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CHE 3214L
Chemical Engineering Laboratory Investigations 1

HEAT TRANSER
Heat Transfer in a Double-pipe Heat Exchanger

A laboratory report submitted to

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by

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1. Introduction

Transport processes deals with momentum and heat exchanges in different systems. Many operations such as equipment design can have a more concise basis through theories. The principle of heat transfer, which can be investigated using a double-pipe heat exchanger, is one of the most important fields of this study.

A double-pipe heat exchanger is the simplest type of heat exchanger. It consists of two tubes, one placed concentrically inside of the other. The annular space through the inner tube are the regions where two fluids will flow. The main driving force of heat transfer is the temperature gradient or the differences in the temperature of the two fluid fluids. The amount of heat transferred between the two fluid media can be calculated using the equation below.

$$\emptyset H = UA\Delta T_{lm} \quad [1]$$

where $\emptyset H$ is the total heat transferred per unit time, U is the overall heat transfer coefficient, A is the overall heat transfer area, and ΔT_{lm} is the logarithmic mean temperature difference.

Transfer of heat happens by three principle means: radiation, convection, and conduction. Radiation takes place in heat exchanger. However, unlike conduction and convection, radiation only becomes significant at relatively high temperature settings. The biggest contribution to heat transfer in a heat exchanger is made through convection where heat is transferred from the hot bulk fluid to the walls of the pipe. On the other hand, conduction occurs as travels through the inner tube wall that separates the two media.

There are two configurations by which a double-pipe heat exchanger can be operated. These are the countercurrent where the two liquids travel in opposite directions and co-current operations where the two liquids travel in the same direction.

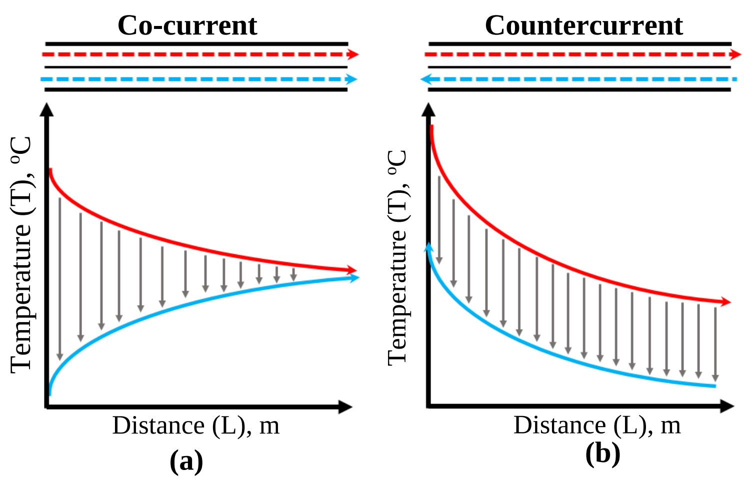

Figure 1. Double-pipe Heat Exchanger in (a) Co-current flow and (b) Countercurrent flow

In co-current flow, both fluids flow in the same direction from one end of the heat exchanger to the other. The driving force in this operation is maximum at the entrance resulting in a large heat transfer rate and gradually decreases along the length of the heat exchanger as shown in Figure 1a. In a countercurrent flow, the two fluids are oriented to leave the tube at opposite ends. As a result, it is possible that the cold fluid will leave at a higher temperature than the hot fluid. This is one of the advantages in operating this type of flow since it signifies a higher amount of heat absorbed from the hot liquid. Contrary to the former configuration, the driving force of a countercurrent operation is relatively constant as shown in Figure 1b.

In the design of heat exchanging equipment, there are several approaches that can be employed. These methods are the log-mean temperature difference (LMTD) method and the number of transfer units (NTU) method. The LMTD method relates the total heat transfer rate to the inlet and outlet fluid temperatures, the overall heat transfer coefficient, and the overall heat transfer area.

$$\Delta T_{lm} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln \frac{\Delta T_{max}}{\Delta T_{min}}} \qquad [2]$$

where ΔT_{max} is the maximum temperature difference, and ΔT_{min} is the minimum temperature difference in unit temperature.

The NTU (Number of Transfer Units) method gives the heat transfer capacity of the heat exchanger. This concept is introduced because the magnitude is less dependent on the flow rate of the fluids than the heat transfer coefficient. This can be defined as a measure of the amount of heat that can be transferred by the heat exchanger.

$$NTU = \left(\frac{UA}{C_{min}}\right) \qquad [3]$$

where NTU is the number of transfer units and C_{min} is the lower thermal capacity value in J/K-s.

In this method, the heat transfer effectiveness is expressed in the form of dimensionless parameter. It is the ratio of the actual heat transfer rate in a heat exchanger to the maximum possible heat transfer rate as if an infinite heat transfer surface area were available. The relation for the effectiveness in terms of heat capacities is dependent on the flow arrangement. The effectiveness for countercurrent and co-current are shown in equations 4 and 5, respectively.

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 - \frac{C_{min}}{C_{max}}\right)\right]}{1 - \left(\frac{C_{min}}{C_{max}}\right)\exp\left[-NTU\left(1 - \frac{C_{min}}{C_{max}}\right)\right]} \qquad [4]$$

$$\varepsilon = \frac{1 - \exp\left[-NTU\left(1 + \frac{C_{min}}{C_{max}}\right)\right]}{1 + \left(\frac{C_{min}}{C_{max}}\right)} \qquad [5]$$

where ε is the effectiveness, and C_{max} is the higher thermal capacity value in J/K-s.

2. Objectives of the Experiment

a) Investigate the effects of varying the inlet temperature, the flow rates, and the flow arrangement (countercurrent or co-current parallel flow) on the temperature profiles in a double-pipe heat exchanger.

b) Determine the overall heat transfer coefficients and the effectiveness of the heat exchange unit at the required operating conditions and compare them with theoretically calculated values.

3. Methodology

3.1 Methodological Framework

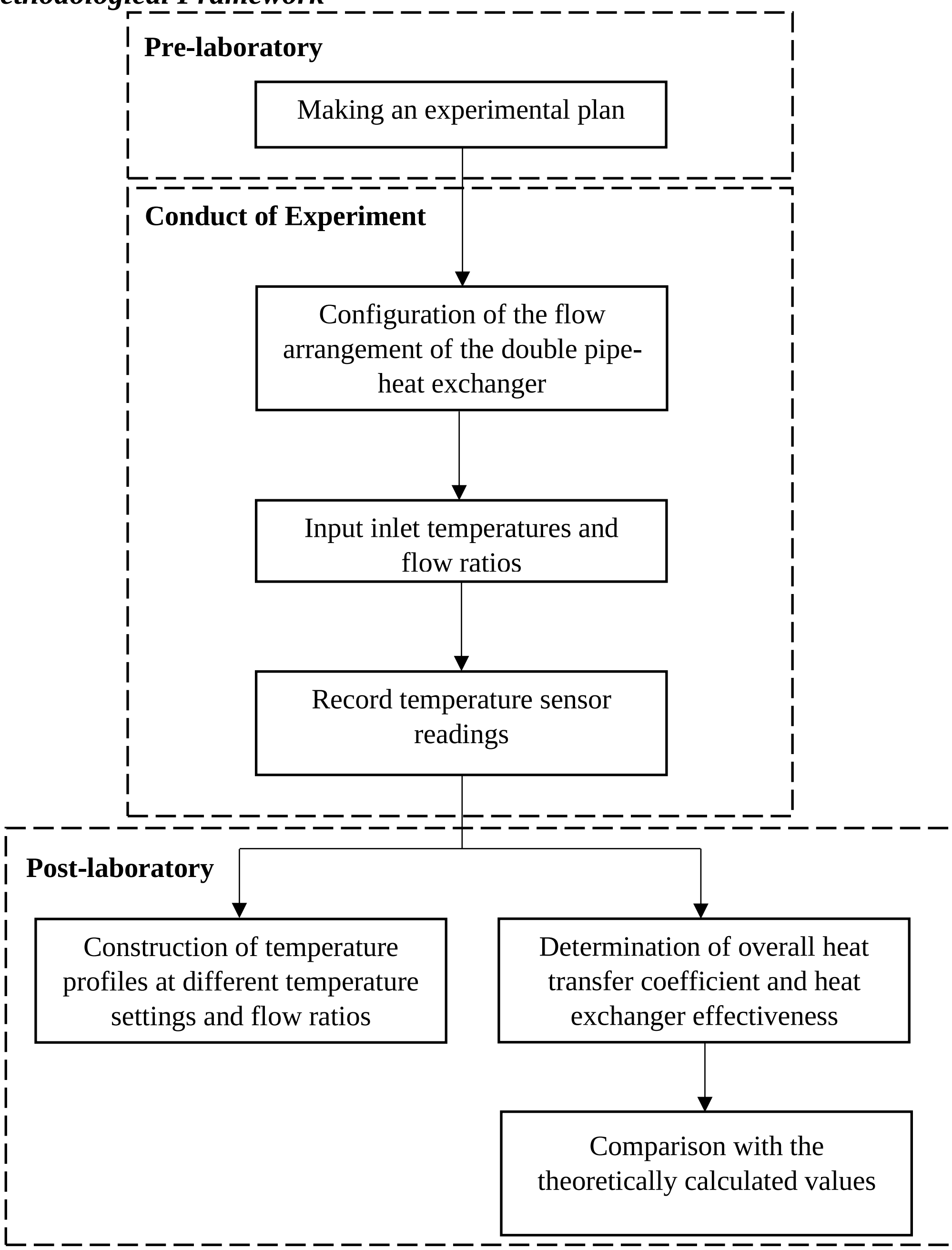

Figure 2. Methodological Framework in the Conduct of Heat Transfer Experiment

An experimental plan was prepared beforehand to create a guide of the flow of the actual conduct of experiment and data processing. The independent and dependent variables as well as the constants of the experimental set up were identified as shown in the tables below.

Table 1. Parameters considered in performing objective 1

Dependent Variables	Independent Variables	Constants
Temperature Profile	Flow Rates, $F_{HOT}:F_{COLD}$	Flow Arrangement Inlet Temperatures, $T_{IN,HOT}$; $T_{IN,COLD}$
	Inlet Temperatures of Hot Water, $T_{IN,HOT}$	Flow Arrangement Flow Rates, $F_{HOT}:F_{COLD}$ Inlet Temperature of Cold Water, $T_{IN,COLD}$
	Flow Arrangement	Flow Rates, $F_{HOT}:F_{COLD}$ Inlet Temperature, $T_{IN,HOT}$; $T_{IN,COLD}$

Table 2. Parameters considered in performing objective 2

Parameter	Experimental Values	Literature Values
overall heat transfer coefficient, U effectiveness of heat exchanger unit, ε	Inlet/ outlet temperatures, T Volumetric flow rates, Inner/ outer diameters, D_i/D_o Pipe length, L Bulk temperature, T_b Wall temperature, T_w	Density, ρ Viscosity, μ Thermal conductivity, k Specific heat capacity, C_p

The temperature sensor readings were recorded and temperature profiles at varying parameters that were considered in Table 1 were constructed. These temperature readings were further processed to determine the overall heat transfer coefficient and heat exchanger effectiveness through equations 1 to 5. These calculated values were quantitatively compared to the theoretical values.

3.2 Materials

Distilled water was used as the fluid medium for the hot water stream while tap water was used for the cold water stream in the double-pipe heat exchanger.

3.3 Equipment

The main equipment that was used to attain the objectives of this experiment is the model Armfield HT30XC double-pipe heat exchanger and a service unit as shown in Figure 3. It is controlled by a computer that serves as the operator interface.

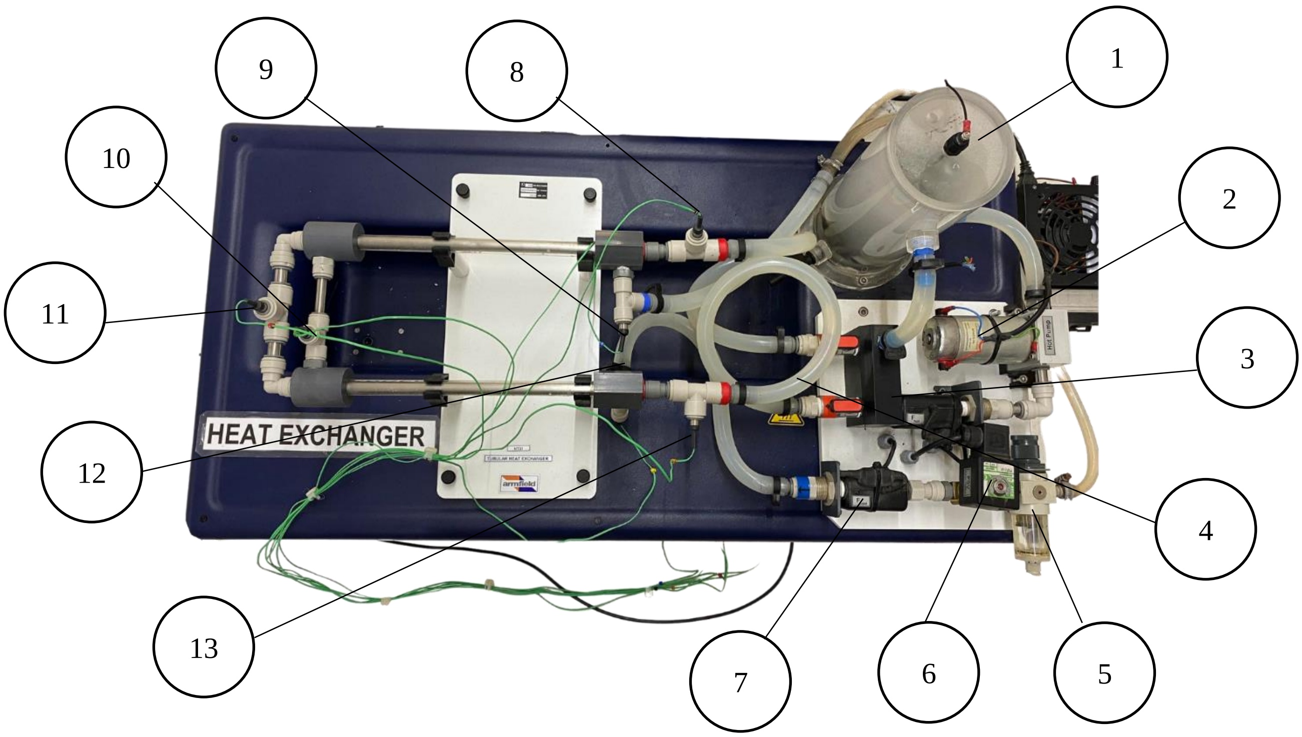

Figure 3. Armfield HT30XC Heat Exchanger Service Unit

The service unit for controlling comprises a (1) hot water vessel which is positioned on the top of the service unit and serves as the reservoir for hot water; (2) a hot water circulation pump which is used to circulate the water through the heat exchanger and back to the hot water vessel; (3) an in-line flow meter which is used to measure the hot water flow rate; (4) flexible tubes which connects the hot water circulator to the unit; (5) a pressure regulator complete with integral filter; (6) a proportioning solenoid valve which is used to control the flow rate of the cold steam; and (7) a second in-line flow meter.

The double pipe heat exchanger as shown in the right side of the figure has two concentric tubes that were made of stainless steel for inner tube which has 9.5 mm outer diameter and 0.6 mm wall thickness, and clear acrylic for outer tube which has 12 mm inner diameter and 3.0 mm wall thickness. Each heat transfer section is 330 mm long.

Six thermocouples or temperature sensors were used to determine the temperature readings at different locations along the tubes. In the countercurrent operation, the thermocouples were installed at the following locations: (8) hot water inlet next to sensor T1 is connected to the red connector at the rear, (11) hot water mid-position, (13) hot water outlet next to sensor T3 is connected to the red connector at the front, (12) cold water inlet next to sensor T4 is connected to the blue connector at the rear, (10) cold water mid-position, and (8) cold water outlet next to sensor T6 is connected to a drain.

A panel located on the front of the service unit contains the MAINS switch (9) for the apparatus to enable or standby; an emergency stop switch (10) which is used to stop the operation for emergency situations; connector (11) on the front panel is to connect the service unit to the computer with a USB port; and (12) input connectors labeled T1-T10.

Figure 4. Front Panel of the HT30XC Heat Exchanger Service Unit

The back panel of the service unit has an input socket (13) where the electrical supply is to be connected. A combined residual current device (RCD) (14) protects the unit from stray currents and (15) output sockets which provides a 23Vdc power supply and 12Vdc auxiliary supply.

Figure 5. Panel located at the rear of the plinth of the HT30XC Heat Exchanger Service Unit

3.4 Procedures

3.4.1 Preliminary Steps

Prior to conducting the experiment, the heat exchanger and its corresponding parts were checked whether they were configured in countercurrent operation where the hot fluid flowing in the inner tube flows in the opposite direction to the flow of cold fluid in the annulus. Set of conditions as shown in the table below were assigned by the instructor. In this case, the conditions assigned were 2C.

Table 3. Temperature and Flow rate Settings

Temperature, °C		Flow rate ratio ($F_{HOT}:F_{COLD}$)			
1	**2**	**A**	**B**	**C**	**D**
70	65	3.0:1.0	2.5:1.0	2.5:1.5	3.0:1.5
60	55	1.0:1.0	1.0:1.0	1.5:1.5	1.5:1.5
50	45	1.0:3.0	1.0:2.5	1.5:2.5	1.5:3.0

However, since the equipment was unable to vary hot water flow; a new set of flow rate ratios were introduced. These were 0.2:3.0, 0.2:2, and 0.2:1.0.

The mains electrical lead was then connected to the mains inlet socket and connected to the power supply. The computer was turned on. Using the USB cable, the USB port on the heat transfer service unit was connected to the USB port on the computer. This was followed by an illumination of the red 'USB POWER' and green 'USB ACTIVE' LEDs. After ensuring that the unit was connected to the computer, the HT30XC software on the computer was launched.

The pressure regulator was set to minimum. This was done by pulling out its grey knob and turning it fully counterclockwise. It was ensured that that the mains switch was off, and the STOP button was pushed in. The RCDs were also switched on by lifting their levers up. The STOP button was then pulled out and the mains switch was turned on. A lighting up of the ENABLED LED light was observed as an indication that the main switch was turned on.

