Numerical Simulation of Heat And Mass Transfer in Adsorbent Bed with Rectangular Fins

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Abstract
A transient two-dimensional numerical model of coupled heat and mass transfers in adsorbent bed of a silica gel/water adsorption chiller system is presented for a plat bed with rectangular fins through the finite element method (FEM) simulations. The bed transient transport behaviors are investigated for all four cycle phases. In the model, a linear driving force equation is used to account for the intra-particle mass transfer resistance. Meanwhile, the refrigerant vapor superficial velocity describing the adsorbate flows is calculated by Darcy's law. The effects of bed configurations including the space of fins and the height of fins on SCP and COP are investigated. The results show that bed transfer processes are enhanced through extended surfaces of fins shape. Furthermore, at a given fins height, SCP increases with the increase in fins number, while at a fixed fins number it decreases as the fins height increases.

Keywords: Adsorbent bed Numerical simulation heat and mass transfers COP SCP

1. Introduction

Adsorption cooling has received much attention in recent years due to its low environmental pollution. This technology has no ozone layers depletion potential and global warming potential; moreover it can make use of low temperature thermal source to realize refrigeration directly. Adsorption cooling has many possible uses in the field of refrigeration and air conditioning. However, the values of the specific cooling power (SCP) and coefficient of performance (COP) are much lower than that expected for vapor compression cycles since the poor heat transfer of the adsorbent bed.

One way to improve the heat transfer of the adsorbent bed is to enhance the thermal conductivity of the adsorbent. For instance, Guilleminot [1]added the Metal or graphite powders even foam metal with good thermal conductivity into adsorbent. Gacciola G el al.[2]used PTFE as the agglomerant to solidify the activated carbon powders, then the effective thermal conductivity of adsorbent were improved to the range from 0.13 to 0.20 W/(m·K)

Cause the main thermal resistance focuses on the side of adsorbent, so increasing the interfacial area for metal-adsorbent contact is the most economic and efficient method to solve this problem. For instance, Guilleminot and Meunier[3] developed a two-dimensional numerical model for an adsorption chiller which equipped annular fins, through the model considered the distribution of temperature, but all the mass transfer resistances of the bed were neglected. Chua et al.[4] numerically analyzed a two-bed, silica gel/water adsorption chiller with annular fins. However the interparticle resistances was not considered.

Gong et al[5]. established an adsorption chiller with annular fins filled with the composite adsorbent that mixing lithium chloride and silica gel. A mathematical model was also developed to analyze the heat and mass transfer in the bed, even though the intra particle mass transfer resistance was considered, the pressure assumed to be constant throughout the bed.

Ilis et al. [6]presented a transient two-dimensional model to represent the heat and mass transfer in an annular adsorbent bed assisted with axial fins. Through ignoring the distribution of the pressure in the bed, the model was simplified by neglecting the convective effects of refrigerant vapor throughout the bed. meanwhile, the governing equations were non-dimensionalised to reduce the mumber of governing parameters, and the bed performance was examined based on the four non-dimensional groups

Yang [7]developed a model on heat and mass transfer in an adsorbent bed with annular fins by using a lumped parameter method to discuss temperature variation and adsorbate distribution. San modeled a multi-bed adsorption heat pump with a plate-fins heat exchangers to calculate the optimum cycle time corresponding to the maximum specific cooling power.

Zhang and Wang[8] developed a three-dimensional model for investigating the pressure and gas flow through the bed using the LDF model for intraparticle mass transport. The relationships of coefficient of performance (COP), specific cooling power (SCP) and longitudinal fins were examined. They found that both COP and

SCP are improved by using longitudinal fins. Li et al. numerical investigated the optimum fins spacing and height. It is found that using the optimized fins spacing and height in the adsorber can produce 2 times cold heat output as much as the previous system.

In fact, there are some detailed researches has been investigated, such as Hamid Niazamnd [9]et al. performed a transient two-dimensional numerical model of combined heat and mass transfer in the adsorbent bed of a silica gel/water with annular fins. Temperature, Velocity of refrigeration vapor and concentration field inside the bed were examined in detail. However, both Li Zhi Zhang's and Hamid Niazmnd's researches, The governing equations contained the heat transfer coefficient at the interface between the metal tube and thermal fluid calculating by the correlation from Dittus and Boelter, but the correlation is just viable in the circular tube, and it is possible that there is no correlation in the existing literatures for what tube in your research with special structure. Actually, it can be detailedly and correctly calculated by the method of conjugate heat transfer.

