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"Experimental Study of Adsorptive Solar-Powered Ice Makers in Agadir (Morocco) - 1 & 2"

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EXPERIMENTAL STUDY OF ADSORPTIVE SOLAR-POWERED ICE MAKERS IN AGADIR (MOROCCO)—1. PERFORMANCE IN ACTUAL SITE

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Abstract—Adsorptive cycles are well adapted to solar refrigeration. Solar ice makers (with an activated carbon + methanol pair) are described. Their major originality lies in the assembly of solar collector and condenser in a unique component: the collecto–condenser. A 'standard' ice maker and two prototypes were tested in actual conditions in Agadir, Morocco and this operation is described. Performances are quite good in Agadir; units produce more than 4 kg/m^2 of ice for more than 60% of the days investigated.

INTRODUCTION

Solar cooling seems to be a very attractive application for solar energy since more cooling is necessary when more solar energy is available. Nevertheless, to date, most efforts for promoting solar cooling have failed.

First efforts in the 1960s and 1970s proposed absorption air conditioners to be powered with solar collectors [1–3].

Other attempts concerned solar powered, desiccant cooling systems [4]. At least one U.S. company did commercialize such units, but currently this alternative does not seem to be ready for major developments.

No major technical problem was encountered and the technical feasibility of solar air conditioners has been proved, but economic success is unlikely in electricity or gas supplied areas. Indeed such systems, resulting from coupling a cooling system with solar collectors, still need auxiliary energy. Where this energy exists, classical systems are very competitive. The authors guess that the economic viability of solar cooling can be reached by integrated thermal cooling systems (which are energetically self-sufficient) installed in remote areas where energy is expensive.

These latter systems include chemically reacting systems [5, 6], liquid absorption units [7–10] and adsorptive units as presented herein [11–16]. The main characteristics of all these latter systems are that the (liquid or solid) absorbent, or the adsorbent, is distributed in the solar collectors and that the units perform daily intermittent cycles. Liquid absorption units generally include a small pump: they need some mechanical auxiliary energy which can be provided by photovoltaic cells. Both adsorption and chemical reacting systems are basically self-sufficient.

The type of solar collectors is another factor of comparison. It depends on the level of temperature required by the process. For example, an efficient water–ammonia absorption solar refrigerator requires a temperature of collector around 150°C. This can be performed generally by collectors which are either parabolic [10] or with evacuated tubes. Adsorptive units require a temperature of collector around 100°C. Then, they can easily be powered by flat-plate collectors, which is an advantage.

Moreover, solid absorbents undergo a large volume change when operating, and liquid absorption units often need a rectification column. Solid adsorption units are free of these inconveniences.

Several technical studies have proven good performance of adsorptive solar cooling systems at the laboratory scale [11–16]. The purpose of the present work was to perform measurements on commercialized adsorptive units in a North African climate (latitude: 30°23′N).

SOLAR ADSORPTIVE CYCLES

Principle of solid adsorption

Adsorption is fixation of gas molecules on the surface of a solid, by Van der Waals bindings. Adsorption is reversible, exothermic, it induces no changes of solid

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volume and does not involve any chemical reaction [17]. When gas is a pure vapour, the equilibrium between the adsorbed and the gaseous phases is divariant: among the most common state variables, m, P and T, two of them define the state of the system, e.g. $m = m(T, P)$,

$$\text{with} \quad \left(\frac{\partial m}{\partial T}\right)_P < 0 \quad \text{and} \quad \left(\frac{\partial m}{\partial P}\right)_T > 0. \qquad (1)$$

Adsorptive refrigerating cycle

Basically, an adsorptive cycle is intermittent: the adsorber containing the adsorbent is alternatively connected with a condenser and with an evaporator (of the adsorbate). The cycle is the succession of two periods. First, a period of heating–desorption–condensation at high pressure, then a period of cooling–adsorption–evaporation at low pressure. High and low pressures are the saturation pressures of the adsorbate at the temperatures of, respectively, the condenser and the evaporator. Figure 1 shows an ideal cycle in the Clapeyron diagram (ln P vs $-1/T$).

Let us begin the cycle at point A, where the adsorber is at low temperature T_a, and low pressure $P_e = P_s(T_e)$. The adsorber is heated, ideally first isolated from any exchanger: the system follows the isoster: $m = m_a = m(T_a, P_e)$. Pressure increases and when it reaches the value $P_c = P_s(T_c)$, (point B), the adsorber is connected with the condenser ideally at constant temperature, T_c. Then, isobaric desorption proceeds while adsorbent temperature increases, up to point R, where adsorbed mass left is $m_r = m(T_r, P_c)$. Condensed liquid amount, $\Delta m = m_a - m_r$, is transferred into the evaporator (e.g. by gravity). Further on, symmetrically on the path ABR, the adsorber is cooled, first isolated, from point R to point D, and then connected to the evaporator (ideally at constant temperature, T_e) from point D back to point A. During this last period, the solid adsorbs the quantity of vapour Δm from the evaporator. This evaporation induces heat extraction, which occurs only during this period: the refrigerating cycle is intermittent.

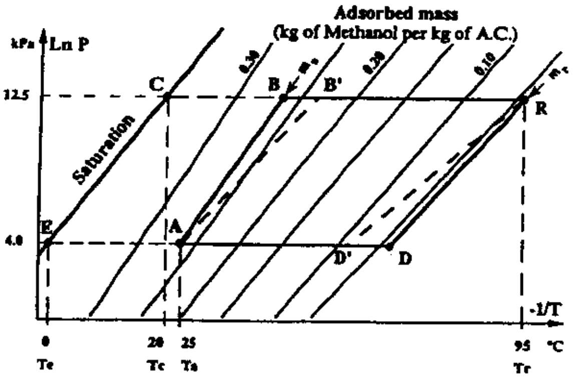

Fig. 1. Ideal adsorptive cycle, A–B–R–D–A, in the Clapeyron diagram (ln P vs $-1/T$) for the AC35 + methanol pair. In the presented cycle, the adsorption state is: $T = 25°\text{C} - P = 4$ kPa (i.e. in equilibrium with an evaporator at 0°C), and the regeneration state is: $T = 95°\text{C} - P = 12.5$ kPa (i.e. in equilibrium with a condenser at 20°C). The cycled amount of adsorbate lies around 0.22 kilogram of methanol per kilogram of A.C. The cycle A–B′–R–D′–A is an example of a real cycle operating in a valveless unit.

An ideal cycle also assumes the presence of at least one valve for alternate connection of the adsorber with the condenser and the evaporator. In a real unit, the cycle is different, mainly for two reasons. First, the adsorber may not be isolated from exchangers at the beginning of heating (or cooling) period, then, exchanger temperatures may change. For instance, in a solar ice maker from which ice is removed every morning, there is no need of any valve between condenser and evaporator. Then, instead of isosteric paths AB and RD, the system follows the paths AB′ and RD′, schematically shown in Fig. 1. The reasons why are explained in Appendix 1. Note that both cycles A–B–R–D–A and A–B′–R–D′–A yield the same cycled mass, Δm, which depends only on points A and R. An example of a real cycle is presented in [14].

Such cycles, ideal or not, operate exclusively with thermal energy.

Solar-powered adsorptive ice makers only comprise an adsorber, which is the solar collector, a condenser, mostly air-cooled, but possibly water-cooled, and an evaporator. There are neither mechanical moving parts nor valves.

Coefficient of performance (COP)

The COP is the ratio of heat extraction during evaporation to heat added to adsorber (collector) during regeneration. Heat balances of collector and of evaporator are described in [13, 14].

In present experiments, solar energy incident on the collectors and weight of produced ice are measured. The net solar COP, COPs, is directly yielded by the ratio of the heat extracted in order to produce the ice actually removed to overall global solar energy incident on the collectors during the considered day: $\text{COPs} = Q_e/E$.

DESCRIPTION OF THE ICE MAKERS

Principle of the experimentation in Agadir

Following conclusions of experimental study of an ice maker prototype, performed in Orsay, and adding a new concept of collecto–condenser, the BLM-France company began commercializing adsorptive solar ice makers using activated carbon (A.C.) + methanol pair, in 1988. Both the French Ministère des Affaires Etrangères and Moroccan Ministère de l'Education Nationale funded experimental study (performed by Agadir and Orsay) of this machine in actual conditions in Morocco. On the other hand some modifications of standardly produced units were supposed to improve performance. Then Agadir, Orsay and BLM decided that two modified units would also be tested in Agadir.

