Design of a Double Pipe Heat Exchanger

Department of Aerospace and Mechanical Engineering

Designed by:
William Koehrsen
Tyler Eston
Arturo Garcia
Theodore Bastian

EMAE 355 Project #1

Professor

Dr. J.R. Kadambi

Teaching Assistant

Scott Rubeo

October 17, 2016

Abstract

A double pipe heat exchanger was designed to facilitate the establishment of a detergent manufacturing plant. The heat exchanger is required to heat benzene for use in the manufacturing process from 27 °C to 40 °C at a flow rate of 1.4 kg/s. Excess water from power generation in the plant is available at a temperature of 70 °C to serve as the heating fluid. The design specifications set the overall rate of heat transfer to heat the benzene at 31754 Watts. Moreover, the heat exchanger was required to operate at the design parameters after one year of continuous operation. The final heat exchanger design calls for a concentric double pipe counterflow configuration with benzene in the inner pipe with an inner diameter of 0.03279 m (nominal diameter 1.25”) and water in the annulus with an inner diameter of 0.06338 m (nominal diameter 2.5”). The pipes are to be made of copper with a length of 8.00 ± 0.05 m. The length of the pipe was deliberately oversized for the initial conditions to ensure adequate operation of the heat exchanger after one year of exposure to the detrimental fouling effects of both benzene and water. Initially, the water will be pumped through the pipe at a mass flow rate of 3.95 kg/s. This results in an overall heat transfer of 45,590 W while ensuring that the water flow remains fully developed and turbulent. After one year of operation and one year of fouling from water and benzene, the water will need to be pumped at a mass flow rate of 4.00 kg/s and the overall heat transfer from the water to the benzene will be 32,080 W. The initial pressure loss of the water including the minor losses in the pipe is 16070 Pascals and rises to 16342 Pa after one year. The pressure loss associated with the benzene remains constant at 9805 Pa as the mass flow rate cannot be altered. The required pump size for the water is 79.2 W to overcome the pressure loss due to the friction and minor losses in the system and the required pump size for the benzene is 15.6 W. These pumps are sized for the final operating condition of the heat exchanger after one year. The pipe materials will cost a total of $462. Combined with the labor costs of machining and fitting the pipes, the total cost of the heat exchanger is projected to be $1260. Parameter comparison allowed for a heat exchanger design that minimizes cost and is able to operate within the design specifications for a minimum of one year before it will require cleaning.

Table of Contents

Section 1: Introduction

A double pipe heat exchanger is required in support of the construction of a manufacturing plant desired by the client. The heat exchanger is needed to heat benzene for use in the manufacturing process from 27°C to 40° C. Management requests the heat exchanger to be of the concentric double pipe style. An adequate supply of water used for power generation purposes is available at 70 °C for use as the heating fluid. The benzene flow rate is static at 1.4 kg/s and the flow rate of water was determined as part of the optimization of the design. The heat exchanger is to be run continuously for one year and should operate within the design requirements after this time period. This requires that the heat exchanger provide at least 31754 W of heat transfer after one year of fouling from water in the annulus and benzene in the inner pipe. The heat exchanger could either be parallel flow or counterflow. Four different pipe sizes (Table 7: Pipe PropertiesTable 7) and two different materials were available for use in the heat exchanger.

Section 2: Nomenclature

Symbol	Description	Units
	Subscript B indicates benzene **Subscript W indicates water**	
$h_{conv,i}$	Inner convective heat transfer coefficient	W/m^{2}K
$h_{conv,o}$	Outer convective heat transfer coefficient	W/m^{2}K
A_a	Area of annulus	m^2
A_p	Area of inner pipe	m^2
D_h	Hydraulic diameter	m
D_e	Equivalent heat transfer diameter	m
ID_a	Inner diameter of outer pipe	m
ID_p	Inner diameter of inner pipe	m
L_c	Characteristic length	m
OD_p	Outer diameter of outer pipe	m
$\dot{Q}$	Rate of heat transfer	W (or J/s)
R_d	Fouling factor	m^{2}K/W
T_F	Final temperature of fluid	°C
T_I	Initial temperature of fluid	°C
$U_{OA,i,f}$	Overall inner heat transfer coefficient after fouling	W/m^{2}K
$U_{OA,i}$	Overall inner heat transfer coefficient	W/m^{2}K
$U_{OA,o}$	Overall outer heat transfer coefficient	W/m^{2}K
c_p	Specific heat	J/kg K
f_d	Darcy friction factor	-
$\dot{m}$	Mass flow rate	kg/s
$r_{w,i}$	Inner wall radius	m
$r_{w,o}$	Outer wall radius	m
Δ	Change in quantity	-
ΔT_1	Temperature difference at station 1	°C

ΔT_2	Temperature difference at station 2	°C
ΔT_{LMTD}	Logarithmic Mean Temperature Difference	°C
ΔP	Difference in pressure	Pa
μ	Viscosity	Pa s
A	Area	m
D	Diameter	m
L	Length	m
Nu	Nusselt number	-
Po	Power	W (or J/s)
Pr	Prandtl number	-
$\dot{V}$	Volumetric flow rate	m^3/s
Re	Reynolds number	-
T	Temperature	°C
V	Velocity	m/s
k	Thermal conductivity	W/m K
α	Thermal diffusivity	m^2/s
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
ϵ	Absolute roughness	m

Section 3: Methods

3.1 Assumptions

Numerous realistic assumptions were employed in the heat exchanger design process to reduce complexity of the calculations. These assumptions represent typical simplification in thermal fluid designs and do not detrimentally impact the adequacy of the final heat exchanger.

- The pipes are hydraulically smooth, meaning that the roughness of the pipe walls is less than the thickness of the laminar sub-layer of the turbulent flow. Both stainless steel and copper pipes used in commercial applications satisfy this assumption.
- The flow through the pipes is fully developed, i.e. the flow profile is identical in the direction of the flow along the length of the pipe. Additionally, the boundary layers on the pipe walls have completely merged. This assumption is valid because the Reynolds number for the flow in the heater exchanger is much greater than 4000, listed in [1] as the minimum Reynolds number for fully developed, turbulent internal pipe flow.
- The fluid properties for benzene and water are relatively constant across the range of operating temperatures of the heat exchanger. Thus, the fluid properties can be evaluated as constants at the mean temperature of the fluids. Moreover, the two fluids are treated as incompressible with constant densities.
- In terms of fouling, the benzene can be treated as an organic liquid and the water can be treated as distilled as it is coming from a power generation use. The fouling factors for benzene and water come from [2]. The water has a smaller fouling factor but it is not negligible in the overall design of the heat exchanger.
- Fouling is assumed to cause insignificant pressure loss in this design. Fouling only affects the overall heat transfer coefficient.
- The design is perfectly insulated thermally from the environment, i.e. there is no heat loss from the water supply of 70°C to the environment outside the outer pipe. Also, there is no loss in pressure from the pipes to the environment.
- The system undergoes continuous operation of heat exchange so the system remains in a steady state, with no time-varying conditions.
- The pipes are horizontal and there is no considerable potential energy difference from the inlet to the outlet. The pressure drop is caused only by the friction in the pipes.
- The flow in the pipes has no rotation and shear stresses are negligible on the pipe walls.
- The thermal entry length for both the water and the benzene is insignificant because both flows are turbulent and fully developed.
- Benzene does not react with the copper tubing over the course of one year [3] and[4].

