



Study of a novel silica gel–water adsorption chiller. Part II. Experimental study

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Abstract

The prototype of a novel silica gel–water adsorption chiller is built and its performance is tested in detail. The experimental results show that the refrigerating capacity (RC) and COP of the chiller are 7.15 and 0.38 kW, respectively, when the hot water temperature is 84.8 °C, the cooling water temperature is 30.6 °C, and the chilled water outlet temperature is 11.7 °C. The RC will reach 6 kW under the condition of 65 °C hot water temperature, 30.5 °C cooling water temperature and 17.6 °C chilled water temperature. The results confirm that this kind of adsorption chiller is an effective refrigerating machine though its performance is not as fine as the prediction results. Also it is well effectively driven by a low-grade heat source. Therefore, its applications to the low-grade heat source are much attractive.

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Keywords: Experiment; Adsorption system; Water; Silica gel; Performance; COP

Etude sur un nouveau refroidisseur à adsorption à gel de silice/eau. Partie II. Etude expérimentale

Mots clés: Expérimentation; Système à adsorption; Eau; Gel de silice; Performance; COP

1. Introduction

In the fields of solar energy and low grade waste heat utilization, a silica gel–water adsorption cooling system should be one of favorable choices for it is much suitable for a 65–85 °C heat source.

As the prediction results, a three-bed silica gel–water adsorption chiller can work effectively under the driving source temperature of 60 °C [1], one two-bed system with mass recovery can be successfully powered by a heat source of 50 °C [2,3], and the dual-mode chiller can use waste heat source with a temperature of 40 °C [4]. These researches are very interesting and exciting, though the efficiency of the systems, when driven by so low grade heat source, is too low to put the chiller into practice easily. Generally, a heat source with 70–85 °C is economical for an silica gel–water adsorption chiller and easily obtained from solar energy or waste heat. So most commercialized adsorption chillers aim

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Nomenclature

COP	coefficient of performance
$C_{p,w}$	specific heat of water, $\text{kJ kg}^{-1} \text{ } ^\circ\text{C}^{-1}$
\dot{G}	mass flow rate, kg s^{-1}
m	adsorbent mass in the chiller, kg
n	times of data acquisition in one cycle
Q	heat power, kW
SCP	specific cooling power, kw kg^{-1}
T	temperature, $^\circ\text{C}$
$\Delta\bar{T}$	time-averaged temperature difference between two temperature sensor, $^\circ\text{C}$

Subscript

h	heat, hot
in	inlet
out	outlet
ref	refrigeration
s	silica gel

Superscript

i	the order number of the data acquisition interval
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at the applications for the heat source with 70–85 $^\circ\text{C}$. In the respect of the experimental study, the highest experimental values of COP for a two-bed conventional silica gel–water adsorption chiller is more than 0.4 with the hot water of 50 $^\circ\text{C}$ and the cooling water of 20 $^\circ\text{C}$ [5]. And Restuccia et al. [6] built a lab-scale adsorption chiller using the macroporous silica gel impregnated with CaCl_2 as the adsorbent. The experimental results showed that the COP was up to 0.6 at the 90–95 $^\circ\text{C}$ heat source. Liu et al. [7] improved such system and designed a silica gel–water adsorption chiller with two adsorption/desorption chambers. The experimental results proved that this chiller was able to produce a cooling power of 6.3 and 9 kW with a COP of about 0.4 and 0.5 depending on different working conditions.

In this work, based on the former work [8], a new type silica gel–water adsorption chiller is built in order to use 65–85 $^\circ\text{C}$ heat source effectively. Its application value is experimentally testified. The experimental results indicate that it is an adsorption chiller with a bright commercialization prospect. Up to now, there have been four chillers to be applied in the real systems such as the solar energy air-conditioning system and building cooling, heating and power (BCHP) system. This study about silica gel–water adsorption cooling technology will benefit the further application and development of the silica gel–water adsorption chiller.

2. Experiments

2.1. Prototype

Fig. 1 is the photograph of the silica gel–water adsorption chiller. The adsorbers, the condensers and the evaporator are housed by a steel shell and divided into three chambers by clapboards. The evaporator lies in the bottom of the chiller and the condensers locates in the upper. The heating/cooling switch system and the vacuum valve are on the top. All the valves are controlled by a PLC in the control

box. The total dimension is 1200 mm (length) \times 1000 mm (width) \times 1550 mm (height).

