University of California Riverside

ME170B Experimental Techniques: Lab 3
Concentric Heat Exchanger

Group A5
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Abstract

The purpose of this experiment is to evaluate the heat transfer efficiency between two fluids flowing in a concentric pipe heat exchanger to determine which flow configuration, parallel or counterflow, is more effective. In the concentric flow heat exchanger, cold and hot water flow either in the same direction (parallel flow) or in opposite directions (counterflow), depending on the valve arrangement. We hypothesized that the counterflow configuration in a concentric pipe heat exchanger will exhibit higher heat transfer efficiency, as indicated by greater overall heat transfer coefficients, higher effectiveness ratios, and lower temperature gradients compared to the parallel flow configuration. To assess the performance of each configuration, key parameters were analyzed, including the heat transfer coefficient, the logarithmic mean temperature difference (LMTD), and the number of transfer units (NTU). Data were collected for three different cold flow rates, with four corresponding hot flow rates for each cold flow rate, under both counterflow and parallel flow conditions. The overall heat transfer coefficients for the inner and outer tubes were determined to be 584.1 W/m^2K and 375.4 W/m^2K for counterflow, and 1240.1 W/m^2K and 356.5 W/m^2K for parallel flow. The calculated **m** values were 0.88 for parallel flow and 0.82 for counterflow, while the effectiveness ratios were found to be 26% for parallel flow and 41.5% for counterflow.

Introduction

Heat exchangers, which facilitate heat transfer between two fluids at different temperatures separated by a solid wall, are critical in many engineering applications, including refrigeration, air-conditioning systems, engines, and computers. Heat exchangers are typically used in engineering systems that require constant heating or cooling of flowing fluid. A common example of this are the evaporators and condensers in the vapor compression refrigeration cycle. The working principle of heat exchangers is that two fluids flow at different temperatures, allowing heat to flow between the two fluids. The higher the temperature difference between the two fluids at any given location, the higher the rate of heat transfer. This raises the question of the most effective flow arrangement between the two fluids.

This lab investigates the effectiveness of two flow arrangements, parallel and counterflow, in a water-water heat exchanger to achieve specific heat-transfer objectives. In the parallel flow configuration, cold and hot fluids enter at the same end, flow in the same direction, and exit at the same end. Conversely, in the counterflow configuration, the fluids enter at opposite ends, flow in opposite directions, and exit at different ends. Temperature data points were recorded throughout the system and plotted as a function of distance, allowing for the calculation of the log mean temperature difference (LMTD) and the overall heat transfer coefficients at the inlet (Ui) and outlet (Uo).

It is hypothesized that a higher rate of heat transfer will be observed in the counter-flow configuration, and that it will maintain a higher temperature difference throughout the length of the heat exchanger. It is also hypothesized that the LMTD for the counterflow configuration will be higher than that of the parallel flow.

Theory

To determine which flow arrangement of heat exchanger is most effective, parallel or counter-flow, one must relate the inlet and outlet temperatures of both the fluids, cold and hot, with the heat transfer rate. To facilitate these calculations it was assumed that the pipes were clean and unfinned, tube resistance was negligible, and no heat was lost to the environment due to the pipes being insulated. Since the fluids are not undergoing phase change and constant specific heats are assumed, the following equations can be used:

$$Q_h = \dot{m}_h c_{p,h} \left(T_{h,i} - T_{h,o}\right) \tag{1}$$

$$Q_C = \dot{m}_c c_{p,c} \left(T_{c,o} - T_{c,i}\right) \tag{2}$$

Where *c* and *h* refer to the cold and hot fluids respectively. Subscripts i and o refer to the inner and outer tube surfaces.

$$\dot{m} = \rho V$$

(3)

Applying analysis of heat exchanger, the heat transfer rate can be expressed similarly to Newton's law of cooling.

$$Q = UA\Delta T_{\mathrm{lm}} \tag{4}$$

Where ΔT_{lm} is the logarithmic mean temperature difference between the outlet and inlet.

$$\Delta T_{\mathrm{lm}} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{5}$$

For parallel flow configuration, the temperature difference is given.

$$\Delta T_1 = T_{h,i} - T_{c,i} \tag{6}$$

$$\Delta T_2 = T_{h,o} - T_{c,o} \tag{7}$$

For counter-flow configuration, the temperature difference is given.

