Study on a dual-mode, multi-stage, multi-bed regenerative adsorption chiller

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Abstract

In this paper, a detailed parametric study on a dual-mode silica gel–water adsorption chiller is performed. This advanced adsorption chiller utilizes effectively low-temperature solar or waste heat sources of temperature between 40 and 95 °C. Two operation modes are possible for the advanced chiller. The first operation mode will be to work as a highly efficient conventional chiller where the driving source temperature is between 60 and 95 °C. The second operation mode will be to work as an advanced three-stage adsorption chiller where the available driving source temperature is very low (between 40 and 60 °C). With this very low driving source temperature in combination with a coolant at 30 °C, no other cycle except an advanced adsorption cycle with staged regeneration will be operational. In this paper, the effect of chilled-water inlet temperature, heat transfer fluid flow rates and adsorption–desorption cycle time effect on cooling capacity and COP of the dual-mode chiller is performed. Simulation results show that both cooling capacity and COP values increase with the increase of chilled water inlet temperature with driving source temperature at 50 and 80 °C in three-stage mode, and single-stage multi-bed mode, respectively. However, the delivered chilled-water temperature increases with chilled-water inlet temperature in both modes.
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Nomenclature

Symbols

C_p	isobaric heat of adsorption (J/kg K)
D_s	surface diffusivity (m²/s)
D_{so}	pre-exponential constant in Eq. (4) (m²/s)
E_a	activation energy (J/kg of mole)
$k_s a_p$	overall mass transfer coefficient (1/s)
m	mass flow rate (kg/s)
$P_s(T)$	saturated vapor pressure (kPa)
$q*$	equilibrium amount adsorbed (kg/kg)
Q_{chill}	cooling capacity (kW)
Q_{hot}	driving heat (kW)
R	real gas constant (J/kg of mole K)
T	temperature (°C)
t	time (s)
W	mass (kg)
Δ	difference

Subscripts

ads	adsorption
chill	chilled water
cond	condenser
cool	cooling water
des	desorption
eva	evaporator
Hex	heat exchanger
hot	hot water
in	inlet
out	outlet
regen	regeneration
s	adsorbent (silica gel)
w	water
water	heat transfer fluid

1. Introduction

An advanced, dual-mode, silica gel–water chiller utilizing low-grade waste heat or renewable energy source between 40 and 95 °C as the driving heat source with a cooling source at 30 °C was introduced by the authors [1]. Two operation modes are possible for the advanced chiller. The first operation mode will be to work as a high efficient conventional chiller where the driving source temperature is between 60 and 95 °C. The

second operation mode will be to work as an advanced, three-stage adsorption chiller where the available driving source temperature will be lower than 60 °C (i.e., between 40 and 60 °C). The main innovative feature of the dual-mode chiller is the ability to operate with not only at small regenerating temperature lift (10 K) but also at mid regenerating temperature lift (50 K) and it is therefore attractive as an energy saver. On the other hand, the use of fixed beds requires the adsorbent to be repeatedly switched between adsorption and desorption modes, which causes severe irreversible heat losses as the result of the batch-cycle operation in three-stage mode. In order to improve the cycle efficiency by reducing irreversibilities in batch-cycle operation, several advanced cycles have been proposed by various authors. Shelton et al. [2] used a thermal wave cycle with adsorber/desorber heat recovery by which heat losses are reduced in fixed-bed cycles. Härkönen and Aittomake [3] describe a ramp-wave cycle with parallel temperature ramps in the fluid and adsorbent bed. Critoph [4] introduced a forced convective thermal wave adsorption cycle. The advantages of the forced convective concept are: (1) the cycle is highly regenerative and hence of high efficiency; (2) there is no complex and expensive heat exchanger within the bed; (3) there is no added thermal mass due to the use of heat exchangers; and (4) the high heat transfer rate will give rapid cycle times. Main disadvantage of the forced convective cycle is the need for the large and expensive gas-to-gas heat exchanger required to transfer recovered energy between the fluid loops.

Meunier [5] and Douss and Meunier [6] analyzed the heat integration problem by introducing cascaded cycles from the same perspective. In this paper, a detailed parametric study on an advanced dual-mode adsorption chiller is performed. The cycles under study operate with $T_{\text{hot in}} = 50$ °C and $T_{\text{hot in}} = 80$ °C, respectively, for three-stage mode and single-stage, multi-bed mode in combination with a coolant inlet at 30 °C. A parametric study is performed to determine the effect of heat transfer fluid (hot, cooling and chilled water) flow rates and adsorption/desorption cycle time on cooling capacity and COP of the dual mode adsorption chiller. The effect of regeneration temperature lifts on irreversible heat losses is also analyzed.

2. Dual-mode adsorption cycle

Two operation modes are possible for the dual-mode adsorption chiller. The first operation mode (six-bed regenerative strategy) will be to work as a highly efficient conventional chiller where the driving source temperature is between 60 and 95 °C. The second operation mode (staged-regeneration strategy) will be to work as an advanced three-stage adsorption chiller where the available driving source temperature is very low (between 40 and 60 °C).

