Performance Analysis of a Heat Pipe Heat Exchanger Under Different Fluid Charges

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Performance Analysis of a Heat Pipe Heat Exchanger Under Different Fluid Charges

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An experimental study was conducted on the effects of the amount of working fluid on the performance of a two-row copper-R134a heat pipe heat exchanger at the air side. The heat pipe heat exchanger operates in the horizontal configuration, and five different fluid charges were studied to cover the underload and overload cases. Moreover, the calculated outlet temperature of the evaporator side, obtained from the heat exchanger simulation, was compared with the experimental results. Based on the present study, it was found that the optimum performance of the copper-R134a heat pipe heat exchanger can be achieved at a fluid charge slightly exceeding the amount required to saturate the wick.

INTRODUCTION

Heat pipe heat exchangers (HPHXs) are heat transfer devices that transfer heat through a phase change. Because of the efficient heat transfer rate and conductivity of the HPHXs, these have been used for cooling and heating purposes.

An HPHX consists of heat pipe tubes that are filled with a proper working fluid. The working fluid absorbs the heat and evaporates in the evaporator section and condenses in the condenser section. The condensed liquid is returned to the evaporator with the aid of the capillary action in the horizontal configuration of heat pipes and gravity in thermosyphon heat exchangers. The performance of a HPHX mainly depends on factors such as the container, the wick structure, and the working fluid.

For a certain type of working fluid, the fluid charge or the amount of working fluid plays a key role in the performance of a HPHX. For instance, Martinez et al. [1] used 3.04 g of ammonia as the working fluid in their research (i.e., slightly exceeding the amount required to saturate the wick). In a study by Kaya and Goldak [2], a heat pipe with completely saturated wick was used for the numerical study of heat and mass transfer in the heat pipes.

The influence of the number of layers and the working fluid loads (fluid charges) on the performance of a screen mesh copper–water heat pipe was investigated by Kempers et al. [3]. Heat pipes with a total length of 177.8 mm, an outer diameter of 9.53 mm, and three layers of screen mesh were used for the test. Five different fluid loads, namely, 50%, 80%, 100%, 120%, and 150% of the amount of water needed to saturate the wick structure, were chosen for this purpose. It was observed that with increasing fluid load, the thermal resistance of the heat pipe also increased. In addition, the heat pipe containing the fluid load close to the amount needed to fully saturate the wick showed small differences in the effective thermal resistance. Based on the results, it was established that the maximum heat transfer in the underload heat pipes was lower than that in the overload heat pipes.

Payakaruk et al. [4] investigated the effects of dimensionless parameters such as Bond, Froude, and Weber numbers on the heat transfer rate of an inclined thermosyphon. A copper tube filled with different working fluids under the filling ratios of 50%, 80%, and 100% was tested at the different inclinational angles. Based on the experiments, it was found that the filling ratio has no effect on the ratio of heat transfer characteristics at any angle to that of the vertical position.

In a different research study by Wong and Kao [5], the effect of fluid charge and wick size on the performance of a single tube horizontal heat pipe was investigated. A transparent heat pipe with two layers of copper mesh as the wick structure was employed in the study. Three different fluid charges, including...
80%, 100%, and 120% of the amount of water needed to saturate the wick structure, were selected for the tests. The heat load to the evaporator section was increased stepwise from 20 to 45 W. It was found that the combination of a fine bottom mesh layer and a fluid charge approximately saturating the wick could lead to the optimum thermal characteristics. It was also seen that with the underload fluid charge and wick with the fine mesh bottom layer, partial dry-out occurred in the evaporator section. The effect of the filling ratio on the steady-state heat transfer performance of a vertical two-phase closed thermosyphon was studied by Jiao et al. [6]. For this purpose, a theoretical model based on the flow patterns was developed. The filling ratio, which was defined as the volume ratio of filled liquid to the whole two-phase closed thermosyphon, is set at 4.5%, 6.37%, 10.1%, 11.8%, 13.5%, 15.2%, 18.5%, and 20.2% in this study. The study showed that the proper range of filling ratio, which can maintain a two-phase closed thermosyphon steady and effective, is between the upper boundary of 10% and the lower boundary of 9%.

