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Heat Exchanger Laboratory Analysis

Thermofluid Mechanics

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1 Introduction

Water to air heat exchangers transfer thermal energy between a circuit of liquid and a stream of air, without mixing of the two fluids. Modern infrastructure relies heavily on this technology in applications such as Heating, Ventilation, and Air Conditioning (HVAC), vehicle radiators, and greenhouses.

Generally, hot or chilled water flows through a circuit of pipes, arranged perpendicular to a stream of air. The surface of the pipes acts as an interface for both fluids to exchange energy through convection.

Two key variables govern performance, the water flow rate, and the air flow rate. This report investigates both variables through TecQuipment's TD1007 Water to Air Heat Exchanger experiments.

Experiment 1 exhibits a static airflow rate, with an orifice differential pressure of 250 Pascals (Pa), and varies the water flow rate, ranging from 0.2 to 1.6 Liters per minute (L/min). Experiment 2 fixes the water flow rate at 0.2 L/min, and varies the airflow rate, ranging from 25 to 250 Pa. Water heater temperature remains at a constant 60 degrees Celsius for both experiments. For each test point, water side and air side heat transfer rates, temperature efficiencies, logarithmic mean temperature difference (LMTD), and the overall heat transfer coefficient (U) are calculated and measured against theoretical expectations.

2 Theory

2.1 Modes of Heat Transfer

Heat transfer occurs primarily via three fundamental mechanisms. Conduction is the transfer of energy between any stationary medium, primarily evaluated by Fourier's law. In the TD1007, conduction occurs across the copper tube walls. Convection is the transfer of energy between a solid and a fluid and is the primary method of energy transfer between the fluid within the pipes, and the fluid flowing over the pipes. All matter above 0 Kelvin radiates energy, however, the amount of energy radiated by the pipes is negligible compared to convection, due to the low emissivity of copper.

2.2 Mass Flow Rates

The water volumetric flow is converted to mass flow rate:

$$\dot{m}_W = \frac{Flow_W}{60000} \cdot \rho_W(\bar{T}_W) [kg s^{-1}]$$

where ρ_W is evaluated at the mean water temperature $\bar{T}_W = \frac{T_{W1} + T_{W2}}{2}$, using the Kell (1975) polynomial, accurate to $\pm 0.003 \text{ kg/m}^3$ over $0-70^\circ\text{C}$.

The air mass flow rate through the orifice is:

$$\dot{m}_A = Cd \cdot A_O \cdot \sqrt{(2 \cdot \Delta P_A \cdot \rho_A)} [kg s^{-1}]$$

where $Cd = 0.6$, $A_O = \pi/4 \cdot (0.10011)^2 = 7.871 \times 10^{-3} \text{ m}^2$, and air density $\rho_A = P/(R \cdot T_{A3})$ are computed at the orifice inlet temperature T_{A3} , and assumed atmospheric pressure $P = 101,325 \text{ Pa}$.

2.3 Heat Transfer Rates

The rate of heat energy emitted by the hot water circuit is:

$$\dot{Q}_W = \dot{m}_W \cdot Cp_W(\bar{T}_W) \cdot \Delta T_W [W] \text{ (heat emission)}$$

where Cp_W is the temperature dependent specific heat capacity of water, and $\Delta T_W = T_{W2} - T_{W1}$. The heat absorbed by the air is:

$$\dot{Q}_A = \dot{m}_A \cdot Cp_A \cdot \Delta T_A [W] \text{ (heat absorption)}$$

where $Cp_A = 1005 \text{ J kg}^{-1} \text{ K}^{-1}$ and $\Delta T_A = T_{A2} - T_{A1}$. In an ideal heat exchanger $\dot{Q}_W = -\dot{Q}_A$. In practice, heat losses to the surroundings always result in a slight inaccuracy.

2.4 Temperature Efficiency

The efficiency of each circuit is the ratio of its actual temperature change to the maximum temperature change possible.

$$\eta_W = \frac{|\Delta T_W|}{T_{W1} - T_{A1}} \cdot 100 \text{ [%]}$$

$$\eta_A = \frac{|\Delta T_A|}{T_{W1} - T_{A1}} \cdot 100 \text{ [%]}$$

The mean temperature efficiency $\bar{\eta} = \frac{\eta_W + \eta_A}{2}$ provides an overall performance metric.

