MEMORANDUM

To: Dr. Stephen Terry

From: David Delgado, Travis Kiser, Brenden Resnick, and Trevor Vook

Date: April 21, 2020

Subject: MAE 412 Group 5 Project Heat Exchanger Final Report

The purpose of this report is to describe the solution that Group 5 has found for the textile dying operation wastewater management system, and our recommendations for the modifications to the system. This document will contain a brief summary of the steps and methods used to find the solution, followed by detailed analysis of the problem. Sample calculations and a copy of the Excel spreadsheet are included.

The most important findings of the report are presented below in a table format.

Item	**Result (¼" Tubes)**	**Result (5/8" Tubes)**	**Result (¾" Tubes)**
Number of Tubes	232	204	195
HX Shell Diameter	10"	12"	14"
Temperature of Make-up Water leaving HX	106.724°F		
Initial Cost	$35,397	$36,841	$41,093
Quarterly Maintenance Cost	$2,320	$2,040	$1,950
Annual Natural Gas Cost Savings	$289,126		
Salvage Value (in end-of-life dollars)	$15,983	$16,635	$18,555
Net Annual Savings	$196,809	$197,929	$198,289
Simple Payback Period	0.18 years	0.19 years	0.21 years
Project Net Present Value	$3,112,641	$3,129,324	$3,131,543

The group recommends that the customer install the recovery heat exchanger using 5/8" diameter tubing; however, all options are very similar and all are acceptable choices for the customer to make based on their own circumstances. Group 5 is thankful for the opportunity to perform this work for you, and we hope that all results are to your satisfaction.

THERMAL SYSTEM DESIGN SOLUTION
FOR WASTEWATER MANAGEMENT SYSTEM

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April 21, 2020

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Executive Summary

The purpose of this document is to outline an optimal solution for the reduction of the temperature of wastewater from a textile dyeing operation. The wastewater is pumped from an atmospheric pit at a temperature of 180°F to a sewer, also vented to atmosphere. City regulations require the wastewater to be cooled to a maximum temperature of 140°F or less. This report considers a heat exchanger with three possible alternatives for the size of the inner tubing: ¼ inch, 5/8 inch, and ¾ inch.

The team recommends the customer install a single-pass heat exchanger with 5/8” tubing. This heat exchanger will have 204 tubes made of 304 stainless steel, with an outer diameter of 5/8” (0.625”) and a wall thickness of 0.049”. This heat exchanger will be fourteen feet long, with one-foot entry and exit plenums on each side, leaving a total tube length of twelve feet. The shell will be constructed of 12-inch diameter insulated pipe.

It is estimated that this heat exchanger will have an overall cost of $36,841; the tubes will cost $14,761, the shell will cost $18,000, and tube installation will cost $4,080. The heat exchanger requires regular cleaning, estimated at a cost of $2,040 a quarter. This heat exchanger will save the plant approximately $197,929 every year for the planned 20 years of operation, and the project has a net present value of $3,129,324.

This heat exchanger was chosen over the heat exchanger with ½" tube size because it has a net present value that is approximately $17,000 higher, for only an increase in initial cost of about $1,500. It was chosen over the ¾" tube size because the ¾" tube heat exchanger has an initial cost that is approximately $4,000 higher, and the net present value is only $2,000 higher; additionally, due to the increased number of tubes found in the 5/8” tube heat exchanger, blockage or damage to a single tube will have a lesser effect on the overall cooling than it would with the ¾" tube heat exchanger.

The wastewater is cooled by plant make-up water; the heating imparted on the make-up water reduces the amount of steam needed to heat it to the desired temperature of 140°F by approximately 5996.8 lb-mass per hour. The reduced operations of the boiler by this amount lead to natural gas cost savings of $289,126 a year; however, the steam flows through a turbine before heating the make-up water, and the reduced rate of steam through the turbine results in reduced power generation. Purchasing enough electricity to make up for this lost production costs $83,036.

The existing pump will continue to work without modification in the new system, thanks to an existing control valve in the line to the sewer. The control valve will be adjusted to ensure that flowrate remains constant through the heat exchanger. Due to the flowrate being kept constant, there will be no changes to the required pump work, and no changes to the electricity cost of running the pump. The pump, however, is not able to completely empty the wastewater pit for cleaning and maintenance; the team recommends shaving the pump impeller, installing a variable-frequency drive, or setting up a temporary sump pump for cleaning purposes, whichever is the most viable choice for the customer.

1.0 Introduction

The team has been tasked with evaluating process performance for a wastewater management and heat recovery system for a textile dyeing operation. 300 GPM of wastewater is pumped from an open storage pit at an average temperature of 180°F to an atmospheric vented sewer. The wastewater is not currently cooled as it goes to the sewer, but the city is now requiring the wastewater to be cooled to no more than 140°F before it reaches the sewer. The existing pump is a Goulds JC 3X4-11 slurry pump operating at 1750 RPM with an 8" impeller.

It is proposed to install a heat exchanger bypass loop to cool the wastewater and recover heat for other plant processes by heating plant make-up water. The goal of this analysis is to evaluate the required heat exchanger specifications for the needed cooling, and the effect that such a heat exchanger would have on the overall system, including the pump and existing valves. The heat exchanger specifications will be analyzed for three given tubing sizes. The effect on the secondary make-up water heat exchanger, and cost reductions from those changes, will also be analyzed. The ability for the existing pump to empty the pit for cleaning purposes, and any NPSH issues that may arise from that, will also be evaluated.

2.0 Current Operations

2.1 Constraints

- Flow rate of wastewater: 300 GPM
- Inlet temperature of wastewater: 180°F
- Desired outlet temperature of wastewater: 140°F
- Make-up water recovery HX inlet temperature: 60°F
- Make-up water secondary HX outlet temperature: 140°F
- Maximum allowable length for recovery HX: 12 feet
- Maximum allowable tube-side velocity of recovery HX: 12 ft/s
- Velocity of fluid through shell: 3 ft/s
- Flowrate of steam from boiler: 125,000 lb/hr
- Properties of steam leaving boiler: 700°F, 400 psig
- Turbine isentropic efficiency: 65%
- Pressure of steam leaving turbine/entering secondary HX: 60 psig
- UA of secondary HX: 150,000 BTU/(hr °F)
- Tube material: 304 Stainless Steel
- Steam leaves secondary HX as saturated liquid condensate at shell pressure
- Tube wall thicknesses: 0.049 inch (½" and 5/8” tubing), 0.062” (¾" tubing)
- Cost of natural gas for boiler: $5/MMBTU
- Boiler Efficiency: 83%
- Pipe diameter (assumed inner diameter): 4 inches
- Goulds JC 3X4-11 slurry pump curve: See Figure A2 (page)
- Pump impeller diameter: 8 inches
- Pump speed: 1750 RPM
- Pit Surface Pressure: 0 psig
- Sewer Pressure: 0 psig
- Pipe Material: Commercial Steel
- Tubing Material: Drawn Metal
- # tubes for ½" Pipe: 232 (see Report 1)
- # tubes for 5/8” Pipe: 204 (see Report 1)
- # tubes for ¾" Pipe: 195 (see Report 1)
- Piping Length and Layout: See Figure A1 (page)

2.2 Assumptions

- Perfectly insulated system
- Pressure of make-up water: 100 psia
- Assume recovery HX is maximum allowable length (12ft)
- Evaluate most values (viscosity, c_p, Pr) at average temperature in heat exchanger
- Make-up water in shell, wastewater in tubes (tubes easier to clean)
- Fouling factor of wastewater: 0.0002 (m^2*K)/W = 4.0884 (s*ft^2*°F)/BTU
- Fouling factor of make-up water: 0.0001 (m^2*K)/W = 2.0442 (s*ft^2*°F)/BTU
- Model heat exchanger as a cross-flow HX with mixed shell fluid
- No friction losses in heat exchanger plenums
- Entire wastewater system at constant atmospheric pressure
- All pipe/tube entrances/exits sharp-edged
- All elbows standard threaded elbows
- HX bypass piping installed in center of 200ft pipe section
- Pump installed in center of 10ft pipe section

3.0 Objectives

The goal of this analysis is to find the following values and results:

- Number of recovery heat exchanger tubes required for each given tube size
- Overall diameter of recovery heat exchanger
- Temperature of make-up water leaving recovery heat exchanger
- Reduction in steam supplied to the secondary heat exchanger
- Effect of steam reduction on steam turbine, resulting fuel cost savings
- Required control valve K value for existing piping system to ensure 300GPM
- Required control valve K values for modified piping system for each given tube size
- Overall head loss of the system for each given tube size
- Change in required pump work for each given tube size
- Change in electricity cost for each given tube size
- Any possible NPSH issues that arise when attempting to pump out the wastewater storage pit for cleaning and maintenance
- Cost of recovery heat exchanger for each given tube size
- Simple payback period for each option
- Net present value of each option

4.0 Summary of Results

Item	Result (¼” Tubes)	Result (5/8” Tubes)	Result (¾" Tubes)
Number of Tubes	232	204	195
HX Shell Diameter	10”	12”	14”
Temperature of Make-up Water leaving HX	106.724°F		
Initial Cost	$35,397	$36,841	$41,093
Quarterly Maintenance Cost	$2,320	$2,040	$1,950
Steam Usage Reduction (lbm/hr)	5996.8		
Annual Natural Gas Cost Savings	$289,126		
Control Valve K value for existing system	46.469		
Control Valve K value for modified system	38.441	39.651	39.995
Overall head loss of system (ft)	65		
Pump Work Change (hp)	0		
Change in Electricity Cost	$0		
Salvage Value (in end-of-life dollars)	$15,983	$16,635	$18,555
Net Annual Savings	$196,809	$197,929	$198,289
Simple Payback Period	0.18 years	0.19 years	0.21 years
Project Net Present Value	$3,112,641	$3,129,324	$3,131,543

NPSH Considerations: The current pump operating conditions can only pump 7.45 feet of wastewater down from the pump’s location. This leaves 2.55 feet of wastewater above the pipe inlet that cannot be pumped, as well as the two feet of wastewater that are below the pipe inlet and cannot be pumped by any pump in the current piping system. In order to pump the remaining 2.55 feet above the inlet while maintaining a flowrate of 300 GPM, either the pump speed must be reduced (which can be done by installing a variable-frequency drive), or the impeller diameter must be reduced by shaving the impeller. To pump out the last two feet, either additional piping must be installed, or a temporary sump pump or other small pump must be installed when it is time for cleaning and maintenance of the pit.

5.0 Recommendation

The group recommends the installation of a single pass heat exchanger with 204 5/8” 304 stainless steel tubes, contained within a 12” diameter shell with internal baffles. The tubes have a wall thickness of 0.049 inches and the heat exchanger has a length of fourteen feet; one-foot plenums on either side result in a tube length of twelve feet. The 5/8” tube option has a net present value of $3.13 million, compared to the 1/2” option, valued at $3.11 million, when analyzed for a twenty-year operation lifetime. This higher net present value also comes with lower maintenance cost and only slightly higher initial cost (~$1,500). Since the NPV is most significantly impacted by the net yearly savings, and these savings are differentiated between the different tubing options only by cleaning costs, it is intuitive to pick the option with the lowest cleaning cost (i.e., 3/4’’ tubing). However, the 5/8’’ tubing option has the next lowest cleaning cost with an additional cost of only $360 per year. The 5/8” tube was chosen over the 3/4” due to having an initial cost approximately $4,000 lower.

Hence, accepting a slightly higher yearly cleaning cost via 5/8’’ tubing is recommended in order to significantly lower the upfront cost of the heat exchanger. Additionally, due to the increased number of tubes found in the 5/8” tube heat exchanger, blockage or damage to a single tube will have a lesser effect on the overall cooling than it would with the 3/4" tube heat exchanger.
The initial cost for our recommended project is $36,841 and by choosing to move forward with this project it will bring a net present benefit of $3,129,324 to the plant.

