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A PROPOSED SOLAR DESALINATION-HEATING SYSTEM WITH HEAT RECOVERY

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نظام شمسي مقترح للتقطير والتسخين مع الإسترجاع الحراري

خلاصة - يتناول هذا البحث جهازا مزدوجا لتسخين وتقطير المياه بالطاقة الشمسية . ويتكون هذا الجهاز بصفة أساسية من مجمع للطاقة الشمسية و مرطب هواء و خزان مياه ومكثف مبرد بالمياه الداخلة . وقد تم وصف الجهاز وطريقة عمله ، مع عرض شامل للتصميم الحراري له . وتتلخص الفكرة الأساسية للجهاز في الإستفادة من كل من الحرارة المحسوسة للهواء والحرارة الكامنة للبخار المصاحب له لتسخين مياه التبريد بالمكثف . والجهاز بسيط في بناءه وتشغيله وكذلك التحكم به . ولذلك فهو يعتبر إختيار مناسب للمناطق النائية التي تعتمد على المياه الجوفية والآبار . وقد تم عرض الأداء الحراري للجهاز عند ظروف تشغيل مختلفة حيث كانت النتائج مشجعة .

ABSTRACT - In the present work, a solar desalination-heating system with heat recovery is proposed. The system includes a solar collector, humidifier, storage tank and a water-cooled condenser. The thermal design of a typical unit is presented. Both latent heat of the condensed vapour and the sensible heat of the circulating air are recovered, and added as a sensible heat to the inlet water in the condenser. The system is simple in construction, operation and control. Therefore, it is a suitable option for remote areas where only the ground water, or wells are available. The thermal performance of the system is also studied under different operating conditions. Results have shown that this system is highly recommended for practical application.

INTRODUCTION

The conventional-type solar still has been subjected to intensive work in both theoretical and practical fields [1-4]. Also, various designs for a solar still have been proposed and investigated. The target of these researches was to enhance and maximize the distillate output from the still. The still output can be enhanced by increasing the evaporating surface, either by using the inclined-try concept [5], or by replacing the basin by an inclined surface covered by wet cloth [6,7]. The rate of energy absorbed mayalso be increased by employing a black dye in the basin water [8]. The rate of evaporation can be enhanced by bubbling dry air in the basin water, which also leads to cooling of the glass cover [9].

On the other hand, Assouad et al [10] have proposed a system with latent heat recovery. The system consists of a humidifier through which atmospheric air is blown to a conventional-type solar still. The wet air from the still flows to a seawater-cooled condenser from which the condensate is collected. The warm water from the condenser is collected in a pond to supply the still basin and the humidifier. Although the system analysis has shown a good efficiency and productivity with respect to the conventional solar still, it seems to have a considerable amount of operation control. Also, the air entering the solar still is saturated with moisture (from the humidifier). Therefore, one can expect that its ability to pick vapour from the still will be reduced.

However, the present system utilizes both latent heat of condensing vapour, and the sensible heat of circulating air. But it differs from that of Assouad et al [10] in both function and theory of operation. In the system. solar radiation is converted to thermal energy for heating the raw water. Some of this water is evaporated and then condensed to supply the drinking water demand.

SYSTEM DESCRIPTION AND OPERATION

The system main components are shown in Fig. 1. Raw water is supplied through the condenser, and circulates between the storage tank, collector and the humidifier. The water absorbs latent heat of condensed vapour and the sensible heat of the circulating air in the condenser, and solar radiation in the collector. In the humidifier, the descending hot water mixes with the ascending air stream from the condenser (or atmosphere). Therefore, a part of this water is evaporated and carried away by the circulating air. This hot moist air flows back to the condenser where both latent heat of the condensed vapour and its sensible heat are given to the raw water. The humidifier output water is mixed with that from the condenser in the storage tank from which the hot water demand is supplied. Also, this tank accommodates any sudden variations in the flow rates and temperatures. The air circulation between the humidifier and condenser is performed by a suitable fan.

The system capacity may be enhanced by employing an auxiliary electric heater in parallel with the solar collector. This heater can be manually or automatically switched on to operate the system at night and off-sunshine periods. If the distilled water is required to be increased, the hot water output can be totally or partially recirculated back to the humidifier through the auxiliary heater. The system contains also some ordinary and nonreturn valves which are useful in controlling water flow in different lines.

