Design Effect of Different Components and Economic Evaluation of an Adsorption Chiller on the System Performance

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A conventional silica gel/water adsorption chiller has been analyzed numerically. A novel non-dimensional mathematical model has been presented to analyze the design effect of different components of an adsorption chiller. The design parameters of this system are characterized by the number of transfer unit, NTU, of different components and the inert material alpha number, α of different components of the systems. Results show that condenser NTU_a has the most influential effect on the system performance, which is followed by adsorber NTU_e. It is also seen that coefficient of performance (COP) and non-dimensional specific cooling capacity increases with the increase of NTU_a and NTU_e, but decreases with the increase of inert material alpha number. A thermo-economic data of the adsorption chiller and some other heat pump systems those are in practical operation are also presented.

1. Introduction

In order to promote environmentally friendly energy utilization systems, one of the major concerns is to develop CFC and HCFC-free refrigeration/heat pump systems, which utilize renewable energy and/or low-temperature waste heat as the driving sources. Adsorption cooling systems are promising for providing a safe alternative to CFC and HCFC-base refrigeration devices. From this context, a number of researchers investigated the possibility of adsorption heat pumping/refrigeration system driven by waste heat or by renewable energy sources [1-3].

The design effect of an adsorber has been studied by Zheng et el. [4] and Alam et el. [5]. A conventional adsorption heat pump/cooling unit consists of four major units, namely, adsorber, desorber, condenser and evaporator. The system performance may deteriorate seriously if one of the major units of an adsorption cooling system is not designed optimally. All the parametric study for conventional adsorption chiller up to now has been conducted based on a fixed design, i.e. in dimensional form. To predict the optimum design conditions of the adsorption system, the model should be defined in non-dimensional form.

From this context, a non-dimensional simulation model has been presented to investigate the design effect of different components of an adsorption chiller in the present study.

2. Mathematical Model for Conventional Adsorption Chiller

The basic conventional adsorption chiller consists of two pairs of heat exchangers, namely an evaporator/adsorber pair and a condenser/desorber pair as shown in Fig.1. The working principle of the conventional chiller is available elsewhere in Saha et al. [2] and Alam et al. [6]. In the present study, it is assumed that the temperature and pressure are uniform throughout the whole adsorber. It is also assumed that the system has no heat loses to the external environment. According to these assumptions, the dynamic behavior of heat and mass transfer inside different components of the adsorption chiller can be written as the following forms.

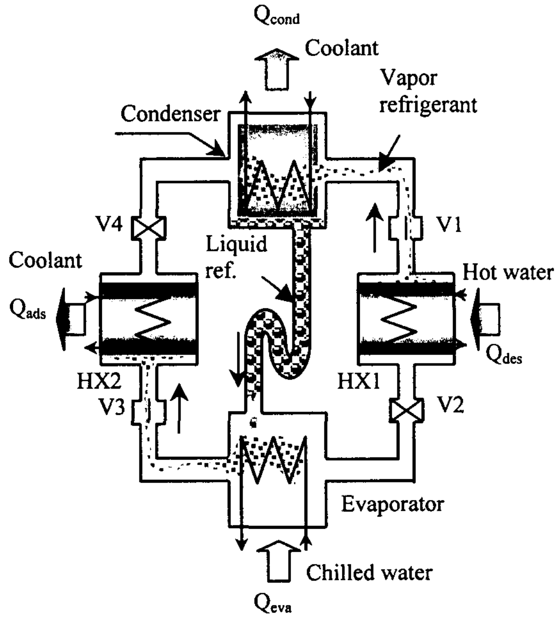

Fig. 1. Schematic diagram of the basic adsorption chiller.

2.1 Energy balance in adsorber/desorber

The energy balance in the silica-gel bed can be written as;

$$\frac{d}{dt}\{(W_sC_s + W_sC_sq + W_{hex}C_{hex})T_b\} = W_s \cdot Q_{st} \cdot \frac{dq}{dt} + \delta[\{\gamma(T_b - T_{con}) - (1-\gamma)(T_b - T_{eva})\} \cdot W_s \cdot C_w \cdot \frac{dq}{dt} \\ \gamma\, \dot{m}_{hot} C_w (T_{hotin} - T_{hotout}) \\ (1-\gamma)\dot{m}_{cool} C_w (T_{coolin} - T_{coolout}) \tag{1}$$

where, δ taking the value 1 or 0 depending whether the valves between the beds and condenser/evaporator are open or not and γ equal to either 1 or 0 depending whether bed roles as desorber or adsorber. The outlet temperature of hot-water and coolant can be expressed as;

$$T_{w,out} = T_b + (T_{w,in} - T_b)\cdot \exp\left(-\frac{U_{hex}A_{hex}}{\dot{m}_w C_w}\right) \tag{2}$$