2.5 Logarithmic Mean Temperature Difference (LMTD)

For a parallel flow heat exchanger, the temperature driving force varies along the length of the copper pipes. The LMTD provides the effective mean driving force:

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln \left(\frac{\Delta T_1}{\Delta T_2} \right)}$$

where $\Delta T_1 = T_{W1} - T_{A1}$ and $\Delta T_2 = T_{W2} - T_{A2}$. Note that parallel flow LMTD is always lower than counter-flows, highlighting the thermodynamic limitations of parallel flow, the cold end temperature difference is reduced as both streams reach equilibrium.

2.6 Overall Heat Transfer Coefficient (U)

The overall heat transfer coefficient U combines the resistances of both the convective boundary layers and the tube walls:

$$U = \frac{\dot{Q}_W}{A_{HX \cdot LMTD}} [W \cdot m^{-2} \cdot K^{-1}]$$

where $A_{HX} = 0.1558 \text{ m}^2$ is the mean heat transfer area of the 32 tube heat exchanger. As $\dot{Q}_W$ is negative, U is also negative in sign. The magnitude $|U|$ is used for comparison, a higher $|U|$ value indicates a more effective heat exchanger for the given operating conditions. U is expected to increase with mass flow rates, as high velocities reduce the convective boundary layer resistance.

3 Apparatus

The TecQuipment TD1007 is a compact bench mounted unit comprised of a water heater circuit, an air duct system, and instruments for measuring relevant variables.

The hot water circuit consists of an electrically heated insulated tank, adjustable up to 60 °C, controlled to ± 2 °C. A centrifugal pump, and a hand-operated valve that regulates the water flow rate through the heat exchanger and insulated tank. Flow is measured by a turbine flow sensor with a digital display, with a resolution of 0.01 L/min. The water inlet and outlet temperatures, T_{W1} & T_{W2} , are measured by K-Type thermocouples, mounted to pipe connectors at the heat exchanger.

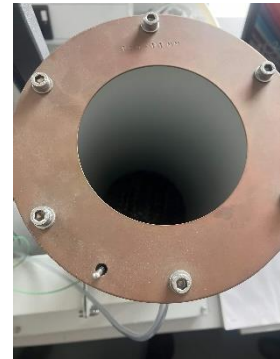


Figure 1, Inlet Duct Orifice

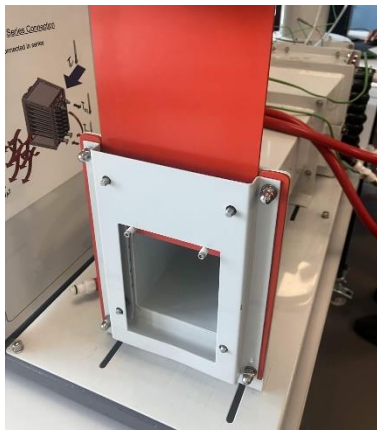


Figure 2, Duct Exit Slide Valve

The air circuit uses an electric centrifugal fan to draw air vertically down a circular inlet duct, turn it 90°, and force it along a horizontal rectangular duct, through the heat exchanger. Airflow is regulated by a slide valve at the duct exit. A calibrated circular orifice plate (Figure 1), with a measured diameter of 100.11mm, is located at the top of the inlet duct. The pressure differential ΔP_A across it is measured by a dedicated pressure sensor and displayed digitally. The air temperature is measured at three points, T_{A1} is upstream of the heat exchanger, T_{A2} is downstream of the heat exchanger, and T_{A3} measures ambient temperature outside of the duct, all via K-type thermocouples, with an accuracy of ± 0.1 °C as per the manufacturer.

The standard 32 tube heat exchanger consists of two bundles of 16 copper pipes. In the parallel configuration used for both experiments, hot water enters all 32 pipes simultaneously, and from the same end. Air passes across the bundles, creating a cross flow. The apparatus constants used in all calculations can be found in Table 1 below.

Table 1, Apparatus Constant and Fixed Parameters

Parameter	Value
Ambient Pressure, P	101,325 Pa
Orifice Diameter, d	100.11 mm
Orifice Area, A _O	$7.871 \times 10^{-3} \text{ m}^2$
Discharge Coefficient, C _d	0.6
Heat Transfer Area, A _{HX}	0.1558 m^2
Specific Heat of Air, C _{pA}	$1005 \text{ J kg}^{-1} \text{ K}^{-1}$
Specific Gas Constant, R	$287 \text{ J kg}^{-1} \text{ K}^{-1}$

4 Procedure

4.1 Experiment 1, Effect of Varying Water Flow Rate

The 32-tube heat exchanger was connected in a parallel flow configuration. The heater setpoint was set to 60 °C and the system was allowed to reach thermal equilibrium. The orifice pressure drop was set to 250 Pa $\pm$ 5 Pa, using the slide valve, fixing the airflow for the duration of the experiment. Standard atmospheric pressure was assumed at 101,325 Pa and used in all calculations. The following steps were then repeated for each water flow setting:

- Set the Water Supply valve to achieve the target flow rate.
- Wait until thermal equilibrium is reached.
- Record T_{W1} , T_{W2} , T_{A1} , T_{A2} , T_{A3} , water flow, and ΔP_A .

