LOW TEMPERATURE DRIVEN HYBRID ADSORPTION COOLING SYSTEM WITH A MECHANICAL BOOSTER PUMP

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ABSTRACT

The use of waste heat of low temperatures is an important problem from the environmental considerations. Notice that adsorption cycles have a distinct advantage over other systems of their ability to produce cooling by using low waste heat and also being absolutely benign for the environment. However the present available adsorption chillers are not suitable to adapt the systems like fuel cell with producing low waste heat less than 60℃. Hence we proposed a new adsorption refrigeration cycle combined with a mechanical booster pump which is placed in between adsorbent beds and condenser to reduce the pressure inside the adsorption bed for regeneration, or evaporator to be adsorbed at pressurized condition, that is, in order to increase an amount of driving refrigerant per cycle. This work deals with the performance testing of a cooling system based on a new adsorption cycle with a cooling capacity of 50 kW, an estimated cooling COP larger than 10.

KEYWORDS

Adsorption refrigeration, Low waste heat recovery, Advanced adsorption cycle, Performance

1. INTRODUCTION

The use of waste heat of low temperatures is an important problem from the environmental considerations. Notice that adsorption cycles have a distinct advantage over other systems because of their ability to produce cooling by using low waste heat, and for being absolutely benign for the environment. However the present available adsorption chillers are not suitable to adapt the systems like fuel cells with producing low waste heat less than 60°C. Hence we proposed a new adsorption refrigeration cycle combined with a mechanical booster pump which is placed in between adsorbent beds and condenser to reduce the pressure inside the adsorption bed for regeneration, or evaporator to be adsorbed at pressurized condition, that is, in order to increase an amount of driving refrigerant per cycle. The former cycle using depressurizing-regeneration was found to be more effective in possessing a wide operational range. It can be expected to have a 5°C chilled water under a 30°C condensation and a 2.4 pressure ratio of the mechanical booster pump.

This work deals with a simulation analysis and the performance testing of a cooling system based on a new adsorption cycle with a cooling capacity of 50 kW, an estimated cooling COP larger than 10.

2. SIMULATION

2.1 SIMULATION MODEL

As shown in Fig.1, the chiller consists of two beds packed with silica-gel as adsorbent, a condenser, a mechanical booster pump and an evaporator.

In the bed-evaporator interaction, adsorption takes place in the adsorbed bed by rejecting heat to external heat source (cooling water), while evaporation of adsorbate occurs in the evaporator which absorbes heat from outside of the system (chilled water) Cooling can be achieved by making use of the heat of evaporation. Simultaneously, in the bed-condenser coupling, desorption takes place in the fully adsorbed bed by receiving heat from external heat source (hot water), while condensation of adsorbate occurs in the condenser to dissipate condensing heat to outside of the system (cooling water). In the following cycle, the two beds change their role mutually, and thus the chiller can work continuously. Since adsorption and condensation are exothermic whereas desorption is endothermic, hence the cooling water and hot water supply are necessary to maintain the process.

In switching to the other process, that is, from the regeneration to the adsorption, the heated bed must be cooled and vice versa, in order to transfer to the next process. For this switching period all vapor valves are closed so as to be isosteric, i.e., neither adsorption nor desorption occurs.

When the mechanical booster pump is used to pump the vapor from the regeneration bed to the condenser, the bed can be regenerated easily and the required temperature of heat source can be reduced.

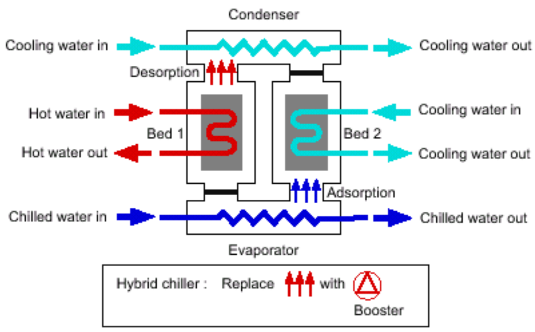

Fig. 1 Schematic of the chiller

Fig.1 Schematic view of the adsorption system with a mechanical booster pump.

