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Study on a solar heat driven dual-mode adsorption chiller

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Environmental concerns and the rising energy cost necessitate looking for renewable energy driven environmentally benign adsorption cooling systems. Solar powered adsorption chillers with non-concentrating flat plate or evacuated tube collectors face the problem of not getting adequate driving source temperature during some months of the year. Multi-staging of the adsorption cycle is then needed to exploit the low driving source temperature. A simulation study of a solar thermal driven dual-mode, four-bed silica gel–water adsorption chiller is undertaken in this work. The solar thermal collector data of Durgapur (23.48 °N, 87.32 °E), India has been used as the heat source for the dual-mode chiller. For a driving source temperature above 60 °C, the chiller works as a single stage four-bed adsorption chiller; while the chiller functions as a two stage adsorption chiller when the driving source temperature falls below 60 °C. With a cooling water temperature of 30 °C, this two stage chiller has been found to produce cooling effect with a driving source temperature as low as 40 °C. Results indicate that the dual-mode chiller is capable of providing cooling throughout the year under the climatic condition of Durgapur, India.

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1. Introduction

Environmental concerns and the rising energy costs necessitate looking for renewable energy driven environmentally benign cooling systems to meet the ever-increasing demand of refrigeration and air-conditioning. Heat driven sorption based cooling systems are gradually emerging as environmentally friendly alternatives to conventional vapor compression based refrigeration cycles. Since the detection of ozone holes in the stratosphere, caused by the CFCs (chlorofluorocarbons) and HCFCs (hydrochlorofluorocarbons) of vapor compression based coolers, interest on adsorption cooling systems have increased a lot [1]. Thermally powered adsorption based chillers have zero ozone depletion potential due to the use of natural refrigerants like water, methanol, ethanol, ammonia, etc. As solar energy or low grade waste heat can be effectively utilised to run these systems, it has the potential to reduce the peak load demand of electricity and the emission of greenhouse gases as well. Besides, the maintenance requirements of adsorption systems are very less due to the absence of rotating parts in the main unit and they do not suffer from corrosion problems also, as in absorption systems. Examples of various adsorption cooling cycles using different adsorbent-refrigerant pairs may be found in the Refs. [2–16].

Solar adsorption refrigeration system seems to be a promising alternative refrigeration device since cooling load of buildings is roughly in phase with solar energy availability. During the past decades much research has been accomplished on solar adsorption refrigeration. Zhai et al. [17] designed a solar-powered air-conditioning system which consisted of a 150 m^2 solar collector and two adsorption chillers with a capacity of 8.5 kW each. The performance of the system under the typical weather condition of Shanghai, China was studied and a daily average solar COP (coefficient of performance) of 0.15 and system COP of 0.35 were obtained. Stitou et al. [18] did experimental investigation on a solar powered thermochemical air-conditioning unit using barium chloride and ammonia pair. The plant used 21.6 m^2 of standard flat plate collector and it could generate a daily cooling of 20 kWh at 4 °C. Pons and Guilleminot [19] worked on an activated carbon–methanol based solar adsorption refrigeration system to produce ice. Ice at −3 °C was produced at the rate of about 6 kg per m^2 of collector area with a net solar COP of 0.12 and a cycle COP of 0.43. Sumathy et al. [20] did a lumped parameter analysis of an adsorption air-conditioning

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Nomenclature

Symbols

α_0	coefficient in Eq. (1), kg (kg of dry adsorbent)$^{-1}$
α_1	coefficient in Eq. (1), kg (kg of dry adsorbent, K)$^{-1}$
α_2	coefficient in Eq. (1), kg (kg of dry adsorbent, K^2)$^{-1}$
α_3	coefficient in Eq. (1), kg (kg of dry adsorbent, K^3)$^{-1}$
β_0	coefficient in Eq. (1), (–)
β_1	coefficient in Eq. (1), K^{-1}
β_2	coefficient in Eq. (1), K^{-2}
β_3	coefficient in Eq. (1), K^{-3}
A	area, m^2
COP	coefficient of performance
C_p	specific heat capacity, J kg^{-1} K^{-1}
D_s	surface diffusion coefficient, m^2 s^{-1}
D_{so}	pre–exponential constant, m^2 s^{-1}
E_a	activation energy, J kg^{-1}
h	enthalpy, J kg^{-1}
ΔH_{st}	isosteric heat of adsorption/desorption, J kg^{-1}
$k_s a_v$	mass transfer coefficient, s^{-1}
m	mass, kg
$\dot{m}$	mass flow rate kg s^{-1}
P	pressure, Pa
P_s	saturation pressure, Pa
q^*	equilibrium uptake, kg kg^{-1}
q	refrigerant amount adsorbed, kg kg^{-1}
Q_{chill}	cooling output, kW
Q_{hot}	driving heat, kW
R	gas constant, J kg^{-1} K^{-1}
R_p	average radius of adsorbent particle, m
SCP	specific cooling power, W kg^{-1}
T	temperature, K
t	time, s
U	heat transfer coefficient, W m^{-2} K^{-1}

