base model vs variable primary

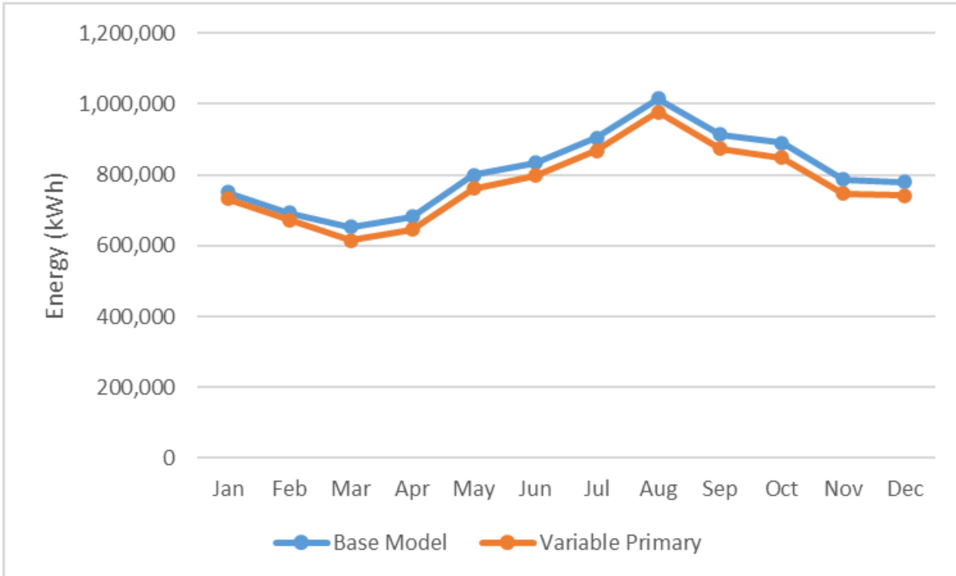


Figure 59: Energy values of base model vs variable primary

Table 20: Energy and demand comparison of base model vs variable primary

Month	Base model demand (kW)	Variable Primary demand (kW)	% Drop
Jan	1,452	1,398	3.71
Feb	1,397	1,344	3.82
Mar	1,294	1,241	4.10
Apr	1,466	1,417	3.36
May	1,566	1,523	2.79
Jun	1,647	1,608	2.37
Jul	1,689	1,652	2.21
Aug	1,759	1,717	2.44
Sep	1,671	1,623	2.88
Oct	1,537	1,483	3.52
Nov	1,366	1,311	4.03
Dec	1,438	1,383	3.79

Month	Base model energy (kWh)	Variable Primary energy (kWh)	% Drop
Jan	750,490	732,794	2.36
Feb	692,853	673,601	2.78
Mar	652,314	615,281	5.68
Apr	681,494	646,146	5.19
May	799,512	761,308	4.78
Jun	834,377	798,481	4.30
Jul	904,597	868,304	4.01
Aug	1,015,629	977,146	3.79
Sep	913,345	874,255	4.28
Oct	890,053	848,996	4.61
Nov	786,772	746,331	5.14
Dec	780,044	741,939	4.88

Table 21: Natural gas comparison of base model vs variable primary

Month	Base model gas consumption (MMBtu)	Variable Primary gas consumption (MMBtu)	Reduction (MMBtu)
Jan	479	483	(4.00)
Feb	473	475	(2.00)
Mar	22	23	(1.00)
Apr	12	13	(1.00)
May	0	0	0.00
Jun	0	0	0.00
Jul	0	0	0.00
Aug	725	590	135.00
Sep	889	758	131.00
Oct	1,259	1,125	134.00
Nov	1,571	1,447	124.00
Dec	2,025	1,924	101.00

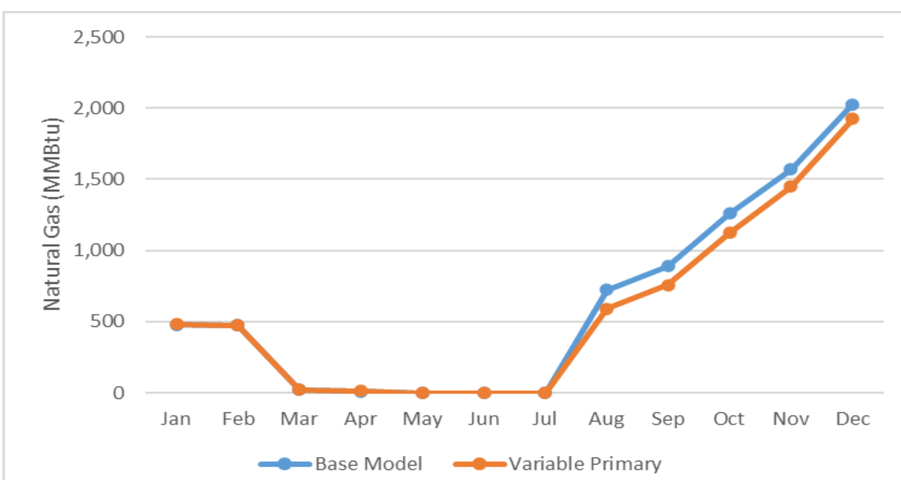


Figure 60: Natural gas consumption of base model vs variable primary

Table 22: Annual cost comparison of base model vs variable primary

Component	Base model	Variable Primary	% Drop
Air System Fans	$167,521	$163,063	2.7
Cooling	$159,559	$155,138	2.8
Heating	$125,502	$121,109	3.5
Pumps	$62,921	$42,616	32.3
Heat Rejection Fans	$17,673	$17,339	1.9
HVAC Sub-Total	**$533,177**	**$499,265**	6.4
Lights	$192,149	$192,860	(0.4)
Electric Equipment	$53,635	$53,834	(0.4)
Non-HVAC Sub-Total	**$245,784**	**$246,694**	(0.4)
Grand Total	**$778,960**	**$745,959**	4.2

The greatest amount of savings will be through decrease in pumping energy. The cost of pumping for the variable primary system is $42,616 which is lower than the $51,578 for the primary secondary system. This decrease in cost was expected due to the reasons discussed above.