2.1. Six-bed regenerative strategy

The proposed six-bed regenerative strategy is an extension from the conventional, two-bed operation. In a conventional adsorption chiller, the inherent restriction in the number of beds resulted in significant temperature fluctuation in all the components. The peak temperature of the condenser outlet that follows shortly after the bed switching adds on to the instantaneous load of the cooling tower. The multi-bed scheme will serve to significantly reduce the peak temperatures in both the evaporator and condenser [6]. The second objective is to improve the recovery efficiency of waste heat or renewable energy

source that can be obtained by using flat plate solar collectors to useful cooling. Such low temperature, industrial waste heat (typically below 100 °C) is commonly purged into the environment. The enthalpy of such waste heat relative to that of the ambient represents a fixed energy expenditure waiting to be tapped for productive uses. However, such a rigid configuration for the string of adsorption chillers represents an under-utilization of these downstream adsorption chillers. The multi-bed scheme will be shown to eliminate such rigid configurations. Thus the main objective of the multi-bed regenerative scheme is to maximize the cooling output of the adsorption chiller for a given investment into the chiller and adsorbent, instead of the COP which is defined as the ratio between cycle averaged cooling capacity rate and that of thermal energy input. Table 1 and Fig. 1 depict the energy utilization schedule and flow configuration for the chiller in 6-bed operational mode, respectively. Referring to Table 1, at a particular instant, Bed 1 is at pre-cooling mode and receives coolant from Bed 6; Bed 2 is in desorption mode (lead desorber) and receives hot water directly from the external heat source; Bed 3 is in desorption mode (intermediate desorber) and receives hot water from Bed 2; Bed 4 is at pre-heating mode and receives hot water from Bed 3; Bed 5 is in adsorption process (lead adsorber) and receives coolant from condenser; and Bed 6 is also in adsorption process (intermediate adsorber) and receives coolant from Bed 5. In this particular instant, Bed 2 and Bed 3 are connected with the condenser and disconnected from the evaporator, while Bed 5 and Bed 6 are connected with the evaporator and disconnected with the condenser. Bed 1 and Bed 4 are isolated from both evaporator and condenser. By capitalizing the phase difference between the three pairs of sorption elements, the heat source from the hottest desorber can be used to regenerate another relatively low-temperature desorber that will act as the intermediate-temperature desorber. The outlet hot water from the intermediate-temperature desorber will further regenerate the third desorber (low-temperature desorber) before being purged into the ambient. Similarly, the coolant from the coldest adsorber can be used in cascaded manner (cooling water outlet from the lowest-temperature adsorber to

Table 1
Energy utilization schemes for the adsorption cycle in multi-bed mode

Bed 1	Sw	Ads (3)	Ads (2)	Ads (1)	Sw	Des (3)	Des (2)	Des (1)
Bed 2	Des (1)	Sw	Ads (3)	Ads (2)	Ads (1)	Sw	Des (3)	Des (2)
Bed 3	Des (2)	Des (1)	Sw	Ads (3)	Ads (2)	Ads (1)	Sw	Des (3)
Bed 4	Sw	Des (3)	Des (2)	Des (1)	Sw	Ads (3)	Ads (2)	Ads (1)
Bed 5	Ads (1)	Sw	Des (3)	Des (2)	Des (1)	Sw	Ads (3)	Ads (2)
Bed 6	Ads (2)	Ads (1)	Sw	Des (3)	Des (2)	Des (1)	Sw	Ads (3)

Ads: operating in adsorption mode (adsorber)

Des: bed operating in desorption mode (desorber)

Sw: switching from adsorber to desorber, and receiving heating stream from heat source or Des $(n/2 - 1)$. Or switching from desorber to adsorber, and receiving coolant from condenser or Ads $(n/2 - 1)$. Here n denotes the total number of beds. $n = 6$ in the present case.

(1): this refers the situation when the sorption element receives either coolant from the condenser or hot water directly from the heat source.

(2): this refers the situation when the sorption element receives hot water from desorber (1) if it is in desorption operation; or receives coolant from adsorber (1) if it is in adsorption mode.

(3): this refers the situation when the sorption element receives hot water from desorber (2) if it is in desorption operation; or receives coolant from adsorber (2) if it is in adsorption mode.

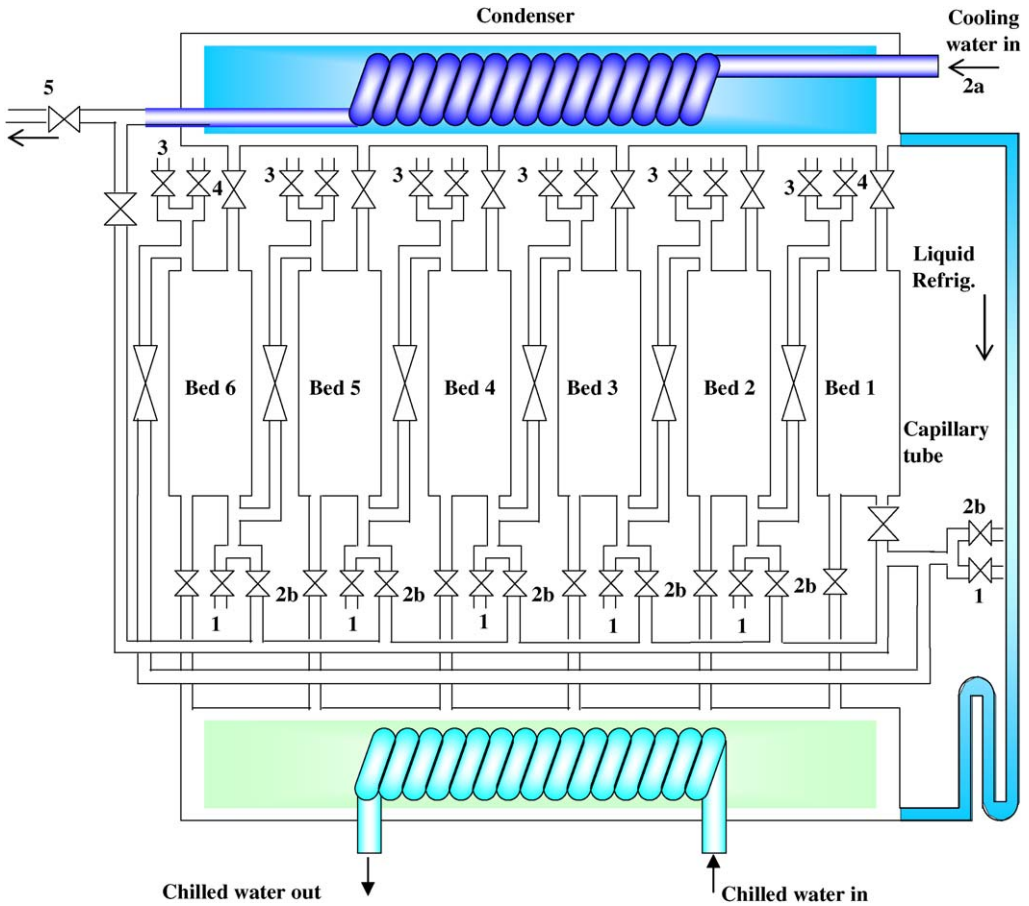*Note*: The width of each box is an indication of the relative time duration over one cycle.