2.2 MATHEMATICAL MODELING

2.2.1Conservation of mass

The rate of adsorption/desorption is governed by the linear driving force kinetic equation.

$$\frac{dq}{dt} = 15\frac{D_{s0}e^{-E_a/(RT)}}{R_p^2}(q^* - q) \qquad (1)$$

Where q^* is given by the following D-A isothermal equation.

$$q^* = q_0 \exp\left[-(A/E)^n\right] \qquad (2)$$

2.2.2 Conservation of energy

The modeling of hot bed and its heat exchanger in the -booster-condenser interaction is discussed here.
Its energy balance can be written as

<u>Adsorbent bed</u>

$$(M_{sg}C_{p,sg} + C_{p,Hex}M_{Hex})\frac{dT_{bed,j}}{dt} + M_{sg}q_{bed,j}\frac{dh_{ads}}{dt} =$$
$$M_{sg}\frac{dq_{bed,j}}{dt}(\Delta H_{ads} + \delta(h_{gs}(T_{cond}) - h_g(P_{cond}, T_{cond}))$$
$$-U_{heating}\frac{Abed}{N}\sum_{K=1}^{N}(T_{bed,j} - T_K) \qquad (3)$$

<u>Heat source</u>

$$\rho_f C_{pf}(T_k)\frac{V_{bed}}{N}\frac{dT_k}{dt} = \frac{dM_{heating}}{dt}\left[h_f(T_{k-1}) - h_f(T_k)\right] +$$
$$U_{heating}\frac{A_{bed}}{N}(T_{bed,J} - T_k) \qquad (4)$$

<u>Condenser</u>

$$\left[C_{pf}(T_{cond})M_{ref,cond} + C_{p,cond}M_{cond}\right]\frac{dT_{cond}}{dt} =$$
$$-\left[h_g(P_{cond}, T_{bed,j}) - h_f(T_{cond})\right]M_{sg}\frac{dq_{bed,j}}{dt}$$
$$-U_{cond}\frac{A_{cond}}{N_2}\sum_{K=1}^{N_2}(T_{cond} - T_K) \qquad (5)$$

<u>Cooling water</u>

$$\rho_f C_{pf}(T_k)\frac{V_{cond}}{N}\frac{dT_k}{dt} = \frac{dM_{cond}}{dt}\left[h_f(T_{k-1}) - h_f(T_k)\right] +$$
$$U_{cond}\frac{A_{cond}}{N}(T_{cond} - T_k) \qquad (6)$$

<u>Mechanical booster pump</u>

The bed pressure in case without the booster pump is given as

$$\frac{dP_{bed}}{dt} = \frac{dp_{sat}(T_{cond})}{dt} \qquad (7)$$

In case of applying the booster pump is given as

$$\frac{dp_{bed}}{dt} = \psi(t) \qquad (8)$$

Where •(t) denotes the bed pressure change caused by the booster. In this analysis, it is assumed by a minus constant for simplification, which means the bed pressure decreases linearly due to depressurizing.

The whole system is finally modeled in the same way to the modeling of cold bed (adsorption process) and expressed by a set of differential equations. A variable order and multi-step solver provided in MATLAB is adopted to solve the equation.

2.2.3 System performance

The system performance of an adsorption cooling chiller can be characterized by the coefficient of performance (COP).

<u>Cooling power</u>

$$Q_{evap} = \frac{dM_{chilled}}{dt}\left[C_{pf}(T^{in}_{chilled})\right]\int_0 (T^{in}_{chilled} - T^{out}_{chilled})\,d\left(\frac{t}{t_{cycle}}\right) \qquad (9)$$

<u>COP</u>

$$COP = \frac{Q_{evap}}{\frac{dM_{heating}}{dt}\left[C_{pf}(T^{in}_{heating})\right]\int_0^1 (T^{in}_{heating} - T^{out}_{heating})\, d\left(\frac{t}{t_{cycle}}\right) + W_{pump}} \quad (10)$$

2.3 RESULTS AND DISCUSSION

Figure 2 to 3 show analyzed results to the experiment which was carried out with using a bench scale adsorption cooling unit with a mechanical booster pump. In the experiment, the cycle operation was made intermittently with and without a mechanical booster pump to see the cooling effect, under the conditions of hot water inlet 60•, cooling water inlet 30•,chilled water inlet 15• and cycle time 10 minute, respectively.

