Universidad de Puerto Rico
Recinto Universitario de Mayagüez
Department of Mechanical Engineering

INME 4032
Mechanical Engineering Lab II

Heat Exchanger Analysis

By
Irvin E. De La Paz
Luis A. Plaza
Hiram González
José R. Vázquez
José M. Otaño

Submitted and Orally Presented to:
Dr. Orlando E. Ruiz. Quiñones

Date of the Experiment: June 16,
Date of Submission: June 17,

Abstract

This experiment consists on taking measurements from a heat exchanger while varying the direction of the flow of the hot and cold fluids. Heat exchangers makes possible the use of machinery and processes that otherwise would generate too much heat to operate. The goals for this laboratory include determining the amount of heat dissipated to the surroundings, the overall efficiency, the temperature efficiency for the hot and cold fluids, overall heat transfer coefficient, the theoretical overall heat transfer coefficient.

The experimental setup consists of a compact heat exchanger in counter and parallel flow that interchanges thermal energy between hot and cold fluids. Data is collected from four thermocouples, two at the cold and hot inlets, and two at the cold and hot outlets. Six measurements of temperature are obtained from each of the thermocouples. The volumetric flow is also measured for both hot and cold fluids.

It was found that the counter flow configuration behaved more closely to the theoretical value, having a maximum error of 44.1%, The maximum error for the parallel configuration was 58.6%, Both errors were not in range of the uncertainty ± 1,298 W/m2*K compared to a theoretical values. Tubing during the experiment was not isolated and contributed to errors in both counter and parallel flows.

Table of Contents

Symbol List and Abbreviations

Table 1 Symbol Table

Symbols:	Meaning
$\boldsymbol{A}$	Cross Sectional Area
$\boldsymbol{C_{p,h}}$	Specific heat for hot fluid
$\boldsymbol{C_{p,c}}$	Specific heat for cold fluid
$\boldsymbol{D}$	Effective Diameter
$\boldsymbol{h_{h,i}}$	Enthalpy for hot inlet fluid
$\boldsymbol{h_{h,o}}$	Enthalpy for hot outlet fluid
$\boldsymbol{h_{c,i}}$	Enthalpy for cold inlet fluid
$\boldsymbol{h_{c,o}}$	Enthalpy for cold outlet fluid
$\boldsymbol{K}$	Thermal Conductivity
$\boldsymbol{\dot{m}_h}$	Mass flow rate for hot fluid
$\boldsymbol{\dot{m}_c}$	Mass flow rate for cold fluid
$\boldsymbol{Nu_D}$	Nussel Number
$\boldsymbol{Pr}$	Prandtl Number
Q	Heat power
Re	Reynolds Number
$\boldsymbol{T_{h,i}}$	Temperature for hot inlet water
$\boldsymbol{T_{c,i}}$	Temperature for cold inlet water
$\boldsymbol{T_{h,o}}$	Temperature for hot outlet water
$\boldsymbol{T_{c,o}}$	Temperature for cold outlet water
U	Overall heat transfer coefficient
$\mathbf{U_d}$	Total Stage Uncertainty
$\mathbf{U_p}$	Total Uncertainty
$\boldsymbol{\eta_h}$	Efficiency in terms hot fluid
$\boldsymbol{\eta_c}$	Efficiency in terms of cold fluid
$\boldsymbol{\eta_m}$	Mean temperature efficiency
$\boldsymbol{\rho}$	Density
$\boldsymbol{\mu}$	Dynamic Viscosity
$\boldsymbol{\nu}$	Kinematic Viscosity

Figure and Table List

Figure List

Table List

Introduction

Heat exchangers are devices that are designed to allow a heat transfer between two fluids. Most of the times, the exchange of thermal energy occurs between two fluids. These types of devices are used in many applications such as refrigeration, cooling engine systems and air conditioning. In order to create the adequate configuration, various parameters are taken into account, such as the fluid properties, the pipe setup and the direction of the fluid. If the flow is directed in the opposite direction, it is called counter flow. In contrast, if the flow follows the same direction of both cold and heat fluids, then it is called a parallel flow.