Fig. 1. Schematic diagram of the advanced adsorption chiller in 6-bed regenerative operation mode.

intermediate-temperature adsorber from where the coolant outlet enters the high-temperature adsorber) before being sent back to the cooling tower.

2.2. Staged regeneration strategy (3-stage operation mode)

If the available heat source temperature is below 60 °C, conventional adsorption chiller is not operational with a cooling source of 30 °C or higher. For practical utilization of these temperatures in adsorption chiller operation, staged regeneration is necessary. Figs. 2(a) and (b) illustrate the adsorption cycle in 3stage operation mode in process A and process C, respectively. The advanced chiller comprises with eight heat exchangers, namely, a condenser, an evaporator and three pairs of adsorber/desorber heat exchangers plus eight refrigerant valves. All six adsorber/desorber heat exchangers are identical. The adsorbent is packed in the adsorber/desorber heat exchangers, which undergoes alternate cooling and heating to allow refrigerant adsorption and desorption. Refrigerant (water), evaporates inside the evaporator, picking up evaporation heat from the chilled water, is
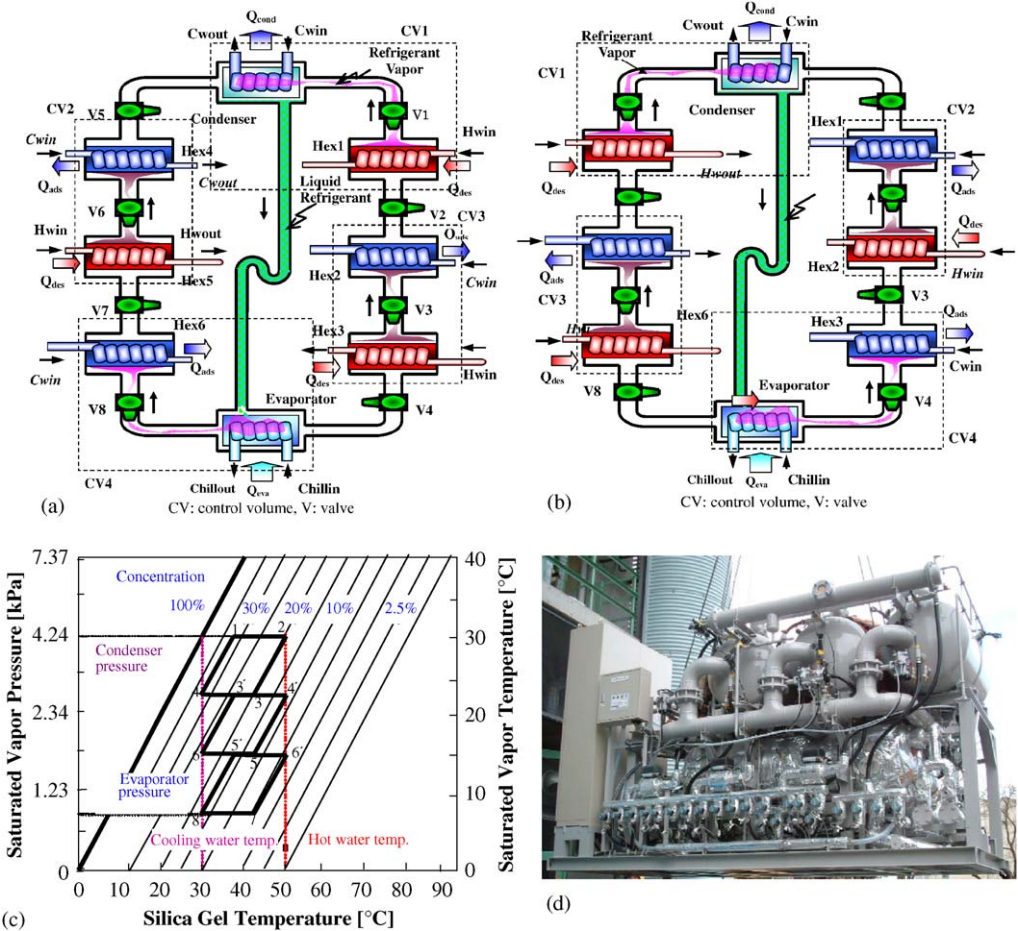
Fig. 2. (a) Schematic of the dual-mode adsorption chiller in 3-stage operation (process A). (b) Schematic of the dual-mode adsorption chiller in 3-stage operation (process C). (c) Dühring diagram of the dual mode adsorption chiller in 3-stage mode. (d) Photograph of the advanced dual-mode adsorption chiller.