2.2. Test system

Fig. 2 shows the test system of the silica gel–water adsorption chiller. This system is composed of hot water system, cooling water system and chilled water system. In the hot water system, a 0.7T hot water tank acts as a role of heat accumulator to keep the hot water inlet temperature stable. The flow rate of the hot water is controlled by valve V2 and the temperature of the hot water is controlled by valve V7. The steam–water heat exchanger is counterflow tube-shell type. The cooling water tank in the cooling water system has a volume of 1 m^3 . The cooling water from the adsorber flows into the cooling water tank and is mixed with the chilled water from the chilled water tank and the tap water. The redundant water is drained out of the cooling water tank to spray drain through an overflow pipe. The

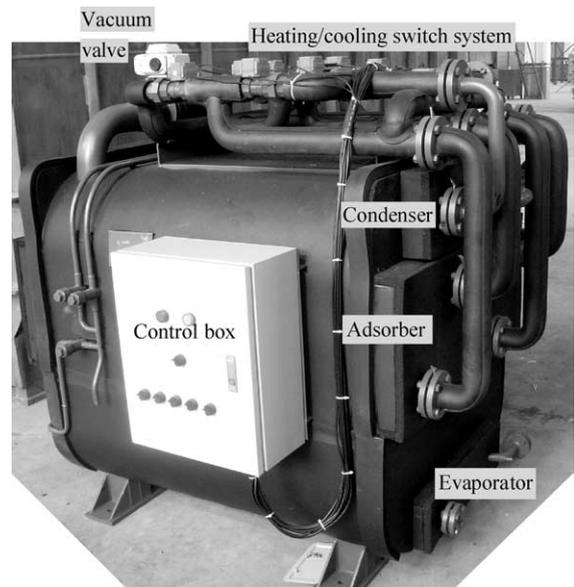


Fig. 1. Photograph of the silica gel–water adsorption chiller.

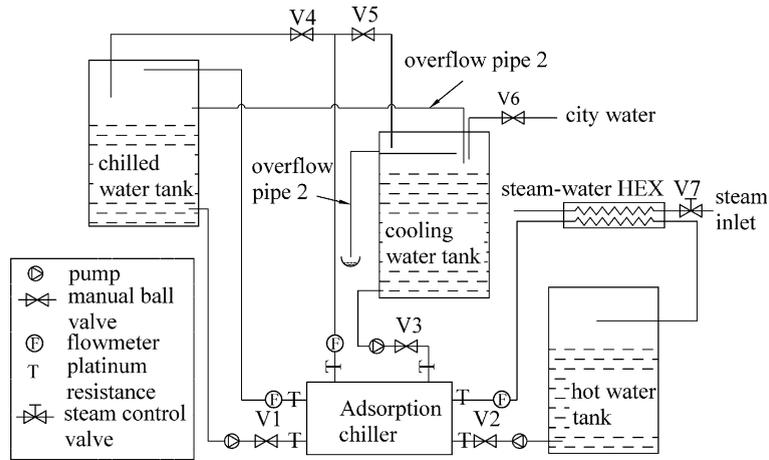


Fig. 2. Test system of the silica gel–water adsorption chiller.

overflow port is very near to the outlet of cooling water in order to drain out more ‘warm’ cooling water but not tap water. The temperature and the flow rate of the cooling water are controlled by valves V6 and V3, respectively. A little of cooling water from the adsorber flows into the chilled water tank through valve V4 to balance the refrigeration output yielded by the chiller. Valves V4 and V1 control the temperature and the flow rate of the chilled water. The redundant water in the chilled water tank is drained into the cooling water tank. The water levels in chilled water tank and cooling water tank are kept by the overflow pipes.

Three flow meters are used to measure the flow rates of hot water, cooling water and chilled water. The precision of the flow meters is 1%FS (full scale). The water inlet and outlet temperatures are measured by 6 Platinum resistances, which are Pt100, grade A. The data are acquired by a Keithly 2700 Multimeter/Data Acquisition System interfaced by a PC. A data acquisition programme reads and stores the data about every 9 s.

2.3. Experimental methods and performance coefficients calculations

The main aim of this work is to test the performance of the novel chiller under different working conditions. So the flow rate of hot water, cooling water and chilled water are given, as 1, 1.53 and 0.5 kg/s and their temperatures are regulated by valves V2, V3 and V1. The experimental work continues under different working conditions. So no less than four cycles is repeated for every working condition in order to slake the influence of the former working condition on the present. All the data used in the calculations of the performance coefficients is obtained from the fourth cycle.

