Potential of Latent Heat Load Removal by Activated Carbons for Energy Savings of Air Conditioning Systems

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Abstract. The study aims to achieve energy savings of air conditioning systems using desiccant dehumidification with low-cost adsorbent. For this purpose, two existing activated carbons were evaluated in terms of water adsorption capacity. The theoretical calculation of effective adsorption from the water adsorption isotherms revealed that the studied activated carbons have promising potential in dehumidification application. In addition to that, the improvement of cooling COP of air conditioners was predicted by simulation of heat pump cycle. The results showed that the COP improved by 50%, which was equivalent to the reduction of electricity input by 30%, with the raise of evaporation temperature from 8.5°C to 18.5°C. The results encouraged the separation of dehumidification load from air conditioners to achieve large energy savings.

1. Introduction

Energy consumption for air conditioning is drastically increasing in Southeast Asian countries, which are under rapid development. On the one hand, penetration of air conditioners to the whole country will significantly improve the amenity of residential spaces, but on the other hand, it would cause serious problems both at a local level and at a global level. On a local level, the power supply would become unstable due to the insufficiency of power generation capacity. Besides, on a worldwide scale, large electricity consumption is responsible for global warming due to the emission of carbon dioxide. Therefore, it is essential to reduce cooling demand and use more efficient alternative air conditioning.

Most countries in Southeast Asia belong to a tropical climate zone where temperature and humidity are high. Cooling load of air conditioners is not only sensible cooling of air but also latent heat removal, which means dehumidification, in this case. Dehumidification by air conditioners is an energy-consuming process because air should be cooled to its dew point to remove moisture as condensate. Therefore, the evaporation temperature of the refrigerant in air conditioners has to be very low, which decreases the coefficient of performance (COP) of air conditioners.
In this context, we focused on desiccant air conditioning capable of minimizing electricity consumption for air conditioning by effectively utilizing low-grade thermal energy. Desiccant air conditioning systems use adsorbent to remove moisture from air directly, and the adsorbent does not require high-quality energy, such as electricity or gas, for regeneration. It is the most effective combination with solar thermal energy that is abundant in a tropical climate. One of the most critical issues is, however, the cost of installation. Especially, novel adsorbents are expensive, and it will raise the system cost. Therefore, we aim to develop a desiccant air conditioning using locally available low-cost materials as adsorbent.

Activated carbon is one of the most promising materials for low-cost adsorbents because it can be produced from various materials, including agricultural residues. The cost of the activation process would also be reduced by using physical activation. The available data of water adsorption by activated carbons are not much, however, because generally activated carbon surface is hydrophobic and water adsorption uptake at lower relative pressure is minimal. Nevertheless, some data showed a high water adsorption capacity by activated carbons, especially at relative pressure larger than 0.6 [1,2].

Using water adsorption isotherms available from literatures, the paper discusses the capability of activated carbon as an adsorbent for desiccant dehumidification and the possibility of energy saving of air conditioning by dehumidification using activated carbons.

2. Water adsorption isotherm

Measurement data of water adsorption onto promising activated carbons were taken from the literature [1], and the adsorption isotherm was modelled as Henry and Sips isotherm equation [3] given as Eqs.(1)-(3).

\[
W = \beta K_H \phi + (1 - \beta) \frac{W_m (K_S \phi)^{1/n}}{1 + (K_S \phi)^{1/n}} \\
K = K_0 \exp \left( -\frac{\Delta H}{RT} \right) \\
\beta = \exp \left( -\alpha \phi \right)
\]

Where, \( W \) and \( W_m \) denote equilibrium adsorption uptake and maximum adsorption uptake, respectively, and \( \phi \) denotes relative pressure. \( K_H \) and \( K_S \) are constants for Henry type and Sips type isotherms, and both of them are given by the form of Eq.(2), where \( R \) is the universal gas constant, \( T \) is temperature, and \( \Delta H \) is adsorption heat. \( \beta \) is a function of relative pressure and given by Eq.(3). \( n \) and \( \alpha \) along with \( W_m \), \( K_{OH}, \Delta H_H, K_{OS}, \Delta H_S \) are fitting parameters.

Figure.1 shows adsorption isotherms for adsorption and desorption, which were reproduced from the data of the activated carbon SAC31 and P20 in [1]. The values of fitting parameters are summarized in Table 1. The adsorption and desorption isotherms of both samples were modeled by Henry and Sips equation except for the adsorption isotherm of P20, which was simply fitted by a sigmoid function due to difficulty of the fitting. The fitting of the desorption isotherm of P20 was not proper at the lower relative pressure region, but it was acceptable because the model would underestimate the desorption capacity due to higher equilibrium adsorption in desorption.
3. Dehumidification capacity under Indonesian climate

The dehumidification capacity of adsorbent can be defined as the effective adsorption amount under given temperature and humidity conditions, which is calculated as a difference in equilibrium adsorption between adsorption and desorption. The adsorption condition, which is equivalent to outdoor air condition, was given as dry-bulb temperature of 32 °C and relative humidity of 80 % considering the weather condition of Jakarta, Indonesia. The desorption condition is given by regeneration air temperature from 40°C to 80°C with the humidity ratio at the outdoor air condition, which is 24 g/kg.

