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Heat exchanger design effect on the system performance of silica gel adsorption refrigeration systems

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Abstract

This article presents a numerical investigation of the heat exchanger design effect on the performance of closed cycle, two-bed adsorption cooling systems with silica gel as adsorbent and water as refrigerant. It is well known that the shorter the cycle time, lower is the performance (cooling capacity and coefficient of performance). A long cycle time is responsible for lower cooling capacity. In this study, a non-dimensional switching frequency, which is inversely proportional to the cycle time, is defined and an optimum switching frequency is derived based on parametric analysis. The effect of other heat exchanger design parameters such as adsorbent number of transfer unit (*NTU*), bed Biot number (*Bi*), the heat exchanger aspect ratio (*Ar*) and the ratio of fluid channel radius to the adsorbent thickness (*Hr*), on the system performance has been investigated. The results show that the switching frequency ω, bed *NTU*, *Ar* and bed *Bi* have strong effects on the system performance. It is also seen that for a given set of design parameters, the system has an optimum switching frequency and the system performance will be declined seriously if the system is not operated at optimum switching frequency. The optimum switching frequency increases with the increase of *NTU*, *Hr* and with the decrease of *Bi* and *Ar*.

Keywords: Adsorption; Silica gel; Switching frequency; Heat exchanger design; System performance

1. Introduction

The use of air conditioning and refrigeration is increasing day by day for providing thermal comfort in industrial and residential areas. This technology requires energy consumption and is responsible for the emission of CO_2 and other green house gases such as CFCs, HCFCs, which are considered major ozone-depleting gases. Adsorption cooling systems are promising for providing a safe alternative to CFC-basis refrigeration devices.

From this context, adsorption refrigeration systems attain considerable attention as they can be driven either by waste heat sources or by renewable energy sources. From the 1970s, interest in solid-vapor adsorption systems was rekindled in view of their energy saving potential. Several heat-pumping and refrigeration applications have been studied using

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Nomenclature

A	heat transfer area
Ar	aspect ratio
Bi	Biot number
c	specific heat
COP	coefficient of performance
D_a	adsorbent thickness
D_f	fluid tube radius
D	$D_a + D_f$
h_f	heat transfer coefficient
h_m	mass transfer coefficient
Hr	heat exchanger thickness ratio
K	constant defined in Eq. (39)
k	thermal conductivity
L	length of a bed
L_v	latent heat of refrigerant
M	mass
$\dot{m}$	mass flow rate
n	constant defined in Eq. (39)
NSCC	non-dimensional specific cooling capacity
NTU	number of transfer unit
P	pressure
q	concentration
$\bar{q}$	non-dimensional concentration
q_e	concentration equilibrium
$\bar{q}_e$	non-dimensional concentration equilibrium
Q_{st}	isoteric heat of adsorption
SCC	specific cooling capacity
T	temperature
t	time
t_{hc}	switching time
x, y	coordinate axes
X, Y	non-dimensional coordinate axes

Greek symbols

α_{f-a}	fluid alpha number
α_{r-a}	refrigerant alpha number
α_{m-a}	inert material alpha number
β	beta number
θ	non-dimensional temperature
λ	lambda number
τ	non-dimensional time
ω	non-dimensional switching frequency

Subscripts

a	adsorber
b	bed
c	cool
con	condenser
eff	effective
eva	evaporator
f	fluid
h	hot
in	inlet
m	metal
max	maximum
out	outlet
r	refrigerant
s	saturation
v	vapor

various adsorbent and adsorbate pairs. Some examples are given in Table 1.

The performance analysis of adsorptive heat pump/cooling systems has been investigated by many researchers, and various methodologies have been proposed. Sakoda and Suzuki [17] proposed a transient model to analyze the influence of operating parameters on the system performance of a solar cooling unit. Saha et al. [14] investigated the influence of operating conditions on the performance of a two-bed silica gel-water adsorption refrigeration system by assuming a similar model. They showed that the system perform-

Table 1
List of literatures in adsorption heat pump/refrigeration area

Pair	Literature	Application area
Zeolite-water	Rothmyer et al. [1], Karagiorgas and Meuiner [2], Tchernev and Emersion [3]	Heat pump
	Guilleminot and Meunier [4]	Solar refrigeration
Zeolite-ammonia	Critoph and Turner [5], Shelton et al. [6,7]	Heat pump
Activated carnon-ammonia	Fuller et al. [8], Zheng et al. [9]	Heat pump
Activated carnon-methanol	Douss and Meunier [10,11], Critoph [12]	Heat pump
Silica gel-water	Boelman et al. [13], Saha et al. [14,15], Chua et al. [16]	Refrigeration
	Sakoda and Suzuki [17,18]	Solar cooling

ance could be improved by optimizing the operating parameters such as operating temperatures, water flow rate and cycle time. A transient dynamic model was developed by Chua et al. [16] to show the effect of operating parameters on the system performance of two-bed silica gel-water adsorption chillers. They also investigated the effect of cycle time and switching time on the system performance. They showed that the cooling capacity might be deteriorated if the cycle time is not optimized. But they did not show how the optimum cycle time could be determined for different configurations of adsorbent beds.

While the feasibility of improving systems performance of solid adsorption heat pumping/cooling systems has been studied, investigation of design parameters on the system performance is scarce. Haji and Worek [19] proposed a model considering only convection term for heat flow to analyze the effect of design and operating parameters on the system performance of a zeolite heat pump system. Fuller et al. [8] studied the effect of design and operating parameters on the performance of spiral type adsorbent reactors of two-bed activated carbon ammonia heat pump systems by considering one-dimensional heat flow. Zheng et al. [9] presented a one-dimensional model to show the effects of design and operating parameters on the systems performance of two-bed activated carbon-ammonia systems. They concluded that the heat exchanger design parameters played an influential role in improving the systems performance. Amar et al. [20] analyzed a two-dimensional model which also took into account the combined heat and mass transfers in the bed to investigate the effects of various operating parameters on the performance of a temperature wave regenerative heat pump. A three-dimensional model was investigated by Zhang and Wang [21] to study the effect of coupled heat and mass transfers in adsorbent beds on the performance of a waste heat adsorption cooling unit. They also studied the effect of reactor configuration on the performance.

