



THE PERFORMANCE ANALYSIS OF A SILICA GEL/WATER ADSORPTION CHILLER AND DYNAMIC HEAT AND MASS TRANSFER CHARACTERISTICS OF ITS ADSORBENT BED: A PARAMETRIC STUDY

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Abstract: The performance of two adsorbent bed silica gel-water adsorption chiller and the influences of its adsorbent bed dimensions, the velocity of heat exchange fluid and the adsorbent particle diameter on the transient distributions of the solid phase temperature, adsorbate concentration and the pressure are numerically examined in this study. A transient two-dimensional local thermal non-equilibrium model that takes into account both internal and external mass transfer resistances is developed to simulate the adsorption chiller considered herein. The internal and external mass transfers are predicted by the linear driving force (LDF) model and Darcy's equation, respectively. It is found that an increase in the adsorbent particle diameter results in the decrease in both the coefficient of performance (COP) and specific cooling power (SCP) of the adsorption chiller. On the other hand, the system performance is nearly independent from the variation of velocity of heat exchange fluid. An increase in the bed thickness leads to an increase in COP and a decrease in the SCP. The influence of adsorbent bed length on the performances of system can be neglected. The results of our simulations provide useful guidelines for the design of this type of adsorption chillers.
Keywords: Adsorption, Cooling, Silica gel/water, LDF, LTNE, COP

ADSORPSİYONLU SİLİKA JEL/SU SOĞUTMA SİSTEMİNİN PERFORMANS ANALİZİ VE ADSORBAN YATAĞININ DİNAMİK ISI VE KÜTLE TRANSFERİ DAVRANIŞI: PARAMETRİK ÇALIŞMA

Özet: İki adsorban yataklı silika jel/su çalışma çiftini kullanan adsorpsiyonlu soğutma sisteminin performansı ve sistemin adsorban yatak boyutlarının, ısı değiştirici akışkan hızının ve adsorban tanecik çapının zamana bağlı olarak katı fazın sıcaklık dağılımı, adsorplanan konsantrasyonu ve basınç dağılımı üzerine etkileri sayısal olarak bu çalışmada incelenmiştir. Zamana bağlı iki boyutlu ve lokal olarak ısıl dengenin olmadığı iç ve dış kütle transfer dirençlerini dikkate alan sayısal bir model söz konusu sistemi incelemek için geliştirilmiştir. İç ve dış kütle transfer dirençleri sırasıyla doğrusal itici güç modeli ve Darcy denklemi vasıtasıyla hesaplanmıştır. Parametrik çalışma neticesinde artan adsorban tanecik çapının sistemin performans katsayısı ve özgül soğutma gücü üzerinde bir düşüşe yol açtığı bulunmuştur. Diğer taraftan, sistem performansının ısı değiştirici akışkan hızının değişiminden neredeyse bağımsız olduğu elde edilmiştir. Adsorban yatak kalınlığındaki bir artış sistemin performans katsayısında yükselmeye ve özgül soğutma gücünde ise düşüşe yol açmıştır. Adsorban yatak uzunluğunun sistem performansı üzerine olan etkisi ihmal edilebilecek düzeydedir. Parametrik çalışmamızın sonuçları bu tip adsorpsiyonlu soğutma sistemlerinin tasarımı için faydalı öneriler ortaya koymaktadır.

Anahtar Kelimeler: Adsorpsiyon, Soğutma, Silika Jel/Su, LDF, LTNE, COP

Nomenclature

a_v	Area of gas–solid interface per unit volume [1/m]	d_p	Average diameter of the adsorbent particle [m]
C_p	Specific heat [J/(kgK)]	E_a	Activation energy of surface diffusion [J/mol]
D_e	Equivalent diffusivity in the adsorbent particles [m ² /s]	H	Convective heat transfer coefficient between the adsorbent bed and cooling fluid [W/(m ² K)]
D_o	Reference diffusivity [m ² /s]	h_g	Interfacial convective heat transfer

	coefficient [W/(m ² K)]
$h_{m,ads}$	Convective heat transfer coefficient between adsorbent bed and the metallic wall [W/(m ² K)]
K	Permeability [m ²]
k_m	Mass transfer coefficient within the adsorbent particles [1/s]
K	Thermal conductivity [W/(mK)]
L	Length of the adsorbent bed [m]
L_s	Latent heat of evaporation [J/kg]
N_{ud}	Nusselt number
P	Pressure [kPa]
Pr	Prandtl number
Q	Heat of adsorption [J/kg]
R	Universal gas constant [J/(molK)]
Re_d	Reynolds number
R_g	Specific gas constant for water vapor [J/(kgK)]
R_1	Inner diameter of the metallic wall [m]
R_2	Outer diameter of the metallic wall [m]
R_3	Outer diameter of the adsorbent bed [m]
r	Radial coordinate [m]
T	Temperature [K]
t	Time [s]
u	Gas phase velocity in radial direction [m/s]
v	Gas phase velocity in axial direction [m/s]
X	Adsorbate concentration [kg _w /kg _{ad}]
X_{∞}	Equilibrium adsorption capacity [kg _w /kg _{ad}]
z	Axial coordinate [m]

Greek symbols

μ	Viscosity [Ns/m ²]
ρ	Density [kg/m ³]
ε_t	Total porosity
ε_b	Bed porosity
ε_p	Particle porosity
λ_{g-e}	Effective thermal conductivity for the gas phase [W/(mK)]
λ_{s-e}	Effective thermal conductivity for the solid phase [W/(mK)]
λ_m	Thermal conductivity of metallic wall [W/(mK)]
λ_f	Thermal conductivity of heat exchange fluid [W/(mK)]

Subscripts

co	Condenser
e	Evaporator
f	Heat exchange fluid
g	Gas phase
i	Initial
m	Metallic wall
re	Regeneration
s	Solid phase
sat	Saturation

INTRODUCTION

The environmental-friendly adsorption cooling (AC) systems are an attractive alternative to the traditional vapor-compression cooling systems as they are

characterized by their low operation and maintenances costs, simple control and the absence of vibration and corrosion problems (Anyanwu, 2000). AC systems can be powered by a low grade heat source such as solar, waste heat or geothermal. Although the AC systems have these advantages, their drawbacks are the intermittent operation, the requirements of special designs to maintain high vacuum, the large volume and weight relative to traditional refrigeration systems, the low specific cooling power (SCP) and the low coefficient of performance (COP) (Anyanwu, 2003). On the other hand, the poor heat and mass transfer within the adsorbent bed are vital to the development and application of the adsorption refrigeration technology (Wang *et al.*, 2010). Hence, an enhancement on the heat and mass transfer conditions inside the adsorbent bed leads to a more efficient system (Anyanwu, 2003). This can be accomplished by using an adsorbent-adsorbate working pair with a high cyclic adsorption capacity and thermal conductivity and a low resistance to adsorbate flow as well as a better heat management during the adsorption cycle. As a result, adsorption cooling systems have attracted much research attention in recent years.

