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Gainesville, FL 32607
February 15, 2021

Caitlyn Simonds
University of Florida
201 Criser Hall PO Box 114000
Gainesville, FL 32611

Dear Ms. Simonds:

The main objective of this lab is to investigate the suitability of using the double pipe heat exchanger apparatus for a research project. The values that are measured and used to characterize the heat exchanger system are as follows: evaluating the log mean temperature difference, LMTD for each flow configuration, computing the product of the overall heat transfer coefficient, and the area, UA for each configuration, calculating the predicted UA using the experimentally produced values, calculating the uncertainty in UA for each flow configuration, estimating the fouling factor, and plotting the two flows and configurations vs. axial position in the heat exchanger. These values were determined by changing flowrates on both pipes, allowing for steady-state, and recording all temperatures.

The pressure in the water system allows for water to run through the double pipe heat exchanger system. Once the critical entry length value is reached by the flow of water, the water is then fully developed. The water will be steady state once the temperature of the water remains fixed or has minimal changes in temperature. As the water flows, it can enter the pipes through a counter flow configuration allowing cold water to mix with hot water in a counter flow condition, or a parallel flow configuration allowing cold water to mix with hot water in a parallel flow condition. Calibration was performed for accurate data. A graduated cylinder measured water that came out after 5 seconds at several different flow rates. Assumptions include constant diameter throughout pipes, steady state after water entry length and minimal temperature changes, incompressible flow, and minimal loss due to valves. To determine the LMTD, the temperature difference at the inlet and outlet is found. The UA is determined using the LMTD, mass flow rate, specific heat, and inlet and outlet temperature for a pipe. The fouling factor is found with actual and experimental UA values, and pipe inlet and outlet areas.

The heat exchanger apparatus consists of a counter flow configuration and a parallel flow configuration with valves that allow for changing between the configurations. There are two rotary flow sensors that measure the flow. Four thermistors measure the temperature at the inlets and exits for both configurations, and two more thermistors that measure the temperature at the midpoint. There is a hot water heater that provides hot water. Measurements are made using a LabVIEW VI that records data. Valves on both pipes allow for the flowrate to be controlled manually. Empirical data was recorded by first turning the heat exchanger circuit breaker on, setting the flow to counter flow configuration, setting hot and cold fluid flow rates at 3 L/min, waiting for steady state conditions, and recording all temperatures. The cold flow was then increased to 6 L/min, steady state conditions was allowed to occur, and then all temperatures were recorded. The same procedure was repeated for the parallel flow configuration. To reduce error, each run allowed to steady state before data collection. To ensure safety, lab goggles and close-toed shoes were worn.

The characterization of the double pipe heat exchanger was done successfully. The LMTD, UA, and fouling factor for several different situations are showcased to be as follows. For parallel flow with cold flow at 3.5 L/min and hot flow at 3.5 L/min $LMTD = 18.7^oC, UA = 145 \pm 4.48\frac{W}{C}, R_f'' = 3.6 \times 10^{-5}\frac{m^2K}{W}$, parallel flow with cold flow at 7 L/min and hot flow at 3.5 L/min $LMTD = 18.4^oC, UA = 168 \pm 5.13\frac{W}{C}, R_f'' = -0.00314\frac{m^2K}{W}$, for counter flow with cold flow at 3.5 L/min and hot fluid flow at 3.5 L/min $LMTD = 19.9^oC, UA = 132 \pm 4.41\frac{W}{C}, R_f'' = 0.000515\frac{m^2K}{W}$, cold flow at 7 L/min and hot fluid flow at 3.5 L/min $LMTD = 19.5^oC, UA = 184 \pm 5.11\frac{W}{C}, R_f'' = -0.00355\frac{m^2K}{W}$. This shows that cold flow effects the UA and R_f'', and the flow configuration effects the LMTD.

Sincerely,

Cyril Moran | Nickolas Saavedra | Kellan Wallace

"On my honor, I have neither given nor received unauthorized aid in doing this assignment."

