Experimental study of a solid adsorption cooling system using flat-tube heat exchangers as adsorption bed

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Abstract

In this paper, a solid adsorption cooling system with silica gel as the adsorbent and water as the adsorbate was experimentally studied. To reduce the manufacturing costs and simplify the construction of the adsorption chiller, a vacuum tank was designed to contain the adsorption bed and evaporator/condenser. Flat-tube type heat exchangers were used for adsorption beds in order to increase the heat transfer area and improve the heat transfer ability between the adsorbent and heat exchanger fins. Under the standard test conditions of 80 °C hot water, 30 °C cooling water, and 14 °C chilled water inlet temperatures, a cooling power of 4.3 kW and a coefficient of performance (COP) for cooling of 0.45 can be achieved. It has provided a specific cooling power (SCP) of about 176 W/(kg adsorbent). With lower hot water flow rates, a higher COP of 0.53 can be achieved.

Keywords: Flat-tube type heat exchanger; Adsorption cooling

1. Introduction

In recent years adsorption refrigeration technology has been attracting more and more attention because it can save energy and is environmentally friendly. Adsorption cycles can be driven by low-grade waste heat or solar energy under 80 °C. They do not have to use ozone-depleting chlorofluorocarbons (CFCs) and do not need electricity or fossil fuels as driving sources.

In this study, silica gel–water has been selected as the adsorbent–adsorbate pair because of the low regeneration temperature of silica gel and the high latent heat of vaporization of water. Additionally, this working pair is non-toxic and steady.

Sakoda and Suzuki [13] have experimentally and analytically investigated a silica gel–water adsorption system for solar cooling. Cho and Kim [4] performed theoretical and experimental studies on a silica gel–water adsorption cooling system. The heat transfer rate of the condenser was found to be the most sensitive parameter having influence on the cold generation capacity.

Boelman et al. [2] have experimentally and numerically studied a commercially available Japanese silica gel–water adsorption chiller. The highest experimental COP values above 0.4 were obtained with hot water inlet temperature $T_{H,in} = 50$ °C and cooling water inlet temperature $T_{cool,in} = 20$ °C. With $T_{H,in} = 85$ °C, $T_{cool,in} = 30$ °C, and $T_{chill,in} = 14$ °C, COP values of 0.34–0.39 can be obtained.

Saha et al. and Boelman et al. [9,3] have analytically investigated the influence of operating conditions and heat exchanger UA-values on the performance of a single-stage, two-bed, silica gel–water adsorption chiller.

To utilize the low-temperature waste heat (50 °C), an advanced 3-stage silica gel–water adsorption cooling system was analytically investigated by Saha et al. [10,11] and experimentally by Saha and Kashiwagi [12]. With $T_{H,in} = 50$ °C, $T_{cool,in} = 30$ °C, and $T_{chill,in} = 12$ °C, a COP of 0.15 is attainable.

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Nomenclature

A	heat transfer area (m^2)	*Subscripts*	
COP	coefficient of performance	chill	chilled water
c_p	isobaric specific heat capacity (J/(kg K))	cool	cooling water
$\dot{m}$	mass flow rate (kg/s)	H	hot water
$\dot{Q}_{ref}$	cooling power (W)	in	inlet
$\dot{Q}_{H}$	heating power (W)	out	outlet
SCP	specific cooling capacity (W/kg)		
T	temperature (°C)		
U	heat transfer coefficient (W/(m^2 K))		

Oertel and Fischer [8] conducted an experimental study of the adsorption cooling system with two different working pair, silica gel–water and silica gel–methanol.

Chua et al. [6,5] have presented analytical studies of a two-bed silica gel–water adsorption chiller with a lump-parameter model and a distributed-parameter model, respectively. Chua et al. [7] have also analytically investigated a multi-bed adsorption chiller to improve the recovery efficiency of low-grade waste heat. With a multi-bed adsorption chiller the chilled water outlet temperature fluctuation can be suppressed.

