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Heat Exchanger Laboratory Analysis

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Introduction

The transfer of heat from one fluid to another is found to be invaluable useful in all forms of industry whether it be within a catering company through the use of refrigeration units or within the HVAC installation of a large office block inducing heat into the air. Such a large range of applications is reflected in the breadth of different type of heat exchangers. Driven by temperature difference a recuperative heat exchangers efficiency is greatly affected by flow direction as discussed within the report. While partial to flow direction a heat exchanger is also affected with fluid flow rates. Focusing on a water to air cross flow heat exchanger an optimum operational area is investigated through the use of laboratory testing. Specifically testing the effect of various water and air flow rates at two different water temperatures.

Heat Exchangers

A heat exchanger in its most simple form is a device which allows the transfer of thermal energy from one fluid to another when a temperature difference is present, be this within a gas or liquid state. Heat transfers from a hot fluid to a cold fluid, driven by the temperature difference, therefore thermal equilibrium is not possible within a heat exchanger. Heat transfer within a heat exchanger exists in two forms conduction and convection, radiation is negligible thus ignored. Conduction, the transfer of heat through direct contact, is present when a solid body is in contact with another or a liquid. Convection on the other hand describes the heat transfer of a fluid only, as thermal energy transfers through a fluid it becomes denser which causes movement and therefore circulation. Forced convection is when the fluid moves at velocity, present in forced draft units. (Gonzalez, 2015)

The main heat exchanger focused upon are recuperative, thus indicating that the thermal transfer occurs through a solid element, such as a steel plate, known as the interface. Thus, no mixing of the two fluids occur. The interface is commonly formed either out a tube or plate structure. The significant difference that exist within heat exchangers are the direction of fluid flow as this impacts the difference in temperatures of the two fluid through the entirety of the circuit.

There are three main types of recuperative heat exchangers that mainly differ with the direction of flow. The first being a parallel flow system where the hot and cold fluid have the same inlet side and flow parallel to each other. The temperature difference is the greatest initially leading to a high degree of heat transfer however this exponentially decreases as the temperature difference decreases, showcased in the left hand graph of figure 1. The second system type is known as a counter flow system, and as the name suggests the fluid flows in opposite directions to each other. The most efficient of the three main types, due to the temperature difference remaining at a constant maximum, right hand side of figure 1. (Butterworth, 1977) The third type of heat exchanger is a cross flow system. Seen on road going vehicles, commonly known as a radiator, operates with the cooling fluid (air) flowing perpendicular to the primary hot fluid, which runs through tubes. A pressure drop, between flow entering and exiting, is present as the fluid passes through the heat exchanger, a spectacle which aids heat transfer through the induction of turbulence. However this has a negative effect as a greater pressure drop means an increases in fluid pumping requirements. (SWEP International, 2019) In the case of an internal combustion engine this will require higher loads on the engine ultimately negatively impacting the system as a whole. In some cases pressure drop is increased through the use of staggered tubes within a cross flow system, pictured in appendix (13), through both the induction of more turbulence and a greater surface area. As the heat transfer coefficient of a gas is low compared to a liquid it is common practise for finned extrusion to be featured on the surface of the pipes. The temperature difference exhibited across the length of the circuit is similar to that of a parallel system.

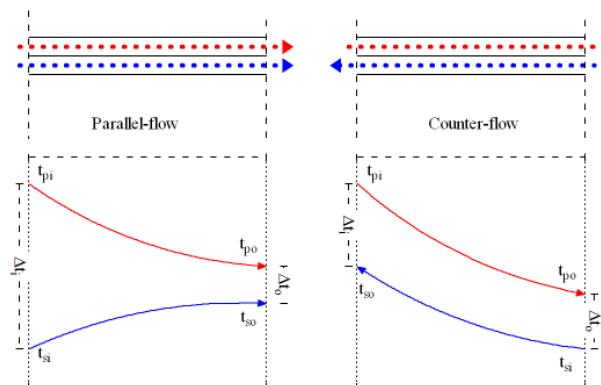


Figure 1 - A set of two graphs illustrating the temperature difference over time for a parallel system and a counter flow system.
https://www.engineeringtoolbox.com/arithmetic-logarithmic-mean-temperature-d_436.html

Experimental Testing

In order to understand the effect of different flow rates of both water and air as well as water fluid temperatures on the heat transfer of a recuperative heat exchanger a TQ TD1007 cross flow water to air heat exchanger configured with 32 tubes was used, featured in figure 2. Cold air is induced through an orifice of a set size, in this case 100.11mm, by an electrical fan pump and forced downstream through the heat exchanger before it exits an variable size orifice (adjustable through a slide pictured in figure 3). For all three test conditions the 32-tube heat exchanger was used and set into the parallel flow configuration. A total of four-minute intervals were issued between each reading in order to allow for thermal equilibrium to occur. Water flow rates were controlled by a valve and set using a digital flow rate meter. The pressure drop was adjusted using the end orifice slide.

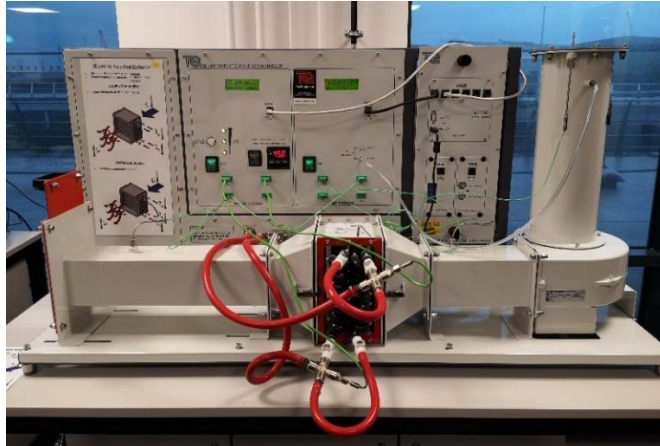


Figure 2 - The TQ TD1007 water to air heat exchanger, configured with the 32 tube cross flow heat exchanger.