These mentioned research studies mostly used a single-tube heat pipe and water as the working fluid, and they have studied the effect of fluid charge on the heat pipe itself rather than at the air side. To this end, this research has been performed and the major objective was the experimental study of the effect of different fluid charges on the performance of a typical two-row HPHX at the air side. The HPHX was operating in the horizontal configuration, and an ozone-friendly refrigerant, R-134a (HFC family), was used as the working fluid.

**EXPERIMENTAL DESCRIPTION**

The following subsections describe the experimental setup as well as the experimental parameters examined in the present research. The experimental setup will be described first and the experimental parameters examined will be given next.

---

### Experimental Setup

To study the thermal performance of the HPHX, a test rig consisting of an air duct of 0.42 × 0.35 m², an HPHX, an electric heater, and a variable-speed fan was set up, as illustrated in Figure 1.

**Figure 1** Schematic diagram for the experimental setup (top view).

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Design specifications of the two-row HPHX</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator and condenser dimensions</td>
<td>420 mm width × 350 mm height</td>
</tr>
<tr>
<td>Adiabatic length (mm)</td>
<td>180</td>
</tr>
<tr>
<td>Number of rows</td>
<td>Two</td>
</tr>
<tr>
<td>Tube material</td>
<td>Copper</td>
</tr>
<tr>
<td>Outer tube diameter (mm)</td>
<td>13.4</td>
</tr>
<tr>
<td>Inner tube diameter (mm)</td>
<td>12.7</td>
</tr>
<tr>
<td>Longitudinal tube spacing (mm)</td>
<td>27.5</td>
</tr>
<tr>
<td>Transverse tube spacing (mm)</td>
<td>31.75</td>
</tr>
<tr>
<td>Vacuum level</td>
<td>The vacuuming process was achieved at a vacuum level as low as 10⁻³ mbar</td>
</tr>
<tr>
<td>Working fluid</td>
<td>R-134a: Liquid density (1.013 bar and 25°C): 1206 kg m⁻³, Boiling point (1.013 bar): –26.6°C, Latent heat of vaporization (1.013 bar at boiling point): 215.9 kJ kg⁻¹</td>
</tr>
<tr>
<td>Wick structure:</td>
<td></td>
</tr>
<tr>
<td>Number of mesh</td>
<td>100 mesh per inch</td>
</tr>
<tr>
<td>Number of layers</td>
<td>3</td>
</tr>
<tr>
<td>Wire material</td>
<td>Stainless steel 304 wire mesh</td>
</tr>
<tr>
<td>Wire diameter (mm)</td>
<td>0.11</td>
</tr>
<tr>
<td>Thickness (mm)</td>
<td>0.612</td>
</tr>
<tr>
<td>Wick porosity</td>
<td>0.67</td>
</tr>
<tr>
<td>Fins:</td>
<td></td>
</tr>
<tr>
<td>Material</td>
<td>Aluminium corrugated</td>
</tr>
<tr>
<td>Type</td>
<td>Wavy plate</td>
</tr>
<tr>
<td>Thickness (mm)</td>
<td>0.15</td>
</tr>
<tr>
<td>Density</td>
<td>12 fin per inch</td>
</tr>
</tbody>
</table>
The whole test rig was insulated by using a 13-mm insulation sheet to minimize heat exchange with the ambient environment. The fresh air is preheated in the condenser section and then passes through an electric heater to be heated to the desired temperature. The warmed air then enters the evaporator section. The two air streams flow through the HPHX in a countercflow configuration.