adsorbed by adsorber 6 via valve 8. The desorption-condensation process corresponds to path $7 \rightarrow 8$ in Fig. 2(c) and is shown by CV4 in Fig. 2(a). Adsorber 2 adsorbs refrigerant from desorber 3 via valve 3 and is shown by CV3 in Fig. 2(a). The adsorption of adsorber 2 is shown by path $5 \rightarrow 6$ and desorption of desorber 3 is shown by path $5' \rightarrow 6'$ in Fig. 2c. Paths $5 \rightarrow 6$ and $5' \rightarrow 6'$ result in path $6 \rightarrow 6'$ as the desorber 3 and adsorber 2 are at the same pressure level. Heat exchanger 4 adsorb refrigerant from desorber 5 via valve 6 that is shown by CV2 in Fig. 2a. As can be seen from Fig. 2c, the adsorption of adsorber 4 and desorption of desorber 5 are shown by paths $3 \rightarrow 4$ and $3' \rightarrow 4'$, respectively. Paths $3 \rightarrow 4$ and $3' \rightarrow 4'$ result in path $4 \rightarrow 4'$. Desorber 1 is connected to the condenser via valve 1. The desorption–condensation process is shown by CV1 and path $1 \rightarrow 2$, respectively, in Figs. 2(a) and (c). The desorbed refrigerant vapor is condensed in the condenser at temperature T_{cond}; cooling water removes the condensation heat. This condensed refrigerant comes back to the evaporator via the capillary tube connecting the condenser

Table 2
Chiller operation time chart in 3-stage mode

Cycle		Desorption, Mode A	Pre-heating/pre-cooling, Mode B	Adsorption, Mode C	Pre-heating/pre-cooling, Mode D
Time (s)		300	30	300	30
Valve	1,3,6,8	Open	Closed	Closed	Closed
	2,4,5,7	Closed	Closed	Open	Closed
Sorption element	1,3,5	Hot water	Coolant	Coolant	Hot water
	2,4,6	Coolant	Hot water	Hot water	Coolant

and evaporator to complete the cycle. The capillary tube is bent for achieving a pressure drop resulting in the refrigerant being in liquid phase in the evaporator. The use of parallel cooling water circuits for the condenser and adsorbers 2, 4 and 6 results in similar temperature levels at the condenser (T_{cond}) and those adsorbers (T_{ads}).

Upon completion of adsorption/desorption process A (300 s) the flows of hot and cooling water are redirected, so that the desorbers 1,3 and 5 switch into adsorption modes and the adsorbers 2, 4 and 6 change into desorption operations (Fig. 2b). All eight refrigerant valves (V1∼V8) are also switched simultaneously, so that refrigerant from the evaporator can be adsorbed by adsorber 3 (CV4 in Fig. 2b). As shown by CV1 in Fig. 2(b), Desorber 4 is then connected to the condenser. Water vapor flows from desorber 2 to adsorber 1 and desorber 6 to adsorber 5 as shown by CV2 and CV3, respectively in Fig. 2(b).

Preceding the start of a new adsorption/desorption process (process A or process C), a pre-heating or pre-cooling process (process B or process D) takes place for 30 s. During process B, sensible heat removed from heat exchanger 1, 3 and 5 is provided to heat exchangers 2, 4 and 6 with all refrigerant valves closed and no adsorption/desorption occurrence. Process D is the reverse of process B. The time chart of the chiller in 3stage operation is shown in Table 2. The photograph of the dual-mode adsorption chiller prototype is shown in Fig. 2(d).

3. Mathematical modeling

The equation used to describe the silica gel–water properties assume an equilibrium process, without hysteresis, and isobaric adsorption/desorption. The modified Freundlich equation [7,8] below, Eq. (1), was chosen for providing a concise analytical expression of experimental data, in the form:

$$q^* = A(T_S)[P_S(T_W)/P_S(T_S)]^{B(T_S)}, \tag{1}$$

where

$$A(T_S) = A_0 + A_1 T_S + A_2 T_S^2 + A_3 T_S^3,$$

$$B(T_S) = B_0 + B_1 T_S + B_2 T_S^2 + B_3 T_S^3,$$

where, q^* is the amount adsorbed in equilibrium. $P_s(T_w)$ and $P_s(T_s)$ are the saturation vapor pressure at water vapor temperature (T_w) and silica gel temperature (T_s),

respectively. The numerical values of $A_0 \sim A_3$ and $B_0 \sim B_3$ are determined by the least square fit to experimental data, which cab be found elsewhere (Saha et al. [1]).

3.1. Adsorption rate

The adsorption process in an adsorbent bed is considered to be controlled by macroscopic diffusion into the particle bed. The adsorption rate is expressed as [9]

$$\mathrm{d}q/\mathrm{d}t = k_s a_p (q^* - q). \tag{2}$$

The overall mass transfer coefficient $(k_s a_p)$ for adsorption is given by

$$k_s a_p = \frac{15 D_s}{R_p^2}. \tag{3}$$

The surface diffusivity D_s is expressed as

$$D_s = D_{so} \, \exp\left(\frac{E_a}{RT}\right). \tag{4}$$

During operation, the evaporated refrigerant would nearly simultaneously reach all the silica gel in the adsorber. In such a case, the intraparticle resistance approaches to zero and modeling the adsorption/desorption rate for the silica gel–water pair in Eq. (2) is justified. In the present treatment, the effect of the refrigerant mass in the gas phase is ignored.

3.2. Mass balance

The mass balance of refrigerant (water) is written by neglecting the gas phase as:

$$\frac{\mathrm{d}Ww}{\mathrm{d}t} + W_s \left(\frac{\mathrm{d}q_{des}}{\mathrm{d}t} + \frac{\mathrm{d}q_{ads}}{\mathrm{d}t}\right) = 0. \tag{5}$$

3.3. Energy balance equations

Adsorption and desorption energy balances, and condenser and evaporator energy balances are considered as Saha et al. [1].