Description of the ice makers

Figure 2 presents the components of the units. They are constructed from only two components connected by a flexible tube.

(a) A collecto–condenser (1 m^2, 90 mm thick) made of two identical stainless steel shells. A grid holds 20 kg of A.C. in the upper shell, which plays the role of solar collector. It is a selective surface (Maxsorb™), single glass, flat plate collector. Internal fins improve

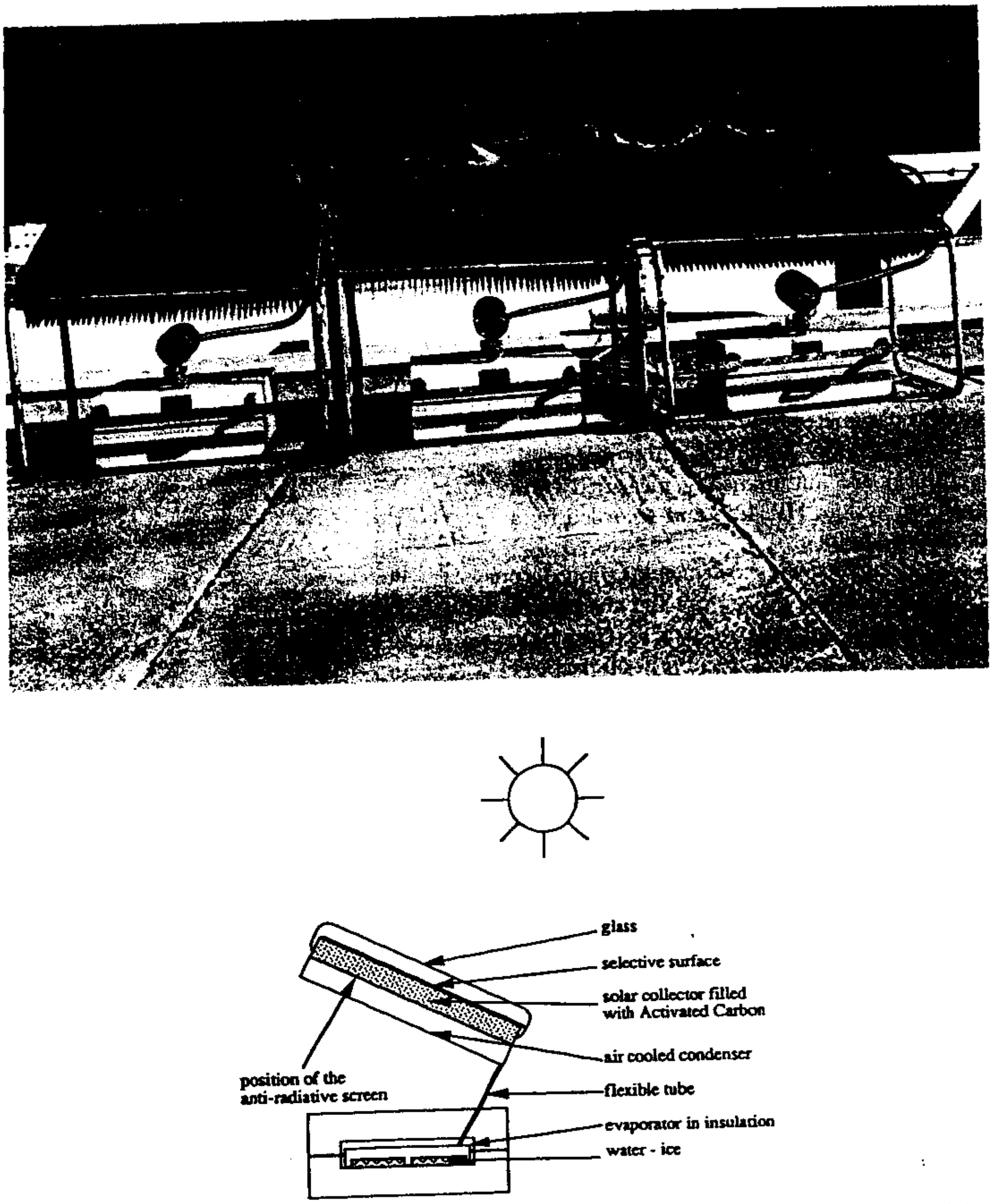

Fig. 2. Schematic of BLM ice makers. Internal and external fins are not presented. The photograph is of three ice makers on the roof of a building at Agadir Science University.

heat transfer between the upper face and the A.C. The bottom shell, with external fins (7.5 m^2), plays the role of air-cooled condenser (natural convection). The main originality of these units, as compared to the split prototype ice maker studied in [12–15], is that BLM has designed this new packaged component: the collecto-condenser [18].

(b) An evaporator (0.3 m^2), which can be immersed in a highly insulated, 5.2 l, water tank.

As previously mentioned, three units are being tested in Agadir. The first one, the standard unit distributed by BLM-France contains the A.C. AC35 supplied by CECA-France. It will be denoted Standard. The second one contains a slightly different A.C., the AC40, also supplied by CECA and which is more activated than AC35. According to Passos [19, 20], this should yield better performance. This unit will be denoted AC40. The third one contains AC35 and an anti radiative screen (ARS) has been installed between the adsorbent and the condenser. This device limits radiative transfer between upper and lower shells of the collecto-condenser, and thus improves the captation efficiency of the collector. This last unit will be denoted ARS.

Temperatures of main components of each unit and meteorological parameters (incident solar power rate on the collector, yielding E by integration, and ambient temperature) are automatically measured by a data acquisition system HP3421, and recorded by a micro-computer Apple//e.

Beside measurements, the experimental procedure followed the instructions of use: 'Set the machine in a sunny place toward south, clean the glass from time to time, fill the water tank every afternoon, remove the produced ice every morning'. The mass and temperature of filling water and of produced ice were recorded.

The photograph shows the three ice makers installed on the roof of a building at Agadir Science University.

EXPERIMENTAL RESULTS

Operation of ice makers began on 25 February 1989. Measurements began on 22 May 1989. The period investigated herein ended on 17 July 1990. From these fourteen months, only 135 days are presented, due to vacations, weekends, operating procedure sometimes different than prescribed, electrical failures.... The Standard unit broke down on 24 February 1990, for a reason still under study. The two other units are still operating with the same performance as in the period described herein.

Example of daily operating conditions

Figure 3 shows typical evolutions of temperatures in the ARS unit from 6 a.m. on 22 May to 6 a.m. on 23 May 1989. After sunrise, the A.C. is heated up from point A to point B′, desorbed vapour condenses in the evaporator (coldest point). The heat released by condensation first melts the ice directly in contact with evaporator walls, which makes the ice removal very easy, as in [13]. After ice removal, the evaporator temperature increases towards condenser temperature. Between point B′ and point R, condensation occurs, mainly in the condenser, but also in the evaporator. Condenser and evaporator temperatures are close, both around 8°C above ambient temperature. After 4 p.m., desorption is over and the water tank is filled. The cooling period is divided into two subperiods. During the first one (4 p.m.–9 p.m.), temperature of the evaporator decreases down to a value lower than 0°C. The liquid water is cooled and undergoes supercooling. In the second one (9 p.m.–6 a.m.), the temperature of the evaporator is roughly constant and water freezes. The simple calculation, developed

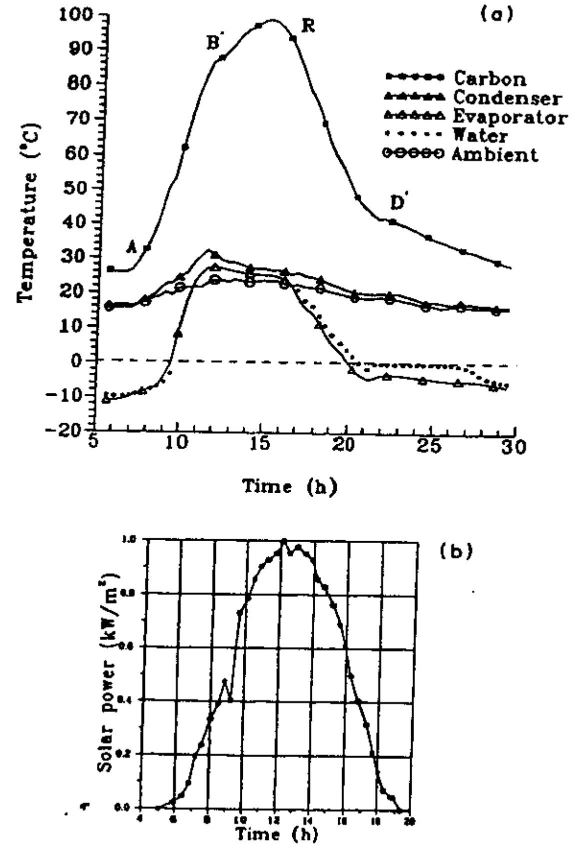

Fig. 3. (a) Recorded temperatures of ARS unit during 22-23 May 1989. (b) Recorded insolation during 22 May 1989. Points A, B′, R and D′ show the states corresponding to points A, B′, R and D′ in the cycle presented in Fig. 1.

