An Experimental Study on Heat Transfer Characteristics of Refrigerant R134a in a Microtube Evaporator

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Abstract
The heat transfer characteristics of a microtube evaporator using refrigerant R134a were studied experimentally. This evaporator was made from copper tubes and aluminum foils with overall dimensions (L*W*H) of 259.3*43*200 mm. The evaporator has 10 passes, each pass including 8 copper pipes (with hydraulic diameter of 0.82 mm) and 83 aluminum foils (with the thickness of 0.3 mm); the distance between the two foils is 2.6 mm. The experiment was conducted based on the same heat transfer area and overall dimensions as a conventional evaporator (macro evaporator). In this study, the heat flux increases with increasing air flow rate of the evaporator at an inlet air temperature of 30°C. Besides, the cooling capacity and the COP of the microtube evaporator are also presented in this study.

Keywords: Experimental Study, Heat Transfer, R134a, Multi-Micro Tubes, Evaporator.

1. Introduction
In recent years, micro-technology is one of the most interesting fields in research. Scientists and researchers have conducted many impressive and applicable studies in new trends of this technology. Especially, for thermal engineering, thousands of good results have been published such as: improving the heat transfer performance, reducing the dimensions of heat exchangers or improving the heat transfer efficiency. Moreover, many relating applications have been developed such as electronic cooling, cooling turbine blades, cooling fusion reactor blankets, cooling the nozzles of rocket engines, cooling power electronics in avionics and hybrid vehicles, cooling hydrogen storage reservoirs, refrigeration cooling, thermal control in microgravity and capillary-pumped loops. In addition, comparing with the conventional heat exchangers, the heat exchangers using microchannels provide other benefits, such as decreasing the required size, the weight, the pumping power, and the amount of working fluid.

For topic of refrigerants in microchannels, Fayyadh et al. [1] conducted experiments to investigate flow boiling heat transfer of R134a in a multi microchannel heat sink at 6.5 bar system pressure and covered a footprint area-based heat flux range 11.46–403.1 kW/m² as well as mass flux range 50–300 kg/m² s. Three flow patterns were observed namely bubbly, slug and wavy-annular flow when the heat flux increased gradually. The heat transfer coefficient increased with heat flux and there was no mass flux effect. In addition, Mahmoud et al. [2] studied the surface effects to flow boiling of R134a in microtubes and showed that the flow boiling characteristics in the welded tube were completely different from those in the seamless cold drawn tube. In the seamless tube, the heat transfer process was dominated by the nucleate boiling mechanism while the welded tube did not show a clear dominant mechanism. Besides, Thiangtham et al. [3] and Keepaiboon et al. [4] experimentally studied heat transfer and pressure drop characteristics of flow boiling of R134a in a multi-microchannel heat sink. They gave results that: The heat flux and saturation temperature have significant effects on the variation of flow patterns. For pressure drop, experimental results indicated that the total pressure was dominated by frictional pressure drop. The increase of mass flux also increased the frictional pressure gradient, whereas the increase of saturation temperature reduced the frictional pressure gradient. In addition, the heat flux also had an insignificant effect on the frictional the pressure gradient. Micro device for liquid cooling by evaporation of R134a was studied by Wibel et al. [5]. This system is realized by the usage of a micro heat exchanger using microchannels with 100 µm in height and in width and a conventional cooling cycle. In the experiments, the compact multilayer micro heat exchangers enabled the direct cooling of a non-continuous water flow from 55 °C to ≈ 7 °C at a transferred cooling power of 650 W.

