



University of Wales Trinity Saint David
School of Engineering

Heat Exchanger Laboratory Analysis
Thermofluid Mechanics
M5X00406

George Jones
P149914
(BEng) Motorsport Engineering Year 2
07.04.2019

Abstract

Heat exchangers are used in all aspects of engineering applications as a method of thermal management. These devices can be in many forms of multiple arrangements according to their specific requirements. This investigation will explore the performance of a multi-pass cross-flow water to air heat exchanger, under measured laboratory conditions. The results of which will be analysed to further understand the relationships between heat transfer and temperature efficiencies within the system, as well as how different mass flow rates and temperatures affect these properties.

Contents

Abstract.....	ii
List of Equations.....	iv
List of Tables	v
Introduction	1
Aims	1
Objectives	1
Literature Review.....	2
Methodology.....	4
Apparatus Configuration.....	4
Test Parameters	6
Experiment 1 - The Effect of Varying Water Flow Rate	6
Experiment 2 - The Effect of Varying Air Flow Rate.....	7
Experiment 3 - Effect of Varying Temperature (Driving Force)	8
Results.....	9
Experiment 1 - The Effect of Varying Water Flow Rate	9
Experiment 2 - The Effect of Varying Air Flow Rate.....	10
Experiment 3 - Effect of Varying Temperature (Driving Force)	11
Analysis	12
Experiment 1 - The Effect of Varying Water Flow Rate	12
Experiment 2 - The Effect of Varying Air Flow Rate.....	14
Experiment 3 - Effect of Varying Temperature (Driving Force)	16
Further Improvements.....	18
Conclusion.....	18
References	19
Appendix	20
Equations	20
Water Density	20
Air Density.....	20
Specific Heat Capacity of Water.....	20
Mass Flow Rate	20
Power Emitted/Absorbed	20
Efficiency	21
Heat Transfer Coefficient.....	21
Nomenclature	22

List of Figures

Figure 1 - Cross-flow and Counter-flow Systems (Cengel, 2002).....	2
Figure 2 - Tecquipment TD1007 Heat Exchanger Equipment.....	4
Figure 3 - Parallel Flow Arrangement Diagram	4
Figure 4 - Standard Tecquipment Heat Exchanger Unit	4
Figure 5 - TD1007 Flow Arrangement	5
Figure 6 - Experiment 1, Power Emitted & Absorbed	12
Figure 7 - Experiment 1, Heat Transfer Coefficient	13
Figure 8 - Experiment 1, Efficiencies	13
Figure 9 - Experiment 2, Power Emitted & Absorbed	14
Figure 10 - Experiment 2, Efficiencies	15
Figure 11 - Experiment 2, Heat Transfer Coefficient	15
Figure 12 - Experiment 3, Power Emitted & Absorbed	16
Figure 13 - Experiment 3, Comparison of Different Initial Water Temperatures	16
Figure 14 - Experiment 3, Efficiencies	17
Figure 15 - Experiment 3, Heat Transfer Coefficient	17
Figure 16 - Nomenclature Table (Tecquipment, 2014)	22

List of Equations

Equation 1 - Heat Transfer	3
Equation 2 - Water Density	20
Equation 3 - Air Density	20
Equation 4 - Specific Heat Capacity of Water	20
Equation 5 - Mass Flow Rate for Water	20
Equation 6 - Mass Flow Rate for Air	20
Equation 7 - Power Emitted/Absorbed	20
Equation 8 - Efficiency of Water Circuit	21
Equation 9 - Efficiency of Air Circuit	21
Equation 10 - Mean Temperature Efficiency	21
Equation 11 - Logarithmic Mean Temperature Difference	21
Equation 12 - Heat Transfer Coefficient	21

List of Tables

Table 1 - Experiment 1, System Data	6
Table 2 - Experiment 1, Air Data	6
Table 3 - Experiment 1, Water Data	6
Table 4 - Experiment 2, System Data	7
Table 5 - Experiment 2, Air Data	7
Table 6 - Experiment 2, Water Data	7
Table 7 - Experiment 3, System Data	8
Table 8 - Experiment 3, Air Data	8
Table 9 - Experiment 3, Water Data	8
Table 10 - Experiment 1, Flow Results	9
Table 11 - Experiment 1, Air Results	9
Table 12 - Experiment 1, Water Results	9
Table 13 - Experiment 2, Flow Results	10
Table 14 - Experiment 2, Air Results	10
Table 15 - Experiment 2, Water Results	10
Table 16 - Experiment 3, Flow Results	11
Table 17 - Experiment 3, Air Results	11
Table 18 - Experiment 3, Water Results	11

Introduction

Aims

- To illustrate the effects on heat exchanger performance by varying water flow rates for a controlled water temperature and air flow.
- To illustrate the effects on heat exchanger performance by varying air flow rates for a controlled water temperature and flow.
- To illustrate the effects on heat exchanger performance by varying the hot water supply temperature for a controlled water flow.

Objectives

A series of laboratory experiments were carried out in turn, using the Tecquipment TD1007 to gather representative data. This data was processed using thermodynamic principles and comparative research to ascertain heat exchanger performance characteristics.

Literature Review

Heat transfer exists where a change in temperature occurs in physical systems. There are three modes of heat transfer being conduction, convection and radiation. Heat exchangers are one method of transferring thermal energy between fluids and predominantly use the convection method (Frank P. Incropera, 2006).

Most modern machines generate heat as a product, or by-product of converting energy to achieve a desired output. Therefore, heat exchangers are widely used in engineering applications as a form of thermal management (Cengel, 2002). There are various configurations of heat exchanger design and the choice of which depends on its specific application requirements (John H Lienhard IV, 2003).

The most common form of heat exchanger within the automotive industry is a variation of the “tube and shell” type. This recuperative heat exchanger configuration can be further split into three interaction formats, being parallel-flow, counter-flow and cross-flow. The cross-flow technique is predominantly used in automotive applications due to relying on ambient air for the cooling fluid in powertrain systems and the ambient air being the heated fluid in vehicle interior HVAC applications (Stoakes, 2011).

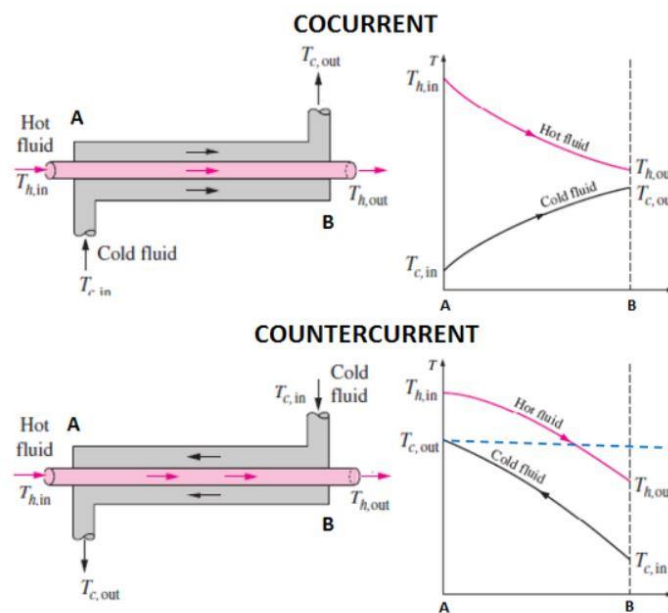


Figure 1 - Cross-flow and Counter-flow Systems (Cengel, 2002)

Counter-flow arrangements offer the greatest heat transfer rate per unit surface area due to their ability to cool the hot circuit outlet to a lower temperature than the cold circuit outlet as highlighted in figure 1.

In fundamental terms, the rate of heat transfer is a product of the surface area of the heat exchanger and temperature difference between the hot and cold circuits.

