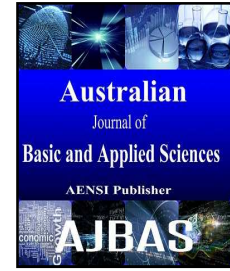




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An Experimental and Simulation Study of the Heat Transfer Coefficient of a Single Mini Channel Condenser for Vapor Compression Refrigeration System

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ABSTRACT

Background: This study presents the experimental investigation of three single mini-channel condensers (100 mm x 50 mm x 18 mm) with hydraulic diameters of 3 mm, 2 mm, and 1mm. The mini-channels made of copper were designed, fabricated, and tested. Each fabricated unit was connected to a vapor compression cycle with R-134a as the refrigerant, which was then placed inside a mini wind tunnel system where forced draft air was introduced to initiate convective heat transfer. The testing and data collection were conducted every five-minute interval for one hour using a Lab View Software. The heat transfer coefficients were calculated from actual experiments at different mass fluxes and Reynolds number during condensation. The calculated heat transfer coefficients were then compared with computed values using the correlations developed by Dittus-Boelter and Lee-Son. The results showed increasing deviation as the diameter was decreased. This deviation could be attributed to the inefficiency of the Dittus-Boelter and Lee-Son correlations in small diameter channels. Moreover, the complexities in the fluid flow process in the fabricated channel might cause the difficulties in attaining very accurate measurements in small channels. The temperature distributions inside the wind tunnel for both air and the refrigerant inside the mini single channel condenser were analyzed and presented using solid works software. The results showed that the temperature of the refrigerant decreases as it releases heat while the temperature of the air increases as it absorbs the heat. These findings are vital for the miniature design of the cooling devices such as car manufacturing, CPU desktop, and refrigeration application.

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INTRODUCTION

One of the main components of air conditioning and refrigeration system is the condenser. In condensers, the refrigerant in a refrigeration system changes the state as the vapor refrigerant is converted to liquid. Its heat transfer coefficient is an important component in understanding the efficiency of the system. For process cooling applications, its design, configuration, and size greatly affect the performance of the system. As the size of the condenser decreases, such as in a mini-channel, the mechanism of fluid flow changes as the capillary effect becomes more prominent. The heat transfer coefficient mechanism is also affected since it is very dependent on fluid flow. Also, some multiple factors can influence the heat transfer coefficient, such as thermal conductivity, specific heat, and viscosity.

Mini-channels are used increasingly with compact heat exchangers. Potential applications of small hydraulic diameter channels include the heat exchangers for electronic equipment, in automotive condensers, and in refrigeration systems. The adoption of the mini-channel also promotes the reduction of the refrigerant charge, which is especially favorable if toxic or flammable refrigerants are used. The phase change of the refrigerants in mini-channels may differ from the conventional channels due to differences in the relative influence of gravity, shear stress and surface tension. The design of small heat exchangers is a must as the size of heat generators such as electronic components reduces to mini scale. The fluid flow inside the mini channel is one of the efficient processes for small-scale cooling. Kandlikar (2006) classified mini channels based on their hydraulic

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diameters (D_h) of 0.5 to 3 mm. There are relatively few studies conducted as regards to mini channels condenser of a refrigeration system.

This study is about the designing, fabrication and testing of a mini channel condenser. The design of the condenser was varied to three mini channels of similar configurations with different channel sizes.

The experimental measurements of the heat transfer coefficient were compared to the existing correlations developed by Dittus-Boelter (1930) and Lee-Son (2009). Finally, the temperature distribution of the fluid flow process inside the single channel condenser was simulated on a computer using a Solid Works software.

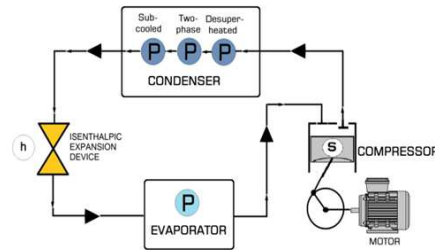


Fig. 1: Analysis on the Condenser.

