# Executive Summary

A pasteurization system was designed to meet processing requirements while staying within design constraints, including meeting 3-A standards for sanitation in the food industry and producing 335,000 gallons of milk per week. This system included a heat exchanger, piping, and a pump in order to heat milk up to a high temperature for a short amount of time before being cooled back down and sent to a cooler. A conceptual analysis was performed in a spreadsheet to determine the length of piping required for a counter flow double pipe heat exchanger using the LMTD method, with pipe diameter and water flow rate as chosen inputs that were varied to give the shortest length. This model drove the decision to switch to a shell and tube heat exchanger due to their higher efficiency and long total pipe length. The shell and tube heat exchanger calculations using the LMTD method were done in MATLAB and incorporated into a Monte Carlo optimization code. By creating a pump head vs cost curve with a line of 3-A sanitary pumps, the heat exchanger was optimized using cost as the objective function. As a result, the pasteurization system consists of a two stage, two tube pass and 1 shell pass heat exchanger with two 9.7-meter sections. Each section was 0.33 m in diameter and had 30 tubes (passed twice). The tubes 1 inch in diameter and packed in a hexagonal scheme. 7.2 kg/s of hot water was required for the second stage of the heat exchanger. Using the optimized parameters for the heat exchanger and pressure loss, a similar shell and tube heat exchanger and the cheapest pump that met the head requirements were used in an economic analysis, along with 3-A standard food grade piping for the holding tube section. Operating costs, maintenance costs of the CIP system, and initial costs of the equipment were taken into account over a 10-year period. As a result, the total cost of the project was predicted to be $173,405, with a payback period of 3.8 days.

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Nomenclature

|  |  |  |
| --- | --- | --- |
| Symbols | Description | Unit |
|  | Heat transfer rate | W |
|  | Mass flow rate | kg/s |
|  | Specific heat capacity | J/(kg·K) |
|  | Temperature | C° |
|  | Temperature difference | C° |
|  | Log mean temperature difference | C° |
|  | Fluid velocity | m/s |
|  | Fluid density | kg/m^3 |
|  | dimeter | m |
|  | Dynamics viscosity | Pa·s |
|  | Reynolds number | - |
|  | Nusselt number | - |
|  | Prandtl number | - |
|  | heat transfer coefficient | W/(m^2·K) |
|  | Thermal conductivity | W/(m·K) |
|  | Effectiveness of heat exchanger | - |
|  | Heat capacity rate ratio | - |
|  | Inlet temperature of the tube-side fluid | C° |
|  | Outlet temperature of the tube-side fluid | C° |
|  | Inlet temperature of the shell-side fluid | C° |
|  | Outlet temperature of the shell-side fluid | C° |
|  | Correction factor | - |
|  | Tube pitch | m |
|  | Pressure drop in the tube side (subscript: t)  and shell side (subscript: s) | Pa |
|  | Friction factor on the tube side (subscript: t)  and shell side (subscript: s) | - |
|  | Mass velocity on the tube side (subscript: t)  and shell side (subscript: s) | kg/m^2·s |
| g | Gravity | m/s^2 |

# **Problem Statement**

Pasteurization is an important process to increase the shelf life of milk. The pasteurization process involves heating milk up to high temperatures for a short amount of time to kill bacteria, then cooling the milk back down for storage. This process involves the controlled transport of milk at different temperatures as well as some form of heating the milk. Penn Dairy requires a pasteurization system design that will ensure that the milk is heated to the correct temperature for a specific amount of time before being chilled to the proper temperature, all while fitting within the space constraints of the facility expansion. The engineering specifications that meet this problem definition are explored in full in Table 1.