$$\Delta T_1 = T_{h,i} - T_{c,o} \tag{8}$$

$$\Delta T_2 = T_{h,o} - T_{c,i} \tag{9}$$

For a pipe of a certain thickness, the overall heat transfer coefficient is different for the inner and outer surfaces. The product of the overall heat transfer coefficient and surface area remain constant.

$$A = U_i A_i = U_o A_o$$

(10)

By utilizing Eqs. 1 and 2, the heat transfer energy can be determined. Solving Eq. 4 for U and substituting the calculated values for Qh and QC, the heat transfer coefficient was computed. The effectiveness ratio of the heat exchanger is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate. This ratio is calculated differently for parallel and counterflow configurations, as outlined below.

$$\epsilon = \frac{1 - \exp[-\mathrm{NTU}(1 + C_r)]}{1 + C_r} \qquad \text{(Parallel flow)}$$

(11)

$$\epsilon = \frac{1 - \exp[-\mathrm{NTU}(1 - C_r)]}{1 - C_r \exp[-\mathrm{NTU}(1 - C_r)]} \quad (C_r < 1) \quad \text{(Counterflow)}$$

(12)

$$\epsilon = \frac{\mathrm{NTU}}{1 + \mathrm{NTU}} \quad (C_r = 1) \quad \text{(Counterflow)}$$

(13)

The NTU (Number of Transfer Units) is a dimensionless parameter commonly used in heat exchanger analysis to evaluate effectiveness without the need to calculate boundary temperatures.

$$\text{NTU} = \frac{UA}{C_{\min}}$$

(14)

And Cr is the heat capacity ratio:

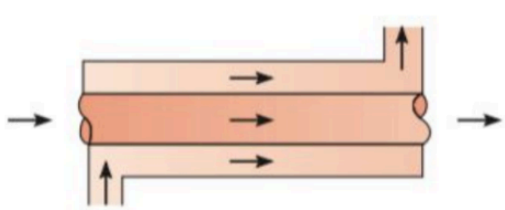
$$C_r = \frac{C_{\min}}{C_{\max}}$$

(15)

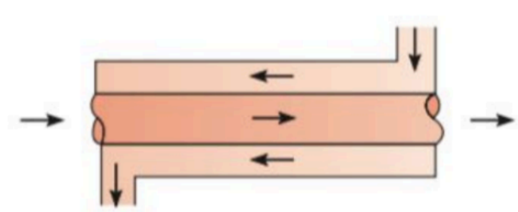
The values of Cmin and Cmax are determined from the heat capacities of the hot water (Ch) and cold water (Cc). Based on the calculated values, the minimum and maximum heat capacities are utilized accordingly.

$$C_h = \dot{m}_h C_{p,h} \tag{16}$$

$$C_c = \dot{m}_c C_{p,c} \tag{17}$$



(a)



(b)

***Figure 1**:* Concentric tube heat exchangers. (a) Parallel flow. (b) Counterflow.

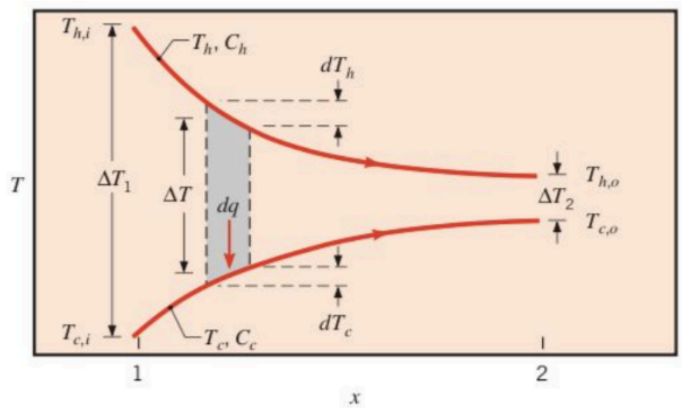


Figure 2: Temperature distributions for a parallel-flow heat exchanger.