A plate heat exchanger or compact heat exchanger is one type of heat transfer device commonly used. It consists of various plates stacked one behind the other. These plates are the path through which heat will be transferred from the hot fluid to the cold fluid. A particular advantage for this configuration is the compactness of the device. It is useful for places where there is a need for a heat exchanger that has small size and the pressures are low to medium.

Heat exchangers are theoretically governed by the laws of thermodynamics. These laws provide the energy balance equation for this system. The efficiency can also be determined using various equations that relate the temperatures.

$$\eta_h = \frac{T_{h,inlet} - T_{h,outlet}}{T_{h,inlet} - T_{c,inlet}}$$

Experimental Setup

This experiment consisted in applying a counter and a parallel flow to a compact heat exchanger (see Figure 1). The first setup consisted in setting up a counter flow in the heat exchanger. Figure 2 shows the schematic for the counter flow setup. Four thermocouples are connected, two to the pipe inlets and two to the outlet pipes. A pump will drive the water through a heater which will increase the waters temperature in a closed loop. The cold water will be provided externally at room temperature. Two volumetric flow meters will measure the speed of volume running through each circuit of fluid. Six values were obtained from each of the four thermocouples.

The same procedure was applied in the second part, except that the fluid was connected in a parallel way. Six measurements of temperature were registered for each thermocouple connected to the system.

Figure 1 - Plate Heat Exchanger

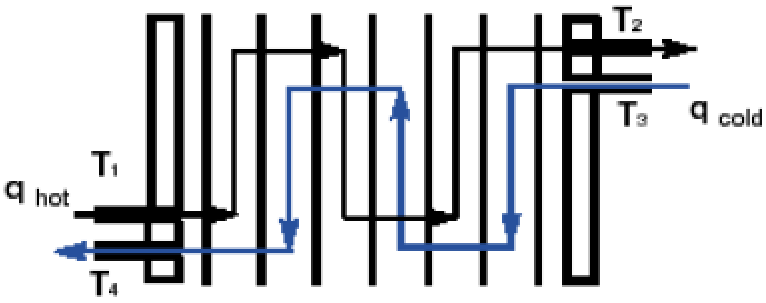

Figure 2 - Design schematic for compact heat exchanger

Experimental Data

Table 2 - Counter flow volumetric and temperature data at inlet and outlet for hot and cold fluid.

Counter Flow							
Run	Flow	L/min	T_ci	T_co	T_hi	T_ho	m^3/s
1	Fhot	2.33	299.6	313.1	325	315	3.88E-05
	Fcold	1.53					2.55E-05
2	Fhot	2.33	299.6	313	325	315	3.88E-05
	Fcold	1.53					2.55E-05
3	Fhot	2.33	299.6	313.2	325	315	3.88E-05
	Fcold	1.53					2.55E-05
4	Fhot	2.33	299.6	313	325	315	3.88E-05
	Fcold	1.53					2.55E-05
5	Fhot	2.33	299.6	313.2	325	315	3.88E-05
	Fcold	1.53					2.55E-05
6	Fhot	2.33	299.6	313.1	324	315	3.88E-05
	Fcold	1.53					2.55E-05

Table 3 - Parallel flow volumetric and temperature data at inlet and outlet for hot and cold fluid.

Parallel Flow							
Run	Flow	L/min	T_ci	T_co	T_ho	T_hi	m^3/s
1	Fhot	2.36	299.8	312.2	317	325	3.93E-05
	Fcold	1.54					2.57E-05
2	Fhot	2.36	299.8	312.2	317	325	3.93E-05
	Fcold	1.54					2.57E-05
3	Fhot	2.36	299.8	312.2	316	325	3.93E-05
	Fcold	1.54					2.57E-05
4	Fhot	2.36	299.8	312.2	317	325	3.93E-05
	Fcold	1.54					2.57E-05
5	Fhot	2.36	299.8	312.2	317	325	3.93E-05
	Fcold	1.54					2.57E-05
6	Fhot	2.36	299.8	312.3	317	325	3.93E-05
	Fcold	1.54					2.57E-05