Thermodynamic investigation on the multi bed adsorption chiller using silica gel-water is carried out by several researchers (Ahmed *et al.*, 2012; Lu and Wang, 2013; Mitra *et al.*, 2014). Although some valuable results are presented in these studies, the proposed models are only focus on the COP values of the systems without providing any information about the transient heat and mass transfer processes. Most of the models proposed previously for heat transfer within the adsorbent bed are based on the Local Thermal Equilibrium (LTE) assumption. It means that there is a thermal equilibrium between the gas and solid phases and hence, single energy equation is sufficient to describe the temperature fields of gas and solid phases. However, local thermal equilibrium assumption cannot be used in some circumstances (Duval *et al.*, 2004). Mhimid (1998) studied the heat and mass transfer in a zeolite bed during water desorption using the local thermal equilibrium and local thermal non-equilibrium models and the results showed that the local thermal equilibrium assumption is not valid in regions with high rates of heat transfer. Jemni and Nasrallah (1995) investigated transient heat and mass transfer in a metal-hydrogen reactor and they concluded that the local thermal equilibrium model is not valid in the whole reactor. Therefore, it is considered that local thermal equilibrium model may not lead to realistic results so two equation model (Local Thermal Non-Equilibrium model) taking into account the heat transfer between the phases is employed in this study.

The mass transfer inside the adsorbent bed is described by two mechanisms, i.e., internal and external mass transfer. Adsorbate flows from a solid adsorbent particle surface to inner points of the particle and through the voids between the solid adsorbent particles are called internal and external mass transfer,

respectively. The internal mass transfer can be evaluated by the Linear Driving Force (LDF) or Solid Diffusion (SD) models. However if the resistance to internal mass transfer is insignificant, adsorption equilibrium can be considered without leading to remarkable deviation from the reality and most of published studies assume adsorption equilibrium. The external mass transfer is typically predicted by Ergun's and Darcy's equation.

The external mass transfer resistance is often ignored and a uniform pressure is employed. The Darcy's equation is used in the present study due to low velocity of gas phase within the adsorbent bed (Wu *et al.*, 2009). The some of previous studies are classified in Table 1 according to proposed model for heat and mass transfer, working pair and model dimension.

Table 1. Classification of proposed models in term of their characteristics

Ref.	Dimension	Internal Mass Transfer	External Mass Transfer	Energy Equation	Working Pair
[12,17,18]	2D	LDF	Darcy's law	LTE	Zeolit13X/Water
[4,5,6,15,30]	Lumped	LDF	Uniform pressure	Uniform temperature	Silica gel/Water
[13]	2D	Adsorption equilibrium	Uniform pressure	LTE	Active carbon/ Ammonia
[14]	1D	LDF	Uniform pressure	LTE	Active carbon/ Methanol
[11,16,21,29]	1D	Adsorption equilibrium	Uniform pressure	LTE	Active carbon/ Ammonia
[19,20,23]	2D	LDF	Darcy's law	LTNE	Silica gel/Water
[22]	2D	LDF	Uniform pressure	LTE	Jiangxi AC809/ Methanol
[26,27]	3D	LDF	Darcy's law	LTE	Zeolithe 13x/water
[28]	3D	LDF	Darcy's law	LTE	Carbon active methanol

In this study, a two dimensional local thermal non equilibrium model is proposed for heat transfer. The external and internal mass transfer resistances are taken into account by linear driving force model (LDF) and Darcy's law, respectively. The effect of design and operating parameters on the performances of two adsorbent bed chiller using silica gel water and distributions of solid phase temperature, adsorbate concentration and the pressure inside the adsorbent bed are presented. The simulation result will provide useful information for the design and transient operation of these kinds of chillers.

MATHEMATICAL MODEL

The proposed mathematical model is based on two dimensional approach with the local thermal non equilibrium between the phases. In this model, the internal and external mass transfer resistances are taken into account by linear driving force model and Darcy's law, respectively. A schematic view of the adsorbent bed is presented in Fig 1. The adsorbent bed is a hollow cylinder which encloses a metal tube for the purpose of heat exchange between the solid adsorbent and the heating or cooling fluid within the tube. The proposed model is based on the following assumptions:

- The adsorbed phase is considered as a liquid and the adsorbate gas is assumed to be an ideal gas.
- The adsorbent bed is composed of uniform-size particles.

- Physical properties such as thermal conductivities, specific heat capacities and viscosity are not a function of temperature.
- The thermal resistance between the metal tube and the adsorbent bed is neglected.
- The pressure in condenser or evaporator during the entire adsorption cycle is assumed to be constant.

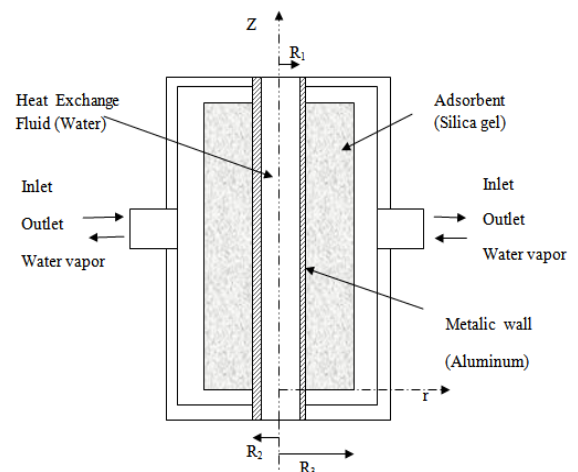


Fig. 1 Schematic diagram of the adsorbent bed

The energy conservation equation for the thermal-fluid is:

$$\rho_f c_{pf} \left[\frac{\partial T_f}{\partial t} + u_f \frac{\partial T_f}{\partial z} \right] = \lambda_f \left(\frac{\partial^2 T_f}{\partial r^2} + \frac{1}{r} \frac{\partial T_f}{\partial r} \right) + \lambda_f \left(\frac{\partial^2 T_f}{\partial z^2} \right) \quad (1)$$