3.2 Design Procedure

Pipe and Fluid Properties

The necessary pipe properties were included in [1] for four different sizes of pipes. It was agreed upon that the entire project would be carried out in SI units which were also provided in [1] (see Table 7). The hydraulic diameter for the annulus, to be used in friction and pressure drop calculations, and the equivalent heat transfer diameter of the annulus, to be used in any heat exchange calculations, were given for all sizes of tubing. The hydraulic diameter must be used in friction calculations because both the inner and the outer walls of the annulus contribute to pressure losses as a result of friction. The heat transfer diameter takes into account the fact that only the one wall of the annulus is involved in heat transfer between the benzene and the water. The benzene was chosen as the fluid in the inner pipe because it was assumed that benzene, as an organic liquid, would have a greater tendency to foul the pipe than the water which was being used for power generation purposes, and it was further assumed that the inner pipe would be easier to clean when necessary. However, even though water does have a smaller tendency to foul as evidenced by its smaller fouling factor as shown in [2], both water and benzene fouling will contribute to a decrease in the overall heat transfer coefficient. Neither benzene nor water was assumed to react significantly with the copper according to the material data sheets for both benzene and copper [2] and [3]. For the inner pipe, the hydraulic diameter and the heat transfer equivalent diameter are both simply the inner diameter of the pipe. For the water flowing in the annulus, care was taken to use the correct diameter when appropriate.

The first step of the design process was to determine the fluid properties that would be used in all calculations. In order to find the correct properties, temperatures had to be specified for both benzene and water. As required by the manufacturing plant, benzene entered the heat exchanger at a temperature of 27° C and exited at a temperature of 40°C. Therefore, it was decided that the fluid properties for benzene should be evaluated at the average temperature of the benzene, or 33.5°C. This assumption was deemed valid because the fluid properties of benzene do not change significantly over the range of temperatures to which the benzene is exposed. The average temperature is a suitable reference point for the fluid properties of benzene and using different properties at different locations along the exchanger would unnecessarily increase the complexity of the design process. The fluid properties for benzene were provided for all the design groups in [5] to ensure uniformity. Next, a suitable temperature for the fluid properties of water had to be determined. Initially, the assumption was made that water would enter at 70°C and exit at the same temperature as the exiting benzene, 40°C. This would only be valid for an infinitely long parallel flow heat exchanger, and thus it was only used as a starting point for the calculations. The fluid properties for water were originally evaluated using [6] at a temperature of 55°C, the assumed average for the water in the heat exchanger. However, after performing preliminary calculations for the heat transfer, it was determined that the ending temperature of the water would be much greater than 40°C and would be closer to 60 or 65 °C. Therefore, it was decided to update the fluid properties for water to a temperature of 65 °C, the average of the water in the preliminary calculations. As is demonstrated in Table 8, this assumption turned out to be valid as the water exited the heat exchanger at a temperature around 68°C resulting in an average temperature of 69°C. The fluid properties of water do not vary significantly across this range and 65°C was chosen as the reference point for water.

Heat Transfer

(Refer to the nomenclature for explanation of variables used in equations)

There are numerous equations to take into consideration in the design of a double pipe heat exchanger. However, the driving equation behind the design relates the overall rate of heat transfer to the overall heat transfer coefficient, the area of the heat exchanger, and the average temperature difference between the heating and cooling fluid found using the Logarithmic Mean Temperature Difference (LMTD) method (from [1]):

$$\dot{Q} = U_{OA,i} A_i \Delta T_{LMTD} \tag{1}$$

This equation allows for the length of the heat exchanger to be determined from $A_i = \pi * D * L$. It was decided that the optimal approach to the design procedure would be to work through equation **(1)** one unknown variable at a time until the needed length of the heat exchanger could be calculated and then optimized.

Given the design specifications in [1], the first calculation that could be completed was the overall rate of heat transfer required to heat the benzene. As demonstrated on pg. 896 of [1], the overall heat transfer associated with an incompressible fluid is:

$$\dot{Q}_{\boldsymbol{b}} = \dot{m}_b c_{p,b} (T_{F,b} - T_{I,b}) \tag{2}$$

From [1], the rate at which benzene gains heat is equal to that at which water loses heat or

$$\dot{Q}_{\boldsymbol{b}} = -\dot{Q}_{\boldsymbol{w}} \tag{3}$$

This calculation yielded the first unknown quantity in equation **(1)**. The overall heat transfer rate was set by the design requirements and therefore had to be met when optimizing the parameters of the heat exchanger that could be altered.

Mass Flow Rate

In order to calculated an average temperature difference using the LMTD method, all four temperatures of the fluids had to be known. The only unknown temperature not given in [1] was the exit temperature of water. Finding the exit temperature first required a specification for the mass flow rate of the water. The most effective way to determine an acceptable mass flow rate for water was to calculate the minimum velocity at which the flow would be fully turbulent. Because the mass flow rate is directly associated with the velocity of the fluid, it was then possible to work out the needed mass flow rate. According to page 746 of [1], internal flow begins to transition from laminar to turbulent at a Reynolds number of 2300 and is fully turbulent at a Reynolds number of 4000. The Reynolds number equation comes from [1]:

$$Re = \frac{V * D}{\nu} \tag{4}$$

When using equation **(4)**, care must be taken to use the inner diameter of the pipe for benzene and the equivalent heat transfer diameter for water when dealing with heat transfer. As all of the necessary properties for benzene were already known, the Reynolds number for benzene in all sizes of the pipe could be calculated. The flow of benzene was clearly turbulent for all pipe sizes. The velocity of a fluid expressed in terms of mass flow rate is:

$$V = \frac{\dot{m}}{\rho * A} \tag{5}$$

Combining equations **(4)** and **(5)**, the minimum flow rate that would result in fully turbulent flow was determined. This flow rate was not the final flow rate used, but served as a helpful starting point for further calculations. All calculations were done in Microsoft Excel which allowed the effect of mass flow rate on other variables to be studied and optimized. Once the mass flow rate had been estimated, the final temperature of the water could be determined by rearranging the heat balance equation **(3)**.

$$T_{F,w} = T_{I,w} + \frac{\dot{Q}_w}{\dot{m}_w c_{p,w}} \tag{6}$$

Logarithmic Mean Temperature Difference (LMTD)

Once the final temperature of the water had been calculated, the second unknown variable in equation **(1)** could be worked out. According to page 906 of [1], the average temperature difference between the fluids must be calculated because this difference varies based on location within the heat exchanger. The LMTD method is away to account for this variation based on location and to determine a suitable average temperature difference. The equation for the LMTD is provided on page 909 of [1]:

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \tag{7}$$

Equation **(7)** is generalizable to both counterflow and parallel flow provided that the temperature differences considered are at the first station and the second station (rather than inlets and outlets).