Alam et al. [1] have numerically investigated the heat exchanger design effect on the system performance of silica gel–water adsorption refrigeration systems. They have shown that the system performance is very much sensitive to the cycle switching frequency. The cycle time of an adsorption refrigeration system is strongly dependent on the heat exchanger. The system performance would be lower if the cycle time is set far from the optimum value.

Silica gel–water adsorption chiller can be used in combination with solar energy because of the possibility using the low-grade solar energy under 80 °C, which can be easily obtained with flat-type collectors or vacuum tube collectors. Although the adsorption chillers are thought to be very promising in the future for the application of solar cooling and waste heat recovery, the wide spread of this technology is not yet possible. The reason is mostly attributed to the poor COP value and higher product cost of adsorption chillers.

In this study, an integrated, one-bed, closed-type silica gel–water adsorption chiller was developed. Flat-tube heat exchangers with corrugated fins were adopted as an adsorber and evaporator/condenser. Experiments were conducted under various operating conditions.

2. Description of adsorption chiller

Fig. 1 shows a schematic view of the adsorption chiller developed in this study. It mainly consists of two heat exchangers, which are made out of aluminum.

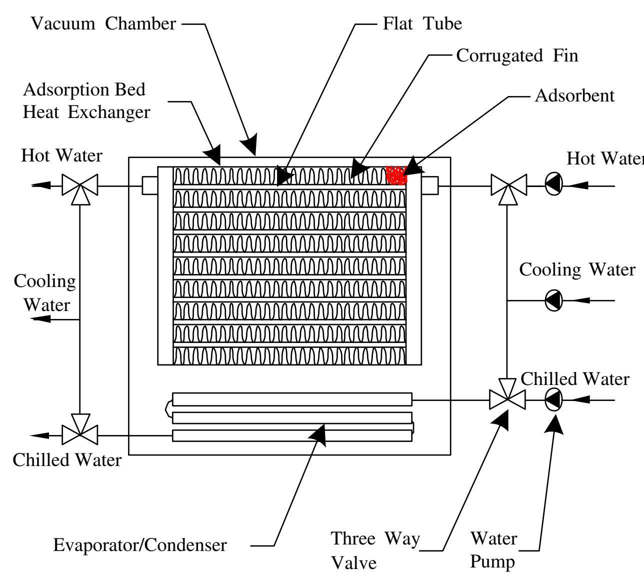

Fig. 1. Schematic of the adsorption chiller.

The upper one, i.e., the adsorber is composed of flat-tube heat exchangers with corrugated fins, between which silica gel (∅0.5–1.5 mm) is packed together with a binder (PVAc). There are 15 vertically arranged adsorption heat exchangers in all, and 24.5 kg silica gel is employed.

The under one, i.e., the evaporator/condenser is also composed of flat-tube heat exchangers, which are horizontally placed on the bottom of the vacuum chamber. Four pieces are used, which are similar to those used for the adsorber. This heat exchanger is used as an evaporator during adsorption cycles and as a condenser during desorption cycles. The characteristics of the flat-tube heat exchangers used are listed in Table 1.

Instead of a two-bed system as usually installed in the commercially available adsorption chiller, only one adsorption bed is used for this chiller here. The two heat exchangers mentioned above are integrated in the same chamber, hence there is no valve used. By means of this kind of design, the construction of the adsorption chiller

Table 1
Characteristics of flat-tube heat exchangers

	Adsorber	Evaporator/condenser
Dimension (mm)	668 × 330.2 × 19	711.6 × 396 × 19
Number of flat tubes	27	33
Row number of fins	28	34
Tube width (mm)	17	17
Fin width (mm)	19	19
Fin pitch (mm)	1.8	1.8
Heat transfer area on fin side (m^2)	4.175	5.414

can be simplified and the production cost of the chiller is diminished.

