

Article

Adsorption Cooler Design, Dynamic Modeling, and Performance Analyses

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Abstract: This paper presents an adsorption cooler (AC) driven by the surplus heat of a solar thermal domestic hot water system to provide cooling to residential buildings. A cylindrical tube adsorber using granular silica gel as adsorbent and water as adsorbate was considered. The AC was modelled using a two-dimensional distributed parameter model implemented in previous adsorption heating and cooling studies. The performance coefficients of the resultant thermally driven cooling system were obtained for a broad range of working conditions. The thermally driven AC was found to have coefficient of performance (COP) of 0.5 and a specific cooling power (SCP) of $44 \text{ W} \cdot \text{kg}^{-1}$ when considering condenser, evaporator, and regeneration temperatures of 30°C , 15°C , and 70°C , respectively. Moreover, the results showed that the AC could be used for refrigeration purposes at temperatures as low as 2°C and that it could also operate during hotter days under temperatures of 42°C .

Keywords: adsorption cooler; thermally driven cooling; solar heat surplus; air cooling; refrigeration



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1. Introduction

Heating and cooling stand as the biggest energy end-use sector in Europe, accounting for 51% of the EU's total energy consumption in 2019, with approximately 75% being obtained from burning fossil fuels [1]. During recent years, the focus is on including renewables in the energy sector and increasing its share. However, considering residential and industrial buildings and processes, the heating and cooling sector is more harmful to the environment than the energy sector itself. More effort must be made in diminishing the negative impacts of thermal energy needs and in moving towards the achievement of the key energy goals set for the next decade through the introduction of new technologies and the improvement of those that are already available. Given that adsorption cooling (AC) can be driven by renewable energy sources and waste heat, this technology can play a significant role in the achievement of these goals. Unlike common vapour compression refrigerators and coolers, AC can work with zero global warming potential (GWP) refrigerants such as water, which is environmentally friendly, abundant in nature, and causes no harm in case of leakages. The low COP is usually pointed out as the major disadvantage of AC technologies. However, ACs are directly driven by thermal energy, meaning that no energy is lost in producing electricity and compensating the heat to work energy conversion efficiency. Due to this, a fairer comparison criterion is required than just comparing the numerical values of adsorption and vapour compression systems' COPs [2]. Moreover, when using thermal energy that is freely obtained, the coefficient of performance (COP) loses its importance. Developing systems that can use waste heat to regenerate AC would lead to the elimination of their major disadvantage; that is, the low COP. However, if the thermal to electric energy conversion efficiency is considered ($\eta \approx 40\%$ in 2019), the COP for vapour compression systems is much lower than commonly reported, which means that AC systems' COP is not a real disadvantage. It is common for the need for cooling to occur when excess heat

is available, such as solar heat, ambient heat, waste heat from machine operation, and industrial processes. Using the surplus of thermal energy, it is possible to thermally drive ACs. Unlike conventional refrigerators and coolers that require electricity to function, ACs can work with excess thermal energy, resorting to just tiny batteries to power the control electronics and drive small circulation pumps.

The European Union has recently reinforced support for heat pump technologies, and it is important to make use of the financial incentives, decided on by policy makers, to carry out further research on AHP technology to minimize its well-identified disadvantages and improve its technology readiness level (TRL). Several studies on adsorption systems have been carried out. These studies are commonly focused on five aspects of adsorption systems: selection of adsorbent–adsorbate working pairs (e.g., [3,4]), development of new adsorbent materials (e.g., [5–9]), adsorbent materials' physical and thermodynamic properties (e.g., [10–12]), adsorber configurations (e.g., [13–16]), and operating conditions (e.g., [17–19]). The capacity for the regeneration of adsorbent material using heat captured by solar thermal collectors is very promising. A comprehensive review of solar adsorption refrigeration systems was presented by Fernandes et al. [20]. Since solar thermal collectors can provide temperatures around 70 °C, adsorbents that can be regenerated at this temperature must be used, silica gel being the most common due to its availability, low cost, and mechanical stability.