Table 1. Engineering specifications for the pasteurizer room including parameters, variables, constraints and benchmarks

|  |  |  |  |
| --- | --- | --- | --- |
| **Category** | **Name** | **Value** | **Unit** |
| **Parameters** | System Pressure | Atmospheric Pressure | Pa |
| Type of heat exchanger | Concentric Counter Flow or Shell and Tube | — |
| Thickness | 1 | mm |
| Hot water supply | ~ 200 | °F |
| **Variables** | Inner Tube Diameter | 0.01-0.025 | m |
| Tube Pitch | — | m |
| Number of Tubes | 1-331 | — |
| Inner Shell Diameter | — | m |
| Baffle Spacing | — | m |
| Water Flow Rate | 4.5-20 | kg/s |
| Which supplies to use | Whey by-products, 100,000 lbs/day softened water @ 105 °F, hot water supply, 70 °F well water, or 200 °F hot water supply | — |
| Heat exchanger lengths | — | m |
| **Constraints** | Milk Mass Flow | 4.491 | kg/s |
| Fins, pumping system, and a heat exchanger | — | Required Concepts |
| Budget | 225,000 | $ |
| Spatial Constraints | Per Figure 1. | — |
| Use of waste products | Whey by-products, 100,000 lbs/day softened water @ 105 °F, or 70 °F well water | — |
| Piping diameter | 75% of the system at 0.75 | in |
| Outdoor piping | Insulated or below the frost line | — |
| Water Type | Softened | — |
| Floor Space | No modifications | — |
| Input Milk Temperature | 40 | °F |
| Output Storage Milk Temperature | 40 | °F |
| Necessary Components | A balance tank, a feed pump, a regenerative preheating section, a centrifugal clarifier, a piping section from the dairy hot water heating system, a holding tube, a boosting pump, a regenerative cooler, cooling sections, related valves, and a process controller | — |
| **Benchmark** | Cost | — | — |
| High-Temperature Short Time (HTST) Treatment | Milk is heated to 162 – 163°F for 20 seconds before cooled | — |

A blueprint of a house

Description automatically generatedA blueprint of a house

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Figure 1. Planned expansion map for the dairy plant renovations, with zoomed view on the right showing the pasteurizer room.

# **Background Information**

A pasteurization system is a specialized thermal treatment setup, where precise temperature control and timing are critical to achieving food safety standards and product consistency. The design must account for thermodynamic principles, such as the heat transfer mechanisms involved in heating and cooling milk, and fluid dynamics to ensure efficient flow and piping through the system. Since the system deals with food products, it must comply with strict regulatory standards to guarantee safety and cleanliness. The key technology of the pasteurization system is the regenerative heating/cooling section. This regenerative section is often used to preheat the cold milk to a certain temperature using the heated milk before it enters the heating section in the water heating system. This system enhances energy efficiency and improves overall system performance by using the heat energy from cooling the pasteurized milk to preheat raw milk.

Adhering to established industry standards, such as 3-A Sanitary Standards [1] for dairy processing equipment, is essential to meet safety and cleanliness requirements. These standards determine the design of the milk pasteurization system to ensure that equipment is easy to clean, uses corrosion-resistant surfaces to prevent contamination, and are safe for food processing. Incorporating these guidelines affects the layout of components to facilitate cleaning and prevent bacterial growth.

In addition to the 3-A standards, compliance with the ASME Section VIII, Division 1 [2] is necessary to meet the mechanical integrity requirements of the system. UG-31 specifies the requirements for tubes and pipes when used as either tubes or shells in pressure vessels. Specifically, UG-27 covers the required thickness for tubes under internal pressure, while UG-28 focuses on the necessary thickness for tubes subjected to external pressure. These provisions ensure that the shell-and-tube heat exchanger can withstand both internal and external pressure during operation. UG-27 outlines the rules for determining the required wall thickness for tubes subjected to internal pressure. The thickness is determined based on the internal pressure, tube diameter, and material yield strength. Additionally, the code mentions that additional thickness should be added to the tube walls to account for wear if the tubes will undergo cleaning operations that may lead to corrosion. Similarly, UG-28 addresses tubes subjected to external pressure, such as atmospheric pressure or vacuum conditions. Tubes exposed to external pressure must be designed with sufficient wall thickness to resist collapse or deformation. If external pressure is a concern, increased thickness is necessary to prevent failure. Moreover, any tubes with threaded ends require additional thickness to compensate for the stress concentrations caused by the threads.

