Design and Analysis of a Newly Installed Plate Heat Exchanger Apparatus for the WPI Unit Operation Laboratory Course

A Major Qualifying Project submitted to the faculty of

WORCESTER POLYTECHNIC INSTITUTE

in partial fulfillment of the requirements for the

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Submitted to

Prof. William Clark: Worcester Polytechnic Institute

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This report represents work of WPI undergraduate students submitted to the faculty as evidence of a degree requirement

Abstract

The goals of this project were to design a plate heat exchanger apparatus and evaluate the potential of replacing the current double pipe heat exchanger by this newly designed plate heat exchanger for the Unit Operation Laboratory Course. Raw data was collected during experiments. The goals for this project were achieved, as the team accomplished all objectives for the Heat Exchanger Unit Operation Experiment. The results obtained from the plate heat exchanger apparatus were consistent with the governing theory compared to the results from the double pipe heat exchanger apparatus.

Acknowledgements

We would like to thank our advisor, Professor W. M. Clark, for his guidance and support throughout this project; without his overall contribution this project would not be possible.

We would also like to thank the lab manager of the WPI UO lab, Tom Partington, for his help with ordering the necessary parts and constructing the mounting stud upon which the plate heat exchanger was mounted.

Contents

1. Introduction

A milestone in the engineering curriculum overall is the study of the transport phenomena which consist of three interconnected branches; heat transfer, mass transfer and fluid mechanics. Heat transfer in particular is of paramount interest to chemical engineers who deal with a plethora of processes which involve the consumption and/or generation of energy, very often in the form of heat. From common daily processes, such as home heating, cooking and others to virtually all industrial processes independently of their scale, heat is transferred - exchanged - between one or more fluids - even in humans the nasal passages operate as heat exchangers during inhaling and exhaling. Heat exchangers are the devices used to transfer heat between one or more fluids and serve as an important processing equipment in various industrial processes ranging from wastewater treatment facilities to petroleum refineries and wineries.

The Unit Operations course offers students majoring in chemical engineering the chance to apply key concepts acquired throughout the chemical engineering sequence through the study and analysis of a variety of experiments. The Unit Operations Laboratory of Worcester Polytechnic Institute utilizes a double pipe heat exchanger to facilitate the analysis and understanding of heat transfer inside a heat exchanger. Water flows inside the inner pipe and steam flows through the annulus formed between the two pipes. A cross section view a double pipe heat exchanger representing the one used in the WPI UO lab is provided in the following schematic.

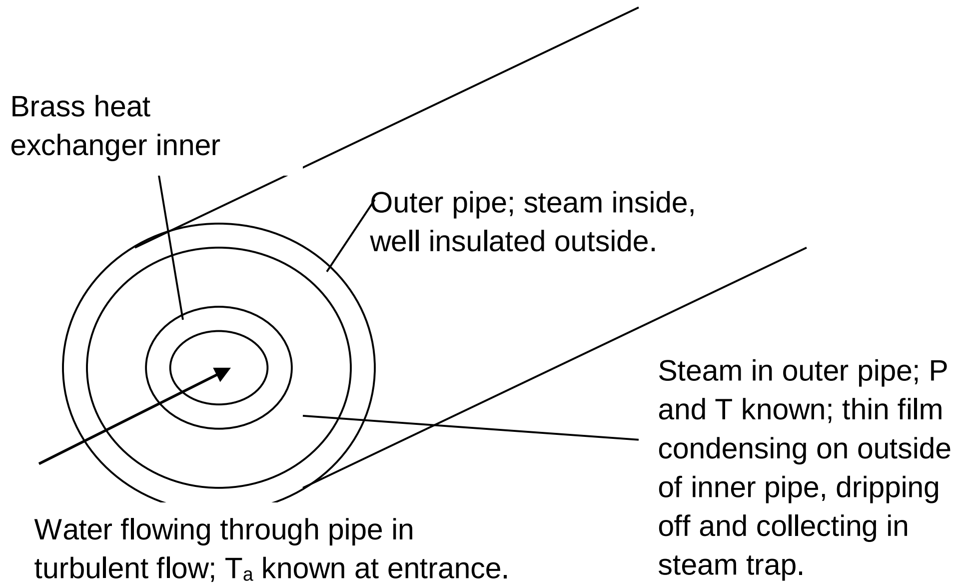


Figure 1: Cross section view of WPI's UO lab double pipe heat exchanger [1]

Students run the experiment and are able to vary the water flowrate, the steam pressure and interchange the flow pattern between co-current (also known as parallel) and countercurrent flow. The main key concepts investigated include the dependency of the overall and individual - that is the water side and steam side - heat transfer coefficients upon operating conditions such as the water flowrate and steam pressure and the effect of the flow pattern on the transfer of heat inside the exchanger. In the double pipe heat exchanger of the WPI Unit Operations Lab where steam condenses in the outer pipe and consequently transfers heat to the water flowing in the inner pipe, it is expected that the flow pattern should not have an impact on the overall transfer of heat between the two fluids - mainly dictated by the heat transfer coefficients - since as steam condenses its temperature remains constant as it undergoes a phase change from vapor to liquid. Therefore, as the temperature of one of the two fluids remains the same so does the temperature gradient across the pipe, the driving force of the transfer of heat between two bodies with different temperatures, and therefore co-current and countercurrent flow are expected to yield the same results. However, students running the experiment often observed higher heat transfer coefficients when operating on co-current flow than in countercurrent. The discrepancies arose mainly due to the configuration of the double pipe heat exchanger and the structure of the experiment itself as often volumes of hot water (condensed steam) were trapped at the water inlet from previous experimental runs leading to ‘distorted’ readings and thus to an overall analysis which would not coincide with the governing theory. Moreover, the cost of the steam supplied by the campus-wide steam plant to the double pipe heat exchanger has also become an issue during the past few years.

The possibility of replacing the current double pipe heat exchanger used in the WPI Unit Operations laboratory due to mainly the aforementioned reasons provides the incentive of this study. In particular, this project focuses on the operation and analysis of a ten-plate heat exchanger which will run on cold and hot water supplied from the building’s utility lines aiming at determining the feasibility of using this process equipment instead of the current double pipe heat exchanger for the Unit Operations Laboratory course at WPI.

2. Background

This section provides relevant information on heat transfer theory focusing mostly on heat transfer inside a heat exchanger and also offers brief background information on the operation of different types of heat exchangers putting emphasis on plate heat exchangers.

2.1 Heat Transfer Theory

Heat transfer is the exchange of thermal energy between physical systems by dissipating heat. The driving force of heat transfer is the temperature difference between the physical systems that heat is exchanged. The theory of heat transfer from one media to another or from one fluid to another fluid is governed by three fundamental principles. First, heat always transfers from a region of high temperature to another region of lower temperature. Second, there must always be a temperature difference between the media for heat to be transferred. Lastly, the heat lost by the region of higher temperature must equal the amount of heat gained by the region of lower temperature except for losses to the surroundings.