This literature review shows that no study has been performed by considering the conjugate heat transfer at the interface between the metal tube and thermal fluid

In this paper, a two-dimensional symmetric non-equilibrium mode which takes into account both the internal and the external mass transfer resistance in the adsorbent of an adsorption cooling system.is proposed. The influences of the bed configurations such as the spaces, height and thickness of fins on the system performance are significantly investigated through the finite element method (FEM) simulations.

2. Mathematical model

A diagrammatic drawing of heat exchanger with fins as an adsorbent bed is shown in Fig.1.heating and cooling required for the adsorbent are supplied by thermal fluid through a metal tube. The spaces between fins are filled with adsorbent. Furthermore, the parameter values and operating conditions used in this model is shown in Table.1.

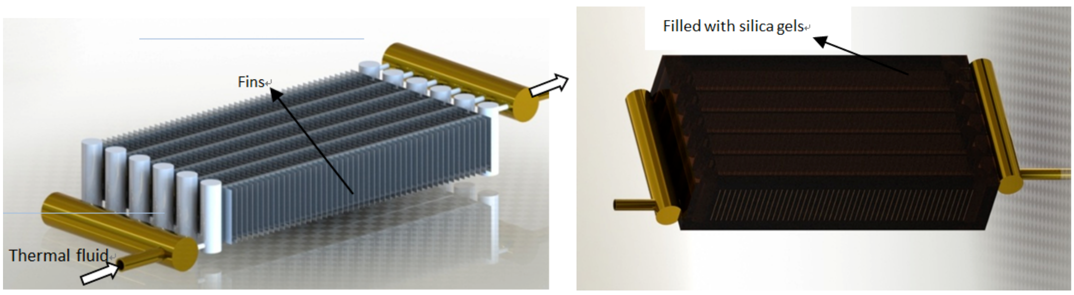


Fig.1 adsorbent bed

Table1. The parameter values and operating conditions

Parameter	Symbol	Value
Length of adsorbent bed	L	120mm
Height of metal tube	H_c	12mm
Fins thickness	δ	1mm
Thermal fluid average velocity	u_f	0.2m/s
Initial pressure	P_0	1000Pa
Initial temperature	T_0	30℃
Inlet thermal fluid temperature during heating	$T_{f,hot}$	70℃
Inlet thermal fluid temperature during cooling	$T_{f,cool}$	30℃
Density of adsorbent	ρ_s	2129kg/m^3
Specific heat of adsorbent	C_{ps}	921 J/(kgK)
Density of gaseous adsorbate	ρ_g	2129 kg/m^3
Specific heat of gaseous adsorbate	C_{pg}	1907 J/(kgK)
Viscosity of gaseous adsorbate	μ_g	10.2E-6 N s/m^2
Heat of adsorption	ΔH	2.37E6 J/kg
Particle diameter	d_p	20μm
Porosity of the particle	θ_p	0.46
Porosity of bed	θ_b	0.36
Equivalent thermal conductivity of the bed	λ_{eq}	0.198 W/(mK)
condensing pressure	P_c	3169Pa
evaporating pressure	P_e	1227Pa

2.1 Assumption

To calculating the heat and mass transfer of adsorbent bed, some assumptions are made:

1. The adsorbed phase and refrigerant vapor are considered as liquid and ideal gas respectively.
2.The particles in the adsorbent bed are standard spherical with uniform size, shape and porosity.
3. There are no heat loss through chamber walls.
4. The only variable thermo physical property of the adsorbent bed parameter is the density of adsorbate gas.
5. The thermal resistance between surface of fins and the adsorbent is neglected.

