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Modelling of an adsorption chiller with adsorbent-coated heat exchangers: Feasibility of a polymer-water adsorption chiller

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ABSTRACT

An analytical model is developed for an adsorption chiller with adsorbent-coated heat exchangers, where adsorbent is deposited on heat exchanger surface in thin film to improve heat and mass transfer characteristics. Approximate solutions are obtained from the simplified governing equations for the heat exchanger and then used to predict performance of the chiller. The analytical model provides the heat and mass fluxes in the system in explicit functions of a few dimensionless numbers including N_t, Ja, γ and C_r. The analytical model is validated via comparison with a two-dimensional numerical model in wide ranges of design and operating parameters. The maximum discrepancy is found ca. 13% in SCP and 0.02 point in COP. Performance of the chiller is discussed regarding the influences of various design and operating parameters. Some experimental results are also analyzed with the analytical model and the results are discussed focusing on the performance of polymer-coated heat exchangers.

1. Introduction

Adsorption cooling has been receiving great interest as an environmentally friendly alternative to the conventional refrigeration technology for its ability to operate with natural refrigerants and low-grade heat sources. In the literature, numerous studies are found regarding working pairs, refrigeration cycles, heat and mass transfer characteristics of adsorbers, lab-scale or commercial chillers [1–6] and theories and modelling techniques are also well summarized [7–9]. Among the popular working pairs, adsorbents include silica gel, activated carbon, zeolite and their composites with some hygroscopic salts and refrigerants include water, alcohol, ammonia and even some HFCs. Although different working pairs may be selected depending on desirable cooling temperature, available heat source temperature, required system size and so on, poor heat and mass transfer performance is a common problem for conventional adsorbents [3]. In order to improve the performance, researchers have been improving the properties of adsorbent and/or the performance of heat exchangers [10,11]. Recently, adsorbent-coated heat exchangers have been proposed as an alternative solution [12–14]. Ouchi et al. [12] synthesized porous alumina film ($\delta_s = 0.148$ mm) on an aluminum plate and measured adsorption equilibrium and adsorption/desorption rate of steam. From the results, it was predicted that a film-type heat exchanger would give 1.5 times larger SCP and 1.4–1.8 times larger COP than a packed-bed heat exchanger. They concluded that the enhancement was attributable to the reduced thermal resistance in the adsorbent film. Freni et al. [13] coated a flat-tube aluminum heat exchanger with SAPO-34 zeolite film ($\delta_s = 0.1$ mm) and compared its performance with that of a granular adsorber where the same type of zeolite grains ($d = 0.6$–0.7 mm) were packed in an identical heat exchanger. It was reported that the coated heat exchanger gave larger mass SCP (675 W/kg for coated, 498 W/kg for granular at 5min cycle period) but smaller volumetric SCP (93 W/liter for coated, 212 W/liter for granular) as well as smaller COP (0.24 for coated, 0.4 for granular). Bendix et al. [14] coated a set of flat-tube heat exchangers with TiAPSO, a titanium exchanged SAPO-34, in varying thickness and measured volumetric cooling powers up to 221 W/liter, which is comparable with that of the granular adsorber in Freni et al. [13]. They concluded that the heat and mass transfer resistances in the coating layer were not the limiting factors in sustaining the high powers measured. However, improving the performance simply by reducing the adsorbent thickness is not always successful because then adsorption capacity is reduced too [2]. A goal of this study is to evaluate feasibility of employing a

