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Study on Thermal Performance of Micro-Channel Separate Heat Pipe for Telecommunication Stations: Experiment and Simulation

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Study on thermal performance of micro-channel separate heat pipe for telecommunication stations: Experiment and simulation

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ABSTRACT

A micro-channel separate heat pipe (MCSHP) as a cooling device for telecommunication stations (TSs) was experimentally investigated in this work. A steady-state mathematical model was built and validated with experimental data. The cooling capacity, inlet and outlet of refrigerant temperatures and refrigerant pressure of evaporator section were measured in an enthalpy difference laboratory (EDL). The average relative errors between simulation and experiment were less than 10%. The effects of geometrical design and environment conditions on thermal performance were analyzed using the validated model. As a result of the simulations, the refrigerant side pressure drop decreased by 96.61% at the flat tube height 3.0 mm compared to 1.4 mm. And the air side pressure drop decreased by 94.49%, when fin height ranged from 4 mm to 20 mm. The cooling capacity increased by 50.65% at the fin pitch 3.0 mm compared to 1.0 mm. These factors were of practical engineering importance in optimum design of MCSHP.

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Keywords: Micro-channel, Steady-state mathematical model, Cooling capacity, Geometrical parameters

Étude de la performance thermique de caloduc séparé à micro-canaux pour les stations de télécommunication: expérience et simulation

Mots clés : Micro canaux ; Modèle mathématique à état stable ; Paramètres géométriques de la puissance de refroidissement

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Abbreviations: EDL, enthalpy difference laboratory; MCSHP, microchannel separate heat pipe system; RTPF, round tubes and plate fins; SHP, separate heat pipe system; TSs, telecommunication stations
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1. Introduction

In recent years, with the rapid development of telecommunication industry, millions of telecommunication stations (TSs) have been built. The communication network in China has grown as the largest one in the world with over 600,000 stations in 2007 (Chen et al., 2009). Operation of such a huge network consumes about 20 billion kWh electricity annually, one-third of which was accounted by TSs (Sun et al., 2014). To ensure the proper operation of the electronics inside, TSs are generally enclosed within a housing and the internal environment is maintained with an air-conditioning system. The electricity consumption of those air conditioning systems for TSs is of great significance, which comprises about 30–50% of the total (Tu et al., 2011). In order to reduce this consumption as well as ensure the indoor air cleanness of TSs for electronic equipment, heat pipe, as a heat dissipation technology, has been widely used in TSs to cool down the indoor air when the outdoor air is cool (Zhou et al., 2013).

Micro-channel separate heat pipe (MCSHP) relies on phase change processes, namely evaporation and condensation, and the circulation of working fluid to exchange heat. Unlike a conventional heat pipe, MCSHP do not have a wick structure; the gravity and micro-channel structure play an important role in providing driven force for the refrigerant circulation. The pressure drops through the evaporator section, condenser section and connection pipes largely affect the refrigerant mass flow.

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rate and thus the cooling capacity; a tiny design or operation negligence could lead to a heavy degradation of performance, even failure of cooling.

It has been reported that the thermal performance is greatly related to the geometrical parameters (Khodabandeh, 2005; Moon et al., 2004; Suman and Kumar, 2005). Experimental studies were performed to investigate the pressure drop in evaporators and the riser of an advanced thermosyphon loop with different dimensions of connecting tube and evaporator channels (Khodabandeh, 2005). Suman and Kumar (2005) compared the thermal characteristics and performance of two different heat pipes with triangular and rectangular shapes by utilizing an analytical model. It was claimed that the capability of the working fluid at the evaporator section of rectangular micro-heat pipes is more depressed than that of triangular ones, because of the larger radius of curvature in rectangular micro-heat pipes. Subsequently, Hung and Seng (2011) proposed a star-groove micro-heat pipe; the effects of various geometrical parameters were analyzed by using a mathematical one-dimensional, steady-state model. The heat transport capacity increased with the increase of cross-sectional area of the star-groove micro-heat pipe, number of corners, and adiabatic section length. On the other hand, the increase in the total length of the micro-heat pipe resulted in the decrease of heat transport capacity. It was observed that there exists an optimum hydraulic diameter and the ratio of the adiabatic section length to the total length when the heat pipe achieves its maximum heat transport capacity. Liang and Hung (2010) made a further research on finned U-shape heat pipes experimentally. It was concluded that the performance could be optimized by changing L-ratios (ratio of the evaporator section length to the condenser section length), pipe diameters and the fin pitches. Among these, the results showed that the performance of heat pipe was strongly governed by the geometrical parameters. However, very little information about the effects of geometrical configuration on thermal performance of MCSHP was provided.

