ORIGINAL ARTICLE

Open Access

CrossMark

Effect of heating time of adsorber-collector on the performance of a solar adsorption refrigerator

Noureddine Cherrad[1,2*], Adel Benchabane[1], Lakhdar Sedira[3] and Amar Rouag[1]

Abstract

This paper presents a numerical study of heat transfer inside the adsorber-collector of a solar adsorption refrigerator using the activated carbon AC35-methanol pair. The objective is to estimate the amount of the heat loss through the adsorber-collector, during the solar heating phase, and to determine the effect of heating time on the thermal efficiency of the system. The numerical results showed that the heating time is the most important factor affecting the amount of energy loss. It has shown that the shorter heating time corresponds to the higher efficiency of the adsorber-collector. In addition, a new optimal coefficient of performance, COP_{optm}, is proposed to determine the number of adsorbers to be added to a machine. This latter is considered for consuming an energy equivalent to that received by the adsorber-collector. These additional adsorbers use a heat transfer fluid, coming from the adsorber-collector, instead of direct heating by solar radiation. An application example is presented using experimental results obtained from the literature. It has shown that the number of the additional adsorbers can reach three adsorbers.

Keywords: Adsorber thermal efficiency, Heat loss, Heat recovery, Optimal coefficient of performance, Solar adsorption refrigerator

Background

The fossil energy is the foundation of the global economy; the consumption, the transport, and the delivery of this energy source are harmful to the human being, the animals, and the nature. We know that the combustion of oil rejects a thousand polluting substances in the atmosphere; one of the well-known direct impact is global warming.

The search for alternative energy sources brought together environmentalists and those who for other reasons do not want to be dependent on oil suppliers. Certainly, other sources of energy may be an alternative to oil, such as solar energy.

Solar power in itself does not present a problem. The problem of its use comes mostly from the investment cost which is relatively expensive compared to fossil energy because the profitability of certain types of solar

systems is still weak. The improvements are not intended only for the materials of the solar installation but also for the performance of the system, which includes all energy losses caused by its components.

The ordinary refrigerating machines, even those that are operated by electricity, are indirect consumer devices of petroleum energy, so a system that must be replaced in the future by an equipment belonging to solar power system technology. In the refrigeration machines powered by the electricity, the compressor is the driving force of the thermodynamic cycle of the system, while in refrigeration machines powered by the sun, the phenomenon of desorption by solar heating can be an alternative to the compression phase. This phenomenon takes place in the adsorber-collector of the solar machine; the more the desorption is important, the more the cold production is high.

The estimation of coefficient of heat loss per unit of time and temperature within the adsorber-collector is already developed by Klein 1975 and discussed by Duffie and Beckman 2013. Jing and Exell 1994 used

* Correspondence: chernoured@gmail.com
[1]Laboratoire de Génie Energétique et Matériaux, LGEM, Université de Biskra, B.P. 145 R.P., Biskra 07000, Algeria
[2]Département de Génie Mécanique, Université de Ouargla, Faculté des Sciences Appliquées, Ouargla 30 000, Algeria
Full list of author information is available at the end of the article

Cherrad *et al. International Journal of Mechanical and Materials Engineering* (2017) 12:7

Page 2 of 9

the coefficient of loss developed by Klein 1975 for simulation and sensitivity analysis of an intermittent solar-powered charcoal/methanol refrigerator. Anyanwu et al. 2001 have made a study of an adsorption refrigerator using the charcoal/methanol pair, where the numerical model takes into account loss coefficient. The results of coefficient of performance (*COP*) are given according to the physical and dimensional characteristics of adsorber-collector. The loss coefficient was also considered by Leite and Daguenet 2000. It was a numerical simulation for instant transfers of heat and mass in each element of the refrigerating adsorption machine during the typical average day of each month. Another use of the coefficient of Klein was in the work of Chekirou et al. 2014, who introduced a modeling of heat and mass transfer in the tubular adsorbent of a refrigerating adsorption machine.

The objective of our study is to analyze the effect of heat loss within the adsorber-collector on the performance of solar adsorption refrigeration machines using AC35-methanol as working pair and to determine the criteria affecting the increase of this amount of energy.

The amount of heat loss through the adsorber-collector was calculated using the coefficient of loss over the solar heating time and an interval of temperatures between the temperature of adsorption (temperature at start of heating) and temperature of generation (temperature at end of heating). The study is done using a numerical program (Fortran) and validated with experimental results obtained from the literature.

Methods
Basic components
The main components of solar adsorption refrigerator are shown in the Fig. 1: the adsorber-collector containing the adsorbent-adsorbate pair (activated carbon AC35-methanol), the air-cooled condenser, and the evaporator within the cold chamber (Rouag et al. 2016).