Improving the overall performance of adsorption cooling systems requires detailed numerical models capable of identifying the optimal values for the several governing parameters, configurations, and designs, which requires considering real-scale systems. Due to the increase in computational capacities, it is now possible to include a higher level of detail in numerical models, resulting in more detailed and accurate analyses of adsorption cooling systems. This work presents a thermally driven cooling system that uses the heat surplus obtained from solar thermal panels during warm seasons to drive an AC for space cooling. In this study, an adsorber that has been used in previous works was filled with granular silica gel. The AC was modelled using a highly detailed two-dimensional distributed parameter model that has been used and validated in previous adsorption heating and cooling work [21–23], which is here adapted to the granular adsorbent. Several simulations were carried out to analyse the impact of the evaporator, condenser, and regeneration temperatures, as well as the cycle time, on the AC's performance. It should be noted that the same base adsorption system used for cooling in summer can be used for heating purposes in winter, simply requiring another heat source, which can also be a renewable heat source.

2. Materials and Methods

2.1. Design

The core of any AC is the adsorber, which can be designed in several configurations. There are numerous review papers available in the literature that analyse the different adsorber configurations and adsorbent packing techniques [24]. The most common designs are the shell-packing, plate-finned, tube, finned-tube, fin-plate, flat-pipe, spiral-plate, coated-tube, and coated-plate designs [25]. Coatings have been considered the most promising packing technique due to their inherent high heat transfer coefficients and compact sizes. However, it is more difficult to build real-scale prototypes using adsorbent coatings and the manufacturing costs are much higher than solutions with granular adsorbent materials. This work considered a cylindrical tube adsorber configuration using a granular adsorbent. Silica gel–water was selected as the adsorbent–adsorbate working pair, given that its regeneration can be achieved at temperatures that can be provided by solar thermal panels and its properties are well-studied and widely reported in the literature. Granular silica gel RD2060 produced by Fuji Silysia Chemical LTD was considered in this study. The adsorber consisted of fifty individual metal tubes packed with silica gel on their outer surfaces. The tubes were enclosed in a vacuum chamber that was alternatively opened to the evaporator or to the condenser, allowing water vapour exchanges. The energy exchanges

required to drive the adsorption/regeneration processes were carried out through a heat transfer fluid (HTF) that was circulated inside the tubes. The adsorber and its schematic are depicted in Figure 1.

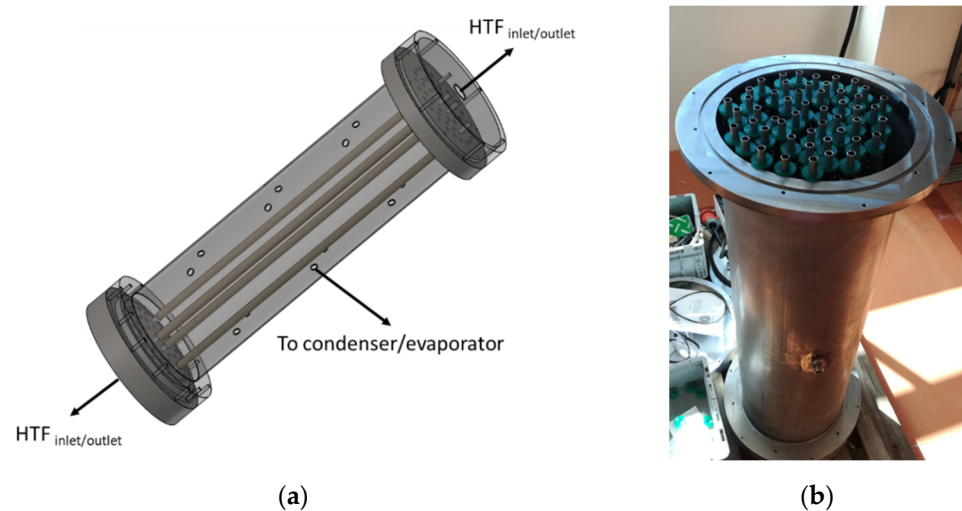


Figure 1. (a) Adsorber schematic; (b) adsorber with fifty tubes covered with silica gel.

2.2. Modeling

Several different physical models have been used to model adsorption heat pumps and refrigerators. An extensive review can be found in [26]. Three types of models are normally considered: thermodynamic models, lumped parameter models, and distributed parameter models. To accurately simulate an adsorption device, considering all the dynamics, distributed parameter models are mandatory. Reference [21] presents an extensive analysis of the physical models for an AHP, comparing different types of models and considering several dimensions. It concludes that a 2D distributed parameter model must be used to accurately simulate an AHP system based on the cylindrical tube design. Therefore, the 2D distributed parameter model was used to describe the AC dynamics and estimate its performance in this work.