The calculated dehumidification capacity is depicted in Fig.2. The dehumidification capacities at different regeneration temperatures were compared between activated carbons and typical silica gels, silica gel types A and B. The adsorption isotherms of silica gels were also taken from the literature [4], and it was reproduced as in Fig.3. The hysteresis between adsorption and desorption was ignored in case of the silica gels because of a lack of data and also it was expected to be small.
The results showed that activated carbons had high dehumidification capacities. It was notable that the dehumidification capacity of SAC31 was approximately twice as much as that of Silica gel type B. In addition, it was interesting that P20 showed a better performance than SAC31 at the regeneration temperature of 40°C. The lower regeneration temperature results in a higher relative pressure in the desorption process. Although P20 has a large hysteresis, the desorption isotherm curve sharply changes at slightly higher relative pressure than that of SAC31. Therefore, P20 could maintain sufficient effective adsorption with a regeneration temperature of 40°C. It should also be noted that the improvement in desorption capacities of SAC31, P20, and Silica gel type B was marginal when the regeneration temperature was raised to a certain level. This is reasonable for adsorbents with so-called S-shaped isotherm because the effective adsorption cannot increase with lowering relative pressure. In other words, these adsorbents would provide their best performances with low regeneration temperature, and that of 50°C would be sufficient for SAC31, for example.

This section discussed the dehumidification capacity of adsorbent, which would have a strong influence on the size of the desiccant dehumidification system. Generally speaking, the larger the dehumidification capacity is, the smaller the system size is. The performance of the system is, however, dominated not only by the dehumidification capacity of the adsorbent but also by the engineering design of the system that affects heat and mass transfer speeds. The dehumidification capacity is one of the critical aspects of the desiccant dehumidification system.
4. COP improvement of an air conditioner

The purpose of the desiccant dehumidification is to bear latent heat removal and to reduce the cooling load of air conditioners. If the air conditioners work only for sensible cooling, the cooling load would be roughly halved compared with the dew point cooling case, and the evaporation temperature of refrigerant could be raised. In this way, the electricity consumption of air conditioners will be largely reduced. Therefore, in this section, the effect of evaporation temperature on the cooling COP of air conditioners was analyzed by detailed thermodynamic cycle simulation.

4.1 Heat pump cycle simulation

The heat pump is the thermodynamic principle of air conditioners, and the thermodynamic performance of the heat pump can be predicted based on the thermophysical properties at state points. Fig.4(a) shows a schematic of a typical heat pump, and Fig.4(b) shows a heat pump cycle on a pressure-enthalpy diagram. In our simulation, not only the thermophysical properties at each state but also heat exchange processes in the evaporator and in the condenser were calculated. Owing to the limitation in availability of the heat exchangers’ specifications, the heat exchangers were modeled as double tube counter flow type heat exchangers, which was used in our group’s experimental setup. The heat transfer model is given in Eqs.(4)-(6).

\[
\frac{dh_r}{dz} = \frac{4a_r}{G_rD_{ei}}(T_t - T_r)
\]

\[
\frac{dh_w}{dz} = \frac{4a_w}{G_wD_{eo}}(T_t - T_w)
\]

\[
T_t = \frac{\alpha_rD_zT_r + \alpha_wD_zT_w}{\alpha_rD_1 + \alpha_wD_2}
\]

Where, \( h \) is enthalpy and \( T \) is temperature. The subscripts \( r \) denotes refrigerant and \( w \) denotes heat transfer fluid (HTF), which was water in this case. The subscript \( t \) represents the heat transfer wall. The wall temperature, \( T_t \), was simplified as in Eq.(6) by ignoring the thermal mass of the tube due to high thermal conductivity of the wall. \( z \) denotes position from the refrigerant inlet. \( G \) is mass velocity. \( D_t \) and \( D_z \) are inner and outer diameters of the inner tube, while \( D_{ei} \) and \( D_{eo} \) are the equivalent diameters of the inner flow path and the annular flow path. \( \alpha_r \) and \( \alpha_w \) are heat transfer coefficients of the refrigerant side and HTF side, respectively. The heat transfer coefficients were calculated by the Dittus-Boelter equation for HTF as well as for single-phase flow of refrigerant. For condensation heat transfer of refrigerant, Yonemoto and Koyama’s equation [5,6] was used, and for evaporation heat transfer, Cavallini’s equation [7] was used. The model also deals with a pressure loss of refrigerant inside the heat transfer tubes by Baba’s equation [8].

The compressor and the expansion valve were not modeled in detail, but the isentropic efficiency was given in the calculation of the compression process. The expansion process was assumed as isenthalpic. The flow chart of the simulation was given in Fig.5.
4.2 Validation of the simulation model

The simulation results were compared with experimental results for validation. For this purpose, experimental data from our research group were used [9]. Although a variety of refrigerants, including refrigerant mixtures, were tested in the experiment, R32 was chosen for our purpose because it is one of the most popular refrigerants with relatively low global warming potential.