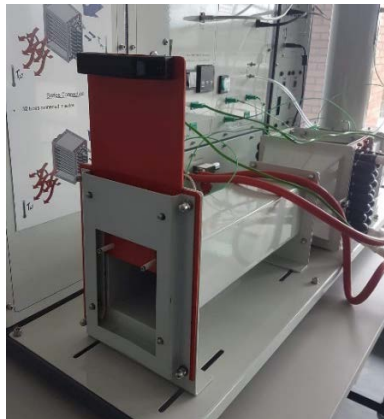


Figure 3 - Variable size orifice used to control the pressure drop with the system.

1.1 The Effect of Varying Water Flow Rate

The first experiment is done in order to observe how different water flow rates affect the performance of the heat exchanger when air flow and water temperature are both fixed.

1.1.1 Procedure

Firstly, ambient air temperature is noted. Water temperature is set at 60 degrees Celsius. The fan is turned on and the orifice pressure drop is set at 250 Pa, however during testing this fluctuated to within ± 2 Pa. Pressure was then kept constant during the whole test. The water supply was similarly set to an initial water flow rate of 2.25 L/minute. Once thermal equilibrium was reached, water inlet 'Tw1' and outlet 'Tw2' as well as Air inlet 'TA1' and outlet 'TA2' temperatures were noted. This process is then repeat adjusting the water flow rate 0.50 L/min each time until 0.25 L/min is reached.

The results are featured in the complete analysis section.

1.2 The Effect of Varying Air Flow Rates

The Second Experiment was aimed at understanding how varying air flow rates affect the efficiency of the heat exchanger considering the water flow rate and temperature was fixed.

1.2.1 Procedure

Water temperature was still held at 60 degrees Celsius. Ambient air pressure was also noted. Once the fan was in operation the slide valve was used to set the airflow pressure drop to an initial 250 Pa. Water flow was set at a constant 25 L/min. After the allotted time for thermal equilibrium measurements were taken similar to the previous experiment. The air pressure drop was altered in increments of 50 Pa until 25 Pa was reached and the recording process repeated.

The results are featured in the complete analysis section.

1.3 The Effect of a Varying Water Temperature

Lastly in order to understand how a difference in water supply temperature would affect the performance of the heat exchanger the temperature was lowered to 45 degrees Celsius and the methodology of experiment two repeated.

1.3.1 Procedure

To start the water temperature is set at a target of 45 degrees Celsius, time was spent waiting for it to drop from the previous 60 degrees. The test machine was run at the lowest pressure drop in order to aid cooling and reduce waiting time. Once the correct temperature was met and the ambient air pressure recorded the pressure drop was reset at 250 Pa. The recording process as well as incremental changes followed in step two were applied for this experiment.

The results are featured in the complete analysis section.

1.4 Complete Analysis

All data collected was entered into an excel workbook as seen in figure 1. Math channels were then further used to understand and graph the result into visual models in order to assess relevant trends. First the change in temperature was found for each test reading of each experiment followed by the average temperature, found through equation 1.