The above-mentioned researches are very interesting and give a clear idea about the dynamic behavior of heat and mass transfers inside the adsorbent bed heat exchangers. In almost all of the above cases they used one-dimensional (in axial direction) heat equation for fluid side even though some used two- or three-dimensional heat equations for adsorbent beds. The fluid side is a vital part of the adsorbent bed heat exchangers. Therefore, it is desirable to investigate the design parameters of the fluid side on the system performance in more detail. However, very little literature has been found on the parametric study of the fluid sides. From this context, in the present investigation, two-dimensional heat equations are considered for both the fluid and adsorbent sides. This study analyzes a set of non-dimensional parameters, which present the different physical design and operating parameters of the system. Parametric study is conducted to show the effects of different non-dimensional parameters on the system performances.

The primary objective of this paper is to analyze the effect of heat exchanger design parameters on the system performance of a two-bed silica gel-water adsorption cooling unit. This paper also examines the effect of switching speed on the system performance. This paper will provide fundamental understandings of the silica gel-water adsorption systems and give useful guidelines regarding designs of adsorbent bed reactors.

2. Cycle description

The adsorption cooling unit analyzed in this paper consists of six major components: two adsorbers, a boiler, a cooling tower, a condenser and an evaporator plus four check valves and a reversible pump as shown in Fig. 1. The adsorbent is packed in the adsorber 1/adsorber 2 heat exchangers, which undergo alternate cooling and heating to allow refrigerant adsorption and desorption, as illustrated in Fig. 1. For description of the cycle, it is assumed that the adsorber 1 is initially to be cold at $T = T_c$, while the adsorber 2 is to be at T_h. In the beginning, hot water at T_h is forced to flow through adsorber 1, and cold water at T_c through adsorber 2. The hot water enters adsorber 1 and goes out to the cooler where the hot water cools down. At the same time, the cold water enters adsorber 2 and goes out to the boiler where the cold water is heated by the heat source. Therefore, the regeneration process begins at adsorber 1 and adsorption process begins at adsorber 2. In the beginning, all the valves are closed, and the pressures of adsorbers 1 and 2 are maintained at P_{eva} and P_{con} corresponding to vapor pressures at temperatures T_{eva} and T_{con}, respectively. When the pressure of adsorber 1 increases to P_{con} and that of adsorber 2 decreases to P_{eva}, the adsorbers 1 and 2 are connected to the condenser and the evaporator, respectively by opening valves V1 and V2, which causes the refrigerant (water) to condense at the condenser and evaporate at the evaporator. After finishing the first half cycle, that is, when the adsorber 1 is sufficiently heated and adsorber 2 sufficiently cooled, all the valves are shut down and the direction of heat transfer fluid is reversed by the reversible pump. In the second half cycle, the regeneration process begins at adsorber 2, and adsorption process begins at adsorber 1. Once the pressures of adsorbers 1 and 2 reach to P_{eva} and P_{con} respectively, then the valves V3 and V4 are opened. Thus, adsorption and desorption processes begin at adsorbers 1 and 2, respectively.

3. Mathematical modeling

A schematic of an adsorbent heat exchanger is shown in Fig. 2a. Side view of the main portion of the adsorbent heat exchanger is traced in Fig. 2b. The adsorbent heat exchanger is divided into two parts. One is the adsorbent side, which is filled with the adsorbent particles and the other is the heat transfer fluid side as illustrated in Fig. 2c. The role of the heat transfer fluid is either to cool down or to heat up the adsorbent particles, which causes the heat exchanger to adsorb refrigerant vapor from the evaporator or to desorb the refrigerant to the condenser.

3.1. Conservation of energy

In this analysis, the following assumptions are considered: (i) the bed has sufficiently unoccupied space and vapor pressure throughout the bed is uniform but varies with time, (ii) the particles are small enough to be regarded as saturated, (iii) constant thermophysical properties, (iv) refrigerant vapor behaves as an ideal gas and (v) adsorbed phase behaves as liquid. According to these assumptions, the energy equation for the heat transfer fluid can be written as,

$$M_f c_f \frac{\partial T_f}{\partial t} = -\dot{m}_f c_f L \frac{\partial T_f}{\partial x} + k_f A_f L \left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2} \right) \quad (1)$$

The energy equation for the bed is expressed as,

$$M_a c_a \left(1 + \frac{c_r}{c_a} q + \frac{M_m c_m}{M_a c_a} \right) \frac{\partial T_b}{\partial t} - \rho_a A_a L Q_{st} \frac{\partial q}{\partial t} = k_{eff} A_a L \left(\frac{\partial^2 T_b}{\partial x^2} + \frac{\partial^2 T_b}{\partial y^2} \right) \quad (2)$$

where, M_f, M_a and M_m are the mass of the heat transfer fluid, the adsorbent and the inert material, respectively, A_f, the heat transfer area of fluid side, A_a, the heat transfer area of adsorbent side, T_f, the temperature of fluid, T_b, the temperature of adsorbent bed and m_f, the mass flow of heat transfer fluid. Other parameters are explained in the nomenclature.