6.0 Appendices

6.1 Figures

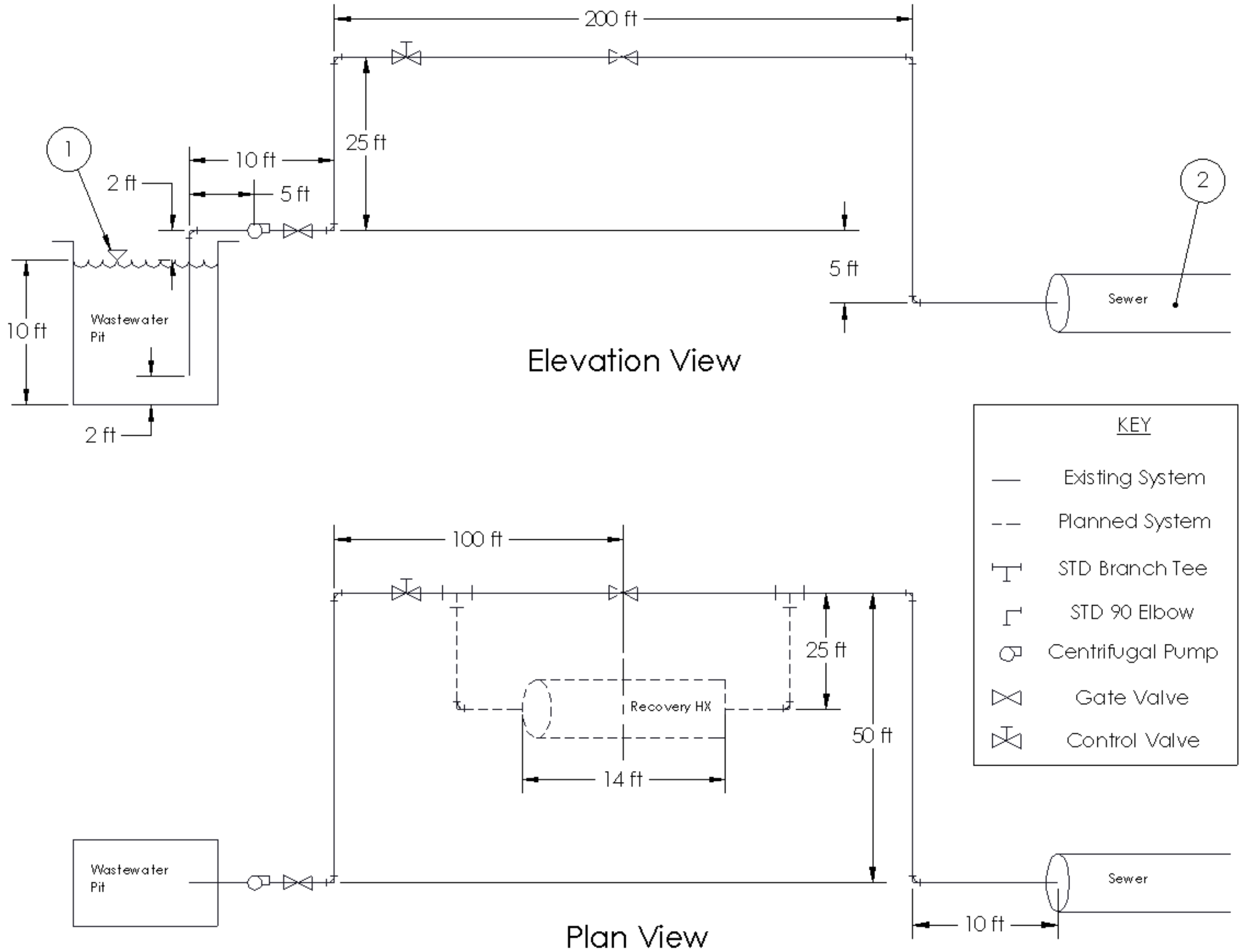


Figure A1. Existing and Planned Piping System Sketch

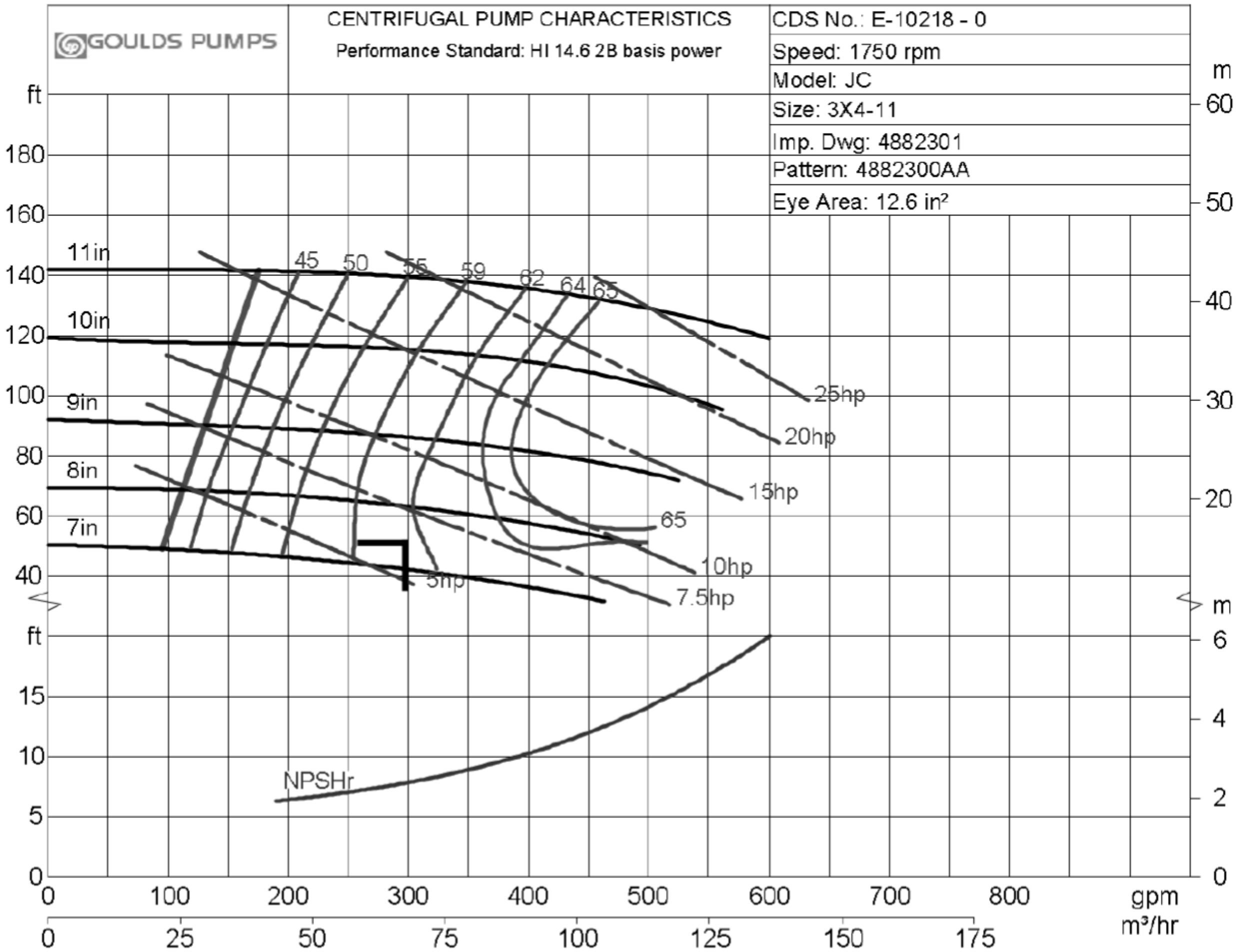


Figure A2. Goulds JC 3X4-11 Pump Curve

Secondary Hx calculations

Givens: steam flowrate: 125000 lbm/hr

Initial steam pressure: 414.7 psia

Initial steam temperature: 700 °F

Turbine Efficiency: 0.65

Turbine Exhaust Pressure: 74.7 psia

Secondary Hx UA: 150000 $\frac{Btu}{hr\,°F}$

Current MW Inlet Temp: 60 °F

Desired MW Outlet Temp: 140 °F

New MW Inlet: 106.724 °F

MW pressure (assume): 100 psia

MW Flowrate: 250 gpm, 2005.208 ft³/hr

Density of MW From City: 62.38358 lbm/ft³

MW Mass Flowrate: 125092.07 lbm/hr

Steam Turbine Calculations

- Enthalpy of steam entering turbine: = @h_pT(414.7,700)

$h_1 = 1362.018$ Btu/lb

- Entropy of steam entering turbine: = @s_pT(414.7,700)

in isentropic turbine would remain the same. $S_1 = S_2$

$S_1 = 1.636722 \frac{Btu}{lb\,°F}$

- Enthalpy of steam from perfect isentropic turbine. = @h_ps(74.7, 1.6367)

- Enthalpy of steam leaving

$h_{2s} = 1188.951$ Btu/lb

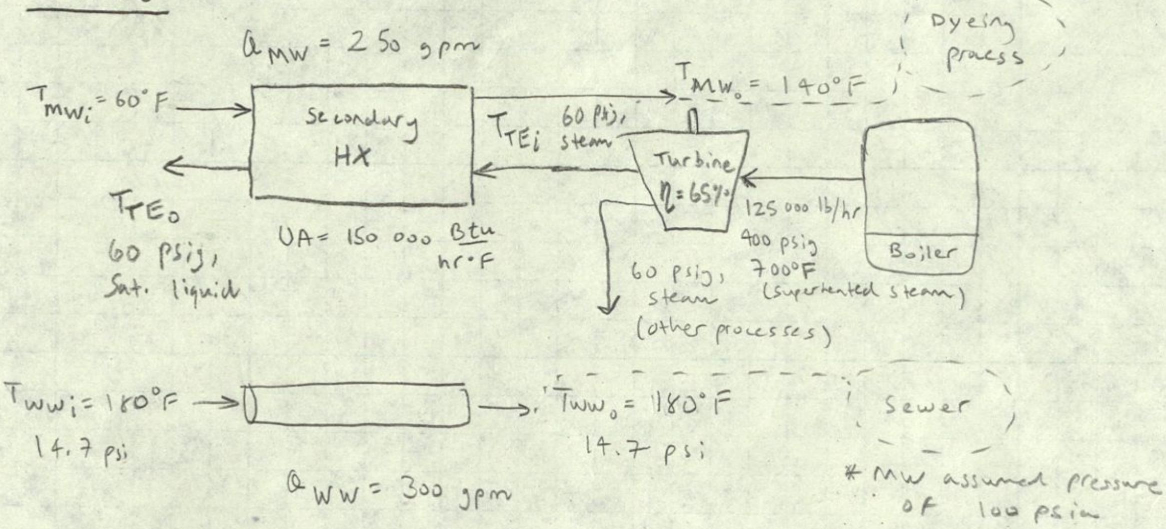
$\eta = 0.65$

$$\eta = \frac{h_1 - h_2}{h_1 - h_{1s}} \qquad .65 = \frac{1362.018 - h_2}{1362.018 - 1188.891}$$

$$h_2 = 1249.52 \frac{Btu}{lbm}$$

We will begin this section by calculating unknowns in the current dyeing operation system. Namely, we will find the steam supplied to the secondary HX in lb/hr. First, here is a schematic of the current system design:

Current



Where properties were evaluated using X Steam Tables (English units):

T_{TE_o} = 307.32°F $h_{TE_o} = 277.3 \frac{Btu}{lb}$ $c_{p,mw}$ @ $\bar{T}_{mw} = 100°F$ $= 0.997702 \frac{Btu}{lb \cdot °F}$

$h_2 = h_1 - \eta(h_1 - h_{2s})$

$\rho_{mw, 60°F} = 62.384 \ lb/ft^3$

(no need to evaluate at higher temp since no velocity limits exist)

Key: MW - make-up water
WW - wastewater
TE - turbine exhaust

Now we can solve for $\dot{m}_{TE}$, steam supplied to secondary HX in lb/hr. Recall Q, volumetric flow rate, can also be written as $\dot{V}$:

$$Q = \dot{V} \Rightarrow \rho = \frac{m}{V} = \frac{\dot{m}}{\dot{V}} = \frac{\dot{m}}{Q}$$

$$\therefore \dot{m}_{MW} = \rho_{MW, 60°F} \, Q_{MW} \qquad (1)$$

$$= \left(62.384 \frac{lb}{ft^3}\right)(250 \text{ gpm})\left(\frac{60 \text{ min}}{hr}\right)\left(\frac{0.133681 \ ft^3}{1 \text{ gallon}}\right)$$

$$\dot{m}_{MW} = 125\,093 \ lb/hr$$

Next, use total heat transfer rate relations to find $\dot{m}_{TE}$:

$$q = \dot{m}_{MW} c_{p,MW}(T_{MW_o} - T_{MW_i}) \qquad (2)$$

$$= (125093 \ lb/hr)\left(0.997702 \frac{Btu}{lb°F}\right)(140 - 60)°F$$

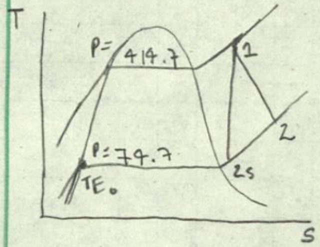
$$= 9.9844 \times 10^6 \ \frac{Btu}{hr}$$

Lastly, use a total heat transfer rate relation for fluid undergoing phase change:

$$q = \dot{m}_{TE}(h_2 - h_{TE_o}) \qquad (3)$$

where $\eta = 0.65$, $h_1 = 1362.02\ \frac{Btu}{lb}$, $h_{2s} = 1188.95\ \frac{Btu}{lb}$; hence $h_2 = 1249.52\ \frac{Btu}{lb}$

$\hookrightarrow T_{TE_i} = 435.37\ °F$

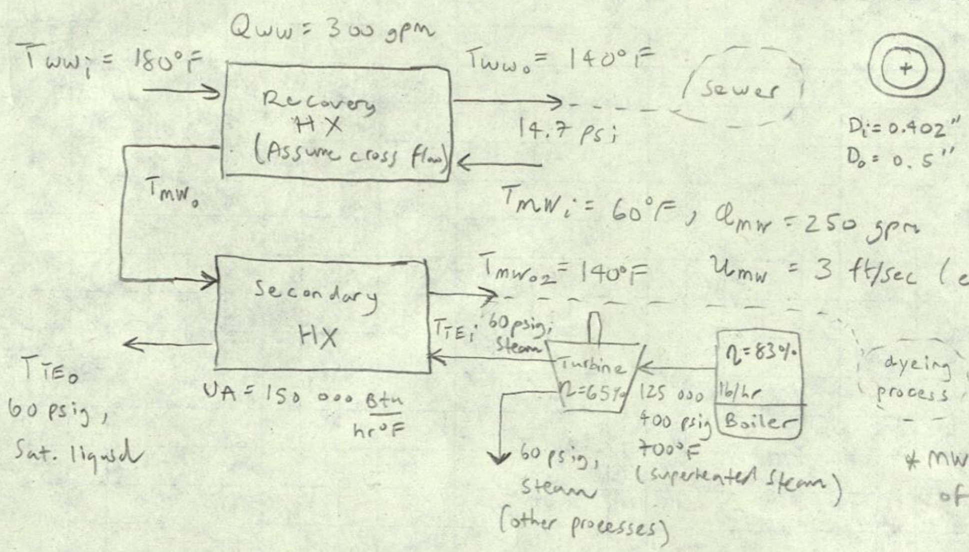


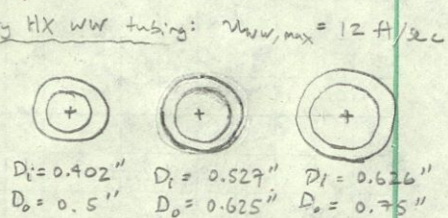
$$\therefore \dot{m}_{TE} = \frac{9.9844 \times 10^6\ \frac{Btu}{hr}}{(1249.52 - 277.3)\ \frac{Btu}{lb}}$$

$$\dot{m}_{TE} = 10\ 270\ \ lb/hr$$

Now, we will look at the desired system consisting of a recovery heat exchanger and the secondary heat exchanger as shown below:

Desired





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where properties are evaluated using X Steam Tables (English Units):

$\overline{T}_{mw} = \frac{100 + 60}{2} = 80°F$ (initial guess) $\qquad \overline{T}_{ww} = \frac{180 + 140}{2} = 160°F$