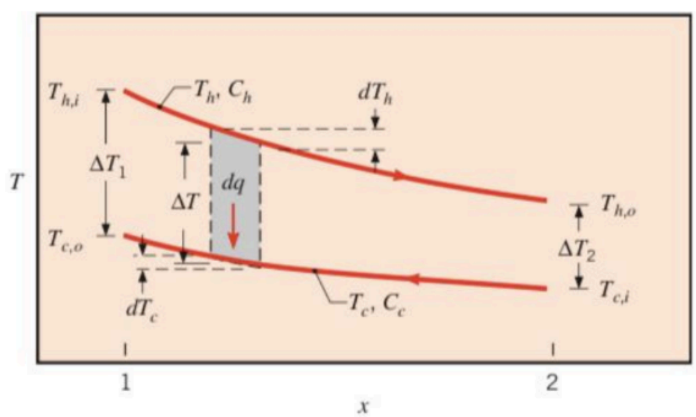


Figure 3: Temperature distributions for a counterflow heat exchanger.

Experimental Methods

Calibration of the concentric tube heat exchanger was not required. The operation of the apparatus involved controlling both hot and cold water flow by adjusting valves as necessary to achieve the desired fluid flow rates. The volume flow rate, measured in gallons per minute (GPM), was regulated using valves A (cold water) and B (hot water). Thermocouples were already strategically installed at the inlets, midpoints, and outlets of the hot and cold water lines to accurately display temperatures and assist in determining the most effective heat exchanger design. Additionally, four directional control valves, located at the base of the apparatus near the water pipes, were used to configure the flow direction. Before conducting the experiment, operators should familiarize themselves with the potential hazards associated with the concentric tube heat exchanger apparatus. The first step in the experiment involves turning on both the hot and cold water lines and allowing the water to flow for several minutes until stable temperature readings are achieved. Once balanced, the operator determines the desired flow configuration, parallel or counter-flow, and adjusts the valves accordingly. Opening valves 1 and 3 (with valves 2 and 4 closed) establishes a parallel flow, while opening valves 2 and 4 (with valves 1 and 3 closed) enables counter-flow. Valves A and B were adjusted to the specified flow rate values to record temperature readings for three different cold water flow rates, each paired with four different hot water flow rates. To ensure accurate temperature measurements, several minutes were allowed between each adjustment of valves A or B. Digital thermometers (Taylor 9940) displayed the thermocouple readings, which captured the temperature changes along the heat exchanger.

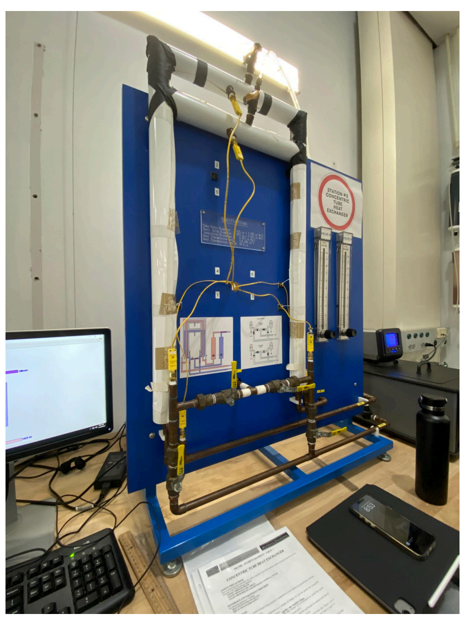


Figure 4: Concentric heat exchanger apparatus.

Results

The data from this experiment demonstrates that counterflow was the most effective flow type for removing heat from the system. The overall heat transfer coefficient was consistently higher for counterflow compared to parallel flow. Using the analyzed data and Equation 5, the average logarithmic mean temperature difference (ΔT_{lm}) for counterflow was 17.7 K, compared to 13.8 K for parallel flow. The experiment utilized three different cold water flow rates and four different hot water flow rates for each cold water flow. Cold water flow rates ranged from 0.3 GPM to 0.7 GPM, while hot water flow rates varied from 0.3 GPM to 0.6 GPM. For counterflow, the overall heat transfer coefficients Ui and Uo were 584.1 $W/m^2 \cdot K$ and 375.4 $W/m^2 \cdot K$, respectively, while for parallel flow, Ui and Uo were 1240.1 $W/m^2 \cdot K$ and 356.5 $W/m^2 \cdot K$, respectively. These coefficients, which quantify heat loss (Ui) and heat gain (Uo), confirm that counterflow reduces heat more effectively than parallel flow. Throughout the experiment, counterflow exhibited a much larger temperature difference between the hot and cold water flows, as illustrated in the temperature versus distance graphs for parallel flow (Figures 5-7) and counterflow (Figures 8-10).

