KETTLE COOLING AND SUSTAINABILITY PROJECT

by

Nico DeGiorgio

Agricultural Systems Management

BioResource and Agricultural Engineering Department

California Polytechnic State University

San Luis Obispo

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TITLE	**:**	**Kettle Cooling and Sustainability Project**
AUTHOR	**:**	**Nico DeGiorgio**
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Senior Project Advisor

Signature

Date

Department Head

Signature

Date

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ABSTRACT

A particular frozen food processing facility uses a once-through cooling process to bring their kettle-cooked product's temperature down from nearly boiling temperatures to the eighty degree Fahrenheit range. This process is water intensive and facility managers are seeking to reduce their potable water consumption. Sales engineers from Air Treatment Corporation, an HVAC&R manufacturers' rep, initiated a number of meetings to propose heat rejection solutions and illustrate their associated payback potentials.

This report contains methods of analyzing a food production process, the technical sales process, thermodynamic principles, refrigeration technologies, and the application of technical knowledge to provide a long-term, system solution. To justify the purchase of a system solution, a detailed engineering economic analysis was conducted to account for the time-value of money and equipment specifications.

The goal of this report is to show the potential for monetary savings by combining technical, system solutions with sustainability.

Keywords: cooling tower, industrial fluid cooler, sales engineering, food processing, heat rejection, cooling, refrigeration, sustainability, HVAC, HVAC&R, BAC, thermodynamics, capital budgeting, engineering economics, industrial water, water use, wastewater, sewage rate, water rate, pro forma financial statement, technical sales, return on investment, closed circuit cooling tower

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LIST OF ABBREVIATIONS

ATC	Air Treatment Corporation
BOD	Biochemical Oxygen Demand (Biological Oxygen Demand)
BTUH	British Thermal Units per Hour
CCCT	Closed Circuit Cooling Tower (industrial fluid cooler)
CTI	Cooling Technology Institute
EOY	End of Year
F	Fahrenheit
HVAC&R	Heating, Ventilation, Air-Conditioning, and Refrigeration
IRR	Internal Rate of Return
MACRS	Modified Accelerated Cost Recovery System
MARR	Minimum Acceptable Rate of Return (hurdle rate)
MIRR	Modified Rate of Return
NAICS	North American Industry Classification System
NPV	Net Present Value
OCCT	Open Circuit Cooling Tower
PP	payback period
ROI	Return on Investment
R Tons	Refrigeration Tons
SGMA	Sustainable Groundwater Management Act
TSS	Total Soluble Solids
TWC	Tomato Water Condensate
WEN	Water Energy Nexus

INTRODUCTION

Water Supply and Utility Background

Due specifically to periods of extended drought across California, Governor Brown issued a statewide mandate declaring heightened regulation on potable water-use and associated waste. (SWRCB 2016). Although the state's measure requires a 20% reduction in water use by 2020, local water agencies will tailor the framework of how they will conduct their conservation efforts. Sonoma County Water Agency relies heavily on precipitation for its freshwater deliveries which requires adequate storage and timely allocation. The unpredictability of atmospheric rivers (CW3E) and a recent reduction of snowpack by 25% (Berg 2017) presents challenges to the water managers to ensure a consistent surface water supply for the future.

Groundwater will also become increasingly difficult to access as time progresses. Local governments in Sonoma County have already posed the idea of developing fees on unsustainable pumping behavior and are empowered to do so by the Sustainable Groundwater Management Act of 2014 (SCSGM 2016). Due to a combination of conservation efforts and the growing challenges in water management, water sales have had a steep decline. This has forced the City of Santa Rosa and Sonoma County Water Agency to increase their rates to restore cash flow for operations and maintenance as seen in Figure 1 below. Supporting documentation provided by the city of Santa Rosa can be found in Appendix O (SRW 2016).

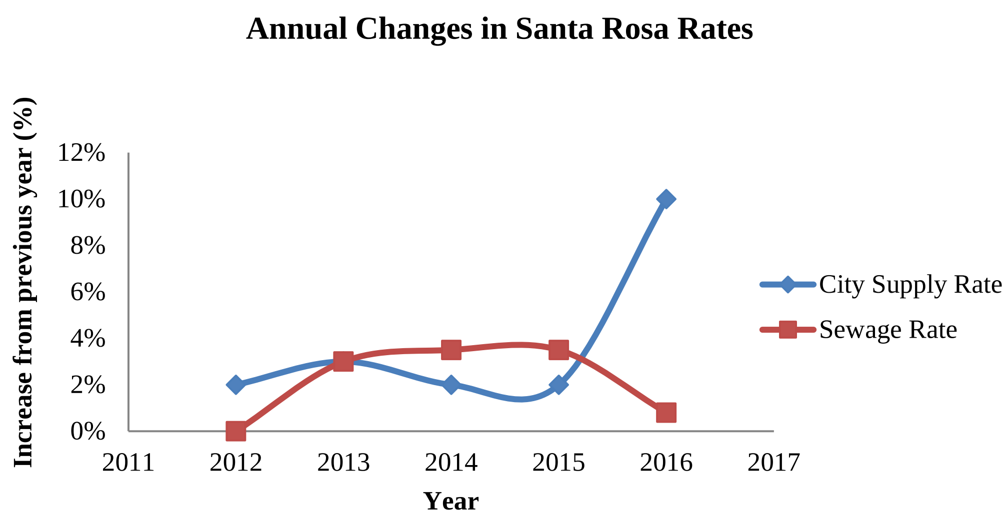


Figure 1 Behavior of annual water and sewage rates

Due to this, industrial consumers can expect rate increases for both water and sewage, despite the fact that their consumption could remain stable. This will force companies to reanalyze the way they use their water and adopt reduced water-consumptive practices. Inefficient use of water, correlated with unsustainable practice, is in opposition to the Reasonable and Beneficial Use Doctrine which could lead to penalties (Wilson 2011).

Industrial Water Use

In Santa Rosa, industrial demand of potable water amounted to 251 acre-feet or over 81 million gallons among 69 customers. This would amount to an average of about 1.2 million gallons per year per industrial user. A particular food processing facility uses high volumes of water for sanitation, cooking, and cooling applications relative to the industrial average for Santa Rosa.

Figure 2 below shows the portion of water that the facility uses relative to all other commercial-type users which includes manufacturers or processors of materials defined by NAICS code sectors 31 to 33 in Santa Rosa (UWMP 2015).

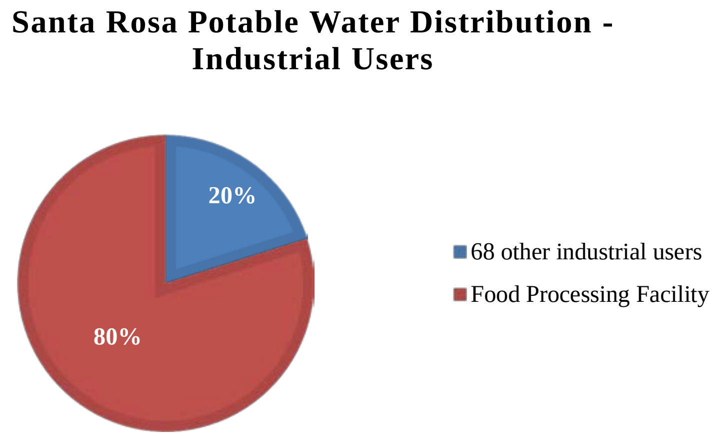


Figure 2 Pie Chart of Industrial Water Users

This level of consumption has led the food processing facility and the city to investigate steps in production that contribute to the highest levels of consumption. In order to do this, the city hired water-use consultants to audit the process to identify areas needing improvement.

Process Assessment

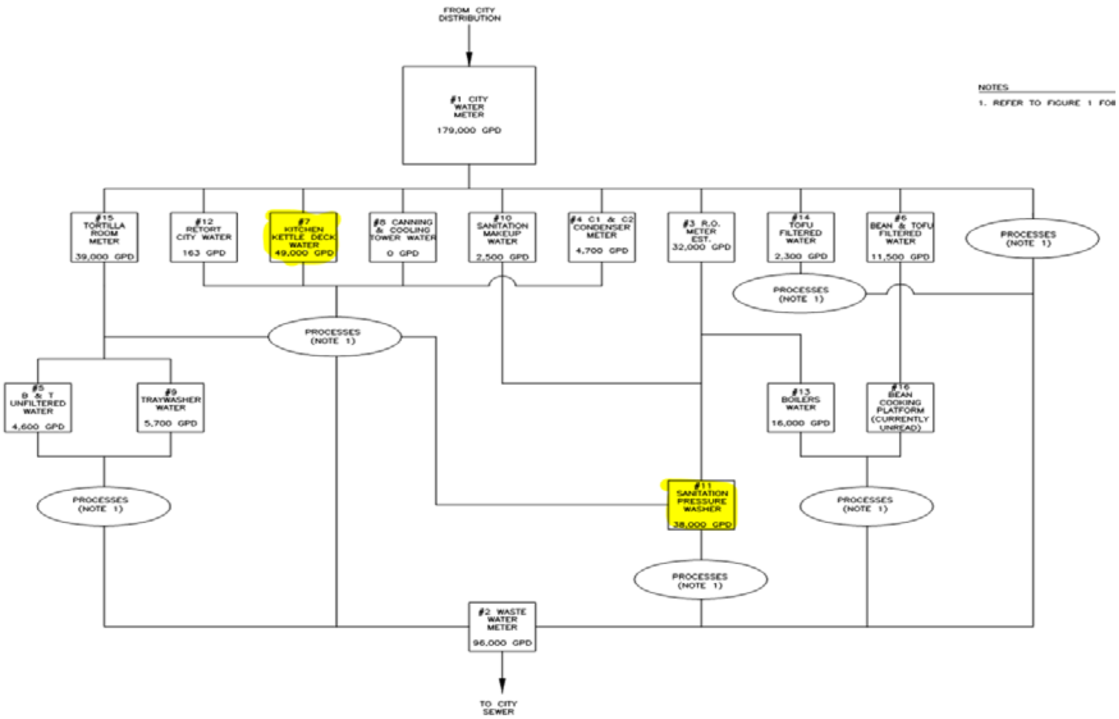


Figure 3 Facility's Water Consumption Diagram

After conducting the facility-wide audit, it was determined that the kettle deck was consuming the largest volumes of water for a single process relative to the other processes throughout the facility. Table 1 summarizes the results of the audit.

Table 1 Food Processes and associated water consumptions

Facility Processes	Acre-Feet/yr	Gallons/yr	Percentage
Total	1500	65,335,000	100%
Kettle Deck	411	17,885,000	27%
Sanitation	318	13,870,000	21%
All other	771	33,580,000	51%

In doing this, various consultants and members of the given food processing facility have come to an agreement that the current once-through kettle cooling operation is critical to the path of improved water-use. It should be noted that more recent information after the audit was released related to water consumption at the kettle deck with values closer to 18 million gallons.

Improvements to this cooling process have been made by members of a manufacturers' representative known as Air Treatment Corporation. They represent a large number of

HVAC&R related manufacturers including: ABB controls and drives, Polaris heat exchangers, and BAC heat rejection equipment (i.e. cooling towers, thermal energy storage, and evaporative condensers). Particular to the water used at the kettle deck, a closed-circuit cooling tower has been proposed to meet the cooling capacity while dramatically reducing the amount of water consumed. Although there are many features involved in the refrigeration process, this report focuses on the cooling of the food product from boiling water temperatures to the eighty degree Fahrenheit range. Improvements are expected to produce large monetary benefits from the reduced water usage and the anticipated increase production capacity. The objective of this report is to outline the methods of Air Treatment Corporation's product solution selection, the sales process, and the anticipated payback with a capital budgeting analysis.

Project Initiation

Sales Cycle

Once the job was prospected, Air Treatment Corporation began to engage in the technical sales process. A sales engineering manager from ATC was sent out to meet with clients in order to further define the problem. After meeting with the plant supervisor, a follow-up meeting was scheduled to learn more about the various constraints that would influence the type of refrigeration solution to be selected. Facility managers expressed that a decision would need to be made before the end of the fiscal year in 2016 and the payback would need to be in the range of three years. The next step in the technical sales process required the sales engineer to synthesize a proposed system solution with these key financial parameters in mind.

The next meeting consisted of a detailed economic analysis, including relevant equipment specifications, in order to illustrate returns on investment. While alternatives were addressed, there was a clear need to implement a recirculating system for the once-through cooling process. Upon further probing, there were a multitude of areas needing improvement for other facility refrigeration process thus expanding the scale of the project. Utilizing the areas of expertise in the room, appropriate data and information were collected in order to begin measuring the extend of savings that could be achieved from load shifting for the proceeding cooling process from 80 to 35 degrees Fahrenheit. This increase in project scope allowed solutions to be split into two phases: kettle cooling upgrades and mechanical cooling upgrades.

While negotiation strategies are important in the sales process, not enough information released in order to mention for this specific project. However, rebates provided by local utilities were leveraged as an incentive to pursue this capital investment.

With the right balance of professional consultation and expression from the clients, ATC closed the deal for cooling improvements at the kettle deck known as the first phase.
The second phase was to replace the air-cooled condensing system, but is outside the scope of this report. For the first project phase, a specific model of an industrial fluid cooler will be selected that best accommodates the food processing facility parameters. Figure 4 below illustrates the basic components of the technical sales cycle.

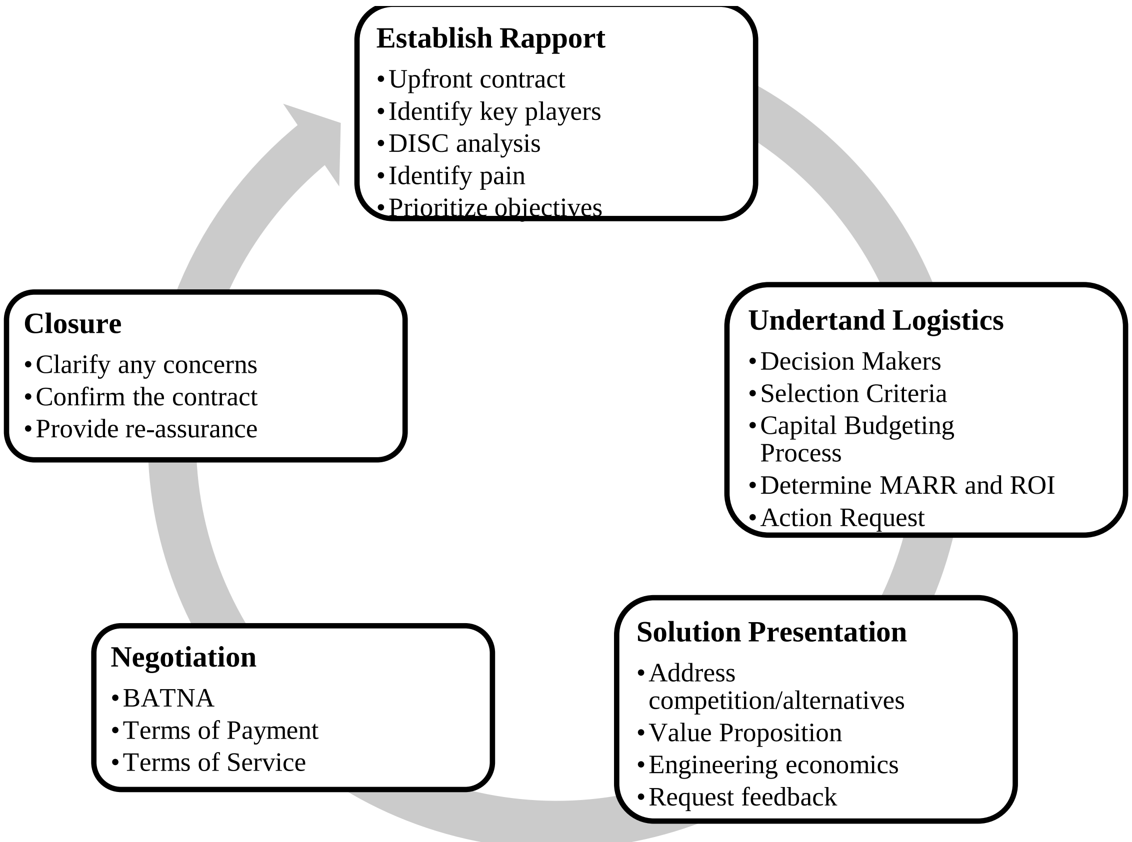


Figure 4 Technical Sales Cycle

Project Parameters

There are many benchmarks that can be applied to an "improved" operation. Facility improvements could result in environmental benefits to the community and further optimize production capacity, however, these are difficult project with information currently available. This report is primarily concerned with the savings incurred from reduced water consumption and improvements to the refrigeration system. A number of variables are involved with the justification of such a project: governmental regulation, utility incentive programs, facility constraints, and company priorities. To create a relevant benchmark of success, the company's priorities were outlined for the cooling solution to the kettle deck operation. The given food processing facility has arranged their list of priorities in the following order:

1) Food safety and quality
2) Water consumption
3) Cooling capacity

This list of company priorities will play an integral role in the decision making process for the best, long-term system solution.

LITERATURE REVIEW

Food processing

Definition

Food processing can be defined as any manipulation of the physical or chemical composition of raw ingredients in order to produce a food product. These processes include, but are not limited to the following: sanitizing, pressurizing, mincing, mashing, liquefying, cooking, baking, cooling, and packaging. Each of these processes has associated consumptions of energy and water. Learning more about the sequence and dependencies between the events in these food processes give us better understanding as to where beneficial improvements can be made.

Food Safety

When food production and processing became industrialized, leading to increased capacity, this provided for tighter tolerances in all aspects of food safety and quality assurance. Due to this, governmental organizations such as the Food and Drug Administration, have implemented provisions like Hazard Analysis and Risk-Based Preventive Controls in order to prevent food borne illnesses. Food processing companies are at minimum expected to produce a product that is deemed “safe” with non-toxic levels pathogenic bacteria, viruses, and other various contaminants. Ranging from food handling to temperature regulation, there are many opportunities for commodities to be spoiled and result in detrimental losses.

More specifically, with frozen foods processing, taking cooked product from boiling temperatures to freezing temperatures crosses a hazardous region for mesophilic microorganisms to thrive and reproduce. Figure 5 shows the growth behavior mesophilic pathogens in food. These bacteria are most replicate in food quickest around 100 Farenheight (38 Celsius) which is dangerous for human consumption because of how close this temperature is to the average human body temperature (AgriLife Extension 2016).

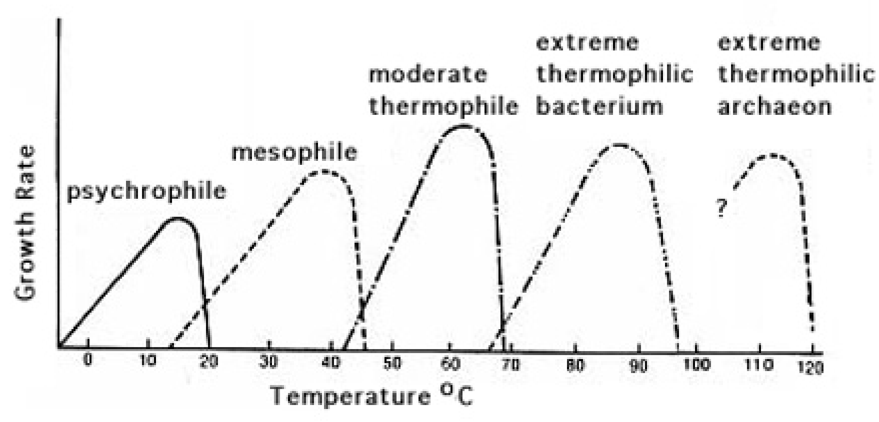


Figure 5 Spectrum of various bacteria’s optimal temperature range (Todar 2008)

Because of this, the FDA has set regulations in the Food Code that require hot foods within the region of 135F to 70F to be cooled in under two hours (Food Code 2013). This is to avoid dangerous levels of bacteria growth, limiting exposure to humans. Popular pathogens in food include, but are not limited to the following: Salmonella, Campylobacter, and E. coli.

Furthermore, refrigerant selection is also important in regards to food safety. Propylene glycol is an example of a food grade refrigerant with water to reduce the freezing point for low temperature applications. Food-grade refrigerants are required when refrigerants come into such close proximity with the product throughout the plumbing of food processing facilities.

Food Quality

Characteristics like taste, appearance, nutritional value, and structure are greatly influenced by a product's temperature profile. Based on consumer preferences, the tolerances for these qualities can be extensive requiring high system precision and redundancy. Specialty food products, which lack GMO, lose the benefits of reduced risk of disease, improved food structure, and longer shelf life. Because the given facility produces a non-GMO product, they are forced to look into their production practices to provide desirable characteristics to stay competitive (UCSC 2005).

One characteristic particularly important to the given facility is the food structure. When foods are frozen, the water contained within the product's plant cell walls expand. This compromises the structure of food product, especially with relatively high water contents. However, this can be avoided by rapidly cooling which forms many smaller internal ice crystals, thus minimizing cell wall rupture (UMich 2014).

In summary, product physical and chemical consistency in food production are extremely important in high volume, automated operations. Ensuring a strong cold chain with proper cooling capacities provides greater resistance to inconsistencies and hazards in system outputs. These considerations play an important role in properly assigning heat rejection solutions to the kettle deck cooling process.

Case Study: Industrial Tomato Processing Operation - Water Energy Nexus

When looking to create an operational definition for success in this project, it is important to put inputs and outputs in comparable terms. For example, contributions from research institutions and the Department of Energy used various water-energy nexus methodologies to analyze a particular tomato processing facility and recovery payback potentials. With an understanding of the tomato paste production process in Figure 6 below, tomato water condensate could be recovered to sanitize, to transport tomatoes in processing flume, or to recover thermal energy. By analyzing the process in terms of comparable flows of heat and water, the TWC was recycled and used for steam generation in the "hot break". The hot break is an essential process in the tomato paste processing industry to provide a desired viscosity. In this case, the circled region indicates where the water was recommended to be reused.

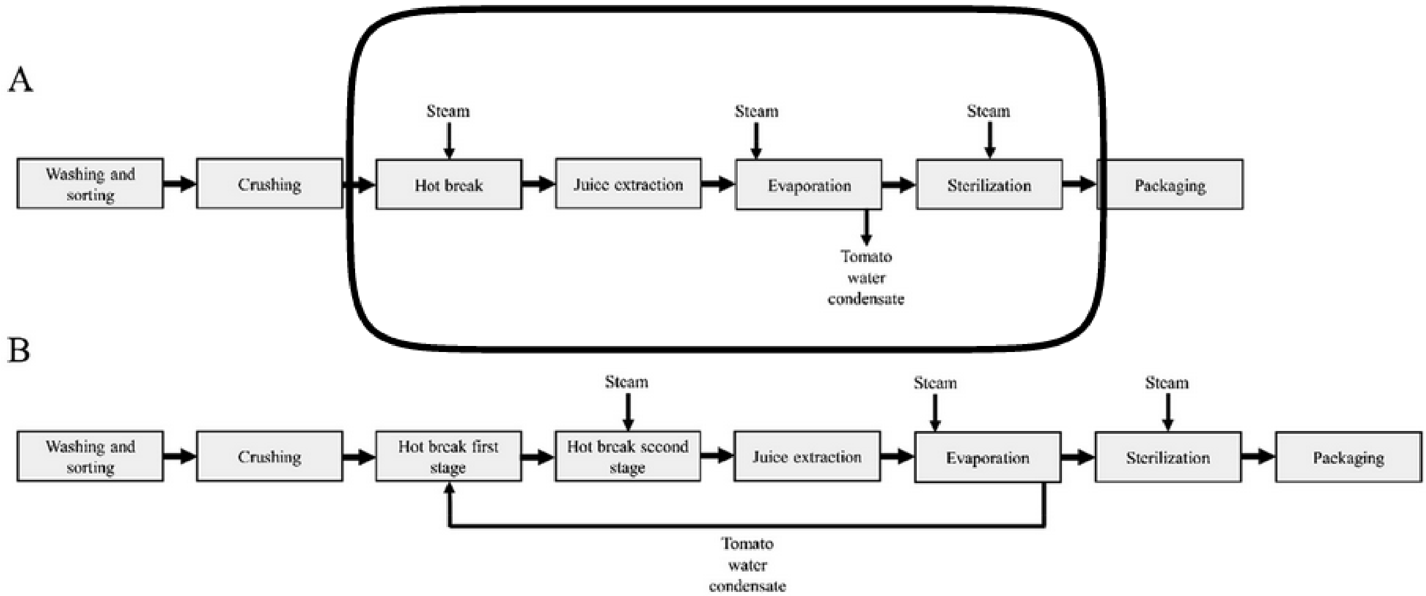


Fig. 1. Flow diagram of processes in tomato paste production under (A) conventional processing and (B) processing with recovered waste heat from tomato water condensate applied to the hot break.

Figure 6 Typical tomato paste food process v. recommended (Amán 2015)

This type of reuse had a cascading effect and allowed for four different sources of savings down the process chain. First off, the cost natural gas burned to create the steam was reduced. Additionally, the volume of water and its associated well pumping cost for steam production were also negated. Tomato water previously was sent to a cooling tower before being disposed of as sewage. Moreover, since the TWC would be recycled for its heat, the facility could avoid energy expenditures related to the cooling tower.

Furthermore, the study emphasized that less volume of water would be drained to wastewater facilities contributing to even more cost-benefits (Amán, Maulhardt, Wong, Kazama, & Simmons 2015).

It's clear that the sustainable strategies implemented produced large annual cost-savings, but each facility must analyze its own process by food type and facility constraints in order to make the best improvements. In summary, two universally important resources are of utmost importance to all production facilities: time and money. However, breaking down the constituents that contribute to slack time and unnecessary expenditures are key to streamlining unique processes to their highest potential. Improved operating efficiency and sustainable improvements are made possible with proper system analysis and adaptations.

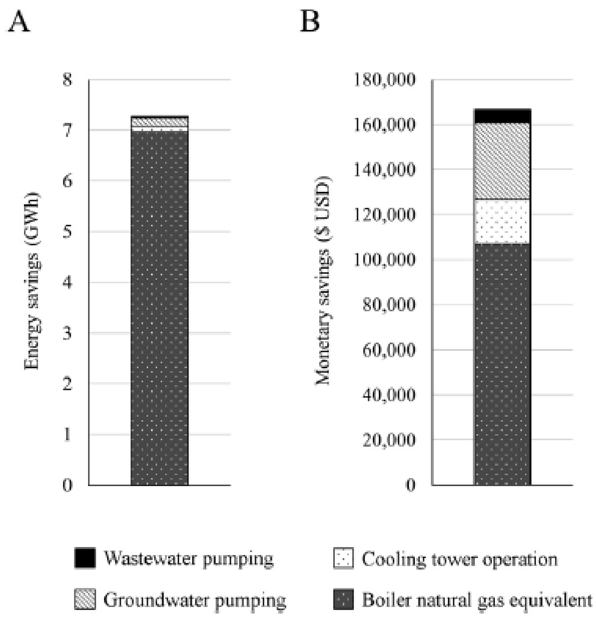


Fig. 2. Estimated seasonal energy and monetary savings from tomato water condensate waste heat recovery and reuse.

Figure 7 Energy savings from tomato water condensate recirculation and reuse (Amán 2015)

Constituents of Financial and Time-Based Constraints

Water-use

Industries can get their water from two categories of sources: public supply and/or groundwater. Water is used in many tasks of food processing: conveyance, cooking, cooling, sanitation, brining, and even as an ingredient in the final product. Food and kindred products manufacturing are among the top three most water consumptive industries in California rate at 1967 gallons per employee per day. As of year 2000, food processing (i.e. dairy, meat, fruit and vegetable, and beverage processing) constitutes 24% of all industrial water use as seen in Table 2 (Gleick 2003)

Table 2 Estimated 2000 Water Use in California's Commercial and Industrial Sectors in AF/year (provided by Pacific Institute)

Commercial Water Use (AF/Year)		Industrial Water Use (AF/Year)	
Schools	251,000	Dairy Processing	17,000
Hotels	30,000	Meat Processing	15,000
Restaurants	163,000	Fruit and Vegetable Processing	70,000
Retail	153,000	Beverage Processing	57,000
Offices	339,000	Refining	84,000
Hospitals	37,000	High Tech	75,000
Golf Courses	229,000	Paper	22,000
Laundries	30,000	Textiles	29,000
		Fabricated Metals	20,000
Unexamined Commercial	621,000	Unexamined Industrial	276,000
Total Commercial (a)	**1,852,000**	**Total Industrial**	**665,000**

For cooling processes, regional water availability and resupply rates will greatly influence where industrial users can expect to get water from. Specific to California, freshwater resources have become increasingly important and at times limited due to the transforming climate. Therefore, the largest consumptions of water, whether it is cooling or sanitation, should be evaluated for improvements.