Water flow rates tested: 1.6, 1.4, 1.0, 0.6 and 0.2 L/min. Although the procedure specifies a maximum of 2.25 L/min, the pump was only able to maintain a maximum flow rate of 1.6 L/min, as such, tested flow rates were amended, and laboratory limitations were noted.

4.2 Experiment 2, Effect of Varying Air Flow Rate

The parallel flow setup and 60 °C heater setpoint were retained. The water flow rate was fixed at 0.2 L/min $\pm$ 0.01 L/min using the water supply valve. Airflow was adjusted by repositioning the slide valve to achieve each target pressure differential. Steps 2-4 from Experiment 1 were repeated for each airflow setting.

Pressure differentials tested: 250, 200, 150, 100, 50 & 25 Pa.

5 Results and Analysis

5.1 Experiment 1

5.1.1 Calculated Results Table

Table 2 presents the calculated results from Experiment 1.

Table 2, Experiment 1 Results

HF (L/min)	$\dot{m}_W$ (kg/s)	$\dot{Q}_W$ (W)	$\dot{Q}_A$ (W)	η_W (%)	η_A (%)	η (%)	C_{EB}	LMTD (°C)	U (W/m ² K)
1.6	0.02621	-679.9	625.3	16.4	14.3	15.4	-0.920	31.55	-138.3
1.4	0.02294	-652.7	625.3	18.2	14.4	16.3	-0.958	30.90	-135.6
1.0	0.01640	-596.9	567.5	23.9	13.5	18.7	-0.951	29.07	-131.8
0.6	0.00985	-515.1	521.2	34.4	12.4	23.4	-1.012	26.91	-122.9
0.2	0.00330	-341.7	347.5	68.7	8.3	38.5	-1.017	18.91	-116.0

