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GRADO EN INGENIERÍA ELECTROMECÁNICA

Especialidad Mecánica

Design of an improved Blue Waters supercomputer cooling system

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Madrid

Julio 2018

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Design of an improved Blue Waters supercomputer cooling system

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DESIGN OF AN IMPROVED BLUE WATERS SUPERCOMPUTER COOLING SYSTEM

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Entidad colaboradora: Universidad de Illinois, Johnsons Controls.

RESUMEN DEL PROYECTO

1. Introducción

El Centro Nacional de Aplicaciones para la Supercomputación (NCSA), está planeando mejorar y expandir el actual superordenador 'Blue Waters' en los próximos 15 meses, el superordenador está localizado en las instalaciones del Centro Nacional 'Petascale'. Se necesita un sistema de refrigeración sofisticado y eficiente para poder eliminar la gran cantidad de calor generada por el superordenador. El nuevo sistema tendrá requisitos de potencia menores que el actual, esto junto a la capacidad que tendrá el sistema para aceptar temperaturas de entrada de agua mayores, nos dará la oportunidad de modificar el sistema actual de refrigeración con el objetivo de reducir los costes anuales, así como el uso de recursos naturales.

El sistema actual de refrigeración tiene varios problemas. Primero, el sistema actual está sobredimensionado, está pensado para refrigerar el actual ordenador que disipa 11 MW de calor de pico y 6 MW de media, mientras que el nuevo disipa 3.8 MW de máxima. Además del superordenador se diseñará un sistema de refrigeración para el centro de almacenamiento de datos que tendrá distintas características térmicas. El sistema tiene dos formas de refrigeración. La primera es con agua refrigerada del campus, tratada en la planta de refrigeración de Oak Street. El gran problema con este método es que es el más caro de los dos, además el coste del agua ha ido incrementándose año a año. El coste actual es de 11.34 $ por MMBTU. La segunda manera de refrigerar el sistema es con el uso de las torres de refrigeración localizadas en la misma instalación. Este método es más barato y más eficiente que el primero. Aun así, el agua del circuito de la torre necesita ser repuesta y tratada químicamente para evitar daños en el equipo.

El sistema está compuesto por dos circuitos de agua: uno en la zona de la torre de refrigeración y otro en el propio edificio. El circuito situado en la zona del edificio es un circuito de agua de temperatura moderada que lleva agua fría a los ordenadores para expulsar el calor producido por estos. En los meses más calientes se utiliza también agua enfriada en la planta de refrigeración de la ciudad para mantener la temperatura del agua suficientemente baja como para refrigerar correctamente los ordenadores. El circuito de la zona del edificio está conectado con el circuito de la torre a través de unos intercambiadores de calor. En la zona de la torre nos encontramos con tres de torres de refrigeración con dos celdas cada una que utilizan enfriamiento por evaporación para refrigerar el agua que proviene de los ordenadores. El objetivo es maximizar el uso de las torres de refrigeración a lo largo del año y reducir la compra de agua refrigerada.

Además de las posibles modificaciones que podamos hacer a los actuales componentes y equipos del actual sistema, exploraremos otra alternativa llamada ‘BlueStream’. Este es un dispositivo desarrollado por Johnson Controls, se trata de un termosifón que funciona junto a una torre de refrigeración. Este equipo elimina parte del calor producido por el superordenador mediante enfriamiento sensible, lo que ayuda a reducir la cantidad de agua usada en las torres de refrigeración, así como aumenta la posibilidad de aumentar el tiempo de uso de estas a lo largo del año.

2. Metodología

Para optimizar el Sistema de refrigeración analizaremos los costes y uso de recursos de las distintas soluciones propuestas por la propia NCSA, Johnson’s Controls y las propuestas por nosotros. Para poder dar un estudio lo más detallado posible utilizaremos datos históricos del tiempo en Champaign, y para facilitar los cálculos utilizaremos el programa de resolución de sistemas de ecuaciones no lineales EES, que nos facilitará el cálculo de los costes del sistema.

Los costes que tendremos en cuenta los hemos dividido en dos grupos, agua y electricidad. En el primer grupo se encuentran el agua que ha sido evaporada en las torres de refrigeración y necesita ser repuesta, además se incluye el agua que es necesario bombear desde la planta de refrigeración en los meses mas calurosos. Por otro lado, está la electricidad, que engloba la potencia requerida en las bombas, de ambos circuitos, y la potencia necesaria para hacer funcionar tanto el ventilador de las torres de refrigeración como de los TSC.

Por último, también se analizará si el actual equipo es el más eficiente. Se estudiará si las actuales bombas se pueden utilizar aun si decidimos cambiar el caudal de agua del sistema. Además, se analizará si podemos mantener el actual entramado de tuberías manteniendo una velocidad del flujo de agua dentro de los limites recomendados.

3. Resultados

La solución que proponemos tiene un coste anual de 170.000 dólares, lo que supone un ahorro de más de 350.000 dólares respecto a si mantuvieran el actual sistema de refrigeración, con un periodo de recuperación de la inversión inicial de tan solo 1.2 años. Además, implementar esta solución supondría un ahorro de más de 14 millones de litros de agua anuales.

Decidimos refrigerar los dos sistemas, el superordenador y el centro de almacenamiento de datos de manera separada. El sistema 1, el superordenador, esta refrigerado for una torre de refrigeración y dos TSC. Mientras que el sistema 2, el dectro de almacenamiento de datos, esta refrigerado por una torre y el agua enfriada proveniente de la planta.

4. Conclusión

Diseñar un sistema de refrigeración de larga escala con tantos componentes y un complejo entramado de tuberías es un gran desafío. Desde el principio nuestro objetivo ha sido tratar de reducir los costes anuales manteniendo la mayor cantidad del equipamiento actual. Con esto en mente, nuestra recomendación final separamos los dos sistemas a refrigerar, manteniendo todo el equipo actual junto con la compra de dos TSC. Esta solución requiere modificar el entramado de tuberías, pero aporta un beneficio considerable, ya que ahorramos 375.000 dólares anualmente. Además del beneficio económico debemos tener en cuenta también que utilizando esta propuesta la NCSA será capaz de ahorrar más de 14 millones de litros de agua anualmente.

DESIGN OF AN IMPROVED BLUE WATERS SUPERCOMPUTER COOLING SYSTEM

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Collaborating Entity: University of Illinois, Johnsons Controls.

Abstract

1. Introduction

The Blue Waters supercomputing facility, located in the National Petascale Center Facility (NPCF), run by the National Center for Supercomputing Applications (NCSA), is planning on upgrading and expanding its current computing equipment in the near future (about 15 months). A sophisticated and efficient cooling system is required to dispose of the large amount of heat generated by the computers. The new supercomputer system will have lower power requirements than the current one as well as a higher acceptable inlet temperature, providing a great opportunity to modify the cooling system in order to cut both monetary and natural resource costs.

The current Blue Waters cooling system has several issues. For one, the current system requires a large amount of energy to cool the current supercomputer that peaks at 11 MW and idles at around 6 MW, while the new one will peak at 3.8 MW. We have to take into consideration that we also have to cool the data storage center, with different thermal conditions.

The current system has two main systems for cooling. The first system is the use of the campus chilled water loop. The main issue with this system is that it is the more expensive method of cooling. The water is pretreated and precooled from the Oak Street Chiller Plant. The cost of chilled water is based on the capacity of heat transferred to the water and amounts to $11.34 per MMBTU. The second system uses the cooling towers that are located at the Blue Waters facility. This is the less expensive and more efficient method of cooling the supercomputer. However, water in this loop constantly needs to be treated with chemicals to prevent fouling. The water also needs constant replenishing because the cooling towers use evaporative cooling, where water is continuously being evaporated to the atmosphere.

The current system is composed of two main separate water loops: a building-side and a tower-side loop. The building-side loop is a medium temperature cooling loop that supplies water to the supercomputers. Chilled water is bled into the building-side loop during warm months in order to keep the water temperature low enough for computer cooling. This building-side loop is also connected to a tower-side water loop through a series of plate and frame heat exchangers. This tower-side loop contains three 2-cell cooling towers that are able to evaporatively cool the medium temperature cooling water during colder months. The goal is to utilize these cooling towers throughout a larger portion of the year, thus decreasing the cost and waste of purchasing chilled water.

In addition to making modifications to current components to suit the new system, there is an alternative cooling system developed by Johnson Controls called BlueStream. This technology involves adding a device, called a Thermosyphon Controller (TSC), in series with the existing cooling towers. This device allows for sensible cooling (i.e. cooling without phase change) of water before the excess heat is sent to the towers to be evaporatively cooled, thus helping reduce the amount of water lost through evaporation as well as increase the portion of the year cooling towers can be utilized.

2. Methodology

In order to optimize the cooling system, we will analyze the operating cost and the use of resources of the different proposed solutions. For a more detailed analysis we will use historical weather data, and to ease the process we will use the non-linear equation solver, EES.

To facilitate the process, we have divided the costs in two different groups, water and electricity. In the first group we include the water that has been evaporated in the cooling towers and the chilled water that need to be pumped from the chiller plant during the warmer months. On the other hand, we have the electricity, which includes the power necessary to make the pumps work, as well as the power needed by the cooling tower and TSC fans.

Furthermore, we will analyze if the actual equipment is the most efficient. We will study if we can maintain the current pumps even if we decided to change the water flow. We will also study if this can affect the piping, and we will change it if necessary, to maintain a viable flow velocity.

3. Results

The recommended solution has an annual operating cost of $170,000, which means more than $350,000 saving in comparison with maintaining the current system. The payback period of adopting this solution is only of 1.2 years. Also, implementing this solution can save more than 3.5 million of gallons of water.

We decided to cool the two systems, supercomputer and data storage, separately. System 1, the computer, will be run of one cooling tower and two TSC. And system 2 will run of one cooling tower and chilled water.

4. Conclusion

Cooling system design on such a large scale involves many components and piping loops interacting at the same time. Early on it became clear that our goal to achieve an optimal cooling system would best be served by utilizing as many existing pumps, heat exchangers, and cooling towers as possible in order to save on initial costs. We ended up analyzing System 1 and System 2 separately because of the larger load and higher inlet temperature required by System 1. Our recommendation is to run System 1 off of one cooling tower and two TSC units and to run System 2 off of one cooling tower cell and chilled water. This configuration had the shortest payback period of just over 1 year compared to the base case. This solution requires substantial repiping but saves around $375 thousand dollars in annual operating costs making these modifications well worth the investment. Furthermore, our recommended solution reduces the water footprint of the facility make it a more sustainable option.

Acknowledgements

I would like to thank our sponsor Johnson Controls for their technical and financial support throughout the project, especially Mr. Tom Carter for his assistance. I would also like to thank the NCSA for their technical assistance, especially Mr. Mo Rantissi. Finally, I would like to thank my faculty advisor Dr. Emad Jassim and T.A. Conrad Smith for their mentorship throughout the semester.

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Chapter 1: Narrative of the project

1. Introduction and Problem Statement

High-performance computing platforms have reached unprecedented total power and power density levels, and this trend does not seem to stop, thanks to the continuous development of IT technology (increased workloads or ultra-dense manycore chips). Removing the heat of this technological monsters has become a critical issue, as the cooling system usually accounts for a significant percentage, around 40% of the total system's power consumption. Due to the high amount of power requires to operate these systems; traditional cooling methods are economically and environmentally unsustainable.

NCSA is planning on turning off its Blue Waters supercomputer within approximately fifteen months and replacing it with a new computing system. While the bid for this system has not been finalized, we have received initial specifications of the anticipated system. Most importantly, the system will require less power and accept warmer inlet temperatures. These conditions are favorable for the design of a new cooling system that is more cost effective, energy efficient, and environmentally friendly.

The current Blue Waters cooling system has several issues that our senior design team has addressed. For one, the current system requires a large amount of energy to cool the current supercomputer that peaks at 11 MW and idles at around 6 MW. Furthermore, water usage costs have only increased over time. The current system has two main systems for cooling. The first system is the use of the campus chilled water loop. The main issue with this system is that it is the more expensive method of cooling. The water is pretreated and precooled from the Oak Street Chiller Plant. The cost of chilled water is based on the capacity of heat transferred to the water and amounts to $11.34 per MMBTU. The second system uses the cooling towers that are located at the Blue Waters facility. This is the less expensive and more efficient method of cooling the supercomputer. However, water in this loop constantly needs to be treated with chemicals to prevent fouling. The water also needs constant replenishing because the cooling towers use evaporative cooling, where water is continuously being evaporated to the atmosphere. These systems are explained in more detail in the next section.

The current system is oversized and provides colder water (50°F) than needed for the new supercomputer. The new supercomputer will have a cooling system with three temperature needs: high temperature for the water-cooled computer system (32°C), medium temperature for the air-cooled data storage racks, and low temperature for other air-cooled equipment. Thus, innovations to the oversized cooling system can offer large returns in the long run.

We explored modification of the current cooling system for the new supercomputers, as well as the design of a new cooling system using Bluestream technology.

1.1 Project Goals and Objectives

Multiple new system configurations were designed to minimize operation costs along with water and energy usage. We explored new options including ground source cooling and thermosyphon technology from Johnson Controls. After ground source cooling proved to be too expensive as an option as shown previously, integrating TSC units became a focus. We also attempted to utilize as much of the existing system components and piping as possible in order to cut down on costs. Our final deliverables for the project include flow diagrams, general equipment arrangement, life-cycle cost analysis, and a recommendation of the optimal cooling solution.

2. State of the art

2.1 Cooling towers

2.1.1 Introduction

Cooling towers are a type of heat exchanger that provides what is called, 'free cooling'. Hot water enters in contact with air that lowers the temperature of the water. During this process, a small percentage of water is evaporated, which needs to be replaced. It is used in industrial processes where water gets heated up, and goes through the cooling tower, once it is cooled down it gets pumped back to the equipment that needs to be cooled. The applications of cooling towers are very diverse, from traditional HVAC heating and cooling systems to industrial processes or applications, such as petroleum plants or refineries to supercomputers.

We can distinguish between natural and forced draft cooling towers and between crossflow and counterflow cooling towers. The difference between the first two ones is that the forced draft cooling tower needs the help of a fan to get the air out, while the induced draft cooling tower move the air out naturally thank to the shape and height of the tower.

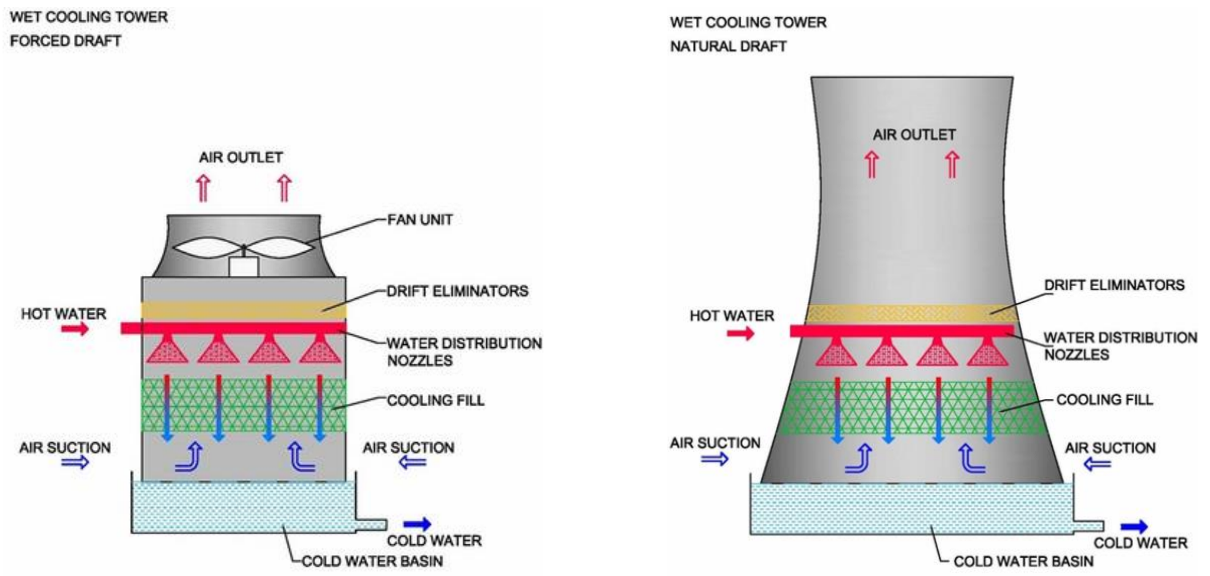

Figure 1. Difference between forced and natural draft cooling towers

The difference between counterflow and crossflow cooling towers is that the in the first ones the air flows upwards, counter to the water flow. While in the crossflow cooling towers the air flows horizontally across the falling water. We can appreciate the difference in the following illustration:

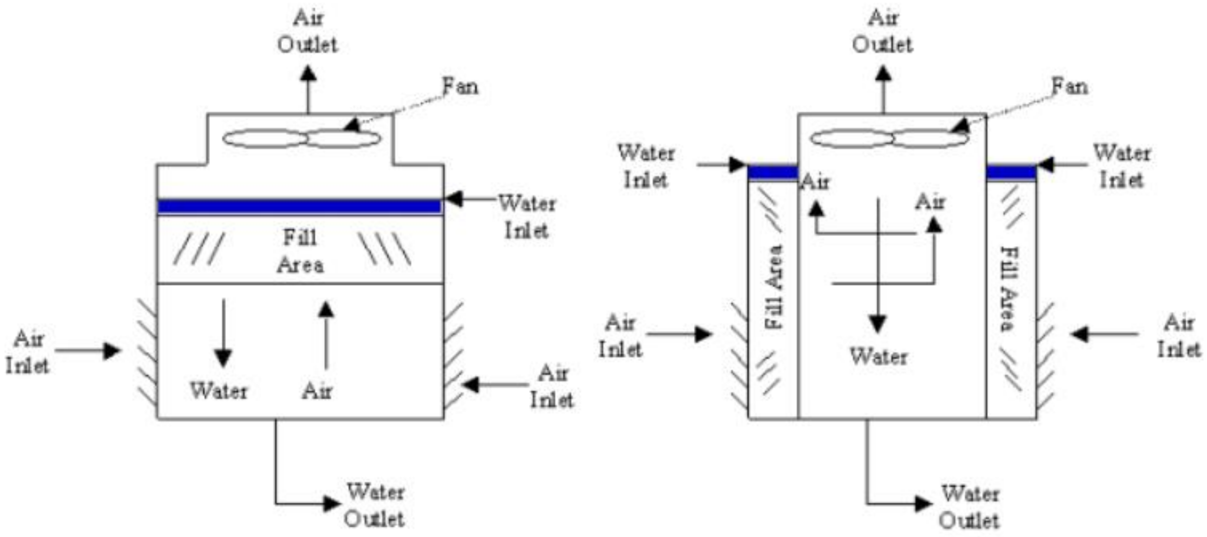

Figure 2. Difference between counterflow and crossflow cooling towers

2.1.2 Experimental approach

The NPCF currently utilizes three (3) Baltimore Aircoil Company (BAC) 31056C cooling towers that reside on top of the north side roof. Each cooling tower consists of two cells—each with a cooling capacity of 1056 refrigeration tons or roughly 3.7 MW. These three (3) existing cooling towers will likely remain in use, and their time of operation throughout the year and function(s) within the cooling loops may be altered to better suit the new system. Given the maximum inlet temperature requirement for the new supercomputer in System 1 at 90°F (32ºC), the cooling towers can be in use to provide free cooling year-round. This assumes that the wet bulb temperature does not exceed 83°F (28ºC) and a 5°F (~3ºC) cooling tower approach. If the design is to serve both Systems 1 and 2, 65°F (18ºC) is required. This reduces the usable time of the cooling towers to roughly 67% of the year. In this case, chilled water would be used for the remaining portion of the year. Separating the systems could allow both 100% of the year for System 1 and 67% of the year for System 2 in order to maximize the usability of the cooling towers. A model for this case is currently being developed.