3. Operation of adsorption chiller

There are two operating modes of the adsorption chiller described here, i.e., desorption/condensation modes and adsorption/evaporation modes. During desorption (regeneration) cycles, the adsorbent (silica gel) in the adsorber is heated by hot water flowing through the heat transfer tubes. Then the water vapor desorbed from the adsorbent condenses on the surface of the condenser, through which cooling water circulates. After completing the desorption process, the operating mode is switched into the adsorption process. Then the adsorber is cooled by cooling water, and the evaporator is supplied with the water to be chilled. Water evaporates from the evaporator and is adsorbed by the adsorbent, so that evaporation proceeds continuously and the chilled water outlet temperature drops. As time goes by, more and more water vapor is adsorbed by the adsorbent, whose adsorptive capacity gradually declines, therefore, the chilled water outlet temperature gradually increases and the cooling power output decreases.

Figs. 2 and 3 show a schematic diagram and a pictorial view of the experimental facility. The entire test system was so constituted, that a steady supply of heat transfer fluid at constant temperature is possible. As a result, the hot water inlet temperature can work within a fluctuation of ±0.5 °C, the cooling water ±0.3 °C, and the chilled water ±0.2 °C. The inlet and outlet temperatures and the temperature in the chamber were measured with PT-100 RTDs with the accuracy of ±0.08 °C. The mass flow rates were measured with turbine-type flowmeters with the accuracy of ±0.5%. The chamber pressure was monitored by pressure transducers with the accuracy of ±0.08% full scale (0–250 mbar). A data acquisition and control unit (Yokogawa DA100) and a computer were used to collect and process the data.

The standard operating conditions are shown in Table 2. The inlet temperatures of the heat transfer fluid are 80 °C for hot water, 30 °C for cooling water, and

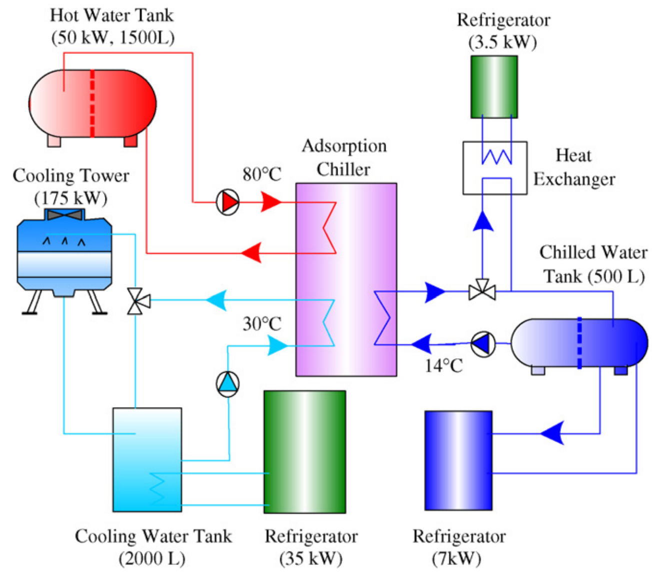

Fig. 2. Schematic of experimental facility.

Fig. 3. Photograph of experimental facility.

Table 2
Opertaing conditions

Standard operating conditions	
Hot water inlet temp.	80 °C
Hot water inlet flow rate	0.48 kg/s
Cooling water inlet temp.	30 °C
Cooling water inlet flow rate	Adsorption 0.6 kg/s Desorption 1.25 kg/s
Chilled water inlet temp.	14 °C
Chilled water inlet flow rate	0.95 kg/s
Cycle time	Adsorption 6 min Desorption 6 min
Varied operating conditions	
Hot water inlet temp.	60–90 °C
Cooling water inlet temp.	20–33 °C
Chilled water inlet temp.	11–20 °C
Cycle time	3–7 min

14 °C for chilled water. The adsorption and desorption cycle times are 6 min. Test runs were conducted for the standard conditions shown in Table 2 and also for various conditions as listed in Table 2.