$\rho_{mw,60°F} = 62.384\ lb/ft^3 \qquad \rho_{ww,180°F} = 60.580\ lb/ft^3$ (velocity limit exists)

$c_{p,mw@\overline{T}_{mw}} = 0.99826\ \frac{Btu}{lb°F} \qquad c_{p,ww@\overline{T}_{ww}} = 1.0005\ \frac{Btu}{lb°F}$

$R''_{f,mw} = R''_{f,o} = 0.0001\ \frac{m^2 K}{W} \qquad R''_{f,ww} = R''_{f,i} = 0.0002\ \frac{m^2 K}{W} = 0.0011357\ \frac{hr\,ft^2\,°F}{Btu}$

$= 0.00056783\ \frac{hr\,ft^2\,°F}{Btu} \qquad \mu_{ww@\overline{T}_{ww}} = 0.962382\ \frac{lb}{ft\text{-}hr} \qquad k_{ww@\overline{T}_{ww}} = 0.381628\ \frac{Btu}{hr\text{-}ft\text{-}°F}$

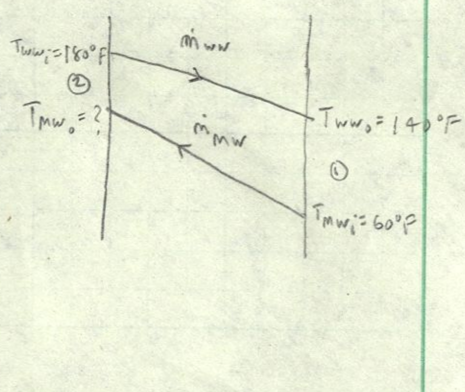
$Pr_{ww@\overline{T}_{ww}} = 2.5230$

First, use eqn (1) to calculate mass flow rates for make-up water and wastewater through the recovery HX:

$$\dot{m}_{ww} = \rho_{ww,180°F}\, Q_{ww}$$

$$= \left(60.580\ \frac{lb}{ft^3}\right)(300\ gpm)\left(\frac{60\ min}{hr}\right)\left(\frac{0.133681\ ft^3}{gallon}\right)$$

$$= 145\ 771\ \frac{lb}{hr}$$



Similarly, for $\dot{m}_{mw}$:

$$\dot{m}_{mw} = \rho_{mw,60°F}\, Q_{mw} = \left(62.384 \frac{lb}{ft^3}\right)\left(250\ gpm\right)\left(\frac{60\ min}{hr}\right)\left(\frac{0.133681\ ft^3}{gallon}\right)$$
$$= 125\,093 \frac{lb}{hr}$$

Use equation (3) to find total heat transfer rate:

$$q = \dot{m}_{ww}\, c_{p,ww@\bar{T}_{ww}} \left(T_{ww_i} - T_{ww_o}\right)$$
$$= \left(145\,771 \frac{lb}{hr}\right)\left(1.0005 \frac{Btu}{lb°F}\right)\left(180 - 140\right)°F$$
$$= 5\,833\,755 \frac{Btu}{hr}$$

Set the result of the previous equivalent to the heat transfer provided by the make-up water, and solve for T_{mw_o}:

$$q = \dot{m}_{mw}\, c_{p,mw@\bar{T}_{mw}} \left(T_{mw_o} - T_{mw_i}\right)$$
$$T_{mw_o} = T_{mw_i} + \frac{q}{\dot{m}_{mw}\, c_{p,mw@\bar{T}_{mw}}}$$
$$= 60°F + \frac{5\,833\,755\ Btu/hr}{\left(125\,093 \frac{lb}{hr}\right)\left(0.99826 \frac{Btu}{lb°F}\right)}$$
$$T_{mw_o} = 106.72°F$$

The accuracy of the previous answer can be improved by evaluating properties at the new $\bar{T}_{mw}$ and solving for T_{mw_o} in the same way as such:

$$\bar{T}_{mw} = \frac{106.72 + 60}{2} = 83.360°F \Rightarrow c_{p,mw@\bar{T}_{mw}} = 0.998104 \frac{Btu}{lb°F}$$

Hence, T_{mw_o} can be reevaluated:

$$T_{mw_o} = T_{mw_i} + \frac{q}{\dot{m}_{mw}\, c_{p,mw@\bar{T}_{mw}}}$$
$$= 60°F + \frac{5\,833\,755\ Btu/hr}{\left(125\,093 \frac{lb}{hr}\right)\left(0.998104 \frac{Btu}{lb°F}\right)}$$
$$T_{mw_o} = 106.72°F \qquad \text{(ANSWER 3)}$$

Now that all temperatures in the diagram are known, it is possible to begin an analysis of the recovery HX's overall heat transfer coefficient. We begin by calculating the internal heat transfer coefficient between the wastewater and the tubes (h_i). Note that these calculations are for tubes with $D_o = 0.5''$, but the same steps can be followed for other tube sizes:

$\boxed{h_i}$:

$$Q_{ww} = u_{ww} A_i N \qquad (4)$$

where $u_{ww,max} = 12\ ft/sec = 43\,200\ ft/hr$,

$A_i = \frac{\pi}{4} D_i^2 = \frac{\pi}{4}\left(\frac{0.402}{12} ft\right)^2 = 0.00088141\ ft^2$, and $Q_{ww} = 300\ gpm = 2406.26\ \frac{ft^3}{hr}$

We can make an initial guess for the optimal number of tubes given the desired design requirements:

$N = 100$ tubes $\Rightarrow$ $u_{ww} = \dfrac{2406.26\ ft^3/hr}{(0.00088141\ ft^2)(100)} = 27\ 300\ ft/hr$

Calculate the Reynold's number for the internal wastewater flow through tubes:

$$Re_D = \frac{(\rho_{ww,180°F})(u_{ww})(D_i)}{\mu_{ww@\bar{T}_{ww}}} = \frac{(60.580\ lb/ft^3)(27\ 300\ ft/hr)(\frac{0.402}{12} ft)}{0.962382\ lb/ft\text{-}hr} \quad (5)$$

$Re_D \approx 57\ 569$ ($Re_D > 10{,}000$ ∴ use Dittus-Boelter equation)

Use Dittus-Boelter Equation to calculate the local Nusselt number:

$$Nu_D = 0.023\, Re_D^{4/5}\, Pr^n_{ww@\bar{T}_{ww}} \text{ where } T_m > T_s \text{ (cooling)} \therefore n = 0.3 \quad (6)$$

$$= 0.023(57\ 569)^{4/5}(2.5230)^{0.3}$$

$Nu_D = 195.19$

Use the relationship below to solve for h_i:

$$Nu_D = \frac{h_i D_i}{k_{ww@\bar{T}_{ww}}} \quad (7)$$

$$h_i = \frac{Nu_D\, k_{ww@\bar{T}_{ww}}}{D_i} = \frac{(195.19)(0.381628\ \frac{Btu}{hr\text{-}ft\text{-}°F})}{(0.402/12\ ft)}$$

$h_i = 2223.6\ \frac{Btu}{hr\text{-}ft^2\text{-}°F}$

Next, calculate the external heat transfer coefficient between the make-up water and the tubes assuming one tube in isolation rather than a bank of tubes. Properties of the make-up water are shown below:

$\boxed{h_o}$: $\bar{T}_{mw} = 83.36°F$ $\quad k_{mw,\bar{T}_{mw}} = 0.35428\ \frac{Btu}{hr\,ft\,°F}$

$\rho_{mw,106.72°F} = 61.922\ \frac{lb}{ft^3}$ $\quad c_{p,mw@\bar{T}_{mw}} = 0.998104\ \frac{Btu}{lb\,°F}$

$\mu_{mw@\bar{T}_{mw}} = 1.990397\ \frac{lb}{ft\text{-}hr}$

$Pr_{mw@\bar{T}_{mw}} = 5.6075$

Find the Reynolds number for external flow over the circular cylinder:

$$Re_D = \frac{(\rho_{mw,106.72°F})(V_{mw})(D_o)}{\mu_{mw@\bar{T}_{mw}}} \quad (8)$$

$$= \frac{(61.922\ \frac{lb}{ft^3})(3\ ft/sec)(\frac{3600\ sec}{hr})(\frac{0.5}{12} ft)}{1.990397\ \frac{lb}{ft\text{-}hr}}$$

$Re_D = 14\ 000$ (turbulent since $Re_D > 2300$)

Calculate the average Nusset number using equation 7.52 in the textbook:

$$\overline{Nu_D} = \frac{\bar{h} D_o}{k_{mm,\overline{T_{mw}}}} = C\, Re_D^{\,m}\, Pr_{mw@T_{mw}}^{1/3} \quad \text{where } Pr > 0.7 \text{ is satisfied} \qquad (9)$$

and from table 7.2 for $Re_D = 14000$
$C = 0.193$, $m = 0.618$

$$\overline{Nu_D} = 0.193(14000)^{0.618}(5.6075)^{1/3}$$

$$= 125.15$$

Therefore, h_o can be calculated as follows:

$$h_o = \frac{k_{mm,\overline{T_{mw}}}}{D_o} Nu_D = \frac{0.35428 \frac{Btu}{hr\,ft\,°F}}{(0.5/12\ ft)}(125.15)$$

$$= 1064.2 \ \frac{Btu}{hr\,ft^2\,°F}$$

Now, the overall heat transfer coefficient for a single tube can be found using the following:

$$\boxed{U_o}: \quad \frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{R''_{f,o}}{A_o} + \frac{\ln(D_o/D_i)}{2\pi k_{304} L} + \frac{R''_{f,i}}{A_i} + \frac{1}{h_o A_o} \qquad (10)$$

$$\frac{1}{U_o} = \frac{D_o}{D_i h_i} + R''_{f,o} + \frac{D_o \ln(D_o/D_i)}{2 k_{304}} + \frac{D_o R''_{f,i}}{D_i} + \frac{1}{h_o}$$

where k_{304} is evaluated at 300K (~ 80°F) => $k_{304} = 14.9 \frac{W}{mK} = 8.6091 \frac{Btu}{hr\,ft\,°F}$

$$\frac{1}{U_o} = \frac{0.5}{(0.402)(2223.6 \frac{Btu}{hr\text{-}ft^2\text{-}°F})} + 0.00056783 \frac{hr\,ft^2\,°F}{Btu} + \frac{(0.5/12\ ft)\ln\left(\frac{0.5}{0.402}\right)}{2(8.6091 \frac{Btu}{hr\,ft\,°F})}$$
$$+ \frac{(0.5)(0.0011357 \frac{hr\,ft^2\,°F}{Btu})}{0.402} + \frac{1}{1064.2 \frac{Btu}{hr\,ft^2\,°F}}$$

$$= 0.0040073 \ \frac{hr\,ft^2\,°F}{Btu}$$

$$U_o = 249.54 \ \frac{Btu}{hr\,ft^2\,°F}$$

Utilize ε-NTU and LMTD methods to evaluate the UA of the recovery HX assuming cross flow HX with wastewater flow unmixed and make-up water flow mixed. These relations will find UA for the entire bank of tubes, and we will solve for N to see if the computed value converges with our initial guess thereof:

$$NTU = \frac{UA}{C_{min}} \qquad (11)$$

$$U_o A_o = C_{min}\, NTU$$

$$U_o(\pi D_o L N) = C_{min}\, NTU$$

$$N = \frac{C_{min}\, NTU}{U_o \pi D_o L} \quad \text{where } L = 12\ ft$$

Find C_{min} using the following relations:

$$C_{mw} = \dot{m}_{mw}\, c_{p,mw@\bar{T}_{mw}} = \left(125\ 093\ \frac{lb}{hr}\right)\left(0.998104\ \frac{Btu}{lb°F}\right) = 124\ 856\ \frac{Btu}{hr°F}$$

$$C_{ww} = \dot{m}_{ww}\, c_{p,ww@\bar{T}_{ww}} = \left(145\ 771\ \frac{lb}{hr}\right)\left(1.0005\ \frac{Btu}{lb°F}\right) = 145\ 844\ \frac{Btu}{hr°F}$$

$$\therefore C_{min} = C_{mw} = 124\ 856\ \frac{Btu}{hr°F} \qquad C_{max} = C_{ww} = 145\ 844\ \frac{Btu}{hr°F}$$

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$$C_r = \frac{C_{min}}{C_{max}} = 0.85609 \qquad (12)$$

Use ε-NTU relation in table 11.4 for cross-flow (single pass) HX with C_{min} (mixed), C_{max} (unmixed)

$$NTU = -\left(\frac{1}{C_r}\right)\ln\left[C_r \ln(1-\varepsilon)+1\right] \qquad (13)$$

where ε is found using the following relation:

$$\varepsilon = \frac{q}{q_{max}} = \frac{C_{ww}(T_{ww_i}-T_{ww_o})}{C_{min}(T_{ww_i}-T_{mw_i})} \qquad (14)$$

$$= \frac{\left(145\ 844\ \frac{Btu}{hr°F}\right)(180-140)°F}{\left(124\ 856\ \frac{Btu}{hr°F}\right)(180-60)°F} = 0.389366$$

Hence, NTU can be evaluated:

$$NTU = -\left(\frac{1}{0.85609}\right)\ln\left[0.85609\ln(1-0.389366)+1\right]$$

$$NTU = 0.64088$$

Finally, N can be computed using equation (11):

$$N = \frac{\left(124\ 856\ \frac{Btu}{hr°F}\right)(0.64088)}{\left(249.54\ \frac{Btu}{hr ft^2 °F}\right)(\pi)\left(\frac{0.5}{12}ft\right)(12\ ft)} = 204.14$$

We would round up the exact value of N to the next whole number, since a fraction of a tube can not be placed in the HX:

$$N = 205 \text{ tubes} \qquad (N_{initial\ guess} = 100 \text{ tubes})$$

Based on the discrepancy between the initial and computed values of N, the value guessed for N is not optimal. We can iterate the calculations above with a new N value until the initial and computed values converge, starting with N = 205. We will find that N = 232 is the optimal number of tubes required via the ε-NTU method. We can confirm this via the LMTD method:

$$N = 232 \Rightarrow \quad u_{ww} = \frac{2406.26\ ft^3/hr}{(0.00088141\ ft^2)(232)} = 11\ 767\ ft/hr$$