Nusselt Number and the Convective Heat Transfer Coefficient

The final unknown variable in equation **(1)** is the overall heat transfer coefficient. However, before this variable could be calculated, the convective heat transfer coefficient of both benzene and water had to be determined. This was done through the use of the Nusselt Number, a dimensionless parameter that is the ratio between the convection and conduction heat transfer in an internal flow. The definition of the Nusselt number is provided on pages 602-604 of [1].

$$Nu \equiv \frac{h * L_C}{k} \tag{8}$$

The characteristic length, L_c, for internal flow is the inner diameter of the inner pipe for benzene and the equivalent heat transfer diameter for water in the annulus. The Nusselt number can also be parameterized as a function of both the Reynolds number and the Prandtl number of a fluid. As given in table 10.5 on page 774 of [1], the Nusselt number can explicitly be calculated:

$$Nu = 0.023 * Re^{0.8} * \mathrm{Pr}^{\mathrm{n}} \tag{9}$$

Equation **(9)** is called the Dittus-Boelter equation where n is 0.4 for a liquid being heated (benzene in the heat exchanger) and 0.3 for a liquid being cooled (water). Several assumptions were employed in order to use this expression. First, the Dittus-Boelter equation requires a Reynolds number > 10,000 and a Prandtl number between 0.7 and 160. Both of these assumptions are correct for the selected design and the fluids used. The other two assumptions are that for internal flow, the walls of the tube are hydraulically smooth (meaning that the laminar sub-layer thickness is greater than the roughness of the tube), and that the equation is evaluated for fully developed flow (meaning the boundary layers have merged). Both of these assumptions are valid in the design because the relative roughness of both stainless steel and the copper are small enough to classify the pipes as hydraulically smooth and the flow will be fully developed based on the Reynolds number. Combining equations **(8)** and **(9)**, the convective heat transfer coefficient of both the water and the benzene was found for the four different tube sizes.

Overall Heat Transfer Coefficient

The final unknown in equation **(1)** is the overall heat transfer coefficient. The overall heat transfer coefficient takes into account the sum of the individual thermal resistances in a system and is defined on page 902 of [1]:

$$U_{OA} \equiv \frac{1}{A \sum_j R_{thermal,j}} \tag{10}$$

For a double pipe heat exchanger, the thermal resistances are associated with convection at the tube inner wall, conduction through the tube wall, and convection at the tube outer wall. The equation for the overall heat transfer coefficient is provided on page 903 of [1]:

$$U_{OA,i} = \frac{1}{\frac{1}{h_{conv,i}} + \frac{r_{w,i}}{k_w} \ln\left(\frac{r_{w,o}}{r_{w,i}}\right) + \frac{r_{w,i}}{r_{w,o}} * \frac{1}{h_{conv,o}}} \tag{11}$$

The overall heat transfer coefficient can be calculated for either the inner surface of the inner tube or the outer surface of the inner tube as long as the appropriate area is employed in equation

(1). The overall heat transfer coefficient was calculated for the inner surface of both copper and stainless steel using properties found on page 770 of [1] and across the range of pipe sizes. Once the overall heat transfer coefficient had been determined, the necessary length of the heat exchanger could be calculated.

Length of Heat Exchanger

Equation **(1)** could be rearranged and combined with the equation for heat exchanger area ($A_i = \pi * D * L$) to yield:

$$L = \frac{\dot{Q}}{U_{OA,i} * \pi * D * \Delta T_{LMTD}} \tag{12}$$

Using this equation, the necessary length of the heat exchanger could be calculated for both materials, both types of heat exchanger, (counterflow and parallel flow), and for the range of pipe sizes. The design process for the initial requirements was now complete. However, the issue of compensating for fouling of the pipe after one year of operation still had to be addressed.

Fouling after One Year

The assumption was that both benzene and water would contribute to the fouling of the pipes and that the fouling would only affect the overall heat transfer coefficient. The fouling effect is calculated through the overall heat transfer coefficient. The equation for the reduction in the overall heat transfer coefficient as the result of fouling is from [9]:

$$U_{OA,i,f} = \frac{1}{(R_{d,B} + R_{d,W}) + \frac{1}{U_{OA,i}}} \tag{13}$$

The fouling factors for benzene and water, given as $R_{d,B}$and $R_{d,W}$, are for one year of operation and were provided by [8] where benzene was assumed to behave as an organic liquid and water was considered distilled from the power generation use. The fouling factor for benzene, 0.00018 W/m^2 k, is twice as great as that of water at 0.00009 W/m^2 k, but both water and benzene do need to be accounted for in order to design a heat exchanger that can operate at the requirements for one year. The fouling factor reduces the effective heat transfer coefficient, therefore increasing the area (and thus length) of the pipe required for the same heat transfer. After calculating the reduced overall heat transfer coefficient, the minimum lengths for the amount of total heat transfer required were determined.

Friction and Pressure Loss

The next step was to determine the friction factor within the pipe and the associated pressure loss. A Moody Chart, located on page 770 of [1], was used to obtain the friction factor rather than an explicit calculation. This chart plots the friction factor versus Reynolds number for a range of relative roughness parameters. The relative roughness of a pipe is the absolute roughness divided by the hydraulic diameter. The absolute roughness values for both copper and stainless steel were provided by [11]. The Reynolds number used in the calculation of the friction factor must be found using the hydraulic diameter. Friction factors for both water and benzene were calculated for all sizes of pipes and for the two different materials. After the friction factors had been determined, it was possible to calculate the pressure loss due to friction. This was done using the Darcy-Weisbach equation provided on page 736 of [1]:

$$\Delta P = f_D * \frac{L}{D} * \frac{1}{2} * \rho * V^2 \tag{14}$$

Equation (**14**) uses the hydraulic diameter for water because both the inner and outer walls cause friction losses within the annulus. Pressure losses were calculated for both benzene and water across the range of pipe sizes.

There are minor pressure losses associated with flow through the end caps in the system. According to pg 796 of [1], minor losses can be calculated from:

$$\boldsymbol{K} \equiv \frac{\boldsymbol{h_{L,minor}}}{\frac{(\boldsymbol{v_{avg}})^2}{\boldsymbol{2*g}}} \tag{15}$$

The loss coefficient, K, can be found for different types of valves and elbows. On page 797 of [1] there is a table and it was assumed that the end caps would act as flanged valves in terms of pressure loss. Minor losses were calculated by adding in the appropriate pressure loss given the diameter of the end caps to the pressure loss calculated using Equation (**14**). The total pressure loss for each fluid is therefore the combination of the pressure loss from friction within the pipe plus the additional pressure loss from losses in the end cap. These pressure losses can be used to calculate the pump needed given the chosen parameters of the heat exchanger system.