Table 1. Characteristic temperatures (in °C) and mass of produced ice (in kg/m^2) for each unit on 22 May, 1989. For this day, $T_d = 24°C$ and $T_n = 16°C$

Unit	T_a	T_r	T_c	T_e	M_i
Standard	24.5	87.5	28	−6.3	5.2
AC40	24.5	87	28	−6	5.2
ARS	28.5	99	27	−6.5	5.2

Table 2. Percentages of (investigated) days yielding a prescribed daily ice production (kg/m^2 per day) for each studied unit

Ice production:	0	0–1	1–2	2–3	3–4	4–5	5.2
AC40	9	4	2	10	16	20	39
ARS	9	1	4	8	14	20	44

in Appendix 2, shows why cooling of the A.C. slows down during the second sub-period, and also why the end of supercooling induces a temperature jump of the A.C., at 9 p.m. The same behaviour was already observed in [14]. At the end of this period, when the whole water is frozen, the ice temperature may lie quite below 0°C. The adsorbent temperature is still higher than ambient temperature.

Table 1 shows the differences between the three units for the day presented in Fig. 3. It shows that Standard and AC40 operate very similarly. In addition, their performances did not exhibit any difference. Therefore, and also because the former stopped operating rather early, only results of the latter will be kept for further study. On the other hand, ARS operates somehow differently. It will be compared with AC40.

Measured ice production

Figure 4 presents measured daily ice productions of AC40 and ARS, M_i, as a function of daily insolation, E. Table 2 presents the detailed proportions of days (among investigated) yielding a prescribed ice production.

For 40% of investigated days, every unit produces 5.2 kg of ice, the temperature of which lies between −3 and −15°C. And the production lies between 4 and 5.2 kg for 20% of investigated days.

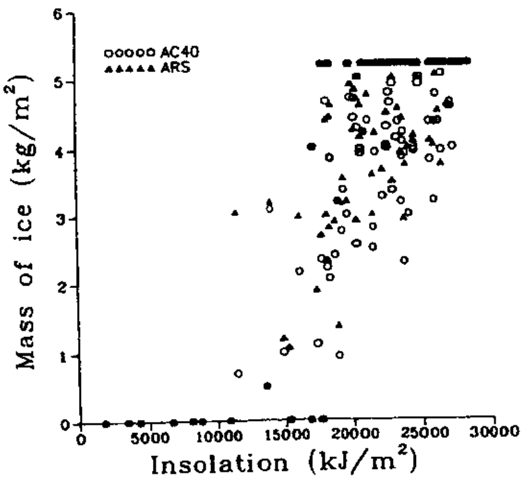

Fig. 4. Experimental ice production as a function of insolation, for AC40 (○) and for ARS (▲). One point represents one day of operation.

The maximal production (5.2 kg) exhibited by Fig. 4, is due to the limited volume of the water tank (5.2 l). Considering the dispersion of points, it can be guessed that the production could have been larger for some days, if the water tank had permitted it. Indeed, measurements show that when ice production is maximal, ice is generally cooled down to temperatures lower than necessary. This shows that heat extraction still developed when the whole water introduced in the water tank was frozen. This limiting effect is confirmed in [21]. On another hand, this limitation is commercially good. A customer will buy this machine for a daily production of 5.2 kg of ice, as claimed by the manufacturer. In Agadir this customer will be satisfied for 40% of days, and rather satisfied (more than 4 kg) for 20% of days. If the water tank were larger, this satisfaction factor would be less.

The heat extraction corresponding to the production of 5.2 kg of ice from water at 20°C lies around 2250 kJ. Figure 4 shows that such an amount of ice can be produced with insolations lying between 18,000 and 28,000 kJ/m^2. This yields a net solar COP, COPs, lying between 0.08 and 0.12.

Figure 4 also shows that ARS has only slightly better performance than AC40. This can be roughly explained as follows: limitation of the heat transfer between adsorbent and condenser improves solar captation efficiency (T_r is indeed larger for ARS, see Table 1), but hinders the cooling of the adsorbent (T_a is also larger for ARS). Both effects almost cancel each other. Further comparison is performed in [21].

Points on Fig. 4 are widely dispersed, because of meteorological conditions. A parametric study of their influences is presented in [21].

IS THE COLLECTO-CONDENSER A GOOD COMPONENT?

From the present study, it can now be concluded that, as it is designed, this component is efficient: the COPs of units studied herein (0.08–0.12) are not much lower than the COPs of the prototype studied in Orsay (0.10–0.12) [14]. As mentioned above, the low value 0.08 is due to the water tank limit. Because of values of COPs higher than 0.10 these units are among the most efficient solar-powered sorption (adsorption or absorption) refrigerating machines [22].

CONCLUSIONS

Solar ice makers using solid adsorption may be quite efficient as shown by experimentation in actual conditions in Agadir, Morocco. Nominal production of 5.2 kg of ice a day is obtained for 40% of days, and a production over 4 kg for 60%. Only 10% of days yield a zero production. The temperature of produced ice lies between −15 and −5°C. The net solar COP lies between 0.08 and 0.12. The three tested units have roughly the same performances.

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NOMENCLATURE

COP	coefficient of performance
COPs	net solar COP
E	incident solar energy on the collectors, J/m^2
HS	global heat losses coefficient of the collecto–condenser, W/K
m	adsorbed mass, kg of methanol per kg of A.C.
m_a	maximal adsorbed mass, kg/kg
m_r	minimal adsorbed mass, kg/kg
M_i	mass of produced ice, kg/m^2 per day
P	pressure, Pa
P_c	pressure of the condenser ($P_c = P_s(T_c)$), Pa
P_e	pressure of the evaporator ($P_e = P_s(T_e)$), Pa
$P_s(T)$	adsorbate saturation pressure at T, Pa
Q_e	heat extraction (net), J/m^2
R	ideal gas constant of the adsorbate, J/kg/K
T	temperature, °C
T_a	adsorption temperature, °C
T_c	condenser temperature, °C
T_d	diurnal ambient temperature, °C
T_e	evaporator temperature, °C
T_n	nocturnal ambient temperature, °C
T_0	ambient temperature, °C
T_r	regeneration temperature, °C
ΔH	isosteric heat of adsorption (often denoted q_{st}), J/kg
ΔH	$= -R \cdot [\partial \ln P/\partial(1/T)]_m$
Δm	desorbed mass during a cycle ($\Delta m = m_a - m_r$), kg/kg
ΣMC_p	total heat capacity: collector + absorbent + adsorbed phase, J/K

Abbreviations

A.C.	activated carbon
AC40	ice maker working with the AC40 + methanol pair
ARS	anti-radiative screen
ARS	ice maker equipped with ARS