By integrating both the 3-A Sanitary Standards for hygiene and the ASME Section VIII, Division 1 codes for mechanical safety, the milk pasteurization system will be designed to be both sanitary and structurally stable, capable of handling the demands of high-temperature short-time (HTST) pasteurization while maintaining the safety.

# **Conceptual Analysis**

Since the problem consists of moving fluids that must be heated and cooled to specific temperatures, the key concepts of the design will be heat transfer by means of heat exchangers as well as required head produced by a pump to overcome major and minor losses throughout the pasteurization system. This system will begin with the regular-grade milk input and hot water produced from the boiler room hot water heating system, which will combine in the first heat exchanger as shown in Figure 2. Then, the hot milk will be held in a holding tank for a specified time before going through a second heat exchanger with the input milk to both cool the pasteurized milk and preheat the unpasteurized milk. Finally, the pasteurized milk that is now partially cooled is sent to a cooler that will lower the temperature to 40 °F and store it there. A pump is needed for the milk which will depend on the diameter and length of the piping as well as any fittings and other minor losses. It is assumed that the hot water is already pumped from the boiler room with a valve to control the flow rate as needed.

A diagram of a water supply system

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Figure 2. Pasteurization system diagram

Some of the other assumptions made include steady flow, constant specific heat, conduction in the axial direction is negligible, and no heat loss to surroundings.

For a detailed model, a counter-flow double-pipe heat exchanger was modeled using the LMTD method. The required milk temperatures and input hot water temperature are known, and the milk mass flow rate is calculated to be 4.491 kg/s in order to meet the 335,000 gallons per week processing requirement, assuming two 8-hour shifts per day, with the system running constantly for those 16 hours. The hot water flow rate and diameters of the heat exchanger were varied to determine the required length of the heat exchanger to meet the temperature requirements of the milk.

To do this, the heat transfer required for the milk was calculated using Equation 1,

where the flow rate, specific heat, and temperature difference are all obtained from the milk. According to an energy balance and assuming the pipes are insulated to neglect heat loss radially outward through the pipe, this heat transfer rate can be equated to the heat transfer rate of the hot water. Thus, Equation 1 can be used again with water parameter values to calculate the outlet temperature of the hot water.

To solve for the surface area, and therefore the length with diameters set as an input, Equation 2 was used,

where the U is the combined heat transfer rate. The LMTD can be determined from Equation 3,

where is the temperature difference between the water input and milk output and is the temperature difference between the water output and milk input. U is obtained by calculating the heat transfer coefficient for the water and the milk using Equations 4, 5, and 6, then combining them in parallel.

With these properties, the required surface area based on convective heat transfer correlations using Nusselt (Nu) and Prandtl (Pr) and Reynolds (Re) numbers were initially calculated. However, the required length of the tubing in our initial model for a counterflow double flow pipe was around 10 km assuming generously sized tubes. In order to minimize the heat exchanger space, the shell and tube configuration may be used to reduce the operating space by incorporating multiple tubes passing through one shell. Correction factors can be used for the LMTD specific to the shell-and-tube geometry to refine the final dimensions. The correction factor will depend on the specific arrangement of multi pass fluid paths. It will be estimated from the LMTD correction factor chart using temperature effectiveness, P, and heat capacity ratio, R, which can be calculated using Equations 7 and 8.

# **Detailed Modeling**

A shell-and-tube heat exchanger is a type of heat exchanger commonly used in industries for transferring heat between two fluids. It consists of a bundle of tubes enclosed within a cylindrical shell. One fluid flows through the tubes, while the other flows around the tubes within the shell. The heat transfer occurs through the tube walls, allowing the fluid to be heated or cooled. This system is used in the pasteurization process due to its effectiveness and ability to meet food safety standards. For the proposed system, a shell-and-tube heat exchanger is composed of two sections: the regenerative section and the water heating section as shown in Figure 3. In the regenerative section, incoming raw milk is preheated by absorbing thermal energy from the outgoing heated pasteurized milk. The water heating section uses a separate hot water system, operating at approximately 200°F, to raise the temperature of the preheated milk to the pasteurization temperature of 162–163°F.