To model the AC adsorber, the energy and mass balance equations (Equations (1) and (2), respectively) were applied to the cylindrical tube geometry:

$$\frac{\partial(\rho C_p T_s)}{\partial t} + \nabla(\rho_v C_{p,v} T_s u) - \nabla(k_s \nabla T_s) - \rho_s(1 - \varepsilon) \Delta H_{ads} \frac{\partial X}{\partial t} = 0, \quad (1)$$

$$\varepsilon \frac{\partial \rho_v}{\partial t} + \nabla(\rho_v u) + \rho_s(1 - \varepsilon) \frac{\partial X}{\partial t} = 0, \quad (2)$$

where

$$\rho C_p = \varepsilon \rho_v C_{p,v} + \rho_s(1 - \varepsilon)(C_s + X C_{p,a}) \quad (3)$$

and

$$\varepsilon = \varepsilon_{bed} + (1 - \varepsilon_{bed}) \varepsilon_p, \quad (4)$$

with ε being the total porosity, ε_p the particle porosity, and ΔH_{ads} the heat of adsorption. The subscripts s and v stand for adsorbent and vapour, respectively.

The momentum balance is expressed by Darcy's Law:

$$u = -\frac{k_D}{\mu} \nabla P \quad (5)$$

The adsorbent bed permeability is obtained with the Blake–Kozeny model [27]:

$$k_D = \frac{d_p^2 \varepsilon_{bed}^3}{150(1 - \varepsilon_{bed})^2} \quad (6)$$

To obtain the adsorbate uptake, the linear driving force (LDF) model can be applied [28]:

$$\frac{dX}{dt} = K_{LDF}(X_{eq} - X) \quad (7)$$

The K_{LDF} coefficient can be evaluated as follows:

$$K_{LDF} = \frac{15D_{ef0}e^{-\frac{E_a}{R'T_s}}}{R_p^2} \quad (8)$$

The equilibrium adsorbate content in the adsorbent for the silica gel–water pair can be calculated as follows:

$$X_{eq} = \frac{Pk_0 e^{\frac{\Delta H_{ads}}{R'T_s}}}{\left[1 + \left(\frac{Pk_0}{q_m} e^{\frac{\Delta H_{ads}}{R'T_s}}\right)^{t_{SG}}\right]^{\frac{1}{t_{SG}}}} \quad (9)$$

The following suppositions were assumed:

- The adsorbent bed is homogenous;
- The evaporator and the condenser are ideal heat exchangers with uniform pressures;
- The adsorbate vapour phase behaves as an ideal gas and the adsorbed phase is in the liquid state;
- The specific heats of the adsorbate vapour and liquid phases are constants;
- The adsorbate vapour around the adsorbent is saturated vapour;
- The thermophysical properties of solid materials do not change with temperature;
- The temperature, adsorbent content, and pressure in the adsorbent bed do not vary along the angular direction.

The energy conservation equation applied to the adsorbent bed can be written as:

$$\begin{aligned} [\rho_s(1-\varepsilon)(C_s + XC_l) + \varepsilon\rho_v C_{pv}] \frac{dT_s}{dt} &= (1-\varepsilon)\rho_s \Delta H_{ads} \frac{dX}{dt} + \frac{k_s}{r} \left(\frac{\partial T_s}{\partial r} + r \frac{\partial^2 T_s}{\partial r^2} \right) \\ &- \frac{C_{pv}}{r} \left(\rho_v T_s u_r + r T_s u_r \frac{\partial \rho_v}{\partial r} + r \rho_v u_r \frac{\partial T_s}{\partial r} + r \rho_v T_s \frac{\partial u_r}{\partial r} \right) \\ &- C_{pv} \left(T_s u_z \frac{\partial \rho_v}{\partial z} + \rho_v T_s \frac{\partial u_z}{\partial z} + \rho_v u_z \frac{\partial T_s}{\partial z} \right) + k_s \left(\frac{\partial^2 T_s}{\partial z^2} \right) \end{aligned} \quad (10)$$

The mass conservation for the adsorbent bed can be obtained as follows:

$$\frac{\partial \rho_v}{\partial t} = -\frac{1}{\varepsilon} \left[\rho_s(1-\varepsilon) \Delta H_{ads} \frac{\partial X}{\partial t} + \frac{1}{r} \left(r u_r \frac{\partial \rho_v}{\partial r} + \rho_v r \frac{\partial u_r}{\partial r} + \rho_v u_r \right) + \rho_v \frac{\partial u_z}{\partial z} + u_z \frac{\partial \rho_v}{\partial z} \right] \quad (11)$$