Micro-channel heat exchangers, which have obvious advantages over Round Tubes and Plate Fins (RTPF) heat exchangers, were attracting increasing attention in air conditioning systems (Khan and Fartaj, 2011; Park and Hrnjak, 2008). Compared to RTPF, micro-channel heat exchangers possess better heat transfer characteristic, lower pressure difference, lower filling ratio and more compact structure. However, the application of micro-channel heat exchangers in heat pipe system was only reported by limited studies. Zhang et al. (2013) experimentally investigated the performance of a new-type flat micro-heat-pipe-array heat exchanger using δ-Al2O3-R141b nanofluids as working fluids. The maximum pressure drop ratio on the air side was 200 Pa/m, which was much less than using RTPF heat exchangers (Lee et al., 2012). Consequently, using the cooling capacity, the refrigerant side pressure drop and the air side pressure drop as the objective function to evaluate the thermal performance of MCSHPs are very essential.

This paper focuses on the effects of geometrical design and environment conditions on thermal performance of MCSHP for TSs. The micro-channel heat exchanger was applied in a separate heat pipe system as the evaporator and condenser section. A steady-state mathematical model of MCSHP was developed for determining the cooling capacity, refrigerant mass flow rate, pressure drop (air side and refrigerant side), and total heat transfer coefficient of evaporator section. The cooling capacity, inlet and outlet of refrigerant temperatures and refrigerant pressure of evaporator section were measured in an enthalpy difference laboratory (EDL). The simulation results were compared with the experimental data for validating the model. Then, the effects of geometrical design and environment conditions on thermal performance of MCSHP were calculated and analyzed by using the validated model.

2. Experimental methods

2.1. Description of MCSHP

MCSHP has an evaporator section and condenser section where the working fluid evaporates and condenses respectively. The condenser section is located above the evaporator so that the condensate is returned to the evaporator by gravity. The vapor ascending and condensate descending tubes are jointed with copper tubes. The evaporator section and condenser section of MCSHP are the micro-channel heat exchangers with identical geometrical parameters and air facing area. The refrigerant flow inside and the air flow outside the heat exchangers are oriented as crossflow. The details and the geometrical dimensions of the micro-channel heat exchanger were shown in Fig. 1. The interior of each flat tube contains several webs (cells), which is utilized to enhance the performance of heat transfer and make the structure more compact. Design of multi-louver fins increases the heat transfer area and improves the efficiency of heat transfer. Flat tube width and fin width have the similar impacts on the heat transfer performance. The geometrical parameters, including flat tube width, flat tube height and number of mini-channel in one flat tube, could influence the refrigerant flow characteristics. Meanwhile, fin pitch, fin height and louver pitch could impact the flow on the air side. The main geometric parameters of the test sample were summarized in Table 1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Core size (mm)</td>
<td>740 × 840 × 25.4</td>
</tr>
<tr>
<td>Number of micro-channel in one flat tube Nw</td>
<td>9</td>
</tr>
<tr>
<td>Flat tube width Bt (mm)</td>
<td>25.4</td>
</tr>
<tr>
<td>Flat tube height Ht (mm)</td>
<td>2</td>
</tr>
<tr>
<td>Flat tube thickness δt (mm)</td>
<td>0.45</td>
</tr>
<tr>
<td>Fin pitch Pf (mm)</td>
<td>1.5</td>
</tr>
<tr>
<td>Fin height Hf (mm)</td>
<td>8</td>
</tr>
<tr>
<td>Fin thickness δf (mm)</td>
<td>0.1</td>
</tr>
<tr>
<td>Louver pitch Pl (mm)</td>
<td>1</td>
</tr>
<tr>
<td>Louver length Lh (mm)</td>
<td>7</td>
</tr>
<tr>
<td>Louver angle θ (deg)</td>
<td>30</td>
</tr>
</tbody>
</table>