Due to geographical and political differences between regions, the structure for water and sewage rates vary greatly. Sewage rates can be based on total volume, demand flow rate, TSS, BOD content, or any combination of these. Dependent on each food process, steps can be taken to optimize the cost of sewage based on a given rate structure. For example, if wastewater treatment plants only charged by TSS and not total volume, theoretically water could be used to dilute effluent, lowering the concentration seen at the wastewater facility, thus reducing sewage rates. Although this is an unrealistic scenario, this is the premise behind food processing system optimization as it relates to water-use.

Electricity

Electrical supply capacity, hours of operation, and building load profile are also common constraints in the food manufacturing and processing industry. With a general understanding of power generation, power distribution, and power quality, this allows us to understand why certain characteristics are considered in system selection.

It is important to keep in mind that electrical utility rates are determined by a provided a rate structure dependent on the following criteria: user-type, demand type, power quality, and time-of-usage. Typical peak hours of operation are from 12 P.M. to 6 P.M. as seen in Figure 8 below.

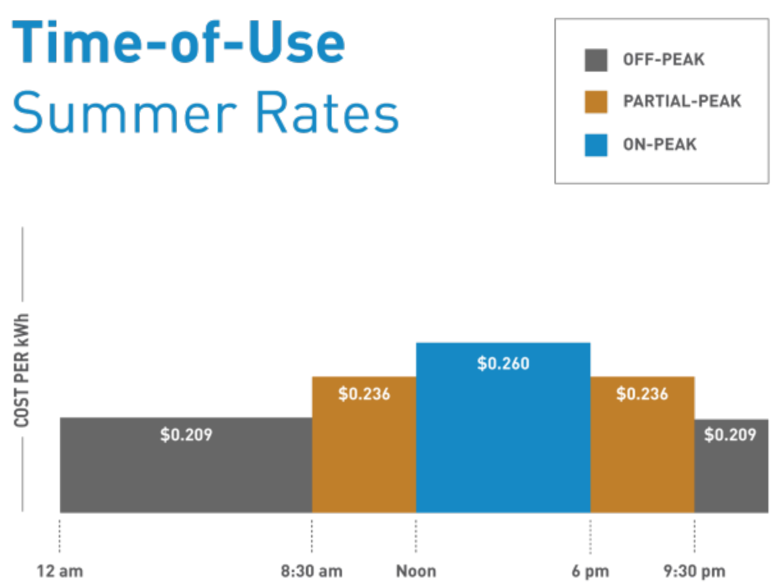


Figure 8 Sample rate structure based solely on time-of-day principle (provided by PG&E)

Though electricity consumption is a relevant variable, water –use efficiency will likely contribute to the greatest source of savings further influencing a system solution.

Fixed Assets

Among many resource constraints, existing facility processing equipment and buildings space availability are variables that system optimizers have the least control over. This is because there are many contingencies outside of cooling optimization that go into the way a facility best processes their food. As a response, sales and application engineers must be diligent to recognize this and propose system solutions that best integrate with existing food processing systems.

Capital goods such as equipment may include, but is not limited to the following: heat exchangers, cooking kettles, blanchers, storage vessels, filters, and pasteurizers. In the case of the given food processing facility, jacketed kettles are used to both cook and cool the food product. The Lee industrial kettles come with a variety of features including: agitators, uniflow jackets, lid-types, and hydraulics for tilting based on capacity. These special features and equipment capacities play an important role in the kettle cooling capacity calculation with appropriately applied assumptions.

The largest and generally the costliest form of capital is the industrial building itself. This is subject to the least amount of change when seeking to implement heat rejection solutions. Refrigeration equipment often consumes a large footprint of area and need to be selected with the necessary constraints in mind for future developments, current processing operations, and scheduled maintenances.

Additionally, minimum spatial requirements are specifically important to consider as well. Compromising the proper spatial requirements of certain heat rejection equipment is detrimental to its performance when the discharge air is recirculated back into the air inlet.

Finally, sound pollution to the surrounding environment could be an issue if facilities are located in residential areas. Proper sound attenuation equipment is available for equipment with noise produced by fans and compressors.

Prospective Operation

Future operation and expansion are also extremely important to consider in providing a system solution. A growing business may be looking into other relevant capital investments that could couple well with the system solution in mind. In the case of the given food processing facility, a Blentec food processor has been purchased to improve the mechanical cooling process. This will be taken into consideration when ATC proposes a system solution for phase 2 of the cooling process.

Refrigeration in Food Processing

Principles and operation of refrigeration technologies

Refrigeration equipment commonly takes advantage of the chemistry of selected refrigerant fluids in order to optimize heat transfer with minimized compressor power expenditure. Generally, most matter experiences three states: as a solid, a liquid, and/or a gas. Depending on the state variables they are prescribed (temperature, pressure, etc.), they can occupy any of these forms. Based on changes of these state variables, substances can experience phase changes going from liquid to gas, solid to gas, etc. Particularly, water has a relatively high latent heat of vaporization. This means that it will accept much more heat while changing phase to a vapor compared to many other fluids at atmospheric pressure. This makes water an effective medium for heat rejection at the right wet-bulb conditions.

Evaporative Cooling Systems

One prime example of an evaporative cooling system is the cooling tower. Cooling towers uniquely take advantage of water's chemistry capable of rejecting equivalent amounts of heat without the use of a compressor. Given certain parameters of relative humidity, dry-bulb, and wet-bulb temperatures, evaporative cooling may be applied. In order to identify the appropriate conditions for this kind of cooling, psychrometric charts are a tool used to approximate the feasibility for a given application.

Where water vapor pressure in the ambient air is greater, water molecules will be less likely to escape a droplet, thus reducing the extent of evaporation. Conversely, lower vapor pressure will contribute to more molecules escaping the confines of a droplet. Wet-bulb temperature serves as an indication for the efficacy of evaporative cooling which is dependent on atmospheric pressure and relative humidity at a given dry-bulb temperature. Dry-bulb only takes into account the gas constituents of air when measuring the average kinetic energy of a substance (i.e. temperature). Understanding the distinction between the dry-bulb and wet-bulb temperatures is important in order to understand when evaporative cooling systems can be implemented.

Facilities where the exit water temperature required is above the regional wet-bulb temperature could consider evaporative cooling as product solution to cooling (BAC 2015). An example of commonly recognized evaporative cooling equipment is the cooling tower as shown in Figure 9.

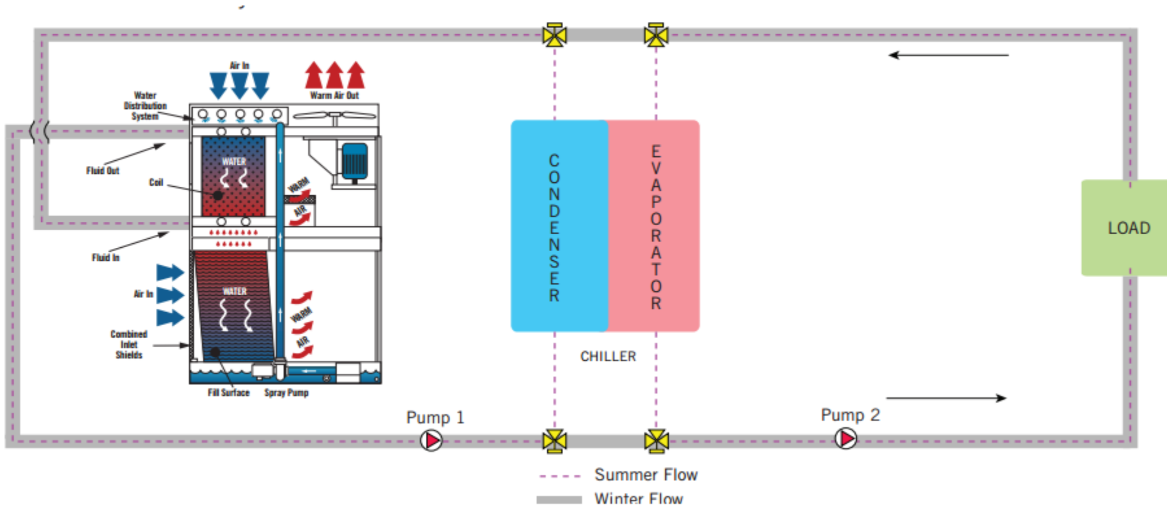


Figure 9 Typical Closed-circuit cooling tower (BAC Product Handbook Vol. 5)

As previously mentioned, cooling towers can be used in conjunction with water-cooled chillers or they be directly applied to the heat load. This equipment removes heat from surrounding air by allowing water and air to passing through channels or fill. Above the fill are water sprayers reducing droplet size with application across the entirety of the top layer of fill. We can expect the reduction in droplet size facilitates a more rapid mass and energy transfer between the air-water boundaries due to the increased droplet surface area-volume ratio. Additionally, the tortuous path the water must take increases the air-water interface time allowing extended evaporation to occur. The evaporation on the draining water rejects heat into the outside environment thus cooling the return water. Figure 10 below illustrates the relationship between the temperature of the water in the circuit and the air passing through the tower.

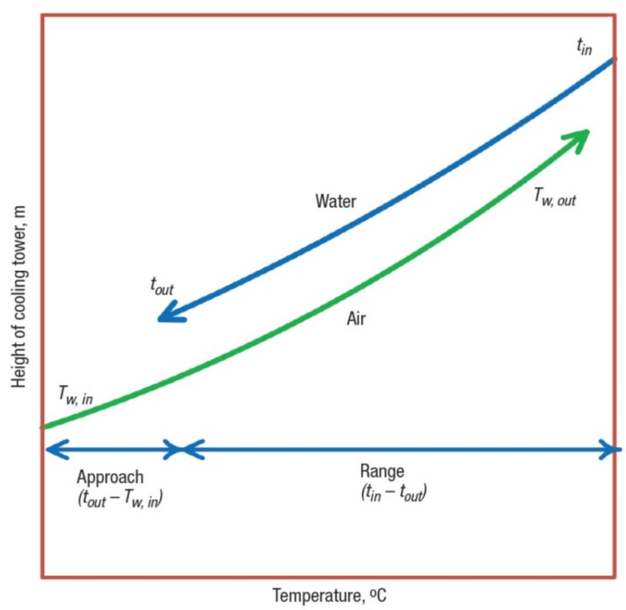


Figure 10 Air and Water Temperature through Tower Profile (Vengateson 2017)

Industrial fluid coolers, also known as closed-circuit cooling towers, utilize heat-exchanging coils to keep the process fluid in a separate circuit. The refrigerant fluid is either potable water or a water-glycol mixture.

The coined term for a cooling tower being the sole heat rejection medium is known as “free cooling”. This is because no mechanical cooling is used, resulting in little to no operating costs. Technically, input work is still required to pump water with associated pipe friction losses and sprayer pressures requirements. Additionally, work-input is required to run the fans for induced inlet air and heaters for defrost cycles in cold regions. However, these amounts are relatively small compared to the amount of input energy normally required to achieve the same level of cooling with a vapor-compression based chiller.

Vapor-Compression Refrigeration Systems

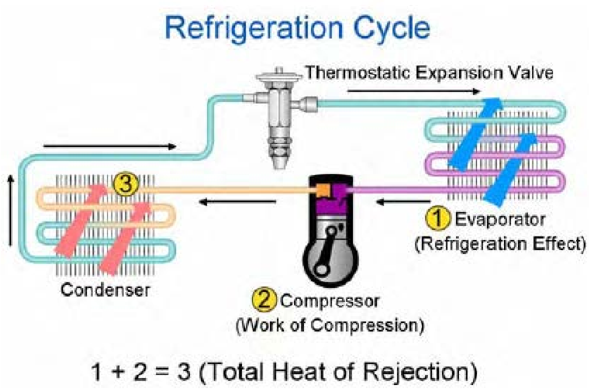


Figure 11 Diagram showing how heat rejection is achieved (provided by Carrier)

Figure 11 illustrates the various components of a vapor-compression refrigeration cycle. This is a popular cycle found in industrial refrigeration equipment. The compressors require great input horsepower in order to achieve those high pressures. The compressor provides the pressure necessary to ensure condensation through the next step for optimal heat rejection into the hot reservoir. In order to avoid high operating expenses, while cooling large loads, application engineers may select thermal energy storage technology over on-demand chillers.

On-demand chillers can provide for immediate blast chilling, but often have high associated energy consumption costs. If operations take place during peak-hours of the electrical grid, this can result in exponentially higher utility rates. Secondly, if a central system chiller needed maintenance, there would not be a reserve source of cooling to keep operation going. Finally, implementing a design with a singular chiller would be difficult in order to meet the varying demand from a food processing facility and from the changes in atmospheric climate. Strategies to avoid these problems require application engineers to explore load shifting strategies and technologies by utilizing thermal energy storage.

Case Study: Cheese Production Facility – Load Shifting

A study on a cheese production facility in Hanford, California serves a great example to the advantages of load shifting with thermal energy storage. Designing a system to build ice overnight when rates are low and demand is not immediately necessary allows the cooling requirements to be appropriately satisfied during operating hours. A multitude of thermal energy storage options were considered, but a dynamic, slurry thermal energy storage system was chosen with 130% reserve storage capacity in the case of scheduled maintenance. In addition, this particular solution did not require a defrosting cycle with its ice insulation-effect inhibiting design. Finally, the dynamic TES system allows for ice storage and production to be separated.

Static systems are designed with the intent that the ice formed around the coils remains until it is melted externally from warm water downstream of load or internally from the coil. However, the dynamic system results in the higher cooling capacity for the short durations. There are two reasons for this: 1) the high surface area of the slurry ice particles and 2) the direct contact of warm water from the load with the slurry ice (Gladis 1997).

These principles of heat transfer and load shifting strategies are pivotal to providing the best long-term solution for a given cooling process like those at the given food processing facility. A deeper look into client expectations, facilities operations, and available technologies will provide for more beneficial, long-term solutions.

HVAC&R Industry Terminology and Thermodynamic Principles

Cooling capacity and total heat rejection are other constraints to be specified in this project. In order to properly analyze a cooling method, it is important to understand the difference between an open system and a closed system in thermodynamics. An open system is where mass and energy crosses the thermal boundary thus displacing heat into surroundings. However, a closed system allows heat energy to cross the boundary, but no mass. These fundamental thermodynamic definitions of systems are the framework behind many refrigeration system technologies.

Furthermore, cooling capacity is defined by tons of refrigeration or the amount of energy required to completely freeze a ton of liquid water at 0 degrees Celsius in a period of 24 hours. In other words, cooling capacity is the amount of transferred heat energy that is capable of being rejected over a certain period of time. This is the HVAC&R industry's standard unit for measuring rate of heat rejection between temperature regions across a certain boundary.

Heat Rejection Performance – Vapor-Compression Systems

Measuring thermal performance is different for each piece of heat rejection technology. For example, chillers using the vapor-compression cycle measure this efficiency with the COP from Equation 1.

$$COP\ (refrigeration) = \frac{Q_{\ in}}{W_c}, \text{ Eqn. 1}$$

$Where,$
$Q_{\ in} = heat\ gained\ through\ the\ evaporator\ coils, kJ\ or\ BTU$
$W_c = work\ expended\ by\ the\ compressor, kJ\ or\ BTU$

The vapor-compression process can be seen in Figure 12 with the pressure-enthalpy diagram. This measure of efficiency accounts for the amount of heat absorbed through the evaporator coil per unit of energy expended by the compressor. It should be noted that this is a theoretical value and could largely misrepresent realistic COP values due to irreversibility of the operation in actual performance. Figures 13 illustrates the difference between ideal and actual operation.

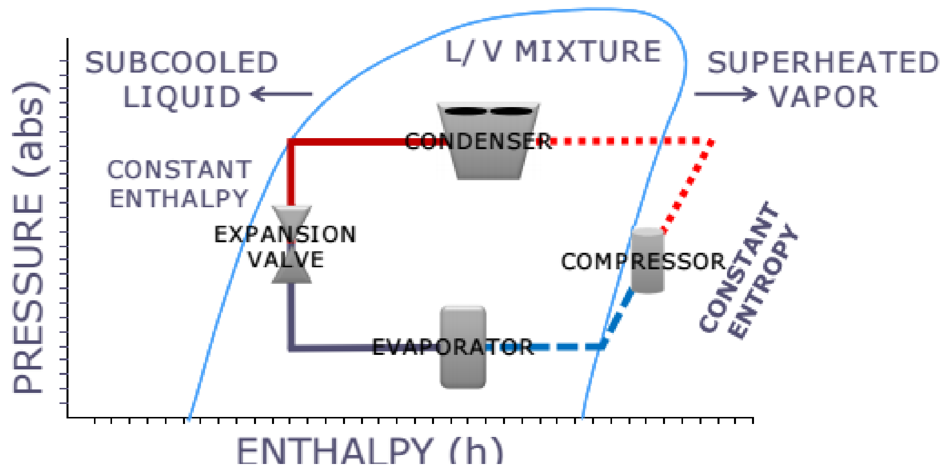


Figure 12 Ideal refrigeration cycle

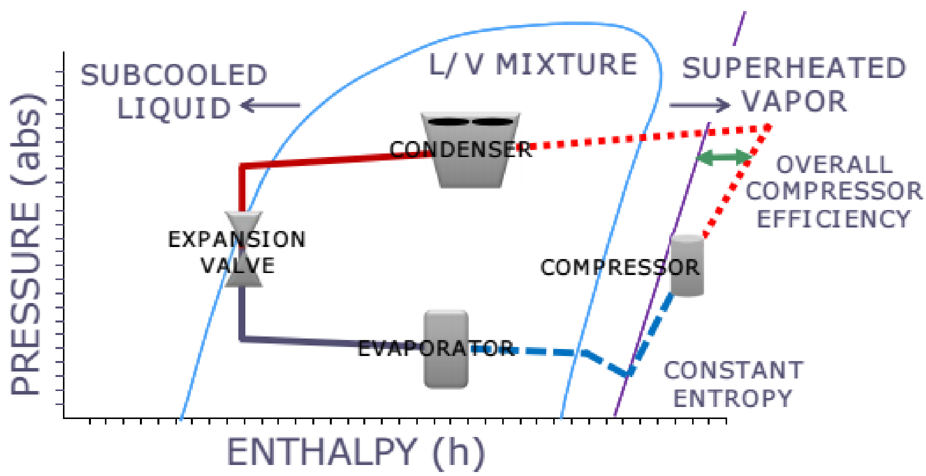


Figure 13 Actual Refrigeration Cycle

Figure 14 below illustrates the enthalpy values at different points in the refrigeration cycle and how they are used to calculate total heat rejection and compressor work.

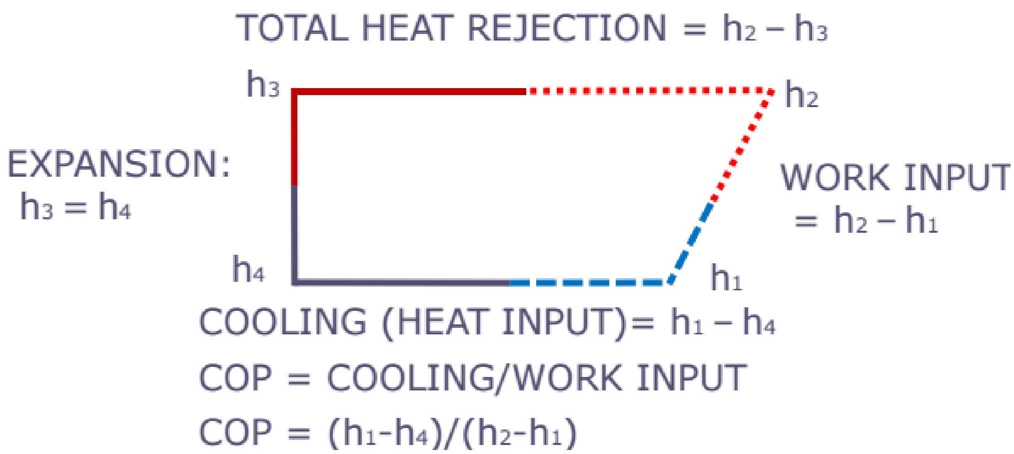


Figure 14 Refrigeration cycle with highlighted enthalpy points

Heat Rejection Performance – Evaporative Systems

Cooling towers are a prime example of an evaporative cooling system. They are often used in conjunction with chillers to absorb the heat from condenser side. This allows for the heat energy to be transferred into the atmosphere leveraging the latent heat of vaporization of water. The thermal performance or efficiency of a cooling tower is based on the following equation:

$$\mu = \frac{(t_i - t_o)}{(t_i - t_{wb})} * 100, \mathrm{Eqn.}\ 2$$

Where,
μ = Cooling tower efficiency, %
t_i = inlet temperature of water to the tower, °F or °C
t_o = outlet temperature of water from the tower, °F or °C
t_{wb} = wet bulb temperature, °F or °C

Simply put, there are two primary things that contribute to improved cooling tower thermal efficiency: range and approach. Range is the difference between the entrance and exit temperatures of the recirculating water. Approach is the proximity of the exit water's temperature to the regional wet-bulb temperature.

This measures of efficiency neglect to factor the energy and water consumption required to accomplish this cooling effect. Furthermore, a decision matrix with cost and unique features should be used to compare the heat rejection options to best meet a given project's definition of success.

While the pressure-enthalpy diagram is useful for measuring the heat rejected in chillers, cooling towers use the psychrometric chart for that information. In order to extract useful information from the psychrometric chart, we must know three independent variables: barometric pressure, dry-bulb temperature, and a value that indicates the concentration of vapor in the air (wet-bulb temperature, humidity ratio, water vapor pressure, dew point, etc.) Table 3 indicates the variables found on the psychrometric chart and how they are obtained.

Table 3 Variables from the Psychrometric Chart (provided by Green Building Advisor)

Quantity	Symbol	Measurable?
Dry bulb temperature	T_{DB}	Yes
Wet bulb temperature	T_{WB}	Yes
Dew point temperature	T_{DP}	Yes
Barometric pressure	p_{bar}	Yes
Water vapor pressure	p_{WV}	No
Relative humidity	RH	Yes
Specific enthalpy	h	No
Specific volume	v	No
Humidity ratio	W	No

The psychrometric chart is a tool used to identify the properties of air based on the amount of water vapor present. With three independent variables and the correct assumptions, this tool allows both the heat rejection and the volume of water evaporated to be calculated for a given cooling tower.

The psychrometric chart below illustrates the sensible and latent heat absorbed by the air as it passes through the tower's fill to the outlet.

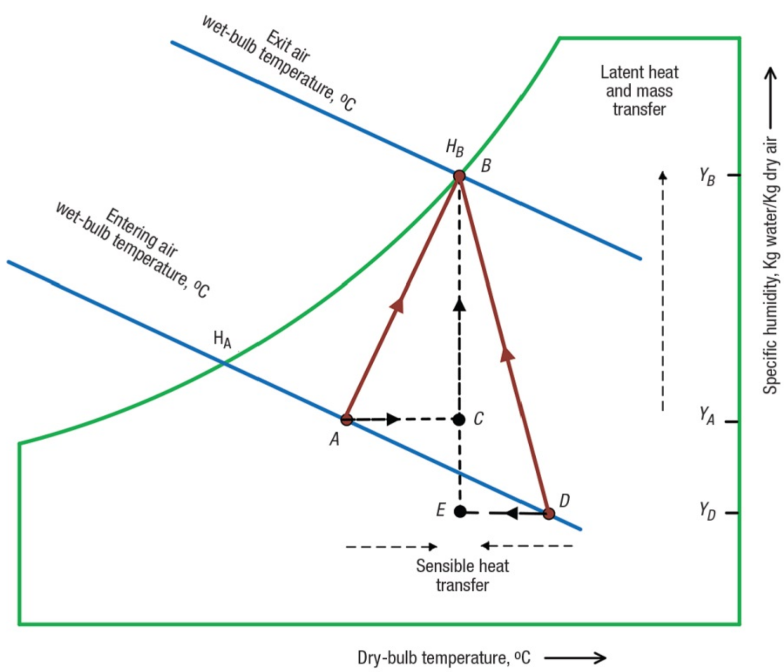


Figure 15 Latent and sensible components of total heat rejection vector

With value for air flow and two known points, the volume of water evaporated from the cooling tower can be computed based on the change in specific humidity or humidity ratio. The increase in the humidity ratio represents the water vapor absorbed by the air as it passes through the cooling tower. In most circumstances, it is safe to assume that air exits the cooling tower at saturated conditions or 100% relative humidity as seen in Figure 15.

Path A-C-B represents conditions where the ambient air is below the temperature of the tower inlet water. Path D-E-B represents the opposite situation, which greatly affects the amount of water the equipment can expect to evaporate for the same amount of total heat rejection. Changes in wet-bulb temperature are the primary drivers for changes in specific enthalpy, which indicate the cooling effect of the equipment. The similarity between the slope of specific enthalpy and the slope of wet-bulb lines verify that strong relationship in Figure 16 below.

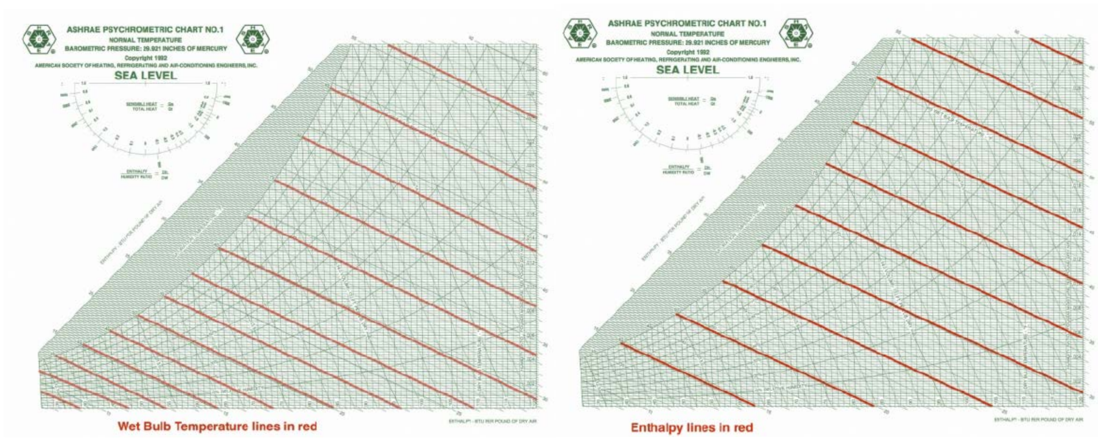


Figure 16 Wet bulb temperature and specific enthalpy lines respectively

Establishing proper dry-bulb conditions for accurate estimates of cooling tower water usage and heat rejection are imperative and require justification.

PROCEDURES AND METHODS

Initial Considerations

Regional Weather Conditions

For the given food processing facility, the current cooling process of 70 F water is being routed through to cool the kettles. Additionally, regional wet-bulb temperatures of 67.5 F indicate that Santa Rosa climate provides favorable conditions for evaporative cooling in this application. These wet-bulb conditions are guaranteed for 99% of the year while 3 to 4 days of the year are expected to have wet-bulb temperatures above this temperature (BAC 2015). Weather data indicates that annual dry-bulb temperature is 71.3 F while higher-end temperatures in summer months reach approximately 80 F (US Climate Data 2017). This information indicates that cooling towers should be considered for a refrigeration solution.