The degree of subcooling is one of the dominant parameters of COP of the heat pump cycle. In the experiment, the degree of subcooling changed with a pressure ratio of the compressor. The comparison of the degree of subcooling as a function of the pressure ratio between the simulation and experiment is depicted in Fig.6. It was shown that the simulation reproduced the relationship between the pressure ratio and the degree of subcooling, which are dominant in COP.
4.3 Improvement of COP

The temperature of HTF was changed to analyze the effect of the supply air temperature of air conditioners on the cooling COP. Although the HTF in the simulation was water, the behavior of the refrigerant side does not matter if the HTF was either water or air. Therefore, the HTF was kept as water because it was validated against the experiment.

Assuming that room air condition is 25°C with a relative humidity of 50%, the air has to be cooled to below 14°C for dehumidification by air conditioners, as shown in Fig. 8. To effectively remove the moisture from the air by condensation and to reach a lower humidity ratio, the air has to be sufficiently cooled, say 10°C. In this case, the enthalpy difference between the room air and the dehumidified air is about 20 kJ/kg. On the other hand, the supply air does not need to be so cold if the air conditioner is not responsible for dehumidification. The supply air can be closer to the room temperature in this case, and
if it is 20°C, for example, the enthalpy difference between the room air and the supply air is a quarter of the former case.

Moreover, the higher supply air temperatures of the air conditioner will improve the COP. The effect of the supply air temperature on the COP was predicted by the simulation. The simulation condition was given in Table 2. In the simulation, the pinch point temperature difference in the evaporator, which appeared at the refrigerant inlet, was given as 1.5 K, and it determined the evaporation pressure. The refrigerant outlet was the superheated vapor, and the degree of superheating depended on the evaporation pressure. The pressure ratio of the compressor was chosen so that the degree of subcooling at the refrigerant outlet of the condenser was kept at about 10°C. Fig.9 shows the effect of the evaporation temperature of refrigerant on the COP, and Fig.10(a) and (b) show the degree of superheating, the degree of subcooling, the pressure ratio and the discharge temperature from the compressor as functions of evaporation temperature.

![Figure 8. Sensible cooling and dehumidification processes in a psychrometric chart.](image)

**Table 2.** Condition of the simulation.

| Condition                        | Value                      |
|----------------------------------|----------------------------|
| Refrigerant                      | R32                        |
| Heat transfer fluid (HTF)        | Water                      |
| Cooling load                     | 2.0 kW                     |
| Temperature of HTF at the inlet of evaporator | 25°C                      |
| Temperature of HTF at the outlet of evaporator | 10°C ~ 20°C               |
| Temperature of HTF at the inlet of condenser | 30°C                      |
| Temperature of HTF at the outlet of condenser | 45°C                      |
| Pinch point temperature difference at evaporator | 1.5 K                      |
| Isentropic efficiency of compressor | 0.76                      |
Figure 9. The effect of evaporation temperature on the COP.

(a) The degree of superheating and the degree of subcooling

(b) The pressure ratio and the discharge temperature from the compressor

Figure 10. The effect of evaporation temperature on the thermodynamic cycle parameters.

The results showed that the COP was improved from 5.20 to 7.75 when the evaporation temperature was raised from 8.5°C, which corresponded to the HTF outlet temperature of 10°C, to 18.5, which corresponded to the HTF outlet temperature of 20°C. It is the improvement of COP by about 50%, and it is equivalent to the reduction of electricity input to the heat pump by about 30% considering that the electricity input is proportional to the inverse of the COP. The increase of COP was mainly due to the smaller pressure ratio with higher evaporation temperature under the condition of a constant degree of subcooling. The smaller pressure ratio also resulted in the lower discharge temperature from the compressor, which was also favorable to the cycle because it reduced the irreversible losses in the condenser.

5. Conclusions
The study investigated the energy saving potential of the dehumidification system using activated carbons. The dehumidification capacity of two promising activated carbons from literature was evaluated based on the outdoor air condition of a city in typical tropical climate, Jakarta, Indonesia. A detailed heat pump cycle simulation also predicted the improvement of COP by reducing the dehumidification load of air conditioners.

The evaluation of the water adsorption potential of the activated carbons revealed that SAC31 had a superior dehumidification capacity than silica gels. In addition to that, it does not require a high
regeneration temperature, and that of 50°C would be sufficient. The simulation analysis of the heat pump cycle showed that the COP of air conditioners would be largely improved by raising the evaporation temperature. The results showed that the COP improved by 50%, which was equivalent to the reduction of electricity input by 30%, with the rise of evaporation temperature from 8.5°C to 18.5°C. Along with reducing the cooling load itself, the electricity consumption of air conditioners would be drastically reduced by separating the dehumidification load from it.

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