Other takeaways from the table above are similar to the ones discussed in the primary secondary loop recommendation. The costs of operating the air system fans, cooling and heating see a marginal drop. This could be because of better utilization of the AHUs with variable flow through its cooling coils which requires slightly lower hp fans. The chiller operates more efficiently because of the variable flow. It does not need to supply a fixed GPM of water and the loading on the chiller can be varied according to the demand of the cooling coils. The heating energy is reduced because of lower reheat required. Other points made in the discussion in the primary secondary recommendation can be stated here since they do not depend on the kind of pumping system implemented.

4.11.2: Cost savings and implementation

Cost savings can be calculated from Table 22: Annual cost comparison of base model vs variable primary,

Cost Savings (Table 17)	=	\$778,960/yr - \$745,959/yr
	=	\$33,001/yr

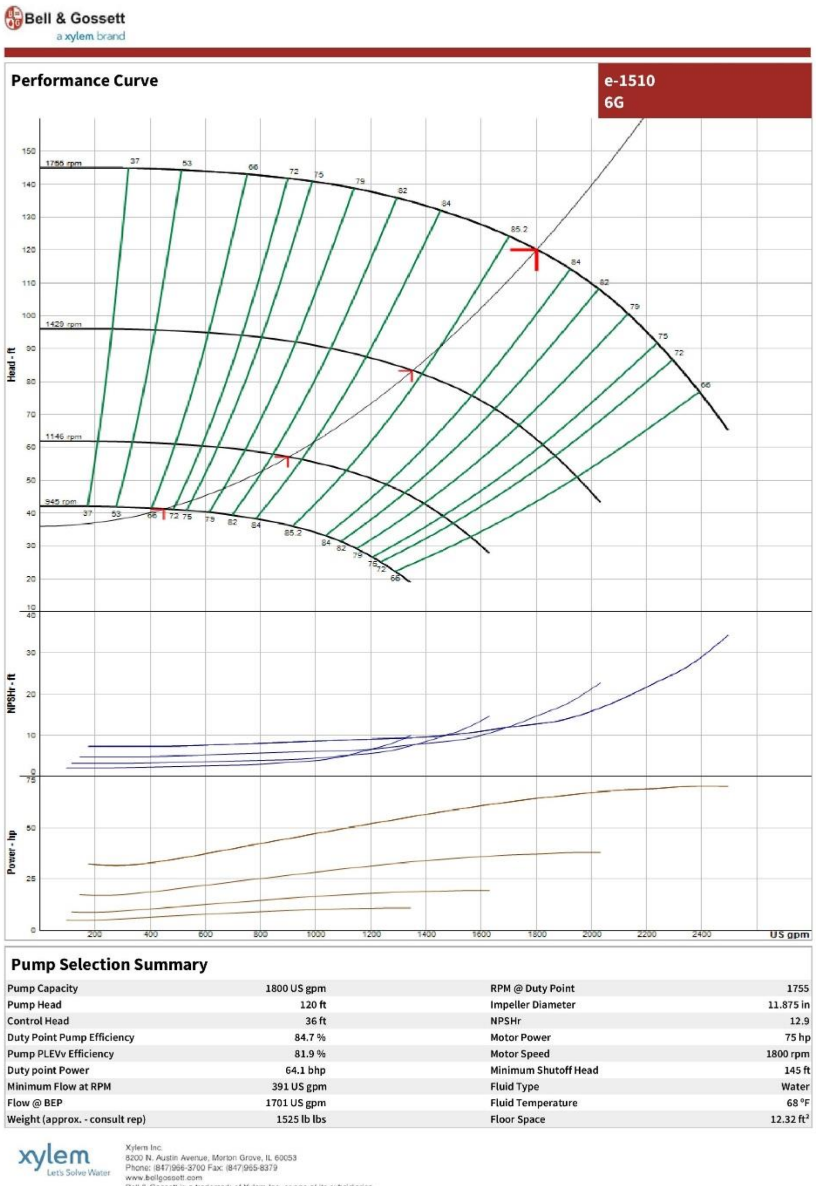
To ensure if the 75 hp pump would satisfy the head and flow requirement, the Bell & Gossett pump sizer was used. The flow rate required would be 2.4 GPM/ton for the 750-ton chiller which is 1,800 GPM. The head values used for the primary secondary loop were 45 ft for the primary side and 85 ft for the secondary side. As discussed before, the head requirement of the pump in a primary-only variable flow system will be lower than the combined head of the primary secondary system. Adding the two head values for the primary secondary system gives a total head of 130 ft. Assuming the conversion to variable primary system reduces the head requirement by 10 ft. Thus, a head of 120 ft was chosen for the primary-only variable flow system. The actual head requirement could be lower than 120 ft but using this value highlights the boost in pump savings as shown in the component costs table of this recommendation.

