Absorption Heat Pump/Refrigeration System Utilizing Ionic Liquid and Hydrofluorocarbon Refrigerants

The ionic liquid butylmethylimidazolium hexafluorophosphate (bmim)(PF₆) and five different hydrofluorocarbon refrigerants were investigated as the working fluid pairs for a waste-heat driven absorption heat pump system for possible applications in electronics thermal management. A significant amount of the energy consumed in large electronic systems is used for cooling, resulting in low grade waste heat, which can be used to drive an absorption refrigeration system if a suitable working fluids can be identified. The Redlich–Kwong-type equation of state was used to model the thermodynamic conditions and the binary mixture properties at the corresponding states. The effects of desorber and absorber temperatures, waste-heat quality, and system design on the heat pump performance were investigated. Supporting experiments using R134a/(bmim)(PF₆) as the working fluid pair were performed. Desorber and absorber outlet temperatures were varied by adjusting the desorber supply power and the coolant temperature at the evaporator inlet, respectively. For an evaporator temperature of 41 °C, which is relevant to electronics cooling applications, the maximum cooling-to-total-energy input was 0.35 with the evaporator cooling capability of 36 W and the desorber outlet temperature in the range of 50 to 110 °C. [DOI: 10.1115/1.4007111]

1 Introduction

The energy consumption of large scale information technology equipment and infrastructure, such as data centers, has been increasing, as the demand for computing and storage capacity rises. The Environmental Protection Agency reported that the energy consumption of servers and data centers in 2006 was more than double that consumed in year 2000 [1]. Cooling is one of the major factors in total energy consumption in data centers accounting for about 30%–50% of the total power drawn from the grid [2]. A variety of novel alternative thermal solutions for electronics cooling have been reported, including thermosyphon [3], loop heat pipes [4], electroosmotic pumping [5], stacked microchannels, [6], impinging jets [7], thermoelectric microcoolers [8], vapor compression refrigeration [9], and absorption based refrigeration systems [10,11]. The cooling systems can be categorized into passive and active; passive cooling systems utilize capillary or gravitational force to circulate the working fluid. Active cooling systems are driven by a pump or a compressor for augmented cooling capacity and improved performance. Active systems driven by a compressor, refrigeration/heat pump systems, offer a further increase in heat removal by insertion of a negative thermal resistance into heat flow path [9]. Among the different candidate technologies, absorption refrigeration offers the compactness, relatively high coefficient of performance (COP) (assuming waste heat is available), making it an attractive option for cooling of high power, high performance electronics [12]. In places where an excess of heat of sufficient quality is available, absorption refrigeration offers an opportunity to recycle the thermal energy, which would otherwise be wasted.

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feasibility of the absorption heat pump system using IL/refrigerant working fluid pair and to evaluate the system performance as a function of control inputs, such as power input to the desorber, and the evaporator temperature. The friction loss and pumping power consumption of the absorption system using various working fluids were evaluated separately [18].

2 Thermodynamic Model

The principal features of the absorption refrigeration cycle are shown in Fig. 1. The cycle resembles that of the vapor compression refrigeration (heat pump) system, except the vapor compressor is replaced with a thermochemical process consisting of an absorber, a liquid pump, a solution heat exchanger, a desorber, and an expansion device. The pressurization in the thermochemical process starts in the absorber, where the refrigerant vapor from the evaporator (state point 2) is isothermally absorbed into the strong-IL solution (state point 10), resulting in a weak IL solution (state point 5) from IL via desorption from the weak-IL solution by the addition of heat (preferably high quality waste heat). The strong-IL solution (state point 10), resulting in a weak IL solution (state point 5) from IL via desorption from the weak-IL solution by the addition of heat (preferably high quality waste heat). The strong-IL returns to the absorber through the solution heat exchanger and expansion device. The condensation/absorption process at the absorber and vaporization/desorption process at the desorber both occur in the liquid phase. This allows use of a liquid pump to create the pressure difference between condenser and evaporator. Although the presence of the absorber and desorber increases the overall system volume, the displacement volume and power consumption for liquid compression are much smaller than those for vapor compression. Table 1 summarizes the main components/processes of an absorption heat pump system using IL/refrigerant mixture as a working fluid.