The SCP and the COP are calculated as follows:

$$\text{COP} = \frac{Q_{\text{ref}}}{Q_{\text{h}}} \quad (1)$$

$$\text{SCP} = \frac{Q_{\text{ref}}}{m_{\text{s}}} \quad (2)$$

Refrigerating capacity:

$$Q_{\text{ref}} = \frac{\sum_{i=1}^n [\dot{G}_{\text{chilled}} C_{p,w} (T_{\text{chilled,in}}^i - T_{\text{chilled,out}}^i) - \dot{G}_{\text{chilled}} C_{p,w} \Delta \bar{T}_{\text{chilled,in,out}}]}{n} \quad (3)$$

and heating power:

$$Q_{\text{h}} = \frac{\sum_{i=1}^n [\dot{G}_{\text{h}} C_{p,w} (T_{\text{h,in}}^i - T_{\text{h,out}}^i) - \dot{G}_{\text{h}} C_{p,w} \Delta \bar{T}_{\text{h,in,out}}]}{n} \quad (4)$$

where $\Delta \bar{T}_{\text{chilled,in,out}}$ and $\Delta \bar{T}_{\text{h,in,out}}$ are the time-averaged temperature differences between the inlet and outlet temperature sensors for the chilled water and hot water, respectively. These differences are measured as: the inlet and outlet temperature sensors are bound together and put into the chilled water tank or the hot water tank for a period of time. The temperatures of the inlet and outlet sensors are collected by a PC. The temperature difference between two temperature sensors for chilled water or hot water is averaged in the testing time. Therefore, the Eqs. (3) and (4) are modified formulas.

3. Error analysis

The temperature differences between the inlet and outlet temperature sensors in chilled water and hot water are shown in Figs. 3 and 4. The time-averaged difference is 0.054 °C for chilled water temperature sensors and 0.13 °C for hot water temperature sensors. The influence of these differences on the heating power or refrigerating capacity is about 2%.

The error of temperature sensors is detected and the Eqs. (3) and (4) have been modified. Further, the Keithly

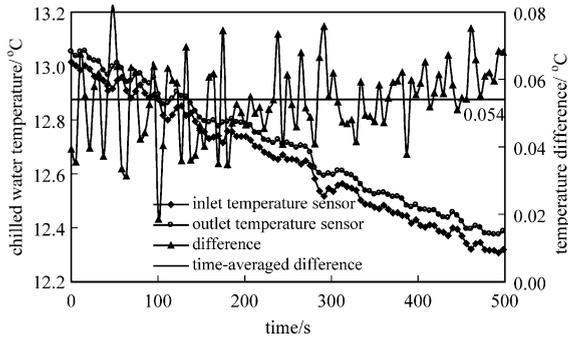


Fig. 3. Measured values of the inlet and outlet temperature sensors in chilled water and their difference.

2700 Multimeter/Data Acquisition System with 6(1/2) bits can be ignored. Therefore, the influence of the precision of temperature sensors on the calculated value of the COP or SCP is slaked. The precision of flow meters has become the main source of the error for the calculated performance coefficients. After calibration using a weighing method, the precision of flow meter is 1% for the chilled water and 7% for hot water. So the relative error is 8% for COP and 1% for SCP.

4. Results and discussions

4.1. Water inlet and outlet temperature variations

Fig. 5 shows the variations of chilled water and refrigerating capacity in three cycles. It indicates that this system has fine operating repeatability and stability. In one whole cycle, the chilled water inlet temperature is controlled to change a little, only 0.8 °C, but the chilled water outlet temperature will changes largely, about 6.3 °C from 14.2 to 20.5 °C. In adsorption process, chilled water outlet temperature quickly declines to the lowest temperature of about 14 °C and slowly rises to about 16 °C due to the weakening adsorption capability of the adsorbent. The corresponding refrigerating capacity is from 14.2 to 8.7 kW. This

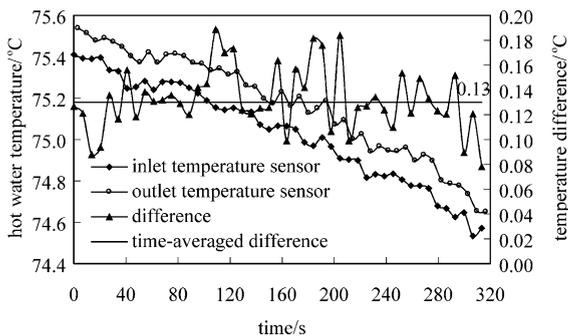


Fig. 4. Measured values of the inlet and outlet temperature sensors in the hot water and their difference.