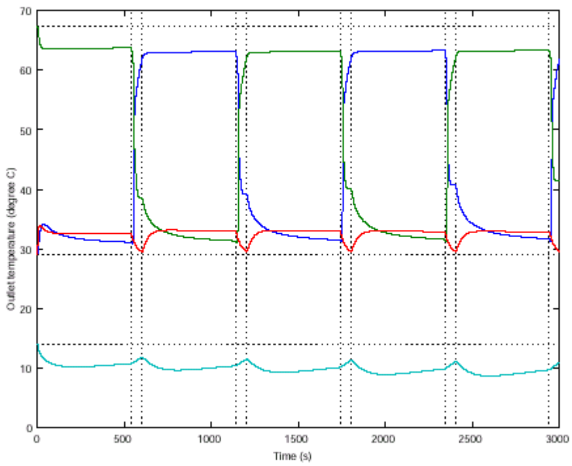

Figure 2 The time dependency of outlet temperatures and pressure in the bed.

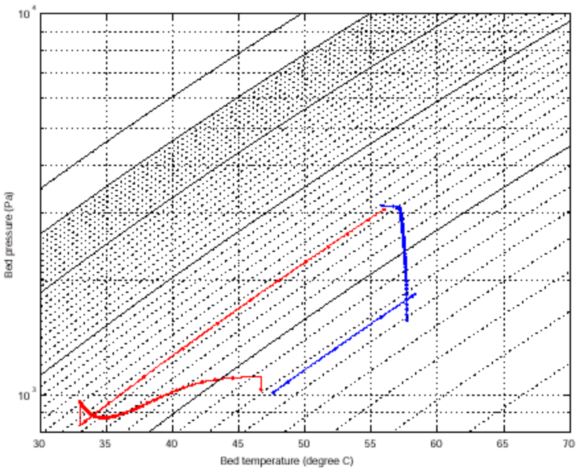

Figure 3 Deuhring diagram of the cycle-steady -state condition of the beds.

The experimental data are also shown in the Fig. 4, where time dependent pressure changes of beds, evaporator and condenser are presented as well as outlet temperatures of condensation, chilled water and the bed temperatures. Notice that the analyzed result is roughly agreed with that of experiment.

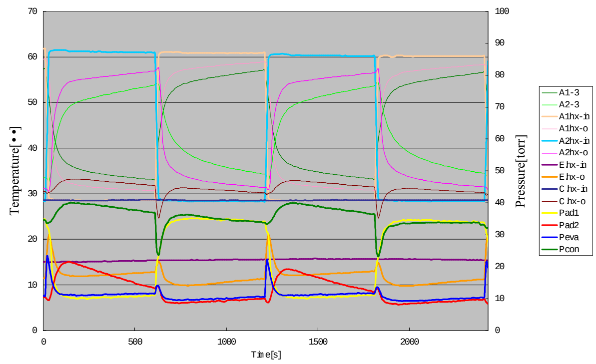

Figure 4 The time dependency of outlet temperatures and pressure in the bed.

It is sure that the analyzed result is quite sensitive to the heat transfer coefficients of heat exchangers. It is hard to estimate the right values of adsorbent bed in the adsorption and desorption processes. However the developed simulation code is considered to be quite effective to predict the performance.

3. PILOT MACHINE

3.1 DESIGN SPECIFICATION

Table 1 shows the specification of a pilot plant which is expected to be installed for testing. The expected COP value is here evaluated by the ratio of the cooling capacity to the pumping power consumption.