2.2 Governing equations

Based on above assumptions, the governing equations are as follows:
The conservation of energy for the thermal fluid can be written as:

$$(\rho C_p)_f \frac{\partial T_f}{\partial t} + \rho_f C_{pf} \boldsymbol{U} \cdot \nabla T_f = k_f \nabla T_f$$

The conservation of energy for the metal tube and fins is:

$$(\rho C_p)_m \frac{\partial T_m}{\partial t} = k_m \nabla T_m$$

Energy balance for adsorbent is:

$$(\rho C_p)_{eff} \frac{\partial T_b}{\partial t} + \rho_g C_{pg} \boldsymbol{U}_g \cdot \nabla T_b = k_{eff} \nabla T_b + \rho_b \Delta H \frac{\partial W}{\partial t}$$

where$(\rho C_p)_{eff}$ is the total heat capacity:

$$(\rho C_p)_{eff} = \theta_b \rho_b C_{b,p} + (1-\theta_b)\rho C_p$$

$$k_{eff} = \theta_b k_b + (1-\theta_b) k_p$$

and the intra-particle mass transfer resistance is considered using the linear dirving force model(Sakoda and Suzuki)$\frac{dW}{dt} = 15 D_{so} \exp\left(-\frac{E_a}{R_u T_b}\right) / (R_p^2 (W^* - W))$

in which $D_{so}$is a pre-exponent constant and E_a is the activation energy of surface diffusion with the value if 2.54E-4m^2s^{-1} and 4.2E4 J mol^{-1} respectively.
$W^*$is the equilibrium uptake at temperature T_b and pressure P.

$$W^* = \frac{1.6E12 \cdot C_w}{[1 + (2E - 12 \cdot C_w)^{1.1}]^{1/1.1}}$$

$$C_w = P exp(\frac{\Delta H}{RT_b})$$

where ΔH is the isosteric heat adsorption.
The refrigerant vapor superficial velocity is calculated by Darcy's law:

$$\boldsymbol{U}_g = -\frac{K_{app}}{\mu} \nabla \boldsymbol{P}$$

where the $K_{app}$is calculated by the following equations(Bird et al., 1960;Lee and Thodos,1983;Ruthven,1984):

$$k_{app} = k_d + \frac{\theta_p \mu}{\tau P} D_{eq}$$

$$k_d = \frac{\theta_b^3 d_p^2}{150(1-\theta_b)^2}$$

$$D_{eq} = \left(\frac{1}{0.02628 \sqrt{\frac{T_b^3/M}{P\sigma^2\Omega}}} + \frac{1}{18.5 d_{pore}\sqrt{T_b/M}}\right)^{-1}$$

$$d_{pore} = 0.6166 d_p$$

$$\tau = \theta_b^{-0.4}$$

where $\sigma = 2.641$ and $\Omega = 2.236$.

$$P = \rho_g R T_b / M$$

2.3 Initial and boundary conditions

$T_f\big|_{z=0} = T_{fhot}$ during the heating

$T_f\big|_{z=0} = T_{fcool}$ during the cooling

$P_y\big|_{y=h} = P_e$ when connected to the evaporator

$P_y\big|_{y=h} = P_c$ when connected to the condenser

$P_y\big|_{y=h} = P_b$ when closed

P_b is calculated by the ideal gas state equation

2.4 Performances

The heat that the adsorber absorbed during a cycle is

$$Q_{in} = \int_0^{t2} (\rho u A C_p)_f (T_{fin} - T_{fout}) dt$$

The mass flow rate of water vapor is calculated by

$$\dot{m}_w = \oint\!\!\oint_{\Sigma} \rho_g \boldsymbol{U}_g dS$$

Which S is area.

