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Transient modeling of an adsorber using finned-tube heat exchanger

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ABSTRACT

This paper presents a heat and mass transfer simulation of an adsorber, taking into consideration the geometry of the heat exchanger and the diffusion in the adsorbent medium. In this model, an increment in the direction of heating/cooling fluid containing one (or more) fin pitch was considered. In the domain of this increment, the distribution of the adsorbent temperature was evaluated in the radial and axial directions. The model is validated by experimental data of different adsorbents, volume flows and temperatures. All data were collected in an adsorber test-bench at Fraunhofer ISE. As a result it was found a coefficient of multiple determination of around 0.94 and an error of 20% during the transient. Finally, the model was run on typical operation for cooling application and compared with data from literature. A numerical energy balance was also evaluated. In summary, the results obtained by a independent group shows the same range of confidence of the model developed here, in terms of the tendency of the curves and even the absolute values.

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1. Introduction

Adsorption chillers have been studied as a promising solution for a more environmentally friendly refrigeration system. However, the large system dimensions are the biggest limitation of this technology [1]. Optimisation of such systems can be done using mathematical models of the heat and mass transfer. Thus, the behaviour of different chillers can be evaluated and different designs can be compared with the help of dynamic simulations [2,3]. An overview of the state-of-the-art of mathematical modeling of adsorbers was done by Yong and Sumathy [4]. Some models consider uniform temperature and pressure, others take into account both heat and mass transfer resistances. The three models most relevant for this paper are briefly described below.

Hajji and Khalloufi [5] presented a theoretical and experimental analysis of a constant-pressure adsorption process. The governing heat and mass transfer equations derived from local thermodynamic equilibrium and the energy balance are solved numerically by finite difference method. They concluded that the effect of the adsorbent thickness and the heat-transfer coefficient between the adsorbent and the heating/cooling fluid have the strongest influence on sorption kinetics and on the cooling capacity of adsorption systems. Latter, they used also a finned structure [6].

Chua et al. [7] developed a two-dimensional (2D) transient model for a two-bed silica gel–water adsorption chiller, where the annular-fins are also considered. They have utilised the finned-tube heat exchanger as adsorptive bed. The adsorption equations consider the heat of adsorption constant and the kinetic is based on mass transfer resistance within the silica gel, which is a function of the equilibrium uptake, the uptake itself and temperature of the adsorbent. Although the finite difference equations were solved using only three nodes in the radial direction and ten in the axial, the simulation results showed a good agreement to the experimental data.

Another 2D model is presented by Maggio et al. [8]. It describes an adsorption cooling machine with internal heat recovery using a double consolidated adsorbent bed, where tubes are coated by a slurry of zeolite powder and inorganic binder. The governing equations describe the energy balance of all components and the mass transfer balance related to the adsorbate gaseous phase through the pores, which is considered mandatory in simulating consolidated configurations. The vapour diffusive velocity is determined by the Ergun's equation and the adsorption enthalpy is not considered constant, but dependant on the uptake. Additional details, as for the adsorbent/adsorbate equilibrium, can be found in their previous work [9].

This paper presents a heat and mass transfer simulation of the adsorber, taking into consideration the geometry of the heat exchanger and the diffusion in the adsorbent medium. In this model,

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Nomenclature

A	change in Gibbs' free energy (J/kg)
a	MDR fitting parameter
b	auxiliary function given by (11)
Cp	specific heat capacity (J/kg K)
d	diameter (m)
E	relative error (%)
h	heat-transfer coefficient (W/m^2 K)
L	total length of the tubes (m)
M	molar mass of adsorbate (kg/mol)
n	number (quantity)
P	pressure (Pa)
q_{ad}	heat of adsorption (J/kg)
q_c	heat of condensation of the adsorbate (J/kg)
R	gas constant, thermal contact resistance (J/mol K, (m^2 K/W))
r	radial direction (m)
s	fin spacing (m)
T	temperature (K)
t	time (s)
v	volume flow rate (m^3/s)
W	adsorption capacity (m^3/kg)
w	adsorption capacity – adsorbate volume per adsorbent mass (m^3/kg)
x	axial direction (m)
z	axial direction in the adsorbent (m)