Table 1 – Geometrical parameters of the test sample.
air temperature and relative humidity as the indoor conditions of TS, and the exterior space for emulating the outdoor environment. Consequently, the evaporator section of MCSHP was installed in the interior space, and the condenser section of MCSHP was installed in the exterior space. The height difference of the two sections was 1.5m. Probes at measuring location ① measured and recorded the outlet air temperatures and relative humidities of the evaporator section. The indoor and outdoor air temperatures and relative humidities were measured and recorded by probes at locations ② and ③.

Fig. 1 – Schematic diagram of the micro-channel heat exchanger.

Fig. 2 – Elevation view of the enthalpy difference laboratory (EDL) and experiment devices.
The experiment was carried out under a simulation of TS conditions, in which the interior space dry air temperature and relative humidity were 28 °C and 45%, respectively, and the exterior space dry air temperature and relative humidity were 18 °C and 40%, respectively. The procedure was outlined as follows:

(1) The standard rating conditions for test was simulated using the EDL.

(2) After evacuating the inside of the MCSHP, R22 was charged into the MCSHP through the fill charge line. The filling charge varied from 2500 g to 3500 g, and a test was performed once at 2500 g, 2700 g, 2900 g, 3100 g, 3300 g, and 3500 g. Six sets of tests were carried out altogether. Measurements were made in a steady-state condition achieved 20 min after the initiation. During the equilibrium, data were recorded every 5 minutes at each steady-state condition and the results were the average of seven data sets. The data sets are the indoor air dry-bulb temperature and relative humidity, outdoor air dry-bulb temperature and relative humidity, air volume, and refrigerant temperature and pressure at the inlet and outlet of evaporator section.

### Table 2 – Major sensors and their errors.

<table>
<thead>
<tr>
<th>Types</th>
<th>Probes 1–3</th>
<th>Refrigerant measurement device</th>
<th>Flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature (°C)</td>
<td>Relative humidity (%)</td>
<td>Temperature (°C)</td>
<td>Pressure (MPa)</td>
</tr>
<tr>
<td>Errors</td>
<td>-15 to 55</td>
<td>-55 to 125</td>
<td>-0.1 to 3.0</td>
</tr>
<tr>
<td>Ranges</td>
<td>0 to 100</td>
<td>0.01</td>
<td>0.03</td>
</tr>
<tr>
<td>1</td>
<td>0.2</td>
<td>2</td>
<td>2025 to 4217</td>
</tr>
</tbody>
</table>

respectively. The temperature sensors were platinum resistance temperature sensors. The volumetric flow rate of air passing through the unit was adjusted by a fan through the control of open nozzles. Base on the mechanism of EDL, the relation between the volumetric flow rate and the differential pressure of the nozzle was presented in Eqs. (1)–(2) according to the ISO standard (ISO-5167, 2003). The air side pressure drop was obtained with differential pressure transducers (series number: FP3002, model: EJA110A-DLS4A-92DA). The temperatures and pressures of R22 at the inlet (location ④) and outlet (location ⑧) of evaporator section were measured by a measurement device (model: HS-6285) of measuring the refrigerant temperature and pressure. The major sensors and their errors and ranges were given in Table 2.

\[ q_{\lambda} = 9.767 \times 10^{-3} \sqrt{\Delta P} \quad \phi = 100 \]  
\[ q_{\lambda} = 21.97 \times 10^{-3} \sqrt{\Delta P} \quad \phi = 150 \]

3. **Modeling**

The modeling of MCSHP includes four modules: the model of evaporator section, condenser section, vapor ascending tube and condensate descending tube. The model structure of evaporator section and condenser section is the same as a micro-channel heat exchanger. Likely, the diameters of vapor ascending tube and condensate descending tube were 19 mm and 16 mm, respectively.