Diagram of a diagram of a milk and milk production

Description automatically generated

Figure 3. Schematic of Shell and Tube Heat Exchanger

The proposed heat exchanger is a two-stage shell and tube design with one shell pass and two tube passes. The number of tubes and the diameter of the tubes are selected, as well as the number of baffles and the shell diameter since they are functions of the tube diameter and number of tubes, but the total length of the exchanger must be found. The governing equation for this design process is Eq. 9,

where F is a correction factor. The total length may therefore be calculated as a function of the total heat transfer, the correction factor, the log mean temperature difference, and the average diameter of the tubing. To begin, Q and the LMTD must be calculated based on known values. The inlet temperature of the hot and cold fluids in the first exchanger, the outlet temperature of the milk from the second exchanger, and the inlet temperature of the water in the second exchanger are known. The outlet temperature of both the hot and cold milk from the first section, and the outlet temperature of the water in the second section are unknown. Both hot outlet temperatures are calculated through the LMTD design process; however, the transition temperature of the milk between heat exchangers is unknown. Theoretically, this temperature could be found iteratively, however, this would make optimization infeasible due to changing fluid properties, so this was specified at 35.6 °C. This value makes property calculations easy and gives a good setpoint for optimal heat transfer between the hot and cold milk. The flow chart of the detailed model is shown in Figure 4. Each step of initialization and calculation was performed to determine the final temperatures of the milk and water in a shell-and-tube heat exchanger.

A screenshot of a diagram

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Figure 4. Conceptual Analysis Flow Chart

Once all properties are found for the average temperatures of each section, the hot outlet temperatures are calculated. The necessary heat transfer is calculated from the mass flow rate of cold inlet milk and outlet temperatures as in Eq. 1. Then the LMTD is calculated using Eq. 3 and a correction factor for shell and tube exchangers are calculated for each section as a function of inlet and outlet temperatures.

To find U, the heat transfer coefficients for each side of the exchanger, the thermal resistance between each side, and the fouling factors for each side must be found. The fouling factors cannot be determined experimentally but may be found in common tables of fouling factors [3]. The heat transfer coefficient for the tubing is a function of the fluid properties, Reynolds and Nusselt numbers, and the tube geometry. The outer heat transfer coefficient must be determined for flow across tubes. This flow is characterized by the crossflow area as in Eq. 11,

where is the pitch of the tubes. The crossflow area determines the maximum velocity of the flow which is assumed in the Reynold’s number calculation for the shell side flow. The outer heat transfer coefficient is then found as a function of the Reynold’s number, fluid properties, and tube geometry (with some correction coefficients to account for losses and assumptions). Finally, U may be calculated as an implicit function of the length of the heat exchanger. A final length is guessed and then the actual length is determined iteratively by calculating and recalculating U and L.

Shell and tube heat exchangers induce significant head loss in the fluids. Thus, pumping requirements are a significant aspect of the design process. The tube side pressure loss can be found using Eq. 12,

(12)

where Gt is a mass velocity for the tube side fluid. The shell side pressure loss can be found using Eq. 13,

(13)

where Gs is the mass velocity for the shell side and De is the shell hydraulic diameter.

Pump selection for the milk was based on the flow rate constraint calculated from the amount of milk required to be processed per week, which was about 70 GPM. A centrifugal pump will be used to pump the raw milk due to the relatively low viscosity of milk and necessity of accurate flow rates to ensure correct holding time and temperature. The TOP-FLO sanitary centrifugal pump line was selected to determine a maximum head - cost relationship because they are all 3-A approved and offer a wide variety of pumps for different requirements. Using values of the maximum head available at 70 GPM for the five different pumps, the head-cost relationship was determined with a linear fit shown in Figure 5. The trend line equation was used in the optimization process to factor in the cost of the pump based on the head loss of each system.