To model the operation of the cooling tower we received from our sponsor, Johnson Controls, a model that related the flow of water with the range and the difference of enthalpies of the inlet and outlet air.

$$y = -0.0308527x^2 - 0.7898894x + 7.9410324;$$

Eqn. 1

where: $x = \ln\left(\frac{Range}{dh_{appr}}\right);$

$$y = \ln(flow)$$

Eqn. 2

This approach is depicted graphically in Figure 3**.**

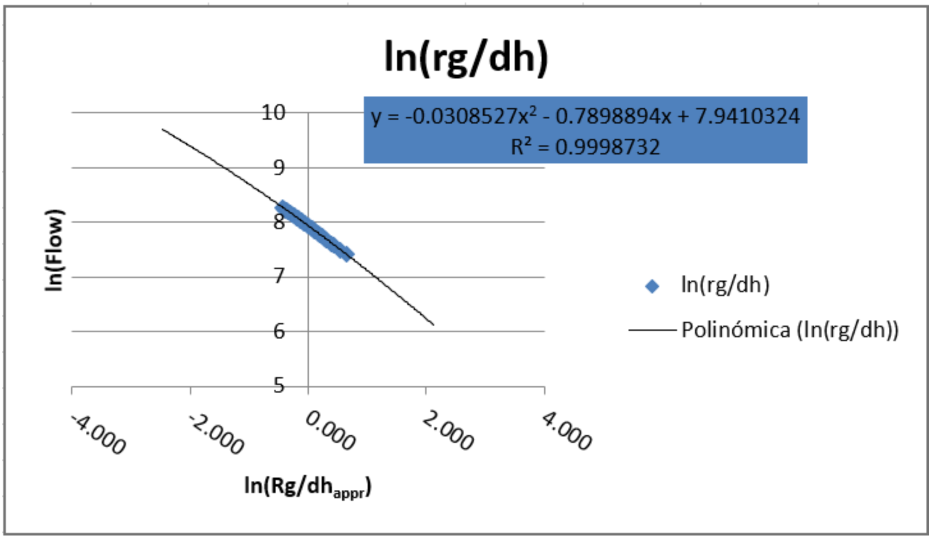

Figure 3. Cooling Tower Model

In our model, water losses are accounted for to have a more detailed cost analysis and a better idea of the water usage. We incorporated equations to calculate evaporative losses (due to the evaporation of water), drift losses (due to water being blown out by the wind) and blowdown losses (water with high mineral concentration that is flush out and replaced by fresh water) in our model.

The evaporative losses are a function of the flow rate, the inlet and outlet temperatures of the cooling tower and the latent heat of vaporization.

$$V_{loss,evap} = V_{cti} * \frac{T_{cti} - T_{cte}}{h_{fg}}$$

Eqn. 3

The drift equation for a BAC 31056 cooling tower is given to be the 0.3% of the total water flow rate:

$$V_{loss,drift} = 0.003 * V_{cti}$$

Eqn. 4

And the blowdown losses are defined as the evaporative losses divided by cycles of concentration (COC) minus 1:

$$V_{loss,bld} = \frac{V_{loss,evap}}{COC - 1}$$

Eqn. 5

The total losses are given by the sum of the evaporative, drift and blowdown losses:

$$V_{loss} = V_{loss,evap} + V_{loss,drift} + V_{loss,bld}$$

Eqn. 6

2.2 Pumps

2.2.1 Introduction

Hydraulic pumps are one of the older inventions of the humankind, and yet one of the most widespread nowadays. A pump is a device which primary function is to move a fluid from one point to another. Pumps can be classified into three different groups depending on the method they used to move the fluid: displacement, direct lift, and gravity.

The first group, the positive displacement pumps, makes the fluid move due to a difference in the volume. They can move a constant flow no matter what the discharge pressure is. Within the positive displacement pumps we can distinguish three different types of pumps:

- Reciprocating-type positive displacement pumps: some examples of this kind of pumps are: plunger pumps or piston pumps.

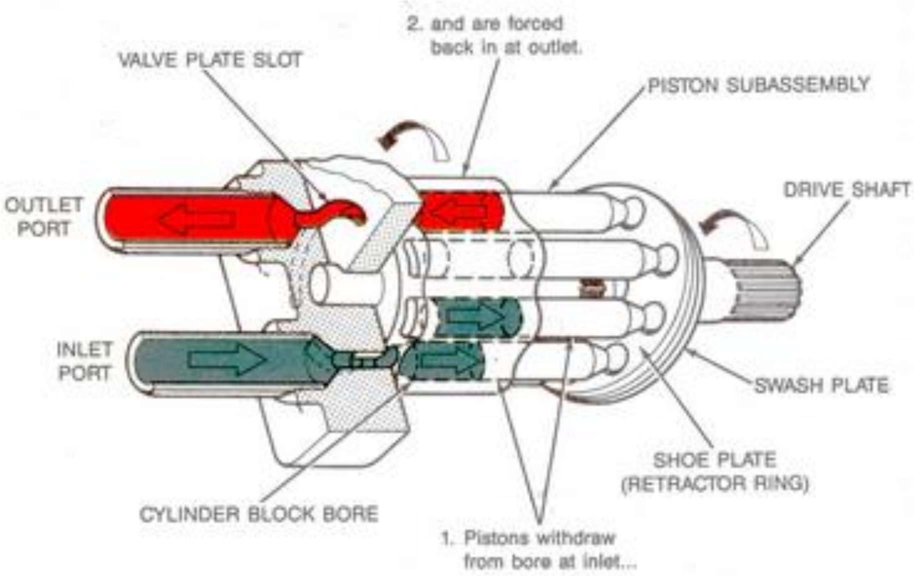

Figure 4. Example of a piston pump

- Rotary-type positive displacement: screw, internal gear, flexible vane or sliding vane pumps.

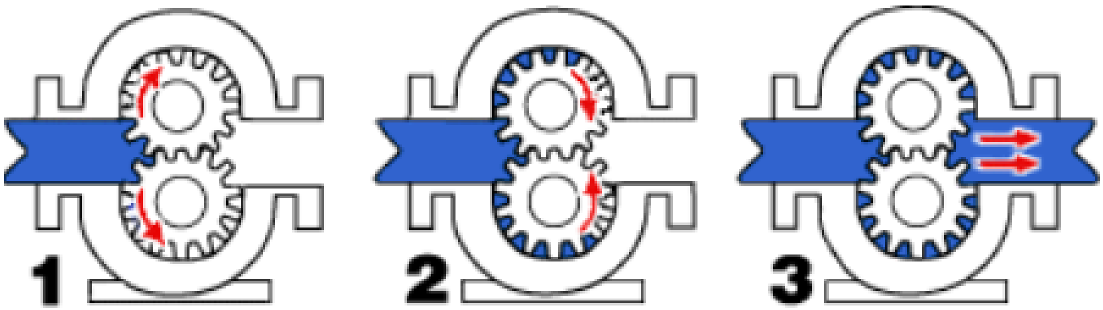

Figure 5. Example of a gear pump

- Linear-type positive displacement: such as rope pumps and chain pumps

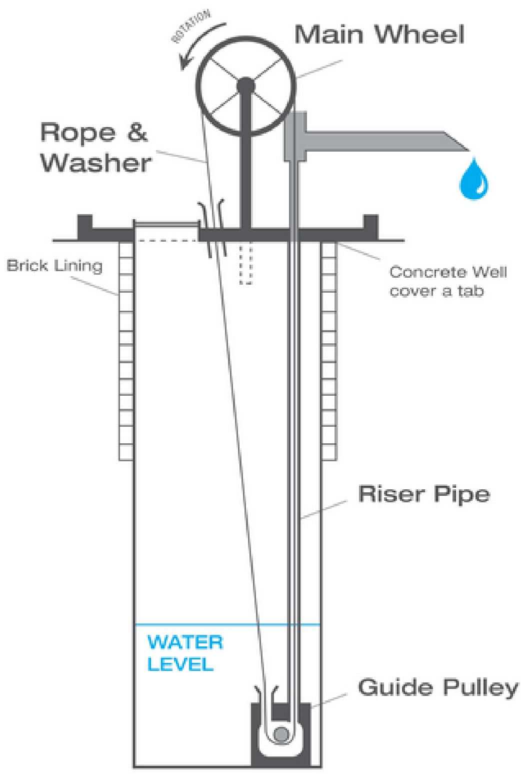

Figure 6. Example of a rope pump

The second type of pumps are the rotodynamic pumps, which will be the one used in this project. This type of pumps adds kinetic energy to the fluid by increasing the flow velocity. This kinetic energy is then transformed to pressure, which can be explained with Bernoulli's equation:

$$\frac{v^2}{2} + gz + \frac{p}{\rho} = constant$$

Eqn. 7

Within the rotodynamic pumps we can distinguish between three different groups:

- Radial-flow pumps: Also known as centrifugal pumps. In these pumps the fluid enters along the axis and is accelerated radially to the exit by the impeller.

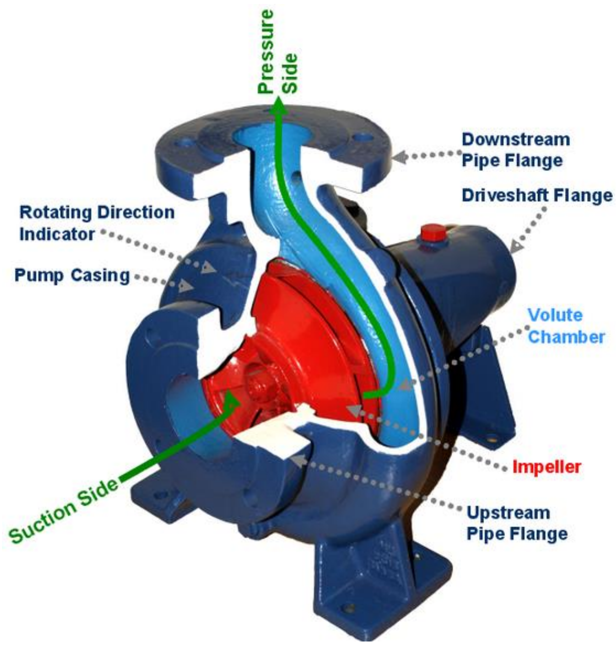

Figure 7. Centrifugal pump

Figure 8. Impeller of a centrifugal pump

- Axial flow pumps: In these cases, the fluid is pushed outward or inward and it moves axially.

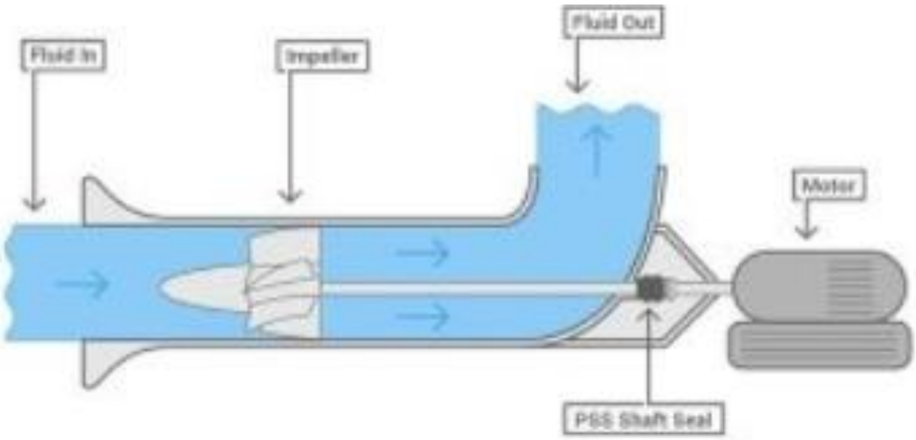

Figure 9. Example of an axial-flow pump

- Diagonal pumps: Or mixed-flow pumps. They are a combination of the last two, the fluid enters the pump axially and it moves radially and axially when going through the impeller.

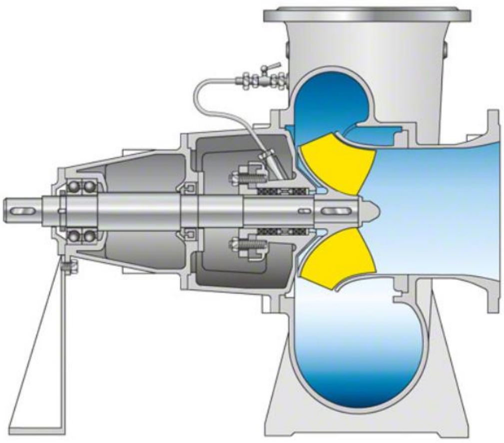

Figure 10. Diagonal pump

The last type of pumps we will discussed are gravity pumps, here the fluid is pumped thanks to the gravitational force. Some example of these kind of pumps are the syphon and the Heron's fountain.

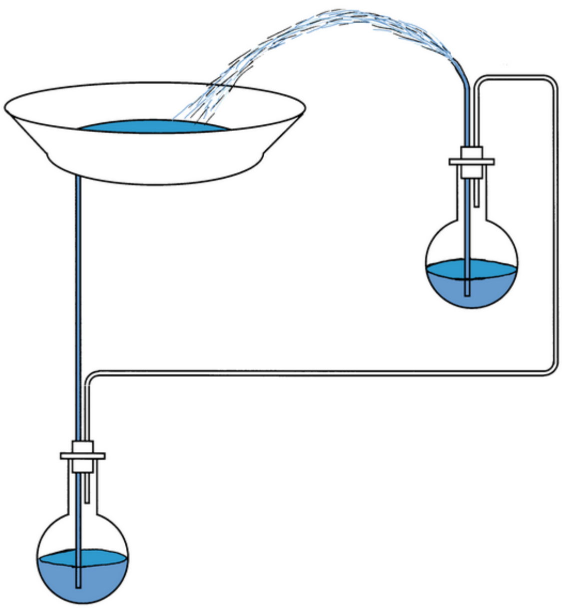

Figure 11. Example of a Heron's fountain

2.2.2 Experimental approach

In the existing system configuration, there are four pumps located on the building-side loop and another four pumps located on the tower-side loop. Ideally, these pumps will be utilized within the new supercomputer system. There are several variables to consider when sizing a pump to a system. Factors include impeller size, impeller speed, pressure losses within the system components (head requirement), motor power, and net positive suction head requirement (NPSHr). Modifications will have to be made to these pumps in order to match the optimal operating point required by the new equipment and flow rates. This is accomplished by creating a pump curve for the system and matching it to a pump performance curve.

Determining the head requirement for the pump involved measuring the pressure differentials across each component of the system. This includes the computer units (CDUs), piping, and heat exchangers. The current piping configuration was assumed to be constant for each new proposed system, as that data is readily available and minimal pipe modifications are needed in System 1 relative to the current setup. The EES model of the existing pipe schematic from the past Blue Waters Team was used as part of the analysis but edited to include the updated flow rate of 1430 gpm. Please note that the volumetric flow rates for the building-side and tower-side loops are shown in Table 1 and were calculated using the following equations:

Water Loop	Flow Rate (m^3/s)	Flow Rate (gpm)
System 1 (Building-side Loop)	0.090	1430
System 1 (Tower-side Loop)	0.090	1430
System 2 (Building-side Loop)	0.021	330
System 2 (Tower-side Loop)	0.090	1430

Table 1. Flow rates for respective water cooling loops

$$Q = \frac{System\ 1\ Power\ Load}{\rho_{water} * c_p * (temp\ diff\ CDUs)} = \frac{dq}{\rho * c_p * (T_{out} - T_{in})} = \frac{3.8\ MW}{(997\ kg/m3) * 4186\ J/kg°C * (42.2 - 32.2)} = 0.09\ m^3/s$$

Eqn. 8

$$Q = \frac{System\ 2\ Power\ Load}{(* c_p * (temp\ diff\ CDUs)} = \frac{dq}{\rho * c_p * (T_{out} - T_{in})} = \frac{0.975\ MW}{(997\ kg/m3) * 4186\ J/kg°C * (29.4 - 18.3)} = 0.021\ m^3/s$$

Eqn. 9

The same procedure was used for the heat exchangers. Relevant equations can be seen within the EES code in the appendix. Data given by the NPCF for the new system included pressure drops across the proposed computers at a specific flow rate. Pressure drop, and flow rate are related by a constant for a single system.