Considering that $r_{tube} \ll L_{tube}$, the variation in the HTF temperature along the radial direction can be disregarded, $\frac{\partial T_f}{\partial r} \approx 0$. Since the HTF velocity is constant, the energy balance equation for the HTF can be written as:

$$\rho_f C_{p,f} \frac{\partial T_f}{\partial t} = k_f \left(\frac{\partial^2 T_f}{\partial z^2} \right) - u_f \rho_f C_{p,f} \frac{\partial T_f}{\partial z} + \frac{4h_{f \rightarrow m}}{d_{in}} (T_m - T_f) \quad (12)$$

The interaction between the adsorbent bed and the HTF occurs through a metal tube. Since the tube thickness is very small and due to the inherent high thermal conductivity of metals, the temperature of the metal tube along the radial direction was considered constant. The energy balance equation for the metal tube was obtained as follows:

$$\rho_m C_m \frac{dT_m}{dt} = k_m \left(\frac{\partial^2 T_m}{\partial z^2} \right) + \frac{4d_{in} h_{f \rightarrow m} (T_f - T_m)}{d_{out}^2 - d_{in}^2} + \frac{4d_{out} h_{m \rightarrow s} (T_s - T_m)}{d_{out}^2 - d_{in}^2} \quad (13)$$

The set of partial differential equations were solved by using the method of lines. The resultant set of ordinary differential equations was solved using the adaptive timestep solver ode15s in Matlab R2020b, adequate for solving stiff ordinary differential equations [29]. More details on the modelling of the HTF and the metal tube, as well as the initial and boundary conditions, can be found in references [21–23].

3. Thermally Driven Cooling Performance and Dynamics

A heat surplus in domestic hot water storage tanks frequently exists when it is necessary to cool down households. This work proposes to use the heat surplus produced by domestic solar thermal panels, which has the main purpose of providing domestic hot water, to drive an AC that can remove heat from a residential building. A basic scheme of such a system is represented in Figure 2. The ideal thermodynamic cycle of the AC is presented in Figure 3.

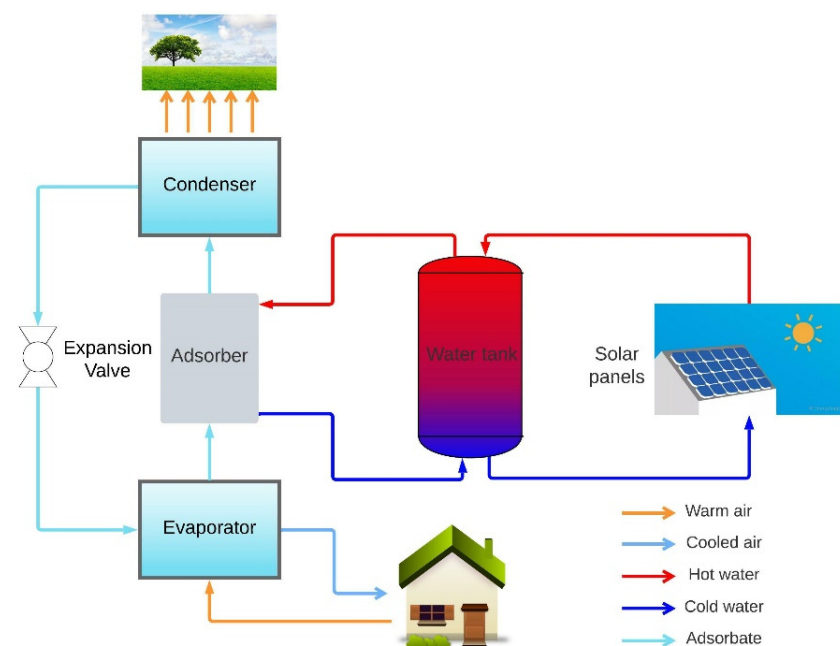


Figure 2. Schematic of an AC driven by solar thermal heat used to remove heat from a residential building.