Table 1 The specification of a pilot plant

Model	Unit	ADRh-50k
Adsorbent/refrigerant	–	Silica-gel/ water
Cooling capacity	kW	50
COP	–	10.2
Chilled water Inlet/outlet temperature Flow rate	°C m^3/H	14.0/9.0 8.6
Cooling water Inlet/outlet temperature Flow rate	°C m^3/H	29/33 30.0
Hot water Inlet/outlet temperature Flow rate	°C m^3/H	55.0/50.0 14.3
Amount of working refrigerant per unit adsorber	Δq kg-H_2O/kg-silica-gel	0.05
Mechanical booster pump Throughput of pumping Power consumption	m^3/min. kW	33.6 4.5

3.2 SIMULATION

A simulation analysis has been conducted to predict the performance of hybrid adsorption refrigeration system. The values for the parameters used in the analysis is given in the Table 2.

The figure 5 to 6 shows the results of temperature profile as adsorbent beds, outlet of cooling water of the condenser and chilled water, in addition to the adsorption cycle.
It was confirmed through the numerical calculations that the design values were satisfied to meet the requirements of specification. However the system performance is much sensitive to the switching time. Since the two bed temperatures should be enough pre-heated or pre-cooled in the mean time.

Table 2 The values for the parameters used in the analysis

D_{s0}	2.54E-4 m^2/s
E_a	4.2E4J/mol
R_p	0.5E-3 m
$dM_{cooling}/dt$	5.0kg/s
$dM_{heating}/dt$	3.97kg/s
$dM_{chilled}/dt$	2.40 kg/s
dM_{cond}/dt	3.333kg/s
$U_{cooling}$	700W/m^2/K
$U_{heating}$	900 W/m^2/K
$U_{chilled}$	2330W/m^2/K
U_{cond}	3460W/m^2/K
A_{bed}	20.358m^2
A_{evap}	11.4m^2
A_{cond}	13m^2
M_{sg}	255 kg
$C_{p,Hex}M_{Hex}$	318.3 kJ/K
$C_{P,evap}M_{evap}$	260.3kJ/
$C_{P,cond}M_{cond}$	109kJ/K
$C_{P,sg}M_{sg}$	232.8kJ/K $C_{P,sg}$=0.92kJ/(kg•K), •=2200kg/m^3 ADS=0.343 m^3
V_{bed}	0.0336m^3
V_{cond}	0.0227m^3
V_{evap}	0.0141m^3
ΔH_{ads}	2640kJ/kg

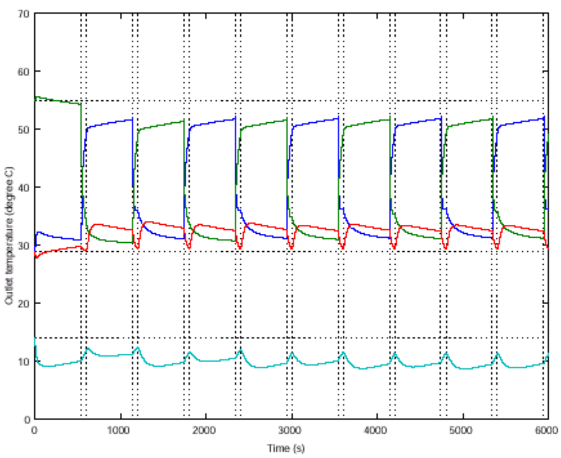

Figure 5 The time dependency of outlet temperatures

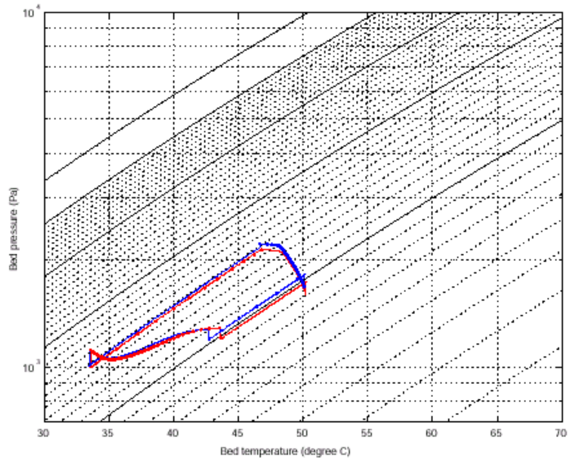

Figure 6 Deuhring di agram of the cycle-steady -state condition of the beds.

