Heat Exchanger Senior Design

**Thermal Systems**

**Diana Tiburcio, Jakob Werle, Yireh Byas, Selinez Soto**

**December 16th, 2023**

A red and white logo

Description automatically generated

Table of Contents

[Abstract 4](#_Toc153663473)

[Introduction 4](#_Toc153663474)

[Problem Statement and Limitations 4](#_Toc153663475)

[Figure 1: Section view of the concentric heat exchanger design hot inner pipe and cold outer pipe 5](#_Toc153663476)

[Engineering Design 5](#_Toc153663477)

[Detailed Analysis 5](#_Toc153663478)

[Hand Calculations 5](#_Toc153663479)

[Sample Calculation 10](#_Toc153663480)

[Analysis of Calculation Results 15](#_Toc153663481)

[Figure 2: Theoretical hot and cold output temperatures of both counter and parallel flows with varying heat exchanger lengths. ​ 15](#_Toc153663482)

[Figure 3: Theoretical hot and cold output temperatures of both counter and parallel flows with varying cold flowrates with the hot flowrate remaining at 0.1 L/s. 15](#_Toc153663483)

[Figure 4: Theoretical hot and cold output temperatures of both counter and parallel flows with varying hot flowrates with the cold flowrate remaining at 0.8 L/s. 16](#_Toc153663484)

[SolidWorks Simulations 16](#_Toc153663485)

[Figure 5: Set-Up of the SolidWorks Simulation Iteration 1 17](#_Toc153663486)

[Figure 6: Simple Solidworks Flow Simulation on Iteration 1 17](#_Toc153663487)

[Figure 7: Setup of the SolidWorks Simulation Iteration 2 18](#_Toc153663488)

[Figure 8: Solidworks Flow Simulation on Iteration 2 18](#_Toc153663489)

[Figure 9: Solidworks Flow Simulation on Iteration 3 19](#_Toc153663490)

[Table 1: Flow Rates Used in Testing the Heat Exchanger System 19](#_Toc153663491)

[Fabrication Methods 20](#_Toc153663492)

[Table 2: Bill of Materials 20](#_Toc153663493)

[Figure 10: Exploded View of Heat Exchanger Components 21](#_Toc153663494)

[Figure 11: Section View of Sensor Collar - Sensor Offset 22](#_Toc153663495)

[Figure 12: Pressure Testing on Sensor Collar 23](#_Toc153663496)

[Figure 13: Overall Dimensions of Final Heat Exchanger Design 23](#_Toc153663497)

[Figure 14: 3D Rendering of Final Heat Exchanger Design 24](#_Toc153663498)

[Figure 15: Fully Assembled Heat Exchanger 24](#_Toc153663499)

[Results 24](#_Toc153663500)

[SolidWorks Results 24](#_Toc153663501)

[Figure 16: Flowlines from SolidWorks Flow Simulation 25](#_Toc153663502)

[Figure 17: Detail View of Flow Entering 90° Tee 25](#_Toc153663503)

[Experimental Results 25](#_Toc153663504)

[Testing Procedure 25](#_Toc153663505)

[Figure 18: Heat Exchanger Testing Apparatus 26](#_Toc153663506)

[Table 3: Cold Inlet Temperatures Measured by Outlet Thermocouple for Parallel and Counter Flow 26](#_Toc153663507)

[Table 4: Parallel Flow Experimental Results 27](#_Toc153663508)

[Table 5: Counter Flow Experimental Results 28](#_Toc153663509)

[Table 6: Summary of Parallel Flow Results 28](#_Toc153663510)

[Table 7: Summary of Parallel Flow Results 29](#_Toc153663511)

[Figure 19: Comparison of Theory, Simulation, and Experimental Results for Parallel Flow 29](#_Toc153663512)

[Figure 20: Comparison of Theory, Simulation, and Experimental Results for Counter Flow 30](#_Toc153663513)

[Conclusion 30](#_Toc153663514)

[References 31](#_Toc153663515)

[Appendix 31](#_Toc153663516)

# Abstract

This design report presents a comprehensive analysis of a double pipe heat exchanger, examining both parallel and counterflow configurations with water as the primary fluid and copper as the chosen material. The design parameters specified that the inlet temperatures for the cold and hot fluids would fall within the ranges of 18-20 degrees Celsius and 60-70 degrees Celsius, respectively. Additionally, the diameters of the pipes were set at 1 and 1.5 inches, with flow rates intentionally kept below 1 liter/s to ensure an optimal temperature outlet difference. The maximum allowable length for the heat exchanger was constrained to 1 meter. The report delves into key aspects of the design, with a primary focus on analyzing the regime of flow rates, determining the overall heat transfer coefficient, and correlating heat transfer rates to predict outlet temperatures for both cold and hot fluids in parallel and counterflow configurations. Theoretical calculations serve as the foundation for this exploration, providing a baseline for comparison with SolidWorks simulations and experimental results. This triad of methodologies facilitates a robust comprehension of the designed heat exchanger, allowing for evaluation of its performance under various conditions. It was determined that maximizing the length of the heat exchanger, maximizing the flowrate of the cold water, and minimizing the flowrate of the hot water would deem optimal results. The iterative nature of the project, incorporating theoretical insights, simulation validations, and experimental adjustments, underscores the commitment to achieving accuracy and efficiency in the pursuit of an optimized heat exchanger design.

# Introduction

The primary goal of this project is to design a double pipe heat exchanger featuring both parallel and counter flow configurations. The investigation focuses on understanding how changes in mass flow rates and heat exchanger length influence the overall heat transfer coefficient. Striving for practicality, the aim is to achieve an efficient heat exchanger design within specified limitations, ensuring optimal cooling for the hot water. Heat exchangers serve as devices facilitating the exchange of thermal energy between two fluids with different temperatures. These fluids, whether liquids or gases, are separated by solid boundaries. The versatility of heat exchangers finds application in various sectors, including space heating, chemical processing, and power production. In essence, this project investigates how to effectively design a concentric heat exchanger by applying mathematical modeling of the system and software simulations and how to analyze the efficiency of a system based off experimental results.

# Problem Statement and Limitations

The engineering project involves the design and evaluation of a double pipe heat exchanger capable of operating in both parallel flow and counterflow configurations. The project focuses on using water as the working fluid and copper as the material for construction. Inlet temperatures are specified, ranging from 18 to 20 ºC for cold water and 60 to 70 ºC for hot water. The inner tube diameters are set at 1 and 1.5 inches, with hot water flowing through the inner tube. Flow rates, constrained to be less than 1 liter/s, are chosen to ensure a reasonable temperature difference for both fluids, and the maximum allowable length of the heat exchanger is 1 meter.