Subscripts

ads	adsorber/adsorbent
bed	adsorbent bed
chill	chilled
cond	condenser
des	desorber
eva	evaporator
hot	hot water
in	inlet
out	outlet
ref	refrigerant
s	saturation
sg	silica gel
w	water

Superscript

ref	refrigerant

system driven by simple flat-plate solar collectors and found optimum values of adsorbent mass and collector area to get a maximum SCP of 150 W m^{-2}. Leite and Daguenet [21] studied the influence of solar collector TIM (transparent insulation material) cover on the performance of an adsorption ice maker in a hot and humid climate. The ice production was found to be 100% more than that obtained for a single cover solar system. El Fadar et al. [22] studied an activated carbon–ammonia based adsorption refrigeration system powered by parabolic trough solar collector coupled with a heat pipe. Use of concentrated collector and high flux density heat pipe made the system more compact and lighter compared to systems with flat plate or evacuated tube collectors. Optimum values of adsorbent mass and aperture width (collector area) were investigated to get maximum COP. Habib et al. [23] performed a theoretical analysis of a solar thermal driven combined adsorption refrigeration system suitable for tropical climates. The cycle combines an activated carbon-R134a cycle with an activated carbon-R507A bottoming cycle and it can generate cooling at as low as −10 °C. Zhai and Wang [24] compared adsorption cooling system in terms of heat storage. They observed that the system without heat storage had better collector efficiency as well as electrical COP compared to the system with heat storage. However, inlet and outlet temperatures of the collector array and the chiller had more fluctuations in the case without heat storage. Louajari et al. [25] studied the influence of externally finned cylindrical adsorber tubes on the performance of a solar adsorption refrigerating machine. An optimal ratio of fin width to tube diameter along with an optimal diameter of adsorber tubes to get the best solar COP has been investigated in this work. Ogueke and Anyanwu [26] studied the effects of different design variables, like collector tube spacing, number of layers of glazing material, adsorbent tube outside diameter, adsorbent packing density and thermal conductivity, etc., on the performance of a solar adsorption refrigerator. Simulation run with the optimal values, obtained through a nonlinear optimization code, showed significant improvements in refrigerator performance. The adsorbent thermal conductivity played an important role in the refrigerator performance, the authors remarked. Dai and Sumathy [27] modeled the performance of an activated carbon–methanol pair solar adsorption cooling system with tubular adsorber in evacuated glass tube insulation. The simulated result did not find any large pressure difference in the 80 mm diameter adsorber tubes during the heating period. Anyanwu and Ogueke [28] did a finite element based transient analysis of a solar-powered adsorption refrigerator using the adsorbent-refrigerant pair of activated carbon–methanol. They found out numerically that the performance of the refrigerator improved with the decrease in condenser pressure and increase in evaporator pressure. Hassan et al. [29] simulated the performance of a tubular adsorption refrigeration system with real time variations in ambient temperature and solar radiation along the day. The pressure distribution across the bed was found to be nearly uniform, only varying with time and no spatial or temporal variation of the thermal conductivity of the activated carbon–methanol bed was observed. Saha el al [30]. proposed a two-stage solar driven silica gel–water adsorption chiller which could be run with 55 °C driving heat source temperature in combination with 30 °C coolant temperature.

One of the problems of solar adsorption chillers is that the chiller cannot be in phase with solar insolation along the whole year, due to low temperature achieved in a number of months. To overcome this problem, Saha et al. [31,32] introduced a dual-mode multi-bed silica gel–water adsorption chiller driven by low grade waste heat which could operate within a temperature range of 40–95 °C. When the driving source temperature is 60–95 °C, the chiller works as a single stage conventional chiller. This novel system functions as a two stage adsorption chiller for a driving source temperature below 60 °C. With a cooling water temperature of 30 °C, this two stage chiller has been found to produce cooling effect with a driving source temperature as low as 40 °C. As a result of this large operating temperature range, this innovative chiller is able to generate cooling throughout the year.

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