The coefficient of performance COP is defined as the ratio of the cooling power $\dot{Q}_{\text{ref}}$ and the heating power $\dot{Q}_{\text{H}}$:

$$\text{COP} = \frac{\dot{Q}_{\text{ref}}}{\dot{Q}_{\text{H}}},$$

where $\dot{Q}_{\text{ref}} = \dot{m}_{\text{chill}} c_{p,\text{water}} (T_{\text{chill,in}} - T_{\text{chill,out}})$ and $\dot{Q}_{\text{H}} = \dot{m}_{\text{H}} c_{p,\text{water}} (T_{\text{H,in}} - T_{\text{H,out}})$.

$\dot{Q}_{\text{ref}}$ and $\dot{Q}_{\text{H}}$ are calculated from the measured flow rates, the isobaric specific heat capacities, and inlet and outlet temperatures.

4. Influence of operating conditions on system performance

4.1. Effect of operating temperature

Fig. 4 shows the effects of hot water inlet temperatures on the cooling power and the COP. The cooling power increases as the hot water temperature is increased from 60 to 90 °C, because the amount of water desorbed increases with higher driving hot water temperatures. However, the COP does not increase monotonically with hot water temperatures. There exists a maximum at about $T_{\text{H,in}} = 80$ °C. This can be attributed to an increase of heat losses due to higher driving temperatures.

Fig. 5 shows the effects of cooling water inlet temperatures on the cooling power and the COP. The tendency is similar to that of the influence of hot water temperatures (Fig. 4). The cooling power increases with decreasing cooling water temperatures, which mean a larger amount of water can be adsorbed during the adsorption cycle. The COP also increases with decreasing cooling water temperatures, and it achieves a maximal value at about $T_{\text{chill,in}} = 22$ °C. The reason is that the larger regeneration temperature lift leads to larger heat losses.

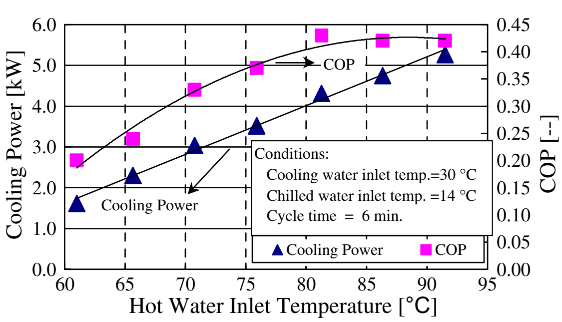

Fig. 4. Effect of hot water inlet temperatures.

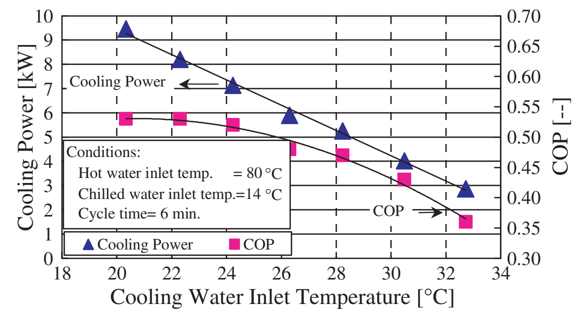

Fig. 5. Effects of cooling water inlet temperature.

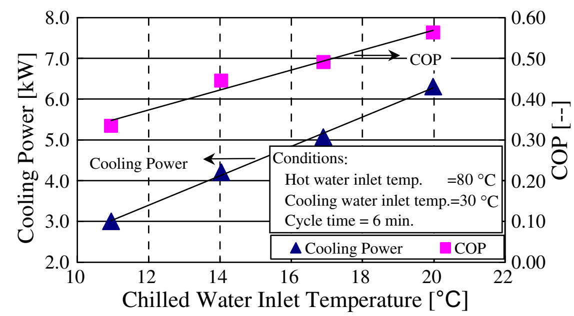

Fig. 6. Effects of chilled water inlet temperatures.