The project objectives include evaluating the properties of both cold and hot water, determining the flow regimes, utilizing appropriate correlations to find heat transfer coefficients, calculating the overall heat transfer coefficient (U), evaluating effectiveness, determining outlet temperatures, assessing pressure drop, and calculating head loss throughout the system.

The investigation involves studying the impact of mass flow rates and the length of the heat exchanger on heat transfer and outlet temperatures under known inlet conditions. Additionally, the project entails performing SolidWorks simulations to complement and validate the results obtained through calculations. The combination of theoretical calculations, empirical correlations, and simulation results will provide a comprehensive understanding of the heat exchanger's performance and guide the design process for optimal efficiency.

A diagram of a pipe with a clock and a temperature chart

Description automatically generated

Figure : Section view of the concentric heat exchanger design hot inner pipe and cold outer pipe

# Engineering Design

## Detailed Analysis

To initiate the heat exchanger design process, we conducted calculations aimed at predicting the final water temperatures, considering parameters such as heat exchanger length, and flow rates of both cold and hot water. Additionally, to gain deeper insights into the impact of controllable factors on system efficiency, we computed parameters such as head loss, pressure differentials, thermal entrance lengths, and flow entrance lengths for both the hot and cold compartments. To streamline this process and facilitate rapid calculations, we developed a MATLAB code. Subsequently, we utilized this code to assess temperature differentials across the system under varying lengths and flow rates. The results were then compared to identify the optimal heat exchanger configuration.

## Hand Calculations

To analyze the heat exchange between a hot and cold fluid in a concentric tube design, the Reynolds Number must be determined before continuing with the calculations. The Reynolds number (ReD) is a dimensionless quantity used to characterize the flow regime in the heat exchanger tubes. It is calculated using the formula:

|  |  |  |
| --- | --- | --- |
|  |  | () |

Where ρ is the fluid density, v is the velocity of the fluid, D is the hydraulic diameter, and μ is the dynamic viscosity. The hydraulic diameter (Dh) is determined by the cross-sectional area (Ac) of the tube and its perimeter (P).

|  |  |  |
| --- | --- | --- |
|  |  | () |
|  |  | () |

The diameters were determined from the McMaster-Carr website from which the pipes were ordered. Additionally, the diameters of the pipes were verified by measuring them using vernier calipers. The velocity (v) is defined as the mass flow rate (ṁ) divided by the product of fluid density (ρ) and the cross-sectional area (Ac).

|  |  |  |
| --- | --- | --- |
|  |  | () |

The Reynolds number is essential for understanding the flow patterns inside the heat exchanger tubes. Using this parameter, the Nusselt number (NuD) and friction factor (f) can be calculated which are crucial for determining convective heat transfer and frictional losses within the heat exchanger. NuD is calculated using the Gnielinski correlation, incorporating the Reynolds number (ReD) and Prandtl number (Pr).

|  |  |  |
| --- | --- | --- |
|  |  | () |

The friction factor (f) is calculated using the Darcy-Weisbach equation and is vital for assessing pressure drop and energy losses in the system.

|  |  |  |
| --- | --- | --- |
|  |  | () |

Where the ε is the pipe roughness coefficient which was determined to be m for copper. The equations for NuD and f provide insights into heat transfer and fluid flow characteristics. From the previously calculated Nusselt number (NuD), thermal conductivity (k), and hydraulic diameter (D), the convective heat transfer coefficient (h) can be determined.

|  |  |  |
| --- | --- | --- |
|  |  | () |

The heat transfer coefficient represents the effectiveness of heat transfer throughout the fluid due to convection. This value can then be used to determine the overall heat transfer coefficient within the heat exchanger. The overall heat transfer coefficient (U) accounts for both convection and conduction throughout the hot and cold water and across the copper pipes. It is calculated using the reciprocals of the individual convective heat transfer coefficients and the thermal resistance of the copper pipes.

|  |  |  |
| --- | --- | --- |
|  |  | () |

The specific heat (C) and heat capacity ratios (Cr) are essential for determining the thermal capacity of the hot and cold water. The hot water had the lower heat capacity and the hot water’s specific heat was used as Cmin and while the cold-water heat capacity was identified as Cmax. These values were used to determine the heat capacity ratio.

|  |  |  |
| --- | --- | --- |
|  |  | () |

|  |  |  |
| --- | --- | --- |
|  |  | () |

The number of transfer units (NTU) can now be calculated as the product of overall heat transfer (U), the area, and the reciprocal of the minimum heat capacity (Cmin).

|  |  |  |
| --- | --- | --- |
|  |  | () |

Where the area (Ao) is the surface area of the entire heat exchanger which means the out diameter of the outermost copper pipe was used is the following equation:

|  |  |  |
| --- | --- | --- |
|  |  | () |

The heat exchanger effectiveness (ε) is determined based on the flow configuration (parallel or counterflow) and the number of transfer units (NTU). The specific correlations between the effectiveness and flow configurations were found on Table 11.3 in the Fundamentals of Heat and Mass Transfer book (Bergman, 2017). For parallel flow, is calculated using the following equation:

|  |  |  |
| --- | --- | --- |
|  |  | () |

While for counterflow, uses:

|  |  |  |
| --- | --- | --- |
|  |  | () |

These equations provide insights into the efficiency of heat transfer in different flow configurations. The maximum possible heat transfer rate (qmax) is determined by the minimum heat capacity (Cmin) and the temperature difference between the hot and cold fluids.

|  |  |  |
| --- | --- | --- |
|  |  | () |

The actual heat transfer rate (q) is then calculated using the effectiveness (ε) and the maximum possible heat transfer rate.

|  |  |  |
| --- | --- | --- |
|  |  | () |

Rearranging the following the heat transfer rate equation:

|  |  |  |
| --- | --- | --- |
|  |  | () |

In addition to utilizing the specific heat resulted in the following equation:

|  |  |  |
| --- | --- | --- |
|  |  | () |

Which was used to calculate the outlet temperatures of both cold and hot water for parallel and counter flow configurations.

The head loss equations account for major and minor losses in the fluid flow, including contraction, expansion, and fittings. The head loss is critical for assessing the energy losses in the heat exchanger, which will in turn influence the system’s efficiency. Various factors, such as velocity, diameter, and geometric parameters, contribute to the head loss calculations.