3.2. Conservation of mass

In this analysis, it is assumed that the rate of change of moisture content in the bed is proportional to the difference between the equilibrium and the actual moisture content. Therefore, the refrigerant mass balance can be expressed as,

$$M_a \frac{\partial q}{\partial t} = h_m (q_e - q), \quad (3)$$

where, q_e and q are the equilibrium uptake and actual moisture content at (T_b, T_s) and h_m is the mass transfer

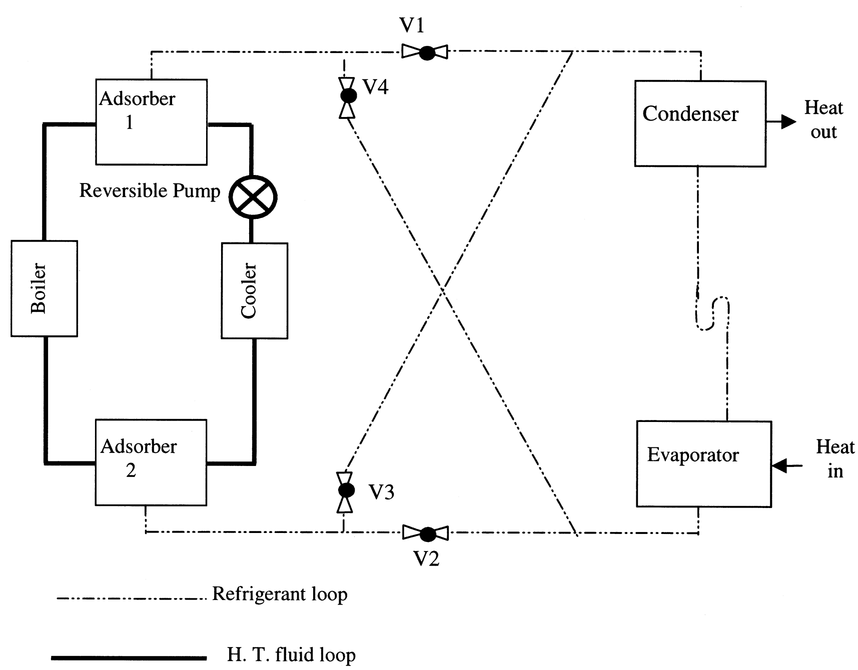

Fig. 1. Schematic of two-bed adsorption refrigeration systems.

coefficient. The initial and boundary conditions for the present problems are as follows:

For regeneration process,

$$T_f(x, y, 0) = T_b(x, y, 0) = T_c, \tag{4}$$

$$T_f - T_h = \frac{\partial T_b}{\partial x} = 0 \quad \text{at } x = 0,$$
$$\frac{\partial T_f}{\partial x} = \frac{\partial T_b}{\partial x} = 0 \quad \text{at } x = L, \tag{5}$$

$$\frac{\partial T_f}{\partial y} = 0 \quad \text{at } y = 0,$$
$$k_f \frac{\partial T_f}{\partial y} = h_f(T_b - T_f) \quad \text{at } y = D_f, \tag{6}$$

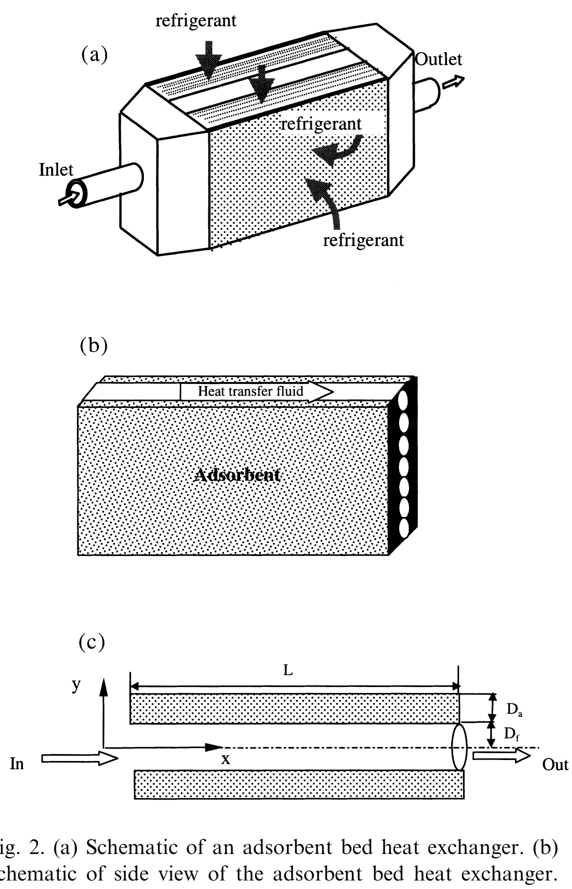

Fig. 2. (a) Schematic of an adsorbent bed heat exchanger. (b) Schematic of side view of the adsorbent bed heat exchanger. (c) Schematic of a flow channel.

$$k_{eff} \frac{\partial T_b}{\partial y} = h_f(T_b - T_f) \quad \text{at } y = D_f,$$
$$\frac{\partial T_b}{\partial y} = 0 \quad \text{at } y = D = D_f + D_a \tag{7}$$

and

$$q = q_e = q_e(T_c, T_{eva}) = q_{max} \quad \text{at } t = 0 \tag{8}$$

For adsorption process,

$$T_f(x, y, t_{hc}) = T_b(x, y, t_{hc}) = T_h, \tag{9}$$

$$T_f - T_c = \frac{\partial T_b}{\partial x} = 0 \quad \text{at } x = L,$$
$$\frac{\partial T_f}{\partial x} = \frac{\partial T_b}{\partial x} = 0 \quad \text{at } x = 0, \tag{10}$$

$$\frac{\partial T_f}{\partial y} = 0 \quad \text{at } y = 0,$$
$$k_f \frac{\partial T_f}{\partial y} = h_f(T_b - T_f) \quad \text{at } y = D_f, \tag{11}$$

$$k_{eff} \frac{\partial T_b}{\partial y} = h_f(T_b - T_f) \quad \text{at } y = D_f,$$
$$\frac{\partial T_b}{\partial y} = 0 \quad \text{at } y = D = D_f + D_a \tag{12}$$

and q is known from previous period.