Manufacturer

Among the many cooling tower manufacturers, Baltimore Air Coil Company will be the equipment manufacturer ATC is representing. BAC offers industry leading technology for both open and closed-circuit cooling towers. In order to determine which should be selected, a decision matrix was developed in order to quantitatively determine between tower types. Based on a series of meetings, the information of qualitative features the client was looking for were be translated into quantitative values. The raw scores were developed according to the client's priority levels of the objectives. Table 4 below indicates which of the two would be selected.

Table 4 Decision Matrix for Open or Closed Circuit Cooling Tower

OCCT or CCCT

Raw Scoring Matrix

1-3 where, 1 = poor, 2 = satisfactory, 3 = excellent

	Food Safety	Food Quality	Capacity Increase	Energy Savings	Treatment Savings	Product Cost
OCCT	2.00	2.00	2.00	3.00	2.00	2.00
CCCT	3.00	2.00	2.00	2.00	3.00	2.00

Importance Factor

0-1 where, 1 = highest importance, 0 = lowest importance

	Food Safety	Food Quality	Increase Capacity	Energy Savings	Treatment Savings	Product Cost
	1.00	1.00	0.50	0.50	0.75	0.50

Weighted Scoring Matrix

	Food Safety	Food Quality	Increase Capacity	Energy Savings	Treatment Savings	Product Cost	**Total**
OCCT	2.00	2.00	1.00	1.50	1.50	1.00	**9.00**
CCCT	3.00	2.00	1.00	1.00	2.25	1.00	**10.25**

A closed-circuit cooling tower has been determined as the better category for a refrigeration solution at the given food processing facility. While the OCCT provides reduced energy costs, it compromises food safety by having an open circuit make contact with food processing equipment inside the facility. Furthermore, the closed-circuit cooling tower has associated water savings because the water does not need to be treated. With this established, a more detailed selection process of which model industrial fluid cooler (i.e. CCCT) will be selected.

There are six main models of cooling towers to select from the BAC provider. The towers vary primarily based on the fan system and the single cell heat rejection capacity ranges. Each product provides unique features which will be outlined in Figure 17 and 18 below.

Manufacturer Product Line

	FXV	FXV DUAL AIR INTAKE	PFI
Model			
Flow and Fan System	Induced Draft Combined Flow, Axial Fan	Induced Draft, Combined Flow, Axial Fan	Induced Draft, Counterflow, Axial Fan
Single Cell Capacity Range	29 - 424 Nominal Tons* 87 - 1,272 USGPM at 95°F to 85°F at a 78°F	344 - 624 Nominal Tons* 1,032 - 1,872 USGPM at 95°F to 85°F at a 78°F	18 - 360 Nominal Tons* 54 - 1,080 USGPM at 95°F to 85°F at a 78°F
UNIQUE FEATURES	• Advanced Coil Technology minimizes scaling and fouling potential • Combined Flow Technology increases cooling capability • Large access doors for easy maintenance access • Single-point wiring for motors and vibration cutout switch • Independent Fan and redundant pump options • Pre-assembled platform packages • Shake table tested up to S_{DS} of 2.4g	• Advanced Coil Technology minimizes scaling and fouling potential • Combined Flow Technology increases cooling capability • Ideal for large tonnage applications • Highest single cell capacity	• Ideal competitor replacement unit • Patent-pending OptiSpray™ Technology lowers operating costs • OptiCoil™ System • Dry operation mode for severe weather operating conditions • Independent Fan and redundant pump options • Shake table tested up to S_{DS} of 3.75g • Single-piece rigging

Figure 17 Closed Circuit Cooling Tower Product Line Set 1

VF1	VFL	HXV HYBRID TOWER	
			Model
Forced Draft, Counterflow, Centrifugal Fan	Forced Draft, Counterflow, Centrifugal Fan	Induced Draft, Combined Flow, Axial Fan	Flow and Fan System
4.1 - 543 Nominal Tons* 12.3 - 1,629 USGPM at 95°F to 85°F at a 78°F	3.9 - 108 Nominal Tons* 11.7 - 324 USGPM at 95°F to 85°F at a 78°F	160 - 305 Nominal Tons* 480 - 915 USGPM at 95°F to 85°F at a 78°F	Single Cell Capacity Range
• Centrifugal fan for indoor and ducted installations • Moderate dry operation for systems that benefit from dry operation • Optional finned coils to enhance dry operation mode	• Low profile to met low height requirements • Centrifugal fan for indoor and ducted installations • Moderate dry operation mode for systems that benefit from dry operation • Optional finned coils to enhance dry operation mode • Single-piece rigging	• Hybrid technology - using sensible, adiabatic, and evaporative heat transfer - for water conservation up to 70% over conventional evaporative products • Maximum dry operation for systems that benefit from dry operation • Suitable for high temperature process fluids greater than 180°F	UNIQUE FEATURES

Figure 18 Closed Circuit Cooling Tower Product Line Set 2

It is important that we calculate the minimum tonnage required to see which of the cooling towers will be needed. To estimate initial feasibility, capacity ranges in the product literature are based on typical conditions where inlet water is 95 F, outlet water is 85 F, and the wet bulb temperature is 78 F. The associated tonnages ranges are based on a nominal 3 gpm per ton of cooling. Selection software will tailor the expected performance of a cooling tower for a design day in Santa Rosa. A “design day” includes information about the cooling process load and the prescribed atmospheric conditions.

The unique features of each tower indicate where their application would best be suited. The PFI towers are designed to operate in climates with high seasonal temperature variation with anticipated dry operations during the winter. These features disqualify the PFI as a potential solution because it would be difficult to justify the inflated cost in a temperate region as Santa Rosa. Additionally, the models equipped with centrifugal fans are used to overcome static head caused by indoor installation and the ductwork. Because of this feature, models with centrifugal fans have higher energy consumption making it irrelevant for the outdoor application at the given food processing facility. Finally, the HXV hybrid tower can be disqualified as a solution because they are best suited for applications with process fluids at temperatures greater than 180 F. This leaves two potential system solutions: the FXV and dual air intake FXV as seen in Figure 17 above. In order to determine which of the two is more feasible, we must calculate the required tons of refrigeration from the current process and develop “design day” parameters for the selection of new equipment.

With an understanding of company priorities and the facility’s current cooling process, the following steps were taken to develop the best system solution:

1. Compile Kettle Cooling Testing Data
2. Calculate the current, total heat transferred out of the product, Q_{out}(BTU)
3. Calculate the current, rate of heat transfer out of the product, $\dot{Q}_{out}$(BTUH or R Tons)
4. Determine the final water temperature of the once-through cooling system,T_f (°F)
5. Establish design day criteria and troubleshoot inputs
6. Select industrial fluid cooler model using BAC selection software
7. Conduct an engineering economic analysis to project returns on investment
8. Conduct a sensitivity analysis to determine integrity of project

Kettle Testing Background

The plant manager at the given food processing facility conducted kettle tests where a temperature probe measured the changes in temperature over the course of approximately two hours at given kettle inlet water temperatures. This raw data collection would allow us to complete the first step to provide the best system solution. Table 5 and Figure 19 are the components of a sample kettle cooling test. A complete summary of kettle testing data can be found in Appendix E.

Figure 19 Kettle No. 18, 500-gal jacketed capacity fitted with agitator

Table 5 Sample Kettle Cooling Test

Test 4, Kettle 18 (Agitator 19 RPM)					
Date	**WIP**	**Back Lid**	**Bypass Valve**	**Flow Rate (GPM)**	**Waste %**
9/29/2015	631219	Open	Open	15	100%
Time	**WIP Temp**	**Cooling Rate**	**Gallons Used**	**Gallons Wasted**	**Gallons Recycled**
8:20 AM	206° F		0	0	0
8:35 AM	158° F	48° F	225	225	0
8:50 AM	130° F	28° F	450	450	0
9:05 AM	117° F	13° F	675	675	0
9:20 AM	106° F	11° F	900	900	0
9:35 AM	97° F	9° F	1125	1125	0
9:50 AM	91° F	6° F	1350	1350	0
10:05 AM	86° F	5° F	1575	1575	0
10:20 AM	82° F	4° F	1800	1800	0
10:35 AM	79° F	3° F	2025	2025	0
2:15	127° F	14.11° F			

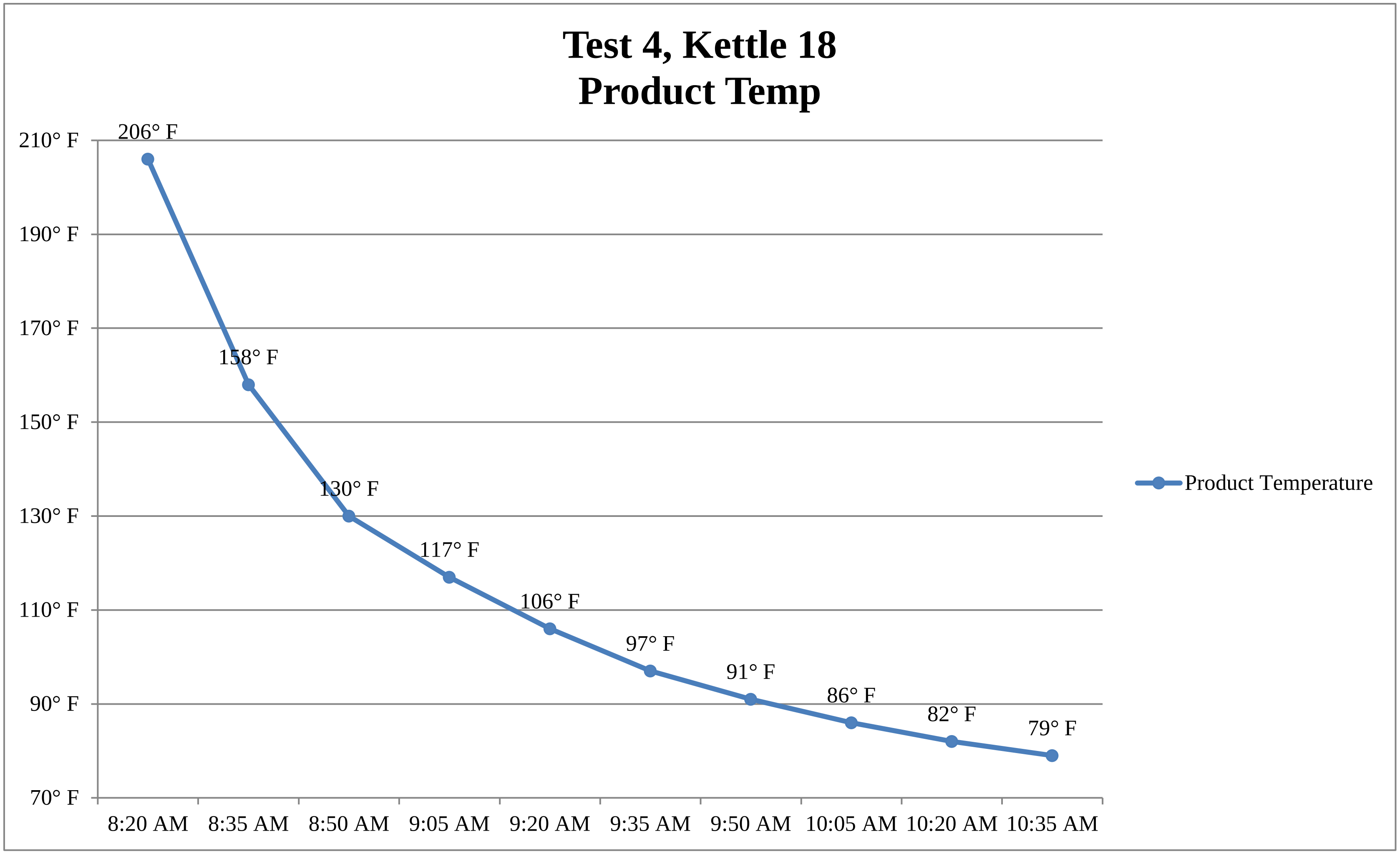


Figure 20 Temperature Profile of Representative Kettle Cooling Test

RESULTS AND DISCUSSION

Compilation of Kettle Testing Data

The kettle cooling test information was collected and organized in a fashion that gave us information about the behavior of the product temperatures as water flowed through the jackets over a certain period of time. That period of time represents the amount of time normally consumed in order to achieve desired cooling of the food product.

Table 6 Summary of Kettle Testing Data with various conditions

Condition	Initial Temp (Water)	Initial Temp (Product)	Final Temp (Product)	Change of Temp (Product)	Total Hours	Jacket Water Flow	Total Water Used	Jacket Water Flow
Average All Tests	70	199	85	-114	2.07	≈15	1799	870
	F	F	F	F	hrs	gpm	gal	gph
Extreme All Tests	73	208	76	-132	1.50	15	1350	900
	max F	max F	min F	delta F	min hrs	max gpm	gal	gph
Representative Test 4, No. 18	70	206	79	-127	2.25	15	2025	900
	F	F	F	F	hrs	gpm	gal	gph

Conclusions from the kettle cooling tests indicate three different process-cooling conditions: average, extreme, and representative conditions. Average conditions were based on averages taken from over 9 sets of kettle tests involving a variety of kettle capacities (i.e. 500 gal, 400 gal, 300 gal). Extreme conditions were based on the highest initial product temperature, lowest product final temperature, warmer inlet water, and the minimum cooling time. The large value heat rejection value from the control-mass thermodynamic problem will explain why the specific temperature, duration, and flow values were chosen. Finally, the representative condition includes a kettle that undergoes a testing process involving a common kettle capacity with expected temperature ranges based on what the head cook has experienced in the past. For these criteria, Kettle No. 18 from Test 4 was deemed representative as seen in Figure 19, Table 5, and Figure 20 above. The variation of values in the data set may result from varying flows through the kettles, size variance in kettles, and varying day-to-day water inlet temperatures. The following considerations should be included in future kettle cooling tests to ensure that readings are most representative:

- Use proper flow measurement and logging technology for cooling water
- All solutions should be agitated at the same speed
- Constituents and characteristics of food product per kettle test should be documented
- Location of the temperature probe should be documented

- Human errors in testing procedures should be documented
- Inlet water temperatures for each kettle test should be documented
- Measurement of product weight should be documented

Developing the Energy Balance for the Closed-System

The following assumptions were made in order to produce a feasible value of tonnage required:

- The food product is contained primarily within the hemispherical portion of the kettle
- Food product is filled to 60% of total kettle capacity
- Sensible heat transfer between food product and cooling water
- Heat rejected from walls of steel kettle to surrounding atmosphere negligible
- Kettle lid was closed, creating a closed system
- Heat primarily conducted through steel jacketed portion
- Isochoric process
- 15 gpm of water flow per kettle

Guided by the zeroth law of thermodynamics, heat transfer can be expected from a variance in temperatures across the boundary. Additionally, the first law of thermodynamics states that energy is neither created nor destroyed, but transferred or transformed instead. This transfer occurs to/from the system, through a defined boundary, from/to a surrounding environment. It is important to note that only changes in energy can be readily observed with this simplicity. Figure 19 below illustrates the schematic used to conduct energy balance computation.

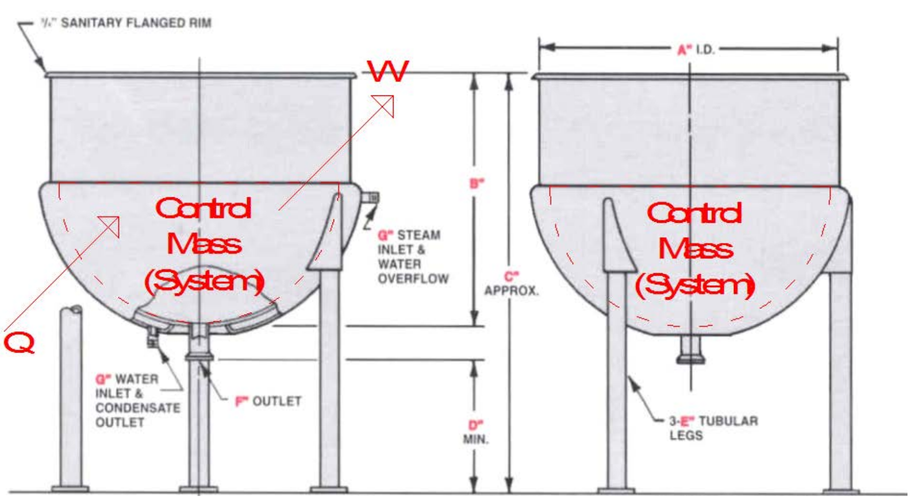


Figure 21 Schematic of control mass for closed-system calculation

For a proper deduction of heat rejected, the fundamental energy balance should be used. The simplified energy balance equation for a closed system is denoted as follows:

$$\Delta E_{total} = \Delta U + \Delta KE + \Delta PE \text{ , Eqn. 3}$$

$$\begin{aligned}
&Where,\\
&\Delta E_{total} = net\ change\ in\ energy\ of\ system, BTU\\
&\Delta U = net\ change\ in\ internal\ energy\ of\ system, BTU\\
&\Delta KE = \ net\ change\ in\ kinetic\ energy\ of\ system\ , BTU\\
&\Delta PE = in\ potential\ energy\ of\ system, BTU
\end{aligned}$$

We know that the system is not changing in kinetic or potential energy, therefore:

$$\Delta E_{total} = \Delta U \ = Q_{system} \ - W_{system}, \text{Eqn. 4}$$

$$\begin{aligned}
&Where,\\
&\Delta E_{total} = total\ change\ in\ energy\ of\ system, BTU\\
&\Delta U \ = change\ in\ internal\ energy\ of\ system, BTU\\
&Q_{system} = heat\ exchanged\ into\ system, BTU\\
&W_{system} = work\ exchanged\ out\ of\ system\ , BTU
\end{aligned}$$

Sign convention for heat and work into system are opposites. While heat flowing in the system is represented as a positive number, work into system is represented by a negative number. The schematic illustrates heat and work and their associated directions.

Since this is an isochoric process (i.e. constant volume), we assume the fluid does no work to the system. Another assumption is that the lids on the kettles are kept closed during the cooking/cooling process. Therefore, this creates a fully "closed" system where no mass crosses the boundary. The change in total energy of the fluid can be described by:

$$\Delta E_{total} = \Delta U \ = Q_{system} = 0 \ = \ Q_{in} - Q_{out}, \text{Eqn. 5}$$

Since there is no heat flow into the product, which produces:

$$\Delta E_{total} = -Q_{out}$$

$$-Q_{out} = \ -Q_{Product\ (system)} = m\ c\ \Delta T, \text{Eqn. 6}$$

$$\begin{aligned}
&Where,\\
&Q_{product} = heat\ rejected\ from\ food\ product, BTU\\
&\Delta T = difference\ between\ final\ and\ initial\ product\ temperature, \text{°F}\\
&m = \ mass\ of\ food\ product, lb\\
&c_{beans} = specific\ heat\ of\ representative\ food\ product, \frac{BTU}{lb\text{°F}}
\end{aligned}$$

Using the mass of our product, a representative heat capacity, and a best condition for temperature change, we can solve for the heat rejected out of the kettles. The following results were obtained by calculating the heat rejected with all three of those conditions. The kettles cook a large variety of products including, but not limited to: vegetables, black-eyed peas, sauces, beans, and some various chili recipes. Beans provided a representative heat capacity and bean density was used to estimate the mass of product in the kettles. A representative mass of 2700 lbs was calculated. Additionally, eight 500-gallon kettles were used to represent a design day heat load. Since each of our kettles requires 15 gpm of water flow, we can expect a total required refrigerant flow of 120 gpm. The following tables contains the results of our calculations.

Total Current Heat Transfer from Product

Table 7 Total heat rejection for each condition

Q product	Per Kettle	All 8 Kettles
Condition	BTU	BTU
Average	-277,000	-2,216,000
Extreme	-321,000	-2,568,000
Rep.	-309,000	-2,472,000

Current Rate of Heat Transfer from Product

The rate of heat transfer is dependent on the amount of time the kettle tests consumed. Most of the tests were around two hours. Test durations will be deduced according to their corresponding condition.

Table 8 Rate of heat rejection for each condition

Q rate	Duration	Per Kettle	All 8 Kettles	Per Kettle	All 8 Kettles
Condition	hrs	BTUH	BTUH	R Tons	R Tons
Average	2.07	-134,000	-1,072,000	-11	-89
Extreme	1.50	-214,000	-1,712,000	-18	-143
Rep.	2.25	-138,000	-1,104,000	-12	-92

Among these three conditions, the average condition was determined to be the best information to use for the selection software. With knowledge of the rate of heat rejection, various methods can be used to determine the temperature of the water leaving the kettle jacket.

Determining Final Water Temperature from Kettle Jacket

There were two methods used in order to determine the final water temperature: using cooling tower equations and using steam tables with control-volume analysis.

Method 1

With proper rearrangement, Equation 7 is used to determine the final water temperature.

$$\Delta T = T_f - T_i = \frac{\dot{Q}}{L_{in} * c_w}\text{ , Eqn. 7}$$

$Where,$
$\dot{Q} = rate\ of\ heat\ rejected, \frac{BTU}{hr}$
$\Delta T = the\ difference\ between\ the\ water\ inlet\ and\ outlet\ temperature,$ °F
$L_{in} = mass\ flow\ rate\ of\ the\ water, \frac{lb}{hr}$
$c_w = specific\ heat\ of\ water, \frac{BTU}{lb°\mathrm{F}}$

Since the assumption was made that the only significant heat transferred is to the jacket water, the water flows in a fixed volume, the following equivalence can be developed:

$$-\dot{Q}_{product} = \dot{Q}_{jacket\ water}\text{, Eqn. 8}$$

Water in this temperature range was considered relatively incompressible so a single value for density could be used. Using Equation 8 above, the final temperature of the water was calculated to be 88 F. This final water temperature value makes sense because our food product and water are approaching thermal equilibrium. We can verify this is a reasonable value by cross-referencing with the T-v diagram and steam tables for water.

Method 2

Using control-volume equation and properties of the initial condition, the final temperature of the water can be determined. Figure 22 below is a schematic for the given problem:

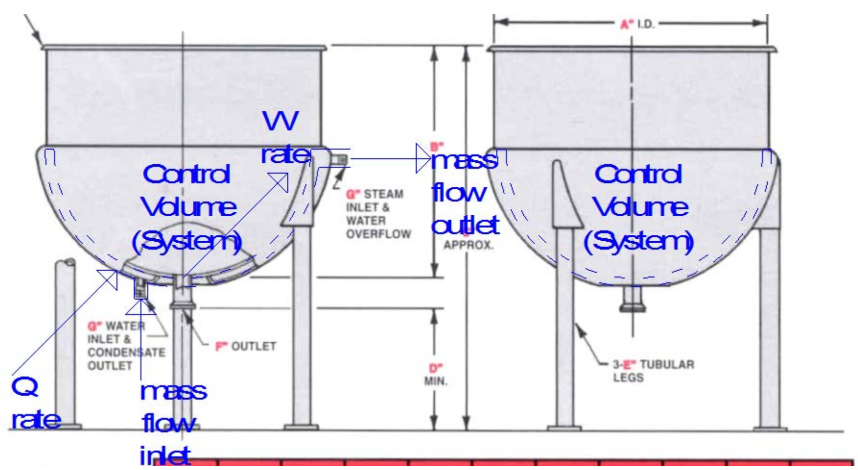


Figure 22 Control-volume schematic

The following assumptions must be applied for simplified analysis:

- Steady-state operation
- Conservation of mass principle
- Liquid is relatively incompressible
- Inlet and outlet diameters are equivalent
- No significant change in potential energy from the inlet and outlet
- No significant change in kinetic energy from the inlet and outlet
- No phase change
- Isochoric process

Due to the behavior of water in the compressible liquid region, the enthalpy for a liquid can be well represented by the enthalpy for a saturated liquid at a given temperature. Figure 23 indicates the unique behavior of compressible liquids that justify the use of the saturated liquid values for the steam tables.

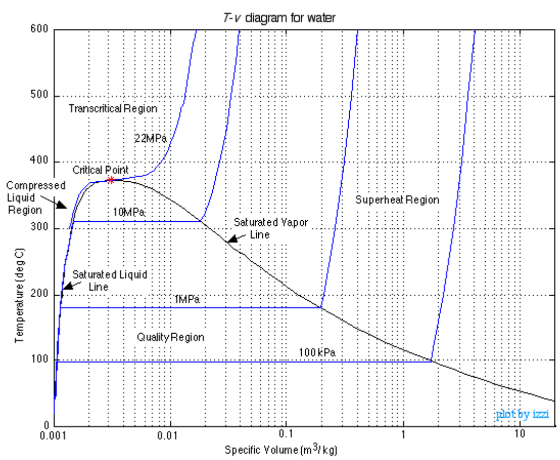


Figure 23 T-υ diagram for Water (provided by Ohio University)

Initially, the control-volume equation is represented as:

$$\frac{\partial E}{dt} = 0 = \dot{Q}_{CV} - \dot{W}_{CV} + \dot{m}[(h_i - h_e) + .5({V^2}_i - {V^2}_e) + g(z_i - z_e) \text{ Eqn. 9}$$

$$Where,$$
$$\frac{\partial E}{dt} = change\ in\ energy\ over\ time, \frac{BTU}{s}$$
$$\dot{Q}_{cv} = rate\ of\ heat\ rejected\ into\ control\ volume, \frac{BTU}{s}$$
$$W_{CV} = rate\ of\ mechanical\ work\ out\ of\ control\ volume, \frac{BTU}{s}$$
$$\dot{m} = mass\ flow\ rate\ of\ the\ water, \frac{lb}{s}$$

$$c_w = specific\ heat\ of\ water, \frac{BTU}{lbF}$$
$$h_i\ = enthalpy\ inlet, \frac{BTU}{lb}$$
$$h_e = enthalpy\ at\ outlet, \frac{BTU}{lb}$$
$$V_i = velocity\ at\ inlet, \frac{ft}{s}$$
$$V_e = velocity\ at\ exit, \frac{ft}{s}$$
$$g = gravitational\ constant, \frac{ft}{s^2}$$
$$z_i = elevation\ at\ inlet, ft$$
$$z_e = elevation\ at\ exit, ft$$

With the applied assumptions, the equation can be simplified to the following:

$$\frac{\partial E}{dt} = 0\ =\ \dot{Q}_{CV} +\ \dot{m}(h_i\ - h_e), \text{Eqn. 10}$$

Since we know two independent properties, inlet temperature and inlet state, we can use the steam tables. Since the saturated liquid can represent the enthalpy values for compressible liquids, a value can be interpolated from the table for the initial enthalpy. Equation 10 above can then be used to determine the exit enthalpy. Finally, another interpolation including both temperature and enthalpies for the inlet and outlet reveals an exit water temperature of 87.7 F.

Table 9 Steam tables for saturated water (provided by Wiley 2011)

		Specific Volume ft^3/lb		Internal Energy Btu/lb		Enthalpy Btu/lb		
Temp. °F	Press. $lbf/in.^2$	Sat. Liquid v_f	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g
64	0.2952	0.01604	1056	32.09	1031.8	32.09	1057.3	1089.4
66	0.3165	0.01604	988.4	34.09	1032.4	34.09	1056.2	1090.3
68	0.3391	0.01605	925.8	36.09	1033.1	36.09	1055.1	1091.2
70	0.3632	0.01605	867.7	38.09	1033.7	38.09	1054.0	1092.0
72	0.3887	0.01606	813.7	40.09	1034.4	40.09	1052.8	1092.9
74	0.4158	0.01606	763.5	42.09	1035.0	42.09	1051.7	1093.8
76	0.4446	0.01606	716.8	44.09	1035.7	44.09	1050.6	1094.7
78	0.4750	0.01607	673.3	46.09	1036.3	46.09	1049.4	1095.5
80	0.5073	0.01607	632.8	48.08	1037.0	48.09	1048.3	1096.4
82	0.5414	0.01608	595.0	50.08	1037.6	50.08	1047.2	1097.3
84	0.5776	0.01608	559.8	52.08	1038.3	52.08	1046.0	1098.1
86	0.6158	0.01609	527.0	54.08	1038.9	54.08	1044.9	1099.0
88	0.6562	0.01609	496.3	56.07	1039.6	56.07	1043.8	1099.9
90	0.6988	0.01610	467.7	58.07	1040.2	58.07	1042.7	1100.7
92	0.7439	0.01611	440.9	60.06	1040.9	60.06	1041.5	1101.6
94	0.7914	0.01611	415.9	62.06	1041.5	62.06	1040.4	1102.4
96	0.8416	0.01612	392.4	64.05	1041.2	64.06	1039.2	1103.3
98	0.8945	0.01612	370.5	66.05	1042.8	66.05	1038.1	1104.2
100	0.9503	0.01613	350.0	68.04	1043.5	68.05	1037.0	1105.0

The two methods of calculating exit temperatures verify that 88 F is a reasonable value to use for outlet temperature of exit water. The appropriate assumptions bulleted above provide for a simple analysis in order to streamline proposal development. This allows sales engineers to quickly determine feasibility of a project and provide system solutions in a shorter time-period keeping businesses profitable. Design engineers are anticipated to apply a much more thorough analysis when the job contract is won. With the appropriate range of inlet and outlet temperatures, wet-bulb temperature, and required refrigeration tons, design day conditions can be developed and then the selection software can be used.