EXPERIMENT 1													
Flow (L/min)	P_amb (bar)	Tank Temp 60	P Drop (pa)	Orifice Diameter (mm)	Orifice A	0.00787127	m ²						
					T _u 1 (Celsius)	T _u 2 (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	m _w (Kg/s)	C _{pu} (J/Kg.K)	Q _u (watts)	η _u (%)
2.25	1.031	249		100.11	60.5	56.7	3.8	58.6	983.90	0.0069	4183.50	586.55	11.18
1.75	1.031	248		100.11	60.8	56	4.8	58.4	984.00	0.0287	4183.42	576.31	14.20
1.25	1.031	251		100.11	60.5	54.2	6.3	57.35	984.52	0.0305	4183.99	540.52	18.81
0.75	1.031	249		100.11	59.5	50.2	9.3	54.85	985.75	0.0123	4182.03	479.23	27.93
0.24	1.031	250		100.11	60.2	39.1	21.1	49.65	988.18	0.0040	4180.32	348.65	63.06
EXPERIMENT 2													
Pdrop (pa)	P_amb (bar)	Flow (L/min)	Orifice Diameter (mm)	T _u 1 (Celsius)	T _u 2 (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	m _w (Kg/s)	C _{pu} (J/Kg.K)	Q _u (watts)	η _u (%)	
250	1.031	0.24	100.11	60.2	39.1	21.1	49.65	988.18	0.0040	4180.32	348.65	62.06	
200	1.031	0.24	100.11	58.1	38.6	19.5	48.35	988.76	0.0040	4179.96	322.37	60.94	
150	1.031	0.24	100.11	58.1	39	19.1	48.55	988.67	0.0040	4180.01	315.73	59.87	
100	1.031	0.24	100.11	58	40.1	17.9	49.05	988.45	0.0040	4180.15	295.84	55.94	
50	1.031	0.24	100.11	58.3	41.9	16.4	50.1	987.98	0.0040	4180.45	270.94	51.09	
25	1.031	0.25	100.11	58.6	43.6	15	51.1	987.52	0.0041	4180.76	258.04	46.44	
EXPERIMENT 3													
Pdrop (pa)	P_amb (bar)	Flow (L/min)	Orifice Diameter (mm)	T _u 1 (Celsius)	T _u 2 (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	m _w (Kg/s)	C _{pu} (J/Kg.K)	Q _u (watts)	η _u (%)	
251	1.031	0.25	100.11	49.4	34.2	15.2	41.8	991.50	0.0041	4178.54	262.39	62.04	
201	1.031	0.24	100.11	48.3	33.8	14.5	41.05	991.80	0.0040	4178.43	240.36	61.18	
149	1.031	0.25	100.11	47.9	34.2	13.7	41.05	991.80	0.0041	4178.43	236.56	59.83	
98	1.031	0.25	100.11	47.8	35.4	12.4	41.6	991.58	0.0041	4178.51	214.07	56.11	
47	1.031	0.25	100.11	47.9	36.8	11.1	42.35	991.28	0.0041	4178.64	191.58	49.12	
25	1.031	0.25	100.11	48.2	37.7	10.5	42.95	991.04	0.0041	4178.71	181.18	46.26	
T _u 1 (Celsius)	T _u 2 (Celsius)	T _u 3 - ambient (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	m _w (kg/s)	C _{pu} (J/Kg.K)	Q _u (watts)	η _u (%)		Mean η (%)	LMTD	U(W/m ² .K)
26.5	31.9	25.7	5.4	29.2	1.19	0.11	1005	622.79	15.88		13.529	29.159	129.11
27	32	26.8	5	29.5	1.18	0.11	1005	575.22	14.79		14.497	28.621	129.24
27	31.8	26.7	4.8	29.4	1.18	0.12	1005	555.62	14.33		16.567	27.579	125.80
26.2	30.5	25.6	4.3	28.35	1.19	0.11	1005	496.63	12.91		20.420	25.908	118.73
26.2	29.4	25.9	3.2	27.8	1.19	0.12	1005	370.66	9.41		35.735	19.374	115.50
T _u 1 (Celsius)	T _u 2 (Celsius)	T _u 3 - ambient (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	Air mass flow rate (kg/s)	1005	Q _u (watts)	η _u (%)		Mean η (%)	LMTD	U(W/m ² .K)
26.2	29.4	25.9	3.2	27.8	1.19	0.12	1005	370.66	9.41		35.735	19.374	115.50
26.1	29.3	25.6	3.2	27.7	1.19	0.10	1005	331.66	10.00		35.469	18.370	112.64
26.2	29.5	25.7	3.3	27.85	1.19	0.09	1005	296.20	10.34		35.110	18.492	109.59
26	29.7	25.2	3.7	27.85	1.19	0.07	1005	271.23	11.56		33.750	19.218	98.80
26.2	30.6	25.4	4.4	28.4	1.19	0.05	1005	227.92	13.71		32.399	19.922	87.29
26.3	31.5	25	5.2	28.9	1.19	0.04	1005	190.33	16.10		31.269	20.573	80.50
T _u 1 (Celsius)	T _u 2 (Celsius)	T _u 3 - ambient (Celsius)	ΔT _u (Celsius)	Average T _u (Celsius)	ρ _u (Kg/m ³)	Air mass flow rate (kg/s)	1005	Q _u (watts)	η _u (%)		Mean η (%)	LMTD	U(W/m ² .K)
24.9	27.3	24.4	2.4	26.1	1.20	0.12	1005	279.34	9.80		35.92	13.89	121.26
24.6	27	24.1	2.4	25.8	1.20	0.10	1005	250.16	10.13		35.65	13.54	113.98
25	27.4	24.6	2.4	26.2	1.20	0.09	1005	215.29	10.48		35.15	13.26	114.51
25.7	28.2	25	2.5	26.95	1.20	0.07	1005	181.70	11.31		33.71	13.29	103.42
25.3	28.5	24.2	3.2	26.9	1.20	0.05	1005	161.11	14.16		31.64	14.28	86.13
25.5	29.1	24.2	3.6	27.3	1.20	0.04	1005	132.12	15.86		31.06	14.53	80.05

Figure 4 - Excel workbook containing the data tables as well as math channels used to compile graphs.

$$\text{Average } T_w (\text{Celsius}) = \frac{T_{w1} + T_{w2}}{2}$$

$$\text{Average } T_A (\text{Celsius}) = \frac{T_{A1} + T_{A2}}{2}$$

Equation 1 - Average temperature of water and air respectively.

The table was further populated by finding the density of water followed by the air density, both found through equations 2 and 3 respectively. The gas constant of dry air was used as detailed below. Following on the water mass flow rate was found through equation 4 and the mass air flow through equation 5.

$$\rho_w = (999.839 + 16.952 * \text{Average } T - 7.990 * 10^{-3} * \text{Average } T^2 - 46.241 * 10^{-6} * \text{Average } T^3 + 105.846 * 10^{-9} * \text{Average } T^4 - 281.030 * 10^{-12} * \text{Average } T^5) / (1 + 16.887 * 10^{-3} * \text{Average } T)$$

Equation 2 - Density of water.

$$\rho_A = \frac{\text{Ambient Pressure (Pa)} - \text{Pressure Drop (Pa)}}{R * (\text{Average } T + 273)}$$

$$R = \text{Gas Constant} = 287.058 \text{ (J/kg.K)}$$

Equation 3 - Air density.

$$\text{Water Mass Flow Rate } (kg/s) = \left(\text{Water Volume } \frac{\text{flow}(L/min)}{60000} \right) * \text{Water Density } (kg.m^{-3})$$

Equation 4 - Water flow rate conversion.