3.1. **The model of micro-channel heat exchanger**

A simulation model was developed for the micro-channel heat exchanger; this model used a finite volume method to predict the micro-channel heat exchanger cooling capacity, refrigerant mass flow rate, pressure drop (air side and refrigerant side), and total heat transfer coefficient of micro-channel heat exchanger. Each flat tube was divided into 100 segments. The properties of air and refrigerant at the outlet of each element were determined by the effectiveness-NTU method. The model was solved by Matlab R2007a and refrigerant thermodynamic properties dynamically link into NIST Refrigerant Database REFPROP software (Lemmon et al., 2002).

The following assumptions were made for constructing this model:

(1) The refrigerant in the micro-channels has a one-dimensional homogeneous flow, and no heat transfer exists along the direction of the refrigerant flow;

(2) the refrigerant flow rate is evenly distributed among the micro-channels.

(3) the refrigerant in the different parallel micro-channels has the same temperature and pressure distribution pattern along the flow direction; and

(4) air flow outside the flat tube is uniform.

Air side heat transfer coefficients \( h \), and air side pressure drop \( \Delta P_a \), were predicted by the Chang and Wang (1997) and Kim and Bullard (2002) correlation, respectively.

When the refrigerant was in single-phase conditions, there were two kinds of refrigerant flow types, namely laminar flow and turbulent flow, of which the determination criterion was given by Bhatti and Shah (1987). The heat transfer coefficients of refrigerants for single-phase conditions (Churchill, 1977a, 1977b; Shah and Bhatti, 1987; Shah and London, 2014; Yin et al., 2001) were as follows.

For laminar flow condition,

\[ Nu = 8.325(1 - 2.2041a + 3.0853a^2 - 2.4756a^3 + 1.0578a^4 - 0.186a^5) \]  
\[ f = 96 \left(1 - 1.3553a + 1.9467a^2 - 1.7012a^3 + 0.9546a^4 - 0.2537a^5 \right) \]
For turbulent flow condition:

\[
\begin{align*}
\text{Nu} &= 4.364^{10} + \left[ \frac{2200 - \text{Re}}{4.364^{10}} + \left( \frac{1}{6.3 + \frac{0.079}{(1 + \text{Pr}^{0.5})^{12}}} \right) \right]^{1/4} \\
f &= \frac{8}{\text{Re}^{12}} + \left[ \frac{1}{2.457} \ln \left( \frac{1}{(7/\text{Re})^{0.5} + 0.277/\text{Re}} \right) \right]^{1/8} + \left( \frac{37530}{\text{Re}} \right)^{1/4}
\end{align*}
\]

(5) (6)

When the refrigerant was in two-phase conditions \(\text{Akers et al., 1958; Yan and Lin, 1999}\):

\[
\text{Nu} = 0.0265 \text{Re}^{0.8} \text{Pr}^{0.17}
\]

(7)

\[
f = 498.3 \text{Re}^{0.74}
\]

(8)

\[
\text{Re}_{\text{eq}} = \frac{G_{\text{eq}} D_{i}}{A_{\text{eq}} ho_{\text{liq}}}
\]

(9)

\[
G_{\text{eq}} = G_{i} \left( 1 - x \right) + x \left( \frac{\rho_{\text{liq}}}{\rho_{\text{vap}}} \right)^{0.5}
\]

(10)

With the Nusselt number and friction factor, the refrigerant-side heat transfer coefficient \(h_{r,i}\) and pressure drop \(\Delta P_{r,i}\) were calculated by Eqs. (11)–(12).

\[
h_{r,i} = \frac{\text{Nu}_{eq} A_{i}}{D_{i}}
\]

(11)

\[
\Delta P_{r,i} = f \frac{L_{i}}{2D_{i} \rho_{i}} \left( \frac{G_{i}}{A_{i}} \right)^{2}
\]

(12)

Heat transfer from the refrigerant to the air undergoes the refrigerant-side heat convection thermal resistance, the wall heat conduction thermal resistance and the air-side heat convection thermal resistance. Therefore the overall heat transfer coefficient in terms of air-side heat transfer area was given as follows:

\[
U = \frac{1}{h_{r,i}} + \frac{1}{\lambda_{w}} A_{w} + \frac{1}{h_{a}} A_{a}
\]