2.2 Energy balance in condenser/evaporator

The condenser heat exchanger energy balance equation can be written as;

$$\frac{d}{dt}\{(W_{ref,hex}C_{ref,hex} + W_{w,ref}C_w)T_{ref}\} = \xi\{-L\cdot W_s \frac{dq_{ad}}{dt} - W_sC_w(T_{ad} - T_{ref})\frac{dq_{ad}}{dt}\} + \dot{m}_{con}C_w(T_{con,in} - T_{con,out}) \tag{3}$$

where, ξ=1 or 0, depending whether the des/adsorber is connected with cond./evap. or not and the heat transfer equation can be expressed as;

$$T_{ref,out} = T_{ref} + (T_{ref,in} - T_b)\exp\left(-\frac{U_{ref}A_{ref}}{\dot{m}_{ref}C_{ref}}\right) \tag{4}$$

2.3 Adsorption Equilibrium

The adsorption equilibrium for silica-gel/water vapor correlated by the following equation;

$$q = A(T_s)(P_{sat}(T_v)/P_{sat}(T_s))^{B(T_s)} \tag{5}$$

where, $A(T_s)$ and $B(T_s)$ are the function of silica gel temperature as discussed by Saha et al. [2].

2.4 Normalization

The set of equations (1)—(4) has been normalized by introducing following transformations;

$$\theta = \frac{T - T_{cool}}{T_{hot} - T_{cool}}, \quad \tau = \frac{t}{t_{cycle}} \text{ and } \bar{q} = \frac{q}{q_{max}} \tag{6}$$

In terms of non-dimensional form, the energy balance equation for adsorber/desorber can be expressed as;

$$(1 + \alpha_{r-s}\bar{q} + \alpha_{b,m-s})\omega\frac{d\theta_b}{d\tau} = \omega\beta\frac{d\bar{q}}{d\tau} + \delta\omega[\{\gamma(\theta_b - \theta_{con}) - (1-\gamma)(\theta_b - \theta_{eva})\}\alpha_{r-s}\frac{d\bar{q}}{d\tau}] \\ + \gamma(1-\theta_b)(1 - \exp(-NTU_a)) - Mr_{hot-cool}\cdot(1-\gamma)\theta_b(1 - \exp(-NTU_a / Mr_{cool-hot})) \tag{7}$$

In the similar way, the energy balance for condenser and evaporator can be written as;

$$(\alpha_{r-s}\mu + \alpha_{ref,m-s})\omega\frac{d\theta_{ref}}{d\tau} = \zeta[\{(\theta_{ad} - \theta_{ref})\cdot\frac{d\bar{q}_a}{d\tau} - \lambda\frac{d\bar{q}_a}{d\tau}]\omega + Mr_{chill-hot}(\theta_{chillin} - \theta_{eva})(1 - \exp(-NTU_e)) \tag{8}$$

2.5 System performance equations

In terms of dimensionless parameters, coefficient of performance (COP) can be written as,

$$\text{COP} = \frac{Q_{eva}}{Q_{in}} = Mr_{chill-hot}\frac{\int_0^1(\theta_{chillin} - \theta_{chillout})d\tau}{\int_0^1(1 - \theta_{hotout})d\tau} \tag{9}$$

and the non-dimensional cooling capacity can be expressed as,

$$\text{NCC} = Mr_{chill-hot}\int_0^1(\theta_{chillin} - \theta_{chillout})d\tau \tag{10}$$

where, NCC can be defined by;

$$\text{NCC} = \frac{\text{cooling capacity}}{\dot{m}_{hot}C_w(T_{hot} - T_{cool})} \tag{11}$$

3. Results and Discussion

The system of non-dimensional differential equations (7)-(9) has been solved numerically by finite difference approximation. The four thermodynamic steps were taken into consideration in the solution process. Two solution strategies have been employed during solution process, namely, the pressurization/depressurization process and the constant pressure process. During the pressurization/depressurization process, the mass transfer into the system is assumed to be constant, i.e., no vapor is allowed to enter or leave the system. The bed pressure can be calculated by checking the mass balance in the bed. The results obtained by the present simulation model compared well with experimental results of Boelman et al. [7]. A comparison of predicted and experimental outlet temperatures is shown in Fig. 2. The input data of temperature in the simulation procedure is taken as the average conditions of experiment as, T_{hotin}=85.2, T_{coolin}=30.1, $T_{chillin}$=13.9 and t_{cycle}=450s. Other values of non-dimensional parameters are calculated from the corresponding dimensional values of Boelman et al.[7]. From Fig.2, it can be observed that the present model agrees well with the experimental data. During the parametric investigation, desired parameter has been varied from a fixed value to a certain label until COP and NCC reach their asymptotic values. The base run parameters are furnished in Table 1. The effect of some most influential non-dimensional parameters on COP and NCC has been discussed in the following paragraphs.