The heat extracted from the evaporator is

$$Q_e = \int_{t3}^{t4} m_w [L_w - C_{p_f}(T_c - T_e)] dt$$

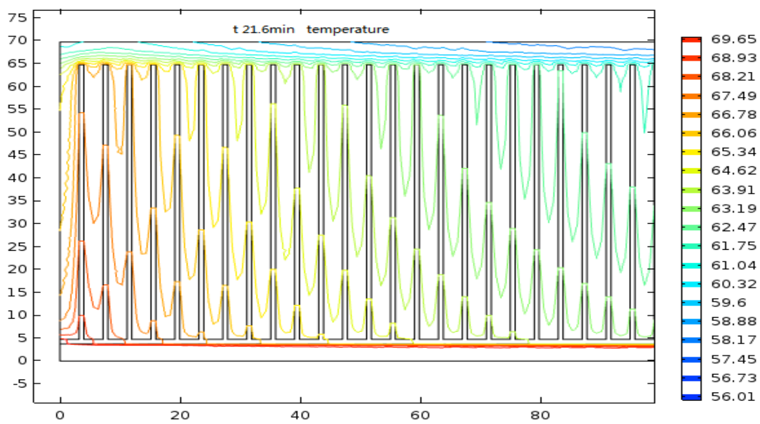
So the coefficient of performance

$$\mathrm{COP} = \frac{Q_e}{Q_{in}}$$

The specific cooling power

$$\mathrm{SCP} = \frac{Q_e}{m_b t_{cycle}}$$

3. Results and discussion



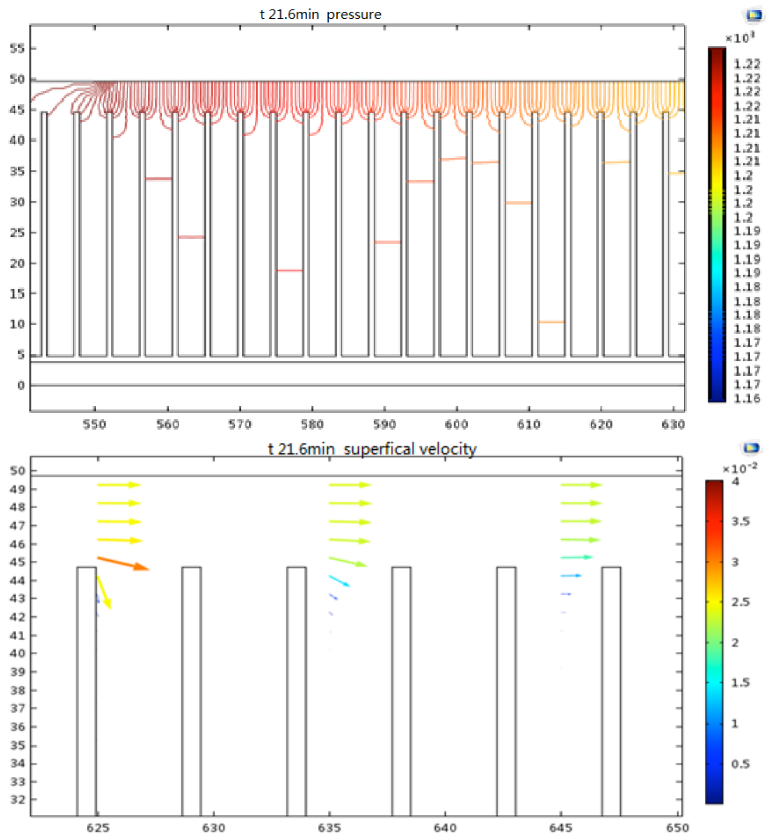
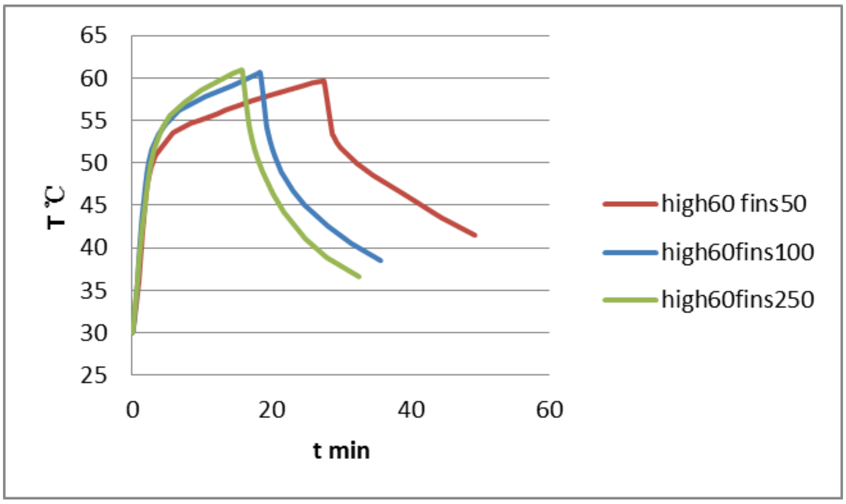


Fig.2 Flow field and pressure distribution of adsorbent bed

The heat and mass transfer in adsorbent bed is calculated by the software COMSOL Multiphysics. The effects of fin height and spacing on the bed performance are discussed in this study as follows.