The energy conservation equation for the metal tube is given by:

$$\rho_m c_{pm} \frac{\partial T_m}{\partial t} = \lambda_m \left(\frac{\partial^2 T_m}{\partial r^2} + \frac{1}{r} \frac{\partial T_m}{\partial r} \right) + \lambda_m \left(\frac{\partial^2 T_m}{\partial z^2} \right) \quad (2)$$

The energy conservation equation for the solid phase of the adsorbent can be written as:

$$\rho_s (1 - \varepsilon_t) (c_{ps} + X c_{pw}) \frac{\partial T_s}{\partial t} = \lambda_{s-e} \left(\frac{\partial^2 T_s}{\partial r^2} + \frac{1}{r} \frac{\partial T_s}{\partial r} \right) + \lambda_{s-e} \left(\frac{\partial^2 T_s}{\partial z^2} \right) + (1 - \varepsilon_a) \rho_s \Delta H \frac{\partial X}{\partial t} - a_v h_g (T_s - T_g) \quad (3)$$

The energy conservation equation for the gas phase can be given by:

$$\begin{aligned} c_{pg} \left[\rho_g \left(\frac{\partial T_g}{\partial t} + u \frac{\partial T_g}{\partial r} + v \frac{\partial T_g}{\partial z} \right) + (1 - \varepsilon_a) \rho_s (T_s - T_g) \frac{\partial X}{\partial t} \right] \\ = \lambda_{g-e} \left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T_g}{\partial r} \right) + \left(\frac{\partial^2 T_g}{\partial z^2} \right) \right] + a_v h_g (T_s - T_g) \end{aligned} \quad (4)$$

The overall mass conservation in the adsorbent bed is given by:

$$\frac{\partial(\varepsilon_t \rho_g)}{\partial t} + \frac{1}{r} \frac{\partial(r u \rho_g)}{\partial r} + \frac{\partial(v \rho_g)}{\partial z} + (1 - \varepsilon_t) \rho_s \frac{\partial X}{\partial t} = 0 \quad (5)$$

The linear driving force (LDF) model is used for describing the adsorption rate (Sakoda and Suzuki, 1984):

$$\frac{\partial X}{\partial t} = k_m (X_\infty - X) \quad (6)$$

Where X_∞ is the adsorbed phase concentration in equilibrium and can be written as (Di *et al.*, 2007):

$$X_\infty = 0.346 \exp(5.6(T_s/T_{sat} - 1)^{1.6}) \quad (7)$$

k_m is the internal mass transfer coefficient given by:

$$k_m = 60. D_e / d_p^2 \quad (8)$$

D_e is the equivalent diffusivity in the adsorbent particles which is expressed as:

$$D_e = D_0 \exp(-E_a/RT_s) \quad (9)$$

The fluid–solid specific surface area for spherical particles is determined by:

$$a_v = \frac{6(1 - \varepsilon_t)}{d_p} \quad (10)$$

The interfacial heat transfer coefficient for the spherical particle is evaluated by:

$$h_g = \frac{\lambda_w}{d_p} (2 + 1.8 R_e^{0.33} P_r^{0.5}) \quad (11)$$

The effective thermal conductivity for the solid and gas phases can be defined as follows:

$$\lambda_{s-e} = (1 - \varepsilon_t) k_s \quad (12)$$

$$\lambda_{g-e} = \varepsilon_t k_g \quad (13)$$

The volume fraction of the gas phase is assumed to be equal to the total porosity, ε_t and may be evaluated using:

$$\varepsilon_t = \varepsilon_b + (1 - \varepsilon_b) \varepsilon_p \quad (14)$$

The velocities of adsorbate in the r and z directions inside the adsorbent bed are determined as follows:

$$u = -\frac{K}{\mu} \frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial P}{\partial r} \right) \quad (15)$$

$$v = -\frac{K}{\mu} \frac{\partial P}{\partial z} \quad (16)$$

Here K is the permeability which can be calculated by the following semi-empirical Blake-Kozeny equation:

$$K = \frac{\varepsilon_a^3 d^2}{150(1 - \varepsilon_a)^2} \quad (17)$$

The heat of adsorption as function of X can be defined by the following equations:

$$Q_{ads} = 3500 - 13400X \quad X \geq 0.05 \quad (18)$$

$$Q_{ads} = 2950 - 1400X \quad X < 0.05 \quad (19)$$

The chiller consists of two beds, a condenser and an evaporator. A schematic view of the chiller is presented in the Fig. 2 and the details of chiller operation are presented in the Ref. (Miyazaki *et al.*, 2009).

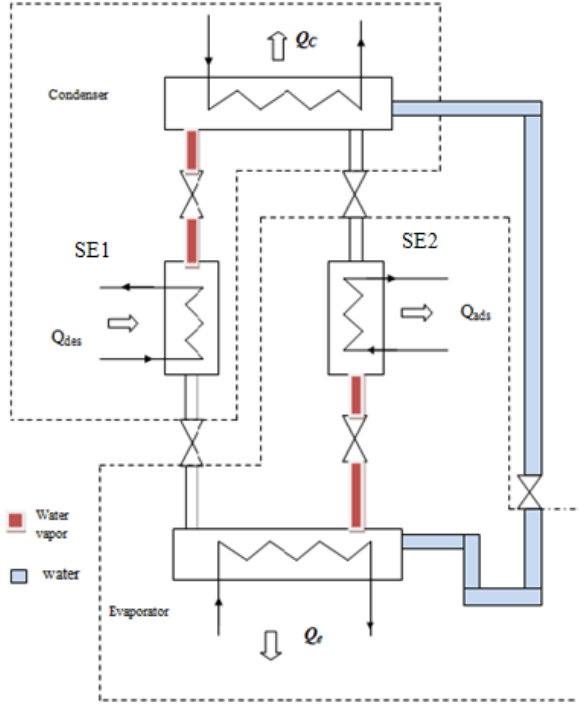


Fig. 2 Schematic diagram of two-bed adsorption refrigeration system.