The HT30XC software was allowed to run with the countercurrent operation chosen. The 'Start COM session button' was clicked to enable USB virtual COM port. To display the diagram screen, the 'view diagram button' was clicked. The 'Power On' button was then clicked to begin the operation. The green 'RUN' LED light on the front panel of the service unit was light up.

Using the software, the cold water flow rate was set. First, on the software, the flow control was adjusted was 0% to 100% stepwise. Then, the cold water supply was turned on. Using the grey knob on the pressure regulator, it was slowly turned clockwise until the flow rate on the computer screen read 0.5 L/min. After obtaining this, the knob was locked by pushing. This was followed by setting the flow control of the cold water back to 0%, stepwise.

The hot water vessel was filled with water until the level is approximately 20 mm from the top or until the conductivity level sensor was submerged in the water. The pump speed was set by choosing the Automatic Mode of Operation while making sure that the low-level indication in the software was not activated. The Flow PID controller window was selected, and the hot water flow rate was set to 0%.

3.4.2 Countercurrent operation: Effect of Flow Rate

In the software heater button, automatic mode of operation was selected with temperature set to 45°C. The cold water flow valve was adjusted to give a cold water flow rate of 3 L/min. In the software, the flow button was clicked to access the hot water flow control mode. The hot water flow valve to give the hot water flowrate value 0.2 L/min. The temperature profile was allowed to stabilize for 20 minutes. When temperatures were stable, six temperatures which were labeled T_1, T_2, T_3, T_4, T_5, and T_6 and the flow rates for both hot and cold water were manually recorded three times. The set of procedures were repeated using the other flow rate ratios

3.4.3 Countercurrent operation: Effect of Driving force

The flow rate ratio between the hot and cold water were set to 0.2:1, and the temperature controller was set to 55°C. The temperatures were allowed to stabilize 20 minutes and were manually recorded thrice. The experiment was repeated using temperature settings of 65^{o}C.

3.4.4 Changing from countercurrent to Co-current operation

The flow rates of both hot and cold water were reduced to change the configuration from countercurrent to co-current operation. From the software, the co-current option was selected. The same steps from 3.4.2 were done.

3.4.5 Shutdown operation

The service unit was shut down by setting the temperature of the hot water to 28^{o}C. The flow rates of the hot and cold water were adjusted to 3 L/min and 2L/min, respectively. The unit was run until T1 has reached 40^{o}C. Both hot and cold water flow rates were adjusted to 0% stepwise. On the software, the heater was turned off, followed by the main switch. The emergency STOP button was pushed in and the RCDs were turned off. The cold water supply was turned off, the software was closed and the computer was shut down.

3.4.6 Data Processing

The data processing needed for the first objective is straightforward. For data recorded at all varying operating conditions, averages were computed for the three different readings. The average values were used for all calculations. Temperature profiles were generated by plotting the temperature readings versus the distance along the length of the heat exchanger.

The experimental overall heat transfer coefficient was calculated using equation 1, with the logarithmic mean temperature difference calculated using equation 2. The mass flow rate was obtained by taking the product of the fluid volumetric flow rate and its density which is function of temperature. Theoretical overall heat transfer coefficient was computed by taking the sum of the resistances due to the films of the fluid on the either side of the wall, the resistances of the tube wall itself, and the resistances due to the deposited scales on the tube walls. The conduction coefficient used was that of stainless steel, based on the average temperature of the bulk temperatures of the hot and cold fluids.

For overall effectiveness computations, the ratio between the actual and the maximum heat transfer rate possible was used to obtain the experimental overall effectiveness and equations 3, 4, and 5 were used to obtain the theoretical overall effectiveness. Finally, percentage of the errors between theoretical and experimental values were computed for both the overall heat transfer coefficient and the overall effectiveness. The same data processing was used for the co-current operation.

4. Results and Discussions

The parameters used during the actual conduct are shown in the table below. As shown in the table, the dimensions of inner and outer tubes are considered such as the different values for the outer diameter for the inner tube, the inner diameter of the outer tube, as well as the cross-sectional area for the inner and outer tube

Table 4. Equipment specifications

Heat Exchanger		
Inner Tube	**Diameter (D_1), *m***	0.0095
	Wall Thickness (W_1), *m*	0.0006
	Hydraulic Diameter (D_i), *m*	0.0083
	Cross-sectional Area (A_1), *m^2*	5.41×10^{-5}
	Heat Transfer Area (A_i), m^2	0.0172
Outer Tube	**Diameter (D_2), *m***	0.012
	Wall Thickness (W_2), *m*	0.003
	Hydraulic Diameter (D_o), *m*	0.0025
	Cross-sectional Area (A_2), *m^2*	4.22×10^{-5}
	Heat Transfer Area (A_o), m^2	0.0197
Steel Thermal Conductivity (k), *W/m·K*		16
Total Length of Heat Transfer Section (L), *m*		0.66
Log Mean Heat Transfer Area (A_{lm}), m^2		0.0184

In this part of the experiment, effects of changing the inlet temperature, the volumetric flow rate, and the flow arrangement on the temperature profile for both countercurrent and concurrent flows are studied.

4.1 Effect of Inlet Temperature on the Temperature Profile in a Double-pipe Heat Exchanger

Effect of varying the inlet temperature on the temperature is studied by having three different hot stream inlet temperatures of 45°C, 55°C, and 65°C at a fixed flow rate ratio of (F_{HOT}:F_{COLD}) 0.2:1 for both flow arrangements. Shown in Figure 6 is the temperature profile of the heat exchanger for a countercurrent operation and shown in Figure 7 is the temperature profile of the heat exchanger for a co-current operation.

4.1.1 Countercurrent operation

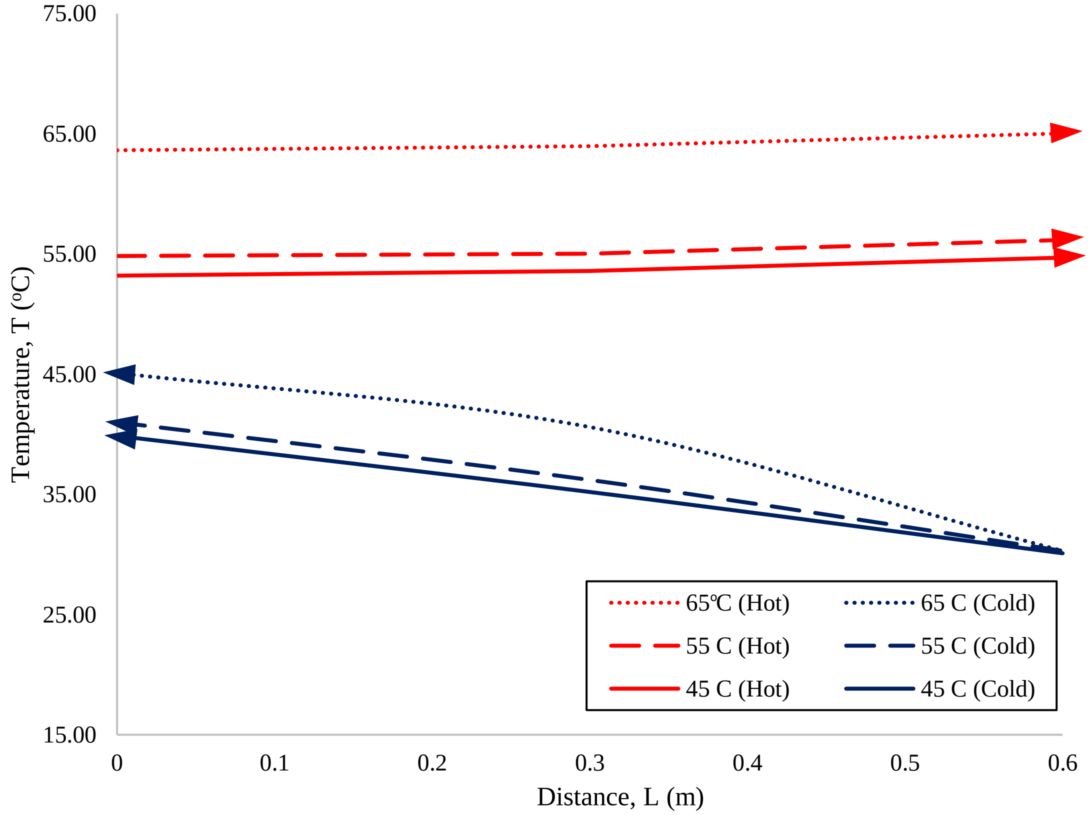

Figure 6. Temperature Profiles of Hot and Cold Fluids at different Inlet Temperatures for a Double-pipe Heat Exchanger operating in a Countercurrent flow at a constant flow ratio of 0.2:1.0

As shown in the graph, as the inlet temperature is increased, there is also an increase in the outlet temperature of the cold stream. It can also be observed from the graph that the temperature difference at each temperature setting is approximately constant throughout the exchanger length. Less variation in the temperature difference throughout the heat exchanger is expected for fluid flowing opposite to each other (Coulson & Richardson, 1999). At 65°C, there is a larger temperature gradient compared to the 45°C and 55°C inlet temperatures. The ΔT_{max} for 65°C is 2.3 and 1.4 times larger relative to 45°C and 55°C, respectively.

As the inlet temperatures of the entering hot fluid is increased, though the difference of the temperatures across the exchanger is approximately constant, the difference becomes larger. This is signified by the increasing distance between the two curves for a pair of flows at each temperature. The inlet temperature of the cold fluid entering is essentially constant because the tap water temperature is not altered. The only factor affecting the temperature difference therefore is the inlet temperature of the entering hot fluid. At higher inlet temperatures of the hot fluid, there is a greater driving force for heat transfer to occur.

Heat transfer in a heat exchanger is governed by Fourier's Law on Heat Conduction and Newton's Law on Cooling. Both principles show that heat transfer is proportional to the temperature difference present. The temperature difference serves as the driving force that affects the transfer of heat. By increasing the inlet temperature of the inner tube, the temperature difference is increased thereby affecting more heat transfer to occur. However, in the counter-current configuration, the cold fluid inlet temperature is kept constant hence its temperature profile would pivot from this point.

4.1.2 Co-current operation

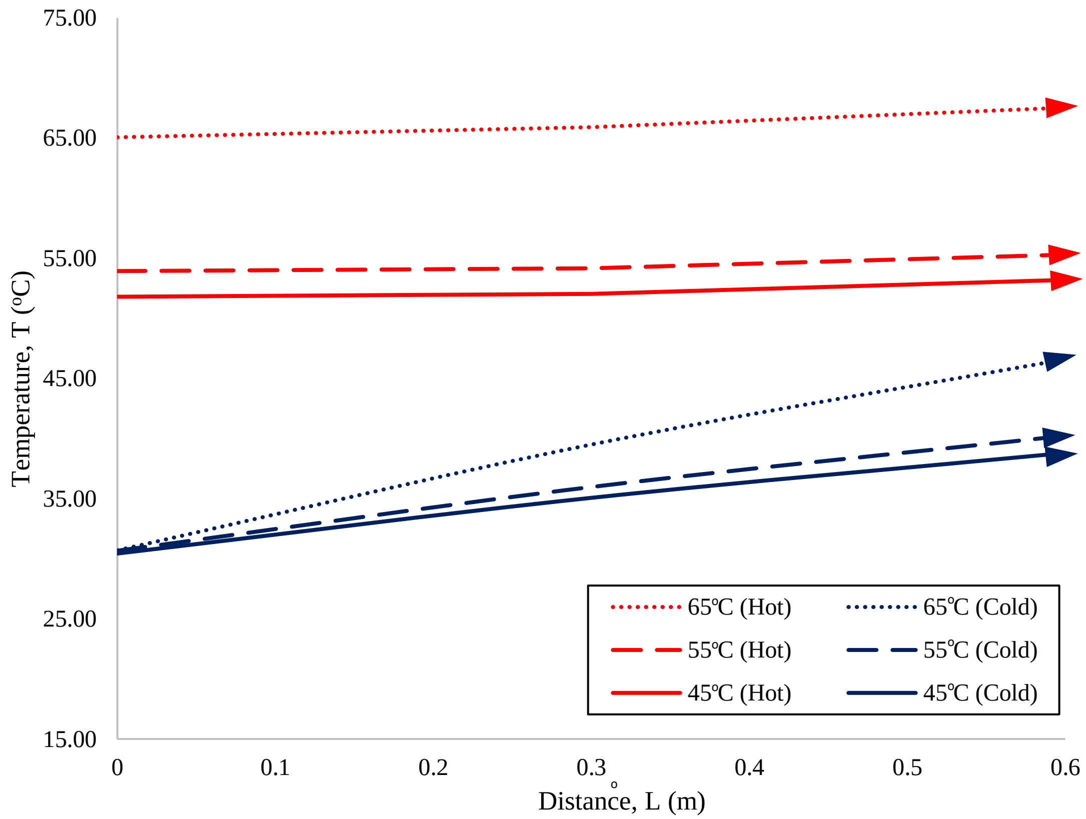

Figure 7. Temperature Profiles of Hot and Cold Fluids at different Inlet Temperatures for a Double-pipe Heat Exchanger operating in a Co-current flow at a constant flow ratio of 0.2:1.0

In a co-current flow arrangement with all conditions kept constant except its inner tube inlet temperature, the resulting temperature profile is shown in Figure 7. Similar to that of the countercurrent flow arrangement, the temperature difference increases with increasing hot fluid inlet temperature. The increase of temperature results to the increase of ΔT_{max} and ΔT_{min} by an average of 55% and 39% respectively. The cold and hot temperature lines approach each other along the pipe length in the co-current regime. The temperature of the streams flowing in the same direction gradually approaches the same value (Coulson & Richardson, 1999). Moreover, at 65°C, there is a larger temperature gradient compared to the 45°C and 55°C inlet temperatures which is also observed in the countercurrent flow arrangement.

It can also be observed from the graph that the change in the temperature is observed at the inlet portion of both fluids because there is a greater temperature difference at this portion while as both fluids flow to the other part of the pipe, there is a minimal change in the temperature of both fluids. This makes the co-current flow suitable for abrupt temperature decreases. As can be noted from the figure that the temperature for both fluids tend to achieve uniformity, and this is true for co-current flows at infinite areas of heat transfer.

Table 5. Temperature differences of the double-pipe heat exchanger at different inner tube temperatures with a 0.2:1 flow ratio.

Inlet temperature ($T_{IN,HOT}$), °C	**Countercurrent flow**		**Co-current flow**	
	Maximum mean temperature difference (ΔT_{max}), °C	**Minimum mean temperature difference (ΔT_{min}), °C**	**Maximum mean temperature difference (ΔT_{max}), °C**	**Minimum mean temperature difference (ΔT_{min}), °C**
45°C	23.10	14.83	22.77	13.00
55°C	24.53	15.17	24.63	13.37
65°C	33.33	19.93	36.83	18.47

As presented in Table 5, an increase in the temperature gradient of the hot and cold streams causes an increase in the ΔT_{max} and ΔT_{min} in the profile. The maximum temperature difference (ΔT_{max}) of the hot and cold water is defined as the difference of the outlet temperature of the hot water and the inlet temperature of the cold water for a countercurrent flow. Heat is being transferred from the hot water to the cold water. The cold water absorbs heat from the hot water which then increases its temperature proportional to the amount of heat that was transferred. The amount of heat transferred is directly proportional to the temperature difference of the two fluids. The greater the temperature difference, the greater is the heat transfer rate. The rate of heat transfer, however, decreases with respect to the length of the heat exchanger.

It is also evident that the minimum mean temperature difference (ΔT_{min}) of the countercurrent flow arrangement is relatively larger than that of the concurrent flow arrangement which suggests that the heat transfer is more effective with a countercurrent arrangement since it offers larger temperature driving forces hence, usually preferred over the co-current flow for this reason (Geankoplis, 2003).

4.2 Effect of Flow Rates on the Temperature Profile in a Double-pipe Heat Exchanger

Aside from the inlet temperatures, the profiles also vary considerably depending on the flow rate. Shown in Figures 8 and 9 are the temperature profiles of the double-pipe heat exchanger in both countercurrent and co-current flows, respectively. For both arrangements, the cold inlet temperatures were kept constant at approximately 30°C and the hot inlet temperature was kept constant at 45°C. However, due to some equipment defects, the hot inlet temperature slightly varies from 46-54°C. To analyze the effect of volumetric flow rate on the temperature profile, three flow rate ratios (F_{HOT}:F_{COLD}) were varied accordingly with a decreasing outer tube flow rate (F_{COLD}); the flow ratio parameters are 0.2: 3, 0.2:2, 0.2:1 L/min.

4.2.1 Countercurrent operation

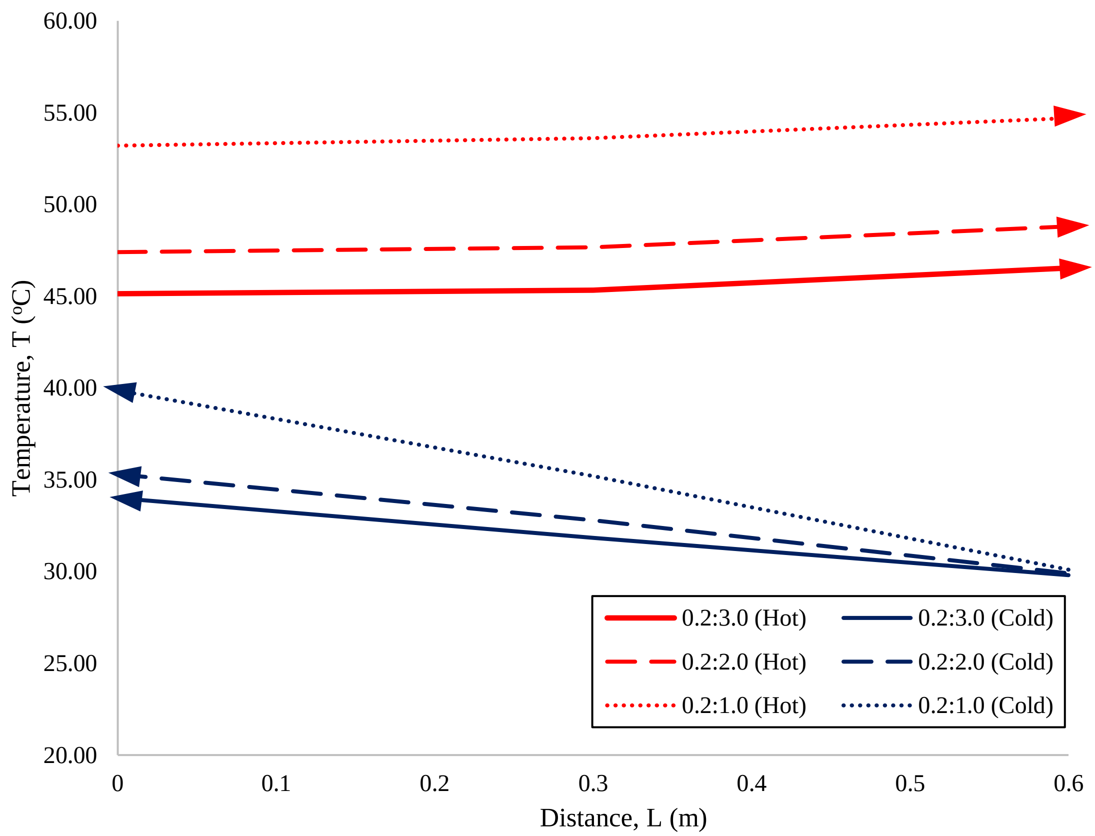

Figure 8. Temperature Profiles of Hot and Cold Fluids at different Flow Ratios for a Double-pipe Heat Exchanger operating in a Countercurrent flow at a constant temperature setting of 45°C.