Table 1: The original data sheet is showcased. The Flowmeter calibration constants were calculated using the LABVIEW VI. The constants were used for the various flow rates and then recorded to approximate the linear output of volume with various voltages. These were then inputted into the heat exchanger for the experiment. The flow rate calculations were calculated using the LABVIEW VI. The data was simultaneously saved and then exported to an excel document to be used for more advanced calculations later. All flow rate values were relatively close to the needed value for the experiment and therefore all values are valid to use to find the UA and other calculations later.

Experimental Unit Location	University of Florida Thermodynamics Lab		
Flowmeter calibration constants	A	B	C
Cold flow	0	1.0438	-0.4107
Hot Flow	0	0.962	0.5808

	COLD	**HOT**	**COLD**	**COLD**	**HOT**	**HOT**
	Flow Rate, $\dot{m}_c$ $\frac{kg}{s}$	**Flow Rate,** $\dot{m}_c$ $\frac{kg}{s}$	**T_{in} (C)**	**T_{out} (C)**	**T_{in} (C)**	**T_{out} (C)**
Counter flow	3.41	3.48	22.41	32.31	53.35	41.29
Counter flow	6.98	3.49	22.40	29.25	53.74	37.71
Parallel flow	3.45	3.52	22.32	31.31	52.98	41.66
Parallel flow	6.96	3.47	22.78	28.27	53.01	38.43

Table 2: The flow rate with the biggest LMTD is counter flow 3.5 vs 7. It also caused a difference in generated the largest UA. The largest UA difference between the experimental and theoretical value was also the counter flow rate 3.5 vs 7. The flow rate with the largest fouling factor was counter flow 3.5 vs 3.5 meaning that there were other substances within the heat exchanger that was causing a resistance to heat flow. There were instances in which the fouling factor was negative, meaning that the substances in the heat exchanger helped the heat flow.

	COLD	**HOT**	**COLD**	**COLD**	**HOT**	**HOT**
	Flow Rate, $\dot{m}_c$ $\frac{kg}{s}$	**Flow Rate,** $\dot{m}_h$ $\frac{kg}{s}$	**LMTD,** C°	**UA experimental** $\frac{W}{C^\circ}$	**UA theoretical** $\frac{W}{C^\circ}$	**Rf"** $\frac{(m^2 \cdot k)}{W}$
Counter flow	3.48 ± 7.561	3.41 ± 9.4665	19.94 ± 4.03	132.57 ± 4.4839	146.83 ± 4.4839	$5.153 \times 10^{-4} \pm 1.83 \times 10^{-5}$
Counter flow	3.49 ± 2.9161	6.98 ± 16.223	19.54 ± 5.90	184.28 ± 5.1267	95.54 ± 5.1267	$-3.014 \times 10^{-3} \pm 3.28 \times 10^{-5}$
Parallel flow	3.52 ± 8.072	3.45 ± 9.4509	18.70 ± 3.04	145.72 ± 4.4084	146.83 ± 4.4084	$3.64 \times 10^{-5} \pm 1.68 \times 10^{-5}$
Parallel flow	3.47 ± 4.0173	6.96 ± 17.3609	18.40 ± 2.95	168.30 ± 5.1090	96.13 ± 5.1090	$-3.55 \times 10^{-3} \pm 3.32 \times 10^{-5}$