Analysis and Results:

Important parameters that need to be defined such as the total area (At). This value is taken from the projected heat transmission area 0.008m^2.

$$At = 0.008(Number\ of\ Active\ Plates)(Number\ of\ sides\ per\ plate) = 0.008(5)(2)$$
$$= .08m^2$$

The temperatures used to determine the properties of the fluids where taken by averaging the inlet and outlet temperature of the cold and the hot stream for all the data sets. These temperatures were also used to tabulate all the other properties needed for the following calculations. The Table 4 below shows the values determined by interpolating in the tables of thermodynamic properties of water found on (Incropera, Dewitt, & D.P, 6th Edition) and (Taylor, 1999).

Table 4 - Thermophysical properties for water, hot (320K) and cold (306K)

Flow Properties	
Density	Kg/m^3
ρ_{hot}	990
ρ_{cold}	995
Specific Heat	J/kg*K
Cp_h	4180
Cp_c	4178
Prantl	-
Pr_h	3.77
Pr_c	5.08
Dynamic Viscosity	N*s/m^2
μ_h	0.000577
μ_c	0.000754
Thermal Conductivity	W/m*K
K_h	0.64
K_c	0.622

The mass flow rate for the streams is calculated by multiplying the density by the volume flow rate. This value of the mass flow rate is used to determine the heat of the fluid. The example is provided below for hot water. Same method was used for cold water.

$$\dot{m}_h = \rho_h(Q_h) = 990x\ \ 3.88\text{E} - 05 = 3.89E - 02\ (kg/s)$$

The Table 5 below shows the tabulated data for the mass flow rates. The values in red are for the hot stream and the values in blue are for the cold stream.

Table 5 – Tabulated values for the mass flow rates

Run	Counter Flow		Parallel Flow	
	Vol. Flow (m^3/s)	(kg/s)	Vol. Flow (m^3/s)	mdot (kg/s)
1	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02
2	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02
3	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02
4	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02
5	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02
6	3.93E-05	3.89E-02	3.93E-05	3.89E-02
	2.57E-05	2.55E-02	2.57E-05	2.55E-02

The amount of heat carried by each water stream is calculated using the formula below. The tabulated values are given in Table 6. One example is provided for demonstration purposes.

$$Q_h = \dot{m}_h C_{p,h}\left(T_{h,i} - T_{h,o}\right) = 3.89E - 02 * \ 4180 * (\ 52.2 - 43.5) = 1323 Watts$$

Table 6 - Tabulated values for the heat carried by each stream

Run	Counter Flow	Parallel Flow
	Q from Fluid (W)	Q from Fluid (W)
1	1478.44	-1323
	-1431.09	1416
2	1494.51	-1323
	-1420.48	1432
3	1494.51	-1323
	-1441.69	1432
4	1494.51	-1323
	-1420.48	1384
5	1510.58	-1323
	-1441.69	1432
6	1478.44	-1334
	-1431.09	1400

The next steps involve calculating the heat lost or gained by the fluids, the percentages of these quantities and their respective efficiencies. The equations below where used to tabulate the values in Table 7.

Heat lost or gained:

$$Q = |Q_h| - |Q_c| = 1478.4 - 1431.0 = 47\ W$$

Percentage of losses or gains:

$$P = \frac{|Q_c|}{|Q_h|} x100 = \frac{1431.0}{1478.4} x100 = 96.8\%$$

Hot Efficiency:

$$\eta_h = \frac{T_{h,inlet} - T_{h,outlet}}{T_{h,inlet} - T_{c,inlet}} x100 = \frac{325 - 315}{(325 - 299.6)} x100 = 36.9\%$$

Cold Efficiency:

$$\eta_c = \frac{T_{c,outlet} - T_{c,inlet}}{T_{h,inlet} - T_{c,inlet}} x100 = \frac{313.1 - 299.6}{325 - 299.6} x100 = 54.2\%$$