3.4 TEST RESULTS AND DISCUSSION

A performance testing has been conducted under the specified conditions for the heat transfer fluid(hot, cooling, and chilled water), while those parameters as a cycle time(10 to 20 min.), a switching time(15 to 60 sec.) and a pumping rates of mechanical booster pump(36 to 50 m^3/mini.), which are not measured values but evaluated values in consideration of expected pumping efficiency at a corresponding rotational speed, were varied in order to obtain an optimum operation. Figure 7 shows the 50 kW pilot machine equipped with the mechanical booster pump.

Figure 7 Pilot machine equipped with the mechanical booster pump.

Figure 8 demonstrates a typical cyclic steady state temperature and pressure profile versus time, where a cycle time is 15 min., a switching time is 30 sec. and a rotational speed of 2007 rpm(pumping rate :52 m^3/min.), respectively.

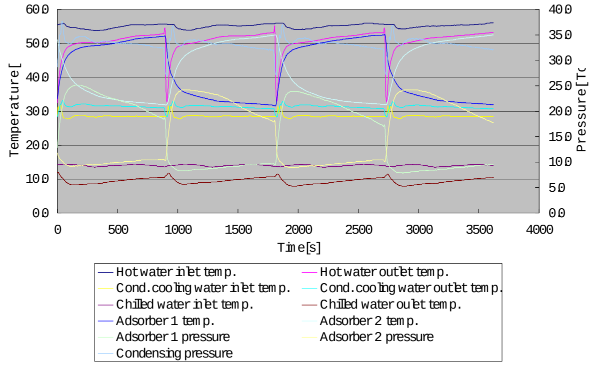

Figure 8 A typical cyclic steady state temperature and pressure prof ile vs. time.

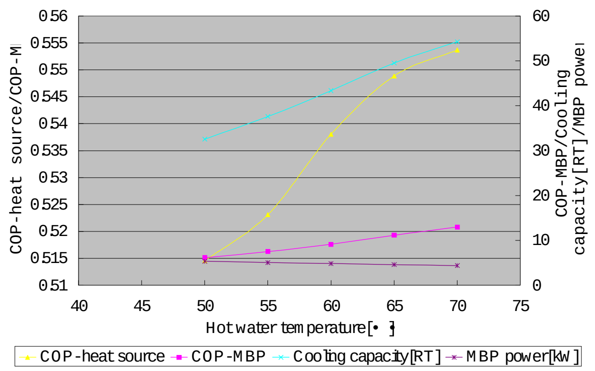

Figure 9 Hot water temperature dependency on cooling capacity & COP.

Figure 9 demonstrates its hot water temperature dependency on a cooling capacity, a power consumption of mechanical booster pump and COP. Notice that the cooling capacity and COP increase with increase of hot water temperature. However they are lower than the specified values of 50 kW and 10, respectively.
This reason is considered to be due to a low activation energy and that the desorption from the micro pores will not take place easily by pumping. Whereas with increase of the adsorber temperature the activation energy gets to be large enough to allow the desorption.

While the lower COP is caused by a low efficiency of inverter by which the motor for pumping is driven. The COP in the figure includes the inverter loss of approx.15%.

It was found that the cooling capacity is increased with the increase of cycle time. The longer cycle time raises the adsorber temperature which is closer to the heat input temperature 55°C and moreover lowers the adsorber pressure. Those effects cause the increase of the working refrigerant water per kg-silicagel (Δq) which is the deference between desorbed and adsorbed states and is directly related with the cooling capacity.