In a countercurrent operation with all conditions kept constant except its outer tube flow rate (F_{COLD}), a higher flow ratio value would result from the decrease in the outer tube flow rate shown in Table 8; or this can be mathematically expressed as 0.2:3 < 0.2:2 < 0.2:1.

Table 6. Flow rate ratios and their equivalent values.

Flow Rate Ratio (F_{HOT}:F_{COLD}), L/min	Ratio value, F_{HOT} / F_{COLD}
0.2:3	0.067
0.2:2	0.1
0.2:1	0.2

These variations will have an effect on the outlet temperatures of the cold fluid. This can be observed in Figure 8 wherein increasing the cold flow rate or the outer tube flow rate will result in a less steeper temperature profile of cold fluid. This implies that decreasing the flow ratio will yield a lower outlet temperature of the cooler fluid. The implication also holds true when there is a faster rate of hot fluid compared to the cold fluid. In this case, there would be a higher outlet temperature of cold fluid since this results in a greater flow ratio thereby generating a steeper profile. The opposite is also true—the slower the hot fluid compared to the cold fluid, the lower the outlet temperature of the cold fluid.

In general, varying the flow rate causes the faster flowing fluid to have a smaller temperature change over the heat transfer length than the other fluid. This makes physical sense since a faster flow rate allows a lesser contact time for a liquid; hence, a smaller change in temperature will happen from the entry point to its exit point. If the processed fluid is much faster, its exit temperature will only be minimal while the other fluid will be greater. On the other hand, if the processed fluid is slower, the effect would be the contrary.

The governing principle of these lies in the convective heat transfer. Convection is the transfer of heat by a bulk movement of liquid. This is a process that is essentially the exchange of energy via mass transfer. Changing the process of fluid flow rate, such as increasing or decreasing the volumetric flow rate of either hot or cold fluids, will change the exit temperatures. These changes in the exit temperature happens because the mean temperature difference will adjust to match the same heat transferred (Geankoplis et al., 2018). This could also be further explained by the equation of heat transfer rate which is

$$\emptyset_H = \left(mC_p\right)_{HOT}(T_{OUT,HOT} - T_{IN,HOT}) = \left(mC_p\right)_{COLD}(T_{OUT,COLD} - T_{IN,COLD}) \qquad [6]$$

where $\emptyset_H$ is the heat transfer rate in kJ/s, m is the mass flow rate of liquid in kg/min, $C_p$is the heat capacity of liquid in kJ·kg^{-1}·^{o}C^{-1}, T_{IN} is the inlet temperature and T_{OUT} is the outlet temperature in ^{o}C.

As established in equation 6, assuming that heat transfer rate does not change, flow rate is inversely proportional to the temperature change. When the flow rate of cold water is increased, there will be a decrease in temperature difference of the cold fluid. The same goes for the hot fluid. When the hot fluid flow rate is increased, there will only be a minimal hot fluid temperature difference.

4.2.2 Co-current operation

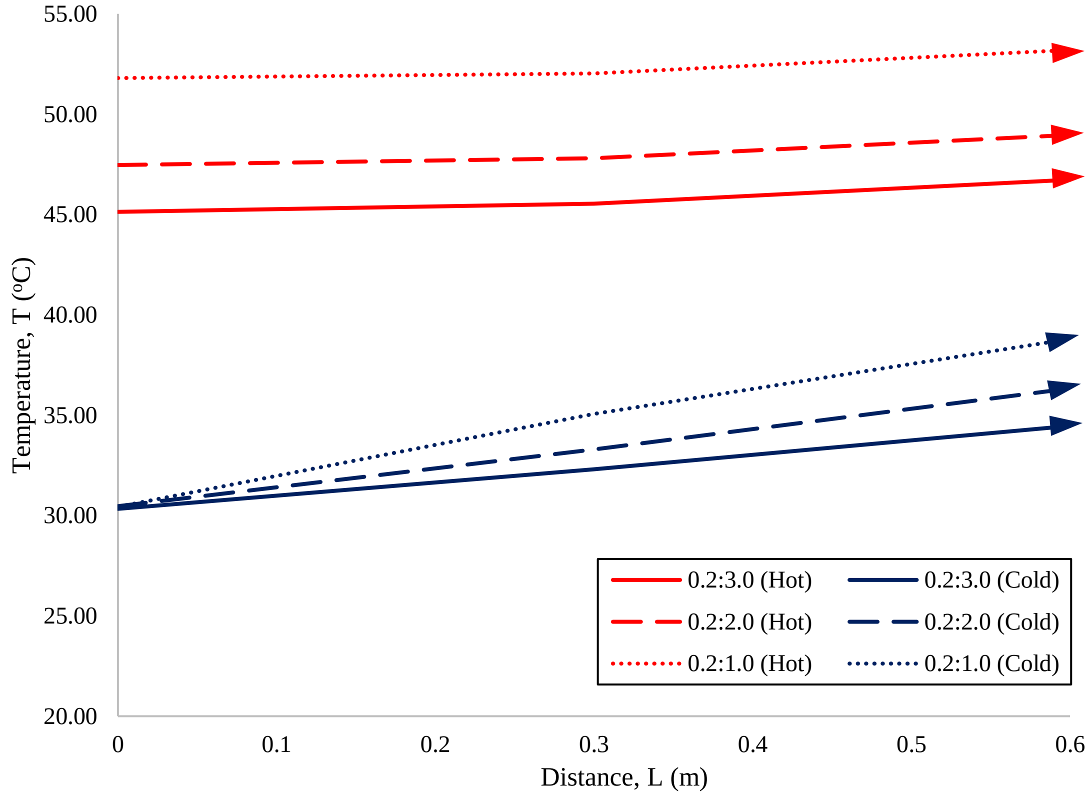

Figure 9. Temperature Profiles of Hot and Cold Fluids at different Flow Ratios for a Double-pipe Heat Exchanger operating in a Co-current flow at a constant temperature setting of 45°C.

In a double-pipe heat exchanger with a co-current configuration with the same set of conditions, an analogous explanation similar to that of the countercurrent can be used to explain the trends. As shown in Figure 9, a lower cold flow rate as denoted by the dotted lines will have a steeper trend as compared to the dashed and bold lines which indicate a higher cold flow rate. A relationship can be established such that the outer tube flow rate is indirectly proportional to the steepness of the line. Given that a steep trend indicates a higher change in temperature, it can be deduced that outer tube flow rate is also indirectly proportional to the change in temperature of cold fluid.

Table 7. Temperature differences of the double-pipe heat exchanger with varying cold flow rates at a temperature of 45°C.

Flow Rate Ratio (F_{HOT}:F_{COLD}), L/min	Countercurrent flow		Co-current flow	
	Maximum mean temperature difference (ΔT_{max}), °C	Minimum mean temperature difference (ΔT_{min}), °C	Maximum mean temperature difference (ΔT_{max}), °C	Minimum mean temperature difference (ΔT_{min}), °C
0.2:3	15.33	12.53	16.40	10.67
0.2:2	17.50	13.50	18.50	11.13
0.2:1	23.10	14.83	22.77	13.00

Shown in Table 7 are the minimum and maximum temperature differences of the double-pipe heat exchanger in co- and countercurrent flows. It can be seen that the maximum and minimum temperatures in both configurations increases as the cold flow rate decreases. An increase in magnitude of the ΔT_{max} and ΔT_{min} from 15.33°C to 23.10°C and 12.53°C to 14.83°C, respectively, can be observed in the countercurrent flow as the cold flow rate decreases from 3 L/min to 1 L/min. This can also be observed in co-current flow wherein the maximum temperature difference rises from 16.40°C to 22.77°C and the minimum temperature difference increases from 10.67°C to 13.00°C. These temperature differences are the driving forces. The greater the temperature difference, the greater is the rate of heat transfer; a greater heat transfer then would occur at low velocities of cold fluid. It is also important to note that there is a large temperature difference that prevails in co-current mode which results in a relatively large heat transfer rate. However, the driving force decreases along the path length of the heat exchanger (Foust, 1980). This can be observed from the drastic change in temperature from ΔT_{max} to ΔT_{min}. On the other hand, in countercurrent mode, the temperature difference along the length of the heat exchanger is relatively constant. This can be explained by the fact that there is a lesser change in temperature from ΔT_{max} to ΔT_{min}. Some errors are attributed to defective temperature sensors causing inconsistencies in the temperature readings and uncontrollable fluctuations in hot water flow rate.

4.3 Effect of Flow Arrangement on the Temperature Profile in a Double-pipe Heat Exchanger

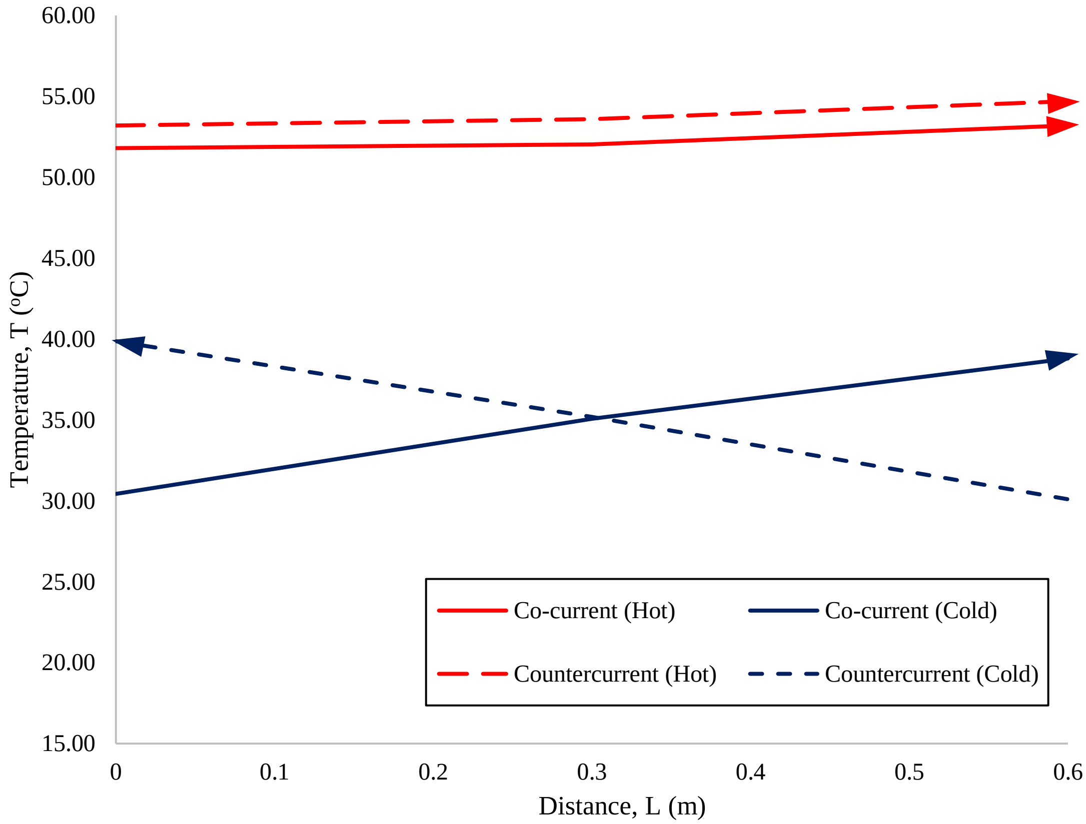

Figure 10. Temperature Profiles of Hot and Cold Fluids for a Double-pipe Heat Exchanger operating in different flow arrangements at a constant temperature setting of 45ºC and flow ratio of 0.2:1.0

To investigate the effect of the flow arrangement to the temperature profile in the heat exchanger, countercurrent operations are compared with the co-current operations as shown in Figure 10. The temperature profiles in the countercurrent operation have a relatively constant temperature gradient along the length of the heat exchanger. Whereas the temperature profiles in the co-current operation experience a decrease in temperature gradient along the length of the heat exchanger. Therefore, heat transfer is constant in countercurrent operation while it varies for co-current with respect to the length of the heat exchanger.

In the countercurrent operation, the hot and cold fluids flow in opposite directions and the temperature difference along the length of the heat exchanger is relatively constant. In the co-current operation, both the hot and the cold fluids enter the exchanger at the same end and flow in the same direction. At the entrance of the exchanger, a large temperature difference prevails, resulting in a relatively large heat-transfer rate. This driving force decreases along the length of the heat exchanger. As a consequence, the heat exchange rate drops asymptotically as the streams approach the end of the exchanger.

The difference between the mean temperature difference with respect to the countercurrent and co-current flow affects the temperature driving force for heat transfer. Because of its larger mean temperature difference, countercurrent flow offers larger temperature driving forces and is usually preferred over the co-current flow for this reason (Geankoplis et al., 2008). At the same exchanger length, heat transfer is more effective with a countercurrent arrangement than in a parallel flow.

4.4 Overall heat transfer coefficients

The overall heat transfer coefficient is a measure of the overall ability of a series of conductive and convective barriers to transfer heat and may be used to calculate rate of heat transfer in heat exchangers. The theoretical overall heat transfer coefficient is the reciprocal of the total resistance to heat transfer from one fluid to another including resistances due to the films of the fluid on either side of the tube wall and that of the tube wall as well (Incropera, 2007). Overall heat transfer coefficient can be determined from experiment as well using the overall temperature difference, heat transfer rates and the heat transfer area which is the average lateral surface area of the tube.

For both flow configurations, the overall heat transfer coefficient was calculated based on the actual flow ratio since the heat exchanger was malfunctioning at the time of the experiment. The flow rate of the hot fluid displayed in the computer could not be modified which resulted in faulty data. To accurately calculate the overall heat transfer coefficient, the actual mass flow rate hot fluid was determined using the expression for the actual heat transfer rate. The actual mass flow rate of the hot fluid was then used to calculate the Reynolds number which is needed to determine the Nusselt number. The Nusselt number was used to calculate the heat transfer coefficient.

Table 8. Experimental and Theoretical Overall Heat Transfer Coefficients for Countercurrent operation

$T_{setting}$ (°C)	Flow Ratio (F_{HOT}:F_{COLD})	U_{theo} (W/m·K)	U_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	3796.06	3638.37	4.15
	0.2:2	3217.61	2768.81	13.95
	0.2:1	2367.01	2109.63	10.87
55	0.2:1	2465.06	2213.18	10.22
65	0.2:1	2630.53	2332.22	11.34

Shown in Table 8 are the theoretical and experimental overall heat transfer coefficients for countercurrent operation under different temperature settings and flow rate ratios. As observed in all the temperature readings and flow ratios, the U_{theo} obtained is higher than the U_{exp}, deviating from 4.15% to 13.95% from the experimental value. The main reason for this discrepancy is that the calculation of heat loss throughout the system was not able to account for the fittings and other factors that contribute to heat loss.

Table 9. Experimental and Theoretical Overall Heat Transfer Coefficients for Co-current operation

$T_{setting}$ (°C)	Flow Ratio (F_{HOT}:F_{COLD})	U_{theo} (W/m·K)	U_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	3678.51	3633.47	1.22
	0.2:2	3261.06	3288.67	0.85
	0.2:1	2331.96	1993.07	14.53
55	0.2:1	2399.77	2089.68	12.92
65	0.2:1	2450.41	2308.59	5.79

Shown in Table 9 are the theoretical and experimental overall heat transfer coefficients for co-current operation under different temperature settings and flow rate ratios. As observed in all the temperature readings and flow ratios, the U_{theo} obtained is higher than the U_{exp}, deviating from 0.85% to 14.53% from the experimental value. Again, the main reason for this difference is that the calculation of heat loss throughout the system was not able to account for the fittings and other factors that contribute to heat loss.

From Table 8 and 9, it can be observed that as the flow rate ratio of the cold fluid is decreased with respect to the hot fluid, setting all other variables constant, the value of the overall heat transfer coefficient decreases. This trend is observed in both configurations. Likewise, at constant flow rate ratio, a similar trend is observed for the overall heat transfer coefficient. As the temperature increases in both countercurrent and co-current operation, the value of the overall heat transfer coefficient increases accordingly. This implies that at higher temperatures and at higher flow ratio of hot water with respect to the cold water, lesser resistances are present since by definition, the theoretical overall heat transfer coefficient is the reciprocal of the sum of all resistances involved in the heat transfer process. Thus, heat is transferred easier to the system when the overall heat transfer coefficient has a greater value.

The difference in heat transfer rates computed indicate that the hot and cold fluids were not in thermal equilibrium. The possible causes for this may be the inconsistent flow rates of water evidenced by fluctuation in flow rate readings given by the equipment which may be attributed to leaking observed in the equipment during the experiment. For well insulated heat exchangers, the heat energy absorbed by the cold fluid is ideally equal to the heat loss by the hot fluid but in actual practice, the differences in computed values may be due to heat loss or gain to the environment.

4.5 Heat Transfer Effectiveness

The heat exchanger effectiveness is defined as the ratio of the heat transferred in the actual heat exchanger to the heat that would be transferred in the ideal heat exchanger (Incropera, 2007).

In the ideal heat exchanger, the maximum possible amount of heat is transferred; this is the difference between the hot and fluids' inlet temperatures. Also, it is the fluid with the lowest thermal capacity that is used to compute the maximum heat that could be transferred between the fluids per unit time. Efficiency is dependent on the heat exchanger geometry, flow configuration and number of transfer units (NTU). NTU is the ratio between the heat transferred for a unit temperature and heat absorbed by the fluid stream that results in a temperature change of the same degree (Geankoplis, 2006).