The cooling power of system is evaluated by the following equation:

$$Q_f = 2\pi\rho_s L_s \iiint_{R_2,0}^{R_3, L t_s} X r dr dz dt \quad (20)$$

The specific cooling power is defined as follow:

$$SCP = \frac{2 \cdot Q_f}{m_s \cdot t_s} \quad (21)$$

where

$$m_s = 2\pi\rho_s \iint_{R_2,0}^{R_3, L} r dr dz \quad (22)$$

t_s is a half of cycle time. The performance coefficient of the system can be found by the following equation:

$$COP = \frac{Q_f}{Q_g} \quad (23)$$

where, Q_g is the amount of heat absorbed by the heat exchange fluid in the preheating and desorption-condensation modes and it is defined by:

$$Q_g = 2\pi U_f C_{pw} \iint_{R_2,0}^{R_3, t_s} (T_f^{Z=0} - T_f^{Z=L}) r dr dt \quad (24)$$

Initial and Boundary Conditions

The initial conditions are listed as below:

For adsorbent bed 01

$$T_s = T_g = T_m = T_a \quad T_f = T_{reg} \quad (25)$$

$$P = P_e \quad X = X_{max}$$

For adsorbent bed 02

$$T_s = T_g = T_m = T_{reg} \quad T_f = T_c \quad (26)$$

$$P = P_{co} \quad X = X_{min}$$

Boundary conditions are given as follows:

Adsorption phase

$$P(R_3, z, t) = P(r, 0, t) = P(r, L, t) = P_e \quad (27)$$

Desorption phase

$$P(R_3, z, t) = P(r, 0, t) = P(r, L, t) = P_{co} \quad (28)$$

Adsorption and desorption phase

$$\frac{\partial P}{\partial r}(R_2, z, t) = 0 \quad (29)$$

Heating and cooling phase

$$\begin{aligned} \frac{\partial P}{\partial r}(R_2, z, t) &= \frac{\partial P}{\partial r}(R_3, z, t) \\ &= \frac{\partial P}{\partial r}(r, 0, t) = \frac{\partial P}{\partial r}(r, L, t) = 0 \end{aligned} \quad (30)$$

$$\frac{\partial T_s}{\partial r}(R_3, z, t) = \frac{\partial T_s}{\partial r}(r, 0, t) = \frac{\partial T_s}{\partial r}(r, L, t) = 0 \quad (31)$$

$$\frac{\partial T_s}{\partial r}(R_2, z, t) = -h_{m_ads}(T_s - T_m) \quad (32)$$

$$\frac{\partial T_m}{\partial r}(R_2, z, t) = -h_{m_ads}(T_m - T_s) \quad (33)$$

$$\frac{\partial T_m}{\partial z}(r, 0, t) = \frac{\partial T_m}{\partial z}(r, L, t) = 0 \quad (34)$$

Heating and cooling phase

$$\frac{\partial T_f}{\partial r}(0, z, t) = \frac{\partial T_f}{\partial z}(r, 0, t) = \frac{\partial T_f}{\partial z}(r, L, t) = 0 \quad (35)$$

$$\frac{\partial T_f}{\partial r}(R_1, z, t) = -h_{f_m}(T_f - T_m) \quad (36)$$

$$\frac{\partial T_m}{\partial r}(R_1, z, t) = -h_{f-m}(T_m - T_f) \quad (37)$$

Here h_{f-m} is the convective heat transfer coefficient between the heat exchange fluid and the metallic wall and it is defined by the following equation

$$h_{f-m} = \frac{\lambda_w}{D} R_e^{0.8} P_r^{0.4} \quad (38)$$

NUMERICAL PROCEDURE

The nonlinear coupled governing partial differential equations under consideration are solved numerically using the finite difference technique. The central differencing, first order upwind scheme, and forward differencing are used to discretize the second order spatial derivatives, convective, and unsteady terms,

respectively. The resulting set of nonlinear algebraic equations is solved iteratively by the combination of the alternating direction implicit (ADI) method, the Newton–Raphson iteration scheme and a block tridiagonal matrix solver algorithm (Thomas algorithm).

The influence of the number of grid points and time steps on the solid phase temperature at nearly thermal equilibrium case and various locations in the computational domain is shown in Table 2. It can be seen from Table 2 that the difference between the results obtained for three different grid sizes (40×10), (15×10) and (30×20) and time steps (0.02, 0.04, 0.1 and 1 s) are quite small. Therefore, the number of grid points and the time step are chosen to be (30×20) and 0.02s, respectively to ensure the reliability of the numerical computations. The convergence criterion used in the simulation program is 10^{-6} .

Table 2. The effect of grid size and time step on the solid phase temperature

$R(m), z(m)$	$\Delta t=0.02$ sec			$\Delta t=0.04$ sec		
	40×10	30×20	15×10	40×10	30×20	15×10
0.023, 0	327.8206	327.8081	327.6185	327.8202	327.8584	327.6688
0.031, 0.175	338.6148	338.5191	338.3353	338.5090	338.5254	338.3430
0.036, 0.35	339.1587	339.1487	339.1432	339.1587	338.1505	339.1419
$R(m), z(m)$	$\Delta t=0.1$ sec			$\Delta t=1$ sec		
	40×10	30×20	15×10	40×10	30×20	15×10
0.023, 0	327.8215	327.8465	327.7017	327.8313	327.8584	327.6688
0.031, 0.175	338.5124	338.5229	338.4242	338.5651	338.5254	338.3430
0.036, 0.35	339.1587	339.1514	339.1446	339.1591	338.1505	339.1419

RESULTS AND DISCUSSIONS

A computer simulation program based on the numerical procedure above is written in MatLab to perform the parametric investigation. The base parameters used in the simulations are listed in Table 3. The effect of adsorbent bed thickness, velocity of the heat exchange fluid, adsorbent bed length and adsorbent particle diameter on the distributions of the solid phase temperature, adsorbate concentration and pressure during the adsorption process as well as the performance of the system are discussed below.

Adsorbent Bed Thickness

The adsorbent bed thickness is an important design parameter that has great influence on the performance of an adsorbent bed. Figs. 3 and 4 depict the effect of bed thickness on the COP and SCP of the adsorption cooling system. It can be seen that the COP slightly increases with an increase in adsorbent bed thickness. On the contrary, the SCP strongly reduces with an increase in adsorbent bed thickness. The reason behind these results is the bad heat and mass transfer conditions within the adsorbent bed. The similar influence of adsorbent bed thickness on the system performance is found by Leong and Liu (2006).