5.1.2 Chart Analysis

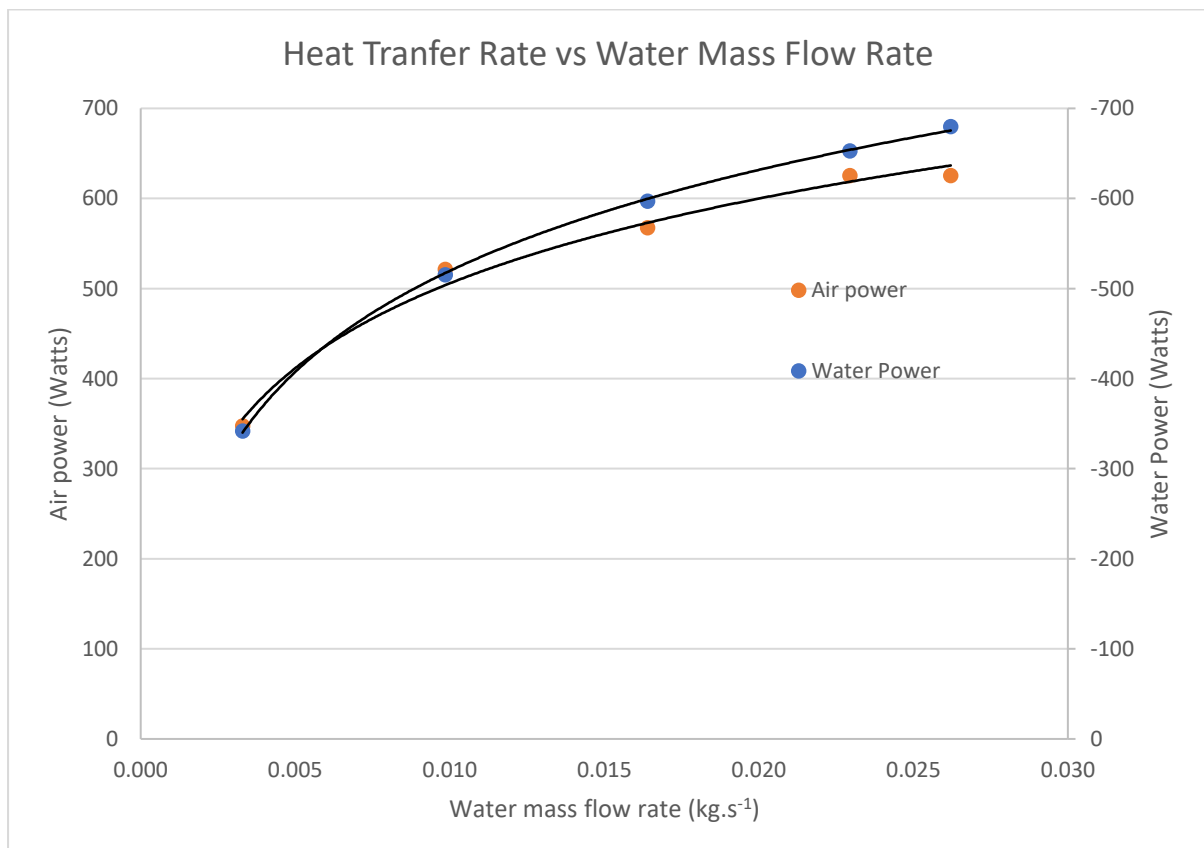


Figure 3, Heat Transfer Rate vs Water Mass Flow Rate

In Figure 3, $\dot{Q}_A$ and $|\dot{Q}_W|$ increase with water mass flow rate, but with diminishing returns, consistent with a power curve that approaches a plateau. At $\dot{m}_W = 0.2$ L/min, $|\dot{Q}_W| = 314.7$ W, increasing to

679.9W at $\dot{m}_w = 1.6$ L/min. The similarity of both $\dot{Q}_w$ and $\dot{Q}_A$'s inverse proportion indicate a good energy balance. The slight divergence towards higher water flow rates is attributable to heat losses from the water circuit and heat exchanger metalwork to the surrounding environment. At higher water flows the higher power throughput and mean temperatures of the pipe works results in a higher rate of parasitic heat loss. The levelling off trend suggests that increasing $\dot{m}_w$ would quickly result in a plateau of heat transfer rate.

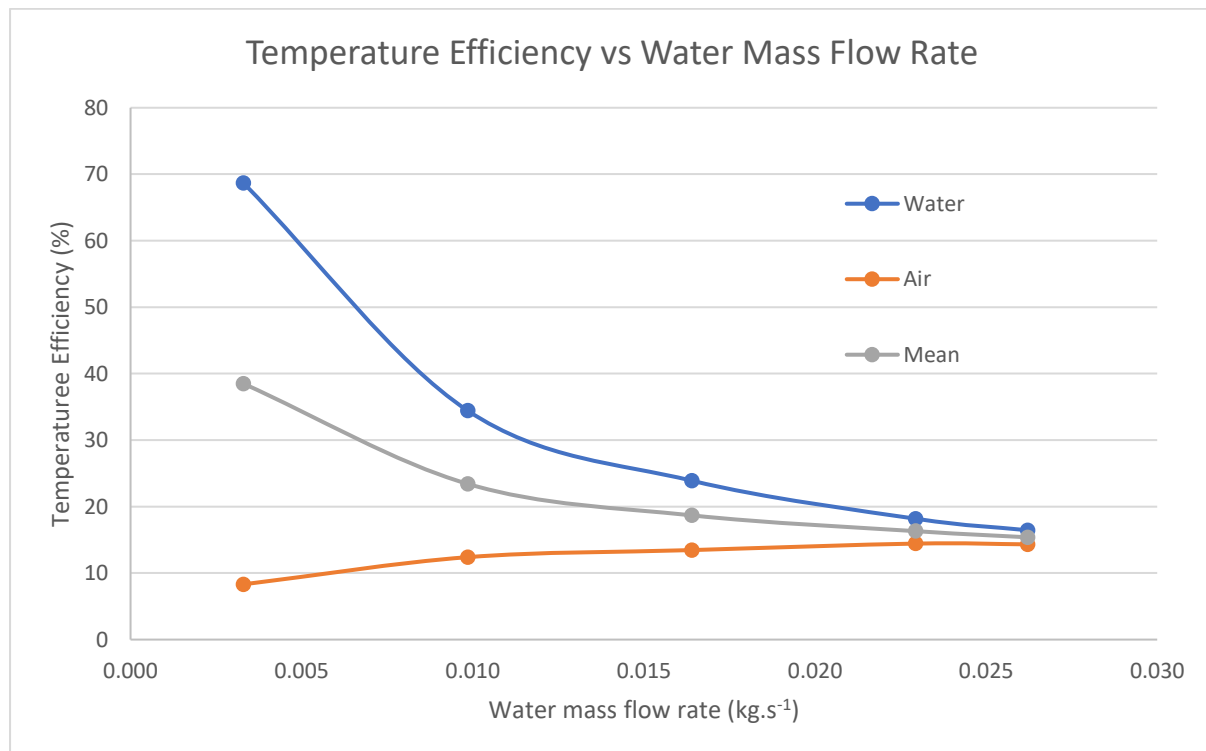


Figure 4, Temperature Efficiency vs Water Mass Flow Rate

As seen in Figure 4, water temperature efficiency η_w decreases with increasing $\dot{m}_w$, from 68.7% at 0.2 L/min to 16.45% AT 1.6 L/min. At lower flow rates, the water spends more time in contact with the heat exchange pipes and therefore undergoes a larger fractional temperature drop.