$$\text{Mass Air flow} = Cd * \text{Orifice Diameter (mm)} * \sqrt{\text{Pressure Drop (Pa)}} * 2 * \rho A$$

Equation 5 - Mass air flow rate

The specific heat capacity of air was assumed to be at 1005 J/kg/K. However, on the other hand the specific heat capacity of water was found through equation 6. Once those figures were obtained the heat emitted by the water and the heat absorbed by the air could be calculated using equation 7 and 8 respectively. A valuable figure, this quantifies the heat transfer rates of both elements leading to an understanding through graphing of both elements against either water flow rate or air flow rate as featured in the next section.

$$CpW \left(\frac{J}{kg} \cdot K \right) = 0.996185 + 0.0002874 \left(\frac{T + 100}{100} \right)^{5.26} + 0.011160 * 10^{-0.036T}$$

Equation 6 - Specific heat capacity of water.

$$QW = \text{Mass flow Rate Water } (kg/s) * CpW * \Delta TW$$

Equation 7 - Heat emitted by water.

$$QA = \text{Mass Air Flow} * CpA * \Delta TA$$

Equation 8 - Heat absorbed by the air.

The following final equations bring both elements together in order to understand how both elements interact. Firstly the temperature efficiencies of both elements were calculated as well as the mean temperature efficiency. This describes the percentage of heat removed from the water or air compared to its initial heat gain. (Kalogirou, 2014) The mean temperature efficiency allows for useful comparison between different heat exchangers. The logarithmic mean temperature difference is worked out in order to measure the driving force of the heat transfer. Finally concluding the math channels the heat transfer coefficient is found using equation 11. A valuable figure as this leads to the understanding of the degree of heat transfer across the boundary layer.

$$\eta W (\%) = \left(\frac{\Delta TW}{TW1 - TA1} \right) * 100$$

$$\eta A (\%) = \left(\frac{\Delta TA}{TW1 - TA1} \right) * 100$$

$$\text{Mean } \eta (\%) = \frac{\eta A + \eta W}{2}$$

Equation 9 - Temperature efficiencies of both water and air as well as mean temperature efficiencies.

$$LMTD = \frac{(TW2 - TA2) - (TW1 - TA1)}{\ln \left(\frac{TW2 - TA2}{TW1 - TA1} \right)}$$

Equation 10 - Logarithmic Mean Temperature Difference.

$$U (W/m^2 * K) = \frac{QW}{0.1558 * LMTD}$$

Equation 11 - Heat transfer coefficient.

1.4.1 Varying Water Flow Rate Results

The following are the results of the first experiment graphed. It is observed that as the flow rate of the water increases heat power of both water and air increases in a similar trend. The slope of both level off after 0.019 kg/s indicating the maximum heat transfer levels are being reached. Interestingly the power transfer slope of water significantly increases after 0.028 kg/s, which is most likely due to a slight variation in testing condition.

Reviewing the temperature efficiencies graph it is clear that as the water flow rate increases the temperature efficiency of water exponentially decreases while that of air slightly increases. This is due to there being a greater change in water temperature than air. As the flow rate of water increases the temperature difference at the end of the circuit decreases leading to a lower water temperature efficiency.

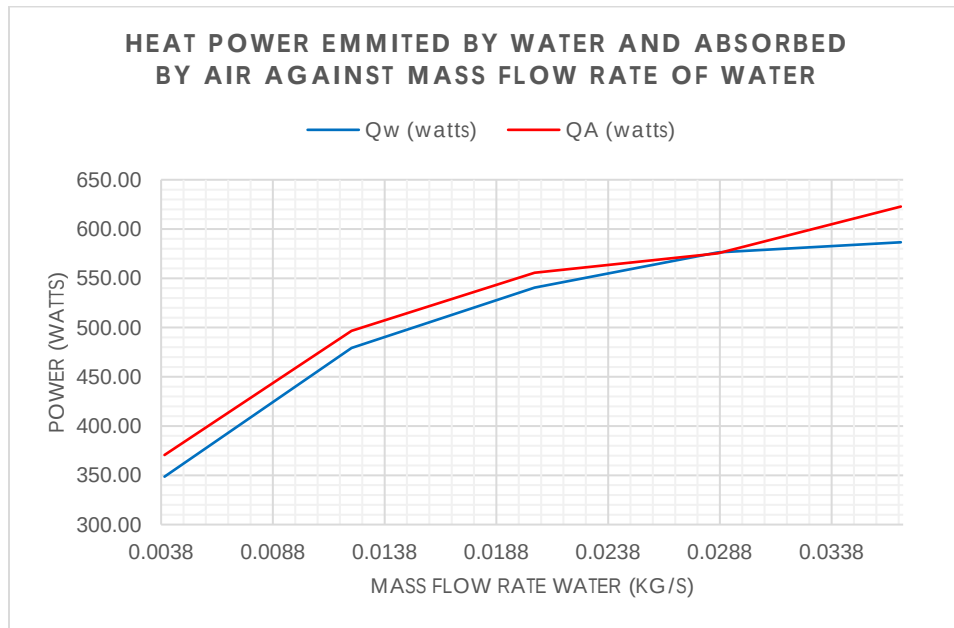


Figure 5 - Heat power emitted and absorbed by the water and air respectively running through the charge cooler.

