Variable flow rates of tube and plate heat exchanges in parallel and counterflow

Robert Hawken

School of Engineering, University of the Sunshine Coast.

1 Introduction

Heat exchangers are used in order to take advantage of the temperature gradient between hot and cold fluids without direct contact. Two common types of heat exchangers are the concentric tube (one pipe running parallel through a larger pipe) and the plate style (multiple layers of stacked plates). Both heat exchangers are fitted for either parallel or counter flow configurations (Cengel and Ghajar 2015).

A parallel heat exchanger is characterised by hot and cold fluids entering at the same orientation. Over a longer distance, the heat transfer rate in a parallel heat exchanger is most likely to become inefficient, unreliable and unpredictable in practice (Wetley, Rorrer & Foster 2020). The log mean temperature difference (L.M.T.D) is used to estimate the average temperature difference between fluids due to the decay style of a parallel system. A counterflowing arrangement involves cold fluid entering the system at the origin of the hot fluid. This ensures a predictable and somewhat linear relationship between temperature gradients as the effective length of heat exchange encompasses the full length of the system rather than the initial inlet (Kakac, Hongtan & Pramuanjaroenkij 2020).

It is primarily hypothesised that a counter flowing heat exchanger will have a higher L.M.T.D and overall heat transfer coefficient (U) over varying flow rates than a heat exchanger in parallel. Secondly, it is hypothesized that a plate heat exchanger will have a higher L.M.T.D and U value than a concentric tube heat exchanger.

1.1 Governing Equations

The coefficient of energy balance (C.E.B) is created by calculating the heat transfer absorbed by the system over the energy emitted by the system.

$$CEB = \frac{\dot{Q}_a}{\dot{Q}_e} \tag{1}$$

L.M.T.D is the natural logarithmic function with respect to the different water temperature recorded. For Parallel flow (Equation 2) and for Counter flow (Equation 3).

$$L.M.T.D = \frac{(TH_2 - TC_2) - (TH_1 - TC_1)}{\ln\left(\frac{TH_2 - TC_2}{TH_1 - TC_1}\right)} \tag{2}$$

$$L.M.T.D = \frac{(TH_1 - TC_2) - (TH_2 - TC_1)}{\ln\left(\frac{TH_1 - TC_2}{TH_2 - TC_1}\right)} \tag{3}$$

Overall heat transfer coefficient values were calculated by the heat transfer rate emitted over the mean heat transfer area by the L.M.T.D. (Appendix 1).

$$U = \frac{\dot{Q}_e}{A \cdot L.M.T.D} \tag{4}$$

2 Experimental Design

2.1 Concentric Tube heat exchanger

In the parallel flow configuration, a TD360a heat exchanger with 60° hot water flow of 3 L/min was applied with variable cold water intake of 3 L/min, 2 L/min, 1 L/min and 0.5 L/min (Figure 1). Intake, central and exit temperatures were recorded for each flow rate after five minutes of equilibrium time.
This experimental design was repeated with the cold water intake reversed into a counter flow arrangement (Figure 1).

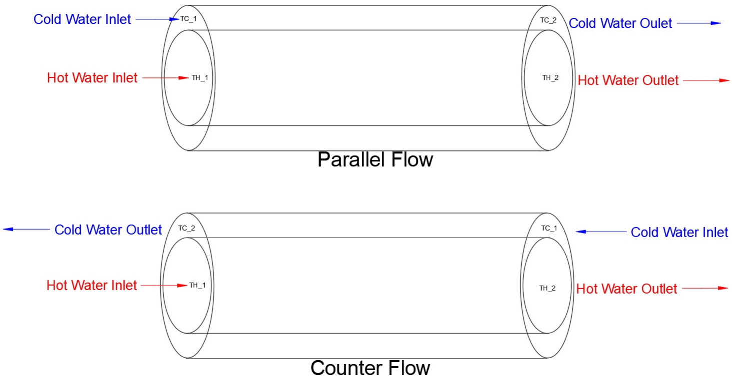


Figure. 1: Water flow directions for Parallel and Counterflow Concentric Tube Arrangements.

