Theo DeGuzman, Marcus Montisano, Xander Song, Jason Zhong

530.335 Heat Transfer Laboratory

Heat Exchanger Project Final Report

April 27, 2023

Section 1: Cost Report

Below are all the purchases made for the project throughout the semester, including costs associated with the final heat exchanger and manufacturing charges.

Table 1: Bill of Materials

No.	**Item**	**Source**	**Individual Price**	**QTY**	**Subtotal**
1	Tupperware Container	Amazon	$19.49	1	$19.49
2	¼” OD Soft Copper Tubing (25 ft)	Amazon	$36.99	1	$36.99
3	½ NPT to ½” Threaded Fitting	McMaster	$4.09	2	$8.18
4	½” to ¼” Reducing Fitting	McMaster	$1.08	2	$2.16
5	¼” OD Solder Connect Fitting	McMaster	$0.40	3	$1.20
6	⅜” OD Copper Tubing (Never arrived)	Amazon	$36.30	0	$0.00
7	½” to ⅜” Reducing Fitting (Not usable because tubing never arrived)	McMaster	$0.90	0	$0.00
8	⅜” OD Solder Connect Fitting (Not usable because tubing never arrived)	McMaster	$0.45	0	$0.00
9	⅜” ID Soft Copper Tubing (Team purchase)	Home Depot	$19.62	1	19.62
10	Polycarbonate Plate 14”x9”	Scrap MESD	$24.09	1	24.09
11	Plastic Sheeting (Team purchase)	Home Depot	$7.99 per roll	0.1	$0.79
12	8/32” Allen Screw	Spare MESD	$10.74 per 100	0.12	$1.29
13	Hose Clamps	Spare MESD	$0.58	4	$2.32
14	Welding Time (15 min)		$30 per hr	0.25	$7.50
				Total	$104.01

Section 2: Modeling

The approach to modeling this heat exchanger was to model it similarly to a shell and tube exchanger with a single shell pass. This neglected the spiral nature of the hot water tubes and that they experience a combination of cross-flow and counter-flow. The cross sections were assumed perfectly circular, with the corrected total pipe lengths. Efficiency was also assumed to be 1.0, which was very close to that measured in the final experiment.

The hot and cold-water temperatures were user input, and those values were used to interpolate the density, specific heat, Prandtl number, and the thermal conductivity. The pipe geometry was hard coded into the simulation, with flow rates included.

From the volumetric flow rate and the cross-sectional area, the velocity and mass flow rate were calculated. From there, the heat capacity of each flow was determined, and the linear speed of each was calculated. This can then be used to calculate Reynold's number. The calculations and equations up to this point are standard and well-documented. From here, the Nusselt number had to be calculated. For the hot water, which was turbulent flow, the following equation was used (Textbook Eqn 8.62):

$$NU_D = \overline{NU_D} = \frac{\left(\left(\frac{f}{8}\right)(Re_D - 1000)\mathrm{Pr}\right)}{1 + 12.7\left(\frac{f}{8}\right)^{\frac{1}{2}}\left(Pr^{\frac{2}{3}} - 1\right)} \tag{1}$$

Where Nu_D is Nusselt number, Re_D is Reynold's number, Pr is Prandtl number, and f is friction factor. For the cold water, which was laminar flow, the equation for Nusselt number is below (Textbook Eqn 8.57):

$$\overline{NU_D} = 3.66 + \frac{.0668 Gz_D}{1 + .04(Gz_D)^{\frac{2}{3}}} \quad \text{where} \quad Gz_D = \left(\frac{D}{x}\right) Re_D Pr \tag{2}$$

Where D is hydraulic diameter and x is total pip length. From these, the convective heat transfer coefficient can be determined:

$$\bar{h} = \frac{k_f \overline{Nu_D}}{x} \tag{3}$$

Where the h is the heat transfer coefficient of the hot and cold flow. From this, the overall heat transfer coefficient and NTU were determined using:

$$U = \frac{1}{\left(\frac{1}{h_{hot}} + \frac{1}{h_{cold}}\right)} \tag{4}$$

$$NTU = \frac{UA}{(\dot{m}c_p)_{min}} \quad (5)$$

Taking the minimum heat capacity between the hot and cold water, and dividing it by the other, gives the heat capacity ratio (C_r). This was used in conjunction with Textbook equation 11.30a to determine the heat exchanger effectiveness:

$$\varepsilon = 2\{1 + C_r + (1 + C_r^2)^{\frac{1}{2}} \times \frac{1+\exp\left[-(NTU)(1+C_r^2)^{\frac{1}{2}}\right]}{1-\exp\left[-(NTU)(1+C_r^2)^{\frac{1}{2}}\right]} \quad (6)$$