The major head loss (hl,major) quantifies the pressure drop due to viscous friction in the main section of the fluid flow. It is calculated using the Darcy-Weisbach equation, which considers the friction factor (f), pipe length (L), hydraulic diameter (D), fluid velocity (v), and gravitational acceleration (g).

|  |  |  |
| --- | --- | --- |
|  |  | () |

This term is crucial for assessing the impact of friction on fluid flow within the heat exchanger tubes. The minor head loss equation, hl,minor accounts for pressure losses associated with additional components such as bends, valves, and fittings. It is calculated using the local loss coefficient (KL), fluid velocity (v), and gravitational acceleration (g).

|  |  |  |
| --- | --- | --- |
|  |  | () |

This equation provides insight into the contribution of minor losses to the overall pressure drop in the heat exchanger system. The contraction head loss equation, hl,contraction, represents the pressure drop resulting from a sudden reduction in pipe diameter. The contraction coefficient (0.44) is applied to account for the energy loss during this process.

|  |  |  |
| --- | --- | --- |
|  |  | () |

Understanding this term is vital for evaluating the impact of geometric changes on fluid flow within the heat exchanger. Similarly, the expansion head loss equation, hl,expansion, calculates the pressure drop due to a sudden increase in pipe diameter. It considers the velocity difference (v2 – v1) across the expansion region.

|  |  |  |
| --- | --- | --- |
|  |  | () |

Lastly, the ball valve and the 90̊ turns have constant local loss coefficients for those specific components in the heat exchanger system. Kball (0.05) represents the loss coefficient for a ball valve, while K90 (0.3) represents the loss coefficient for a 90-degree bend. These coefficients are used in the hl,minor equation to quantify the impact of these components on overall pressure losses.

In the selected heat exchanger design, the piping through which the cold-water passes has two ball valves, two 8” copper pipes, two 90̊ turns, one contraction, one expansion, and a 39” copper pipe where the heat exchange occurs. The copper pipe sections inflict major head loss while all other components cause minor head loss. The total cold-side head loss equation, hl,cold, combines various components' contributions to assess the overall pressure drop in the cold fluid flow path.

|  |  |  |
| --- | --- | --- |
|  |  | () |

However, the internal pipe that contains the hot water only has one ball valve and a 57” copper pipe. The equation hl,hot calculates the total head loss in the hot fluid flow path by combining contributions from major losses in a 57-inch length of pipe and minor losses associated with a ball valve. The major loss term considers frictional losses occurring along the specified length of the pipe, considering factors such as length, diameter, velocity, and gravitational acceleration. Meanwhile, the minor loss term quantifies the additional head loss introduced by the presence of a ball valve in the hot fluid pathway.

|  |  |  |
| --- | --- | --- |
|  |  | () |

The pressure drop occurring in the fluid flow and is calculated as a function of the Darcy-Weisbach friction factor (f), fluid density (ρ), mean velocity (um), hydraulic diameter (Dh), and length of the pipe (L).

|  |  |  |
| --- | --- | --- |
|  |  | () |

This expression captures the impact of various parameters on the pressure loss, providing a quantitative measure of the resistance encountered by the fluid as it flows through the system. The result, ∆P, is crucial for assessing the energy requirements and efficiency of the piping network.

The equation for turbulent flow entrance length, Le, is vital for determining the distance required for the flow to transition into fully developed turbulent conditions. It is calculated using a correlation involving the Reynolds number (ReD) and the hydraulic diameter (Dh).

|  |  |  |
| --- | --- | --- |
|  |  | () |

This length is instrumental in understanding when the fluid flow achieves a stable, fully turbulent profile, which is essential for accurate heat transfer and pressure drop predictions within the heat exchanger. The turbulent thermal entrance length, Lth, is determined by the hydraulic diameter (Dh) and provides the distance over which the temperature profile stabilizes as the fluid enters the heat exchanger.

|  |  |  |
| --- | --- | --- |
|  |  | () |

This length is critical for predicting when the thermal conditions within the heat exchanger reach a fully developed turbulent state. A precise understanding of the turbulent thermal entrance length is essential for accurate thermal analysis and optimal heat exchange efficiency.

## Sample Calculation

All the constants used throughout the calculations were determined using linear interpolation and the two initial hot and cold temperatures.

**Hot Constants**

* Hot flow rate: m = 0.1 L/s
* Hot temperature: Th = 65 ℃
* k = 0.6585
* Pr = 2.7414
* = 4187.3
* μ = 0.000432
* ρ = 979.5
* m
* m

**Hot Calculations**

**Cold Constants**

* Cold flow rate: m = 0.8 L/s
* Cold temperature: Tc = 19 ℃
* k = 0.6014
* Pr = 7.1558
* = 4182.7
* μ = 0.00103
* ρ = 998.0
* m
* m

**Cold Calculations**

**Combined Calculations for Temperature**

**Parallel Flow**

**Counter Flow**

**Other Calculations for Hot:**

Pressure

Entrance Lengths

Head loss

**Other Calculations for Cold:**

Pressure

Entrance Lengths

Head loss

## Analysis of Calculation Results

Based on the calculations, a MATLAB code was created and utilized to efficiently determine the resulting output temperatures at different heat exchanger lengths in addition to the cold and hot flower rates. These values were compiled into digestible graphs depicted below to aid in the determination of optimal heat exchanger design and optimal operational parameters. Initially, the influence of the length of the heat exchanger was analyzed since that was the most influential and controllable aspect of the system.

Figure : Theoretical hot and cold output temperatures of both counter and parallel flows with varying heat exchanger lengths. ​

It can be observed in Figure 2, a longer heat exchanger will result in a larger temperature difference within the fluids. Thus, the theoretical results finalized the decision to make the heat exchanger a meter long to remain within the constraints of the project while still achieving the maximum heat transfer.

Figure : Theoretical hot and cold output temperatures of both counter and parallel flows with varying cold flowrates with the hot flowrate remaining at 0.1 L/s.

Another controllable aspect of the heat exchanger was the flowrates at which the cold and hot water were flowing through the system. The influence of the cold-water flowrate was depicted in Figure 3, where the hot water flowrate used in the theoretical calculations was maintained at 0.1 L/s while the cold flow rate was changed. It was determined that the faster the cold water flowed through the heat exchanger, the larger the temperature difference within both cold and hot water. This is because the cold water is not given sufficient time to warm up and new, cold water is frequently entering the system. Utilizing these theoretical results, when the experimental trials were conducted, the highest possible flowrate that was achievable while keeping it consistent was applied for the cold water.