REFERENCES

1. R. Chung *et al*., A study of a solar air conditioner. *Mech. Engng* **85**, 31–35 (1963).
2. J. C. V. Chinnappa, Solar operation of ammonia–water multistage air conditioning cycles in the tropics. *Solar Energy* **16**, 165–170 (1974).
3. R. L. Oonk, W. A. Beckman and J. A. Duffie, Modeling of the CSU heating/cooling system. *Solar Energy* **17**, 21–28 (1975).
4. T. R. Penney, *Proc. Desiccant Cooling and Dehumidification Opportunities for Buildings Workshop*, Solar Energy Conservation Training Institute, Chattanooga, Tennessee, U.S.A., 10–11 June (1986).
5. P. Worsøe-Schmidt, Solar refrigeration for developing countries using a solid absorption cycle. *Int. J. Ambient Energy* **4**, 115–124 (1983).
6. M. Balat et G. Crozat, Conception et étude d'un prototype de présérie de réfrigérateur solaire basé sur une réaction solide–gaz. *Int. J. Refrig.* **11**, 308–314 (1988).
7. J. C. V. Chinnappa, Experimental study of the intermittent vapor absorption refrigeration cycle employing the refrigerant–absorbent systems ammonia–water and ammonia–lithium nitrate. *Solar Energy* **5**, 1–18 (1961).
8. W. Sloetjes, J. Haverhals, K. Kerkdijk, I. Osman Ahmed, H. Saber, S. Shaa Eldin, R. Porsius, A. Stolk, A. El Karib, K. Yousif and H. Wardi Hassan, Operational results of the 13 kW/50 m^3 solar-driven cold store in Khartoum, the Sudan. *Solar Energy* **41**(4), 341–347 (1988).
9. M. D. Staicovici, An autonomous solar ammonia–water refrigeration system. *Solar Energy* **36**, 115–124 (1986).
10. M. Clerx and G. J. Trezek, Performance of an aqua–ammonia absorption solar refrigeration at sub-freezing evaporator conditions. *Solar Energy* **39**, 379–389 (1987).
11. D. I. Tchernev, Utilisation of solar energy for refrigeration and air conditioning. *Refrigeration Science and Technology, Paris* **205–211** (1982).
12. A. Boubakri, Ph. Grenier and M. Pons, Utilisation of activated carbon and methanol pair to solar ice production. *Proc. JITH Conf., Rabat, Morocco* **1**, 69–79 (1985).
13. M. Pons and J. J. Guilleminot, Design of an experimental, solar-powered, solid adsorption ice maker. *J. Solar Energy Engng* **108**, 332–337 (1986).
14. M. Pons and Ph. Grenier, Experimental data on a solar-powered ice maker using activated carbon and methanol solid adsorption pair. *J. Solar Energy Engng* **109**, 303–310 (1987).
15. M. Pons, Ph. Grenier and A. Boubakri, Experimental, solar-powered, solid adsorption ice maker. *Proc. ASME/JSME/JSES Solar Energy Conf. Honolulu, Hawaï, U.S.A.* 867–872 (1987).
16. Ph. Grenier, J. J. Guilleminot, F. Meunier and M. Pons, Solar powered solid adsorption cold store. *J. Solar Energy Engng* **110**, 192–197 (1988).
17. D. M. Ruthven, *Principles of Adsorption and Adsorption Processes*. Wiley-interscience, New York (1984).
18. G. Paeye and A. Guyader, Brevet français No 2574530.

19. E. F. Passos, F. Meunier and J. C. Gianola, *Heat Recovery Systems* **6**, 259–264 (1986).
20. E. F. Passos, J. P. Escobedo and F. Meunier, *Solar Energy* **42**, 103–111 (1989).
21. A. Boubakri, M. Arsalane, B. Yous, L. Ali-Moussa, M. Pons, F. Meunier and J. J. Guilleminot, Experimental study of adsorptive solar-powered ice makers in Agadir (Morocco)—2. Influences of meteorological parameters. *Renewable Energy* **2**(1), 15–21 (1992).
22. J. Bougard, The state of art of solar-driven sorption refrigeration machines. To be published in Pcdgs of I.I.F.-I.I.R. Conf., Dresden, 1990, Commissions B2–C2–D1–D2/3.

APPENDIX 1

When the system is in state A, an increase in adsorbent temperature tends to increase pressure. If the adsorber is still connected to the evaporator, vapor will be desorbed and condensed in the evaporator. In a refrigerator, heat release in the evaporator must absolutely be prevented. Then the adsorber must be isolated, e.g. with a valve. Conversely, for an ice maker, once ice is removed, it does not matter whether condensation occurs in the evaporator, and heats it from roughly 0°C up to condenser temperature: there is no need for a valve. Then, the cycle will not be the ideal cycle A–B–R–D–A described in Fig. 1. Indeed, the amount of adsorbate that condenses in the evaporator for heating it is obviously desorbed, then the system does not follow the isosteric path AB, but a path AB′, schematically shown in Fig. 1. Symmetrically, at the beginning of cooling period, the evaporator is at the same temperature as the condenser (around ambient temperature). While the adsorber cools down, the evaporator is progressively cooled by evaporation down to freezing temperature. Then the system does not follow the isosteric path AD, but a path AD′, schematically shown in Fig. 1.

Moreover, ambient temperature is not constant and condenser area is finite. Then, condenser temperature is not constant, and desorption is not isobaric. After point D′, evaporation pressure is more likely to be constant: while the water freezes, the evaporator temperature remains roughly constant.

APPENDIX 2

Let us consider the cooling of the collecto-condenser, assuming its temperature is homogeneous. The collecto-condenser is cooled by natural convection, and as it is connected with the evaporator, the A.C. adsorbs methanol. Then the heat balance is written:

$$(\Sigma MC_p)\cdot\frac{dT}{dt}-M\cdot\frac{dm}{dt}\cdot\Delta H=-HS\cdot(T-T_0) \quad \text{(A1)}$$

as $dm=(\partial m/\partial T)_P\cdot dT+(\partial m/\partial P)_T\cdot dP$; and $dP=(dP_s/dT)\cdot dT_c$; eq. (A1) transforms in:

$$\left[(\Sigma MC_p)-M\cdot\left(\frac{\partial m}{\partial T}\right)_P\cdot\Delta H\right]\cdot\frac{dT}{dt}=-HS\cdot(T-T_0)$$
$$+M\cdot\left(\frac{\partial m}{\partial P}\right)_T\cdot\left(\frac{dP_s}{dT}\right)\cdot\Delta H\cdot\frac{dT_c}{dt}. \quad \text{(A2)}$$

As $(\partial m/\partial T)_P$ is negative and ΔH positive, the factor of dT/dt in l.h.s. is positive. And, as $(\partial m/\partial P)_T$ and (dP_s/dT) are positive, the factor of dT_c/dt in r.h.s. is positive too. Then dT/dt is an increasing function of dT_c/dt and dT/dt is negative when dT_c/dt vanishes (the first term of r.h.s. is negative). This simply explains why the temperature decrease of the collecto–condenser is slower when water freezes (T_c roughly constant) than before (T_c decreasing). On another hand, when supercooling takes end, there is a short but rapid increase in T_c: dT_c/dt is positive. Equation (A2) shows that if dT_c/dt is large enough, dT/dt can be positive too, as exhibited in measurements (Fig. 3).

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EXPERIMENTAL STUDY OF ADSORPTIVE SOLAR-POWERED ICE MAKERS IN AGADIR (MOROCCO)—2. INFLUENCES OF METEOROLOGICAL PARAMETERS

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Abstract—Performances of the solar powered ice-makers tested in Agadir, Morocco, as described in [1], are studied. Experimental data are sufficient for establishing a statistical correlation between heat extraction and meteorological parameters: insolation, diurnal and nocturnal temperatures. The influences of these parameters are also evaluated through operation analysis. Both approaches are rather consistent. Nocturnal temperature appears to be a quite important factor. Indeed, cold nights (around 10°C) largely increase production. Meteorological conditions for producing either just no ice at all or 5.2 kg, are investigated, as well as an evaluation of yearly production: in Agadir, the average daily production lies around 4 kg/m². With a nocturnal cooling system, the units would produce around 1 kg more. From the economic point of view, small units such as those tested herein are too expensive, but solarization of a pre-existing cold store in isolated areas looks competitive.

INTRODUCTION

Solid adsorption can be efficiently used for solar refrigeration. Three adsorptive solar powered ice makers are sited in Agadir, Morocco. They are made of only two components: the collecto–condenser and the evaporator. They operate with an activated carbon (A.C.)+methanol pair. The first unit is denoted Standard, with the A.C. denoted AC35. The second one is denoted AC40, with the A.C. denoted AC40. And the third one, denoted ARS, operates with AC35 and is equipped with an Anti-Radiative Screen (ARS) that limits heat transfer between the adsorbent and the condenser [1].