Heat can be transferred through four fundamental methods, also known as modes of heat transfer, and include advection, conduction or diffusion, convection and radiation. Briefly, advection describes the transfer of heat by the flow or movement of a fluid or a body respectively from one location to another and is dependent upon displacement and momentum. Conduction or diffusion refers to the transfer of heat between solids or fluids that are in physical contact and arises from the rapid movement/vibration of atoms and molecules which transfer part of their energy (in the form of heat) to adjacent neighboring particles. Convection is the heat transfer caused by the mass motion of a fluid when the heated fluid is forced to move away from the source of heat carrying energy with it in the form of heat. Finally, radiation refers to the transfer of heat due to the movement of charged subatomic particles. Contrary to the other modes of heat transfer radiation does not require a medium to take place; radiation can occur in absolute vacuum. In a heat exchanger, heat is transferred through convection and conduction

2.2 Heat Transfer in Heat Exchanger

In a heat exchanger, heat is first transferred from hot fluid to the surface separating the two fluids (boundary) by convection. Next, heat is transferred through the boundary by conduction. Then heat is transferred from the boundary to the cold fluid by convection (Cengel et al., 2008). An ideal plate heat exchanger can be imagined as a closed system, so energy is conserved in this system, which means the heat lost from the hot water equals to the heat gained by the cold water. In this project, the system is not an ideal one, as there are heat losses to the surroundings, so the optimal operating condition can be determined when the heat transfer efficiency is highest.

There are some factors that affect the heat transfer rate which can be deduced from the heat transfer equation, the governing equation for the heat transfer rate particularly inside a heat exchanger.

$$Q = U * A * \Delta T_{LM}$$

Temperature difference between the two fluids provides the driving force for heat exchange. Thus a greater temperature difference will result in a greater heat transfer rate. Moreover, according to theory a higher fluid flow rate (higher fluid velocity) results in a higher heat transfer rate. The overall heat transfer coefficient, expressed as U, also dictates the rate at which heat transfer takes place. The overall heat transfer coefficient (U) is a measure of the overall ability of a series of conductive and/or convective barriers to promote, or equally resist, transfer of heat. It is a function of the individual heat transfer coefficients associated with each body or fluid participating in the heat exchange process along with the heat transfer coefficient associated to the medium (solid as the pipe wall or fluid as in the air) where the transfer of heat takes place and is a measure of the medium's ability to promote or resist heat transfer and is a function of the medium's inherent properties (such as its material, thickness), the conditions (temperature and pressure) and the mode of heat transfer. Thermal conductivity of the building material affects the heat transfer rate in a heat exchanger. Heat exchangers made of materials with higher thermal conductivity (often denoted as k) provide higher heat transfer rate. A larger heat transfer surface area also generates a higher transfer rate according to the heat transfer principle. The surface area available for heat transfer is dictated by the design and manufacture of the exchanger in order to meet the heat load requirements. In a tube and shell heat exchanger which consists of a collection of relatively thin, long tubes which provide the surface area available for heat transfer, the more tubes contained in the bundle the greater the surface area. The tube length also affects the heat transfer rate, as the outside diameter and metal thickness of the tubes does.

2.3 Types of heat exchangers

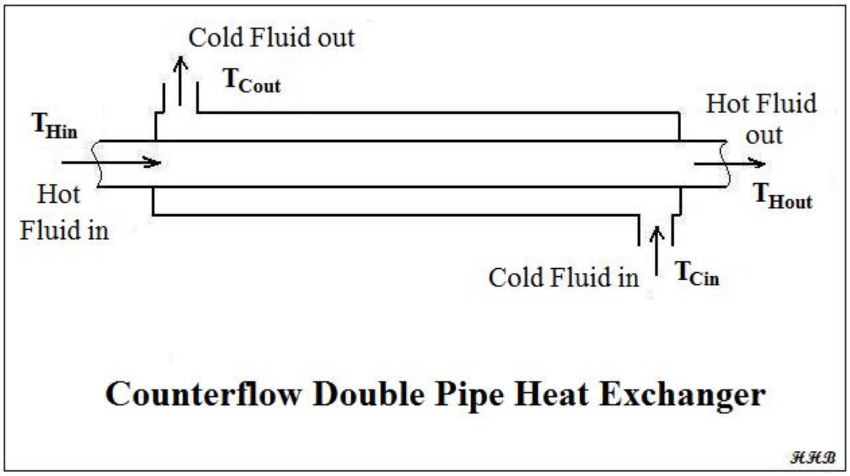


Figure 2: Double Pipe Heat Exchanger in countercurrent flow

Double pipe heat exchangers are the simplest used currently in industrial processes. They take their name from their construction where one pipe is placed inside another pipe of bigger diameter. Heat is exchanged between fluids running through the inside pipe and the annulus formed between the in between pipes while the flow pattern is limited to either countercurrent or co-current flow. Due to this configuration, double pipe heat exchangers are not very efficient in heat transfer area while they take up considerable amount of space, the pipes need to be wider and longer in order to obtain a larger heat transfer area. Counter-current flow will result in a better heat transfer coefficient compared with co-current flow.

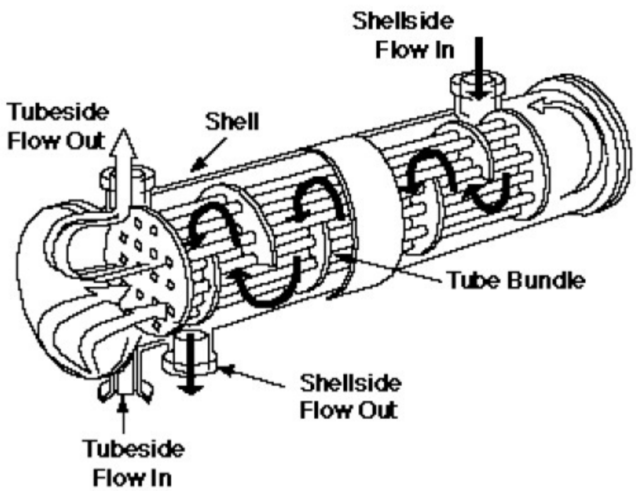


Figure. 3 Shell and Tube Heat Exchanger

The shell and tube heat exchanger is one of the most commonly used heat exchanger for industrial processes. All possible flow patterns are attainable in a shell and tube heat exchanger; co-current (or parallel) flow where the fluids flow in the same direction, countercurrent flow where the fluids flow in opposite direction and cross-flow where the fluids flow at right angles to each other. A shell and tube heat exchanger is made up of a series of tubes called a tube bundle and an outside shell covering them. One fluid passes through the tubes and another passes through the shell-side. The heat transfer coefficient is affected by the flow pattern. Careful consideration is put in choosing which fluid is placed in the tube-side or the shell-side of the exchanger in order to achieve efficient heat transfer and avoid excess fouling. Tube and shell heat exchangers consist of multiple inner tubes which significantly increase the total surface area available for heat transfer, rendering them to be a more efficient option than a double pipe heat exchanger.