Pump Sizing

The final aspect of the design procedure was to select pumps that could handle the pressure loss of both the water and the benzene. The equation for power required to overcome a pressure loss is from [10]:

$$Po = \Delta P * \dot{V} \tag{16}$$

The volumetric flow rate can be calculated from the mass flow rate and the density:

$$\dot{V} = \left(\frac{1}{\rho}\right) * \dot{m} \tag{17}$$

Final pump sizes provided in Table 1 represent the maximum pump required for the system. This pump size will occur after one year for the water as it requires more power to drive the water at a higher mass flow rate. The pump required for the benzene remains constant as it was assumed

that the fouling of the benzene does not significantly impact the overall friction factor of the pipe and only has an effect on the overall heat transfer coefficient. Pump sizes provided are the absolute minimum given the expected pressure loss from friction and from minor losses and should be rounded up when the heat exchanger is implemented.
The final specifications of the heat exchanger system were obtained by optimizing this procedure within the design parameters. When completed in Excel, this method rapidly allowed properties to be calculated across the range of water flow rates, across the entire array of pipe sizes, for both available materials, and for counterflow or parallel flow style heat exchangers. Furthermore, it was possible to see the direct effects of altering one variable, which made for efficient optimization of the modifiable parameters.

3.3 Selection and Optimization of Parameters

Based on the design specifications outlined in [1], five parameters could be altered in the design of the concentric double pipe heat exchanger:

1. **Mass flow rate of water**
2. **Style of heat exchanger**: counterflow or parallel flow
3. **Material for heat exchanger**: stainless steel or copper
4. **Size of pipes**: four choices as shown in Table 7
5. **Length of heat exchanger**

These parameters were selected and optimized based on the design procedure outlined. The design procedure was fully implemented in Microsoft Excel, which allowed the influenced of any single variable to be studied and optimized for cost and under the constraint of the required heat transfer. The primary aim was to minimize the needed length of the heat exchanger within the requirement of an overall heat transfer rate of 31754 W for one year of continuous operation.

The first variable chosen was the mass flow rate of water. In order for the assumptions employed in the design to be valid, the flow of the water in the annulus had to be fully developed and turbulent. Also, turbulent flow transfers heat more efficiently than laminar flow. According to [1], the Reynolds number for the transition to laminar to turbulent flow is 2300 for internal pipe flow and 4000 marks the regime of fully developed turbulent flow. Based on these numbers and equations **(4)** and **(5)**, to achieve the necessary conditions required a minimum velocity of 1.8 m/s for the water with an upper bound found to be 3.0 m/s. Working from the velocity to the mass flow rate, this requirement could be met in all but the 4” by 3” pipe with a mass flow rate of 3.95 kg/s. This was selected to be the initial mass flow rate, and this decision also eliminated the 4” by 3” pipe size from consideration.

The next variable pinned down was the style of heat exchanger. Here there were only two options, counterflow and parallel flow. Once the mass flow rate of water had been specified, the final temperature of the water could be calculated from equation **(6)** and the logarithmic mean temperature difference (LMTD) could be found. Performing the calculation demonstrated that the LMTD for counterflow was always greater than that for parallel flow. As the overall heat transfer rate is directly related to the LMTD as shown in equation **(1)**, it was concluded that counterflow should be chosen to allow for a shorter pipe to be used in the heat exchanger. There were no known benefits to a parallel flow style of heat exchanger, and based on the numbers, counterflow was clearly the superior design choice.

Material selection was the next step in the optimization procedure. As with the style of heat exchanger, this choice was relatively straightforward. Based on the overall length calculations as shown in Table 14, copper required a shorter length of pipe for every pipe size. This was because the thermal conductivity of copper is much greater than that of stainless steel and by equation **(11)**, the overall heat transfer coefficient is therefore higher. In addition to allowing for a shorter overall pipe length, copper is the less expensive material per unit as shown in Table 3. Selecting copper as the pipe material both reduced the amount of material needed and decreased the cost of the heat exchanger for a set length. Corrosion factors of the material were not taken into consideration, but if the heat exchanger were required to last for longer than one year or were exposed to adverse environmental conditions, the choice of material would require more consideration, but given the design requirements, copper was the more cost-effective option. In addition, both the materials safety data sheet for benzene and copper cleared the use of copper piping in this design and ensured that there would not be any chemical reaction which could cause safety issues and/or undesired performance. The MSDS used are attached at the end of this report.

The second to last choice to be made was the size of the pipe. The 4” by 3” pipe had already been eliminated from contention because the velocity at the selected mass flow rate did not match the determined requirements. That left three options of pipe size. Selection of the pipe size was guided by Table 14 and by

Table 17. These showed that the 2.0” by 1.25” pipe had the shortest minimum length but also the greatest pressure loss for the water. This is because the pressure loss is proportional to the velocity squared, and the smaller annulus of this pipe size results in a considerable increase in water velocity. However, the 2.5” by 1.25” pipe size had the second shortest length and had the second lowest pressure loss out of the three pipes being considered. For this combination of results, it was decided that the 2.5” by 1.25” pipe offered the best combination of a short length and a relatively small pressure drop. This would optimize the cost of the heat exchanger by allowing for less copper material to be used while also reducing the power and the size of the pump needed to drive the water through the heat exchanger. By employing Excel to make calculations for all parameters, the problem of choosing the final pipe size was solved by analyzing the information available to find the most efficient piping.

The final variable that needed to be determined was the length of the piping. As can be seen in Table 14 and

Table 15, the size of the needed pipe varied considerably after one year of operation. Based on the minimum required length of the pipe after one year, it was deemed that the length should be 8.00 m. The tolerance on this length was set as ± 0.05 m (5 cm) in order to reduce the cost of machining and this range would still allow for adequate heat transfer. After selecting the pipe length of 8.00 m, the calculations were re-run at the initial mass flow rate of water of 3.95 kg/s. At this mass flow rate, it was discovered that not enough heat transfer would take place to raise the temperature of the benzene to that required in the manufacturing process. Therefore, the mass flow rate of the water was incrementally increased until the necessary heat transfer occurred. At 4.00 kg/s, the 2.5” by 1.25” copper pipe with a length of 7.00 m operating in the counterflow style could provide a heat transfer to the benzene of 32080 W (see Table 11). As the requirement to heat the benzene to 40°C only needs 31754 W, there is a margin of safety should any of the calculations be slightly inaccurate.

The parameters selected allow for a cost-effective heat exchanger that is able to operate within the requirements outlined in [1] for one year of continuous use. Further successful operation beyond this timeframe and outside of these operating conditions cannot be guaranteed. For a summary of the parameters and the final design, please refer to Table **1**.

Section 4: Results

4.1 Heat Exchanger Specifications

Type M copper tubing will be used in the concentric double pipe counterflow heat exchanger. The nominal diameter of the outer tube is 2.5” and the nominal diameter of the inner tube is 1.25.” The total length of the heat exchanger is 7.00 m to ensure proper operation after one year of fouling. The required heat transfer rate is 31754 W. Below are the specifications for the design.