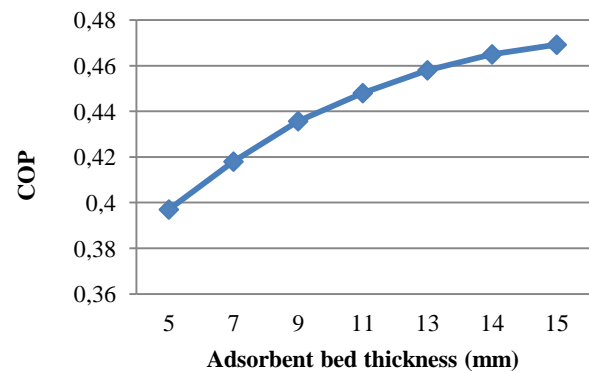


Figure 3. Variation of COP with adsorbent bed thickness

Figs. 5, 6 and 7 show the distributions of solid phase temperature, adsorbate concentration and the pressure in the adsorbent bed for adsorbent bed thicknesses of R= 5, 11 and 15 mm and at times of $t= 20, 200$ and 400 sec after beginning of adsorption phase. It can be seen that an increase in the adsorbent bed thickness leads to significant variation on the distributions of solid phase temperature and adsorbate concentration but not pressure distribution. The reason behind these findings is that the thermal resistances through the adsorbent bed become considerable when the adsorbent bed thickness is increased. Therefore,

increasing adsorbent bed thickness results in an increase in the cycle time. The adsorbent bed thickness should be kept as smaller as possible to reduce the cycle time and as a result of this, an enhancement on the specific cooling power can be achieved.

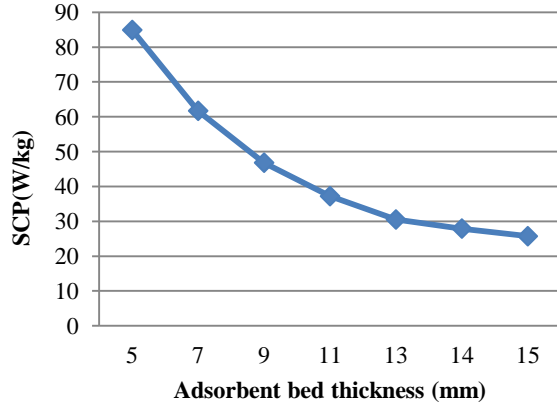


Figure 4. Variation of SCP with adsorbent bed thickness.

Table 3. The base parameters used in the simulations.

Parameter	Value	Unit
C_{pg}	1800	J/kgK
C_{ps}	924	J/kgK
D_p	$1.4e^{-3}$	m
D_o	$2.54e^{-4}$	m^2/s
E_a	$4.2e^4$	J/mol
k_g	0.024	W/mK
k_s	0.198	W/mK
L	0.35	m
P_{co}	4.246	kPa
Pe	1.228	kPa
R_3	0.036	m
R_2	0.021	m
R_1	0.02	m
T_{con}	40	$^{\circ}C$
T_{reg}	90	$^{\circ}C$
ϵ_b	0.37	
ϵ_p	0.64	
μ_g	$1.5e^{-5}$	kg/ms
ρ_s	2027	kg/m^3
ρ_m	2700	kg/m^3
C_{pm}	900	J/kgK
k_m	237	W/mK
ρ_f	1000	kg/m^3
C_{pf}	4180	J/kgK
k_f	0.6	W/mK
U_f	0.6	m/s

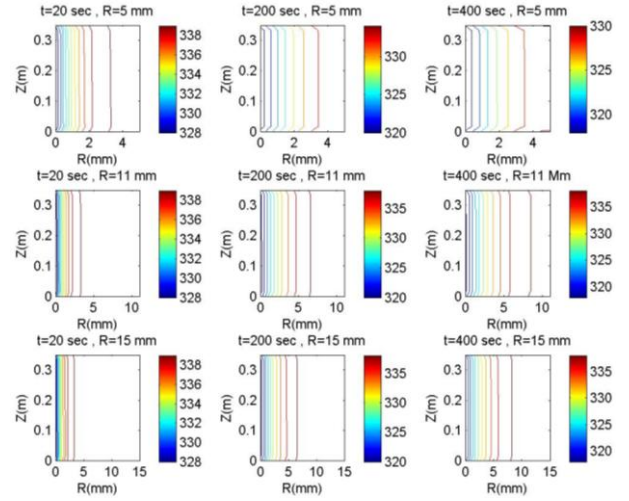


Figure 5. The influence of adsorbent bed thickness on the distribution of solid phase temperature.

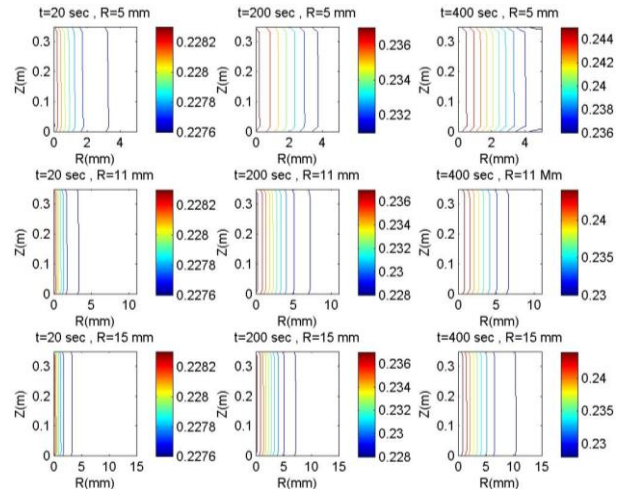


Figure 6. The influence of adsorbent bed thickness on the distribution of adsorbate concentration in the adsorbent bed.

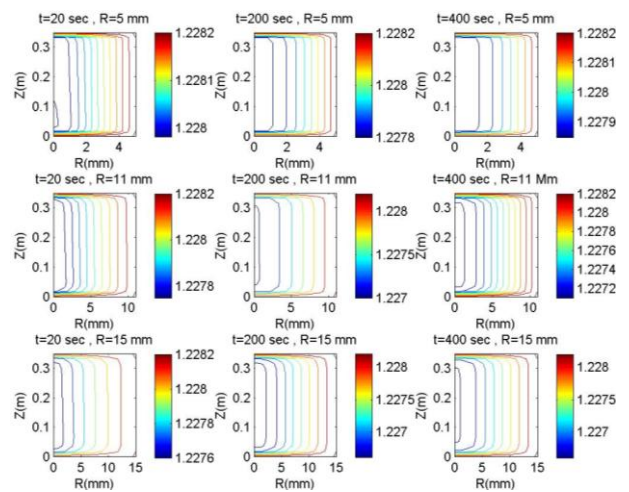


Figure 7. The influence of adsorbent bed thickness on the distribution of pressure.