Figure 61 illustrates that a 75 hp VFD pump can supply chilled water to all zones being cooled using the chiller. Thus, the two 50 hp pumps and the 15 hp chiller pump can be turned off and will result in energy and cost savings,

Demand Reduction	=	(15 hp + 2 x 50 hp) x 0.746 kW/hp
	=	85.79 kW
Energy Savings	=	85.79 kW x 5,400 hrs/yr
	=	463,266 kWh/yr
Cost Savings	=	85.79 kW x 12 months x \$12.96/kW
		+ 463,266 kWh x \$0.0503/kWh
	=	\$36,644/yr

Total cost savings = $36,664/yr + $33,001/yr

= $69,665/yr



Pump Capacity	1800 US gpm	RPM @ Duty Point	1755
Pump Head	120 ft	Impeller Diameter	11.875 in
Control Head	36 ft	NPSHr	12.9
Duty Point Pump Efficiency	84.7 %	Motor Power	75 hp
Pump PLEVv Efficiency	81.9 %	Motor Speed	1800 rpm
Duty point Power	64.1 bhp	Minimum Shutoff Head	145 ft
Minimum Flow at RPM	391 US gpm	Fluid Type	Water
Flow @ BEP	1701 US gpm	Fluid Temperature	68 °F
Weight (approx. - consult rep)	1525 lb lbs	Floor Space	12.32 ft²

Figure 61: Pump sizing for 1,800 GPM and 120 ft of head

For implementing this recommendation, the following considerations need to be made –

1. Some of the upgrades required to create the primary secondary chilled water distribution system will be carried over to this recommendation. This includes elimination of the low ΔT issue on the air side, adding VFD to the 75 hp pump and modifying the 3-way valves.
2. All pumps except the 75 hp pump will need to be turned off and piping modifications need to be done to make sure the areas being served by the two 50 hp pumps are now being served by the 75 hp pump.
3. Running a primary-only variable flow system requires the chiller to support variable flow through its evaporator. Older chillers are not capable of handling variable flow which could be the case for the chiller of this plant. According to the ASHRAE handbook, the approximate lifespan of a centrifugal chiller is about 25 years and it is economical to have the entire unit replaced rather than work on overhauling it. The plant should consider buying a new 750-ton chiller which is capable of handling variable flow.

This recommendation borrows implementation costs from the previous recommendation and adds newer components,

Table 23: Implementation costs

Item	Unit Price (materials and labor)[5]	Cost
60 feet of 12" Steel Pipe	$230 / foot	**$13,800**
Three 12" Gate Valves	$7,450 each	**$22,350**
Three 12" Tee Joints	$2,475 each	**$7,425**
VFD for 75 hp Pump	$7,710	**$7,710**
Addition of Three 2-way Valves (Globe)	$11,630 each	**$34,890**
Inspection and cleaning of cooling coils	$2,000 each	**$10,000**
New 750-ton water cooled centrifugal chiller	$369,100	**$369,100**
Engineering	50% of material and labor costs	**$230,137**
Total Cost of Project (excluding asbestos abatement)		**$695,412**

Simple Payback Period = Implementation cost / Cost savings

= $695,412 / $69,665/yr

= 10 years

Therefore, the total cost comes out to be $695,412 and with $69,665 as cost savings, the simple payback period will be 10 years. This recommendation could be a tough sell for the upper management but should be considered if the plant is planning resume manufacturing operations again.

[5] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

4.12: Constant air volume to variable air volume conversion

This recommendation builds on the primary secondary recommendation made in Section 4.9. The couple of recommendations before this dealt with the water side operation of the chiller plant. This recommendation will look at changing the air side of the HVAC system. Since converting the RTUs to CHW and the primary-only variable flow systems had unfavorable payback periods, these options will not be explored here.

4.12.1: Variable Air Volume Systems

As opposed to CAV, a VAV system will vary the amount of air being supplied to the space. Better part load performance is achieved by means of lower reheat requirement. A significant amount of energy is saved in running the fans because they can vary their speeds and do not need to push a constant amount of air [37].

VAV boxes contain dampers which change position based on the thermostat set point of the zone to change the flow rate of air. When the air dampers close, less air flows into the zone. This results in a pressure rise in the duct. A pressure sensor, usually located in the most remote region of the ductwork, sends a signal to the AHU. The AHU will then lower the speed of its fans to reduce air supply. This helps in ensuring that no more than the required air flow occurs through all the VAV units associated with the AHU. The speed of the return fan will be adjusted as per the supply fan speed.