Air efficiency η_A shows an opposing trend, rising from 8.31% at 0.2 L/min to 14.44% at 1.4 L/min. It is of note that the efficiency drops slightly at 1.6 L/min to 14.32%, this does not indicate deteriorating air efficiency. The ΔT_A is identical at both points (5.4 °C), indicating that $\dot{Q}_A$ plateaued at 1.4 L/min. The small decrease arises because the water inlet temperature rose slightly at the higher flow rate, increasing the denominator of the efficiency expression without a corresponding increase in ΔT_A .

The mean efficiency $\bar{\eta}$ decreases overall from 38.5% to 15.4%, primarily due to the large reduction in η_w outweighing the modest gains in η_A .

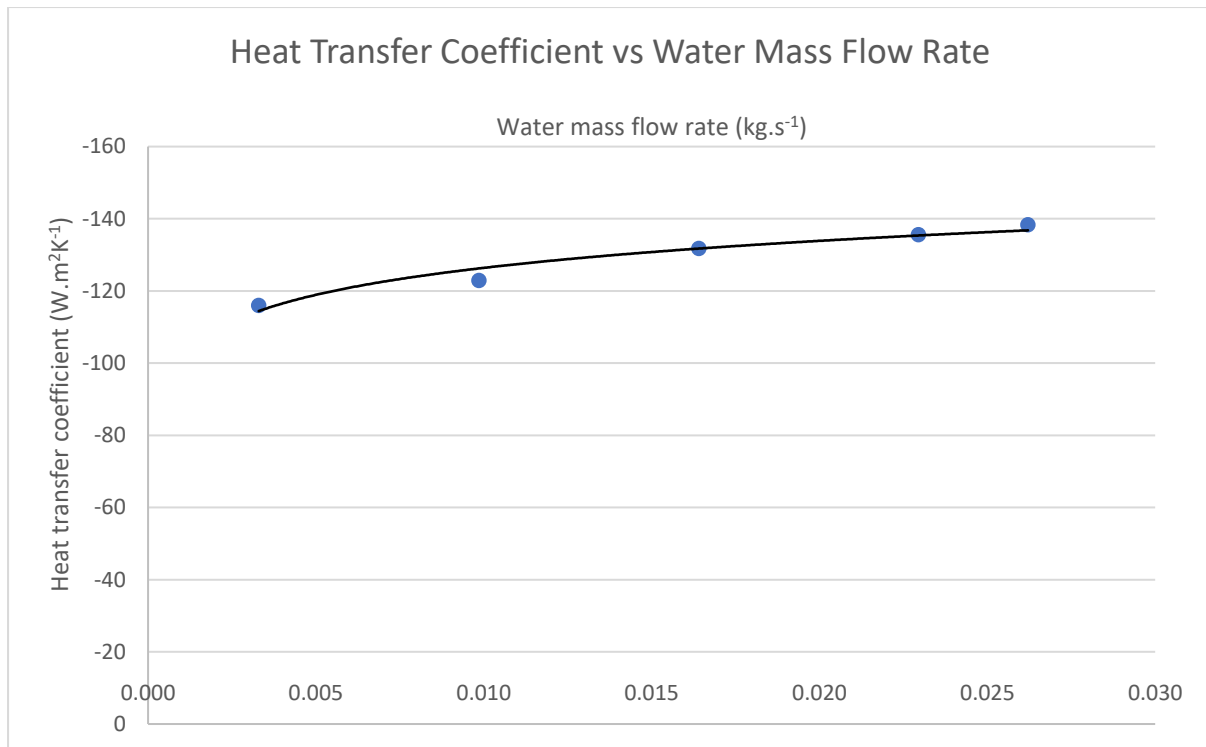


Figure 5, Overall Heat Transfer Coefficient vs Water Mass Flow Rate

Figure 5 shows how $|U|$ increases from 116 to 138 $\text{W m}^2 \text{K}^{-1}$ across the test range, but the rate of increase visibly diminishes with higher $\dot{m}_w$ values. The initial rise reflects improved convective heat transfer between the water and the pipes, due to the Reynolds number increasing and reducing the thermal boundary layer thickness. However, beyond a certain point external convective resistance dominates, and further increases in internal velocity produce smaller gains in heat transfer.