Benchmarking of the refrigerant/ionic liquid combination and system-level simulations were carried out. Energy and mass conservation equations for all components comprising the system were simultaneously solved to determine the heat and workloads. The overall energy balance for the system is given in Eq. (1):

\[ \dot{Q}_d + \dot{Q}_e + W_p = \dot{Q}_a + \dot{Q}_c \]  

where \( W_p \) is the liquid pump work and \( \dot{Q} \) is the heat input/output. The subscripts d, c, and a represent the desorber, evaporator, condenser, and absorber, respectively. All values of heat are expressed as positive (magnitude) values regardless of the direction (in or out) of heat flow. The solution heat exchanger is assumed to be ideal, i.e., to follow an isobaric process and with 100% heat exchanger efficiency. The energy conservation for a subsystem consisting of a regenerative heat exchanger and a pump is given by Eq. (2):

\[ (h_9 - h_8)(m_w - m_t) = (h_7 - h_5)m_w - W_p \]  

where h is enthalpy; the subscript numbers correspond to the locations shown in the system diagram, Fig. 1, and \( m_w \) and \( m_t \) are mass flow rates of the weak-IL solution and refrigerant leaving the desorber. Energy conservation for the desorber is given by Eq. (3):

\[ \dot{Q}_d = h_8(m_w - m_t) + h_3m_t - h_7m_w \]  

Similarly, heat rejected at the absorber is given by Eq. (4):

\[ \dot{Q}_a = h_5m_w - h_{10}(m_w - m_t) - h_2m_t \]  

Energy conservation for the condenser and the evaporator yield the respective heat loads, Eqs. (5) and (6):

\[ \dot{Q}_c = (h_4 - h_3)m_t \]  

\[ \dot{Q}_e = (h_2 - h_1)m_t \]  

The cooling-to-total-energy (CE) is then defined as the heat removed at the evaporator divided by the power supplied to the desorber and the pump, Eq. (7):

\[ CE = \frac{\dot{Q}_e}{\dot{Q}_d + W_p} \]  

In this analysis, the RK equation of state (EOS) was used to calculate the thermodynamic properties of the fluids. Binary interaction parameters were introduced to improve the accuracy of the model. Several assumptions were made for convenience in computation: (1) pressure drops in the system heat exchangers are neglected; (2) the expansion process is isenthalpic; (3) the compression process is isentropic; (4) state 4 is saturated liquid refrigerant; (5) state 2 is saturated vapor refrigerant; and (6) the vapor quality at state 10 and state 5 is zero.

The general RK–EOS can be written in the following form, Eqs. (8)–(10) [19]:

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**Table 1 Main components and the processes of an absorption refrigeration system using IL/refrigerant as working fluids**

<table>
<thead>
<tr>
<th>Component</th>
<th>State</th>
<th>Process</th>
</tr>
</thead>
<tbody>
<tr>
<td>Evaporator</td>
<td>1 → 2</td>
<td>Heat absorption from chip</td>
</tr>
<tr>
<td>Absorber</td>
<td>2, 10 → 5</td>
<td>Refrigerant absorption/condensation into IL</td>
</tr>
<tr>
<td>Pump</td>
<td>5 → 6</td>
<td>Isentropic pressurization</td>
</tr>
<tr>
<td>Solution HX</td>
<td>6 → 7, 8 → 9</td>
<td>Regenerative preheating</td>
</tr>
<tr>
<td>Desorber</td>
<td>7 → 8, 3</td>
<td>Refrigerant desorption/vaporization from IL</td>
</tr>
<tr>
<td>Condenser</td>
<td>3 → 4</td>
<td>Heat rejection to ambient</td>
</tr>
<tr>
<td>Expansion device</td>
<td>4 → 1, 9 → 10</td>
<td>Isenthalpic expansion</td>
</tr>
</tbody>
</table>
The fugacity coefficient can be calculated from Ref. [20]:

\[
p = \frac{RT}{V - b} - \frac{a(T)}{V(V + b)}
\]

\[
a(T) = 0.42748 \frac{R^2 T^2}{P_c} x(T)
\]

\[
b = 0.08664 \frac{RT_c}{P_c}
\]

The subscript \( c \) represents the critical properties of the substance. \( P \) is the pressure, \( T \) is the temperature, \( V \) is the molar volume, \( R \) is the gas constant, and \( a \) and \( b \) are constants. The temperature-dependent function of the \( x \) parameter is expressed by Eq. (11):

\[
x(T) = \sum_{i=0}^{3} \beta_i (1/T_c - T)^i, \quad T_c \equiv T/T_c
\]

The parameter \( \beta_0 \) is determined so as to yield the vapor pressure of each pure compound, the refrigerant and the ionic liquid alone [16]. The critical properties along with \( \beta \) values are summarized in Table 2.

Three binary interaction parameters (BIPs) \( \tau, l, \) and \( k \) were introduced in the \( a \) and \( b \) parameters for \( N \) component mixtures [20],

\[
a = \sum_{i,j=1}^{N} \sqrt{a_i a_j f_{ji}(T)(1 - li_j)x_i x_j}, \quad a_i = 0.42748 \frac{R^2 T^2}{P_{ci}} x(T)
\]

\[
f_{ji}(T) = 1 + \tau_{ij}/T, \quad \text{where} \quad \tau_{ij} = \tau_{ji} \text{ and } \tau_{ii} = 0
\]

\[
b = \frac{1}{2} \sum_{i,j=1}^{N} (b_i + b_j)(1 - li_j)(1 - ki_j) x_i x_j, \quad b_i = 0.08664 \frac{RT_{ci}}{P_{ci}}
\]

where, \( li_j = li_j; li_i = 0; ki_j = ki_j; ki_i = 0. \)

At equilibrium for an \( N \)-component system, the chemical potential of each species/phase must be the same giving Eq. (15):

\[
x_i^a \phi_i^a = x_i^b \phi_i^b = x_i^c \phi_i^c = \ldots, \quad (i = 1, \ldots, N)
\]

where \( x_i^a, x_i^b, x_i^c, \ldots \) are the mole fractions of the \( i \)-th species in the \( a, b, c, \ldots \) coexisting phases and \( \phi_i^a, \phi_i^b, \phi_i^c, \ldots \) are the fugacity coefficients of the \( i \)-th species in the \( a, b, c, \ldots \) corresponding phases. The fugacity coefficient can be calculated from Ref. [20]:

\[
\ln \phi_i = \int_{-\infty}^{\infty} \left\{ \frac{\partial Z}{\partial n_i} \right\}_{T, V, n_j} \frac{1}{n_i} \frac{dV}{V} - \ln Z
\]

\[
Z = \frac{PV}{RT}
\]

The equilibrium condition, Eq. (15), is used to compute the solubility of a refrigerant in an IL at a given temperature and pressure. An iterative computational code was used to find the BIPs for each binary working fluid mixture of a refrigerant and an IL.

The objective function was to minimize the nonlinear least square difference between the computed pressure and measured pressure value with the equilibrium condition (Eq. (15)) used as a constraint. The solubility data for HFC and (bmim)(PF\(_6\)) mixtures are taken from Shiflett and Yokozeki [21]. The optimum BIPs found for different working fluids are summarized in Table 3.