Fig. 6 shows the effects of chilled water inlet temperatures. Both of the cooling power and the COP significantly rise with increasing chilled water temperatures, because the higher chilled water temperature results in higher system pressure and therefore, larger amount of water adsorbed.

4.2. Effect of flow rate

Fig. 7 shows the influence of hot water flow rates. The cooling power increases with increasing flow rates at the lower range of flow rates, whereas the COP decreases. At higher range of flow rates, the cooling power as well as the COP varies slightly.

Fig. 8 shows the influence of chilled water flow rates. As the flow rate is increased, both the cooling power and the COP rise due to improved heat transfer rates.

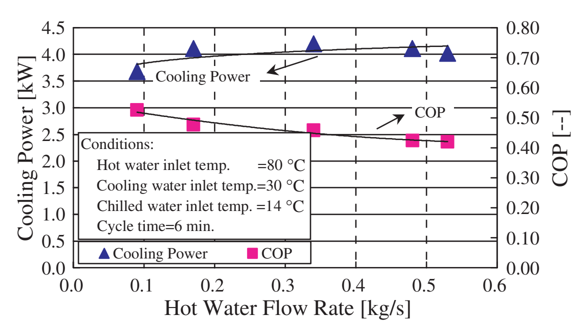

Fig. 7. Effects of hot water flow rates.

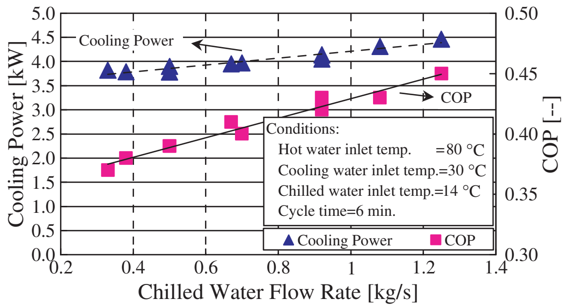

Fig. 8. Effects of chilled water flow rates.

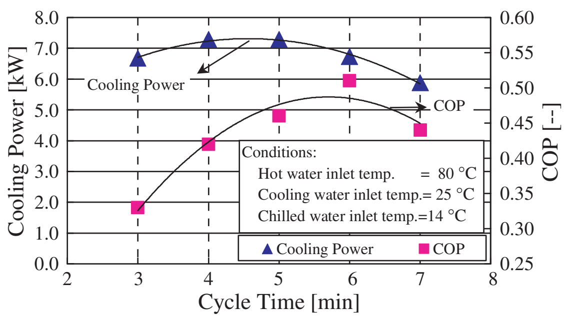

Fig. 9. Effects of cycle times.

4.3. Effect of cycle time

Fig. 9 shows the effects of adsorption/desorption cycle times (condition: 80/25/14 °C). There exists a peak between 4 and 5 min for the cooling power and at about 6 min for the COP. For shorter cycle times, the desorption process could be incomplete, and that results in diminished adsorptive capacity of silica gel and therefore, lowered cooling powers and COPs. Furthermore, the sensible heat for cooling of the evaporator after operating mode switch should be also an important reason for the clearly reduced COP. For longer cycle times, the cooling power decreases because of the rapid diminution of adsorptive capacity of silica gel during the last few minutes.

5. Conclusions

The effects of operating conditions on the cooling power and COP of a closed-type adsorption chiller with silica gel as the adsorbent and water as the adsorbate have been experimentally investigated. The following conclusions were drawn from the foregoing discussion:

1. The system performance is strongly dependent on the operating conditions such as the operating temperatures, flow rates, and cycle times.
2. Under the standard test conditions a cooling power of 4.3 kW and a COP of 0.45 can be obtained. The corresponding SCP is about 176 W/(kg silica gel). With lower hot water flow rates, a COP of 0.53 can be achieved.
3. To obtain an optimal performance, an appropriate cycle time should be selected. There exists a maxima COP value with the cycle time about 6 min under the operating conditions in this study.

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