The Velocity of Heat Exchange Fluid

The effect of the velocity of heat exchange fluid on the performance coefficients are shown in Figs. 8 and 9. The performance coefficients slightly increase with the increasing value of velocity of heat exchange fluid up to 0.3 m/s and beyond that value its effect on the performance coefficients is negligible. The effect of the velocity of heat exchange fluid such as $U_f=0.1, 0.3$ and 0.6 m/s on the distributions of solid phase temperature, adsorbate concentration and the pressure in the adsorbent bed for the time $t= 20, 100$ and 400 sec are shown in Figs. 10, 11 and 12 respectively.

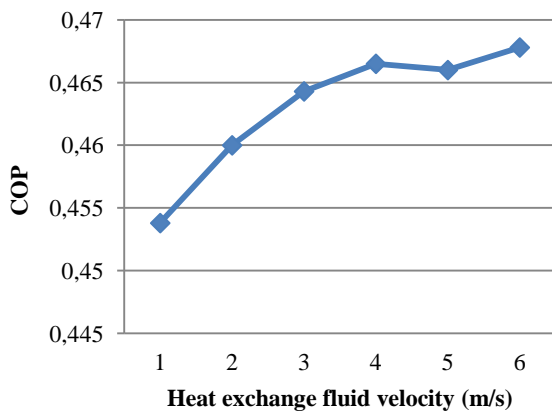


Figure 8. The variation of COP with heat exchange fluid velocity.

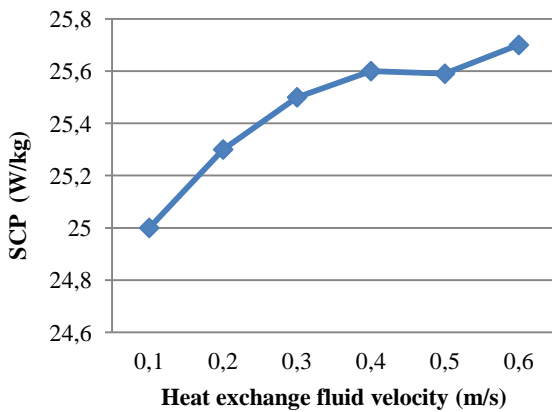


Figure 9. The variation of SCP with heat exchange fluid velocity.

The distributions of solid phase temperature, adsorbate concentration and the pressure in the adsorbent bed show a strong dependence on the variation of velocity of heat exchange fluid at the beginning of the adsorption phase. This is due to fact that the temperature gradient in the axial direction for low value of velocity of heat exchange fluid causes to heat transfer between heat exchange fluid and metallic wall and this temperature gradient becomes insignificant with the time progress. In other words, the cycle time decreases with the increasing value of velocity of heat exchange fluid. The optimal velocity of the heat exchange fluid should be in a range of 0.1–0.3 m/s.

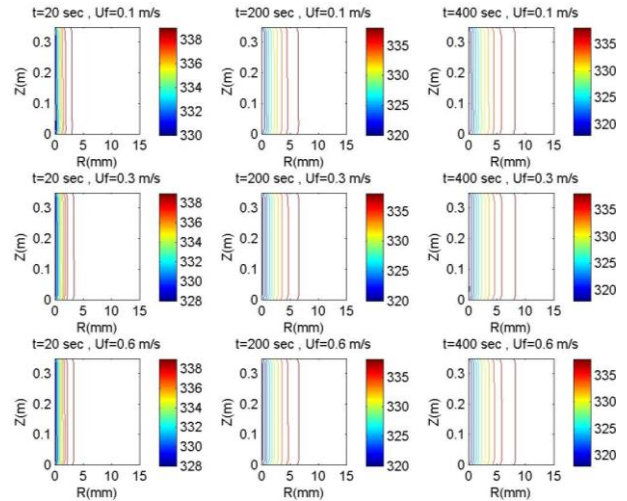


Figure 10. The influence of heat exchange fluid velocity on the distribution of solid phase temperature.

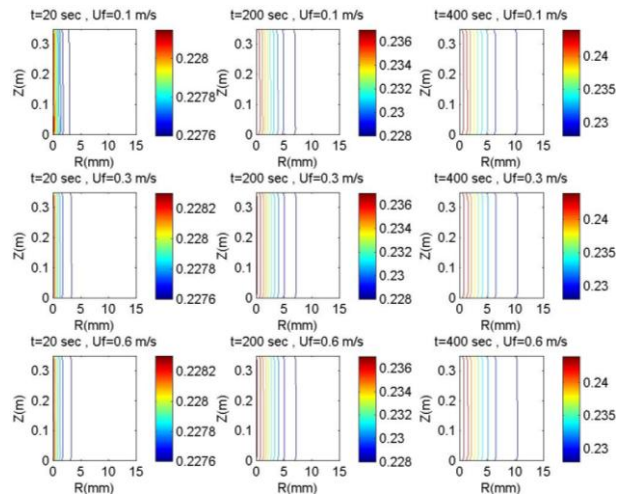


Figure 11. The influence of heat exchange fluid velocity on the distribution of adsorbate concentration in the adsorbent bed.

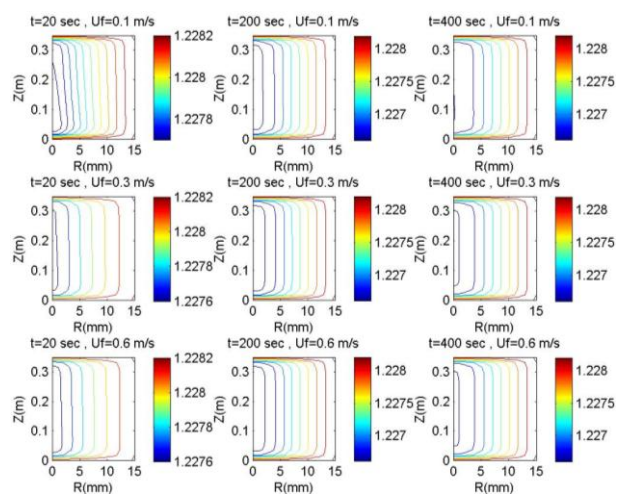


Figure 12. The influence of heat exchange fluid velocity on the distribution of pressure.

Length of Adsorbent Bed

The variations of performance coefficients with adsorbent bed length are shown in Figs. 13 and 14. It is clear that the coefficient of performance and specific cooling power increase only slightly with the increasing value of the adsorbent bed length. The effect of adsorbent bed lengths of 0.35, 0.65 and 0.85 m on the distributions of solid phase temperature, adsorbate concentration and the pressure in the adsorbent bed at time $t=20, 100$ and 400 sec are presented in Figs. 15, 16 and 17, respectively.