Equation 1 - Heat Transfer

$$\dot{Q} = U \times A \times \Delta T$$

Where $\dot{Q}$ = Overall Heat Transfer Rate (W), U = Heat Transfer Coefficient (W/m²k), A = Surface area (m²), T = Temperature

As the temperature difference is dynamic, it is necessary to substitute this value with the logarithmic mean temperature difference (LMTD) (*equation 11*) when considering the overall temperature difference across hot and cold circuits in a heat exchanger (Kay, 1985)

Various methods can be applied to increase surface area of a heat exchanger. The addition of small fins is the preferred option in the automotive industry as the cost of production and overall exchanger dimensions can be reduced for the cross-flow configuration, as opposed to the maximum reliability strategy mentioned by (Rodriguez, 1997). The most common material used in automotive heat exchangers is aluminium due to the compromise between cost, corrosion resistance and relative consequence of failure. However, for optimal heat transfer rates through conduction to the separating wall, a material with greater thermal conductivity can be used (Tritt, 2004).

The overall efficiency of heat exchangers can also be improved by implementing a multi-pass model, whereby the cooling, or heating fluid is repetitively passed through the working section as opposed to a single pass route (Thulukkanam, 2000).

Automotive heat exchangers pose a unique challenge through the range in which they are typically used. Free convection occurs in the engine coolant heat exchanger whilst the vehicle is static. This mode of convection is severely limited by natural heat transfer and will be less effective with an increase in ambient temperatures. Another contributing factor is the low water pump speed at engine idle, reducing the coolant mass flow rate through the heat exchanger.

Forced convection also occurs in the application of engine coolant heat exchangers, where an auxiliary fan is used to increase the air mass flow rate where coolant temperatures are increased whilst stationary or at low speeds (Hucho, 1974). Additional heat transfer also occurs when the vehicle speed (relative air velocity) is increased. This scenario can also be described as forced convection from an increased air mass flow rate, with a rise in coolant mass flow rate being applied through the relative water pump speed.

Methodology

Apparatus Configuration

The Tecquipment TD1007 water to air heat exchanger as illustrated in *figure 2* was used for all practical aspects of this investigation.

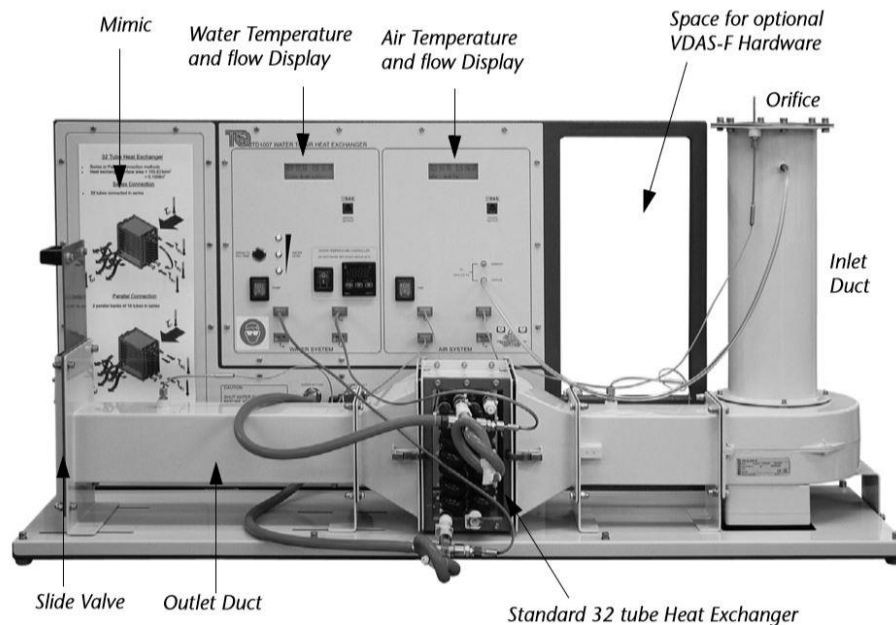


Figure 2 - Tecquipment TD1007 Heat Exchanger Equipment

The TD1007 was set up with its standard 32 tube heat exchanger unit in the parallel flow arrangement and remained in this formation for all procedures. The heat exchanger connectivity was checked in accordance with the supplied mimic sheet before the first test commenced. For additional context the temperature-controlled water passes through the working section from bottom to top, per *figures 3 and 4*. Water flow was adjusted using the hand-controlled valve.

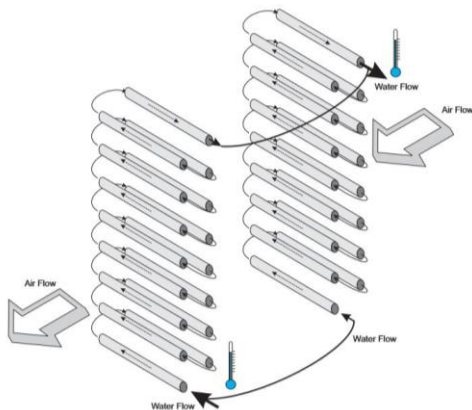


Figure 3 - Parallel Flow Arrangement Diagram

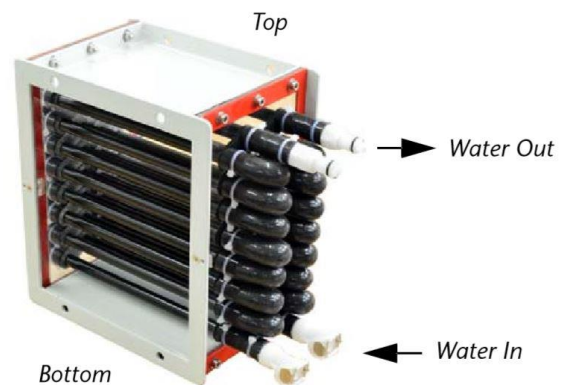


Figure 4 - Standard Tecquipment Heat Exchanger Unit

Interactions between both fluids occur through a controlled forced convection process. *Figure 5* demonstrates the governed fluid paths. The application of thermocouples provided representative temperature readings, both into and out of the fluid paths. The possibility for an intermediary thermocouple is built into the TD1007, however this option is only available with the system in its series configuration (Tecquipment, 2014).

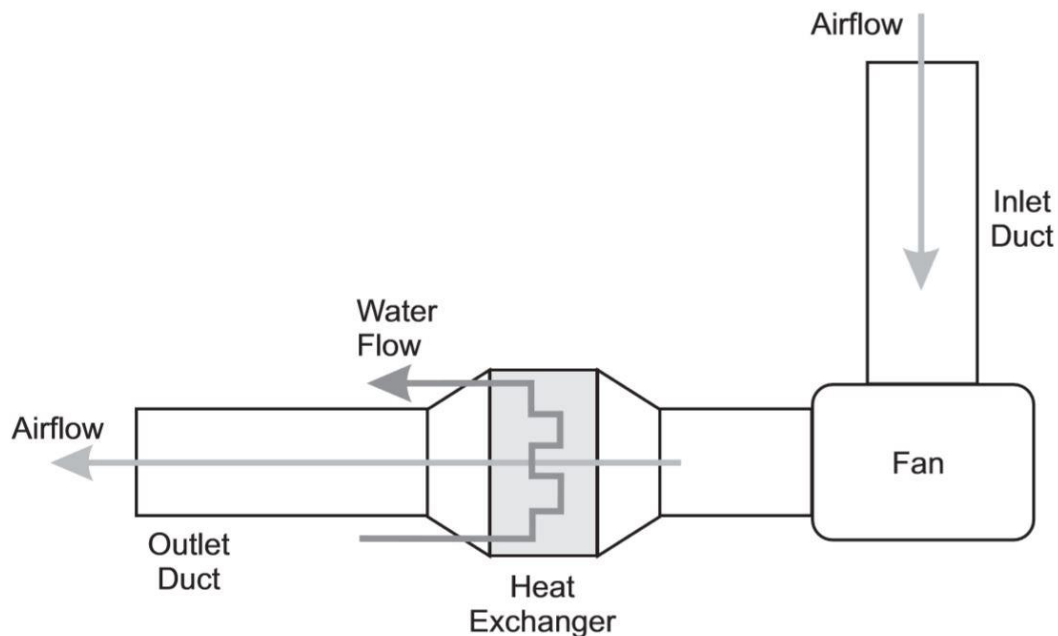


Figure 5 - TD1007 Flow Arrangement

The inlet orifice remained consistent at a calibrated diameter of 0.10011m as displayed in the test parameters table (*tables 1-9*). The air was pulled through the inlet duct and accelerated by a centrifugal fan, before being turned through 90°. From this, it passed through a horizontal duct before an expansion into the working section. The air then flowed through the perpendicular upstream and downstream tube bundles of the heat exchanger before being ejected through an adjustable slide valve positioned in the outlet duct.