The pressure differential of the heat exchangers, computers, and piping were added together to obtain the total system pressure differential. This value and the specific flow rate given by the NPCF were used to obtain this constant value for the system.

The constant flow rate for the CDUs was calculated as 0.000016 psi/(gpm)^2 using Table 2 and Equation 8. These values were obtained from the manufacturer. Unfortunately, no data was given for the proposed operating conditions of maximum inlet temperature. Therefore, it was assumed that the system behavior will be the same as the given data, thus using a constant *c* value as shown.

	For 4 Cabinets	For 19 Cabinets
Primary Flow Rate (gpm)	232 (0,015 m^3/s)	1102 (0,07 m^3/s)
Temperature In (°C)	25.6	25.6
Temperature Out (°C)	42.9	42.9
Pressure Drop (psi)	19.4 (13,38 bar)	19.4

Table 2. Data given from manufacturer of System 1 CDUs

$$c = \frac{Pressure\ Drop}{(Volumetric\ Flow\ Rate)^2} = \frac{dP}{Q^2} = \frac{19.4\ psi}{(1102\ gpm)^2} = 0.000016$$

Eqn. 10

	For 19 Cabinets
Primary Flow Rate (gpm)	1430 (0,09 m^3/s)
Temperature In (°C)	32.2
Temperature Out (°C)	42.2
Pressure Drop (psi)	32.7 (22,55 bar)

Table 3. Data calculated for desired flow and temperature conditions

$$dP = c * Q^2 = 0.000016 * 1430^2 = 32.7\ psi\ (22.55\ \text{bar})$$

Eqn. 11

The objective was to determine if the pump could meet the requirements of the system at its maximum operation. An inlet/outlet temperature difference of 10°C was used during this calculation, and the equation yielded a pressure difference of 32.7 psi (22.55 bar) across the computer units, Table 3 and Equation 10. Using the pressure drop of the computer, pipe, and heat exchanger along with the constant obtained from given data, an accurate system curve can be obtained for the building-side loop System 1. The tower-side loop pressure drop needs to account for the vertical head it travels pumping water to the top of the cooling tower, roughly ~60 ft (18.3 m.). This, along with piping and heat exchanger pressure drops make up the total differential pressure for the tower-side loop. System curves can be fitted to a pump performance curve based on the operational flow rate and designated pressure. The pressure drop for each component in feet of head are shown below, in Table 4.

	Building-Side, System 1	Tower-Side
Piping Pressure Drop (head-ft)	0.53 (0,016 bar)	0.33 (0,01 bar)
Heat Exchanger Pressure Drop (head-ft)	1.73 (0,052 bar)	1.73
Computer/CT Pressure Drop (head-ft)	77.42 (2.31 bar)	60 (1,79 bar)
Total Pressure Drop (head-ft)	79.8 (2,38 bar)	62 (1,85 bar)
Operating Flow Rate (gpm)	1430 (0,09 m^3/s)	1430

Table 4. Pressure Drop Across System Components

2.3 Heat Exchangers

2.3.1 Introduction

A heat exchanger is a device which transfer heat between two or more fluids. Depending on the heat exchanger we can find single or two-phase fluids, and they maybe separated or in direct contact. We can categorize heat exchangers in two different ways: the first considers the flow configuration, and the second classification is based on the equipment used to manufacture them.

The first classification is based on the flow configuration, here we can distinguish four basic flow configurations:

- Counterflow: The two fluids flow parallel to each other but in opposite directions. This is the most efficient configuration since it can transfer the most heat per unit mass.

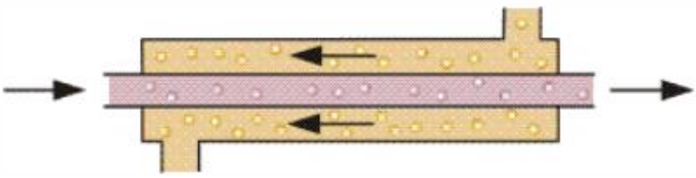

Figure 12**.** Counterflow configuration of a heat exchanger

- Parallel: Both fluids flow parallel to each other with the same direction. This configuration provides more uniform wall temperatures.

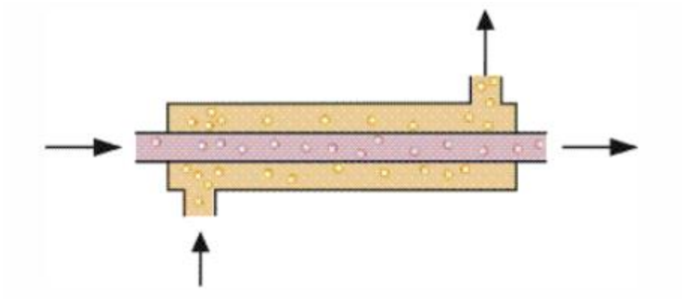

Figure 13. Parallel flow configuration of a heat exchanger

- Crossflow: In this configuration both fluids flow at right angles between each other. These kind of heat exchangers offer an intermediate efficiency between heat transfer and wall temperatures

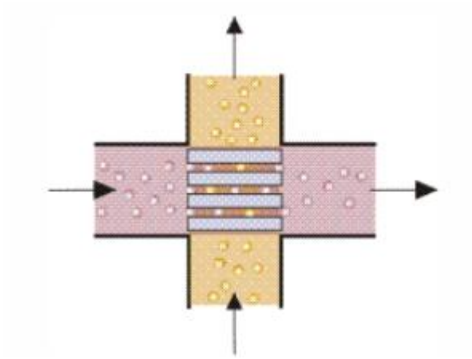

Figure 14. Crossflow configuration of a heat exchanger

- Hybrids: They are combination of both the above. For example, a cross/counter flow.

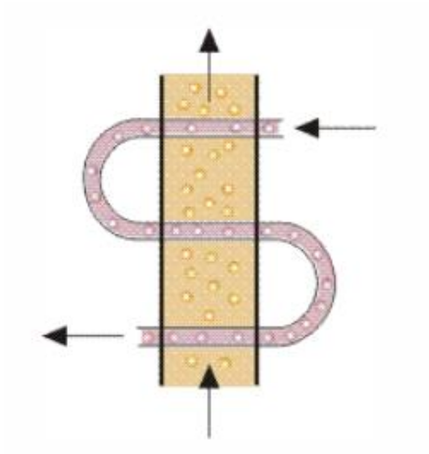

Figure 15. Cross/counter flow configuration of a heat exchanger

The second classification is according to the construction of the heat exchanger. We have two main different groups, recuperative heat exchangers, which have separate flow path for each fluid and regenerative heat exchanger, which has a single flow path.

The main two types of regenerative of heat exchanger are Static and Dynamic. In this type of heat exchangers, the hot fluid flow through a matrix and this heat is released when the cold fluid passes through it.

We have different types of recuperative heat exchangers, which are shown below:

- Shell and tube heat exchangers: as it name says this type of heat exchanger consist on a series of tubes, where the hot/cold fluid flows through and the other fluid runs over the tubes.

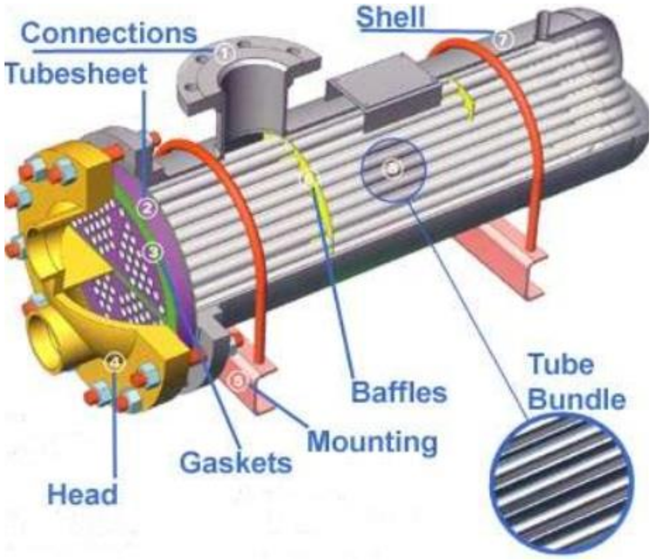

Figure 16. Example of a shell and tube heat exchanger

- Plate heat exchanger: These heat exchangers has several with a large surface area place one over the other. Nowadays they are very common in HVAC applications. One of the main advantages of this type of heat exchanger is that the plates are interchangeable.

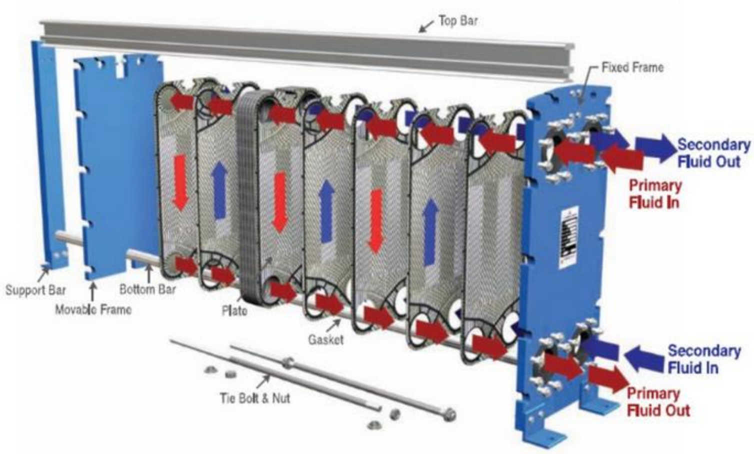

Figure 17. Example of a plate heat exchanger

- Plate and shell heat exchangers: Combines the two previous technologies. This type of heat exchanger offers high heat transfer, high pressure and high operating temperature.

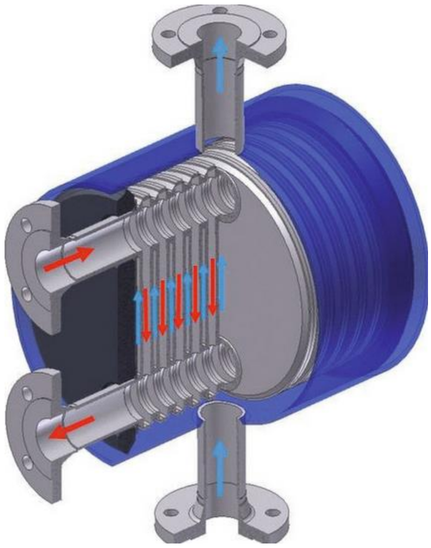

Figure 18. Plate and shell heat exchanger

The ones mention above are the most typical heat exchangers but not the only ones, we also have plate fin heat exchangers, pillow plate heat exchanger or fluid heat exchangers. We have to mention also the direct contact heat exchangers, where both fluids exchange heat in the absence of a wall, the most typical direct contact heat exchanger is the cooling tower, which we talk about before.

2.3.2 Experimental approach

The current EES models account for no change in the existing heat exchanger layout. Several models have been created that consider different applications of the existing heat exchangers for open and closed building-side water cooling loops. These models will be further analyzed later.

The heat exchanger pressure drop was calculated to determine the head requirements of the system in different situations. The pressure drop across the heat exchanger can be defined by:

$$\Delta p_{HX} = -f * \rho * u^2 * \frac{L_{p,HX}}{2 * D_{h,HX}}$$

Eqn. 12

Where Lp is the effective length, u is the flow velocity, Dh is the hydraulic diameter,$\rho$is the density of water and f is the friction factor. The friction factor is found using the following equation:

$$f = m * Re^{-n}$$

Eqn. 13

Where m and n are coefficients provided by the manufacturer, m is equal to 13.5 and n is equal to 0.135. This equation is only valid for turbulent flow. The Reynolds number is defined by the following equation:

$$Re = \frac{D_h * \rho * u}{\mu}$$

Eqn. 14

Where μ is the viscosity of water.

2.4 Ground source cooling

Ground source cooling is an option that we looked into early on in the project cycle. Using Geothermal cooling would involve installing wells and pumps in order to send the heat from the facility down below the surface of the earth. This has a benefit because the earth's temperature is relatively stable underground year-round, as opposed to weather conditions at the surface. The looping in our system needs to be vertical instead of horizontal because with the large thermal load there simply would not be enough space for shallow horizontal piping.

Finally, it is important to choose whether an open-loop or a closed-loop system is preferred. Open-loop systems exchange heat underground with a water source. Open-loop system are around 50% less expensive to install in most cases but require more maintenance and permits must be obtained because water sources are often polluted. A closed-loop system is the more expensive option to install and must be used when no water source is available. However, this system requires less maintenance after installation. For our system, we did not choose an open-loop system because the Mahomet aquifer (as shaded in yellow) is not close enough to the Petascale facility (represented by the purple star) as shown in Figure 19. There is still groundwater present underneath the facility, but it wouldn't be as effective as the aquifer. This makes a closed-loop our preferred choice.

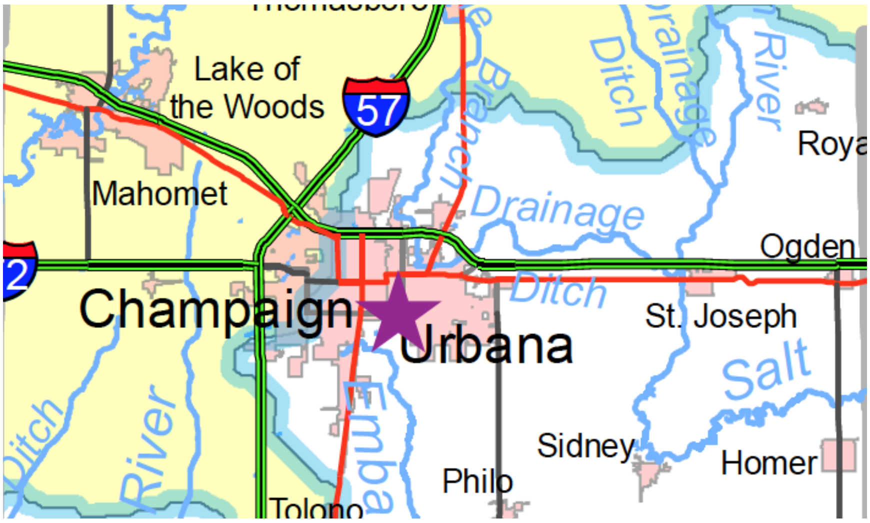

Figure 19. Map of Mahomet aquifer

Using a closed system we based our installation cost on the total heat load of the system in tons. According to the US Department of Energy, when pricing a geothermal system, the vertical looping cost is $1500 per ton. Taking the total heat load of 1365 tons (4.8MW), we estimate an installation cost of $2.05 million dollars. This cost was much larger than other options making it an infeasible choice.

2.5 Piping

The pressure losses in the piping needed to be accounted for to determine the head required for the pumps. We have two different kinds of pressure losses: major losses (due to friction in the pipes) and minor losses (due to different features in the piping system).

The major losses are defined by the following equation:

$$\Delta p_{major} = -\lambda * \frac{\rho * u^2}{2} * \frac{L}{D}$$

Eqn. 15

Where λ is the friction factor, ρ is the density of water, u is the flow velocity, L is the effective length, and D is the diameter of the pipe. We calculated the friction factor using the Colebrook equation:

$$\frac{1}{\sqrt{\lambda}} = -2 * \mathrm{LOG}\left(\frac{2.51}{Re * \sqrt{\lambda}} + \frac{\varepsilon}{3.72 * D}\right)$$

Eqn. 16

ε is the equivalent roughness, in these case is 0.0000675 for commercial steel pipes. The Reynolds number can be found with the following equation:

$$Re = \frac{\rho * u * L}{\mu}$$

Eqn. 17

Minor losses are defined as:

$$\Delta p_{minor} = -\frac{1}{2} * K_{L,Total} * \rho * u^2$$

Eqn. 18

Where $K_{L,Total}$ depends on the number of features of the piping, such as bends or tees. We can calculate its value with Eqn. 13:

$$K_{L,Total} = K_{L,90} * N_{L,90} + K_{L,45} * N_{L,45} + K_{L,tee} * N_{L,tee}$$

Eqn. 19

K is the loss coefficient for each feature and *N* the number of features.

The total pressure losses are the sum of both minor and major losses. The pressure drops were calculated separately for each side of the cooling system, tower and building-side.

2.6 BlueStream TSC

The TSC, pictured in Figure 20, is a dry heat rejection device supplied with piping, valves, controls, instrumentation and a VFD for powering the fans. The TSC utilizes a refrigerant that recirculates naturally to convey the heat from its evaporator to the condenser.

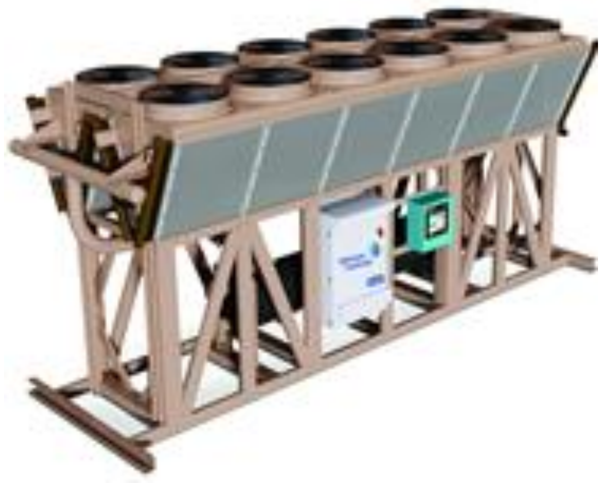

Figure 20. BlueStream Thermosyphon Controller

The evaporator is a tube and shell style flooded evaporator with the water on the tube side and the refrigerant on the shell side. The condenser is an air cooled single pass tube-fin condenser with an enclosed VFD for powering the fans and fan speed control. This control provides two main operating modes: minimum operating cost mode and maximum water saving mode. In the first one the system balances the energy and water uses to yield the lowest system operating cost. With the maximum water saving mode, the system will use the least amount of water by maximizing the dry heat rejection operation.