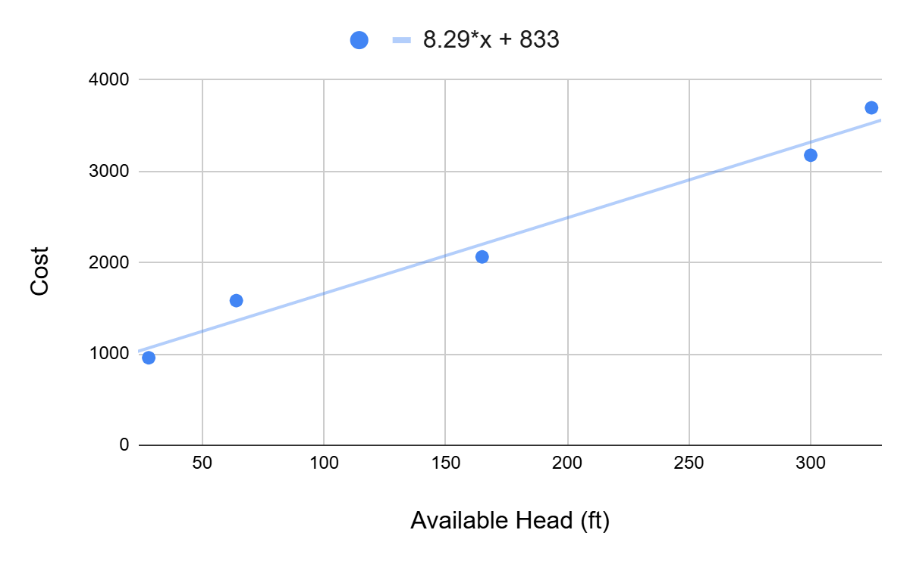


Figure 5. Available Head and Cost Relationship

The holding tube section of the pasteurization system must hold the milk at high temperature for 20 seconds. Since 75% of the piping in the system must have a diameter of 0.75 inches, and the milk flow rate required is fixed in order to meet processing demands, the holding tube section must be 2780 m long. Due to floor space constraints, this tubing must run in many parallel passes before going into the regenerative heating section of the shell and tube. This is not very feasible, however the 0.75 inch piping must be adhered to. Calculating the Reynold’s number in the holding pipes to be 3811 and using the Blasius approximation for the major loss friction coefficient, the major head loss from the holding tubes was found to be 17,360,884 m. Obviously, this is infeasible and will therefore be ignored when selecting a pump.

# **Design Selection**

The design selection process was conducted through MATLAB. A Monte Carlo optimization script was set up with tube diameter, hot water mass flow rate, and number of tubes as the optimization variables and cost as the optimization function. No constraints were applied directly to the system, but the variable bounds were based on reasonable values from physical heat exchangers and recommendations from Nang Sein Mya et al. (2019)[6]. A random value for each variable was chosen for each iteration of the optimizations (with one million iterations in total) and then the heat exchanger design with the lowest resulting cost was chosen.

The optimization process is depicted in Figure 6 as a flow chart. The calculation process in Figure 6 uses the same initialization shown in Figure 3. An optimization process was introduced to determine the optimal values for the following parameters: tube length, number of tubes, total volume, and total cost. These optimized values were obtained through iterative calculations using a for loop in MATLAB. The total volume of the system was calculated by summing the shell volume and tube volume. This total volume was then used to estimate the total cost of the system. The total cost comprised two main components: material cost and head cost. The head cost includes the cost of the tube-side both through and return of the system, and the cost of the shell side. The optimization chart presents the results of all the optimized parameters for the final design.