3.4. COP and cooling capacity (Q_{chill})

The COP and cooling capacity are defined by the following equations:

$$\mathrm{COP} = Q_{chill}/Q_{hot}, \tag{6}$$

$$Q_{chill} = m_{chill} \, Cp_{water} \int_0^{t_{cycle}} (T_{chillI\ in} - T_{chill\ out}) \mathrm{d}t, \tag{7}$$

where

$$Q_{hot} = m_{hot} \, p_{water} \int_0^{t_{cycle}} (T_{hot\ in} - T_{hot\ out}) \, \mathrm{d}t.$$

Here t_{cycle} denotes the total cycle time.

4. Results and discussion

To clarify the practical operational limits and also the performance characteristics, cycle simulation runs were performed for the dual-mode, regenerative multi-bed and non-regenerative multi-stage chiller. Since the main objective is to use low temperature waste heat or renewable energy sources (below 100 °C) as the driving source, the investigation was conducted for hot water temperature between 40 and 95 °C. In the present paper, the effect of chilled-water temperature, heat transfer fluid (hot and cooling water) mass flow rate and adsorption/desorption cycle time on the chiller cooling capacity and COP are described.

4.1. Chilled water temperature

Figs. 3(a) and (b) show the effect of chilled-water inlet temperature on cooling capacity and COP with fixed hot and cooling water inlet temperatures. Hot water inlet temperature is chosen as 80 °C and 50 °C for regenerative multi-bed and non-regenerative multi-stage operation, respectively. Cooling water inlet temperature in both modes is taken as 30 °C. For the multi-bed operation mode, cooling capacity increases from 3.51 to 7.88 kW and COP increases from 0.38 to 0.56, respectively, as the chilled-water inlet temperature increases from 8 to 24 °C. We can observe similar trend for both cooling capacity and COP for multi-stage operation mode. Cooling capacity increases from 1.03 to 2.31 kW and COP increases from 0.17 to 0.25, respectively, as the chilled-water inlet temperature increases from 8 to 24 °C. This can be attributed from the fact that higher chilled-water inlet temperature causes higher amount of refrigerant being evaporated. However, with increasing chilled-water inlet temperature, the delivered chilled-water temperature rises from 3.83 to 15.63 °C and 3.47 to 14.18 °C, respectively, for multi-bed regenerative mode and multi-stage non-regenerative mode (see Fig. 3c). In order to obtain delivered chilled temperature around 7 °C, we should operate the dual-mode chiller with chilled-water inlet temperatures between 10 and 14 °C in both operation modes. Another noteworthy finding is that, with relatively higher chilled-water inlet temperature (above 14 °C) the cooling quality is better for multi-stage operation mode than that of the regenerative multi-bed mode.