A valve is set at inlet and outlet of adsorber-collector, to control the flow of the refrigerant from the adsorbent towards the condenser during the desorption phase, and from the evaporator towards the adsorbent during the adsorption phase.
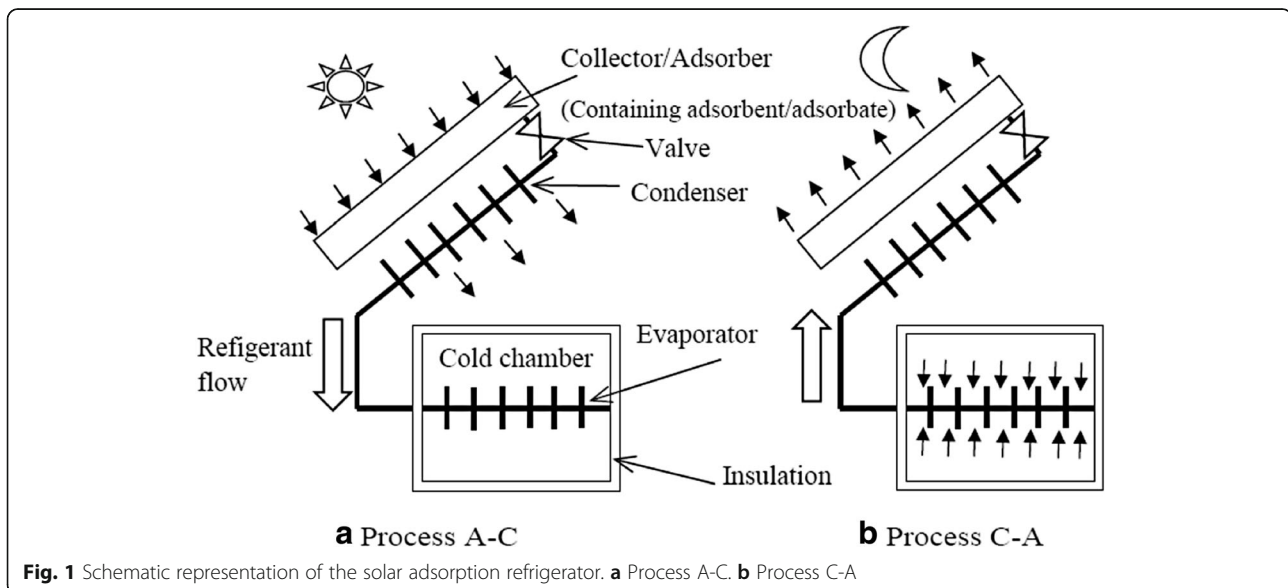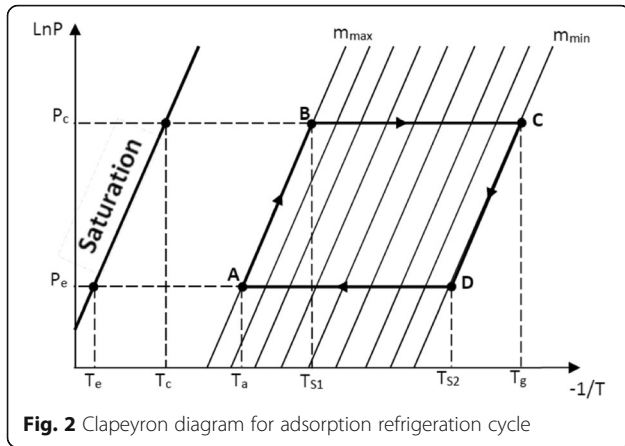
Intermittent cycle
Four thermodynamic processes form the intermittent ideal cycle of solar-powered adsorption refrigerator are shown in Fig. 2.

First, the adsorber-collector is heated by solar energy, which increases the pressure of the adsorbent-adsorbate pair up to the condensing pressure P_c necessary for the condenser (processes A–B). In state B of cycle, the valve at the inlet of condenser is open, and the continuing of the heating at constant pressure induces the desorption of refrigerant (adsorbate) from the adsorbent towards the air-cooled condenser to be condensed at condensing temperature T_c (processes B–C).

The desorbed mass during processes B–C is the difference between the maximal mass of adsorbate in state B and the minimal mass of adsorbate in state C.

The maximum temperature (generating temperature T_g) reached by the adsorbent will be in state C. At this value of temperature, the valve is closed and solar irradiance

Fig. 1 Schematic representation of the solar adsorption refrigerator. **a** Process A-C. **b** Process C-A

Fig. 2 Clapeyron diagram for adsorption refrigeration cycle

starts to decrease, which induces the drop of temperature and pressure of adsorber-collector (processes C–D).

When the pressure reaches the value P_e, which reigns in the evaporator, the valve is open, and the continuing of the cooling of adsorbent at constant pressure induces the evaporation of refrigerant at an evaporating temperature T_e from the evaporator towards the adsorbent by the adsorption phenomena (processes D–A). By this way, the evaporated refrigerant extracts heat and generates cold production within the cold chamber.

The adsorbed mass during processes D–A is the difference between the maximal mass of adsorbate in state A and the minimal mass of adsorbate in state D.

This cycle is said to be intermittent because the cold production starts only at sunset.

Adsorbed mass
Dubinin and Astakov proposed a state equation for mass adsorbed by microporous medium in equilibrium with polymodal distribution of pore size (Leite and Daguenet 2000):

$$X(T,P) = W_0 \rho_l(T) exp\left\{ -D\left[T.ln\left(\frac{P_s(T)}{P}\right)\right]^n\right\} \quad (1)$$

Where W_0 is the maximum volume adsorbed, ρ_l is the density of the adsorbate in the liquid state, D is the coefficient of affinity, T is the temperature of the adsorber-collector, P_s is the saturation pressure of the adsorbate, P is the equilibrium pressure of adsorbent-adsorbate pair, and n is the parameter of adjustment of the D–A equation.

The adsorbate mass concentration maximal X_{max} and minimal X_{min} are calculated by Eq. (1), respectively, at the adsorption temperature T_a and the generating temperature T_g as the following:

$$X(T_a, P_e) = X(T_{s1}, P_c) = X_{max} \quad (2)$$

$$X(T_g, P_c) = X(T_{s2}, P_e) = X_{min} \quad (3)$$

Where T_{s1} and T_{s2} are respectively the temperature at start of desorption and temperature at start of adsorption.

During the adsorption, the evaporating pressure is $P_e = P_s (T_e)$, and for the desorption, the condensing pressure is $P_c = P_s (T_c)$. T_e and T_c are respectively the evaporating temperature and the condensing temperature.