2.0 Theoretical Considerations:

2.1 Vapor Compression Cycle:

The condenser is a heat exchanger that usually rejects all the heat from the refrigeration system. The condenser accepts hot, high-pressure refrigerant, usually a superheated gas, from the compressor and reject heat from the gas to some cooler substance. As energy is removed from the gas it condenses and this condensate is drained so that it may continue its path back through the expansion valve or capillary to the evaporator. The condenser is divided into three regions characterized by the thermodynamic states, such as the de-superheated vapor, the change of phase (two-phase), and the subcooled liquid regions. The refrigerant condenses at a negligible temperature difference from the ambient conditions.

2.2 Dimensionless parameters for fluid flow:

Many dimensionless parameters are generally used in fluid mechanics to characterize the heat transfer and fluid flow behavior. Some important dimensionless parameters associated with heat transfer and fluid flow, which are used in the current study, are briefly described below.

2.2.1 Reynolds number (Re):

Reynolds number (Re) is a very significant parameter in fluid mechanics. It is named after the British engineer and physicist, Osbourne Reynolds. It is defined as the ratio of inertia force to the viscous force and used to characterize the nature and flow regime of fluids. The general expression of Reynolds number is

$$Re = \frac{\text{inertia force}}{\text{viscous force}} = \frac{\rho v L}{\mu} \quad (1)$$

In this equation:

ρ is density of flowing fluid.

μ is viscosity of the flowing fluid

L is a characteristic length of the

problem. For pipe flow, $L=D$

where L is the pipe diameter

v is a characteristic velocity of the

problem. For pipe flow, $v = v_m$

where v_m is the mean velocity

It specifies the flow characteristics, such as laminar, transitional, or turbulent regime. In a flow, usually, the fluid viscous force acts for managing stability of flow, while fluid inertia is liable for disorganizing the flow. For small Reynolds numbers, the inertial force is not that significant, therefore, the flow is laminar and smooth. On the other hand, for high Reynolds number, the inertia plays an important role, and it is dominant. As a result, the flow is turbulent.

2.2.2 Prandtl number (Pr):

The Prandtl number (Pr) is a dimensionless number, which is widely used in heat transfer approximations. It is named after the German physicist Ludwig Prandtl. It is defined as the ratio of momentum diffusivity (kinematic viscosity) to the thermal diffusivity and expressed as

$$Pr = \frac{\text{momentum diffusivity}}{\text{thermal diffusivity}} = \frac{c_p \mu}{k} \quad (2)$$

In this equation:

c_p is the constant-pressure specific heat of the flowing fluid.

k is the conductivity of the flowing fluid.

The Pr is related to viscosity, specific heat, and thermal conductivity of the fluid. It describes the development of a thermal boundary layer thickness on the heat transfer surface. The thermal boundary layer is a barrier to the heat transfer. The smaller Pr means that the heat diffuses very quickly than the velocity (momentum).

2.2.3 Nusselt number (Nu):

Nusselt number (Nu) was named after a German Engineer Ernst Kraft Wilhelm Nusselt. It is a very important dimensionless parameter in convective heat transfer fields, which is generally expressed as

$$Nu = \frac{\text{Convective heat transfer}}{\text{Conductive heat transfer}} = \frac{hL}{k} \quad (3)$$

It depends on channel geometry, fluid heat transfer coefficient, and conductivity. For a circular geometry channel, $L = D_{ch}$. Therefore, the expression for Nu becomes

$$Nu = \frac{hD_{ch}}{k} \quad (4)$$

where L is the characteristic length, D_{ch} is the channel diameter, h is the heat transfer coefficient, and k is the thermal conductivity of liquid at bulk temperature.

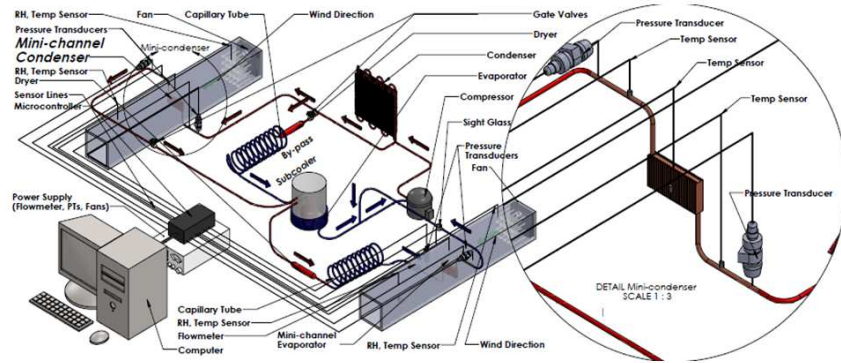


Fig. 2: Experimental Set-up.