Mean Temperature Efficiency:

$$\eta_m = \frac{\eta_h + \eta_c}{2} = \frac{36.9 + 54.2}{2} x100 = 45.6\%$$

Table 7 - Efficiencies and other parameters

Run	Cross Flow				Parallel Flow			
	Heat Lost or Gained (W)	Losses or Gains (%)	Eff η (%)	Mean Temp Eff ηm (%)	Heat Lost or Gained (W)	Losses or Gains (%)	Eff η (%)	Mean Temp Eff η_m (%)
1	47.4	96.8	36.9 54.2	45.6	93	93.4	34.3 48.8	41.5
2	74	95	37.3 53.8	45.6	109.3	92.4	34.5 48.6	41.6
3	52.8	96.5	37.3 54.6	46	109.3	92.4	34.6 48.8	41.7
4	74	95	37.3 53.8	45.6	60.5	95.6	33.7 49.2	41.5
5	68.9	95.4	37.5 54.2	45.8	109.3	92.4	34.5 48.6	41.6
6	47.4	96.8	37.1 54.4	45.8	66.1	95.3	34 49.4	41.7

After these parameters are calculated the logarithmic mean temperature difference (LMTD) is tabulated. The behavior of these type of systems depends completely on the type of flow. For a counter flow heat exchanger the LMTD is almost constant, for the case of parallel flow this difference is reduced until both streams achieve an equilibrium temperature. The

Table 8 below shows the values tabulated in order to determine the LMTD. In addition some important variables are defined.

$$\Delta T_1 = T_{hi} - T_{ci} = 325 - 299.6 = 24.9$$

$$\Delta T_2 = T_{ho} - T_{co} = 315 - 313.1 = 2.2$$

$$\Delta \mathrm{T_{LMTD}} = \frac{\Delta \mathrm{T_1} - \Delta \mathrm{T_2}}{\ln\left(\Delta \mathrm{T_1} / \Delta \mathrm{T_2}\right)} \mathrm{CorrectionFactor} = \frac{24.9 - 2.2}{\mathrm{Ln}\left(\frac{24.9}{2.2}\right)} 0.95 = 8.9$$

Table 8 - LMTD values for the hot and cold stream

Run	Counter Flow				Parallel Flow			
	ΔT1	ΔT2	LMTD (K)	Corrected LMTDF (K)	ΔT1	ΔT2	LMTD (K)	Corrected LMTDF (K)
1	24.9	2.2	9.4	8.9	25.4	4.3	11.88	11.286
2	24.9	2.2	9.4	8.9	25.5	4.3	11.91	11.314
3	24.9	2	9.1	8.6	25.4	4.2	11.78	11.191
4	24.9	2.2	9.4	8.9	25.2	4.3	11.82	11.229
5	25.1	2.1	9.3	8.8	25.5	4.3	11.91	11.314
6	24.8	2.1	9.2	8.7	25.3	4.2	11.75	11.163

Finally the overall heat transfer coefficient can be calculated from the experimental values. The Table 9 in next page shows the values tabulated for the heat transfer coefficient.

$$U = \frac{Q_h}{A\Delta T_{LMTD}} \left[\frac{W}{m^2K}\right] = \frac{47.4}{0.08 * 8.9} = 4159\frac{W}{m^2K}$$

Table 9 - Experimental overall heat transfer coefficient

Run	Counter Flow U (W/m2*K)	Parallel U (W/m2*K)
1	4159	2931
2	4204	2923
3	4331	2956
4	4204	2946
5	4288	2923
6	4232	2987

The analytical calculations where done by determining the convection factor for the hot stream and the cold stream. In order to do this the number of Reynolds must be determined. The

Table 10 shows the values tabulated for the Reynolds number

$$Re = \frac{Q(\rho)(d_h)}{A\mu} = \frac{3.89\,E - 02x990x0.003}{0.008 * 0.000577} = 1790$$

Table 10 - Values in the red show the hot stream Reynold's and values in blue show the cold stream Reynold's for Counter Flow