Thermal efficiency of the adsorber-collector
The total energy received by the adsorber-collector Q_{ge} can be divided in five parts:

$$Q_{ge} = Q_1 + Q_2 + Q_3 + Q_{des} + Q_L \quad (4)$$

The heats Q_1, Q_2, and Q_3 are respectively energy supplied to heat the adsorbent, energy supplied to heat the tubes in metal, and energy supplied to heat the adsorbate mass. They can be calculated as the following (Chekirou et al. 2011):

$$Q_1 = m_d Cp_d(T_g - T_a) \quad (5)$$

m_d is the adsorbent mass and Cp_d is the specific heat of adsorbent.

$$Q_2 = m_t Cp_t(T_g - T_a) \quad (6)$$

m_t and Cp_t are respectively the mass and specific heat of metallic tubes containing the adsorbent.

$$Q_3 = m_d \left(X_{max} \int_{T_a}^{T_{S1}} Cp_l(T)dT + \int_{T_{S1}}^{T_g} X(T)Cp_l(T)dT \right) \quad (7)$$

Cp_l is the specific heat of adsorbate in liquid state.

The heat necessary for desorption process Q_{des} is given by (Chekirou 2008):

$$Q_{des} = m_d nD \int_{T_{S1}}^{T_g} X(T) T^n \left(ln\frac{P_s(T)}{P_c}\right)^{n-1} \frac{q_{st}^2(T)}{rT^2} dT \quad (8)$$

Isosteric heat q_{st} is calculated as the function of the pressure and temperature:

$$q_{st} = L(T_c) + rTln\left(\frac{P_s(T)}{P_c}\right) + \left[\frac{\alpha rT}{nD}\right]\left[Tln\left(\frac{P_s(T)}{P_c}\right)\right]^{(1-n)} \quad (9)$$

L is the latent heat of the adsorbate, r is the particular gas constant of the adsorbate, and α is the thermal expansion coefficient of the adsorbate.

According to Anyanwu et al. 2001, the heat loss from the lateral surfaces of adsorber-collector is assumed to be negligible.

Cherrad *et al. International Journal of Mechanical and Materials Engineering* (2017) 12:7

Page 4 of 9

The global loss coefficient is given by Duffie and Beckman 2013 as follows:

$$U_L = U_t + U_b \qquad (10)$$

U_t is the loss coefficient from the top of the adsorber-collector, and U_b is the loss coefficient from the bottom of the adsorber-collector.

where:

$$U_t(T_p, T_{am}) = \left[\frac{n_g}{\frac{c_0}{T_p}\left(\frac{T_p - T_{am}}{n_{g+f}}\right)^e} + \frac{1}{h_w} \right]^{-1}$$
$$+ \left[\frac{\sigma(T_p + T_{am})(T_p^2 + T_{am}^2)}{(E_p + 0.00591 n_g h_w)^{-1} + \frac{2n_g + f - 1 + 0.133 E_p}{E_g} - n_g} \right] \qquad (11)$$

with:

$$e = 0.43\left(1 - \frac{100}{T_p}\right)$$

$$f = (1 + 0.089 h_w - 0.1166 h_w E_p)(1 + 0.07866 n_g)$$

$$c_0 = 520(1 - 0.000051 \Omega^2)$$

$0° < \Omega < 70°$, and for $70° < \Omega < 90°$ we take $\Omega = 70°$

$$h_w = 2.8 + 3.0 W_V$$

T_p is the wall temperature of the top of adsorber-collector, T_{am} is the ambient temperature, n_g is the number of glass cover of adsorber-collector, E_p is the emissivity of the top wall of the adsorber-collector, E_g is the emissivity of the glass cover of the adsorber-collector, Ω is the adsorber-collector inclination, and W_v is the wind velocity.

Chekirou et al. 2014 took U_b as the constant.

To estimate the amount of total heat loss towards the air ambient during processes A–C (Fig. 2), the following equation is used:

$$Q_L = S.(U_t + U_b).(Tp_{max} - Tp_{min}).t_{max} \qquad (12)$$

Using Eq. (12), we cover the full time of heating process t_{max} on a range of wall adsorber-collector temperatures from Tp_{min} (temperature of adsorber-collector wall at the start of heating) to Tp_{max} (temperature of adsorber-collector wall at the end of heating).

The temperatures Tp_{max} and Tp_{min} given in Eq. (12) can be substituted respectively by the temperatures T_g and T_a, so:

$$Q_L = S.(U_t + U_b).(T_g - T_a).t_{max} \qquad (13)$$

And for Eq. (11), the temperature ambient T_{am} is considered to be equal to T_a.

It is known that the refrigerator adsorber-collector needs a useful energy Q_u and the remainder is a lost energy Q_L; therefore,

$$Q_u = Q_{ge} - Q_L \qquad (14)$$

While the recovery heat Q_r from the adsorber-collector is a part or all the lost energy Q_L, so $Q_r \le Q_L$, it varies from 0 to Q_L, then we can write:

$$Q_r = Q_L\left(1 - \frac{t_h}{t_{max}}\right) \qquad (15)$$

Equation (15) means that $Q_r = 0$ for a heating time $t_h = t_{max}$ and $Q_r = Q_L$ for heating time $t_h = 0$.

On the other hand, the thermal efficiency of the solar-powered adsorber-collector can be expressed as follows:

$$\eta = \frac{Q_u}{Q_{ge}} = \frac{Q_{ge} - Q_L}{Q_{ge}} = 1 - \frac{Q_L}{Q_{ge}} \qquad (16)$$

Coefficient of performance

The coefficient of performance of an adsorptive cooling system is equal to the heat extracted in the evaporator per the heat received in the adsorber-collector (Douss and Meunier 1988):

$$COP = \frac{Q_{ev}}{Q_{ge}} \qquad (17)$$

Where:

$$Q_{ev} = m_d(X_{max} - X_{min})\left[L(T_e) - \int_{T_e}^{T_c} Cp_l(T)dT\right] \qquad (18)$$

The first term of Eq. (18) represents the heat absorbed by the evaporation of the refrigerant (adsorbate) at the evaporation temperature T_e. The second term represents the sensible heat necessary to bring the condensate of its condensing temperature T_c to evaporating temperature T_e (Chekirou et al. 2011).