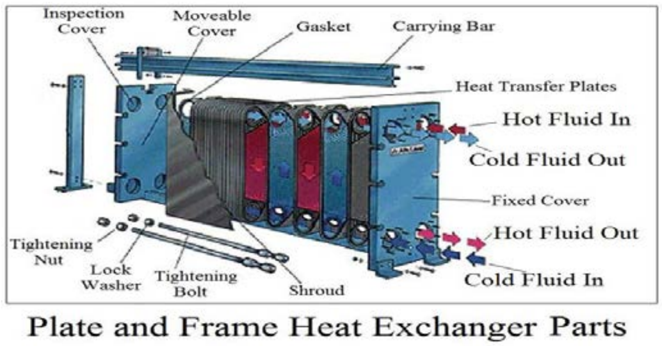


Figure. 4 Plate Heat Exchanger

A plate heat exchanger consists of a number of heat transfer plates which are held in place between a fixed plate and a loose pressure plate to form a complete unit. Each heat transfer plate has a gasket arrangement which provides two separate channel systems. The arrangement of the gaskets results in through flow in single channels, so that the primary and secondary media are in counter-current flow. The media cannot be mixed because of the gasket design. The plates are corrugated, which creates turbulence in the fluids as they flow through the unit. This turbulence, in association with the ratio of the volume of the media to the size of the exchanger, gives an effective heat transfer coefficient.

In a plate heat exchanger, heat is transferred through a metal heat exchanger plate. A plate heat exchanger usually has higher heat transfer than a double pipe heat exchanger or shell and tube heat exchanger. Also, plate heat exchangers are more space efficient than double pipe heat exchanger, because of the compact size configuration and the higher heat transfer coefficient they provide. Plate heat exchangers' configuration can be easily modified to achieve higher heat transfer capacity by simply adding additional plate heat exchangers.

2.4 Comparison of Plate and Double Pipe heat exchangers

Plate heat exchangers have higher surface area to volume ratio than conventional double pipe heat exchangers. The plates in a plate heat exchanger are designed to generate high turbulence, so that plate heat exchangers usually offer a superior heat transfer coefficient. The high turbulence also provides a self-cleaning effect. When compared to a traditional double pipe heat exchanger the fouling of the heat transfer surfaces is considerably reduced. The advantages of a plate heat exchanger are encompassed in its compact size, the higher heat transfer coefficient they offer, and their easiness of cleaning and maintenance. It is estimated that in order to achieve the same amount of work, a plate heat exchanger only requires about ⅓ to ⅕ the surface area of a conventional double pipe heat exchanger. The double pipe heat exchanger in the UO lab occupies significantly more space than the newly installed plate heat exchanger. In addition, due to its small size and low operating pressure less pumping for the plate heat exchanger is required, thus the plate heat exchanger is less expensive to purchase and to operate. Contrary to double pipe heat exchangers, plate heat exchangers operate well even with a small temperature difference between fluids, but do not function as well with large temperature differences. Thus, the plate heat exchanger can work well with the utility water, and does not require steam, which can make the heat exchanger experiment viable after the steam supply is stopped. Fouling factor, denoted as R_f, should be much lower for a plate heat exchanger than in a double pipe one. This is due to the design of plate heat exchangers which provides a much higher turbulence and thereby thermal efficiency than in a typical double pipe heat exchanger. A typical k value for a plate heat exchanger operating with water, as the one under investigation in this report, ranges between 6000-7500 W/^{2}C while for a double pipe one is only about 2000-2500 W/^{2}C. From the thermal design margin equation applied upon the design of a heat exchanger, $M = k * R_f$, and the aforementioned thermal conductivity values, it is clear that fouling factor values (R_f) for a plate heat exchanger are considerably lower than in a double pipe one. However, since the plate heat exchanger used for this project was brand new, fouling was assumed negligible and excluded from the analysis.

3. Methodology

3.1 Process development and apparatus installation

The team began designing the process of the experiment under the guide of the project advisor. The process requires a continuous hot flow and cold flow, the utility water was chosen as the source of cooling and heating for the exchanger. The inlet and outlet of the hot water through the heat exchanger was connected to detachable joints, this setup allowed the team to switch between co-current and counter-current flow patterns without the need of rebuilding the system.

In this project a 10-plate heat exchanger manufactured by *Duda Energy LLC* was used. The exchanger itself had already been purchased but several extra parts needed to be ordered. These include the tees, joints and the fittings used to establish the connections between the hoses connecting the exchanger and the rotameters through which the water flows. Measurements of the available hoses' diameters were taken and the selection and ordering of the appropriate threads and fittings was done accordingly in order to assure a safe and effective connection which would prevent leakages and withstand fluctuations in flowrates and thus pressures. Several combinations and trials were tested until the final leakage-free configuration was established. An appropriate location in the WPI UO lab for the plate heat exchanger to be installed was found which provides available space and easiness for more than one person to operate the equipment while a drain area for the water outlets is in close vicinity. A metal plate was constructed by the lab manager meeting the size specifications and welded onto a metallic rod and finally the plate heat exchanger was mounted on the plate frame.

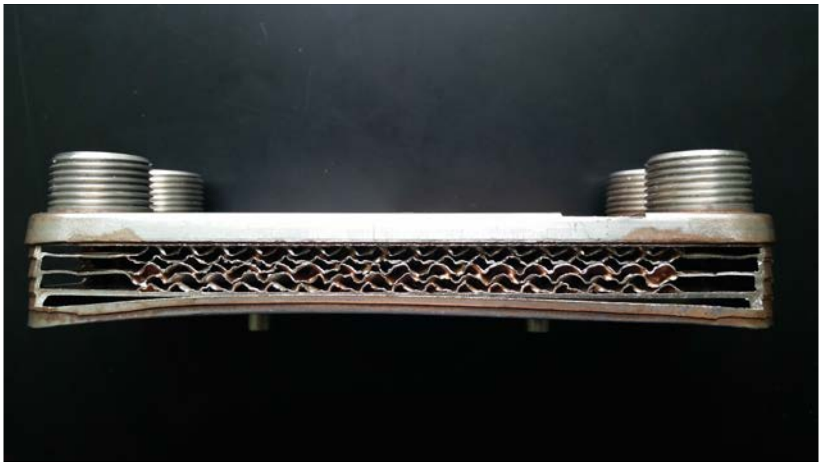


Figure. 5 Side View of the Installed Plate Heat Exchanger

According to the product specifications the total surface area available for heat transfer provided by this specific model is 0.12 m^2 per heat transfer plate. This is a copper brazed model with copper welding and Stainless Steel 304 plates **(**duda diesel**).** Stainless Steel is generally a good corrosion resistive material, however for this project only hot and cold water were used, so corrosion was not concerned. Figure 5 is a picture of the plate heat exchanger used for this project .