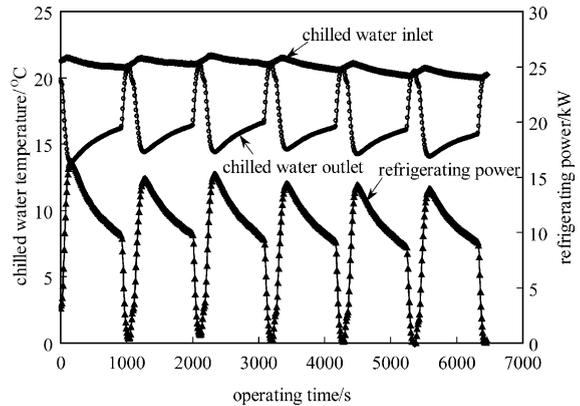


Fig. 5. Chilled water temperature and refrigerating capacity variations with operating time (heating time: 900 s; mass recovery time: 180 s).

temperature will sharply increase near to the inlet temperature during the mass recovery process because the evaporators have no refrigeration output in this phase. After the adsorption capacity of the desorbed adsorbent is restored, the heat recovery process starts and the desorbed adsorbent is cooled to adsorb. So the adsorbent begins to generate refrigeration output. The chilled water outlet temperature falls down. The decrease is not large in the limit of the cooling capacity in the heat recovery process.

Fig. 6 shows the hot water and cooling water temperature variations with the operating time. Inlet temperatures of the cooling water and hot water are both stable, which are kept close to 80 and 31 °C, respectively. The outlet temperatures will largely vary with the operating time in one cycle. As like as the other adsorption cooling system, the heating power and cooling power of the adsorbent are both decreasing with the operating time in one half cycle. The time-averaged differences between inlet temperature and outlet temperature for hot water and cooling water are 7 and 6 °C, respectively. In the mass recovery process, there are interesting changes for the outlet temperatures. The hot

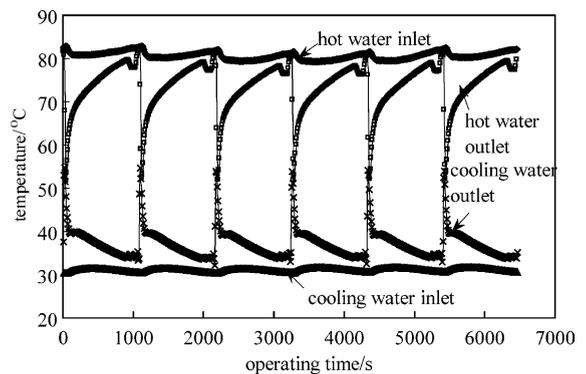


Fig. 6. Hot water and cooling water temperature variations with operating time (heating time: 900 s; mass recovery time: 180 s).

water outlet temperature will decrease first and then increase, but cooling water outlet temperature increases first and decreases last during the mass recovery process, as shown in Fig. 6. The pressure difference between the adsorber and the desorber is much large in the beginning of the mass recovery process when the adsorber continues to be cooled and the desorber heated. So the adsorber has a great adsorption capacity from the desorber at this time. The desorber will absorb much heat to desorb more and the adsorber will release much heat to adsorb more. Then the hot water outlet temperature will sharply decrease and the cooling water outlet temperature promptly increases when the inlet temperatures are constant. The hot water outlet temperature will increase and the cooling water outlet temperature decrease again when the adsorber is near to fullness and the desorber can't desorb more, and when the pressures in the adsorbers are nearly equal to each other. These temperature variations express the thermodynamic characteristics of the adsorber.

4.2. Effects of heating/cooling time

Heating/cooling time is another key parameter to influence the performance of the adsorption chiller. Theoretically, the shorter heating/cooling time is, the higher the refrigerating capacity is and the lower the COP is, even though the hot water and chilled water temperatures are different. But in practice, the adsorption/desorption and condensation/evaporation processes will not completely accomplish and the refrigerating potential of the adsorber cannot exert deeply if the heating/cooling time is too short. So a heating/cooling time between 800 and 900 s is adopted if a highest refrigerating capacity is required for this chiller, as shown in Fig. 7. And a 900 s heating/cooling time will be indispensable if a higher COP is necessary. The RC will drop after the heating/cooling time exceeds 900 s because the adsorbent nearly completely finishes desorption/adsorption process. Also the COP will increase lightly and even drop after 900 s heating/cooling for the RC promptly decreases. Therefore, 900 s heating/cooling time is optimal for this chiller. Fig. 8 shows the COP variations with the heating/cooling time.