The design specifications of the HPHX are listed in Table 1. An HPHX consists of 22 copper tubes in two rows of 11 tubes, designed and fabricated, as illustrated in Figure 2. The tubes are in staggered configuration and three layers of stainless-steel wire mesh were pressed against the internal tube wall as a wick structure. The total length of each tube is 1020 mm, with 420 mm in the evaporator and condenser sections, and 180 mm for the adiabatic section. The desired air temperature of the evaporator inlet was maintained by using a 5.5-kW electric heater with a control box operating with an on/off position.

The dry-bulb temperature (DBT) of the various air states was measured using resistance temperature detector (RTD) sensors, and a data logging system was used to record and convert the output signal from the sensors to degrees Celsius. The recorded data were analyzed by DASYLab software. All the RTD sensors were calibrated in the temperature range of 10 to 50°C against a laboratory Dry-Well calibrator with an accuracy of ±0.1°C. An extra duct with a straightener was added to the evaporator outlet to determine the air velocity based on ANSI/ASHRAE [7] requirements.

In order to establish the desired coil face velocities, the log-Tchebycheff rule for rectangular ducts [7] was employed for the velocity traverse study. For this purpose, the duct cross section was divided into 25 grid points and the mean duct air velocity was evaluated by measuring air velocity at the grid points. A digital flowmeter with an accuracy of ±0.015 m s⁻¹ was used to measure the air velocity.

**Experimental Parameters**

Parameters considered in this research were fluid charge, coil face velocity, and evaporator inlet air temperature.
According to Figure 1, process 3–4, the heat transfer occurs from air to the refrigerant, which leads to a temperature reduction in the passing air stream. Therefore, this section can be used as a precooling process in heating, ventilation, and air conditioning (HVAC) systems. For this purpose, the HPHX could be installed in air-conditioning systems as a way to precool the supply of air before it reaches the cooling coils. As a result, the energy demand of the air-conditioning system will be reduced. Normally, in these situations, the HPHX evaporator section will be in direct contact with the ambient air; therefore, to study the effect of different fluid charges on the HPHX performance, a range of temperatures close to the temperatures in Kuala Lumpur, Malaysia, were selected as the evaporator inlet temperatures. The average daily temperature of Kuala Lumpur is at 23.1–33.5°C [8]. Coil face velocities representing those that could typically occur in practice were established in the system. The condenser inlet air DBT was maintained at a nominal temperature of 24°C as the representative DBT of the climate chamber for future research.

The porosity of the wick was estimated based on the porosity model used for wire meshes [9]. To calculate the fluid charge, the void volume in the wick and the wick porosity were considered. Five different fluid charges were selected for the present research: 80%, 90%, 100%, 110%, and 120% of the fluid to saturate the wick, to cover the underload and overload charges. These correspond to 15.52 g, 17.40 g, 19.40 g, 21.34 g, and 23.28 g of R134a, respectively. The 100% charge is defined as 100% charge = volume of the total wick × porosity of the wick.

The working fluid was filled in the tubes after the evacuation of the noncondensable gases using a vacuum pump.

Thus, in this research, tests were conducted as follows:

- Fluid charge values of 80%, 90%, 100%, 110%, and 120%.
- Face velocity values in the range of 1.5 to 2.5 m s⁻¹ with a 0.2 increment.
- Evaporator inlet DBT values of 27, 29, 31, 33, and 35°C.

The DBT 24°C was not tested in this series of experiments because of the room-temperature limitation. Therefore, there was a total of 150 (5 × 6 × 5) experimental test runs. Data were categorized based on the nominal values just listed and are presented and discussed in the present paper.

All data recorded were at 1-s intervals, and for each measured variable, the mean value for 60 continuous data items was calculated to guarantee proper averaging. During the data collection period, the system was operating at steady-state equilibrium conditions, defined when every temperature reading varied by less than 0.1°C in 1 s. The mean values were used for the calculations.