2.2 Plate heat exchanger

After each flow rate was tested, a TD360b plate heat exchanger was installed and both parallel and counterflow variable cold flow rates were repeated with 3 minute equilibrium time (Figure 2).

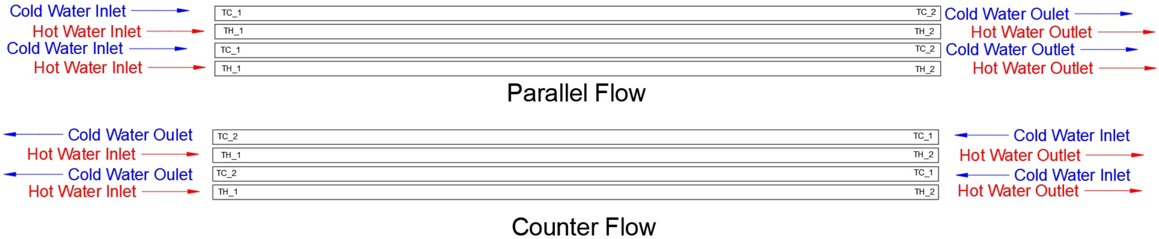


Figure. 2: Water flow directions for Parallel and Counterflow Plate Arrangements

2.3 Numerical Analysis

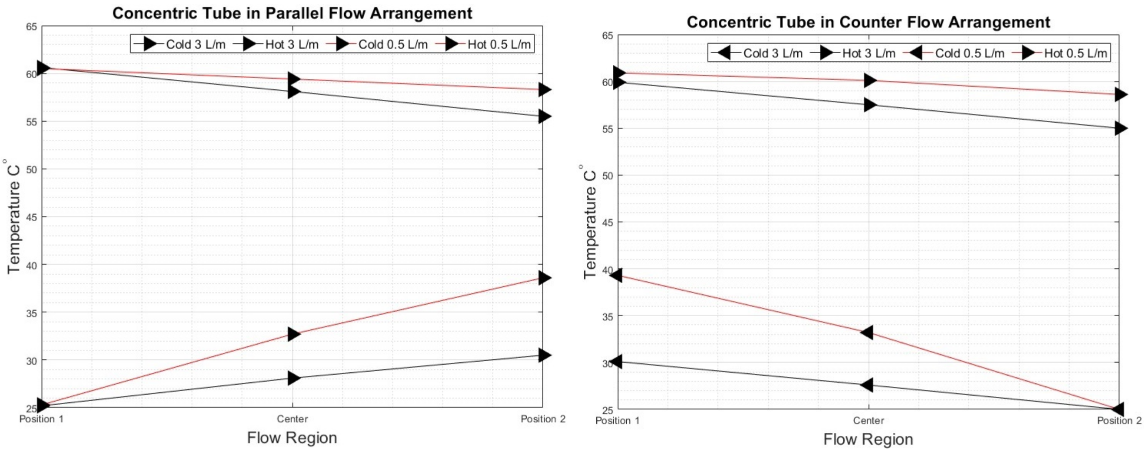
Graphical illustrations were completed using MATLAB 2022Ra edition and line diagrams with AutoCAD 2023. TD360a/b Bench top heat exchanger by TecQquipment (2018) included the relevant formulae used for the Governing equations in section 1.1. Density, specific heat capacity, heat transfer rate and temperature efficiency calculations were completed via Microsoft Excel 365 (Appendix 1).

3 Results and Discussion

3.1 Concentric Tube In Parallel and Counter Flow Arrangement

Temperature recordings for concentric tube in parallel flow arrangement is displayed across all four cold flow rates at Position 1 (inlet) and Position 2 (outlet)(Figure 3a). The 0.5 L/min flow rates resulted in the largest temperature increase of cold water reaching $\approx 40^\circ$ at both outlet instances (Position 2). This suggests the slower flow rate acts more like a condenser with a larger change in cold water temperature than hot water (Wetley et.al. 2020).

Conversely the temperature difference of the 3 L/min flow rates remained steady, suggesting the increased flow rate, reduced heat transfer (Figure 3a). Figure 3b displays concentric tube in the counter flow arrangement. In support of the trends outlined above, 0.5 L/min flow rate experienced the highest temperature difference in counter flow arrangement as seen in (Naphon 2006). However there was no significant temperature difference between flow directions at the same flow rate (Figure 3b). Parallel flows can cause a higher thermal stress in materials due to the large temperature difference at the inlet (Position 1) (Thulukkanam, 2013).