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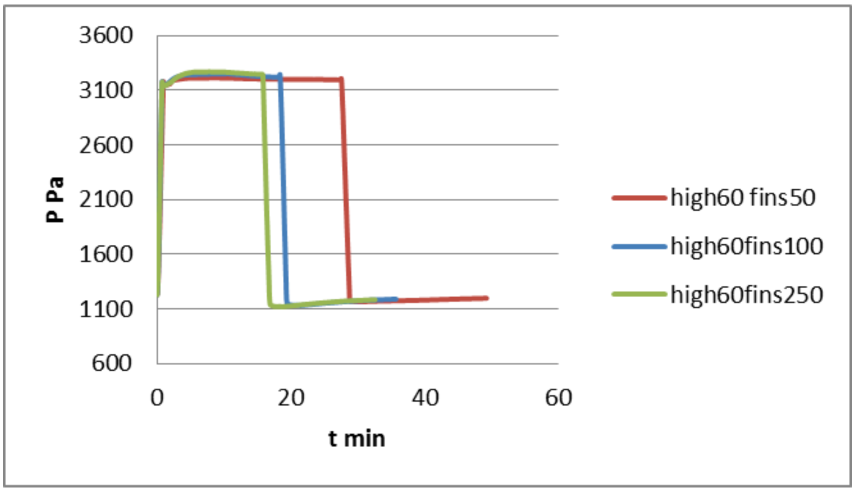

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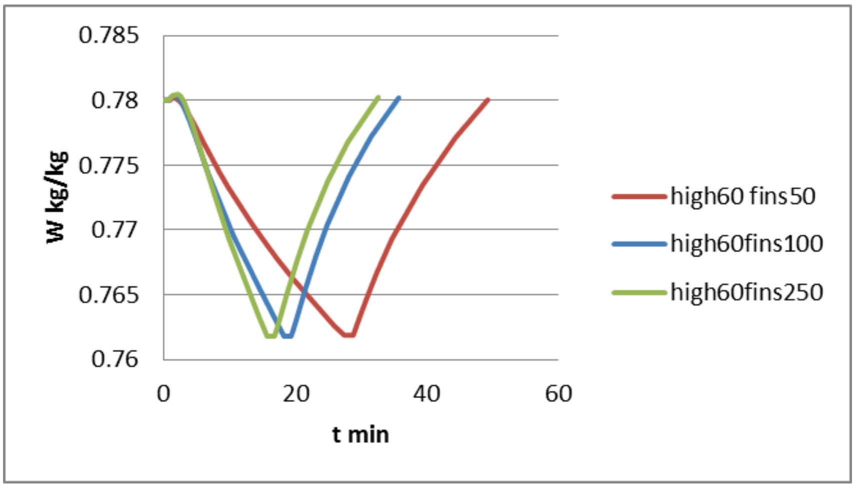

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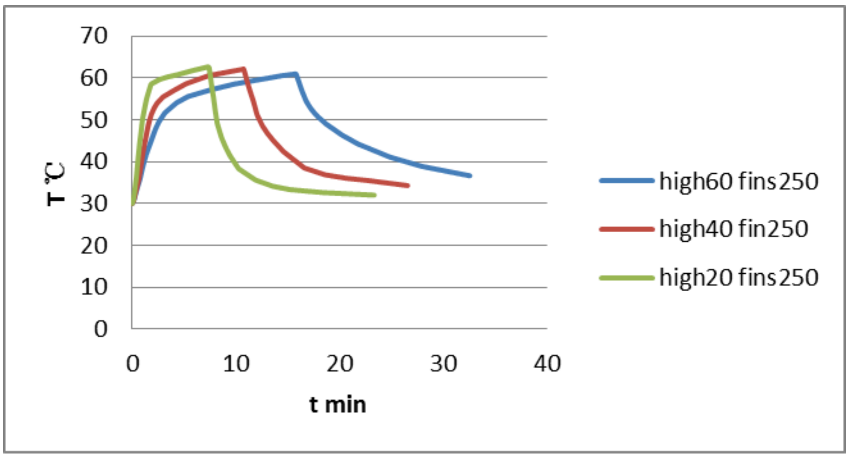
Fig.3 Comparing the time variations of the averaged bed for different numbers of fins (a) temperature, (b) pressure for bed and (c) average adsorbed amount