The BIPs found were used to generate a solubility plot at different temperature and pressure conditions. Figures 2–6 show the solubility of various HFCs in (bmim)(PF\(_6\)), where the different lines are isotherms modeled by the EOS, and the symbols indicate experimental data at the same temperatures used to develop the EOS correlations. It can be seen that the quality of the RK correlation is adequate for an entire range of experimental parameters to be useful for the system performance calculations.

The EOS was then used to find the enthalpy values at the point of interest, Eq. (18):
H = H^R + \int_{T_0}^{T} \sum_{i=1}^{N} x_i C_{pi}^0 dT + H_0 \quad \text{(18)}

where $H^R$ is the residual enthalpy, $T_0$ is the reference temperature set as 273.15 K, and $H_0$ is an arbitrary constant (an enthalpy at the reference state). The ideal-gas heat capacity of the $i$-th species, $C_{pi}^0$, is modeled in Eq. (19):

$$C_{pi}^0 = C_{pi}^{0} + C_{pi}^{1} T + C_{pi}^{2} T^2 + C_{pi}^{3} T^3$$ \quad \text{(19)}

The coefficients in Eq. (19) are given in Table 4. Using the heat capacity for each species in the mixture, the enthalpy for the RK–EOS is given by Eq. (20):

$$H = \left( \frac{a}{b} - \frac{T}{b} \frac{da}{dT} \right) \ln \frac{V}{V + b} + RT(Z - 1) + \int_{T_0}^{T} \sum_{i=1}^{N} x_i C_{pi}^0 dT + H_0 \quad \text{(20)}$$

### 3 Experimental

An experimental setup for a laboratory scale absorption refrigeration system using R134a/(bmim)(PF$_6$) mixture as a working fluid was built and operated, as shown in Fig. 7. The microfluidic channel heat/mass exchangers were fabricated in copper and used for an evaporator, a condenser, an absorber, and a desorber. A magnetic gear pump was employed for the circulation of the R134a/(bmim)(PF$_6$) mixture. A 150 ml stainless-steel cylinder was installed for the separation of refrigerant-vapor from the mixture. Vapor/noncondensables were trapped in the 50 ml liquid-receiver located at the outlet of the condenser to guarantee liquid flow through the expansion device. Orifice-type metering valves were adopted for the expansion devices for fine adjustment of

<table>
<thead>
<tr>
<th>Pure compound</th>
<th>$C_{p}^0$ (J mol$^{-1}$)</th>
<th>$C_{p}^1$ (J mol$^{-1}$K$^{-1}$)</th>
<th>$C_{p}^2$ (J mol$^{-1}$K$^{-2}$)</th>
<th>$C_{p}^3$ (J mol$^{-1}$K$^{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(bmim)(PF$_6$) [15]</td>
<td>-2.214</td>
<td>0.57685</td>
<td>$-3.854 \times 10^{-4}$</td>
<td>$9.785 \times 10^{-8}$</td>
</tr>
<tr>
<td>R32 [16]</td>
<td>20.34</td>
<td>0.07534</td>
<td>$1.872 \times 10^{-4}$</td>
<td>$-3.116 \times 10^{-8}$</td>
</tr>
<tr>
<td>R125 [16]</td>
<td>16.58</td>
<td>0.33983</td>
<td>$-2.873 \times 10^{-4}$</td>
<td>$8.870 \times 10^{-8}$</td>
</tr>
<tr>
<td>R134a [16]</td>
<td>12.89</td>
<td>0.30500</td>
<td>$-2.342 \times 10^{-4}$</td>
<td>$6.852 \times 10^{-8}$</td>
</tr>
<tr>
<td>R143a [16]</td>
<td>5.740</td>
<td>0.31388</td>
<td>$-2.595 \times 10^{-4}$</td>
<td>$8.410 \times 10^{-8}$</td>
</tr>
<tr>
<td>R152a [16]</td>
<td>8.670</td>
<td>0.2394</td>
<td>$-1.456 \times 10^{-4}$</td>
<td>$3.392 \times 10^{-8}$</td>
</tr>
</tbody>
</table>
flow rates, as well as the system cooling capability. Kapton heaters were attached on the backsides of the evaporator and desorber to simulate the heat dissipation from chip, and waste heat as an energy source, respectively. The absorber and the condenser were cooled by secondary water loops. Pressure transducers were placed at the evaporator inlet, the absorber outlet, the desorber inlet, the condenser outlet, and the separator mixture outlet. The temperatures at the inlets and outlets of all components were measured using T-type (copper-constantan) thermocouples. The heat/mass exchangers, the separator and the liquid-receiver were insulated using glass-fiber insulation. The copper tubes (0.635 cm) were thermally insulated using elastomeric material (ethylene propylene terpolymer). As shown in Fig. 8, the heat/mass exchangers have microchannels with 1 mm cross-sectional area and the channel pitch of 1.5 mm. The evaporator and condenser have channel lengths of 1.5 cm and 3 cm, respectively, while the absorber and desorber have the channel lengths of 5 cm. Relatively large cross-sectional area of the absorber and the desorber was allowed to reduce the pressure drop due to the high viscosity of ILs.