The TSC is also relatively low maintenance due to the fact that there is no compressor. The fans and heat exchanger are easy to clean as needed, and freeze protection is a standard as a result of low pressure drops and fittings insulated with controlled heaters.

The BlueStream technology can be applied in new or existing facilities working together with a cooling tower, they can provide up to 30-80% of savings in water annually.

The rejection capacity of the TSC at three different flow rates: 600, 800, and 1000 gpm (0.038, 0.05, 0.063 m^3/s) is given by Figure 21. Depending on the application, each TSC can save up to 5 million gallons of water annually. Each TSC can provide up to 386 refrigeration tons of cooling capacity with 95ºF (35ºC) entering water and 85ºF (29,4ºC) leaving water at a 40ºF (4,44ºC) ambient dry bulb temperature.

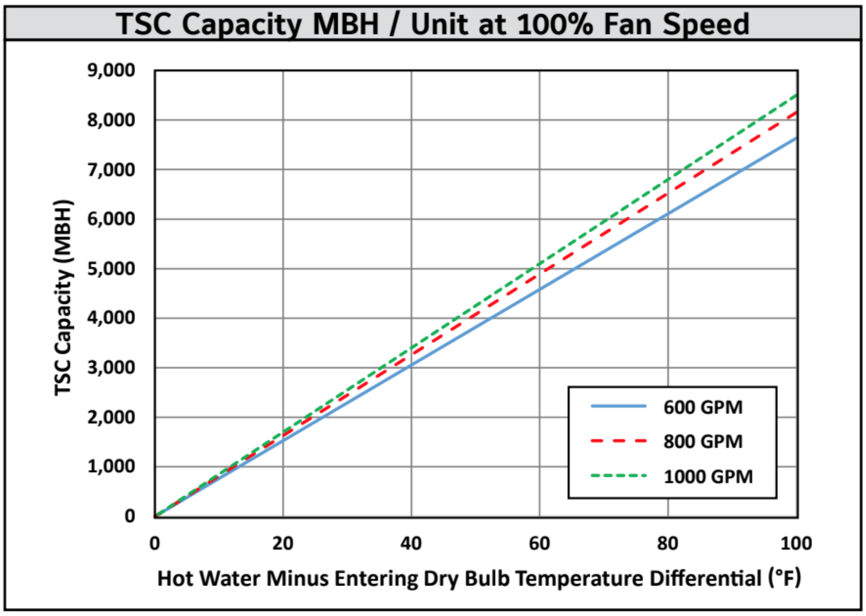

Figure 21. Heat rejection capacity of the TSC.

In order to minimize water consumption for the cooling process, implementation of the BlueStream TSC was analyzed and considered. We propose adding 1-2 TSC units to the building-side loop to decrease water usage. With two TSC(s) installed in the building-side loop, it is estimated that the cooling towers would not need to be in operation for dry bulb temperatures below about 28°F (-2,22ºC). Otherwise, the TSC(s) will aid in cooling the water as much as possible before interaction with the cooling tower heat exchange.

Chapter 2: Experimental procedure

1. Introduction

In this chapter we will develop how we obtain our results and we will explain in detail the procedure we followed. We have chosen EES to facilitate the calculations.

We will use weather data and all the information provided by our sponsors to model five different solution to the cooling system and we will recommend the one with the lowest operating cost and the one with higher water savings.

2. EES modeling

The team designed a model for the five proposed solutions of the cooling system using *Engineering Equation Solver* (EES) software. EES uses the Newton-Raphson method to solve algebraic equations. Also, EES has the ability to compute output tables from a parametric table of input values, allowing us to use hourly weather data in order to give a more detailed cost analysis.

In modeling the system, we assumed constant properties of water in the entire system. We modeled both sides, tower and building, separately with the heat exchanger as the common element. The tower-side loop was modeled as a closed-loop where the temperature changes in the cooling tower were calculated according to Equation 1. The total losses due to evaporation, drift and blowdown are explained in the corresponding section of the cooling towers. We neglected the temperature rise across the pumps, since it was revealed to be insignificant. The pumps must meet the head requirements in order to compensate for the pressure losses in the cooling tower, piping and heat exchanger. The TSC cooling capacity was modeled with the data provided by our sponsor.

In the building-side loop we have to distinguished between the two systems, each system was modeled separately in each case. Both systems work in a closed-loop, where the water flow rate is constant. The benefit of this configuration is that the pressure losses are constant through the year. This simplified the selection of the pumps since the head requirement will be the same.

The pressure losses in the piping were calculated as the sum of the major (due to friction) and minor (due to features in the piping) losses. The number of bends, tees and other features were obtained through a detailed analysis of the building schematic.

For a more detailed cost analysis we used Excel files to analyze the data obtained from the parametric tables. For more detailed information, refer to the EES code in the Appendix.

3. Proposed Solutions

3.1 Proposed Solution 1

In the first proposed solution we maintain the current configuration of the system, shown in Figure 22. The cooling tower can handle the load of System 1 during the entire year without the necessity of bleeding in chilled water. Only one pump is needed in each side to maintain the 1430 gpm (0,09 m^3/s) flow. System 2 runs solely on chilled water.

The annual operating cost of this solution is the highest, more than $540,000 a year due to the high cost of chilled water. We will use this solution as a reference in order to compare the annual cost of the other solutions.

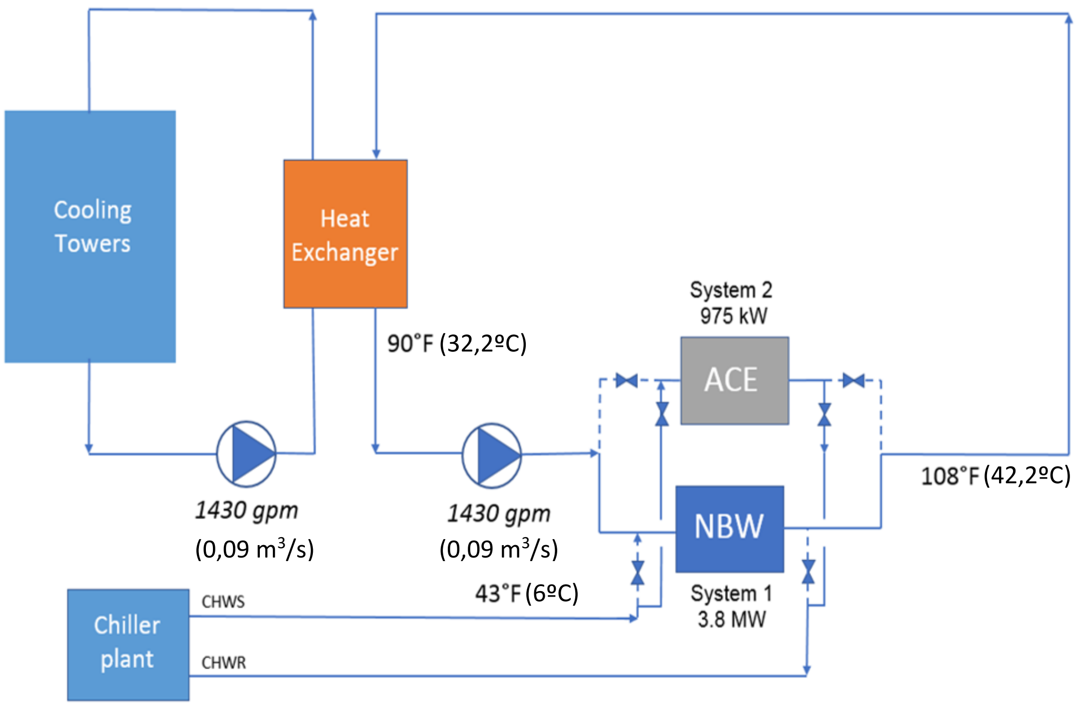

Figure 22. Proposed solution #1 – Base Case

3.2 Proposed Solution 2

Proposed Solution 2 can be seen in Figure 23. The difference between this solution and the previous one is the addition of two TSC's in System 1. The reason why we used two TSC's in System 1 can be seen in Table 5, where it is the combination with the lowest payback period.

SYSTEM 1	Cooling Tower Water Cost ($)	Combined Electric ($)	Annual Operating cost ($)	Annual water usage (millions of gallons/liters)	Payback Period (TSC base + installation)
CT	95,580	23,691	119,271	14.96 (56,8)	-
CT + 1 TSC	66,470	30,541	97,011	9.1 (34,5)	6.7
CT + 2 TSC	35,150	34,190	69,340	5.85 (22,1)	6.0
CT + 3 TSC	20,827	34,753	55,580	1.4 (5,3)	7.1

Table 5. Payback period of the installation of 1,2 or 3 TSC's in System 1

With this configuration we are able to save $50,000 a year thanks to the addition of the TSC's, the reason behind is that the TSC's can handle the load on their own 15% of the year. Besides reducing our annual operating cost, the two TSC's provides significant savings on water, with this configuration we can save 9 million of gallons of water each year comparing to Proposed Solution 1.

The main disadvantage of this solution is the high cost of the chilled water. In System 2, we dedicated 85% of our annual operating cost to pay for the chilled water.

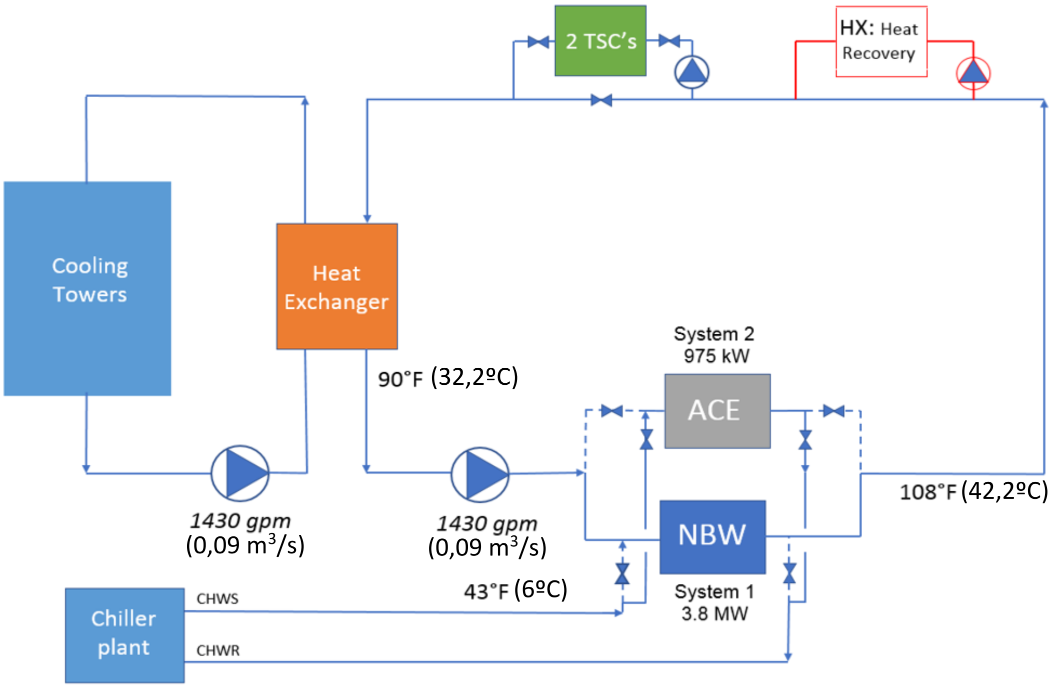

Figure 23. Proposed solution #2

3.3 Proposed Solution 3

In order to reduce the high cost of chilled water in System 2 we decided to add a TSC in System 2, maintaining the previous configuration in System 1, where the cooling tower handle the load with the two TSC's. The reason why we only add one in System 2 are the flow requirements of the TSC, the minimum flow going through a TSC must be higher than 200 gpm (0,013 m^3/s).

With this configuration the annual operating cost is reduced by $250,000. But piping modifications are required in order to implement this solution. We need to repurpose one heat exchanger to separate the loop of the chilled water from the System 2 loop. The addition of the heat exchanger mitigates the pressure drop of the chilled water that is been bled into the current loop. It also helps maintaining the water clean, which minimizes the maintenance of the TSC. This solution is shown in Figure 24.

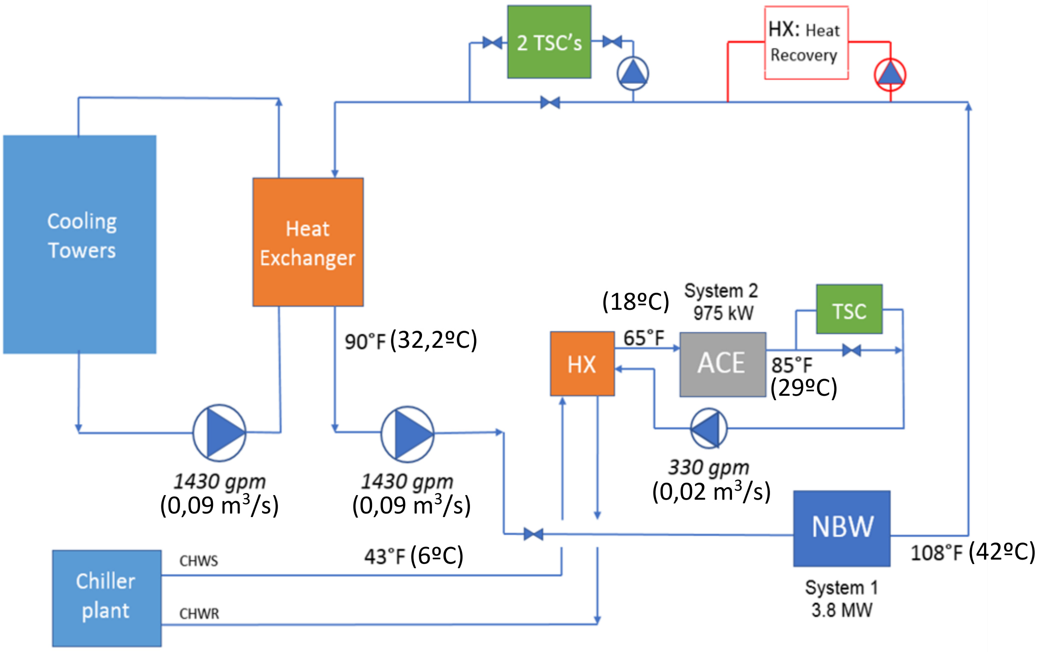

Figure 24. Proposed solution #3

3.4 Proposed Solution 4

Proposed Solution 4 can be seen in Figure 25. This solution is different than all previous solutions in the sense that there is separation between System 1 and 2 via the cooling tower loops. This is important in a feasibility standpoint because separation of the systems cooling tower side maximizes the use of the available equipment currently installed in the NPSF. Available equipment used for cooling includes but is not limited to the cooling towers, heat exchangers, and pumps.

As shown in the diagram, System 1 uses a cooling tower and two (2) TSCs. This setup for the System 1 building-side loop has been refined such that it maximizes water savings while, at the same time, minimizing annual operational costs and payback period. In this design, Heat Exchanger 2 (HX 2) is connected to the chilled water line. In the current setup for Blue Waters, chilled water is bled into the building-side loop. This is not ideal because issues of pressure difference occur in the pipes when trying bleed in the chilled water. Using HX 2 to separate the two loops mitigates this issue. As stated previously, the cooling tower and two TSCs can handle the 3.8 MW load of System 1 and inlet temperature requirement of 90°F (32,2ºC) all year-round based on the EES model and average hourly weather data for Champaign, Illinois. Therefore, use of the chilled water heat exchange will be for emergency cases only—such as, cooling component failure or rare large deviations from average hourly weather data.

System 2 building-side loop interacts with a cooling tower via a heat exchanger. Since, the cooling tower can only handle the entire 975 kW load for a little more than half the year, based on EES modeling and average hourly weather data, the loop proceeds to flow through Heat Exchanger 4 (HX 4) where it interacts with the separated chilled water loop. Only enough chilled water is used in order to meet the 65°F (18,3ºC) inlet temperature requirement for the storage equipment that makes up System 2.

Proposed Solution #4 leads to one of the lowest annual operating cost of $170,903 with more than $400,000 in annual chilled water savings solely from utilizing a cooling tower cell to help cool the load. Furthermore, now that the dirty chilled water loop does not bleed into the building-side loops for each system, the result are closed building-side loops. Therefore, the quality of the water flowing through the computers and storage can be controlled. Good quality water can also result in better heat exchange as well.