SIMULATION PROCEDURE

The system performance is simulated under the following conditions:

1. Single glass cover flat-plate collector with ordinary black paint absorber.
   The collector has the following values :
   Tilt angle = 30° South facing ($\gamma = 0$) Latitude = 30° N
   $U_L = 6\ W/m^2$ $(\alpha\tau)_e = 0.8$ $F' = 0.9$
2. Relative humidity of air entering the humidifier = 0.4
   and the absolute pressure = 1.02 bar

3. Pressure drop in the humidifier = 0.02 bar

The following assumptions are made to simplify calculations:

a) Heat capacitance of the system components are neglected with respect to that of water.
b) The air leaving humidifier is saturated, with a temperature equal to that of entering water.
c) Water leaves the humidifier at a temperature equal to that of entering air
d) Ideal heat exchange in the condenser.
e) Heat losses from the humidifier, condenser and pipes are neglected.
f) Steady state operation.

According to these assumptions, the energy balance equation for a unit collector area is given by :

$$\dot{m}\, cp_w\ (T_{w4} - T_{w3}) = F'[\ S - U_L\ (T_p - T_a\ )\ ]\ , \tag{1}$$

Where, $\dot{m}$ is the mass flow rate per unit collector area, T_p is the absorber plate mean temperature,
and T_a is the ambient air temperature.
In the calculations, T_p is approximated by $(T_{w3}+T_{w4})/2$ for simplicity.
The humidifier energy balance equation for 1 Kg water entering is given by :

$$h_{f4} + w_1 m_{da} h_{v1} = m_{da} cp_a (T_{a2} - T_{a1}) + w_2 m_{da} h_{v2} + m_o\ h_{f5}\ , \tag{2}$$

and the mass balance equation :

$$1 - m_o = m_{da}(w_2 - w_1)\ , \tag{3}$$

where, $w_1 = (0.622\ P_v\ ) \ /\ (P_{a1} - P_v)$;

$w_2 = (0.622\ P_s\ ) \ /\ (P_{a2} - P_s)$;

and $m_o = \dot{m}_o \ /\ \dot{m}$ = Hot water output ratio,

$m_{da} = \dot{m}_{da} / \dot{m}$ = Dry air ratio.

The condenser heat and mass balance equations (for 1 Kg water circulated) are given by :

$$cp_w(T_{w2} - T_{w1}) = m_{da} cp_a(T_{a2} - T_{a1}) + m_v\ h_{fg2}\ , \tag{4}$$

$$m_v = m_{da}(w_2 - w_1\ )\ , \tag{5}$$

where, $m_v = \dot{m}_v \ /\ \dot{m}$ = Condensate ratio.

The heat balance equation for the storage tank, assuming complete mixing yields,

$$T_{v3} = (T_{v3} + m_o T_{v5}) / (1 + m_o) \tag{6}$$

Calculations are carried out for three different meteorological conditions; namely summer, spring (equinox) and winter. The total solar radiation on a tilted surface was estimated by the clear day model (for latitudes less than 45°) following Anderson [9],

$$H = Z\ [\ \cos\alpha \sin\beta \cos(\gamma_s - \gamma) + \sin\alpha \cos\beta\ ] + 0.5\ Z\ A\ (1 + \cos\beta)$$
$$+0.5\ \rho_s Z\ (\ 1 - \cos\beta\ )\ (\ \sin\alpha + B\ )\ , \tag{7}$$

where, $Z = H_o \ \mathrm{Exp}(\ -A / \sin\alpha\ )$.

The value of ρ_s is taken as 0.2, and constants H_o, A, B are given by,

	H_o, W/m^2	A	B
June 21,	1088.3	0.205	0.134
March 21,	1151.2	0.177	0.092
December 23,	1233.2	0.142	0.057

The ambient air temperature T_a, is calculated from,

$$T_a = 0.5\ (T_{ax} + T_{an}) - 0.5\ (T_{ax} - T_{an})\ \cos[\ \pi\ (\tau - 3) / 12\ ]\ ,$$

where, T_{an}, T_{ax} are the minimum and maximum ambient air temperatures respectively.
System of equations (1-7) are solved by iteration with a time step of 30 minutes, from sunrise to sunset hours for the three selected conditions (days).