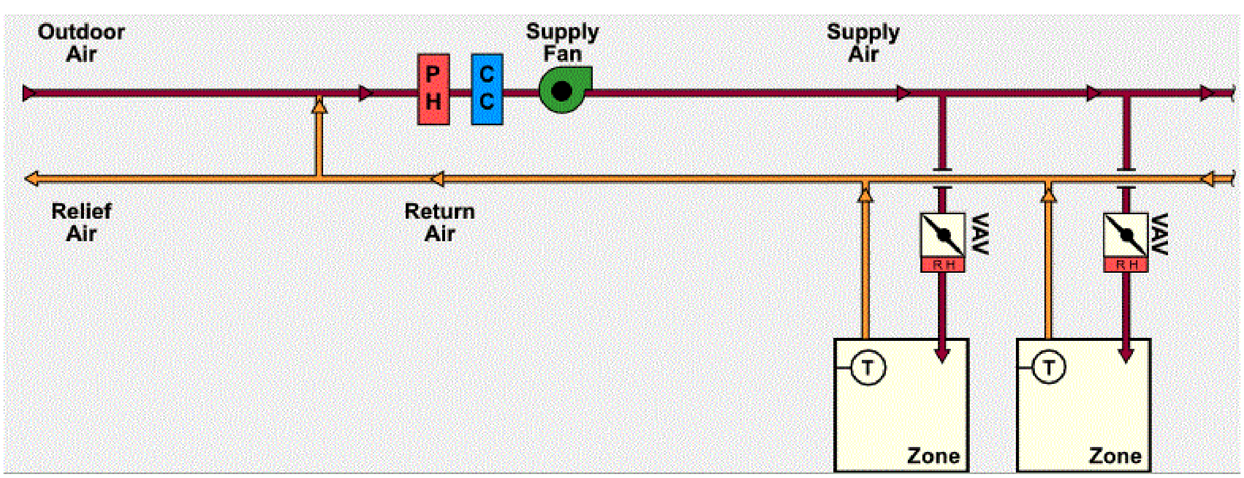


Figure 62: Variable air volume system

Converting a CAV system to VAV would require a high investment which may not be very attractive to the plant manager. Assuming the system has not been touched since the time it was fit in, there could be a host of issues that would need to be dealt with before going in to change it. Asbestos is one major issue that the plant may have to deal with if they want to change the systems. The team found some asbestos near the chilled water pipes which could indicate that there is more of it around the ducts and wherever insulation is needed in and around the AHUs. Working on removal of asbestos will add significantly to the cost of the entire project.

Considering the age of the system, new controls would need to be fit in for efficient operation of the VAV system. Some of the older units could still be operated on pneumatic actuators instead of electric controls. A newer control system would require extra training for the plant technicians which is another add-on in terms of cost. An important aspect of changing the control system would be to look at the duct pressures and ensure they are reset to what the VAV system demands. For a CAV system, the duct pressure would be relatively higher than what it is for VAVs. If the pressure is not changed the effectiveness of VAV will be reduced significantly [38]. A common incident that can occur during warmer conditions is, the duct pressures will be increased to push more air through but when the climate cools off, the plant technicians forget to reset the pressures back to their original values thus resulting in greater energy consumption.

An additional issue with renovating the system would be complying to standards. If a system has been built during a time when the building standards were not as stringent as they are today, they can continue to function with the system they have. If any renovation work is done, the state could come in and inspect it according to the standards that exist today. Ensuring minimum requirements have been met would mean that the plant would need to invest a lot of time, effort and capital into this project to cover all their bases.

A solution that could be considered as a workaround could be to install VFDs on the fans of AHUs. A CAV system uses constant air flow and temperature control based on the most critical zone of the entire plant. If this zone’s cooling requirement reduces, the VFD will lower the fan speed. This could result in other spaces not getting enough cool air to adequately remove the heat. Therefore, without terminal boxes, this becomes a tricky operation from a controls standpoint. Installing a VFD on the fans of AHUs will probably need manual monitoring which is a drawback but then it will help save significant amount of energy as the airflow rates can be changed by looking at the needs of all spaces. The fans could be set at a lower speed during lower occupancy periods which can go hand in hand with supply air temperature resets. Utilizing such a system, in some cases, resulted in thermal energy consumption due to reheating was reduced by 40% and the fan power savings were about 60% [39].

With complete information on the current system, a recommendation can be made on how the conversion from CAV to VAV would look like. If the system does not need a complete overhaul, retrofits could prove to be a cheaper option. There are options available on the market which replace the terminal boxes and come with actuators fit in. Further research would be needed to find cost effective solutions for the exact system that is at the plant.

The simulation has been set up with the assumption that with a change to VAV system, the plant will use demand controlled ventilation. This is a newer airflow control strategy which is implemented in buildings with VAV systems. With a DCV system, the CO_2 level in a space is monitored and the ventilation rate is changed based on that. If there is a greater number of people in a room, the CO_2 level will rise. The ventilation rate of the room can thus be increased to maintain the CO_2 level within ASHRAE recommended levels. This leads to better airflow control for spaces. With a constant volume system, the occupancy level is ignored and the air flow rate is constant which leads to higher energy consumption. DCV helps in saving heating or cooling energy by changing the ratio of recirculated to outside air whereas, the overall fan energy consumption will remain the same since minimum airflow rates to spaces have to be maintained as per ASHRAE guidelines [40].

4.12.2: Energy savings

The following figures and tables will illustrate the results obtained after running the simulation and the results have been compared with the base model energy, demand and natural gas consumption values.