(13)

The effectiveness of heat exchanger \(\varepsilon\) \(\text{Yang and Tao, 2006}\) in single phase conditions,

\[
\varepsilon = 1 - \exp \left[ \frac{1}{C_{r}} \text{NTU}^{0.22} \left( \exp \left( -C_{r} \cdot \text{NTU}^{0.78} \right) - 1 \right) \right]
\]

(14)

Where \(C_{r} = \frac{(GC)_{\text{min}}}{(GC)_{\text{max}}}\)

(15)

The effectiveness of heat exchanger \(\varepsilon\) \(\text{Yang and Tao, 2006}\) in two phase conditions,

\[
\varepsilon = 1 - e^{-\text{NTU}}
\]

(16)
(1) Input micro-channel heat exchanger geometrical parameters, inlet refrigerant state parameters and operational conditions.
(2) Assume the initial heat transfer rate $Q_0$.
(3) Obtain the vapor quality and then judge the state of refrigerant.
(4) Calculate refrigerant side heat transfer coefficient and pressure drop governed by Eqs. (3)–(12).
(5) Calculate the heat transfer rate of every segment $Q(i)$ by $\epsilon$-NTU method according to Eqs. (13)–(20), and compare with initial heat transfer rate $Q_0$ and return step (2) until $Q(i)$ converged.
(6) $i = i + 1$, and set the outlet condition of next segment, return to step (3) until $i > N$.
(7) Output the outlet of refrigerant state parameters, refrigerant mass flow rate, pressure drop of air side and refrigerant side and heat transfer coefficient.

3.2. The model of vapor ascending tube and condensate descending tube

The flow of refrigerant in the vapor ascending tube and condensate descending tube was an isothermal process. The total pressure drop was calculated by Eq. (21).

$$\Delta P = (\lambda L + \sum \xi \rho_{\text{ref}} V_{\text{ref}}^2) \frac{V_{\text{ref}}^2}{2g} + (\lambda L + \sum \xi \rho_{\text{ref}} V_{\text{ref}}^2) \frac{V_{\text{ref}}^2}{2g}$$

(21)

The pressure increase caused by the height difference was calculated by Eq. (22).

$$\Delta P = (\rho_{\text{ref}} - \rho_{\text{ref}}) g H_{\text{ev}}$$

(22)

4. Results and discussion

Before examining the effects of geometrical parameters and environment conditions on the thermal performance of the MCSHP, the numerical results of the steady-state mathematical model was validated. Cooling capacity, inlet and outlet refrigerant temperature and refrigerant pressure of evaporator section were tested through EDL. The refrigerant mass flow rate was calculated using the enthalpy difference method. Comparison between simulated results from the calibrated model and measured results of MCSHP was shown in Table 3 and Fig. 5.

The average relative errors $\sigma$ between measured value and simulated value were calculated by Eq. (23).

$$\sigma = \frac{\text{Measured value} - \text{Calculated value}}{\text{Measured value}} \times 100\%$$

(23)

As shown in Table 3, the average relative errors of pressure between the measured and simulated were 0.7%, 2.82%, respectively, and the average relative errors of temperature were 2.37% and 3.26%, respectively. In Fig. 5, hollow triangles were the cooling capacity and refrigerant mass flow rates obtained from the experiments and the solid squares were the simulated results with the calibrated model. Different colors represented the different experiment sets. For the refrigerant mass flow rate ranging between 52 kg/h to 68 kg/h, the coefficient of determination $R^2$ between the calculated and measured results was 0.9936, which demonstrated a very good agreement.

Since this model neglected the effect of the microchannel heat exchanger header on the heat transfer rate of MCSHP thus changed the results of refrigerant pressure and temperature and the cooling capacity, there would exist discrepancy between the experiments and the simulation. Even so, these errors were in a reasonable range and the model was good enough for engineering applications.