To investigate the effect of desorber power (waste heat) input on the evaporator cooling capacity (varied up to ~40 W), the desorber power has been increased up to ~200 W to keep the evaporator temperature constant at 41 °C with the condenser and absorber coolant (secondary fluid) inlet temperatures of 22 °C. In addition, the effects of absorber and condenser coolant inlet temperatures were also investigated by varying both coolant inlet temperatures between 30 °C and 45 °C with the desorber power maintained at 120 W. The measured parameters were temperature, absolute pressure, and electrical power input. Uncertainty in the temperature reading was 0.1 K for calibrated thermocouples relative to each other. The uncertainty in absolute pressure measurement was 0.25% of the maximum range 2068.43 kPa (300 psi). The uncertainty in the output value of the electrical power transducer is given as 0.14% of the measured value. The heat transfer at the evaporator $Q_e$ and desorber $Q_d$ were measured in the experimental setup. The ratio of $Q_e/Q_d$ is defined here as $CE_{th}$, which is the same as CE in Eq. (7) except the liquid pump work has been neglected, giving Eq. (21):

$$CE_{th} = \frac{Q_e}{Q_d}$$  (21)

The pumping work was ignored in Eq. (21) because the experimental system was not designed for efficient pumping due to the excessive use of sensors and piping. In addition, the pumping work is normally a very minor part of the total energy input (denominator of Eq. (7)). Applying error propagation analysis to Eq. (21) gives the estimated uncertainty of 0.2%.

### Table 5: Theoretical performances and state conditions ($T_f/T_e/T_d = 35/50/25\,\text{°C}, \quad Q_e = 100\,\text{W}$)

<table>
<thead>
<tr>
<th>Fluid pair</th>
<th>$P_e, P_d$ (kPa)</th>
<th>$P_{cr}, P_{cs}$ (kPa)</th>
<th>$f$</th>
<th>$x_s$ (mole %)</th>
<th>$x_w$ (mole %)</th>
<th>$Q_d$ (W)</th>
<th>CE</th>
</tr>
</thead>
<tbody>
<tr>
<td>R32/(bmim)(PF$_6$)</td>
<td>3144</td>
<td>1688</td>
<td>4.40</td>
<td>55.9</td>
<td>76.4</td>
<td>184.9</td>
<td>0.533</td>
</tr>
<tr>
<td>R125/(bmim)(PF$_6$)</td>
<td>2540</td>
<td>1376</td>
<td>9.48</td>
<td>20.6</td>
<td>36.2</td>
<td>542.4</td>
<td>0.180</td>
</tr>
<tr>
<td>R134a/(bmim)(PF$_6$)</td>
<td>1319</td>
<td>665</td>
<td>6.60</td>
<td>29.4</td>
<td>49.7</td>
<td>250.7</td>
<td>0.395</td>
</tr>
<tr>
<td>R143a/(bmim)(PF$_6$)</td>
<td>2310</td>
<td>1262</td>
<td>47.0</td>
<td>17.3</td>
<td>22.3</td>
<td>842.5</td>
<td>0.114</td>
</tr>
<tr>
<td>R152a/(bmim)(PF$_6$)</td>
<td>1178</td>
<td>596</td>
<td>8.14</td>
<td>32.7</td>
<td>53.6</td>
<td>232.8</td>
<td>0.426</td>
</tr>
</tbody>
</table>