From equations (5), (6), (7):

$$\boxed{h_i}: \quad Re_D = \frac{(60.580\ lb/ft^3)(11\ 767\ ft/hr)\left(\frac{0.402}{12}ft\right)}{0.962382\ lb/ft\text{-}hr} = 24\ 814$$

$$Nu_D = 0.023\,(24814)^{4/5}\,(2.5230)^{0.3}$$

$$= 99.554$$

$$h_i = \frac{\left(0.381628 \frac{Btu}{hr\,ft\,°F}\right)}{\left(0.402/12\ ft\right)}(99.554) = 1134.1\ \frac{Btu}{hr\,ft^2\,°F}$$

The external heat transfer coefficient between the make-up water and the tubes, h_o, remains constant since the make-up water properties and 3 ft/sec fluid velocity requirement across the tubes are not affected by the number of tubes, N. Hence, $h_o = 1064.2\ \frac{Btu}{hr\,ft^2\,°F}$ and U_o for N=1 is found using equation (10):

$$\boxed{U_o}: \quad \frac{1}{U_o} = \frac{0.5}{(0.402)\left(1134.1\ \frac{Btu}{hr\,ft^2\,°F}\right)} + 0.00056783\ \frac{hr\,ft^2\,°F}{Btu} + \frac{\left(\frac{0.5}{12}ft\right)\ln\left(\frac{0.5}{0.402}\right)}{2\left(8.6091\ \frac{Btu}{hr\,ft\,°F}\right)}$$

$$+ \frac{(0.5)\left(0.0011357\ \frac{hr\,ft^2\,°F}{Btu}\right)}{0.402} + \frac{1}{1064.2\ \frac{Btu}{hr\,ft^2\,°F}}$$

$$= 0.0045447\ \frac{hr\,ft^2\,°F}{Btu}$$

$$U_o = 220.04\ \frac{Btu}{hr\,ft^2\,°F}$$

Now, utilize LMTD method to evaluate UA of the recovery HX assuming cross flow HX with wastewater flow unmixed and make-up water flow mixed. Find UA for the entire bank of tubes:

$$q = F\,U_oA_o\,\Delta T_{lm} \qquad (15)$$

$$= F\,U_o(\pi D_o L N)\,\Delta T_{lm}$$

$$N = \frac{C_{ww}(T_{ww_i} - T_{ww_o})}{F\,U_o\,\pi D_o L\,\Delta T_{lm}} \qquad \text{where } L = 12\ ft$$

Find the log mean temperature difference, ΔT_{lm}, in the recovery HX:

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1/\Delta T_2)} \qquad (16)$$

where $\Delta T_1 = T_{ww_o} - T_{mw_i} = 80°F$

$\Delta T_2 = T_{ww_i} - T_{mw_o} = 73.28°F$

$$= \frac{(80 - 73.28)°F}{\ln\left(80/73.28\right)}$$

$$\Delta T_{lm} = 76.591°F$$

Next find the correction factor, F, using figure 11S.4 in the supplemental textbook material:

$$R = \frac{T_i - T_o}{t_o - t_i} \qquad (17)$$

where "t" assigned to tube-side fluid (WW) temp

$$R = \frac{60-106.72}{140-180} = 1.168$$

$$P = \frac{t_o - t_i}{T_i - t_i} \qquad (18)$$

$$P = \frac{140-180}{60-180} = 0.33333$$

$\therefore F = 0.95$ (interpolated from figure 11 5.4)

Hence, N can be calculated from equation (15):

$$N = \frac{\left(145\ 844 \frac{Btu}{hr\,°F}\right)(180-140)°F}{(0.95)\left(220.04 \frac{Btu}{hr\,ft^2\,°F}\right)(\pi)\left(\frac{0.5}{12} ft\right)(12\ ft)(76.591°F)}$$

$N = 231.97 \Rightarrow N = 232$ tubes (for $\frac{1}{2}''$ tube) (ANSWER 1)

After rounding up the exact value of the calculated number of tubes, we find that N = 232. This confirms that 232 is the optimal number of tubes required since it matches the initial guess. Hence, the ε-NTU and LMTD methods give the same result in this case. An important thing to note is that there is a simpler equivalent form of the solution to the number of tubes for both methods, which facilitate Excel calculations. For ε-NTU method, we begin with equation (11):

$$(U_o A_o)_{overall} = C_{min}\, NTU$$

$$(U_o A_o)_{single\ tube} = U_o \pi D_o L$$

The solution for N was originally written as follows:

$$N = \frac{C_{min}\, NTU}{U_o \pi D_o L}$$

We may rewrit it as:

$$N = \frac{(U_o A_o)_{overall}}{(U_o A_o)_{single\ tube}} \qquad (19)$$

Similarly, for LMTD method we begin with equation (15):

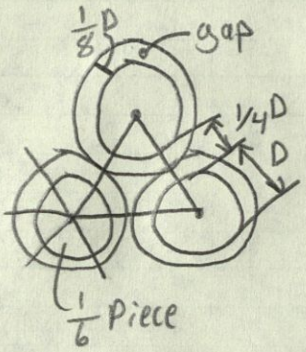
$$(U_o A_o)_{overall} = \frac{q}{F\, \Delta T_{lm}}$$

$$(U_o A_o)_{single\ tube} = U_o \pi D_o L \qquad \text{where } N = \frac{q}{F U_o \pi D_o L\, \Delta T_{lm}}$$

$$\therefore N = \frac{(U_o A_o)_{overall}}{(U_o A_o)_{single\ tube}}$$

Tube Packing and Shell Size Calculations

- We choose the most efficient, equilateral, packing of our tubes.
- Tubes must have $\frac{1}{4}D$ space between them.



- The triangle contains 3, 1/6 pieces.
- Each triangle contains $\frac{1}{2}$ of a pipe.
- Each triangle side is $\frac{5}{4}D$ long.

Area calculations for $\frac{1}{2}$" tube:

Side length: $L = \frac{5}{4}\left(\frac{1}{2}''\right) = \frac{5}{8}'' = 0.625''$

Triangle height: $h = 0.625 \sin(60°) = 0.5413''$

Triangle Area: $A = \frac{1}{2}bh = \frac{1}{2}(.625'')(0.5413'') = 0.1691\,in^2$

Total Area: (For 232 needed tubes)

$$2 \cdot A \cdot N_{tubes} = 0.1691\,in^2 \cdot 232 \cdot 2 = 78.48\,in^2$$

Required shell ID:

$$A = \frac{\pi D^2}{4} \qquad D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{4 \cdot 78.48\,in^2}{\pi}}$$

$$D = 9.9962\text{ in}$$ (ANSWER 2)

Round up to a 10" pipe.

We can solve for the new steam supplied to the secondary HX in lb/hr, $\dot{m}_{TE}$, as a result of increasing the temperature of the make-up water entering the secondary HX. Properties are evaluated using X Steam Tables (English Units):

$$\bar{T}_{MW} = \frac{106.72 + 140}{2} = 123.36\,°F \qquad \rho_{MW,\,60°F} = 62.384 \ \text{lb/ft}^3$$

(flow rate going in to recovery HX is the same going in to secondary HX)

$$c_{p,\,mw@\bar{T}_{MW}} = 0.997986 \ \frac{Btu}{lb\,°F}$$

Recall, the mass flow rate of the make-up water can be found using equation (1)

$$\dot{m}_{MW} = \rho_{MW,\,60°F}\ Q_{MW} = \left(62.384\ \frac{lb}{ft^3}\right)(250\ gpm)\left(\frac{60\ min}{hr}\right)\left(\frac{0.133681\ ft^3}{1\ gallon}\right)$$

$$\dot{m}_{MW} = 125\,093\ \text{lb/hr}$$

Use equation (2) to find the total heat transfer rate:

$$q = \dot{m}_{MW}\, c_{p,\,mw@\bar{T}_{MW}} \left(T_{MW_{O2}} - T_{MW_O}\right)$$

$$= \left(125\,093\ \frac{lb}{hr}\right)\left(0.997986\ \frac{Btu}{lb\,°F}\right)(140 - 106.72)\,°F$$

$$q = 4\,154\,722\ \frac{Btu}{hr}$$

Use equation (3) to calculate the total heat transfer rate for steam undergoing phase change:

$$q = \dot{m}_{TE}\,(h_2 - h_{TE_o})$$

$$\dot{m}_{TE} = \frac{q}{h_2 - h_{TE_o}} = \frac{4\,154\,722\ \frac{Btu}{hr}}{(1249.52 - 277.3)\ \frac{Btu}{lb}}$$

$$\dot{m}_{TE} = 4273.4\ \frac{lb}{hr}$$

Note, the steam that heats the make-up water in the secondary HX has the same properties as those in the current design. Namely, $T_{TE_o} = 307.32°F$ and $T_{TE_i} = 435.37°F$. Hence, the reduction in steam supplied to the secondary HX can be found:

$$\Delta\dot{m}_{TE} = (\dot{m}_{TE})_{current} - (\dot{m}_{TE})_{desired} \qquad (20)$$

$$= 10270 - 4273.4\ \frac{lb}{hr}$$

$$\Delta\dot{m}_{TE} = 5996.6\ \frac{lb}{hr} \qquad \text{(ANSWER 4)}$$

- Steam Cost Savings

$\eta_{boiler} = 83\%$ Gas cost of Boiler $.000005 \$/Btu$

- Assume boiler feed water is at 14.7 psia 60°F

$h_{FW} = 28.11959 \frac{Btu}{lbm}$

- Enthalpy of steam leaving boiler at 700°F, 414.7 psia

$h_{steam} = 1362.0177 \frac{Btu}{lbm}$

heat saved:

$$Q = \dot{m}_{\substack{steam \\ saved}} \left(\Delta h_{\text{in boiler}}\right) \qquad (21)$$

$$Q = 5996.7734 \frac{lbm}{hr}\left(1362.0177 \frac{Btu}{lbm} - 28.11969\right)$$

$$Q = 7999084.905 \frac{Btu}{hr}$$

Gas saved:

$$Q_{BtuNG} = \frac{Q}{\eta_{Boiler}} = \frac{7999084.905 \, Btu/hr}{.83} \qquad (22)$$

$$Q_{BtuNG} = 9637451.693 \frac{Btu}{hr}$$

Cost saved:

$$\text{hourly Savings} = Q_{BtuNG} \cdot \text{Gas cost} \qquad (23)$$

$$= 9637451.693 \frac{Btu}{hr} \cdot 0.000005 \frac{\$}{Btu}$$

Operation yearly: 6000 hrs

$$= \boxed{48.19 \, \$/hr}$$

$$\text{yearly savings} = 48.19 \frac{\$}{hr} \cdot 6000 hr$$

$$\text{yearly savings} = \boxed{289123.55} \, \$ \qquad \text{(ANSWER 5)}$$

5

6.2 Sample Calculations

We start by analyzing the piping schematic to gather known values for the existing system (see Figure A1). We know that the wastewater flowing through the system is fully developed, steady, incompressible pipe flow and has a temperature of 180°F. The pressure can be assumed as atmospheric (P_w=14.7psia). This assumption is made since both the pit (point 1) and sewer (point 2) lines are vented to atmosphere and are not pressurized. From this information we used the X-Steam Tables to find all other water properties. Likewise, we can assume zero fluid velocity at points 1 and 2 due to zero pressurization. We are also given a 4in pipe diameter made of steel. Munson Table 8.1 was referenced for the equivalent roughness of commercial steel. The last pieces of given information regard the pump performance. We are given a flowrate of 300GPM and a pump impeller diameter of 8in. From the provided pump curve, we determine the pump head, power, and NPSH at 300GPM, as well as the shutoff head and speed. These values are summarized in Table 1.

Table 1. Given and Known Values for Existing Piping System

Item	Value	Item	Value
Wastewater temperature (T_{WW})	180 °F	Pipe diameter (D)	4 in.
Wastewater pressure (P_{WW})	14.7 psia	Pipe roughness (ε)	0.00015 ft
Wastewater density (ρ_{WW})	60.580 lb/ft^3	Flowrate (Q)	300 GPM = 0.6684 ft^3/sec
Wastewater viscosity (μ_{WW})	0.00023166 lb/ft/sec	Impeller diameter (d)	8 in.
Inlet and outlet pressure (P_1 , P_2)	14.7 psia	Pump speed (v)	1750 RPM
Inlet and outlet velocities (V_1 , V_2)	0 ft/sec	Pump head at Q (H_{pump})	62 ft
Inlet height (Z_1)	10 ft	Pump power at Q (P_{pump})	7.6 hp
Outlet height (Z_2)	7 ft	Pump NPSH at Q (NPSHr)	8 ft
Total pipe length (L)	385 ft	Pump shutoff head (TDH)	69 ft
Acceleration due to gravity (g)	32.2 ft/sec^2	Turbine head ($H_{turbines}$)	0 ft

Hence, a full Bernoulli's Equation between points 1 and 2 can be written for the existing system:

$$\frac{P_1}{\rho_{WW}g} + Z_1 + H_{pump} = \frac{P_2}{\rho_{WW}g} + Z_2 + H_{turbines} + \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)$$

After cancelling out known quantities the equation above can be simplified to:

$$Z_1 + H_{pump} = Z_2 + \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)$$

Where the velocity of wastewater through the pipes (V), the Darcy friction factor (f), and the sum of the pipe losses associated with pipe fittings (ΣK) is unknown. The task for problem 1 is to find the K loss coefficient for the control valve, so we will solve for ΣK then subtract known losses associated with the other pipe fittings at the end. First, we need to calculate V and f to determine head losses due to flow through the pipes. The average velocity of the wastewater through the 4in diameter pipe is calculated using the following equation:

$$Q = VA$$

$$V = \frac{Q}{\frac{\pi}{4}(D)^2} = \frac{0.6684\ ft^3/sec}{\frac{\pi}{4}\left(\frac{4}{12}ft\right)^2} = 7.6593\frac{ft}{sec}$$