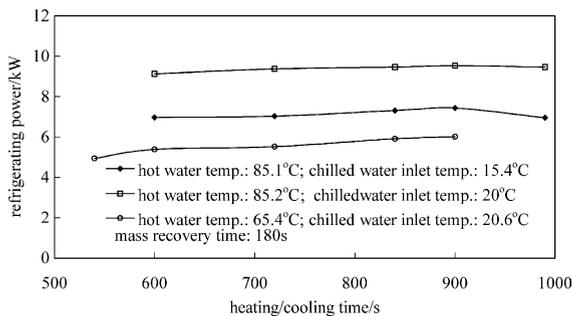


Fig. 7. Refrigerating capacity (RC) variations with the heating/cooling time.

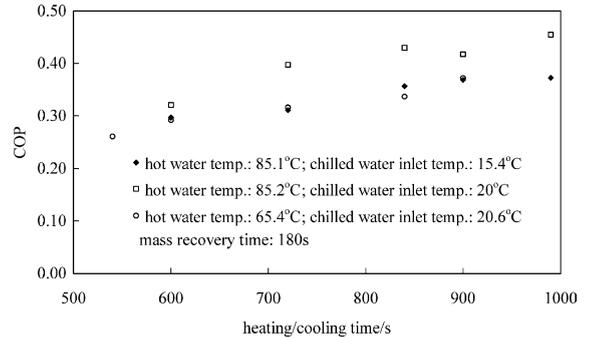


Fig. 8. COP variations with the heating/cooling time.

4.3. Effects of mass recovery time

Mass recovery process can improve the performance of the adsorption chiller driven by a low temperature heat source. Fig. 9 shows the SCP variations with the mass recovery time. For this chiller, mass recovery not only makes the desorber desorb fully but also realize the heat exchanger between two evaporators to save some refrigerating output. This process seems as the second adsorption/desorption process. But it is not always effective to improve COP or RC since the chiller has not refrigeration output during the mass recovery process. Long mass recovery time causes a full desorption process and then its refrigerating capacity increases. Simultaneously, the ratio of non-refrigeration output time for the adsorption chiller in the cycle time is rising. So the mass recovery time has an optimal value. The best mass recovery time for this chiller would be between 120 and 180 s depending the working conditions though the SCP is not very sensitive to the mass recovery time as shown in Fig. 9. The chiller works at a fixed cycle time mode and the mass recovery time and the heating/cooling time must be confirmed before the chiller comes into industrialization. The selection of the proper mass recovery time as well as the proper heating/cooling time is important to the chiller for the energy density of an adsorption chiller is too small to abandon any potential improvement of the performance.

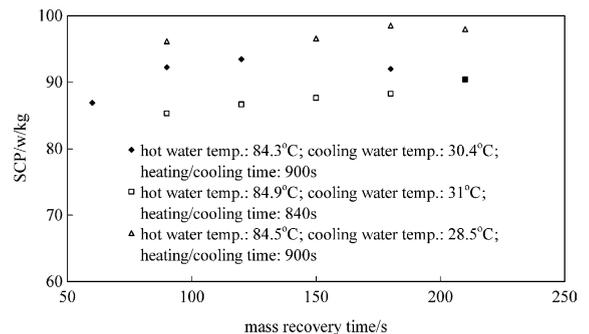


Fig. 9. SCP variations with the mass recovery time.

4.4. Effects of hot water temperature

As the driving force, the hot water temperature is very important to the adsorption refrigerating system. A high hot water temperature causes a large temperature lift for the system at the same cooling water temperature. Then the large temperature lift results in a high refrigerating capacity, as shown in Fig. 10. However, the adsorption chiller will lose its competitive power if the hot water temperature is higher than 90 °C because an absorption chiller has predominated in the market of higher than 90 °C hot water temperature. Furthermore, a higher hot water temperature has caused a higher refrigerating capacity as well as a higher heating power. Heightening the hot water temperature will not markedly increase the system COP after this temperature is higher than 80 °C, as shown in Fig. 11. This result is very close to Saha’s study.

Therefore, a heat source higher than 90 °C is not recommended due to its lower COP and weaker competitive power than an absorption chiller, though the chiller can well work at this temperature.