Figure : Theoretical hot and cold output temperatures of both counter and parallel flows with varying hot flowrates with the cold flowrate remaining at 0.8 L/s.

Similarly, the effect of the hot water flowrate was studied theoretically, and it was determined that applying a slower flow rate for the hot water deemed the largest temperature differences (Figure 4). However, it is important to note that the variation in output temperatures was much less than when the cold flowrate was varied by similar increments. The increased temperature difference when reducing the hot flowrate is due to the hot water being exposed to the cold water for longer due to the increased hydraulic retention time. The reduced influence of the hot water flowrate can be attributed to the cold water having a much larger influence on the heat transfers occurring throughout the system.

## SolidWorks Simulations

For this project, a SolidWorks flow simulation was used to verify MATLAB calculations and visualize the effects of changes applied to the heat exchanger system. To begin building the simulation, models of the various pipes and fittings were modeled or imported from McMaster-Carr and assembled. Next, a flow simulation study was created, and the system parameters were specified. All parameters were kept constant except for the mass flow rates of the hot and cold pipes, which were the variables that defined each iteration. The heat exchanger was designed to support parallel flow or counter flow by making a simple change to the inlet and outlet settings.

In the beginning stages of the project, the system, shown in Figure 5, was modeled extremely simply to gain basic preliminary results and some of the values were found to be invalid. For this original design, the length was chosen to be 18in long with a hot inlet temperature of 65°C flowing through the outer pipe, and a cold inlet temperature of 19°C flowing through the inner pipe. The mass flow rates of the hot and cold pipes were 0.75kg/s and 0.8kg/s, respectively. This system is shown in Figure , while the flow through the system is shown in Figure 6.

A drawing of a pipe

Description automatically generated

Figure : Set-Up of the SolidWorks Simulation Iteration 1

A screenshot of a computer

Description automatically generated

Figure : Simple Solidworks Flow Simulation on Iteration 1

The conclusions drawn from this simulation were that the cold pipe temperature increased to about 20.75°C, while the hot pipe decreased to about 63.8°C. It was decided that changes would be necessary in the next iteration. These included increasing the temperature difference within each pipe by trying slower mass flow rates, in addition to aiming for more accurate results through increasing mesh resolution, and adding more accurate coupling models into the simulation, by importing them from McMaster-Carr.

Since iteration 1 revealed certain design changes that needed to be made, these were considered for iteration 2. The mesh resolution was increased for better accuracy of results, and the couplings from McMaster-Carr were imported into the model, as shown in Figure . Additionally, the configuration of flows was changed, and the hot water was run through the inner tube, while the cold water was run through the outer tube. This is shown in Figure .

A pipe with a long pipe

Description automatically generated with medium confidence

Figure : Setup of the SolidWorks Simulation Iteration 2

A screen shot of a video game

Description automatically generated

Figure : Solidworks Flow Simulation on Iteration 2

This simulation resulted in larger changes in temperature for the hot and cold water, in addition to closer results to the theoretical values. This ultimately demonstrated that the project was on the right track, and that this configuration of inner and out pipe flows was more desirable.  The iteration 3 design is the same as in iteration 2 which was shown in Figure .

A red and black line

Description automatically generated with medium confidence

Figure : Solidworks Flow Simulation on Iteration 3

For iteration 3, the heat exchanger was chosen to be 1 meter long with a hot inlet temperature of 65°C, and a cold inlet temperature of roughly 11°C. Since the cold water for the system was provided by a hose with pipes that extend out of the building, its temperature was affected by the day-to-day temperature changes outside. It was lower than the desired 19°C and, on average, read values around 11°C at the inlet cold temperature probe. To aid in the accuracy of the simulation results though, the inlet cold testing temperatures, shown in Table 3, were used to set up the simulation values for the cold water flow.

Table : Flow Rates Used in Testing the Heat Exchanger System

A table with text and numbers

Description automatically generated

As stated previously, each test was defined by changes in the mass flow rates of the hot and cold water coming into the heat exchanger. For the first test the system was set up with the cold water flowing at 0.8kg/s within the outer pipe and the hot water flowing at 0.1kg/s within the inner pipe. After obtaining data on this combination of flow rates, other testing iterations with varying values were examined.

The SolidWorks simulation data was used to gain an understanding of the system and verify that the project was on target. This can be seen in the adjustments and design changes that were made for each iteration. In the beginning of the project, the simulation was roughly built, and the hot and cold flows were not being run in their ideal configurations. As more of the system was understood through the software and in comparing the simulation data with theoretical data, many changes were made to get the system to make it as optimal as possible. After getting the simulation to a point where it was able to validate the theoretical numbers, it could finally be used as a comparison for the experimental data. This will be seen in the Experimental Results section.

## Fabrication Methods

#### Bill of Materials

Table : Bill of Materials

 The cost breakdown of the heat exchanger project is shown in Table 2. The college of engineering provided all copper tubing, sensors, and test bench mounting hardware. With the school’s contribution of material, the actual cost for the design team was $227.71. However, the total cost of building a heat exchanger would be $336.93.

Notable choices that helped reduce the overall cost of the project was selecting inexpensive sensor mounting material and deciding to use only 3 valves.

Diagram of a machine parts

Description automatically generated with medium confidence

Figure : Exploded View of Heat Exchanger Components

Figure 10 shows an exploded view of the final assembly with item numbers detailed. This view also served as an assembly check to ensure all parts could be fit together prior to purchasing anything.

#### Component Choices

The choice for components was based on constraints of meeting design requirements, cost, and manufacturability. Certain parts such as the heat exchanger tubing was predetermined by the project guidelines so adjacent hardware was chosen to work with it. The typical method of joining copper tubing is by soldering, so all copper fittings and hardware were narrowed to those can be soldered.

To meet the design requirement of having parallel and counterflow, two valves were chosen to control the hot flow, and one valve was chosen to control cold flow. Having 3 valves allowed all configurations of flow to be achieved. The choice of valve type mainly came down to price and adjustability. It was determined that using a ball valve would generate sufficient control of flow rate while only costing ~$28 when compared to higher precision globe valves which costed upwards of 3x as much.