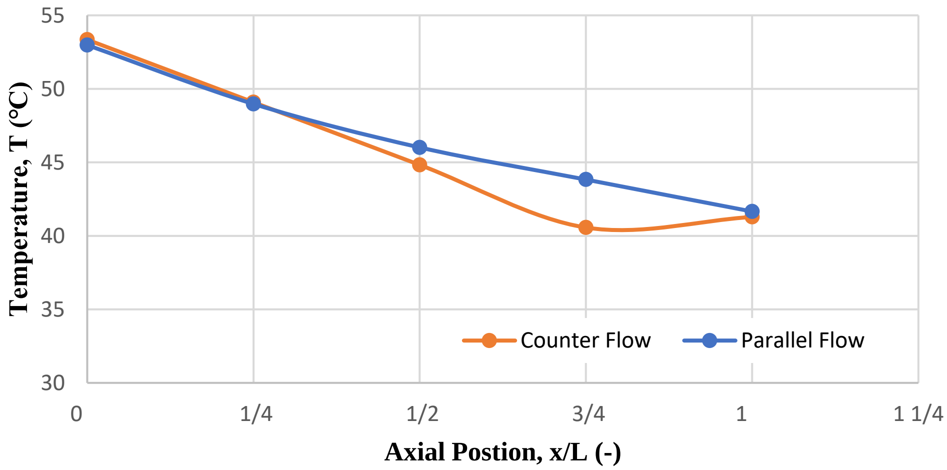

Figure 1: This is a comparison of temperature compared to axial position for equal flows. The predicted calculation was done using the ε-NTU method knowing only the flow rates and T_{in} *and* T_{out}. The counter flow shows a larger decrease in temperature along the pipe. This means the heat flow in the counter flow is larger than in the parallel flow.

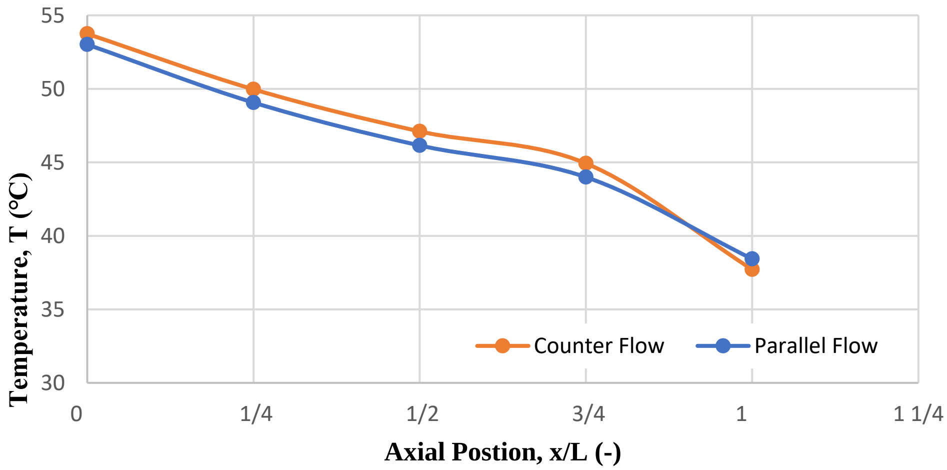

Figure 2: This is a comparison of temperature compared to axial position for unequal flows. The predicted calculation was done using the ε-NTU method knowing only the flow rates and T_{in} and T_{out}. The graph shows a relatively same degree in decrease for both flows until the last portion of the pipe. The counter flows then has a substantially larger decrease in temperature and shows that the heat flow in the counter flow is larger than the parallel flow.

Table 3: The uncertainties in each of the parameters for the actual overall heat transfer coefficient (UA) for each flow configuration. Mass flow rates for both the hot and cold flows contributed the most to the uncertainty in UA. The uncertainty of the mass flow rate for the hot flow is much higher than the cold mass flow rate due to the difference in density of the fluid. The parameter that contributed least to the uncertainty in UA was the specific heat capacity of the flows (C_p). The uneven flow configurations for both heat exchangers resulted in almost double the uncertainty in the mass flow rate of the hot flow compared to the even flow rates. It also caused the difference in the uncertainties of the mass flow rates for the hot and cold flows to be increased substantially. The flow configuration and heat exchanger that had the least uncertainty was the counter flow with even flowrates.

Flowrate and Flow Type	$T_{h,in}$ (°C)	$T_{h,out}$ (°C)	$C_{p,h}$ (J/(kg·°C))	$C_{p,c}$ (J/(kg·°C))	$\dot{m}_h$ (kg/s)	$\dot{m}_c$ (kg/s)	$T_{c,in}$ (°C)	$T_{c,out}$ (°C)	UA (W/°C)
Parallel Flow 3.5 vs. 3.5	± 0.1561	± 1.4042	$\pm 4.85 \times 10^{-7}$	$\pm 4.51 \times 10^{-7}$	± 9.4665	± 7.5609	± 0.1539	± 1.3635	± 4.4839
Parallel Flow 3.5 vs. 7.0	± 0.1113	± 1.9004	$\pm 8.36 \times 10^{-7}$	$\pm 6.53 \times 10^{-7}$	± 16.223	± 2.9161	± 1.0523	± 4.0797	± 5.1267
Counter Flow 3.5 vs. 3.5	± 0.0806	± 0.9041	$\pm 4.85 \times 10^{-7}$	$\pm 4.71 \times 10^{-7}$	± 9.4509	± 8.0712	± 0.0669	± 0.8605	± 4.4084
Counter Flow 3.5 vs. 3.5	± 0.0463	± 1.4585	$\pm 9.19 \times 10^{-7}$	$\pm 9.59 \times 10^{-7}$	± 17.369	± 4.0173	± 0.4763	± 2.7342	± 5.1090