Table 10. Experimental and Theoretical Heat Exchanger Effectiveness for Countercurrent operation

$T_{setting}$ (°C)	Flow Ratio (F_{HOT}:F_{COLD})	ε_{theo} (W/m·K)	ε_{exp} (W/m·K)	Percent Error (%)
	0.2:3	0.2590	0.2500	3.47
45	0.2:2	0.3222	0.2857	11.34
	0.2:1	0.4323	0.3972	8.13
55	0.2:1	0.4474	0.4136	7.56
65	0.2:1	0.4649	0.4261	8.33

Shown in Table 10 are the theoretical and experimental values of heat exchanger effectiveness for countercurrent operation under different temperature settings and flow rate ratios. From the table, the experimental and theoretical values range from 0.25 to 0.46. As observed in all the temperature readings and flow ratios, the ε_{theo} obtained is higher than the ε_{exp}, deviating from 3.47% to 11.34% from the experimental value. When the increasing flow rate ratio is taken into account, there is an observed increase in the heat exchanger effectiveness. At 45°C the value goes from 0.2590 to 0.4323 for the theoretical and from 0.2500 to 0.3972 for the experimental value. Moreover, when the increasing temperature is taken into account and the flow ratio is held constant, there is also an observed increase in the heat exchanger effectiveness.

Table 11. Experimental and Theoretical Heat Exchanger Effectiveness for Co-current operation

$T_{setting}$ (°C)	Flow Ratio (F_H:F_C)	ε_{theo} (W/m·K)	ε_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	0.2577	0.2524	2.04
	0.2:2	0.3182	0.3168	0.47
	0.2:1	0.4175	0.3676	11.96
55	0.2:1	0.4341	0.3881	10.59
65	0.2:1	0.4568	0.4321	5.41

Shown in Table 11 are the theoretical and experimental values of heat exchanger effectiveness for co-current operation under different temperature settings and flow rate ratios. From the table, the experimental and theoretical values range from 0.25 to 0.46. As observed in all the temperature readings and flow ratios, the ε_{theo} obtained is higher than the ε_{exp}, deviating from 0.47% to 11.96% from the experimental value. At 45°C the value goes from 0.2577 to 0.4175 for the theoretical and from 0.2524 to 0.3676 for the experimental value. Moreover, when the increasing temperature is taken into account and the flow ratio is held constant, there is also an observed increase in the heat exchanger effectiveness.

From the calculated experimental and theoretical values of the heat-exchanger effectiveness, the countercurrent operation of the heat exchanger resulted in slightly higher magnitudes of ε as compared to the co-current operation. One advantage of the countercurrent flow arrangement from co-current flow is that the outlet temperature of the cold fluid can approach the highest temperature of the hot fluid. When this applies, a greater effectiveness is achieved by the countercurrent flow regime (Incropera, 2007). From this, it can be inferred that high effectiveness would mean more heat is transferred when the hot and cold fluid flow counter currently with each other.

In countercurrent flow, the two fluids leave the tube at opposite ends, and so the exit temperature of the hot fluid can be lower than the exit temperature of the colder fluid. However, in a co-current flow, the exit temperature of the hot fluid will always be greater than the exit temperature of the cold fluid. This is the first indication that countercurrent flow is better than co-current flow: not all heat transfer carried out in countercurrent flow can be realized in co-current flow. One of the important advantages of countercurrent flow is the possibility of extracting a higher proportion of the heat content of the heating fluid. It is important to note that the LMTD value for countercurrent flow is much larger than for co-current flow at the same terminal temperature (operating conditions). This explains why theoretical ε for countercurrent flow is higher compared to co-current flow as stated earlier.

5. Conclusions

Varying the flow rate ratio results in the faster flowing fluid having a smaller temperature change over the heat transfer length than the other fluid. The faster the hot fluid compared to the cold fluid, the higher the outlet temperature of the cold fluid. The opposite is true. The slower the hot fluid compared to the cold fluid, the lower the outlet temperature of the cold fluid.

For the type of flow arrangement, the countercurrent has a constant rate of heat transfer whereas in the co-current operation, the rate of heat transfer decreases with respect to the length of the heat exchanger thus it can be inferred that the countercurrent flow is preferred to the concurrent flow arrangement.

The overall heat transfer coefficient is a measure of the overall ability of the heat exchanger for heat transfer amidst the convective and conductive barrier of the heat exchanger unit. It is the reciprocal of the resistances that occur within the system. A higher value of the overall heat transfer coefficient, U indicates that the heat exchanger unit is effective for heat transfer. The overall heat transfer coefficient is expected to be higher in a countercurrent flow arrangement than in a co-current double-pipe heat exchanger. This is attributed to the fact that LMTD is higher in a countercurrent flow than in a co-current flow. The greater the driving force, the more efficient is the heat transfer.

The effectiveness of the heat exchanger unit is defined as the ratio between the actual heat transfer and the maximum possible heat transfer. The closer is the effectiveness to unity, the more effective is the heat transfer process. In the experiment, the effectiveness is in the range of 0.25-0.46. Some values of the effectiveness obtained for co-current were greater than that in countercurrent which deviates from what is expected.

References

Coulson, J.M., Richardson, J.F. (1999). Coulson and Richardson's Chemical Engineering Volume 1 - Fluid Flow, Heat Transfer and Mass Transfer (6th Ed). USA: Elsevier

Geankoplis, C. J. (2003). Principles of Transport Processes. New Jersey, New Jersey, USA: Pearson

Geankoplis, C. J., Hersel, A., & Lepek, D. J. (2018). Transport Processes and Separation Process Principles. Pearson.

Geankoplis, C.J. (2006). Transport Processes and Unit Operations, 4th edition, Prentice Hall, New Jersey.

Incropera, F. P., DeWitt, D. P., Bergman, T. L., & Lavine, A. S. (2007)Fundamentals of Heat and Mass Transfer, 6th ed.

Çengel, Y. A. (2007). Heat and mass transfer: A practical approach. Boston: McGraw-Hill.

ANNEX

Data Processing & Analysis Report

University of San Carlos
Department of Chemical Engineering

CHE 3214L Chemical Engineering Laboratory Investigations 1

Data Processing & Analysis Report

(Form CHE 3214L-3)

Prepared and submitted by:

Chillann Kirk D. Illaga | Jerold J. Monsanto | Jean Mercy C. Rosal

Experiment 5: **Heat Transfer in Double Pipe Heat Exchanger**

Objectives of the Experiment

1. Investigate the effects of varying the inlet temperature, the flow rates, and the flow arrangement (countercurrent or cocurrent parallel flow) on the temperature profiles in a double-pipe heat exchanger.
2. Determine the overall heat transfer coefficients and the effectiveness of the heat exchanger unit at the required operating conditions and compare them with theoretically calculated values.

Results & Discussion

Heat transfer, unlike state functions, is a process function; thus, the quantity of heat transferred in a thermodynamic process that changes the state of a system depends on how that process occurs, not just on the net difference between the initial and final states of the process. In its most basic form, heat transfer is focused on only two things: temperature and heat flow. The amount of thermal energy accessible is represented by temperature, whereas heat flow is the movement of thermal energy from one location to another.

For this experiment, the parameters used during the actual conduct are shown in the table below. As shown in the table, the dimensions of inner and outer tubes are taken into account such as the different values for the outer diameter for the inner tube, the inner diameter of the outer tube, as well as the cross sectional area for the inner and outer tube.

Table 1.1. Equipment specifications

	Heat Exchanger	
Inner Tube	**Diameter (D_1), *m***	0.0095
	Wall Thickness (W_1), *m*	0.0006
	Hydraulic Diameter (D_i), *m*	0.0083
	Cross-sectional Area (A_1), *m^2*	5.41E-05
	Heat Transfer Area (A_i), m^2	0.0172
Outer Tube	**Diameter (D_2), *m***	0.012
	Wall Thickness (W_2), *m*	0.003
	Hydraulic Diameter (D_o), *m*	0.0025
	Cross-sectional Area (A_2), *m^2*	4.22E-05
	Heat Transfer Area (A_o), m^2	0.0197
Steel Thermal Conductivity (k), *W/m·K*		16
Total Length of Heat Transfer Section (L), *m*		0.66
Log Mean Heat Transfer Area (A_{lm}), m^2		0.0184

From these parameters listed, the inner tube heat transfer area (A_i) and the outer tube heat transfer area (A_o) were calculated using the formula

$$A = \pi DL \qquad [1.1]$$

where A is the heat transfer area of a tube, D is the diameter of the tube, and L is the tidal length of the heat transfer section. This value is of importance as this suggests where the heat transfer occurs. Since in a double-pipe heat exchanger, two flow arrangement are possible which are the cocurrent and countercurrent, the log mean heat transfer area must also be calculated using the formula

$$A_{lm} = \frac{A_o - A_i}{ln\frac{A_0}{A_i}} \qquad [1.2]$$

where A_{lm} is the log mean heat transfer area, A_i is the inner tube heat transfer area, and A_o is the outer tube heat transfer area. The inner and outer tube heat transfer areas can be used to sum up all resistances hence, calculating the overall heat transfer coefficient which are to be tackled in the second part of this paper.

In this part of the experiment, effects of changing the inlet temperature, the volumetric flow rate, and the flow arrangement on the temperature profile for both countercurrent and concurrent flows are studied.

1. Effect of Inlet Temperature on the Temperature Profiles in a Double-pipe heat exchanger

Effect of varying the inlet temperature on the temperature is studied by having three different hot stream inlet temperatures of 45ºC, 55ºC, and 65ºC at a fixed flow rate ratio of (F_{HOTt}:F_{COLD}) 0.2:1 for both flow arrangements. Shown in Figure 1.1 is the temperature profile of the heat exchanger for a countercurrent operation and shown in Figure 1.2 is the temperature profile of the heat exchanger for a cocurrent operation.

1.1. Countercurrent operation

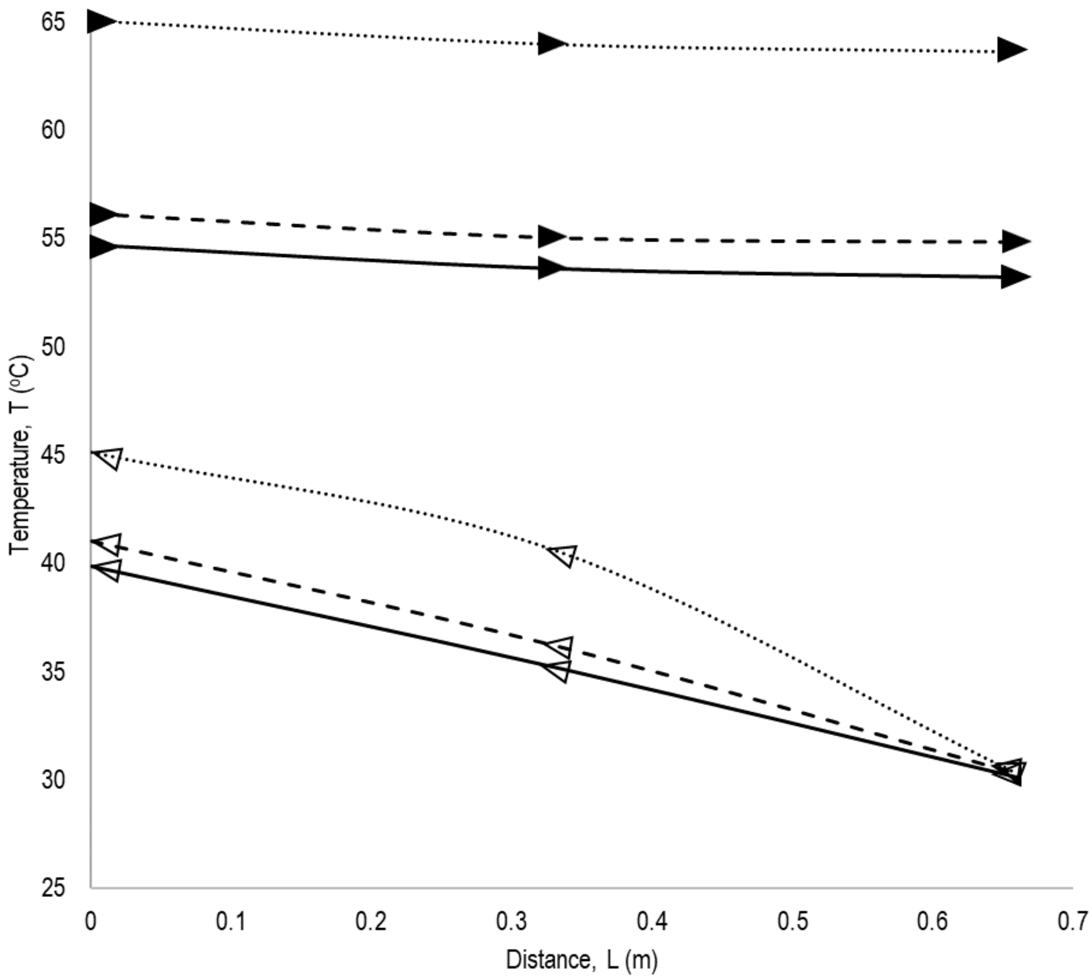

Figure 1.1. Temperature Profiles of Different Inlet Temperatures of the Double-pipe Heat Exchanger Operating in a Countercurrent Flow at a flow ratio of 0.2:1 (arrows: direction of flow; bold line: 45ºC; dashed line: 55ºC; dotted line: 65ºC; closed symbols: hot fluid; open symbols: cold fluid).

As shown in the graph, as the inlet temperature is increased, there is also an increase in the outlet temperature of the cold stream. It can be observed from the graph that the temperature difference at each temperature setting is approximately constant throughout the exchanger length. Less variation in the temperature difference throughout the heat exchanger is expected for fluid flowing opposite to each other (Coulson & Richardson, 1999). At 65°C, there is a larger temperature gradient compared to the 45°C and 55°C inlet temperatures. The ΔT_{max} for 65°C is 2.3 and 1.4 times larger relative to 45°C and 55°C, respectively.

As the inlet temperatures of the entering hot fluid is increased, though the difference of the temperatures across the exchanger is approximately constant, the difference becomes larger. This is signified by the increasing distance between the two curves for a pair of flows at each temperature. The inlet temperature of the cold fluid entering is essentially constant because the tap water temperature is not altered. The only factor affecting the temperature difference therefore is the inlet temperature of the entering hot fluid. At higher inlet temperatures of the hot fluid, there is a greater driving force for heat transfer to occur.

Heat transfer in a heat exchanger is governed by Fourier's Law on Heat Conduction and Newton's Law on Cooling. Both principles show that heat transfer is proportional to the temperature difference present. The temperature difference serves as the driving force that affects the transfer of heat. By increasing the inlet temperature of the inner tube, the temperature difference is increased thereby affecting more heat transfer to occur. However, in the counter-current configuration, the cold fluid inlet temperature is kept constant hence its temperature profile would pivot from this point.

1.2. Cocurrent operation

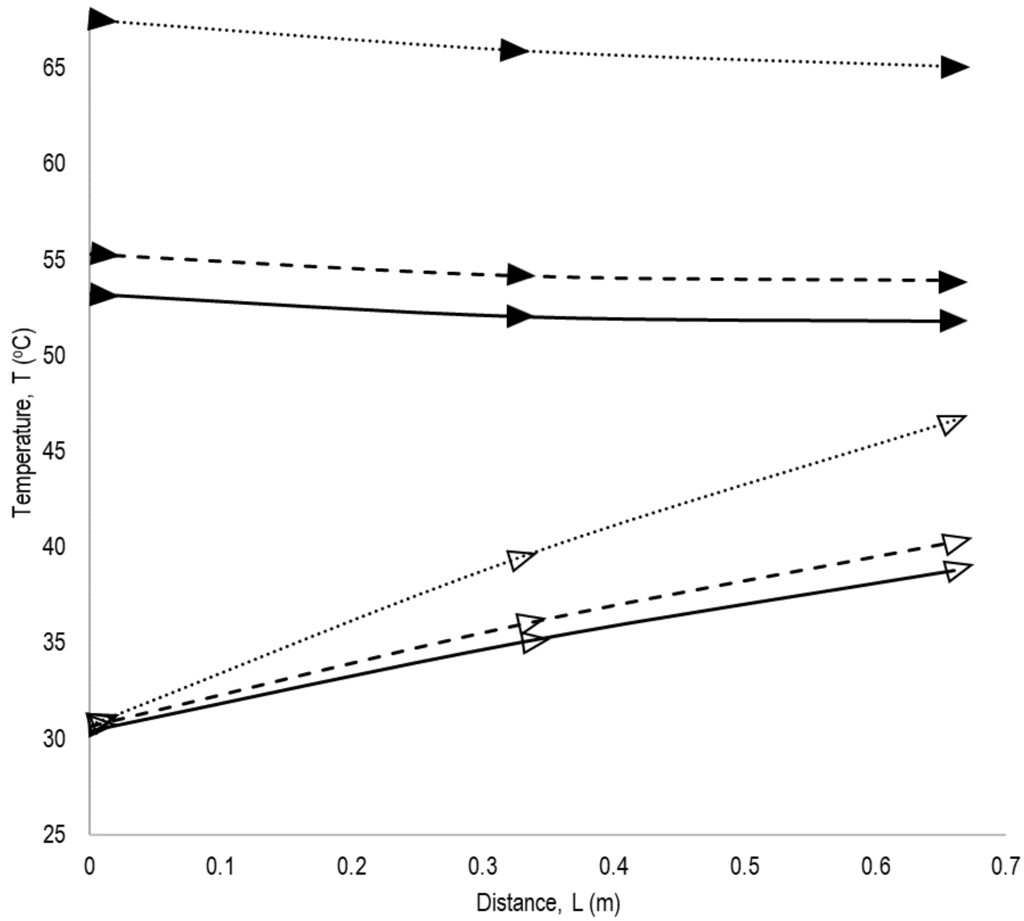

Figure 1.2. Temperature Profiles of Different Inlet Temperatures of the Double-pipe Heat Exchanger Operating in a Cocurrent Flow at a flow ratio of 0.2:1 (arrows: direction of flow; bold line: 45°C; dashed line: 55°C; dotted line: 65°C; closed symbols: hot fluid; open symbols: cold fluid).