4.5. Effects of chilled water temperature

Figs. 12 and 13, respectively show the refrigerating capacity and COP variations with the different chilled water inlet temperature. Inlet temperature increase of 1 °C for chilled water can cause about 5% improvement of refrigerating capacity and more than 4% that of COP at the same hot water temperature, the same cooling water temperature, the same heat recovery time and the same heating/cooling time. So increasing the chilled water inlet temperature is advantageous to improving the performance of the system. In the experiment, the highest inlet temperature of the chilled water is about 20 °C and the system can supply lower than 16 °C chilled water when the hot water temperature is about 85 °C. So it well meets the requirement of fan-coil units with a large circulating flow rate of the chilled water.

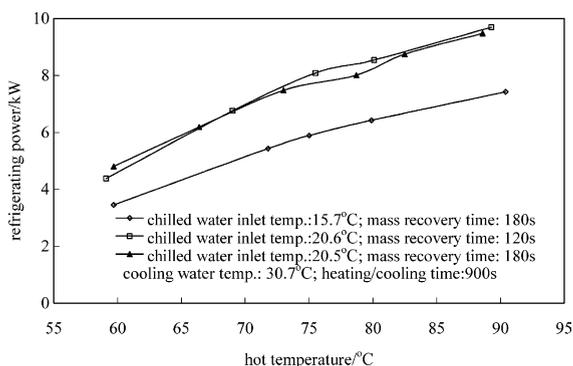


Fig. 10. Refrigerating capacity (RC) variations with the hot water temperature.

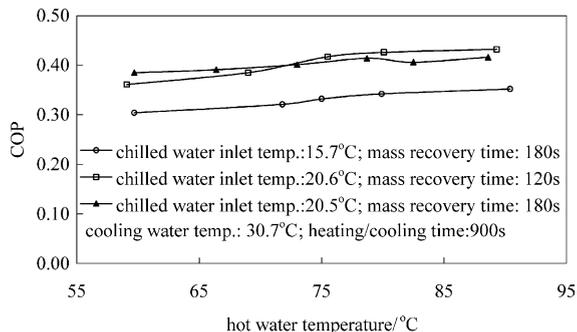


Fig. 11. COP variations with the hot water temperature.

4.6. Effects of cooling water temperature

The cooling water temperature is one of decisive factors on the performance of the adsorption chiller because it influences not only on the condensation process but also on the adsorption process. A high cooling water temperature increases the condensation temperature and adsorption temperature. As a result, both the desorption and adsorption processes are worsened. Then more refrigerating potential is lost. There is no crystallization trouble for an adsorption chiller. So the lower cooling temperature is, the higher the performance of the chiller is. Decreasing cooling water temperature has a similar effect on the system as well as increasing chilled water temperature. When the cooling water temperature drops 1 °C, SCP can increase more than 4%. Fig. 14 shows the SCP and COP variations with the cooling water temperature from 28.4 to 36.4 °C. The COP variation is not very regular for the COP is determined by RC and heating power.

4.7. Comparisons with the former work

The trend of the COP and SCP for this chiller is quite similar to that for the former conventional chiller, but the values of them are quite different. Table 1 shows the comparisons of the experimental results between this work

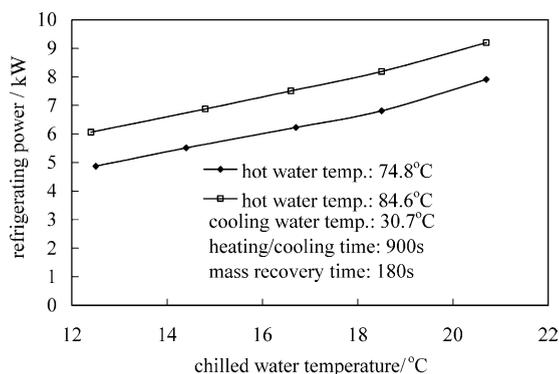


Fig. 12. Refrigerating capacity (RC) variations with the chilled water inlet temperature.

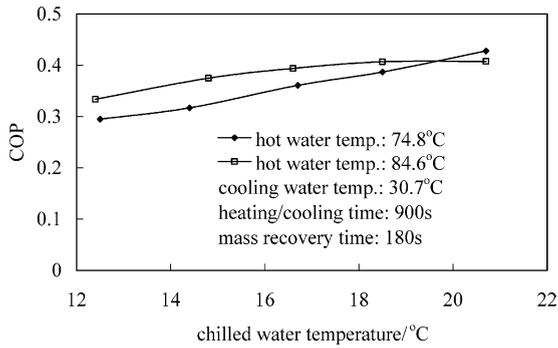


Fig. 13. COP variations with the chilled water inlet temperature.

and the former work at a typical working condition with the hot water temperature at 85 °C and the chilled water inlet temperature at 14 °C. As shown in Table 1, the novel chiller is quite better than the conventional chiller [5] for COP and even better than an improved (in structure) chiller [7] for the COP and SCP. Therefore, this novel chiller is another effective cooling machine compared with the conventional chiller and the improved chiller.