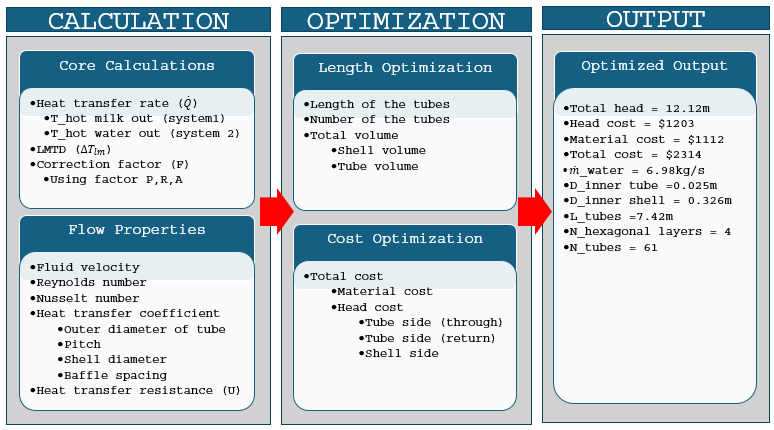


Figure 6. Optimization Flow Chart

The objective function was formulated based on material and pumping costs for the heat exchanger. With the input variables and knowledge of the heat transfer needed to bring the milk to the proper temperature for heat exchanger, the length of all the heat exchanger could be found. The length fully determines the geometry of the heat exchanger and so a material cost can be found based on the mass of the stainless steel and the price per unit of mass. Since the mass flow rates involved in the process are also known and the geometry is fully defined, pumping head can be found based on flow characteristics and frictional effects of the tubing. A price estimate for the pumping was based on a general search of pump price ranges which was then fitted to a linear model.

The optimization process settled on two stages, two tube pass and 1 shell pass heat exchanger with two 9.7-meter sections. Each section will contain will be 0.33 m in diameter and will have 30 tubes (passed twice). The tubes will be 2.5 cm or 1 inch in diameter and will be packed in a hexagonal scheme. 7.2 kg/s of hot feed water will be required for the second stage of the heat exchange. The total material cost is $1100, and the total pumping cost is $1210, for a total cost of $2310. The final heat exchanger dimensions and associated costs are shown in Table 2.

Table 2. Heat exchanger geometry, flow rates, and costs

|  |  |
| --- | --- |
| **Geometry** | |
| Number of tubes | 30 (two passes) |
| Inner Shell Diameter | 0.33 m |
| Inner Tube Diameter | 2.5 cm |
| Hot Water Flow Rate | 7.15 kg/s |
| Milk Flow Rate | 4.49 kg/s |
| Length | 9.66 m (per exchanger) |
| Thickness | 1 mm |
| Number of Baffles | 59 (per exchanger) |
| **Costs** | |
| Material Cost | $1105 |
| Pumping Cost | $1209 |
| Cost | $2314 |

After obtaining the heat exchanger specifications from the optimization program, a heat transfer distributor, HeatX, was called to try to obtain a quote for a heat exchanger with similar parameters because they had an option for 3-A standard shell and tube heat exchangers. The quote was supposed to be for a BEMH TEMA type shell and tube heat exchanger with 30 tubes, a shell diameter of 12 inches, tube diameter of 1 inch, and length of 20 feet, as shown in Figure 7. However, after initial communication, the distributor did not follow up with a quote. Instead, a similar spec heat exchanger was found online with an associated price already given. Two heat exchangers will be purchased, one for the milk regenerative section, and one for the water heating section due to the infeasibility in length if these heat exchangers were combined and split into two sections. The tube diameter and length chosen are said to be the ‘standard’, which matches well with the results from the optimization process [4].

A close-up of a spiral tube

Description automatically generated

Figure 7. SolidWorks drawing of proposed heat exchanger design with some critical dimensions noted.

Based on the head loss of the heat exchanger calculated after optimization, which was about 40 feet, a pump was selected that provided 40 feet of head at a flow rate of 70 GPM as stated previously. The same pump product line was used for this selection as was used when determining the head-cost curves for the optimization code. From this, the cheapest pump that meets the demand of the system is the TF-C114 pump run at 60 Hz with a 3.5-inch diameter impeller, as shown in Figure 8.

A graph with lines and numbers

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Figure 8. Pump curve for TF-C114 pump, showing the operating point for the system in red. [5]