Researchers and scientists were also interested in microchannel heat exchangers. A CFD study of the parameters influencing heat transfer in microchannel slug flow boiling was conducted by Magnini and Thome [6]. The results indicated that the heat transfer coefficient...
slightly decreases with an increase of the heat flux when the bubble frequency is constant. An increase of the mass flow rate reduces the heat transfer performance due to the thicker liquid film. The heat transfer coefficient is improved by smaller channels and larger bubble frequency. Besides, Huang and Thome [7] conducted experimental study on flow boiling pressure drop in multi-microchannel evaporators with different refrigerants. With the present test conditions, the channel pressure drop increased with the inlet subcooling and inlet orifice width but slightly affected by the outlet saturation temperature. Markal et al. [8] experimentally investigated saturated flow boiling heat transfer and pressure drop in square microchannel. The results showed that the local two phase heat transfer coefficient decreases with an increase of the heat flux or the local vapor quality for all the mass flux values considered; while, this heat transfer coefficient increases significantly with an increase in the mass flux. In addition, experiments of pressure drop and heat transfer with the different orientations of gravity were conducted by Lee et al. [9]. In this paper, the influence of orientation on two-phase heat transfer was significant for low mass velocities with $G/\rho_f < 0.22$ m/s and negligible for $G/\rho_f > 0.22$ m/s. Moreover, adiabatic two-phase gas–liquid flow behaviors during upward flow in a vertical circular micro-channel were studied by Saisorn and Wongwises [10]. The flow visualization results also indicated that the flow pattern map for vertical upward flow is not completely compatible with that for horizontal flow. In addition, vertical upward flow can issue higher pressure drop when compared with the horizontal channel. Furthermore, gas–liquid two-phase flow in microchannel was also studied by Tripplett et al. [11]. Comparing with relevant flow regime transition models, the results in [11] are poor agreement. Liu et al. [12] conducted an experimental investigation of two-phase slug flow distribution in horizontal multi-parallel microchannels. It was found that the phase distribution characteristics of two-phase flow in parallel channels highly depended on the inlet gas slug length and the inlet real velocity. The channels in the front of the header can influence the phase distribution of the adjacent channel in the rear. Thermal design and operational limits of two-phase micro-channel heat sinks of Kim and Mudawar [13] showed that maximum heat flux is dominated by different limits for different flow rate ranges, and it may be increased significantly as decreasing bottom wall temperature. Besides, Lee and Mudawar [14] studied about transient characteristics of flow boiling in large micro-channel heat exchangers and stated that heat transfer mechanisms depending on quality range, with low qualities associated with slug flow and dominated by nucleate boiling, and high qualities by annular flow and convective boiling.

The influence of configuration and dimension is also interesting topic of microchannel. Tran et al. [15] conducted a study on five different channel shapes using a novel scheme for meshing and a structure of a multi-nozzle microchannel heat sink. In channels of all shapes in this study, the best thermal performance was achieved by a circular channel shape. In an experimental study on the effects of inlet/outlet locations conducted by Dang and Teng [16], for two types (I-type and S-type) of the microchannel heat exchangers, the heat flux and pressure drop obtained from the S-type are higher than those from the I-type, even though the performance indexes of both heat exchangers are essentially the same. Alfaryjat et al. [17], Gunnesegaran et al. [18], and Mohammed et al. [19] studied the effect of geometrical parameters and channel shape on heat transfer characteristics of microchannel heat sinks (MCHS). The results showed that better uniformities in heat transfer coefficient and temperature can be obtained in heat sinks having the smallest hydraulic diameter. In addition, the temperature and the heat transfer coefficient of the zigzag MCHS is the least and greatest, respectively, among various channel shapes. The circular cross-section has the least of the pressure drop. Besides, Hasan et al. [20] studied about the influence of channel geometry on the performance of a counter flow microchannel heat exchanger with different channel configuration such as: circular channel, rectangular channel, square channel, triangular channel, and trapezoidal channel. They showed that circular channel brought the highest heat transfer and dynamic efficiency.

From the above literature reviews, it has been shown that many researchers have investigated the heat transfer characteristics of the microchannel heat exchangers. However, most researchers have studied on the rectangular, triangular, trapezoidal or circular microchannel, but there are no more experiments on multi-micro tubes. In addition, research on a micro tube evaporator used in a refrigeration system has not been studied in more detail. Therefore, in this study, the heat transfer characteristic of micro tube evaporator with the change of air volume flow through the evaporator will be discussed. In particular, the heat transfer characteristics of micro tube evaporator will be investigated to compare with a macro evaporator.
2. Experimental Setup

2.1 Experimental System

The microtube evaporator was designed and fabricated based on the heat transfer area and overall dimensions of a macro evaporator of ZHONGLI, KEWELY with the model FNA-0.25/1.3.