2.2 Heat Transfer Coefficient:

The Dittus-Boelter (1930) correlation is a common correlation useful for many applications. This correlation is applicable when forced convection is the sole mode of heat transfer. This is an equation quite similar to McAdams (1954) equation at which the application is presented in the literature, which is not restricted to small diameters (Owhaib, 2007). The equation is expressed below as

$$h_L = 0.023(Re)^{0.80}(Pr)^{0.40} \left(\frac{k}{D}\right) \quad (5)$$

Another correlation developed by Son-Lee (2009) is the single-phase heat transfer coefficient (h_L), which is expressed as

$$h_L = 0.034(Re)^{0.80}(Pr)^{0.30} \left(\frac{k}{D}\right) \quad (6)$$

where k is the thermal conductivity of R-134a, D is the hydraulic diameter, Re is the Reynolds number, and Pr is the Prandtl number.

3.0 Description of the Experimental Work:

Three mini-channel condensers (100 mm x 50 mm x 20 mm copper block) with 1 mm, 2 mm, and 3 mm hydraulic radius for each were fabricated and installed one at a time in the experimental rig as shown in Figure 2. Fins were machined on each face. Each condenser had 24 fins with a width of 2 mm, a length of 50 mm and height of 5 mm. A fan was installed downstream of the heat exchanger to guarantee sufficient air flow across the condenser inside the insulated long duct on the refrigerant-side at the inlet and outlet section in the rig. The condenser set-up was placed in a mini wind tunnel system to simulate the cooling system.

Two temperature sensors and two pressure transducers were installed on the sufficient air flow side across the condenser inside the insulated long duct used in the rig. Two temperature sensors and

two pressure transducers were installed on the refrigerant side at the inlet and outlet of the mini-channel condenser. Another temperature sensor was also installed at the middle of the mini channel condenser. The refrigerant mass flow rate was measured at the outlet of the expansion valve. A valve that passed through the evaporator was used to control the mass flow rate. On the air-side, two temperature sensor spaced equally were installed at the inlet and two at the outlet side. The air velocity over the mini-channel condenser was measured using a hot wire anemometer. Every start of the experiment, the system was vacuumed for about 30-40 min to remove trapped air, followed by leak test before charging the system with the refrigerant R134a. The temperature, flow rate, and pressure readings were automatically recorded, stored and processed by a data acquisition system using LabVIEW software. The data logger was set to scan the data at an interval of 5 minutes.

4.0 Data Analysis and Evaluation of Heat:

Exchanger Performance:

The results related to the heat transfer coefficient and pressure drop were obtained from the readings of temperature sensors, pressure transducers, mass flow meter and hotwire anemometer, installed in the test section. The heat rejected in the condenser was calculated using the relation

$$Q = \dot{m}_s (h_{in} - h_{out}) \quad (7)$$

where \dot{m}_s is the refrigerant mass flow rate and obtained directly from the flow meter, h_{in} is the enthalpy entering the mini channel condenser corresponding to measured pressure and temperature conditions, and h_{out} is the enthalpy leaving the condenser corresponding to measured pressure and temperature conditions.

The average values of experimental heat transfer coefficient were calculated at the average inside surface temperature of the condenser (Suhayla, 2011):

$$Q = h_r A_i (T_{sat} - T_{si}) \quad (8)$$

where h_r is the refrigerant side heat transfer coefficient, A_i is the inside area of mini channel, T_{sat} is the saturation temperature of refrigerant, and T_{si} is the temperature of the inner wall surface.