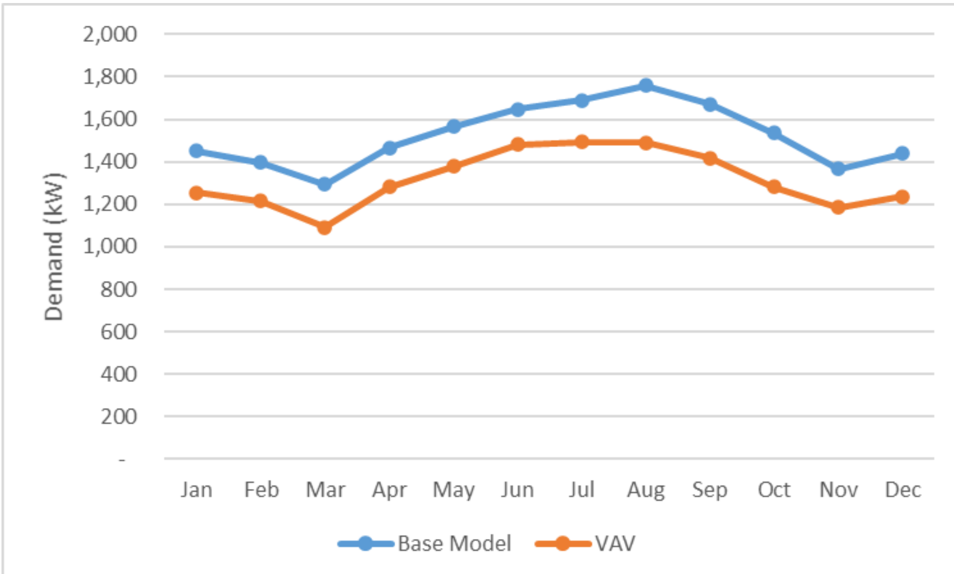


Figure 63: Demand values of base model vs VAV

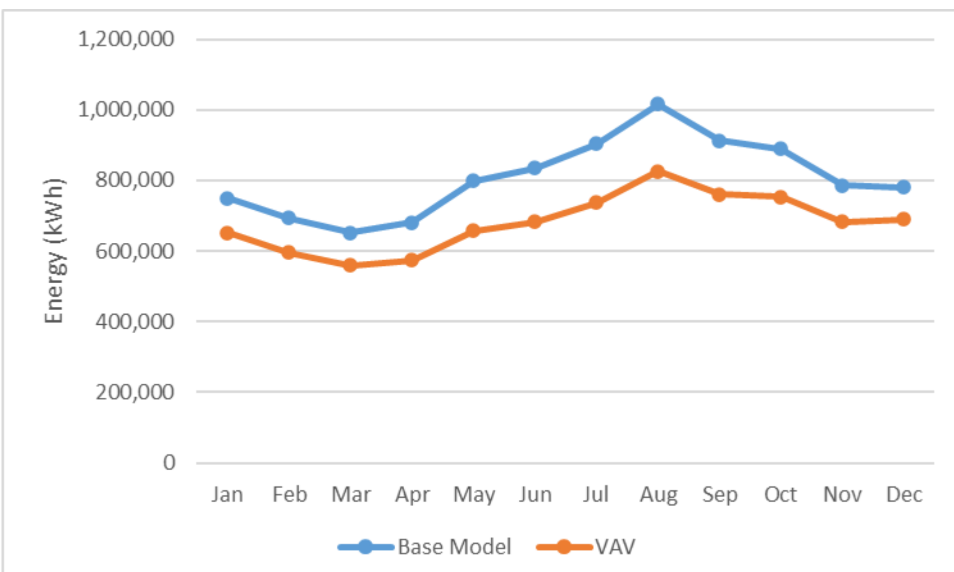


Figure 64: Energy values of base model vs VAV

Table 24: Energy and demand comparison of base model vs VAV

Month	Base model demand (kW)	VAV demand (kW)	% Drop
Jan	1,452	1,256	13.47
Feb	1,397	1,215	13.03
Mar	1,294	1,092	15.63
Apr	1,466	1,282	12.51
May	1,566	1,379	11.93
Jun	1,647	1,481	10.10
Jul	1,689	1,493	11.60
Aug	1,759	1,489	15.37
Sep	1,671	1,418	15.14
Oct	1,537	1,281	16.64
Nov	1,366	1,185	13.24
Dec	1,438	1,235	14.10

Month	Base model energy (kWh)	VAV energy (kWh)	% Drop
Jan	750,490	651,641	13.17
Feb	692,853	595,300	14.08
Mar	652,314	559,115	14.29
Apr	681,494	573,554	15.84
May	799,512	657,752	17.73
Jun	834,377	682,786	18.17
Jul	904,597	737,413	18.48
Aug	1,015,629	826,265	18.64
Sep	913,345	761,300	16.65
Oct	890,053	753,752	15.31
Nov	786,772	683,065	13.18
Dec	780,044	690,419	11.49

Table 25: Natural gas comparison of base model vs VAV

Month	Base model gas consumption (MMBtu)	VAV gas consumption (MMBtu)	Reduction (MMBtu)
Jan	479	123	356
Feb	473	142	331
Mar	22	5	17
Apr	12	3	9
May	0	0	0
Jun	0	0	0
Jul	0	0	0
Aug	725	366	359
Sep	889	526	363
Oct	1,259	872	387
Nov	1,571	1,188	383
Dec	2,025	1,507	518

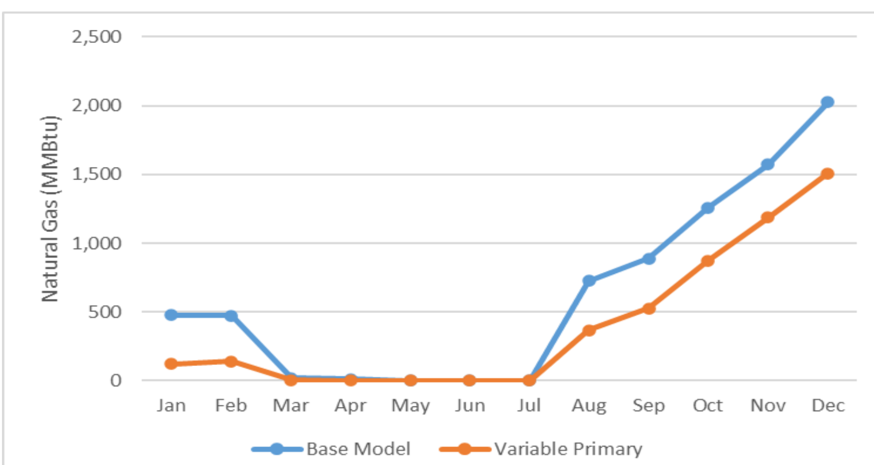


Figure 65: Natural gas consumption of base model vs VAV

Table 26: Annual cost comparison of base model vs VAV

Component	Base model	VAV	% Drop
Air System Fans	$167,521	$86,162	48.6
Cooling	$159,559	$140,387	12.0
Heating	$125,502	$105,716	15.8
Pumps	$62,921	$53,713	14.6
Heat Rejection Fans	$17,673	$16,327	7.6
HVAC Sub-Total	**$533,177**	**$402,305**	**24.5**
Lights	$192,149	$193,758	(0.8)
Electric Equipment	$53,635	$54,156	(1.0)
Non-HVAC Sub-Total	**$245,784**	**$247,915**	**(0.9)**
Grand Total	**$778,960**	**$650,220**	**16.5**