**THEORY RELEVANT TO THE PRESENT RESEARCH**

In the present research, sensible effectiveness was employed to determine the HPHX performance at the air side. The effectiveness is the most relevant parameter to characterize the HPHX performance [10]. The effectiveness in Eq. (1) is defined as the ratio of the actual heat transfer to the maximum possible heat transfer in the heat exchanger,

$$\varepsilon = \frac{Q_{act}}{Q_{max}} \quad (1)$$

and

$$Q_{act} = m_e(h_3 - h_4) \quad (2)$$

and

$$Q_{max} = m_{min}(h_3 - h_1) \quad (3)$$

From Eqs. (1)–(3), the total effectiveness can be determined by

$$\varepsilon_{tot} = \frac{m_e}{m_{min}} \left(\frac{h_3 - h_4}{h_3 - h_1}\right) \quad (4)$$

where $h_4$ is the evaporator outlet air specific enthalpy (state 4), $h_3$ is the evaporator inlet air specific enthalpy (state 3), and $h_1$ is the condenser inlet air specific enthalpy (state 1) as illustrated in Figure 1; $m_e$ is the mass flow rate of the evaporator side and $m_{min}$ is the smaller of the mass flow rates in the evaporator and condenser sides (i.e., the mass flow rate which is smaller). However, in the present research, the mass flow rate is equal in the evaporator and condenser side. Therefore, Eq. (4) can be rewritten as

$$\varepsilon_{tot} = \left(\frac{h_3 - h_4}{h_3 - h_1}\right) \quad (5)$$

DBT can be used instead of specific enthalpy in Eq. (5) to calculate the sensible effectiveness as given in Eq. (6). In the HPHX under study, by assuming $h \cong c_p T$ for the air streams at the measuring points, the sensible and total effectiveness can be considered almost equal ($\varepsilon_{sen} = \varepsilon_{tot}$). Therefore, in the present study, the sensible effectiveness is used to determine the HPHX performance at the air side, and is obtained based on the temperature measurements:

$$\varepsilon_{sen} = \left(\frac{T_3 - T_4}{T_3 - T_1}\right) \quad (6)$$

To determine the sensible effectiveness, the following assumptions are made:

1. The HPHX operates as a sensible heat transfer device.
2. The HPHX operates under the steady-state condition during each test (i.e., the mass flow rate for both the sections is steady and the temperature at all measuring points is steady).
3. The HPHX is fully insulated and no external energy is supplied into or lost from it.
4. The energy transfer between the evaporator and condenser sections of the HPHX is equal and opposite, and since the mass flow rate is also equal in the two sections, and assuming $c_p$ is constant then $(T_3 - T_4) = (T_2 - T_1)$.
5. The airflow is steady and the air is well mixed (uniform) at the measuring points.
The uncertainty in calculating the energy balance ratio (EBR) and sensible effectiveness was estimated as about 0.7% and 0.6%, respectively. The method used for the uncertainty evaluation is based on the calculation of the relative error squares, referred to in the readings for $T_1$, $T_2$, $T_3$, and $T_4$, adding them and extracting the square root (root sum square method).

**SIMULATION OF THE HPHX**

The method using the effectiveness number of transfer units (NTU) was employed to predict the heat transfer performance of the HPHX in the present research [11]. For constant specific heat and heat transfer properties throughout the heat exchanger, the effectiveness can be written as [12]

$$\varepsilon = \frac{1}{1 - e^{-NTU(1-C_{\text{min}}/C_{\text{max}})}}$$

(7)

In the evaporator and condenser sections of the HPHX, the hot and cold air streams are in cross flow with the vapor inside the tubes. Because of the phase change in the heat pipe sections, the maximum heat capacity rate ($C_v$) is several orders of magnitude larger than the minimum heat capacity. Therefore, $C_{\text{min}}/C_{\text{max}} = C_e/C_v = C_c/C_v = 0$. This yields the effectiveness-NTU equations for a single row HPHX as [13, 14] as the following.