Results and discussion

Tables 1 and 2 give all the relevant data of numerical computation.

The pressure of saturation P_s, the latent heat L, density ρ_l, and the mass specific heat of adsorbate (methanol) in the liquid state Cp_l are estimated by interpolation of the data given in Table 3.

For each reference of validation, i.e., Douss and Meunier 1988 and Lemmini and Errougani 2007, the mass m_d of the adsorbent (activated carbon AC35) has been taken the same as that considered by the authors. Lemmini and

Table 1 Data used in the present numerical study

Symbol	Parameter	Value
Cp_d	Specific heat of adsorbent	920 J kg^{-1} K^{-1}
Cp_t	Specific heat of tubes (copper) containing the adsorbent	380 J kg^{-1} K^{-1}
E_g	Emissivity of glass cover	0.88[a]
E_P	Emissivity of the top wall of adsorber-collector	0.1[a]
n_g	Number of glass cover	1[a]
r	Particular gas constant	259.34 J kg^{-1} K^{-1}
S	Adsorber-collector surface	0.73 m^2
t_{max}	Maximum time of heating process A–C (Fig. 2)	12 h
U_b	Loss coefficient from the bottom of the adsorber-collector	0.9 W K^{-1a}
W_v	Wind velocity	1 m s^{-1}
a	Thermal expansion coefficient of adsorbate	0.00126 K^{-1b}
Ω	Adsorber-collector inclination	34 °C[a]
σ	Stefan-Boltzmann constant	5.670367E-8 W m^{-2} K^{-4}

[a]Data obtained from the work of Chekirou et al. 2014
[b]Data obtained from the work of Goodwin 1987

Errougani 2007 have given the value of the adsorbent mass, which is equal to 14.5 kg, and for the machine studied by Douss and Meunier 1988, it is estimated by 12.5 kg.

The same for the mass m_t of metallic tubes containing the adsorbent, which are estimated to be equal to 45.68 and 26.024 kg for Douss and Meunier 1988 and Lemmini and Errougani 2007 respectively.

The maximum time t_{max} of heating processes A–C (Fig. 2) is considered for all day, so it is an average of 12 h.

Validation of numerical model
First, we have validated our numerical model by the results given by Douss and Meunier 1988. The authors have studied an experimental unit of an adsorption refrigeration machine that operates by electrical heating instead of the solar heating; therefore, heat loss Q_L cannot be estimated by Eq. (13), which is destined to solar

Table 2 Values of Dubinin-Astakhov equation parameters

Symbol	Parameter	Value
D	Coefficient of affinity	5.02E-07[a]
n	Parameter of adjustment of D-A equation	2.15[b]
W_0	Maximum adsorbed volume	0.000425 m^3 (of adsorbate) kg^{-1} (of adsorbent)[b]

[a]Data obtained from the work of Douss and Meunier 1988
[b]Data obtained from the work of Douss & Meunier 1988; Pons & Grenier 1986 and Critoph 1988

Table 3 Thermophysical properties of methanol along the saturation line (Bejan and Kraus 2003)

T [K]	P_s [MPa]	ρ_l [kg m^{-3}]	Cp_l [kJ kg^{-1} K^{-1}]	L [kJ kg^{-1}]
200	6.1 × 10^{-6}	880.28	2.2141	1289.99
250	0.00081	831.52	2.3121	1234.79
300	0.0187	784.51	2.5461	1166.17
325	0.0603	760.74	2.7223	1124.564
350	0.1617	735.84	2.9362	1075.936
375	0.3748	708.86	3.1891	1017.33
400	0.7737	678.59	3.4912	944.57
425	1.4561	643.38	3.8713	852.59
450	2.5433	600.49	4.4067	739.19
475	4.1688	544.35	5.3805	610.48
500	6.5250	451.53	9.9683	391.09

adsorption adsorber-collectors. Douss and Meunier 1988 have given this amount of heat; it is equal to 365 kJ. For the temperatures' data, we used the data of working fluid (WF) by Douss and Meunier 1988.

The compared results of *COP* are shown in Figs. 3, 4, and 5.