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Nomenclature

C	thermal mass, kJ K^{-1}
C_r	thermal capacity ratio
C_p	specific heat, kJ kg^{-1} K^{-1}
c	water concentration, kg m^{-3}
D	mass diffusivity, m^2 s^{-1}
d	diameter, m
h_{ti}	solid-side heat transfer coefficient at vapor-adsorbent interface, kW m^{-2} K^{-1}
h_{mi}	solid-side mass transfer coefficient at vapor-adsorbent interface, m s^{-1}
i_{ads}	heat of adsorption, kJ kg^{-1}
i_{fg}	heat of evaporation, kJ kg^{-1}
Ja	Jakob number
k	thermal conductivity, kW m^{-1} K^{-1}
L	fluid channel length, m
m	mass, kg
$\dot{m}$	mass flow rate, kg s^{-1}
N_t	number of transfer units
Nu	Nusselt number
$\dot{n}$	mass flux, kg m^{-2} s^{-1}
p	pressure, kPa
Q	heat quantity, kJ
$\dot{Q}$	heat transfer rate, kW
$\dot{q}$	heat flux, kW m^{-2}
Sh	Sherwood number,
T	temperature, K
t	time, s
t^*	dimensionless time ($=t/\tau$),
U_m	metal-to-adsorbent overall heat transfer coefficient, kW m^{-2} K^{-1}
U_w	fluid-to-metal overall heat transfer coefficient, kW m^{-2} K^{-1}
u	mean velocity in fluid channel, m s^{-1}
w	water uptake (=mass ratio of water to dry adsorbent)
x	coordinate in flow direction, m
y	coordinate perpendicular to flow direction, m

Greek symbols

α	thermal diffusivity
β	equilibrium constant
δ	thickness, m
ε	effectiveness
Γ	mass flow rate per perimeter ($=\rho u \delta$)
γ	heat transfer coefficient ratio
ϕ	extended surface ratio ($=A_{total}/A_{base}$)
τ	process (half cycle) period, s
ρ	density, kg m^{-3}
θ	dimensionless heat flux
ω	dimensionless mass flux

Superscripts

0	initial state
1	final state

Subscripts

0	reference state
a	advection
ads	adsorbent or adsorption process
c	cooling period
DS	dry adsorbent
eq	equilibrium
h	heating period
hex	heat exchanger
i	inlet or interface
l	liquid
m	metal or mean
o	outlet
pen	penetration theory
des	desorption process
s	adsorbent or solid
v	vapor
w	wall or heat transfer fluid

polymer desiccant as adsorbent. The polymer desiccant considered in this study is being used to fabricate desiccant wheels, which were tested and proven to work efficiently at low heating temperature [15]. Thanks to its large water-adsorption capacity (4–5 times larger than silica gels), it provides enough adsorption power even when it is very thinly coated on the wheels ($\delta_s < 0.1$ mm), which is important for enhancement of heat and mass transfer performance. Another goal is to develop a simple analytical model for an adsorption chiller. In Ref. [9], various modelling works in the past are categorized into three groups, i.e. thermodynamic, lumped parameter and heat and mass transfer models. When the local condition in a system is interested, a heat and mass transfer model, i.e. a two- or three-dimensional transient model is desirable. But such a model requires much time and effort not only to build but also excessive computing resources to use it, which is not desirable especially in the early stage of research and development. For this reason, this study is intended to develop a simple integral model that can predict global performance of a system accurately enough for the purpose of preliminary design and analysis. In the following, the governing equations for an adsorption heat exchanger are given in Section 2 and their solution with a 2D numerical model is described in Section 3. An analytical model introduced in Section 4 and it is validated via comparison with the numerical model in Section 5.1–3. Finally, the analytical model is applied to experimental analyses for demonstrating its application in Section 5.4.

2. System and governing equations

Fig. 1(a) shows a schematic diagram of a simple two-bed adsorption chiller, where two identical adsorption heat exchangers (AHE1, AHE2) are connected to a condenser (CON) and an evaporator (EVA) via a 4-way valve. The valve rotates 90° at a certain interval (τ) so that each adsorption heat exchanger is alternatively connected to the condenser or the evaporator. Although not shown, (cooling and heating) hydronic circuits are also provided to heat up a heat exchanger when it is connected to the condenser and cool down when it is connected to the evaporator. Fig. 1(b) illustrates the periodic operating conditions for the heat exchangers. In the period of $-\tau \le t \le 0$, AHE1 plays as an adsorber being exposed to low pressure (p_{eva}) and low temperature (T_{cwi}) in connection with the evaporator and the cooling water circuit. On the other hand, AHE2 plays as a desorber being exposed to high pressure (p_{con}) and high temperature (T_{hwi}) in connection with the condenser and the hot water circuit. In the next period $0 \le t \le \tau$, AHE1 and AHE2 alternate their roles as the pressure and hydronic connections are changed. Note that the pressure and