The following groups of transformation are introduced into Eqs. (1)–(12) to normalize the governing equations,

$$\theta = \frac{T - T_c}{T_h - T_c}, \quad \tau = \frac{t}{t_{hc}}, \quad X = \frac{x}{L}, \quad Y = \frac{y}{D} \quad \text{and}$$
$$\bar{q} = \frac{q}{q_{max}} \tag{13}$$

Therefore, the resulting non-dimensional equations are as follows:

Energy equation for the heat transfer fluid

$$\alpha_{f-a} w \frac{\partial \theta_f}{\partial \tau} = -\frac{\partial \theta_f}{\partial X} + \frac{NTU \cdot Kr}{Bi(1 + Hr)Ar} \left(\frac{\partial^2 \theta_f}{\partial X^2} + Ar^2 \frac{\partial^2 \theta_f}{\partial Y^2} \right) \tag{14}$$

Energy equation for the bed

$$\left[(1+\alpha_{r-a}\bar{q}+\alpha_{m-a})\frac{\partial\theta_b}{\partial\tau} - \frac{\partial\bar{q}}{\partial\tau}\right]\frac{w\cdot Bi(1+Hr)\cdot Hr\cdot Ar}{NTU} = \frac{\partial^2\theta_b}{\partial X^2}+Ar^2\frac{\partial^2\theta_b}{\partial Y^2} \tag{15}$$

The mass balance equation can be expressed as

$$\frac{d\bar{q}}{d\tau} = \mu(\bar{q}_e - \bar{q}) \tag{16}$$

Initial and boundary conditions are as follows:

For regeneration process

$$\theta_f(X,Y,0) = \theta_b(X,Y,0) = 0 \tag{17}$$

$$\theta_f - 1 = \frac{\partial\theta_b}{\partial X} = 0 \quad \text{at } X=0, \qquad \frac{\partial\theta_f}{\partial X} = \frac{\partial\theta_b}{\partial X} = 0 \quad \text{at } X=1 \tag{18}$$

$$\frac{\partial\theta_f}{\partial Y} = 0 \quad \text{at } Y=0, \qquad \frac{\partial\theta_f}{\partial Y} = \frac{Bi(1+Hr)}{Kr}(\theta_b-\theta_f) \quad \text{at } Y=\frac{Hr}{1+Hr} \tag{19}$$

$$\frac{\partial\theta_b}{\partial Y} = 0 \quad \text{at } Y=1, \qquad \frac{\partial\theta_b}{\partial Y} = Bi(1+Hr)(\theta_b-\theta_f) \quad \text{at } Y=\frac{Hr}{1+Hr} \tag{20}$$

and

$$\bar{q} = 1 \tag{21}$$

For adsorption process

$$\theta_f(X,Y,1) = \theta_b(X,Y,1) = 1 \tag{22}$$

$$\theta_f = \frac{\partial\theta_b}{\partial X} = 0 \quad \text{at } X=1, \qquad \frac{\partial\theta_f}{\partial X} = \frac{\partial\theta_b}{\partial X} = 0 \quad \text{at } X=0 \tag{23}$$

$$\frac{\partial\theta_f}{\partial Y} = 0 \quad \text{at } Y=0, \qquad \frac{\partial\theta_f}{\partial Y} = \frac{Bi(1+Hr)}{Kr}(\theta_b-\theta_f) \quad \text{at } Y=\frac{Hr}{1+Hr} \tag{24}$$

$$\frac{\partial\theta_b}{\partial Y} = 0 \quad \text{at } Y=1, \qquad \frac{\partial\theta_b}{\partial Y} = Bi(1+Hr)(\theta_b-\theta_f) \quad \text{at } Y=\frac{Hr}{1+Hr} \tag{25}$$

and

$$\bar{q} = \text{known from previous period.} \tag{26}$$

The following non-dimensional parameters are used in this analysis:

Non-dimensional bed switching frequency

$$\omega = \frac{M_a c_a}{\dot{m}_f c_f t_{hc}} \tag{27}$$

Number of transfer unit

$$NTU = \frac{h_f A_f}{\dot{m}_f c_f} \tag{28}$$

Bed Biot number

$$Bi = \frac{h_f D_a}{k_{eff}} \tag{29}$$

Inert material alpha number

$$\alpha_{m-a} = \frac{M_m c_m}{M_a c_a} \tag{30}$$

Fluid alpha number

$$\alpha_{f-a} = \frac{M_f c_f}{M_a c_a} \tag{31}$$

Refrigerant alpha number

$$\alpha_{r-a} = \frac{c_r}{c_a} q_{max} \tag{32}$$

Adsorbent Beta number

$$\beta = \frac{Q_{st} q_{max}}{c_a(T_H - T_c)} \tag{33}$$

Thermal conductivity ratio

$$Kr = \frac{k_f}{k_{eff}} \tag{34}$$

Aspect ratio

$$Ar = \frac{L}{D} \tag{35}$$

Heat exchanger thickness ratio

$$Hr = \frac{D_f}{D_a} \tag{36}$$

Non-dimensional mass transfer coefficient

$$\mu = \frac{h_m t_{hc}}{M_a} \tag{37}$$

Lambda number

$$\lambda = \frac{Q_{st}}{L_v} \tag{38}$$

4. Thermodynamic property model

4.1. Equilibrium state

To determine the moisture content in the bed at every moment, an equilibrium equation is needed. In this analysis, Freundlich's equation [22] is chosen for equilibrium state and defined by

$$q_e = K \cdot \left(\frac{P_s(T_v)}{P_s(T_b)}\right)^{1/n} \tag{39}$$

where q_e is amount adsorbed in equilibrium with pressure P_s (T_v). $P_s(T_v)$ and $P_s(T_b)$ are the saturation vapor pressures at temperatures T_v (water vapor) and T_b (silica gel bed), respectively. Chihara and Suzuki [22] suggested the values of K and n for the silica gel-water pair as $K = 0.346$ kg/kg and $n = 1.6$ from the experimental results, where K denotes the limiting amount adsorbed at $P_s(T_v)/P_s(T_b) = 1$. The saturation pressure and temperature are related by the Antonie's equation [23].