For the evaporator side:

$$\varepsilon_{\text{ev}} = 1 - \exp(-NTU)$$

(8)

For the condenser side:

$$\varepsilon_{\text{cn}} = 1 - \exp(-NTU)$$

(9)

The NTU value for the evaporator and condenser sides are as follows, respectively:

$$\text{(NTU)}_e = \frac{(UA)_e}{C_e}$$

(10)

$$C_e = (mc_p)_e$$

(11)

$$\text{(NTU)}_c = \frac{(UA)_c}{C_c}$$

(12)

$$C_c = (mc_p)_c$$

(13)

For an HPHX with $n$ rows of heat pipe tubes, the effectiveness-NTU equations are as follows [13, 14]:

For the evaporation side:

$$\varepsilon_{\text{en}} = \left(\frac{1 - \varepsilon_{\text{en}}}{1 - \varepsilon_{\text{en}}/C_c}\right)^n - 1$$

(14)

heat transfer engineering

For the condenser side:

$$\varepsilon_{\text{cn}} = \left(\frac{1 - \varepsilon_{\text{cn}}}{1 - \varepsilon_{\text{cn}}/C_c}\right)^n - \frac{C_e}{C_c}$$

(15)

For $C_e/C_v = 0$ and $C_c/C_v = 0$ the preceding equations will be in the form of

$$\varepsilon_{\text{en}} = 1 - (1 - \varepsilon_{\text{en}})^n$$

(16)

and

$$\varepsilon_{\text{cn}} = 1 - (1 - \varepsilon_{\text{cn}})^n$$

(17)

Then the overall effectiveness (total effectiveness) of the HPHX is written as:

If $C_e > C_c$

$$\varepsilon_{\text{tot}} = \left(\frac{1}{\varepsilon_{\text{en}}} + \frac{C_e}{C_c}\right)^{-1}$$

(18)

and

If $C_c > C_e$

$$\varepsilon_{\text{tot}} = \left(\frac{1}{\varepsilon_{\text{en}}} + \frac{C_c}{C_e}\right)^{-1}$$

(19)

where $\varepsilon_{\text{tot}}$ is the overall or total effectiveness of the HPHX; $NTU = UA/C_{\text{min}}$ is the number of transfer units, $U$ is the overall heat exchanger heat transfer coefficient, $A$ is the heat transfer area, and $C$ is the heat capacity rate.

$U$ can be written as [15]

$$U = \frac{1}{[R_{e.s.e} + R_{h.p} + R_{e.s.c}]A}$$

(20)

where $R_{e.s.e}$ is the external surface thermal resistance of the evaporator side, $R_{e.s.c}$ is the external surface thermal resistance of the condenser side, and $R_{h.p}$ is the thermal resistance of the heat pipe structure.

The dominant thermal resistances in the heat pipe structure; that is, the wall thermal resistance and wick structure thermal resistance were considered for the estimation of the thermal resistance of the heat pipe structure [9, 16].

The thermal resistance exists between the airflow and heat pipe external surface. Since the surfaces of the tubes were fitted with fins to improve the heat transfer, the mean heat transfer coefficient ($\alpha$) can be estimated using the correlation for the flow over the finned tube banks as

$$Nu = \alpha D_{\text{hydraulic}}/k_{\text{air}}$$

(21)

and the Nusselt number is given by (Hewitt [17])

$$Nu = 0.19 \left(\frac{a}{b}\right)^{0.2} \left(\frac{\xi}{D_{\text{tube}}}\right)^{0.18} \left(\frac{L_{\text{fin}}}{D_{\text{tube}}}\right)^{-0.14} \text{Re}^{0.65} \text{Pr}^{0.33}$$

(22)

where $a$ is the tube distance in a row, $b$ is the distance between the tubes in two successive rows, $L_{\text{fin}}$ is the fin length in the gas flow direction, $D_{\text{tube}}$ is the tube diameter, and $\xi$ is the fin spacing.
The influence of fluid charge on sensible effectiveness at face velocity of 1.7 m s\(^{-1}\).