To record pressure and temperature data from the testing apparatus, one pressure line and one thermocouple had to be mounted at the inlet and outlet of each flow. Both sensors were preselected for the project so designing a mounting method that captured accurate results was the main consideration. As seen in Figure ,to ensure that fluid flow would not be impeded and subsequently effect data, the thermocouple probe was offset such that the tip barely entered the stream. In efforts to further isolate the thermocouple from temperature inaccuracies due to conduction through its mounting location, a material with a low heat transfer coefficient was sought out. The final choice was high density polyurethane (HDPE) as its heat transfer coefficient was ~0.4 W/m\*K. This material was also chosen for being relatively low cost when compared to copper, at $37.25/ft vs. $251.58/ft.

A drawing of a machine

Description automatically generated

Figure : Section View of Sensor Collar - Sensor Offset

#### Assembly Methods

Preparing a workflow before parts arrived allowed the team to efficiently execute the fabrication process. The process was broken down into the following steps:

1. Place part orders with machine shop advisors
2. Prepare for manufacturing
   1. Print part drawings
   2. Produce CAM toolpaths using Fusion 360
3. Confirm correct parts and quantities upon shipment arrival
4. Cut copper tubing using pipe cutter tool
5. Machine sensor collars using HAAS CNC mill
6. Dry fit tubing and joints to ensure proper fitment
7. Prepare all solder joints by scuffing and applying flux compound
8. Solder first set of joints using blowtorch
9. Press sensor collar onto assembly using hydraulic press
10. Solder remaining set of joints
11. Drill and tap sensor holes
12. Mount heat exchanger to test bench
    1. Secure assembly into mounting brackets
    2. Hook up water lines
    3. Screw in sensors

All parts were pressure tested during after manufacturing as shown below in Figure 12.

A white pipe with a yellow handle and a green pliers

Description automatically generated

Figure : Pressure Testing on Sensor Collar

#### Final Design

A drawing of a line

Description automatically generated

Figure : Overall Dimensions of Final Heat Exchanger Design

Dimensions of the final design are shown in Figure . This drawing was used during manufacturing to validate proper sizing.

A metal bar with a pipe

Description automatically generated with medium confidence

Figure : 3D Rendering of Final Heat Exchanger Design

Figure displays a SolidWorks rendering of the finalized heat exchanger design.

A brown bar on a grey surface

Description automatically generated

Figure : Fully Assembled Heat Exchanger

In Figure , the finished build of the heat exchanger is shown.

# Results

## SolidWorks Results

Figure 12 displays the parallel flow trajectories through the heat exchanger system. Both flows enter from the left side of the system and exit to the right. The cold flow through the outer pipe and the hot flow through the inner pipe are moving as expected. There are not areas of strangely moving flow and it can be seen that it moves uniformly throughout. Additionally, the hot water flowing in the inner pipe can be seen cooling down as it moves from left to right and its color changes from red to orange to yellow. This type of plot helps to verify that the simulation is working properly, and results are accurate.

Early on in the project, there were issues with vortices forming in the area where the pipes meet the shoulder fittings since there is changing geometry. A close up of the left shoulder fitting is shown in Figure and was included to show that no vortices or odd flow is occurring there after making adjustments. The final simulation demonstrated uniform and smooth flow there.

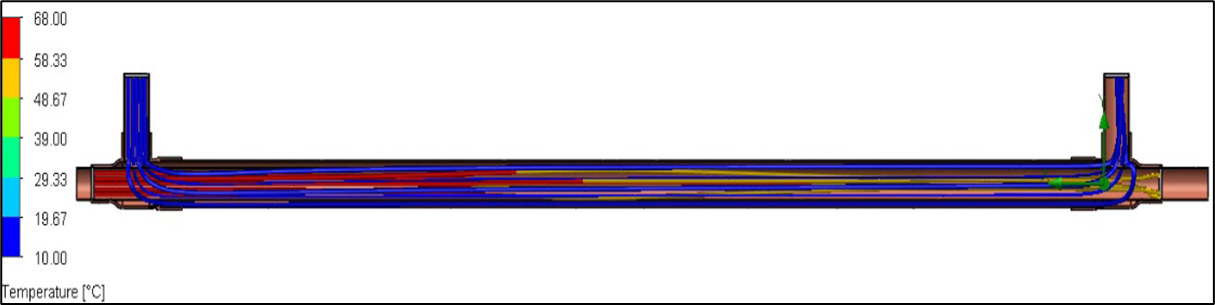


Figure : Flowlines from SolidWorks Flow Simulation

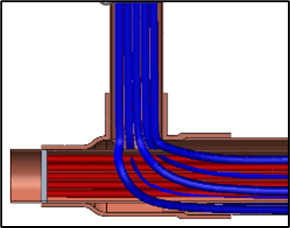


Figure : Detail View of Flow Entering 90° Tee

## Experimental Results

## Testing Procedure

To begin our testing process, allow the hot water in the tank to heat to approximately 65 degrees Celsius. When testing first began, the heating time would take around four hours with one heater. Once another heater was purchased, the heating took about two hours. While waiting for the water to heat to the desired temperature, the thermocouples are installed, and pressure lines are connected. Next, the water lines are attached to the heat exchanger, either in parallel or counterflow configuration. The cold water from the spigot is then opened and the hot water pump is turned on. Once both cold and hot waters are on, the flow rates are adjusted to the values shown in Table 1, for trial specification. Once the temperatures have stabilized, the hot and cold inlet and outlet temperatures are read, three times per trial. Through each trial, the hot water is allowed to reheat which takes about fifteen minutes.

A machine on a table

Description automatically generated

Figure : Heat Exchanger Testing Apparatus

Throughout the testing process, there were some complications that occurred. The thermocouples experienced some failures and calibration errors. To try and overcome these problems, an attempt was made to rebuild the thermocouples, and offset values were created to get as close to accurate readings as possible. Also due to this issue, the values for the cold water inlet had to be recorded with the cold water outlet thermocouple as there was only one probe on the cold side that could provide reasonable readings. This was done by running only the cold water through the system until the outlet temperature stabilized, and then taking that reading as the inlet temperature. After that, the hot water was turned on and the hot inlet and outlet, and cold outlet were recorded as normal. Another issue that occurred when testing was cavitation in the hot flow. As a result, the pressure readings for the hot flow were inaccurate and unreliable.

Table : Cold Inlet Temperatures Measured by Outlet Thermocouple for Parallel and Counter Flow

|  |  |
| --- | --- |
| A table with blue and white text  Description automatically generated | A screenshot of a computer  Description automatically generated |
|  |  |

These values in Table , were used to calculate the theoretical data and run the SolidWorks simulation more accurately after testing was completed. The table shows the “inlet” cold temperatures recorded by the outlet cold flow thermocouples. The reason for recording data in this manner was previously explained in the Testing Methods section. The cold inlet values, shown in Table were entered into the simulation as the cold water temperature, and the simulations were re-run, so that the values could align more closely with the adjusted theoretical data that also was adjusted to use these temperatures.