All data outputs were manually captured by photographing the water and air temperature/flow displays simultaneously, as the Tecquipment Versatile Data Acquisition Software (VDAS) was not available to use on this occasion. Results were compiled from each procedure in Excel for further analysis.

Test Parameters

The ambient climatic conditions were measured and recorded prior to the first test. It was assumed that no change in ambient pressure values would occur through the course of the experiment.

The controlled and measured investigation parameters can be observed below.

Experiment 1 - The Effect of Varying Water Flow Rate

- The water heater tank temperature was set to 60°C.
- The Slide valve was adjusted to achieve an orifice pressure drop of 250Pa (+/- 5.)
- The water supply valve was adjusted to achieve a water flow of 2.25 L/min (+/- 0.01.)
- The process was repeated for 1.75L/min, 1.25L/min, 0.75L/min and 0.25L/min water flow rates with an equalisation period of 6 minutes between recordings.

Table 1 - Experiment 1, System Data

System Data				
	Ambient Pressure (Bar)	Orifice Diameter (m)	Orifice Area (m ²)	Surface Area (m ²)
1	1.029	0.10011	0.00787127	155.823
2	1.029	0.10011	0.00787127	155.823
3	1.029	0.10011	0.00787127	155.823
4	1.029	0.10011	0.00787127	155.823
5	1.029	0.10011	0.00787127	155.823

Table 2 - Experiment 1, Air Data

Air Data				
	Pressure Difference (Pa)	Ambient Air Temp (°C)	Air Temp In (°C)	Air Temp Out (°C)
1	249	24.8	25.2	30.7
2	249	24.9	25.2	30.5
3	250	25	25.3	30.4
4	248	24.8	25.2	29.8
5	250	25.2	25.3	28.6

Table 3 - Experiment 1, Water Data

Water Data				
	Water Flow (L/min)	Water Tank Temp (°C)	Water Temp In (°C)	Water Temp Out (°C)
1	2.26	60	60.9	57
2	1.75	60	60.9	55.9
3	1.25	60.1	60.5	53.8
4	0.75	59.8	59.9	50
5	0.25	60.1	60.2	38.7
INDEPENDENT VARIABLE				

Experiment 2 - The Effect of Varying Air Flow Rate

- The water heater tank temperature was set to 60°C.
- The Slide valve was adjusted to achieve an orifice pressure drop of 250Pa (+/- 5.)
- The water supply valve was adjusted to achieve a water flow of 0.25 L/min (+/- 0.01.)
- The process was repeated for air flow rates of 200Pa, 150Pa, 100Pa and 50Pa pressure drop with an equalisation period of 6 minutes between recordings.

Table 4 - Experiment 2, System Data

System Data				
	Ambient Pressure (Bar)	Orifice Diameter (m)	Orifice Area (m ²)	Surface Area (m ²)
1	1.029	0.10011	0.00787127	0.1558
2	1.029	0.10011	0.00787127	0.1558
3	1.029	0.10011	0.00787127	0.1558
4	1.029	0.10011	0.00787127	0.1558
5	1.029	0.10011	0.00787127	0.1558
6	1.029	0.10011	0.00787127	0.1558

Table 5 - Experiment 2, Air Data

Air Data				
	Pressure Difference (Pa)	Ambient Air Temp (°C)	Air Temp In (°C)	Air Temp Out (°C)
1	250	25.2	25.3	28.6
2	200	24.9	25.4	28.7
3	149	25.1	25.6	28.9
4	99	24.9	25.7	29.4
5	50	24.7	25.9	30.4
6	25	25	26.4	31.6
INDEPENDENT VARIABLE				

Table 6 - Experiment 2, Water Data

Water Data				
	Water Flow (L/min)	Water Tank Temp (°C)	Water Temp In (°C)	Water Temp Out (°C)
1	0.25	60.1	60.2	38.7
2	0.24	59.9	58.7	38.4
3	0.24	60.1	58.5	39
4	0.24	60.1	58.8	40
5	0.24	60	59.1	42
6	0.24	60.1	59.6	44

Experiment 3 - Effect of Varying Temperature (Driving Force)

- The water heater tank temperature was set to 45°C.
- The Slide valve was adjusted to achieve an orifice pressure drop of 250Pa (+/- 5.)
- The water supply valve was adjusted to achieve a water flow of 0.25 L/min (+/- 0.01.)
- The process was repeated for air flow rates of 200Pa, 150Pa, 100Pa and 50Pa pressure drop with an equalisation period of 6 minutes between recordings.

Table 7 - Experiment 3, System Data

System Data				
	Ambient Pressure (Bar)	Orifice Diameter (m)	Orifice Area (m ²)	Surface Area (m ²)
1	1.029	0.10011	0.00787127	155.823
2	1.029	0.10011	0.00787127	155.823
3	1.029	0.10011	0.00787127	155.823
4	1.029	0.10011	0.00787127	155.823
5	1.029	0.10011	0.00787127	155.823
6	1.029	0.10011	0.00787127	155.823

Table 8 - Experiment 3, Air Data

Air Data				
	Pressure Difference (Pa)	Ambient Air Temp (°C)	Air Temp In (°C)	Air Temp Out (°C)
1	250	24.7	25.3	27.5
2	200	25.3	25.4	27.7
3	149	24.8	25.5	27.9
4	103	24.7	25.6	28.2
5	50	24.7	25.8	29
6	26	24.7	26.2	29.7

Table 9 - Experiment 3, Water Data

Water Data				
	Water Tank Temp (°C)	Water Flow (L/min)	Water Temp In (°C)	Water Temp Out (°C)
1	45	0.24	48.8	34
2	45.8	0.24	48.5	34.4
3	45	0.24	48.5	34.9
4	44.9	0.24	48.7	35.6
5	45.7	0.24	48.9	37
6	45.6	0.24	49.1	38.3
INDEPENDENT VARIABLE				

Results

Experiment 1 - The Effect of Varying Water Flow Rate

Table 10 - Experiment 1, Flow Results

Flow Results				
	Logarithmic Mean ΔT (LMTD) (°C)	Heat Transfer Coefficient (U)	Mean Temperature Efficiency (%)	CEB
1	30.76099935	0.124765766	13.16526611	0.988142836
2	30.25838474	0.125952757	14.42577031	0.958904519
3	28.89960633	0.126303686	16.76136364	0.965351484
4	26.79939043	0.120887421	20.89337176	0.977081714
5	20.00078382	0.117582701	35.53008596	0.969497599

Table 11 - Experiment 1, Air Results

Air Results							
	Δ Temp (°C)	Average Temp (°C)	Density of Air (kg/m³)	Mass Flowrate (kg/s)	Spec. Heat Capacity (J/Kg/K)	Heat Absorbed (W)	Temperature Efficiency (%)
1	5.5	27.95	1.203101672	0.106910021	1005	590.9451389	15.40616246
2	5.3	27.85	1.202698014	0.106910021	1005	569.4562248	14.84593838
3	5.1	27.85	1.202294627	0.107124484	1005	549.066544	14.48863636
4	4.6	27.5	1.203101672	0.106695126	1005	493.2515669	13.25648415
5	3.3	26.95	1.201488665	0.107124484	1005	355.278352	9.455587393

Table 12 - Experiment 1, Water Results

Water Results							
	Δ Temp (°C)	Average Temp (°C)	Density of Water (kg/m³)	Mass Flowrate (kg/s)	Spec. Heat Capacity (J/Kg/K)	Absolute Heat Emitted (W) (-)	Temperature Efficiency (%)
1	-3.9	58.95	973.0833055	0.036652805	4183.652659	598.0361518	10.92436975
2	-5	58.4	973.4119445	0.028391182	4183.42036	593.8612382	14.00560224
3	-6.7	57.15	974.1538443	0.020294872	4182.908836	568.7737092	19.03409091
4	-9.9	54.95	975.4425973	0.012193032	4182.06389	504.8212037	28.53025937
5	-21.5	49.45	978.568367	0.004077368	4180.263105	366.4561445	61.60458453