Run	Reynolds
1	1790
	885
2	1792
	885
3	1789
	885
4	1787
	885
5	1792
	885
6	1789
	886

The Chilton-Colburn correlation has determines that for the current Reynolds values the Nusselt to be used is 4.36. With this value, it is possible to solve for the convection heat transfer for both of the streams. The Table 11 shows the corresponding values for each stream, as above these values are color coded.

$$h = \frac{(k)Nu_d}{d} = .64 * \frac{4.36}{0.003} = 15100\left(\frac{\mathrm{W}}{\mathrm{m}^2 * \mathrm{k}}\right)$$

Table 11 - Convection heat transfer for each stream in Counterflow

Run	Conv. Coeff. h (W/m^2*k)
1	15100
	14675
2	15100
	14675
3	15100
	14675
4	15100
	14675
5	15100
	14675
6	15100
	14675

The analytical overall heat transfer coefficient is the determined from the following equation.

$$U = \frac{1}{{}^{1}/_{\mathrm{h_h}} + {}^{1}/_{\mathrm{h_c}}} = \frac{1}{{}^{1}/_{15100} + {}^{1}/_{14675}} = 7442$$

Finally the analytical values are tabulated in Table 12. The percentage of error between these values can be seen also from the Table 12.

Table 12 - Percentage error between calculations. Both parallel and counter flows are shown.

	Parallel Flow			Counter Flow		
Run	Experimental U (W/m2*K)	Analytical U (W/m2*K)	%Error	Experimental U (W/m2*K)	Analytical U (W/m2*K)	%Error
1	3137	7442	58	4159	7442	44
2	3165	7442	57	4204	7442	44
3	3200	7442	57	4331	7442	42
4	3080	7442	59	4204	7442	44
5	3165	7442	57	4288	7442	42
6	3135	7442	58	4232	7442	43

Uncertainty analysis:

The overall heat coefficient (U) is given by;

$$U = \frac{Q_h}{A * \Delta T_{LMTD}} = \frac{\dot{m} * C_p * \Delta T}{A * \frac{(\Delta T1 - \Delta T2)}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)}}$$

All but ΔT, ΔT_1 and ΔT_2 are assumed to be accurate. Thus the first step is to calculate temperature uncertainty. To calculate the instrument uncertainty of the temperature, the zero order uncertainty (U_0) and instrument component error (U_c) must be computed. For a 95% confidence level U_{0e1} =± ½ resolution = 0.05 °C. To estimate the data scatter the standard deviation was calculated for each temperature set and the highest value is used. From Table 13, U_{0e1}=±0.117 °C. The total instrument uncertainty is $U_0 = U_{0e1} + U_{0e2} = 0.167 ºC$

Table 13 Standard deviation of temperature measurement

Standard Deviation (°C)				
	T_ci	T_co	T_hi	T_ho
Counter Flow	0.000	0.089	0.098	0.052
Parallel Flow	0.000	0.041	0.041	0.117

Thus the precision error is; $U_0 = U_{0e1} + U_{0e2} = 0.05 + 0.117 = \mp 0.16°\text{C}$. The instrument component error is taken from the web page (Thermocouple Technical Information) and is $U_c = \mp 1.5°C$. Thus the total stage uncertainty is;

$$U_d = \sqrt{{U_0}^2 + {U_C}^2} = \sqrt{0.167^2 + 1.5^2} = 1.509\ ºC$$

Because ΔT, ΔT1 and ΔT2 are the difference between temperatures, the uncertainty for each of the factors is the sum of each of the uncertainty thus for the first measurement the factors are;

$$\Delta T = (325.2 \mp 1.509) - (316.5 \mp 1.509) = 8.7 \mp 3.08$$

$$\Delta T1 = 25.4 \mp 3.08$$

$$\Delta T2 = 4.27 \mp 3.08$$

Next the sensitivity indexes are calculated using the Kline-McClintock second power relation. All computations are done using the first set of data for the parallel flow run. Because the complex nature of the function of U, the sensitivity indexes were computed numerically using MathCad.