4.2. Hot, cooling and chilled-water mass flow rates

The effect of heat transfer fluid (hot and cooling water) flow rates on cooling capacity and COP are shown in Figs. 4(a) and (b). Both cooling capacity and COP increase with higher heat transfer fluid flow rates in both multi-bed and multi-stage operation mode. Increasing the heat transfer fluid flow rate is highly effective in the range below 0.6 kg/s, above that the increase in cooling capacity and COP is only marginal. The effect of chilled water mass flow rate on cooling capacity and COP is shown in Figs. 5(a) and (b). As can be observed from Figs. 5(a) and (b), there is a tendency for cooling capacity and COP to increase with chilled water flow in the entire range studied. However, as can be seen from Fig. 5(c), the chilled-water outlet temperature visibly increase with higher water flow rates, which makes it impractical to utilize high flow rates of chilled water.
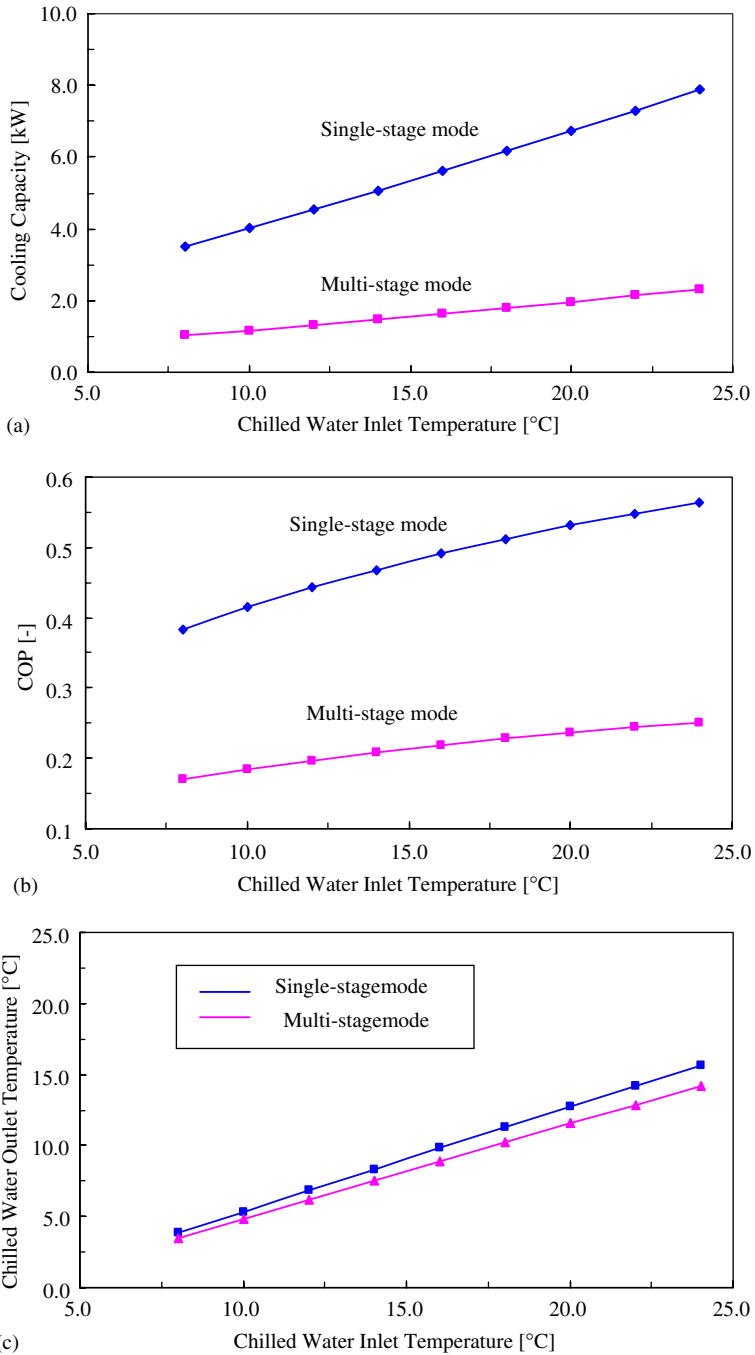
Fig. 3. (a) Chilled-water temperature effect on cooling capacity. (b) Chilled-water temperature effect on COP. (c) Effect of chilled-water inlet temperature on its outlet temperature.
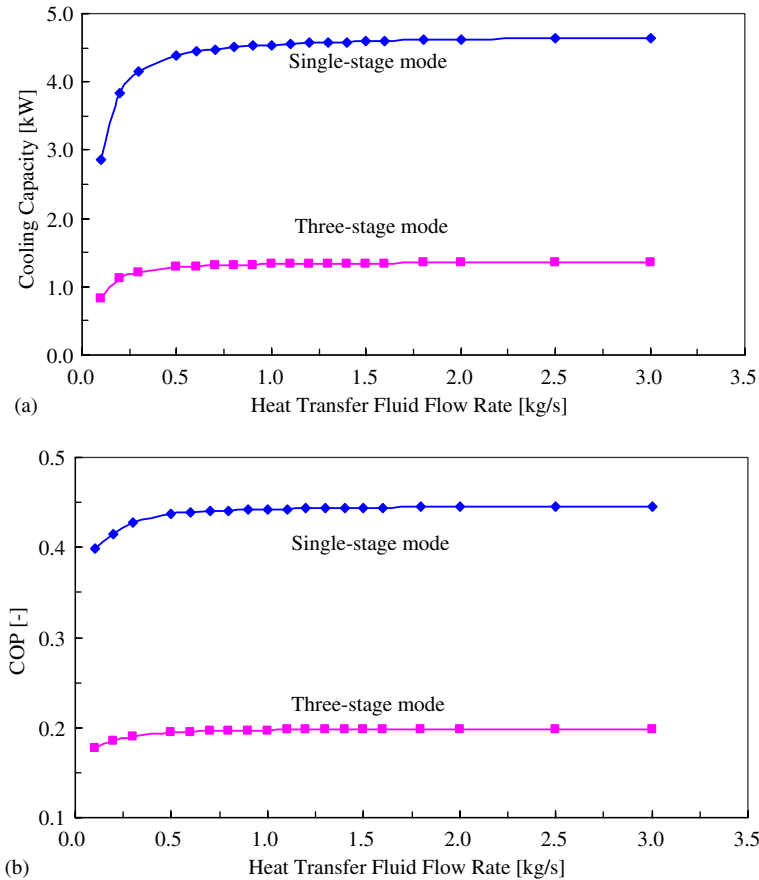
Fig. 4. (a) Heat transfer fluid flow rate effect on cooling capacity. (b) Heat transfer fluid flow rate effect on COP.

4.2.1. Adsorption–desorption cycle times

Figs. 6(a), (b) show the effect of adsorption–desorption cycle times on cooling capacity and COP. The highest cooling capacity values (Fig. 6a) were obtained for cycle times between 180 and 360 s. When cycle times are shorter than 180 s, there is not enough time for adsorption and desorption to occur satisfactorily, and cooling output decreases abruptly. For cycles longer than 360 s, on the other hand, cooling capacity decreases gradually since adsorption tends to become less intense after the first 5 min, gradually decreasing as the adsorbent approaches its equilibrium condition. The COP (Fig. 6b) increases uniformly with cycle time. This occurs because of lower consumption of driving heat with longer cycles.