Establishing Design Day Conditions and Troubleshooting Inputs

Codes, Standards, and Ratings Systems

Upon initial selection, design variables were input as follows: 120 gpm flow, 88 F degree tower inlet water, 70 F tower outlet temperature, and a 67.5 F wet-bulb temperature. Pure water is used as our refrigerant because of our temperature range of application. However, warnings showed up with the list of equipment selections because software inputs/design day conditions did not meet Cooling Technology Institute's standards.

Codes, standards, and rating systems decided by ASHRAE and the Cooling Technology Institute have provided minimum performance requirements for heat rejection equipment. Based on test procedures CTI ATC-105S and CTI STD-201 RS, this will provide a performance rating for closed-circuit cooling towers. For propeller or axial fan closed-circuit cooling towers, the minimum performance requirement is ≥16.1 gpm/HP at conditions of 102 F/ 90 F/ 75 F (ASHRAE Standard 90.1). These temperatures represent the inlet water temperature, outlet water temperature, and regional entrance wet-bulb temperature respectively.

Since BAC selection software follows those standards and warnings were indicated, specifically the minimum approach parameter, the original software selection criteria needed to be adjusted. This re-selection is required because the leaving fluid temperature must be 5 F above regional wet bulb temperature according to the CTI STD-201 limits of thermal certification as seen in Table 10.

Table 10 CTI STD-201 thermal certification limits for Cooling Towers/Closed-Circuit Coolers

	SI Units	IP Units
Wet Bulb Temperature	10ºC to 32.2ºC	50ºF to 90ºF
Maximum Process Fluid Temperature	51.7ºC	125ºF
Minimum Range	2.2ºC	4ºF
Minimum Approach	2.8ºC	5ºF
Barometric Pressure	91.4 kPa to 105 kPa	27 in Hg to 31 in Hg

Consequentially, in order to achieve the same total heat rejection, the range must be kept the same but at a pair of higher inlet and outlet water temperatures. A second, modified selection was made with the following criteria: FXV model, 120gpm, 95F/75F/67.5F, pure water, and with a price ranked listing of the results. Previous methods for calculating exit water temperature verify that water can expect to leave at 93 F for the second selection, but an intentional 95 F was applied to provide an additional safety factor for thermal performance. Table 11 illustrates the modifications to design day conditions.

Table 11 Design Day Conditions

Design Day	Tower Inlet Temp	Tower Outlet Temp	Ambient Dry Bulb Temp	Wet Bulb Temp	Range/ Approach	Heat Rejection Rate	Fan Speed	Tower Efficiency
	F	F	F	F	F	BTUH	%	%
Original	88	70	80	67.5	18/3.5	1,080,000	50%	88%
Modified	95	75	80	67.5	20/7.5	1,200,000	50%	73%

Additional Selection Requirements

Once the design day criteria had been input, there were additional selection requirements a selector could have used. This included a maximum fluid pressure drop, number of tower units, model accessories, and limits on sound pollution, total horsepower, and dimensions. After the construction site had been surveyed for its available space, there were no other requirements necessary to further refine selection.

BAC Software Selection

Although modified selection reflects a lower tower efficiency, it provides system redundancy for non-ideal weather conditions and complies with CTI standards. The software produced a list of models with the top two options being the FXV-0806B-28D-L and the FXV-806B-32D-K. Both options were absent of warnings and met thermal performance standards. The options were low first cost selection (LFC) and recommended selection (Rec) based on payback impact respectively.

Baltimore Aircoil Company, Inc.

Closed Circuit Cooling Tower Selection Program

Version:	7.5.3 NA
Product data correct as of:	March 03, 2017

Project Name:	CCCT Solution
Selection Name:	Final Selection LFC
Project State/Province:	California
Project Country:	United States
Date:	April 27, 2017

Model Information

Product Line:	FXV/FXV3
Model:	FXV-0806B-28D-L
Number of Units:	1
Coil Type:	Standard Coil
Coil Finning:	None
Fan Type:	Standard Fan
Fan Motor:	(1) 15.00 = 15.00 HP/Unit
Total Standard Fan Power:	Full Speed, 15.00 BHP/Unit
Total Pump Motor Power:	(1) 2.00 = 2.00 HP/Unit
Intake Option:	None
Internal Option:	None
Discharge Option:	None

Design Conditions

Fluid:	Water
Flow Rate:	120.00 USGPM
Entering Fluid Temp.:	95.00 °F
Leaving Fluid Temp.:	75.00 °F
Wet Bulb Temp.:	67.50 °F
Fluid Pressure Drop:	0.86 psi
Reserve Capability:	5.16%

Thermal performance at design conditions and standard total fan motor power is certified by the Cooling Technology Institute (CTI).

Engineering Data, per Unit

Unit Length: 05' 11.75" + 01' 06.00" (Pump) = 07' 05.75" (Total)

Unit Width:	08' 05.75"
Unit Height:	18' 01.00"
Approximate Shipping Weight:	7,034 lbs
Heaviest Section:	4,788 lbs
Approximate Operating Weight:	10,314 lbs
Approximate Remote Sump Operating Weight:	9,349 lbs
Air Flow:	38,740 CFM
Spray Water Flow:	290 USGPM
Coil Volume:	102 U.S. gallons

Coil Connections:

(1) 4" Coil Inlet and Outlet, Based on 120.00 USGPM Flow per Unit

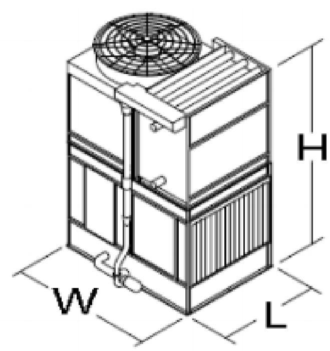
Remote Sump Connections: (1) 6"

Heater kW Data (Optional)

0°F (-17.8°C) Ambient Heaters:	(1) 4 kW
-20°F (-28.9°C) Ambient Heaters:	(1) 6 kW

Minimum Distance Required:

From Solid Wall:	4 ft
From 50% Open Wall:	3 ft



Energy Rating:

23.29 per ASHRAE 90.1, ASHRAE 189 and CA Title 24.

Note: These unit dimensions account for the selected fan type for the standard cataloged drive configuration, but they do not account for other options/accessories. Please contact your local BAC sales representative for dimensions of units with other options/accessories.

Figure 24 LCF tower selected

Baltimore Aircoil Company, Inc.

Closed Circuit Cooling Tower Selection Program

Version:	7.5.3 NA
Product data correct as of:	March 03, 2017

Project Name:	CCCT Solution
Selection Name:	Final Selection Rec
Project State/Province:	California
Project Country:	United States
Date:	April 27, 2017

Model Information

Product Line:	FXV/FXV3
Model:	FXV-0806B-32D-K
Number of Units:	1
Coil Type:	Standard Coil
Coil Finning:	None
Fan Type:	Standard Fan
Fan Motor:	(1) 10.00 = 10.00 HP/Unit
Total Standard Fan Power:	Full Speed, 10.00 BHP/Unit
Total Pump Motor Power:	(1) 2.00 = 2.00 HP/Unit
Intake Option:	None
Internal Option:	None
Discharge Option:	None

Design Conditions

Fluid:	Water
Flow Rate:	120.00 USGPM
Entering Fluid Temp.:	95.00 °F
Leaving Fluid Temp.:	75.00 °F
Wet Bulb Temp.:	67.50 °F
Fluid Pressure Drop:	0.95 psi
Reserve Capability:	1.48%

Thermal performance at design conditions and standard total fan motor power is certified by the Cooling Technology Institute (CTI).

Engineering Data, per Unit

Unit Length:	05' 11.75" + 01' 06.00" (Pump) = 07' 05.75"	(Total)
Unit Width:	08' 05.75"	
Unit Height:	18' 01.00"	
Approximate Shipping Weight:	7,176	lbs
Heaviest Section:	4,930	lbs
Approximate Operating Weight:	10,578	lbs
Approximate Remote Sump Operating Weight:	9,613	lbs
Air Flow:	35,170	CFM
Spray Water Flow:	290	USGPM
Coil Volume:	117	U.S. gallons

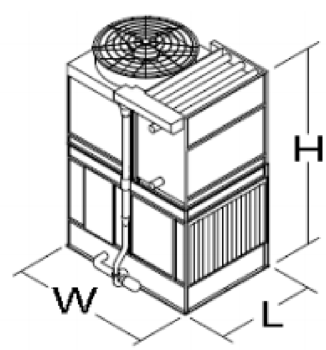
Coil Connections:

(1) 4" Coil Inlet and Outlet, Based on 120.00 USGPM Flow per Unit

Remote Sump Connections:	(1)	6"

Heater kW Data (Optional)

0°F (-17.8°C) Ambient Heaters:	(1)	4 kW
-20°F (-28.9°C) Ambient Heaters:	(1)	6 kW



Minimum Distance Required:

From Solid Wall:	3.5 ft
From 50% Open Wall:	3 ft

Energy Rating:

32.00 per ASHRAE 90.1, ASHRAE 189 and CA Title 24.

Note: These unit dimensions account for the selected fan type for the standard cataloged drive configuration, but they do not account for other options/accessories. Please contact your local BAC sales representative for dimensions of units with other options/accessories.

Figure 25 Rec cooling tower selection

Table 12 BAC software selection results

Selection Criteria: 120 gpm, 95F/75F/67.5F, pure water, price ranking							
BTUH rejection		-1,200,000		-100	Tons	Efficiency	73%
Selection Type	**Product**	**Qty**	**Model**	**Series**	**Total Fan Motor (HP)**	**Total Pump Motor (HP)**	**Warnings Exist**
LFC	FXV	1.00	FXV-0806B-28D-L	no	15.00	2.00	No
Rec	**FXV**	**1.00**	**FXV-0806B-32D-K**	**no**	**10.00**	**2.00**	**No**
	Unit Height	**Fluid Pressure Drop (psi)**	**Reserve Capability (%)**	**Price Rank**	**Payback (Years)**	**Energy Rating (USGPM /HP)**	**Warnings Exist**
LFC	18′ 01.00″	0.86	5.16	1.00	--	23.29	No
Rec	**18′ 01.00″**	**0.95**	**1.48**	**1.02**	**1.07**	**32.00**	**No**

Now that we have models of cooling towers, another decision matrix was created to objectively select the best system solution.

Table 13 Decision Matrix for FXV Model

FXV Comparison Table				
Raw Scoring Table				
1-3 where, 1 = poor, 2 = satisfactory, 3 = excellent				
	Product Cost	Energy Rating	Reserve Capacity	
28D-L	2.00	2.00	2.25	
32D-K	1.75	3.00	2.00	
Importance Factor				
0-1 where, 1 = highest importance, 0 = lowest importance				
	Product Cost	Energy Rating	Reserve Capacity	
	0.50	0.75	1.00	
Weighted Scoring Matrix				**TOTAL**
28D-L	1.00	1.50	2.25	4.75
32D-K	**0.88**	**2.25**	**2.00**	**5.13**

Based on the decision matrix, the FXV-806B-32D-K is the best, long-term system solution. Although the first model provides a reserve capacity, it does not compare to the level of impact the energy savings of the second model. This was the model recommended for best payback in terms of annual energy savings. Specifics on various savings will be described in the engineering economics portion of this report.

Engineering Economics of Product Selection

Capital Budgeting Metrics

For this report, the five payback metrics used are: return-on-investment, the payback period, net present value, internal rate of return, and modified internal rate of return. Each method of budget analysis gives decision makers unique insight into the feasibility of the project.

Return on Investment

This value can be given as a percentage or dollar amount comparing the cost of investment to the expected returns over a certain period. This term is very familiar to facility managers when it comes to making decisions for capital investments. Although this does not include the time value of money, this should be included in sales presentations to indicate feasibility and facilitate a simple understanding among decision makers. For the pro forma spreadsheet, the ROI was computed over the first three years and did include discounted cash flows.

$$Simple\ ROI = \frac{Investment\ Revenue - Investment\ Cost}{Investment\ Cost}, \text{Eqn. } 11$$

Payback Period

This method provides a rough estimation as to how quickly a project can be paid off. This is also useful for determining the feasibility of the project during the sales presentation. This will allow decision makers to understand how the project needs to be financed and how it fits within the vision of the company's priorities. Normally in sales presentations, this method does not account for the time-value of money. However, cash flows on the pro forma statements have all future cash flows discounted taking the time-value of money into account.

$$PP\ (yrs) = n\ (yrs\ until\ full\ recovery) - \frac{cumulative\ cash\ flow\ at\ year\ n}{year\ n\ cash\ flow}, \text{Eqn. } 12$$

Net Present Value

This indicates how the initial cash outflow (investment) can be compared to discounted future cash inflows and outflows. A positive NPV value tells us that our future cash flows justify the initial investment taking the time-value of money into consideration. Determining a representative discount rate is important and will be further explored.

$$NPV = \sum_{t=1}^{T} \frac{C_t}{(1+r)^t} - C_o\ , \text{Eqn. } 13$$

$$\begin{aligned}&Where,\\&C_t = net\ cash\ inflow\ during\ the\ period\ t\\&C_o = total\ initial\ investment\ costs\\&r = discount\ rate\\&t = number\ of\ time\ periods\end{aligned}$$

Internal Rate of Return

This value indicates the discount rate at which future cash flow benefits from a project would have a net present value of zero. The higher the IRR, the more valuable a capital investment appears. This method allows for easier interpretation, regardless of financial scale, with percent values. Although the time-value of money is considered, the IRR calculation assumes that positive cash flows from the project can be reinvested at a return rate identical to that of the initial project. To accommodate for this, the MIRR constructs a more conservative reinvestment rate and includes the finance rate.

$$NPV = 0 = \sum_{t=1}^{T} \frac{C_t}{(1 + IRR)^t} - C_o, \text{Eqn. 14}$$

$$\begin{aligned}&Where,\\&C_t = net\ cash\ inflow\ during\ the\ period\ t\\&C_o = total\ initial\ investment\ costs\\&r = discount\ rate\\&t = number\ of\ time\ periods\end{aligned}$$

Modified Internal Rate of Return

This method offers a more realistic value for rate of return. This is because the water savings could not be directly reinvested in another identical cooling tower because our operation only requires one solution of that capacity. Thus, the reinvestment rate of return for this project is based on the cost of capital. Since this is a private company, information normally used to calculate the cost of capital is not available. Therefore, the cost of capital for publicly traded food processors was used for the reinvestment rate. From a sample of 87 different food processors, a cost of capital was determined to be 5.76% (NYU 2017). Additionally, any negative cash flows would be discounted by a finance rate of 6%. This represents a typical interest rate at which banks will loan to industrial customers on construction jobs. The resulting percentage values are always lower than the IRR.

$$MIRR = \sqrt[n]{\frac{FV\ (positive\ cash\ flows, reinvestment\ rate)}{-PV\ (negative\ cash\ flows,\ finance\ rate)}} - 1, \text{Eqn. 15}$$

Minimum Acceptable Rate of Return

This is a value that is used by facility managers to decide whether a project is attractive or not. An acceptable project is deemed when the IRR or MIRR exceeds the hurdle rate (MARR). Typically, the hurdle rate developed by financial mangers is dependent on the company's cost of capital and risk tolerance.

Pro Forma Income Statement

This type of financial document is a tool used by budget analysts to understand the resulting cash flows from anticipated changes in operation ranging from acquisitions to large capital investments. This document will allow sales engineers and decision makers to compute payback metrics to justify or nullify the project. The full pro forma spreadsheet for the 8-year budget analysis can be found in Appendix M. The following subtopics are items affecting or representing various cash flows.

Initial Investment Costs

Determining relevant cash flows includes any positive or negative change in cash as a result of the new investment. The most obvious cash flow would be the net capital investment the company must make for the cooling tower including the following components: installation, freight cost, sales tax, and the equipment itself. This fixed cost is primarily justified by the decrease in processed water contributing to a positive, gradient cash flow series.

Water Rates

The pro forma financial statements are in terms of discrete years (2017, 2018, etc.) despite the fact that water rates are increased on a semi-annual basis in the months of January and June. Based on averages of historical water rate increases, an annual increase of 5% was applied to projected water rates. Figure 26 below and includes historical water rates with projections of their future values of water rate. Appendix O verifies the historical water rates.

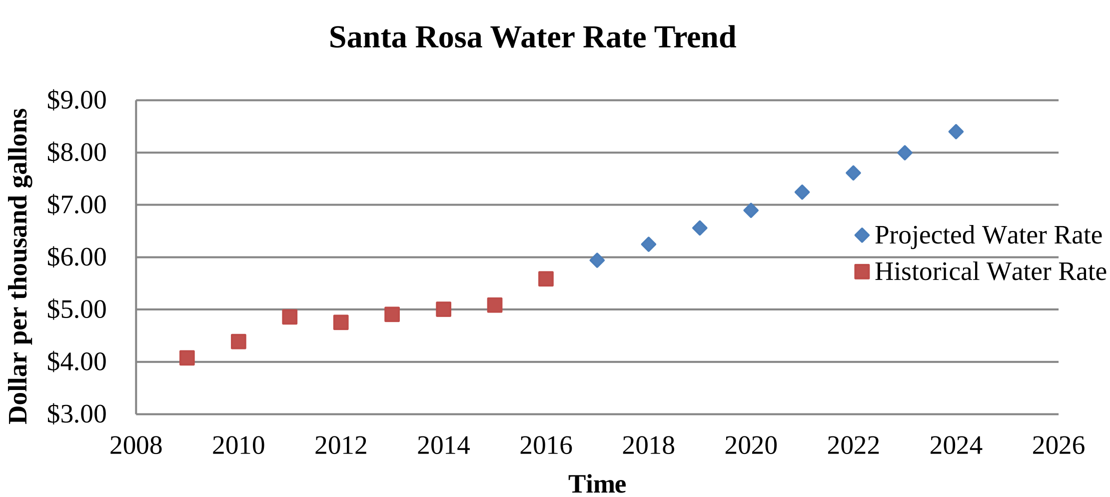


Figure 26 Plot of Water Rate v. Time

The historical data, the Sonoma County Water Agency website, and public media outlets suggest that this rate increase is a feasible value to apply. Representing the changes in the water rates was an important step in the budget analysis because neglecting those changes would have led to a discrepancy of approximately $85,000 dollars less in the net present value for the 8-year budget analysis.

Sewage Rate

In addition to the water that delivered, the company must pay for the water it sends back to the wastewater treatment plant. The sewage rate for this particular application is based solely on the volume of water sent back, not necessarily the quality of the water (i.e. B.O.D., TSS). However, the gradual sewage rate increases were computed similarly to the methodology applied to water rates previously discussed. Figure 27 below. Appendix O verify the historical sewage rates.

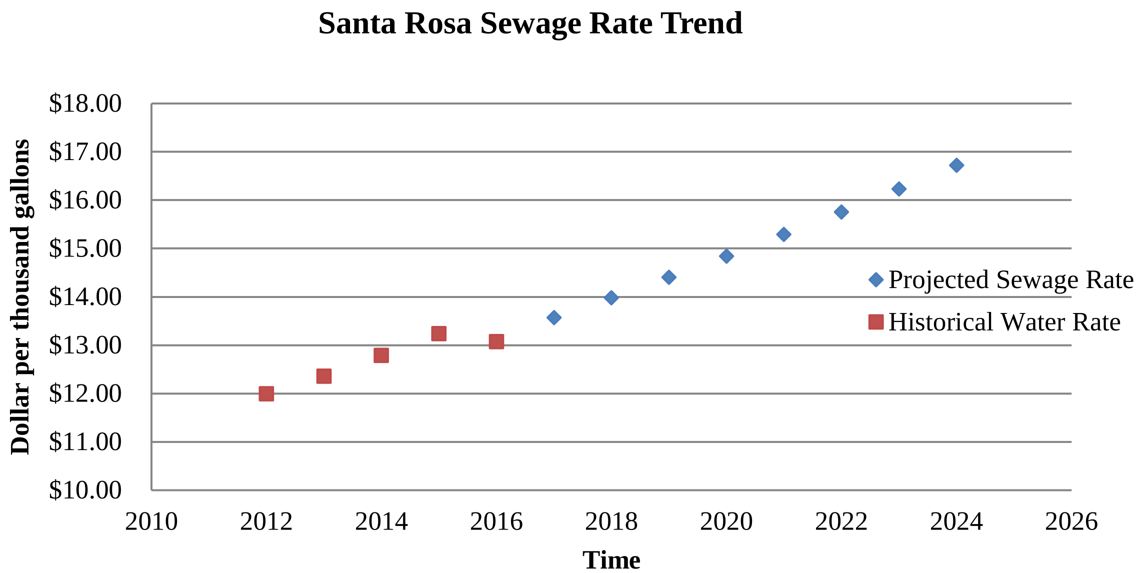


Figure 27 Plot of Sewage Rate v. Time

This rate increase is accounted for in the pro forma financial statements in terms of discrete years with an annual 3% rate increases. Neglecting the changes in sewage rates would have led to a discrepancy of approximately $114,000 dollars less in the net present value for the 8-year budget analysis.

Cooling Tower Energy Consumption

The next relevant cash flow would be the energy consumption from running the new piece of heat rejection equipment. This energy consumption is due to the running fan, the sump pump, and the circulation pump that goes into the facility. PG&E electricity provider applies complicated algorithms and rate structures in order to encourage sustainable energy consumption. Among the many things that influence the electricity rate, power-quality and the demand charge associated with current rate plans will not be considered in the scope of this study. However, factors like the projected rate increases over the 8-year period, time-of-day rate structure, and customer type will be considered. The most recent electricity rates publicly available, for industrial customers (E-20) requiring primary power distribution (i.e. supplied voltage between 2,000 and 50,000 volts), are in Figure 28 outlining the $ per kWh and distribution of operating hours.

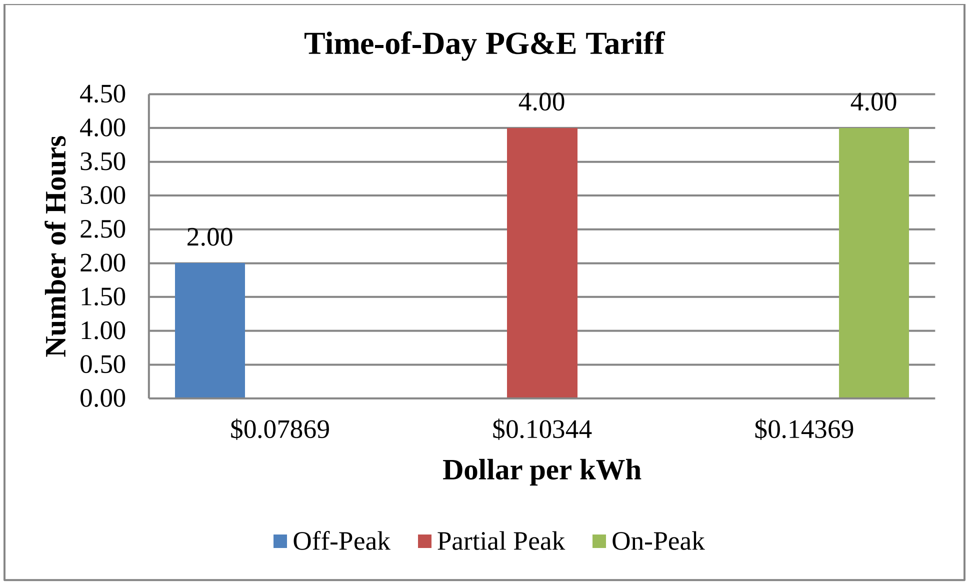


Figure 28 Cooling tower energy load profile

The rates in the bar chart above are associated with 2016 values, but will be applied over the entirety of the year to simplify analysis and eventually provide a more conservative NPV per year. The distribution of hours of cooling-tower operation was based on the facility's standard operating procedures of cooking and cooling five batches of product within the period of 3:30 AM and 9:30 PM. Each batch of food product is cooled within two hours to meet FDA standards for food safety, to retain food quality, and maintain current food-processing capacity. Finally, over the 8-year budget analysis, the electricity rates are anticipated to increase each year by 3%.

The power requirements for the fan and the sump pump were based strictly off the nominal horsepower ratings listed on the tower datasheet generated by BAC software. Although a variable frequency drive will be used to adjust the fan speed, thus lowering power consumption, full-speed operation was considered exclusively for the energy consumption analysis. In reality, tower fans can be expected to be operating at speeds as low as 50% of maximum.

The circulating pump power consumption was based on interpretations from the pump curve and worst case pumping scenario for the system curve. With a designated operating point, one could determine a representative power consumption for the energy analysis. Two scenarios were considered for operating points: normal and worst-case operation. Normal operation is based on operation of 8 kettles of 500-gallon capacity operating simultaneously on the main kettle deck. However, the worst-case scenario was based on the to the simultaneous cooling of 8, 500-gallon kettles hydraulically furthest from the pumping station. Based on interpretation of the piping layout diagram provided, head and flow requirements were developed to represent the most power intensive pumping scenario. The following head and flow requirements were largely based on kettle design, friction loss estimates, static head, NPSH, and changes in elevation. Pipe friction losses were based on the Hazen Williams equation to provide rough estimates on pumping costs. Friction loss calculations will need to be reviewed by the refrigeration design engineer using the Darcy–Weisbach equation to account for turbulent flow and the temperature gradients throughout the pumping system. Furthermore, friction losses across various valves and the glycol heat exchanger were included as rough estimates, but also require further investigation. Finally, a deeper understanding of kettle deck operation will provide an optimized pump selection based on a representative system curve. By providing a slightly oversized pump station and variable frequency drive, operators can adjust pump speed according to what minimizes energy cost while maintaining production capacity. Once the head requirements for both scenarios were considered, the following equations could be applied to estimate the required motor horsepower for the pump station.

$$Water\ HP = \frac{TDH * GPM}{3960}, \text{Eqn. 16}$$

$$Brake\ HP = \frac{WHP}{Pump\ Eff.}, \text{Eqn. 17}$$

$$Motor\ HP = \frac{Brake\ HP}{Motor\ Eff.}, \text{Eqn. 18}$$

With normal operation and worst case operation requiring 5.7HP and 8.8HP respectively, a nominal 10HP motor would be selected for the capital budget analysis. In reality, a variable frequency drive will allow operators to adjust motor speed allowing for adjustable flow and head. Therefore, expenses on the pro forma financial statements will be inflated compared to what the food processing facility will actually incur.

Cooling Tower Water Usage

Another annual cost associated with our capital investment would be the water losses required to cool the refrigerant returning back to the heat load in our closed-loop. Calculating the total water usage is important for the pro forma financial statements to investigate payback on project. Total water use by cooling towers can be described by Equation 19 below.

$$Water\ Use = Evaporative\ Loss + Blowdown + Drift, \text{Eqn. 19}$$

The first category of water loss would be due to evaporation. Simply put, the spray water in the tower making contact with the refrigerant coils, which are plumbed into the kettle jackets, allows the heat to be rejected into the atmosphere via evaporation. This is represented by the latent component of heat transfer discussed in the literature review.