$$\frac{\partial U}{\partial \Delta T} = 342.47$$

$$\frac{\partial U}{\partial \Delta T1} = 75.16$$

$$\frac{\partial U}{\partial \Delta T2} = 248.91$$

And thus uncertainty is given by;

$$U_p = \sqrt{(342.47 * 3.08)^2 + (75.16 * 3.08)^2 + (248.91 * 3.08)^2} = \pm 1{,}298$$

The following figures were also produced as part of the requirements for the laboratory. The figure 3 corresponds to counter flow heat gain:

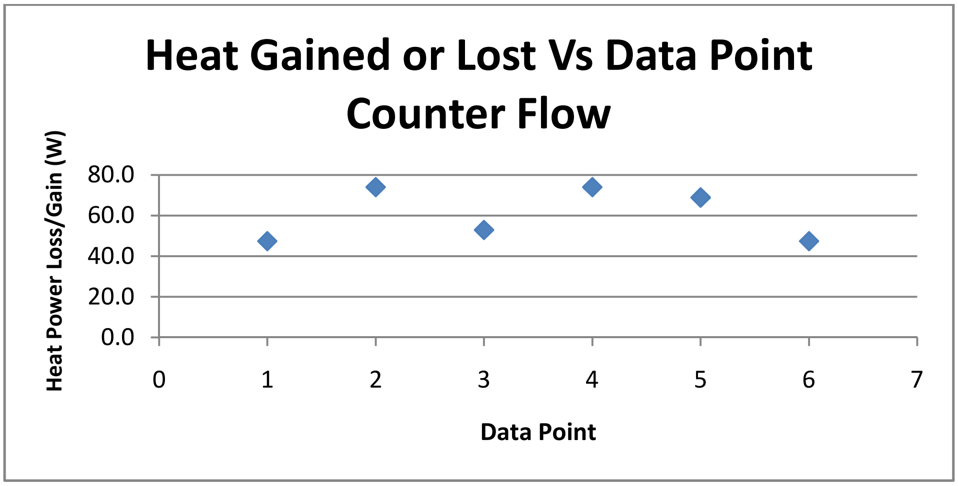

Figure 3 – Heat exchange vs. Data set for counter flow

For heat gain and loss in parallel flow figure 4 was made.

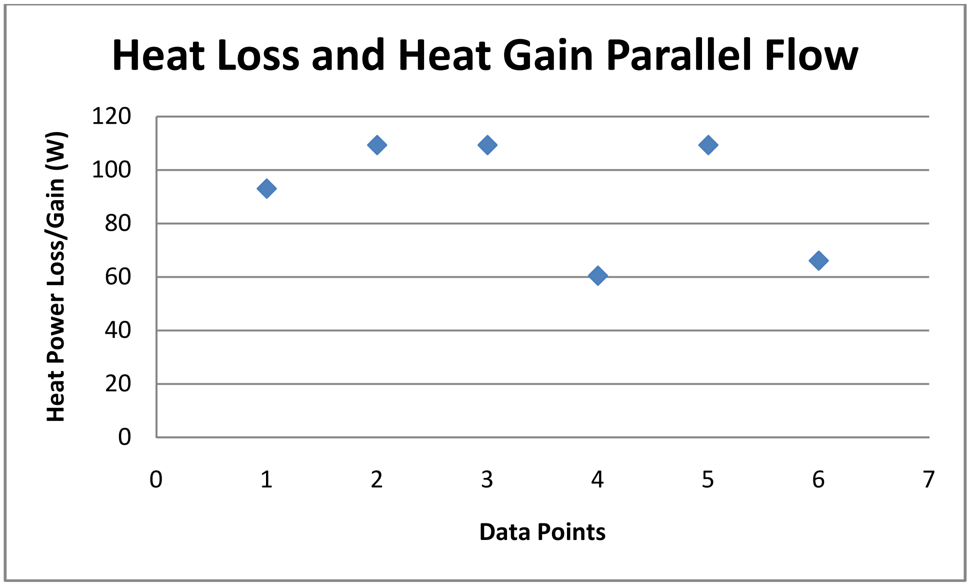

Figure 4 - Heat exchange vs. Data set for parallel flow

These following figures are comparisons between parallel and counter flow mean temperature (figure 6) and LMTD method (Figure 5).