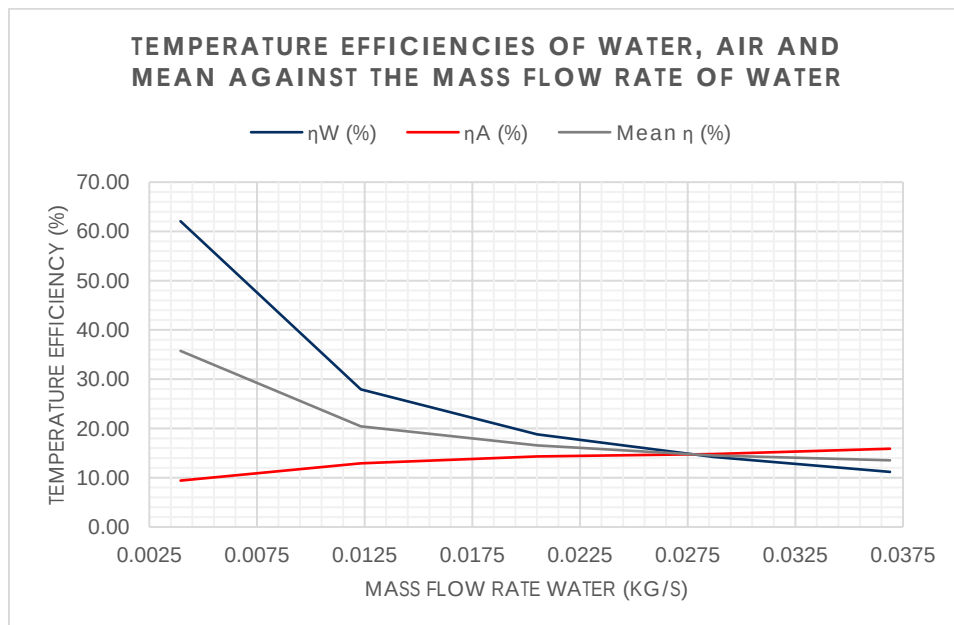


Figure 6 - The effect of water mass flow rate on the efficiencies of the charge cooler elements.

Thirdly through the heat transfer coefficient graph the effect of water flow rate on the performance of the heat exchanger can be understood. As the flow rate of water increases the heat transfer coefficient follows in increase, however after 0.0288 kg/s it is seen to actually decrease thus indicating that maximum efficiency has been achieved and an increase in flow would only further negatively impact its performance.

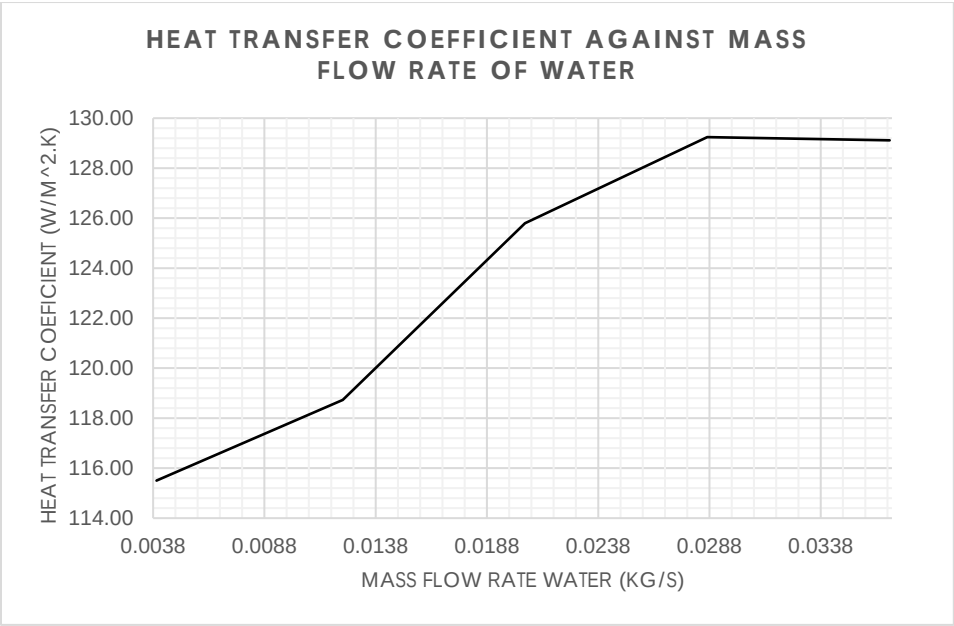


Figure 7 - Effect of water flow rate on the total heat transfer coefficient.

1.4.2 Varying Air Flow Rates Results

Graphing the results for experiment 2 yields similar results to those of experiment one when looking at the heat power transfer, however, a constant angle of slope is generally exhibited by both water and air indicating that the maximum level has not yet been reached. This trend is also reflected in figure 10, as the heat transfer coefficient exhibits a steep slope of increases as the air flow rate increases, almost linear up until 0.09 kg/s. Power transfer of air and water intersect after 0.10 kg/s of air, while at lower flow rates a larger separation is apparent leading to the conclusion that there is apparent heat loss through external factors such as the heat exchanger attachments.