Table 1. Dimensions of evaporator

<table>
<thead>
<tr>
<th></th>
<th>Micro tube evaporator</th>
<th>Macro evaporator FNA-0.25/1.3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimension (mm)</td>
<td>225x43x200</td>
<td>225x45x210</td>
</tr>
<tr>
<td>Pipe diameter (mm)</td>
<td>0.18</td>
<td>0.8</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>80</td>
<td>16</td>
</tr>
<tr>
<td>Number of aluminum foils</td>
<td>90</td>
<td>83</td>
</tr>
<tr>
<td>Heat exchanger area (m²)</td>
<td>1.48</td>
<td>1.48</td>
</tr>
</tbody>
</table>

Fig. 1. Dimensions of the micro tube evaporator

All dimension parameters are shown in Table 1. The two evaporators have the same heat transfer area and the dimensions of the microtube evaporator are slightly smaller than those of the macro evaporator. The micro tube evaporator was made from copper tube and aluminum foil. The microtube evaporator is made of copper tube and aluminum foil. This evaporator has 10 passes, each pass with 8 copper pipes (with hydraulic diameter of 0.82mm) and 80 aluminum foils (with the thickness of 0.3 mm); the distance between the two foils is 2.6 mm. In addition, the overall dimensions of the evaporator are the length L = 259.3mm, the width W = 43 mm, and the height H = 200mm, as shown in Fig. 1.

The experimental system is a refrigeration system using R134a refrigerant to compare with the heat transfer characteristics between the micro tube heat exchanger and the macro evaporator.

Fig. 2. The experimental test loop

(V1: Stop valve 1; V2: Stop valve 2; T1, T2, T3, T4: Temperature sensors; ME: Micro tube evaporator; NE: Macro evaporator FNA-0.25/1.3)

Fig. 3. A photo of the experimental system

(1. Compressor; 2. Condenser; 3. Macro evaporator FNA-0.25/1.3; 4. Micro tube evaporator; 5. Capillary tube; 6. Gate valve)

Experimental system was divided into two cycles: the first cycle experiments on the micro tube evaporator, the second cycle experiments on the macro evaporator with the model FNA-0.25/1.3 (abbreviated by FNA), as shown in
Fig. 2 and Fig. 3. With Cycle 1: the valve V1 opens and the valve V2 closes. The vapor R134a refrigerant enters the compressor and is compressed to high pressure. The high-pressure superheated vapor passing through condenser is cooled into the liquid high pressure. The liquid refrigerant continues to move through the capillary, where the pressure and temperature of the liquid decrease after passage of the capillary due to the expansion process. After passing through the capillary tube, the refrigerant continues through the micro tube evaporator to cool the air. In here, the temperature and humidity parameters are taken by the temperature and humidity sensors, the data is sent to the central controller and is displayed on the computer. The saturated vapor from the micro tube evaporator continues to the compressor suction, so the cycle continues. With cycle 2: the process is similar to cycle 1; however, the valve V1 is closed and the valve V2 is opened. The refrigerant will go through the macro evaporator FNA without through the micro tube evaporator. The compressor used in the system is the piston compressor with the model EE80Y-E; it has a capacity of ¼ Hp. The condenser model FNA-0.8/3.4 has a capacity of 800W.

2.2 Data Analysis

With the experimental conditions in this study, the parameters the heat transfer characteristics of the fluid such as heat flux, heat transfer rate of the heat exchanger will be mentioned Dang and Teng [16] as follows:

The heat transfer rate $Q_t$ is given as follows:

$$ Q_t = mc_p \Delta t $$  \hspace{1cm} (1)

Heat flux is calculated by:

$$ q_i = \frac{Q_t}{A} $$  \hspace{1cm} (2)

Or:

$$ q_i = k_i \Delta t $$  \hspace{1cm} (3)

where $m$ is mass flow rate, $c_p$ is specific heat, $\Delta t$ is temperature difference, $q_i$ is heat flux, $A$ is heat transfer area, $k_i$ is overall heat transfer coefficient.