The next source of water loss is called drift loss. This loss is usually composed of only .1% to .3% of refrigeration circulation rate (e.g. at .1% and 100gpm refrigerant flow, 0.1gpm of drift loss). These consist of liquid droplets that get forced out of the outlet due to high air flow. Another loss is the required bleed-off of the water in the sump tank of the cooling tower. The selected cooling tower has been designed for 3 cycles of concentration. Due to all of these various losses, the remaining water eventually increases in concentration of bicarbonates and other constituents that could potentially compromise cooling tower operation. To prevent this, water is systematically bled-off in order to prevent costly maintenance caused by the formation of deposit on the coil surface.

For this report, two approaches were used to compute water usage: 1) interpretation from the psychometric chart with rule-of-thumb equations 2) utilizing an online cooling tower water-loss calculator.

Method 1

For the first method, psychrometric chart software by Greenheck was used to provide accurate values for properties at the inlet and exit of the cooling tower. The following assumptions were used for the computations:

- Mass and energy transfer scenario
- Total energy rejected by tower absorbed by air
- Conservation of mass principle through tower system
- Exit air at saturated conditions (i.e. 100% relative humidity)
- Atmosphere at a pressure of 1atm
- Fan running at half-speed
- Affinity law applies
- Average specific volume acceptable for water loss estimation
- 80 F dry bulb inlet temperature representative for a “design day”

These assumptions allow us to use the psychrometric chart to indicate the location for the two points, compare the change in enthalpy, and dictate the change in humidity ratio. The change in humidity ratio indicates the amount of water vapor added into the air as it passes through the cooling tower as seen in Figure 29.

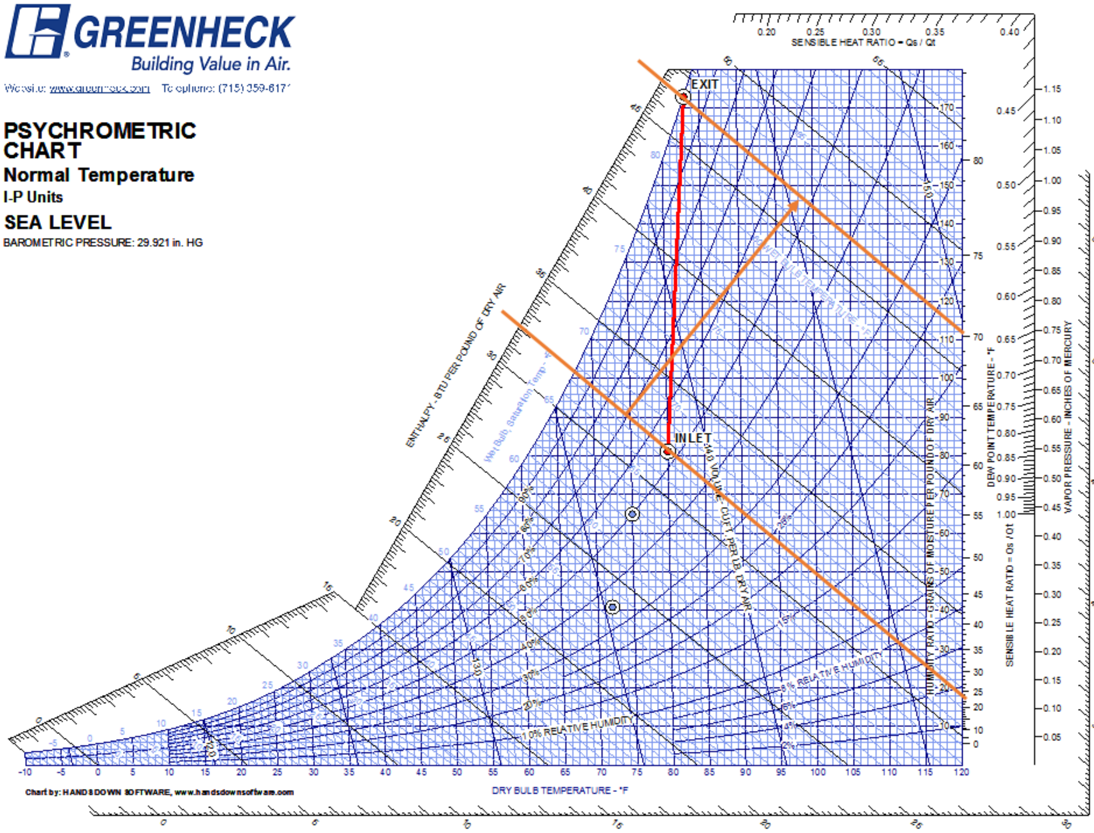


Figure 29 Psychrometric chart with indicated process and the associated enthalpy change

After adjustments to exit dry-bulb temperatures were made to correspond to a 1,200,000 BTUH rate of heat rejectetion and tower specifications, Equation 19 indicated the evaporative loss to be 1.95 gpm.

$$air\ flow\ \left(\frac{ft^3}{min}\right) * air\ density\ \left(\frac{lb\ dry\ air}{ft^3}\right) * \Delta W\ \left(\frac{lb\ H20}{lb\ dry\ air}\right) = Evap\ Loss\ \left(\frac{lb\ H20}{min}\right) \text{Eqn. 19}$$

For drift loss, the rule-of-thumb of .3% of circulating tower flow was used which amounted to 0.36 gpm. Blowdown used an equation based off the percentages of evaporative and drift loss relative to circulating flow. Equation 20 below, with three cycles of concentration, indicates that 0.62 gpm were lost due to blowdown.

$$Blowdown\ (gpm) = circulating\ flow\ *\ \left[\frac{\%\ Evap\ loss}{cycles\ of\ concentration\ -1} - \%\ Drift\right], \text{Eqn. 20}$$

This amounted to a total water usage of 2.93 gpm for Method 1.

Method 2

The second method required input information about tower flow, range, ambient wet-bulb temperature, cycles of concentration, and drift rate. Figures 30 and 31 illustrate the input variables and the output for water usage respectively.

Operating Conditions

Tower Water Flow	120	*gpm*	27	*m^3/h*
Hot Water Temperature	95.00	°F	35.00	°C
Cold Water Temperature	75.00	°F	23.89	°C
Wet-Bulb Temperature	67.50	°F	19.72	°C
Drift Rate	0.005	%		
Concentrations	3			

Figure 30 Water Usage Calculator Inputs (provided by SPX Cooling)

Water Usage

Evaporation	2.24	*gpm*	0.51	*m^3/h*
Drift	0.01	*gpm*	0.00	*m^3/h*
Blow down	1.11	*gpm*	0.25	*m^3/h*
Total Usage	3.35	*gpm*	0.76	*m^3/h*

Figure 31 Water Usage Calculator Outputs (provided by SPX Cooling)

Method 2 amounted to a total of 3.35 gpm of industrial fluid cooler water usage. The following graph was generated by the calculator to indicate water usage based on varying wet bulb temperature and cooling tower range.

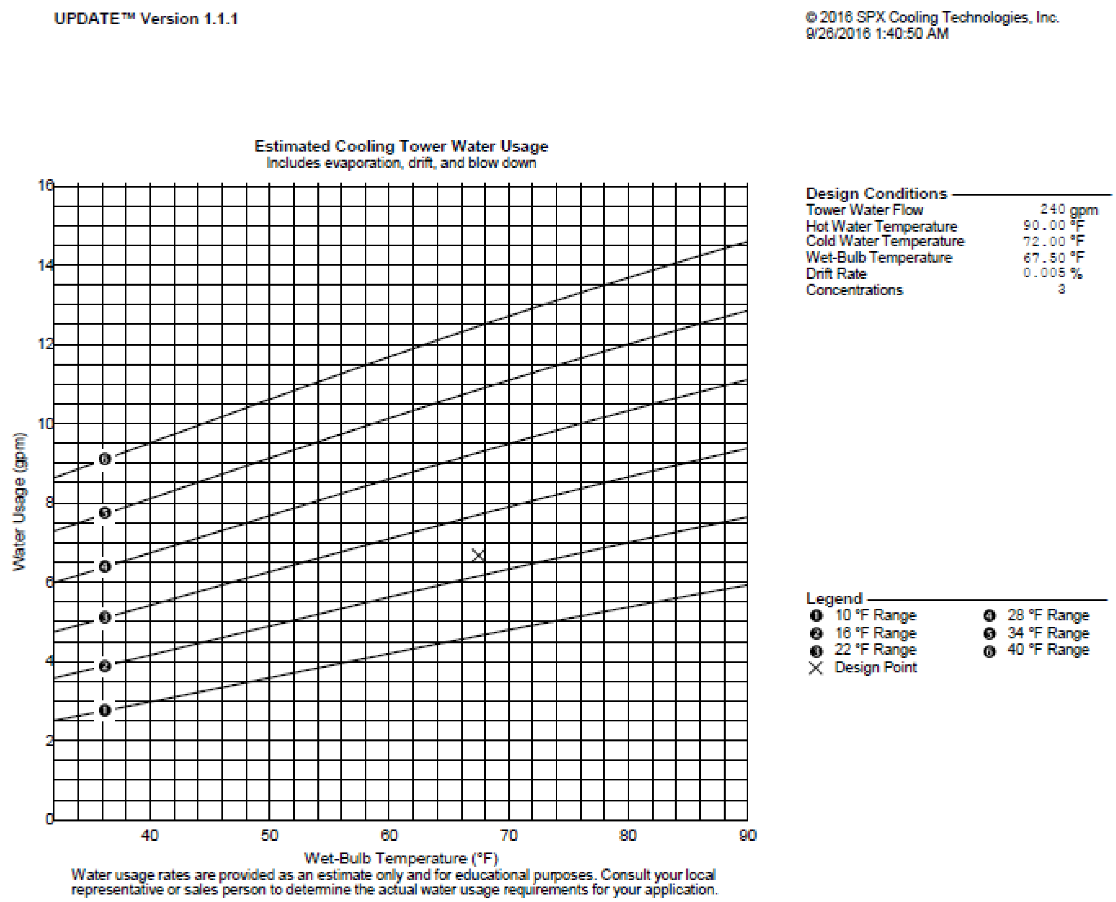


Figure 32 Water Usage Graph (provided by SPX Cooling)

Both Method 1 and 2 provided reasonable values of 2.93 gpm and 3.35 gpm respectively. It should be taken into consideration that changing ambient dry bulb inlet temperature, tower aging, and fan speed will produce varying water loss values. However, the inlet dry bulb temperature, fan speed, and wet bulb temperatures selections were intentional to represent a "design day". With these considerations, a 3 gpm of water use will be used for the pro forma financial statements to illustrate realistic payback period and return on investment. The distribution of water use from Method 1 was used and the blowdown value was increased by 0.07 gpm. Table 14 indicates volume of water consumed by each category of use and their relative percentages of total water use. It is important to separate these types of water use because blowdown must be accounted for in the sewage costs as well. Table 14 below was developed in accordance with 3110 operational hours per year.

Table 14 Distribution of industrial fluid cooler water use

Blowdown	Drift	Evaporative	Total Water Use	
127,828	67,176	363,883	558,887	gal/year
23%	12%	65%	100%	

Miscellaneous cash flows

This equipment also has an associated annual cost for maintenance and inspection. The cost associated for the technician services is two negative cash flows of $1500 dollars per year.

When looking to determine opportunity cost of the cooling tower investment, it is difficult to develop a substantial cost compared to the benefits this investment provides. The unit will take-up approximately 143 square-feet and is placed behind the building in a remote location. Due to the immense returns on investment, opportunity cost related to space can be neglected. The only opportunity cost associated with this investment is not using the cash in another type of investment (stock market, bond, annuity, etc.). Since this is covered by the MIRR, the pro forma statements will not double count opportunity cost in the pro forma financial statement.

On a final note, a boost in productivity may occur due to this improvement in the cooling process. However, the positive cash flows associated with the increased operational capacity is important, but outside the scope of this budget analysis.

Depreciation

Depreciation of the cooling tower product was received from the IRS website with tables of asset class codes and their associated MACRS. The cooling tower was found to have a 7-year recovery period (OK State 2007). It should also be noted that the salvage value of the cooling tower is zero considering how specialized the tower is to this particular facility. This would make the cooling tower a particularly difficult product to resell.

Discount Rate

In determining the discount rate, the factors that influence its selected value are the internal required rate of return and internal financial advisory.

The required rate of return is influenced primarily by the current risk-free rate of return and inflation. The risk-free rates are commonly associated with the returns offered on treasury bonds. Treasury bills at the end of year 2016 indicate an approximate rate of return of .5% for the maturity of a year-length treasury bond. An approximate inflation rate of 2% was determined based on typical values observed in the 5 years prior to 2016. The sum of these two would produce an approximate required rate of return of 2.5%.

The liquidity of assets and the health of the food processing industry influence the risk tolerance of the internal financial advisory for the given food processing facility. Since much of this facility's fixed assets contribute to its equity, the company's liquidity ratio is anticipated to be low. Additionally, the company's current assets consist mostly of raw food as inventory, which is a nondurable good, thus further contributing to a lower liquidity ratio. This ratio indicates how easily the company could transform capital into liquid funds in the case of an emergency to pay off debts. These financial and market behavior allow managers tailor the minimum acceptable rate of return to their respective industry. Accounting for project risk and the liquidity of the facility, an additional 3.5% should contribute to the minimum attractive rate of return.

Since the MARR was estimated to be 6%, this will also be used as the discount rate. Managers must adjust the pro forma financial spreadsheet to use a discount rate that reflects the given facility's precise minimum attractive rate of return.

Capital Budget Analysis Results

Table 15 Summary of important rates used to develop pro forma financial spreadsheet

Annual Rate	Category
2.0%	inflation
0.5%	T-bill yield
6.0%	discount/finance
5.8%	reinvestment (cost of capital)
3.0%	sewage rate increase
5.0%	water rate increase

Table 16 Results for various corporate finance metrics

ROI = EOY 3	**32%**	**NPV =**	**$ 1,230,042.47**	**IRR =**	**42%**	**MIRR =**	**21%**	**Payback Period =**	**2.31**	**years**

Table 16 above shows that this project is highly profitable. By EOY 3, the company can expect to make their money back with a recovery of $212,672. The net present value indicates that this project is worth over 1.2 million dollars in year 2016. Furthermore, the IRR and MIRR are likely to exceed the company's hurdle rate which further encourages initiating the project ATC has to offer.

Sensitivity Analysis

Before engaging in lofty financial investments, a sensitivity analysis should be used to test the integrity of the project potential changes in variables contributing to significant cash flows. This analysis was conducted for each individual scenario that could jeopardize the profitability of this capital investment:

1. A reduction in the cost of water by 25%,50%,75%, and 100% for all 8 years
2. A reduction in the cost of sewage by 25%,50%,75%, and 100% for all 8 years
3. An electric energy rate increase of two to three times the normal rate for all 8 years
4. A complete failure of the original cooling tower followed by a complete reinvestment of the original net capital investment amount: $664,600

Scenario 1: Reduction of the Cost of Water

The measure of the success of the project can be interpreted from its resilience to the reduction of the water rate in the sensitivity analysis. The excel document tables indicate that if the water rate was reduced to zero, for all 8 years and ceteris paribus, the sewage rate savings would still provide a NPV of $664,601 and an IRR of 27%. With further testing, profitability was most sensitive to changes in the sewage rate.

Scenario 2: Reduction of the Cost of Sewage

Keeping all other factors constant, if there was no sewage rate applied to the facilities wastewater, for eight consecutive years, this would make the project unprofitable with a NPV of -$7,626 and an IRR of 5.7%. This negative net present value makes sense for an investment project considering the discount rate is 6%. Essentially, the earnings are being discounted at a higher rate than the potential for returns. However, this scenario is highly unlikely considering the sewage rates are trending to rise 3% annually and the facility will continue to discharge water to treatment facilities.

Scenario 3: Energy Rate Price Hikes

The capital investment was also resilient to increases in energy costs two to three times the normal rate described by the E-20 industrial consumer rates from PG&E. Despite the tripled energy costs, for all 8 years, the net present value was largely unaffected with a NPV as high as $1,170,053.

Scenario 4: Complete Failure of Industrial Fluid Cooler

Also, it was determined that the cooling tower would have to break down 3 consecutive years for EOY 1, EOY 2, and EOY 3 to produce the first negative present value. Even if the machinery were to break down twice, for EOY 1 and EOY 2, the project would still have a NPV of $316,187. This was tested by adding a negative -$664,600 cash flow into the maintenance category ceteris paribus. This demonstrates the large future cash flows in savings that come from the water and sewage savings.

Final Engineering Economics Remarks

Both the sensitivity analysis and the financial metrics indicate that the industrial fluid cooler capital investment would be a highly profitable investment at the given food processing facility.

RECOMMENDATIONS

The kettle cooling and sustainability project can expect three outcomes: water and sewage savings equivalent to an average of $397,342 annually, a 97% decrease in water usage, and an improved refrigeration process contributing to improved food safety. Additionally, the local water authorities of Santa Rosa have provided rebates amounting to $200,000 dollars to assist with the initial investment.

After the kettle cooling improvements, the given food processing facility should invest in the improvement of the second phase of cooling. The system solution has a glycol heat exchanger plumbed into it to streamline this improvement. The upgrade would allow the facility to:

1) Eliminate the bucket cooling process by using the kettles as heat exchangers
2) Lower the end cooling temperature in phase two
3) Improve the flexibility of both phase 1 and 2 cooling processes
4) Provide the capability to further reduce phase 1 cooling time
5) Ensure resilience of cooling tower performance on days with high wet-bulb temperatures.
6) Meet the increase in capacity of incoming food processing equipment

Improved monitoring of flow rates and temperature profiles of the kettles are also important for future facility improvements. Additionally, an outline of the food processing facility’s daily kettle deck operation should be documented so that the pumping station and distribution system can be further optimized. Furthermore, an internal case study should be conducted by the given food processing facility to measure how much more product is being produced as a result of the system solution.

Currently, the facility kettle decks produce an average of 4663 lbs per operational hour with 3110 operating hours per year (See Appendix B). Since kettle deck is responsible for 77% of the total food production, improvements to the kettle deck are likely to result in greater annual production output. Assuming there are no upstream bottlenecks and the demand for the product was there, this would translate into increased annual sales. This extrinsic benefit of increased revenue could be applied to the pro forma financial statements further justifying the project. The conclusions of such a study would further incentivize future project improvements to increase in production capacity at the kettle deck of similar food processing facilities. Systems solutions, such as Kettle Cooling and Sustainability Project, are relevant to many industrial and commercial facilities by coupling profitability with sustainability.

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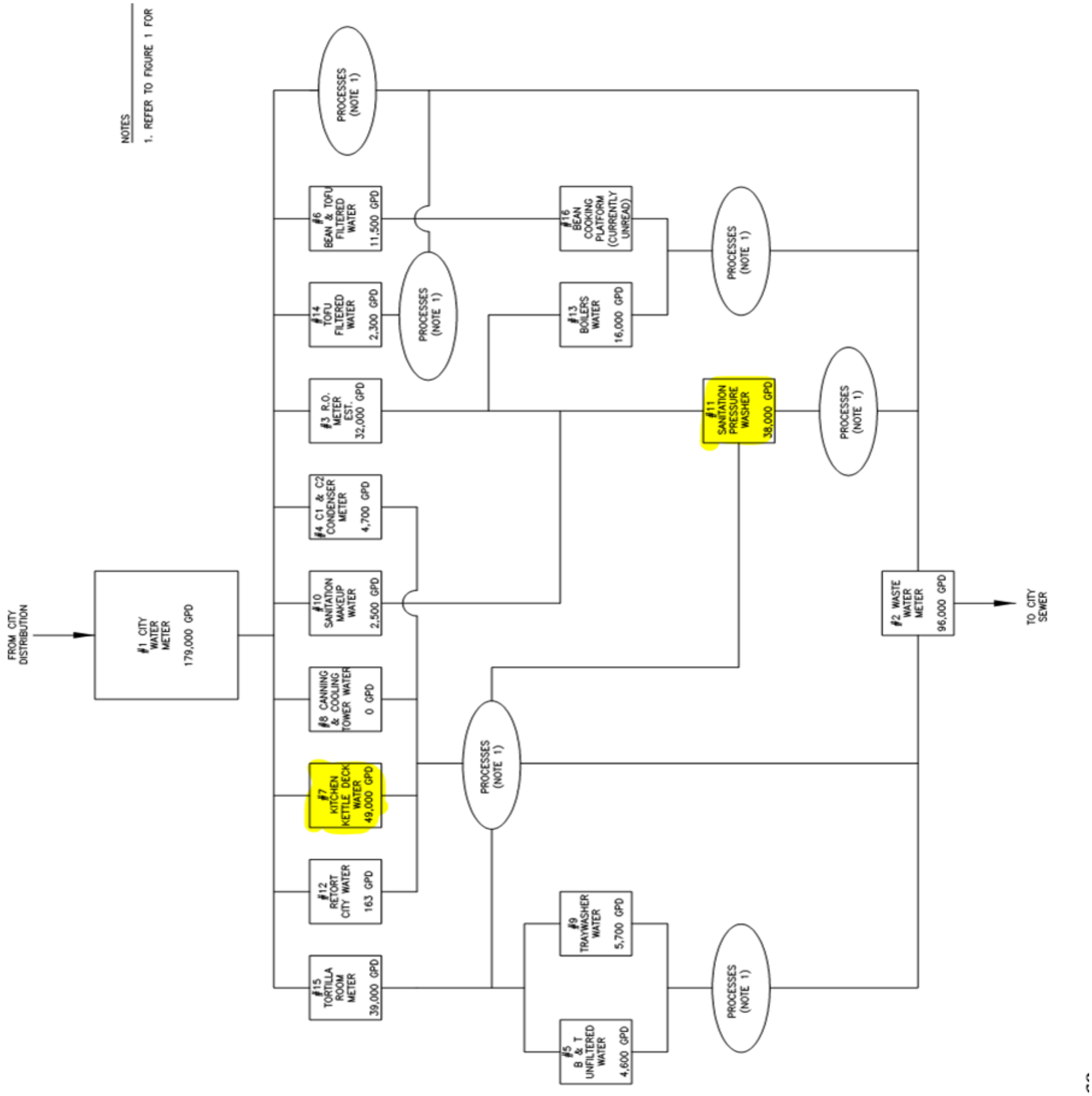
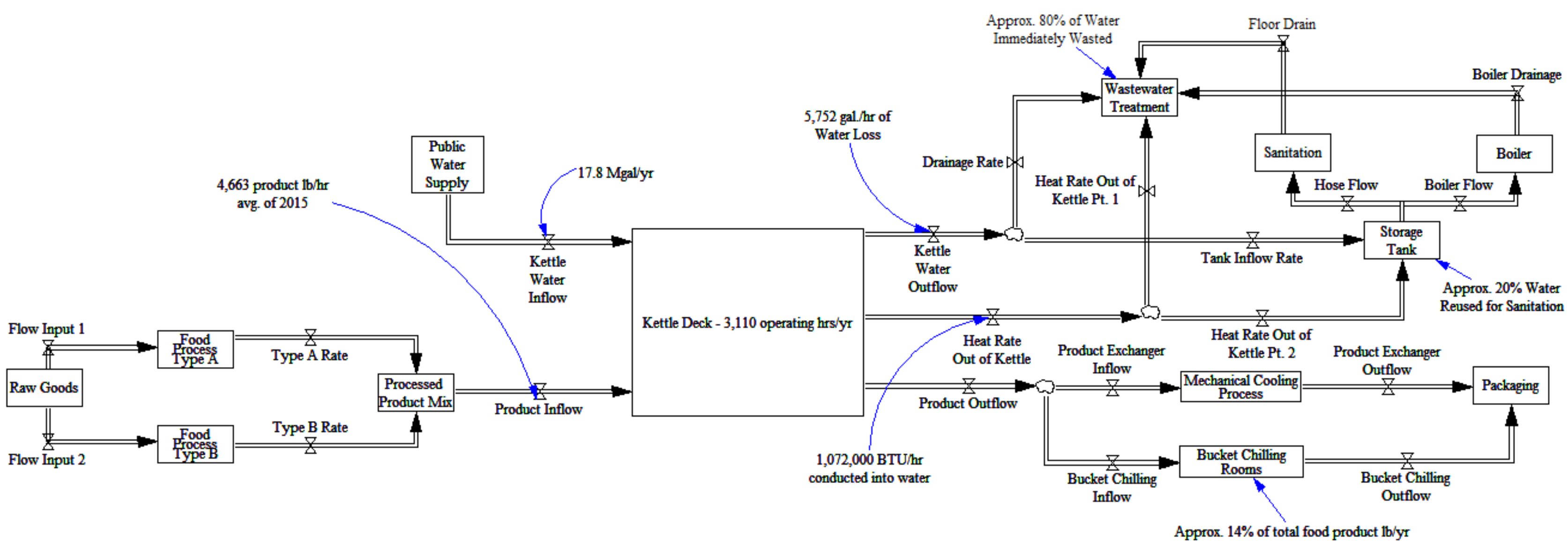
APPENDICIES

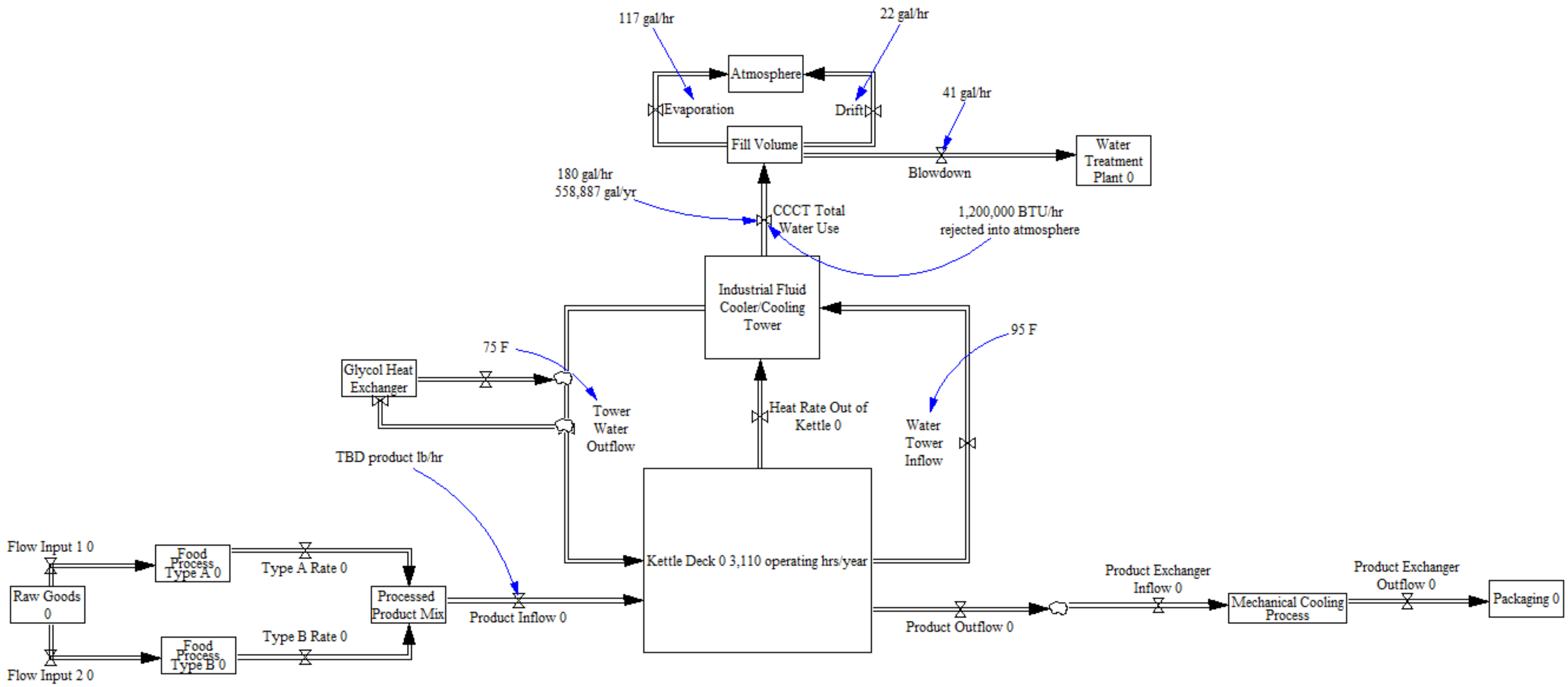
APPENDIX A



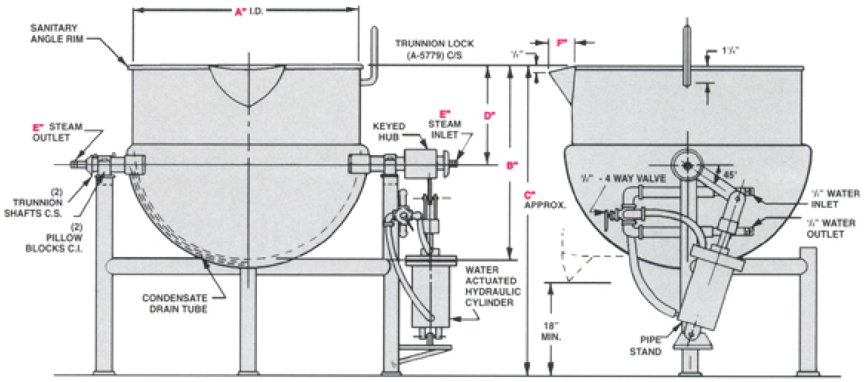
APPENDIX B


Approx. 80% of Water Immediately Wasted
Floor Drain
Boiler Drainage
Wastewater Treatment
5,752 gal./hr of Water Loss
Public Water Supply
Sanitation
Boiler
Drainage Rate
Heat Rate Out of Kettle Pt. 1
4,663 product lb/hr avg. of 2015
17.8 Mgal/yr
Hose Flow
Boiler Flow
Storage Tank
Kettle Water Inflow
Kettle Water Outflow
Tank Inflow Rate
Approx. 20% Water Reused for Sanitation
Kettle Deck - 3,110 operating hrs/yr
Flow Input 1
Food Process Type A
Type A Rate
Heat Rate Out of Kettle
Heat Rate Out of Kettle Pt. 2
Product Exchanger Inflow
Product Exchanger Outflow
Raw Goods
Processed Product Mix
Product Inflow
Mechanical Cooling Process
Packaging
Product Outflow
Food Process Type B
Type B Rate
Flow Input 2
1,072,000 BTU/hr conducted into water
Bucket Chilling Rooms
Bucket Chilling Inflow
Bucket Chilling Outflow
Approx. 14% of total food product lb/yr