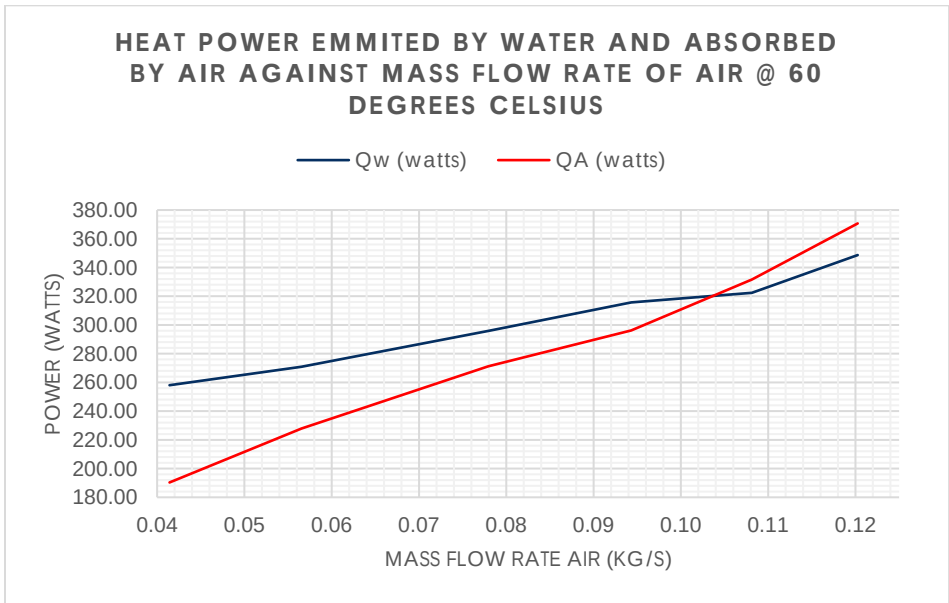


Figure 8 - The influence mass flow rate of air has on emitted and absorbed heat respectively of water and air at a water chamber temperature of 60 degrees Celsius.

The temperature efficiency graph showcases an interesting change compared to the previous experiment as with the increase in air flow the water efficiency increases while a slight decrease in air efficiency is exhibited. The mean temperature follows the increase in water temperature efficiency.

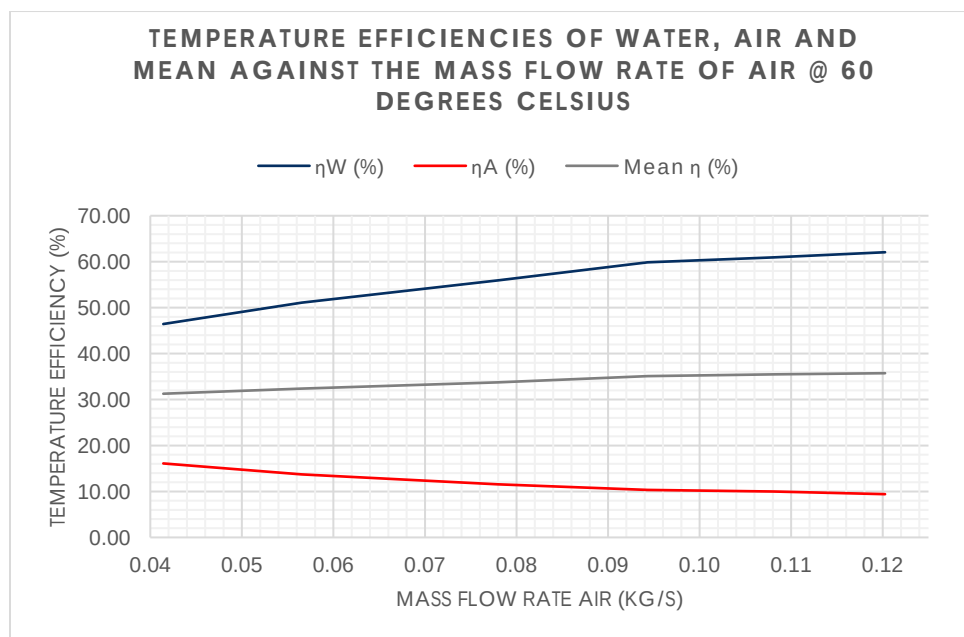


Figure 9 - Comparing the temperature efficiencies of water, air and mean against the mass flow rate of air at 60 degrees Celsius.

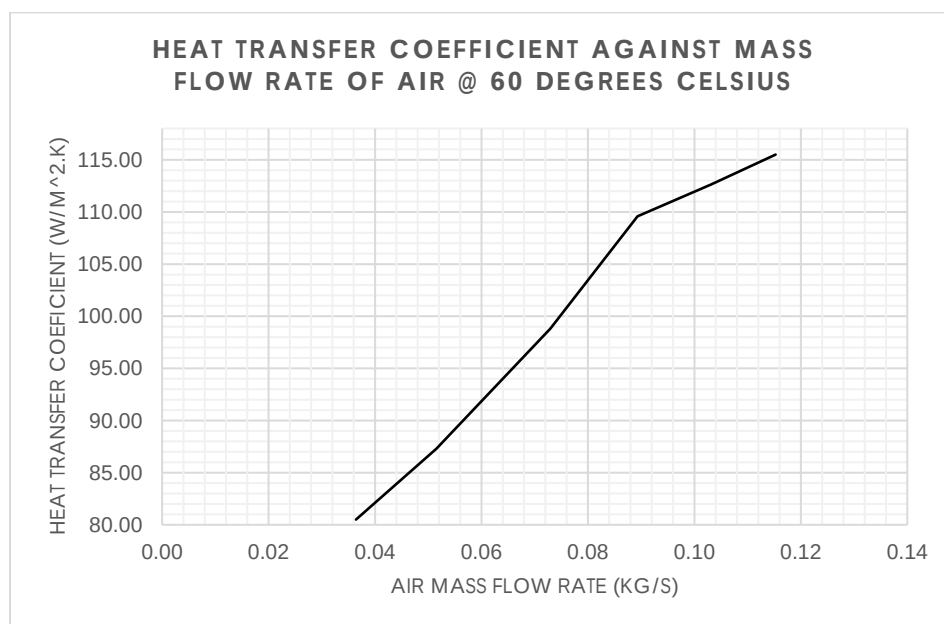


Figure 10 - Impact of air mass flow rate on the total heat transfer coefficient at a water tank temperature of 60 degrees Celsius.