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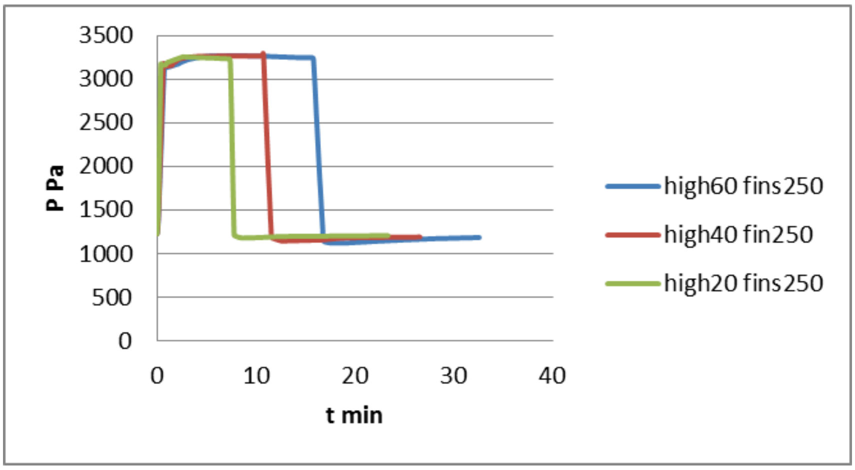

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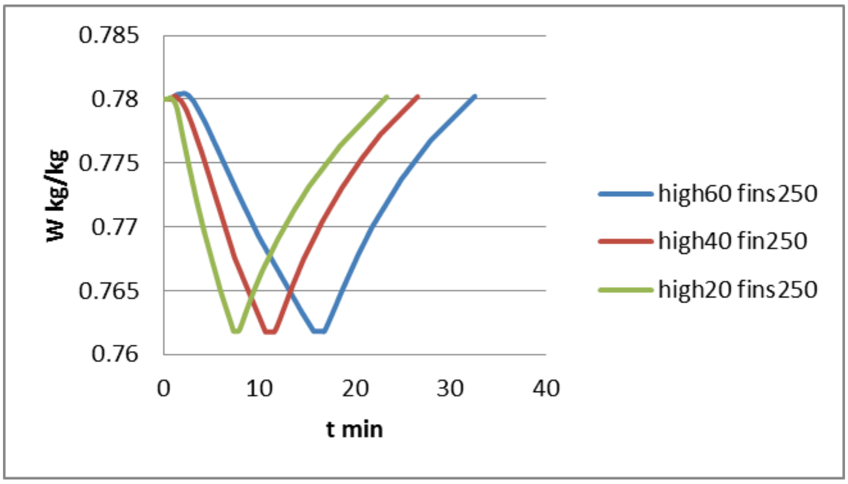
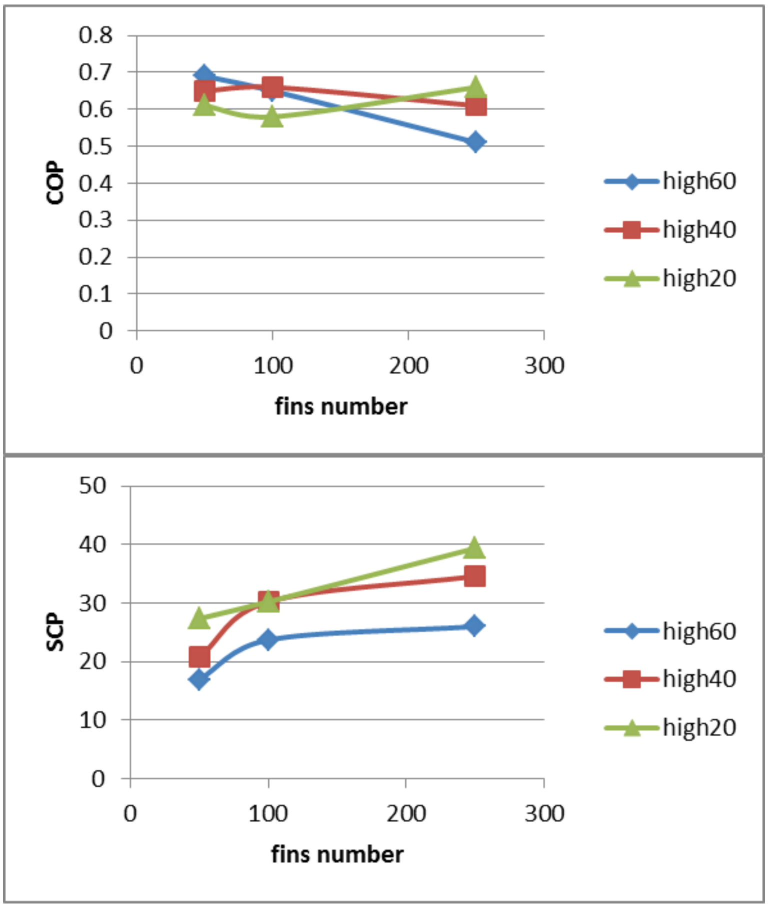