Next, we need to calculate the Reynolds number for the flow to determine if the Darcy friction factor will be based on a laminar or turbulent flow analysis. The water properties used in the Reynolds number calculation are based on a wastewater temperature of 180°F and atmospheric pressure (i.e., conditions at point 1 in Figure A1) as shown:

$$Re = \frac{\rho_{WW,180°\mathrm{F}} VD}{\mu_{WW,180°\mathrm{F}}} = \frac{(60.580\ \mathrm{lb/ft^3})(7.6593\ ft/sec)(\frac{4}{12}ft)}{0.00023166\,\frac{\mathrm{lb}}{\mathrm{ft\cdot sec}}}$$

$$Re = 667\ 645$$

Hence, the flow through the pipes is turbulent and the friction factor will depend on the fluid density and pipe roughness. This factor can be approximated using the Haaland equation:

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{Re} + \left(\frac{\frac{\varepsilon}{D}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{667645} + \left(\frac{\frac{0.00015ft}{\frac{4}{12}ft}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = 7.6609$$

$$\therefore f = 0.017039$$

Next, we need to solve for the total pipe length, L, in the existing system. This can be done by referring to Figure A1. In the elevation view, we take horizontal pipes to represent piping in the x-direction, while vertical pipes are in the y-direction. In the plan view of the system, we take horizonal pipes to represent the x-direction, while vertical pipes are in the z-direction. Making sure not to double count piping in the x-direction within the two views, we can calculate the total pipe length by taking the pipe inlet in the wastewater pit as the measurement starting point:

$$L_x = 10ft + 200ft + 10ft = 220ft$$

$$L_y = 8ft + 2ft + 25ft + 25ft + 5ft = 65ft$$

$$L_z = 50ft + 50ft = 100ft$$

$$L = L_x + L_y + L_z = 385ft$$

We can now solve for ΣK using the simplified Bernoulli's Equation:

$$Z_1 + H_{pump} = Z_2 + \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)$$

$$\Sigma K = \frac{2g}{V^2}\left(Z_1 - Z_2 + H_{pump}\right) - \frac{fL}{D}$$

$$\Sigma K = \frac{2\left(32.2\,\frac{ft}{sec^2}\right)}{(7.6593\;ft/sec)^2}(10ft - 7ft + 62ft) - \frac{(0.017039)(385ft)}{\left(\frac{4}{12}ft\right)}$$

$$\Sigma K = 51.675$$

The overall head losses associated with pipe fittings in the existing system is not the solution to Problem 1. We want to know the K loss coefficient for the control valve. So, the next step is to use the Crane reference to find head losses associated with all other pipe fittings. We know that for the given pipe diameter of 4in. the friction factor, f_T, is 0.017. From Figure A1, we also know the existing system consists of a sharp inlet, 7 standard 90° elbows, 1 gate valve (β=1, θ=0), 1 control valve, and a sharp exit. The equations and values associated with these fittings are shown in Table 2.

Table 2. Head Losses for Pipe Fittings in Existing System

Item (Qty)	Equation	Value
Sharp inlet (x1)	$K_{in} = 0.5$	$K_{in} = 0.5$
STD 90° elbows (x7)	$K_{elb} = (7)(30)(f_T)$	$K_{elb} = 3.57$
Gate valve (x1)	$K_{GV} = (8)(f_T)$	$K_{GV} = 0.136$
Sharp exit (x1)	$K_{exit} = 1.0$	$K_{exit} = 1.0$
Control valve (x1)	$K_{CV} = \Sigma K - K_{in} - K_{elb} - K_{GV} - K_{exit}$	$K_{CV} = 46.469$

<u>Answer 1:</u> Therefore, the K loss coefficient for the control valve is approximated as 46.469 for the existing system.

Next, we look at the planned piping system incorporating the recovery heat exchanger (HX), as well as additional piping and fittings. Like the steps for problem 1, we will begin by stating given and known values. We know that wastewater is cooled down from 180°F to 140°F after leaving the recovery HX per city requirements. Therefore, the average wastewater temperature flowing through the tubes, $\overline{T}_{WW}$, is 160°F. The piping after the HX will be analyzed separately from the piping before the HX. Hence, new water properties need to be evaluated at 160°F and 140°F where the wastewater pressure remains constant at 14.7psia. In Table 3, the subscript "1" is used to refer to the piping system before the HX and "2" is after the HX. Water properties remain the same for the piping before the HX as those shown in Table 1. Table 4 lists the givens for the HX ½'' tube flow. The subscript "tubes" is designated to values representing flow through the HX. The tube roughness is taken from Munson Table 8.1. The tube inner diameter, tube length, and number of tubes are taken from previous calculations (see page).

Table 3. Given and Known Values for Planned Piping System

Item	Value	Item	Value
Wastewater temperature ($T_{WW,2}$)	140 °F	Pipe diameter (D_{pipe})	4 in.
Wastewater pressure ($P_{WW,2}$)	14.7 psia	Pipe roughness (ε_{pipe})	0.00015 ft
Wastewater density ($\rho_{WW,2}$)	61.378 lb/ft^3	Flowrate (Q_1)	300 GPM = 0.6684 ft^3/sec
Wastewater viscosity ($\mu_{WW,2}$)	0.0003134 lb/ft/sec	Wastewater velocity in pipes ($V_{1,\ pipe}$)	7.6593 ft/sec
Minor head losses after HX ($\Sigma K_{2,\ pipe}$)[1]	4.56	Minor head losses before HX ($\Sigma K_{1,\ pipe}$)	unknown
Pipe length (L_2)[2]	208 ft	Pipe length (L_1)[2]	213 ft

Notes:
1. Minor head losses through the piping after the HX results from one sharp inlet from the plenum to the rest of the piping, one STD branch tee, four STD 90° elbows and one sharp exit at the sewer outlet.
2. L_1 and L_2 are calculated by assuming the HX bypass is in the center of the 200ft pipe and knowing that the new total pipe length is 421ft.

Table 4. Given and Known Values for Planned HX ½'' Tubing

Item	Value	Item	Value
Average wastewater temperature ($\overline{T}_{WW}$)	160 °F	Tube inner diameter (D_{tubes})	0.402 in.
Wastewater pressure ($P_{WW,tubes}$)	14.7 psia	Tube roughness (ε_{tubes})	0.000005 ft
Wastewater density ($\rho_{WW,180°F}$)[1]	60.580 lb/ft^3	Flowrate (Q)	300 GPM = 0.6684 ft^3/sec
Wastewater viscosity ($\mu_{WW,tubes}$)	0.00026733 lb/ft/sec	Number of tubes (N_{tubes})	232
Tube length (L_{tubes})	12 ft	Minor head losses through tubes (ΣK_{tubes})[2]	1.5

Notes:
1. Wastewater density for the tubes is not evaluated at the average wastewater temperature, but instead at the higher temperature of 180°F due to the velocity limit that exists on the system. All other properties are evaluated at 160°F.
2. Minor head losses through tubes result from one sharp entrance into the tubes from the plenum and one sharp exit from the tubes to the exit plenum.

Now, we can write the simplified Bernoulli's equation for the planned piping system assuming ½'' tubing is used. We know that the pressure at points 1 and 2 is atmospheric and therefore cancels out. We also know the turbine head remains zero. Hence, the Bernoulli's equation for the planned piping system can be written as such:

$$Z_1 + H_{pump} = Z_2 + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{180°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{140°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{tubes}$$

Note that the velocity of wastewater flow through the piping, V_{pipe}, will be different before and after the HX due to its dependence on a changing flowrate. We will have to calculate the wastewater velocity in the piping after the HX. Furthermore, the Darcy friction factor will remain the same for the piping before the HX (i.e., at temperature 180°F) due to its dependence on fluid density and pipe roughness. However, the Darcy friction factor for the piping after the HX (i.e., at temperature 140°F) will need to be evaluated. The sum of the pipe losses associated with pipe fittings before the HX, $\Sigma K_{1,pipe}$, is also unknown. This value will be used to calculate the new K loss coefficient for the control valve. Furthermore, the velocity of wastewater through the tubes (V_{tubes}) and the Darcy friction factor associated with the tubes (f_{tubes}) will need to be evaluated. We begin by calculating the velocity of the wastewater in the piping after the HX, $V_{2,pipe}$, which has properties of water at 140 °F and atmospheric pressure:

$$Q_1 = \frac{\dot{m}_1}{\rho_1}, Q_2 = \frac{\dot{m}_2}{\rho_2}, \dot{m}_1 = \dot{m}_2$$

$$\therefore Q_2 = \frac{\rho_{1,180°F}}{\rho_{2,140°F}} Q_1 = \frac{60.580\,\frac{lb}{ft^3}}{61.378\,\frac{lb}{ft^3}} \cdot 0.6684\,\frac{ft^3}{sec} = 0.65971\,\frac{ft^3}{sec}$$

$$V_{140°F,pipe} = \frac{Q_2}{\frac{\pi}{4}(D)^2} = \frac{0.65971\ ft^3/sec}{\frac{\pi}{4}\left(\frac{4}{12}ft\right)^2} = 7.5597\ ft/sec$$

Next, we calculate the Reynolds number for the pipe flow after the HX:

$$Re = \frac{\rho_{WW,2} V_{2,pipe} D_{pipe}}{\mu_{WW,2}} = \frac{(61.378\ \mathrm{lb/ft^3})(7.5597\ ft/sec)(\frac{4}{12}ft)}{0.0003134\,\frac{\mathrm{lb}}{\mathrm{ft}\cdot\mathrm{sec}}}$$

$$Re = 493\ 512$$

Hence, the flow through the piping after the HX is turbulent and the friction factor is approximated using the Haaland equation:

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{Re} + \left(\frac{\frac{\varepsilon_{pipe}}{D_{pipe}}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{493511} + \left(\frac{\frac{0.00015ft}{\frac{4}{12}ft}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = 7.6111$$

$$\therefore f_{140\text{°F,pipe}} = 0.017262$$

Next, we calculate the velocity of wastewater through the tubes:

$$Q = (VA)_{tubes} = V_{tubes}\left(N_{tubes}\frac{\pi D_{tubes}^2}{4}\right)$$

$$V_{tubes} = \frac{Q}{N_{tubes}\left(\frac{\pi}{4}\right)(D_{tubes})^2} = \frac{0.6684\ ft^3/sec}{(232)\left(\frac{\pi}{4}\right)\left(\frac{0.402}{12}ft\right)^2} = 3.2687\frac{ft}{sec}$$

We can use the velocity through the tubes to find the Reynolds number for flow through the HX as follows:

$$Re = \frac{\rho_{WW,180\text{°F}}V_{tubes}D_{tubes}}{\mu_{WW,tubes}} = \frac{(60.580\ \text{lb/ft}^3)(3.2687\ ft/sec)(\frac{0.402}{12}ft)}{0.00026733\frac{\text{lb}}{\text{ft}\cdot\text{sec}}}$$

$$Re = 24\,814$$

Hence, the flow through the HX tubes is turbulent and the friction factor associated with the tubes is approximated using the Haaland equation:

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{Re} + \left(\frac{\frac{\varepsilon_{tubes}}{D_{tubes}}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = -1.8\log_{10}\left(\frac{6.9}{24814} + \left(\frac{\frac{0.000005ft}{\frac{0.402}{12}ft}}{3.7}\right)^{1.11}\right)$$

$$\frac{1}{f^{\frac{1}{2}}} = 6.3641$$

$$\therefore f_{tubes} = 0.02469$$

Finally, we can solve for ΣK_{pipe} using the simplified Bernoulli's Equation for the planned piping system:

$$Z_1 + H_{pump} = Z_2 + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{180°\mathrm{F},pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{140°\mathrm{F},pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{tubes}$$

$$\Sigma K_{1,pipe} = \frac{2g}{V_{1,pipe}^2}\left(Z_1 + H_{pump} - Z_2 - \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{tubes} - \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{140°\mathrm{F},pipe}\right) - \left(\frac{fL}{D}\right)_{180°\mathrm{F,pipe}}$$

$$\Sigma K_{1,pipe} = \frac{2(32.2ft/sec^2)}{(7.6593ft/sec)^2}\left(10ft + 62ft - 7ft - \left(\frac{(3.2687ft/sec)^2}{2\left(\frac{32.2ft}{sec^2}\right)}\left(\frac{(0.02469)(12ft)}{\left(\frac{0.402}{12}ft\right)} + 1.5\right)\right)_{tubes} - \left(\frac{(7.5597ft/sec)^2}{2(\frac{32.2ft}{sec^2})}\left(\frac{(0.017262)(208ft)}{(\frac{4}{12}ft)} + 4.56\right)\right)_{140°\mathrm{F},pipe}\right) - \left(\frac{(0.017039)(213ft)}{\left(\frac{4}{12}ft\right)}\right)_{180°\mathrm{F,pipe}}$$

$$\Sigma K_{1,pipe} = 43.647$$

We want to know the new K loss coefficient for the control valve. So, the next step is to use the Crane reference to find head losses associated with all new pipe fittings (f_T is still equivalent to 0.017). The equations and values associated with these fittings are shown in Table 5.

Table 5. Head Losses for Pipe Fittings before HX in Planned System

Item (Qty)	Equation	Value
Sharp inlet (x1)	$K_{in} = 0.5$	$K_{in} = 0.5$
STD 90° elbows (x5)	$K_{elb} = (5)(30)(f_T)$	$K_{elb} = 2.55$
Gate valve (x1)	$K_{GV} = (1)(8)(f_T)$	$K_{GV} = 0.136$
STD branch tee (x1)[1]	$K_{BT} = (1)(60)(f_T)$	$K_{BT} = 1.02$
Sharp exit (x1)[2]	$K_{exit} = 1.0$	$K_{exit} = 1.0$
Control valve (x1)	$K_{CV} = \Sigma K_{1,pipe} - K_{in} - K_{elb} - K_{GV} - K_{BT} - K_{exit}$	$K_{CV} = 38.441$
Notes: 1. Branch tees are added to the system to function as 90° elbows. 2. The additional sharp exit is the pipe exit into the plenum of the HX		

Note that the overall head loss in the planned piping system, h_L, is a constant for all tubing options at the given design flow since the control valve ensures that the following equation always holds true:

$$Z_1 + H_{pump} = Z_2 + h_L$$

Where

$$h_L = \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{180°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{140°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{tubes}$$

Hence, the overall head loss is a function of the pump head rise and the change in elevation between the inlet and exit points:

$$h_L = Z_1 - Z_2 + H_{pump} = 10ft - 7ft + 62ft = 65ft$$

<u>Answer 2a:</u> Therefore, the overall head loss through the proposed piping system at the design flow (i.e., Q=300GPM) is 65ft. This is true for all tubing options since the overall head loss is a function of the pump head rise and the difference in elevation between points 1 and 2 in Figure A1.