### 4 Results and Discussion

The system efficiency was evaluated at different desorber outlet temperatures $(T_d)$, which is the highest operating temperature in the system. The temperature of the heat source (e.g., waste heat) for powering the system needs to be higher than this temperature. Lower desorber outlet temperature is, therefore, preferred to utilize low thermodynamic quality waste heat. Figure 9 shows the effect of desorber outlet temperature on CE where the condenser and evaporator saturation temperatures are set at 50 °C and 25 °C, respectively, while the absorber outlet temperature was maintained at 35 °C. The state conditions, along with the CE value at a desorber outlet temperature of 80 °C are summarized in Table 5. The circulation ratio is defined as $f = m_f/m_e$, and $x_s$ and $x_w$ are the refrigerant mole fractions in the strong and weak mixture solutions, respectively. The two operating pressure conditions are determined by the saturated pressures at the corresponding condenser and evaporator temperatures. At high desorber temperatures, the CE tends to decrease, as more cooling will be required to maintain the desired condenser and absorber operating temperatures. Also, the remaining refrigerant content in the strong-IL solution is sufficiently low at high temperatures and the impact of a further increased desorber outlet temperature on the cooling capacity is small. Eq. (3) can be rewritten as Eq. (22):

$$\frac{Q_d}{m_e} = f (h_8 - h_7) - h_8 + h_3$$  (22)

The desorber outlet temperature changes both the enthalpies and the circulation ratio in Eq.(22). An increase in the desorber outlet temperature increases the enthalpies ($h_3$ and $h_8$), which

![Fig. 8 (a) Absorber microchannel and (b) cover plate with inlet and outlet ports](image)

![Fig. 9 Desorber outlet temperature effect on system performance for working fluid (bmim)(PF$_6$) and HFC refrigerants. $T_f/T_e/T_d = 50/25/35\,\text{°C}$](image)
requires larger desorber power supply \((Q_d)\) and thus the decrease of COP in Fig. 9. On the other hand, the circulation ratio is decreased by the increase of the desorber outlet temperature that leads to the reduction of the desorber power input. The circulation ratio was determined by Eq. (23):

\[
f_m = \frac{m_w}{m_r} = \frac{1 - x^m_s}{x^m_w - x^m_s} \tag{23}
\]

where \(x^m_s\) and \(x^m_w\) are the mass fractions of refrigerant in the strong- and the weak-IL solutions, respectively. Since as shown in the solubility curves in Figs. 2–6, an increase of the desorber outlet temperature brings about a reduction of refrigerant mass fraction of the strong-IL solution, the circulation ratio is decreased by Eq. (22) (Fig. 10). Pumping power is also reduced due to the decreased solution flow rate \((m_r)\). Finally, both the smaller desorber heat input and the pumping power result in improved CE at an increased desorber outlet temperature. This is, however, in conflict with the effect of the increased desorber outlet enthalpies, which calls for a reduction of CE. The results shown in Fig. 9 suggest that the effect of the desorber outlet temperature increase on the circulation ratio and the solution flow rate is not that profound. Thus, the CE tends to level off from the maximum value, which is likely due to the increase of desorber outlet enthalpies. Note that the CE trends for the five HFC refrigerants examined in Fig. 9 match those of the molar solubility of the refrigerants in \((\text{bmim})(\text{PF}_6)\), i.e., \(\text{R}32 > \text{R}152a > \text{R}134a > \text{R}125 > \text{R}143a\). This result emphasizes the importance of the affinity between absorbent (IL) and refrigerants in order to achieve the highest system performance.