Reference [1] presents these units, and their performances in terms of ice production. The purpose of the present study is to investigate their performances in terms of heat extraction, and to correlate them to insolation, diurnal and nocturnal temperatures. A detailed impact of the ARS will also be studied.

ANALYSIS OF HEAT EXTRACTION

Evaluation of heat extraction from the measurements

The water introduced in the water tank, M_w, undergoes a temperature decrease, ΔT_w. If it reaches freezing temperature, some ice is produced, M_i. When the whole water is frozen ($M_i = M_w$), the ice may undergo a temperature decrease ΔT_i. The net heat extraction, Q_e, can easily be deduced from measurements of M_w, ΔT_w, M_i and ΔT_i. Therefore, Q_e will be considered as the measured heat extraction

$$Q_e = M_w \cdot C_{pw} \cdot \Delta T_w + L_f \cdot M_i + M_i \cdot C_{pi} \cdot \Delta T_i. \quad (1)$$

Figure 1 presents Q_e vs E for the 135 investigated days and for the ARS unit. For a better under-

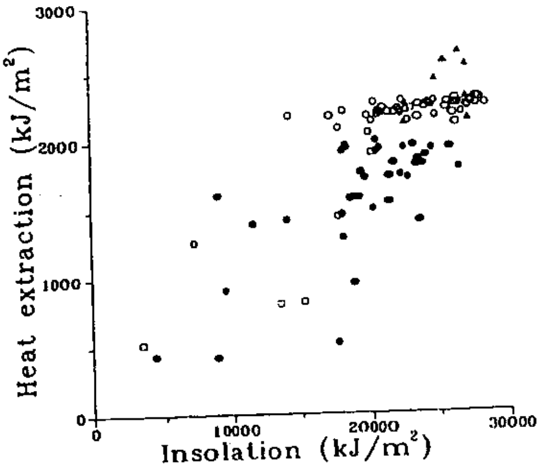

Fig. 1. Heat extraction for the 135 investigated days and for the ARS unit as a function of insolation. ○: points undergoing the Water Tank Limit effect; □: points undergoing the First Changing Day effect; ●: points kept for statistical study; ▲: points with enlarged water tank.

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standing, we have separated the points in different groups. Except for those points denoted ▲, which will be explained later on, Fig. 1 shows many points with Q_e lying between 2200 and 2400 kJ/m^2, denoted ○. As explained in [1], for these days, the whole water introduced in the water tank was frozen. The limited volume of the water tank induces a limited ice production. In addition, it induces a limited heat extraction. Indeed, in such conditions, one has: $M_i = M_w = 5.2$ kg, and $\Delta T_i \neq 0$. Thus, at the end of evaporation, T_e decreases (for instance, this occurred on the day presented in Fig. 3 in [1]). Individual influences of characteristic temperatures are described in the Appendix. $(\partial Q_e/\partial T_e)$ is positive, then Q_e is lower than what it could have been: the limited volume of the water tank affects performance. This effect will be denoted as the Water Tank Limit (WTL).

The non-periodicity of meteorological conditions induces another effect. For instance, during a poorly insolated day following a sunny day, operating conditions are absolutely not cyclic. Of course, this affects performance. The few days concerned (denoted □ in Fig. 1) by this effect will be denoted by First Changing Day (FCD). Consequences of WTL and FCD effects will be discussed later on.

The remaining points, denoted ● in Fig. 1,† still present a wide dispersion. A parametrical study will be performed with only latter points.

PARAMETRICAL STUDY

Before studying on meteorological influences, the number of investigated parameters must be fixed. Let us begin from the description of the cycle [1]. It is clear and well known that heat extraction depends on the four temperatures, T_a, T_e, T_r and T_c (see also the Appendix). According to the measurements performed in Agadir, T_e is almost constant when there is no WTL (between -5 and 0°C), and does not depend on meteorological conditions. This was expected because T_e mainly depends on the ice melting point. Therefore, Q_e depends on the three other temperatures and parametric study will be performed with three parameters. Of course insolation and ambient temperature are involved. Probably so are wind conditions, but the complete lack of experimental data on wind velocity prevented investigation of its influence. Moreover, alternating day–night is well represented by two values of ambient temperature. As points R and A (see Figs 1 and 3 in [1]) are the most important in the cycle, we have considered ambient temperature values at about 4 p.m. (when desorption ends), denoted by the diurnal temperature, T_d, and at sunrise (around 6 a.m.), denoted by the nocturnal temperature, T_n. The correlation between Q_e, E, T_d and T_n can now be studied.

Correlated between heat extraction and meteorological parameters

We kept the points satisfying the two following conditions:

(i) $Q_e < 2200$ kJ/m^2 (i.e. no WTL),
(ii) difference of insolation between the considered day and the day before, lower than 7500 kJ/m^2 (i.e. not a FCD).

Then a linear correlation between Q_e, E, T_d and T_n is calculated from these points by the least square method. It results in:

$$Q_e = 1000 + 0.095E - 18T_d - 53T_n, \quad \text{for AC40,} \quad (2)$$

$$Q_e = 1600 + 0.085E - 18.5T_d - 67T_n, \quad \text{for ARS,} \quad (3)$$

(with E and Q_e in kJ/m^2, T_d and T_n in °C).

Figure 2 presents comparison of experimental Q_e and values calculated with eqs (2) and (3), respectively. Some dispersion remains, that might be due to wind conditions, but is much less than in Fig. 1: the agreement is satisfactory.

Discussion of coefficients

(a) *Influence of insolation*: $(\partial Q_e/\partial E)$. This coefficient can be interpreted as an average marginal COPs. For

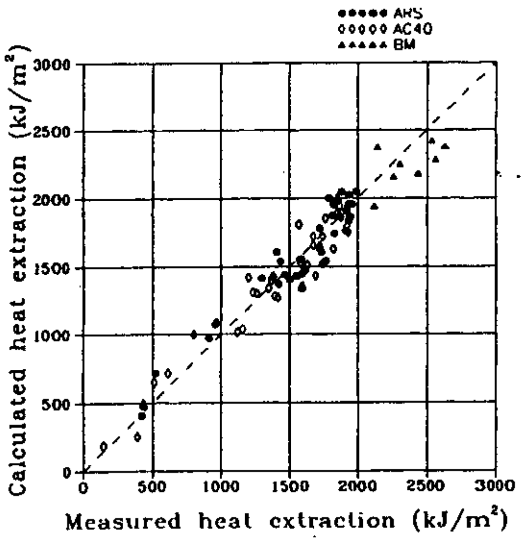

Fig. 2. Comparison of experimental heat extraction with values calculated with help of eqs (2) and (3). ● : points for ARS unit; ◇ : points for AC40 unit; ▲: points for ARS unit with enlarged water tank.

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† Figure 1 presents the performance of ARS. As shown in Fig. 4 in [1], results for AC40 are very similar.

the days investigated in this section, experimental COPs lies between 0.06 and 0.11. This is quite consistent with statistical values of $(\partial Q_e/\partial E)$ in eqs (2) and (3): 0.095 for AC40, and 0.085 for ARS.

(b) *Influence of diurnal temperature*: $(\partial Q_e/\partial T_d)$. In a first approximation, it can be assumed that when T_d changes (and only T_d), T_r and T_c will vary in equal quantity. The Appendix permits the evaluation of $(\partial Q_e/\partial T_d)$, yielding: -13 kJ/m²/K for AC40, and -14.6 for ARS. These values are rather in good agreement with statistical values in eqs (2) and (3): -18 and -18.5, respectively.

(c) *Influence of nocturnal temperature*: $(\partial Q_e/\partial T_n)$. A direct correlation between T_n and T_a, as previously between T_d, T_r and T_c, is not correct. Indeed, T_a also depends on E (via T_r) and on Q_e (because the heat of adsorption slows down the cooling of the adsorbent, see Appendix 2 of [1]). Another linear correlation,

$$T_a = a\cdot E + b\cdot Q_e + c\cdot T_n + d \qquad (4)$$

is established from experimental data with the least squares method. It yields:

$$T_a = 220\times10^{-6}E + 2.11\times10^{-3}Q_e + 0.86T_n + 1.46 \quad \text{for the AC40} \qquad (5)$$

$$T_a = 67\times10^{-6}E + 5.27\times10^{-3}Q_e + 1.60T_n + 1.03 \quad \text{for the ARS} \qquad (6)$$

(with E and Q_e in kJ/m², T_a and T_n in °C).