RESULTS AND DISCUSSIONS

The two parts of the mini-channel condenser (heat exchanger) before they were brazed together through oxy-acetylene are shown in Figure 3. Figure 4 shows the assembled mini channel condenser. Copper is the material of choice for the condenser in this study because of its durability, mechanical strength, and high thermal conductivity. During the brazing process, the surface area of the heat exchanger was carefully observed to avoid clogging

particularly near the entry points of the refrigerant. The variation in the brazing material fill in each channel may also have a marked effect on performance. If some brazing material goes into the channel, it will cause local channel size to become smaller resulting in a reduction in the channel cross-sectional area. This local change in the cross-sectional area may have a drastic effect on the characteristics of the pressure drop. It is worthwhile to note, that this brazed assembly was designed to withstand a pressure of up to 11 bars otherwise exceeding this limitation will damage them. For that reason, these designs were tap after the main condenser. As a result, these mini-channel condensers were only capable of performing the subcooled region out of three different regions namely the de-superheated, saturation and subcooled. It is observed that the resident time of the refrigerant within the heat exchanger decreases which causes a small amount of heat to be absorbed from the surrounding.

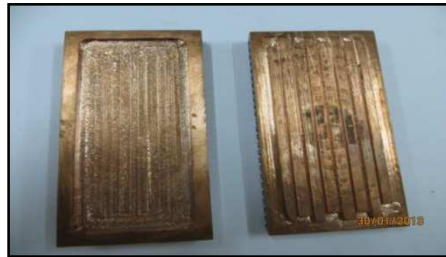


Figure 3. Interior part of mini-channel (Unassembled)



Fig. 4: Mini-Channel Condenser.

The theoretical values of the average heat transfer in the refrigerant side of R134a computed from Dittus-Boelter (1930) and Lee-Son (2009) correlations at experimental data were compared with experimental values of heat transfer coefficient from equation 8 as shown in figures 5 to 7. As can be seen in figures 5 to 7, the experimental heat transfer coefficients values ranged from 13 to 98 $W/m^2 \cdot ^\circ K$ for the 1 mm, from 2600 to 9000 $W/m^2 \cdot ^\circ K$ for the 2 mm, and from 3900 to 5200 $W/m^2 \cdot ^\circ K$ for the 3 mm. The heat transfer coefficient is affected by the mass flux and the size of the channel. The heat transfer coefficient is increased with the rise in the mass flux of R134a. Figures 5 to 7 consistently show the calculated heat transfer coefficient values of both

Dittus-Boelter (1930) and Lee-Son (2009) correlations that over-predicted the experimental results. Also from the same figures, deviations are visible from the existing correlations of heat transfer coefficient. The discrepancy of this heat transfer coefficient from this correlation appears to be related to difficulties in achieving accurate measurements (Mirza, 2010), particularly for the 1 mm channel size as shown in figure 5. The primary cause for these deviations is related to difficulties in getting accurate measurements at small flow rates in small channels, especially for the 1 mm channel diameter having a mass flow of 2 g/s. It is worthwhile to note that as the channel dimensions become smaller, the surface roughness and the measurement error become

relatively more significant. Many authors have noted this difficulty in attaining the precise measurements on smaller channels that always subject to error. These researchers include Cavallini *et al.* (2006), Bergles *et al.* (2010), and Koyama *et al.* (2003). The difficulties may arise from very small flow rates and amounts of heat, and small dimensions of the test

sections. Another factor to be considered as mentioned by Kandlikar (2006), which is frequently neglected, is the uniformity of the cross-sectional dimensions along the channel flow length. If unaccounted, this error significantly alters the experimental results.

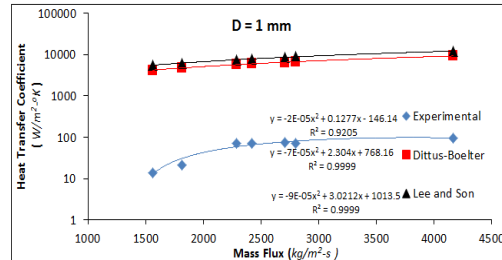


Fig. 5: Variation of heat transfer coefficient with mass flux for condenser size of 1 mm.