The overall energy consumption of the facility reduces drastically in all around. The largest drop is seen in the air system fans category which is to be expected with a CAV to VAV conversion. The fans consume energy according to the CFM requirement of the spaces and combined with the variable speed pumping, the savings obtained are significantly greater than a CAV system with variable secondary. The cooling and heating energy values also drop by large margins. This can be attributed to better load management with spaces reaching setpoint more efficiently with the help of VAV boxes and requiring lower terminal reheating.

4.12.3: Cost savings and implementation

Cost savings can be calculated from Table 22: Annual cost comparison of base model vs variable primary,

Cost Savings (Table 17)	=	\$778,960/yr - \$650,220/yr
	=	\$128,740/yr

The 15 hp pumps should remain operational and the energy consumption has been accounted for through the simulations. The plant could turn the two 50 hp pumps off and add piping such that the 75 hp pump can supply the entire plant's chilled water similar to Section 4.6.3,

Demand Reduction	=	(2 x 50 hp) x 0.746 kW/hp
	=	74.6 kW
Energy Savings	=	74.6 kW x 5,400 hrs/yr
	=	402,840 kWh/yr
Cost Savings	=	74.6 kW x 12 months x \$12.96/kW
		+ 402,840 kWh x \$0.0503/kWh
	=	\$31,864/yr
Total cost savings	=	\$31,864/yr + \$128,740/yr
	=	\$160,604/yr

The implementation of this recommendation will add on to the one discussed in Section 4.9 which changed the constant primary pumping to a primary secondary system.

Therefore, the water side changes that need to be implemented here are the same as the ones made in Section 4.9.3,

1. Creating a headered arrangement for the chiller pumps.
2. Opening up the bypass line and making it operational.
3. Solving the issues of low ΔT.
4. Adding a VFD on the 75 hp secondary loop pump.
5. Converting the 3-way valves in the AHUs to 2-way valves.

Additional changes need to be made on the air side. The current system in place is CAV with terminal reheat. As discussed before, the AHU will cool the air to the lowest temperature requirement from a space in a HVAC zone and other spaces will have to heat the incoming cold air according to the setpoint. There are no terminal boxes present which could be retrofit with any dampers to convert the system to a VAV. Therefore, new VAV terminal boxes will need to be purchased and installed in the seven spaces that the chiller serves. It is assumed that the plant will install seven VAV boxes which can be connected to the supply ducts. A standard CFM value for all the spaces combined came out to be 20,000 CFM after simulating the VAV system. For the implementation it is assumed that ten VAV terminals would need to be purchased, each with a 2,000 CFM rating. Many HVAC equipment manufacturers have VAV terminal boxes which are DCV-ready. The CO_2 sensors are pre-installed and programmed which makes it easier for installation.

The AHU controls will need to be updated such that the airflow control strategy adopted is demand controlled ventilation. The supply fans would need to be installed with VFDs for the entire setup to efficiently function as a VAV system. The Carrier HAP simulation resulted in the supply fan being rated at 270 hp for the area being served by the chiller. It was assumed that the entire space is served by 6 AHUs which results in each AHU having a 50 hp fan. This is assuming that each AHU is equally sized which they are probably not. However, the implementation costs should not stray far from what is assumed here.

It is also assumed that all the motors are compatible with variable frequency drives. After successful installation of the VAV terminals, commissioning service will be required to test the entire system.

This recommendation borrows implementation costs from Section 4.9.2: and adds newer components,

Table 27: Implementation costs

Item	Unit Price (materials and labor)[6]	Cost
70 feet of 12" Steel Pipe (Includes Headered Pumping)	$230 / foot	**$16,800**
Addition of Three 2-way Valves (Globe)	$11,630 each	**$34,890**
Three 12" Gate Valves	$7,450 each	**$22,350**
Three 12" Tee Joints	$2,475 each	**$7,425**
VFD for 75 hp Pump	$7,710	**$7,710**
VFD for 50 hp Fans	$5,420 each	**$32,520**
VAV Terminal Boxes with HW Reheat	$26,975 each	**$269,750**
Controls upgrade (VAV boxes and AHUs)	$90,000	**$90,000**
Inspection and cleaning of cooling coils	$2,000 each	**$10,000**
Engineering	$210,312	**$245,722**
Total Cost of Project (excluding asbestos abatement)		**$737,167**

Simple Payback Period = Implementation cost / Cost savings

= $737,167 / $160,604/yr

= 4.6 years

[6] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

Therefore, the total cost comes out to be $737,167 and with $160,604 as cost savings, the simple payback period will be 4.6 years. With a very high implementation cost of this recommendation, it would be difficult for the facility's upper management to green light this project. However, the plant could convert to the primary secondary loop first and then go ahead with the VAV conversion each terminal box at a time. The savings with the VAV system would be clearly visible with the fan speeds throttling down as more terminal boxes get added.