Analysis of the data from both configurations indicates that the heat transfer coefficient at the inlet depends on the Reynolds and Prandtl numbers. The calculated values of m were 0.88 for parallel flow and 0.82 for counterflow, showing a 1.97% difference compared to the literature value of 0.8. The heat exchanger temperatures reached equilibrium much more quickly in the counterflow configuration, as evidenced by the steeper slopes of the graphs, confirming its higher efficiency. The effectiveness ratio, determined using the NTU method, was 26% for parallel flow and 41.5% for counterflow. These results are visually represented in the graphs below.

Figures 5-7 show the results for 12 parallel flow heat exchanger experiments. Each figure has a specified cold water flow rate, and contains experiments for 4 hot water flow rates. The temperature profile of the cold water and hot water stream through the heat exchanger are displayed. Figures 8-10 show the same results for the counterflow heat exchanger experiments. Figures 11 & 12 showcase the relationship of effectiveness and NTU for the parallel and flow heat exchanger respectively.

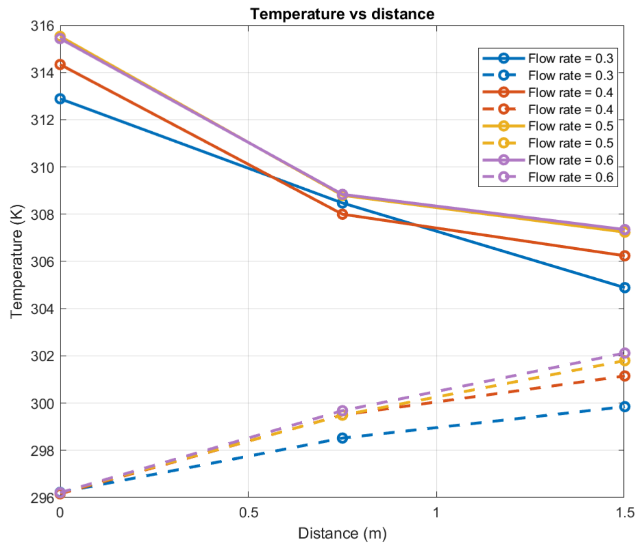


Figure 5: Parallel Flow Temperatures with cold water flow rate at 0.3 GPM at various hot flow rates

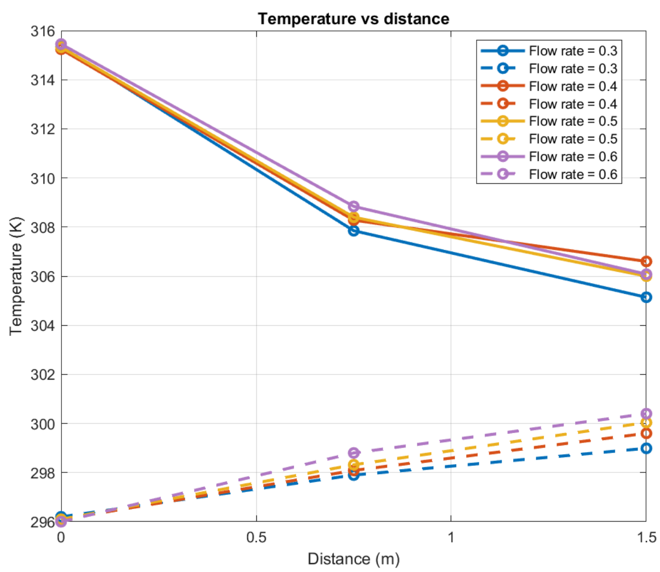


Figure 6: Parallel Flow Temperatures with cold water flow rate at 0.5 GPM at various hot flow rates