5.2 Experiment 2

5.2.1 Calculated Results Table

Table 3 presents the calculated results from Experiment 2.

Table 3, Experiment 2 Results

ΔP (Pa)	$\dot{m}_A$ (kg/s)	$\dot{Q}_W$ (W)	$\dot{Q}_A$ (W)	η_w (%)	η_A (%)	$\bar{\eta}$ (%)	C_{EB}	LMTD (°C)	U (W/m ² K)
250	0.11528	-312.9	336.0	66.6	8.5	37.6	-1.074	18.43	-109.0
200	0.10313	-310.2	310.9	66.2	8.8	37.5	-1.002	18.39	-108.2
150	0.08930	-303.2	269.2	64.9	8.8	36.9	-0.888	18.69	-104.1
100	0.07292	-293.5	249.2	62.6	10.0	36.3	-0.849	19.05	-98.9
50	0.05157	-274.1	212.5	58.7	12.1	35.4	-0.776	19.50	-90.2
25	0.03645	-254.7	175.8	54.6	14.2	34.4	-0.690	20.04	-81.6

5.2.2 Chart Analysis

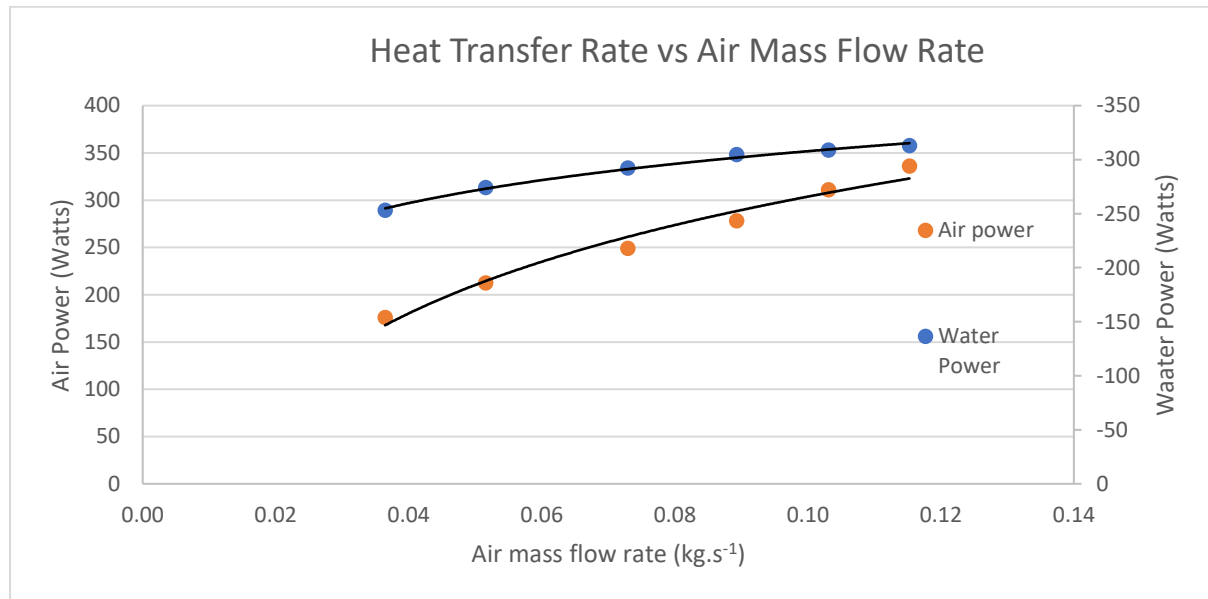


Figure 6, Heat Transfer Rate vs Air Mass Flow Rate

Figure 6 displays an increase in $\dot{Q}_A$ and $|\dot{Q}_W|$ of approximately linear fashion with air mass flow rate, in contrast to the diminishing returns observed in Experiment 1. $|\dot{Q}_W|$ increases from 254 W at $\dot{m}_A = 0.0365 \text{ kg s}^{-1}$ to 312.9 W at $\dot{m}_A = 0.1153 \text{ kg s}^{-1}$. Notably, the energy balance coefficient C_{EB} decreases significantly at low airflows ($C_{EB} = -0.690$ at $\Delta P = 25 \text{ Pa}$), improving to -1.074 at maximum airflow. At low $\dot{m}_A$ the air temperature rise per unit mass is very large, meaning the duct walls and machine casing absorb energy from the hot water that is not then transferred to the lower temperature air stream. As $\dot{m}_A$ increases, this loss fraction becomes relatively smaller. There is a slight trend towards a plateau, indicating that although heat transfer would continue to improve with increases in air flow rate, its efficacy would eventually stagnate.