4.2. System performance equations

The system performance of an adsorption cooling unit can be characterized by the coefficient of performance (COP) and specific cooling capacity (SCC).

The energy input during a half cycle

$$Q_{in} = \dot{m}_f c_f \int_0^{t_{hc}} (T_{in} - T_{out})\mathrm{d}t \tag{40}$$

The heat extracted from the evaporator is

$$Q_{eva} = \int_0^{t_{hc}} \left[L_v(T_{eva}) - c_r(T_{con} - T_{eva})\right]\frac{\mathrm{d}q}{\mathrm{d}t}\mathrm{d}t \tag{41}$$

The coefficient of performance is defined as

$$\mathrm{COP} = \frac{Q_{eva}}{Q_{in}} = \frac{\int_0^{t_{hc}} \left[L_v(T_{eva}) - c_r(T_{con} - T_{eva})\right]\frac{\mathrm{d}q}{\mathrm{d}t}\mathrm{d}t}{\dot{m}_f c_f \int_0^{t_{hc}} (T_{in} - T_{out})\mathrm{d}t} \tag{42}$$

In terms of non-dimensional parameters defined previously, COP can be expressed as

$$\mathrm{COP} = \frac{\omega\left[\lambda\beta - \alpha_{r-a}(\theta_{con} - \theta_{eva})\right]\bar{q}_{max}}{\int_0^1 (1 - \theta_{out})\mathrm{d}\tau} \tag{43}$$

The specific cooling capacity is given by

$$\mathrm{SCC} = \frac{Q_{eva}}{M_a t_{hc}} \tag{44}$$

Introducing a definition of non-dimensional specific cooling capacity NSCC as follows

$$\mathrm{NSCC} = \frac{\mathrm{SCC}}{\dot{m}_f c_f (T_h - T_c)/M_a}, \tag{45}$$

one may obtain

$$\mathrm{NSCC} = \omega\left[\lambda\beta - \alpha_{r-a}(\theta_{con} - \theta_{eva})\right]\bar{q}_{max} \tag{46}$$

5. Solution methodologies

The set of non-dimensional partial differential equations, which governs the heat and mass transfers

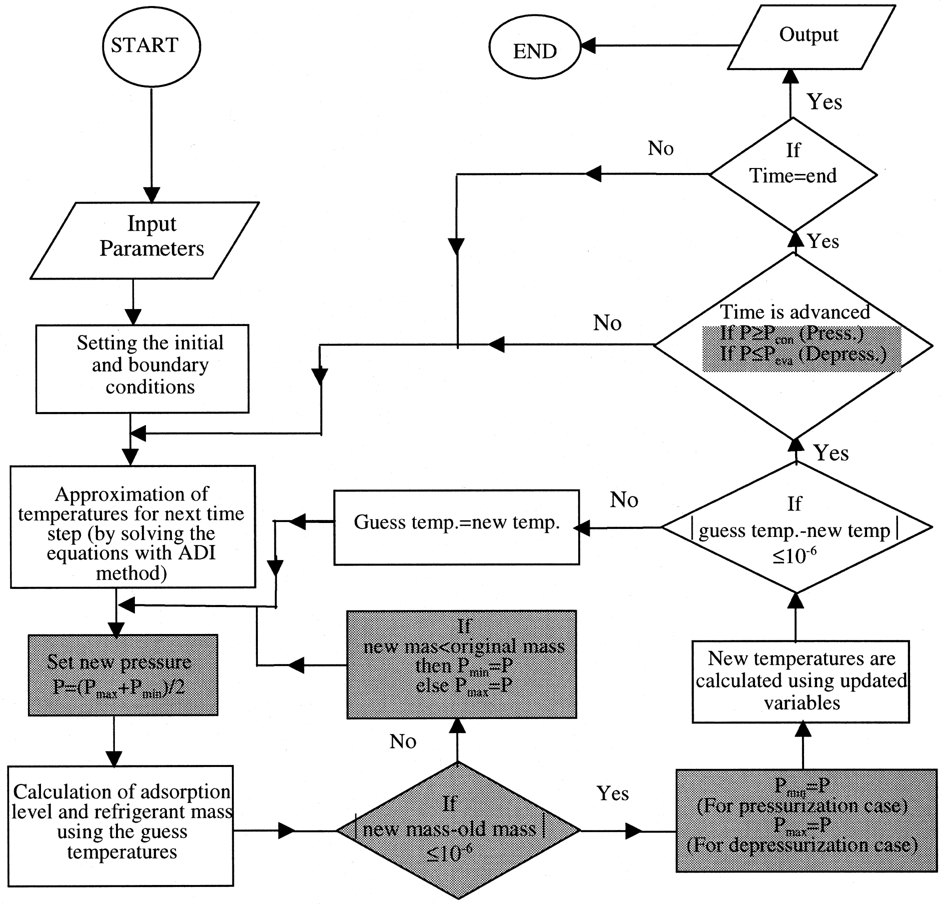

Fig. 3. Sequence of computer program.

of the systems, was solved numerically by using the finite difference scheme. The entire computational domain is divided into a number of equal step discrete elements. The spatial second derivative term is approximated by the second order central difference scheme and spatial first derivative term (convection term) is estimated by the quadratic upstream differencing scheme (QUDS) [24]. An alternating direction implicit (ADI) method is employed to solve the nonlinear set of equations (14)–(16) with the initial and boundary conditions (17)–(26).