The Reynolds number is determined based on the hydraulic diameter:

\[
\text{Re} = \rho V D_{hydraulic}/\mu \tag{23}
\]

The heat transfer in the external surface is usually estimated based on the fin efficiency [18]:

\[
\eta_{fin} = \tanh(mL_{fin})/(mL_{fin}) \tag{24}
\]

and

\[
m = \left(\frac{2\alpha_{fin}(1 + \delta_{fin}/L_{fin})}{k_{fin}b_{fin}}\right) \tag{25}
\]

where \(\alpha_{fin}\) in Eq. (25) is the external surface heat transfer coefficient and given by Eq. (21).

Then the thermal resistance of the external surface can be determined using Eq. (26):

\[
R_{ex} = \frac{1}{\eta_{fin}\alpha_{fin}A_{fin}} \tag{26}
\]

The thermal resistance for \(N\) tubular heat pipe wall is given by (Shah and Sekulic [15])

\[
R_{wall} = \frac{\ln(D_{o}/D_{i})_{wall}}{2\pi L_{tube}Nk_{wall}} \tag{27}
\]

Figure 3 The influence of fluid charge on sensible effectiveness at face velocity of 1.7 m s\(^{-1}\).
Three layers of wire mesh screen were used as the wick structure in the heat pipes. The thermal conductivity of multiple layers of screen mesh material with the working fluid is given by Chang [19] as

$$k_{eff} = \frac{k_f}{(1 + A)^2} \left\{ \psi^2 A \left[ \frac{\Psi A}{\psi - \pi B(1 - k_f/k_s)/2} \right. \right.$$ 

$$\left. + \frac{2[1 + A(1 - \psi)]}{\psi - \pi B(1 - k_f/k_s)/4} \right] + [1 + A(1 - \psi^2)] \right\}$$  \hspace{1cm} (28)

with $B = D_{wire}/t$, $A = D_{wire}/w$, $w$ is the opening width of the mesh, $D_{wire}$ is the wire diameter, and $t$ is the thickness of a single layer of wire mesh. For the multiple layers, $t = t_{wick}/n_0$.

In Eq. (28) $\psi$ is the constant parameter dealing with the contact condition of wires in the mesh. Then the thermal resistance of the wick for $N$ tubular heat pipes can be estimated by Shah and Sekulic [15] as

$$R_{wick} = \frac{\ln(D_0/D_i)_{wick}}{2\pi L_{wick} N k_{eff}}$$  \hspace{1cm} (29)

After calculation of the effectiveness from Eq. (18) or (19), the outlet temperature of the evaporator was obtained from the following equation [14]:

$$T_{e, outlet} = T_{e, inlet} - \epsilon_{tot}(T_{e, inlet} - T_{c, inlet})$$  \hspace{1cm} (30)

The comparisons between the experimental results and theoretical values were made for the heat pipes that were under the 100% fluid charge (i.e., the saturated wick).

**RESULTS AND DISCUSSION**

The energy balance ratio (EBR) is defined as the energy extracted from the air passing through the evaporator divided by the energy transferred to the condenser side. The EBR values for the tests are tabulated in Table 2. It can be seen from Table 2 that the EBR values for most of the tests examined are not equal to 1 (note: EBR value should be equal to unity if all the energy extracted from the air passing through the evaporator side were transferred to the air passing through the condenser side). There are three possible reasons for the EBR deviation from unity: possible intervening heat transfer between evaporator and condenser sections, possible heat transfer to the ambient air because of insufficient insulation, and not having a uniform temperature profile at the measuring stations because of the incomplete mixed air.

For checking any possible intervening heat transfer between the evaporator and condenser sides and any possible heat transfer to the surrounding air, the intervening heat transfer test was done with zero fluid charge for all the face velocities in the temperature range of study. It was found that for all the tests, the maximum temperature difference between the air streams in the evaporator and condenser sections was 0.1°C. Therefore, the heat transfer through conduction of tubes and heat transfer to the surrounding air were negligible. It was obvious that incomplete mixed air (not uniform) at the measuring points caused the deviation of the EBR from unity.