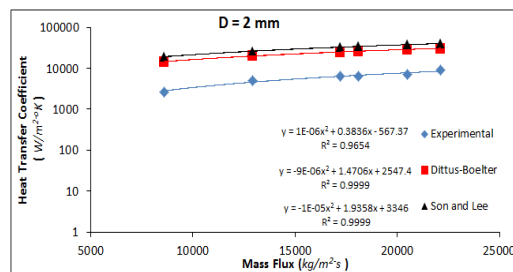


Fig. 6: Variation of heat transfer coefficient with mass flux for condenser size of 2 mm.

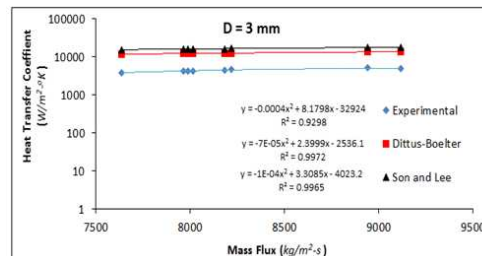


Fig. 7: Variation of heat transfer coefficient with mass flux for condenser size of 3 mm.

The results of the simulation on the refrigerant temperature of hydraulic diameter $D=1\text{mm}$, $D=2\text{mm}$ and $D=3\text{mm}$ are shown in Tables 1, 2 and 3 with the corresponding temperature profiles in Figures 8, 9 and 10. The inlet temperatures for hydraulic diameters of $D=1\text{mm}$, $D=2\text{mm}$ and $D=3\text{mm}$ at the refrigerant side are at 36.38°C , 41.687°C , 37.513°C , respectively. At the air side, the upstream temperatures at hydraulic diameters of $D=1\text{mm}$, $D=2\text{mm}$ and $D=3\text{mm}$, are at 32.398°C , 28.127°C , 32.068°C , respectively. From Figures 8, 9 and 10, one can see that the fluid temperature changes along the mini channel condenser and along the wind tunnel during simulation. These temperature changes help us to clearly understand the heat exchange between two fluids, namely the R-134a and air in the

mini heat exchanger. The simulation results show the temperature profiles along the flow paths of both air and the refrigerant. The colors indicate the increasing temperatures of air as it absorbs heat and the decreasing temperatures of refrigerant as it releases heat.

It is also observed that the results obtained from the calculations done on the simulation reports show the refrigerant outlet temperature and temperature values of air downstream are 35.24°C , 41.43°C , 37.44°C , and 32.65°C , 30.88°C , 32.79°C , respectively. The comparison of both measured values presented as significant change in the temperature. This change is due to the exclusion of some of the important parameters like frictional losses, surface roughness, capillary effect, etc.

Table 1: Refrigerant Conditions, Air Conditions and Simulation Results on Temperature from D=1 mm.

Boundary Conditions				
Refrigerant			Air	
	Pressure(MPa)	Temperature(°C)	Upstream	32.398 °C
Inlet	1.0383	36.380	Wind Velocity	1.233 m/s
			Relative	65.014%
			Barometric	100.660 kPa
	Simulation	Experimental	Simulation	Experimental
	Outlet		Downstream	
Temperature(°C)	35.24	36.362	32.65	33.03

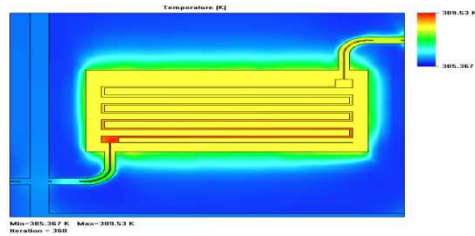


Fig. 8: Temperature Profile of the mini-channel for D=1 mm in hydraulic diameter .

Table 2: Refrigerant Conditions, Air Conditions and Simulation Results on Temperature from D=2 mm.