CHAPTER 5: CONCLUSIONS

5.1: Summary of recommendations

The following table lists all the five recommendations made throughout this thesis,

Table 28: Summary of cost savings, implementation costs and payback periods

Recommendation	Cost savings	Implementation cost	Simple payback period
Chilled water pump upgrade	$36,644/yr	$65,575	2 years
Primary secondary loop	$55,265/yr	$149,175	2.7 years
Primary secondary loop + RTUs converted to CHW	$23,401/yr	$344,010	15 years
Primary-only variable flow system	$69,665/yr	$695,412	10 years
Primary secondary + Constant air volume to variable air volume conversion	$160,604/yr	$737,167	4.6 years

The result of comparing all recommendations is that the plant should look to convert to a primary secondary loop since it is something that can be easily implemented by the plant. Most infrastructure and equipment are present and with a few additions they can get the system up and running. Conversion to a VAV air side system is also something worth considering. With a high implementation cost the plan of conversion could allow for a slow transition. This helps the plant manager present a case in favor of conversion since he will have enough data at each stage to show actual dollars being saved on the energy bills.

The recommendations have been made with the assumption that the base model created most closely represents the actual energy consumption of the facility. The base model was made such that it matched the energy and gas bills to a certain degree. With more information on the actual HVAC systems, an even more detailed energy model can be created. However, this does not detract from the benefits of all recommendations discussed in this thesis.

5.2: Additional recommendations

5.2.1: Variable speed drive fans and pumps on the cooling tower loop

In addition to the VFDs installed for the primary secondary loop in section 4.9, the plant can work on converting the condenser loop to variable pumping and change the cooling tower fans to variable speed fans. The cooling tower loop must have been sized to handle the plant's needs when they were manufacturing products so it might be oversized for the current needs.

The following tables compare the primary secondary (P-S) and the current recommendation – Primary Secondary with VFDs on the Cooling Tower side (P-S CT VFD).

Table 29: Energy and demand comparison of P-S vs P-S CT VFD

Month	Base model demand (kW)	P-S CT VFD demand (kW)	% Drop
Jan	1,417	1,406	0.78
Feb	1,364	1,352	0.82
Mar	1,260	1,249	0.90
Apr	1,434	1,422	0.86
May	1,538	1,525	0.86
Jun	1,622	1,609	0.85
Jul	1,665	1,651	0.83
Aug	1,730	1,716	0.80
Sep	1,639	1,625	0.81
Oct	1,501	1,489	0.81
Nov	1,330	1,319	0.82
Dec	1,402	1,391	0.83

Month	Base model energy (kWh)	P-S CT VFD energy (kWh)	% Drop
Jan	743,603	735,281	1.12
Feb	684,643	675,207	1.38
Mar	634,193	619,813	2.27
Apr	664,052	648,748	2.30
May	780,564	764,640	2.04
Jun	817,088	802,262	1.81
Jul	887,455	872,523	1.68
Aug	1,000,288	980,378	1.99
Sep	895,384	878,169	1.92
Oct	870,680	851,231	2.23
Nov	768,676	748,989	2.56
Dec	765,460	746,406	2.49

Converting the condenser side does not affect the natural gas consumption and thus, has not been shown.

Table 30: Annual cost comparison of P-S vs P-S CT VFD

Component	P-S	P-S CT VFD	% Drop
Air System Fans	$162,900	$163,019	(0.1)
Cooling	$156,089	$155,712	0.2
Heating	$121,136	$121,136	0.0
Pumps	$51,578	$46,304	10.2
Heat Rejection Fans	$17,415	$15,873	8.9
HVAC Sub-Total	**$509,117**	**$502,044**	**1.4**
Lights	$192,667	$192,808	(0.1)
Electric Equipment	$53,775	$53,816	(0.1)
Non-HVAC Sub-Total	**$246,442**	**$246,624**	**(0.1)**
Grand Total	**$755,559**	**$748,669**	**0.9**

With the addition of VFDs on fans and pumps on the condenser side of the chiller loop, significant savings can be only be obtained through reduction in cost of operating the pumps and heat rejection fans. All other costs practically remain unchanged which is evident from the table above.

Additional cost savings can be calculated from Table 30,

Additional Cost Savings = $755,559 - $748,669

= $6,890/yr

To change the controls of the constant speed cooling tower fans to VFD, the current horsepower of the fans needs to be calculated. The maximum tons of cooling that the plant's chillers can provide is 1,750 tons. ASHRAE 90.1 – Table 6.8.1G deals with performance requirements for heat rejection equipment. It specifies the ratio of the flow rate of condenser water to the fan horsepower for an open circuit cooling tower with an axial fan. The performance requirement of 38.2 GPM/hp is recommended. The flow rate of the condenser pumps is assumed to be 3 GPM/ton, as opposed to 2.4 GPM/ton, to account for the heat generated by the chiller compressor. Therefore, for 1,750 tons, the flow rate will be 5,250 GPM which results in a total fan horsepower of 140 hp. Upon inspection of the tower using Google Earth, it is observed that the tower has 2 cells which results in each cell with a 70 hp fan. Two VFDs can be added to each fan to reduce the fan energy consumption. According to the flow rate for the 750-ton chiller, the horsepower of the cooling tower fan comes out to be around 60 hp. Replacing the fans would need significant effort to update the tower design and controls. Adding lower hp fans would limit the plant's heat rejection capacity if they do decide to resume manufacturing. Thus, this option will not be explored.