The four steps of thermodynamic process were taken into consideration in the solution process. The solution techniques applied in this analysis are divided mainly into two strategies; one is pressurization/depressurization process and the other is constant pressure process. During the pressurization/depressurization process, the mass transfer into the system is assumed to be constant, i.e., no vapor mass is allowed to enter/leave the system. The bed pressure can be calculated by checking the mass balance in the bed. The sequence of program is shown in Fig. 3. The whole flow chart given in Fig. 3 is for the pressurization/depressurization process. For the constant pressure process, the shaded parts from Fig. 3 should be eliminated. The convergence criteria for all cases used in this program is 10^{-6}. The values taken for the base run of this analysis are presented in Table 2.

6. Results and discussions

In the present analysis, a set of non-dimensional

Table 2
Base run parameters

T_c = 20°C	NTU = 50	Kr = 2.0	Q_{st} = 2800 kJ/kg
T_h = 80°C	Bi = 0.5	μ = 0.34	L_v = 2500 kJ/kg
T_{con} = 20°C	Ar = 10	$\alpha_{f-a} = 0.1$	q_{max} = 0.34 kg/kg
T_{eva} = 14°C	Hr = 0.5	$\alpha_{m-a} = 0.2$	

parameters, which present a physical characteristic of the system, is presented and discussion of the effect of the parameters on the system performance is illustrated in the following subsections.

6.1. Switching frequency, ω

It is well known that the shorter the cycle time, the lower is the performance. However, long cycle time may cause waste of energy. An experimental analysis by Boelman et al. [13] showed that COP increases monotonically with cycle time, at least until 1300 s. The same agreement has been observed by Saha et al. [14] and Chua et al. [16]. Their analyses were conducted based on a same machine installed at Tokyo University of Agriculture and Technology. They discussed the effect of cycle time on the system performance for given design parameters of the heat exchanger. However, they did not investigate the effect of design parameter on the system performance. It is a major operational question how to determine an optimum bed switching frequency, because there should be an optimum cycle time for an adsorption refrigeration system.

In this analysis, a non-dimensional switching frequency is presented which is defined as $\omega = M_a c_a / \dot{m}_f c_f t_{hc}$, i.e., inversely proportional to the time of heating (desorption) process or cooling (adsorption) process (t_{hc}).

Physically, ω means the ratio of the required time (t_{rh}), to take a heat capacity by the adsorbent materials, to the switching time (t_{hc}). The required time (t_{rh}) is an internal characteristic time of the system, while t_{hc} is an external time constant which can be controlled during experiments. For a given amount of adsorbent (silica gel) in the system, an optimum operating time should exist to take a given heat capacity ($M_a c_a$) by the heat transfer fluid. In an ideal system with no heat and mass transfer resistance, and infinite heat transfer area, the optimum switching frequency must be 1.0 to take just required heat capacity ($M_a c_a$) by the mass flow rate of the heat transfer fluid $\dot{m}_f$. In actual system, the optimum ω depends on design parameters, which will be discussed in the following subsections.

6.2. Number of transfer units (NTU)

The number of transfer units (*NTU*) is one of the most important design parameters of a heat exchanger. It represents the heat transfer characteristic inside the reactor of the adsorption refrigeration systems. It is defined as the ratio of the heat transfer at the interface of fluid/tube to the advection of energy in the fluid.

The effect of *NTU* on the system performance is shown in Fig. 4a and b. From these figures, it can be observed that the system performance, namely, COP and NSCC increase as the *NTU* increases. It is well known that a bigger heat exchanger gives a higher value of *NTU*. But the question is how much one can increase the size of a heat exchanger without limit because the pressure drop across the heat exchanger increases with size, which will deteriorate the systems performances. Fig. 4a and b show that an increase in *NTU* leads to decrease in the rate of improvement of systems performance. Therefore, it may be concluded that there should be an optimum value of *NTU* for maximum COP and NSCC. The optimum value of *NTU* for the present base conditions is considered as 50.

The effect of switching frequency for different *NTU* is also presented in Fig. 4a and b. From these figures, one may see that there is an optimum switching frequency for each *NTU* and this optimum value increases with the increase of *NTU*. It is also observed that the system performance may be deteriorated if the

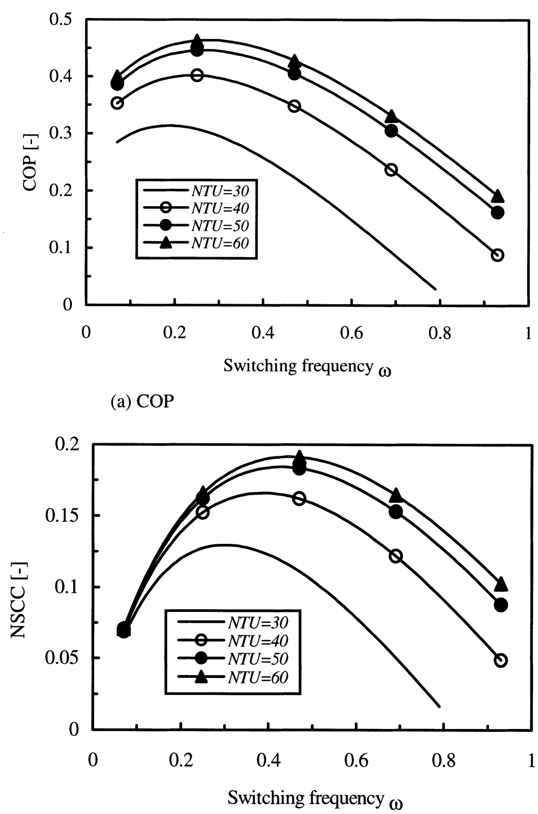

Fig. 4. COP and NSCC for different *NTU* vs. switching frequency ω.