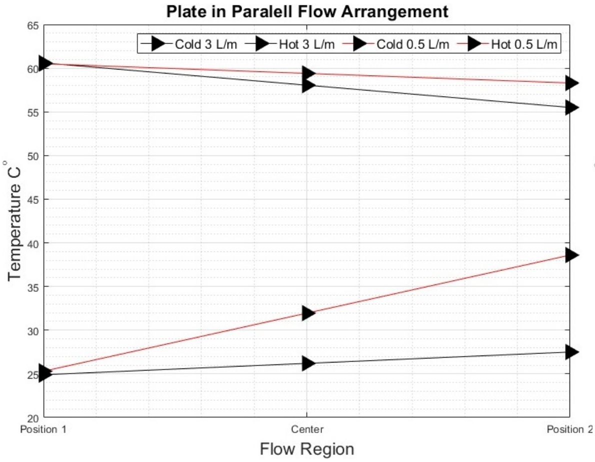
(a) *Concentric Tube Parallel Flow* (b) *Concentric Tube in Counter Flow*

Figure. 3: Inlet and outlet temperature of concentric tube heat exchanger in parallel and counter flow arrangements.

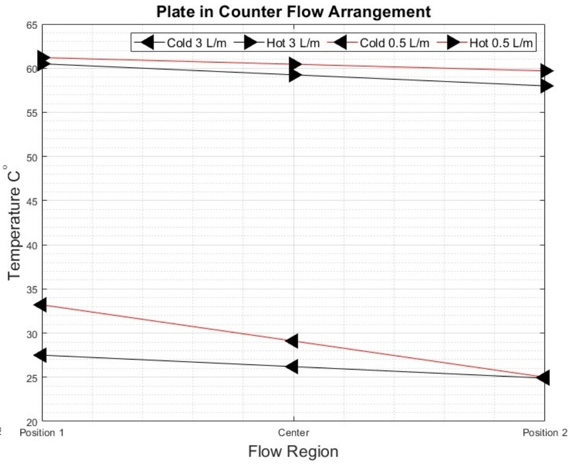
3.2 Heat Plate in Parallel and Counter Flow Arrangement

The inlet, average and outlet temperature recordings for the heat plate in the parallel and counter flow arrangement are displayed across all four cold flow rates in Figure 4 (a&b). In line with section 3.1, it is seen that the 0.5 L/min flow rates resulted in the largest temperature increase of cold water ($\approx 13^\circ$ parallel & $\approx 8^\circ$ counterflow) compared to the 3 L/min flow rate ($\approx 2.5^\circ$ parallel & counterflow).

As stated, a greater cold and hot water temperature change occurred in parallel flow rather than the counter flow arrangement (Figure 4b). This is contrary to the stated hypothesis and current literature. Thulukkanam (2013) states that a counterflow system is the most efficient single pass system at the same flow rates and temperatures. Due to the nature of parallel and counter flow inlet temperature difference, it is more appropriate to use L.M.T.D (as both inlet and outlet temperatures are known) and U values outlined in Table 1 (Thulukkanam 2013).