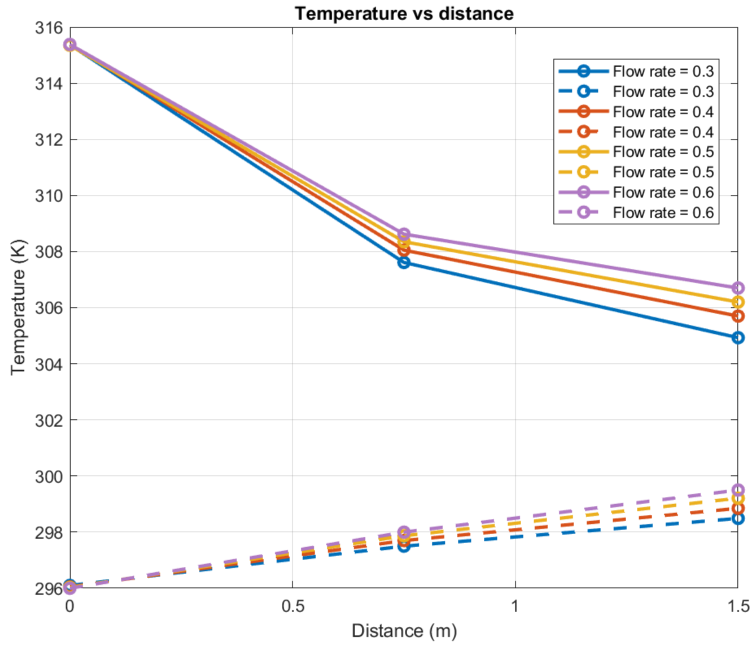


Figure 7: Parallel Flow Temperatures with cold water flow rate at 0.7 GPM at various hot flow rates

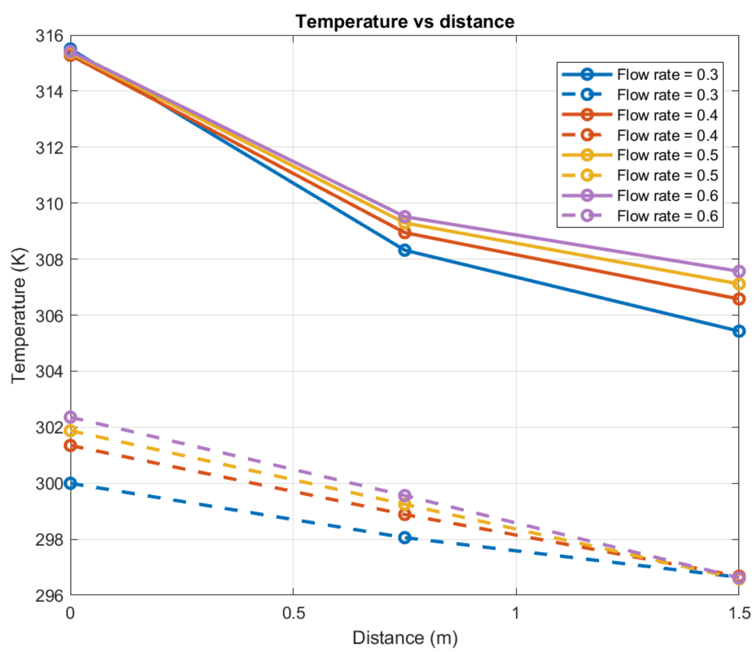


Figure 8: Counter Flow Temperatures with cold flow rate at 0.3 GPM at various hot flow rates

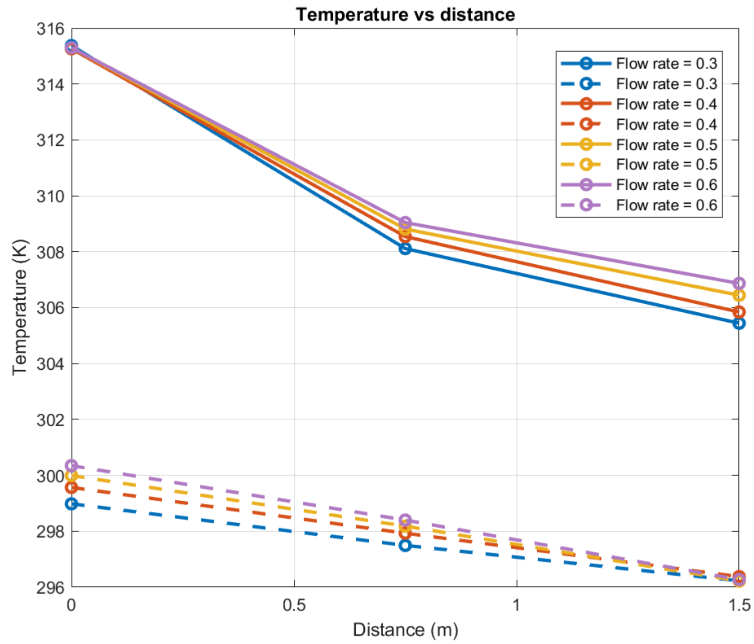


Figure 9: Counter Flow Temperatures with cold flow rate at 0.5 GPM at various hot flow rates