The following calculations are of the parallel flow configuration at 3.5 lpm for the cold and hot flow. In the following calculation ΔT_{lm} is the log mean temperature difference, $T_{h,in}$ is the temperature of the hot fluid inlet, $T_{c,in}$ is the temperature of the cold fluid at the inlet, $T_{h,out}$ is the temperature of the hot fluid at the outlet, $T_{c,out}$ is the temperature of the cold fluid at the inlet, ΔT_1 is the temperature difference between the hot and cold fluids at the inlet, and ΔT_2 is the temperature difference between the hot and cold fluids at the outlet.

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \textit{ Bergman, p. 654, Eqn. 11.15}$$

$$\Delta T_1 = T_{h,in} - T_{c,in} \textit{ Bergman, p. 654, Eqn. 11.16}$$

$$\Delta T_1 = (52.9756\ ℃) - (22.3162\ ℃)$$

$$\Delta T_1 = 30.6594\ ℃$$

$$\Delta T_2 = T_{h,out} - T_{c,out}$$

$$\Delta T_2 = (41.6559\ ℃) - (31.3088\ ℃)$$

$$\Delta T_2 = 10.3471\ ℃$$

$$\Delta T_{lm} = \frac{(10.3471\ ℃) - (30.6594\ ℃)}{\ln\left(\frac{10.3417\ \cancel{℃}}{30.6594\ \cancel{℃}}\right)}$$

$$\Delta T_{lm} = \frac{-20.3123\ ℃}{-1.08623}$$

$$\Delta T_{lm} = 18.6997\ ℃$$

In the following calculation q is the total rate of heat transfer, UA is the actual total heat transfer coefficient multiplied by the area, ΔT_{lm} is the log mean temperature difference, $\dot{m}_h$ is the mass transfer of the hot fluid, $C_{p,h}$ is the specific heat capacity of the hot water, $T_{h,in}$ is the temperature of the hot water at the inlet, and $T_{h,out}$ is the temperature of the hot water at the outlet.

$$q = UA \cdot \Delta T_{lm} \textit{ Bergman, p. 654, Eqn. 11.14}$$

$$UA = \frac{q}{\Delta T_{lm}}$$

$$q = \dot{m}_h \cdot C_{p,h} \cdot \left(T_{h,in} - T_{h,out}\right) \textit{ Bergman, p. 651, Eqn. 11.6b}$$

$$q = \left(0.057706\ \frac{kg}{s}\right) \cdot \left(4182.45\ \frac{J}{kg \cdot ℃}\right) \cdot (52.9756\ ℃ - 41.6559\ ℃\ )$$

$$q = \left(241.354\ \frac{\cancel{kg} \cdot J}{s \cdot \cancel{kg} \cdot \cancel{℃}}\right) \cdot (11.3197\ \cancel{℃})$$

$$q = 2724.929\ W$$

$$UA = \frac{2724.929\ W}{18.6997\ ℃}$$

$$UA = 145.72\ \frac{W}{℃}$$

In the following calculation UA is the predicted overall heat transfer coefficient multiplied by area, h_i is the convection coefficient for the hot flow, h_o is the convection coefficient for the cold flow, D_o is the outer diameter of the copper pipe, D_i is the inner diameter of the copper pipe, A_i is the surface area of the inside of the copper pipe, A_o is the surface area of the outside of the copper pipe, k is the thermal conductivity of the copper pipe, and L is the effective length of the heat exchanger.