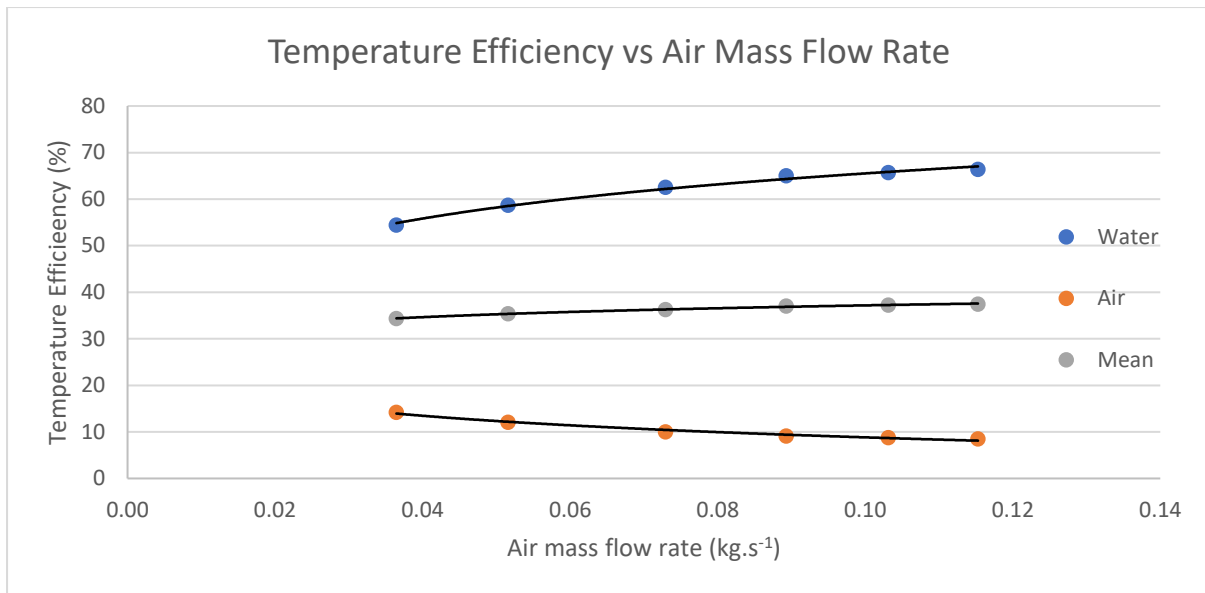


Figure 7, Temperature Efficiency vs Air Mass Flow Rate

As seen in Figure 7, η_w increases with increasing $\dot{m}_A$, from 54.6% to 66.6%, while air efficiency η_A decreases from 14.2% to 8.5%. As more air flows per second across the tubes, each unit mass of air picks up less heat, so efficiency drops. However, the increased external convection draws heat from the water more effectively, increasing the internal temperature drop, and thus η_w . The mean efficiency rises slightly, from 34.32% to 37.43%, highlighting the trade-off between air efficiency and water efficiency.