Two rotameters were borrowed from the WPI Work Shop, one measuring up to 8.65GPM and was used for the cold water, and the other measuring up to 5 GPM and was used for the hot water flowrate. Before actually mounting the rotameters, the maximum flow capacity of each rotameter was measured to determine their suitability for the experimental runs to follow, to prevent overflow or insufficient flow, which both could affect the accuracy of the readings. The rotameters and the plate heat exchanger were then bolted on a mounting stud and finally mounted to a convenient location in the WPI Unit Operations laboratory. After hooking up the flowmeters and heat exchangers to the water sources, calibration of the rotameters was performed to examine the accuracy of the reading. The volume of the hot and cold water found in one minute were measured, and compared with the reading from the flow meters. This was repeated for several flowrate readings and a calibration curve for each rotameter was produced which can be found in the *Appendix* section.

Finally, pressure gage and thermocouples were connected to the heat exchanger to measure temperature of the inlet and outlet fluids and pressure difference between the cold and the hot side.

3.2 Test of the system

The main problems usually encountered with heat exchangers are as follows:

1. Fouling**:** This is caused by deposits of scale, dirt, sand and/or other solid particles on the conducting surfaces. Coke formation in furnace tubes and other causes of semi-blockage of tubes will drastically decrease efficiency in an exchanger. Such problems will result in shutdown for cleaning and possible tube and other parts replacements. Many of these problems can be avoided by proper operation and fluid treatment - filtration, corrosion inhibition, furnace firing control.
2. Air pockets**:** The formation of air pockets in exchanges due to improper venting at start up, or build of gas from light materials, will affect the heat transfer rate. This can be avoided by venting all air or gas out at start-up and periodically venting gases as required.
3. Leakage**:** Most leakages occur due to gasket failure - replacement of gaskets might be necessary after some time of operation. Tube failure generally occurs due to corrosion, excessive pressure or by failure of the welded or rolled fitting of the tubes into the tube-sheets

The system needs to be tested before actual operation to prevent any problems mentioned above. Additionally, to obtain a better idea of how the system operates and whether or not the system will be suitable for this project.

3.3 Improvements made to the system

During the test process of the system temperature of the inlet hot water was fluctuating considerably, which would affect the results of this experiment, so another plate heat exchanger was added to the system, to control the inlet hot water temperature.

3.4 Experimental process

In this project comparison is the key part, thus the experiment was designed to compare the difference between counter-current and co-current flows, as well as the effect of increasing flowrate of one stream and keeping the other constant. The data sets were collected when flowrate of cold was kept at 20% (of the 8.65 GPM rotameter's rating) and the hot flowrate was increased from 0.5 GPM to 5.0 GPM by increments of 0.5 GPM; cold kept at 40% and 60% and increasing the hot flowrate as described above respectively. Following, the group kept the hot stream flowrate constant and changed the flowrate of cold stream. Several data sets were collected. Finally, the system was switched to co-current flow pattern and experiments were conducted by keeping the cold flowrate constant at a 20% rating. The data will show the effect of varying the flowrates of the inlet streams on the temperature levels of the outlet streams.

3.5 Data Analysis

The raw data collected during the experiment was used to calculate the heat duty and the overall heat transfer coefficient for the plate heat exchanger. The data was taken when the system reached a steady state during each run or otherwise when the system achieved thermal equilibrium. That was established when the temperature readings remained constant, however some minimal fluctuation would often occur even after this point due to potential fluctuations in the flowrates as a result of the building's utility consumption and the precision of the temperature indicator itself providing an uncertainty in the magnitude of the first decimal for each reading. Additionally, the heat duty was calculated independently for hot water and cold water streams. The heat transferred was determined using the following equation:

$$\mathrm{Q} = \mathrm{m} * \mathrm{Cp} * (\mathrm{Tout} - \mathrm{Tin})$$

where Cp is the heat capacity of water, m is the mass flow rate of the stream. Since Cp is a function of temperature the following power equation was used for Cp of water:

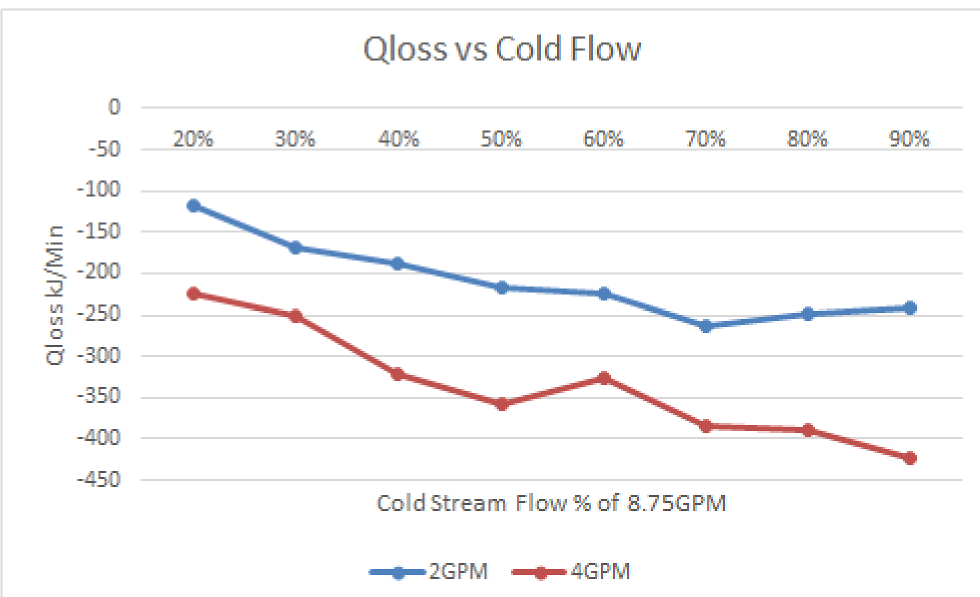
$$Cp = \frac{4.1868kJ}{kg} * K(1.0038 - 2.2459 * 10^{3}\,\frac{(Tout - Tin)}{2} + 2.6257 * 10^{-6}\,\frac{(Tout - Tin)^{2}}{4}).$$