Experiment 2 - The Effect of Varying Air Flow Rate

Table 13 - Experiment 2, Flow Results

	Flow Results			CEB
	Logarithmic Mean ΔT (LMTD) ($^{\circ}\text{C}$)	Heat Transfer Coefficient (U)	Mean Temperature Efficiency (%)	
1	20.00078382	117.6000588	35.53008596	0.969497599
2	19.13361202	111.4759945	35.43543544	0.956242335
3	19.30669924	106.1122638	34.65045593	0.85931187
4	19.75973426	99.92517581	35.41033435	0.704301873
5	20.541208	87.38130207	34.09785933	0.63306194
6	21.11989452	77.48342103	31.3253012	0.694368745

Table 14 - Experiment 2, Air Results

	Air Results						
	ΔT ($^{\circ}\text{C}$)	Average Temp ($^{\circ}\text{C}$)	Density of Air (kg/m^3)	Mass Flowrate (kg/s)	Spec. Heat Capacity ($\text{J}/\text{Kg}/\text{K}$)	Heat Absorbed (W)	Temperature Efficiency (%)
1	3.3	26.95	1.201488665	0.107124484	1005	355.278352	9.455587393
2	3.3	27.05	1.202698014	0.095815052	1005	317.7706184	9.90990991
3	3.3	27.25	1.201891511	0.082701212	1005	274.2785694	10.03039514
4	4.5	28.15	1.203505601	0.047907526	1005	216.6617853	13.67781155
5	5.2	29	1.202294627	0.033875736	1005	177.0345981	15.90214067
6	5.2	29	1.202294627	0.033875736	1005	177.0345981	15.6626506

Table 15 - Experiment 2, Water Results

	Water Results						
	ΔT ($^{\circ}\text{C}$)	Average Temp ($^{\circ}\text{C}$)	Density of Water (kg/m^3)	Mass Flowrate (kg/s)	Spec. Heat Capacity ($\text{J}/\text{Kg}/\text{K}$)	Heat Emitted (W)	Temperature Efficiency (%)
1	-21.5	49.45	978.568367	0.004077368	4180.263105	366.4561445	61.60458453
2	-20.3	48.55	979.0666058	0.003916266	4180.011775	332.3118072	60.96096096
3	-19.5	48.75	978.9562114	0.003915825	4180.066545	319.1839645	59.27051672
4	-18.8	49.4	978.5961456	0.003914385	4180.248815	307.6263085	57.14285714
5	-17.1	50.55	977.9543116	0.003911817	4180.587118	279.6481466	52.29357798
6	-15.6	51.8	977.2497515	0.003908999	4180.97747	254.9576137	46.98795181

Experiment 3 - Effect of Varying Temperature (Driving Force)

Table 16 - Experiment 3, Flow Results

Flow Results				
	Logarithmic Mean ΔT (LMTD) ($^{\circ}\text{C}$)	Heat Transfer Coefficient (U)	Mean Temperature Efficiency (%)	CEB
1	13.2275313	0.117959966	36.17021277	0.974164929
2	13.25011516	0.112186482	35.4978355	0.956171709
3	13.45007927	0.106586422	34.7826087	0.892957702
4	13.79186006	0.10010145	33.98268398	0.835183661
5	14.24002931	0.088035395	32.68398268	0.788714856
6	14.60115306	0.077892851	31.22270742	0.686364992

Table 17 - Experiment 3, Air Results

	Air Results						
	Δ Temp (°C)	Average Temp (°C)	Density of Air (kg/m³)	Mass Flowrate (kg/s)	Spec. Heat Capacity (J/Kg/K)	Heat Absorbed (W)	Temperature Efficiency (%)
1	2.2	26.4	1.203505601	0.107124484	1005	236.8522347	9.361702128
2	2.3	26.55	1.201086089	0.095815052	1005	221.4764916	9.956709957
3	2.4	26.7	1.203101672	0.082701212	1005	199.4753232	10.43478261
4	2.6	26.9	1.203505601	0.068760235	1005	179.670494	11.25541126
5	3.2	27.4	1.203505601	0.047907526	1005	154.0706029	13.85281385
6	3.5	27.95	1.203505601	0.034546608	1006	121.6386072	15.28384279

Table 18 - Experiment 3, Water Results

	Water Results						
	Δ Temp (°C)	Average Temp (°C)	Density of Water (kg/m³)	Mass Flowrate (kg/s)	Spec. Heat Capacity (J/Kg/K)	Heat Emitted (W)	Temperature Efficiency (%)
1	-14.8	41.4	982.889428	0.003931558	4178.482867	243.1336088	62.9787234
2	-14.1	41.45	982.8635417	0.003931454	4178.490471	231.6283672	61.03896104
3	-13.6	41.7	982.7339288	0.003930936	4178.529187	223.3872028	59.13043478
4	-13.1	42.15	982.4998664	0.003929999	4178.601786	215.1269265	56.70995671
5	-11.9	42.95	982.0813516	0.003928325	4178.739934	195.343858	51.51515152
6	-10.8	43.7	981.6862092	0.003926745	4178.879781	177.2214617	47.16157205

Analysis

This section will examine all findings for each test procedure (as described in methodology and illustrated in results). All values for power are considered as absolute for comparative measures. These otherwise negative values are incorporated only as part of the sign convention within thermodynamics to indicate a transfer of energy is occurring.

Experiment 1 - The Effect of Varying Water Flow Rate

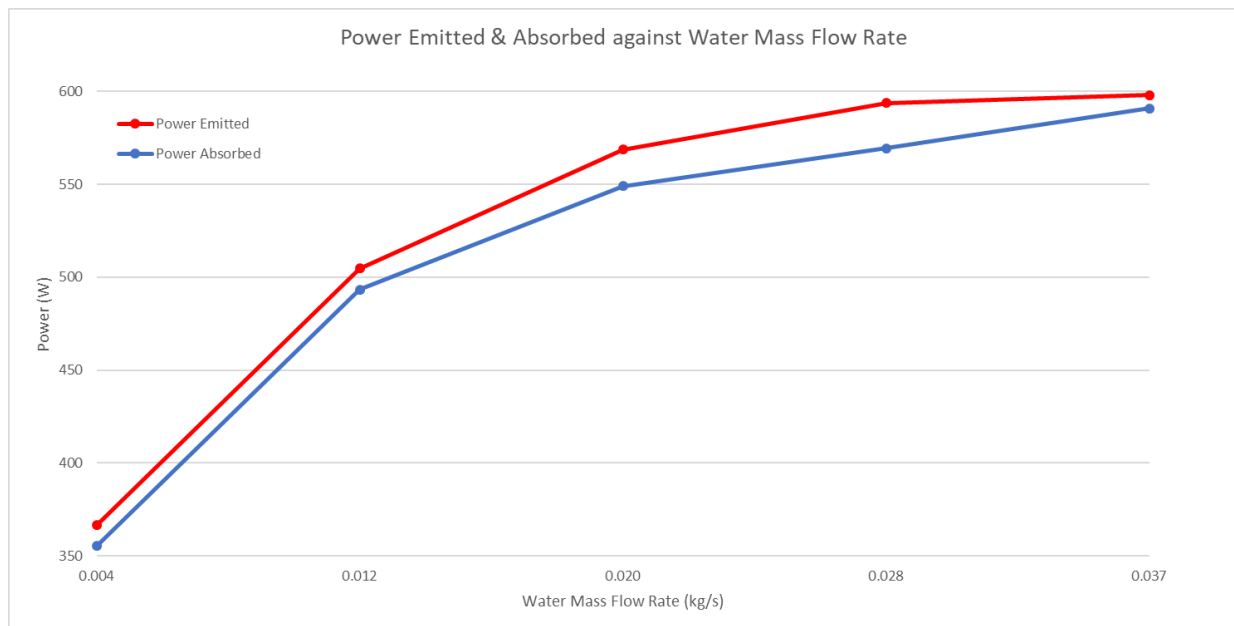


Figure 6 - Experiment 1, Power Emitted & Absorbed

Figure 6 demonstrates an increase in heat transfer with water mass flow rate. The percentage difference of the region between the power emitted and power absorbed is 2.91%. This indicates some energy loss to surroundings, as expected with any non-ideal system. Some potential errors may also be caused by inaccurate thermocouples. The maximum heat transfer rate is expected to occur with an increased water mass flow rate, as the power emitted/absorbed both begin to plateau. This theory is further supported by the decreasing mean temperature efficiency in figure 8.