Table : Parallel Flow Experimental Results

A screenshot of a computer

Description automatically generated

Table contains the experimental data for the parallel flow data experimentations. The table shows how not only were the inlet and outlet temperatures taken, but so was the bath temperature. These temperatures were included to show what the initial hot inlet temperature should have been if the thermocouples were properly working. The table shows the three trials ran for each different flow rate. The average inlet and outlet temperature is recorded and the temperature changes for each set of flow rates for the parallel configuration.

Table : Counter Flow Experimental Results

A screenshot of a computer

Description automatically generated

Table 5 contains the experimental data for the counter flow data experimentations. As stated previously, the bath temperature is recorded to compare to the hot inlet temperature. Unlike with the parallel flow trials, the thermocouples were working better, so the inlet temperature reading was more accurate. Again, the average temperature is taken and the temperature change between the inlet and outlet temperatures are recorded for the counterflow configuration.

Table : Summary of Parallel Flow Results

A screenshot of a computer

Description automatically generated

In Table , the theoretical, SolidWorks, and experimental data is shown, as well as the percent errors for the theoretical and SolidWorks data compared to the experimental.

The SolidWorks percent error values are fairly high, reaching values of up to 31%. Focusing on the simulation and theoretical values, they are seen to be closer to each other than to the experimental data, which indicates that the experimental data is the one that is way off.

On the day of the parallel flow experimental testing, the thermocouples were reading extremely odd numbers, and even displaying different values if the wire was just moved slightly. Thus, the faulty thermocouples were the cause for the inaccuracies in the data recorded for parallel flow.

Table : Summary of Parallel Flow Results

A screenshot of a computer screen

Description automatically generated

The theoretical, SolidWorks, and Experimental data for the counter flow tests is shown in Table , as well as the percent errors for the theoretical and SolidWorks data compared to the experimental, again.

The SolidWorks data percent errors for this testing is much lower than that of the parallel testing data, resulting in values only as high as 7%. This is a reasonable range for this parameter since it is under 10%. The reason for this lower percent error trend is that the thermocouples on the day of the counter flow experimental testing were working well and providing good data.

A graph of different temperature levels

Description automatically generated with medium confidence

Figure : Comparison of Theory, Simulation, and Experimental Results for Parallel Flow

Figure 19 provides an overview of the results obtained through the 3 methods of determining outlet temperatures for hot and cold water using parallel flow.

A graph of different temperature levels

Description automatically generated with medium confidence

Figure : Comparison of Theory, Simulation, and Experimental Results for Counter Flow

Figure provides an overview of the results obtained through the 3 methods of determining outlet temperatures for hot and cold water using counter flow.

# Conclusion

The aim of this project was to design a double pipe heat exchanger that could support both parallel and counter flow configurations, while also providing the largest temperature change within the hot and cold flows running through it. Theoretical calculations were performed using MATLAB and hand calculations, and a SolidWorks simulation was created to verify these results. The most optimal design was chosen using both of these. A physical system was then fabricated and tested to acquire experimental results, that could be analyzed.

Based off the calculations that were thoroughly depicted in the report, digestible graphs that represented the influence that the heat exchanger length, cold and hot flowrates had on the efficiency of the system were analyzed. Based off Figure 2, it was determined to make the heat exchanger 1 meter long to maximize the temperature differences between the fluids. Furthermore, after analyzing Figures 3 and 4, it was decided to utilize the slowest possible hot flowrate and the fastest possible cold flowrate to optimize the heat exchanger. Based on these initial design selections, the selection of other required system components such as copper fittings, adapters, valves, and sensor sleeve were made. The final selection of components and manufacturing methods were chosen to meet the specific design criteria determined by theoretical analysis and simulations. The final cost of the heat exchanger project is $227. A fabrication process was established and executed to build the heat exchanger using contemporary techniques. Once fabrication was complete, data collection commenced.

Throughout the experiments, issues with the thermocouples for the parallel testing led to many inaccuracies, thus garnering a high percentage error for the experimental and SolidWorks results. On the other hand, the percentage errors for the counterflow testing, the thermocouples readings were more accurate. As a result, the percentage errors for both the experimental and Solidworks, in counter flow, were less than 10 percent. However, the trends observed across the experimental and theoretical results aligned. Those trends included that increasing the cold-water flowrate increased the temperature reduction of the hot water and that the cold water had relatively minor changes in temperature. Furthermore, it was interesting to note that reducing the flowrate of the cold water had a drastically larger effect on the efficiency of the system when compared to raising the flowrate of the hot water. Stemming from the results of the project, it was deemed that buying individual thermal couples would improve the quality of the experimental results. Furthermore, being able to extend the heat exchanger would also improve the heat exchanger’s efficiency.

# References

Bergman. (2017).

# Appendix

MATLAB Code

clc

clear

format long

% Givens

T\_c\_i = 19; % C

T\_h\_i = 65; % C

TcK = T\_c\_i+273.15;

ThK = T\_h\_i +273.15;

D\_c\_i = 1.5\*0.0254; % m

D\_c\_o = 1.625\*0.0254; % m

D\_h\_i = 1\*0.0254; % m

D\_h\_o = 1.125\*0.0254; % m

D\_i\_o = 2.375\*0.0254; %m

%^insulation outer diameter

m\_c = 0.8; %kg/s

m\_h = 0.1; %kg/s

L = 1; % m

A\_o = pi()\*D\_c\_o\*L;

A\_c = pi()\*((D\_c\_i^2)-(D\_h\_o^2))/4;

A\_h = pi()\*(D\_h\_i^2)/4;

D\_c\_hy = 4\*A\_c/(pi\*D\_c\_i);

%D\_c\_hy = D\_c\_i-D\_h\_o;

k\_cu = 398; % W/m · K

eps = 1.5\*10^-6; % m

thermo\_prop = [273.15, 0.00611, 1.000, 206.3, 2502, 4.217, 1.854, 1750,...

8.02, 569, 18.2, 12.99, 0.815, 75.5, 68.05, 273.15; 275 0.00697,...

1.000, 181.7, 2497, 4.211, 1.855, 1652, 8.09, 574, 18.3, 12.22, 0.817,...

75.3, 32.74, 275; 280, 0.00990, 1.000, 130.4, 2485, 4.198, 1.858,...