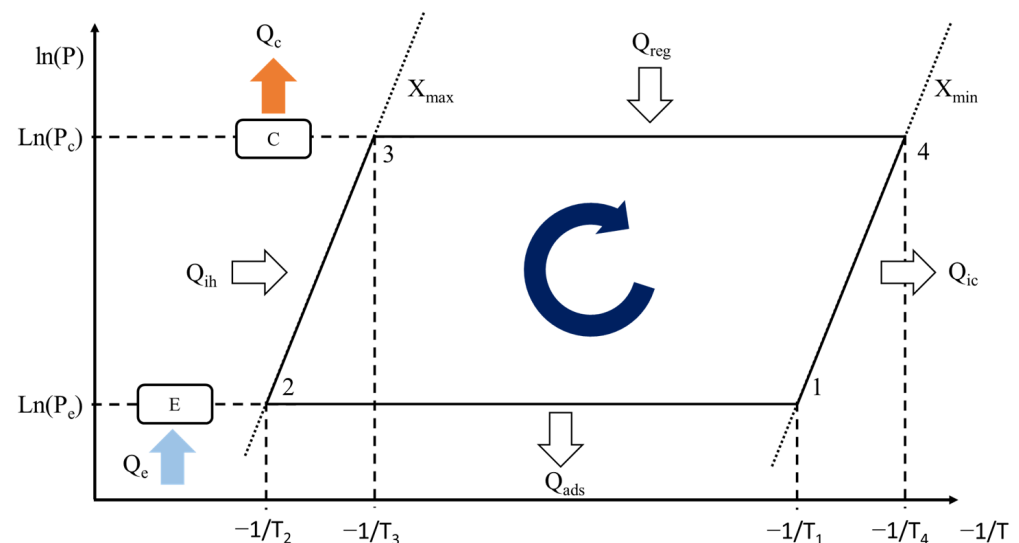


Figure 3. Ideal thermodynamic cycle of the AC.

The performance of an AC is defined by its cooling COP and specific cooling power (SCP), which can be obtained using Equations (10) and (11), respectively:

$$\text{COP} = \frac{Q_e}{Q_{ih} + Q_{reg}} \quad (14)$$

$$\text{SCP} = \frac{Q_e}{m_s \tau_{cyc}} \quad (15)$$

Heat is provided to the adsorber during the isosteric heating and regeneration phases. The isosteric heat can be obtained as follows:

$$Q_{ih} \approx \int_{T_{initial}}^{T_{final}} [m_s (C_s + X_{max} C_{p,a}) + m_m C_m] dT \quad (16)$$

The heat provided during the regeneration phase can be calculated as follows:

$$Q_{reg} \approx \int_{T_{initial}}^{T_{final}} [m_s (C_s + X C_{p,a}) + m_m C_m] dT - \int_{X_{max}}^{X_{min}} m_s \Delta H_{ads} dX \quad (17)$$

The heat captured by the evaporator, corresponding to the cooling effect, can be obtained as follows:

$$Q_e = m_s \Delta X \Delta H_v + \int_{T_c}^{T_e} m_s \Delta X C_{p,a} dT \quad (18)$$

Table 1 contains the main parameters used in the simulations, which were selected with the aim of describing the use of the adsorber in this work to remove heat from a resident building, as shown in Figure 2. The selected temperatures corresponded to a warm summer day in the region of Aveiro, Portugal. The condenser was outside at an ambient temperature of 30 °C. The evaporator temperature was set to 15 °C, which may be considerably low for space cooling but takes into account that the air increases its temperature along the path from the evaporator to the home division that is being cooled (an approximately 3 °C temperature increase was considered along that path). The regeneration temperature corresponded to the typical temperature inside a domestic hot water reservoir heated by solar thermal collectors. Net water at 18 °C was used to remove heat from the adsorber during adsorption. For the application considered in this work, the performance of the AC was characterized by a COP of 0.5 and an SCP of 44 W·kg^{−1}. The performance coefficients may be lower than those of conventional coolers, but it should be highlighted that the energy used to drive the AC was surplus energy that would otherwise be wasted. Furthermore, the total cooling power of the system can be improved by increasing the mass of adsorbent material in the adsorber or by using two or more adsorbers in parallel (one being regenerated when the other is adsorbing water) [30–32].