1.4.3 Varying Water Temperature Results

A lower temperature of 45 °C of experiment three reveals a general decrease in heat power transfer compared to 60 °C, which is to be expected considering the lower temperature difference. The formation of the trend lines remain similar exhibiting the same intersection however now occurring before 0.10 kg/s. It appears a higher air flow rate would yield greater power transfer however it may soon plateau. The temperature efficiency graph shows the exact same trends as at 60 °C while the heat transfer coefficient exhibits the same steep linear plot however with an interesting sudden decrease after 0.09 kg/s followed by a sudden continuation

of positive increase after 0.10 kg/s of flow. This is possibly due to the fact that at that data point the water flow reading was at 0.24 L/min instead of 0.25L/min as in all other cases.

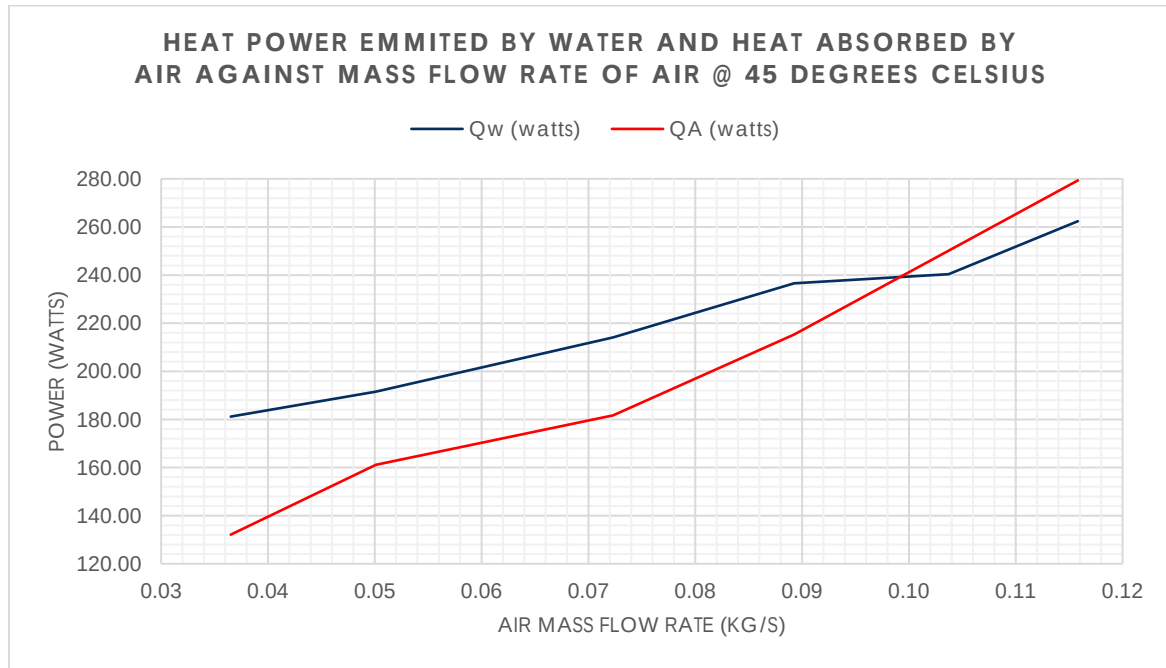


Figure 11 - The heat power of water and air against air mass flow rate at 45 degrees Celsius water tank setting.