c

Fig.4 Comparing the time variations of the averaged bed for different height of fins (a) temperature, (b) pressure for bed and (c) average adsorbed amount

Form Fig.3 and Fig.4, it is clear that the cycle time is effected by fins geometry parameters acutely. The value of cycle time increase with the height of fins and decease with the number of fins. As we all known, the value of cycle time directly influents the SCP of chiller system. So in order to discuss the influence of geometry of fins on performance coefficients, it is necessary to demonstrate the COP and SCP as Fig.5 showed.



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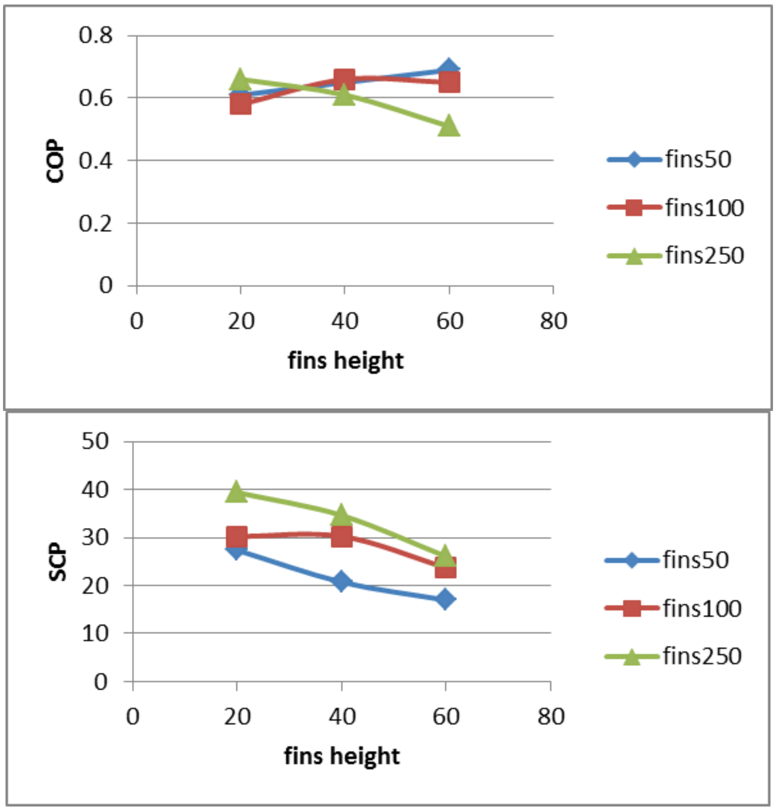

b

Fig.5 (a)variations of the COP and SCP with fins numbers with different fins height. (b) Variations of the COP and SCP with fins height with different fins numbers.

It is obvious that the SCP increase with the fins numbers but decrease with the fin height. This can be clear explained by the cycle time as Fig.3 and Fig.4. The COP decreases with the decrement of fins numbers except at the condition which fins height is 20mm. in addition, the COP increases with the fin height except at the condition which fins number is 250. Therefor there is an optimization point considering the balance between COP and SCP.

4. Conclusion

This paper presents a transient two dimensional modeling of a silica gel/water adsorption chiller employing fins to enhance transfer processes. The model can be employed to test the conditions with different geometry of fins such as height and spcing (fins numbers). The cycle time, SCP and COP is also calculated in this paper.

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