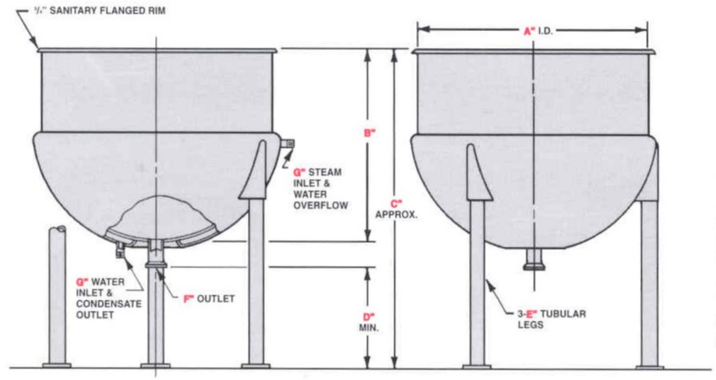
APPENDIX C


117 gal/hr
22 gal/hr
Atmosphere
Evaporation
Drift
41 gal/hr
Fill Volume
Water Treatment Plant 0
Blowdown
180 gal/hr
558,887 gal/yr
CCCT Total Water Use
1,200,000 BTU/hr rejected into atmosphere
Industrial Fluid Cooler/Cooling Tower
95 F
75 F
Glycol Heat Exchanger
Tower Water Outflow
Heat Rate Out of Kettle 0
Water Tower Inflow
TBD product lb/hr
Flow Input 1 0
Food Process Type A 0
Type A Rate 0
Kettle Deck 0 3,110 operating hrs/year
Raw Goods 0
Processed Product Mix
Product Inflow 0
Product Exchanger Inflow 0
Product Exchanger Outflow 0
Mechanical Cooling Process
Packaging 0
Product Outflow 0
Type B Rate 0
Food Process Type B 0
Flow Input 2 0

APPENDIX D



CAPACITY		25	30	40	50	60	75	80	100	125	150	200	250	300
DIMENSIONS	A	23	23	25 1/2	29 1/2	32	32	36	36	38	42	48	52	54
	B	20 1/4	22	23 1/2	23	24	27 1/2	25	30	34	34 1/2	36	39	42
	C	44	46	47 1/2	48	49 1/2	53	50 1/2	55 1/2	60	61 1/2	64	67	70
	D	12 1/2	14	14 1/4	12 3/4	12 1/2	16	11 1/2	16 1/2	19 1/2	19	17 1/2	18 1/2	20 1/2
	E	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	1	1	1	1
	F	2 3/4	3	3	3	3 1/2	3 1/2	3 1/2	3 1/2	4	4	5	5	5



CAPACITY		5	10	15	20	25	30	40	50	60	75	80
DIMENSIONS	A	16 3/4	17 5/8	18 3/4	22	23	23	25 1/2	29 1/2	32	32	36
	B	10	14	16 1/2	17	20 1/4	22	23 1/2	23	24	27 1/2	25
	C	26 1/8	30 1/8	32 5/8	33 1/8	36 3/8	38 1/8	39 5/8	39 1/8	40 1/8	43 5/8	41 1/8
	D	15	15	15	15	15	15	15	15	15	15	15
	E	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	2
	F	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	2
	G	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	3/4	1

CAPACITY		100	125	150	200	250	300	400	500	600	750	1000
DIMENSIONS	A	36	38	42	48	52	54	58	62	66	68	72
	B	30	34	34 1/2	36	39	42	47	51	54	62	72
	C	46 1/8	50 1/8	50 5/8	52 1/8	55 1/8	58 1/8	63 1/8	67 1/8	70 1/8	78 1/8	88 1/8
	D	15	15	15	15	15	15	15	15	15	15	15
	E	2	2	3	3	3	3	3	3	3	3	3
	F	2	2	2	2	2	2	2	2	3	3	3
	G	1	1	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	2	2	2

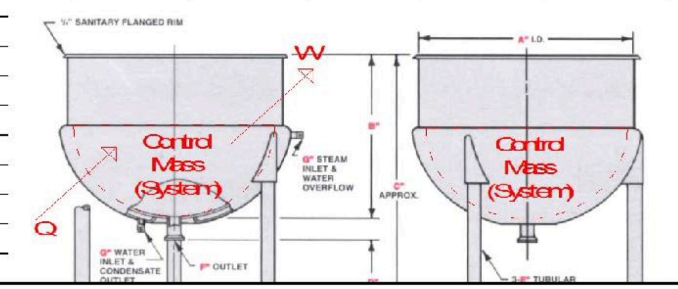
APPENDIX E

Test 1	Ti (Water)	Ti (Product)	Tf (Product)	Delta T	Agitator	Time Start	Time End	Hrs	Min	Tot. Hrs	Flow	Volume	Total test flow
Kettle No.	F	F	F	F	y/n	h:mm:ss	h:mm:ss				gpm	gal	gph
3		206	87	-119		4:50:00 PM	7:25:00 PM	2.00	35.00	2.58	14	2170	840
8		206	87	-119		6:05:00 PM	7:45:00 PM	1.00	40.00	1.67	15	1500	900
10		208	87	-121		5:00:00 PM	6:45:00 PM	1.00	45.00	1.75	14	1470	840
Test 2													
8		203	87	-116	y	2:10:00 PM	4:10:00 PM	2.00	0.00	2.00	14	1680	840
9		207	79	-128	y	2:10:00 PM	4:10:00 PM	2.00	0.00	2.00	15	1800	900
10		204	88	-116	y	2:10:00 PM	4:10:00 PM	2.00	0.00	2.00	14	1680	840
Test 3													
9		204	80	-124		8:55:00 AM	10:25:00 AM	1.00	30.00	1.50	14	1260	840
10		198	83	-115		8:55:00 AM	10:25:00 AM	1.00	30.00	1.50	15	1350	900
Test 4													
16		203	80	-123	y	8:20:00 AM	10:35:00 AM	2.00	15.00	2.25	15	2025	900
17		204	87	-117	y	8:20:00 AM	10:35:00 AM	2.00	15.00	2.25	14	1890	840
18		206	79	-127	y	8:20:00 AM	10:35:00 AM	2.00	15.00	2.25	15	2025	900
Test 5													
17	73	166	82	-84	y	8:00:00 PM	10:30:00 PM	2.00	30.00	2.50	14	2100	840
18	73	166	76	-90	y	8:00:00 PM	10:30:00 PM	2.00	30.00	2.50	15	2250	900
Test 6													
18	73	205	91	-114	y	1:00:00 PM	2:50:00 PM	1.00	50.00	1.83	14	1540	840
Test 7													
18	68	205	85	-120	y	9:35:00 AM	11:25:00 AM	1.00	50.00	1.83	15	1650	900
17	68	207	85	-122	y	9:35:00 AM	11:25:00 AM	1.00	50.00	1.83	15	1650	900
Test 8													
9	67	195	93	-102		10:00:00 AM	12:30:00 PM	2.00	30.00	2.50	14	2100	840
9	67	180	90	-90		4:15:00 PM	6:45:00 PM	2.00	30.00	2.50	15	2250	900

Condition	Ti (Water)	Ti (Product)	Tf (Product)	Delta T	Tot. Hrs	Flow	Volume	Total test flow
Average	70	199	85	-114	2.07	15	1800	870
All Tests	F	F	F	F	hrs	gpm	gal	gph
Extreme	73	208	76	-132	1.50	15	1350	900
All Tests	max F	max F	min F	delta F	min hrs	max gpm	gal	gph
Representative	70	206	79	-127	2.25	15	2025	900
Test 4, No. 18		F	F	F	hrs	gpm	gal	gph

APPENDIX F

	A	B	C	D	E	F	G	H	I
1	Given:								
2		Variable	Value	Unit	Description				
3		No. of Kettles		8					
4		Vkettle		500	gal	(total kettle volume)			
5		Annual Water Consumption		17.89	mil. gal.				
6		Kettle Dimensions			inches	(see APPENDIX D)			
7		No. of cooks per day		5					
8		Operation Days		311					
9		Kettle Testing Data							
10		Plant Location: Santa Rosa							
11		Coolant q		15	gpm				
12	Req'd:								
13	i)	Total heat transferred out of the beans, Qout (BTU)							
14	ii)	Rate of heat transfer using							
15		information provided by the kettle							
16		cooling tests:							
17	iii)	Final water temperature once passed through kettle jacket, Tf (F)							
18	Assume:								
19	a.	Kettle jacketed area is a hemi-sphere							
20	b.	Food product filled only to jacketed portion							
21	c.	Sensible heat transfer (no phase change of refrigerant)							
22	d.	Control mass calculation for part i) and ii)							
23	e.	Control volume calculation for part iii)							
24	f.	According to the facility manager, kettle's volume filled with 2/3rd of the product							
25	g.	Water audit based on 365 days per year							
26	h.	kettle lids closed							
27	Schematic:								
28									
29									
30									
31									
32									
33									
34									
35									
36									
37									



Sol'n:

i)	System:	Beans (food inside kettle)
	Boundary:	Steel (kettle walls)
	Surrounding:	Water (in kettle jacket)

Guided by the zeroth law of thermodynamics, heat transfer can be expected from a variance in temperatures across the boundary.

Additionally, the first law of thermodymaics states that energy is neither created nor destroyed, only transferred or transformed to/from the system, through a defined boundary, and from/to a surrounding environment.

It is important to note that only **<u>changes</u>** in energy can be readily observed with this simplicity.

The simplified energy balance equation for a closed system is denoted as follows:

$$\Delta E\ total = 0 = \Delta U + \Delta KE + \Delta PE \text{ , Eqn. 1}$$

We know that the system is not changing in kinetic or potential energy, therefore:

$$\Delta E\ total = 0 = \Delta U = Q_{net} - W_{net}\text{, Eqn. 2}$$

where,
Q = heat, + into system, - out of system
W = work, - into system, + out of system

*Note the sign convention is relative to the system

Since this is an isochoric process (constant volume), we assume the fluid does no work on or by. Another assumption is that the lids on the kettles are kept closed during the cooking/cooling process. Therefore, this creates a fully "closed" system where no mass crosses the boundary. Therefore, the change in total energy of the fluid can be described by:

$$\Delta E_{total} = \Delta U_{total} = Q_{net} = 0 = Q_{surrounding} - Q_{system} \text{ Eqn. 3}$$

$$Q_{surrounding} = Q_{system} \text{ Eqn. 4}$$

Equation 4 says that the quantity of heat removed from the cooked food (system), is transferred to and from the water-jacket (surrounding).

Now that we have simplified the equation, we can use the "kettle cooling test" information to compute how much sensible heat is rejected from the cooked product with the current once-through cooling process.

With the inital and final temperature points of the food product, we know that the **sensible** heat transfer equation can be used.

$Q_{Product} = m\,c\,\Delta T$, Eqn. 5

$Q = m\,(lb) * c\left(\frac{BTU}{lb * F}\right) * \Delta T\ (F)$

where,
m= mass
c = heat capacity
ΔT = temperature change

Note: There is no expected phase change of the product during heating.

It is important to understand the conditions under which the the kettle tests were conducted in order to achieve a representative value which will be used to size our system for sizing our system.

I've selected three methods we can use to approximate the total heat transferred:

a) average values from all kettle tests — test 10.1
b) most extreme cooling conditions
c) most representative kettle test conditions

The most representative kettle would be one which was guarenteed to have the lid closed for the duration of the cooling test, was agitated, and only represented the 500 gal container

Representative weight for the food product is given by the product of the density of beans and 60% of the volume of the 500gal container. Our representative food mass is 2679lbs.

Heat capacity indicates how much heat energy a certain substance can absorb per degree-mass change in temperature.

We found a reasonable value from the following resource below ----

HERE

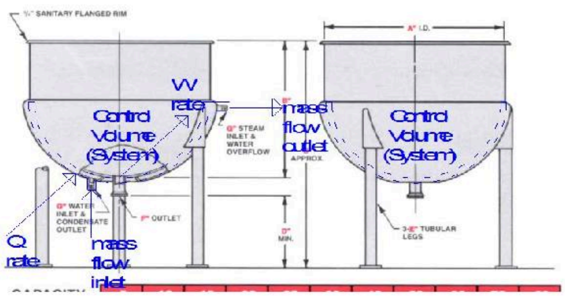
a) Obtaining the averages from each of the kettle cooling tests will give us appropriate

	A	B	C	D	E	F	G	H	I
130		values in order to meet the needs of our design.							
131									
132		m	2,700	lbs					
133		c	0.90	BTU/(lb*F)					
134		ΔT	-114	F					
135		Ti	199	F					
136		Tf	85	F					
137	b)	In order to ensure our operation is properly cooled, we can apply the most extreme case							
138		scenario (i.e. max. water TiP, min. water TfP, and longest test duration)							
139									
140		m	2,700	lbs					
141		c	0.90	BTU/(lb*F)					
142		ΔT	-132	F					
143		Ti	208	F					
144		Tf	76	F					
145	c)	m	2,700	lbs					
146		c	0.90	BTU/(lb*F)					
147		ΔT	-127	F					
148		Ti	206	F					
149		Tf	79	F					
150									
151									
152									
153		Q product	Per Kettle	All 8 Kettles	Duration				
154		Condition	BTU	BTU	hrs				
155	a)	Average	-277,000	-2,216,000	2.07				
156	b)	Extreme	-321,000	-2,568,000	1.50				
157	c)	Rep.	-309,000	-2,472,000	2.25				
158									
159	ii)	The rate of heat transfer is dependent on the amount of time the kettle tests consumed.							
160		Most of the tests were around two hours. Test durations will be deduced similarly to the							
161		corresponding three methods.							
162									
163									
164									
165		Q rate	Per Kettle	All 8 Kettles	Per Kettle	All 8 Kettles			
166		Condition	BTU/hr	BTU/hr	R Tons	R Tons			
167	a)	Average	-134,000	-1,072,000	-11	-89			
168	b)	Extreme	-214,000	-1,712,000	-18	-143			
169	c)	Rep.	-138,000	-1,104,000	-12	-92			
170		**I have chosen to select method A as the best value for sizing our cooling tower.**							
171									
172	iii)	To calculate the temperature of the return water to the cooling tower, we must use the							
173		bring to mind the following equation:							
174					$-Q_{Product} = Q_{Water}$				
175					or				

	A	B	C	D	E	F	G	H	I
176				$-Q_{system} = Q_{surroundings}$, Eqn. 4					
177									
178									
179									
180									
181		We can assume the heat contributed due to friction in the kettle jacket to be negligible.							
182		Knowing the properties of water, specifically that it is being sensibly heated, we can apply							
183		**sensible** heat transfer equation again.							
184									
185				$Q = m(lb) * c\left(\frac{BTU}{lb*F}\right) * \Delta T$ (F)					
186									
187									
188				(+)	heat into system				
189		Q system		134,000	BTUH	(from "average" condition)			
190									
191		Twater f	=	Qsystem/(mC) +Twater i					
192									
193		m (water)	=	(gal./min) (cooling time) (lb/gal)					
194			=	lb					
195		mass flow	=	7,486	lb/hr				
196									
197		1 kJ/kgK	=	0.24	BTU/(lbF)				
198									
199		C* (water)	=	4.18	kJ/(kgK)				
200			=	1.00	BTU/(lbF)				
201		T (water)	=	70.00	F	$\Delta T = T_f - T_i = \frac{\dot{Q}}{L_{in} * c_w}$, Eqn. 3 ,			
202									
203									
204		**Twater f**	**=**	**88**	**F**				
205		**Approximate ΔT =**		**18**	**F**				
206									

T-v diagram and steam table verification of T final (T outlet) of water

I. Sketch



II. Given

state 1 & 2 are in compressible liquid

Q out	-277000	BTU
v flow	15.00	gal/min
Dia. Inlet	1.50	inch
A inlet	0.0123	ft^2
cook time	2.07	hours

III. Assume

Control volume

Steady-state operation

Liquid relatively incompressible

Conservation of mass

m dot inlet = m dot outlet = m dot

A inlet = A outlet = A

Pressure at inlet	60.31	psi	*non-significant p-loss
Pressure at outlet	54.56	psi	

ΔKE = 0, ΔPE = 0

Fluids highly incompressible, so enthalpy is largely a funciton of temperature

Due to the behavior of water in the compressible liquid region, the enthalpy for a liquid can be well represented by the enthalpy for a saturated liquid **at a given temperature.**

IV. Analyze

a. find final temperature of water

$$\frac{\partial E}{dt} = 0 = \dot{Q}_{CV} - \dot{W}_{CV} + \dot{m}[(h_i - h_e) + .5(V^2{}_i - V^2{}_e) + g(z_i - z_e)$$

Simplify first law

$$\frac{\partial E}{dt} = 0 = \dot{Q}_{CV} + \dot{m}[(h_i - h_e)]$$

Inlet Water 70 F

Qcv	-277000	(+) into system	
Q dot cv	-133852	BTU/hr	
Q dot cv	-37	BTU/s	
Wcv	0.00	BTU/hr	isochoric process
spec. vol	0.01606	ft^3/lb	at 70 degree F
v dot	15.00	gal/min	
m dot	2.08	lb/s	
h i	38.09	BTU/lb	
h e	55.96	BTU/lb	
Interpolate!			
T		h	
F		BTU/lb	
	70.00		38.09
	x		55.96
	96.00		64.06
Tf	87.9	F	

Inlet Water 75 F

Qcv	-277000	(+) into system	
Q dot cv	-133852	BTU/hr	
Q dot cv	-37	BTU/s	
Wcv	0.00	BTU/hr	isochoric process
spec. vol	0.01606	ft^3/lb	at 70 degree F
v dot	15.00	gal/min	
m dot	2.08	lb/s	
h i	43.09	BTU/lb	
h e	60.96	BTU/lb	
Interpolate!			
T		h	
F		BTU/lb	
	75.00		43.09
	x		60.96
	96.00		64.06
Tf	92.9	F	

Variance in the final temperatures of the food product are a result of varying flows through the kettles, size varience in kettles, and varying inlet water temperature dependent on the kettle testing date. Additionally, the final temperature of the water is indicative of initial cooling conditions when the rate of heat transfer is highest.

As the temperature of the food product approaches 85F, we can expect exit water temperatures to be lower due to the decreased rate of heat transfer. This behavior of the final water temperature can be verified by observing the temperature profile for test 4, Kettle 18 in and newtons law of cooling.

311		Since BAC selection software follows the standards and codes listed above, specifically the	
312		minimum approach parameter, I will need to reconsider my software selection. This re-	
313		selection is required because the leaving fluid temperature must be 5F above regional wet	
314		bulb temperature according to the CTI STD-201 limits of thermal certification (CTI).	

Additionally, we must make sure that our design criteria indicates a heat rejection above the value required by the average conditions.

2 Checks		Reserve	
Minimum Approach Requirement	OKAY	2.5	F
Design rejection meets demand	OKAY	12%	

What are the CTI STD-201 limits of thermal certification?

As stated in STD 201RS, the limits of certification are different for cooling towers/closed circuit coolers and evaporative condensers.

For Cooling Towers and Closed-Circuit Coolers:

	SI Units	IP Units
Wet Bulb Temperature	10°C to 32.2°C	50°F to 90°F
Maximum Process Fluid Temperature	51.7°C	125°F
Minimum Range	2.2°C	4°F
Minimum Approach	2.8°C	5°F
Barometric Pressure	91.4 kPa to 105 kPa	27 in Hg to 31 in Hg

Sonoma County WB Temperature Those percentages represent the percentage of time the wet-bulb temperature ill be above the listed wet-bulb e.g. For Sonoma Co., the wet-bulb temp is expected to go above 69.2 for .4% of the year {35 hours} or above 67.5 for 1% of the year {88 hours}. This is important because the evaportive cooling affect will be compromised for that short period of time. Depending on the application, this may be more or less important.

Low wet-bulb temperatures indicate that the air (based on low humidity levels) allow for the water-coated thermometer to evaporate, thus removing heat, thus lowering temperatures.

WB	WB	DD Heating	DD Cooling
0.40%	1.00%	3047	375
69.20	67.50		

*How much variation from the base temperature of 65F, extra heating or cooling required

** WB and DD from http://www.baltimoreaircoil.com/english/resource-library/file/651

To ensure redundancy of design, I selected used a delta T of 20F (increasing the range by 2 degrees). This adjustment increased the required cooling tower thermal efficiency and rate of heat rejection. With these considerations, the following selection was made with critera below:

Selection Criteria: 120 gpm, 95F/ 75F/ 67.5F, pure water, price ranking

Inlet Temp	95.0	Outlet Temp	75.0	Wet-Bulb	67.5

	A	B	C	D	E	F	G	H	I
361		**BTUH rejection**		**-1,200,000**			**-100 Tons**	**Efficiency**	**73%**
362		**Selection Type**	**Product**	**Qty**	**Model**	**Series**	**Total**	**Total**	**Warnings**
363							**Fan**	**Pump**	**Exist**
364							**Motor**	**Motor**	
365							**(HP)**	**(HP)**	
366		**LFC**	FXV	1.00	FXV-0806B-28D-L	no	15.00	2.00	No
367		**Rec**	**FXV**	**1.00**	**FXV-0806B-32D-K**	**no**	**10.00**	**2.00**	**No**
368			**Unit**	**Fluid**	**Reserve**	**Price**	**Payback**	**Energy**	**Warnings**
369			**Height**	**Pressure**	**Capability**	**Rank**	**(Years)**	**Rating**	**Exist**
370				**Drop**	**(%)**			**(USGPM**	
371				**(psi)**				**/ HP)**	
372		**LFC**	18′ 01.00″	0.86	5.16	1.00	--	23.29	No
373		**Rec**	**18′ 01.00″**	**0.95**	**1.48**	**1.02**	**1.07**	**32.00**	**No**
374									

APPENDIX G

Temp. °F	Press. lbf/in.²	Specific Volume ft³/lb Sat. Liquid v_f	Specific Volume ft³/lb Sat. Vapor v_g	Internal Energy Btu/lb Sat. Liquid u_f	Internal Energy Btu/lb Sat. Vapor u_g	Enthalpy Btu/lb Sat. Liquid h_f	Enthalpy Btu/lb Evap. h_{fg}	Enthalpy Btu/lb Sat. Vapor h_g
64	0.2952	0.01604	1056	32.09	1031.8	32.09	1057.3	1089.4
66	0.3165	0.01604	988.4	34.09	1032.4	34.09	1056.2	1090.3
68	0.3391	0.01605	925.8	36.09	1033.1	36.09	1055.1	1091.2
70	0.3632	0.01605	867.7	38.09	1033.7	38.09	1054.0	1092.0
72	0.3887	0.01606	813.7	40.09	1034.4	40.09	1052.8	1092.9
74	0.4158	0.01606	763.5	42.09	1035.0	42.09	1051.7	1093.8
76	0.4446	0.01606	716.8	44.09	1035.7	44.09	1050.6	1094.7
78	0.4750	0.01607	673.3	46.09	1036.3	46.09	1049.4	1095.5
80	0.5073	0.01607	632.8	48.08	1037.0	48.09	1048.3	1096.4
82	0.5414	0.01608	595.0	50.08	1037.6	50.08	1047.2	1097.3
84	0.5776	0.01608	559.8	52.08	1038.3	52.08	1046.0	1098.1
86	0.6158	0.01609	527.0	54.08	1038.9	54.08	1044.9	1099.0
88	0.6562	0.01609	496.3	56.07	1039.6	56.07	1043.8	1099.9
90	0.6988	0.01610	467.7	58.07	1040.2	58.07	1042.7	1100.7
92	0.7439	0.01611	440.9	60.06	1040.9	60.06	1041.5	1101.6
94	0.7914	0.01611	415.9	62.06	1041.5	62.06	1040.4	1102.4
96	0.8416	0.01612	392.4	64.05	1041.2	64.06	1039.2	1103.3
98	0.8945	0.01612	370.5	66.05	1042.8	66.05	1038.1	1104.2
100	0.9503	0.01613	350.0	68.04	1043.5	68.05	1037.0	1105.0

Temp. °F	Press. lbf/in.²	Specific Volume ft³/lb Sat. Liquid v_f	Specific Volume ft³/lb Sat. Vapor v_g	Internal Energy Btu/lb Sat. Liquid u_f	Internal Energy Btu/lb Sat. Vapor u_g	Enthalpy Btu/lb Sat. Liquid h_f	Enthalpy Btu/lb Evap. h_{fg}	Enthalpy Btu/lb Sat. Vapor h_g
70	0.3632	0.01605	867.7	38.09	1033.7	38.09	1054.0	1092.0
72	0.3887	0.01606	813.7	40.09	1034.4	40.09	1052.8	1092.9
74	0.4158	0.01606	763.5	42.09	1035.0	42.09	1051.7	1093.8
76	0.4446	0.01606	716.8	44.09	1035.7	44.09	1050.6	1094.7
78	0.4750	0.01607	673.3	46.09	1036.3	46.09	1049.4	1095.5
80	0.5073	0.01607	632.8	48.08	1037.0	48.09	1048.3	1096.4
82	0.5414	0.01608	595.0	50.08	1037.6	50.08	1047.2	1097.3
84	0.5776	0.01608	559.8	52.08	1038.3	52.08	1046.0	1098.1
86	0.6158	0.01609	527.0	54.08	1038.9	54.08	1044.9	1099.0
88	0.6562	0.01609	496.3	56.07	1039.6	56.07	1043.8	1099.9
90	0.6988	0.01610	467.7	58.07	1040.2	58.07	1042.7	1100.7
92	0.7439	0.01611	440.9	60.06	1040.9	60.06	1041.5	1101.6
94	0.7914	0.01611	415.9	62.06	1041.5	62.06	1040.4	1102.4
96	0.8416	0.01612	392.4	64.05	1041.2	64.06	1039.2	1103.3
98	0.8945	0.01612	370.5	66.05	1042.8	66.05	1038.1	1104.2
100	0.9503	0.01613	350.0	68.04	1043.5	68.05	1037.0	1105.0

APPENDIX H

Total water use by a cooling tower is given by the formula below:

$$Water\ Use = Evaporative\ Loss + Blowdown + Drift$$

Calculating the total water usage is important for the pro forma financial statements to investigate payback on project. Computation can involve two different methods.