The performance of the AC was obtained for several working conditions and control parameters, aiming to investigate its behaviour under different working conditions. A segregated approach was considered, varying one parameter at a time and assuming the reference values for the remaining parameters. The analysis included the evaporator, condenser, and regeneration temperatures, as well as the cycle time, which was considered as a control parameter. Figure 4 shows the evolution of the performance coefficients for different evaporator and condenser temperatures. As expected, the performance of the AC improved for higher evaporator temperatures and lower condenser temperatures. Lower evaporator temperatures and higher condenser temperatures deteriorated the performance of the AC. Notwithstanding the loss of performance, the system could work on very hot days with a COP of 0.31 and an SCP of 26 W·kg^{−1}, corresponding to condenser temperatures as high as 42 °C. Moreover, the thermally driven cooling system could be used for refrigeration purposes at temperatures as low as 2 °C, corresponding to a COP of 0.25 and an SCP of 18 W·kg^{−1}.

Table 1. Reference parameters used to calculate the COP and SCP.

Parameter	Value	Unit
C_s	921	$\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$
$d_{in,tube}$	0.01	m
$h_{m\rightarrow s}$	30	$\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
k_s	0.198	$\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$
L_{tube}	1	m
m_s	5.35	kg
t_{ads}	600	s
T_c	30	$^{\circ}\text{C}$
T_e	15	$^{\circ}\text{C}$
$T_{f,ads}$	18	$^{\circ}\text{C}$
$T_{f,reg}$	70	$^{\circ}\text{C}$
v_{HTF}	0.05	$\text{m}\cdot\text{s}^{-1}$
ρ_s	2027	$\text{kg}\cdot\text{m}^{-3}$
σ_s	0.002	m

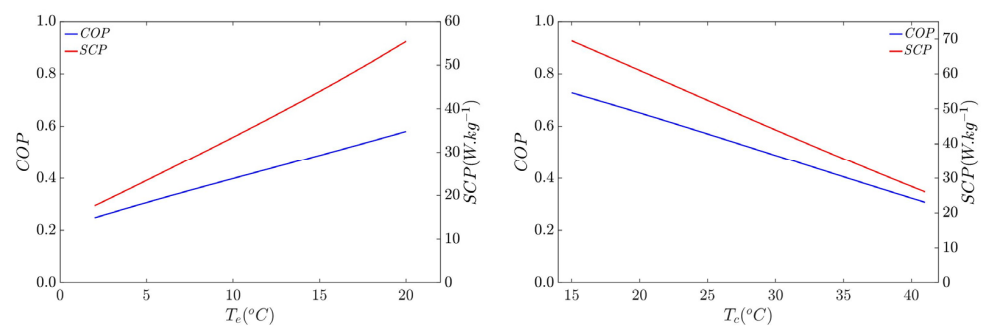
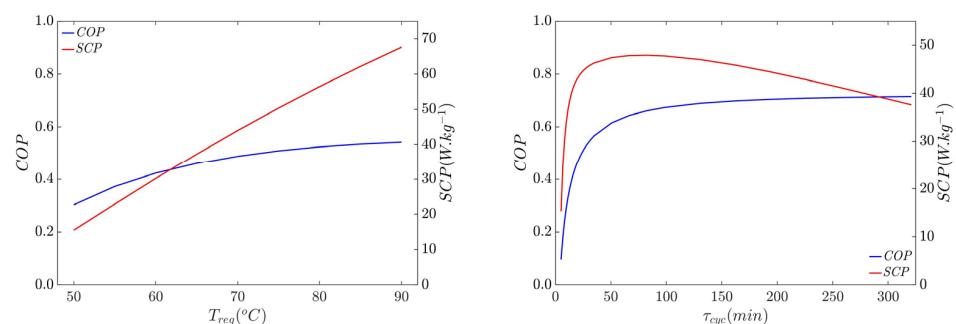
**Figure 4.** COP and SCP variation with the evaporator (on the (left)) and condenser (on the (right)) temperatures.

Figure 5 (left) depicts the influence of the regeneration temperature on the AC's performance for a cycle time of 22.2 min and of the cycle time (right) for a regeneration temperature of 70 $^{\circ}\text{C}$. It is noticeable that variations in the regeneration temperature affected the SCP more severely than the COP, given that higher regeneration temperatures led to faster heat exchange between the HTF and the adsorbent material. Identifying the optimal cycle time is one of the most important parameters in every adsorption system, its identification usually being complex due to the opposite behaviours of the COP and SCP. In this particular case, these opposite behaviours were present for cycle times higher than 83 min, where the COP kept increasing with the cycle time and the SCP started to decrease. As the cycle time increased, the SCP decreased, since the extra energy that was obtained by running longer (and more complete) cycles did not compensate for the increase in the cycle duration. The optimal balance between the COP and the SCP must be found for each individual adsorption system and, in this case, for each individual AC.