In a cocurrent flow arrangement with all conditions kept constant except its inner tube inlet temperature, the resulting temperature profile is shown in Figure 1.2. Similar to that of the countercurrent flow arrangement, the temperature difference increases with increasing hot fluid inlet temperature. The increase of temperature results to the increase of ΔT_{max} and ΔT_{min} by an average of 55% and 39% respectively. The cold and hot temperature lines approach each other along the pipe length in the co-current regime. The temperature of the streams flowing in the same direction gradually approaches the same value (Coulson & Richardson, 1999). Moreover, at 65°C, there is a larger temperature gradient compared to the 45°C and 55°C inlet temperatures which is also observed in the countercurrent flow arrangement.

It can also be observed from the graph that the change in the temperature is observed at the inlet portion of both fluids because there is a greater temperature difference at this portion while as both fluids flow to the other part of the pipe, there is a minimal change in the temperature of both fluids. This makes the cocurrent flow suitable for abrupt temperature decreases. As can be noted from Figure 1.2 that the temperature for both fluids tends to achieve uniformity and this is true for cocurrent flows at infinite areas of heat transfer.

Table 1.2. Temperature differences of the double-pipe heat exchanger at different inner tube temperatures with a 0.2:1 flow ratio.

Inlet temperature ($T_{IN,HOT}$), ^oC	Countercurrent flow		Cocurrent flow	
	Maximum mean temperature difference (ΔT_{max}), °C	Minimum mean temperature difference (ΔT_{min}), °C	Maximum mean temperature difference (ΔT_{max}), °C	Minimum mean temperature difference (ΔT_{min}), °C
45°C	23.10	14.83	22.77	13.00
55°C	24.53	15.17	24.63	13.37
65°C	33.33	19.93	36.83	18.47

As presented in Table 1.2, an increase in the temperature gradient of the hot and cold streams causes an increase in the ΔT_{max} and ΔT_{min} in the profile. The maximum temperature difference (ΔT_{max}) of the hot and cold water is defined as the difference of the outlet temperature of the hot water and the inlet temperature of the cold water for a countercurrent flow. Heat is being transferred from the hot water to the cold water. The cold water absorbs heat from the hot water which then increases its temperature proportional to the amount of heat that was transferred. The amount of heat transferred is directly proportional to the temperature difference of the two fluids. The greater the temperature difference, the greater is the heat transfer rate. The rate of heat transfer, however, decreases with respect to the length of the heat exchanger. From the table, it is also evident that the minimum mean temperature difference (ΔT_{min}) of the countercurrent flow arrangement is relatively larger than that of the concurrent flow arrangement which suggests that the heat transfer is more effective with a countercurrent arrangement since it offers larger temperature driving forces hence, usually preferred over the co-current flow for this reason (Geankoplis, 2003).

2. Effect of Flow rates on the Temperature Profiles in a Double-pipe heat exchanger

Aside from the inlet temperatures, the profiles also vary considerably depending on the flow rate. Shown in figures 2.1 and 2.2 are the temperature profiles of the double-pipe heat exchanger in both countercurrent and cocurrent flows, respectively. For both arrangements, the cold inlet temperatures were kept constant at approximately 30°C and the hot inlet temperature was kept constant at 45°C. However, due to some equipment defects, the hot inlet temperature slightly varies from 46-54°C. To analyze the effect of volumetric flow rate on the temperature profile, three flow rate ratios (F_{HOT}: F_{COLD}) were varied accordingly with a decreasing outer tube flow rate (F_{COLD}); the flow ratio parameters are 0.2: 3, 0.2:2, 0.2:1 L/min.

2.1. Countercurrent operation

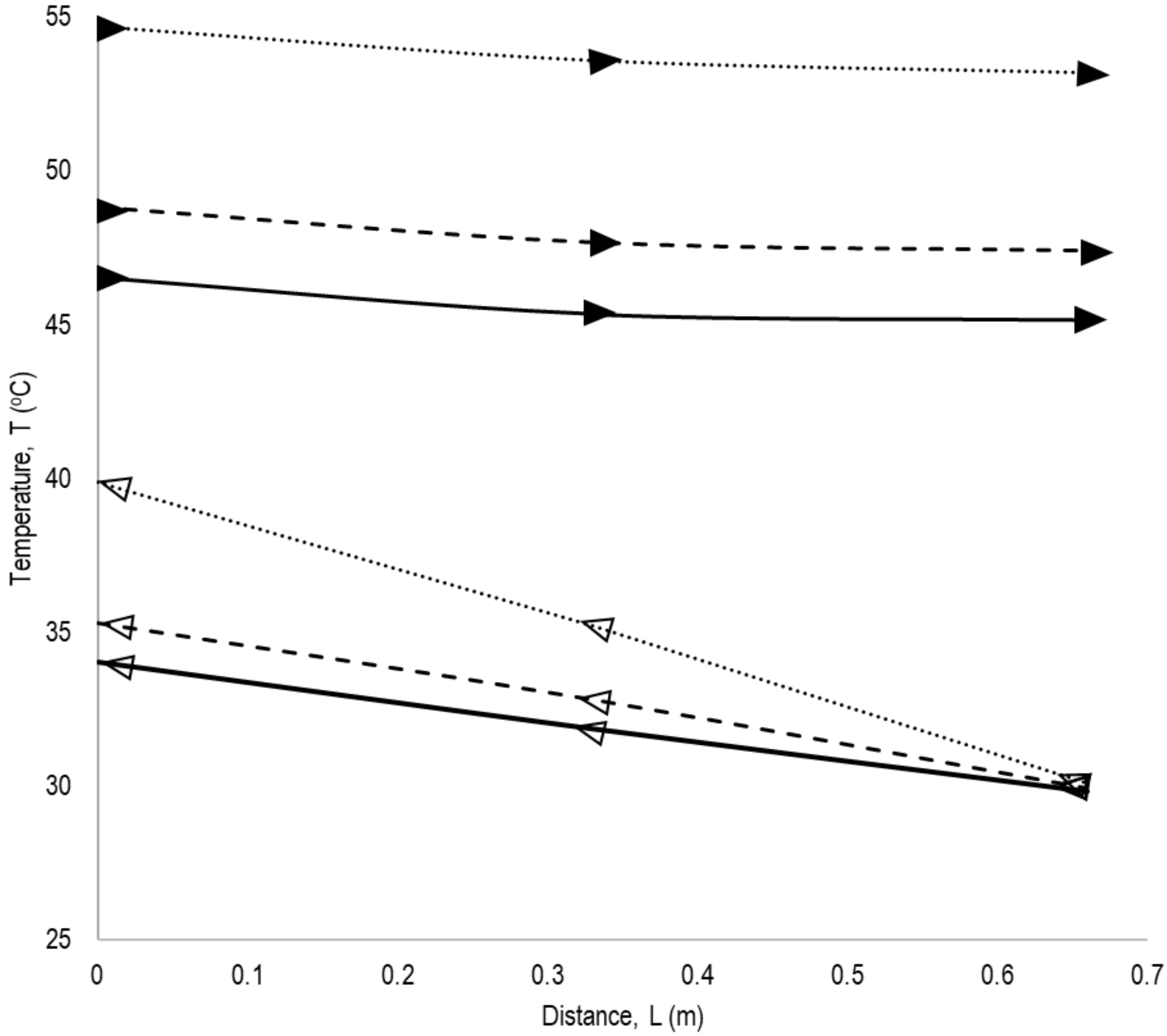

Figure 2.1. Temperature Profiles of Different Flow Ratios of the Double-pipe Heat Exchanger Operating in a Countercurrent Flow at a Temperature Setting of 45°C (arrows: direction of flow; bold line: 0.2:3 flow ratio; dashed line: 0.2:2 flow ratio; dotted line: 0.2:1 flow ratio; closed symbols: hot fluid; open symbols: cold fluid).

In a countercurrent operation with all conditions kept constant except its outer tube flow rate (F_{COLD}), a higher flow ratio value would result from the decrease in the outer tube flow rate shown in table 2.1; or this can be mathematically expressed as 0.2:3 < 0.2:2 < 0.2:1.

Table 2.1. Flow rate ratios and their equivalent values.

Flow Rate Ratio (F_{HOT}:F_{COLD}), *L/min*	**Ratio value, F_{HOT} / F_{COLD}**
0.2:3	0.067
0.2:2	0.1
0.2:1	0.2

These variations will have an effect on the outlet temperatures of the cold fluid. This can be observed in figure 2.1 wherein increasing the cold flow rate or the outer tube flow rate will result in a less steeper temperature profile of cold fluid. This implies that decreasing the flow ratio will yield a lower outlet temperature of the cooler fluid. The implication also holds true when there is a faster rate of hot fluid compared to the cold fluid. In this case, there would be a higher outlet temperature of cold fluid since this results in a greater flow ratio thereby generating a steeper profile. The opposite is also true—the slower the hot fluid compared to the cold fluid, the lower the outlet temperature of the cold fluid.

In general, varying the flow rate causes the faster flowing fluid to have a smaller temperature change over the heat transfer length than the other fluid. This makes physical sense since a faster flow rate allows a lesser contact time for a liquid; hence, a smaller change in temperature will happen from the entry point to its exit point. If the processed fluid is much faster, its exit temperature will only be minimal while the other fluid will be greater. On the other hand, if the processed fluid is slower, the effect would be the contrary.

The governing principle of these lies in the convective heat transfer. Convection is the transfer of heat by a bulk movement of liquid. This is a process that is essentially the exchange of energy via mass transfer. Changing the process of fluid flow rate, such as increasing or decreasing the volumetric flow rate of either hot and cold fluids, will change the exit temperatures. These changes in the exit temperature happens because the mean temperature difference will adjust to match the same heat transferred (Geankoplis et al., 2018). This could also be further explained by the equation of heat transfer rate which is

$$\emptyset_H = \left(mC_p\right)_{HOT}(T_{OUT,HOT} - T_{IN,HOT}) = \left(mC_p\right)_{COLD}(T_{OUT,COLD} - T_{IN,COLD}) \quad [2.1]$$

where $\emptyset_H$ is the heat transfer rate in kJ/s, m is the mass flow rate of liquid in kg/min, C_p is the heat capacity of liquid in kJ·kg^{-1}·°C^{-1}, T_{IN} is the inlet temperature and T_{OUT} is the outlet temperature in °C.

As established in equation 2.1, assuming that heat transfer rate does not change, flow rate is inversely proportional to the temperature change. When the flow rate of cold water is increased, there will be a decrease in temperature difference of the cold fluid. The same goes for the hot fluid. When the hot fluid flow rate is increased, there will only be a minimal hot fluid temperature difference.

2.2. Cocurrent operation

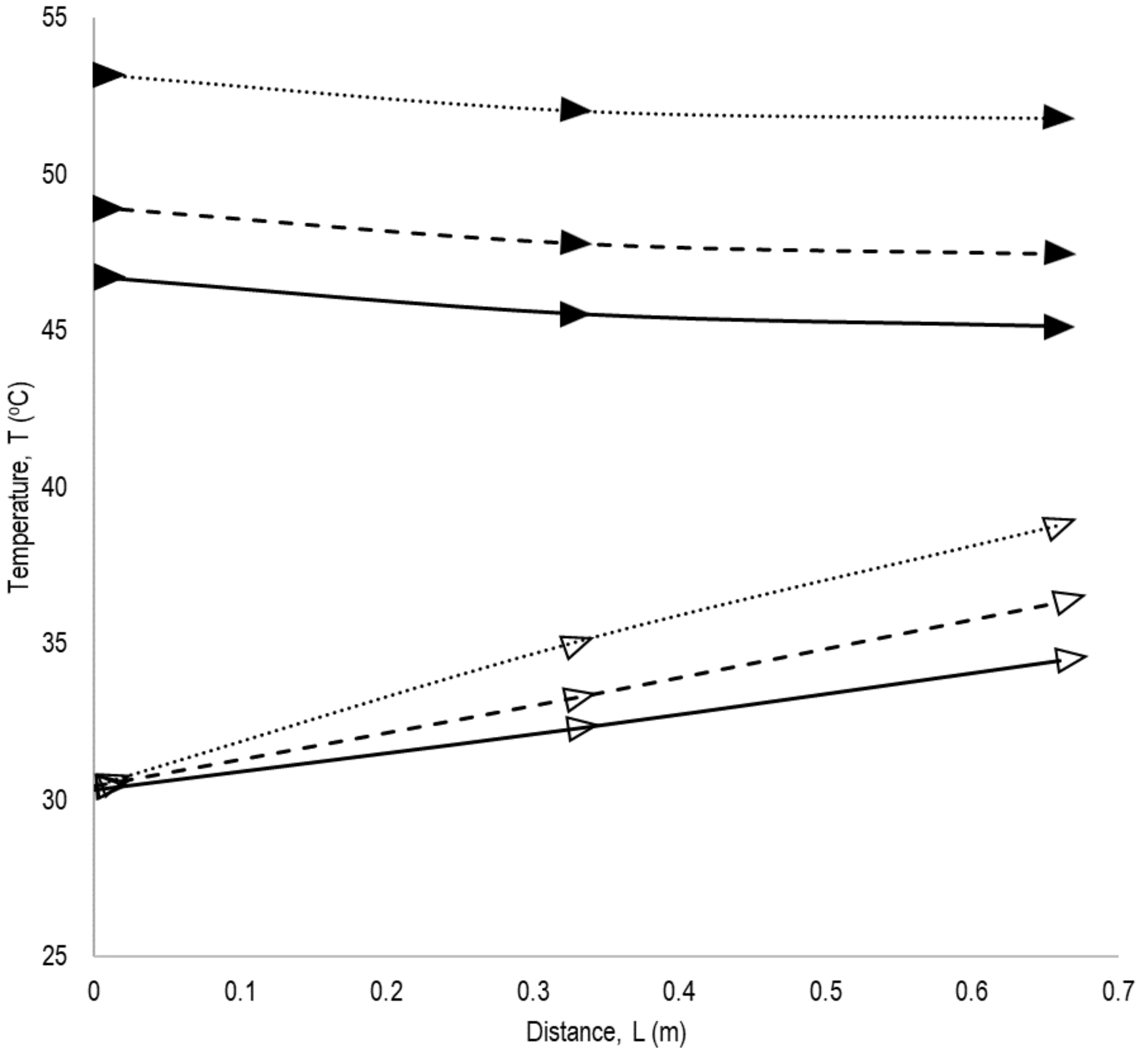

Figure 2.2. Temperature Profiles of Different Flow Ratios of the Double-pipe Heat Exchanger Operating in a Cocurrent Flow at a Temperature Setting of 45°C (arrows: direction of flow; bold line: 0.2:3 flow ratio; dashed line: 0.2:2 flow ratio; dotted line: 0.2:1 flow ratio; closed symbols: hot fluid; open symbols: cold fluid).

In a double-pipe heat exchanger with a cocurrent configuration with the same set of conditions, an analogous explanation similar to that of the countercurrent can be used to explain the trends. As shown in figure 2.2, a lower cold flow rate as denoted by the dotted lines will have a steeper trend as compared to the dashed and bold lines which indicate a higher cold flow rate. A relationship can be established such that the outer tube flow rate is indirectly proportional to the steepness of the line. Given that a steep trend indicates a higher change in temperature, it can be deduced that outer tube flow rate is also indirectly proportional to the change in temperature of cold fluid.

Table 2.2. Temperature differences of the double-pipe heat exchanger with varying cold flow rates at a temperature of 45°C.

Flow Rate Ratio (F_{HOT}:F_{COLD}), *L/min*	Countercurrent flow		Cocurrent flow	
	Maximum mean temperature difference (ΔT_{max}), *°C*	Minimum mean temperature difference (ΔT_{min}), *°C*	Maximum mean temperature difference (ΔT_{max}), *°C*	Minimum mean temperature difference (ΔT_{min}), *°C*
0.2:3	15.33	12.53	16.40	10.67
0.2:2	17.50	13.50	18.50	11.13
0.2:1	23.10	14.83	22.77	13.00

Shown in table 2 are the minimum and maximum temperature differences of the double-pipe heat exchanger in co- and countercurrent flows. It can be seen that the maximum and minimum temperatures in both configurations increases as the cold flow rate decreases. An increase in magnitude of the ΔT_{max} and ΔT_{min} from 15.33°C to 23.10°C and 12.53°C to 14.83°C, respectively, can be observed in the countercurrent flow as the cold flow rate decreases from 3 L/min to 1 L/min. This can also be observed in cocurrent flow wherein the maximum temperature difference rises from 16.40°C to 22.77°C and the minimum temperature difference increases from 10.67°C to 13.00°C. These temperature differences are the driving forces. The greater the temperature difference, the greater is the rate of heat transfer; a greater heat transfer then would occur at low velocities of cold fluid. It is also important to note that there is a large temperature difference that prevails in cocurrent mode which results in a relatively large heat transfer rate. However, the driving force decreases along the path length of the heat exchanger (Foust, 1980). This can be observed from the drastic change in temperature from ΔT_{max} to ΔT_{min}. On the other hand, in countercurrent mode, the temperature difference along the length of the heat exchanger is relatively constant. This can be explained by the fact that there is a lesser change in temperature from ΔT_{max} to ΔT_{min}. Some errors are attributed to defective temperature sensors causing inconsistencies in the temperature readings and uncontrollable fluctuations in hot water flow rate.