Initially:		After 1 Year:	
$\dot{m}_w$:	3.95 kg/s	$\dot{m}_w$:	4.00 kg/s
V_w:	1.83 m/s	V_w:	1.86 m/s
$U_{OA,i}$:	1,569 W/m^{2}k	$U_{OA,i}$:	1,104 W/m^{2}k
$\dot{Q}$:	45590 W	$\dot{Q}$:	32080 W
Δp_w:	16070 Pa	Δp_w:	16342 Pa
Δp_b:	9805 Pa	Δp_b:	9805 Pa
		Water Pump Power (accounting for minor losses):	79.2 W
		Benzene Pump Power (accounting for minor losses):	15.6 W

Table 1: Heat Exchanger Specifications

4.2 Engineering Drawings

Figure 1: Overall Schematic

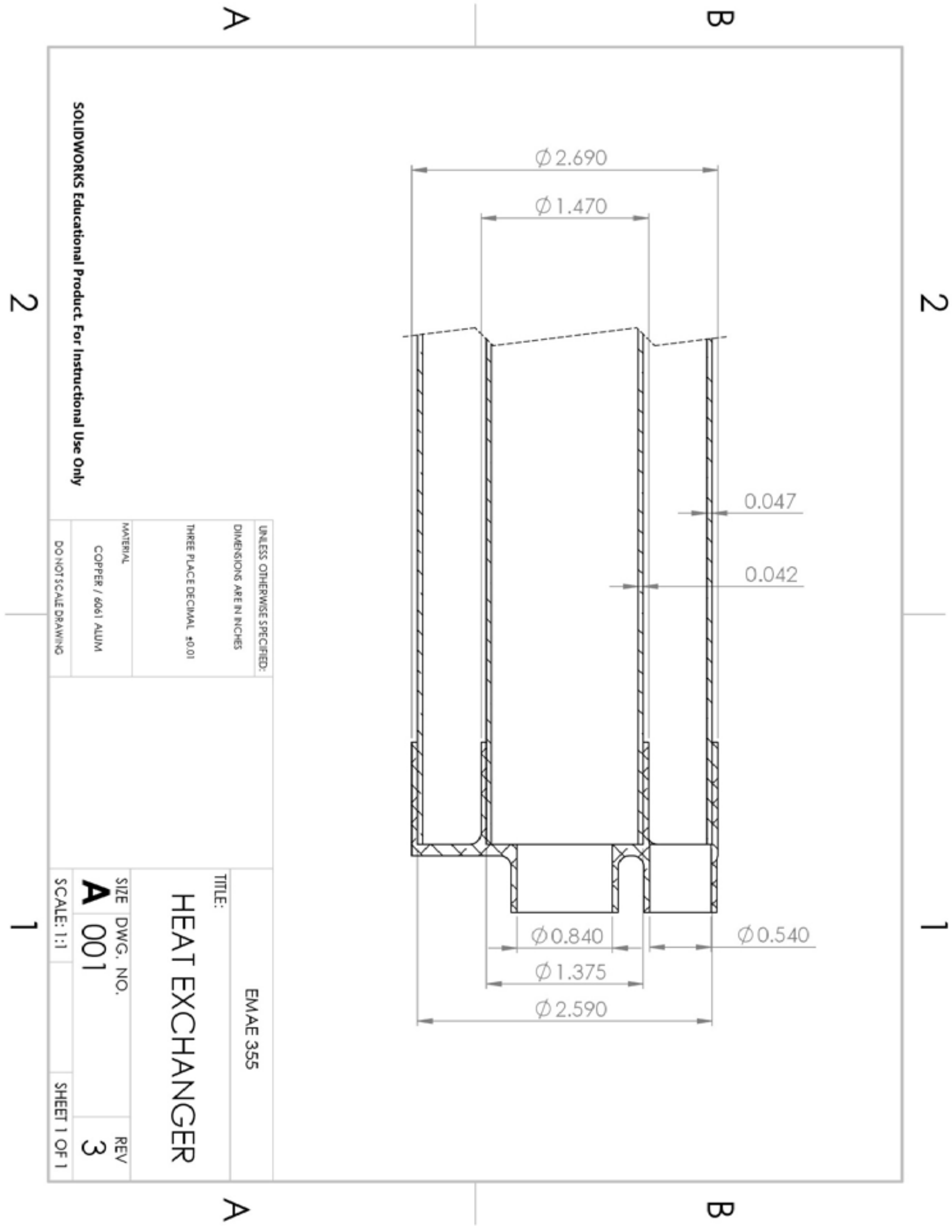

Figure 2: Section View of Aluminum End Cap

4.3 Bill of Materials and Cost Analysis

Part Number	Description	Quantity	Price	Subtotal
01	Copper tube, 2.5” Nominal, 25’ Long	1	$309.77	$309.77
02	Copper tube, 1.25” Nominal, 25’ Long	1	$108.92	$108.92
03	6061 Aluminum End Cap	2	$22.01	$44.02
Parts Total				$462.72

Table 2: Bill of Materials (from [13])

Prices Listed in Dollars per Foot							
Size	Copper Inner	Copper Outer	SS Inner	SS Outer	Copper Total	SS Total	SS / Copper
2 x 1 1/4	4.36	8.54	11.73	16.27	12.90	28.00	217.05%
2 1/2 x 1 1/4	4.36	12.39	11.73	20.93	16.75	32.66	194.99%
3 x 2	8.54	16.48	16.27	25.66	25.02	41.93	167.59%
4 x 3	16.48	31.80	25.66	31.51	48.28	57.17	118.41%

Table 3: Cost Comparison of Stainless Steel versus Copper Pipe (from [13])

Cost Analysis

Given the cost to CNC machine the end caps and a plumbing engineer’s hourly rate for assembly and fitment [9], the labor and machining cost totals to approximately $800, for a grand total of approximately $1,260.

Regarding the material selected for the pipes themselves, copper was chosen over stainless steel primarily because of the thermal conductivity-to-cost ratio. For the exact same dimensions, copper pipes (in the desired length and thickness) were found to be less expensive than their stainless steel counterparts [10]. Even with fouling accounted for, copper pipe in the appropriate length was cheaper than the more fouling-resistant stainless steel keeping in mind that in either scenario, the heat exchanger has been designed to require yearly maintenance to remove interior fouling.

Additionally, it is not feasible to purchase a pre-made end cap for these exact pipes and flow rates, so one was designed to be machinable using a CNC mill and only two fixture methods to keep costs down. 6061 Aluminum alloy was chosen as the cap material due to its abundance, low cost, easy machinability, and high corrosion resistance.

Section 5: Discussion

5.1 Method for Addressing Fouling

The heat exchanger was required to provide the necessary heat transfer of 31754 W for one year of continuous operation. The issue that arises in the operation of the heat exchanger is the fouling in the inner pipe from the benzene and in the outer pipe from the water. This fouling needed to be accounted for in the design of the heat exchanger. In order to accomplish this, the length was selected as 8.00 m and it was decided that the mass flow rate of water would gradually be increased over the course of operation. The length of 8.00 m ensures there will be enough surface area to transmit heat so that even after a year of fouling, the heat exchanger can still be successfully operated within its intended parameters. Initially, the system will pump 3.95 kg of water through the exchanger per second which computes to a velocity of 1.83 m/s. With a clean pipe and that velocity, this achieves a heat transfer of 45,590 W, which is slightly greater than the initial requirement. However, after a year of fouling, the initial flow rate of water is not enough to achieve 31,754 W of heat transfer, the amount of energy necessary to sufficiently heat the benzene. To increase the heat transfer, the speed of water pumped will be raised to 1.86 m/s which results in 4.00 kg of water flow through the exchanger every second. With that increased flow rate and the area of the 8.00 m pipe, the final heat transfer rate of the double pipe heat exchanger will be 32,080 W, enough to heat the benzene to 40 degrees C.