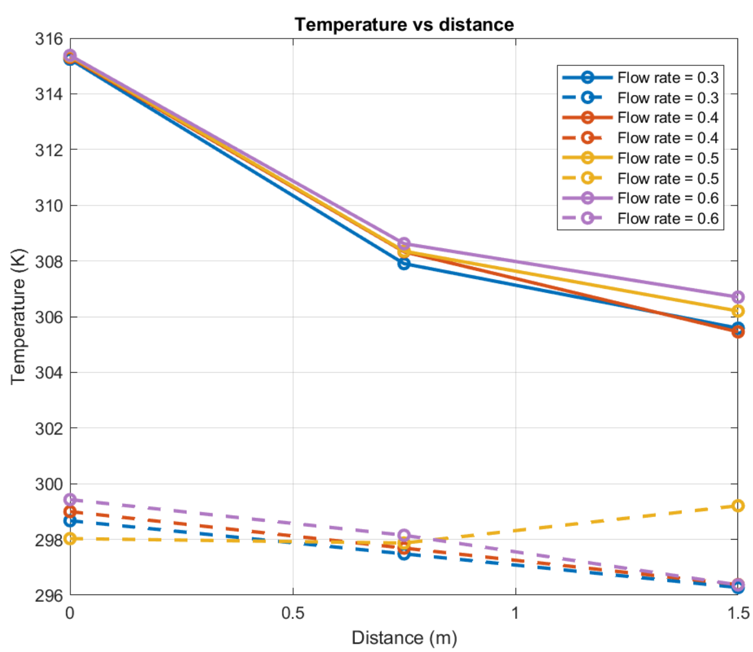


Figure 10: Counter Flow Temperatures with cold flow rate at 0.7 GPM at various hot flow rates

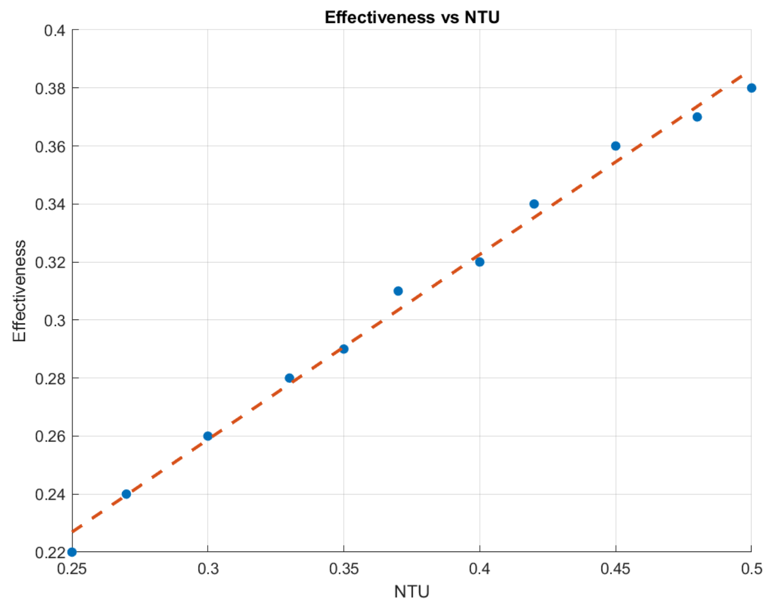


Figure 11: Relationship of effectiveness and NTU for parallel flow heat exchanger.