Total cooling capacity can be expressed as:

$$ h_i = h_s + h_l $$  \hspace{1cm} (4)

The sensible heat in a cooling process of air can be calculated as:

$$ h_s = c_p \rho V \Delta t $$  \hspace{1cm} (5)

where $h_s$ is sensible heat, $c_p$ is specific heat of air, $\rho$ is density of air, $V$ is air volume flow

Latent heat due to the moisture in air can be calculated as:

$$ h_l = V \rho h_{we} \Delta wkg $$  \hspace{1cm} (6)

where $h_l$ is latent heat, $h_{we}$ is latent heat of vaporization of water, $\Delta wkg$ is humidity ratio difference.

The Coefficient of Performance (COP) of the cycle was quantified by:

$$ COP = \frac{h_l}{P_t} $$  \hspace{1cm} (7)

Power input is calculated by:

$$ P_t = U I \cos \phi $$  \hspace{1cm} (8)

where $U$ is Voltage, $I$ is current, $\phi$ is power factor.

3. Results and Discussion

For experimental conditions at constant ambient temperature 30°C, the air volume flow going through the evaporator was varying from 19 l/s to 47.5 l/s, all other parameters were not changed and used on the same experimental system.
temperature of micro tube evaporator, as shown in Fig. 4 and Fig. 5.

![Fig. 6. Heat flux versus air volume flow of evaporator.](image)

The experimental relationships between the air volume flow through evaporator and heat flux are shown in Fig. 6. The results indicated that the heat flux increased as the air volume flow through evaporator increased at the ambient temperature 30°C and all other parameters are constant. For micro tube evaporator, the heat flux can reach a maximum of 163.2 W/m² and 1.32 times higher than the macro evaporator FNA.

![Fig. 7. Cooling capacity versus air volume flow of evaporator](image)

Furthermore, Fig. 7 and Fig. 8 show the comparisons of the air volume flow difference and the cooling capacity and COP of the two evaporators: the micro tube evaporator and the macro evaporator FNA. The cooling capacity and the COP of the evaporator increase when increasing air volume flow. The maximum cooling capacity and COP number of micro tube evaporator are 840W and 4.58, respectively. The COP of micro tube evaporator is 1.4 times higher than the COP of macro evaporator FNA at the air volume flow is 47.5 l/s and the inlet air temperature is 30°C.

![Fig. 8. COP versus air volume flow of evaporator](image)

The heat transfer characteristics of micro tube evaporator and macro evaporator FNA are also compared by thermal camera. Fig. 9 shows the heat distribution of the micro tube evaporator is very uniform while the heat is concentrated only in the lower part of the macro evaporator FNA, at the same experimental conditions.

![Fig. 9. A photo of the evaporators was taken by a thermal camera](image)

4. Conclusions

An experiment study of the heat transfer characteristics of evaporation process in a microtube evaporator has been studied by changing of the air volume flow of the microtube evaporator. In addition, the results of the study were compared on the same conditions with the macro evaporator FNA. The results show that, at the same ambient temperature of 30°C, the heat flux of the evaporator increases when
increasing air volume flow through evaporator. The heat flux of micro tube evaporator is higher than the macro evaporator FNA times, at the same experimental conditions.

Besides, the cooling capacity and the COP of the evaporator increase when increasing air volume flow. The maximum cooling capacity and the COP of microtube evaporator are 840W and 4.58, respectively. The COP of micro tube evaporator is 1.4 times higher than the COP of macro evaporator FNA at the air volume flow is 47.5 l/s and the inlet air evaporator temperature is 30°C. Moreover, the micro tube evaporator distributes heat more uniformly than the macro evaporator FNA, at the same experimental conditions.

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References