4.1. Effect of flat tube width and fin width

Flat tube width and fin width have the same change trend. Figure 6a plots cooling capacity and pressure drop (refrigerant side) of evaporator section with respect to the flat tube width and fin width. The cooling capacity increased with the increase of the flat tube width and fin width, and it ranged from 2550 W to 3400 W. It is because the total heat transfer area of each element increased with increasing flat tube width and fin width. In addition, the refrigerant side heat transfer coefficient was dependent on the Nusselt number, refrigerant thermal conductivity and hydraulic diameter according to Eq. (11). However, as Fig. 6b shown, with the increase of flat tube

| Table 3 – Comparison of the refrigerant temperature and pressure of the evaporator section between the simulated and measured results. |
|-----------------|-----------------|-----------------|-----------------|
| The average relative errors | Inlet of the evaporator section | Outlet of the evaporator section |
| Temperature | Pressure | Temperature | Pressure |
| $\sigma$ | 2.37% | 0.7% | 3.26% | 2.82% |

Fig. 5 – Comparison of the cooling capacity of the evaporator section between the simulated and measured results.
width and fin width, the effectiveness of the evaporator section increased gradually, while the increasing rate decreased. The refrigerant side of pressure drop decreased with increasing flat tube width and fin width. It varied from 1.7 kPa to 0.08 kPa, when the flat tube width and fin width increased, which enhanced the effective flow area of the flat tube and decreased the flow rate of refrigerant, thus reduced the flow resistance on the refrigerant side.

4.2. Effect of flat tube height

Fig. 7 shows that the cooling capacity and refrigerant side of pressure drop decreased with increasing flat tube height. The varied range was 3130–3390 W and 0.11–3.25 kPa, respectively. The effective flow area of flat tube increased with the flat tube height increased, then the flow velocity of refrigerant in the tube and the refrigerant side of heat transfer coefficient both decreased. In addition, when the flat tube height changed from 1.4 mm to 3.0 mm, the number of flat tube decreased under the condition of constant micro-channel heat exchanger width. Therefore, the cooling capacity and the pressure drop of refrigerant side decreased.

4.3. Effect of number of micro-channel in one flat tube

Fig. 8 presents cooling capacity and refrigerant side of pressure drop increased with increasing number of micro-channel in one flat tube. The change range was 3194–3209 W and 0.05–0.7 kPa, respectively. Because of the increasing number of micro-channel in one flat tube, the refrigerant flow area reduced and the flow velocity of the refrigerant increased, the refrigerant side heat transfer coefficient increased, therefore, the cooling capacity and the pressure drop of refrigerant side increased.
4.4. Effect of fin pitch

Fig. 9a shows cooling capacity and pressure drop (air side) of the evaporator section with respect to the fin pitch. The cooling capacity decreased with the increase of the fin pitch, and it ranged from 3450 W to 2290 W. Air flow rate and the Colburn j factor of air side (as shown in Fig. 9b) decreased with increasing fin pitch, the heat transfer coefficient of the air side decreased. Meanwhile, the fin pitch has physical impact on the area of heat transfer. At the condition of constant micro-channel heat exchanger length, the area of heat transfer decreased with the increase of the fin pitch; therefore, the cooling capacity decreased. The air side pressure drop decreased with increasing fin pitch, and it varied from 481 Pa to 69 Pa. The friction coefficient of the air side (as shown in Fig. 9b) decreased with the increase of fin pitch, and the pressure drop of air side decreased with the increase of fin pitch.

4.5. Effect of fin height

Fig. 10a describes that cooling capacity and air side of pressure drop decreased with increasing fin height. The change range was 2468–3475 W and 32.89–1310.48 Pa, respectively. When the width of micro-channel heat exchanger was constant, the fin height has physical impact on the number of flat tubes and

Fig. 9 – (a) Effect of fin pitch on cooling capacity and pressure difference (air side). (b) Effect of fin pitch on Colburn j-factor and friction coefficient.
the area of heat transfer. Because the heat transfer area of air side increased with increasing fin height, the heat transfer rate increased. In addition, air flow rate passing through evaporator section decreased owing to the air flow area enlarged with increase of fin height, which could decrease the heat transfer coefficient. The fin height and the number of flat tubes together impact on the cooling capacity. Therefore, the cooling capacity was proportional to the product of every element's total heat transfer area $A_a$, air side of heat transfer coefficient $h_a$ and the number of flat tubes, of which decreased with the increase of fin height, as shown in Fig. 10b. As a result, the cooling capacity was decreased. The air flow rate passing through the evaporator section decreased due to effective windward area increased with increase of fin height, which could lead to decrease in pressure drop of air side. While the fin height increased below 12 mm, the pressure drop of air side decreased rapidly, however, once the fin height was higher than 12 mm, the pressure drop of air side decreased slowly.