3. Effect of Flow Arrangement on the Temperature Profiles in a Double-pipe heat exchanger

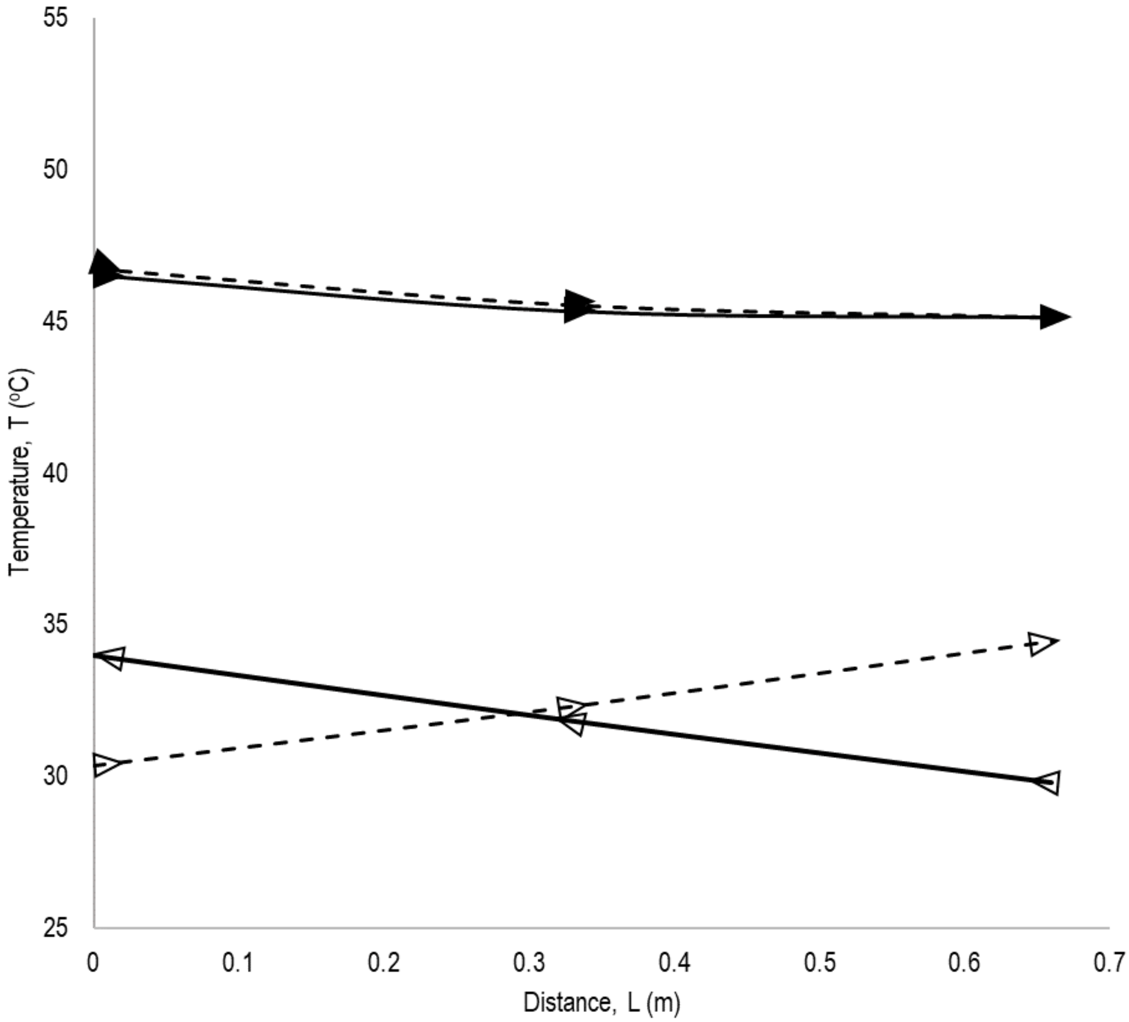

Figure 3. Temperature Profiles of the Double-pipe Heat Exchanger Operating in different flow arrangements at a flow ratio of 0.2:3 (arrows: direction of flow; bold line:countercurrent flow arrangement; dashed line: cocurrent flow arrangement; closed symbols: hot fluid; open symbols: cold fluid).

To investigate the effect of the flow arrangement to the temperature profile in the heat exchanger, countercurrent operations are compared with the co-current operations as shown in Figure 3. The temperature profiles in the countercurrent operation have a relatively constant temperature gradient along the length of the heat exchanger. Whereas the temperature profiles in the co-current operation experience a decrease in temperature gradient along the length of the heat exchanger. Therefore, heat transfer is constant in countercurrent operation while it varies for co-current with respect to the length of the heat exchanger.

In the countercurrent operation, the hot and cold fluids flow in opposite directions and the temperature difference along the length of the heat exchanger is relatively constant. In the co-current operation, both the hot and the cold fluids enter the exchanger at the same end and flow in the same direction. At the entrance of the exchanger, a large temperature difference prevails, resulting in a relatively large heat-transfer rate. This driving force decreases along the length of the heat exchanger. As a consequence, the heat exchange rate drops asymptotically as the streams approach the end of the exchanger.

The difference between the mean temperature difference with respect to the countercurrent and co-current flow affects the temperature driving force for heat transfer. Because of its larger mean temperature difference, countercurrent flow offers larger temperature driving forces and is usually preferred over the co-current flow for this reason (Geankoplis et al., 2008). At the same exchanger length, heat transfer is more effective with a countercurrent arrangement than in a parallel flow.

4. Overall heat transfer coefficients

The overall heat transfer coefficient is a measure of the overall ability of a series of conductive and convective barriers to transfer heat and may be used to calculate rate of heat transfer in heat exchangers. The theoretical overall heat transfer coefficient is the reciprocal of the total resistance to heat transfer from one fluid to another including resistances due to the films of the fluid on either side of the tube wall and that of the tube wall as well (Incropera, 2007). Overall heat transfer coefficient can be determined from experiment as well using the overall temperature difference, heat transfer rates and the heat transfer area which is the average lateral surface area of the tube.

The heat transfer process occurring in a double-pipe heat exchanger involved two prevailing heat transfer mechanisms namely conduction and convection. Heat is transferred convectively from the hot fluid to the interface of the fluid and the wall of the inner pipe. This mechanism, like all other forms of transfer, experiences a certain convective resistance before the heat can penetrate through this interface. Furthermore, once the heat has already penetrated past through the interface, heat then flows through the thickness of the inner pipe. This form of heat transfer mechanism is called conduction. The heat being transferred conductively experiences a certain resistance before finally reaching the interface between the wall of the inner pipe and the cold fluid, which transfers heat via convection. These resistances dictate the ease of flow of heat from the hot to cold fluid. The greater the resistance, the smaller is the overall heat transfer coefficient, hence a lesser rate of heat transfer between the two fluid mediums (Çengel, 2007).

For both flow configurations, the overall heat transfer coefficient was calculated based on the actual flow ratio since the heat exchanger was malfunctioning at the time of the experiment. The flow rate of the hot fluid displayed in the computer could not be modified which resulted in faulty data. To accurately calculate the overall heat transfer coefficient, the actual mass flow rate hot fluid was determined using the expression for the actual heat transfer rate. The actual mass flow rate of the hot fluid was then used to calculate the Reynolds number which is needed to determine the Nusselt number. The Nusselt number was used to calculate the heat transfer coefficient.

Table 4.1. Experimental and Theoretical Overall Heat Transfer Coefficients for Countercurrent Operation

$T_{setting}$ (°C)	Flow Ratio (F_H:F_C)	Actual Flow Ratio	U_{theo} (W/m·K)	U_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	8.58:2.99	3796.06	3638.37	4.15
	0.2:2	7.60:1.96	3217.61	2768.81	13.95
	0.2:1	6.56:1.00	2367.01	2109.63	10.87
55	0.2:1	8.05:1.00	2465.06	2213.18	10.22
65	0.2:1	10.91:1.02	2630.53	2332.22	11.34

Shown in table 4.1 are the theoretical and experimental overall heat transfer coefficients for countercurrent operation under different temperature settings and flow rate ratios. As observed in all the temperature readings and flow ratios, the U_{theo} obtained is higher than the U_{exp}, deviating from 4.15% to 13.95% from the experimental value. The main reason for this discrepancy is that the calculation of heat loss throughout the system was not able to account for the fittings and other factors that contribute to heat loss.

Table 4.2. Experimental and Theoretical Overall Heat Transfer Coefficients for Co-current Operation

$T_{setting}$ (°C)	Flow Ratio (F_H:F_C)	Actual Flow Ratio	U_{theo} (W/m·K)	U_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	7.54:2.90	3678.51	3633.47	1.22
	0.2:2	7.93:2.02	3261.06	3288.67	0.85
	0.2:1	6.20:1.03	2331.96	1993.07	14.53
55	0.2:1	6.95:1.01	2399.77	2089.68	12.92
65	0.2:1	6.44:0.96	2450.41	2308.59	5.79

Shown in table 4.2 are the theoretical and experimental overall heat transfer coefficients for co-current operation under different temperature settings and flow rate ratios. As observed in all the temperature readings and flow ratios, the U_{theo} obtained is higher than the U_{exp}, deviating from 0.85% to

14.53% from the experimental value. Again, the main reason for this difference is that the calculation of heat loss throughout the system was not able to account for the fittings and other factors that contribute to heat loss.

From table 4.1 and 4.2, it can be observed that as the flow rate ratio of the cold fluid is decreased with respect to the hot fluid, setting all other variables constant, the value of the overall heat transfer coefficient decreases. This trend is observed in both configurations. Likewise, at constant flow rate ratio, a similar trend is observed for the overall heat transfer coefficient. As the temperature increases in both countercurrent and cocurrent operation, the value of the overall heat transfer coefficient increases accordingly. This implies that at higher temperatures and at higher flow ratio of hot water with respect to the cold water, lesser resistances are present since by definition, the theoretical overall heat transfer coefficient is the reciprocal of the sum of all resistances involved in the heat transfer process. Thus, heat is transferred easier to the system when the overall heat transfer coefficient has a greater value.

The difference in heat transfer rates computed indicate that the hot and cold fluids were not in thermal equilibrium. The possible causes for this may be the inconsistent flow rates of water evidenced by fluctuation in flow rate readings given by the equipment which may be attributed to leaking observed in the equipment during the experiment. For well insulated heat exchangers, the heat energy absorbed by the cold fluid is ideally equal to the heat loss by the hot fluid but in actual practice, the differences in computed values may be due to heat loss or gain to the environment.

5. Heat transfer effectiveness

The heat exchanger effectiveness is defined as the ratio of the heat transferred in the actual heat exchanger to the heat that would be transferred in the ideal heat exchanger (Incropera, 2007). In the ideal heat exchanger the maximum possible amount of heat is transferred; this is the difference between the hot and fluids' inlet temperatures. Also, it is the fluid with the lowest thermal capacity that is used to compute the maximum heat that could be transferred between the fluids per unit time. Efficiency is dependent on the heat exchanger geometry, flow configuration and number of transfer units (NTU). NTU is the ratio between the heat transferred for a unit temperature and heat absorbed by the fluid stream that results in a temperature change of the same degree (Geankoplis, 2006).

Table 5.1. Experimental and Theoretical Heat Exchanger Effectiveness for Counter-Current Operation

$T_{setting}$ (°C)	Flow Ratio (F_H:F_C)	ε_{theo} (W/m·K)	ε_{exp} (W/m·K)	Percent Error (%)
45	0.2:3	0.2590	0.2500	3.47
	0.2:2	0.3222	0.2857	11.34

	0.2:1	0.4323	0.3972	8.13
55	0.2:1	0.4474	0.4136	7.56
65	0.2:1	0.4649	0.4261	8.33

Shown in table 5.1 are the theoretical and experimental values of heat exchanger effectiveness for countercurrent operation under different temperature settings and flow rate ratios. From the table, the experimental and theoretical values range from 0.25 to 0.46. As observed in all the temperature readings and flow ratios, the $\boldsymbol{\varepsilon}_{theo}$ obtained is higher than the $\boldsymbol{\varepsilon}_{exp}$, deviating from 3.47% to 11.34% from the experimental value. When the increasing flow rate ratio is taken into account, there is an observed increase in the heat exchanger effectiveness. At 45°C the value goes from 0.2590 to 0.4323 for the theoretical and from 0.2500 to 0.3972 for the experimental value. Moreover, when the increasing temperature is taken into account and the flow ratio is held constant, there is also an observed increase in the heat exchanger effectiveness.

Table 5.2. Experimental and Theoretical Heat Exchanger Effectiveness for Co-Current Operation

$T_{setting}$ (°C)	Flow Ratio (F_H:F_C)	$\boldsymbol{\varepsilon}_{theo}$ (W/m·K)	$\boldsymbol{\varepsilon}_{exp}$ (W/m·K)	Percent Error (%)
45	0.2:3	0.2577	0.2524	2.04
	0.2:2	0.3182	0.3168	0.47
	0.2:1	0.4175	0.3676	11.96
55	0.2:1	0.4341	0.3881	10.59
65	0.2:1	0.4568	0.4321	5.41

Shown in table 5.1 are the theoretical and experimental values of heat exchanger effectiveness for co-current operation under different temperature settings and flow rate ratios. From the table, the experimental and theoretical values range from 0.25 to 0.46. As observed in all the temperature readings and flow ratios, the $\boldsymbol{\varepsilon}_{theo}$ obtained is higher than the $\boldsymbol{\varepsilon}_{exp}$, deviating from 0.47% to 11.96% from the experimental value. At 45°C the value goes from 0.2577 to 0.4175 for the theoretical and from 0.2524 to 0.3676 for the experimental value. Moreover, when the increasing temperature is taken into account and the flow ratio is held constant, there is also an observed increase in the heat exchanger effectiveness.

From the calculated experimental and theoretical values of the heat-exchanger effectiveness, the countercurrent operation of the heat exchanger resulted in slightly higher magnitudes of ε as compared to the co-current operation. One advantage of the countercurrent flow arrangement from co-current flow is that the outlet temperature of the cold fluid can approach the highest temperature of the hot fluid. When this applies, a greater effectiveness is achieved by the countercurrent flow regime (Incropera, 2007). From this, it can be inferred that high effectiveness would mean more heat is transferred when the hot and cold fluid flow counter currently with each other.

In countercurrent flow, the two fluids leave the tube at opposite ends, and so the exit temperature of the hot fluid can be lower than the exit temperature of the colder fluid. However, in a co-current flow, the exit temperature of the hot fluid will always be greater than the exit temperature of the cold fluid. This is the first indication that countercurrent flow is better than co-current flow: not all heat transfer carried out in countercurrent flow can be realized in co-current flow. One of the important advantages of countercurrent flow is the possibility of extracting a higher proportion of the heat content of the heating fluid. It is important to note that the LMTD value for countercurrent flow is much larger than for co-current flow at the same terminal temperature (operating conditions). This explains why theoretical ε for countercurrent flow is higher compared to co-current flow as stated earlier.

Conclusions

Increasing the inlet temperature of the hot fluid increases the temperature gradient of both fluids for both the countercurrent and concurrent flow arrangement and essentially increases the heat transfer rate as temperature difference is the driving force for heat transfer. Moreover, the higher the inlet temperature of the hot stream, the greater the ΔT_{max} is for both operations.

Varying the flow rate ratio results in the faster flowing fluid having a smaller temperature change over the heat transfer length than the other fluid. The faster the hot fluid compared to the cold fluid, the higher the outlet temperature of the cold fluid. The opposite is true. The slower the hot fluid compared to the cold fluid, the lower the outlet temperature of the cold fluid.

For the type of flow arrangement, the countercurrent has a constant rate of heat transfer whereas in the co-current operation, the rate of heat transfer decreases with respect to the length of the heat exchanger thus it can be inferred that the countercurrent flow is preferred to the concurrent flow arrangement.

The overall heat transfer coefficient is a measure of the overall ability of the heat exchanger for heat transfer amidst the convective and conductive barrier of the heat exchanger unit. It is the reciprocal of the resistances that occur within the system. A higher value of the overall heat transfer coefficient, U indicates that the heat exchanger unit is effective for heat transfer. The overall heat transfer coefficient is expected to be higher in a countercurrent flow arrangement than in a co-current double-pipe heat exchanger. This is attributed to the fact that LMTD is higher in a countercurrent flow than in a co-current flow. The greater the driving force, the more efficient is the heat transfer.

The effectiveness of the heat exchanger unit is defined as the ratio between the actual heat transfer and the maximum possible heat transfer. The closer is the effectiveness to unity, the more effective is the heat transfer process. In the experiment, the effectiveness is in the range of 0.25-0.46. Some values of the effectiveness obtained for co-current were greater than that in countercurrent which deviates from what is expected.

References

Coulson, J.M., Richardson, J.F. (1999). Coulson and Richardson's Chemical Engineering Volume 1 - Fluid Flow, Heat Transfer and Mass Transfer (6th Ed). USA: Elsevier

Geankoplis, C. J. (2003). Principles of Transport Processes. New Jersey, New Jersey, USA: Pearson

Geankoplis, C. J., Hersel, A., & Lepek, D. J. (2018). Transport Processes and Separation Process Principles. Pearson.

Geankoplis, C.J. (2006). Transport Processes and Unit Operations, 4th edition, Prentice Hall, New Jersey.

Incropera, F. P., DeWitt, D. P., Bergman, T. L., & Lavine, A. S. (2007)Fundamentals of Heat and Mass Transfer, 6th ed.

Çengel, Y. A. (2007). *Heat and mass transfer: A practical approach*. Boston: McGraw-Hill.

ANNEX 1: RAW DATA SHEETS

[Please see attached scanned hard copy of raw data sheets]

Table 1. *Countercurrent operation:* variation of flow rate

COUNTERCURRENT OPERATION								
Temperature Controller Setting: 45°C								
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 3								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Outlet**	**Middle**	**Inlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.2	45.2	45.4	46.6	2.98	34.0	31.9	29.8
2	0.3	45.1	45.3	46.5	2.92	34.0	31.8	29.8
3	0.2	45.1	45.3	46.5	3.06	34.0	31.8	29.8
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 2								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Outlet**	**Middle**	**Inlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.2	47.4	47.6	48.8	1.97	35.4	32.8	29.9
2	0.2	47.4	47.7	48.8	1.87	35.3	32.8	29.9
3	0.7	47.4	47.7	48.8	2.03	35.2	32.8	29.9
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Outlet**	**Middle**	**Inlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**

1	0.3	53.1	53.5	54.6	0.95	39.8	35.2	30.1
2	1.0	53.2	53.6	54.7	1.03	40.0	35.2	30.1
3	0.3	53.3	53.7	54.8	1.03	39.8	35.2	30.1
Observations:								
• The cold-water flowrate did not reach the constant desired flowrate. Instead, it fluctuated around the desired flowrate. • The hot water flowrate on the software is not adjustable. The flowrate fluctuated around the value of 0.2 L/min.								

Table 2. *Countercurrent operation:* variation of inlet temperature.

COUNTERCURRENT OPERATION								
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1								
Temperature Controller Setting: 55°C								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Outlet**	**Middle**	**Inlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.3	54.8	55.1	56.2	0.84	41.0	36.2	30.3
2	0.3	54.9	55.0	56.2	0.81	41.0	36.2	30.3
3	0.3	54.9	55.0	56.1	0.84	41.0	36.2	30.3
Temperature Controller Setting: 65°C								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Outlet**	**Middle**	**Inlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.2	53.7	54.0	55.1	1.03	39.8	35.6	30.3
2	0.2	53.7	54.0	55.1	1.04	40.1	35.6	30.3

3	0.2	53.5	53.9	54.9	0.99	40.7	35.6	30.3
Observations:								

- The set temperature setting of 55°C and 65°C were not achieved. Instead, the temperature of the hot fluid only ranged from 53.7-56.2°C.
- In hot fluid stream, temperature increases. from the outlet to the inlet, while in cold fluid stream, temperature decrease from inlet to outlet.