The number of transfer unit, NTU is one of the most important design parameters of a heat exchanger. There are four heat exchangers in a conventional adsorption chiller, namely, adsorber, desorber, condenser and evaporator. In this paragraphs the effect of adsorber/desorber NTU on COP and NCC has been discussed. In the present model, it is assumed that NTU of adsorber and desorbers are identical. This is valid because, the heat transfer coefficient of adsorber and desorber can make identical by adjusting the flow rates of hot and cooling water. According to Saha et al. [2], the heat transfer coefficients of adsorber and desorber nearly same. Converting these values of heat transfer coefficient into NTU, the author of this study found that the adsorber/desorber NTU values are almost identical.

The effect of adsorber number of transfer unit, NTU_a on COP and NCC is depicted in Fig. 3. Both COP and NCC increase as NTU_a increases. It is well known that the higher the NTU, the better heat transport inside the adsorber reactor, which results in better performance. It may also be seen that the effect of NTUa on both COP and NCC is negligible if NTUa is greater than 2. Therefore, the optimum value of NTUa is considered as 2.0 for the present base run conditions.

Evaporator number of transfer unit, NTUe is defined as the ratio of heat transfer at the interface of chilled water/tube to the advection of energy in chilled water. Figure 4 depicts the influence of evaporator number of transfer unit, NTUe on COP and NCC. Increasing NTUe is equivalent to the increase in convective heat transfer between the chilled water and tube wall relative to energy supplied by the input chilled water, which results in higher cooling output and better performance. It is also seen that increasing NTUe is no longer valuable when NTUe is greater than 2.0. Therefore, the optimum value of NTUe for the base run case has been taken as 2.0 with the variation limit less than 5%.

Table 1. Base Run Parameters.

$Mr_{cool-hot}$=1.23	$\alpha_{b,m-s}$=1.87
$Mr_{con-hot}$=1.0	$\alpha_{con,m-s}$=0.23
$Mr_{chill-hot}$=0.5	$\alpha_{eva,m-s}$=0.12
NTUa=2.0	Q_{st} = 2800 kJ/kg
NTUc=1.0	L = 2500 kJ/kg
NTUe=2.0	q_{max} = 0.34 kg/kg

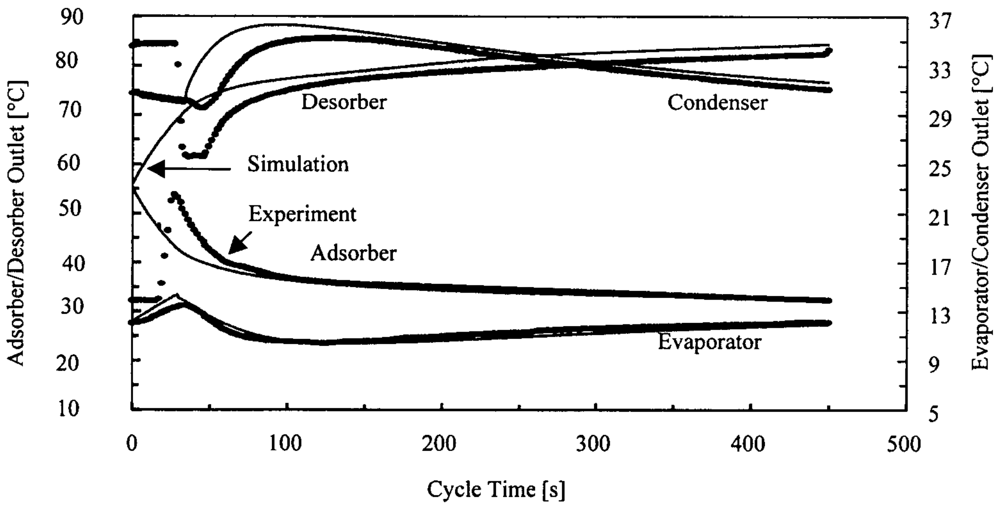

Fig. 2. Outlet temperature profiles of different components.