4.6. Effect of louver pitch

Fig. 11a presents that cooling capacity and pressure drop (air side) decreased with increasing louver pitch, and the change range was 3210–3430 W and 263.13–1959.3605 Pa, respectively. The main function of louver structure was to destroy the boundary layer of air flow. Fig. 11b shows that Colburn j factor of air side and friction coefficient decreased with the increase of the louver pitch, leading to a lower heat transfer coefficient on the air side. Consequently, the cooling capacity and air side of pressure drop both decreased.

4.7. Indoor and outdoor temperature difference

The effect of temperature difference between indoor and outdoor air was also analyzed by using this model. Temperature difference between indoor and outdoor air was one of the important factors affecting the MCSHP thermal performance (Zhou et al., 2013; Zhu et al., 2013). The exterior space temperature was varied from 8 °C to 22 °C, while the interior space temperature was maintained at 28 °C. As shown in Fig. 12, cooling capacity increased rapidly when the temperature difference between indoor and outdoor air increased from 6 °C to 8 °C. However, this trend slowed down and became more linear when the temperature difference between indoor and outdoor air increased from 10 °C to 20 °C. The cooling capacity change trends identified here were similar to the experimental research on SHP heat transfer performance (Li et al., 2010; Zhu et al., 2013). The cooling capacity was increased by 135% with temperature difference between indoor and outdoor increased from 6 °C to 8 °C. Another 40% increase was achieved when the temperature increased from 10 °C to 20 °C, which fully showed the MCSHP has a wide prospect of saving energy application (Chaudhry et al., 2012; Srimuang and Amatachaya, 2012). As the indoor and outdoor difference varying from 6 °C to 8 °C, the heat transfer mode of evaporator section changed from overheated steam to phase change, the cooling capacity increased rapidly. But with the temperature difference between indoor and outdoor air increasing gradually, the increase of the cooling capacity was mainly dependent on the temperature of returning refrigerant to the evaporator section. The mechanism of heat transfer change from phase change to sensible heat exchange, and the cooling capacity increased by the inlet temperature of evaporator section decreased rate.
5. Conclusions

A separate micro-channel heat pipe system was proposed, and the relating theoretical model was developed to analyze effects of geometrical design and environment conditions on thermal performance of micro-channel separate heat pipe for telecommunications stations. The cooling capacity, inlet and outlet of refrigerant temperature and refrigerant pressure were tested in an enthalpy difference laboratory. The simulation results were in good agreement with experiment results. The average error between model and experiment results was less than 10%. The cooling capacity, air side and refrigerant side pressure drop predicted in the present study can serve as a useful analytical tool for separate heat pipe design and performance analysis. Besides, the following conclusion can be drawn based on the simulation of the heat pipe heat exchange device:

1. The maximum influence on the pressure drop of refrigerant side was caused by the flat tube height. The refrigerant side pressure drop decreased by 96.61% at the flat tube height 3.0 mm compared to 1.4 mm. The maximum influence on the pressure drop of air side was caused by fin height, followed by fin pitch, louver pitch minimum. The air side pressure drop decreased by 94.49%, when fin height changed from 4 mm to 20 mm.
2. The maximum influence on the cooling capacity was caused by fin pitch. The cooling capacity increased by 50.65% at the fin pitch 3.0 mm compared to 1.0 mm.
3. The effect of temperature difference between indoor and outdoor on SHP was analyzed, which showed that the cooling capacity increased by 135% with the increasing of the indoor and outdoor temperature difference from 6 °C to 8 °C.

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