1422, 8.29, 582, 18.6, 10.26, 0.825, 74.8, 46.04, 280; 285, 0.01387,...

1.000, 99.4, 2473, 4.189, 1.861, 1225, 8.49, 590, 18.9, 8.81, 0.833,...

74.3, 114.1, 285; 290, 0.01917, 1.001, 69.7, 2461, 4.184, 1.864, 1080,...

8.69, 598, 19.3, 7.56, 0.841, 73.7, 174.0, 290; 295, 0.02617, 1.002,...

51.94, 2449, 4.181, 1.868, 959, 8.89, 606, 19.5, 6.62, 0.849, 72.7,...

227.5, 295; 300, 0.03531, 1.003, 39.13, 2438, 4.179, 1.872, 855, 9.09,...

613, 19.6, 5.83, 0.857, 71.7, 276.1, 300; 305, 0.04712, 1.005, 29.74,...

2426, 4.178, 1.877, 769, 9.29, 620, 20.1, 5.20, 0.865, 70.9, 320.6,...

305; 310, 0.06221, 1.007, 22.93, 2414, 4.178, 1.882, 695, 9.49, 628,...

20.4, 4.62, 0.873, 70.0, 361.9, 310; 315, 0.08132, 1.009, 17.82, 2402,...

4.179, 1.888, 631, 9.69, 634, 20.7, 4.16, 0.883, 69.2, 400.4, 315; 320,...

0.1053, 1.011, 13.98, 2390, 4.180, 1.895, 577, 9.89, 640, 21.0, 3.77,...

0.894, 68.3, 436.7, 320; 325, 0.1351, 1.013, 11.06, 2378, 4.182,...

1.903, 528, 10.09, 645, 21.3, 3.42, 0.901, 67.5, 471.2, 325; 330,...

0.1719, 1.016, 8.82, 2366, 4.184, 1.911, 489, 10.29, 650, 21.7, 3.15,...

0.908, 66.6, 504.0, 330; 335, 0.2167, 1.018, 7.09, 2354, 4.186, 1.920,...

453, 10.49, 656, 22.0, 2.88, 0.916, 65.8, 535.5, 335; 340, 0.2713,...

1.021, 5.74, 2342, 4.188, 1.930, 420, 10.69, 660, 22.3, 2.66, 0.925,...

64.9, 566.0, 340; 345, 0.3372, 1.024, 4.683, 2329, 4.191, 1.941,...

389, 10.89, 664, 22.6, 2.45, 0.933, 64.1, 595.4, 345];

% Linear Interpolation Values

T\_c\_low = 0; T\_c\_up = 0; mu\_c\_low = 0; mu\_c\_up = 0; k\_c\_low = 0;...

k\_c\_up = 0; Pr\_c\_low = 0; Pr\_c\_up = 0; C\_c\_up = 0; C\_c\_low = 0;

mu\_h\_low = 0; mu\_h\_up = 0; k\_h\_low = 0; k\_h\_up = 0; Pr\_h\_low = 0;...

Pr\_h\_up = 0; C\_h\_up = 0; C\_h\_low = 0;

k\_cu\_low = 0; k\_cu\_up = 0;

rho\_c = 0; rho\_h = 0;

for i = 1:16

% Cold Values

if abs(TcK-thermo\_prop(i,1)) <= 5

% x = value, y = temperature

if (TcK-thermo\_prop(i,1)) > 0

% lower values

T\_c\_low = thermo\_prop(i,1);

mu\_c\_low = thermo\_prop(i,8);

k\_c\_low = thermo\_prop(i,10);

Pr\_c\_low = thermo\_prop(i,12);

C\_c\_low = thermo\_prop(i,6);

rho\_c\_low = thermo\_prop(i,3);

else

% upper values

T\_c\_up = thermo\_prop(i,1);

mu\_c\_up = thermo\_prop(i,8);

k\_c\_up = thermo\_prop(i,10);

Pr\_c\_up = thermo\_prop(i,12);

C\_c\_up = thermo\_prop(i,6);

rho\_c\_up = thermo\_prop(i,3);

end

end

% Hot Values

if abs(ThK-thermo\_prop(i,1)) <= 5

% x = value, y = temperature

if (ThK-thermo\_prop(i,1)) > 0

% lower values

T\_h\_low = thermo\_prop(i,1);

mu\_h\_low = thermo\_prop(i,8);

k\_h\_low = thermo\_prop(i,10);

Pr\_h\_low = thermo\_prop(i,12);

C\_h\_low = thermo\_prop(i,6);

rho\_h\_low = thermo\_prop(i,3);

else

% upper values

T\_h\_up = thermo\_prop(i,1);

mu\_h\_up = thermo\_prop(i,8);

k\_h\_up = thermo\_prop(i,10);

Pr\_h\_up = thermo\_prop(i,12);

C\_h\_up = thermo\_prop(i,6);

rho\_h\_up = thermo\_prop(i,3);

end

end

end

% Cold Interpolations

mu\_c = ((((TcK-T\_c\_up)\*(mu\_c\_low-mu\_c\_up))/(T\_c\_low-T\_c\_up))+mu\_c\_up)\*10^-6; % N · s/m2

k\_c = ((((TcK-T\_c\_up)\*(k\_c\_low-k\_c\_up))/(T\_c\_low-T\_c\_up))+k\_c\_up)\*10^-3; % W/m · K

Pr\_c = (((TcK-T\_c\_up)\*(Pr\_c\_low-Pr\_c\_up))/(T\_c\_low-T\_c\_up))+Pr\_c\_up;

c\_c = ((((TcK-T\_c\_up)\*(C\_c\_low-C\_c\_up))/(T\_c\_low-T\_c\_up))+C\_c\_up)\*1000; % J/kg · K

rho\_c = 1/(((((TcK-T\_c\_up)\*(rho\_c\_low-rho\_c\_up))/(rho\_c\_low-T\_c\_up))+rho\_c\_up)\*10^-3); % kg/m^3

% Hot Interpolations

mu\_h = ((((ThK-T\_h\_up)\*(mu\_h\_low-mu\_h\_up))/(T\_h\_low-T\_h\_up))+mu\_h\_up)\*10^-6; % N · s/m2

k\_h = ((((ThK-T\_h\_up)\*(k\_h\_low-k\_h\_up))/(T\_h\_low-T\_h\_up))+k\_h\_up)\*10^-3; % W/m · K