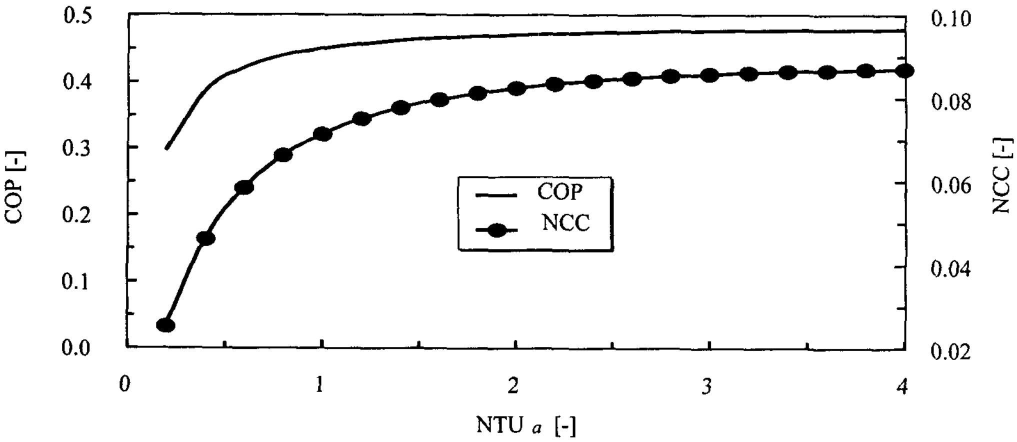

Fig. 3. Effect of adsorber number of transfer unit, NTUa on COP and NCC.

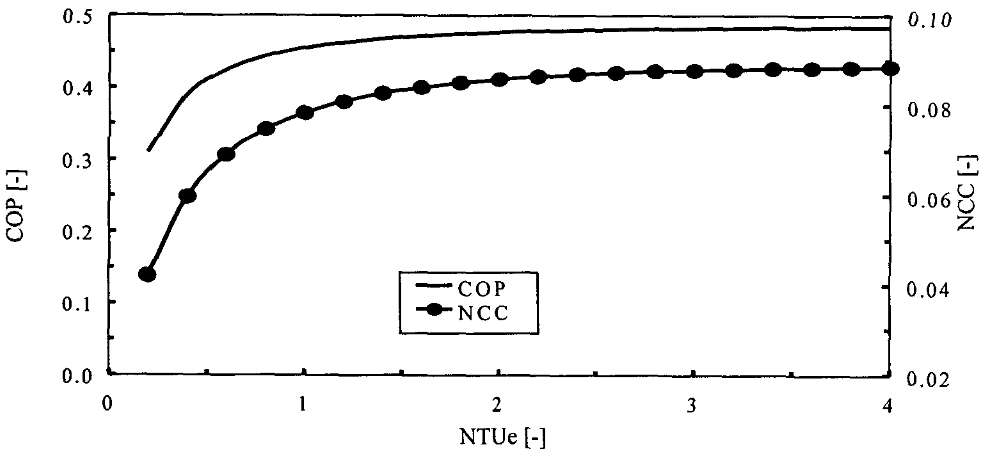

Fig. 4. Effect of evaporator number of transfer unit, NTUe on COP and NCC.

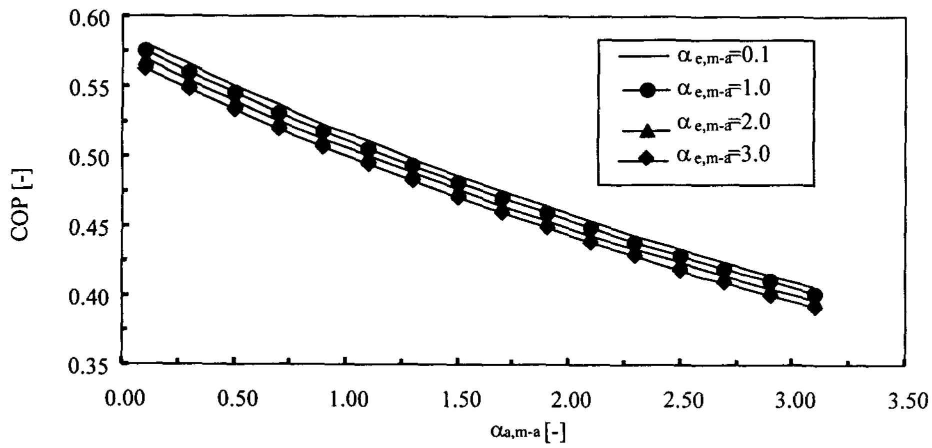

Fig. 5. Effect of inert material alpha number on COP.