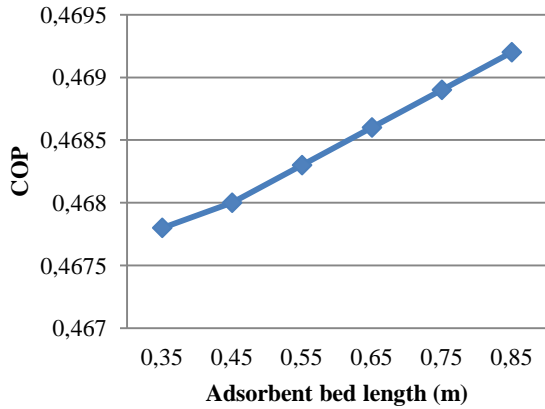


Figure 13. The variation of COP with the length of adsorbent bed.

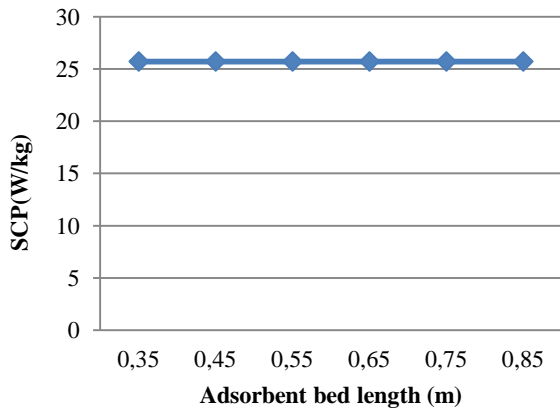


Figure 14. The variation of SCP with the length of adsorbent bed.

It is clear that the influence of adsorbent bed length on distribution of solid phase temperature, pressure and adsorbate concentration is slightly affected with the increasing of the length of adsorbent bed and this finding is also expressed by Leong and Liu (2006). Hence, the choice of adsorbent bed length depends on the cooling power wishes to product.

Adsorbent Particle Diameter

The effect of particle diameter on the performance coefficients is shown in Figs. 18 and 19. It can be seen that the coefficient of performance and specific cooling power increases with the decreasing value of adsorbent particle diameter.

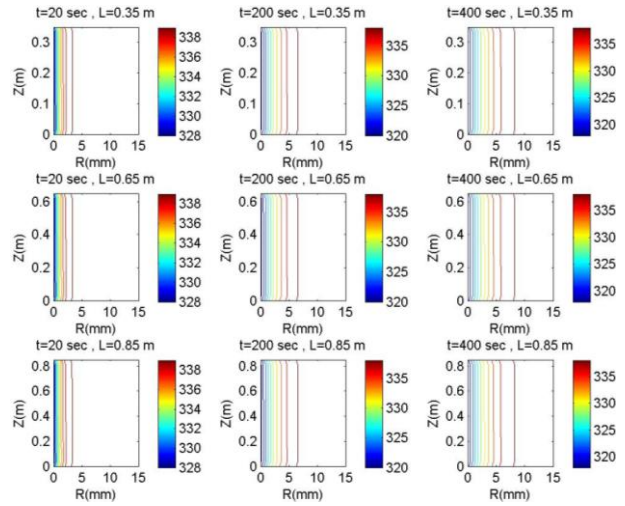


Figure 15. The influence of adsorbent bed length on the distribution of solid phase temperature.

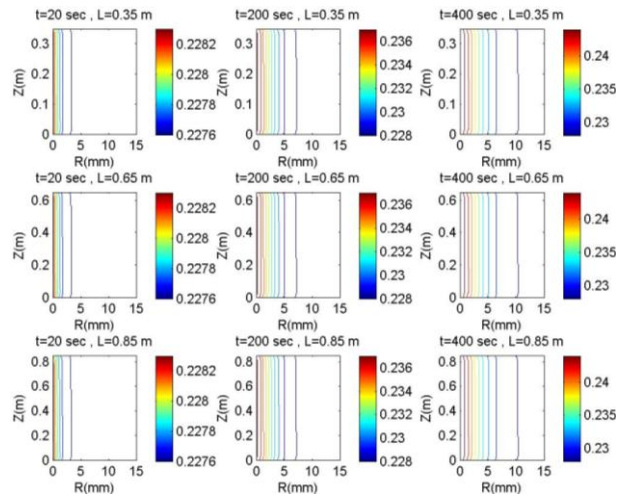


Figure 16. The influence of adsorbent bed length on the distribution of adsorbate concentration in the adsorbent bed.

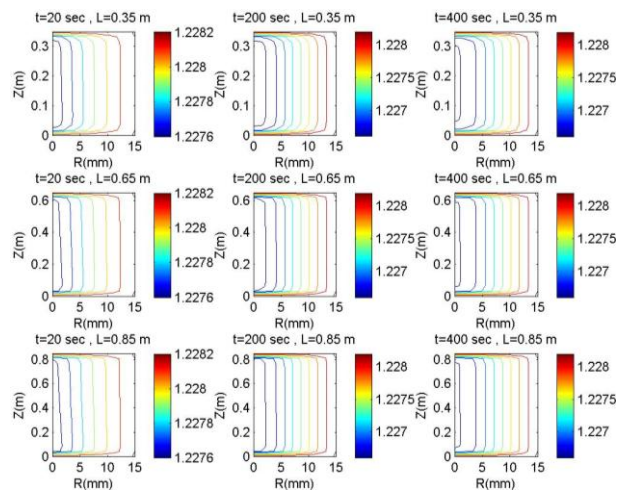


Figure 17. The influence of adsorbent bed length on the distribution of pressure.

The effect of adsorbent particle diameters of 1.2, 1.8 and 2.2 mm on the distributions of solid phase temperature, adsorbate concentration and the pressure in the adsorbent bed at time $t=20, 100$ and 400 sec are shown in Figs. 20, 21 and 22. The distributions of adsorbate concentration, the solid phase temperature and the pressure in the adsorbent bed are strongly affected by the variation of particle diameter.