Method 1 Psychrometric Chart and Rule of Thumb Equations
Method 2 Online Water Loss Calculator

Both of these methods will be used to cross reference each other to ensure a representative value for water loss.

Given:

Tin	95	F	water
Tout	75	F	water
wet bulb	67.5	F	
dry bulb	80	F	*santa rosa high summer temperature
v dot	120	gpm	water
m dot	59,832	lb/hr	water
density	8.31	lb/gal	water
approach	7.5	F	
Qdot out	-1,200,000	BTUH	energy rejected by selected cooling tower
3 cycles of concentration			

Req'd:

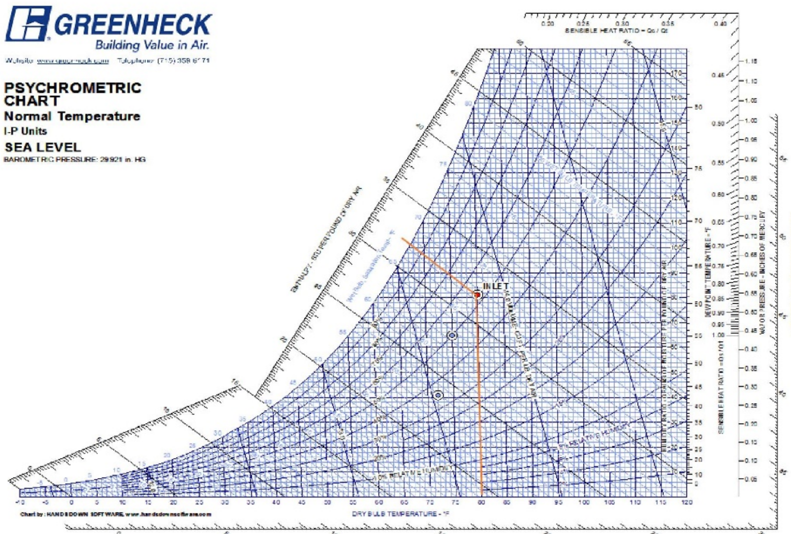
a) Determine evaporative "make-up" water component
b) Determine drift
c) Determine blowdown

Assume: mass+energy transfer problem
total energy rejected by tower absorbed by air (water vapor + dry air)
increase in humidity ratio due solely from evaopration of water through tower
exit air at 100% relative humidity
1 atm barometric pressure
fan running at half speed, affinity law indicates half mass flow
average specific volume of INLET and EXIT conditions used to calculate density

Sol'n:

Method 1

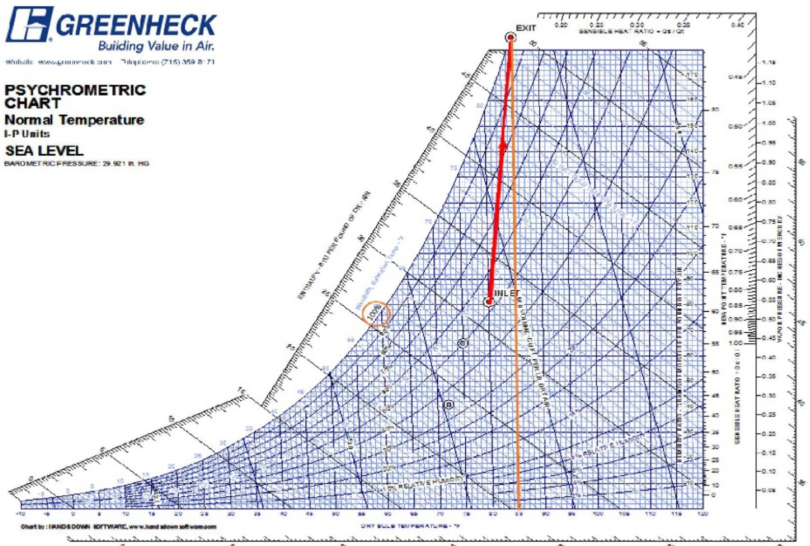
a) Determine evaporative "make-up" water component



For exit conditions, use 100% RH and ***tentative*** dry bulb temperature in order to plot a point. This can be seen in first psychrometric chart below.

Once point is plotted, use software to connect the dots into a process. The software indicates how much energy enters the air based on the difference of enthalpy. This change in enthalpy can be seen in the second chart below.

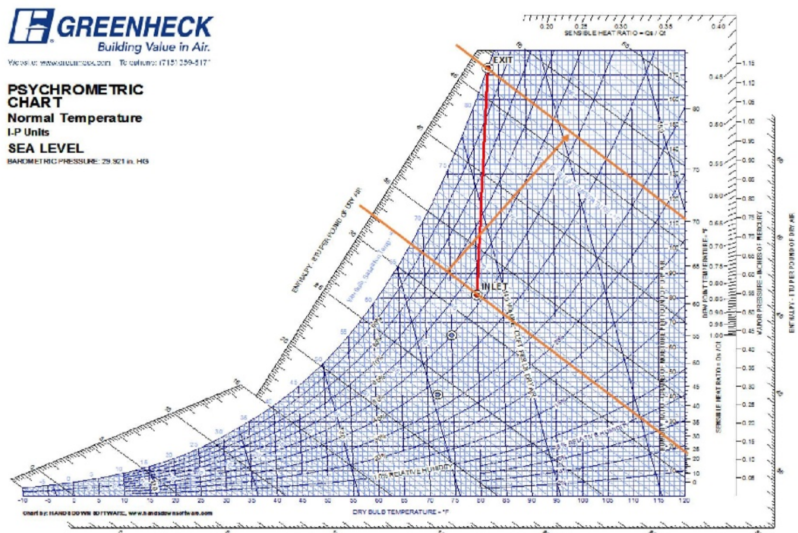
Since the total heat rejected from the tower is assumed to enter the air, a representative dry bulb temperature will be assigned based on a total rate of heat rejection of: approximately 1,200,000 BTUH



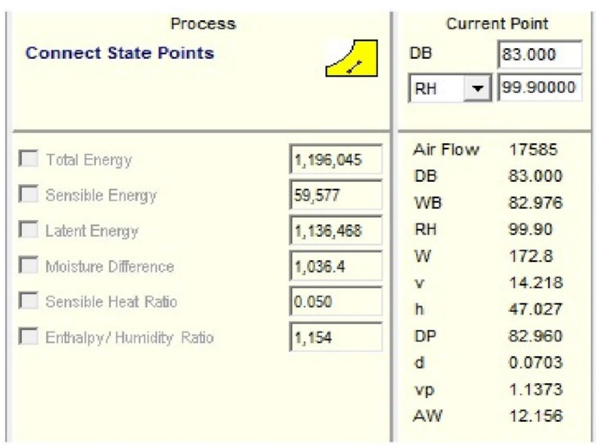
The software indicates that the total heat accepted by the air to be 1,400,000 BTUH which is slightly higher than expected conditions. By testing other dry bulb exit air temperatures, 83F was determined to be the most accurate. Equation 6 is the hand calculation used by the software to determine the total energy gained by the air.

$$air\ flow\left(\frac{ft^3}{min}\right) * density\ \frac{lb\ dry\ air}{ft^3} * enthalpy\ change\ \left(\frac{BTU}{lb\ dry\ air}\right) * \frac{60min}{hr}$$
$$= heat\ rejection\ rate\ \left(\frac{BTU}{hr}\ or\ BTUH\right)$$

Qdot	1,206,000 BTUH	*positive value because heat absorbed into air (water vapor + dry air)



The image below indicates the conditions for the EXIT state (current point) and the process between the two points. It should be noted that this is at conditions of half fan speed. The sensible and latent components of total energy can be seen below as well.



Now that we have established a EXIT state point, we can now use this point to identify the humidity ratio for both INLET and EXIT points. Taking the difference between these two will indicate the increase in humidity ratio which is equvalient to the amount of evaporated water from the tower per lb dry air.

Method 2 A Water Usage Calculator required the following inputs.

a)

b)

c)

Operating Conditions

Tower Water Flow	120	gpm	27	m^3/h
Hot Water Temperature	95.00	°F	35.00	°C
Cold Water Temperature	75.00	°F	23.89	°C
Wet-Bulb Temperature	67.50	°F	19.72	°C
Drift Rate	0.005	%		
Concentrations	3			

With this information, the calculator produced the following results

Water Usage

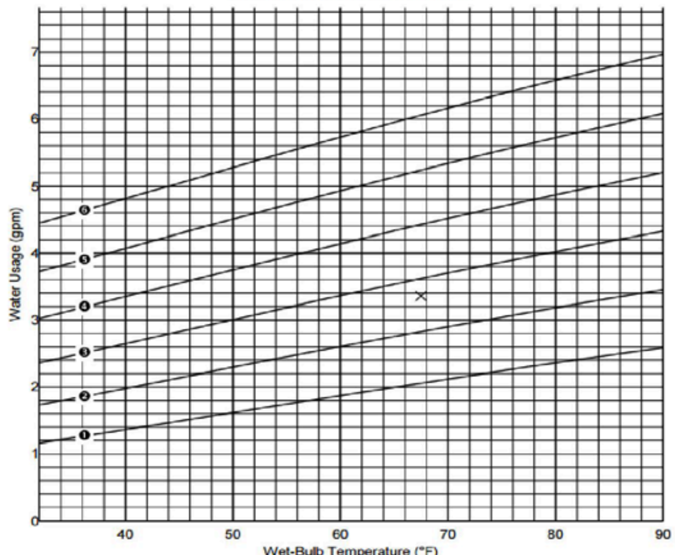
Evaporation	2.24	gpm	0.51	m^3/h
Drift	0.01	gpm	0.00	m^3/h
Blow down	1.11	gpm	0.25	m^3/h
Total Usage	3.35	gpm	0.76	m^3/h

The following graph was generated by the calculator to indicate water usage based on varying wet bulb temperature and cooling tower range

UPDATE™ Version 1.1.1

5/14/2017 5:07:47 PM

Estimated Cooling Tower Water Usage
Includes evaporation, drift, and blow down



Water usage rates are provided as an estimate only and for educational purposes. Consult your local representative or sales person to determine the actual water usage requirements for your application.

Design Conditions

Tower Water Flow	120 gpm
Hot Water Temperature	95.00 °F
Cold Water Temperature	75.00 °F
Wet-Bulb Temperature	67.50 °F
Drift Rate	0.005 %
Concentrations	3

Legend

- ❶ 10 °F Range
- ❷ 16 °F Range
- ❸ 22 °F Range
- ✕ Design Point
- ❹ 28 °F Range
- ❺ 34 °F Range
- ❻ 40 °F Range

Both Method 1 and 2 provided reasonable values. It should be taken into consideration that changing ambient dry bulb inlet temperature, tower aging, and fan speed will produce varying water loss values. However, the inlet dry bulb temperature, fan speed, and wet bulb temperatures selections were intentional to represent a "design day".

3gpm of water use during tower operation will be used for the pro forma financial statements to illustrate realistic payback period and return on investment. To accommodate, blowdown for Method 1 was raised by .07gpm

Below is the distribution of water use between each category relative to total water use:

Annual Usage

3110 operational hours

Blowdown	Drift	Evaporativ	Total Water Use	
127,828	67,176	363,883	558,887	gal/year
23%	12%	65%	100%	

APPENDIX I

Baltimore Aircoil Company, Inc.

Closed Circuit Cooling Tower Selection Program

Version:	7.5.3 NA
Product data correct as of:	March 03, 2017

Project Name:	CCCT Solution
Selection Name:	Final Selection Rec
Project State/Province:	California
Project Country:	United States
Date:	April 27, 2017

Model Information

Product Line:	FXV/FXV3
Model:	FXV-0806B-32D-K
Number of Units:	1
Coil Type:	Standard Coil
Coil Finning:	None
Fan Type:	Standard Fan
Fan Motor:	(1) 10.00 = 10.00 HP/Unit
Total Standard Fan Power:	Full Speed, 10.00 BHP/Unit
Total Pump Motor Power:	(1) 2.00 = 2.00 HP/Unit
Intake Option:	None
Internal Option:	None
Discharge Option:	None

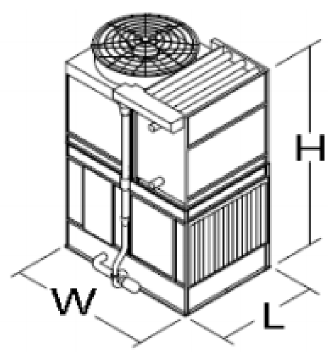
Design Conditions

Fluid:	Water
Flow Rate:	120.00 USGPM
Entering Fluid Temp.:	95.00 °F
Leaving Fluid Temp.:	75.00 °F
Wet Bulb Temp.:	67.50 °F
Fluid Pressure Drop:	0.95 psi
Reserve Capability:	1.48%

Thermal performance at design conditions and standard total fan motor power is certified by the Cooling Technology Institute (CTI).

Engineering Data, per Unit

Unit Length:	05' 11.75" + 01' 06.00" (Pump) = 07' 05.75"	(Total)
Unit Width:	08' 05.75"	
Unit Height:	18' 01.00"	
Approximate Shipping Weight:	7,176	lbs
Heaviest Section:	4,930	lbs
Approximate Operating Weight:	10,578	lbs
Approximate Remote Sump Operating Weight:	9,613	lbs
Air Flow:	35,170	CFM
Spray Water Flow:	290	USGPM
Coil Volume:	117	U.S. gallons



Coil Connections:

(1) 4" Coil Inlet and Outlet, Based on 120.00 USGPM Flow per Unit

Remote Sump Connections:	(1) 6"	
Heater kW Data (Optional)		
0°F (-17.8°C) Ambient Heaters:	(1) 4	kW
-20°F (-28.9°C) Ambient Heaters:	(1) 6	kW

Minimum Distance Required:

From Solid Wall:	3.5	ft
From 50% Open Wall:	3	ft

Energy Rating:

32.00 per ASHRAE 90.1, ASHRAE 189 and CA Title 24.

Note: These unit dimensions account for the selected fan type for the standard cataloged drive configuration, but they do not account for other options/accessories. Please contact your local BAC sales representative for dimensions of units with other options/accessories.

APPENDIX J

Scenario 1, 8 Kettles, Area 2

Location	Loc. Description	Nominal Pipe Diameter	U/S Pressure	U/S head	Velocity	U/S Velocity Head	Flow	Change in Distance	Change in Elevation	Misc. Friction Loss	D/S Friction Loss	No. of turns
letter		in	psi	ft	ft/s	ft	gpm	ft	ft	ft	ft	
A	Pump Outlet	3.00	78.00	180.18	5.45	0.46	120.00	15.00	0.00	2.31	0.65	4.00
B	Tower Entrance	3.00	76.72	177.22	5.45	0.46	120.00	0.00	7.00	1.18	0.00	0.00
C	Tower Exit	3.00	73.18	169.05	5.45	0.46	120.00	7.00	-7.00	0.58	0.30	1.00
D*	Tower GL	3.00	75.83	175.17	5.45	0.46	120.00	32.00	0.00	9.04	1.38	7.00
E	Wall GL (3" mani)	3.00	71.32	164.75	5.45	0.46	120.00	40.00	40.00	0.58	1.72	1.00
F	Supply Wall RL	1.25	53.10	122.67	3.92	0.24	15.00	32.00	0.00	0.58	2.20	1.00
G	Supply RL (right turn)	1.25	51.90	119.89	3.92	0.24	15.00	160.00	0.00	0.58	11.01	1.00
H	Supply RL (left turn)	1.25	46.88	108.30	3.92	0.24	15.00	73.00	0.00	0.58	5.02	1.00
I	Supply RL end	1.25	44.46	102.70	3.92	0.24	15.00	40.00	-40.00	0.58	2.75	1.00
J	Supply GL	1.25	60.33	139.37	3.92	0.24	15.00	16.00	3.00	0.58	1.10	1.00
K	Kettle Inlet	1.25	58.31	134.69	3.92	0.24	15.00	0.00	0.00	13.28	0.00	3.00
L	Kettle Exit	1.25	52.56	121.41	3.92	0.24	15.00	8.00	-3.00	1.73	0.55	3.00
M	Submain Entrance	1.50	52.52	121.33	8.17	1.04	45.00	8.00	0.00	0.58	1.68	1.00
N	Submain Exit/ Return GL	1.50	51.55	119.07	8.17	1.04	45.00	40.00	40.00	0.58	8.41	1.00
O	Return RL	1.50	30.34	70.08	8.17	1.04	45.00	73.00	0.00	0.58	15.35	1.00
P	Return RL (right turn)	1.50	23.44	54.15	8.17	1.04	45.00	145.00	15.00	0.58	30.49	1.00
Q	Return (expansion)	2.50	3.63	8.39	6.86	0.73	105.00	10.00	0.00	2.00	0.82	0.00
R	Return RL (left turn)	2.50	2.41	5.57	6.86	0.73	105.00	8.00	0.00	0.58	0.66	1.00
S	Return RL (tot. flow)	2.50	1.78	4.11	7.84	0.96	120.00	18.00	0.00	0.00	1.88	0.00
T	Return Wall RL	2.50	0.96	2.23	7.84	0.96	120.00	40.00	-40.00	0.58	4.19	1.00
U	Wall GL (3" mani)	3.00	16.43	37.96	5.45	0.46	120.00	20.00	0.00	1.16	0.86	2.00
V	Expansion Tank Losses	3.00	15.56	35.94	5.45	0.46	120.00	5.00	0.00	0.58	0.22	1.00
W	Pump Inlet	3.00	15.22	35.15	5.45	0.46	120.00	0.00	0.00	0.00	0.00	1.00

Note: Friction losses based on engineering drawings that could not be included in the Appendices due to confidentiality

NPSHA	34.17 ft
Remaining Head	39.77 ft
Hvp (@ 90F)	1.60 ft
SF	4.00 ft

2 Grundfos

$$Hf\ (segment) = 10.5\left(\frac{GPMseg}{c}\right) * (Lsegment) * (ID)^{-4.87}\ Eq.1$$

Bernollis Equation = $\boldsymbol{P1} + \boldsymbol{Z1} + \left(\frac{v^2}{2g}\right) = P2 + \boldsymbol{Z2} + \left(\frac{v^2}{2g}\right) + \boldsymbol{Hf}$

$$Q = V * A$$

$$NPSHA = Remaining\ Head\ - Hvp\ - SF$$

NPSHA > NPSHR

It should be noted that the Hazen Williams equation is typically accepted in agricultural and civil applications. Darcy-Weibach equations should be used for final design.

According to pump curve, our NPSHR is well below NPSHA for both worst case and normal operating conditions.

APPENDIX K

Product photo could vary from the actual product

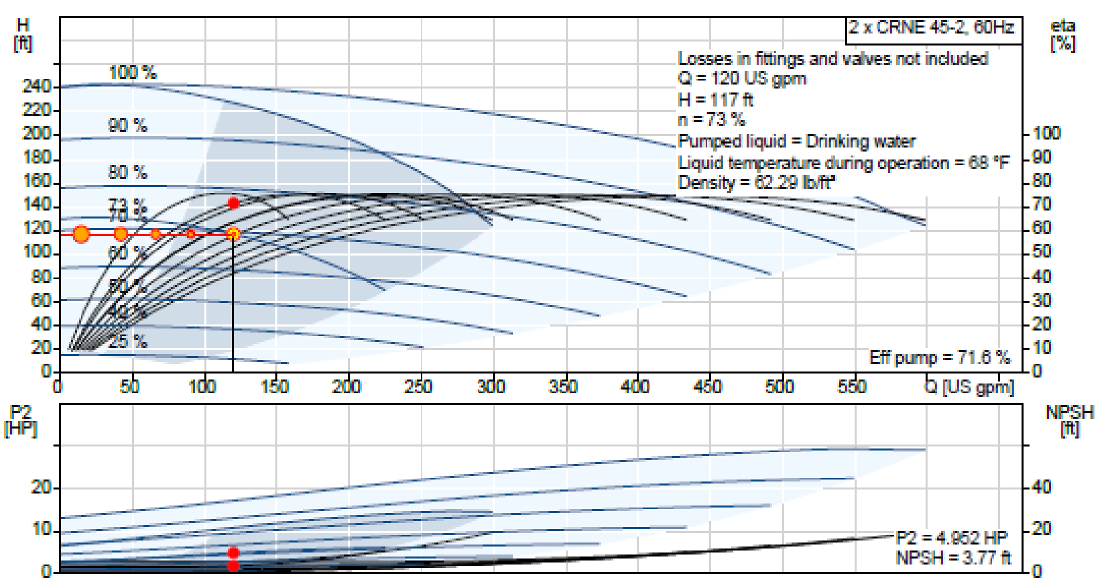
CRNE 45-2

Vertical, multistage centrifugal pump with integrated frequency converter. Pump materials in contact with the liquid are in stainless steel (EN 1.4301)

Conditions of Service	
Flow:	120 US gpm
Head:	117 ft
Efficiency:	59.9 %
Liquid:	Drinking water
Temperature:	68 °F
NPSH required:	3.77 ft
Viscosity:	1 cSt
Specific Gravity:	1.000

Pump Data	
Max pressure at stated temperature:	232 psi / 250 °F
Liquid temperature range:	32 .. 32 °F
Maximum ambient temperature:	104 °F
Shaft seal:	HQQE
Flange standard:	ANSI
Pipe connection:	3"
Product number:	98183877

Motor Data	
Rated power - P2:	14.75 HP
Rated voltage:	460 V
Main frequency:	60 Hz
Enclosure class:	IP55
Insulation class:	F
Motor protection:	YES
Motor type:	160AA



APPENDIX L

$ estimate of e-usage

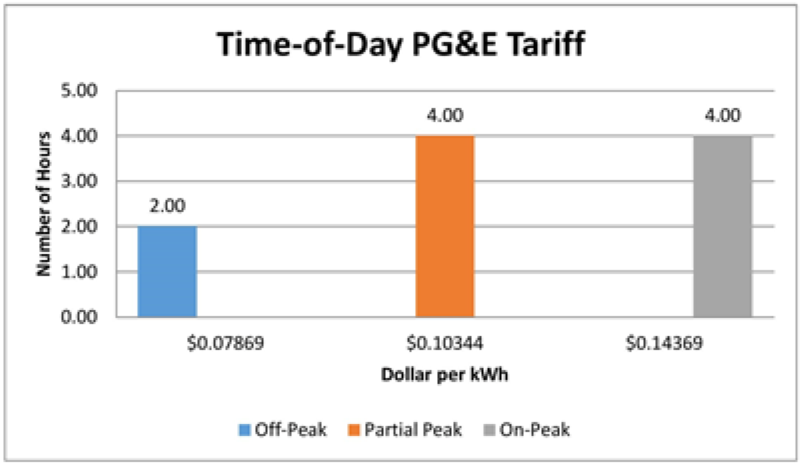
Given:	CCCT operation hours			10	hours/day
	Operational Days			311	days/work year
	Operation times		330am		to 9pm
	No. of batches			5	batches/day
	Fan HP			10	total HP
	Sump Pump HP			2	total HP
	Circulating Pump HP			10	total HP
	Unit No.			1	
	Model of Selection		FXV-0806B-32D-K		
Req'd:	i)	Determine hour of operation			
	ii)	Determine times of operation			
		*must consider peak rates for electricity			
	iii)	Determine industrial energy rate (E-20)			
		https://www.pge.com/tariffs/electric.shtml			
	iv)	Determine total horsepower input into CCCT			
		a) fan			
		b) sump pump			
		c) circulating pump			
	v)	Determine energy consumption		E - 20 rates	
	vi)	Determine cost of energy usage		$0.14369	per kWh
				$0.10344	per kWh
Sol'n:	i)	Hours of op.		$0.07869	per kWh
		3110 hours per year			

	Off-Peak	Partial Peak	On-Peak	
ii)				
iii)	$ 0.07869	$ 0.10344	$ 0.14369	per kWh
	2.00	0.00	0.00	hours
	0.00	4.00	0.00	
	0.00	0.00	4.00	

Estimated Kettle Operation Schedule

3:30:00 AM	4:30:00 AM	5:30:00 AM	6:30:00 AM	7:30:00 AM
		2		
8:30:00 AM	9:30:00 AM	10:30:00 AM	11:30:00 AM	12:30:00 PM
	2			
1:30:00 PM	2:30:00 PM	3:30:00 PM	4:30:00 PM	5:30:00 PM
2				4
6:30:00 PM	7:30:00 PM	8:30:00 PM	9:30:00 PM	
			4	

note: Demand charges will be neglected. This takes a full understanding of the facility operation. Power factor adjustment will also be neglected



vi)	Fan HP		10	total HP	7.5	kW
	Sump Pump HP		2	total HP	1.5	kW
	Circulating Pum		10	max HP	7.5	
	Unit No.		1			
	Total Power		12	HP	16.4	kW
	Off-Peak	Partial Peak		On-Peak		
	$ 0.07869	$ 0.10344		$ 0.14369		
	2.00	4.00		4.00		
	$ 2.58	$ 6.79		$ 9.43		

Total Cost of Energy Per Day $ 18.80
Total Cost of Energy Per Work Year* $ 5,846.47
*assuming energy rates are relatively constant
**rates based on Oct 1st 2016
Assume 3% increase per year
https://eec.ucdavis.edu/files/02-06-2014-The-Future-of-Electricity-Prices-in-California-Final-Draft-1.pdf
pg. 27

Power	=	(F*x)/t	=	(E)/t
	=	E/t * t (hours) * $/E		
	=	$ for annual energy consumption		

Ciculating Pump Horsepower Computation

Circulating Pump HP was based on interpretations from the pump curve and worst case pumping scenario for the system curve. This is where the highest flow and head would be required. This was based on friciton losses and elevation changes of the hydraulically furthest kettles in the facility. The selected pipe sizes and kettle deck operation should be further investigated to provide an optimized pump selection based on the system curve. By providing a slightly oversized pump station and variable frequency drive, operators can adjust pump speed according to what minimizes energy cost while maintaining production capacity.

Based on interpretation of the piping layout diagram provided, the following head and flow requirements were developed to represent the most power intensive pumping scenario. This occurs when the 8 hydraulically furthest, highest capacity kettles require cooling flow simultaneously. The following head and flow requirements were largely based on kettle design, friction loss estimates, static head, NPSH, and changes in elevation. Computations were based on Hazen Williams equations to provide rough estimates on pumping costs. Friction loss calculations will need to be reviewed by the refrigeration design engineer using the Darcy–Weisbach equation to account for turbulent flow and the temperature gradients throughout the pumping system.

With all this considered, the following values were deemed acceptable to estimate power req'd

	Normal Op.	Worst Case	
TDH	117	180	ft
Flow	120	120	gpm

The minimum energy req'd for each scenario to pump the water is calculated with the following:

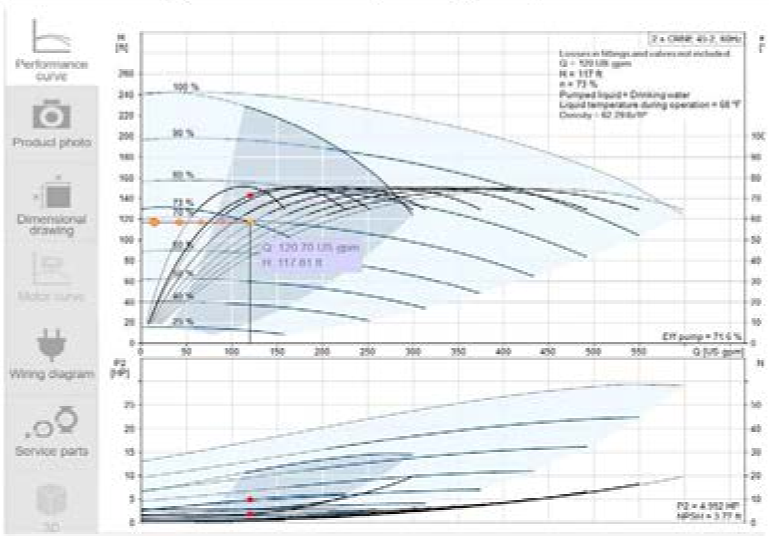
$$Water\ HP\ = \frac{TDH\ * GPM}{3960} \qquad Brake\ HP = \frac{WHP}{Pump\ Eff.} \qquad Motor\ HP = \frac{Brake\ HP}{Motor\ Eff.}$$

The following efficiencies were provided by Grundfos Pump Selection Software:

Model: CRNE 45-2 A-G-G-E-HQQE

	Normal Op	Worst Case	
Motor eff	86.2%		
Pump eff	71.6%		
WHP	3.55	5.45	hp
BHP	4.95	7.62	hp
Motor HP	5.74	8.84	hp

With a variable frequency drive, operators will be able to adjust motor rotational speed allowing for adjustable flow and head. A nominal **10 HP motor** was selected for conservative savings estimates on the capital budget analysis.