4.2.2. Control strategy based on cycle time

The above discussed behavior of gradual decrease in cooling capacity (in kW) with longer cycle times can be utilized as a control strategy for partial load operation. Fig. 6(a) provides a guideline for the design and optimization of a control system aimed not only at avoiding excessive on/off operation, but also at providing maximum cooling capacity
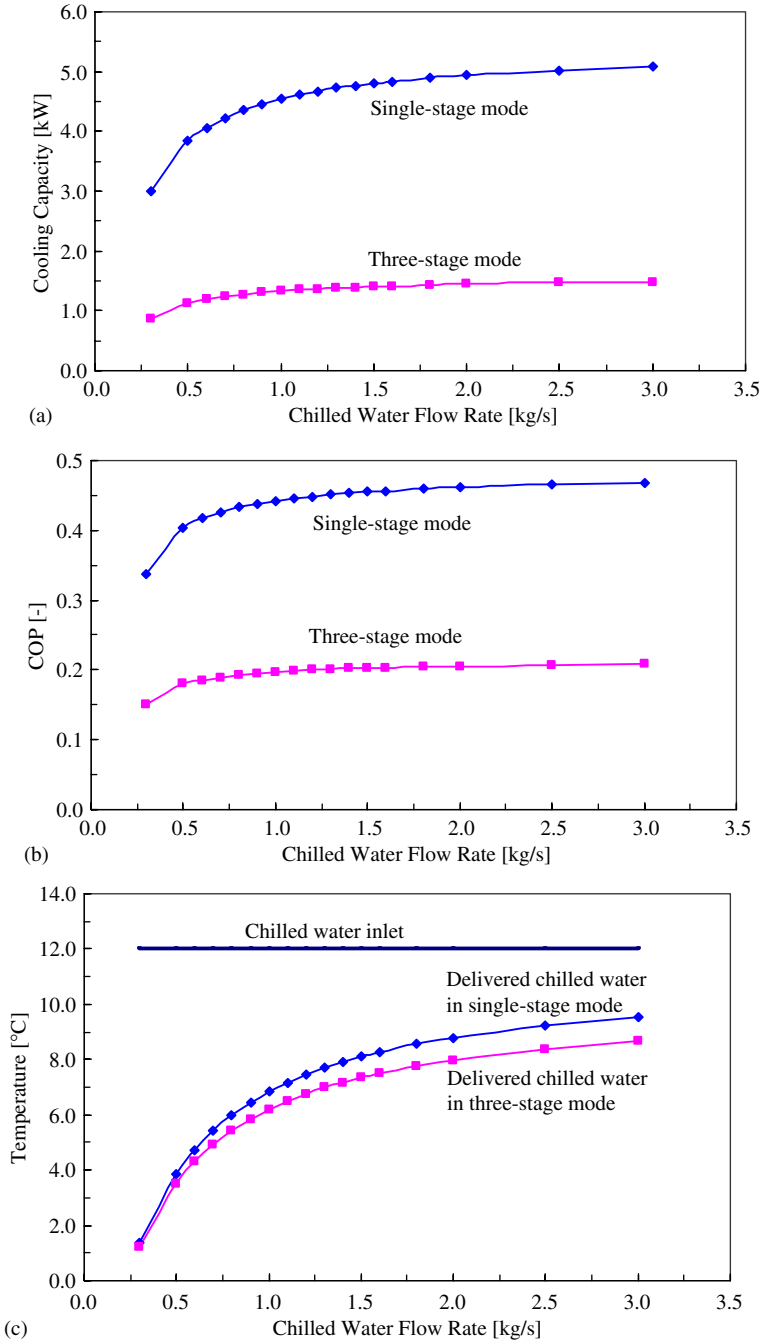
Fig. 5. (a) Chilled-water flow rate effect on cooling capacity. (b) Chilled-water flow rate effect on COP. (c) Effect of chilled-water flow rate on its outlet temperature.
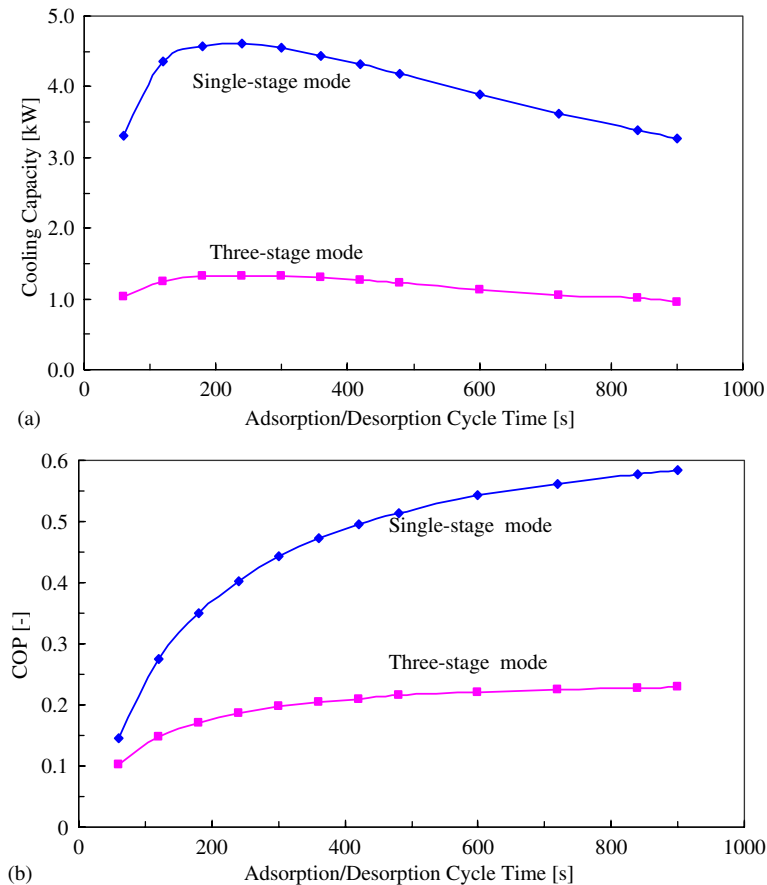
Fig. 6. (a) Adsorption/desorption cycle time effect on cooling capacity. (b) Adsorption/desorption cycle time effect on COP.

during high-demand periods (cycles between 200 and 360 s) and enhanced COP in low-demand conditions (longer cycles).