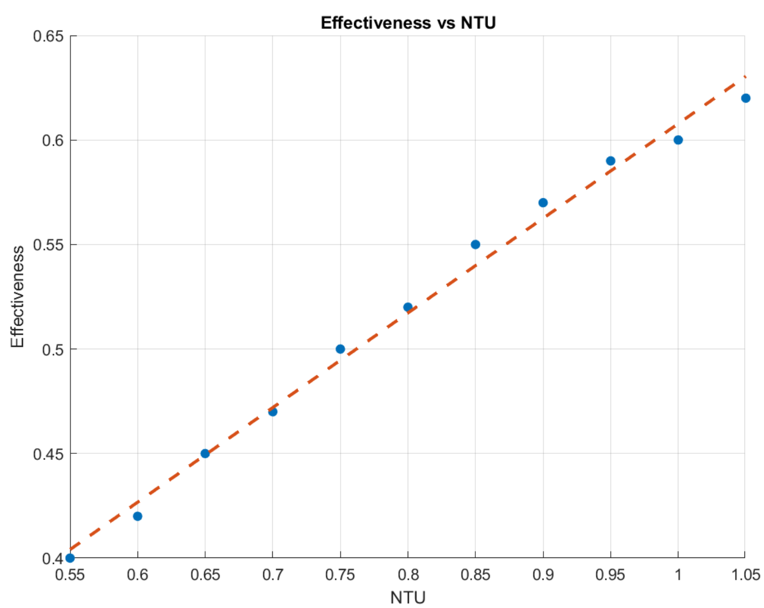


Figure 12: Relationship of effectiveness and NTU for counter flow heat exchanger.

While these results, there were several sources of uncertainty. Figure 10 shows an outlier measurement in cold water temperature in the counter-flow configuration that deviates greatly from the expected measurement. This highlights a source of uncertainty with the thermocouple measurements, indicating that they may not be entirely accurate. Another source of uncertainty is possible cavitation within the tubes of the heat exchanger. Since the tubes were covered in insulation, it was not possible to view any bubbles that may have been forming. Another important source of uncertainty in this experiment was the water heating device which supplied different temperature water throughout the various experiments. Despite these sources of uncertainty, the results of the experiment still show the expected relationship between variables.

Discussion

The results of this experiment highlight the differences between parallel flow and counter flow concentric heat exchangers. Given that the counter-flow configuration had a higher LMTD and rate of heat transfer, the hypothesis of the experiment is rendered valid. Additionally, the results imply that Ui is a function of Reynolds (Re) and Prandtl (Pr) numbers, and the effectiveness ratio (ε) was determined using the number of transfer units (NTU) method, providing a comprehensive evaluation of the heat exchanger's performance.

While this experiment provided profound insights and analysis into the working principle of a concentric heat exchanger, it is important to mention the sources of error, uncertainty and limitations of the experiment. The most significant source of error and uncertainty in this experiment is the unknown calibration status of the thermocouples. Due to the nature of the apparatus, the thermocouples are already fixed and inserted to probe different temperatures throughout the heat exchanger. For this reason, their calibration status is unknown, along with their precision and accuracy. Another source of uncertainty was the viscous and fluid losses experienced by the fluid as they flow through the heat exchanger. Since this apparatus involved vertical tubing as seen in Figure 4, it is possible that the viscous losses of the fluid are not negligible and also that cavitation could have been occurring in certain sections of the piping. Given these sources of error and uncertainty, the conclusions of this experiment may be somewhat limited to the theoretical insights implicated by the results.

Conclusion

The objective of this lab was to determine which flow arrangement in a water-water heat exchanger, parallel or counterflow, is most effective for achieving a specific heat-transfer objective. Based on the analytical and data analysis results, the counterflow configuration proved to be more effective than the parallel flow in a concentric heat exchanger. The overall heat transfer coefficients for the counterflow configuration were found to be 584.1 W/m²·K for the inner tube and 375.4 W/m²·K for the outer tube, while for the parallel flow, they were 1240.1 W/m²·K and 356.5 W/m²·K, respectively. In the counterflow arrangement, the hot fluid passed through the inner tube to minimize heat loss, while the outer tube was designated for the cold stream to reduce pressure drop. However, the experiment was subject to potential errors, including difficulty maintaining constant flow rates for both hot and cold water. Additionally, time constraints prevented temperatures from fully stabilizing, which could have led to less accurate readings. Cavitation bubbles within the apparatus also impacted the water flow rates, introducing another source of error.

Recommendations for improving the experiment include incorporating a visual representation of the cross-section of the concentric tube heat exchanger to enhance understanding of its structure and flow dynamics. Additionally, adjusting valves 5 and 6, either by closing or opening them, could help stabilize the pressure in valves A and B, resulting in more consistent flow rates and more accurate temperature readings.