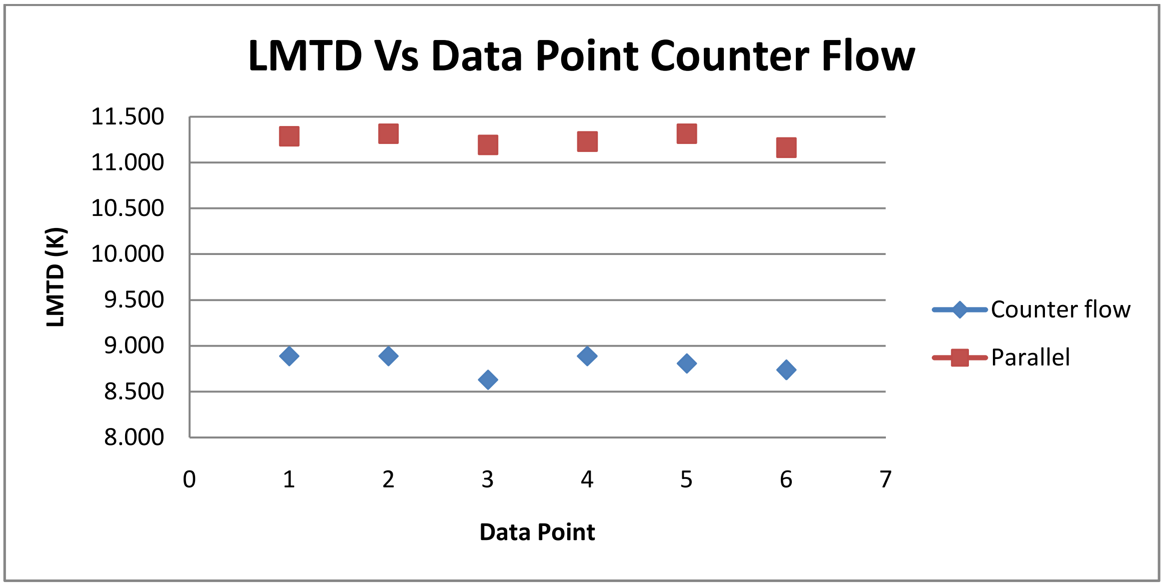

Figure 5 - LMTD vs. data set comparison between parallel and counter flows

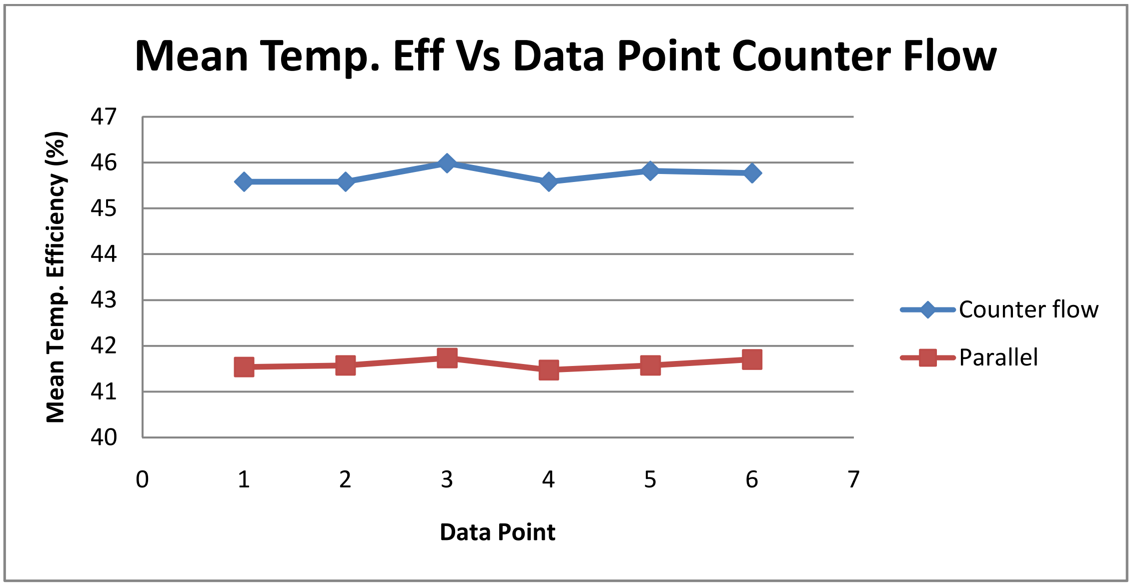

Figure 6 - Mean temperature efficiency vs. data set comparison between parallel and counter flows

Discussion

Through the analysis of the data obtained throughout the experiment for a plate heat exchanger, the properties of said exchanger could be observed under the different flow conditions. Data was obtained for the plate heat exchanger in two configurations: parallel flow and counter flow. The first observation made when interpreting the temperature data was that although in parallel flow there was slightly larger temperature difference at the exits of the heat exchanger, the counter flow configuration was found to be more efficient. This can be attributed to the fact that, while in the parallel configuration, the heat exchanger lost more heat to the environment, as seen in **Error! Reference source not found.**.

After calculating the mean temperature efficiency, log mean temperature difference, and the overall heat exchanger efficiency the experimental overall heat transfer coefficient (U_{exp}) was calculated for both configurations. This value was compared to the theoretical overall heat transfer coefficient obtained for the system. Using this comparison it was determined that the counter flow configuration behaved more closely to the theoretical value, having a maximum error of 44.1%. This error was not in range of the uncertainty analysis with the experimental value being $4159 \pm 1{,}298$ W/m2*K compared to a theoretical value of 7442 W/m2*K. The maximum error for the parallel configuration was 58.6%, this was also not within the range of uncertainty determined, being $3080 \pm 1{,}298$ W/m2*K. The increase in error in the parallel configuration, as mentioned before, can be attributed to the configuration losing more heat to the environment than the counter flow configuration.