For the 750-ton chiller, the condenser flow rate will be 2.250 GPM. Using Google Earth, the height of the cooling tower was approximated to be 20 ft which is the static head. Additional 30 ft of losses can be assumed to be overcome by the condenser pump due to friction losses and drop in head as water passes through the condenser and the cooling tower. Sizing a pump for 2,250 GPM and 50 ft of head, through the Bell & Gossett website, a 50 hp is recommended.

However, the pumps being used at the plant must have been sized for the scenario where all three chillers must have been operating. The pump then would have had a flow requirement of 3 GPM/ton for the 750-ton and two 500-ton chillers resulting in 5,250 GPM of flow through the condenser loop. During the visit, the team noted that the pump might be around 120-150 hp. This is confirmed by the pump sizer which gives a suggested horsepower of 120 hp. Since it is cheaper to purchase a VFD for a 50 hp pump as compared to the 120 hp pump, the plant can save energy by using one of the 50 hp pumps from the secondary loop that was turned off for the recommendation discussed in Section 4.9.

Demand, energy and cost savings obtained by turning off the 120 hp pump and using the 50 hp pump have been calculated below,

Demand Reduction	=	(120 – 50) hp x 0.746 kW/hp
	=	52.22 kW
Energy Savings	=	52.22 kW x 5,400 hrs/yr
	=	281,988 kWh/yr
Cost Savings	=	52.22 kW x 12 months x $12.96/kW
		+ 281,988 kWh x $0.0503/kWh
	=	$22,305/yr

The total cost savings will be a sum of the savings obtained above in addition to the ones calculated from Table 30 and from the recommendation in Section 4.9.

Total cost savings	=	$22,305/yr + $6,890/yr + $55,265/yr
	=	$84,460/yr

This recommendation borrows implementation costs from Section 4.9 and adds newer components,

Table 31: Implementation costs

Item	Unit Price (materials and labor)[7]	Cost
70 feet of 12" Steel Pipe (Includes Headered Pumping)	$230 / foot	**$16,800**
Three 12" Gate Valves	$7,450 each	**$22,350**
Three 12" Tee Joints	$2,475 each	**$7,425**
VFD for 75 hp Pump	$7,710	**$7,710**
Addition of Three 2-way Valves (Globe)	$11,630 each	**$34,890**
Inspection and cleaning of cooling coils	$2,000 each	**$10,000**
VFD for 50 hp Pump	$5,420	**$5,420**
VFDs for the 70 hp Fans	$7,710 each	**$15,420**
Engineering	50% of material and labor costs	**$60,000**
Total Cost of Project (excluding asbestos abatement)		**$180,000**

Simple Payback Period = Implementation cost / Cost savings

= $180,000 / $84,460/yr

= 2.2 years

Therefore, the total cost comes out to be $180,000 and with $84,460 as cost savings, the simple payback period will be 2.2 years. The payback period for this recommendation is slightly lower than the original recommendation where the chilled water loop was converted to a primary secondary system. Thus, by adding VFDs to the condenser loop, the cost increases but with significant cost savings the payback period is better.

[7] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

5.2.2: Removal of asbestos

Throughout all implementation tables, the asbestos removal costs were ignored. This recommendation briefly looks at the potential costs and steps needed to tackle this issue. The plant was found to have asbestos in the mechanical room near the chiller pumps. Presence of asbestos in other parts of the plant could not be confirmed and thus will not be considered.

RSMeans provides the key steps for asbestos removal,

1. A contractor with experience in handling asbestos and someone who knows the federal and state regulations for disposal of the product should be hired.
2. An industrial hygienist will create a plan to get rid of asbestos in the plant.
3. When removing asbestos and carrying it outside the plant, the area around the place where asbestos is located needs to be monitored for air quality. A path needs to be designated which will be used to carry it outside the facility.
4. HEPA filters will be installed to ensure maximum efficiency in filtration of any harmful suspended particles.
5. A decontamination zone will be set up for its safe removal which will be inspected by the industrial hygienist.
6. All the asbestos is removed, safely bagged and carried out of the plant for its disposal.
7. The area in contact with asbestos is thoroughly cleaned and the area is declared safe after the entire process.

The following table outlines a rough cost of the entire process. It is by no means an accurate listing of the costs that could be incurred during the actual process. This recommendation aims to outline the process and try to give the reader an idea as to how the costing for the process of removal is done. This implementation cost table was made under the following assumptions,

1. Mechanical room (7,500 square feet) is the only space in the plant with asbestos.
2. Only 200 ft of piping has asbestos for insulation.
3. The entire process can be completed within three days with the help of three trained personnel.
4. About 15-20 cubic feet of asbestos will be removed from the entire space.

Table 32: Implementation costs

Item	Unit Price (materials and labor)[8]	Cost
Asbestos removal planning	$1,325	**$1,325**
2,000 CFM filtration device	$870	**$870**
Personal protective equipment	$500 each	**$1,500**
HEPA vacuum	$1,425 each	**$1,425**
Disposable fiber drums	$19 each	**$95**
Preparation of containment area	$0.33 / square foot	**$2,475**
Adding separation barriers	$5.5 / square foot	**$23,900**
Bulk asbestos removal	$5.6 / linear foot	**$1,120**
OSHA testing	500 / day	**$1,500**
Handling, disposal	$23.3 / drum	**$117**
Cleaning of contaminated zone	$1.03 / square foot	**$7,725**
Total Cost of Project		**$42,052**

[8] Costs from Means' Mechanical Cost Data, 2019 and Grainger's website

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