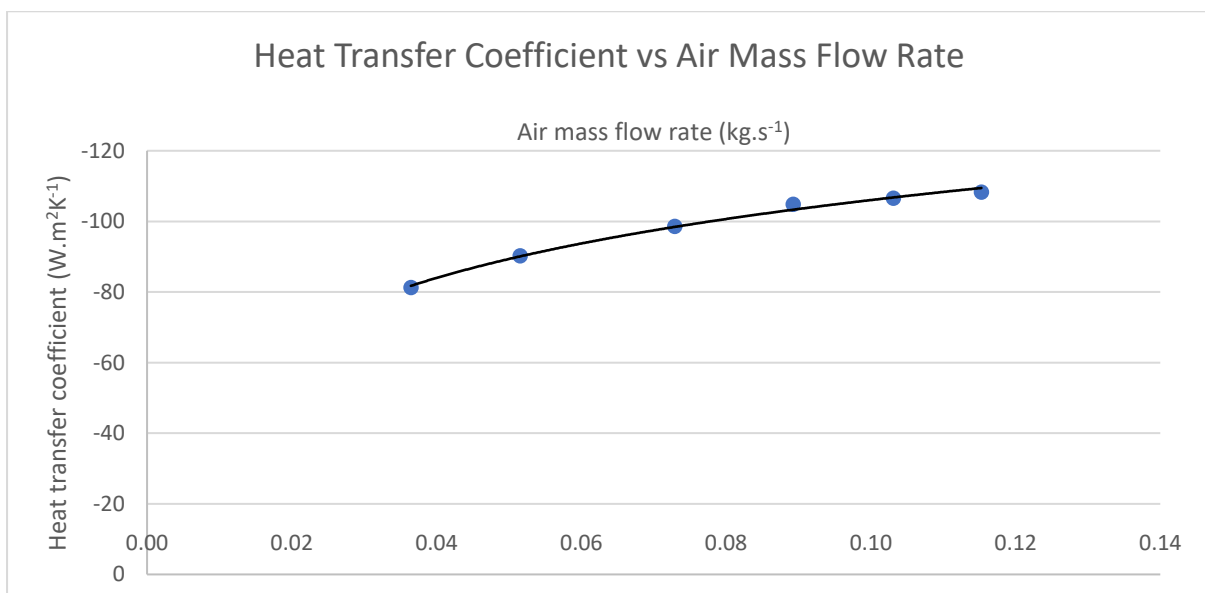


Figure 8, Overall Heat Transfer Coefficient vs Air Mass Flow Rate

Figure 8 shows an increase in $|U|$ with higher $\dot{m}_A$, with a slight diminishing in the curve. The gains per unit increase in $\dot{m}_A$ drop sharply after the first two data points. As both internal and external

resistances become comparable at higher airflows, the linear relationship between $|U|$ and $\dot{m}_A$ naturally degrades regardless of water flow rate.

6 Conclusions

6.1 Experimental Findings

In Experiment 1, as $\dot{m}_w$ increased from 0.2 to 1.6 L/min, $\dot{Q}_A$ increased from 347.5 W to 625.3 W, and $\dot{Q}_w$ decreased from -341.7 W to -679.9 W. The two series are inversely proportional, diverging from the x-axis in opposite directions. Both exhibit diminishing returns at higher flows, with the curves visibly flattening towards the higher $\dot{m}_w$ values as the external convective resistance became the controlling variable. The improvement in $|U|$ from 116 to 138 W/m²K reflects enhanced convection under thermally developing laminar conditions. The energy balance coefficient began close to -1 at lower water flows, and diverged progressively at higher flows, as the higher tube temperatures at greater flow rates drove larger parasitic heat losses to surroundings.

In Experiment 2, both heat transfer rates and $|U|$ increased with $\dot{m}_A$, but contrary to the manual's predicted linear trend, diminishing returns emerged beyond $\dot{m}_A = 0.089$ kg/s, with the final two data points effectively plateauing at 106.6 and 108.3 W/m²K. As external fluid resistance falls with increasing flow it approaches equilibrium with the fixed internal fluid resistance, at which point the external resistance ceases to be the sole source of controlling resistance, and further airflow gains diminishing improvements. The energy balance deteriorated significantly at low airflows, where fixed duct heat losses represent a large fraction of a small total heat transfer, recovering to near ideal values as airflow increased.

Water temperature efficiency decreased with increasing water flow rate, but increased with increasing air flow rate. Conversely, air efficiency showed the opposite behaviour in both experiments, highlighting the degree to which the internal and external fluids approach thermal equilibrium with increases in general flow rate. The marginal drop in η_A from 14.44% to 14.32% between 1.4 and 1.6 L/min in Experiment 1 does not indicate deteriorating air efficiency but arises purely due to the water's initial temperature rising slightly at the higher water flow rate.

6.2 Sources of Experimental Error

A large source of uncertainty in the experiment is thermocouple accuracy. K type thermocouples have a rated accuracy of ± 0.1 °C. With ΔT_A values of 2-5 °C and ΔT_w values of 6.2-25 °C, this represents an error as large as 5% for the air and 1.6% for the water.

The water flow sensor reads to 0.01 L/min resolution. At the fixed 0.20 L/min used throughout experiment 2, this is a $\pm 5\%$ resolution uncertainty, at higher flow rates in experiment one the relative impact is less noticeable.

6.3 Potential Improvements

A pump capable of reaching the specified 2.25 L/min would allow the Experiment 1 plateau to be characterised more fully.

Insulating the air duct would reduce parasitic heat losses, lowering effects on C_{EB} deviation at low air flows.

Sensors with a higher accuracy and resolution would allow for less leeway each way when setting variables, reducing the potential error.

7 References

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