Figure 8 also illustrates a rapid decrease in water temperature efficiency as the mass flow rate of water is increased. This is due to an increase in power emitted from the water circuit as more heat can be transferred to the cooler air within the same time interval. The increase in air temperature

efficiency is considerably smaller than the decrease in water temperature efficiency as the air mass flow rate remains constant whilst the water mass flow rate increases.

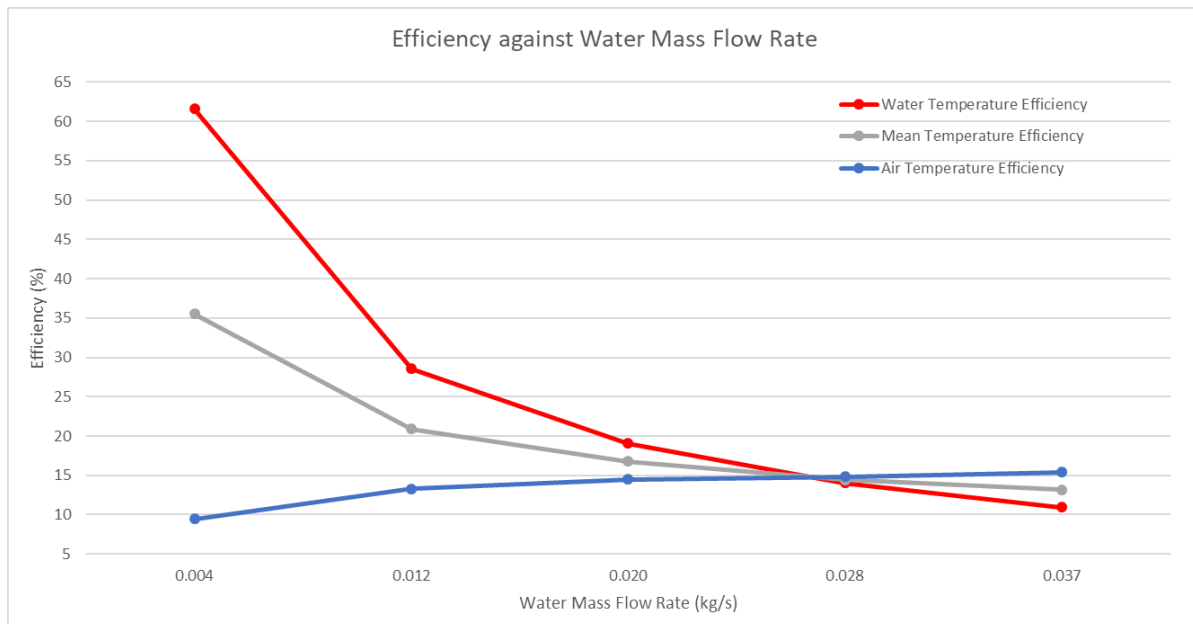


Figure 8 - Experiment 1, Efficiencies

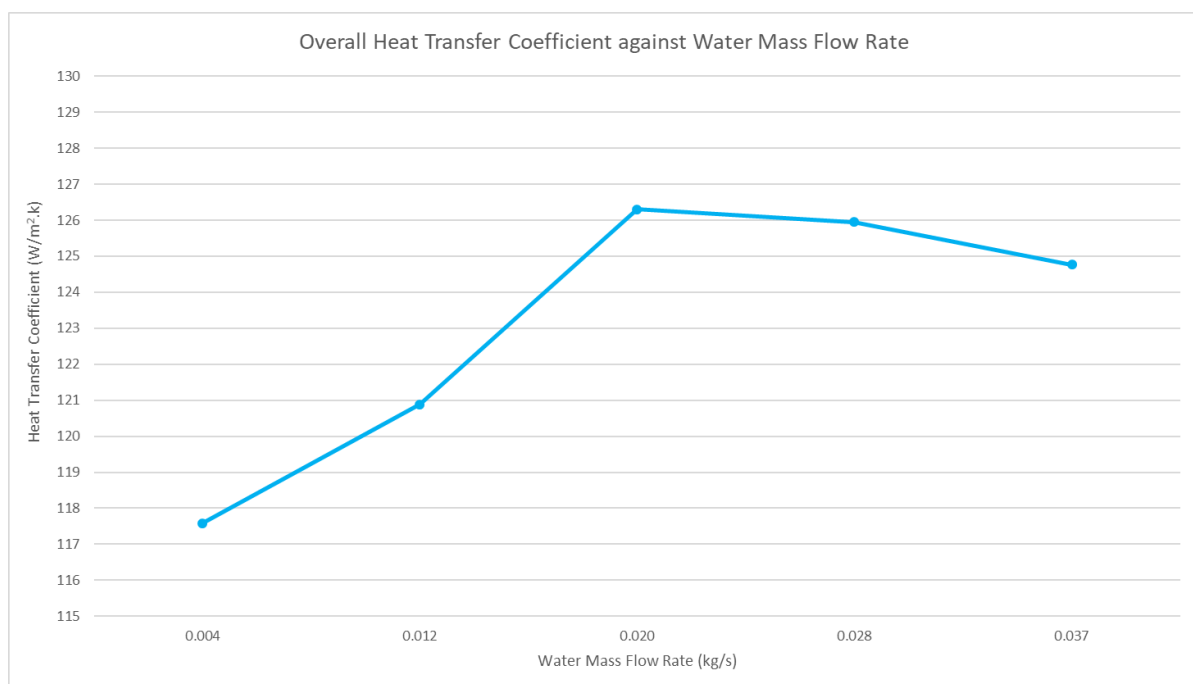


Figure 7 - Experiment 1, Heat Transfer Coefficient

The curve in *figure 7* shows a general increase in heat transfer coefficient as the water mass flow rate is increased. Initial results were used to calculate the overall heat transfer coefficients, with any small variations being exaggerated through the application of *equation 12*. It would be recommended to repeat this experiment to gain improved accuracy as small errors are present.