In addition its COP can be improved by the decrease of Δp which is the pressure difference between suction and delivery pressures. The power consumption is roughly proportional to Δp, in this case condensing pressure is constant at 40 torr. Since the suction pressure can be raised by the increase of adsorber temperature to the heat input 55°C, and so results in the decrease of Δp.

4. CONCLUSION

In this study,
1) we developed a simulation program of a transient two-bed adsorption chiller with a mechanical booster pump.
2) It was found that the simulation program was effective to predict its performance.
3) Based on our analysis it was confirmed the design values were adequate to obtain the specified performance.
4) The pilot machine was successfully operated and demonstrated the nominal cooling capacity 50 kW and COP 10 as expected. This success allows us widely to use low grade waste heat as a fuel cell cogeneration.

5. REFERENCE

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NOMENCLATURE

$C_{p,cond}$	specific heat capacity of condenser heat exchangeing material; J/kg K
$C_{p,evap}$	specific heat capacity of evaporator heat exchangeing material; J/kg K
$C_{p,Hex}$	mass weighted average specific heat capacity of heat exchanger tube-fin assembly; J/kg K
$C_{p,sg}$	specific heat capacity of silica-gel; J/kg K
C_{pf}	specific heat capacity of liquid refrigerant water; J/kg K
COP	ratio of cycle averaged cooling capacity to power consumption of booster pump:-
D_{s0}	pre-exponent constant in the kinetic equation; m^2/s
E_a	activation energy of surface diffusion; J/mol
h_{ad}	partial enthalpy of adsorbate in adsorbent-adsorbate system; J/kg
h_f	specific enthalpy of liquid water; J/kg
h_g	specific enthalpy of gaseouswater; J/kg
M_{cond}	mass of condenser heat exchanger tube; kg
M_{evap}	mass of evaporator heat exchanger tube; kg

M_{Hex}	mass of heat exchanger tube-fin assemblyin bed; kg
$M_{ref,evap}$	liquid refrigerant mass in evaporator; kg
M_{sg}	silica-gel mass; kg
P	pressure; Pa
P_{cond}	condenser pressure; Pa
P_{evap}	evaporator pressure; Pa
P_{sat}	saturate vapor pressure; Pa
q	fraction of refrigerant adsorbed by the adsorbent; kg/kg of dry adsorbent
q_0	limit of adsorption by the adsorbent; kg/kg of dry adsorbent
E	characteristics energy; J/mol K
n	constant; -
q^*	fraction of refrigerant which can be adsorbed by the adsorbent under saturation condition; kg/kg of dry adsorbent
Q_{evap}	cycle averaged cooling capacity; W
R	universal gas constant: J/mol K
R_p	average radius of silica-gel
t	time; s
t_{cycle}	cycle time; s
T	temperature; °C
$T_{bed,}$	bed temperature; °C
T_{cond}	condenser temperature; °C
T_{evap}	evaporator temperature; °C
T_K	temperature of the Kth element of the bed heat exchanger (k=1,….,N_{bed}); °C
$U_{chilled}$	evaporator heat transfer coefficient; W/m^2 K
U_{cond}	condenser heat transfer coefficient; W/m^2 K
$U_{cooling}$	adsorber heat transfer coefficient; W/m^2 K
$U_{heating}$	desorber heat transfer coefficient; W/m^2 K
V_{bed}	internal volume of heat exchanger tubes in the bed; m^3
V_{cond}	internal volume of heat exchanger tubes in the condenser; m^3
V_{evap}	internal volume of heat exchanger tubes in the evaporator; m^3
$dM_{chilled}/dt$	chilled water flow rate; kg/s
dM_{cond}/dt	condenser cooling water flow rate; kg/s
dM_{cond}/dt	condenser coolant flow rate; kg/s
$dM_{cooling}/dt$	coolant flow rate through adsorber; kg/s
$dM_{heating}/dt$	heat source flow rate through desorber; kg/s
ΔH_{ads}	isosteric heat of adsorption; J/kg
δ	flag which governs desorber transients; -
ρ_f	density of liquid water; kg/m^3