In an ideal closed system, the heat load generated from the hot water stream must equal the heat absorbed by the cold water stream, however this system is not an ideal one and due to heat losses to the environment the two heat loads were not found to be equal. The heat loss is to thus determined by $Qloss = Qh - Qc$. The overall heat transfer coefficient was calculated using equation: $U = Q/(A * \Delta T_{LM})$, where ΔT_{LM} is the log mean temperature difference and is calculated by the following equation: $\Delta T_{LM} = \frac{(\boldsymbol{Th,in - Tc,out}) - (\boldsymbol{Th,out - Tc,in})}{\boldsymbol{Ln}(\frac{\boldsymbol{Th,in - Tc,out}}{\boldsymbol{Th,out - Tc,in}})}$. The estimation of the surface area available for heat transfer is not straightforward when it comes to a plate heat exchanger. There are contradicting methods found in literature as to what is the most accurate way of determining the actual or effective heat transfer area of a plate heat exchanger, with some views arguing that all number of plates should be included in the calculation, others supporting that the top and bottom plates do not contribute in the transfer of heat and therefore should be excluded from the calculation while others offering an area to volume ratio method for calculating the effective heat transfer surface area. The group decided to exclude the top and bottom plates from the calculation which were regarded as insulation barriers were no heat was transferred through these plates and based on the model's specifications as mentioned above where each plate provides a 0.12 m^2 surface area, the total heat transfer surface area for this plate heat exchanger was found to be (10-2)*0.12= 0.096m2. It should also be noted here, that due to the sinusoidal corrugated pattern of each plate surface the effective heat transfer surface area is likely to be in fact larger. However, its exact value is hard to be determined yet it is perhaps the greatest advantage offered from such a plate heat exchanger, which despite its compact size provides through this construction a great available heat transfer surface area per volume and thus high overall heat transfer coefficients. The fouling factor was disregarded because the heat exchanger was brand new and had not been used earlier to this experiment.

4. Results and Discussion

4.1 Heat Loss

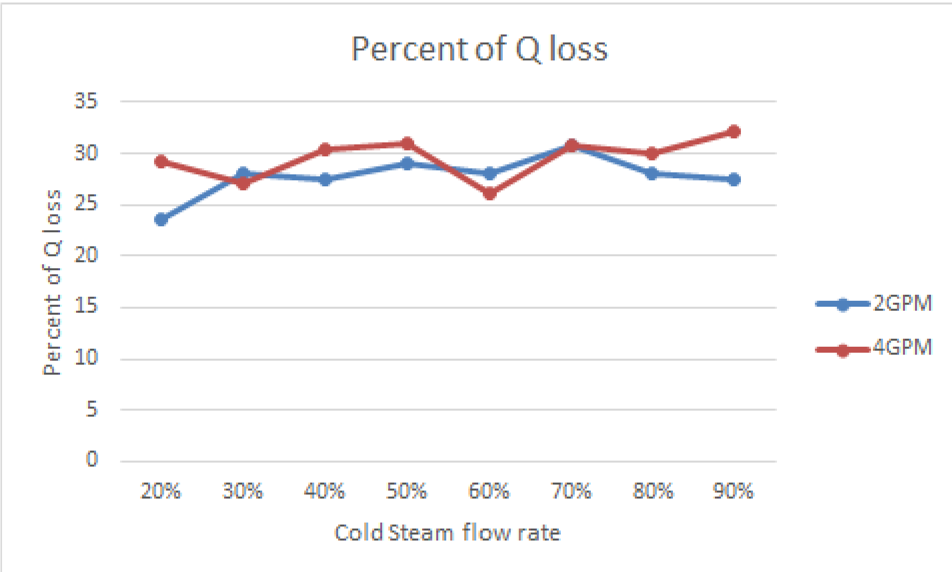
The following graph indicates the trend of heat loss due to flow rates change of cold stream. More heat was emitted to the surrounding when increasing the flow of cold stream and keeping hot stream constant. This is consistent with Newton's law of cooling which suggests that the rate of heat loss from a body is proportional to the temperature difference between the body and its surroundings. The heat loss approximate doubles in absolute magnitude as the flow rate of hot stream doubles. There are some points in the graph that do not match the consistency of the trend, because the temperature of hot stream inlet and the flow rate of cold stream were not constant, affected by the water usage in the building.



Graph. 1 Q_{loss} vs Cold Stream Flow rate

4.2 Percentage of heat loss

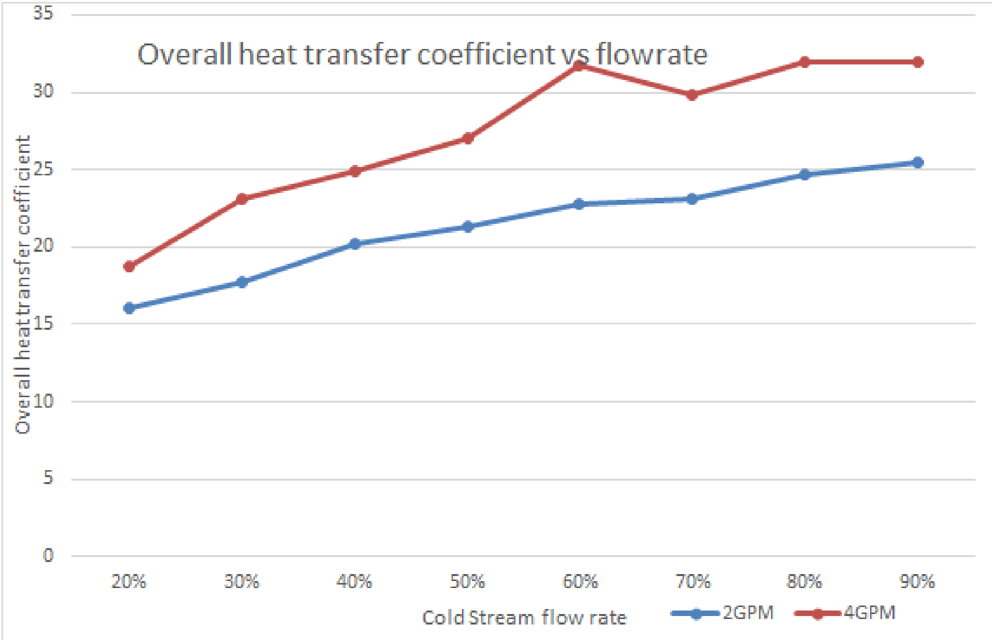
The percentage of heat loss was calculated by dividing the heat loss (Q_h-Q_c) by the amount of heat generated from hot stream (Q_h). The results showed that heat loss ranges between 24%-32% of heat transferred from hot side. This is because the heat exchanger was not insulated, and indicates the experiment raw data was consistent, since the percentage of heat loss was constant.