When waste heat can be utilized to supply heat at the desorber, a more practical system COP is defined in Eq. (24):

\[
\eta = \frac{Q_e}{W_p} \tag{24}
\]

Equation (24) is the more usual figure of merit for absorption refrigeration/heat pump systems where waste heat is used and does not appear in the performance metric. Equation (24) removes the contribution of the heat input at the desorber from the system’s thermal efficiency (Eq. (7)). The coefficient of performance \(\eta\) is plotted with respect to desorber outlet temperature in Fig. 11. As previously discussed, at higher desorber outlet temperatures, the pumping work is reduced due to the increase of circulation ratio, which leads to an enhanced system efficiency. When waste (free) heat is provided at the desorber to drive the system, the operating pressure range becomes important as the pumping work is the product of the liquid volumetric flow rate and pressure difference between the absorber and desorber; i.e., \(W_p \sim V_l \times \Delta P\). It is surprising that R32 still shows relatively high performance, even though the pressure difference is twice as large as that of R152a and R134a (Table 5). This again suggests that high solubility of a refrigerant in the absorbent should be a primary factor in the selection of working fluids for the cycle.

Once the absorbent/refrigerant are selected, to ensure maximum efficiency of the system, the operating conditions must be optimized for a given pair of working fluids. To that end, the effect of absorber temperature can be observed by comparing Fig. 12 with Fig. 9, where the absorber inlet temperature was lowered from \(35^\circ\text{C}\) to \(26^\circ\text{C}\). The results show that the COP increases for all HFC refrigerants due to the increased refrigerant solubility at the absorber at lower temperatures. It is more evident from the Raoult’s law which gives Eq. (25):

\[
x_w \sim \frac{P_s}{P_{sat}(T_s)} \tag{25}
\]

At the lower absorber outlet temperature \((T_s = 26^\circ\text{C})\), the refrigerant mole fraction in the weak-IL solution is increased, resulting in the decrease of the circulation ratio and thus the increase of COP. Another noticeable change is the smaller COP difference between R32, R152a, and R134a. This is
because the solubility of all refrigerants saturates in (bmim)(PF₆) at nearly 0.97.

System design is also important for efficient operation of the system. A solution heat exchanger is an essential component, as heat that would be otherwise rejected to the ambient can be recovered to preheat the circulating solution. Therefore, the required heat supply at the desorber will decrease, which would lead to higher system efficiency. To quantify the efficacy of the heat exchanger to the system performance, the CE of a system without a solution heat exchanger was evaluated in Fig. 13. When compared to the results in Fig. 9, in the case of R143a the CE was increased 14-fold and on average the CE was doubled in the presence of the the heat exchanger. Also, it was observed that the CE reaches its maximum at the desorber outlet temperature around 80°C, which is different from the trend in Fig. 9. When the pressure is fixed, the solubility changes become less sensitive to increasing temperature (Figs. 2–6). This means that the desorber outlet temperature changes have a more significant effect on the solubility, as well as circulation ratio (Eq. (21)) at low desorber outlet temperatures. Therefore, at low desorber outlet temperature, the effect of the circulation ratio decrease is a more dominant mechanism than that of an increased desorber temperature difference, and thus, CE increases. However, at a high desorber outlet temperature, the variation of the circulation ratio is suppressed and the effect of an increased desorber temperature difference becomes dominant, which necessitates an increased desorber power supply and resulting CE decrease.