RESULTS AND DISCUSSION

Figure 2. shows the total solar radiation on the tilted collector surface, and the ambient temperature for the three days at which calculations are performed. In order to generalize the results, the system performance is obtained for a unit collector area, and each flow rate is given as a ratio to that of circulated water. For steady state operation, the amount of water in the storage tank should be kept unchanged. Therefore, the water flow rates through the collector and condenser are equal, while the amount of hot water output is the same as that coming from the humidifier.

Figures 3, 4 and 5 show the variations in hot water temperature and the condensate ratio with time in the selected days, for a flow rate of 10 $Kg/hr.m^2$. The dry air ratio and thermal efficiency for heating only = $\dot{m}\ cp_w (T_{wd} - T_{w1})/H$, are also given. Condensate ratio is a good measure to the system efficiency for desalination. The following observations can be seen from these figures :

1. The maximum temperature of hot water output ranges between 51 and 57°C depending on the time of the year, and occurs around 1 p.m. This temperature is comparable to that of an ordinary solar heating system.

2. The condensate ratio increases with time to a maximum value at 1 p.m. This maximum occurs in equinox, sense the solar radiation is normal to the collector surface at that time. Its value ranges between about 0.11 in 21 December and 0.15 in 21 March.
3. Dry air ratio, which is the minimum amount of dry air required for 1 Kg water entering the humidifier, decreases with time and has a minimum value at 1 p.m. It decreases as the water temperature increases. Therefore, a minimum value of about zero is required on 21 March.
4. The change in thermal efficiency with time is marginal. It has an average value of about 0.5 over the year for this particular flow rate.

On the other hand, the collector flow rate strongly affects the system performance up to about 40 $Kg/hr.m^2$ as shown in Fig.6. Hot water temperature and condensate ratio decrease as the flow rate increase, but the dry air ratio and thermal efficiency increase in the same sense. In fact, a compromise between the hot and distillate water demands is needed to decide the operating flow rate. However, a considerable value of condensate ratio, with a reasonable output hot water temperature can be attained if the operating flow rate ranges between 10 and 20 $Kg/hr.m^2$. At this range of flow rate, the dry air ratio is low (which means a lower power consumption for air circulation), and the thermal efficiency is reasonable.

CONCLUSION

A solar desalination-heating system with heat recovery is proposed and theoretically investigated. Results have shown that the system can operate all over the year, with a thermal efficiency of 40-50% , for water heating . The fresh water yield of the system can reach about 15% of the input raw water, when the collector flow rate is 10 $Kg/hr.m^2$. Results of this work can be a suitable guide for more theoretical and experimental study in this field. The system which is simple and easy to control, is highly recommended for practical application.

NOMENCLATURE

C_p	specific heat at constant pressure.
F'	collector plate efficiency factor.
H	total solar radiation on a tilted surface.
h	specific enthalpy.
$\dot{m}$	mass flow rate.
m	ratio of mass flow rate to that of the collector.
P	absolute pressure.
P_v	vapour pressure.
S	absorbed solar energy by the collector absorber.
T	temperature.
U_L	collector overall heat loss coefficient.
w	absolute humidity.
α	solar altitude angle.

$(\alpha\tau)_e$ collector effective absorption transmittance coefficient.
β collector tilt angle.
γ, γ_s collector and azimuth angles respectively.
τ local time in hours.
ρ_s surrounding reflectivity.