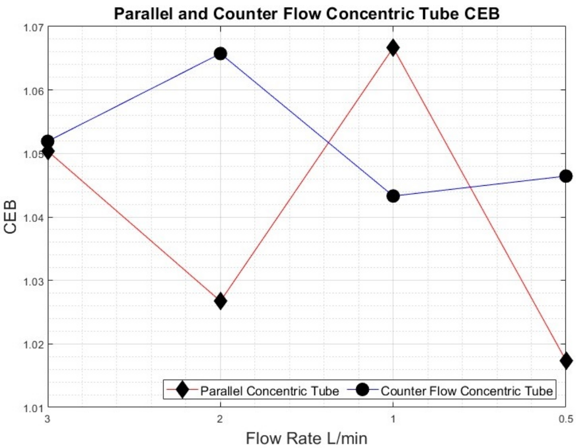
(a) *Plate in Parallel Flow*



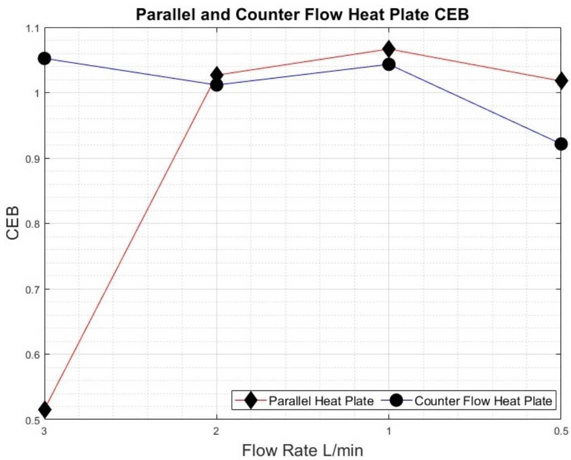
(b) *Plate in Counter Flow*

Figure. 4: Inlet and outlet temperature of plate heat exchanger in parallel and counter flow arrangements.

The coefficient of energy balance (C.E.B) for the heat plate in the parallel and counter flow arrangement is displayed across all four cold flow rates (Figure 5a & b). It is seen that the parallel direction in both heat exchangers has a higher variance in coefficients $\approx 0.5 - 1.05$. This indicates an unsteady relationship between energy absorbed and emitted. In both heat exchangers, the counter flow direction shows a steady heat exchange between heat emitted and absorbed ≈ 1.05. Further calculations of L.M.T.D and U are needed for final conclusions (Thulukkanam, 2013).



(a) *Concentric Tube CEB for Parallel Counterflow*



(b) *Heat Plate CEB for Parallel Counterflow*

Figure. 5: Coefficient of energy balance with change in flow rates between Concentric Tube and Heat Flow arrangement

3.3 Analysis of L.M.T.D and Overall Heat transfer coefficient

Table 1 refers to the L.M.T.D values for the concentric tube and plate heat exchanger across both flow directions at 3 L/min and 0.5 L/min. It is evident that there is no significant difference ($\approx$ 1%) between the L.M.T.D values in the concentric tube between Parallel and Counter flow operations at either 3 L min or 0.5 L min.

However in the plate system, there is a significant temperature difference in regards to 3 L/min ($\approx$ 4%) and counter flow ($\approx$ 17%), both in favour of counter flow. This indicates the highest L.M.T.D value is the counter flow plate at 3 L/min (33.05) and the lowest in parallel plate (26.7°. This is supportive of the current implementation of heat exchangers in applications (Cengel and Ghajar 2015).

As heat exchangers are considered steady state devices, it is more suitable to use a counterflow system, due to the stable L.T.M.D profiles for engineering design (Kakac et. al. 2020)
Table 1 also indicates that the highest L.M.T.D occurred at a flow rate of 3 L/min as more fluid is available for heat transfer than at 0.5 L/min (Cengel and Ghajar 2015).

Table 1: L.M.T.D (°C) Tube and Plate Heat Exchanger with Variable Flow rates

Cold Flow	Parallel Tube	Counter Tube	% Δ	Parallel Plate	Counter Plate	% Δ
3 (L/min)	29.89	29.90	-0.002	31.69	33.05	4.277
0.5 (L/min)	26.70	27.16	1.70	26.70	31.23	16.95

At 3 L/min, parallel tube was more efficient $\approx$ 4% and counter flow was $\approx$ 3% more efficient at 0.5 L/min, however both are not significantly different (Table 2). There was a 112% and 70% difference between efficiency in favor of the parallel configuration over the counterflow in a plate heat exchanger. This is contrary to the current research and expectations of findings. It is expected that a plate heat exchanger would have a higher U value, due to, two cold flows surrounding each hot flow (Cengel and Ghajar 2015). It is also expected that counter flow arrangements have a higher L.M.T.D, due to the consistent temperature gradient profile patterns outlined in section 3.1 (Hesselgreaves, Law & Reay 2016).

