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Realistic minimum desorption temperatures and compressor sizing for activated carbon + HFC 134a adsorption coolers

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HIGHLIGHTS

► Evaluation of realistically minimum desorption temperatures.
► Design of adsorption compressors.
► Activated carbon + HFC 134a system.
► Validation of the model through experimental data.
► Identification of critical processes.

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ABSTRACT

A low thermal diffusivity of adsorption beds induces a large thermal gradient across cylindrical adsorbers used in adsorption cooling cycles. This reduces the concentration difference across which a thermal compressor operates. Slow adsorption kinetics in conjunction with the void volume effect further diminishes throughputs from those adsorption thermal compressors. The problem can be partially alleviated by increasing the desorption temperatures. The theme of this paper is the determination the minimum desorption temperature required for a given set of evaporating/condensing temperatures for an activated carbon + HFC 134a adsorption cooler. The calculation scheme is validated from experimental data. Results from a parametric analysis covering a range of evaporating/condensing/desorption temperatures are presented. It is found that the overall uptake efficiency and Carnot COP characterize these bounds. A design methodology for adsorber sizing is evolved.

1. Introduction

Sorption refrigeration cycles are construed to be one of the means of waste heat recovery. Among them solid sorption cycles have the benefits of dispensing with solution heat exchangers and solution pumps. Solid sorption cycles based on silica gel, zeolite and activated carbon as adsorbents and water, alcohols, ammonia, carbon dioxide and HFC refrigerants as adsorbates have been investigated extensively in the literature [1–4]. Industry generally prefers operation of refrigeration cycles under pressures above but close to atmospheric pressures. Although, much has been said about the positive aspects of adsorption cooling, seldom a realistic appreciation of thermal exigencies has been provided. Fig. 1 shows a schematic diagram of a typical adsorption cooler. Saha et al. [5] derive conditions of minimum desorption temperature for a few adsorbent + refrigerant combinations based on the assumptions of equilibrium conditions prevailing in the adsorption beds, no thermal gradients between the heating medium and the core of the adsorption bed and the absence of void volume effect. Saha et al. [6] expand that approach to multistage thermal compression which further reduces the temperature at which the heat source should be. Banker et al. [7] have shown that the core of a cylindrical adsorber never reaches the heating medium temperature within finite cycle times that are practical. As a result, adsorption occurs at

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Nomenclature

C	uptake, specific adsorptance, concentration (g g^{-1})
COP	coefficient of performance
c_p	specific heat (J kg^{-1} K^{-1})
E	characteristic energy of the adsorption system (J mol^{-1})
Fo	Fourier number
h	enthalpy (kJ kg^{-1})
Δh_{st}	isosteric heat of adsorption (J kg^{-1})
m	mass (kg)
n	structural heterogeneity parameter in Dubinin–Astakhov equation
p	pressure (bar)
Q	refrigeration load (W)
R	gas constant only in Eq (18) (J mol^{-1} K^{-1})
R	radius of adsorber (m)
r	radial coordinate
T	temperature (°C)
t	temperature (°C)
v	specific volume (cm^3 g^{-1})
W_0	limiting volume of adsorption space of the adsorbent (cm^3 g^{-1})
α	thermal diffusivity (m^2 s^{-1})
β	coefficient of thermal expansion (K^{-1})
k	thermal conductivity (W m^{-1} K^{-1})
ρ	density (kg m^{-3})
η	efficiency
τ	time (s)

Superscripts

$'$, $''$	non-ideal operating conditions

Subscripts

a	adsorbed phase
ad	adsorption
av	average
a, b, c, d	states of ideal adsorption compression cycle
b	normal boiling point
c	cooling
ch	activated carbon
con	condensing
des	desorption
ev	evaporator
f	saturated liquid
fg	vaporization
g	vapor phase of refrigerant
h	heating
op	operating
ref	refrigerant
s	saturation
u	uptake

a temperature higher than the purported adsorption temperature and desorption occurs at a lower temperature. Fig. 2 illustrates the differences between ideal and real cycles on pressure–concentration–temperature plane. Here a–b–c–d is an ideal adsorption cycle which shrinks to a'–b'–c'–d' because of the differences between the cooling/heating media and the mean temperature of the adsorber. It gets further modified as to a''–b'–c''–d' due to void volume effect [8]. The solution of Saha et al. [6] is based on a temperature at which $C_b = C_d$. Srinivasan et al. [9] introduce the concept of uptake efficiency (similar to volumetric efficiency of a positive displacement compressor) which is the ratio of actual to ideal uptake difference across which the adsorber operates. Thus, with reference to Fig. 2, the overall uptake efficiency can be defined as

$$\eta_{\text{u-overall}} = \frac{C_{b'} - C_{a'}}{C_b - C_a} \tag{1}$$

Banker [10] has shown that the measured overall uptake efficiency is only of the order of 20–40% for the case of activated carbon + HFC 134a experimental heat recovery cooler. A designer has the input data of required evaporating temperature for a given ambient condition (which dictates the adsorption and condensing temperatures). To weigh the potential of a thermally driven solid sorption cooler as an effective waste heat recovery device, it is imminent to realistically assess what the minimum temperature of the heat source should be that will drive the adsorption cooler. This paper attempts to provide a criterion that links the Carnot COP and the overall uptake efficiency. A practical design approach is also suggested.

2. Formulation of the problem

The requirement of the temperature at which refrigeration is required (t_{ev}), the cooling load (Q) and the temperature at which heat rejection occurs (t_{ad} and t_{con}) are the primary inputs. The last parameter is governed, broadly, by the local ambient conditions. In the case of adsorption refrigeration cycles, though heat rejection occurs in the adsorber and the condenser, invariably the ambient forms the heat sink and hence in further analysis the adsorption and condensing temperatures are taken to be the same [11]. Thus, with reference to Fig. 2, p_b (saturation pressure of the refrigerant corresponding to t_{ev}), t_b, the temperature of the cooling medium and the condensing pressure ($p_{con} = p_d$) which is the saturation pressure of the refrigerant at the condensing temperature, are defined. We designate t_b ($=t_{ad}$) and t_d ($=t_{des}$) as notional adsorption and desorption temperatures. Srinivasan et al. [9] proposed that the uptake efficiency of the compressor should be at least 77%, that is

$$\eta_u = \frac{C_{b'} - C_{a''}}{C_{b'} - C_{a'}} > 0.77 \tag{2}$$

It is apparent that $\eta_{\text{u-overall}}$ will be much smaller than 77%. The problem posed is what should be the desorption temperature (t_d) for a given $\eta_{\text{u-overall}}$ and what is the possible range of the latter?

It is assumed that the heating and cooling of the adsorber occur on its outer surface. This is a meaningful proposition because for small adsorbers it is quite difficult to incorporate a heat exchanger inside them. The casing of the adsorber is assumed to be isothermal at the temperature of the heating/cooling medium. This is a logical assumption since the wall thickness required to withstand low operating pressures of metallic housings of the adsorber can be very small. Further, the thermal diffusivity of the container is quite large compared to that of its contents. The one dimensional unsteady radial heat conduction equation given below

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = \frac{1}{\alpha}\frac{\partial T}{\partial \tau} \tag{3}$$

needs to be solved subject to the following initial condition of $T(r,0) = t_b$ and boundary conditions of $T(R,t) = t_d$; $\partial T(t)/\partial r|_{r=0} = 0$ for the heating phase (process b'–d' in Fig. 2) with subscripts b and