References:

[1] Incropera, F.P., and DeWitt, D.P., Fundamentals of Heat and Mass Transfer, 5th edition, John Wiley & Sons, New York, 2000.

Statement of Contributions:

Elijah Perez: Experimental Design, Theory & Derivations, Data Analysis, Lab Report
Soham Saha: Experimental Design, Data collection, Lab Report Editing
Alex Pham:: Experimental Design, Data collection

Appendix:

Q = energy transferred by heat

$\dot{m}$ = mass flow rate

cp = fluid heat capacity

T = temperature

ρ = density

U = heat transfer coefficient

A = heat transmission area

ε = effectiveness ratio

Parallel Flow:

Cold Water Flow Rate 1: 0.3 GPM = X m^3/s

Hot Water Flow Rates	T1 Hot Intake	T2	T3	T4	T5	T6
.3	39.8901	33.475	31.8657	23.2134	25.5264	26.857
.4	41.5327	35.0594	33.3482	23.1665	26.1607	28.1588
.5	42.3678	35.8459	34.3581	23.2075	26.4916	28.8048
.6	42.4476	36.074	34.7449	23.3351	26.6859	29.1204

Table 1: Constant Cold Flow = 0.3 GPM with varying Hot water flow rates (Parallel Flow)

Cold Water Flow Rate 2: 0.5 GPM

Hot Water Flow Rates	T1	T2	T3	T4	T5	T6
.3	42.3479	34.8515	32.688	23.14	24.8742	25.9887
.4	42.2403	35.254	33.0822	23.0828	25.0933	26.569
.5	42.3247	35.5655	33.6056	23.0625	25.3201	27.0412
.6	42.4515	35.8425	34.1375	23.0625	25.5357	27.3882

Table 2: Constant Cold Flow = 0.5 GPM with varying Hot water flow rates (Parallel Flow)

Cold Water Flow Rate 3: 0.7 GPM

Hot Water Flow Rates	T1	T2	T3	T4	T5	T6
.3	42.3768	34.6184	31.9287	23.0919	24.4966	25.4968
.4	42.308	35.0815	32.7224	23.0542	24.6819	25.8511
.5	42.3507	35.3582	33.2007	23.0388	24.8726	26.2129
.6	42.383	35.6231	33.6588	23.0221	25.0206	26.5032

Table 3: Constant Cold Flow = 0.7 GPM with varying Hot water flow rates (Parallel Flow)

Counter Flow:

Cold Water Flow Rate 1: 0.3 GPM

Hot Water Flow Rates	T1	T2	T3	T4	T5	T6
.3	42.5036	35.3199	32.427	26.9962	25.056	23.6414
.4	42.2801	35.9459	33.5753	28.3459	25.8836	23.6768
.5	42.3662	36.2937	34.1145	28.875	26.2477	23.5863
.6	42.4174	36.5165	34.5649	29.3575	26.5571	23.6221

Table 4: Constant Cold Flow = 0.3 GPM with varying Hot water flow rates (Counter Flow)

Cold Water Flow Rate 2: 0.5 GPM

Hot Water Flow Rates	T1	T2	T3	T4	T5	T6
.3	42.3754	35.113	32.4417	25.9799	24.4912	23.2251
.4	42.2623	35.5424	32.842	26.5661	24.928	23.3734
.5	42.3025	35.814	33.4432	26.991	25.1816	23.2144
.6	42.3665	36.0426	33.8658	27.3452	25.3981	23.267

Table 4: Constant Cold Flow = 0.5 GPM with varying Hot water flow rates (Counter Flow)

Cold Water Flow Rate 3: 0.7 GPM

Hot Water Flow Rates	T1	T2	T3	T4	T5	T6
.3	42.2421	34.9019	32.5844	25.6723	24.4821	23.2735
.4	42.3029	35.326	32.4519	25.9992	24.6881	23.377
.5	42.3353	35.6853	33.1512	26.4295	24.9984	23.3648
.6	42.3458	35.8786	33.4767	26.6988	25.1494	23.3663

Table 4: Constant Cold Flow = 0.7 GPM with varying Hot water flow rates (Counter Flow)