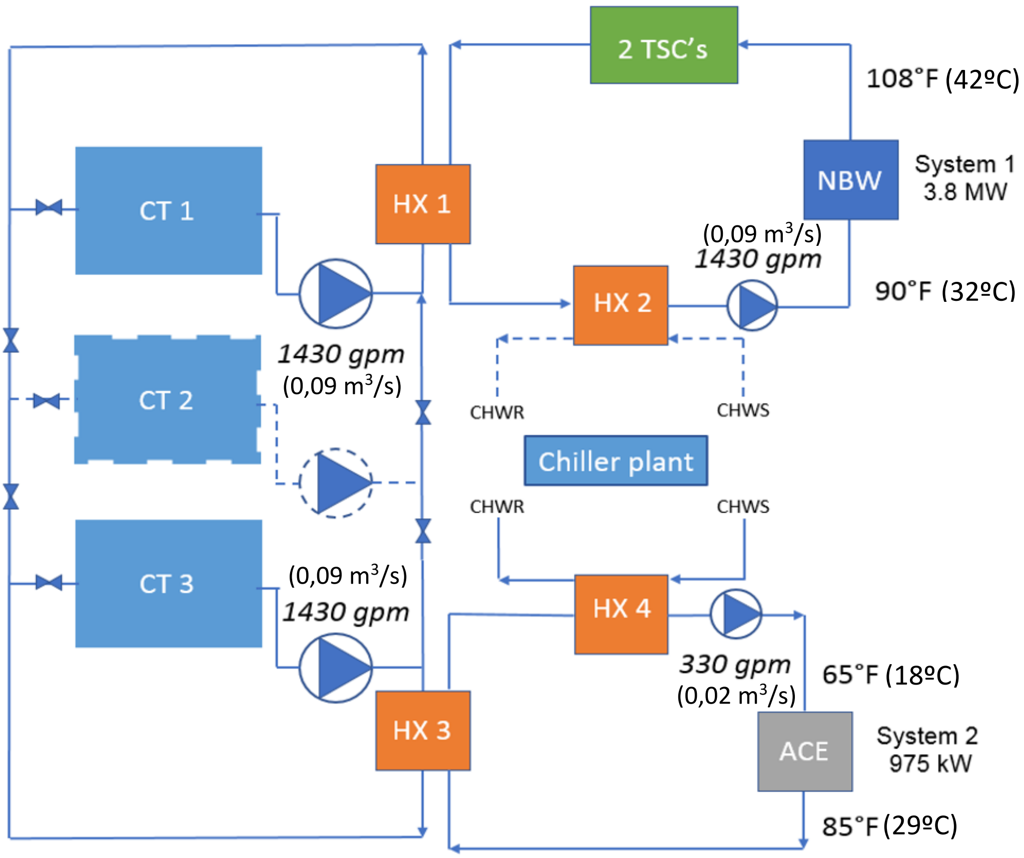

Figure 25. Proposed solution #4

3.5 Proposed Solution 5

Proposed Solution 5 can be seen in Figure 26. The design of this solution is the same as Solution #4 except for the addition of a TSC in series with the building-side loop before the heat exchange with the cooling towers. Similar pipe modification costs to Solution #4 would be incurred with this design. The TSC in System 2 saves even more water leading to more of an environmental impact. The annual operating cost for this design is the lowest at $169,223. Even though it has the lowest annual operating cost, it does not have the lowest payback period as a result of the upfront costs that would be incurred by the additional TSC.

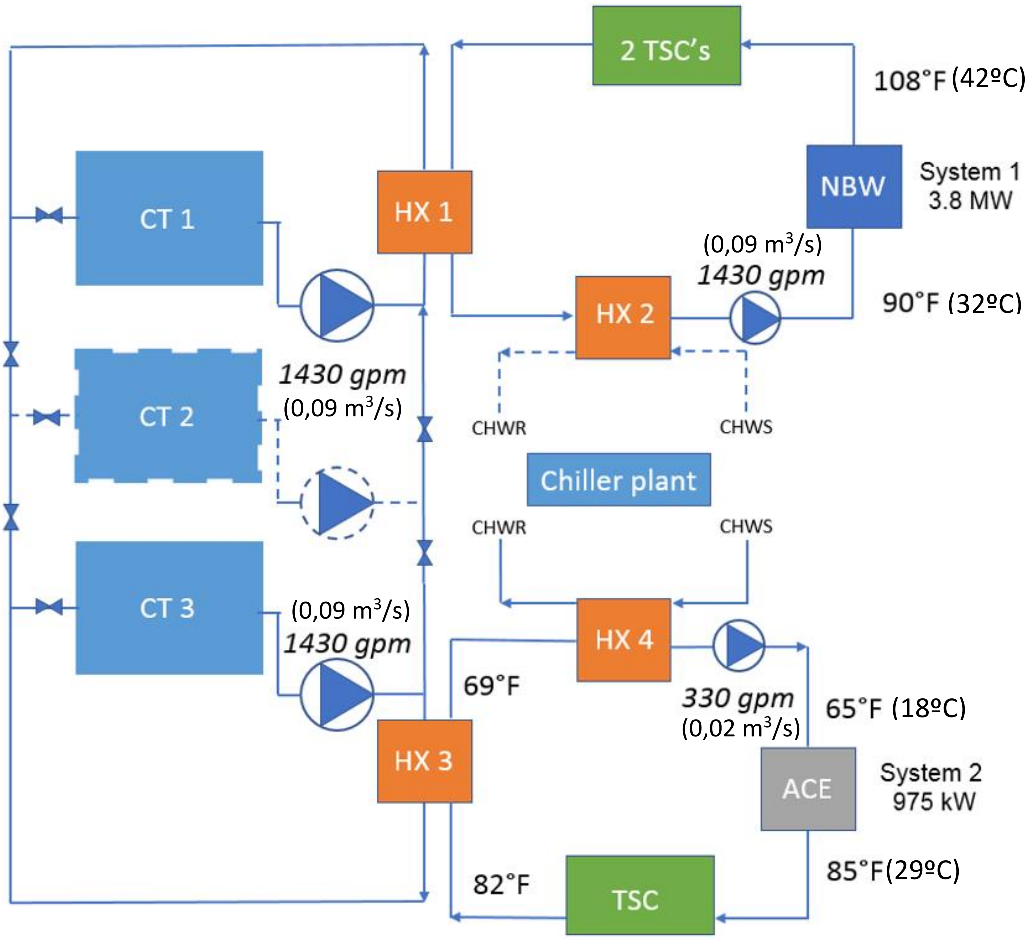

Figure 26. Proposed solution #5

3.6 Summary of Operation Costs

When comparing the five proposed solutions, there are a multitude of factors to take into account. These factors are summarized in Table 6, below. The main two columns to be aware of are "*Annual Operating Cost*" and "*Payback Period (years)*." The annual operating cost is a summation of the costs for chilled water, cooling tower make-up water, and electricity. Electricity cost is due to operating the pumps, as well as the fans for the cooling towers and TSC units.

Solution 1 gives the simulated base case with no optimization. As previously discussed, the optimal number of TSC units for System 1 was determined to be 2. In Solution 2, these are added to the system, though only ~$50,000 in annual savings are realized. This is because most of the operating cost is due to System 2 running purely on the expensive chilled water loop. Once a single TSC is added to System 2, the operating cost drops over $300,000 annually from the base case. Solutions 3, 4 and 5 explore using either a TSC or cooling tower cell (or combination thereof) for System 2. Using both a cooling tower cell and a TSC unit with System 2 leads to the lowest annual operating cost. However, this additional TSC would come at a financial cost that outweighs the ~$1,000 in annual operational savings.

Evaluating the payback period also helps greatly in determining the feasibility of an investment. There are many factors that go into calculating a full payback period, including accounting for inflation, interest rates, etc. For the purposes of this project, a simple payback period was calculated using the annual savings, the capital investment for the the TSC units and their installation, and a rough estimate of repiping costs associated with each solution. The payback period is based off the base case meaning that it is the additional time needed after base case modifications have been made.

Based on the calculated annual operating costs and projected simple payback period, we recommend that NCSA implement our Proposed Solution 4 for the cooling system of the New Blue Waters.

Solution	Cooling Equipment	Annual Chilled Water Cost	Annual Cooling Tower Water Cost	Combined Electric	Annual Operating Cost	Est. Pipe Modification Costs	Annual water usage (gallons)	Annual water savings (gallons)	Payback Period (years)
1	**System 1:** CT **System 2:** CHW	$424,100	$ 95,580	$ 23,691	543,371	$ 288,300	14,500,000	-	-
2	**System 1:** CT + 2 TSC's **System 2:** CHW	$424,100	$ 35,150	$ 34,190	493,440	$ 288,300	5,850,000	8,650,000	6.0
3	**System 1:** CT + 2 TSC's **System 2:** CHW + 1 TSC	$150,400	$ 35,150	$ 47,283	232,833	$ 407,340	5,850,000	8,650,000	1.8
4	**System 1:** CT + 2 TSC's **System 2:** CHW + 1 CT Cell	$ 57,905	$ 59,610	$ 53,388	170,903	$ 426,250	10,744,000	3,756,000	1.2
5	**System 1:** CT + 2 TSC's **System 2:** CHW + 1 CT Cell + 1 TSC	$ 45,636	$ 59,457	$ 64,128	169,221	$ 426,250	10,744,000	3,756,000	1.6

Table 6. Summary of operation costs among proposed solutions

4. Budget

No expenses were incurred during the completion of this project. EES software was free and accessible to all students. All travel involved local site visits.

Chapter 3: Results

1. Recommended Solution

1.1 Detailed Diagram of Recommended Solution

Illustrated in Figure 28 is a detailed diagram of the recommended solution. As shown in the upper left of the figure, the water that returns from System 1 computers will be at 108°F (42,2ºC) where it then exchanges heat with two TSCs that will sensibly cool the water down a few degrees, depending on the outdoor dry bulb temperature. Next, the water will exchange heat with Heat Exchanger 1 (HX 1) that is connected on one side to the System 1 cooling tower loop. At this point, the water will have been cooled to the necessary 90°F (32,2ºC) inlet temperature. As noted above, the cooling tower and 2 TSCs should meet the full load year-round. Therefore, connection to HX 2 with the chilled water loop would only be used as an emergency cooling form.

The return water from System 2 will exchange heat with Heat Exchanger 3 (HX 3) that is connected to the System 2 cooling tower loop. When the full load cannot be met by the cooling tower loop, chilled water is used to cool the water down to the necessary 65°F (18,3ºC) via HX 4.

In the System 1 cooling tower loop, both cells of Cooling Tower 1 (CT 1), 1 sump (SUMP 1), and 1 pump are used. In the System 2 cooling tower loop, one cell of Cooling Tower 3 (CT 3), 1 sump (SUMP 3), and 1 pump are used.

Any components in Figure 28 that are shown in gray or with dashed lines represent redundancy among the cooling systems. For example, there is a 2*n* redundancy in the cooling tower loop. That means that there is a back cell for each cell in use. The pumps also exhibit a 2*n* redundancy as there a backup pump for each one in use. The sumps, however, have an *n*+1 redundancy since there is only one back up sump for the total number of sumps in use.

A display of pipe dimensions is also included. Any lines that are shown in blue represent existing piping that does not need to be replaced. All lines in red represent new piping that would need to be installed.

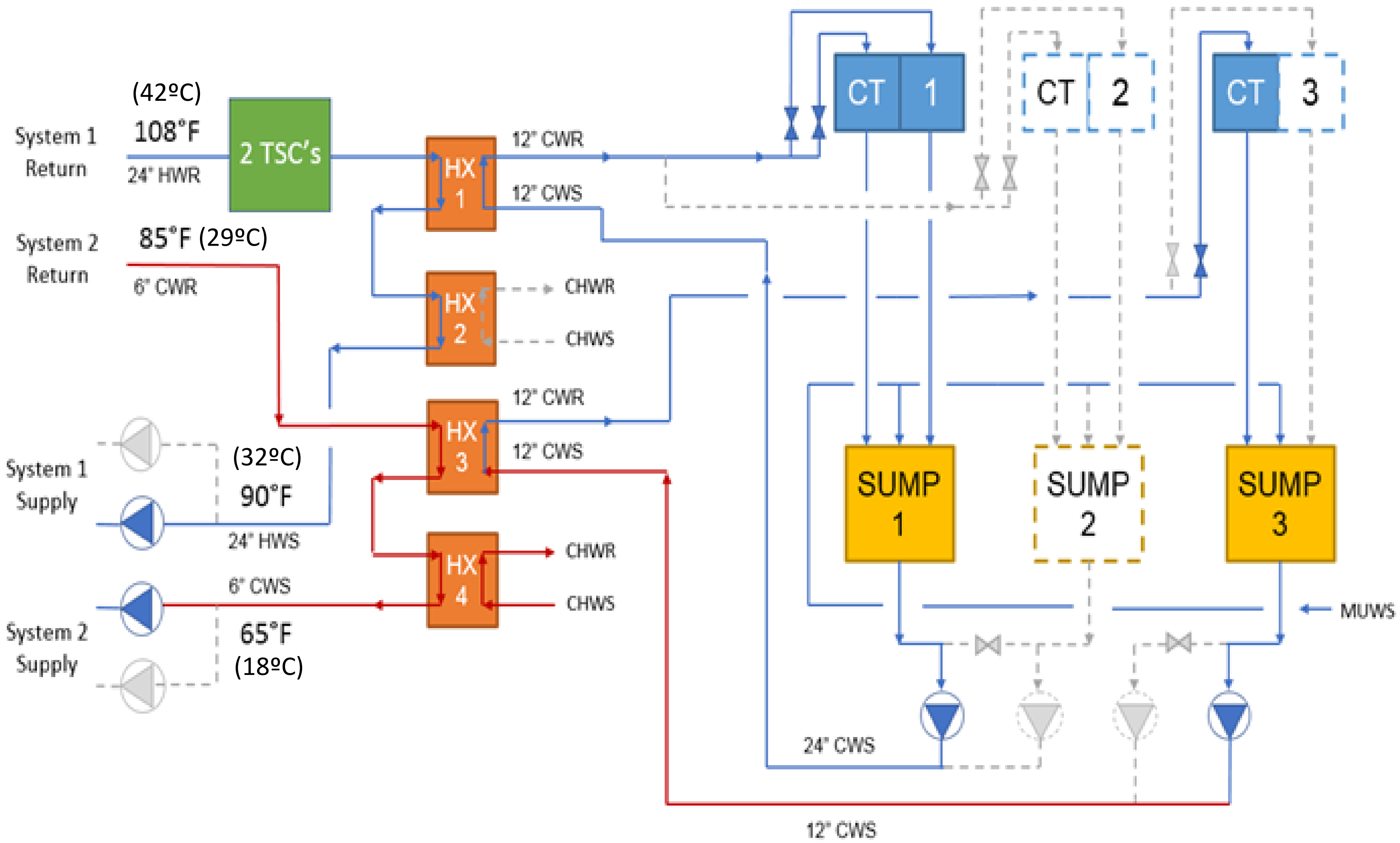

Figure 27. Detailed diagram of recommended solution (solution #4)

1.2 Location of TSCs

The TSC units are vital components of our recommended system. That being said, placement of the TSC units required significant consideration due to their large size and limited space around the current Petascale facility. After discussion with the NCSA, we decided to place the TSC units on the north side of the building as seen in Figure 29. This spot is ideal because of its proximity to the mechanical room. Integrating the TSC units into the current piping system will be simpler with the TSC's as close to the pump and piping equipment currently housed in the mechanical room. The space available to the north of the building is around 40' x 60' and each TSC is sized around 30' x 8.5'. This will allow for placement of the two TSC units with room to spare.

Figure 28. TSC installation location

1.3 Piping Requirements

Based on the guidelines received from the NCSA, the preferred velocity through the pipes should be between 4-7 ft/s (1-3 m/s) to prevent cavitation and air pockets. In our recommended solution a flow rate is selected of around 1430 gpm (0,09 m^3/s) for the cooling tower loop and System 1 and around 330 gpm (0,02 m^3/s) for System 2. We used basic fluid dynamic analysis in equation 14 to see what the velocity would be in the largest current pipes at Blue Waters (24" in diameter). We found that in the largest pipes the velocity would drop to around 1 ft/s (0,3 m/s) meaning that the pipes are slightly oversized for our proposed flow rate. To solve this, we found optimal pipe sizes for the tower and building loops as shown in Table 7. We assumed that new piping implemented would be at these proposed sizes and existing piping would be left in place due to the large expense of replacing all the piping at the facility. By utilizing smaller piping with the new system additions the flow rate will be close to the required 4 ft/s (1,2 m/s) and the flow rate can be increased in the oversized pipe to periodically flush out air from the system when required.

$$1430\,\frac{Gal}{min} * 134\,\frac{ft^3}{Gal} * \frac{1}{60}\,\frac{min}{sec} * \frac{1}{\pi*(1^2)}\,\frac{1}{ft^2} = 1.017\,\frac{ft}{sec}$$

Eqn. 20

	Flow Rate (gpm)	Pipe Size (ID)
Cooling Tower Side	1430 (0,09 m^3/s)	12” (30,5 cm)
Building-Side System 1	1430 (0,09 m^3/s)	18” (45,72 cm)
Building-Side System 2	333 (0,02 m^3/s)	6” (15,24 cm)

Table 7. Optimal pipe sizing

To find the costs for piping we used RSMeans data from 2017. Because the new supercomputer equipment does not have a specific placement location yet, all piping lengths are estimates. We estimated piping length from mechanical piping drawings provided by the NCSA. The length of pipe needed could change depending on how the final system is placed in the facility. We divided the piping cost into three components of insulation, hangers, and the pipes themselves. The costs we used are included in Table 8 and include the price per part as well as labor to install. An additional 50% was added to the sum of costs in order to allow for a conservative estimate. Overall piping costs for each proposed solution can be found in the summary of operation costs section under Table 6.

Pipe Diameter	PVC Piping	30mm Pipe Insulation	Clevis Hangers/Supports
18” (45,72 cm)	$325 per ft. (1070$ per m.)	$12.85 per ft. (42$ per m.)	$111 each
12” (30,48 cm)	$141 per ft. (463$ per m.)	$10.85 per ft. (36$ per m.)	$35 each
6” (15,24 cm)	$63 per ft. (207$ per m.)	$7.80 per ft. (26$ per m.)	$20 each

Table 8. Estimated piping costs

1.4 Pump Modifications

The existing pumps within the system should be sufficient to run both building-side and tower-side loops. With a total of eight pumps available for usage (4 per loop), this proposal only requires the use of three. One will be sufficient for the building-side loop at a maintained flow rate of 1430 gpm. (0,09 m^3/s). Two will be used for the cooling towers since the recommended solution includes two separate tower-side loops. There is not enough available data on System 2 to size a pump for that specific loop. Further information on pressure drop across the two heat exchangers will be necessary before performing the calculations.

In order to meet the new requirements of the system, the impeller speed and diameter must be modified. Each impeller speed setting has its own performance curve to go along with it. Currently, the pump impellers are operating at 1770 rpm, a speed which exceeds the needs of all loops. The catalogue specific to the existing pumps was found online which included all performances curves for various impeller speeds. It was determined that an impeller speed of 1180 rpm yielded the optimal curve. The impeller was sized according to the system curve’s position on the performance curve. The operating points of building and tower-side pumps are shown in Figure 30 and Figure 31 respectively. Details on the head and impellers are contained in Table 9 and Table 10.