<u>Answer 2b:</u> We also determined that the new K loss coefficient associated with the control valve for the proposed piping system is approximately 38.441. This is true only for the ½'' tubing on which the calculations in the previous steps are based. In order to find the new K loss coefficient of the control valve for the other tubing options, one would have to use the appropriate tubing properties to evaluate ΣK_{pipe} (i.e., tube inner diameter, tube velocity, and the Darcy friction factor associated with the tubes).

Problem 3 asks to compute the pump work change and electrical cost savings/increase as a result of changing the piping system from existing to proposed. Table 1 lists the pump power for a flowrate of 300GPM as 7.6hp per the Goulds pump curve, which is approximately 5.667kW. This holds true for both the existing and planned systems since the control valve ensures the pump operates at the same point on the curve (i.e., same impeller diameter and flowrate, therefore same head rise).

<u>Answer 3a:</u> Hence, the pump work stays the same between the piping systems (i.e., zero pump work change). Pump work is approximately 7.6hp or 5.667kW. This is true for all tubing options.

The electrical costs associated with the operation of the pump can be computed using the given electricity cost of \$0.070/kWh. Thus, the hourly pump electricity costs can be calculated as follows:

$$Hourly\ pump\ electricity\ costs = \frac{\$0.070}{\text{kWh}} \times 5.667\text{kW} = \frac{\$0.3967}{\text{hr}}$$

With a given uptime of 24 hours per day, 5 days per week, and 50 weeks per year, we can calculate the total hours the plant operates per year:

$$Plant\ uptime\ per\ year = \frac{24\ hrs}{day} \times \frac{5\ days}{week} \times \frac{50\ weeks}{year} = \frac{6000\ hrs}{year}$$

Hence, the yearly pump electricity costs can be approximated as follows:

$$Yearly\ pump\ electricity\ costs = \frac{\$0.3967}{\text{hr}} \times \frac{6000\ hrs}{year} = \$2380/year$$

Note that this cost is based on a pump work of 7.6hp. Since this value remains constant between the two systems, we know that the pump electrical costs will also remain the same.

<u>Answer 3b:</u> Therefore, there are zero pump electrical cost savings/increases as a result of modifying the piping system. In both cases, the yearly electrical costs associated with the pump operation are approximately $2380 per year. This is true for all tubing options.

Next, we determine the feasibility of pumping the pit down to the bottom of the inlet pipe. In other words, we must identify if any NPSH issues exist. From Figure A1, we know that the bottom of the inlet pipe is at an elevation of 2ft. We have also assumed that the pump lies in the middle of the 10ft horizontal pipe. We know the pump height is 12ft. In order to find how far below the pump height the pump can operate, Z_{max}, before cavitation we will rely on the following equation:

$$Z_{max} = \frac{P_{atm}}{\gamma} - \Sigma h_L - \frac{P_v}{\gamma} - NPSHr$$

Where Σh_L represents the overall head loss between the free surface and the pump impeller inlet. In other words, all head losses before flow reaches the pump. Hence, the equation can be rewritten as follows:

$$Z_{max} = \frac{P_{atm}}{\gamma} - \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)_{pump} - \frac{P_v}{\gamma} - NPSHr$$

Known quantities for this scenario are listed in Table 6. Note that the wastewater flow velocity through the pipe section remains consistent with previous calculations due to a constant flowrate and pipe diameter. The Darcy friction factor has already been evaluated for the water properties experienced before flow reaches the pump. However, the length referenced in this equation only consists of the pipe length before reaching the pump and is therefore a new value as shown in Table 6. P_v refers to the water vapor pressure evaluated at 180°F using X-Steam Tables. Lastly, NPSHr was evaluated using the Goulds pump curve and a given flowrate of 300GPM.

Table 6. Given and Known Values for Piping System Before Pump

Item	Value	Item	Value
Wastewater temperature (T_{WW})	180 °F	Pipe diameter (D)	4 in.
Wastewater pressure (P_{WW})	14.7 psia	Pipe roughness (ε)	0.00015 ft
Wastewater specific gravity (γ)[1]	60.580 lbf/ft^3	Flowrate (Q)	300 GPM = 0.6684 ft^3/sec
Atmospheric pressure (P_{atm})	14.7 psia = 2116.8 lbf/ft^2	Pipe length (L)[2]	15 ft
Water vapor pressure (P_v)	7.5196 psia = 1082.8 lbf/ft^2	Pipe velocity (V)	7.6593 ft/sec
Acceleration due to gravity (g)	32.2 ft/sec^2	Darcy friction factor (f)	0.017039
Pump NPSH at Q (NPSHr)	8 ft	Sum of K loss coefficients (ΣK)[3]	1.01

Notes:
1. The specific gravity is calculated by taking the product of acceleration due to gravity and water density at 180°F and 14.7psia. Then a conversion factor of 1 lbf = 32.2lbm-ft/sec^2 is applied to give us specific gravity in units of lbf/ft^3.
2. New pipe length assumes that the pump lies in the middle of the 10ft horizontal pipe run shown in Figure A1.
3. Sum of K loss coefficients before reaching the pump consists of a sharp entrance and one elbow for which K_{in} = 0.5 and $K_{elb} = 30f_T$ = (30)(0.017) = 0.51.

Hence, Z_{max} can be evaluated by plugging in all known values into the previous equation:

$$Z_{max} = \frac{P_{atm}}{\gamma} - \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)_{pump} - \frac{P_v}{\gamma} - NPSHr$$

$$Z_{max} = \frac{2116.8 lbf/ft^2}{60.580 lbf/ft^3} - \frac{(7.6593 ft/sec)^2}{2(32.2 ft/sec^2)}\left(\frac{(0.017039)(15ft)}{(\frac{4}{12}ft)} + 1.01\right)_{pump} - \frac{1082.8 lbf/ft^2}{60.580 lbf/ft^3} - 8ft$$

$$Z_{max} = 7.450\ ft$$

<u>Answer 5a:</u> **This tells us that the pump can only operate safely without cavitation when the water level is at a maximum of 7.450ft below the pump height. Since the bottom of the inlet pipe is 10ft below the pump height, this means that we cannot pump the pit down to the bottom of the inlet pipe using the given design conditions. Hence, the system will encounter NPSH issues at a water level of 7.450ft below the pump height. Specifically, 2.55ft of wastewater above the pipe inlet cannot be pumped in this case.**

To determine if the existing pump can be used at a different flowrate to pump the pit down to the bottom of the inlet pipe, we will calculate the NPSHr for a Z_{max} of 10ft. If Z_{max} is equal to 10ft then we can indeed pump the pit down to the bottom of the inlet pipe. We will use the same equation as before, except we will rearrange the terms to solve for NPSHr:

$$NPSHr = \frac{P_{atm}}{\gamma} - \frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)_{pump} - \frac{P_v}{\gamma} - Z_{max}$$

$$NPSHr = \frac{2116.8 lbf/ft^2}{60.580 lbf/ft^3} - \frac{(7.6593 ft/sec)^2}{2(32.2 ft/sec^2)}\left(\frac{(0.017039)(15 ft)}{(\frac{4}{12} ft)} + 1.01\right)_{pump} - \frac{1082.8 lbf/ft^2}{60.580 lbf/ft^3} - 10 ft$$

$$NPSHr = 5.450\ ft$$

<u>**Answer 5b:**</u> **This value of NPSHr is too low for what can be achieved by the current pump at the given speed. Even at a flowrate of 200GPM, NPSHr is only reduced to 6ft which still does not allow us to pump down to the bottom of the inlet pipe (Z_{max} of 9.450ft and 0.55ft above the pipe inlet cannot be pumped). With the specified flowrate of 300GPM, we can only pump down to 2.55ft above the pipe inlet. Therefore, an external solution will need to be adopted to solve the NPSH issue and pump down to the pipe inlet level. In order to pump the remaining 2.55 feet above the inlet while maintaining a flowrate of 300 GPM, either the pump speed must be reduced (which can be done by installing a variable-frequency drive) or the impeller diameter must be reduced by shaving the impeller. These changes will minimize NPSHr in the same way that reducing the flowrate minimizes the value, thereby getting us closer to a Z_{max} of 10ft. This can be proven using the following affinity law:**

$$\boldsymbol{C_{Q1} = C_{Q2}}$$

$$\boldsymbol{\frac{Q_1}{n_1 D_1^3} = \frac{Q_2}{n_2 D_2^3}}$$

$$\boldsymbol{\frac{Q_2}{Q_1} = \left(\frac{n_2}{n_1}\right)\left(\frac{D_2}{D_1}\right)^3}$$

Where Q is the wastewater flowrate, n denotes the pump speed, and D represents the impeller diameter. Therefore, our goal of pumping down to the pipe inlet level can be achieved by either reducing the pump speed or reducing the impeller diameter. Reducing the impeller diameter will have a larger effect due to the presence of an exponent in the equation above.

There is two feet of wastewater in the pit below the pump inlet piping. In order to fully empty the pit, either additional piping must be installed, or a temporary pump, such as a sump pump that can be immersed in the water, must be installed in order to pump out that last 2 feet of wastewater.

Table 8: Given and Known Values for Economic Analysis

Item	Value	Item	Value
# of tubes (1/2”)	232	Hx Length (ft)	12
Tube Price per foot ($/ft)	5.66	Shell Diameter (in)	10
Shell Pipe Cost ($/in)	1500	Installation Cost ($/tube)	20
Cleaning Cost ($)	10	Cleanings per year	4
Yearly lost power (kWh)	1,186,335	Gas Saving Benefit ($)	289,125.59
Power cost ($/kWh)	0.07	Turbine Energy Output (BTU/lb)	112.49
Turbine Flowrate w/o Recovery Hx (lbm/hr)	125000	Conversion for BTU/hr to kW	0.000293
Turbine Flowrate w/ Recovery Hx (lbm/hr)	119003.2	Discount Rate	5%
Inflation Rate	3%	Time Period (years)	20

To find the initial cost of the heat exchanger we must include the material costs of the tubes, the pipe used for the shell, and the installation cost for the project. Calculate the total cost of the tubing using the number of tubes, length, and cost per foot of tubing

$$C_T = \#\ of\ tubes \cdot Hx\ length \cdot tube\ price = 232\ tubes \cdot 12\ ft \cdot \$5.66/ft = \$15{,}757.44$$

Calculate the total cost of the shell using the required shell diameter to contain all tubes and the cost per inch of diameter. The cost per inch of diameter is a given value assuming the full length of our heat exchanger.

$$C_S = Shell\ diameter \cdot pipe\ cost\ per\ inch = 10\ in \cdot \$1500/in = \$15{,}000.00$$

The installation cost is based on the number of tubes and a fixed cost per tube to install multiplied together.

$$C_i = \#\ of\ tubes \cdot install\ cost\ per\ tube = 232\ tubes \cdot \$20.00/tube = \$4{,}640.00$$

To calculate the total initial cost (C) you sum tube cost (C_T), shell cost (C_S), and installation cost (C_i). This is a one-time cost for the following economic analysis.

$$C = C_T + C_S + C_i = \$15{,}757.44 + \$15{,}000.00 + \$4{,}640.00 = \$35{,}397.44$$

<u>Answer 1:</u> Thus the total cost of the heat exchanger with ½’’ tubing is approximately $35,397.44. The same procedure would be used to calculate the total cost associated with the other heat exchanger options.

To determine the simple payback period, we will need to have the net yearly benefit of the project. The yearly costs are subtracted from the yearly savings to get the benefit. First calculate the annual cleaning cost (C_c) by multiplying together the number of tubes, cleanings per year, and the cost per tube of cleaning.

$$C_c = Cost\ per\ tube \cdot \#\ of\ tubes \cdot quarters\ per\ year = \$10/tube \cdot 232\ tubes \cdot 4 = \$9{,}280$$

To find the two different turbine powers you multiply the flowrate of steam, the energy from each pound of steam, and a conversion factor to get kW of power from BTU/hr.

$$Turbine\ Power\ = \ turbine\ flowrate \cdot energy\ per\ lb\ of\ steam \cdot conversion$$

$$Turbine\ Power\ existing = \ 125000 lbmhr \cdot 112.49 BTUlbm \cdot (2.93 * 10^{-4}) kW/BTU/hr\ \ = \ 4121.06\ kW$$

$$Turbine\ Power\ with\ Hx = \ 119003.2 lbmhr \cdot 112.49 BTUlbm \cdot (2.93 * 10^{-4}) kW/BTU/hr\ = 3923.35\ kW$$

The power lost is found by subtracting the lower energy production from the turbine with a heat exchanger in the system from the existing turbine energy production.

$$\begin{aligned} Power\ Lost\ &= \ Turbine\ Power\ [existing]\ -\ Turbine\ Power\ [with\ Hx] \\ &= \ 4121.06\ kW\ -\ 3923.35\ kW \end{aligned}$$

The yearly lost power is calculated by multiplying by the power lost during operation and the yearly uptime of the facility.

$$Yearly\ Lost\ Power = Power\ lost \cdot Yearly\ Uptime = 197.71\ kW \cdot 6000\ hrs$$

The cost of lost power is calculated from the yearly lost power multiplied by the given power cost.

$$C_{pl} = yearly\ lost\ power \cdot power\ cost = 1{,}186{,}235\ kWh \cdot \$0.07/kWh = \$83{,}036.47$$

The net yearly benefit can be found by subtracting the total initial cost and the lost power production from the gas savings that was calculated in previous calculations (see page).

$$NYB = gas\ savings - C_{pl} - C_c = \$289{,}125.59 - \$83{,}036.47 - \$9{,}280.00 = \$196{,}809.12/year$$

The simple payback in years can be found by dividing the total initial cost by the net yearly benefit and then multiplying by 12 for the payback in months.

$$Payback = \frac{C}{NYB} \cdot \frac{12\ months}{1\ year} = \frac{\$35{,}397.44}{\$196{,}809.12/year} \cdot \frac{12\ months}{1\ year} = 2.158\ months$$

<u>Answer 2:</u> This tells us that the cost of installing the heat exchanger with ½'' tubing will be paid back to the plant in the form of gas savings after approximately 0.18 years (i.e., 2.16 months). The calculation of yearly savings considers the yearly cleaning cost and cost of lost power due to reduced turbine operations. The simple payback period for the other tubing options can be calculated using the same procedure as above.