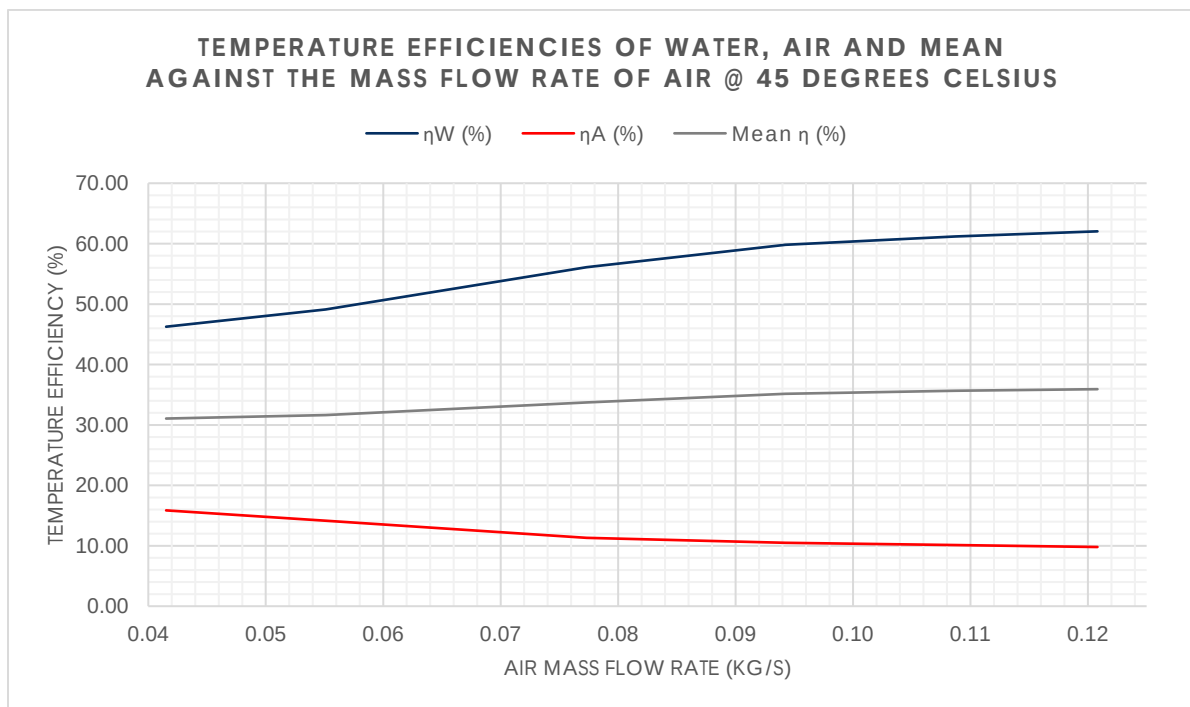


Figure 12 - The effect of air mass flow rate of temperature efficiencies at a water tank temperature of 45 degrees Celsius.

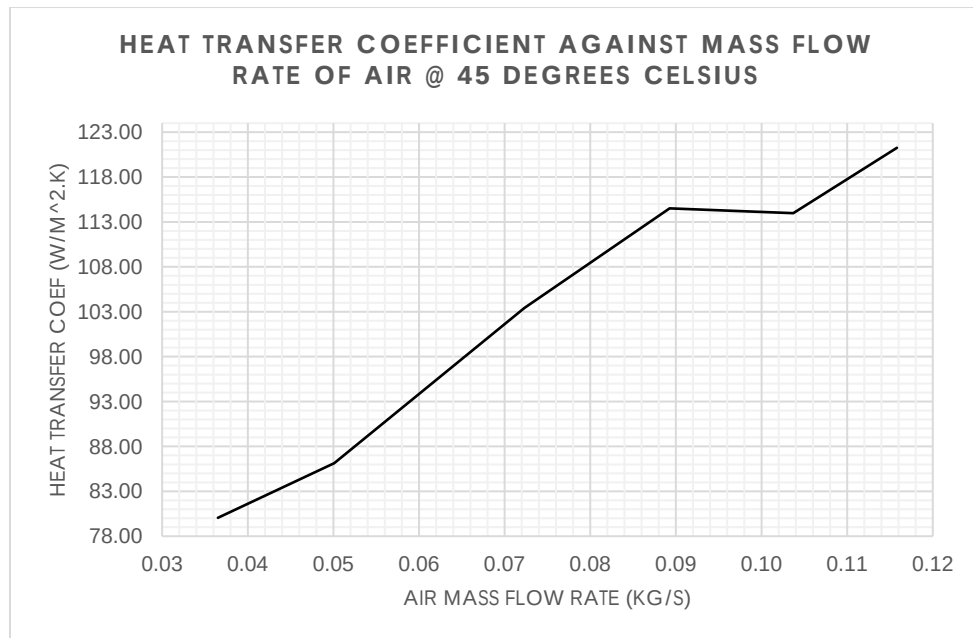


Figure 13 - Total heat transfer coefficient against air mass flow rate at a water tank temperature of 45 degrees Celsius.

Conclusion

Heat exchangers play an important role within industries, with a wide range of different types to suit different operational conditions. Hence understanding the specific capabilities will best determine how efficiently it will perform. Therefore this investigation into the effects of both air and water flow rates through a recuperative cross flow heat exchanger yielded a firm understanding of its performance capabilities given two different temperatures. The discovered ideal operation condition would be a water flow rate of 0.004 kg/s as further flow increase exponential decreases temperature efficiency. While a maximum air flow rate greater than 0.12 kg/s, due to both heat transfer coefficient and heat power transfer graphs showing further scope for increase, would further improve mean temperature efficiency. At 45°C the graphs yielded similar trends as when the water temperature was at 60°C, however the decrease in temperature difference, being the driving factor, showed a general decrease in power transfer values, especially leading to an earlier decrease in heat transfer slope. These results were all assuming constant air temperature, which unfortunately was not the case as an increase in ambient temperature was completely plausible considering the numerous influx of lab technicians observing the experiment. The gas constant was assumed for dry air, humidity levels were not taken into consideration. Thus, ideally this would be repeated in a precise temperature and humidity controlled environment. As flow rates fluctuated they were altered in order to remain within tolerance, however this most likely affected reading and the period of thermal equilibrium. Self-controlling accuracy mechanisms would greatly be beneficial, as well a new heat exchanger containment structure made out of a material of low thermal conductivity in order to mitigate thermal losses. Greater thermal equilibrium time may also have allowed for better result accuracy. Further investigation into a single row as well as in-line cross flow heat exchanger configurations would be of interest, however a substantial understanding of the current heat exchanger has been gained to then be further considered for a multitude of applications.

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[Accessed 06 04 2019].

Appendix

$$Tw = Tw2 - Tw1$$

$$TA = TA2 - TA1$$

Equation 12 - Change in water and air temperature respectively.

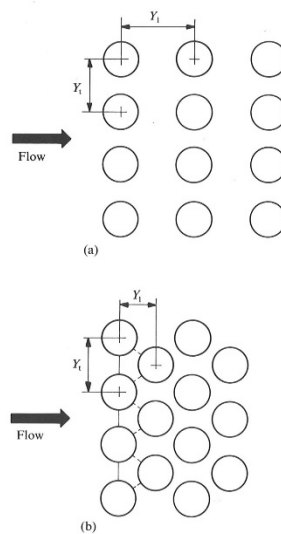


Figure 14 - Diagram showcasing in-line (a) and staggered (b) heat exchanger tube arrangements. (Butterworth, 1977)