This design only accounted for the effects of fouling on the overall heat transfer coefficient. The pressure drop in the pipe could also be directly affected if the fouling significantly increased the relative roughness of the pipe, which would lead to a higher friction factor and loss a greater pressure loss. Under the simplified scenario, the fouling only decreased the overall heat transfer coefficient as shown in equation (13) and Table 13. An additional assumption made was that benzene has the fouling properties of an organic liquid and therefore the fouling factor can be found from [8]. Also, the water was assumed to be distilled as it was heated in the power generation process as given in [7]. These assumptions were made based on the design specifications provided and the manager's instructions.

If the heat exchanger needs to continue operations beyond a year, fouling will continue to occur until the point where the heat exchanger no longer provides the necessary heat transfer. To counter any fouling beyond the heat exchanger's intended maintenance free operating period, the flow rate of water will need to be increased or the exchanger will need to be taken out of service for cleaning as necessary.

5.2 Alternative Designs

Several parameters were chosen in the design of the heat exchanger; the sizing of the tubing, the material for the tubing, the heat exchanger flow type, and the mass flow rate of the water. Below are comparative alternate designs that were considered. Different design decisions clearly show the benefit of choosing one design over the other.

The final design is type M copper tubing with 2.5" nominal diameter outer tubing and 1.25" nominal diameter inner tubing with counterflow. Below are specifications for the aforementioned piping, and the varying performance of counter vs parallel flow:

Design 1 Parallel Flow		Design 2 Counterflow	
Tube length	5.61 m	Tube Length	5.57 m
LMTD	35.0118	LMTD	35.2503
Δp_w:	7808 Pa	Δp_w:	7431 Pa
Δp_b:	5888 Pa	Δp_b:	5582 Pa
Water Pump Power	143.23 Pa	Water Pump Power	136.31 W
Benzene Pump Power	111.11 Pa	Benzene Pump Power	105.33 W

Table 4: Comparison of Counter and Parallel Flow

Cross Flow heat exchangers decrease the length of piping needed and the pump power needed; hence they can save cost of material, space needed, and power usage.

Below is a table that shows the advantage of copper tubing versus stainless steel for the same size piping in cross flow heat exchange:

Design 3 Stainless Steel		Design 2 Copper	
Tube Length	6.12 m	Tube Length	5.57 m
$U_{OA,i}$:	1430	$U_{OA,i}$:	1569
Δp_W:	8512 Pa	Δp_W:	7432 Pa
Δp_B:	6419 Pa	Δp_B:	5582 Pa
Water Pump Power	156.14 W	Water Pump Power	136.31 W
Benzene Pump Power	121.13 W	Benzene Pump Power	105.33 W

Table 5: Comparison of Stainless Steel and Copper

The Copper piping has a higher thermal conductivity which helps increase the total heat transfer coefficient and consequently reduces the length of tubing needed. Because less tubing is needed, the copper pipe heat exchanger also has a lower pressure loss which means less power is needed to pump the fluid.

Below are designs that show the specifications for different sized pumps for copper tubing with crossflow heat exchange, neglecting minor losses:

Design 2: 2.5" x 1.25"		Design 4: 2"x1.25"		Design 5: 3"x2"		Design 6: 4"x3"	
Tube Length	5.57 m	Tube Length	4.95 m	Tube Length	6.91 m	Tube Length	9.07 m
Δp_W :	7432 Pa	Δp_W :	47761 Pa	Δp_W :	12900 Pa	Δp_W :	11237 Pa
Δp_B :	5583 Pa	Δp_B :	4962 Pa	Δp_B :	723 Pa	Δp_B:	138 Pa
Water Pump Power	136.33 W	Water Pump Power	177.162 W	Water Pump Power	23.471 W	Water Pump Power	156.04 W
Benzene Pump Power	105.36 W	Benzene Pump Power	93.64 W	Benzene Pump Power	56.4 W	Benzene Pump Power	49 W

Table 6: Comparison of Pipe Sizing

The piping needed for Design 5 and Design 6 are significantly larger than the piping for Design 2 and Design 4, so Design 2 and Design 4 have space and material cost advantages. Designs 4, 5, and 6 require one pump to have at least 2 times or more power than the other pipe. More powerful pumps will cost more than the pumps required for Design 2.

Taking into account all of the variables, the final design minimizes both the length the cost of the heat exchanger. Furthermore, the final design is able to meet both the initial constraints and the requirement that it remain operable after one year of continuous use.

5.3 Maintenance Schedule

The heat exchanger was designed for an initial service free period of one year of continuous operation. After a year, there will be enough fouling that cleaning the exchanger will be necessary otherwise either the exchanger will no longer be able to properly heat the benzene or the flow rate of the water will need to be increased beyond the intended maximum velocity the exchanger was designed for.

To clean the inside pipes of the exchanger, the system must be taken out of service and disconnected from the benzene and water supply lines in order for the technician to access the inside. For initial cleaning, a rotary tube cleaner should be used to remove most of the foreign material. Next, descaling chemicals should be run through the pipes to detach any remaining deposits. Third, the sides of the pipes should be buffed until smooth in order to remove any blemishes in the interior surface of the pipes which may serve as future sites for fouling to happen. Finally, the pipe should be washed out to remove any traces of chemicals which may react with the benzene and any leftover material from the defouling process. Only after the pipe has been cleaned should it be put back into service.

Section 6: Conclusion

A concentric double pipe heat exchanger was designed for use in the manufacturing process of a detergent producing plant. Five variables associated with the heat exchanger could be modified in order to meet the requirement of heating benzene in the inner pipe from 27 °C to 40 °C at a mass flow rate of 1.4 kg/s. This required a total heat transfer of 31754 W to be provided by water from a power generation purpose entering the heat exchanger at 70°C. The mass flow rate of water in the annulus is initially designed to be 3.95 kg/s in order to ensure a velocity that corresponds to a fully developed and turbulent flow. The mass flow rate of water will gradually be increased to 4.00 kg/s after one year of operation. The overall length of the heat exchanger was set at 8.00 ± 0.05 m which, combined with the increased mass flow rate of water, will provide the necessary heat transfer after one year of fouling from the benzene and the water. Counterflow will be used in the heat exchanger as it results in a greater logarithmic mean temperature difference and hence a shorter length for the heat exchanger. Furthermore, copper will be used in place of stainless steel because it has a significantly higher thermal conductivity which reduces the minimum length of the heat exchanger. Copper also has a cost benefit per unit over stainless steel which means that not only will less material need to be used, but the material that is used will be less expensive for a given length. The size of the tubing is a nominal outer diameter of 2.5" and a nominal inner diameter of 1.25." This pipe size provides the second highest overall heat transfer coefficient as shown in Table 13 while reducing the pressure loss as compared to the pipe with the greatest overall heat transfer coefficient as demonstrated in

Table 17. The 2.5 x 1.25 size pipe has the second shortest length while facilitating the use of a less powerful pump which optimizes both the cost and performance of the heat exchanger. The required pump size for the water is 79.2 W to overcome the pressure loss due to the friction and minor losses in the system and the required pump size for the benzene is 15.6 W. The pump for water is overpowered initially but is sized to provide the correct flow rate after the required one year of continuous operation in the heat exchanger. The final cost for the heat exchanger is estimated to be $1260 of which $800 is labor. The initial total heat transfer rate of the concentric double pipe heat exchanger is 45,590 W and the heat transfer rate after one year of operation is 32,080 W. The heat exchanger will thus perform adequately for at least one year before the mass flow rate must be further increased, or the pipes cleaned.