Subscripts
a air
da dry air
f liquid
o output
s saturated condition
v vapour
w water
1,2, point location

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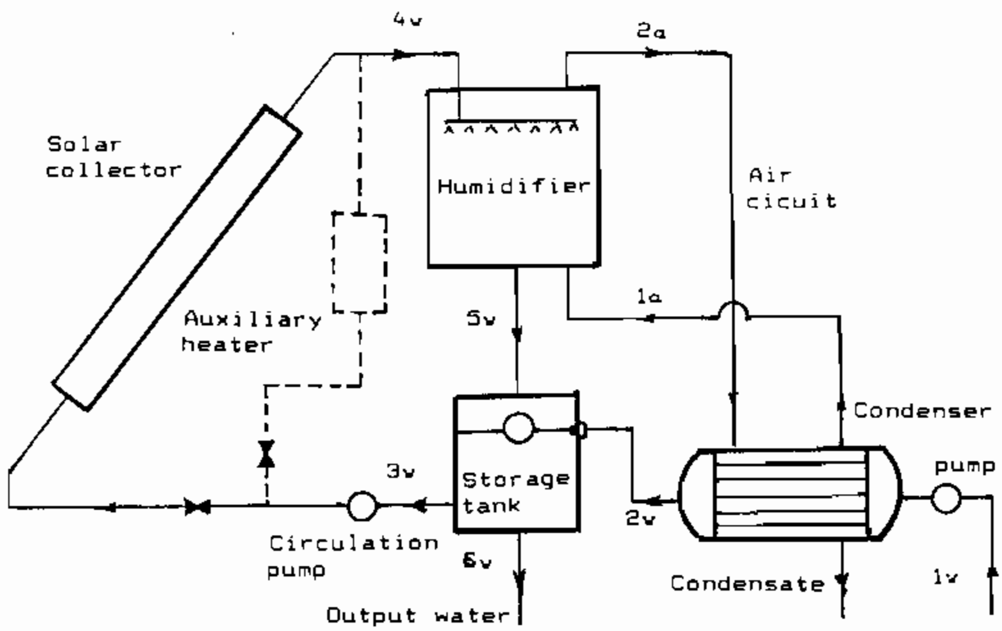

Fig. 1. Schematic diagram of the system.

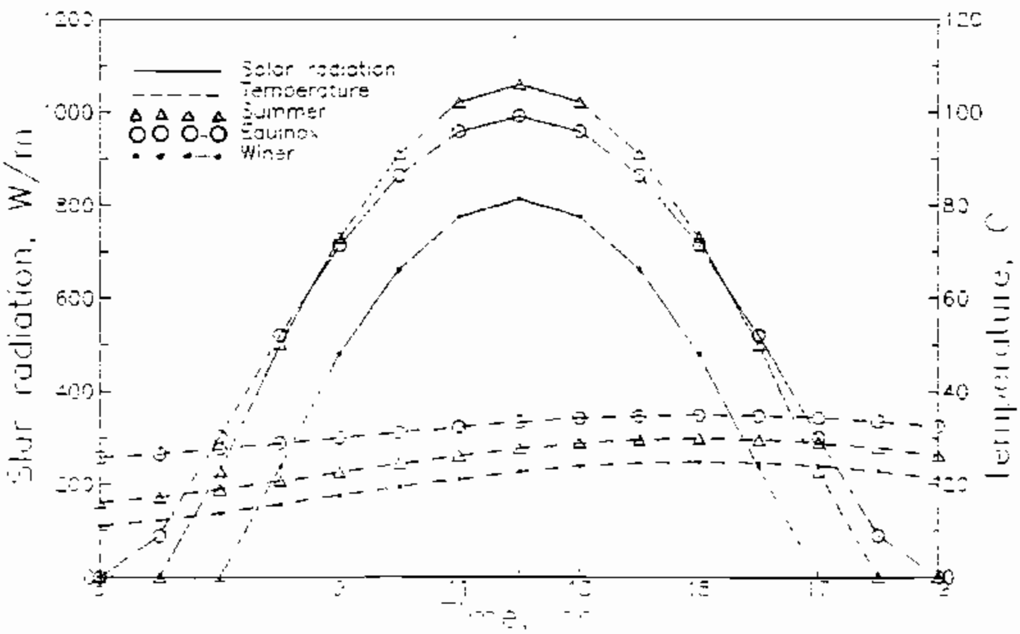

Fig. 2. [illegible]