Finally, when analyzing the heat exchanger it is important to consider the factors that may have contributed to the error obtained in the data analysis. Many of these factors were taken into consideration when performing the uncertainty analysis such as the tolerances of the equipment used. Another factor that could have contributed to an increase in error are the friction losses within the water tubes that occur after the flow rate is measured. Finally the conditions within the plate heat exchanger are difficult to ascertain and could have effects on its efficiency.

Conclusion

In this experiment we observed the properties and behaviors of heat exchangers. In our case we studied a plate heat exchanger in a parallel flow configuration and a counter flow configuration. Using thermocouples at the entrances and exit flows of the heat exchanger we were able to determine how a plate heat exchanger behaved with both flow configurations. When the overall heat transfer coefficients for both flow configurations were compared to values obtained theoretically, it was determined that the counter flow system was more efficient.

The errors between the experimental and theoretical values were calculated and compared with the tolerances obtained in the uncertainty analysis. The errors were found to be just within the acceptable values given by the uncertainty analysis. In the future, in order to reduce the error, it is recommended that the plate heat exchanger have a layer of insulation to reduce heat losses to the environment. It may also be beneficial to move the flow meters closer to the entrances of the heat exchanger if possible, to reduce the effect of friction loss on the flow.

References

Incropera, Dewitt, & D.P. (6th Edition). *Fundamentals of Heat and Mass Transfer*. New York: John Wiley and Sons.

Taylor, R. P. (1999). *Analysis and Design of Energy Systems*. New Jersey: Prentice Hall, Inc.

Thermocouple Technical Information. (n.d.). Retrieved June 16, 2009, from Peak Sensors: http://www.peaksensors.co.uk/thermocoupledatasheets.html

Appendix