Experiment 2 - The Effect of Varying Air Flow Rate

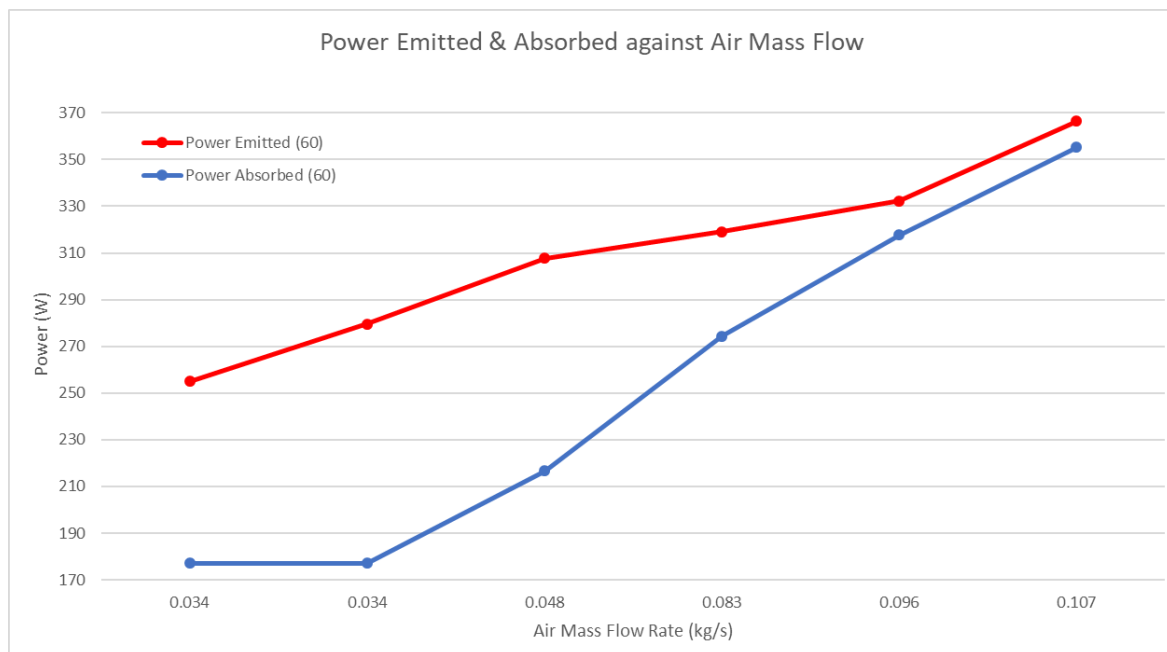


Figure 9 - Experiment 2, Power Emitted & Absorbed

Figure 9 shows an increase in power absorbed with an increased air mass flow rate in a linear fashion. Large difference between emitted and absorbed power at low air mass flow rate indicates much greater heat losses are present in the system. As the air mass flow rate is increased, the losses are reduced. It is expected that the heat transfer would increase further at greater air mass flow rates.

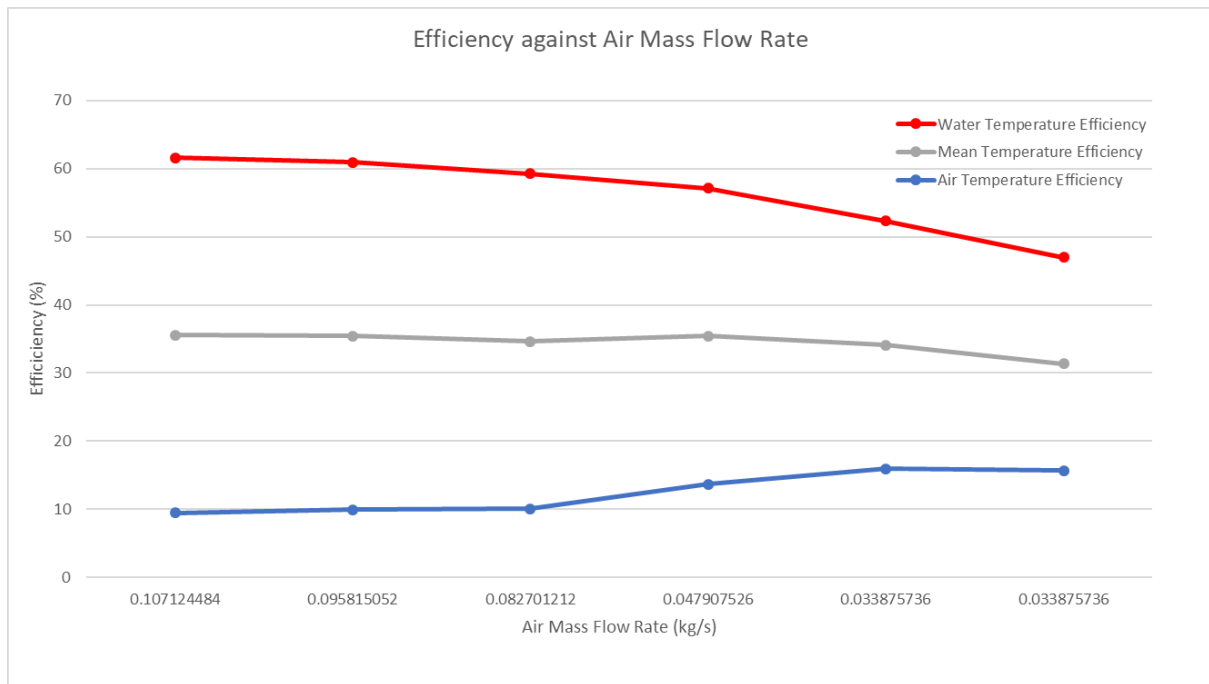


Figure 10 - Experiment 2, Efficiencies

The mean temperature efficiency displayed in *figure 10* suggests a stable efficiency of approximately 35% is achieved from an air mass flow rate of 0.083kg/s onwards. It is evident that a larger mean efficiency can be maintain by increasing the air mass flow rate over the water mass flow rate.

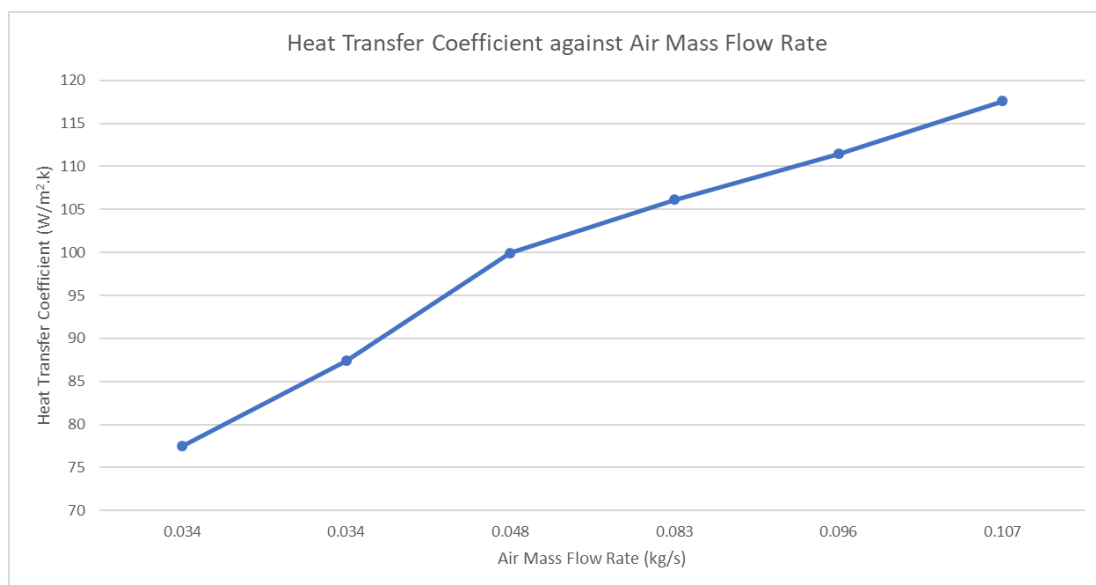


Figure 11 - Experiment 2, Heat Transfer Coefficient

The trend in *figure 11* indicates an increase in overall heat transfer coefficient per increase of air mass flow rate. This trend is linear at the highest tested values which indicates a predicted further increase if extrapolated.