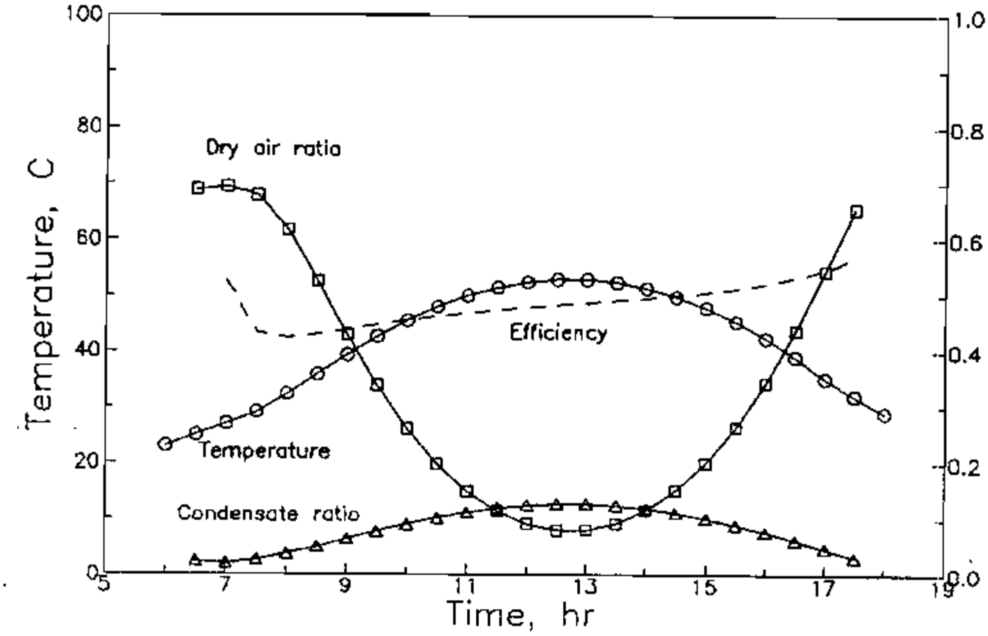

Fig. 3. System performance in a Summer day. 21 June ($\dot{m}$=10 Kg/hr.m^2).

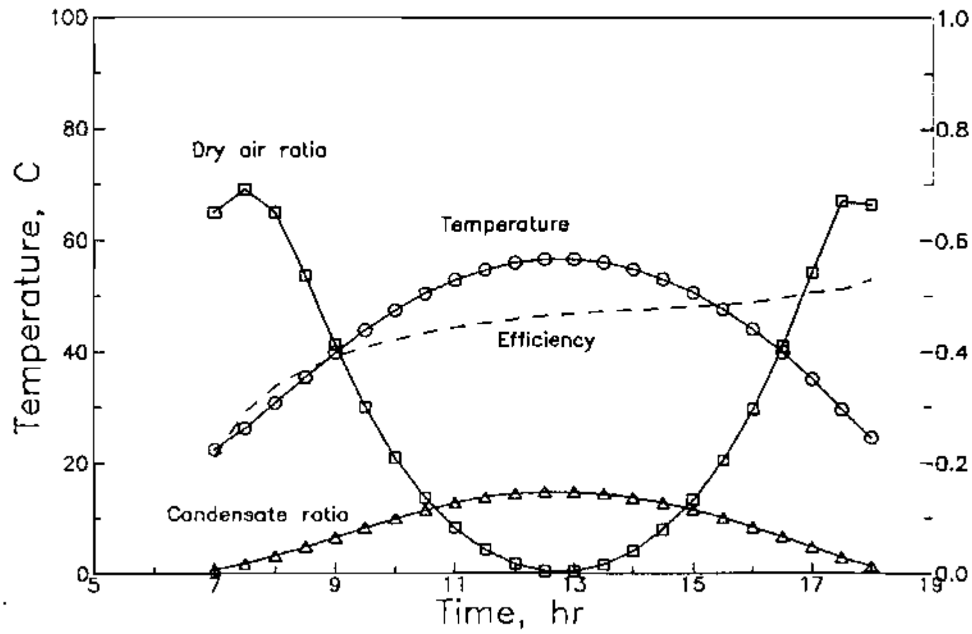

Fig. 4. System performance in a Spring day. 21 March ($\dot{m}$=10 Kg/hr.m^2).