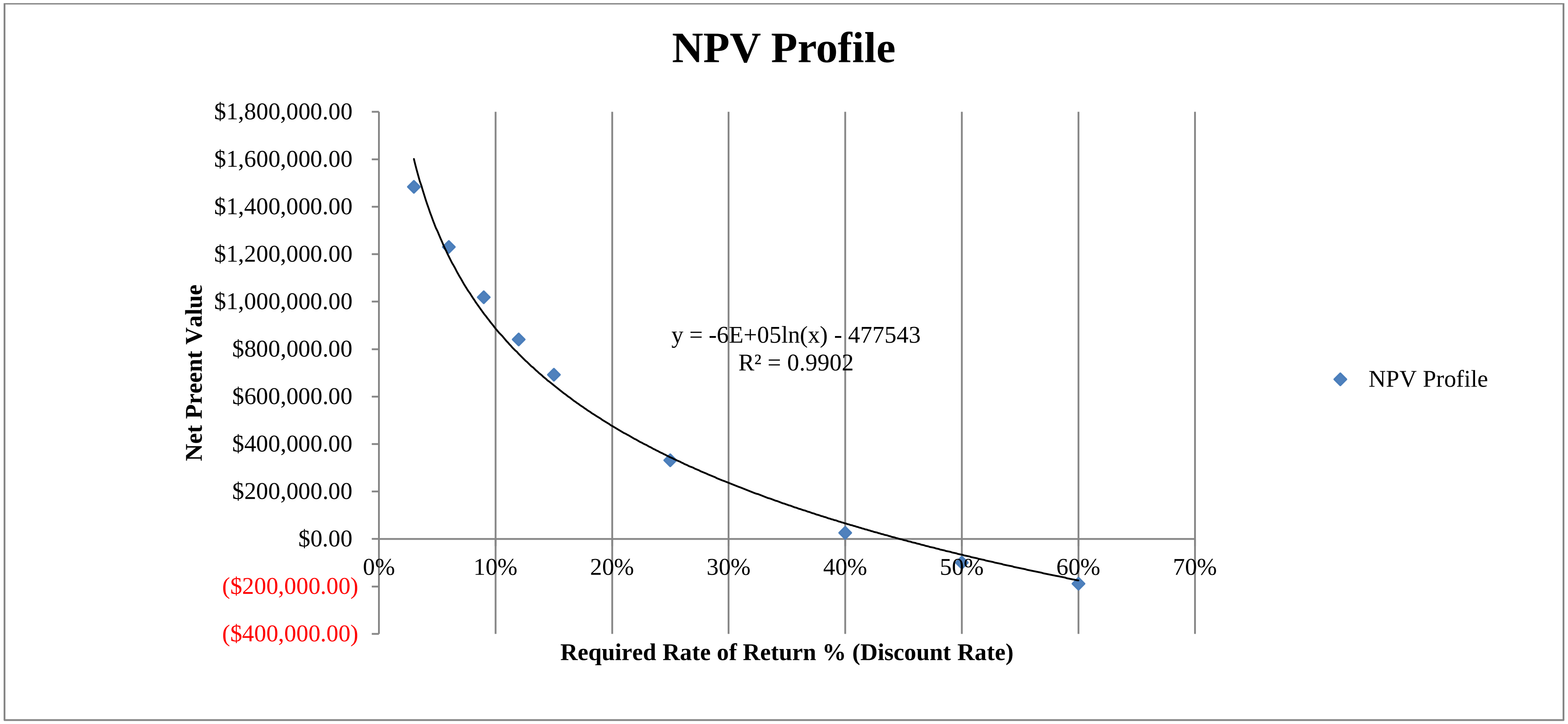
Sizing result

Type	CRNE 45-2	
Quantity * Motor	2 * 14.75 HP	
Flow	120	US gpm
H total	117	ft
Power P1	4.409	kW
Power P2	4.952	HP
Eff pump	71.6	%
Eff motor	86.2	%
Eff pump+mtr	59.9	% =Eta pump * Eta motor
Eff total	59.9	%
Flow total	11055980	gal/year
Consumption	13681	kWh/Year
Price	On request	
Total costs	On request	/10Years
Life cycle cost		/10Years

APPENDIX M

	MARCS	0.1429	0.2449	0.1749	0.1249	0.0893	0.0892	0.0893	0.0446
	Cash Flow	2017	2018	2019	2020	2021	2022	2023	2024
	Year 0	EOY 1	EOY 2	EOY 3	EOY 4	EOY 5	EOY 6	EOY 7	EOY 8
I. Investment Cash Flows									
Net Capital Investment	$ (664,600.00)								
Installation	$ (284,000.00)								
Freight	$ (6,500.00)								
CCCT FXV model	$ (374,100.00)								
Equipment	$ (344,000.00)								
Sales Tax	$ (30,100.00)								
II. Operating Cash Flows									
Revenues		$ 349,165.53	$ 361,888.63	$ 375,106.12	$ 388,838.50	$ 403,107.18	$ 417,934.51	$ 433,343.81	$ 449,359.46
Gallons of Water and Sewage Saved		17885000	17885000	17885000	17885000	17885000	17885000	17885000	17885000
Water Rate per 1000 gal.		$5.95	$6.25	$6.56	$6.90	$7.25	$7.61	$8.00	$8.40
Sewage Rate per 1000 gal.		$13.58	$13.99	$14.41	$14.85	$15.29	$15.76	$16.23	$16.72
Expenses		$ (13,915.01)	$ (14,311.40)	$ (14,723.16)	$ (15,150.93)	$ (15,595.38)	$ (16,057.19)	$ (16,537.11)	$ (17,035.88)
Cost of Water and Sewage from CCCT		$ (5,068.54)	$ (5,289.53)	$ (5,520.63)	$ (5,762.33)	$ (6,015.12)	$ (6,279.53)	$ (6,556.11)	$ (6,845.45)
Gallons of Water Used		558887	558887	558887	558887	558887	558887	558887	558887
Water rate per 1000 gal.		$5.95	$6.25	$6.56	$6.90	$7.25	$7.61	$8.00	$8.40
Sewage rate per 1000 gal.		$13.58	$13.99	$14.41	$14.85	$15.29	$15.76	$16.23	$16.72
Cost of Energy from CCCT		$ (5,846.47)	$ (6,021.87)	$ (6,202.52)	$ (6,388.60)	$ (6,580.26)	$ (6,777.66)	$ (6,980.99)	$ (7,190.42)
Maitainance		$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)
EBITDA		$ 335,250.52	$ 347,577.24	$ 360,382.96	$ 373,687.57	$ 387,511.81	$ 401,877.31	$ 416,806.70	$ 432,323.58
Depreciation		$ 94,971.34	$ 162,760.54	$ 116,238.54	$ 83,008.54	$ 59,348.78	$ 59,282.32	$ 59,348.78	$ 29,641.16
EBIT		$ 240,279.18	$ 184,816.70	$ 244,144.42	$ 290,679.03	$ 328,163.03	$ 342,594.99	$ 357,457.92	$ 402,682.42
Taxes (less)		$ (60,069.80)	$ (46,204.17)	$ (61,036.11)	$ (72,669.76)	$ (82,040.76)	$ (85,648.75)	$ (89,364.48)	$ (100,670.61)
NOPAT		$ 180,209.39	$ 138,612.52	$ 183,108.32	$ 218,009.28	$ 246,122.27	$ 256,946.24	$ 268,093.44	$ 302,011.82
Depreciation (more)		$ 94,971.34	$ 162,760.54	$ 116,238.54	$ 83,008.54	$ 59,348.78	$ 59,282.32	$ 59,348.78	$ 29,641.16
OCF		$ 275,180.73	$ 301,373.06	$ 299,346.86	$ 301,017.82	$ 305,471.05	$ 316,228.56	$ 327,442.22	$ 331,652.98
III. Terminal Year Cash Flows									
Salvage Value									$0.00
IV. Net Cash Flow									
	$ (664,600.00)	$ 275,180.73	$ 301,373.06	$ 299,346.86	$ 301,017.82	$ 305,471.05	$ 316,228.56	$ 327,442.22	$ 331,652.98
V. Results									

ROI = EOY 3	32%	NPV =	$ 1,230,042.47	IRR =	42%	MIRR =	21%	Payback Period =	2.31 years


NPV Profile
$1,800,000.00
$1,600,000.00
$1,400,000.00
$1,200,000.00
$1,000,000.00
$800,000.00
$600,000.00
$400,000.00
$200,000.00
$0.00
($200,000.00)
($400,000.00)
Net Preent Value
0%
10%
20%
30%
40%
50%
60%
70%
y = -6E+05ln(x) - 477543
R² = 0.9902
NPV Profile
Required Rate of Return % (Discount Rate)

NPV	Rate
$1,484,422.85	3%
$1,230,042.47	6%
$1,018,577.90	9%
$841,332.30	12%
$691,605.29	15%
$332,291.75	25%
$25,499.83	40%
($98,953.09)	50%
($187,973.04)	60%

Neglecting changes in...

water rate	$	(85,603.92)
sewage rate	$	(114,924.63)
both	$	(200,528.55)

Table A-1. **3-, 5-, 7-, 10-, 15-, and 20-Year Property Half-Year Convention**

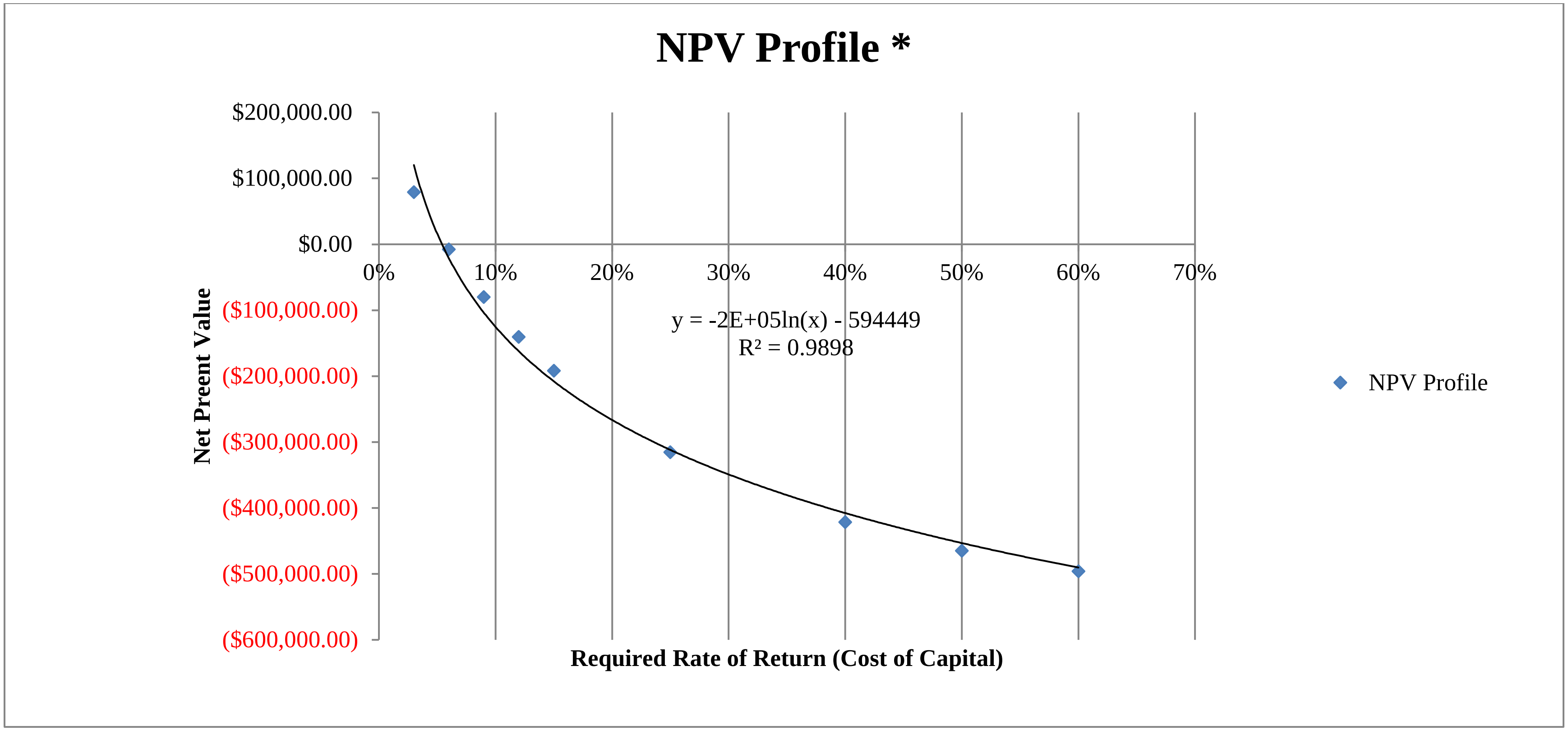
Year	Depreciation rate for recovery period					
	3-year	5-year	7-year	10-year	15-year	20-year
1	33.33%	20.00%	14.29%	10.00%	5.00%	3.750%
2	44.45	32.00	24.49	18.00	9.50	7.219
3	14.81	19.20	17.49	14.40	8.55	6.677
4	7.41	11.52	12.49	11.52	7.70	6.177
5		11.52	8.93	9.22	6.93	5.713
6		5.76	8.92	7.37	6.23	5.285
7			8.93	6.55	5.90	4.888
8			4.46	6.55	5.90	4.522
9				6.56	5.91	4.462
10				6.55	5.90	4.461
11				3.28	5.91	4.462
12					5.90	4.461
13					5.91	4.462
14					5.90	4.461
15					5.91	4.462
16					2.95	4.461
17						4.462
18						4.461
19						4.462
20						4.461
21						2.231

6% discount rate

6% discount/finance rate
3% sewage rate increase
5.8% reinvestment rate
5% water rate increase

APPENDIX N

			MARCS	0.1429	0.2449	0.1749	0.1249	0.0893	0.0892	0.0893	0.0446
			Cash Flow	2017	2018	2019	2020	2021	2022	2023	2024
			Year 0	EOY 1	EOY 2	EOY 3	EOY 4	EOY 5	EOY 6	EOY 7	EOY 8
I. Investment Cash Flows											
Net Capital Investment			$ (664,600.00)								
	Installation		$ (284,000.00)								
	Freight		$ (6,500.00)								
	CCCT FXV model		$ (374,100.00)								
		Equipment	$ (344,000.00)								
		Sales Tax	$ (30,100.00)								
II. Operating Cash Flows											
Revenues				$ 106,351.47	$ 111,735.52	$ 117,392.13	$ 123,335.10	$ 129,578.94	$ 136,138.88	$ 143,030.91	$ 150,271.85
	Gallons of Water and Sewage Saved			17885000	17885000	17885000	17885000	17885000	17885000	17885000	17885000
	Water Rate per 1000 gal.			$5.95	$6.25	$6.56	$6.90	$7.25	$7.61	$8.00	$8.40
	Sewage Rate per 1000 gal.			$0.00	$0.00	$0.00	$0.00	$0.00	$0.00	$0.00	$0.00
Expenses				$ (12,169.84)	$ (12,513.48)	$ (12,870.90)	$ (13,242.69)	$ (13,629.46)	$ (14,031.86)	$ (14,450.56)	$ (14,886.26)
	Cost of Water and Sewage from CCCT			$ (3,323.37)	$ (3,491.61)	$ (3,668.38)	$ (3,854.09)	$ (4,049.20)	$ (4,254.19)	$ (4,469.56)	$ (4,695.83)
	Gallons of Water Used			558887	558887	558887	558887	558887	558887	558887	558887
	Water rate per 1000 gal.			$5.95	$6.25	$6.56	$6.90	$7.25	$7.61	$8.00	$8.40
	Sewage rate per 1000 gal.			$0.00	$0.00	$0.00	$0.00	$0.00	$0.00	$0.00	$0.00
	Cost of Energy			$ (5,846.47)	$ (6,021.87)	$ (6,202.52)	$ (6,388.60)	$ (6,580.26)	$ (6,777.66)	$ (6,980.99)	$ (7,190.42)
	Maitainance			$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)	$ (3,000.00)
	Repair										
EBITDA				$ 94,181.63	$ 99,222.04	$ 104,521.23	$ 110,092.42	$ 115,949.49	$ 122,107.02	$ 128,580.35	$ 135,385.59
Depreciation				$ 94,971.34	$ 162,760.54	$ 116,238.54	$ 83,008.54	$ 59,348.78	$ 59,282.32	$ 59,348.78	$ 29,641.16
EBIT				$ (789.71)	$ (63,538.50)	$ (11,717.31)	$ 27,083.88	$ 56,600.71	$ 62,824.70	$ 69,231.57	$ 105,744.43
Taxes (less)				$ 197.43	$ 15,884.63	$ 2,929.33	$ (6,770.97)	$ (14,150.18)	$ (15,706.18)	$ (17,307.89)	$ (26,436.11)
NOPAT				$ (592.28)	$ (47,653.88)	$ (8,787.98)	$ 20,312.91	$ 42,450.53	$ 47,118.53	$ 51,923.68	$ 79,308.32
Depreciation (more)				$ 94,971.34	$ 162,760.54	$ 116,238.54	$ 83,008.54	$ 59,348.78	$ 59,282.32	$ 59,348.78	$ 29,641.16
OCF				$ 94,379.06	$ 115,106.66	$ 107,450.56	$ 103,321.45	$ 101,799.31	$ 106,400.85	$ 111,272.46	$ 108,949.48
III. Terminal Year Cash Flows											
Salvage Value											$ -
Total Disposal Cash Flow											$ -
IV. Net Cash Flow											
			$ (664,600.00)	$ 94,379.06	$ 115,106.66	$ 107,450.56	$ 103,321.45	$ 101,799.31	$ 106,400.85	$ 111,272.46	$ 108,949.48
V. Results											
			NPV =	$ (7,623.25)	**IRR =**	5.7%	**MIRR =**	5.7%	**Payback Period =**	6.35	years



NPV Profile *
$200,000.00
$100,000.00
$0.00
($100,000.00)
($200,000.00)
($300,000.00)
($400,000.00)
($500,000.00)
($600,000.00)
0%
10%
20%
30%
40%
50%
60%
70%
Net Preent Value
Required Rate of Return (Cost of Capital)
y = -2E+05ln(x) - 594449
R² = 0.9898
NPV Profile

Sensitivity Analysis

- Will reduce cost of water by 25%,50%,75%, and 100%
- Will reduce cost of sewage by 25%,50%,75%, and 100%
- Will increase the electricity rates by 100%
- Complete failure of cooling tower

							NPV	Rate
Water Reduction		NPV	IRR	PP				
	25%	$ 1,088,682.12	38%	2.47	years		$79,063.90	3%
	50%	$ 947,321.77	35%	2.67	years		($7,623.25)	6%
	75%	$ 805,961.43	31%	2.9	years		($79,809.84)	9%
	100%	$ 664,601.08	27%	3.18	years		($140,420.30)	12%
Sewer Reduction		NPV	IRR	PP			($191,709.73)	15%
	25%	$ 920,626.04	34%	2.74	years		($315,229.18)	25%
	50%	$ 611,209.61	25%	3.39	years		($421,407.32)	40%
	75%	$ 301,793.18	16%	4.41	years		($464,765.42)	50%
	100%	$ (7,626.25)	6%	6.35	years		($495,918.68)	60%
Energy rate increase								
		NPV	IRR					
2x		$1,200,048.19	42%					
3x		$1,170,053.92	40%					
Complete Breakdown								
		NPV	IRR					
EOY 8		$ 917,308.77	39%					
EOY 5		$ 857,571.63	35%					
EOY 3		$ 811,534.24	29%					
EOY 2		$ 786,423.74	26%					
EOY 1		$ 759,806.62	22%					
EOY 1&2		$ 316,187.89	12%					
EOY 1,2,3		$ (102,320.34)	4%					

APPENDIX O

Meter Code	Size	1-Jan-16	1-Jul-15	1-Jan-15	2-Jul-14	########	16-Dec-13	1-Jul-13	1-Jan-13	1-Jul-12	14-Jan-12	1-Jul-11	1-Jan-11	1-Jul-10		15-Jan-10		1-Jan-09
Wastewater Fixed Service Charges															**Wastewater Fixed Service Charges**			
1	5/8"	22.74	21.60	21.60	20.87	20.87	20.16	20.16	20.16	19.57	19.57	19.00	19.00	17.76	**SCWA pass-through rates** began July 1, 2010 **No change to wastewater rates**	17.76	**4 Tiered rate** structure began Jan 15, 2010	16.60
2	1"	54.75	48.10	48.10	46.47	46.47	44.90	44.90	44.90	43.59	43.59	42.32	42.32	39.55		39.55		36.96
3	1.5"	108.11	102.63	102.63	99.16	99.16	95.81	95.81	95.81	93.02	93.02	90.31	90.31	84.40		84.40		78.88
4	2"	172.15	178.80	178.80	172.75	172.75	166.91	166.91	166.91	162.05	162.05	157.33	157.33	147.04		147.04		137.42
5	3"	321.56	396.66	396.66	383.25	383.25	370.29	370.29	370.29	359.5	359.50	349.03	349.03	326.20		326.20		304.86
6	4"	535.00	701.61	701.61	677.88	677.88	654.96	654.96	654.96	635.88	635.88	617.36	617.36	576.97		576.97		539.22
7	6"	1068.60	1572.94	1572.94	1519.75	1519.75	1468.36	1468.36	1468.36	1425.59	1425.59	1384.07	1384.07	1293.52		1,293.52		1,208.90
Wastewater Usage Charges															**Wastewater Usage Charges**			
Usage Charge per 1,000 Gallons		13.08	13.24	13.24	12.79	12.79	12.36	12.36	12.36	12.00	12.00	11.65	11.65	10.89	**SCWA pass-through rates** began July 1, 2010 **No change to wastewater rates**	10.89	**4 Tiered rate** structure began Jan 15, 2010	
Water Fixed Service Charges															**Water Fixed Service Charges**			
1	5/8"	10.78	12.52	12.52	11.92	11.92	11.35	11.35	11.35	9.85	9.85	8.35	8.35	7.73	**SCWA pass-through rates** began July 1, 2010 **No change to fixed water rates**	7.73	**4 Tiered rate** structure began Jan 15, 2010	
2	1"	24.18	22.80	22.80	21.71	21.71	20.68	20.68	20.68	17.95	17.95	15.21	15.21	14.08		14.08		13.04
3	1.5"	46.51	43.58	43.58	41.50	41.50	39.52	39.52	39.52	34.30	34.30	29.07	29.07	26.92		26.92		24.93
4	2"	73.31	73.23	73.23	69.74	69.74	66.42	66.42	66.42	57.65	57.65	48.86	48.86	45.24		45.24		41.89
5	3"	135.83	171.26	171.26	163.10	163.10	155.33	155.33	155.33	134.84	134.84	114.27	114.27	105.81		105.81		97.97
6	4"	225.16	291.34	291.34	277.47	277.47	264.26	264.26	264.26	229.39	229.39	194.40	194.40	180		180		166.67
7	6"	448.46	637.82	637.82	607.45	607.45	578.52	578.52	578.52	502.18	502.18	425.58	425.58	394.06		394.06		364.87
Water Usage Charges															**Water Usage Charges**			
Usage Charge per 1,000 Gallons		5.25	4.95	4.86	4.86	4.79	4.79	4.79	4.70	4.65	4.55	4.50	4.41	4.09	**SCWA pass-through rates** began July 1, 2010	3.98	**Tier 1**: Up to sewer cap	3.83
		6.14	5.7	5.59	5.59	5.51	5.51	5.51	5.41	5.36	5.24	5.19	5.08	4.71		4.58	**Tier 2**: Water use above sewer cap up to 8,000k	4.77
		N/A	7.11	6.98	6.98	6.88	6.88	6.88	6.75	6.68	6.53	6.47	6.34	5.89		5.73	**Tier 3**: Water use 8,001k to 30,000k above sewer cap	7.16
		N/A	10.68	10.48	10.48	10.33	10.33	10.33	10.14	10.04	9.81	9.71	9.51	8.83		8.59	**Tier 4**: Water use over 30,000k above sewer cap	
Z = Y, Multi-Unit Residential and Commercial Water Rates															**Z = Y, Multi-Unit Residential and Commercial Water Rates**			
Z = Y		5.25	4.95	4.86	4.86	4.79	4.79	4.79	4.70	4.65	4.55	4.5	4.41	4.09	**SCWA pass-through rates** began July 1, 2010	3.98	All use billed at Tier 1 rate	3.83
Multi-Unit Res		5.59	5.46	5.36	5.36	5.28	5.28	5.28	5.18	5.13	5.01	4.96	4.86	4.51		4.39	All use billed at commodity rate	4.08
Commercial / Industrial		5.59	5.46	5.36	5.36	5.28	5.28	5.28	5.18	5.13	5.01	4.96	4.86	4.51		4.39	All use billed at commodity rate	4.08

APPENDIX P

California Polytechnic State University	06/01/2016
BioResource and Agricultural Engineering Department	DeGiorgio, Nicolo
ASM Senior Project Contract	007551319 ASM

Project Title

Kettle Cooling and Sustainability Project

How Project Meets Requirements for the ASM Major

ASM Project Requirements - The ASM senior project must include a problem solving experience that incorporates the application of technology and the organizational skills of business and management, and quantitative, analytical problem solving.

Application of agricultural technology	The project will involve the study of current cooking and cooling systems for food process.
Application of business and/or management skills	Using engineering economics, selected technologies can then be analyzed for their return on investments by working with utility companies to discuss rebate potential. Additionally, the interaction with the customer puts my soft-skills to the test in a professional environment conducting a sale.
Quantitative, analytical problem solving	The technology involved will have parameters that must be designed for to meet financial, energy-water consumptive, and production parameters.

Capstone Project Experience - The ASM senior project must incorporate knowledge and skills acquired in earlier coursework (Major, Support and/or GE courses).

ASM Approach - *"Senior projects for students in the Agricultural Systems Management major must include a problem solving experience that incorporates the* ***application of technology*** *and the application of business or management skills. Agricultural systems management involves the application of quantitative, analytical processes for developing solutions to technological, business or management problems associated with* ***agricultural production, processing, or the distribution*** *of agricultural products and support services to agricultural or related industries. A systems approach, interdisciplinary experience and agricultural training in* ***specialized areas*** *are common features of this type of problem solving. "*

incorporates knowledge/ skills from these key courses	AGB 310 Agribusiness Credit and Finance (time-value of money, discount rate, depreciation, capital budget analysis, pro forma income statements) BRAE 203 Agricultural Systems Analysis (engineering economics, flow network diagrams, project cost-estimation) BRAE 301 Hydraulic and Mechanical Power Systems (selection, application, and use of hydraulic components and mechanical power transmission equipment) BRAE 324 Principles of Agricultural Electrification (basic power and circuits calculations, power factor improvement, reading circuit diagrams, basic power distribution design) BRAE 342 Agricultural Materials (stress, strain, mat'l selection, etc.) BRAE 432 Agricultural Buildings (principles of building heat loss/rejection, food storage selection, environmental factors consideration) BRAE 532 Pumps and Wells (pump curve familiarity, well and sump pump design, pump station maintenance, efficiency improvements) Statistical Methods for Engineers General Chemistry for Engineers I (section on **Heat Transfer**) Physics I & II (section on **Thermodynamics**) Technical Writing Thermodynamics I