From there, the heat transfer out of the hot water and into the cold water were predicted, and from these values, temperatures could be determined:

$$q_{hot} = \varepsilon \cdot q_{max} = \varepsilon \cdot (C_{min})(\Delta T) \quad (7)$$

$$q_{cold} = q_{hot} \cdot \eta_{assumed} \quad (8)$$

$$T_{out} = T_{in} \pm \frac{q}{C_{fluid}} \quad (9)$$

Finally, to calculate the pressure drop, an absolute roughness of 0.03 was assumed and used to calculate relative roughness and pull a friction factor from a Moody diagram. This, combined with the Reynold's number, results in a friction factor of .04 for hot water, and .32 for cold. The following equation also takes in geometric constants and dynamic pressure to calculate pressure drop:

$$\Delta P = \frac{f\left(\frac{1}{2}\rho v^2\right)L_{equiv}}{D_H} \quad (10)$$

This simulation now can output all the values needed.

The predicted value for F_1 was 1180.1 W.

Section 3: Results

Table 1: 180 second average values measured on final heat exchanger and an Armfield HT31

Performance Value	Final Heat Exchanger	Armfield HT31
$T_{h,i}$ (℃)	29.98	30.11
$T_{h,o}$ (℃)	23.32	26.26
$T_{c,i}$ (℃)	3.60	5.00
$T_{c,o}$ (℃)	13.08	8.50
$\dot{V}_h$ (l/min)	2.00	2.01
$\dot{V}_c$ (l/min)	1.48	2.02
ΔP_{cold} (kPa)	0.41	9.31
ΔP_{hot} (kPa)	1.83	2.95

Table 2: Measured and derived values of various dimensional, cost, performance figures of the final heat exchanger compared to an Armfield HT31 heat exchanger.

Performance Value	Final Heat Exchanger	Armfield HT31
m_e (kg)	2.165	0.76
V_e (m^3)	0.0093	0.0063
x_e (USD)	$104.52	N/A
$q_{h\rightarrow}$ (W)	912.3	538.4
$q_{\rightarrow c}$ (W)	967.2	492.9
η	1.06	0.92
U ($\frac{W}{m^2k}$)	2899	1355
ε	.339	0.153
NTU	2.593	0.179

Table 3: Calculated figures of merit for the final heat exchanger compared to those of an Armfield HT31 heat exchanger.

Figure of Merit	Final Heat Exchanger	Armfield HT31
F1 = $q_{h\rightarrow}$ (W)	912.3	538.4
F2 = $\frac{q_{h\rightarrow}}{V_e}$ $(\frac{W}{m^3})$	98194	85460
F3 = $\frac{q_{h\rightarrow}}{m_e}$ $(\frac{W}{kg})$	421.4	708.4
F4 = $\frac{q_{h\rightarrow}}{\Delta P_{max}}$ $(\frac{W}{kPa})$	504.0	57.8
F5 = $\frac{q_{h\rightarrow}}{x_e}$ $(\frac{W}{USD})$	8.771	N/A

Section 4: Discussion

Our heat exchanger compared well to the predicted values. It is important to note that during the experiment, the cold-water pump experienced an issue unrelated to our heat exchanger. Therefore, the MATLAB code was modified to reflect the lower cold water flow rate. The equivalent length coefficients of the fittings were also tweaked to match the measured values more closely. After these adjustments, the measured rate of heat removed was 912 W, or about 29% lower than the simulated value of 1180 W. This difference could be due to many of the assumptions made. For example, the box was assumed to be a perfect insulator, but it is not and therefore the cold water may have absorbed heat from the environment, causing it to draw less heat out of the hot water. Similarly, no consideration was taken for the fact that the hot water tubes were coiled, or that the cold water acted in a combination of cross- and counter-flow over the hot water tubes. The tubes and baffles were also assumed to be perfectly circular, but due to minor flattening in the tubes, and a non-rigid cylindrical bag-like baffle, this was likely not the case. All of these factors and other assumptions may account for this discrepancy. However, in terms of simulating fluids and heat transfer, getting on the same order of magnitude is considered good, making our 29% discrepancy imply that our model well-reflects out heat exchanger.

The Armfield heat exchanger was able to remove around 540 W of heat from the hot water. Our final heat exchanger was able to remove 912 W in less favorable conditions, with a broken cold water pump. Despite the difference, directly comparing the two heat exchangers' performances shows ours was able to remove almost 400 W, or almost 70% more heat than the Armfield heat exchanger. Ours was also more efficient, and also had twice the overall heat transfer coefficient. It was larger and bulkier that the Armfield, which caused it to perform worse in figure of merit three, heat transfer per unit mass. However, it outperformed Armfield in every other category, especially heat transfer over pressure drop, where it won out by a factor of ten. This was because our exchanger had a much longer tube than the Armfield, allowing for more heat transfer through the pipe, while also maintaining a fairly large cross-sectional area to limit pressure drops. The coiled nature of the tubes and ruffled surface of the baffles initiated turbulence to minimize the chance that water flowed through the center of this larger cross sections without exchanging much heat.