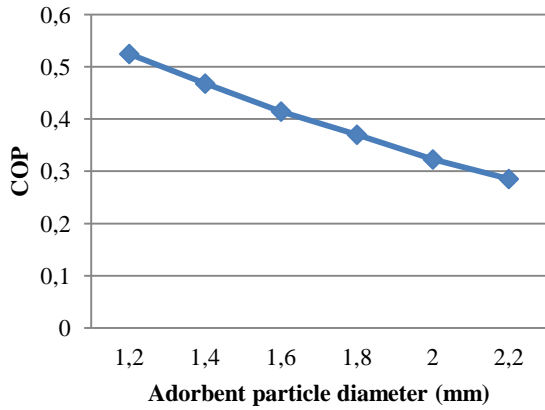


Figure 18. The variation of COP with adsorbent particle diameter.

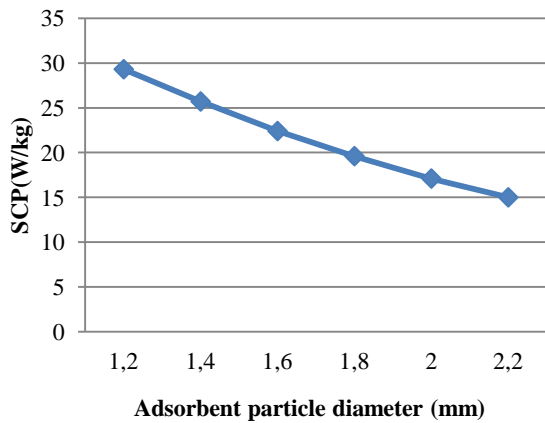


Figure 19. The variation of SCP with adsorbent particle diameter.

It is clear from Figs. 20, 21 and 22 that the adsorbate concentration in the adsorbent bed and the pressure decrease with increasing particle diameter and the solid phase temperature increase with an increase in the particle diameter. The reason behind these results that the specific area of adsorbent increases with decrease of particle diameter and hence, the particle diameter should be kept as smaller as possible.

CONCLUSIONS

A parametric study on the performance of adsorption chiller with two adsorbent beds based on the silica gel/water working pair and the distributions of solid phase temperature, adsorbate concentration and pressure inside the adsorbent bed is carried out by using a two dimensional local thermal non-equilibrium model.

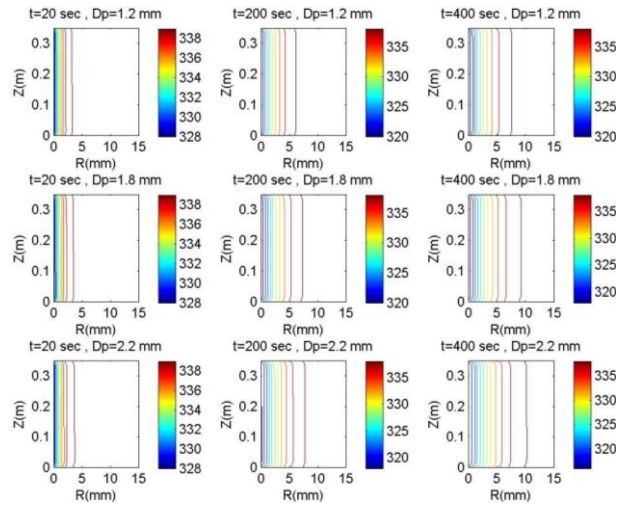


Figure 20. The influence of adsorbent particle diameter on the distribution of solid phase temperature.

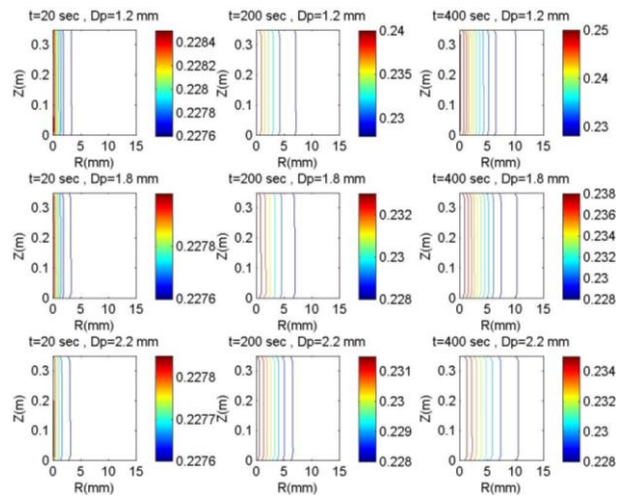


Figure 21. The influence of adsorbent particle diameter on the distribution of adsorbate concentration in the adsorbent bed.

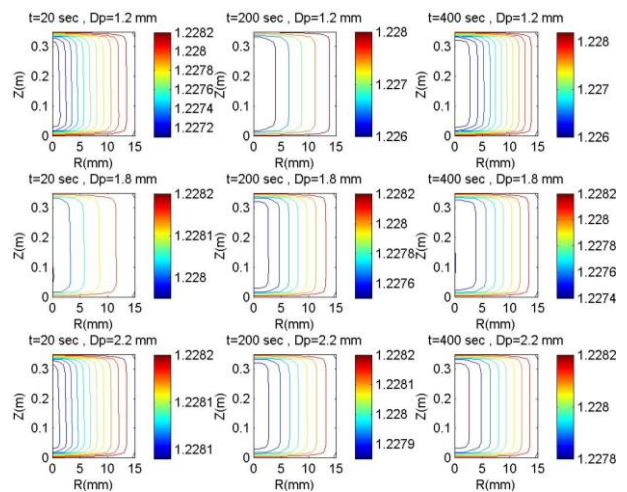


Figure 22. The influence of adsorbent particle diameter on the distribution of pressure.

The following conclusions from the present study are drawn:

- The COP of system is positively affected by the increasing value of the adsorbent bed thickness but the same effect on the SCP is not observed.
- The distributions of solid phase temperature and the adsorbate concentration strongly depend on the adsorbent bed thickness but not pressure.
- The velocity of heat exchange fluid which is greater than 0.3 m/s does not lead to any significant variations on the specific cooling power and coefficient of performance. The optimal velocity of heat exchange fluid needs to be within a range of 0.1-0.3 m/s
- The distributions of solid phase temperature, adsorbate concentration and the pressure show a strong dependence on the variation of velocity of heat exchange fluid especially at beginning of adsorption phase.
- The influence of adsorbent bed length on the coefficient of performance and specific cooling power of system is negligibly small. The adsorbent bed length depends on the cooling power wish to product.
- The distribution of solid phase temperature, adsorbate concentration and the pressure does not depend on the length of adsorbent bed.
- A decrease in the adsorbent particle diameter results an increase in the performance coefficient and specific cooling power of the system.
- The distributions of solid phase temperature, adsorbate concentration and the pressure depend strongly on the particle diameter of adsorbent.

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