The net present value will be calculated using a discount rate of 5% and an inflation rate of 3% over a 20-year period. The present value at year 0 will be $35,397.44. The present value of the annual savings over n lifetime years can be found by using the equation:

$$PV_{annual} = \frac{NYB}{1+f} \cdot \left(\frac{(1+i')^n - 1}{i' \cdot (1+i')^n}\right)$$

Where the effective interest rate, i', can be found by using the inflation rate, *f*, and the discount rate, *i*, as such:

$$i' = \frac{i-f}{1+f} = \frac{0.05 - 0.03}{1+0.03} = 0.0194 = 1.94\%$$

Which means PV_{annual} over an estimated life of n=20 years can be calculated using the previously stated equation:

$$PV_{annual} = \frac{NYB}{1+f} \cdot \left(\frac{(1+i')^n - 1}{i' \cdot (1+i')^n}\right) = \frac{\$196809.12}{1.03} \cdot \left(\frac{1.01942^{20} - 1}{0.01942 \cdot 1.01942^{20}}\right) = \$3{,}142{,}014.91$$

Then, the salvage value after inflation can be found by using the inflation rate, *f*, and the given constraint that the heat exchanger will salvage for a quarter of its initial cost.

$$S = 0.25 \cdot initial\ cost\ HX \cdot (1+f)^{20} = 0.25 \cdot \$35{,}397.44 \cdot (1.03)^{20} = \$15{,}982.93$$

Next, the present value of the salvage worth after 20 years can be calculated using the given discount rate, *i*, as such:

$$PV_S = \frac{S}{(1+i)^{20}} = \frac{\$15982.93}{1.05^{20}} = \$6{,}023.80$$

Hence, the net present value, *NPV*, can be found by adding the present value of the annual savings and the present value of the salvage worth over a period of 20 years, and subtracting the initial cost associated with the HX.

$$NPV = PV_{annual} + PV_S - C = \$3{,}142{,}014.91 + \$6{,}023.80 - \$35{,}397.44 = \$3{,}112{,}641$$

<u>Answer 3:</u> **The project net present value (NPV) is approximately $3,112,641 for the ½'' tubing option. The NPV of the other tubing options can be calculated using the same procedure as above. They are differentiated by only a few thousand dollars, with the ¾'' tubing option providing the highest NPV. Therefore, the installation of a recovery heat exchanger is a sound economic decision. Furthermore, this installation satisfies the requirements of the city to reduce the wastewater temperature to 140°F before discharge into the sewer. All calculations above are based on meeting this requirement precisely. Future engineering studies could benefit from varying the wastewater discharge temperature to determine an optimum NPV.**

Results

Heat Exchanger Construction

Item	1/2" Tubing	5/8" Tubing	3/4" Tubing
Number of Tubes (LMTD)	232	204	195
Number of Tubes (ε-NTU)	232	204	195
Recovery HX Diameter (LMTD) (in)	9.996	11.717	13.747
Recovery HX Diameter (ε-NTU) (in)	9.996	11.717	13.747
Recovert HX Diameter (Rounded) (in)	10	12	14

Heat Transfer Values

Heat Rate (BTU/s)	1620.46
Make-up Water Exit Temp (°F)	106.72
Steam Reduction to Secondary HX (lb/hr)	5996.82
Boiler Fuel Cost Savings ($/hr)	48.19

Piping, Pumping, and Control Valve Values

Q1 (K for control valve before modification)	46.469		
Tube Size	1/2"	5/8"	3/4"
Q2a (Overall Head Loss) (ft)	65	65	65
Q2b (New K Value for control valve)	38.441	39.651	39.995
Q3a (Pump Work Change)	0	0	0
Q3b (Electrical Cost Change)	0	0	0
Q5 (NPSH Issues)	Yes; can only pump down to 2.55ft above pipe inlet; to pump down to pipe inlet need to either shave impeller, install VFD, or install secondary pump		

Economic Analysis

Item	1/2" Tubing	5/8" Tubing	3/4" Tubing
Recovery HX Inital Cost (including installation)	$ 35,397.44	$ 36,841.44	$ 41,092.80
Project Simple Payback Period (years)	0.180	0.186	0.207
Project Net Present Value	$ 3,112,641.33	$ 3,129,323.62	$ 3,131,543.07

Given Values and Basic Calculations/Conversions

Item	Value	Units	Conversions	
Wastewater (WW) Flowrate	300	gpm	1155	in^3/sec
WW Inlet Temp (Current Disch.)	180	°F		
Desired WW Discharge Temp	140	°F		
Make-up Water (MW) Flowrate	250	gpm	962.5	in^3/sec
MW Inlet Temp	60	°F		
Desired MW Discharge Temp	140	°F		
Recovery HX Tube Max Length	12	feet	144	inch
Steam Flowrate	125000	lb/hr		
Steam Pressure	400	psig		
Steam Temperature	700	°F		
Turbine Isentropic Efficiency	65%			
Turbine Exhaust Pressure	60	psig		
Secondary HX UA Value	150000	BTU/(hr-°F)		
Recovery HX Shellside Velocity	3	ft/sec	36	in/sec
Max Recovery HX Tubeside V	12	ft/sec	144	in/sec
k of 304 SS (300K)	14.9	W/mK	0.002391409	BTU/(s ft °F)
1/2" Tube OD	0.5	inch		
5/8" Tube OD	0.625	inch		
3/4" Tube OD	0.75	inch		
1/2" Tube ID	0.402	inch		
5/8" Tube ID	0.527	inch		
3/4" Tube ID	0.626	inch		
1/2" Tube Inner C.S. Area	0.126923485	in^2	0.000881413	ft^2
5/8" Tube Inner C.S. Area	0.218127847	in^2	0.001514777	ft^2
3/4" Tube Inner C.S. Area	0.307778691	in^2	0.002137352	ft^2

Tube Number & Velocity Calculations

Tube Diameter	1/2"	5/8"	3/4"
Number of Tubes (LMTD)	**232**	**204**	**195**
Number of Tubes (ε-NTU)	**232**	**204**	**195**
Cross-sectional area (in^2) (LMTD)	29.4462	44.4981	60.0168
Cross-sectional area (in^2) (ε-NTU)	29.4462	44.4981	60.0168
Velocity thru Tubes (ft/s) (LMTD)	3.2687	2.1630	1.6037
Velocity thru Tubes (ft/s) (ε-NTU)	3.2687	2.1630	1.6037
Outer Surface Area (in^2) (LMTD)	52477.1637	57679.6411	66161.9413
Outer Surface Area (in^2) (ε-NTU)	52477.1637	57679.6411	66161.9413
Inner Surface Area (in^2) (LMTD)	42191.6396	48635.4734	55223.1670
Inner Surface Area (in^2) (ε-NTU)	42191.6396	48635.4734	55223.1670

Cost Calculations

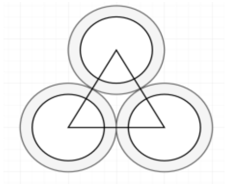
	Costs	For Tubes	Installation	Shell Cost	Total Capital Cost
1/2" Tube (per ft.)	$5.66	$15,757.44	$4,640.00	$15,000.00	**$35,397.44**
5/8" Tube (per ft.)	$6.03	$14,761.44	$4,080.00	$18,000.00	**$36,841.44**
3/4" Tube (per ft.)	$6.92	$16,192.80	$3,900.00	$21,000.00	**$41,092.80**
Install per tube:	$20.00				
Shell (per in of diam.)	$1,500.00				

Costs are calculated using the rounded values for shell diameter, found in cells Q23:R25.
Costs are based on the largest number of tubes calculated (the most expensive HX.)

Tube Packing & Shell Size Calculations

Most efficient packing method for circles: Triangle packing (equilateral triangle)
D/4 space between tubes: Equivalent to packing circles with diameter (5*OD/4)

Inner circle: Tube
Outer circle: Required space between tubes
Triangle: Representation of tube packing area
Side of triangle = distance between centers of tubes



Length of side of triangle:	1/2" Tube:	0.625	inch
	5/8" Tube:	0.78125	inch
	3/4" Tube:	0.9375	inch
Height of triangle:	1/2" Tube:	0.5413	inch
	5/8" Tube:	0.6766	inch
	3/4" Tube:	0.8119	inch
Area of triangle:	1/2" Tube:	0.1691	in^2
(triangle contains 1/2 of a tube)	5/8" Tube:	0.2643	in^2
	3/4" Tube:	0.3806	in^2
Packing efficiency		0.9069	

		LMTD	ε-NTU	
Area occupied by tubes:	1/2" Tube:	78.48	78.48	in^2
(inside shell area)	5/8" Tube:	107.83	107.83	in^2
	3/4" Tube:	148.43	148.43	in^2
Required shell ID:	1/2" Tube:	9.996	9.996	inch
(minimum)	5/8" Tube:	11.717	11.717	inch
	3/4" Tube:	13.747	13.747	inch
Rounded up for mfg.:	1/2" Tube:	10	10	inch
	5/8" Tube:	12	12	inch
	3/4" Tube:	14	14	inch

Reasoning: Cylindrical HX shells are usually just a section of larger pipe with heads welded or bolted on. You can't realistically go to a manufacturer and ask for a 11.72" pipe, but you can ask for a 12" pipe.

Givens, Tabulated Values, and Basic Calculations

Tbar(WW):	160	°F	Flow(WW):	0.668402778	ft^3/s
Tbar(MW):	83.362	°F	Flow(MW):	0.557002315	ft^3/s
P(WW):	14.7 psia				
P(MW):	100 psia				
From XSteam:					
Density(WW):	60.5804	lb/ft^3	Mdot(WW):	40.49210053	lbm/s
Density(MW):	62.3836	lb/ft^3	Mdot(MW):	34.7477995	lbm/s
Cp(WW):	1.0005	BTU/lbmR			
Cp(MW):	0.9981	BTU/lbmR			
From Tables:			C(WW):	40.51162181	BTU/Rs
R"(f) (WW) (Metric)	0.0002	(m^2*k)/W	C(MW)	34.68166026	BTU/Rs
R"(f) (MW) (Metric)	0.0001	(m^2*k)/W	Cmin:	34.68166026	BTU/Rs
R"(f) (WW) (US Customary)	4.0884	(s*ft^2*°F)/BTU	Cr	0.856091628	
R"(f) (MW) (US Customary)	2.0442	(s*ft^2*°F)/BTU			

MW Temp Out Calculations

T(in) (WW):	180	°F
T(out) (WW):	140	°F
T(in) (MW):	60	°F
T(out) (MW):	**106.724**	**°F**
Q=C(WW)*ΔT(WW):	1620.464872	BTU/s

Internal Heat Transfer Coefficient (Wastewater) (LMTD)

	1/2"	5/8"	3/4"
Density(WW (lbm/ft^3):	60.58	60.58	60.58
Velocity(WW) (ft/s):	3.27	2.16	1.60
ID (ft):	0.0335	0.0439	0.0522
Dynamic Viscosity (lbm/ft*s):	0.0002673	0.0002673	0.0002673
Re:	24814	21527	18959
Pr:	2.52	2.52	2.52
Nu:	99.52	88.82	80.24
Kwater (BTU/(s*ft*°F)):	0.000106	0.000106	0.000106
h(i) (BTU/(s*ft^2*°F)):	0.31492	0.21441	0.16306

Internal Heat Transfer Coefficient (Wastewater) (ε-NTU)

	1/2"	5/8"	3/4"
Density(WW (lbm/ft^3):	60.58	60.58	60.58
Velocity(WW) (ft/s):	3.27	2.16	1.60
ID (ft):	0.0335	0.0439	0.0522
Dynamic Viscosity (lbm/ft*s):	0.0002673	0.0002673	0.0002673
Re:	24814	21527	18959
Pr:	2.52	2.52	2.52
Nu:	99.52	88.82	80.24
Kwater (BTU/(s*ft*°F)):	0.000106	0.000106	0.000106
h(i) (BTU/(s*ft^2*°F)):	0.31492	0.21441	0.16306

External Heat Transfer Coefficient (Make-up Water)

	1/2"	5/8"	3/4"	
Density(MW) (lbm/ft^3):	61.92	61.92	61.92	
Velocity(MW) (ft/s):	3	3	3	
OD (ft):	0.0417	0.0521	0.0625	
Dynamic Viscosity (lbm/(ft*s)):	0.0005517	0.0005517	0.0005517	
Re	14029	17536	21044	
C	0.193	0.193	0.193	Re 4000 < Re < 40000: Table 7.2
m	0.618	0.618	0.618	
Pr:	5.59	5.59	5.59	
Nu	125.19	143.70	160.84	
Kwater (BTU/(s*ft*°F):	0.000098	0.000098	0.000098	
h(o) (BTU/(s*ft^2*°F):	0.29575	0.27158	0.25331	

LMTD Method to Find UA

LMTD:	76.5888	°F
P	0.3333	
R	1.1681	
F	0.95	
UA=Q/(F*LMTD):	22.27155768	BTU/(°F-s)

ε-NTU Method to Find UA

Qmax = Cmin*(ΔT(in))	4161.799	BTU/s
ε = Q/Qmax	0.389366	

Model HX as cross-flow with mixed shell (Cmin mixed)

$C_{\min}$ (mixed), $C_{\max}$ (unmixed) $$\mathrm{NTU} = -\left(\frac{1}{C_r}\right)\ln[C_r \ln(1-\varepsilon)+1]$$

NTU	0.640885	
UA = NTU*Cmin	22.22694	BTU/(°F*s)