Table 3. *Co-current operation:* variation of flow rate.

CO-CURRENT OPERATION								
Temperature Controller Setting: 45°C								
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 3								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Inlet**	**Middle**	**Outlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.4	45.1	45.5	46.7	3.16	34.6	32.3	30.3
2	0.2	45.1	45.5	46.7	2.92	34.5	32.3	30.3
3	0.2	45.2	45.6	46.8	2.62	34.3	32.3	30.4
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 2								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Inlet**	**Middle**	**Outlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.2	47.4	47.7	48.8	2.09	36.3	33.3	30.4
2	0.2	47.4	47.8	49.0	1.99	36.3	33.3	30.5
3	0.2	47.6	47.9	49.1	1.99	36.4	33.3	30.5
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1								

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet	Middle	Inlet	F_{cold} (L/min)	Inlet	Middle	Outlet
		T_1 (°C)	T_2 (°C)	T_3 (°C)		T_4 (°C)	T_5 (°C)	T_6 (°C)
1	0.1	51.7	51.9	53.1	1.06	38.6	35.0	30.4
2	0.2	51.8	52.1	53.2	1.01	38.9	35.1	30.5
3	0.2	51.9	52.1	53.3	1.01	38.9	35.1	30.4
Observations:								

- The cold-water flowrate did not reach the constant desired flowrate. Instead, it fluctuated around the desired flowrate.
- The hot water flowrate on the software is not adjustable. The flowrate fluctuated around the value of 0.2 L/min.

Table 4. *Co-current operation:* variation of inlet temperature.

CO-CURRENT OPERATION								
Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 :1								
Temperature Controller Setting: 55°C								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Inlet**	**Middle**	**Outlet**
		T_1 (°C)	**T_2 (°C)**	**T_3 (°C)**		**T_4 (°C)**	**T_5 (°C)**	**T_6 (°C)**
1	0.2	54	54.1	55.3	0.95	40.2	36.0	30.6
2	0.2	53.9	54.2	55.3	1.07	40.3	36.0	30.6
3	0.2	53.9	54.2	55.3	1.01	40.2	35.9	30.6
Temperature Controller Setting: 65°C								
No.	**Hot Fluid**				**Cold Fluid**			
	F_{hot} (L/min)	**Outlet**	**Middle**	**Inlet**	**F_{cold} (L/min)**	**Inlet**	**Middle**	**Outlet**

		T_1 (°C)	T_2 (°C)	T_3 (°C)		T_4 (°C)	T_5 (°C)	T_6 (°C)
1	0.2	65.0	65.9	67.5	0.98	46.6	39.5	30.7
2	0.2	65.1	65.9	67.5	0.96	46.8	39.5	30.7
3	0.1	65.1	65.9	57.6	0.93	46.4	39.5	30.7

Observations:

- Leaks can be observed in the pressure regulator and tubing connections.
- The measured flow rates of hot and cold water fluctuate ranging from 0.1-0.4 L/min for hot flow rates while 2.62-3.16 L/min, 1.99-2.09 L/min, 1.01 – 1.06 L/min for cold water flow rates with 0.2:3, 0.2:2, and 0.2:1 ratio, respectively.
- In hot fluid stream, temperature increases. from the outlet to the inlet, while in cold fluid stream, temperature decrease from inlet to outlet.

ANNEX 2: Processing of Data

Table A2-1. Equipment specifications and fluid properties

Heat Exchanger		
Inner Tube	**Diameter (D_1), *m***	0.0095
	Wall Thickness (W_1), *m*	0.0006
	Hydraulic Diameter (D_i), *m*	0.0083
	Cross-sectional Area (A_1), *m*^{2}	5.41E-05
	Heat Transfer Area (A_i), m^2	0.0172
Outer Tube	**Diameter (D_2), *m***	0.012
	Wall Thickness (W_2), *m*	0.003
	Hydraulic Diameter (D_o), *m*	0.0025
	Cross-sectional Area (A_2), *m*^{2}	4.22E-05
	Heat Transfer Area (A_o), m^2	0.0197
Steel Thermal Conductivity (λ), *W/m·K*		16
Total Length of Heat Transfer Section (L), *m*		0.66
Log Mean Heat Transfer Area (A_{lm}), m^2		0.0184

**Conductivity, Density and Viscosity data are taken from R. H. Perry and C. H. Chilton, Chemical Engineers' Handbook, 5th ed.New York: McGraw-Hill Book Company, 1973.*

Sample calculations:

Tube Cross-sectional Area (A_1 or A_2)

The tube cross-sectional area is determined by using the formula in getting the area of a circle.

$$A_1 = \frac{1}{4}\pi D_i^2 = \frac{1}{4}\pi(0.0083\ m)^2 = 5.41 \times 10^{-5}\ m^2 \quad \text{[A2-1.1]}$$

Heat Transfer Area (A_o or A_i)

The heat transfer area is equal to the lateral surface area of the tube.

$$A_o = \pi DL = \pi(0.0083\ m)(0.66\ m) = 0.0172\ m^2 \quad \text{[A2-1.2]}$$

Log Mean Heat Transfer Area (A_{lm})

The logarithmic mean area is the ratio between the difference of the outer and inner tube heat transfer areas and the difference of the natural logarithm of such heat transfer areas.

$$A_{lm} = \frac{A_o - A_i}{\ln\left(\frac{A_o}{A_i}\right)} = \frac{0.0197\ m^2 - 0.0172\ m^2}{\ln\left(\frac{0.0197\ m^2}{0.0172\ m^2}\right)} = 0.0184\ m^2 \quad \text{[A2-1.3]}$$

Table A2-2. Effect of Variation of Flowrate in different configurations at 45^{o}C.

Countercurrent flow									
Flowrate ratio		**F_{HOT}, L/min**	**In (T_3), °C**	**Mid (T_2), °C**	**Out (T_1), °C**	**F_{COLD}, L/min**	**Out (T_4), °C**	**Mid (T_5), °C**	**In (T_6), °C**
0.2:3	**1**	0.2	46.6	45.4	45.2	2.98	34	31.9	29.8
	2	0.2	46.5	45.3	45.1	2.92	34	31.8	29.8
	3	0.2	46.5	45.3	45.1	3.06	34	31.8	29.8
Mean		**0.20**	**46.53**	**45.33**	**45.13**	**2.99**	**34.00**	**31.83**	**29.80**
0.2:2	**1**	0.2	48.8	47.6	47.4	1.97	35.4	32.8	29.9
	2	0.2	48.8	47.7	47.4	1.87	35.3	32.8	29.9
	3	0.7	48.8	47.7	47.4	2.03	35.2	32.8	29.9
Mean		**0.37**	**48.80**	**47.67**	**47.40**	**1.96**	**35.30**	**32.80**	**29.90**
0.2:1	**1**	0.3	54.6	53.5	53.1	0.95	39.8	35.2	30.1
	2	0.1	54.7	53.6	53.2	1.03	40	35.2	30.1
	3	0.3	54.8	53.7	53.3	1.03	39.8	35.2	30.1
Mean		**0.23**	**54.70**	**53.60**	**53.20**	**1.00**	**39.87**	**35.20**	**30.10**
Cocurrent flow									
Flowrate ratio		**F_{HOT}, L/min**	**In (T_3), °C**	**Mid (T_2), °C**	**Out (T_1), °C**	**F_{COLD}, L/min**	**In (T_4), °C**	**Mid (T_5), °C**	**Out (T_6), °C**
0.2:3	**1**	0.4	46.7	45.5	45.1	3.16	30.3	32.3	34.6
	2	0.2	46.7	45.5	45.1	2.92	30.3	32.3	34.5
	3	0.2	46.8	45.6	45.2	2.62	30.4	32.3	34.3
Mean		**0.27**	**46.73**	**45.53**	**45.13**	**2.90**	**30.33**	**32.30**	**34.47**
0.2:2	**1**	0.2	48.8	47.7	47.4	2.09	30.4	33.3	36.3
	2	0.2	49	47.8	47.4	1.99	30.5	33.3	36.3
	3	0.2	49.1	47.9	47.6	1.99	30.5	33.3	36.4
Mean		**0.20**	**48.97**	**47.80**	**47.47**	**2.02**	**30.47**	**33.30**	**36.33**
0.2:1	**1**	0.1	53.1	51.9	51.7	1.06	30.4	35	38.6
	2	0.2	53.2	52.1	51.8	1.01	30.5	35.1	38.9
	3	0.2	53.3	52.1	51.9	1.01	30.4	35.1	38.9
Mean		**0.17**	**53.20**	**52.03**	**51.80**	**1.03**	**30.43**	**35.07**	**38.80**

Sample calculations:

Countercurrent operation, hot water flow rate and inlet at 0.2:3 flow ratio

Mean Temperature Reading (T_n)

The mean temperature reading is the average of the temperature readings in three runs.

$$T_3 = \frac{T_{3_1} + T_{3_2} + T_{3_3}}{3} = \frac{46.6°C + 46.5°C + 46.5°C}{3} = 46.53°C \quad [A2-2.1]$$

Mean Flow Rate Reading (F_{HOT} or F_{COLD})

The mean flow rate reading is the average of the flow rate readings in three runs.

$$F_{HOT} = \frac{F_{HOT_1} + F_{HOT_2} + F_{HOT_3}}{3} = \frac{0.2\ \frac{L}{min} + 0.2\ \frac{L}{min} + 0.2\ \frac{L}{min}}{3} = 0.2\ \frac{L}{min} \quad [A2\text{-}2.1]$$

Table A2-3. Effect of Variation of Inlet temperatures in different configurations at flow ratio of 0.2:1.

Temperature (T), °C		F_{HOT}, L/min	In (T_3), °C	Mid (T_2), °C	Out (T_1), °C	F_{COLD}, L/min	Out (T_4), °C	Mid (T_5), °C	In (T_6), °C
Countercurrent flow									
45	**1**	0.30	54.60	53.50	53.10	0.95	39.80	35.20	30.10
	2	0.10	54.70	53.60	53.20	1.03	40.00	35.20	30.10
	3	0.30	54.80	53.70	53.30	1.03	39.80	35.20	30.10
Mean		**0.23**	**54.70**	**53.60**	**53.20**	**1.00**	**39.87**	**35.20**	**30.10**
55	**1**	0.30	56.20	55.10	54.80	1.04	41.00	36.20	30.30
	2	0.30	56.20	55.00	54.90	0.96	41.00	36.20	30.30
	3	0.30	56.10	55.00	54.80	0.99	41.00	36.20	30.30
Mean		**0.30**	**56.17**	**55.03**	**54.83**	**1.00**	**41.00**	**36.20**	**30.30**
65	**1**	0.20	65.10	64.00	63.70	1.03	44.50	40.60	30.30
	2	0.20	65.10	64.00	63.70	1.04	45.10	40.60	30.30
	3	0.20	64.90	63.90	63.50	0.99	45.70	40.60	30.30
Mean		**0.20**	**65.03**	**63.97**	**63.63**	**1.02**	**45.10**	**40.60**	**30.30**
Cocurrent flow									
Temperature (T), °C		**F_{HOT}, L/min**	**In (T_3), °C**	**Mid (T_2), °C**	**Out (T_1), °C**	**F_{COLD}, L/min**	**In (T_4), °C**	**Mid (T_5), °C**	**Out (T_6), °C**
45	**1**	0.10	53.10	51.90	51.70	1.06	30.40	35.00	38.60
	2	0.20	53.20	52.10	51.80	1.01	30.50	35.10	38.90
	3	0.20	53.30	52.10	51.90	1.01	30.40	35.10	38.90
Mean		**0.17**	**53.20**	**52.03**	**51.80**	**1.03**	**30.43**	**35.07**	**38.80**
55	**1**	0.20	55.30	54.10	54.00	0.95	30.80	36.00	40.20
	2	0.20	55.30	54.20	53.90	1.07	30.60	36.00	40.30
	3	0.20	55.30	54.20	53.90	1.01	30.60	35.90	40.20
Mean		**0.20**	**55.30**	**54.17**	**53.93**	**1.01**	**30.67**	**35.97**	**40.23**
65	**1**	0.20	67.50	65.90	65.00	0.98	30.70	39.50	46.60
	2	0.20	67.50	65.90	65.10	0.96	30.70	39.50	46.80
	3	0.10	67.60	65.90	65.10	0.93	30.70	39.50	46.40
Mean		**0.17**	**67.53**	**65.90**	**65.07**	**0.96**	**30.70**	**39.50**	**46.60**

Sample calculations:

Countercurrent operation, hot water flow rate and inlet at 45°C.

Mean Temperature Reading (T_n)

The equation is used in determining the mean temperature reading. Refer to equations A2-2.1.

Mean Flow Rate Reading (F_{HOT} or F_{COLD})

The equation is used in determining the mean flow rate reading. Refer to equations A2-2.2.

Table A2-3. Temperature differences of the double-pipe heat exchanger with varying flow rates and inlet temperatures.

Countercurrent flow					
Flow rate ratio (F_{HOT}:F_{COLD}), L/min	**ΔT_{min}, °C**	**ΔT_{max}, °C**	**Inlet Temperature (T_{in}), °C**	**ΔT_{min}, °C**	**ΔT_{max}, °C**
0.2:3	12.53	15.33	**45**	14.83	23.10
0.2:2	13.50	17.50	**55**	15.17	24.53
0.2:1	14.83	23.10	**60**	19.93	33.33
Cocurrent flow					
Flow rate ratio (F_{HOT}:F_{COLD}), L/min	**ΔT_{max}, °C**	**ΔT_{min}, °C**	**Inlet Temperature (T_{in}), °C**	**ΔT_{max}, °C**	**ΔT_{min}, °C**
0.2:3	16.40	10.67	**45**	22.77	13.00
0.2:2	18.50	11.13	**55**	24.63	13.70
0.2:1	22.77	13.00	**60**	36.83	18.47

Sample calculations:

Flow ratio of 0.2:3

Maximum Mean Temperature Difference in Countercurrent Operation (ΔT_{max})

The maximum mean temperature difference in countercurrent operation is the difference between the temperature in hot temperature outlet and cold temperature inlet.

$$\Delta T_{max} = T_1 - T_6 = 45.13°C - 28.80°C = 15.33°C \quad [A2\text{-}3.1]$$

Maximum Mean Temperature Difference in Cocurrent Operation (ΔT_{max})

The maximum mean temperature difference in countercurrent operation is the difference between the temperature in hot temperature inlet and cold temperature inlet.

$$\Delta T_{max} = T_1 - T_4 = 46.73°C - 30.33°C = 16.40°C \quad [A2\text{-}3.2]$$

Minimum Mean Temperature Difference in Countercurrent Operation (ΔT_{min})

The minimum mean temperature difference in countercurrent operation is the difference between the temperature in hot temperature inlet and cold temperature outlet.

$$\Delta T_{min} = T_3 - T_4 = 46.53°C - 34.00°C = 12.53°C \quad [A2\text{-}3.3]$$

Minimum Mean Temperature Difference in Cocurrent Operation (ΔT_{min})

The minimum mean temperature difference in countercurrent operation is the difference between the temperature in hot temperature outlet and cold temperature outlet.

$$\Delta T_{min} = T_3 - T_6 = 45.13°C - 34.47°C = 10.67°C \quad [A2\text{-}3.4]$$

Table A2-4. Experimental Overall Heat Transfer Coefficient and Heat Exchanger Unit Effectiveness

$T_{setting}$ (°C)	Flow Type	Flow ratio (F_{HOT}:F_{COLD})	F_{HOT} (m³/s)	F_{COLD} (m³/s)	ΔT_{lm} (K)	$T_{HOT,ave}$ (°C)	$T_{COLD,ave}$ (°C)	k_H (factor)	k_C (factor)	ρ_{HOT} (kg/m³)	ρ_{COLD} (kg/m³)	m_{HOT} (kg/s)	m_{COLD} (kg/s)	Cp_{HOT} (J/kg-K)	Cp_{COLD} (J/kg-K)	C_{HOT} (W/K)	C_{COLD} (W/K)	T_{hi}-T_{ci} (K)	φ_{Hmax} (W)	φ_H (W)	ε	U (W/m K)
45.00	**Countercurrent**	0.2:3	0.0000038	0.0000498	13.92	45.87	31.90	1.42	0.23	989.89	995.09	0.14	0.05	4184.78	4182.92	592.64	207.43	16.80	3484.75	871.19	**0.25**	**3638.37**
		0.2:2	0.0000038	0.0000327	15.41	48.10	32.60	4.26	0.35	988.91	994.86	0.13	0.03	4185.08	4183.01	524.35	135.94	18.90	2569.32	734.09	**0.29**	**2768.81**
		0.2:1	0.0000038	0.0000167	18.66	53.95	34.99	-3.53	0.99	986.21	994.07	0.11	0.02	4185.86	4183.33	451.43	69.31	24.60	1704.99	677.15	**0.40**	**2109.63**
	Cocurrent	0.2:3	0.0000045	0.0000483	13.32	45.93	32.40	1.46	0.32	989.85	994.93	0.12	0.05	4184.79	4182.99	520.48	201.15	16.40	3298.91	832.77	**0.25**	**3633.47**
		0.2:2	0.0000033	0.0000337	14.51	48.22	33.40	4.62	0.52	988.85	994.60	0.13	0.03	4185.10	4183.12	547.21	140.07	18.50	2591.32	820.82	**0.32**	**3288.67**
		0.2:1	0.0000028	0.0000172	17.43	52.50	34.62	-5.00	0.86	986.90	994.19	0.10	0.02	4185.67	4183.28	426.84	71.40	22.77	1625.68	597.58	**0.37**	**1993.07**
55.00	**Countercurrent**	0.2:1	0.0000050	0.0000167	19.48	55.50	35.65	1.22	1.30	985.45	993.84	0.13	0.02	4186.07	4183.42	553.32	69.29	25.87	1792.64	741.45	**0.41**	**2213.18**
	Cocurrent	0.2:1	0.0000033	0.0000168	18.62	54.60	35.45	0.85	1.20	985.88	993.91	0.11	0.02	4185.95	4183.39	477.94	69.99	24.63	1723.89	669.12	**0.39**	**2089.68**
65.00	**Countercurrent**	0.2:1	0.0000033	0.0000170	26.06	64.33	37.70	0.76	3.35	980.90	993.11	0.18	0.02	4187.24	4183.69	746.69	70.63	34.73	2453.07	1045.36	**0.43**	**2332.22**
	Cocurrent	0.2:1	0.0000033	0.0000160	26.61	66.30	38.65	1.70	6.41	979.82	992.76	0.11	0.02	4187.51	4183.82	440.27	66.46	36.80	2445.60	1056.66	**0.43**	**2308.59**