switching frequency is set far from the optimum value. The optimum values of ω ranged 0.2–0.35 for COP requirement and 0.25–0.45 for NSCC requirement. To determine the optimum cycle time for an adsorption refrigeration system, one needs only one optimum point of switching frequency. The final optimum point can be determined between the optimum points of COP and NSCC depending on the requirements of COP and NSCC.

6.3. Bed Biot number (*Bi*)

Bi is a Biot number of an adsorbent bed heat exchanger, which implies the heat transfer characteristic of the heat exchanger. It is defined as the ratio of conductive resistance of the adsorbent bed to the convective resistance in the heat transfer fluid. An increase in *Bi* is equivalent to the increase of conductive resistance or decrease in convective resistance.

Fig. 5a and b show the effect of *Bi* on the system performance. From these figures, it is seen that both COP and NSCC decrease with increasing *Bi*. The reason is that the *Bi* of an adsorbent heat exchanger depends not only on convective heat transfer resistance but also on conductive resistance of the adsorbent bed. A higher value of conductive resistance in the adsorbent layer gives a smaller heat transport in the adsorbent bed. This means the conductive resistance in the adsorbent bed is more dominant than convective resistance. And these results agree with the results of Zheng et al. [9].

One may also observe from Fig. 5a and b that there exists an optimum switching frequency for both COP and NSCC. These figures indicate that COP and NSCC are not optimized at the same switching frequency as it is observed for the case of different *NTU*. It is seen that the optimum ω increases with decreasing *Bi*.

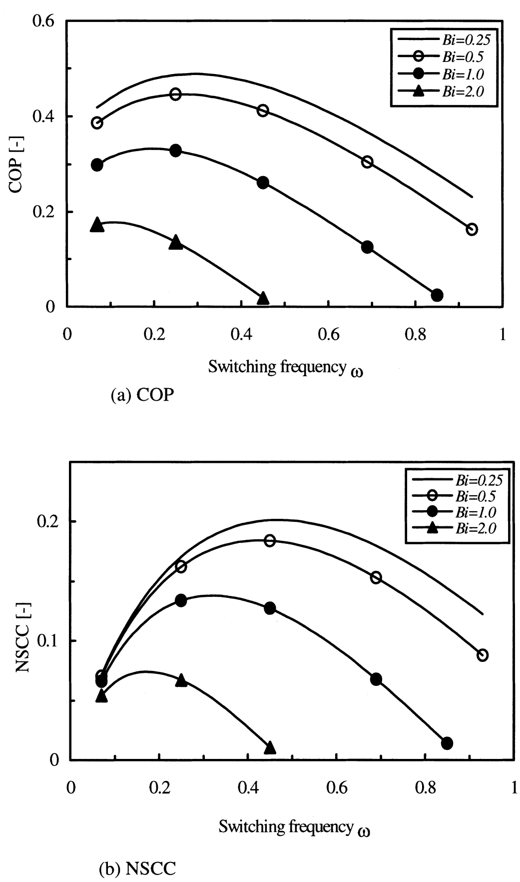

Fig. 5. COP and NSCC for different *Bi* vs. switching frequency ω.

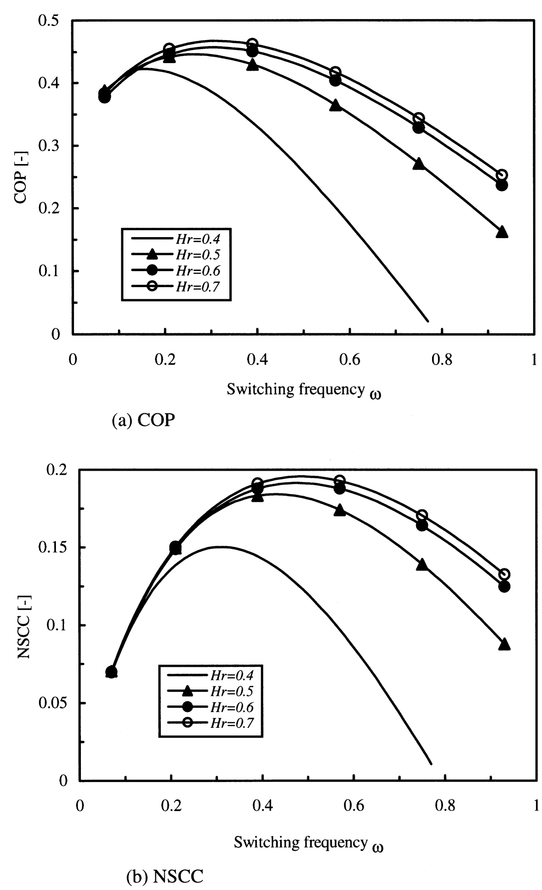

Fig. 6. COP and NSCC for different *Hr* vs. switching frequency ω.