The influence of fluid charge on the sensible effectiveness is shown in Figures 3 and 4, which are the representatives of results for the range of conditions tested. It was observed that the sensible effectiveness was increased as the fluid charge increased from 80% to 100% and then slightly decreased from 110% to 120%. These results imply that at the fluid charges lower than the amount needed to saturate the wick (i.e., 100%),

---

**Figure 7** Experimental and theoretical comparison of evaporator outlet DBT to evaporator inlet DBT at face velocity of 1.7 m s⁻¹.

**Figure 8** Experimental and theoretical comparison of evaporator outlet DBT to evaporator inlet DBT at face velocity of 1.9 m s⁻¹.
the heat transport between the evaporator and the condenser through the working fluid is low. This might be due to the fact that although thermal resistance is low at fluid charges lower than 100%, nevertheless, the amount of working fluid (i.e., refrigerant) being insufficient to saturate the wick caused low heat transfer in the heat pipe. As the fluid charge increases, the performance of the HPHX gets better because of enough liquid in the heat pipes. When the fluid charge reaches to a level of overload (i.e., 120%), the sensible effectiveness slightly decreases. This may be because of the fact that when the fluid charge is significantly more than the amount needed to saturate the wick, it causes the evaporator to flood and subsequently disturbs the evaporation and condensation processes of the working fluid.

Figures 5 and 6 show the effect of the evaporator inlet DBT on the HPHX sensible effectiveness. It is evident that for fluid charges lower than the amount needed to saturate the wick, the sensible effectiveness was reduced as the evaporator inlet DBT increased. In other words, the heat transfer capability of the HPHX was diminished corresponding with the increase of the evaporator inlet DBT. This is because of the fact that by increasing DBT, the heat flux to the container increases and since thermal resistance in these amounts of fluid charges (80% and 90%) is lower than the other fluid charges, the possibility of the dry-out phenomenon in the evaporator can be expected. As a result, the evaporator works with a shorter effective length and its performance declines. At higher fluid charges, the effect of the evaporator inlet DBT on the sensible effectiveness was not significant and the sensible effectiveness was almost constant in the temperature range of the study.

The influence of a higher DBT on the sensible effectiveness was not studied since the main objective of the present research was to find out the optimum fluid charge in the temperature range of 27–35°C for the purpose of future research.

Evaporator outlet temperatures were calculated from Eq. (30) and compared with the experimental data, as shown in Figures 7–9 at three representative face velocities. The comparison of the experimental data and theoretical values indicates that there is an acceptable agreement between the experimental data and theoretical values with a maximum deviation of 3%. Moreover, the uncertainty of recorded data for the evaporator outlet temperature based on the RTD sensor accuracy indicates uncertainty about 1% using the root sum square method. The uncertainty of the experimentally measured evaporator outlet temperatures is shown as the error bands in Figures 7–9.

The sensible effectiveness range, which was the main purpose of the present experimental research, is tabulated in Table 3, which shows the sensible effectiveness range (min–max) for the face velocities in the temperature range of the study.

For convenience, the reference for sensible effectiveness ranges (min–max) for two representative face velocities are presented in Figures 10 and 11. As can be seen from the figures, the sensible effectiveness has the optimum amount at 110% fluid charge. In other words, the optimum sensible effectiveness of

<table>
<thead>
<tr>
<th>Fluid Charge (%)</th>
<th>Sensible Effectiveness (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>70</td>
<td>10.37–21.21</td>
</tr>
<tr>
<td>80</td>
<td>17.05–25.99</td>
</tr>
<tr>
<td>90</td>
<td>26.09–28.07</td>
</tr>
<tr>
<td>100</td>
<td>26.04–28.8</td>
</tr>
<tr>
<td>110</td>
<td>23.16–26.6</td>
</tr>
<tr>
<td>120</td>
<td>23.16–26.6</td>
</tr>
</tbody>
</table>

Figure 10 The influence of fluid charge on sensible effectiveness range (min–max) at face velocity of 1.5 m s⁻¹.