Pr\_h = (((ThK-T\_h\_up)\*(Pr\_h\_low-Pr\_h\_up))/(T\_h\_low-T\_h\_up))+Pr\_h\_up;

c\_h = ((((ThK-T\_h\_up)\*(C\_h\_low-C\_h\_up))/(T\_h\_low-T\_h\_up))+C\_h\_up)\*1000; % J/kg · K

rho\_h = 1/(((((ThK-T\_h\_up)\*(rho\_h\_low-rho\_h\_up))/(rho\_h\_low-T\_c\_up))+rho\_h\_up)\*10^-3); % kg/m^3

% Calculated Constants

%%%%%%% the area of inner - outer

v\_c = m\_c/(rho\_c\*A\_c);

v\_h = m\_h/(rho\_h\*A\_h);

ReD\_c = (rho\_c\*v\_c\*D\_c\_hy)/mu\_c;

ReD\_h = (rho\_h\*v\_h\*D\_h\_i)/mu\_h;

% ReD\_c = (m\_c\*(D\_c\_i-D\_h\_o))/((pi()/4)\*((D\_c\_i^2)-(D\_h\_o^2))\*mu\_c);

% ReD\_h = (m\_h\*4)/(pi()\*D\_h\_i\*mu\_h);

f\_h = (-1.8\*log10(((eps/(3.7\*D\_h\_i))^1.11)+(6.9/ReD\_h)))^-2;

f\_c = (-1.8\*log10(((eps/(3.7\*D\_c\_hy))^1.11)+(6.9/ReD\_c)))^-2;

NuD\_h = ((f\_h/8)\*(ReD\_h-1000)\*Pr\_h)/(1+(12.7\*((f\_h/8)^(1/2))\*((Pr\_h^(2/3))-1)));

NuD\_c = ((f\_c/8)\*(ReD\_c-1000)\*Pr\_c)/(1+(12.7\*((f\_c/8)^(1/2))\*((Pr\_c^(2/3))-1)));

% NuD\_c = 0.023\*(ReD\_c^0.8)\*(Pr\_c^0.4); % cold water is getting heated

% NuD\_h = 0.023\*(ReD\_h^0.8)\*(Pr\_h^0.3); % hot water is getting cooled

h\_h = (k\_h\*NuD\_h)/D\_h\_i;

h\_c = (k\_c\*NuD\_c)/(D\_c\_hy);

U = 1/(((1/h\_h)+(log(D\_h\_o/D\_h\_i)/(2\*pi()\*L\*k\_cu))+(1/(h\_c))+(log(D\_c\_o/D\_c\_i)/(2\*pi()\*L\*k\_cu))));

C\_h = c\_h\*m\_h;

C\_c = c\_c\*m\_c;

if C\_c < C\_h

C\_min = C\_c;

C\_max = C\_h;

else

C\_min = C\_h;

C\_max = C\_c;

end

C\_r = C\_min/C\_max;

NTU = (U\*(pi()\*D\_c\_o\*L))/C\_min;

q\_max = C\_min\*(T\_h\_i-T\_c\_i);

% Parallel Flow

eps\_p = (1-exp(-NTU\*(1+C\_r)))/(1+C\_r);

q\_p = eps\_p\*q\_max;

T\_c\_o\_p = (q\_p/(C\_c))+T\_c\_i;

T\_h\_o\_p = (-q\_p/(C\_h))+T\_h\_i;

% Counter Flow

eps\_c = (1-exp(-NTU\*(1-C\_r)))/(1-(C\_r\*exp(-NTU\*(1-C\_r))));

q\_c = eps\_c\*q\_max;

T\_c\_o\_c = (q\_c/(C\_c))+T\_c\_i;

T\_h\_o\_c = (-q\_c/(C\_h))+T\_h\_i;

% OTHER VALUES

Le\_c = 0;

Le\_h = 0;

% Flow Entrance Length

if ReD\_c > 2300

Le\_c = 4.4\*(ReD\_c^(1/6))\*D\_c\_hy

else

Le\_c = 0.06\*ReD\_c\*D\_c\_hy

end

if ReD\_h > 2300

Le\_h = 4.4\*(ReD\_h^(1/6))\*D\_h\_i

else

Le\_h = 0.06\*ReD\_h\*D\_h\_i

end

% Thermal Entrance Length

if ReD\_c > 2300

Lt\_c = 10\*D\_c\_hy

else

Lt\_c = 0.05\*ReD\_c\*Pr\_c\*D\_c\_hy

end

if ReD\_c > 2300

Lt\_h = 10\*D\_h\_i

else

Lt\_h = 0.05\*ReD\_h\*Pr\_h\*D\_h\_i

end

% Pressure

L\_tot = 57\*0.0254; % m

pressure\_c = f\_c\*rho\_c\*(v\_c^2)\*L/(2\*D\_c\_hy)

pressure\_h = f\_h\*rho\_h\*(v\_h^2)\*L/(2\*D\_h\_i)

% Headloss

v\_c\_i = m\_c/(rho\_c\*A\_h);

hl\_c = (0.44\*(v\_c^2))/(2\*9.81)

hl\_e = ((v\_c-v\_c\_i)^2)/(2\*9.81)

K\_ball = 0.05;

K\_90 = 0.3;

% COLD

Hl\_c = (2\*majorheadloss(v\_c\_i,f\_c,8\*0.0254,D\_h\_i))+majorheadloss(v\_c,f\_c,L,D\_c\_hy)+(2\*minorheadloss(v\_c\_i,K\_ball))+(2\*minorheadloss(v\_c,K\_90))+hl\_e+hl\_c

Hl\_h = majorheadloss(v\_h,f\_h,57\*0.0254,D\_h\_i)+minorheadloss(v\_h,K\_ball)

% printing

disp(['Length: ',num2str(L),' m'])

fprintf('\n')

disp(['U: ',num2str(U)])

fprintf('\n')

disp(['Initial Hot Temperature: ',num2str(T\_h\_i)])

disp(['Initial Cold Temperature: ',num2str(T\_c\_i)])

fprintf('\n Parallel Flow \n')

disp(['Heat Transfer Rate: ',num2str(q\_p)])

disp(['Final Hot Temperature: ',num2str(T\_h\_o\_p)])

disp(['Final Cold Temperature: ',num2str(T\_c\_o\_p)])

fprintf('\n Counter Flow \n')

disp(['Heat Transfer Rate: ',num2str(q\_c)])

disp(['Final Hot Temperature: ',num2str(T\_h\_o\_c)])

disp(['Final Cold Temperature: ',num2str(T\_c\_o\_c)])