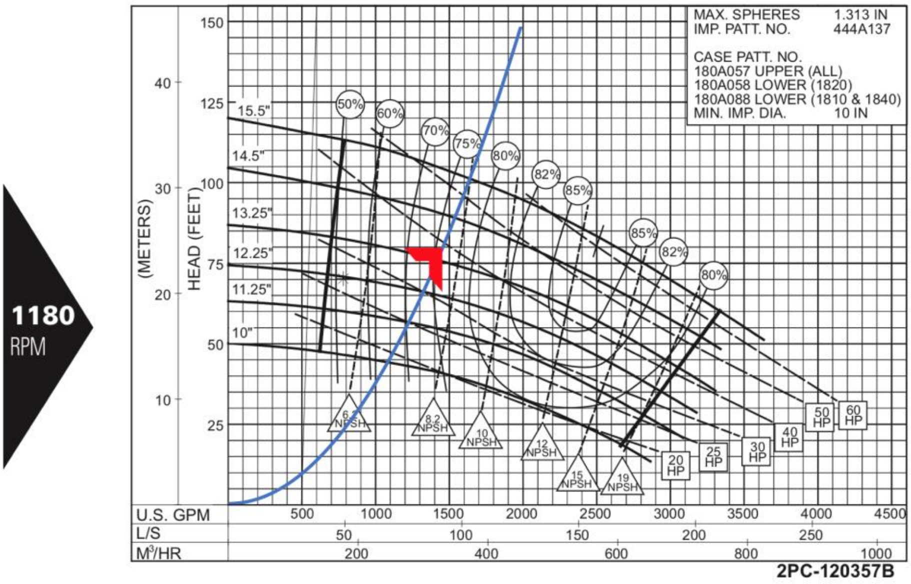

Figure 29. Building-side system requirement curve along with the existing pump performance curve at 1180 rpm

	Old	New
Impeller Size (in)	11.31 (28,73 cm)	13.25 (33,66 cm)
Impeller Speed (rpm)	1775	1180
Head (ft-head)	120 (3,6 bar)	**79.8 (2,38 bar)**
Flow Rate (gpm)	2315 (0,146 m^3/s)	**1430 (0,09 m^3/s)**

Table 9. Pump information for Building-Side

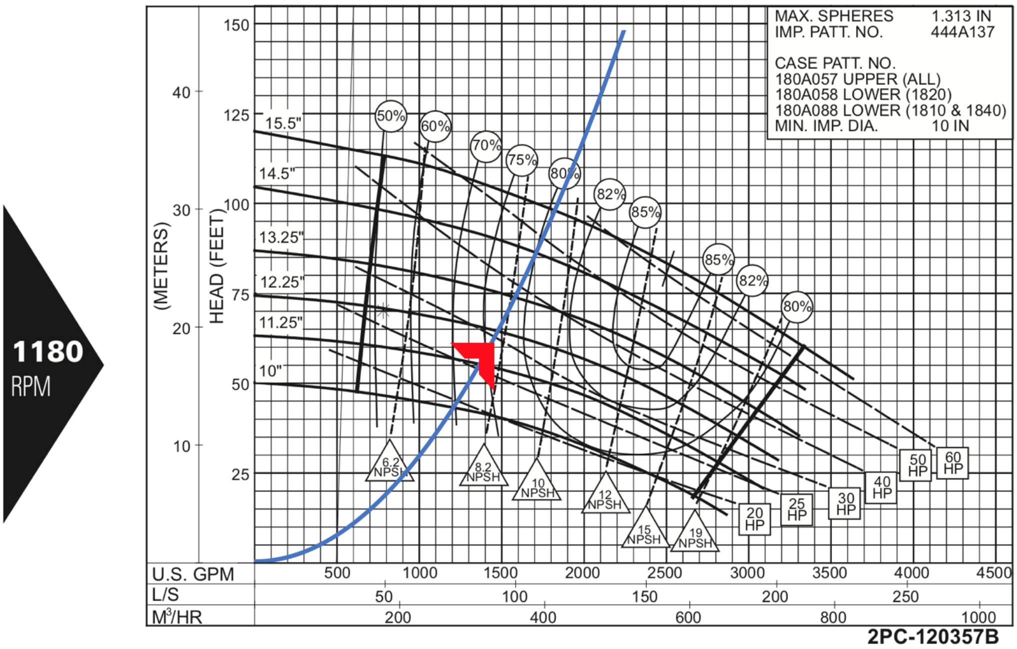

Figure 30. Both tower-side system requirement curves along with the existing pump performance curve at 1180 rpm

	Old	New
Impeller Size (in)	13.375 (33,97 cm)	12.25 (31,11 cm)
Impeller Speed (rpm)	1775	1180
Head (ft-head)	125 (3,74 bar)	**62 (1,85 bar)**
Flow Rate (gpm)	10000 (0,063 m^3/s)	**1430 (0,09 m^3/s)**

Table 10. Pump Information for Tower-Side

1.5 Sustainability and Water Savings

In addition to our recommended solution being more cost-effective, it is also more sustainable than the base case. Water is the most valuable resource on this planet and in today's age more and more efforts are being made to reduce overall water usage. The Mahomet Aquifer specifically is an important resource in central Illinois as it serves over half million people each day [8]. Water costs across the country are on the rise, and Champaign, IL is no exception to this trend. Therefore, by implementing a system that uses less water, we are not only helping the environment we are also creating a more successful solution down the road. The sustainability in our system is represented in both make-up water and chilled water usage reductions on an annual basis.

In our recommended solution we have introduced a cooling tower to not only System 1, but also System 2 as well. All these cooling towers would usually result in an increase in make-up water needed to replace what is lost to evaporation. However, because our larger System 1 load involves 2 TSC's, we will save make-up water over the base case. These TSC units ease the load on the cooling towers throughout most of the year. Our simulations show an annual savings in water of 3.75 million gallons per year (14 million liters) with our recommended system, from 14.5 million (55 million liters) with the base case down to 10.75 million (40 million liter). This is the amount of water that 30 average american households use in an entire year.

Our water savings are not restricted solely to make-up water. We will also save in chilled water use as well. In fact the savings are even more drastic with chilled water than with make-up water. This is due to the introduction of a cooling tower in the System 2 loop so that the system is no longer solely served by chilled water. From the base case we are saving around 25,000 MMBTU per year in chilled water. To put this in perspective, our recommended solution uses less than 15% of the amount of chilled water from the base case. One of the largest buildings on our campus, the Illini Union, uses around this much chilled water each year illustrating that these savings are substantial.

Chapter 4: Conclusion

1. Conclusion

Cooling system design on such a large scale involves many components and piping loops interacting at the same time. Early on it became clear that our goal to achieve an optimal cooling system would best be served by utilizing as many existing pumps, heat exchangers, and cooling towers as possible in order to save on initial costs. We ended up analyzing System 1 and System 2 separately because of the larger load and higher inlet temperature required by System 1. Our recommendation is to run System 1 off of one cooling tower and two TSC units and to run System 2 off of one cooling tower cell and chilled water. This configuration had the shortest payback period of just over 1 year compared to the base case. This solution requires substantial repiping but saves around $375 thousand dollars in annual operating costs making these modifications well worth the investment. Furthermore, our recommended solution reduces the water footprint of the facility make it a more sustainable option.

Appendix 1: EES Code

Case 1 + pressure losses

```
{Cooling Tower Loop System 1 Only}

V_dot_twr=1460 {10000 gpm in m3/s}

N_ct=1 {number of cooling towers}

V_dot_cti=V_dot_twr/N_ct

V_dot_bld=1460


T_bldi=108 {inlet tower temperature in degrees F}

T_blde=85 {inlet tower temperature in degrees F}

T_wb=20 {wet bulb temperature in F}

T_db=52 {dry bulb temperature in F}

c_p=4.187 {kj/kgK}

Rv=461.9 {J/kgK}

Rho=1.225 {kg/m^3}

Range=T_cti-T_cte {Range of the cooling tower}

pt=101.325 {kPa}

CoC=3.5

cost_elect=.0782 {$/kW*hr}

hours_yr=8760

eff=1.0


{outlet temperature of cooling towers}


Q_HX=V_dot_bld*c_p*(T_bldi-T_blde)

V_dot_twr*c_p*Range=Q_HX


{Cooling tower model}

T_int=T_cte+(0.3*Range)


h_int=enthalpy(AirH2O,T=((T_int-32)/1.8),B=((T_int-32)/1.8),P=pt)*.43
```

h_out=**enthalpy**(***AirH2O***,***T***=((T_wb-32)/1.8),***B***=((T_wb-32)/1.8),***P***=pt)*.43

x1=Range/(h_int-h_out)

x=**ln**(x1)

y=**ln**(V_dot_cti)

y=(-0.0308527*(x^2))-(0.7898894*x)+7.9410324

" Cooling Tower Exit"

{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at cooling tower exit is difference between inlet and losses. }

V_dot_loss = (0.003*(V_dot_cti*(6.31e-5)) +(V_dot_cti*(6.31e-5))* c_p * (((T_cti-32)/1.8) - ((T_cte-32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }

V_dot_cte = V_dot_cti - V_dot_loss

" Makeup"

{ Assume makeup water at the same temperature as the cold water. }

V_dot_mkp = V_dot_loss

T_mkp = T_cti

"Blowdown"

V_dot_blow=V_dot_mkp/(CoC-1)

{Cooling Tower Cost System 1 Only}

m_yr=525600

per_yr=1

cooling_tower_time=m_yr*per_yr

{Hourly Cost}

m_hr=60

cooling_tower_time_hourly=m_hr

vol_wat_hr=cooling_tower_time_hourly*(V_dot_loss+V_dot_blow)

makeup_water_cost=.00418 {$/gal}

cooling_tower_water_cost_hr=makeup_water_cost*vol_wat_hr

vol_wat_yr=cooling_tower_time*(V_dot_loss+V_dot_blow)

cooling_tower_water_cost=makeup_water_cost*vol_wat_yr/1000 {in thousands of dollars}

{Fan Power}

atmos=pt {kPa}

Max_HP=75 {hp}

h_intatmos=**enthalpy**(***AirH2O***,***T***=(T_int-32)/1.8,***B***=(T_int-32)/1.8,***P***=pt)*.43

h_WBatmos=**enthalpy**(***AirH2O***,***T***=(T_wb-32)/1.8,***B***=(T_wb-32)/1.8,***P***=pt)*.43

dhatmos=h_intatmos-h_WBatmos

coeff1=**exp**(-0.0308527***ln**((x1)^2) - 0.7898894***ln**(x1) + 7.9410324)

coeff2=(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos))^0.667

GPMBase=(coeff1*(dhatmos/(h_int-h_out))*coeff2)*3

TwrPct = V_dot_twr/GPMBase

TwrACFM =
V_dot_twr*(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))*
TwrPct^0.946

TwrkW =0.746*Max_HP * (TwrACFM / V_dot_twr)^3 *
(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))

N_cooling_fans=2

hours_cooling_yr=hours_yr

cooling_fan_cost=TwrkW*N_cooling_fans*hours_cooling_yr*cost_elect*.746/(eff*1000)

{Pump Costs}

Building_Pump_kW=0.746*25

Tower_Pump_kW=0.746*40

Building_Pump_Cost=Building_Pump_kW*cost_elect*hours_yr/((.75)*1000) {75% efficiency}

Tower_pump_Cost=Tower_Pump_kW*cost_elect*hours_yr/((.75)*1000)*per_yr

{System 2 - CHW ONLY}

Energy_Sys2 = 975 {in kW}

CHW_cost = 0.04965 {$/kW-hr}

Cost_CHW = CHW_cost*hours_yr*Energy_Sys2/1000 {in thousands of dollars}

{Total Cost}

Total_COST = Tower_pump_Cost+cooling_fan_cost+cooling_tower_water_cost+Building_Pump_Cost+Cost_CHW

{HXR Data}

{Heat Transfer Data}

{Geometry}

rho_water=998

HXR_no = 4 { # of heat exchangers }

N_plates = 659 { # of plates per exchanger }

N_cp = (N_plates-1)/2 { # of channels for fluid flow }

d_HXR = 1.9E-3 { spacing between plates }

t_HXR = 0.4E-3 { thickness of each plate }

W_HXR = 0.57 { width of each plate }

H_HXR = 1.478 { height of each plate }

Dh_HXR = 2*d_HXR { hydraulic diameter of flow area }

A_HXR = H_HXR * W_HXR * (N_plates-1) { total heat transfer area }

k_plate = 16.2 { thermal conductivity of each plate }

{ Fluid Properties}

T_cti_HX=(T_cti-32)/1.8

T_cte_HX=(T_cte-32)/1.8

T_bldi_HX=(T_bldi-32)/1.8

T_blde_HX=(T_blde-32)/1.8

V_dot_twr_HX=V_dot_twr*6.309E-5

V_dot_bld_HX=V_dot_bld*6.309E-5

```
T_ctavg = (T_cti_HX + T_cte_HX)/2 { Average temperature of tower-side stream }

T_bwavg = (T_bldi_HX + T_blde_HX)/2 { Average temperature of computer-side stream }

{ Properties of water for both streams are taken to be constant over the temperature range of interest.
}

k_ctx = 600E-3 { thermal conductivity of tower-side stream,
W/mK }

mu_ctx = 1225E-6 { dynamic viscosity of tower-side stream,
Ns/m^2}

Pr_ctx = 8 { Prandtl number of tower-side stream }

k_bwx = 600E-3 { thermal conductivity of computer-side stream,
W/mK }

mu_bwx = 1225E-6 { dynamic viscosity of computer-side stream,
Ns/m^2}

Pr_bwx = 8 { Prandtl number of computer-side stream }

mu = 1225E-6 { dynamic viscosity of water,
Ns/m^2}


" Heat Transfer Correlations"

{ Calculates flow velocity (m/s), Reynolds number (using hydraulic diameter), Nusselt number (for
chevron-style counter-flow heat exchangers), and heat transfer coefficient for both tower-side and
computer-side streams. }

u_ctx = (V_dot_twr_HX/HXR_no) / (N_cp * d_HXR*W_HXR)

Re_ctx = (Dh_HXR*rho_water*u_ctx)/mu_ctx

Nus_ctx = 0.317*(Re_ctx)^(0.65) * (Pr_ctx)^(0.4)

h_ctx = (Nus_ctx*k_ctx)/Dh_HXR

u_bwx = (V_dot_bld_HX/HXR_no) / (N_cp * d_HXR*W_HXR)

Re_bwx = (Dh_HXR*rho_water*u_bwx)/mu_bwx

Nus_bwx = 0.317*(Re_bwx)^(0.65) * (Pr_bwx)^(0.4)

h_bwx = (Nus_bwx*k_bwx)/Dh_HXR

{ Calculates overall heat transfer coefficient U (W/m^2K) using plate resistance and convective
resistance for each stream. }

1/U = 1/h_bwx + 1/h_ctx + t_HXR/k_plate
```

"HXR Pressure Drop Data"

{ Computes tower-side and computer-side pressure drop in heat exchangers. }

Lp_HXR = 1.193 { length of flow channel }

m_HXR = 13.5 { fitted constant }

n_HXR = 0.135 { fitted constant }

{ Heat exchanger friction factor and pressure drop calculations for tower-side and computer-side streams, respectively. }

f_ctx = m_HXR * (Re_ctx)^(-n_HXR)

DELTAP_ct_HXR = - (f_ctx*rho_water*u_ctx^2*Lp_HXR/(2*Dh_HXR)) / 1000 {kPa}

f_bwx = m_HXR * (Re_bwx)^(-n_HXR)

DELTAP_bw_HXR = - (f_bwx*rho_water*u_bwx^2*Lp_HXR/(2*Dh_HXR)) / 1000 {kPa}

"Pipe Pressure Drop Data"

{ This section of code models the pressure drop in the pipes in the tower-side loop}

"General Considerations"

rough = 0.0675E-3 { inside pipe roughness for commercial steel pipes }

xi_90 = 0.3 { loss coefficient for 90 degree bends }

xi_45 = 0.2 { loss coefficient for 45 degree bends }

xi_tee = 1.0 { loss coefficient for T bends }

"Blue Waters Side"

{ Pipe pressure drop for the computer-side loop is computed as the sum of major and minor losses for each size of pipe. }

DELTAP_bw_pipes =
DELTAP_maj_24+DELTAP_maj_18+DELTAP_maj_16+DELTAP_maj_12+DELTAP_maj_8+DELTAP_min_24+
DELTAP_min_18+DELTAP_min_16+DELTAP_min_12+DELTAP_min_8

"Pipe Geometry"

Dia_24 = 0.6096 { diameter (in meters) of 24 inch diameter pipe }

Dia_18 = 0.4572 { diameter (in meters) of 18 inch diameter pipe }

Dia_16 = 0.4064 { diameter (in meters) of 16 inch diameter pipe }

Dia_12 = 0.3048 { diameter (in meters) of 12 inch diameter pipe }

Dia_8 = 0.2032 { diameter (in meters) of 8 inch diameter pipe }

L_24 =103 { length (in meters) of 24 inch diameter pipe }

L_18 = 74.4 { length (in meters) of 18 inch diameter pipe }

L_16 = 79.25 { length (in meters) of 16 inch diameter pipe }

L_12 = 4.9 { length (in meters) of 12 inch diameter pipe }

L_8 = 6.7 { length (in meters) of 8 inch diameter pipe }

Bend90_24 = 10 { # of 90 degree bends in 24 inch diameter pipe }

Bend90_18 = 1 { # of 90 degree bends in 18 inch diameter pipe }

Bend90_16 = 0 { # of 90 degree bends in 16 inch diameter pipe }

Bend90_12 = 1 { # of 90 degree bends in 12 inch diameter pipe }

Bend90_8 = 2 { # of 90 degree bends in 8 inch diameter pipe }

Bend45_24 = 2 { # of 45 degree bends in 24 inch diameter pipe }

Bend45_18 = 0 { # of 45 degree bends in 18 inch diameter pipe }

Bend45_16 = 0 { # of 45 degree bends in 16 inch diameter pipe }

Bend45_12 = 0 { # of 45 degree bends in 12 inch diameter pipe }

Bend45_8 = 2 { # of 45 degree bends in 8 inch diameter pipe }

Tees_24 = 5 { # of T junctions in 24 inch diameter pipe }

Tees_18 = 0 { # of T junctions in 18 inch diameter pipe }

Tees_16 = 0 { # of T junctions in 16 inch diameter pipe }

Tees_12 = 2 { # of T junctions in 12 inch diameter pipe }

Tees_8 = 2 { # of T junctions in 8 inch diameter pipe }

"Flow Conditions"