In this case, T_a is the only characteristic temperature, heat extraction depends on: T_n obviously does not influence T_r and T_c, and, as previously mentioned, T_c does not change. Then, derivating eq. (4) yields:

$$\mathrm{d}Q_e = \left(\frac{\partial Q_e}{\partial T_a}\right)\cdot \mathrm{d}T_a = \left(\frac{\partial Q_e}{\partial T_a}\right)\cdot[a\cdot \mathrm{d}E + b\cdot \mathrm{d}Q_e + c\cdot \mathrm{d}T_n], \qquad (7)$$

where $(\partial Q_e/\partial T_a)$ is the marginal influence of T_a on heat extraction, as defined in the Appendix. By resorting terms in $\mathrm{d}Q_e$, and with $\mathrm{d}E = 0$, eq. (7) yields:

$$\left(\frac{\partial Q_e}{\partial T_n}\right) = \frac{c\cdot\left(\frac{\partial Q_e}{\partial T_a}\right)}{1 - b\cdot\left(\frac{\partial Q_e}{\partial T_a}\right)}. \qquad (8)$$

With values of $(\partial Q_e/\partial T_a)$ extracted from the Appendix, eq. (8) yields the following values of $(\partial Q_e/\partial T_n)$: -113 and -76 kJ/m²/K, for AC40 and for ARS, respectively. For ARS, the agreement with the statistical value (-67) is quite good. The discrepancy for AC40 (statistical value $= -53$) can be due to an inaccurate equation of state, or to another effect not accounted for in this interpretation. This point is still under study.

Anyway, the influence of nocturnal ambient temperature is much more significant (at least three times) than for the diurnal temperature.

First Changing Day effect

Few such days were collected: four first bad days and six first sunny days (denoted □ in Fig. 1), and conclusions are not easily drawn. Comparing measured and calculated (with eqs (2) and (3)) values of heat extraction yields an extra heat extraction lying between 300 and 500 kJ/m² for three first bad days and a difference lower than 200 kJ/m² for the other days. Thus it seems that first bad days have a better performance than they should. This can be explained as follows. In normal operation in a sequence of sunny days, adsorption temperature, T_a, is 8–12°C higher than ambient temperature, T_n. At the end of the first bad day, T_a is lower than the day before. This decrease in T_a induces an extra heat extraction (see Table 1 in the Appendix), which occurs only because it is the first bad day. The difference between T_a and T_n, which exists in normal operation, acts as a cold storage, which is consumed during the first bad days. This cold storage is probably restored during sunny days. We expected first sunny days to have lower heat extraction than evaluated with eqs (2) and (3), but did not find it out. The cold storage is probably not restored in one sunny day but in several.

This effect was mentioned in a study by simulation [2], but has not yet been pointed out experimentally.

Water Tank Limit effect

We formerly said that the accumulation of points with Q_e lying between 2200 and 2400 kJ/m², when insolation lies between 20,000 and 30,000 kJ/m² (see Fig. 1), is due to the water tank limit. For ascertaining this point, in May 1990, some experiments were performed with an enlarged water tank (9.5 l) for unit ARS. Corresponding points are denoted ▲ in Fig. 1. They show that heat extraction can exceed 2500 kJ/m². Our interpretation (heat extraction limited by water tank volume) is correct.

Comparison of these latter points with values calculated with eq. (3) does not yield a difference greater than dispersion on other points (see Fig. 2, points denoted ▲). Extrapolation of eqs (2) and (3) is then expected to be valid.

CUT-OFF LIMIT

The statistical correlations, eqs (2) and (3), yield some answers to the user's question: in which con-

Table 1. Average values of T_a, T_r, T_c and T_e, and individual marginal influences of T_a, T_r, T_c and T_e on heat extraction (in kilojoule per kilogram of A.C.) for each studied unit

Unit	T_r	T_a	T_c	T_e	$\partial q_c^*/\partial T_r$	$\partial q_c^*/\partial T_a$	$\partial q_c^*/\partial T_c$	$\partial q_c^*/\partial T_e$
AC40	81.7	23.6	33.5	−4.2	5.43	−9.14	−6.08	9.74
ARS	89.6	27	32.5	−5.6	4.42	−6.28	−5.15	6.93

ditions do these units produce no ice at all, and when is nominal production (5.2 kg) achieved?

In order to define the quantity no ice at all, we have assumed a systematical (e.g. automatic) filling of the water tank, pouring 5.2 l of water in it. The sensible heat released by this quantity between ambient and freezing temperatures is 500 kJ. In such conditions, a heat extraction of 500 kJ/m^2 will produce no ice (note that 500 kJ are enough for freezing 1 kg of water).

It was mentioned in [1] that production of 5.2 kg of ice corresponds to a heat extraction of 2250 kJ/m^2.

Let us now introduce two cut-off values of insolation. E_0 is the threshold insolation, below which no ice is produced ($Q_c = 500$ kJ/m^2). E_5 is the minimal insolation for nominal ice production of 5.2 kg ($Q_c = 2250$ kJ/m^2). Actually, E_0 and E_5 are not values, but functions of T_n and T_d, according to eqs (2) and (3).

Remember that in places in Morocco like Rabat, Marrakech or Agadir, daily insolation, E, lies between 20,000 and 30,000 kJ/m^2 for 62–73% of days, and that 2000 and 30,000 kJ/m^2 are lower and upper limits of E. The dependance of E_0 and E_5 on T_d and T_n, mainly on T_n, will be investigated with prescribed values of $(T_d - T_n)$: low values of $(T_d - T_n)$ are schematic of tropical climates, high values correspond to desertic (Saharan) climates. In Agadir, the average value of $(T_d - T_n)$ is 8°C, and climate is rather temperate: Agadir is situated on the Atlantic Coast.

Figure 3 presents the functions E_0 and E_5 vs T_n; with $(T_d - T_n) = 0$ and 16°C. It appears that there is no large difference between the studied units, and between the climates: nocturnal temperature is a major factor. When nights are cold ($T_n \approx 10$°C), some ice is produced even with very low insolation ($E_0 < 5000$ kJ/m^2), and as soon as insolation exceeds 20,000 kJ/m^2, 5.2 kg of ice are produced. When nights are warm ($T_n \approx 30$°C), insolation must exceed 20,000 kJ/m^2 in order to produce some ice, and nominal production can never be achieved.

These conclusions are quite important for development of this technology. In Agadir, the average value of T_n is 17°C, which leads to good performances: more than 4 kg of ice per day and per square meter for 60% of days. Performances will not be as good in climates with warmer nights. On the other hand, in cold night climates, they will be excellent.

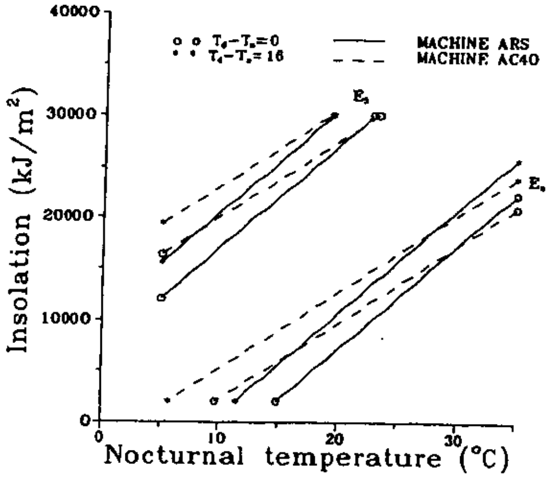

Fig. 3. Meteorological conditions for a zero (E_0) and for a nominal (E_5) production. Full line: ARS unit, dashed line: AC40 unit, 0: $T_d - T_n = 0$°C (tropical climate), *: $T_d - T_n = 16$°C (desert climate).