Boundary Conditions				
Refrigerant			Air	
	Pressure(MPa)	Temperature(°C)	Upstream	28.127 °C
Inlet	1.1753	41.687	Wind Velocity	1.233 m/s
			Relative	86.541 %
			Barometric	100.465 kPa
	Simulation	Experimental	Simulation	Experimental
	Outlet		Downstream	
Temperature(°C)	41.43	40.169	30.88	28.775

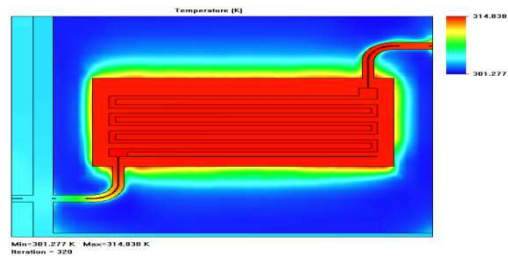


Fig. 9: Temperature Profile of mini channel for D=2 mm in hydraulic diameter .

Table 3: Refrigerant Conditions, Air Conditions and Simulation Results on Temperature from D=3 mm.

Boundary Conditions				
Refrigerant			Air	
	Pressure(MPa)	Temperature(°C)	Upstream	32.06781 °C
Inlet	1.1109	37.51	Wind Velocity	1.233 m/s
			Relative	64.270%
			Barometric	100.637 kPa
	Simulation	Experimental	Simulation	Experimental
	Outlet		Downstream	
Temperature(°C)	37.44	35.22	32.79	32.36

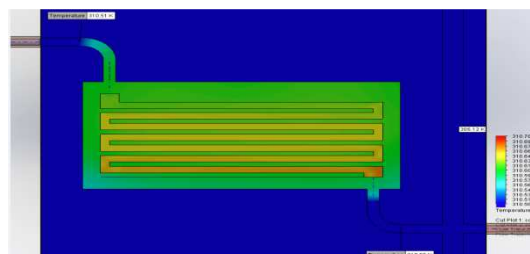


Fig. 10: Temperature Profile of mini channel for D = 3 mm in hydraulic diameter.

The experimental heat transfer coefficients against Reynolds number are presented in Figures 11, 12 and 13, with the corresponding hydraulic diameters of D=1mm, D=2mm, and D=3mm, respectively. Each curve in the figures displays the

equation of experimental heat transfer coefficient as a function of Reynolds number with the corresponding value of the coefficient of determination. These functions were obtained by the curve fitting or regression with least squares using

Microsoft Excel. The equations on the graphs could only be used within the range of Reynolds number. There were 6 data points for D=1mm, 5 data points for D=2mm, and 8 data points for D=3 mm, being considered during the fitting of the equation. The

corresponding ranges of Reynolds number were 14000-47000, 110000-350000, and 53000-93000, respectively. It is observed that the experimental heat transfer coefficient increases as the Reynolds number increases.

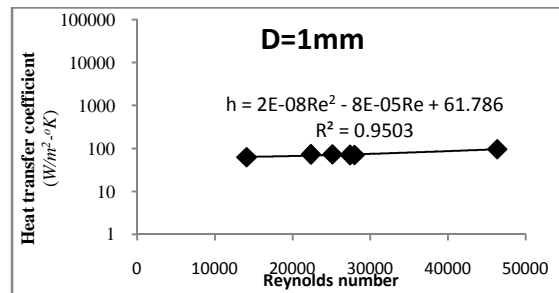


Fig. 11: Variation of experimental values of heat transfer Coefficient and Reynolds number for D=1 mm.

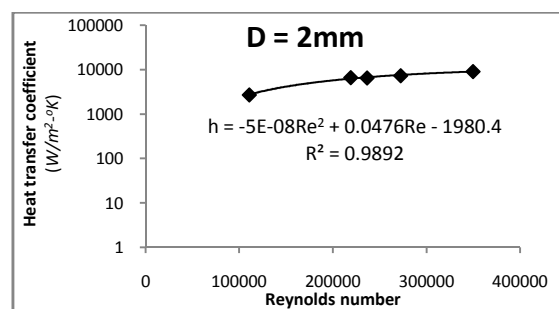


Fig. 12: Variation of experimental values of heat transfer Coefficient and Reynolds number for D=2 mm.