Figure 14(a) shows the experimentally measured CE_{Th} of the absorption refrigeration system using R134a/(bmim)(PF₆) mixture as the working fluid pair. Similar to the predictions in Fig. 13, CE_{Th}s range from 0.1 to 0.4 without solution heat exchanger. Also, the system performance reaches its maximum at the desorber outlet temperatures around 75°C ~ 80°C, which is consistent with the theoretical results shown in Fig. 13. Considering that practical imperfections of the system as well as the details of heat/mass exchange processes in the system components have not been fully reflected in the model, the system-level predictions are in fairly good agreement with the experimental data. A more detailed model would be useful for understanding/characterizing the effects of flow maldistributions in microfluidic channel, kinetics of absorption/desorption processes, liquid-vapor separations, and heat exchanger effectiveness. It was observed that with the smaller solution expansion valve opening, the system showed better performance (Fig. 14(a)). This is because less desorber power (Q_d) is consumed with the narrowed solution expansion valve flow path, which brings about the reduced solution flow rate (Fig. 14(b)). The evaporator cooling capacity increased with increasing desorber outlet temperature (Fig. 14(c)), since refrigerant flow rate (m_r) increases with the reduction of the circulation ratio (Eq. (21)). However, as the desorber outlet temperature increases, the circulation ratio reduction by the decrease of x_m becomes insignificant (Fig. 10), which results in the saturation of the evaporator cooling capacity enhancement in Fig. 14(c). It is interesting that the effect of the expansion valve opening is negligibly small on the evaporator cooling capacity, which needs further study.
5 Conclusion

Ionic liquid (bmim)(PF_6) and five HFC refrigerants were evaluated as working fluid pairs for an absorption refrigeration/heat pump system. The Redlich–Kwong EOS was employed to predict the mixture properties and to define the states of the thermodynamic cycle. The system efficiency showed strong dependency on the affinity between the refrigerant and absorbent. However, when waste heat is available, operating pressure ranges also become an important practical factor that defines the system efficiency. Furthermore, the results show that the operating conditions and system/component designs are essential in order to improve the system performance. A laboratory-scale experimental setup for the absorption heat pump system using R134a/(bmim)(PF_6) mixture as a working fluid was built to evaluate the feasibility of the system. With the desorber outlet temperature ranging from 50°C to 110°C, the system was operated with the maximum COP of 0.35 and evaporator cooling capacity of 36 W without a solution heat exchanger. As predicted, adding a solution heat exchanger could boost the COP up to 0.8, and it is expected that the adjustment of the operating conditions can bring further increase of COP. Further studies on the system performance optimization, more efficient desorber and absorber designs, and low vapor pressure, high solubility IL/refrigerant combinations are suggested.

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Nomenclature

\( CE \) = cooling-to-total-energy
\( CE_{Th} \) = cooling-to-total-thermal energy
\( \text{COP} \) = coefficient of performance
\( C_{p} \) = ideal-gas heat capacity (J/mol)
\( f \) = circulation ratio
\( H \) = molar enthalpy (J/mol)
\( H^{R} \) = residual enthalpy (J/mol)
\( h \) = specific enthalpy (J/kg)
\( k, l \) = adjustable binary interaction parameter
\( m \) = mass flow rate (kg/s)
\( P \) = pressure (Pa)
\( Q \) = heat transfer rate (W)
\( R \) = gas constant, 8.314 J/mol-K
\( T \) = temperature (K)
\( V \) = molar volume (cm^3/mol)
\( W_{p} \) = pumping work (W)
\( x_{w} \) = liquid phase refrigerant mole fraction
\( x_{m} \) = liquid phase refrigerant mass fraction

Greek Symbols

\( \eta \) = system efficiency utilizing waste heat
\( \varphi \) = fugacity coefficient

Subscripts

0 = reference state
1,…, 10 = state numbers indicated in Fig. 1
a = absorber
c = condenser
d = desorber
e = evaporator
l = liquid
r = refrigerant
s = strong-ionic liquid solution
sat = saturation
w = weak-ionic liquid solution

References