Table 2: U ($\frac{W}{M^2 \cdot K}$) Tube and Plate Heat Exchanger with Variable Flow rates

Cold Flow	Parallel Tube	Counter Tube	% Δ	Parallel Plate	Counter Plate	% Δ
3 (L/min)	1756.32	1687.80	4.06	1656.88	778.51	112.82
0.5 (L/min)	847.83	871.27	-2.69	847.83	494.08	71.59

3.4 Errors and Recommendations for Variable Flow Rate

The primary errors in this experiment is that TC_1 should always be below 20° at every instance, which was not adhered to. The second major error was a 5 minute equilibrium time for the concentric tube and 3 minutes for the plate heat exchanger rather 5 minutes for both. As it was not possible to measure the temperature in the centre of the plate exchanger, the average was used between inlet and outlet.

Appendix 1 indicates that the centre temperature was not always the average for the concentric tube arrangement and could influence the plate results. When "no significant difference" was determined between heat exchangers, the plate heat exchanger results could become significantly different if both tests were measured at 5 minute equilibrium times. However due to the nature of exponential temperature change, it was deemed unsuitable for exponential extrapolation to estimate plate heat exchanger data at 5 minutes (Appendix 2).

4 Conclusion

As seen there are numerous expected and unexpected outcomes from this experimental design, further recommendations will be considered before replication and systemic review of results from other TD 360 results. Due to the vast array of variables, the conclusion will be outlined in separate points per variable.

4.1 Variable Flow rates

1. 0.5 L/min flow rates had the highest increase in outlet cold water temperature for the concentric tube compared to 3 L/min.
2. 3 L/min flow rates had the highest decrease in outlet hot water temperature for the concentric tube compared to 0.5 L/min.
3. The same trends of flow rates of 1 & 2 were seen in the plate heat exchanger.
4. This indicates that a 0.5 L/min flow rate should be used if that aim is to heat up cold water and 3.0 L/min should be used if the aim is to decrease the temperature of hot water for either heat exchanger.

4.2 Parallel or Counter flow orientation

1. There was no significant difference in both outlet temperatures of the same flow rate between either flow direction in the concentric tube heat exchanger.
2. There was a significant difference in cold water outlet temperatures, with the largest difference in the parallel flow direction for the plate heat exchanger.

4.3 Conclusion of L.M.T.D

1. There was no significant difference between L.M.T.D values at the same flow rate in the concentric tube.
2. There was a significant difference between L.M.T.D values at the same flow rates in the plate heat exchanger.
3. The highest L.M.T.D values occurred in the counter flow arrangement as expected however the largest mean difference was at 3 L/min.

4.4 Conclusion of Heat transfer rate

1. There was no significant difference between thermal transfer rate in concentric tube heat exchanger at the same flow rate.
2. There was a significant difference between thermal transfer rate in plate heat exchanger with the parallel configuration the preferred option.

References

1. Abu-Khader, M.M. (2012). Plate heat exchangers: Recent advances. Renewable and Sustainable Energy Reviews, [online] 16(4), pp.1883–1891. doi:10.1016/j.rser.2012.01.009.

2. Hesselgreaves, J.E., Law, R. and Reay, D. (2016). Compact Heat Exchangers : Selection, Design and Operation. [online] Elsevier Science. Available at: https://www.elsevier.com/books/compact-heat-exchangers/hesselgreaves/978-0-08-100305-3 [Accessed 26 Nov. 2019].

3. Thulukkanam. K (2013). Heat Exchanger Design Handbook, Second Edition. Crc Press.

4. Naphon, P. (2006). Second law analysis on the heat transfer of the horizontal concentric tube heat exchanger. International Communications in Heat and Mass Transfer, 33(8), pp.1029–1041. doi:10.1016/j.icheatmasstransfer.2006.05.001.

5. S Kakac, Hongtan Liu and A Pramuanjaroenkij (2020). Heat exchangers : selection, rating, and thermal design. Boca Raton, Fl: Crc Press.

6. TecQuipment. (2018). Heat Exchanger Service Module. [online] Available at: https://www.tecquipment.com/heat-exchanger-service-module.

7. Welty, J. Rorrer. G, and Foster. D (2020). Fundamentals Of Momentum, Heat, And Mass Transfer. 6th Ed. S.L.: John Wiley.