4.8. Comparisons with the simulated results

The trend of the COP and SCP shown by the simulated results in Ref. [8] is well consistent with experimental results as described above. However, the effects of the heating/cooling time and the mass recovery time on the COP is higher a bit in the simulated results than in the experimental results. And the best mass recovery time and the best heating/cooling time for the COP or SCP shown in the experimental results are quite different from those shown in the simulated results. They are both longer in the experiments than in the simulations, which indicates that the heat and mass transfer performance of the adsorbers is not good enough.

The experimental results shown above are quite different from the simulated results in Ref. [8] on the values. The values of the tested performance coefficients are much lower than the simulated ones at the same working condition. There are some reasons to cause these large differences:

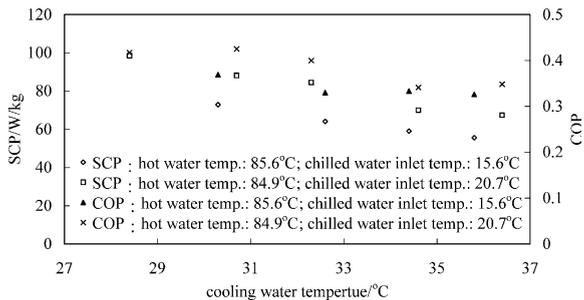


Fig. 14. SCP and COP variations with the cooling water temperature.

firstly, the assumptions taken in the simulations do not accord with the practical. For example, the heat transfer among the methanol evaporator (ME), the working water evaporator (WE) and the idle water evaporators are ignored in the simulation. Secondly, some liquid methanol is rudimental in the pipes of the working WE, such as WE1, because the pipes are arrayed horizontally and the liquid methanol condensed in the pipes of WE1 can't be discharged completely. So the rudimental methanol in WE1 will evaporate after switch and then condensate in the WE2 or the ME, depending on of which the temperature is lower. Then the heat transfer occurs between the two WEs or between the WE1 and the ME. Moreover, a too short space between the WEs and the ME also causes unavoidable heat transfer between the ME and idle WE for the boiling liquid methanol in the ME is easy to rush out the surface of the idle WE during the desorption/adsorption process. Thus, some of refrigerating output is lost. These two losses of refrigeration output come from the evaporator. The negative temperature difference for the chilled water, as shown in Fig. 15, proves that the loss of refrigeration output in the evaporator is large but is ignored in the simulation.

Table 2 lists their comparisons for two typical working conditions. At the same temperatures of the hot water and the cooling water, a higher inlet temperature of the chilled water causes a lower difference percentage. This is because the temperature difference between the idle WE and ME or between the working WE and the idle WE is smaller and the heat transfer between them decreases when the inlet temperature of the chilled water is high. Then the loss of refrigeration output in the evaporator decreases.

The large difference between the experimental results and the simulated results also shows a potential improvement for the prototype. The system performance must be improved if the effective measures are taken to avoid the loss of the refrigeration output in the evaporator.

5. Conclusions

In this chiller, only one vacuum valve is installed between two adsorption/desorption vacuum chambers to

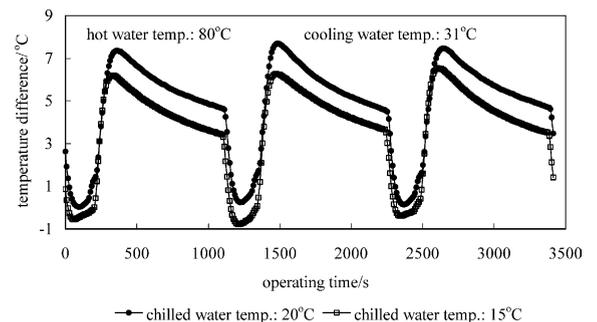


Fig. 15. Temperature difference for chilled water.