4.3. Irreversible heat losses

The regenerative temperature lifts (temperature difference between hot and cooling water inlet) effect on irreversible heat losses of batched cycle operations are illustrated with reference to the simulated pressure–temperature (P–T) diagram of Fig. 7. The simulated P–T diagrams with standard operating conditions (see Table 3) for both multi-bed and multi-stage modes are depicted in this figure. The adsorbent temperature at the end of adsorption is about $2\,°C$ higher than the coolant inlet temperature for both modes. However, at the end of desorption, adsorbent temperature in both modes is about $1.5\,°C$ lower than the hot water inlet temperature. These temperature differences ($\Delta T_{ads-water}$ and $\Delta T_{des-water}$), of which the former is particularly detrimental to cooling capacity, occur because the 330 s cycle time (adsorption–desorption cycle time 300 s and pre-heat/pre-cool cycle time 30 s) is not long enough for the adsorbent temperature to change by 50 and 20 K
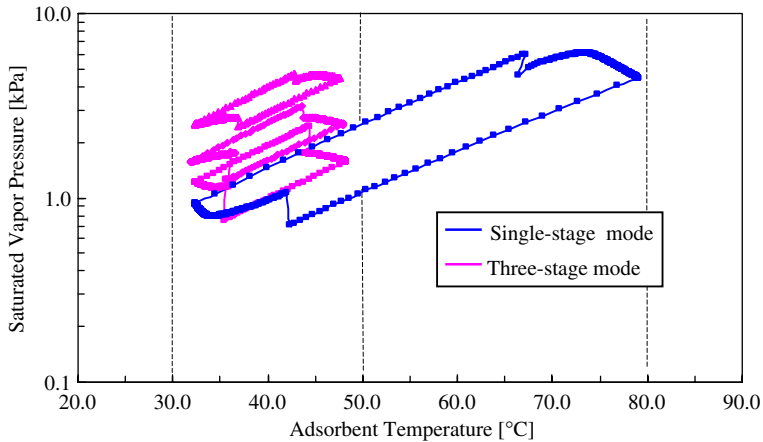
Fig. 7. Pressure–temperature diagram of dual-mode adsorption cycle (rated conditions).

Table 3
Standard operating conditions

	Hot water in		Cooling water in		Chilled water in	
	Temp (°C) flow (kg/s)		Temp (°C) flow (kg/s)		Temp (°C) flow (kg/s)	
Multi-bed Mode	80	0.57	30	0.57	12	0.084
Multi-stage	80	0.57	30	0.57	12	0.084
Cycle time	Adsorption/Desorption		Pre-heating/Pre-cooling			
	300 s		30 s			

between driving heat source and cooling water temperatures, respectively, for single-stage multi-bed and multi-stage modes. Improvements in the sorption element, such as lower thermal capacitance and higher thermal conductance could contribute to shorten the time requirement of sensible heating and cooling. Another noteworthy observation from Fig. 7 is that in the both operation modes, adsorbent cooling is clearly slower than desorption heating, which suggests not only that adsorption may take longer time to occur than desorption, but also that removal of adsorption heat may be more problematic than addition of heat for desorption.

5. Conclusions

The practical operational limits and performance characteristics of an innovative dual-mode, multi-stage, multi-bed regenerative adsorption chiller powered by near-ambient temperature renewable energy or waste heat sources were investigated. The main conclusions are as follows:

(1) The main advantage in the innovative dual-mode cycle is its ability to utilize effectively low-grade renewable energy or waste heat of temperature between 40 and 95 °C as the driving heat sources.

(2) Both cooling capacity and COP increase with higher chilled-water inlet temperature and the delivered chilled-water inlet temperature also increases with increasing chilled-water inlet temperature suggests that increasing chilled-water inlet temperature above 14 °C results in declination of cooling quality.

(3) Cooling capacity and COP increase with the increase of hot and cooling water flow rates in both modes of the dual-mode adsorption chiller. However, the increase of heat transfer fluid flow rates above 1.0 kg/s is no longer beneficial.

(4) Both cooling capacity and COP increase moderately with the increase in chilled-water flow rate. However, the delivered chilled-water temperature increases sharply with the increase of chilled-water flow rates in both operation modes of the advanced chiller.

(5) In both operation modes, long cycle times (above 360 s) result in COP gains and cooling capacity losses, which make them suitable for partial load operations.

(6) Irreversible heat losses increase with regeneration temperature lift, as a result of repeated switching between driving heat source and coolant temperatures.

(7) From the above perspectives, the use of unexploited renewable energy or waste heat at near-ambient temperature may offer attractive possibilities for improving energy efficiency.

Acknowledgements

References

[1] Saha BB, Koyama S, Kashiwagi T, Akisawa A, Ng KC, Chua HT. Waste heat driven dual-mode, multi-stage, multi-bed regenerative adsorption system. Int J Refrigerat 2003;26:749–57.

[2] Shelton SV, Wepfer WJ, Miles DJ. Square wave analysis of the solid—vapor adsorption heat pump. Heat Recover Systems CHP 1989;9(3):223–47.

[3] Härkönen M, Aittomäki A. Analytical model for the thermal wave adsorption heat pump cycle. Heat Recover System CHP 1992;12(1):73–80.

[4] Critoph RE. Forced convection adsorption cycle with packed bed heat regeneration. Int J Refrigerat 1999;22(1):38–46.

[5] Meunier F. Theoretical performances of solid adsorbent cascading cycles using the zeolite-water and active carbon–methanol pairs: four case studies. Heat Recover Systems CHP 1986;6(6):491–8.

[6] Douss N, Meunier F. Experimental study of cascading adsorption cycles. Chem Eng Sci 1989;44(2):225–35.

[7] Saha BB, Boelman EC, Kashiwagi T. Computer simulation of a silica gel–water adsorption refrigeration cycle—the influence of operating conditions on cooling output and COP. ASHRAE Trans 1995;39(2):348–57.

[8] Saha BB. Performance analysis of advanced adsorption cycle. PhD. thesis, Tokyo University of Agriculture and Technology, 1997.

[9] Sakoda A, Suzuki M. Fundamental study on solar powered adsorption cooling system. J Chem Eng Japan 1984;17(1):52–7.