Graph. 2 Percentage of Q_{loss} vs Cold Stream Flow rate

4.3 Overall heat transfer coefficient versus cold flow rate

The previous graph showed the trend of increasing absolute value of heat loss when flow rates increase. So it is reasonable to find that the overall heat transfer coefficient will increase as flow rates increased. In theory, an increasing flow rate and thus an increasing fluid velocity essentially decreases the thickness of the boundary layer of turbulent flow, leads less resistance for convective heat transfer, and this will increase the overall heat transfer coefficient, and the following data corresponds to the theory.

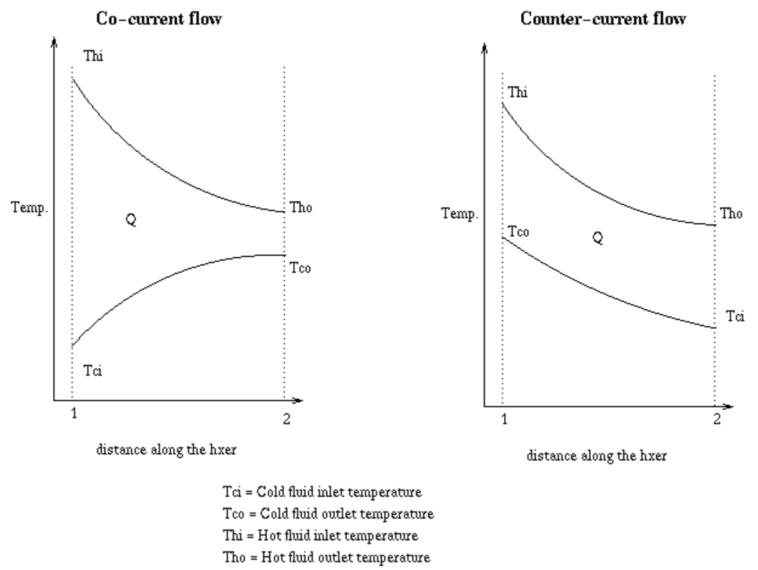


Graph. 3 Overall Heat Transfer Coefficient vs Cold Stream Flow rate

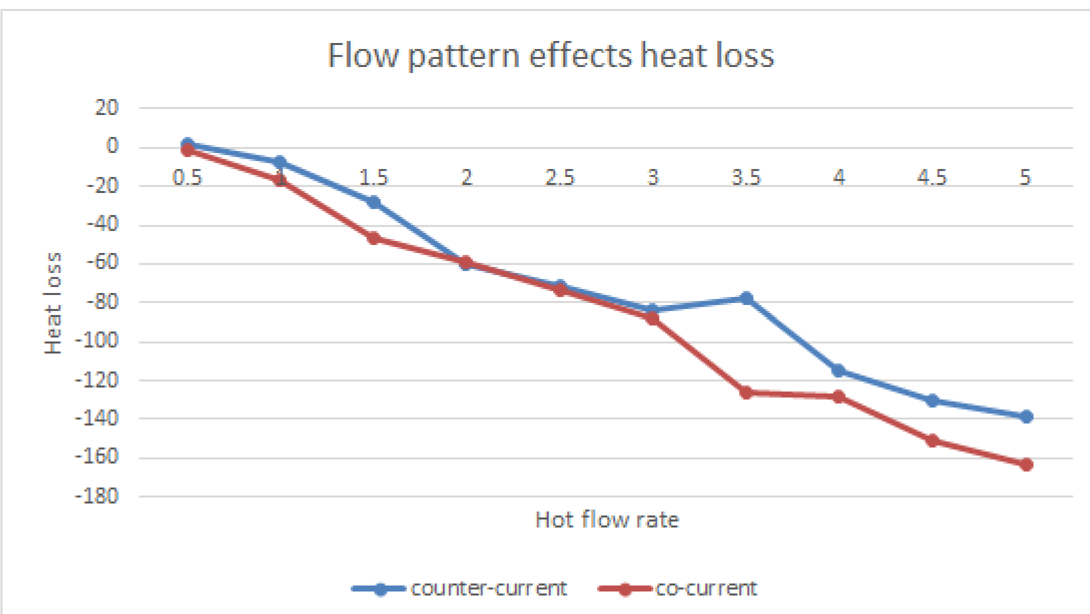
4.4 Effects of flow pattern

The collected and processed data showed that when the streams are counter-current, the heat loss is less than the heat loss of co-current for the same flow rate values. Additionally, counter-current flow has higher value of overall heat transfer coefficient. This corresponds to the heat transfer principle. Heat transfer rate is proportional to the driving force, which is the log mean temperature difference. Co-current flow provides a large temperature difference at the inlet end, but the temperature difference gets smaller towards the outlet end.

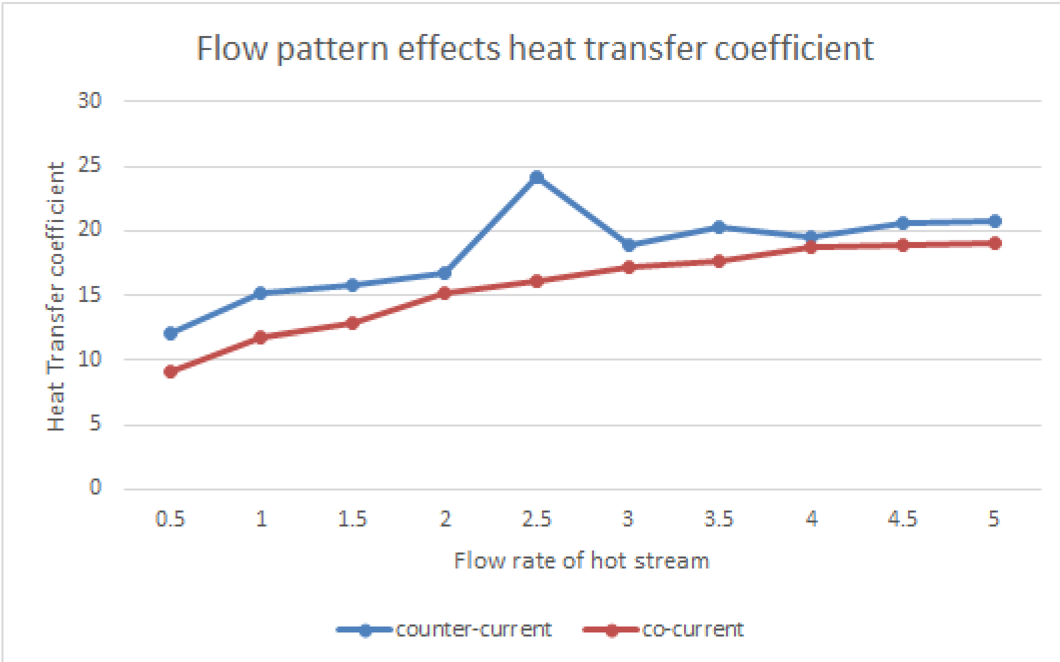
Figure. 6 Temperature vs Distance from Stream Entrance



However, counter-current flow will provide a higher driving force for the same heat exchanger operating under the same condition.