6.4. Heat exchanger thickness ratio (Hr)

A non-dimensional parameter Hr is defined here as the ratio of the radius of fluid channel to the thickness of adsorbent bed. The effect of different Hr on the system performance is illustrated in Fig. 6a and b. It is seen that an increase in Hr leads to increase in both COP and NSCC. An increase in Hr is analogous to a decrease in the thickness of adsorbent bed or to an increase in the radius of heat transfer fluid channel. That means, the thinner the adsorbent bed, higher is the performance, which is an expected result as it is observed for the case of different Bi. If h_f is much smaller than h_a (which means the convective resistance in fluid side is dominant), the effect of D_f on the system performance is complicated because h_f decreases with increasing D_f while A increases. If h_f is comparable or larger than h_a, the effect of D_f on the system performance is somewhat straight forward; the system performance increases with increasing D_f. However, one can not increase D_f without limit for a given set of non-dimensional parameters NTU, Bi, Ar, etc., because there is a tacit relation among them. There should be an optimum allocation between D_f and D_a for the fixed values of those parameters. One may see that the rate of change in both COP and NSCC is very small when the value of Hr is greater than 0.5. When the Hr is increased from 0.4 to 0.5, the COP increases from 0.42 to 0.45 and NSCC increases from 0.15 to 0.18, which means about 7% improvement in COP and 20% in NSCC. However, increasing the value of Hr from 0.5 to 0.6, the COP improves from 0.45 to 0.46 and NSCC from 0.18 to 0.19, that is, only 0.02% improvement in COP and 6% in NSCC. Therefore, the optimum value of Hr for the present basis conditions is considered as 0.5.

Fig. 6a and b also show that the optimum switching frequency ω is different for COP and NSCC and this optimum value increases as Hr increases. The optimum ω for COP requirement is lower than that for NSCC requirement.

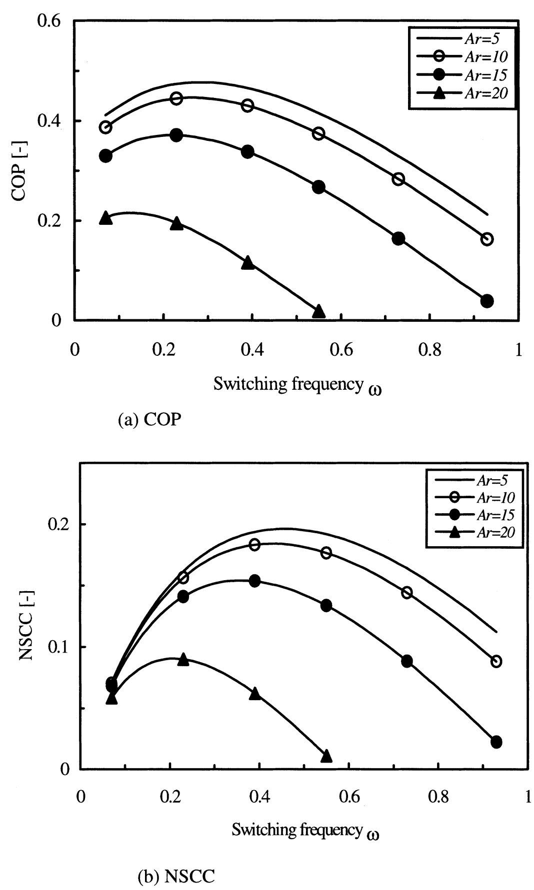

Fig. 7. COP and NSCC for different Ar vs. switching frequency ω.

6.5. Aspect ratio (Ar)

The aspect ratio of a heat exchanger is defined as the ratio of length to the width of the heat exchanger. The effect of the aspect ratio on the COP and NSCC has been presented in Fig. 7a and b. It can be seen that the COP as well as NSCC increases as Ar decreases. An increase in Ar is equivalent to increasing of L or decreasing of D. That means, the system performance increases with decreasing L and increasing D. In this present parametric analysis, we vary only Ar without changing any other non-dimensional parameter. Increasing L without changing any other parameter means that the fixed amount of heat (from heat transfer fluid) is used to heat up more space, resulting in a poor performance. Therefore, an optimum length L of a heat exchanger is needed to get maximum performance for a given diameter. It is observed for different Bi and Hr that the thinner the adsorbent bed, the better the performance. The width of the heat exchanger, D is the sum of the bed thickness, D_a and the radius of fluid channel, D_f. One cannot increase the width of a heat exchanger with the increasing of bed thickness. But, the only possible way to an increase in D is by increasing the radius of fluid channel. The possibility of the increase in radius of fluid channel is discussed in the previous subsection. From the discussions regarding Hr and Ar, it can be concluded that D_f plays an influential role in improving the system performance.

Fig. 7a and b also show that COP as well as NSCC improve slightly when Ar is less than 10. Reducing Ar by half from the value 20 to 10, COP improves from 0.22 to 0.45 and NSCC from 0.09 to 0.18. However, reducing Ar by half from 10 to 5, COP gains only from 0.45 to 0.48 and NSCC from 0.18 to 0.2. Therefore, the optimum Ar for the base run case is considered as 10. An increase in optimum ω has been also observed with the decrease of Ar.

7. Conclusions

The effect of design parameters on the system performance in two-bed silica gel-water, adsorption refrigeration system was investigated using numerical techniques. The following conclusions were drawn from this parametric study:

1. The system performance is very much sensitive to the switching frequency, ω. There is an optimum ω for a given set of design parameters. The optimum ω for the present base run case is estimated as 0.27 for COP requirement and 0.43 for NSCC requirement.
2. It is seen that the COP and NSCC may not be optimized at a same switching frequency. The optimum ω for COP requirement is lower than that for NSCC requirement.
3. Optimum ω as well as system performance increases with the increase of NTU and Hr, but decreases with the increase of Bi and Ar.
4. It may be concluded that the cycle time of an adsorption refrigeration system is strongly dependent on the configuration of the heat exchanger. The system performance will be lower if the cycle time of the system is set far from the optimum cycle time.

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