Table 1
Comparisons of the experimental results between this work and the former work at a typical working condition

Reference	Hot water temperature (°C)	Cooling water temperature (°C)	Chilled water temperature (°C)		Refrigerating power (kW)	COP	SCP (W/kg)
			Inlet	Outlet			
This work	85	30.8	14	10.7	6.6	0.371	63.4
[5]	85	32	14	11.5	7.36	0.28	–
	85	30	14	10.5	10.3	0.34	–
[7]	85	30	14	12.8	2.79	0.212	52.8

increase its performance when it is driven by a low temperature heat source. The operating reliability of the chiller rises highly because of fewer moving parts. The experimental results have testified that the chiller designed in Ref. [8] is an available and valuable refrigerating device, the experimental results is not as good as the simulated results.

The COP and RC are very close to Akahira's computational results [2], as shown in the experimental results. The experimental results also show that the refrigerating capacity (RC) and COP of the chiller are 7.15 and 0.38 kW, respectively, when the hot water temperature is 84.8 °C, the cooling water temperature 30.6 °C, chilled water outlet temperature 11.7 °C. And the RC will reach 6 kW under the condition of 65 °C hot water temperature, 30.5 °C cooling water temperature and 17.6 °C chilled water outlet temperature. Therefore, this kind of adsorption chiller is a high-effective refrigerating machine. It is well effectively driven by a low grade heat source such as 65–85 °C.

For this chiller, a 900 s heating/cooling time will be a better choice for RC and COP when the chiller works at a fixed heating/cooling time mode as shown in the experimental results. And the suitable mass recovery time is between 120 and 180 s. However, a 180 s mass recovery time is needed for a 65 °C heat source and a 120 s mass recovery time is necessary for a 85 °C heat source. The temperature range of the heat source is well suitable for solar energy. So the chiller can effectively utilize solar energy.

The comparison between the simulated results and the

Table 2
Comparisons between the simulated results and the experimental results

$T_h=85\text{ }^\circ\text{C}$, $T_{\text{cooling}}=31\text{ }^\circ\text{C}$		Simulated	Experimental	Difference (%)
$T_{\text{chilled,inlet}}=20\text{ }^\circ\text{C}$	COP	0.651	0.451	30.7
	RC/kW	12.6	10.16	19.4
$T_{\text{chilled,inlet}}=15\text{ }^\circ\text{C}$	COP	0.588	0.369	37.2
	RC/kW	10.4	7.43	28.6

experimental results indicates that this chiller has a potential of improvement if the refrigeration output loss of the evaporator is avoided in the structure design.

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References

- [1] B.B. Saha, S. Koyama, J.B. Lee, K. Kumahara, K.C.A. Alam, et al., Performance evaluation of a low-temperature waste heat driven multi-bed adsorption chiller, *Int J Multiphase Flow* 29 (2003) 1249–1263.
- [2] A. Akahira, K.C.A. Alam, Y. Hamamoto, A. Akisawa, T. Kashiwagi, Mass recovery adsorption refrigeration cycle—improving cooling capacity, *Int J Refrigeration* 27 (2004) 225–234.
- [3] E.C. Boelman, B.B. Saha, T. Kashiwagi, Experimental investigation of a silica gel–water adsorption refrigeration cycle—the influence of operating conditions on cooling output and COP, *ASHRAE Trans* 101 (2) (1995) 358–366.
- [4] B.B. Saha, S. Koyama, T. Kashiwagi, A. Akisawa, K.C. Ng, H.T. Chua, Waste heat driven dual-mode, multi-stage, multi-bed regenerative adsorption system, *Int J Refrigeration* 26 (2003) 749–757.
- [5] E.C. Boelman, B.B. Saha, T. Kashiwagi, Experimental investigation of a silica gel–water adsorption refrigeration cycle—the influence of operating conditions on cooling output and COP, *ASHRAE Trans* 101 (Part 2) (1995) 358–366.
- [6] G. Restuccia, A. Freni, S. Vasta, Yu. Aristov, Selective water sorbent for solid sorption chiller: experimental results and modeling, *Int J Refrigeration* 2004; 27284–27293.
- [7] Y.L. Liu, R.Z. Wang, Z.Z. Xia, Experimental performance of a silica gel–water adsorption chiller, *Appl Therm Eng* 2 (2005) 3359–3375.
- [8] D.C. Wang, J.Y. Wu, Z.Z. Xia, R.Z. Wang, H. Zhai, W.D. Dou, Study of a novel silica gel–water adsorption chiller. Part I. Design and performance prediction. *Int J Refrigeration*, in press, doi:10.1016/j.ijrefrig.2005.03.001.