Greek symbols

ΔS	differential molar entropy (J/kg K)
α	coefficient of thermal expansion of the adsorbate (1/K)
δ	fin thickness (m)
λ	thermal conductivity (W/m K)
ρ	density (kg/m^3)

Subscripts

a	adsorbate (working fluid)
ch	adsorber chamber
f	fin
i	inner
o	outer
s	solid adsorbent
sat	saturated
t	tube
w	water (heat-transport fluid)

an increment in the x direction (direction of heating/cooling fluid) containing one (or more) fin pitch was considered. In the domain of this increment, the distribution of the adsorbent temperature was evaluated in the r and z directions.

2. Mathematical model

The thermal behaviour of the adsorber is strongly dependent on the conductivity of the adsorbent and the heat-transfer coefficient between the metal and the solid adsorbent. Therefore, the finned-tube heat exchanger was used to increase the heat-transfer area. For the geometric model of this heat exchanger, a simplification based on annular-fins was made, as can be seen in Fig. 1, with the following assumptions:

1. the heat transfer in the adsorbent medium is two-dimensional (axial and radial);
2. the thermal gradient in radial direction is neglected both for water flow and tubes ($r_i \ll L$);
3. the fin thickness is small enough to consider the heat transfer only in radial direction;
4. a thermal contact resistance is considered at the interface tube/fin;
5. the convective effects and pressure drops are negligible in the adsorber chamber;
6. no heat losses to the ambient are considered due to the adsorber is inside a vacuum chamber, where the heat transfer to the ambient is restricted to radiation. As the temperature is close to the ambient, this heat loss can be neglected. However, for a chiller simulation, heat losses are considered;
7. regarding thermal conductivity, the adsorbent is treated as a continuous medium;
8. all adsorbent particles have the same properties (including shape and size); they are uniformly distributed throughout the adsorbent, and in local thermal equilibrium with the adsorbate and the surrounding gaseous phase;
9. the gaseous phase behaves as an ideal gas;
10. the properties of the heat-transfer fluid and the adsorbate are considered as temperature dependent.

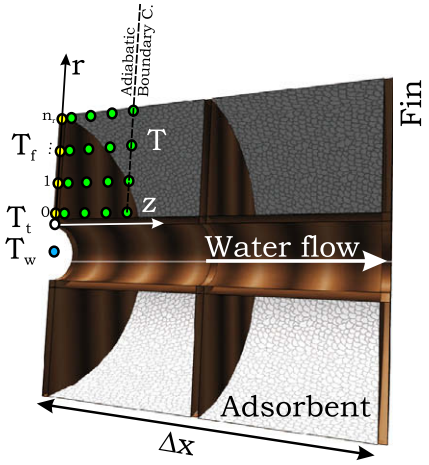

Fig. 1. Schematic of the adsorber control-volume.

The simulation is divided in n_x nodes that contain n_f fins, as shown in Fig. 1. It is calculated for each node one mean temperature for the water flow T_w, another for the tube T_t, n_r points for the fin and a thermal gradient in 2D for the adsorbent as the flowchart of Fig. 2. The model take into consideration the thermal contact resistance between the fin and the tube. It can simulate fins smelted, pressed or even with a air-gap. The chamber of the adsorber is under vacuum and it is big enough to make the pressure drop during adsorption close to zero. This was also the reason to not consider heat losses to the ambient. As pellets are often used as adsorbent, assumption (7) and (8) are about the treatment of it as an unique solid in terms of heat conduction and diffusion.

2.1. Adsorption equations

Starting on the rigorous thermodynamic expressions, the heat of adsorption is calculated by the Gibbs-Helmholtz equation:

$$q_{ad} = q_c + A - T_s\,\Delta S \tag{1}$$

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