$$UA = \left[\frac{1}{h_i \cdot A_i} + \frac{\ln\left(\frac{D_o}{D_i}\right)}{2 \cdot \pi \cdot k \cdot L} + \frac{1}{h_o \cdot A_o}\right]^{-1} \quad \textit{Bergman, p. 650, Eqn. 11.5}$$

$$UA = \left[\frac{1}{\left(4020.33\ \frac{W}{\cancel{m^2} \cdot K}\right) \cdot (0.1046\ \cancel{m^2})} + \frac{\ln\left(\frac{0.0128\ \cancel{m}}{0.0109\ \cancel{m}}\right)}{2 \cdot \pi \cdot \left(403\ \frac{W}{\cancel{m} \cdot K}\right) \cdot (3.05\ \cancel{m})} + \frac{1}{\left(1843.86\ \frac{W}{\cancel{m^2} \cdot K}\right) \cdot (0.1229\ \cancel{m^2})}\right]^{-1}$$

$$UA = \left[\left(2.377\ \times 10^{-3}\ \frac{K}{W}\right) + \left(2.081 \times 10^{-5}\ \frac{K}{W}\right) + \left(4.413\ \times 10^{-3}\ \frac{K}{W}\right)\right]^{-1}$$

$$UA = \left[6.811\ \times 10^{-3}\ \frac{K}{W}\right]^{-1}$$

$$UA = 146.825\ \frac{W}{{}^{\circ}\mathrm{C}}$$

In the following equation R_f'' is the fouling factor, $(UA)_A$ is the actual overall heat transfer coefficient multiplied by the area, $(UA)_T$ is the predicted overall heat transfer coefficient, A_i is the surface area of the inside of the copper pipe, and A_o is the surface area of the outside of the copper pipe.

$$R_f'' = \frac{\frac{1}{(UA)_A} - \frac{1}{(UA)_T}}{\frac{1}{A_i} + \frac{1}{A_o}}$$

$$R_f'' = \frac{\frac{1}{145.719\ \frac{W}{K}} - \frac{1}{146.825\ \frac{W}{K}}}{\frac{1}{0.1046\ m^2} + \frac{1}{0.1229\ m^2}}$$

$$R_f'' = \frac{\left(3.3776 \times 10^{-5}\ \frac{K}{W}\right)}{17.696\ m^{-2}}$$

$$R_f'' = 2.9204 \times 10^{-6}\ \frac{m^2 \cdot K}{W}$$

In the following calculation $T_{h,out,\,½}$ is the temperature of the hot fluid at half the length of the pipe, $T_{h,in}$ is the temperature of the hot fluid at the inlet, $T_{c,in}$ is the temperature of the cold fluid at the inlet, ε is the effectiveness of the parallel-flow heat exchanger, $C_{p,h}$ is the specific heat capacity of the hot flow, $\dot{m}_h$ is the mass flow rate of the hot fluid, C_{min} is the smallest heat capacity rate of the two flows, and q_{max} is the maximum rate of heat transfer possible.

$$T_{h,out,\frac{1}{2}} = T_{h,in} - \frac{\varepsilon \cdot q_{max}}{\dot{m}_h \cdot C_{p,h}}$$

$$q_{max} = C_{min} \cdot \left(T_{h,in} - T_{c,in}\right)$$

$$q_{max} = \left(241.353\ \frac{J}{s \cdot \cancel{K}}\right) \cdot (52.9756\ \cancel{℃} - 22.3162\ \cancel{℃})$$

$$q_{max} = 7399.738\ W$$

$$T_{h,out,\frac{1}{2}} = (52.9756\ ℃) - \frac{0.2271 \cdot (7380.46\ \cancel{W})}{\left(0.057706\ \frac{\cancel{kg}}{\cancel{s}}\right) \cdot \left(4180.144\ \frac{\cancel{J}}{\cancel{kg} \cdot ℃}\right)}$$