V_dot_24 =V_dot_bld*6.309E-5 { Flow rate through 24 inch diameter pipe }

V_dot_18 = V_dot_bld*6.309E-5 { Flow rate through 18 inch diameter pipe }

V_dot_16 = V_dot_bld*6.309E-5/2 { Flow rate through 16 inch diameter pipe }

V_dot_12 = V_dot_bld*6.309E-5/4 { Flow rate through 12 inch diameter pipe }

```
V_dot_8 = V_dot_bld*6.309E-5/4 { Flow rate through 8 inch diameter pipe }


"24 Inch Pipe Analysis"

{ Computes minor loss coefficient, stream velocity, friction factor (lambda), Reynolds number, and major
and minor pressure losses for 24 inch diameter pipe. }

xi_24 = xi_90 * Bend90_24 + xi_45 * Bend45_24 + xi_tee * Tees_24

u_24 = V_dot_24/(pi*(Dia_24/2)^2)

1/sqrt(lambda_24) = -2 * log10( 2.51/(Re_24*sqrt(lambda_24)) + (rough/Dia_24)/3.72 )

Re_24 = (rho_water*u_24*L_24)/mu

DELTAP_maj_24 = - ( lambda_24*(L_24/Dia_24)*((rho_water*u_24^2)/2) ) / 1000

DELTAP_min_24 = - xi_24 * 0.5*rho_water*u_24^2 /1000


"18 Inch Pipe Analysis"

{ Computes minor loss coefficient, stream velocity, friction factor (lambda), Reynolds number, and major
and minor pressure losses for 18 inch diameter pipe. }

xi_18 = xi_90 * Bend90_18 + xi_45 * Bend45_18 + xi_tee * Tees_18

u_18 = V_dot_18/(pi*(Dia_18/2)^2)

1/sqrt(lambda_18) = -2 * log10( 2.51/(Re_18*sqrt(lambda_18)) + (rough/Dia_18)/3.72 )

Re_18 = (rho_water*u_18*L_18)/mu

DELTAP_maj_18 = - ( lambda_18*(L_18/Dia_18)*((rho_water*u_18^2)/2) ) / 1000

DELTAP_min_18 = - xi_18 * 0.5*rho_water*u_18^2 / 1000


"16 Inch Pipe Analysis"

{ Computes minor loss coefficient, stream velocity, friction factor (lambda), Reynolds number, and major
and minor pressure losses for 16 inch diameter pipe. }

xi_16 = xi_90 * Bend90_16 + xi_45 * Bend45_16 + xi_tee * Tees_16

u_16 = V_dot_16/(pi*(Dia_16/2)^2)

1/sqrt(lambda_16) = -2 * log10( 2.51/(Re_16*sqrt(lambda_16)) + (rough/Dia_16)/3.72 )

Re_16 = (rho_water*u_16*L_16)/mu

DELTAP_maj_16 = - ( lambda_16*(L_16/Dia_16)*((rho_water*u_16^2)/2) ) / 1000

DELTAP_min_16 = - xi_16 * 0.5*rho_water*u_16^2 / 1000


"12 Inch Pipe Analysis"
```

```
{ Computes minor loss coefficient, stream velocity, friction factor (lambda), Reynolds number, and major
and minor pressure losses for 12 inch diameter pipe. }

xi_12 = xi_90 * Bend90_12 + xi_45 * Bend45_12 + xi_tee * Tees_12

u_12 = V_dot_12/(pi*(Dia_12/2)^2)

1/sqrt(lambda_12) = -2 * log10( 2.51/(Re_12*sqrt(lambda_12)) + (rough/Dia_12)/3.72 )

Re_12 = (rho_water*u_12*L_12)/mu

DELTAP_maj_12 = - ( lambda_12*(L_12/Dia_12)*((rho_water*u_12^2)/2) ) / 1000 / 4

DELTAP_min_12 = - xi_12 * 0.5*rho_water*u_12^2 / 1000 / 4


"8 Inch Pipe Analysis"

{ Computes minor loss coefficient, stream velocity, friction factor (lambda), Reynolds number, and major
and minor pressure losses for 8 inch diameter pipe. }

xi_8 = xi_90 * Bend90_8 + xi_45 * Bend45_8 + xi_tee * Tees_8

u_8 = V_dot_8/(pi*(Dia_8/2)^2)

1/sqrt(lambda_8) = -2 * log10( 2.51/(Re_8*sqrt(lambda_8)) + (rough/Dia_8)/3.72 )

Re_8 = (rho_water*u_8*L_8)/mu

DELTAP_maj_8 = - ( lambda_8*(L_8/Dia_8)*((rho_water*u_8^2)/2) ) / 1000 / 4

DELTAP_min_8 = - xi_8 * 0.5*rho_water*u_8^2 / 1000 / 4
```

Case 2

```
{Cooling Tower Loop System 1 Only}
V_dot_twr=1460 {1460 gpm }
N_ct=1 {number of cooling towers}
V_dot_cti=V_dot_twr/N_ct
V_dot_bld=1460

T_tsci=108 {inlet tower temperature in degrees F}
T_blde=85 {inlet tower temperature in degrees F}
T_wb=20 {wet bulb temperature in F}
T_db=52 {dry bulb temperature in F}
c_p=4.187 {kj/kgK}
Rv=461.9 {J/kgK}
Rho=1.225 {kg/m^3}
Range=T_cti-T_cte {Range of the cooling tower}
pt=101.325 {kPa}
CoC=3.5
cost_elect=.0782 {$/kW*hr}
hours_yr=8760

{2 TSC's}
CC_1000=N_TSC*1.465*(T_tsci-T_db)/60 {cooling capacity of a TSC with a water flow rate of 1000 gpm}
CC_800=N_TSC*1.435*(T_tsci-T_db)/60 {cooling capacity of a TSC with a water flow rate of 800 gpm}
V_dot_tsc=N_TSC*V_dot_twr/2 {The TSC are in paralell}
V_dot_through=V_dot_bld-(V_dot_tsc)
T_tsco=((T_tsci-32)/1.8-((CC_800)/((V_dot_tsc*(6.31e-5)*c_p))))*1.8+32
T_bldi=((V_dot_tsc*T_tsco)+(V_dot_through*T_tsci))/(V_dot_tsc+V_dot_through)

{TSC Electricty Cost}
power_per_tsc=20 {kW}


N_TSC=2
eff=1.0
```

```
fan_speed_per=0.5

Tsc_elect_cost=power_per_tsc*N_TSC*hours_yr*cost_elect*fan_speed_per/(eff*1000)


{outlet temperature of cooling towers}


Q_HX=V_dot_bld*c_p*(T_bldi-T_blde)

V_dot_twr*c_p*Range=Q_HX


{Cooling tower model}

T_int=T_cte+(0.3*Range)


h_int=enthalpy(AirH2O,T=((T_int-32)/1.8),B=((T_int-32)/1.8),P=pt)*.43

h_out=enthalpy(AirH2O,T=((T_wb-32)/1.8),B=((T_wb-32)/1.8),P=pt)*.43


x1=Range/(h_int-h_out)

x=ln(x1)

y=ln(V_dot_cti)

y=(-0.0308527*(x^2))-(0.7898894*x)+7.9410324



" Cooling Tower Exit"

{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at

cooling tower exit is difference between inlet and losses. }

V_dot_loss = (0.003*(V_dot_cti*(6.31e-5)) +(V_dot_cti*(6.31e-5))* c_p * (((T_cti-32)/1.8) - ((T_cte-
32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }

V_dot_cte = V_dot_cti - V_dot_loss


" Makeup"

{ Assume makeup water at the same temperature as the cold water. }

V_dot_mkp = V_dot_loss

T_mkp = T_cti


"Blowdown"
```

```
V_dot_blow=V_dot_mkp/(CoC-1)


{Cooling Tower Cost System 1 Only}

m_yr=525600

per_yr=.85

cooling_tower_time=m_yr*per_yr


{Hourly Cost}

m_hr=60

cooling_tower_time_hourly=m_hr

vol_wat_hr=cooling_tower_time_hourly*(V_dot_loss+V_dot_blow)

makeup_water_cost=.00418 {$/gal}

cooling_tower_water_cost_hr=makeup_water_cost*vol_wat_hr


vol_wat_yr=cooling_tower_time*(V_dot_loss+V_dot_blow)

cooling_tower_water_cost=makeup_water_cost*vol_wat_yr/1000 {in thousands of dollars}



{Fan Power}


atmos=pt {kPa}

Max_HP=75 {hp}

h_intatmos=enthalpy(AirH2O,T=(T_int-32)/1.8,B=(T_int-32)/1.8,P=pt)*.43

h_WBatmos=enthalpy(AirH2O,T=(T_wb-32)/1.8,B=(T_wb-32)/1.8,P=pt)*.43

dhatmos=h_intatmos-h_WBatmos

coeff1=exp(-0.0308527*ln((x1)^2) - 0.7898894*ln(x1) + 7.9410324)

coeff2=(volume(AirH2O,T=78.95,B=T_wb,P=pt)/volume(AirH2O,T=78.95,B=T_wb,P=atmos))^0.667

GPMBase=(coeff1*(dhatmos/(h_int-h_out))*coeff2)*3


TwrPct = V_dot_twr/GPMBase
```

```
TwrACFM =
V_dot_twr*(volume(AirH2O,T=78.95,B=T_wb,P=pt)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))*
TwrPct^0.946

TwrHP =0.746*Max_HP * (TwrACFM / V_dot_twr)^3 *
(volume(AirH2O,T=78.95,B=T_wb,P=atmos)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))

N_cooling_fans=2

hours_cooling_yr=hours_yr*.85

cooling_fan_cost=TwrHP*N_cooling_fans*hours_cooling_yr*cost_elect*.746/(eff*1000)




{Pump Costs}

Building_Pump_kW=0.746*25

Tower_Pump_kW=0.746*40

Building_Pump_Cost=Building_Pump_kW*cost_elect*hours_yr/((.75)*1000) {75% efficiency}

Tower_pump_Cost=Tower_Pump_kW*cost_elect*hours_yr/((.75)*1000)*per_yr


{System 2 - CHW ONLY}

Energy_Sys2 = 975 "in kW"


CHW_cost = 0.04965 "$/kW-hr"

Cost_CHW = CHW_cost*hours_yr*Energy_Sys2/1000 "in thousands of dollars"


{Total Cost}

Total_COST =
Tower_pump_Cost+cooling_fan_cost+cooling_tower_water_cost+Building_Pump_Cost+Tsc_elect_cost+
Cost_CHW
```

Case 3

{Cooling Tower Loop System 1 Only}

```
V_dot_twr=1460 {10000 gpm in m3/s}

N_ct=1 {number of cooling towers}

V_dot_cti=V_dot_twr/N_ct

V_dot_bld=1460


T_tsci=108 {inlet tower temperature in degrees F}

T_blde=85 {inlet tower temperature in degrees F}

T_wb=55 {average wet bulb temperature in F}

T_db=60 {dry bulb temperature in F}

c_p=4.187 {kj/kgK}

Rv=461.9 {J/kgK}

Rho=1.225 {kg/m^3}

Range=T_cti-T_cte

pt=101.325 {kPa}

CoC=3.5


{2 TSC's}

CC_1000=N_TSC*1.465*(T_tsci-T_db)/60 {cooling capacity of TSC with 1000 gpm going through}

CC_800=N_TSC*1.435*(T_tsci-T_db)/60 {cooling capacity of TSC with 800 gpm going through}

V_dot_tsc=N_TSC*V_dot_twr/2

V_dot_through=V_dot_bld-(V_dot_tsc)

T_tsco=((T_tsci-32)/1.8-((CC_800)/((V_dot_tsc*(6.31e-5)*c_p))))*1.8+32

T_bldi=((V_dot_tsc*T_tsco)+(V_dot_through*T_tsci))/(V_dot_tsc+V_dot_through)


{TSC Electricty Cost}

power_per_tsc=20 {kW}

cost_elect=.0782 {$/kW*hr}

hours_yr=8760

N_TSC=2

eff=1.0

fan_speed_per=0.5
```

```
Tsc_elect_cost=power_per_tsc*N_TSC*hours_yr*cost_elect*fan_speed_per/(eff*1000)

{outlet temperature of cooling towers}
Q_HX=V_dot_bld*c_p*(T_bldi-T_blde)
V_dot_twr*c_p*Range=Q_HX
T_int=T_cte+(0.3*Range)

h_int=enthalpy(AirH2O,T=((T_int-32)/1.8),B=((T_int-32)/1.8),P=pt)*.43
h_out=enthalpy(AirH2O,T=((T_wb-32)/1.8),B=((T_wb-32)/1.8),P=pt)*.43

x1=Range/(h_int-h_out)
x=ln(x1)
y=ln(V_dot_cti)
y=(-0.0308527*(x^2))-(0.7898894*x)+7.9410324

" Cooling Tower Exit"
{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at
cooling tower exit is difference between inlet and losses. }
V_dot_loss = (0.003*(V_dot_cti*(6.31e-5)) +(V_dot_cti*(6.31e-5))* c_p * (((T_cti-32)/1.8) - ((T_cte-
32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }
V_dot_cte = V_dot_cti - V_dot_loss

" Makeup"
{ Assume makeup water at the same temperature as the cold water. }
V_dot_mkp = V_dot_loss
T_mkp = T_cti

"Blowdown"
V_dot_blow=V_dot_mkp/(CoC-1)

{Cooling Tower Cost System 1 Only}
m_yr=525600
```

```
per_yr=.85

cooling_tower_time=m_yr*per_yr


{Hourly Cost}

m_hr=60

cooling_tower_time_hourly=m_hr

vol_wat_hr=cooling_tower_time_hourly*(V_dot_loss+V_dot_blow)

makeup_water_cost=.00418 {$/gal}

cooling_tower_water_cost_hr=makeup_water_cost*vol_wat_hr


vol_wat_yr=cooling_tower_time*(V_dot_loss+V_dot_blow)

cooling_tower_water_cost=makeup_water_cost*vol_wat_yr/1000 {in thousands of dollars}



{Fan Power}

atmos=pt {kPa}

Max_HP=75 {hp}

h_intatmos=enthalpy(AirH2O,T=(T_int-32)/1.8,B=(T_int-32)/1.8,P=pt)*.43

h_WBatmos=enthalpy(AirH2O,T=(T_wb-32)/1.8,B=(T_wb-32)/1.8,P=pt)*.43

dhatmos=h_intatmos-h_WBatmos

coeff1=exp(-0.0308527*ln((x1)^2) - 0.7898894*ln(x1) + 7.9410324)

coeff2=(volume(AirH2O,T=78.95,B=T_wb,P=pt)/volume(AirH2O,T=78.95,B=T_wb,P=atmos))^0.667

GPMBase=(coeff1*(dhatmos/(h_int-h_out))*coeff2)*3


TwrPct = V_dot_twr/GPMBase


TwrACFM =
V_dot_twr*(volume(AirH2O,T=78.95,B=T_wb,P=pt)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))*
TwrPct^0.946

TwrHP =0.746*Max_HP * (TwrACFM / V_dot_twr)^3 *
(volume(AirH2O,T=78.95,B=T_wb,P=atmos)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))

N_cooling_fans=2

hours_cooling_yr=hours_yr*.85

cooling_fan_cost=TwrHP*N_cooling_fans*hours_cooling_yr*cost_elect*.746/(eff*1000)
```

```
{Pump Costs}

Building_Pump_kW=0.746*25

Tower_Pump_kW=0.746*40

Building_Pump_Cost=Building_Pump_kW*cost_elect*hours_yr/((.75)*1000) {75% efficiency}

Tower_pump_Cost=Tower_Pump_kW*cost_elect*hours_yr/((.75)*1000)*per_yr


{System 2 - CHW for part of year with TSC}


{To get accurate data you have to run hourly weather data with a parametric table and sum the hourly
cost in an Excel file}


{Cooling Tower Loop System 2 Only}

V_dot_twr2=1460 {gpm}

N_ct2=1 {number of cooling towers}

V_dot_cti2=V_dot_twr2/N_ct2

V_dot_bld2=330

Energy_Sys2 = 975 {in kW}

CHW_cost = 0.04965 {$/kW-hr or $14.50/Mbtu}

{Pump Costs}

        {same as Sys 1}


{TSC Electricty Cost}

T_tsci2 = 85

N_TSC_2=1

Tsc_elect_cost_2=0.4*power_per_tsc*N_TSC_2*hours_yr*cost_elect*fan_speed_per/(eff*1000)


{Mixed Mode}

TSC_coolcap = (1320/60)*(T_tsci2-T_db)


Energy_Sys22 = Energy_Sys2-TSC_coolcap

Cost_CHW_hourly = CHW_cost*Energy_Sys22 "dollars/hr"

Cost_CHW =Cost_CHW_hourly*hours_yr/1000 "dollars/hr"
```

{Total Cost}

Total_COST =
Tower_pump_Cost+cooling_fan_cost+cooling_tower_water_cost+Building_Pump_Cost+Tsc_elect_cost+Cost_CHW+Tsc_elect_cost_2