Nocturnal temperature is an important factor for these ice makers. This point can be explained by the characteristics of the adsorptive pair. For the A.C.+methanol pair, the performance of the cycle is strongly dependent on the adsorption temperature T_a, see [3] and Table 1. With another pair, e.g. zeolite+water, the units will not behave in the same way. Besides, serious improvement can be achieved by means of a nocturnal cooling system. The presently studied units have no nocturnal cooling system, unlike the prototype studied in Orsay [4–7], where dampers could be open during nights: then air could flow between collector and glass. When operating, this system yielded a value of T_a reduced by 4°C. According to Table 1 in the Appendix, this would yield an extra cold production of about 730 kJ/m^2 (1.5 kg of ice) for AC40, and about 500 kJ/m^2 (1 kg of ice) for ARS.

IMPACT OF ANTI-RADIATIVE SCREEN

It was mentioned in [1] that the ARS has only minimal influence on ice production. According to statistical correlations, eqs (2) and (3), the ARS improves the average performance by 120 kJ/m^2.

From Table 1 in the Appendix, the reason why this device does not change performance can be developed. On one hand, regeneration is improved: regeneration temperature is increased because of better captation efficiency (reduced rear losses) and condenser temperature is lower because of reduced heat transfer from adsorbent to condenser. On the other hand, adsorption is hindered. Indeed, adsorption temperature is higher, mainly for two reasons: initial temperature before cooling, T_r, is larger, and in addition the cooling rate is reduced (reduced rear losses). These two competing effects nearly cancel. The finally slightly better performance of ARS is also confirmed by the lower average value of T_e. (But as mentioned about the WTL effect, lower value of T_e reduces the improvement due to ARS.) The improvement due to ARS, as evaluated in the Appendix, is 180 kJ/m², which is rather consistent with the statistical value (120).

In addition, the ARS unit seems to have a stronger sensitivity on T_n. This point appears very clearly on Fig. 3. With a rather low T_n (e.g. $T_n = 15$, $T_d = 23$, $E = 7500$), the ARS unit produces 1 kg of ice when AC40 just chills the water. For such days, the improvement of captation efficiency has unbalanced positive consequences on performance.

The ARS would bring maximal improvements in units equipped with a nocturnal cooling system, and with a consequently enlarged water-tank.

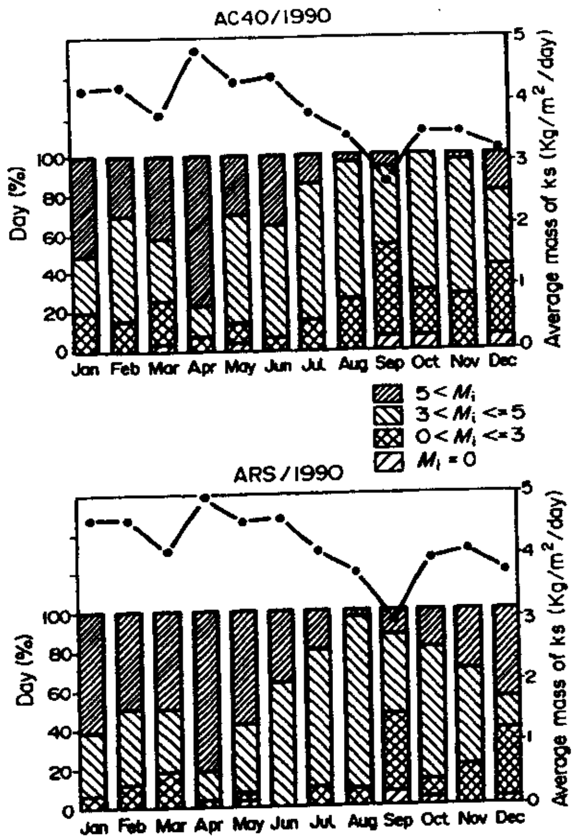

Fig. 4. Curve: evaluation of monthly averaged daily ice production, with help of eqs (2) and (3) and with meteorological data in Agadir. Histogram: proportion of days, in each month, yielding a prescribed ice production: (zero), (0–3), (3–5), (over 5 kg). Summation of daily ice production yields following total yearly production: 1410 kg/m² for AC40, and 1580 for ARS.

ECONOMIC EVALUATION

Evaluation of yearly ice production

From 22 May 1989 to 17 July 1990, full experimentation was not performed every day, but meteorological parameters were recorded. Then, from E, T_d and T_n, Q_e can be evaluated with help of eqs (2) and (3). Ice production, M_i, is then deduced from eq. (1), where $\Delta T_i = 0$, $\Delta T_w = 20$ and $M_w = 5.2$ kg (systematic filling of the water tank). M_i must lie between 0 and 5.2 kg. Figure 4 presents the monthly averaged M_i, and the proportions, within each month, of days yielding M_i either zero, less than 3 kg, between 3 and 5 kg, or more than 5 kg. Seasonal variations appear quite well. Production is maximal in April, because of sunny days and cold nights, and minimal in September, because of cloudy days. Production is better from January to June than from July to December.

The average daily production lies around 4 kg/m² which yields a yearly total production of about 1500 kg/m².

Economic evaluation

In isolated areas, e.g. in Senegal (Africa), ice may cost around 0.2 U.S.\$/kg, mainly because of transportation. Assuming 5 years as a reasonable pay back period, such units should cost less than 1500 U.S.\$/m², including transportation. Small units like those described herein were produced in France at a cost of around 2000 U.S.\$/m², not including transportation. This is too expensive. However, considering now solarization of a pre-existing cold store, the overcost mainly concerns the collecto–condensers. As they can be produced for less than 1000 U.S.\$/m², such a solution looks competitive.

CONCLUSION

Experimental study of the three units tested in real conditions in Agadir allows us to draw some conclusions.

For more than 60% of investigated days, each unit produces more than 4 kg of ice per square meter. The yearly averaged production lies around 4 $\mathrm{kg/m^2}$. Neither an alternative A.C., the AC40, nor the implementation of an anti-radiative screen improves average performance, although the latter modifies operating conditions.

Meteorological parameters have been investigated. The net solar COP lies between 0.08 and 0.12. Nocturnal ambient temperature is a very important factor: when its exceeds 23°C the units cannot produce the nominal quantity of ice (5.2 kg), and when it exceeds 30°C, production is very low.

The collecto–condenser is a good concept, but some improvements could be performed. Collecto–condensers equipped with an anti-radiative screen and with a nocturnal cooling system, would produce daily at least 1 $\mathrm{kg/m^2}$ more.

Economic viability of small units such as those tested in Agadir, is unlikely, but solarization of existing refrigeration systems in isolated areas looks competitive.

Statistical correlations are established from experimental data. They represent the units behaviour. With them, heat extraction of those units in other climates can be predicted very easily. To this end, necessary meteorological data are: daily insolation, diurnal temperature (around 4 p.m.) and nocturnal temperature (around 6 a.m.).

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NOMENCLATURE

COP	coefficient of performance
C_{pi}	specific heat of ice, J/kg/K
C_{pl}	specific heat of liquid absorbate, J/kg/K
C_{pw}	specific heat of liquid water, J/kg/K
E	incident solar energy on the collectors, $\mathrm{J/m^2}$
$E_0(E_5)$	values of E for production of "no ice" (5.2 kg of ice)
E'	parameter of Dubinin–Astakhov equation, kJ/kg
L	latent heat of evaporation of adsorbate, J/kg
$L = -R\cdot[\mathrm{d}\ln P_s/\mathrm{d}(1/T)]$	
L_f	latent heat of fusion of ice, J/kg
m	adsorbed mass, kg of methanol per kg of A.C.
m_a	maximal adsorbed mass, kg/kg
m_r	minimal adsorbed mass, kg/kg
M	mass of adsorbent in the adsorber (collector), $\mathrm{kg/m^2}$
M_i	daily production of ice, $\mathrm{kg/m^2}$
M_w	mass of water introduced in the water tank, $\mathrm{kg/m^2}$
n	exponent of Dubinin–Astakhov equation
P	pressure, Pa
P_c	pressure of the condenser ($P_c = P_s(T_c)$), Pa
P_e	pressure of the evaporator ($P_e = P_s(T_e)$), Pa
$P_s(T)$	saturation pressure of the adsorbate at T, Pa
q_e^*	heat extraction per kg of A.C. (gross), J/kg
Q_e	heat extraction (net), $\mathrm{J/m^2}$
R	ideal gas constant of the adsorbate, J/kg/K
T	temperature, °C
T_a	adsorption temperature, °C
T_c	condenser temperature, °C
T_d	diurnal ambient temperature, °C
T_e	evaporator temperature, °C
T_n	nocturnal ambient temperature, °C
T_r	regeneration temperature, °C
W_0	volume of micropores in adsorbent, $\mathrm{m^3/kg}$
Δm	desorbed mass during a cycle ($\Delta m = m_a - m_r$), kg/kg
ΔT_i	temperature decrease undergone by the ice after total freezing
ΔT_w	temperature decrease undergone by water before freezing
$\rho_{(T)}$	volumetric mass of adsorbed phase, $\mathrm{kg/m^3}$