**Some data are in reference to tables A2-1, A2-2, and A2-3.*

Sample Calculations:

At constant temperature 45°C with an expected flow ratio of 0.5:0.5; countercurrent

Mass Flow Rate of Cold Liquid ($\dot{m}_{COLD}$)

The mass flow rate of liquid can be calculated by multiplying the volumetric flow rate to its density. Additional conversion factors such as cubic meters to cubic centimeters and liters to milliliters are also used.

$$\dot{m_{COLD}} = F_{HOT}\rho_{HOT} = \left(3\frac{L}{min}\right)\left(1\frac{cm^3}{mL}\right)\left(\frac{1000\ mL}{1\ L}\right)\left(\frac{1\ min}{60\ sec}\right)\left(\frac{1m^3}{10^6 cm^3}\right)\left(989.89\frac{kg}{m^3}\right) = 0.0495\frac{kg}{s} \quad \text{[A2-4.1]}$$

Mass Flow Rate of Hot Liquid ($\dot{m}_{HOT}$)

The amount of heat absorbed by the cold fluid is equal to the amount of heat given off by the hot fluid or represented by $Q_{HOT} = Q_{COLD}$; hence, from the heat equation, the mass flow rate of the hot fluid can be determined.

$$\dot{m}_{HOT} = \frac{\dot{m}_{COLD}Cp_{COLD}(T_3 - T_4)}{\frac{T_1 - T_6}{Cp_{HOT}}} = \frac{0.0495\frac{kg}{s} \times 4182.92\ \frac{J}{kg \cdot K}(307 - 302.8)K}{\frac{(319.6 - 318.1)K}{4184.78\frac{J}{kg \cdot K}}} = 0.14162\frac{kg}{s} \quad \text{[A2-4.2]}$$

Thermal Capacity Rate (C_{HOT})

The thermal capacity rate is the quantity of heat a flowing fluid of a certain mass flow rate can absorb or release per unit temperature change per unit time.

$$C_{HOT} = m_{HOT}Cp_{HOT} = \left(0.416\frac{kg}{s}\right)\left(4184.78\frac{J}{kg \cdot K}\right) = 592.63\frac{J}{K \cdot s} \quad \text{[A2-4.3]}$$

Log Mean Temperature Different (ΔT_{lm})

The logarithmic mean temperature difference is the ratio between the difference of the maximum temperature difference and the minimum temperature difference and the difference of the natural logarithm of such temperature differences.

$$\Delta T_{lm} = \frac{\Delta T_{max} - \Delta T_{min}}{\ln\left(\frac{\Delta T_{max}}{\Delta T_{min}}\right)} = \frac{15.3K - 12.6K}{\ln\left(\frac{15.3K}{12.6K}\right)} = 13.9K \quad \text{[A2-4.4]}$$

Heat Flow Rate (ΦH)

The rate of heat flow is proportional to the thermal capacity rate and to the temperature difference between the ends of the pipe.

$$\Phi H_{max} = C_{COLD}(T_1 - T_4) = \left(207.43\frac{W}{K}\right)(319.75\ K - 302.95\ K) = 3484.75\ W \quad \text{[A2-4.5]}$$

Experimental Heat Exchanger Effectiveness

The experimental heat exchanger effectiveness is the ratio of the actual heat transfer to the maximum possible heat transfer.

$$\varepsilon = \frac{\emptyset_{Hmax}}{\emptyset_H} = \frac{3484.75\ W}{871.19\ W} = 0.250 \qquad [A2\text{-}4.6]$$

Experimental Overall Heat Transfer Coefficient

The overall heat transfer coefficient is determined experimentally from the measured overall temperature difference and heat transfer rate.

$$U_{exp} = \frac{\emptyset_H}{A\Delta T_{LM}} = \frac{871.19W}{0.01720m^2 \times 13.9K} = 3638.37\frac{W}{m^2 \cdot K} \qquad [A2\text{-}4.7]$$

Table A2-5. Theoretical Overall Heat Transfer Coefficient and Heat Exchanger Unit Effectiveness

$T_{setting}$ (°C)	Flow Type	Flow ratio (F_{HOT}:F_{COLD})	μ_{HOT} (kg/m.s)	μ_{COLD} (kg/m.s)	K_{HOT} (W/m.K)	K_{COLD} (W/m.K)	$N_{PR,HOT}$	$N_{PR,COLD}$	v_{HOT}	v_{COLD}	$N_{Re,HOT}$	$N_{Re,COLD}$	Nu_{HOT}	Nu_{COLD}	h_i (W/m.K)	h_o (W/m.K)	U_{theo}	c_{min}	c_{max}	NTU	ε_{theo}
45	Countercurrent	0.2:3	0.000588	0.000766	0.63	0.61	3.91	5.23	0.06	1.18	32263.23	3846.0822	160.51	32.90	12180.94	8062.31	3796.06	207.43	592.64	0.31	0.26
		0.2:2	0.000566	0.000755	0.63	0.61	3.74	5.15	0.06	0.79	29694.13	2600.4613	147.59	23.90	11250.00	5866.33	3217.61	135.94	524.35	0.41	0.32
		0.2:1	0.000513	0.000720	0.64	0.62	3.35	4.88	0.06	0.39	28197.68	1362.9038	135.54	13.96	10449.89	3442.26	2367.01	69.31	451.43	0.59	0.43
	Cocurrent	0.2:3	0.000587	0.000758	0.63	0.61	3.90	5.17	0.06	1.18	28383.00	3885.0189	144.76	33.01	10987.96	8099.28	3678.51	201.15	520.48	0.31	0.26
		0.2:2	0.000564	0.000743	0.63	0.61	3.73	5.06	0.06	0.79	31054.55	2642.1143	152.83	24.04	11652.22	5908.87	3261.06	140.07	547.21	0.40	0.32
		0.2:1	0.000525	0.000725	0.64	0.62	3.44	4.92	0.06	0.39	26033.40	1353.1066	128.52	13.92	9880.55	3430.97	2331.96	71.40	426.84	0.56	0.42
55	Countercurrent	0.2:1	0.000500	0.000711	0.64	0.62	3.26	4.82	0.06	0.39	35438.19	1380.4214	160.92	14.02	12443.60	3462.34	2465.06	69.29	553.32	0.61	0.45
	Cocurrent	0.2:1	0.000507	0.000713	0.64	0.62	3.31	4.84	0.06	0.39	30182.76	1375.1003	142.43	14.00	10994.61	3456.26	2399.77	69.99	477.94	0.59	0.43
65	Countercurrent	0.2:1	0.000436	0.000683	0.65	0.62	2.80	4.61	0.06	0.39	54791.28	1435.1492	214.51	14.21	16869.09	3524.30	2630.53	70.63	746.69	0.64	0.46
	Cocurrent	0.2:1	0.000424	0.000671	0.66	0.62	2.71	4.52	0.06	0.39	33242.17	1460.9534	141.98	14.30	11206.87	3553.08	2450.41	66.46	440.27	0.63	0.46

**Some data are in reference to tables A2-1, A2-2, and A2-3, A2-4.*

Sample calculations:

At constant temperature 45°C with an expected flow ratio of 0.5:0.5; countercurrent

Reynold's number (N_{Re})

The Reynold's number can be obtained by employing the equation,

$$N_{Re_{HOT}} = \frac{4\emptyset m_H}{\pi D_1 \mu_H} = \frac{4\left(0.14162\,\frac{kg}{s}\right)}{(3.1416)\ (0.0095m)\left(5.88\times10^4\,\frac{kg}{m\cdot s}\right)} = 32263.23$$ [A2-5.1]

Prandtl's number (Pr)

The Prandtl's number can be obtained by employing the equation,

$$Pr_{HOT} = \frac{Cp_{HOT}\mu_{HOT}}{K_{HOT}} = \frac{4184.78\ \frac{J}{kg.K}\times 0.00058830\,\frac{kg}{m\cdot s}}{0.62988356\,\frac{W}{K}} = 3.908509836$$ [A2-5.2]

Nusselt's Number (Nu)

The Nusselt's number is the ratio of convective to conductive heat transfer at a boundary in a fluid.

$$Nu_{HOT} = 0.023\left({N_{Re,HOT}}^{0.8}{N_{Pr,HOT}}^{0.4}\right) = 0.023(32263.23^{0.8}3.90851^{0.4}) = 160.51$$ [A2-5.3]

Heat transfer coefficient (h)

The heat transfer coefficient is the proportionality constant between the heat flux and the thermodynamic driving force for the flow of heat. It is also the ratio between the Nusselt's number multiplied by the convective coeffiecient, and the diameter of the pipe.

$$h_i = \frac{Nu_{HOT}K_{HOT}}{D_i} = \frac{160.5087\left(0.61270508\,\frac{W}{K}\right)}{0.0083\ m} = 12180.94\,\frac{W}{m.K}$$ [A2-5.4]

Maximum Thermal Capacity Rate (C_{max})

The thermal capacity rate is the quantity of heat a flowing fluid of a certain mass flow rate can absorb or release per unit temperature change per unit time.

$$C_{max} = m_{HOT}Cp_{HOT} = \left(0.416\,\frac{kg}{s}\right)\left(4184.78\,\frac{J}{kg\cdot K}\right) = 592.63\,\frac{J}{K\cdot s}$$ [A2-5.5]

Minimum Thermal Capacity Rate (C_{min})

The same equation will be used with A2-4.3 in finding the minimum thermal capacity rate; however, the mass flows and the heat capacity of cold fluid is used instead.

$$C_{min} = m_{COLD}Cp_{COLD} = \left(0.0496\frac{kg}{s}\right)\left(4182.92\frac{J}{kg.K}\right) = 207.43\frac{W}{K} \quad \text{[A2-5.6]}$$

Number of Transfer Units

The NTU can be taken as a measure of the amount of heat that can be transferred by the heat exchanger. It is also the ratio between the heat transferred for a unit temperature driving force and the heat absorbed by the fluid stream that results to a temperature change of the same degree.

$$NTU = \frac{U_{theo}A_i}{C_{min}} = \frac{3796.063765\frac{W}{m^2.K}(0.0185\ m^2)}{207.43\frac{W}{K}} = 0.31495 \quad \text{[A2-5.7]}$$

Theoretical Heat Exchanger Effectiveness

The theoretical heat exchanger effectiveness in terms of heat capacities depends upon the flow arrangement. The effectiveness for countercurrent and cocurrent single pass heat exchangers are given using the NTU method, respectively.

$$\varepsilon = \frac{1-\exp\left[-NTU\left(1-\frac{C_{min}}{C_{max}}\right)\right]}{\left(1-\frac{C_{min}}{C_{max}}\right)\exp\left[-NTU\left(1-\frac{C_{min}}{C_{max}}\right)\right]} = \frac{1-\exp\left[-0.31495\left(1-\frac{207.43}{592.64}\right)\right]}{1-\left(\frac{207.43}{592.64}\right)\exp\left[-0.2845\left(1-\frac{207.43}{592.64}\right)\right]} = 0.25 \quad \text{[A2-5.8]}$$

$$\varepsilon = \frac{1-\exp\left[-NTU\left(1+\frac{C_{min}}{C_{max}}\right)\right]}{\left(1+\frac{C_{min}}{C_{max}}\right)} = \frac{1-\exp\left[-0.31472\left(1+\frac{201.15}{520.48}\right)\right]}{1+\left(\frac{201.15}{520.48}\right)} = 0.252 \quad \text{[A2-5.9]}$$

Theoretical Overall Heat Transfer Coefficient

The theoretical heat transfer is the sum of resistances due to the films of fluid on either side of the inner tube wall, the resistance of the tube wall itself, and, in certain cases, the resistance due to the deposited scales on the tube walls (fouled tubes).

$$U = \frac{1}{\frac{1}{h_i}+\frac{r_0-r_i}{kA_{lm}}+\frac{A_i}{h_oA_o}} = \frac{1}{\frac{1}{12180.94}+7.29688\times10^{-5}+\frac{0.0185\ m}{8062.31(0.0197m)}} = 3796.06\frac{W}{m^2.K} \quad \text{[A2-5.10]}$$

Raw Data Sheets

Table 1. *Countercurrent operation:* variation of flow rate

COUNTERCURRENT OPERATION

Temperature Controller Setting: 45 °C

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 3

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Outlet T_4 (°C)	Middle T_5 (°C)	Inlet T_6 (°C)
1	0.2	45.2	45.4	46.6	2.98	34.0	31.9	29.8
2	0.3	45.1	45.3	46.5	2.92	34.0	31.8	29.8
3	0.2	45.1	45.3	46.5	3.06	34.0	31.8	29.8

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 2

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Outlet T_4 (°C)	Middle T_5 (°C)	Inlet T_6 (°C)
1	0.2	47.4	47.6	48.8	1.97	35.4	32.8	29.9
2	0.2	47.4	47.7	48.8	1.87	35.3	32.8	29.9
3	0.7	47.4	47.7	48.8	2.03	35.2	32.8	29.9

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Outlet T_4 (°C)	Middle T_5 (°C)	Inlet T_6 (°C)
1	0.3	53.1	53.5	54.6	0.95	39.8	35.2	30.1
2	1.0	53.2	53.6	54.7	1.03	40.0	35.2	30.1
3	0.3	53.3	53.7	54.8	1.03	39.8	35.2	30.1

Observations:

The cold-water flow rate did not reach the constant desired flow rate. Instead, it fluctuated around the desired flow rate.
The hot water flow rate on the software is not adjustable. The flow rate fluctuated around the value of 0.2 L/min.

Table 2. *Countercurrent operation:* variation of inlet temperature.

COUNTERCURRENT OPERATION

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1

Temperature Controller Setting: 55 °C

No.	Hot Fluid				Cold Fluid			
		Outlet	Middle	Inlet		Outlet	Middle	Inlet
	F_{hot} (L/min)	T_1 (°C)	T_2 (°C)	T_3 (°C)	F_{cold} (L/min)	T_4 (°C)	T_5 (°C)	T_6 (°C)
1	0.3	54.8	55.1	56.2	1.04	41.0	36.2	30.3
2	0.3	54.9	55.0	56.2	0.96	41.0	36.2	30.3
3	0.3	54.8	55.0	56.1	0.99	41.0	36.2	30.3

Temperature Controller Setting: 65 °C

No.	Hot Fluid				Cold Fluid			
		Outlet	Middle	Inlet		Outlet	Middle	Inlet
	F_{hot} (L/min)	T_1 (°C)	T_2 (°C)	T_3 (°C)	F_{cold} (L/min)	T_4 (°C)	T_5 (°C)	T_6 (°C)
1	0.2	63.7	64.0	65.1	1.03	44.5	40.6	30.3
2	0.2	63.7	64.0	65.1	1.09	45.1	40.6	30.3
3	0.2	63.5	63.9	64.9	0.99	45.7	40.6	30.3

Observations:

The set temperature setting of 55 °C and 65 °C were not achieved. Instead, the temperature of the hot fluid only ranged from 53.7 - 56.2 °C
In the hot fluid stream, temperature increases from the outlet to the inlet, while in cold fluid stream, temperature decreases from inlet to outlet.

Table 3. *Co-current operation:* variation of flow rate.

CO-CURRENT OPERATION

Temperature Controller Setting: 45 °C

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 3

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Inlet T_4 (°C)	Middle T_5 (°C)	Outlet T_6 (°C)
1	0.4	45.1	45.5	46.7	3.16	30.3	32.3	34.6
2	0.2	45.1	45.5	46.7	2.92	30.3	32.3	34.5
3	0.2	45.2	45.6	46.8	2.62	30.4	32.3	34.3

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 2

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Inlet T_4 (°C)	Middle T_5 (°C)	Outlet T_6 (°C)
1	0.2	47.4	47.7	48.8	2.09	30.4	33.3	36.3
2	0.2	47.4	47.8	49.0	1.99	30.5	33.3	36.3
3	0.2	47.6	47.9	49.1	1.99	30.5	33.3	36.4

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Inlet T_4 (°C)	Middle T_5 (°C)	Outlet T_6 (°C)
1	0.1	51.7	51.9	53.1	1.06	30.4	35.0	38.6
2	0.2	51.8	52.1	53.2	1.01	30.5	35.1	38.9
3	0.2	51.9	52.1	53.3	1.01	30.4	35.1	38.9

Observations:

the cold water flow rate did not reach the constant desired flowrate. Instead, it fluctuated around the desired flow rate.

The hot water flow rate on the software is not adjustable. The flowrate fluctuated around the value of 0.2 L/min.

Table 4. *Co-current operation:* variation of inlet temperature.

CO-CURRENT OPERATION

Flow Rate Ratio (F_{hot}:F_{cold}): 0.2 : 1

Temperature Controller Setting: 55 °C

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Inlet T_4 (°C)	Middle T_5 (°C)	Outlet T_6 (°C)
1	0.2	54	54.1	55.3	0.95	30.8	36.0	40.2
2	0.2	53.9	54.2	55.3	1.07	30.6	36.0	40.3
3	0.2	53.9	54.2	55.3	1.01	30.6	35.9	40.2

Temperature Controller Setting: 65 °C

No.	Hot Fluid				Cold Fluid			
	F_{hot} (L/min)	Outlet T_1 (°C)	Middle T_2 (°C)	Inlet T_3 (°C)	F_{cold} (L/min)	Inlet T_4 (°C)	Middle T_5 (°C)	Outlet T_6 (°C)
1	0.2	65.0	65.9	67.5	0.98	30.7	39.5	46.6
2	0.2	65.1	65.9	67.5	0.96	30.7	39.5	46.8
3	0.1	65.1	65.9	67.6	0.93	30.7	39.5	46.9

Observations:

Leaks can be observed in the pressure regulator and tubing connections. In hot fluid stream, temperature increases from the outlet to the inlet, while in cold fluid stream, temperature increases from inlet to outlet.