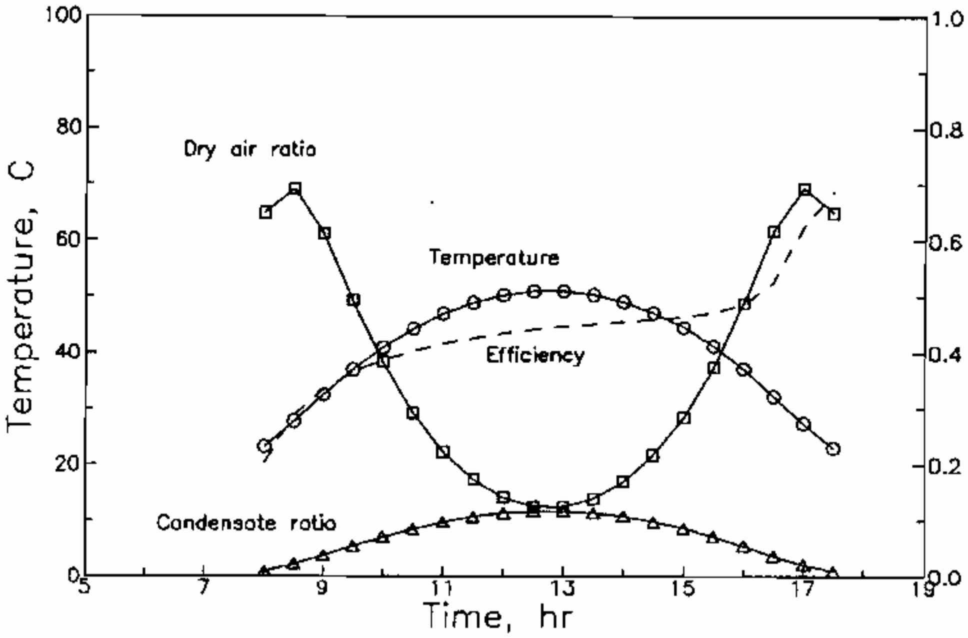

Fig. 5. System performance in a Winter day. 21 December ($\dot{m}$=10 Kg/hr.m^2).

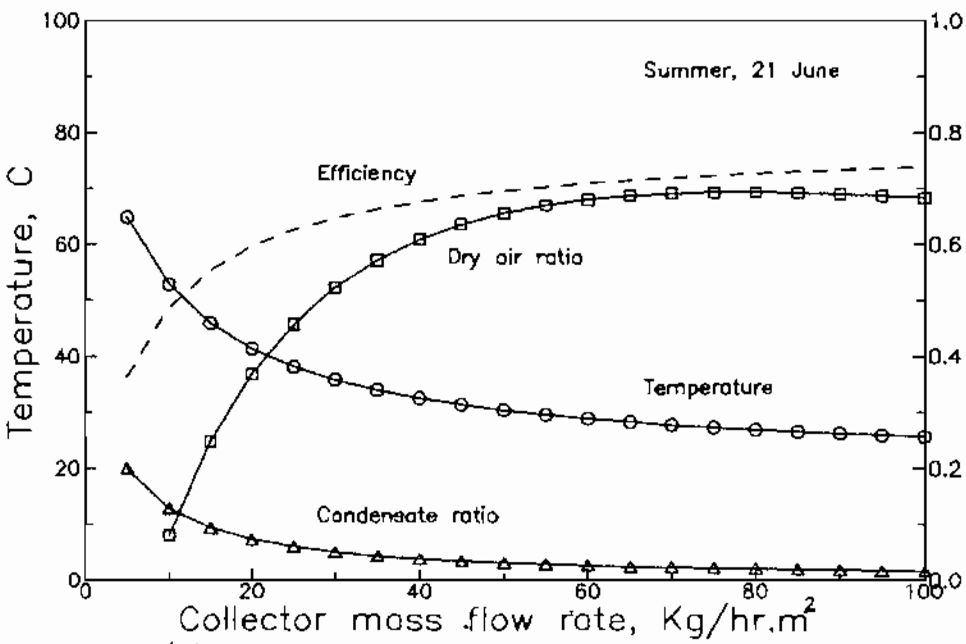

Fig. 6. Effect of collector mass flow rate on the system performance.