Section 7: Acknowledgements

Our design team would like to thank the following individuals for their assistance with the design process. The heat exchanger would not have been completed without their generous counsel.

Professor Kadambi: for providing a suggested method for the design process, general all-around advice, and for being willing to answer our numerous questions related to every aspect of a heat exchanger.

Teaching Assistant Scott Rubeo: for general assistance and for being willing to read and suggest improvements to an initial draft of this report.

Section 8: References

[1] S. Turns, *Thermal-Fluid Sciences: An Integrated Approach*. Cambridge: Cambridge University Press, 2006.

[2] "Engineering Page > Heat Exchangers > Typical Fouling Factors", *Engineeringpage.com*, 2016. [Online]. Available: http://www.engineeringpage.com/technology/thermal/fouling_factors.html. [Accessed: 18- Sep- 2016].

[3] "Fouling and Reduced Heat Transfer in Heat Exchangers", *Engineeringtoolbox.com*, 2016. [Online]. Available: http://www.engineeringtoolbox.com/fouling-heat-transfer-d_1661.html. [Accessed: 18- Sep- 2016].

[4] Cerro Flow Products LLC, "Copper Safety Data Sheet", St. Louis, 2016.

[5] Muskingum University, "Benzene Safety Data Sheet", New Concord, 2016.

[6] J. Kadambi, "A Suggested Method for the Heat Exchanger Project", EMAE 355 Lecture Notes, 2016.

[7] J. Kadambi, "Project #1 Design Requirements", EMAE 355 Lecture Notes, 2016.

[8] J. Kadambi, "Benzene Fluid Properties", EMAE 355 Lecture Notes, 2016.

[9] CustomPartNet, "Machining cost Estimator," in *CustomPartNet*, 2009. [Online]. Available: http://www.custompartnet.com/estimate/machining/. Accessed: Sep. 19, 2016.

[10] "McMaster-Carr," 2016. [Online]. Available: http://www.mcmaster.com/. Accessed: Sep. 18, 2016.

[11] *NIST12 - Fluid Properties Database*. National Institute of Standards and Technology, 2016.

[12] "Power Gained by Fluid from a Pump or a Fan", *Engineeringtoolbox.com*, 2016. [Online]. Available: http://www.engineeringtoolbox.com/pump-fan-power-d_632.html. [Accessed: 18- Sep- 2016].

[13] "Pipe Roughness Data", *Efunda.com*, 2016. [Online]. Available: http://www.efunda.com/formulae/fluids/roughness.cfm. [Accessed: 19- Sep- 2016]."Power Gained by Fluid from a Pump or a Fan", *Engineeringtoolbox.com*, 2016. [Online]. Available: http://www.engineeringtoolbox.com/pump-fan-power-d_632.html. [Accessed: 18- Sep- 2016].

[14] "18.5 Heat Exchangers", *Web.mit.edu*, 2016. [Online]. Available: http://web.mit.edu/16.unified/www/SPRING/propulsion/notes/node131.html. [Accessed: 19- Sep- 2016].

Section 9: Academic Integrity Statement

This report was created by the group members listed below in compliance with Case Western Reserve University's Academic Integrity policy available at: https://students.case.edu/community/conduct/aiboard/policy.html

By signing below, each group member is giving their affirmation to the above statement.

Name:	Signature:
William Koehrsen______________	______________________________
Tyler Eston____________________	______________________________
Arturo Garcia___________________	______________________________
Theodore Bastian_________________	______________________________

Appendices

I: Data Tables

Size	ID_a	ID_p	OD_p	A_p	A_a	D_h	D_e
2 x1 1/4	0.05102	0.03279	0.03493	0.0008444	0.001086	0.01609	0.03959
2 1/2 x 1 1/4	0.06338	0.03279	0.03493	0.0008444	0.002196	0.02845	0.08007
3 x 2	0.07572	0.05102	0.05398	0.002044	0.002214	0.02174	0.05223
4 x 3	0.09998	0.07572	0.07938	0.004503	0.002901	0.0206	0.04654

Table 7: Pipe Properties**Error! Not a valid link.**

Substance		ṁ	Ti	Tf	ρ	cp	k	ν	Pr	α	μ
Water	Initial	3.95	70.00	68.08	980.55	4187.3	0.659	4.42E-07	2.753	1.61E-07	4.33E-04
	After One Year	4.75	70.00	68.40	980.55	4187.3	0.659	4.42E-07	2.753	1.61E-07	4.33E-04
Benzene		1.4	27.00	40	878.6	1744.76	0.143	6.84E-07	7.05	9.10E-08	6.01E-04

Table 8: Fluid Properties

		ṁ					
Benzene		1.4	V	Re for heat transfer	Re for friction	Nu	h, conv
2 x1 1/4			1.887	90463.61	90463.61	463.64	2021.98
2 1/2 x 1 1/4			1.887	90463.61	90463.61	463.64	2021.98
3 x 2			0.780	58148.74	58148.74	325.56	912.50
4 x 3			0.354	39173.23	39173.23	237.35	448.25
Water		3.95					
2 x1 1/4			3.709	332246.76	135030.32	814.40	13556.21
2 1/2 x 1 1/4			1.834	332309.36	118074.20	814.52	6703.77
3 x 2			1.819	215004.47	89492.58	574.95	7254.26
4 x 3			1.389	146212.24	64717.92	422.33	5980.20

Table 9: Initial Fluid Calculations

		ṁ					
Benzene		1.4	V	Re for heat transfer	Re for friction	Nu	h, conv
2 x1 1/4			1.887	90463.61	90463.61	463.64	2021.98
2 1/2 x 1 1/4			1.887	90463.61	90463.61	463.64	2021.98
3 x 2			0.780	58148.74	58148.74	325.56	912.50
4 x 3			0.354	39173.23	39173.23	237.35	448.25
Water		4.75					
2 x1 1/4			4.461	399537.25	162378.23	943.88	15711.42
2 1/2 x 1 1/4			2.206	399612.52	141987.96	944.02	7769.56
3 x 2			2.188	258549.68	107617.65	666.35	8407.57
4 x 3			1.670	175824.85	77825.35	489.48	6930.95

Table 10: Fluid Calculations after One Year

	$\dot{Q}$ needed	$\dot{Q}$ achieved initial	$\dot{Q}$ achieved after one year
Benzene	31754.632	39982	31990
Water	-31754.632	-39982	-31990

Table 11: Heat Balance

	Initial	After One Year
	ΔT_{LMTD}	ΔT_{LMTD}
Crossflow	35.40	35.25
Parallel Flow	35.20	35.01