Graph. 4 Effects of Flow pattern on Heat Loss



Graph. 5 Effects of Flow pattern on Heat Transfer Coefficient

5. Conclusions and recommendations

5.1 Conclusions

The team determined the heat loss of the heat exchanger increased as the cooling water flow rate was increased. However, the percentage of heat loss remained constant. According to the data, about 24%-32% heat transferred from the hot stream was lost to the surrounding, irrespectively of the conditions under which the system was running. The overall heat transfer coefficient also increased as the cooling water flow rate was increased. These results were consistent with the theory and replicate the trend the team observed from the double pipe heat exchanger in the Unit Operation Lab. The heat transfer coefficient was calculated for both hot and cold streams, and the heat loss to the surrounding was calculated. Unlike the double pipe heat exchanger that was running steam, the calculation for plate heat exchanger running water in this experiment only requires simple heat exchanging equation, and the Wilson Plot method does not apply in this case. By comparing the results achieved by running the plate heat exchanger and double pipe heat exchanger, it is clear that both systems can show the impact of flow rate on heat transfer coefficient, but only plate heat exchanger will show impact of flow configuration on heat transfer coefficient. Taking into account the exchanger’s small compact size and its significantly smaller energy input requirement, utilizing water from the utility lines contrary to stem supplied by the campus-wide steam plant, the new plate heat exchanger is deemed to be a proper substitution for the current double pipe heat exchanger in the UO lab.

5.2 Recommendations

The results obtained prove the necessity of further testing and experimenting on the heat exchanger for better understanding. The overall trend of the results is consistent, however, some discrepancies were found in the results. More experiments and improvement to the system will solve these issues and improve the experimental analyses.

The team observed that the reading of the rotameters kept changing. Because the utility water was used for this experiment, so the water usage in the building would significantly affect the flow rate of the inlet water. If the inlet water was running from a continuous source, then the flow rate would not change. By adding a reservoir, the utility water would go to the water tank first, and by controlling the valves on the water tank, the flow rate that goes into the heat exchanger could be maintained constant. Additionally, if the utility water gets mixed with the water stored in the reservoir first, the temperature change of inlet stream could be minimized as well.

The goal of this project is to find out if this new plate heat exchanger system would be a good substitution for the current double pipe heat exchanger. The results obtained from this experiment, determined the dependency of heat loss and overall heat transfer coefficient on flowrates, the dependency of heat loss and overall heat transfer coefficient on flow pattern. All the objectives for the double pipe heat exchanger experiment can be achieved by running the plate heat exchanger apparatus, and plate heat exchanger provides better results for the effects of flow directions.

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Appendix

Appendix A Experimental Data

counter- current

Hot	Cold	Pin	Po	T1	T2	T3	T4
2 GPM	20%	38	40	37.2	50.1	13.3	24.3
	30%	37	40	34.6	50	12.5	21.1
	40%	32.5	38	32.3	49.9	11.4	19
	50%	29	38	30.7	49.9	10.5	17.1
	60%	26	37	29.6	50.1	9.9	15.9
	70%	19	34.5	28.5	50.3	9.3	14.6
	80%	15	34	27.2	49.7	8.3	13.3
	90%	10.5	33	25.9	48.4	7.7	12.2
3GPM	90%	10	32	28	45.6	7.3	12.6
	80%	16	33.5	28.3	45.3	7	12.8
	70%	20	35	29	45.9	6.8	13.3
	60%	24.5	37	29.7	46.8	6.6	14.4

	50%	29	37.5	31.6	47.5	6.6	15.1
	40%	32	38	33.1	48.1	6.5	16.6
	30%	37	39.5	35	48.5	6.5	18.4
	20%	39	40	37.7	48.9	6.6	21.2
4GPM	20%	38	40	39.9	49.6	6.6	22.3
	30%	36.5	40	38	49.7	6.5	20.1
	40%	33	37	36.2	49.5	6.4	17.7
	50%	29	37	34.7	49.2	6.3	16.2
	60%	24	36	33.4	49	6.2	15.8
	70%	20	35	32.6	48.3	6.2	14
	80%	15	34	31.4	47.7	6.1	13.3
	90%	9	32	30.4	46.9	6.1	12.4
1.5GPM	80%	32	14	16.6	22.6	12.9	14
	60%	35	22	16.7	22.2	12.7	14
	40%	37	30	17.1	22	13.1	14.6

	20%	39	33	17.5	21.4	12.8	15
0.5	20%			15.4	17.7	14.3	14.9
1				17.3	20.8	14.3	15.9
1.5				20	24.6	14.1	17
2				22.1	27.5	13.5	17.7
2.5				24.5	29.8	13.6	18.9
3				26.6	31.9	13.5	20
3.5				28.4	33.4	13.5	21.1
4				29.9	34.9	13.5	21.5
4.5				30.4	35.2	13.5	22
5				30.5	34.9	13.5	22
5	40%			28.8	34.5	13.4	19.8
4				26.7	33	13.5	18.9
3				23.8	30.2	13.5	17.7
2				20.1	26.3	13.5	16

1				15.9	20.3	13.5	14.4
1	60%			15.7	20.3	13.5	14.3
2				19.4	25.7	13.5	15.5
3				22.9	30	13.5	16.7
4				25.5	32.4	13.4	17.7
5				27.6	34.2	13.4	18.5
5	80%			26.3	33.4	13.4	17.4
4				24.3	31.6	13.3	16.6
3				21.8	29.2	13.3	15.8
2				18.6	25.5	13.2	14.8
1				15.2	19.8	13.2	13.8

Co-Current

Hot	Cold	Pi	Po	T1	T2	T3	T4
0.5	20%	25	24	14.9	17.6	13.4	14

1	26	24.5	19	24.5	13	15.4
1.5	26	24	22.1	28.3	12.8	16.5
2			23.8	29.7	12.8	17.6
2.5			25.9	31.6	12.8	18.6
3			27.5	33	12.7	19.4
3.5			29.8	35.6	12.7	20.5
4			32.5	38.2	12.9	22.1
4.5			33.8	39.3	13	22.7
5			34.8	40	13	23.1