$$T_{h,out,\frac{1}{2}} = (52.9756\ ℃) - \frac{1680.203}{241.353\ \frac{1}{℃}}$$

$$T_{h,out,\frac{1}{2}} = (52.9756\ ℃) - (6.9616\ ℃)$$

$$T_{h,out,\frac{1}{2}} = 46.014\ ℃$$

The following uncertainty calculation is from the counter-flow exchanger at uneven flowrates. In the following equation $\dot{m}_h$ is the mass flow rate of the hot flow, $\dot{m}_c$ is the mass flow rate of the cold flow, $C_{p,h}$ is the specific heat capacity of the hot flow, $C_{p,c}$ is the specific heat capacity of the cold flow, $T_{h,in}$ is the temperature of the hot flow at the inlet, $T_{h,out}$ is the temperature of the hot flow at the outlet, $T_{c,out}$ is the temperature of the cold flow at the outlet, $T_{c,in}$ is the temperature of the cold flow at the inlet, *UA* is the overall heat coefficient. The uncertainty of each variable will be represented by '*U*' of the variable in question; the uncertainty of *UA* is *U(UA).*

$$UA = \frac{\left[\dot{m}_h \cdot C_{p,h} \cdot \left(T_{h,in} - T_{h,out}\right) + \dot{m}_c \cdot C_{p,c} \cdot \left(T_{c,out} - T_{c,in}\right)\right]}{\dfrac{(\mathrm{T_{h,out}} - \mathrm{T_{c,in}}) - (\mathrm{T_{h,in}} - T_{c,out})}{\ln\left(\dfrac{\mathrm{T_{h,out}} - \mathrm{T_{c,in}}}{\mathrm{T_{h,in}} - T_{c,out}}\right)}}$$ *Bergman, p. 654, Eqn. 11.14*

$$U(UA) = \sqrt{\left(\frac{\partial UA}{\partial \dot{m}_h} U(\dot{m}_h)\right)^2 + \left(\frac{\partial UA}{\partial C_{p,h}} U\left(C_{p,h}\right)\right)^2 + \left(\frac{\partial UA}{\partial T_{h,in}} U\left(T_{h,in}\right)\right)^2 + \left(\frac{\partial UA}{\partial T_{h,out}} U\left(T_{h,out}\right)\right)^2 + \left(\frac{\partial UA}{\partial \dot{m}_c} U(\dot{m}_c)\right)^2 + \dots}$$

$$\sqrt{\left(\frac{\partial UA}{\partial C_{p,c}} U\left(C_{p,c}\right)\right)^2 + \left(\frac{\partial UA}{\partial T_{c,out}} U\left(T_{c,out}\right)\right)^2 + \left(\frac{\partial UA}{\partial T_{c,in}} U\left(T_{c,in}\right)\right)^2}$$

$$U(UA) =$$

$$\sqrt{\left((5818.083) \cdot (2.436\ x\ 10^{-3}\ )\right)^2 + \left((0.08026) \cdot (0.0405)\right)^2 + \left((-15.107) \cdot (0.10151)\right)^2 + \left((21.523) \cdot (0.10302)\right)^2 + \dots}$$

$$\sqrt{\left((2485.3) \cdot (0.002712)\right)^2 + \left((0.06919) \cdot (0.06919)\right)^2 + \left((78.275) \cdot (0.10018)\right)^2 + \left((-84.6919) \cdot (0.1000)\right)^2}$$

$$U(UA) = \sqrt{(17.473) + (9.186\ x\ 10^{-7}) + (0.0462) + (1.458) + (3.949) + (9.589\ x\ 10^{-7}) + (2.734) + (0.4762)}$$

$$U(UA) = 5.109\ \frac{W}{°\mathrm{C}}$$

Lab Report Participation Log

Name	**Date**	**Hours Worked**	**Description of Tasks Performed**
Cyril Moran	3/01/2022	8	Calculated most of the UA, LMDT, and all uncertainties.
Nickolas Saavedra	02/25/2022	1	Uploaded lab data
Cyril Moran	3/02/2022	2	Formatted pages 6-7
Cyril Moran	3/05/2022	6	Formatted pages 6-7, calculated UA and all uncertainties
Cyril Moran	3/23/2022	2	Added uncertainties of Rf and theoretical UA
Kellan Wallace	3/23/2022	8	Worked on pages 2-5, calculated uncertainty of LMDT
Nickolas Saavedra	3/21/2022	3	Wrote letter
Nickolas Saavedra	3/22/2022	7	Finished letter, Worked on page 2 and 3, formatted report,
	Total Hours:	37	