The average relative errors between the results compared in the Figs. 3, 4, and 5 are respectively 4.77, 5.22, and 9.96%.
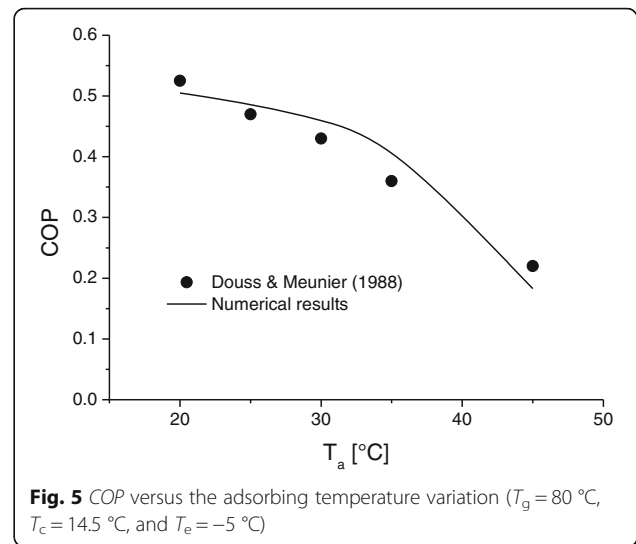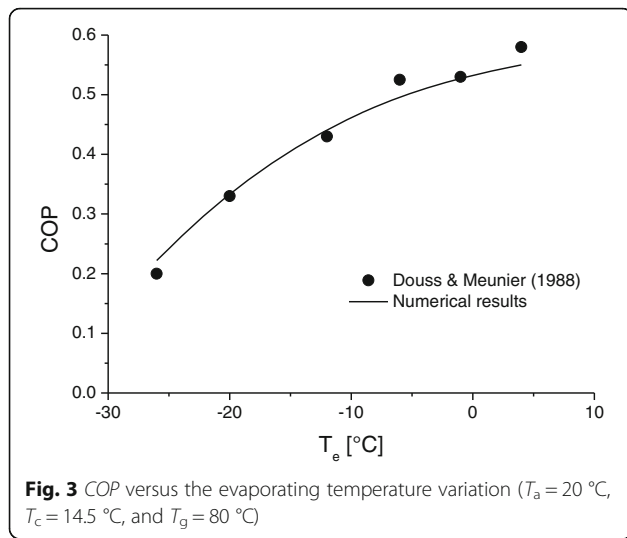
The validation with the work of Douss and Meunier 1988 was made for the four calculated heats, i.e., Q_1, Q_2, Q_3, and Q_{des}, while Q_L is given by the author. Our numerical model takes into account the prediction of this quantity (Q_L), that is why we are going to carry out a second validation with Lemmini and Errougani 2007 to validate all the total energy Q_{ge}, ensuring by this way the validation of Eq. (13).

Lemmini and Errougani 2007 studied an experimental prototype of a solar adsorption refrigeration machine, where the recorded results are given in Table 4.

We have calculated the cooling production quantities Q_{ev} obtained by Lemmini and Errougani 2007 for eleven tested days by using Eq. (17), where *COP* and Q_{ge} are known. After that, we have predicted the operating temperatures of the studied machine.

The advantage of the developed numerical program is the prediction of operating temperatures T_a, T_c, and T_g for given cold production Q_{ev} and evaporating temperature T_e. The calculated temperatures will be used to estimate the amount of heat Q_{ge}.

The numerical results of *COP* variation using Eq. (17) compared with those obtained during the tested days by Lemmini and Errougani 2007 are shown in Table 4 and Fig. 6. The comparison shows an average relative error of 5.03%.

Cherrad *et al. International Journal of Mechanical and Materials Engineering* (2017) 12:7

Page 6 of 9

Fig. 3 *COP* versus the evaporating temperature variation ($T_a = 20$ °C, $T_c = 14.5$ °C, and $T_g = 80$ °C)

Fig. 5 *COP* versus the adsorbing temperature variation ($T_g = 80$ °C, $T_c = 14.5$ °C, and $T_e = -5$ °C)

Thermal efficiency and performance coefficient

The obtained results for the amounts of heats estimated, the thermal efficiency of the adsorber-collector, and the coefficient of performance of solar adsorption refrigerator in this subsection are based on the data considered for validation with the results of Lemmini and Errougani 2007.

As shown in Fig. 7, for one adsorber-collector, 25% of total received heat during t_{max} is useful and the rest must be recovered. The thermal efficiency takes two ends, maximum and minimum, which are 100% for th = 0 and 25% for $t_h = t_{max}$. This can be deduced from Eq. (16).

The center of this curve takes a parabolic form, where it must coincide with the recovered heat quantity $Q_r(t_h)/Q_{ge}(t_{max}) = 50\%$ and the quantity of heat received $Q_{ge}(t_h)/Q_{ge}(t_{max}) = 50\%$ at $t_h/t_{max} = 0.33$.

At this time where $Q_u(t_h)/Q_{ge}(t_{max}) = Q_L(t_h)/Q_{ge}(t_{max}) = 25\%$ and using Eq. (16), the value of thermal efficiency η is
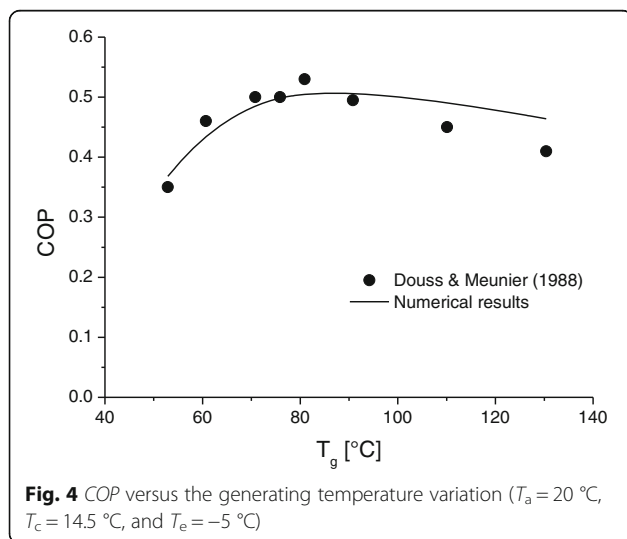
50%. It is the moment where the three quantities must take the same value which equals to 50% and corresponds to $t_h/t_{max} = 0.33$.