**Figure 5.** COP and SCP variation with the regeneration temperature (on the (left)) and the cycle time (on the (right)).

4. Conclusions

This work presents an AC driven by surplus heat from thermal solar collectors suitable for residential space cooling applications. This system can be used to provide cooling to residential buildings during the hot summer months solely using the inherent heat surplus that is generated by solar thermal collectors during warm seasons. The AC uses silica gel–water as the working pair and was modelled with a two-dimensional distributed parameter model. The model was used to simulate the thermally driven cooling system and calculate its performance coefficients. Considering the reference parameters, which corresponded to a hot summer day in the Aveiro region, the thermal cooling system had a COP of 0.5 and a total cooling power of approximately 235 W. A larger unit would be required to provide cooling to households, what can be accomplished either by increasing the adsorbent mass in the adsorber or by using several adsorbers in parallel. Analysis of the system main governing parameters of the thermally driven cooling was carried out. The results showed that, despite the lower performance for lower evaporator temperatures and higher condenser temperatures, the system could accommodate evaporator temperatures as low as 2 °C and condenser temperatures as high as 42 °C, demonstrating that it is still viable on hotter days. Using the heat surplus of solar thermal systems, the thermally driven cooling system could work close to its maximum COP and deliver more than half of its maximum SCP. Despite the low performance of the thermally driven cooling system, this technology is worthy of further research and development, considering that the energy required to drive the system is freely obtained (clean and cheaper). The same base adsorption system can be used for cooling purposes in summer or for heating purposes in winter.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

AC	Adsorption cooling
C	Specific heat ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)
C_p	Constant pressure specific heat ($\text{J} \cdot \text{kg}^{-1} \cdot \text{K}^{-1}$)
COP_c	Cooling coefficient of performance
d	Diameter (m)
D_{ef0}	Effective diffusion coefficient ($\text{m}^2 \cdot \text{s}^{-1}$)
E_a	Activation energy ($\text{J} \cdot \text{kg}^{-1}$)
HTF	Heat transfer fluid
$h_{f \rightarrow m}$	Fluid–metal convective heat transfer coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)
$h_{m \rightarrow s}$	Adsorbent–metal heat transfer coefficient ($\text{W} \cdot \text{m}^{-2} \cdot \text{K}^{-1}$)
k	Thermal conductivity ($\text{W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$)
k_0	Pre-exponential coefficient ($\text{kg} \cdot \text{kg}^{-1} \cdot \text{Pa}^{-1}$)
k_D	Blake–Kozeny permeability (m^2)
K_{LDF}	LDF constant (s^{-1})
L	Tube length (m)
m	Mass (kg)

P	Pressure (Pa)
Q	Heat (J)
q_m	Monolayer specific capacity ($\text{kg}\cdot\text{kg}^{-1}$)
r	Radial coordinate (m)
R_p	Particle radius (m)
R'	Adsorbate specific gas constant ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)
SCP	Specific cooling power ($\text{W}\cdot\text{kg}^{-1}$)
t	Time (s)
t_{SG}	Non-dimensional Toth constant
T	Temperature (K)
u	Adsorbate velocity ($\text{m}\cdot\text{s}^{-1}$)
v	Velocity ($\text{m}\cdot\text{s}^{-1}$)
X	Adsorbate concentration in the adsorbent ($\text{kg}_a\cdot\text{kg}_s^{-1}$)
z	Axial longitudinal coordinate (m)
Subscripts	
a	Adsorbate
ads	Adsorption
bed	Adsorbent bed
c	Condenser/cooling
cyc	Cycle
e	Evaporator
eq	Equilibrium
f	Fluid
h	Heating
ic	Isosteric cooling
ih	Isosteric heating
in	Inner
ini	Initial
m	Metal
max	Maximum
min	Minimum
out	Outer
p	Particle
reg	Regeneration
s	Adsorbent
v	Vapour/vaporization
Greek letters	
ΔH_{ads}	Adsorption heat ($\text{J}\cdot\text{kg}^{-1}$)
ΔH_v	Heat of vaporization ($\text{J}\cdot\text{kg}^{-1}$)
ε	Adsorbent bed porosity
μ	Dynamic viscosity ($\text{Pa}\cdot\text{s}$)
ρ	Adsorbate density ($\text{kg}\cdot\text{m}^{-3}$)
σ	Thickness (m)
τ	Cycle time (s)

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