Equation to Find 1/UA for a Single Tube

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{R''_{f_o}}{A_o} + \frac{\ln\frac{D_o}{D_i}}{2\pi k_{304} L} + \frac{R''_{f_i}}{A_i} + \frac{1}{h_o A_o}$$

$$\frac{1}{UA} = \frac{1}{h_i D_i \pi L} + \frac{R''_{f_o}}{D_o \pi L} + \frac{\ln\frac{D_o}{D_i}}{2\pi k_{304} L} + \frac{R''_{f_i}}{D_i \pi L} + \frac{1}{h_o D_o \pi L}$$

$$\frac{1}{UA} = (A) + (B) + (C) + (D) + (E)$$

Number of Tubes Calculation

	1/2"	5/8"	3/4"
h(i) (BTU/(s*ft^2*°F)) (LMTD):	0.3149	0.2144	0.1631
h(i) (BTU/(s*ft^2*°F)) (ε-NTU):	0.3149	0.2144	0.1631
h(o) (BTU/(s*ft^2*°F):	0.2957	0.2716	0.2533
R''(f)(i) (s*ft^2*°F)/BTU)	4.0884	4.0884	4.0884
R''(f)(o) (s*ft^2*°F)/BTU)	2.0442	2.0442	2.0442
D(i) (ft)	0.0335	0.043916667	0.052166667
D(o) (ft)	0.0417	0.0521	0.0625
k(304) (BTU/(s ft °F))	0.0024	0.0024	0.0024
L (ft)	12	12	12
(A) (LMTD)	2.5143	2.8171	3.1184
(A) (ε-NTU)	2.5143	2.8171	3.1184
(B)	1.3014	1.0411	0.8676
(C)	1.2099	0.9459	1.0023
(D)	3.2372	2.4694	2.0789
(E)	2.1526	1.8753	1.6755
1/UA (for one tube) (LMTD)	10.4154	9.1488	8.7426
1/UA (for one tube) (ε-NTU)	10.4154	9.1488	8.7426
UA for one tube (LMTD method)	0.0960	0.1093	0.1144
UA for one tube (ε-NTU method)	0.0960	0.1093	0.1144
Number of Tubes (LMTD)	**232**	**204**	**195**
Number of Tubes (ε-NTU)	**232**	**204**	**195**

Givens

Steam Flowrate (lbm/hr)	125000
Initial Steam Pressure (psia)	414.7
Initial Steam Temp (°F)	700
Turbine Efficiency	0.65
Turbine Exhaust Pressure (psia)	74.7
Secondary HX UA (BTU/hr*°F)	150000
Current MW Inlet T (°F)	60
Desired MW Outlet T (°F)	140
New MW Inlet T (°F)	106.724
MW Pressure (assumed) (psia)	100
MW Flowrate (gpm)	250
MW Flowrate (ft^3/hr)	2005.208
Density of MW from City (lbm/ft^3)	62.38358
MW Mass Flowrate (lbm/hr)	125092.1
Exit Density of MW (lbm/ft^3)	61.39583

Steam Turbine Calculations

Specific entropy of steam entering turbine (BTU/(lb*°F))	1.636721624
h(1) (BTU/lb) (h of steam entering turbine)	1362.017744
h(2s) (BTU/lb) (h of steam leaving perfect isentropic turbine)	1188.951415
h(2) = h(1) - η(h(1)-h(2s)) (BTU/lb) (h of steam leaving turbine	1249.52
Temperature of steam leaving turbine (°F)	435.3695235

$$\eta = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

Savings

Steam Savings (lbm/hr)	5996.81562
Boiler Efficiency	83%
Cost of Gas for Boiler ($/BTU)	$0.000005
Enthalpy of Boiler FW (BTU/lb)	28.11959366
Enthalpy of Steam (BTU/lb)	1362.017744
Q to Heat Saved Water (BTU/hr)	7999141.263
Q of Saved Gas (BTU/hr)	9637519.594
Cost Savings for Gas ($/hr)	$48.19
Operating Hours per Year	6000
Cost Savings per Year	$289,125.59

Assume boiler feed water is at 14.7psia, 60°F

Current Scenario

T(avg) of MW (°F)	100
cp of MW at T(avg) (BTU/(lbm*°F))	0.997702
C of MW (BTU/(hr*°F))	124804.6
Q (BTU/hr)	9984364
Steam inlet temperature (°F)	435.3695
Steam inlet enthalpy h(in) (BTU/lbm)	1249.52
Steam outlet temp (Tsat@60psig) (°F)	307.3208
Steam outlet enthalpy h(out) (BTU/lbm)	277.3121
Steam mass flowrate = Q/(h(in)-h(out)) (lbm/hr)	10269.73

With Added Recovery HX

T(avg) of MW (°F)	123.362
cp of MW at T(avg) (BTU/(lbm*°F))	0.997986
C of MW (BTU/(hr*°F))	124840.2
Q (BTU/hr)	4154185
Steam inlet temperature (°F)	435.3695
Steam inlet enthalpy (BTU/lb)	1249.52
Steam outlet temp (Tsat@60psig) (°F)	307.3208
Steam outlet enthalpy h(out) (BTU/lb)	277.3121
Steam mass flowrate = Q/(h(in)-h(out)) (lbm/hr)	4272.918

Given Values and Basic Calculations/Conversions

In Place Now

Inlet Height (ft)	10	
Pump Height (ft)	12	
Valve Height (ft)	37	
Outlet Height (ft)	7	
Total Pipe Length (ft)	385	
Pipe Diameter (in)	4	
Pipe Roughness (ft)	0.00015	(Table 8.1, Munson)
Pit Temperature (degF)	180	
Inlet Pressure (psig)	0	
Outlet Pressure (psig)	0	
Flowrate @180F (GPM)	300	
Flowrate @180F (ft^3/s)	0.6684	
Impeller Diameter (in)	8	
Average Water Velocity thru Pipe @180F (ft/s)	7.659	
Pump Head (ft)	62	
Pump Power (hp)	7.6	
Pump NPSH (ft)	8	(from pump curve)
Pump Shutoff Head (ft)	69	
Pump Speed (RPM)	1750	
Relative Roughness	0.00045	
Water Density @180F (lbm/ft^3)	60.5790	
Dynamic Viscosity @180F (lb/ft/hr)	0.8340	
Dynamic Viscosity @180F (lb/ft/s)	0.0002317	
Re in Pipe @180F	667642	
1/sqrt(f) @180F	7.6609	
Darcy friction factor f @180F	0.017039	
Acceleration due to gravity (ft/s^2)	32.2	
Water Density @140F (lbm/ft^3)	61.378	
Flowrate @140F (ft^3/s)	0.6597	
Average Water Velocity thru Pipe @140F (ft/s)	7.560	
Dynamic Viscosity @140F (lb/ft/hr)	1.128	
Dynamic Viscosity @140F (lb/ft/s)	0.0003134	
Re in Pipe @140F	493491	
1/sqrt(f) @140F	7.611	
Darcy friction factor f @140F	0.017262	

Planned

HX Height (ft)	37		
Total Length Before HX (ft)	213	Assume HX bypass in	
Total Length After HX (ft)	208	center of 200ft line	
HX Tube Roughness (ft)	0.000005	(Table 8.1, Munson)	
HX Tube Length (ft)	12		
Tube Specific #s:	**1/2"**	**5/8"**	**3/4"**
Tube Inner Diameter (in)	0.402	0.527	0.626
# of Tubes	232	204	195
Relative Roughness	0.000149254	0.000114	0.0000958
Flowrate thru each Tube (ft^3/s)	0.002881	0.003276	0.003428
C.S. Area of 1 Tube (ft^2)	0.000881	0.001515	0.002137
Velocity thru Tube (ft/s)	3.269	2.163	1.604
Re	24814	21527	18959
1/sqrt(f)	6.364135513	6.265853	6.172904203
Darcy friction factor f	0.024690004	0.025471	0.02624345

Power/Cost Calculations

Cost of Electricity($/kWh)	$	0.07
hp to kW Conversion (hp/kW)	1.341	
Pump Power (kW)	5.667	
Hourly Pump Electricity Cost	$	0.40
Plant Uptime per Year (hr)	6000	
Yearly Pump Electricity Cost	$	2,380.31

Same pump operating at same flowrate (thanks to the control valve); same power consumption! No increase in pump shaft work or in electricity costs

K Calculations (Existing System)

$$\frac{P_1}{\rho g} + z_1 + H_{pump} = \frac{P_2}{\rho g} + z_2 + H_{turb} + \frac{v^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)$$

Solve for sum of K:

K (solve for):	51.67467
LHS:	72
RHS:	72

Solver: Change J9 to make difference between J10 and J11 = 0

K values (Existing System)	
fT	0.017
Sharp Inlet (x1)	0.5
Elbow (x7)	3.57
Sharp Exit (x1)	1
Gate Valve	0.136
K_CV	**46.469**

K Calculations (Planned System)

$$\frac{P_1}{\rho g} + Z_1 + H_{pump} = \frac{P_2}{\rho g} + Z_2 + H_{turb} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{180°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{140°F,pipe} + \left(\frac{V^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)\right)_{tubes}$$

Solve for sum of K for each tube size:

	1/2"	**5/8"**	**3/4"**
K (solve for):	43.647	44.857	45.201
LHS:	72	72	72
RHS:	71.99963	72.00034	72.000107
K_CV	**38.441**	**39.651**	**39.995**

Solver: Change J24:L24 in order to make the sum of the squared differences between J25:L25 and J26:L26 = 0

Control valve ensures that overall head loss is always equal to pump head rise + height difference (z1-z2) for all tube sizes!

Overall Head Loss (ft) **65**

K values (HX Tube)	
Sharp Inlet (x1)	0.5
Sharp Exit (x1)	1

K values (New System before HX)	
fT	0.017
Sharp Inlet (x1)	0.5
Sharp Exit (x1)	1
Elbow (x5)	2.55
Branch Tee (x1)	1.02
Gate Valve	0.136

K values (New System after HX)	
Sharp Inlet (x1)	0.5
Branch Tee (x1)	1.02
Elbow (x4)	2.04

NPSH Problems Calculations

$$z_{max} = \frac{p_{atm}}{\gamma} - \frac{v^2}{2g}\left(\frac{fL}{D} + \Sigma K\right)_{before\ pump} - \frac{p_v}{\gamma} - NPSH_R$$

How far into pit can we pump?

L before pump (ft)	15	Assume pump is positioned in middle of 10ft horiz. pipe (5ft on either side before elbows)
Sum K before pump	1.01	
NPSHr of pump (ft)	8	
Water vapor pressure (psia)	7.5196	
Water specific gravity (lbf/ft^3)	60.579	
Velocity (ft/s)	7.6593	
Acceleration due to gravity (ft/s^2)	32.2	
Atmospheric pressure (psia)	14.7	
z_max (ft)	**7.450**	

Pump can only pump 7.45ft down from pump height w/o cavitation: leaves 2.55ft above inlet

To pump to pipe inlet (z=10ft):

Max allowable NPSH (ft) **5.449734** Too small: impossible for this pump at this speed to have this low an NPSHr

NPSH scales with v --> scales with Q

To reduce NPSH, reduce flowrate - affinity laws say that to reduce flowrate, reduce speed (install VFD) or impeller diameter (shave impeller); could also install temporary sump pump when cleaning

Answers

Q1 (K for CV before modification)	46.469		
Tube Size	1/2"	5/8"	3/4"
Q2a (Overall Head Loss) (ft)	65	65	65
Q2b (New K Value for CV)	38.441	39.651	39.995
Q3a (Pump Work Change)	0	0	0
Q3b (Electrical Cost Change)	0	0	0
Q5 (NPSH Issues)	Yes; can only pump down to 2.55ft above pipe inlet; to pump down to pipe inlet need to either shave impeller, install VFD, or install secondary pump		

Economic Calculations

Cost	Amount			Frequency
	1/2" Tubing	5/8" Tubing	3/4" Tubing	
Tubing	$ 15,757.44	$ 14,761.44	$ 16,192.80	Initial
Shell	$ 15,000.00	$ 18,000.00	$ 21,000.00	Initial
Installation	$ 4,640.00	$ 4,080.00	$ 3,900.00	Initial
Overall Initial Cost	**$ 35,397.44**	**$ 36,841.44**	**$ 41,092.80**	Initial
Cleaning	*$ 2,320.00*	*$ 2,040.00*	*$ 1,950.00*	*Quarterly*
Convert to yearly:	$ 9,280.00	$ 8,160.00	$ 7,800.00	Yearly
Lost Power Prod.	$ 83,036.47	$ 83,036.47	$ 83,036.47	Yearly
Benefits	**Amount**			**Frequency**
Gas Savings	$289,125.59	$289,125.59	$289,125.59	Yearly
Salvage (year zero $)	*$ 8,849.36*	*$ 9,210.36*	*$ 10,273.20*	*End*
Salvage (20 yrs of inflation)	*$ 15,982.93*	*$ 16,634.93*	*$ 18,554.54*	*End*
Salvage (Y0 w/ interest):	$ 6,023.80	$ 6,269.53	$ 6,993.01	Initial
Net Yearly Benefit NYB:	$196,809.12	$197,929.12	$198,289.12	Yearly
Overall Initial Cost	$ (35,397.44)	$ (36,841.44)	$ (41,092.80)	Initial
Present Worth of NYB	$3,142,014.98	$3,159,895.53	$3,165,642.86	Initial
Present Worth of Salvage	$ 6,023.80	$ 6,269.53	$ 6,993.01	Initial
NPV	**$ 3,112,641.33**	**$ 3,129,323.62**	**$ 3,131,543.07**	
Simple Payback Pd (yrs)	**0.1799**	**0.1861**	**0.2072**	
Simple Payback Pd (mths)	**2.1583**	**2.2336**	**2.4868**	

Power Production Calculations

Existing System	
Energy output of turbine (Btu/lb)	112.49
Steam flowrate through turbine (lbm/hr)	125000
Power output of turbine (Btu/hr)	14061639.23
Power output of turbine (kW)	4121.059655
W/ Recovery HX	
Energy output of turbine (Btu/lb)	112.49
Steam flowrate through turbine (lbm/hr)	119003.1844
Power output of turbine (Btu/hr)	13387038.77
Power output of turbine (kW)	3923.353776
Power production lost (kW)	197.7058793
Yearly Power Production Lost (kWh)	1186235.276

Project Parameters

Cost per kWh	$ 0.07
Yearly Uptime (hrs)	6000
Discount Rate	5%
Life (years)	20
Inflation Rate	3%
Effective Interest Rate	1.94%

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