Case 4

```
{Cooling Tower Loop System 1 Only}

V_dot_twr=1460 {10000 gpm in m3/s}

N_ct=1 {number of cooling towers}

V_dot_cti=V_dot_twr/N_ct

V_dot_bld=1460


T_tsci=108 {inlet tower temperature in degrees F}

T_blde=90 {inlet tower temperature in degrees F}

T_wb=20 {average wet bulb temperature in F}

T_db=30 {dry bulb temperature in F}

c_p=4.187 {kj/kgK}

Rv=461.9 {J/kgK}

Rho=1.225 {kg/m^3}

Range=T_cti-T_cte

pt=1 {atm}

atmos = 1

CoC=3.5


{2 TSC's}

CC_1000=N_TSC*1.465*(T_tsci-T_db)/60

CC_800=N_TSC*1.435*(T_tsci-T_db)/60

CC_600=N_TSC*1.380*(T_tsci-T_db)/60

V_dot_tsc=N_TSC*1000

V_dot_through=V_dot_bld-(V_dot_tsc)

T_tsco=((T_tsci-32)/1.8-((CC_1000)/((V_dot_tsc*(6.31e-5)*c_p))))*1.8+32

T_bldi=((V_dot_tsc*T_tsco)+(V_dot_through*T_tsci))/(V_dot_tsc+V_dot_through)


{TSC Electricty Cost}

power_per_tsc=20 {kW}

cost_elect=.0782 {$/kW*hr}

hours_yr=8760

N_TSC=1
```

```
eff=1.0

fan_speed_per=0.5

Tsc_elect_cost=power_per_tsc*N_TSC*hours_yr*cost_elect*fan_speed_per/(eff*1000)


{outlet temperature of cooling towers}

Q_HX=V_dot_bld*c_p*(T_bldi-T_blde)

V_dot_twr*c_p*Range=Q_HX

T_int=T_cte+(0.3*Range)

h_int=enthalpy(AirH2O,T=T_int,B=T_int,P=pt)

h_out=enthalpy(AirH2O,T=T_wb,B=T_wb,P=pt)

x1=Range/(h_int-h_out)

x=ln(x1)

y=ln(V_dot_cti)

y=(-0.0308527*(x^2))-(0.7898894*x)+7.9410324



" Cooling Tower Exit"

{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at
cooling tower exit is difference between inlet and losses. }

V_dot_loss = (0.003*(V_dot_cti*(6.31e-5)) +(V_dot_cti*(6.31e-5))* c_p * (((T_cti-32)/1.8) - ((T_cte-
32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }

V_dot_cte = V_dot_cti - V_dot_loss


" Makeup"

{ Assume makeup water at the same temperature as the cold water. }

V_dot_mkp = V_dot_loss

T_mkp = T_cti


"Blowdown"

V_dot_blow=V_dot_mkp/(CoC-1)


{Cooling Tower Cost System 1 Only}

m_yr=525600
```

per_yr=.85

cooling_tower_time=m_yr*per_yr

{Hourly Cost}

m_hr=60

cooling_tower_time_hourly=m_hr

vol_wat_hr=cooling_tower_time_hourly*(V_dot_loss+V_dot_blow)

makeup_water_cost=.00418 {$/gal}

cooling_tower_water_cost_hr=makeup_water_cost*vol_wat_hr

vol_wat_yr=cooling_tower_time*(V_dot_loss+V_dot_blow)

cooling_tower_water_cost=makeup_water_cost*vol_wat_yr/1000 {in thousands of dollars}

{Fan Power}

Max_HP=75 {hp}

h_intatmos=**enthalpy**(***AirH2O***,***T***=T_int,***B***=T_int,***P***=pt)

h_WBatmos=**enthalpy**(***AirH2O***,***T***=T_wb,***B***=T_wb,***P***=pt)

dhatmos=h_intatmos-h_WBatmos

coeff1=**exp**(-0.0308527***ln**((x1)^2) - 0.7898894***ln**(x1) + 7.9410324)

coeff2=(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos))^0.667

GPMBase=(coeff1*(dhatmos/(h_int-h_out))*coeff2)*3

TwrPct = V_dot_twr/GPMBase

TwrACFM =
V_dot_twr*(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))*
TwrPct^0.946

TwrkW = 0.746*Max_HP * (TwrACFM / V_dot_twr)^3 *
(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))

N_cooling_fans=2

hours_cooling_yr=hours_yr*.85

cooling_fan_cost=TwrkW*N_cooling_fans*hours_cooling_yr*cost_elect/(eff*1000)

```
{Pump Costs}

Building_Pump_kW=0.746*25

Tower_Pump_kW=0.746*40

Building_Pump_Cost=Building_Pump_kW*cost_elect*hours_yr/((.75)*1000) {75% efficiency}

Tower_pump_Cost=Tower_Pump_kW*cost_elect*hours_yr/((.75)*1000)*per_yr



{-------System 2-------}


{For a more detailed cost analysis run parametric table with hourly weather data}

{System 2 - CHW for part of year with TSC}

Energy_Sys2 = 975 "in kW"


CHW_cost = 0.04965 "$/kW-hr or $14.50/Mbtu"

Cost_CHW = .4*CHW_cost*hours_yr*Energy_Sys2/1000 "in thousands of dollars"



{Cooling Tower Loop System 2 Only}

V_dot_twr2=1460 {gpm}

N_ct2=1 {number of cooling towers}

V_dot_cti2=V_dot_twr2/N_ct2

V_dot_bld2=330


T_bldi2=85 {inlet tower temperature in degrees F}

T_blde2=65 {inlet tower temperature in degrees F}


Range2=T_cti2-T_cte2



{outlet temperature of cooling towers}

Q_HX2=V_dot_bld2*c_p*(T_bldi2-T_blde2)
```

```
V_dot_twr2*c_p*Range2=Q_HX2

T_int2=T_cte2+(0.3*Range2)

h_int2=enthalpy(AirH2O,T=T_int2,B=T_int2,P=pt)

h_out2=enthalpy(AirH2O,T=T_wb,B=T_wb,P=pt)

x12=(Range2/(h_int2-h_out2))

x2=ln(x12)

y2=ln(V_dot_cti2)

y2=(-0.0308527*(x2^2))-(0.7898894*x2)+7.9410324


" Cooling Tower Exit"

{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at

cooling tower exit is difference between inlet and losses. }

V_dot_loss2 = (0.003*(V_dot_cti2*(6.31e-5)) +(V_dot_cti2*(6.31e-5))* c_p * (((T_cti2-32)/1.8) -
((T_cte2-32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }

V_dot_cte2 = V_dot_cti2 - V_dot_loss2


" Makeup"

{ Assume makeup water at the same temperature as the cold water. }

V_dot_mkp2 = V_dot_loss2

T_mkp2 = T_cti2


"Blowdown"

V_dot_blow2=V_dot_mkp2/(CoC-1)


{Cooling Tower Cost System 1 Only}

per_yr2=0.6 {value determined using weather data}

cooling_tower_time2=m_yr*per_yr2

vol_wat_yr2=cooling_tower_time2*(V_dot_loss2+V_dot_blow2)

cooling_tower_water_cost2=makeup_water_cost*vol_wat_yr2/1000 {in thousands of dollars}


{Cooling Tower Fans Cost}

N_cooling_fans2=1
```

cooling_fan_cost2=TwrkW2*N_cooling_fans2*hours_yr*cost_elect/(eff*1000)

{Pump Costs}

{same as Sys 1}

{Fan Power}

h_intatmos2=**enthalpy**(***AirH2O***,***T***=T_int2,***B***=T_int2,***P***=pt)

h_WBatmos2=**enthalpy**(***AirH2O***,***T***=T_wb,***B***=T_wb,***P***=pt)

dhatmos2=h_intatmos2-h_WBatmos2

coeff12=**exp**(-0.0308527***ln**((x12)^2) - 0.7898894***ln**(x12) + 7.9410324)

coeff22=(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt))^0.667

GPMBase2=(coeff12*(dhatmos2/(h_int2-h_out2))*coeff22)*3

TwrPct2 = V_dot_twr2/GPMBase2

TwrACFM2 =
V_dot_twr2*(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=pt)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))*
TwrPct2^0.946

TwrkW2 = 0.746*Max_HP * (TwrACFM2 / V_dot_twr2)^3 *
(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)/(**volume**(***AirH2O***,***T***=78.95,***B***=T_wb,***P***=atmos)))

{Total Cost}

Total_COST =
Tower_pump_Cost+cooling_fan_cost+cooling_tower_water_cost+Building_Pump_Cost+Tsc_elect_cost+
Cost_CHW+cooling_fan_cost2+cooling_tower_water_cost2+Tower_pump_Cost+Building_Pump_Cost

Case 5

```
{Cooling Tower Loop System 1 Only}

V_dot_twr=1460 {10000 gpm in m3/s}

N_ct=1 {number of cooling towers}

V_dot_cti=V_dot_twr/N_ct

V_dot_bld=1460

T_tsci=108 {inlet tower temperature in degrees F}

T_blde=85 {inlet tower temperature in degrees F}

T_wb=20 {average wet bulb temperature in F}

T_db=25 {dry bulb temperature in F}

c_p=4.187 {kj/kgK}

Rv=461.9 {J/kgK}

Rho=1.225 {kg/m^3}

Range=T_cti-T_cte

pt=101.325 {kPa}

CoC=3.5

{2 TSC's}

CC_1000=N_TSC*1.465*(T_tsci-T_db)/60 {cooling capacity of TSC with 1000 gpm going through}

CC_800=N_TSC*1.435*(T_tsci-T_db)/60 {cooling capacity of TSC with 800 gpm going through}

V_dot_tsc=N_TSC*V_dot_twr/2

V_dot_through=V_dot_bld-(V_dot_tsc)

T_tsco=((T_tsci-32)/1.8-((CC_800)/((V_dot_tsc*(6.31e-5)*c_p))))*1.8+32

T_bldi=((V_dot_tsc*T_tsco)+(V_dot_through*T_tsci))/(V_dot_tsc+V_dot_through)

{TSC Electricty Cost}

power_per_tsc=20 {kW}

cost_elect=.0782 {$/kW*hr}

hours_yr=8760

N_TSC=2

eff=1.0

fan_speed_per=0.5
```

```
Tsc_elect_cost=power_per_tsc*N_TSC*hours_yr*cost_elect*fan_speed_per/(eff*1000)

{outlet temperature of cooling towers}
Q_HX=V_dot_bld*c_p*(T_bldi-T_blde)
V_dot_twr*c_p*Range=Q_HX
T_int=T_cte+(0.3*Range)

h_int=enthalpy(AirH2O,T=((T_int-32)/1.8),B=((T_int-32)/1.8),P=pt)*.43
h_out=enthalpy(AirH2O,T=((T_wb-32)/1.8),B=((T_wb-32)/1.8),P=pt)*.43

x1=Range/(h_int-h_out)
x=ln(x1)
y=ln(V_dot_cti)
y=(-0.0308527*(x^2))-(0.7898894*x)+7.9410324

" Cooling Tower Exit"
{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at
cooling tower exit is difference between inlet and losses. }
V_dot_loss = (0.003*(V_dot_cti*(6.31e-5)) +(V_dot_cti*(6.31e-5))* c_p * (((T_cti-32)/1.8) - ((T_cte-
32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }
V_dot_cte = V_dot_cti - V_dot_loss

" Makeup"
{ Assume makeup water at the same temperature as the cold water. }
V_dot_mkp = V_dot_loss
T_mkp = T_cti

"Blowdown"
V_dot_blow=V_dot_mkp/(CoC-1)

{Cooling Tower Cost System 1 Only}
m_yr=525600
```

```
per_yr=.85

cooling_tower_time=m_yr*per_yr


{Hourly Cost}

m_hr=60

cooling_tower_time_hourly=m_hr

vol_wat_hr=cooling_tower_time_hourly*(V_dot_loss+V_dot_blow)

makeup_water_cost=.00418 {$/gal}

cooling_tower_water_cost_hr=makeup_water_cost*vol_wat_hr


vol_wat_yr=cooling_tower_time*(V_dot_loss+V_dot_blow)

cooling_tower_water_cost=makeup_water_cost*vol_wat_yr/1000 {in thousands of dollars}



{Fan Power}

atmos=pt {kPa}

Max_HP=75 {hp}

h_intatmos=enthalpy(AirH2O,T=(T_int-32)/1.8,B=(T_int-32)/1.8,P=pt)*.43

h_WBatmos=enthalpy(AirH2O,T=(T_wb-32)/1.8,B=(T_wb-32)/1.8,P=pt)*.43

dhatmos=h_intatmos-h_WBatmos

coeff1=exp(-0.0308527*ln((x1)^2) - 0.7898894*ln(x1) + 7.9410324)

coeff2=(volume(AirH2O,T=78.95,B=T_wb,P=pt)/volume(AirH2O,T=78.95,B=T_wb,P=atmos))^0.667

GPMBase=(coeff1*(dhatmos/(h_int-h_out))*coeff2)*3


TwrPct = V_dot_twr/GPMBase


TwrACFM =
V_dot_twr*(volume(AirH2O,T=78.95,B=T_wb,P=pt)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))*
TwrPct^0.946

TwrHP =0.746*Max_HP * (TwrACFM / V_dot_twr)^3 *
(volume(AirH2O,T=78.95,B=T_wb,P=atmos)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))

N_cooling_fans=2

hours_cooling_yr=hours_yr*.85

cooling_fan_cost=TwrHP*N_cooling_fans*hours_cooling_yr*cost_elect*.746/(eff*1000)
```

```
{Pump Costs}

Building_Pump_kW=0.746*25

Tower_Pump_kW=0.746*40

Building_Pump_Cost=Building_Pump_kW*cost_elect*hours_yr/((.75)*1000) {75% efficiency}

Tower_pump_Cost=Tower_Pump_kW*cost_elect*hours_yr/((.75)*1000)*per_yr


{System 2 - CHW for part of year with TSC}


{To get accurate data you have to run hourly weather data with a parametric table and sum the hourly
cost in an Excel file}


{Cooling Tower Loop System 2 Only}

V_dot_twr2=1460 {gpm}

N_ct2=1 {number of cooling towers}

V_dot_cti2=V_dot_twr2/N_ct2

V_dot_bld2=330


{inlet tower temperature in degrees F}


Range2=T_cti2-T_cte2


{outlet temperature of cooling towers}

T_blde2=if(T_wb,64,65,65,T_cte2)

Q_HX2=V_dot_bld2*c_p*(T_bldi2-T_blde2)

V_dot_twr2*c_p*Range2=Q_HX2

T_int2=T_cte2+(0.3*Range2)

h_int2=enthalpy(AirH2O,T=T_int2,B=T_int2,P=pt)

h_out2=enthalpy(AirH2O,T=T_wb,B=T_wb,P=pt)

x12=(Range2/(h_int2-h_out2))

x2=ln(x12)

y2=ln(V_dot_cti2)

y2=(-0.0308527*(x2^2))-(0.7898894*x2)+7.9410324
```

" Cooling Tower Exit"

{ Compute water losses in cooling towers as sum of drift losses + evaporative losses. Flow rate at cooling tower exit is difference between inlet and losses. }

```
V_dot_loss2 = (0.003*(V_dot_cti2*(6.31e-5)) +(V_dot_cti2*(6.31e-5))* c_p * (((T_cti2-32)/1.8) - ((T_cte2-32)/1.8)) / 2290)/(6.31e-5) { drift + evaporation }

V_dot_cte2 = V_dot_cti2 - V_dot_loss2
```

" Makeup"

{ Assume makeup water at the same temperature as the cold water. }

```
V_dot_mkp2 = V_dot_loss2

T_mkp2 = T_cti2
```

"Blowdown"

```
V_dot_blow2=V_dot_mkp2/(CoC-1)
```

{Cooling Tower Cost System 1 Only}

```
per_yr2=1

cooling_tower_time2=m_yr*per_yr2

vol_wat_yr2=cooling_tower_time2*(V_dot_loss2+V_dot_blow2)

cooling_tower_water_cost2=makeup_water_cost*vol_wat_yr2/1000 {in thousands of dollars}
```

{Cooling Tower Fans Cost}

```
N_cooling_fans2=1

cooling_fan_cost2=TwrkW2*N_cooling_fans2*hours_yr*cost_elect/(eff*1000)
```

{System 2 - CHW for part of year with TSC}

```
Energy_Sys2 = 975 {in kW}

CHW_cost = 0.04965 {$/kW-hr or $14.50/Mbtu}
```

{Pump Costs}

```
{same as Sys 1}

{Fan Power}

h_intatmos2=enthalpy(AirH2O,T=T_int2,B=T_int2,P=pt)

h_WBatmos2=enthalpy(AirH2O,T=T_wb,B=T_wb,P=pt)

dhatmos2=h_intatmos2-h_WBatmos2

coeff12=exp(-0.0308527*ln((x12)^2) - 0.7898894*ln(x12) + 7.9410324)

coeff22=(volume(AirH2O,T=78.95,B=T_wb,P=pt)/volume(AirH2O,T=78.95,B=T_wb,P=pt))^0.667

GPMBase2=(coeff12*(dhatmos2/(h_int2-h_out2))*coeff22)*3


TwrPct2 = V_dot_twr2/GPMBase2


TwrACFM2 =
V_dot_twr2*(volume(AirH2O,T=78.95,B=T_wb,P=pt)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))*
TwrPct2^0.946

TwrkW2 = 0.746*Max_HP * (TwrACFM2 / V_dot_twr2)^3 *
(volume(AirH2O,T=78.95,B=T_wb,P=atmos)/(volume(AirH2O,T=78.95,B=T_wb,P=atmos)))



{TSC Electricty Cost}

T_tsci2 = 85

N_TSC_2=1

Tsc_elect_cost_2=0.4*power_per_tsc*N_TSC_2*hours_yr*cost_elect*fan_speed_per/(eff*1000)


{Mixed Mode}

TSC_coolcap = (1320/60)*(T_tsci2-T_db)

{T_bli2 depends on the cooling capacity of the TSC}

T_bldi2=T_tsci2-20*TSC_coolcap/Energy_Sys22

Energy_Sys22 = Energy_Sys2-TSC_coolcap

Cost_CHW_hourly = CHW_cost*Energy_Sys22 "dollars/hr"

Cost_CHW=Cost_CHW_hourly*hours_yr/1000


{Total Cost}
```

Total_COST =
Tower_pump_Cost+cooling_fan_cost+cooling_tower_water_cost+Building_Pump_Cost+Tsc_elect_cost+Cost_CHW+Tsc_elect_cost_2

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