Experiment 3 - Effect of Varying Temperature (Driving Force)

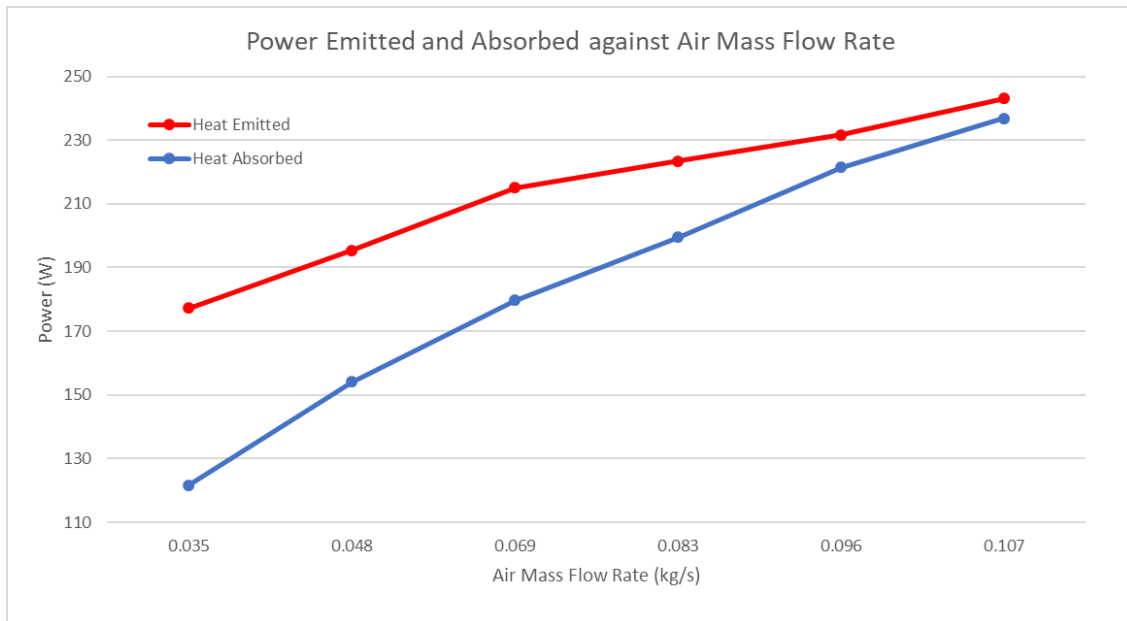


Figure 12 - Experiment 3, Power Emitted & Absorbed

By varying the initial temperature to a reduced 45°C, whilst maintaining the air mass flow rate increases from experiment 2, it can be seen in *figure 12* that a similar trend occurs as previously, except with a reduction in losses to surroundings at a lower flow rate. This indicates a possible link to losses from initial temperatures. This appears logical as an increased water temperature would provide an increased temperature differential with the system surroundings, thus promoting further

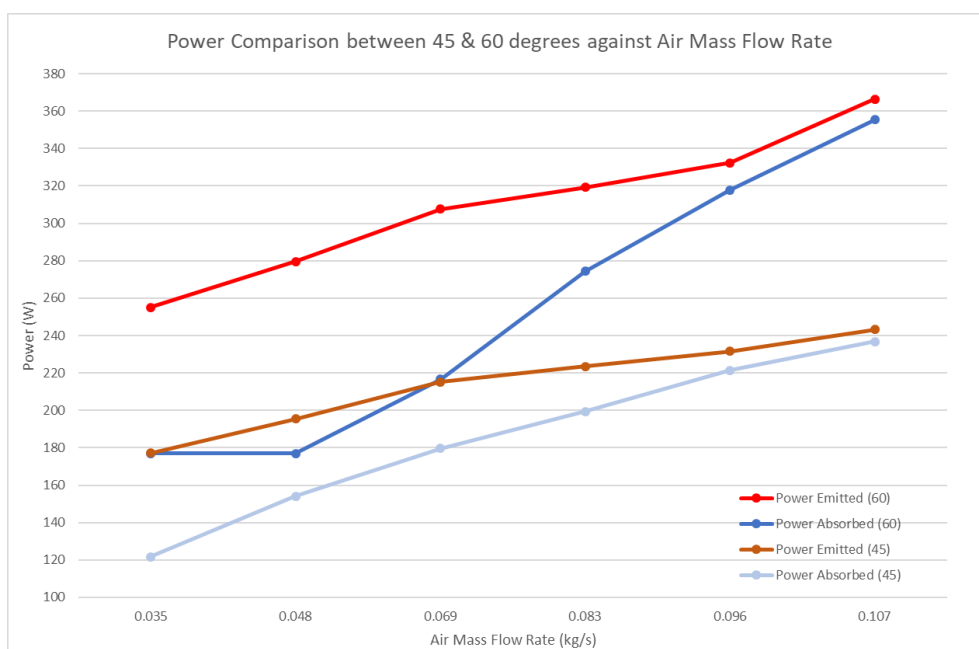


Figure 13 - Experiment 3, Comparison of Different Initial Water Temperatures

heat transfer. When compared directly per *figure 13*, the overall Power is reduced as expected but the trends are consistent.

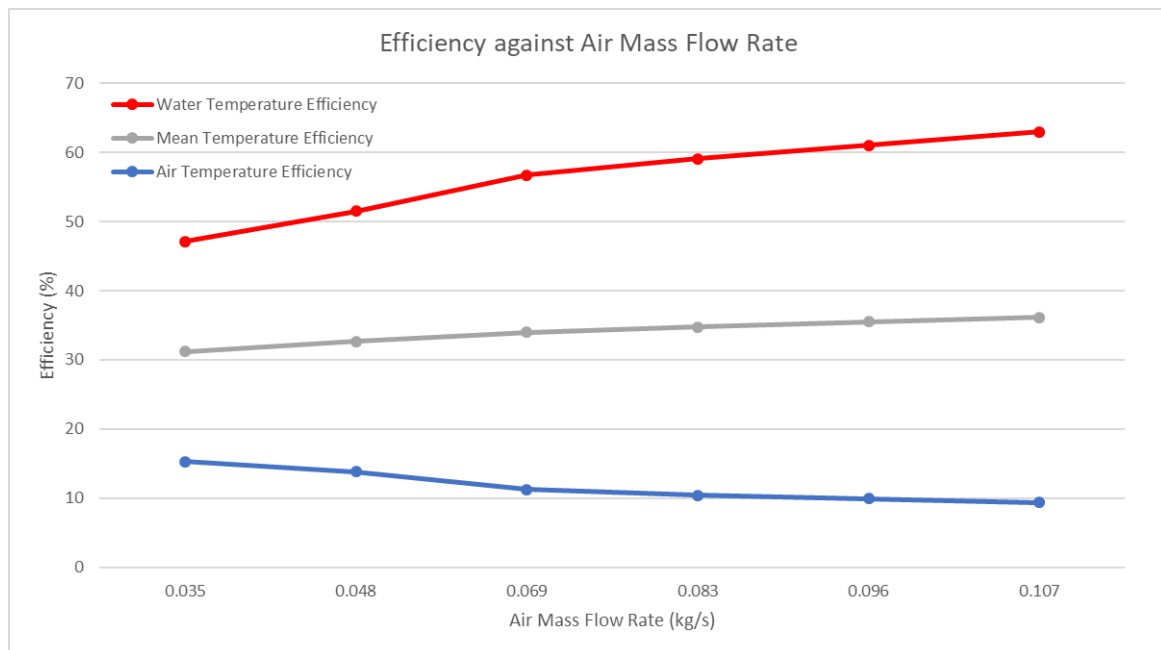


Figure 14 - Experiment 3, Efficiencies

The efficiency of the system is not affected by the change in initial water temperature, as illustrated in *figure 14*. A similar heat transfer coefficient curve is also produced from a reduction in water temperature *figure 15*.