The calculated efficiency of our final heat exchanger was 106% or 1.06. Tests conducted beforehand indicated that there was no mixing inside our heat exchanger, so this was likely not the cause. This was most likely triggered by the cold-water absorbing heat from the outside of the exchanger, driving the cold water temperature higher, causing the efficiency number to artificially inflate. For our heat exchanger, despite the fact it appeared to have better-than-perfect efficiency, this is actually a detriment. This is because heat into the cold water increases its temperature, which decreased the magnitude of the temperature gradient inside the exchanger, providing less potential difference to drive heat out of the hot water.

A major item limiting the performance of our design was the power of the motor not being able to pump hot water into the smaller diameter (¼”) of our original pipe, which was 25 feet long. This caused us to use a larger diameter pipe (⅜”) to ensure we could withstand the pressure drop. However, this larger pipe was harder to bend by hand, required a larger bend radius, and forced us to use less coils in our final design. Having fewer coils and decreasing the total pipe length to 10 feet, reduced the amount of surface area between the hot and the cold water, hindering the performance of the heat exchanger. Another limiting factor was the box we chose was not as sturdy as advertised. While it was technically air- and watertight under no pressure, any amount of pressure caused the box to flex and leak violently. This required the addition of a sealed, polycarbonate lid with fasteners and sealant, which increased total mass by more than a pound. This caused the exchanger to be heavy despite previous design choices to make it lighter, resulting in a worse score in figure of merit three. This lid did work to seal the container though, with a singular drip forming while the exchanger was filling that sealed itself once the exchanger was full, due to internal pressure closing the gaps in the internally applied sealing material.

If we were to iterate on this heat exchanger, we would drop back down to a smaller tube size. Sizing up from ¼” OD to ⅜” ID was a large jump, but a design decision made to ensure our heat exchanger would function without having to risk pressure drop issues. This would allow a longer tube, more turbulent internal flow, and overall more heat exchange. The bag-baffles worked excellently, and if given more time, could likely be sealed. We believe this to be the case since they just barely dripped into the outer box with our rudimentary hose-clamp solution. If the bag-baffles were sealed, the entire exterior box could be removed, saving a large amount of weight.

We used welding in our heat exchanger, which is an unusual manufacturing method for this project. Due to supply chain issues causing a substitute tube purchase and consequently a lack of correctly sized fittings, we welded the reduction fittings to the tube to fill large gaps between improperly sized fittings. It would not have been possible to solder the pipes together. Teaching how to weld copper may have helped other teams who saw similar issues, and it was critical for allowing our heat exchanger to perform without internal mixing. Unlike the proposal, we opted not to use 3D printed baffles, as the light plastic ones we used were far cheaper, lighter, manufacturable, and likely more effective than a 3D print. However, metal 3D printing copper would open a host of new opportunities for the design, as 3D printed flow channels are quickly being adapted by industry to produce extremely optimized heat exchangers. For example, this is now standard for the cooling units on the exterior of rocket engine nozzles. Beyond this, a CNC tube bender would have been very helpful in bending our tubes. Additionally, a manual tube bender that could fit our size tube also could have been helpful (our tube was too large for the one provided), but it may or may not have been more efficient than sand-packing and hand bending.

We learned many things in this design project that may help for projects in the future. First, you should always do calculations for components before buying them if planning to use them in a realistic test or prototype. We also learned a lot about sealing, and that a small pressure buildup can lead to a large force on the inside of a container, causing it to burst open. We also learned that using silicone sealant on the inside is far better than on the outside, because the pressure inside the vessels presses the sealant into place instead of trying to push it out.

One way to improve this design project for next year would be to have more deadlines earlier in the semester. We placed an order for parts as early as possible, before leaving for spring break, and had all parts available when we got back from break. However, between the first and final evaluations, we realized we needed more parts, and the parts did not come in in time. If the first evaluation was before spring break, we would have more time to order parts and work on the final design. Similarly, if groups were allowed to test their systems under pressure for leaks before the first evaluation, fewer issues would be discovered during the evaluation. Another suggestion would be to require groups to do estimated pressure drop calculations before ordering pipe. The reason we did not do this calculation was because we did not have a maximum pressure drop number for comparison before ordering the pipe, so providing this number up front will be helpful to future teams.