Table 12: ΔT_{LMTD}

	Before Fouling	Before Fouling	After fouling	After Fouling
	U_oa,i	U_oa,i	U_oa,i	U_oa,i
Pipe Sizes	Stainless Steel	Copper	Stainless Steel	Copper
2 x1 1/4	1590.85	1765.55	1251.41	1357.04
2 1/2 x 1 1/4	1429.84	1569.42	1162.56	1253.18
3 x 2	759.82	813.16	676.49	718.44
4 x 3	399.66	417.56	375.91	391.70

Table 13: Overall Heat Transfer Coefficient

	Copper	Copper	Stainless Steel	Stainless Steel
	Crossflow	Parallel Flow	Crossflow	Parallel Flow
Pipe Size	Minimum Length (m)	Minimum Length (m)	Minimum Length (m)	Minimum Length (m)
2 x1 1/4	4.953	4.987	5.497	5.534
2 1/2 x 1 1/4	5.572	5.610	6.116	6.158
3 x 2	6.912	6.959	7.397	7.447
4 x 3	9.069	9.131	9.475	9.540

Table 14: Minimum Length of Pipe Initially

	Copper	Stainless Steel
	Crossflow	Crossflow
Pipe Size	Minimum Length (m)	Minimum Length (m)
2 x1 1/4	6.418	6.959
2 1/2 x 1 1/4	6.949	7.491
3 x 2	7.791	8.274
4 x 3	9.628	10.032

Table 15: Minimum Length of Pipe after One Year

	Relative Roughness		Friction Factor		Friction Factor	
	Copper	Stainless Steel	Copper	Copper	Stainless Steel	Stainless Steel
Pipe Sizes			Benzene	Water	Benzene	Water
2 x1 1/4	9.32E-05	1.24E-04	0.021	0.023	0.022	0.024
2 1/2 x 1 1/4	5.27E-05	7.03E-05	0.021	0.023	0.022	0.024
3 x 2	6.9E-05	9.20E-05	0.02	0.025	0.021	0.026
4 x 3	7.28E-05	9.71E-05	0.021	0.027	0.022	0.028

Table 16: Darcy Friction Factors

	Initially	Initially		After One Year	
	Crossflow	Crossflow		Crossflow	Crossflow
	Copper	Copper		Copper	Copper
	Water	Benzene		Water	Benzene
Pipe Sizes	ΔP	ΔP		ΔP	ΔP
2 x1 1/4	67500.1	7013.2		97610.8	7013.2
2 1/2 x 1 1/4	9336.3	7013.2		13501.0	7013.2
3 x 2	13065.3	732.6		18893.4	732.6
4 x 3	8673.5	106.8		12542.6	106.8

Table 17: Pressure Loss (Without Minor Losses)

In Watts	Initial	Initial		Final	Final
Pipe: 2.5" X 1.25"	Water	Benzene		Water	Benzene
Power Required (no losses)	37.6	11.2		65.4	11.2
Power Required (minor Losses)	63.8	14.2		91.6	14.2

Table 18: Pump Power Requirements

II: Sample Calculations for Final Design

Included are the calculations for the final design. Some of the numbers may be in slight disagreement with the data tables provided. Rounding errors in the calculations presented below are to blame for any discrepancy. For the final design, numbers calculated in Excel were assumed to be correct and were the figures used in the final design specifications.

Heat rate into Benzene:

$$\dot{Q}_B = 1.4 * 1744.76(40 - 27) = 31754 \text{ W}$$

Heat rate from Water:

$$\dot{Q}_B = -31754 \text{ W}$$

Benzene fluid velocity:

$$V = \frac{1.4}{878.6 * 0.0008444} = 1.877 \frac{m}{s}$$

Water Initial Fluid Velocity:

$$V = \frac{3.95}{980.55 * 0.002196} = 1.834\ m/s$$

The water Reynolds Number for heat transfer is:

$$Re = \frac{1.834 * \ 0.08007}{4.42 * 10^{-7}} = 332{,}309$$

The benzene Reynolds Number is:

$$Re = \frac{1.877 * \ 0.03279}{6.84 * 10^{-7}} = 90463$$

Water final temperature is:

$$T_{W,F} = 70 + \frac{-31754}{3.95 * 4187.3} = 68.08\ ℃$$

The Log Mean Temperature Difference is:

$$\Delta T_{LMTD} = \frac{(70 - 40\ ) - (68.08 - 27)}{\ln[\frac{(70 - 40)}{(68.08 - 27)}]} = 35.2503\ °C$$

The Nusselt Number for water:

$$Nu = 0.023 * 332309^{0.8} * 2.4753^{0.3}=814.5$$

The Nusselt Number for Benzene:

$$Nu = 0.023 * 90463^{0.8} * 7.05^{0.4} = 463.64$$

The Convective Heat transfer coefficient for water:

$$h_{conv,W} = \frac{0.659 * 814.5}{0.08007} = 6703.7 \frac{W}{m^2K}$$

The Convective Heat transfer for benzene

$$h_{conv,B} = \frac{0.143 * 463.64}{0.03279} = 2022 \frac{W}{m^2K}$$

The overall initial heat transfer coefficient of the inner surface:

$$U_{OA,i} = \frac{1}{\frac{1}{2022} + \frac{0.016395}{0.659} \ln\left(\frac{0.017465}{0.016395}\right) + \frac{0.016395}{0.017465} * \frac{1}{6703.7}} = 1569.5 \frac{W}{m^2K}$$

The computed minimum needed length for the heat exchanger is:

$$L = \frac{31754}{1429.8 * \pi * 0.03279 * 35.25} = 5.57\, m$$

The overall heat transfer coefficient after fouling:

$$U_{OA,i,f} = \frac{1}{(0.00018 + 0.0009) + \frac{1}{1570}} = 1102 \frac{W}{m^2k}$$

Minimum length accounting for fouling:

$$L = \frac{31754}{1102 * \pi * 0.03279 * 35.4} = 7.85\, m$$

Pressure loss in water (outer) pipe before fouling:

$$\Delta P = 0.023 * \frac{8.0}{0.03279} * \frac{1}{2} * 980.55 * 1.834^2 + 5400 = 16070\, Pa$$

Pressure loss in benzene (inner) pipe before fouling:

$$\Delta P = 0.021 * \frac{5.57}{0.02845} * \frac{1}{2} * 878.6 * 1.877^{2} + 1790 = 9805\, Pa$$

Pressure loss of water after fouling:

$$\Delta P = 0.023 * \frac{8}{0.03279} * \frac{1}{2} * 980.55 * 1.86^2 + 5400 = 16342\ Pa$$

Pressure loss of benzene after fouling:

$$\Delta P = 0.021 * \frac{8}{0.02845} * \frac{1}{2} * 878.6 * 1.877^2 + 1790 = 9805\ Pa$$

Volumetric flow rate for water:

$$\dot{V} = \frac{3.95}{980.55} = 0.004844 \frac{m^3}{s}$$

Volumetric flow rate for benzene:

$$\dot{V} = \frac{1.4}{878.6} = 0.00159\ \frac{m^3}{s}$$

Power of the water pump accounting for fouling and minor losses is:

$$Power = 16342 * 0.00403 = 79.2\ W$$

Power of the benzene pump accounting for fouling and minor losses is:

$$Power = 9805 * 0.00159 = 15.6\ W$$