The shorter heating times correspond to higher efficiencies of the adsorber-collector. It can allow to recover up to 75% of total heat received by the adsorber-collector during t_{max} with zero heat loss. This means that improving design of the adsorber-collector is required to ensure the useful amount of solar energy necessary for system functioning during a very short time. So, it is recommended to use a heat transfer fluid instead of direct heating by solar radiation. Thus, we can isolate the upper side of the adsorber-collector that represents the large hole of the heat, and we command the heating time by the temperature and the flow rate of the heat transfer fluid. The fluid will flow in

Fig. 4 *COP* versus the generating temperature variation ($T_a = 20$ °C, $T_c = 14.5$ °C, and $T_e = -5$ °C)

Table 4 Numerical results of *COP* variation compared with those calculated during the tested days by Lemmini and Errougani 2007

Day	COP[a]	COP[b]	Relative error [%]
02 April	9.00	9.65	7.22
03 April	8.00	7.86	1.75
04 April	11.00	10.10	8.18
05 April	9.00	8.49	5.67
07 April	6.90	7.54	9.28
10 April	6.00	5.73	4.50
11 April	10.20	10.69	4.80
12 April	8.60	8.55	0.58
13 April	7.90	7.73	2.15
14 April	6.00	6.24	4.00
15 April	9.00	9.65	7.22

[a]Data obtained from the work of Lemmini and Errougani 2007
[b]Numerical results obtained in the present work

Cherrad *et al. International Journal of Mechanical and Materials Engineering* (2017) 12:7
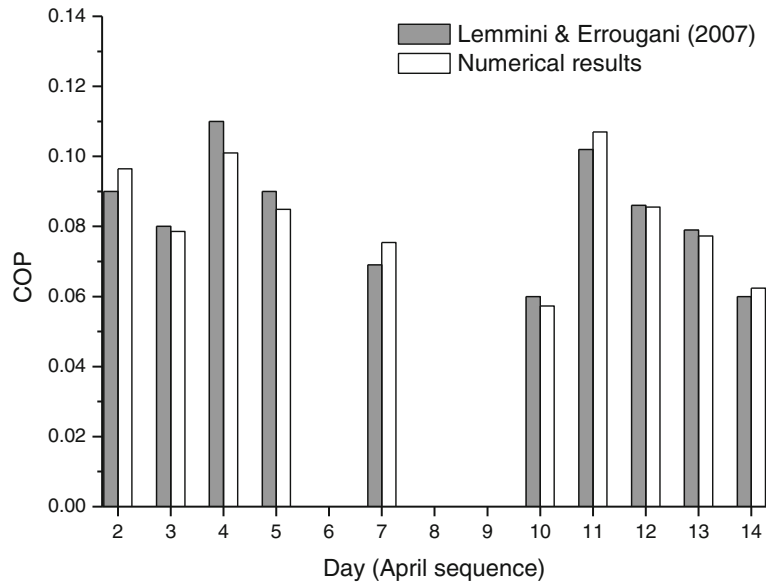
Page 7 of 9

Fig. 6 Comparison of the numerical *COPs* of the present study with those obtained by Lemmini and Errougani 2007

the tubes of a heat exchanger set inside the adsorber-collector. Even if the prior heating of heat transfer fluid by solar energy takes time, we can use a system of energy storage in the form of heat. Thus, we ensure the heating of the adsorber-collector even at night, and the cycle of the machine becomes continuous instead of intermittent cycle with a very good performance.

With a recovery energy system, the number of additional adsorbers can be added to the machine with a total received energy Q_{ge} by only one additional adsorber is Num_{re}, so we can write:

One additional adsorber needs $\rightarrow Q_u$ (*)
Number of additional adsorbers (Num_{re}) to be added need $\rightarrow Q_r$ (**)

From (*), (**), Eqs. (14) and (15):

$$Num_{re} = \frac{Q_r}{Q_u} = \frac{Q_L\left(1 - \frac{t_h}{t_{max}}\right)}{Q_{ge} - Q_L} \tag{19}$$

As shown in Fig. 8, for a heating time t_h less than 4 h (t_{max} is assumed to be equal to 12 h), we can recover a
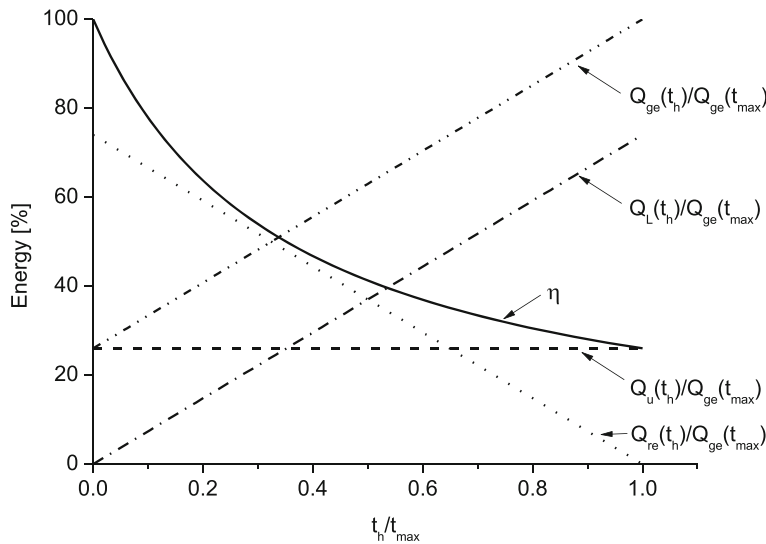

Fig. 7 Energy ratios and efficiency of adsorber-collector as function of the ratio of heating time and maximum heating time ($T_a = T_c = 25$ °C, $T_e = 0$ °C, and $T_g = 90$ °C)

Cherrad *et al. International Journal of Mechanical and Materials Engineering* (2017) 12:7

Page 8 of 9

useful energy for functioning of two to three additional adsorbers.