Abbreviations

A.C.	activated carbon
AC40	unit operating with the AC40 + methanol pair
ARS	anti-radiative screen
ARS	unit equipped with ARS
FCD	first changing day
WTL	water tank limit

REFERENCES

1. A. Boubakri, M. Arsalane, B. Yous, L. Ali-Moussa, M. Pons, F. Meunier and J. J. Guilleminot, Experimental study of adsorptive solar-powered ice makers in Agadir (Morocco)—1. Performance in actual site. *Renewable Energy* **2**(1), 7–13 (1992).
2. F. Lemmini, Thesis, Université Mohamed V, Faculté des Sciences, Rabat, Morocco (1990).
3. Ph. Grenier, J. J. Guilleminot, I. Ismail and M. Pons, *Proc. XVIth I.I.R. Conf., Paris* 353–361 (1983).
4. A. Boubakri, Ph. Grenier and M. Pons, Utilisation of activated carbon and methanol pair to solar ice production. *Proc. JITH Conf., Rabat, Morocco* **1**, 69–79 (1985).
5. M. Pons and J. J. Guilleminot, Design of an experimental, solar-powered, solid adsorption ice maker. *J. Solar Energy Engng* **108**, 332–337 (1986).
6. M. Pons and Ph. Grenier, Experimental data on a solar-powered ice maker using activated carbon and methanol solid adsorption pair. *J. Solar Energy Engng* **109**, 303–310 (1987).
7. M. Pons, Ph. Grenier and A. Boubakri, Experimental, solar-powered, solid adsorption ice maker. *Proc. ASME/JSME/JSES Solar Energy Conf., Honolulu, Hawaii U.S.A.* 867–872 (1987).
8. B. P. Bering, M. N. Dubinin and V. V. Serpinski, On thermodynamics of adsorption in micropores. *J. Colloids Interface Sci.* **38**, 185–194 (1972).
9. M. Pons and Ph. Grenier, A phenomenological adsorption equilibrium law... *Carbon* **24**, 615–625 (1986).

APPENDIX

According to the description of the cycle, the adsorption and regeneration points (points A and R in Fig. 1 in [1]) are defined by four temperatures, even when the cycle is not ideal: T_a, T_e, T_r and T_c. Considering 1 kg of adsorbent and the cycled amount of adsorbate, $\Delta m = m_a - m_r$, the gross heat extraction is yielded by eq. (A1):

$$q_e^* = \int_{T_c}^{T_e} \Delta m \cdot C_{p1} \cdot \mathrm{d}T + \int_{m_r}^{m_a} L_{(T)} \cdot \mathrm{d}m. \qquad \text{(A1)}$$

Assuming that the system follows the ideal path R–D–A during adsorption–evaporation period (see Fig. 1 in [1]), and that C_{p1} is constant between T_e and T_c, eq. (A1) transforms in:

$$q_e^* = (m_a - m_r) \cdot [L_{(T_e)} - C_{p1} \cdot (T_c - T_e)]. \qquad \text{(A2)}$$

Derivating eq. (A2) yields:

$$\mathrm{d}q_e^* = (\mathrm{d}m_a - \mathrm{d}m_r) \cdot [L_{(T_e)} - C_{p1} \cdot (T_c - T_e)]$$
$$+ (m_a - m_r) \cdot [(\mathrm{d}L_{(T)}/\mathrm{d}T) \cdot \mathrm{d}T_e) - C_{p1} \cdot (\mathrm{d}T_c - \mathrm{d}T_e)], \qquad \text{(A3)}$$

where

$$\mathrm{d}m_a = (\partial m/\partial T)_P \cdot \mathrm{d}T_a + (\partial m/\partial P)_T \cdot (\mathrm{d}P_s/\mathrm{d}T) \cdot \mathrm{d}T_e, \qquad \text{(A4a)}$$

$$\mathrm{d}m_r = (\partial m/\partial T)_P \cdot \mathrm{d}T_r + (\partial m/\partial P)_T \cdot (\mathrm{d}P_s/\mathrm{d}T) \cdot \mathrm{d}T_c. \qquad \text{(A4b)}$$

Partial derivatives in eqs (A4) are calculated: with $T = T_a$ and $P = P_e = P_s(T_e)$ in eq. (A4a), and with $T = T_r$ and $P = P_c = P_s(T_c)$ in eq. (A4b). For computing these partial derivatives, one needs an equation of state. We used the well-known equation of Dubinin–Astakhov, as presented in [8, 9]. The correlation $m = m(T, P)$ is written:

$$m = W_0 \cdot \rho_{(T)} \cdot \exp\left[-\left(\frac{\mathrm{R} \cdot T \cdot \ln \dfrac{P_{s(T)}}{P}}{E'}\right)^n\right]. \qquad \text{(A5)}$$

For AC35, values of parameters are the ones published in [9], i.e.

$$W_0 = 425 \times 10^{-6}\ \mathrm{m^3/kg}\ \text{(of A.C.)}, \quad n = 2.15,$$

and $E' = 221$ kJ/kg (of methanol).

For AC40, values of parameters were determined by fitting eq. (A5) on experimental data provided by the manufacturer of AC40, CECA, France. These data lie in the domain ($20 < T < 100$°C) and ($0.7 < P < 12.7$ kPa). The corresponding values are:

$$W_0 = 611 \times 10^{-6}\ \mathrm{m^3/kg}\ \text{(of A.C.)}, \quad n = 1.53$$

and $E' = 173$ kJ/kg (of methanol).

Values of $(\partial q_e^*/\partial T_x)$, with $T_x = T_a$, T_e, T_c or T_r, can be computed with help of eqs (A3)–(A5). Table 1 presents the average operating temperatures of AC40 and ARS units, and the corresponding values of $(\partial q_e^*/\partial T_x)$.

Multiplying eq. (A1) by M, and subtracting losses (cooling of evaporator and water tank, and heat input in the water tank from ambient air through insulation) yields the net heat extraction. As only marginal influences are investigated in this section, it will be assumed that losses are not affected by marginal variations. Marginal influences can then be written: $M \cdot (\partial q_e^*/\partial T_x)$; with $T_x = T_a$, T_e, T_c or T_r.

Influence of meteorological parameters

Operating temperatures are modified by meteorological conditions, especially ambient temperature. Once the influence of the latter on the former is established, its influence on performance can be evaluated with the help of Table 1.

Effect of ARS

In order to evaluate this effect, unit ARS should be compared with Standard, because the ARS is the only difference between them. Unfortunately, failure of this latter unit prevents direct comparison. However, let us remember that AC40 and Standard units operated very similarly, as mentioned in [1]. Then, operating temperatures of AC40, presented in Table 1, can be assumed to be the ones of Standard. Let us denote ΔT_x, with $x = a - r - c$ or e, the difference of average operating temperatures between ARS and AC40. For instance, $\Delta T_a = (T_a$ for ARS$) - (T_a$ for AC40$)$. ΔT_xs are the differences of operating conditions induced by the ARS. As the A.C. in ARS and Standard is AC35, the effect of ARS on performance will be evaluated via eq. (A6):

$$\Delta Q_e = \left(\sum_{x=a,r,c,e} \left(\frac{\partial q_e^*}{\partial T_x}\right) \cdot \Delta T_x\right) \cdot M, \qquad \text{(A6)}$$

where $(\partial q_e^*/\partial T_x)$s are the ones in Table 1 for ARS.