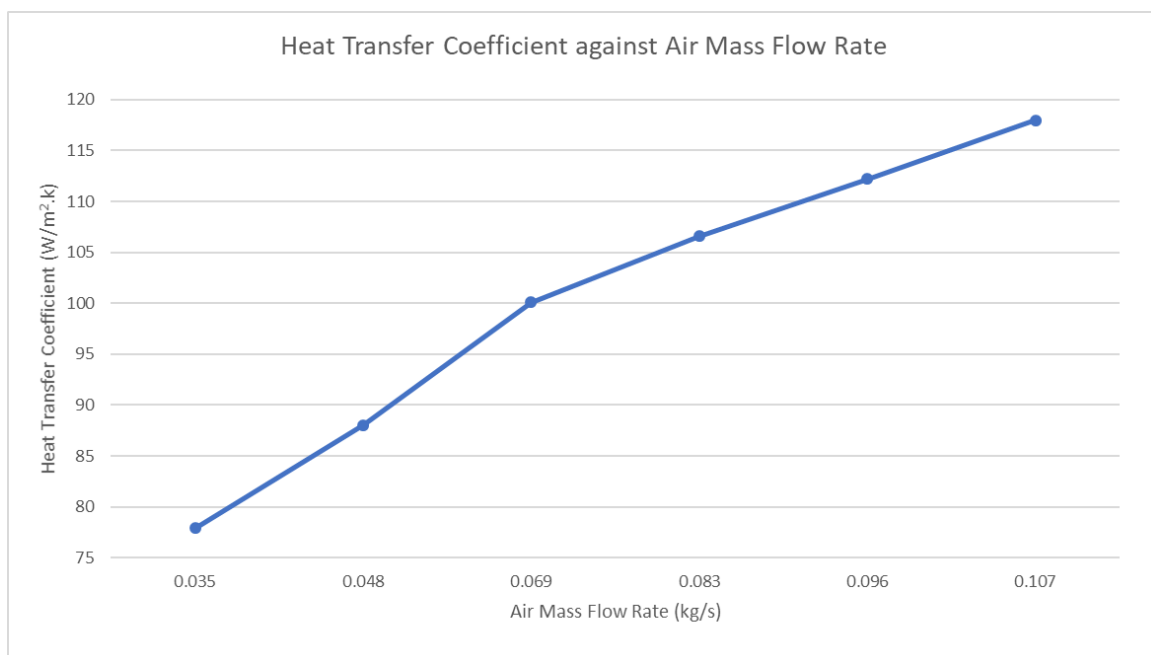


Figure 15 - Experiment 3, Heat Transfer Coefficient

Further Improvements

Some potential measuring errors occurred through the three test procedures. These errors could be mitigated by calibrating the thermocouples, repeating the entire experimental procedure and increasing the equilibrium time given for each reading.

An additional investigation could be carried out to account for the losses observed in each power emitted/absorbed graph. It is understood that a portion of these losses are transferred into the system surroundings, however, an exact measurement of this would be beneficial to demonstrate all modes of heat transfer taking place.

Conclusion

The experiments confirm the fundamental theories outlined previously, following conventional thermodynamic principles. An increase in water mass flow rate results in peak in power being reached at around 0.04kg/s, whereas an increase in air mass flow rate at a constant water rate produces a linear rise in power with no obvious limit observed. A higher initial water temperature improves the magnitude of power; however, this has no effect on the overall system efficiency or heat transfer coefficient. The greatest loss to surroundings (or potential measuring errors) occur with varying air mass flow rates at a higher initial water temperature.

The maximum heat transfer from all experiments is achieved at the highest water mass flow rate. This is most likely due to a turbulent flow regime within the tube bundles, causing an agitated flow. The heat transfer appears to be reduced during lower water mass flow rates (laminar flow). This is a result of the build-up of a boundary layer on the outermost region of the pipe. This boundary layer experiences skin friction and acts as an insulating blanket for the faster moving inner section, thus reducing heat transfer (Beckers, 1956).

The results are in line with those expected from a theoretical multi-pass cross-flow heat exchanger (Luben Cabezas-Gómez, 2006).

References

- Beckers, H. L., 1956. *Heat Transfer in Turbulent Tube Flow*. s.l.:Springer Netherlands.
- Cengel, Y. A., 2002. *Heat Transfer: A Practical Approach*. 2nd ed. Nevada: McGraw-Hill Higher Education.
- Frank P. Incropera, D. P. D. T. L. B. A. S. L., 2006. *Introduction to Heat Transfer*. 5th ed. s.l.:John Wiley & Sons Inc.
- Hucho, K. D. E. a. W.-H., 1974. *A Rational Approach To Automotive Radiator Systems Design*, Germany: SAE International.
- John H Lienhard IV, J. H. L. V., 2003. *A Heat Transfer Textbook*. 3rd ed. Cambridge, Massachusetts: Phlogiston Press.
- Kay, J. M., 1985. *Fluid Mechanics and Transfer Processes*. Cambridge: Cambridge University Press.
- Luben Cabezas-Gómez, H. A. N. a. J. M. S.-J., 2006. Thermal Performance of Multipass Parallel and Counter-Cross-Flow Heat Exchangers. *Journal of Heat Transfer*, 129(3), pp. 282-290.
- Rodriguez, P., 1997. *Selection of Materials for Heat Exchangers*, Kalpakkam: Indira Gandhi Center for Atomic Research.
- Stoakes, G., 2011. *Level 2 Principles of Light Vehicle Maintenance and Repair Candidate Handbook (Light Vehicle Technology)*. s.l.:Heinemann.
- Tecquipment, 2014. *TD1007 Water to Heat Exchanger User Guide*. s.l.:Tecquipment Ltd.
- Thulukkanam, K., 2000. *Heat Exchanger Design Handbook*. s.l.:CRC Press.
- Tritt, T. M., 2004. *Thermal Conductivity: Theory, Properties, and Applications (Physics of Solids and Liquids)*. s.l.:Springer US.

Appendix

Equations

Water Density

Equation 2 - Water Density

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

Air Density

Equation 3 - Air Density

$$\rho_a = \frac{P}{RT}$$

Specific Heat Capacity of Water

Equation 4 - Specific Heat Capacity of Water

$$C_{pW} = 0.996185 + 0.0002874 \left(\frac{T + 100}{100} \right)^{5.26} + (0.011160 \times 10^{-0.036T}) \times C_{pW@15^\circ}$$

Mass Flow Rate

Equation 5 - Mass Flow Rate for Water

$$\dot{m}_W = \frac{\text{Vol Flow (Lmin)}}{60000} \times \rho$$

Equation 6 - Mass Flow Rate for Air

$$\dot{m}_a = C_d \times A_o \sqrt{\Delta P \times 2 \times \rho_a}$$

Power Emitted/Absorbed

Equation 7 - Power Emitted/Absorbed

$$Q = \dot{m} \times \dot{C}_p \times \Delta T$$

Efficiency

Equation 8 - Efficiency of Water Circuit

$$\eta_w = \frac{T_{w1} - T_{w2}}{T_{w1} - T_{a1}} \times 100$$

Equation 9 - Efficiency of Air Circuit

$$\eta_a = \frac{T_{a2} - T_{a1}}{T_{w1} - T_{a1}} \times 100$$

Equation 10 - Mean Temperature Efficiency

$$\bar{\eta} = \frac{\eta_w + \eta_a}{2}$$

Heat Transfer Coefficient

Equation 11 - Logarithmic Mean Temperature Difference

$$LMTD = \frac{(T_{w2} - T_{a2}) - (T_{w1} - T_{a1})}{\ln\left(\frac{T_{w2} - T_{a2}}{T_{w1} - T_{a1}}\right)}$$

Equation 12 - Heat Transfer Coefficient

$$U = \frac{\dot{Q}_w}{A \times LMTD}$$

Nomenclature

Symbol	Definition	Units
A	Heat Transfer Area	m^2
A_O	Orifice Area	m^2
C	Heat capacity	$J.K^{-1}$
C_d	Coefficient of discharge (orifice)	-
c_p and c_{pA} and c_{pW}	Specific Heat Capacity, of air and water at Constant Pressure (atmospheric pressure for air)	$J.kg^{-1}K^{-1}$
$LMTD$	Logarithmic Mean Temperature Difference	K or °C where shown
ρ and ρ_A and ρ_W	Density, of air and water	$kg.m^{-3}$
η_o	Overall Efficiency	%
C_{EB}	Energy Balance Coefficient	-
$\bar{\eta}_A$ and $\bar{\eta}_W$	Mean Temperature Efficiencies	%
Δp	Orifice pressure drop	Pa
Q	Heat energy transferred for a unit mass	$J.kg^{-1}$
$\dot{Q}$	Heat energy transferred for a unit time (heat transfer rate - or power)	W
$\dot{Q}_A$ and $\dot{Q}_W$	Heat energy (power) absorbed by the air and emitted by the water in a unit time	W
R	Specific gas constant for dry air	287 J/kg.K
T, T_W, T_A	Temperature, of water and air	K or °C where shown
ΔT	Change in Temperature	K or °C where shown
$\bar{T}$	Mean Temperature	K or °C where shown
U	Heat Transfer Coefficient	$W.m^{-2}K^{-1}$
$\dot{m}_W$ and $\dot{m}_A$	Mass Flow Rate (hot water circuit and air circuit)	$kg.s^{-1}$

Figure 16 - Nomenclature Table (Tecquipment, 2014)