The recovery of energy loss Q_L leads us to propose a new optimal *COP*, it may be given by:

$$COP_{optm} = (Num_{re} + 1).COP \qquad (20)$$

By Eq. (20), we can see that a high optimal coefficient of performance COP_{optm} corresponds to a heating of shorter time t_h. If $t_h = t_{max}$, no energy recovery (by Eq. (19) $Qr = 0 \Rightarrow Num_{re} = 0$) and *COP* optm would take the value of the normal *COP*.

As shown in Fig. 9, with an energy recovery system from one adsorber-collector, the COP_{optm} value can reach up to four times the *COP* in the normal case.
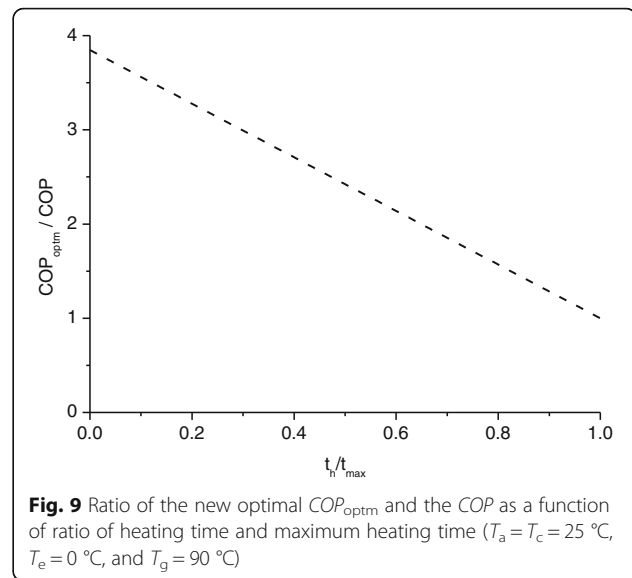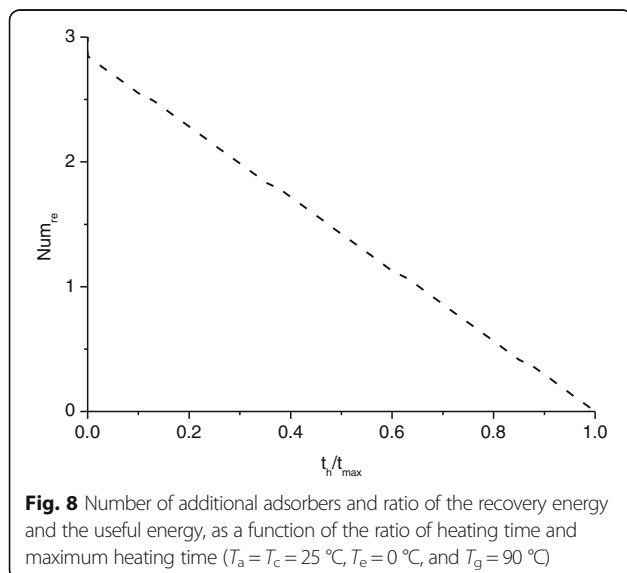
Conclusions

In the present paper, a numerical model was developed to predict the different amounts of heat received by the adsorber-collector of a solar-powered adsorption refrigeration machine using activated carbon AC35/methanol as a working pair.

Four amounts of heat are useful for the operation of system, while the fifth amount is an energy loss. The estimate of this lost energy showed that the time of heating of the adsorber-collector is a very important factor affecting its thermal efficiency.

The shorter heating time allows to cut off the continuation of heat loss into the external environment and gives a good performance.

If we use a system heated by solar energy, the reduction of the heating time is somewhat difficult because it is related to the radiation intensity, which is as function of the time. So, it is useful to use a heat transfer fluid

Fig. 9 Ratio of the new optimal COP_{optm} and the *COP* as a function of ratio of heating time and maximum heating time ($T_a = T_c = 25$ °C, $T_e = 0$ °C, and $T_g = 90$ °C)

heated by the sun with storage system using the water for example instead the direct heating by solar radiation.

Furthermore, the performed study allows to define a new optimal coefficient of performance (COP_{optm}) more generalized. It takes the value of normal coefficient of performance (*COP*) for zero recovery of energy loss, and greater value for more energy recovery, keeping the same total energy necessary for the functioning of the system.

The usefulness of this optimal coefficient of performance is to determine the number of additional adsorbers that can be added to the machine for an equivalent consumption to energy received by the adsorber-collector.

The cold production increases with the number of additional adsorbers. Even though the days when solar radiation is low, the decrease in cold production will be better than that without an energy recovery system.

Fig. 8 Number of additional adsorbers and ratio of the recovery energy and the useful energy, as a function of the ratio of heating time and maximum heating time ($T_a = T_c = 25$ °C, $T_e = 0$ °C, and $T_g = 90$ °C)

Nomenclature

COP Coefficient of performance
COP_{optm} Optimal coefficient of performance
Cp_d Specific heat of adsorbent, J kg^{-1} K^{-1}
Cp_l Specific heat of adsorbate in liquid state, J kg^{-1} K^{-1}
Cp_t Specific heat of tubes in metal containing the adsorbent, J kg^{-1} K^{-1}
D Coefficient of affinity
E_g Emissivity of glass cover of adsorber-collector
E_P Emissivity of top wall of adsorber-collector
L Latent heat of adsorbate, J kg^{-1}
m Adsorbate mass, kg
m_d Adsorbent mass, kg
m_t Mass of metallic tubes containing the adsorbent, kg
n Parameter of adjustment of D–A equation
n_g Number of glass cover of adsorber-collector