# **Economic Analysis**

After optimizing the shell and tube heat exchanger design using material costs and a pump head vs. cost curve fit, an economic analysis was performed based on actual available heat exchangers and pumps from the optimized design parameters. The total cost of the pasteurization system will depend on the initial costs of the pump, heat exchanger, and piping, as well as the predicted maintenance cost of the system. The economic analysis will use the current prime interest rate, which is 7.75%. Typical heat exchangers have a lifetime of about 20 years, while pumps last about 10 years. A heat exchanger for sale from Cooper Industries, LLC with similar specs to one section of the optimized specs cost $23,500, however approximating the cost of upgrading to the 3-A standard to be double the price, the heat exchanger proposed will have an initial cost of $47,000. Since two must be purchased, this value is doubled. The pumping system costs were approximated from a study done on small-scale dairy processing facilities based on consulting key industry experts. The cost of the pasteurizer pump in the study was $5,000, however the pump required for the proposed system is only $1,583. Therefore, the ratio of the proposed system pump cost to the system in the study was calculated and applied to the other values from the study. As a result, the total pumping system consisting of the HTST pump, HTST sanitary valves, holding tube section, and CIP system totaled to an initial cost of $13,138.9 [7]. It was assumed that the system would be used until its end of life and therefore would have no salvage value. The maintenance cost of a pasteurization system mostly comes from the cost of cleaning using clean-in-place (CIP) systems. From a life cycle assessment on CIP systems in dairy plants using a conventional nitric acid and sodium hydroxide cleaning method with hot water disinfectant, it was estimated that CIP uses 264 gallons of water and 110 kWh of electricity per clean [8]. A new study showed that only one CIP is needed per day in a milk pasteurizer [9]. The average cost of water per 100 gallons in Pennsylvania for commercial use was found to be $1.8, and $0.2 per kWh of electricity in PA [10,11]. Assuming that the plant runs 365 days a year, this results in a yearly total maintenance cost of $9764.5, without taking into account the money lost from the downtime of the plant during CIP. A summary of these values is shown in Table 3.

Table 3. Pasteurization system cost analysis values.

|  |  |  |
| --- | --- | --- |
|  | Heat Exchanger | Pump System |
| Initial Cost | ($94,000) | ($13,138.9) |
| Maintenance Cost (CIP) | ($9764.5) | |
| Life | 20 | 10 |
| Interest Rate | 7.75% | |

Using the Net Present Worth method and assuming a 10-year lifespan, the total project cost was projected to be $173,404.79, which is less than the $225,000 budget approved for the pasteurization system.

In order to recover all initial investment plus the time value of money, the Discounted Payback period was calculated. This required the calculation of the annual net cash flow (NCF). Since the milk is being used to produce various dairy products, it will be assumed that the 335,000 gallons per week is split evenly between milk, butter, cheese, and yogurt. According to the United States Department of Agriculture, the price of these products is $21.43 (per hundred pounds), $2.73 (per pound), $1.95 (per pound), and $1.23 (per pound), respectively [12]. It will also be assumed that it takes around six gallons of milk to produce one gallon of butter, one gallon to make one pound of cheese, and one gallon to make one gallon of yogurt, according to various sources. Assuming that the operating profit margin for an estimated revenue of over $1 million is 18.1% [13], the net profit per week was calculated to be around $275,000. Accounting for the CIP maintenance costs of the pasteurization system along with the net profit, while assuming no other inflows or outflows, the annual net cash flow of the plant is around $14.3 million. This is clearly an absurd value for the size of the plant, and is a result of the massive processing requirement imposed. The discounted payback period was calculated to be 0.0454 years, or 16.4 days.

# **Design Summary and Conclusions**

The pasteurizer system was designed to efficiently heat and cool milk while maintaining 3-A sanitary standards and adhering to the budget constraints. The final design utilized a two-stage shell-and-tube heat exchanger due to its high thermal efficiency and compact configuration. This system contained two sections, each 9.7 meters in length with a shell diameter of 0.33 m, containing 30 hexagonally packed tubes per section, resulting in a compact and cost-efficient design. The system also contained a centrifugal pump so that the basic flow and head requirements will be achieved while maintaining cost efficiency. Critical design decisions including the selection of centrifugal pumps and specific heat-exchanger configurations were made using Monte Carlo optimization with the cost as the objective function. However, the limitations include the exclusion of the holding tube from head loss calculations due to the infeasibility of its design constraints, potential heat loss in the long holding tube, and reliance on outdated data for the economic analysis. Additionally, assumptions such as the feasibility of producing 335,000 gallons of milk per week and results such as the unrealistic net cash flow calculations highlight limitations in the analysis that could be improved with more realistic and updated inputs.

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