In an adsorbent bed heat exchanger, some materials e.g. aluminum or copper are used to separate the heat transfer fluid from adsorbent and to extend the heat transfer surfaces (e.g. fins). These materials are known as the inert material as they do not adsorb or desorb refrigerant vapor. In this study, these inert materials are characterized by a non dimensional parameter, namely, inert material alpha number, $\alpha_{m\text{-}a}$ that is defined as the heat capacitance ratio of the inert mass to the adsorbent mass. The effects of adsorber/desorber and evaporator inert material alpha number on COP are depicted in Fig. 5.

It can be seen that COP improves with the decrease of $\alpha_{a,m\text{-}a}$. The reason is that the more inert material is added to the adsorber/desorber, the more energy is needed to heat up or cool down the structure, which is responsible for lowering system performance. It may also be seen that an increase in evaporator inert material alpha number, $\alpha_{e,m\text{-}a}$ leads to decrease in system performance. However, this effect is negligible. Therefore, it may be concluded that the amount of inert mass should be minimized to achieve maximum performance.

Table 2. Typical Thermo-economic Data of the Conventional Adsorption System and Various Heat Pumps.

System Type	Heat Source	COP	Temperature Glide [K]	Initial Investment [US$/kW]	Refrigerant	Source
Conventional Absorption	75~95°C	0.6~0.8	40~70	800	H_2O	HPTCJ (2001)
Conventional Adsorption	60~90°C	0.3~0.8	30~60	1000	H_2O	HPTCJ (2001)
Heat Pump	Exhaust air	3.1	20~25	2500~3000	R290	Berntsson (1999)
Heat Pump	Soil	2.5~2.7	3	1300~1600	R134a and R407C	Berntsson (1999)
Heat Pump	Rock	3.0	3	1500~1700	R134a and R407C	Berntsson (1999)

4. Thermo-economic Analysis

The economics of air-conditioning and heat pump machines to a large extent are determined by the required heat exchangers area to operate the cycle. From economic viewpoint, the installed heat exchanger area determines both the initial cost of a system and, via the COP, the operating or running cost (energy cost). Waste heat powered absorption and adsorption systems are characterized by high initial costs and relatively low energy and maintenance costs in comparison with the vapor compression systems. Table 2 shows the thermo-economic data of the conventional silica gel-water adsorption chiller and some other heat pump systems those are in practical operation. Straightforward economic comparison between the adsorption cooling system and commercially available heat pump systems are not possible as the chiller is rarely available in market or still in the R&D stage. As can be seen from Table 2, the adsorption system requires huge initial investments, even though the initial costs are lower than those of the cited heat pump systems, and thus the economic benefits may not be realized in a short time. The advantages of the system is that it does not require electricity or fossil fuels as the driving energy source i.e., no energy cost and has relatively low maintenance cost with a lifespan of around 20 years. Widespread dissemination of the proposed system will not only benefit in energy efficiency gains but also in the reduction of the cost of pollution abatement. The COP levels provided in Table 2 include the electricity consumption for circulation pumps on the heat source side of all three heat pump systems.

5. Conclusions

A new simulation model has been presented to investigate performance of a conventional adsorption chiller. The model shows good agreement with the experimental data in the case of different components temperature profiles. The following conclusions can be drawn from the present parametric study and thermo-economic evaluation;

(i) Both COP and NCC are sensitive to the adsorber/desorber number of transfer unit, NTUa, followed by the evaporator number of transfer unit, NTUe. Condenser number of transfer unit, NTUc has less influence on COP and NCC.

(ii) Among the inert mass of different heat exchanger, the inert mass in adsorber/desorber has the strongest effect. System performances improve with the reduction of inert mass in heat exchanger.

(iii) The over all performance of a chiller can be improved by optimizing the design and operating parameters. The optimum values of the different *NTU* for the base run case are as follows;
NTUa=NTUe=2.0, NTUc=1.0

(iv) The advantages of the silica gel-water based adsorption system is that it does not require electricity or fossil fuels as the driving energy source i.e., the system has no energy cost and has relatively low maintenance cost.

(v) Further study on adsorber/desorber heat exchanger design in different configuration is needed to find the effect of heat exchanger design on the system performance of an adsorption heat pump/ refrigeration systems.

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