Num_{re} Number of additional adsorbers

r Particular gas constant of adsorbate, J kg^{-1} K^{-1}

P Equilibrium pressure of adsorbent-adsorbate pair, Pa

P_c Condensing pressure of adsorbate, Pa

P_e Evaporating pressure of adsorbate, Pa

P_s Saturation pressure of adsorbate, Pa

Q_{des} Heat necessary for desorption process, J

Q_{ev} Heat load in evaporator (cooling production), J

Q_{ge} Total energy received by adsorber-collector, J

q_{st} Isosteric heat, J kg^{-1}

Q_1 Energy supplied to heat the adsorbent, J

Q_2 Energy supplied to heat the tubes in metal containing the adsorbent, J

Q_3 Energy supplied to heat the adsorbate mass, J

Q_L Heat loss by adsorber-collector during the time of heating (processes A–C of Fig. 2), J

Q_r Recovered heat from adsorber-collector, J

Q_u Useful energy for functioning of system, J

S Adsorber-collector surface, m^2

T Temperature, K

T_a Adsorption temperature (temperature at end of adsorption), K

T_{am} Ambient temperature, K

T_c Condensing temperature of adsorbate, K

T_e Evaporating temperature of adsorbate, K

T_g Generating temperature (temperature at end of desorption), K

T_p Wall temperature of the top of adsorber-collector, K

Tp_{max} Maximal temperature of heated walls of adsorber-collector (including upper and bottom parts), K

Tp_{min} Minimal temperature of heated walls of adsorber-collector (including upper and bottom parts), K

t_h Heating time of adsorbent, s

t_{max} Maximum time of heating processes A–C (Fig. 2), s

U_b Loss coefficient from the bottom of the adsorber-collector, W K^{-1}

U_L Global loss coefficient from the adsorber-collector surfaces, W K^{-1}

U_t Loss coefficient from the top of the adsorber-collector, W K^{-1}

T_{s1} Temperature at start of desorption, K

T_{s2} Temperature at start of adsorption, K

W_v Wind velocity, m s^{-1}

W_0 Maximum adsorbed volume, m^3 (of adsorbate) kg^{-1} (of adsorbent)

X_{max} Maximal adsorbate mass concentration in 1 kg of adsorbent, kg kg^{-1}

X_{min} Maximal adsorbate mass concentration in 1 kg of adsorbent, kg kg^{-1}

Greek symbols

α Thermal expansion coefficient of adsorbate, K^{-1}

ΔT_{am} Variation of ambient temperature, K

ΔT_p Variation of upper wall temperature of adsorber-collector, K

η Thermal efficiency of adsorber-collector

Ω Adsorber-collector inclination, °

ρ_l Density of the adsorbate in the liquid state, kg m^{-3}

σ Stefan-Boltzmann constant, W m^{-2} K^{-4}

Author details
[1]Laboratoire de Génie Energétique et Matériaux, LGEM, Université de Biskra, B.P. 145 R.P., Biskra 07000, Algeria. [2]Département de Génie Mécanique, Université de Ouargla, Faculté des Sciences Appliquées, Ouargla 30 000, Algeria. [3]Laboratoire de Génie Mécanique, LGM, Université de Biskra, B.P. 145 R.P., Biskra 07000, Algeria.

References
Anyanwu, E. E., Oteh, U. U., & Ogueke, N. V. (2001). Simulation of a solid adsorption solar refrigerator using activated carbon/methanol adsorbent/refrigerant pair. *Energy Conversion and Management, 42*, 899–915.

Bejan, A., & Kraus, A. D. (2003). *Heat transfer handbook*. New York: Wiley.

Chekirou, W. (2008). *Etude et analyse d'une machine frigorifique solaire à adsorption*. Alegria: University of Constantine. Doctorat Thesis.

Chekirou, W., Boussehain, R., Feidt, M., Karaali, A., & Boukheit, N. (2011). Numerical results on operating parameters influence for a heat recovery adsorption machine. *Energy Procedia, 6*, 202–216.

Chekirou, W., Chikouche, A., Boukheit, N., Karaali, A., & Phalippou, S. (2014). Dynamic modelling and simulation of the tubular adsorber of a solid adsorption machine powered by solar energy. *International Journal of Refrigeration, 39*, 137–151.

Critoph, E. (1988). Performance limitations of adsorption cycles for solar cooling. *Solar Energy, 41*(1), 21–31.

Douss, N., & Meunier, F. (1988). Effect of operating temperatures on the coefficient of performance of active carbon-methanol systems. *Heat Recovery Systems & CHP, 8*(5), 383–392.

Duffie, J. A., Beckman, W. (2013). *Solar engineering of thermal Processes* (4th ed.). New York: Wiley.

Goodwin, R. D. (1987). Methanol thermodynamic properties from 176 to 673 K at pressures to 700 bar. *Journal of Physical Chemistry, 16*(4), 799–892.

Jing, H., & Exell, R. H. B. (1994). Simulation and sensitivity analysis of an intermittent solar-powered charcoal/methanol refrigerator. *Renewable Energy, 4*(1), 133–149.

Klein, S. A. (1975). Calculation of fiat-plate collector loss coefficients. *Solar Energy, 17*, 79–80.

Leite, A. P. F., & Daguenet, M. (2000). Performance of a new solid adsorption ice maker with solar energy regeneration. *Energy Conversion and Management, 41*, 1625–1647.

Lemmini, F., & Errougani, A. (2007). Experimentation of a solar adsorption refrigerator in Morocco. *Renewable Energy, 32*, 2629–2641.

Pons, M., & Grenier, P. H. (1986). A phenomenological adsorption equilibrium law extracted from experimental and theoretical considerations applied to the activated carbon + methanol pair. *Carbon, 24*(5), 615–625.

Rouag, A., Benchabane, A., Labed, A., & Boultif, N. (2016). Thermal design of air cooled condenser of a solar adsorption refrigerator. *Journal of Applied Engineering Science & Technology, 2*(1), 23–29.