Appendix B Analyzed data

counter- current

Hot	Cold	LMTD	Qh kj/min	Qc	Uh	Uc	Qloss
2 GPM	20%	24.83789	-499.649	382.202	-20.9546	16.02902	-117.447
	30%	25.3481	-598.152	430.366	-24.5807	17.6856	-167.786

		7				2	
	40%	25.57499	-685.288	496.4253	-27.9117	20.21935	-188.863
	50%	25.993	-748.927	532.1567	-30.0132	21.32612	-216.771
	60%	26.28683	-800.801	575.5905	-31.7333	22.80889	-225.21
	70%	26.60259	-852.823	589.7133	-33.3937	23.09116	-263.11
	80%	26.70097	-880.898	632.8444	-34.3659	24.68872	-248.053
	90%	26.17658	-880.898	638.609	-35.0543	25.4127	-242.289
3GPM	90%	26.37369	-1045.38	751.4668	-41.2887	29.68026	-293.912
	80%	26.5068	-1009.06	733.443	-39.6542	28.82291	-275.62
	70%	27.0678	-1003.01	722.263	-38.5996	27.7952	-280.751

		3		3		7	
	60%	27.4883	-1015.11	746.763	-38.4676	28.29852	-268.349
	50%	28.54029	-942.609	683.8988	-34.4034	24.96101	-258.71
	40%	28.98099	-888.358	657.8815	-31.9304	23.6463	-230.477
	30%	29.29272	-798.181	593.3134	-28.3838	21.09858	-204.868
	20%	29.3672	-660.492	505.249	-23.4279	17.92139	-155.243
4GPM	20%	30.20073	-767.599	542.6481	-26.4756	18.71672	-224.95
	30%	30.54015	-927.941	676.7851	-31.6503	23.08386	-251.156
	40%	30.78917	-1056.73	735.0589	-35.7515	24.86869	-321.671
	50%	30.6424	-1153.62	795.295	-39.2165	27.0354	-358.327

		8				2	
	60%	30.1004	-1242.67	917.245	-43.0043	31.74255	-325.423
	70%	30.17786	-1250.77	865.4569	-43.1737	29.87348	-385.317
	80%	29.61737	-1299.45	909.0566	-45.7026	31.97226	-390.39
	90%	29.1027	-1315.68	892.2546	-47.092	31.93628	-423.43
1.5GPM	80%	5.809615	-170.193	139.8344	-30.5157	25.07238	-30.3586
	60%	5.850888	-155.923	125.3686	-27.7599	22.32008	-30.5545
	40%	5.526787	-138.82	98.64941	-26.1642	18.59305	-40.1706
	20%	5.506332	-110.366	77.19637	-20.8786	14.60371	-33.1695

0.5	20%	1.819526	-18.7568	21.09127	-10.7381	12.07461	2.334464
1		3.872628	-63.7916	56.18051	-17.1588	15.11154	-7.61113
1.5		6.714169	-130.277	101.6792	-20.2118	15.77497	-28.5979
2		9.186942	-207.511	147.0456	-23.5288	16.67285	-60.4655
2.5		8	-257.071	185.3294	-33.4728	24.13143	-71.7421
3		12.49039	-310.497	226.9859	-25.8947	18.93004	-83.5116
3.5		13.55848	-343.209	265.0726	-26.3679	20.36492	-78.1361
4		14.84953	-393.594	278.8991	-27.6099	19.56425	-114.695
4.5		14.97389	-426.124	296.1646	-29.6436	20.60285	-129.96
5		14.8558	-434.748	296.164	-30.4838	20.7665	-138.583

		2		6		9	
5	40%	15.04729	-564.016	418.6037	-39.0447	28.97835	-145.412
4		13.64505	-496.65	353.5921	-37.9144	26.99331	-143.058
3		11.36453	-375.402	275.3854	-34.4092	25.24167	-100.017
2		8.31322	-238.467	164.2319	-29.8805	20.57865	-74.235
1		3.891121	-80.276	59.22939	-21.4902	15.85591	-21.0466
1	60%	3.787493	-83.9437	77.19307	-23.0869	21.23025	-6.75063
2		7.854809	-242.34	192.7238	-32.138	25.5581	-49.6164
3		11.23743	-416.788	307.9444	-38.6346	28.54526	-108.844

4		13.35785	-544.315	413.2916	-42.4466	32.22913	-131.024
5		14.93745	-653.729	489.7446	-45.588	34.15246	-163.985

5	80%	14.39441	-703.648	506.8421	-50.9202	36.67817	-196.806
4		12.89678	-576.128	418.4722	-46.5336	33.7998	-157.656
3		10.76477	-434.545	317.3082	-42.0493	30.70473	-117.237
2		7.750297	-265.598	203.2818	-35.6974	27.32178	-62.3167
1		3.640957	-83.9437	76.316	-24.016	21.83377	-7.6277

Co-Current

Hot	Cold	LMTD					
0.5	20%	2.39871	-22.0287	21.0912	-9.5662	9.15910	-0.93745

	5		7		1	
1	7.44271	-100.469	84.19538	-14.0614	11.78382	-16.2732
1.5	10.50045	-175.906	129.6126	-17.4502	12.85785	-46.2929
2	11.54126	-226.852	167.9393	-20.4747	15.15751	-58.9126
2.5	13.04994	-276.597	202.6999	-22.0784	16.17983	-73.897
3	14.19155	-322.286	233.9178	-23.656	17.16968	-88.3687
3.5	16.07927	-398.479	271.9874	-25.8147	17.62021	-126.491
4	17.79266	-449.048	320.3039	-26.2894	18.7521	-128.745
4.5	18.62112	-488.65	337.5229	-27.3351	18.88105	-151.127
5	19.24615	-514.253	351.2843	-27.8331	19.01269	-162.969

Appendix C Sample Calculation for heat loss (Run 1) and percentage of Qloss

Use the Rotameter calibration equation to find the actual water flow rate

1.2599*2GPM-0.1186=2.4 GPM

Converting volumetric water flow to SI units

2.4GPM*3.785liter/gallon=9.084liter/min

Converting volumetric water flow to mass flow rate

9.084liter/min* 1kg/liter=9.084kg/min

Calculate the heat capacity

4.1868kj/kg*K(1.0038-2.2459*10^3(37.2-50.1)/2+2.6257*10^-6((37.250.1)/2)^2)=4.26kJ/kg*K

Calculate the heat transferred

Qh=9.084kg/min * 4.26kJ/kg*K * (37.2-50.1)=-499.65 kJ/min

Qc=382.202kJ/min

Qloss=Qh+Qc=-117.45kJ/min

Overall heat transfer coefficient

U=Q/(A*Tlm)=-20.95kJ/m2*K*min

Qloss%=(Qloss/Qh)*100%=(117.45/499.65)*100%=23.5%

Appendix D Calibration Curve for Rotameters