Figure 11 The influence of fluid charge on sensible effectiveness range (min–max) at face velocity of 2.5 m s⁻¹.
CONCLUSIONS

In the present research, the effect of fluid charge on the performance of a typical two-row horizontal HPHX was investigated. The HPHX was tested under different fluid charges that represented both underload and overload cases, and different face velocities in the range of 27–35°C for the evaporator inlet DBT. Inlet air temperature for the condenser was kept constant at 24°C during the tests. It was found that in the temperature range of the study, for all the face velocities, the sensible effectiveness was lower for the fluid charges under the amount required to saturate the wick, and this may be due to the insufficient working fluid to saturate the wick and possibility of a dry-out in the evaporator section. At the higher fluid charges, a slight decrease in the sensible effectiveness was observed, and this might be due to the heat pipe flooding and consequently disturbing the evaporation and condensation processes of the refrigerant. The comparison between the experimental data and simulated values for the saturated situation showed good agreement for the evaporator outlet temperature. Based on the experimental results, it was confirmed that the optimum effectiveness of the copper-R134a HPHX can be achieved at a fluid charge slightly exceeding the amount needed to saturate the wick.

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NOMENCLATURE

\( a \) 
- tube distance in a row in Eq. (22), mm

\( A \)
- heat transfer area, \( \text{m}^2 \)

\( B \)
- dimensionless parameter defined by \( D_{wire}/t \) in Eq. (28)

\( b \)
- distance between the tubes in two successive rows in Eq. (22), mm

\( c_p \)
- specific heat of the ambient air, J kg\(^{-1}\) K\(^{-1}\)

\( C \)
- heat capacity, J s\(^{-1}\) K\(^{-1}\)

\( D \)
- diameter, m

\( DBT \)
- dry bulb temperature, °C

\( EBR \)
- energy balance ratio

\( F.C \)
- fluid charge, %

\( HPHX \)
- heat pipe heat exchanger

\( h \)
- specific enthalpy, k J kg\(^{-1}\)

\( k \)
- thermal conductivity, W m\(^{-1}\) K\(^{-1}\)

\( L \)
- length, m

\( m \)
- mass flow rate, kg s\(^{-1}\)

\( NTU \)
- number of heat transfer units

\( Nu \)
- Nusselt number

\( N \)
- number of the tubes

\( n \)
- number of rows

\( nol \)
- number of layers

\( Pr \)
- Prandtl number

\( Q \)
- heat transfer rate, W

\( R \)
- thermal resistance, k W\(^{-1}\)

\( Re \)
- Reynolds number

\( RTD \)
- resistance temperature detector

\( T \)
- temperature, °C

\( t \)
- thickness, m

\( U \)
- heat transfer coefficient, W m\(^{-2}\) °C\(^{-1}\)

\( V \)
- coil face velocity, m s\(^{-1}\)

\( w \)
- opening width of the mesh, mm

Greek Symbols

\( \varepsilon \)
- effectiveness

\( \eta \)
- fin efficiency

\( \delta \)
- fin thickness, m

\( \alpha \)
- heat transfer coefficient, W m\(^{-2}\) K\(^{-1}\)

\( \rho \)
- density, kg m\(^{-3}\)

\( \mu \)
- viscosity, N s m\(^{-2}\)

\( \zeta \)
- fin spacing, m

\( \psi \)
- constant parameter dealing with the contact condition of wires in the mesh

Subscripts

\( act \)
- actual

\( c \)
- condenser

\( cn \)
- condenser number of rows

\( c1 \)
- single row condenser

\( e.s \)
- external surface

\( eff \)
- effective

\( e \)
- evaporator
en evaporator number of rows  
e1 single row evaporator  
f fluid phase  
h.p heat pipe  
i inner  
max maximum  
min minimum  
o outer  
S solid phase  
sen sensible  
tot total  
v vapor  

REFERENCES


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