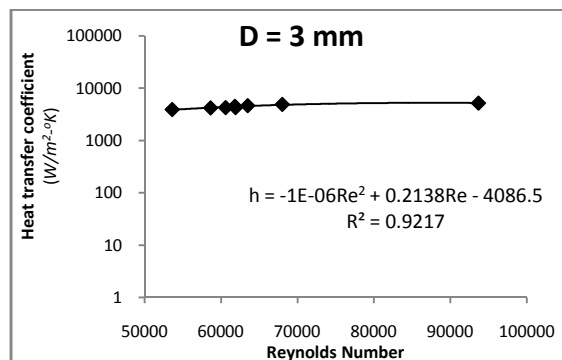


Fig. 13: Variation of experimental values of heat transfer Coefficient and Reynolds number for D=3 mm.

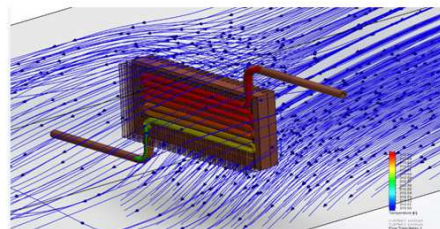


Fig. 14: Flow of air perpendicular to the fins inside the wind tunnel.

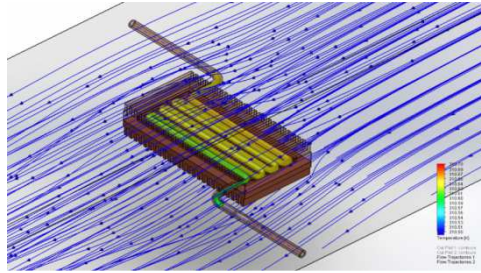


Fig. 15: Flow of air parallel to the fins inside the wind tunnel.

Table 4: Average heat transfer coefficient of air on fins orientation.

Orientation of Fins	Heat transfer coefficient of air (W/m ² /K)		
	D = 1 mm	D = 2 mm	D = 3 mm
Flow of air perpendicular to the fins	3.76	2.76	1.83
Flow of air parallel to the fins	3.52	2.62	1.76

Figures 14 and 15 show the flow of air perpendicular and parallel to the fins. The values of average heat transfer coefficient for air flow perpendicular to the fins and airflow parallel to the fins with hydraulic diameters of D=1 mm, D=2 mm and D=3 mm are shown in Table 4.

As revealed in Table 4, the average heat transfer coefficient of air consistently exhibits that the air flow perpendicular to the fins is slightly higher than the airflow parallel to the fins. The reason for this difference in the heat transfer coefficient of air could be attributed to the distribution of air on the fins.

6. Conclusions And Recommendations:

Based on the results of this study, the following conclusions are drawn.

1. The fabricated mini channel condensers used in this experiment were only capable of withstanding the pressure of up to 11 bars.
2. The heat transfer coefficient increases with increase in the mass flux in the experimental results and in the calculated values of both Dittus-Boelter and Lee-Son correlations. However, the calculated values of both correlations over-predicted the experimental results. This indicates the inappropriateness of the correlations in small channels.
3. The experimental heat transfer coefficient increases as Reynolds number increases.
4. The simulation results clearly show the temperature profiles along the flow paths of both air and the refrigerant. These temperature profiles are represented in colors to indicate the increasing temperatures of air as it absorbs heat and the decreasing temperatures of refrigerant as it releases heat.
5. The average heat transfer coefficient of air consistently shows that the flow of air perpendicular to the fins is slightly higher than the flow of air parallel to the fins.

The following are recommended for further studies:

1. To conduct more experimental runs using different refrigerant fluids, pressures above 10 bars and hydraulic radii of 1.5 mm, 2.5 mm, and 3.5 mm.
2. To perform similar experiments using condenser made of materials having high thermal conductivity and incorporate this property in the design and fabrication of a cooling system to withstand pressures beyond 11 bars.
3. To analyse a boundary layer for small tubes and channels and utilize the result along with the experimental data in the formulation of a new empirical equation for calculating the heat transfer coefficient.
4. To obtain the thermal distribution and temperature profile inside the mini-channel condensers used in this experiment through a thermal imaging system for better measurement of the local fluid temperatures as well as the surface temperature.
5. To characterize the geometry of the mini-channel used in this experiment in terms of the heat transfer coefficient and the temperature distribution through rigorous investigation of the flow process.

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