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Analysis of an innovative water desalination system using low-grade solar heat

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Abstract

This paper presents a theoretical analysis and preliminary experimental results for an innovative water desalination system using low-grade solar heat. The system utilizes natural means (gravity and atmospheric pressure) to create a vacuum under which water can be rapidly evaporated at much lower temperatures and with less energy than conventional techniques. The system consists of an evaporator connected to a condenser. The vapor produced in the evaporator is driven to the condenser where it condenses and is collected as a product. The effect of various operating conditions, namely, withdrawal rate, depth of water body in the evaporator, temperature of the heat source, and condenser temperature, on the system performance were studied. Numerical simulations and preliminary experimental results show that the performance of this system is superior to a flat-basin solar still, and the output may be twice that of a flat-basin solar still for the same input. Vacuum equivalent to 4 kPa (abs) or less can be created depending on the ambient temperature at which condensation takes place.

Keywords: Low-grade solar heat; Solar desalination; Vacuum; Solar still; Desalination

1. Introduction

Fresh water resources are under heavy pressure due to population growth rates and pollution caused by industrial wastes. A very small fraction, about 0.3%, of the available water resources is available as fresh water [1]. A drinking water shortage is expected to become one of the biggest problems facing the world. To compensate for this, desalination of saline water appears to be the best solution, since the only inexhaustible source of water is the ocean.

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Desalination processes consume significant amounts of energy, and many countries in the world, particularly those suffering from severe water shortages, cannot afford the energy required for desalination. Fortunately, many of those countries lie in areas with high insolation rates. Therefore, solar desalination can be a suitable alternative, provided efficient technologies are developed to utilize the solar energy in a cost-effective way. Solar energy can be used to produce fresh water directly in a solar still or indirectly where the thermal energy from a solar energy system is supplied to a desalination unit.

A number of efforts have been made to develop and improve the performance of solar desalination systems, particularly solar stills. To increase the temperature of the water inside the still, some researchers [2–4] suggested coupling the still to solar collectors. The results showed an improvement in the still's performance. One of the main reasons behind the low efficiency of solar stills, which is about 30–40% [1], is the loss of the latent heat of condensation to the environment and the sensible heat carried away by the condensate. The use of latent heat of condensation to preheat the feed water has shown good improvement in the still's performance [5,6]. The use of latent heat of condensation of one stage to evaporate water in another stage, as in multi-effect stills, has been studied by many researchers showing very good improvement in the still's performance [7,8]. Other researchers [9,10] have investigated the concept of evaporation at low temperatures under vacuum conditions and reported good improvement in the system performance. However, they used vacuum pumps, which require additional energy input to the system.

This paper presents a theoretical analysis and preliminary test results of a solar desalination system which utilizes natural forces — atmospheric pressure and gravity — to create a vacuum. The new concept has the advantages of vacuum distillation without requiring additional energy to create the vacuum.

2. Proposed system and its operating principle

The atmospheric pressure is equivalent to the hydrostatic pressure generated by a column of water of about 10 m high. So, if a column of height more than 10 m and closed from the top is filled with water and the water is allowed to fall under the effect of gravity, it will fall to a height of about 10 m, creating a vacuum in the part above. The desalination system analyzed in this study makes use of the above concept, which was first proposed by Sharma and Goswami [11]. It consists of a solar heating system and an evaporator and a condenser at a height of about 10 m above ground level, connected via pipes to a saline water supply, concentrated brine discharge, and a fresh water tank, all at the ground level. Fig. 1 shows a schematic of the system. A vacuum is created by balancing the hydrostatic and the atmospheric pressures in the supply and discharge pipes.

The evaporator has a provision to supply it with solar or other low-grade thermal energy through a closed loop heat exchanger. The feed water and concentrated brine discharge pass through a tube-in-tube heat exchanger in order to extract the maximum possible energy from the hot brine. The evaporator is connected to a condenser, which dissipates the heat of condensation to the environment. Provisions are made for periodically flushing the system and restarting it, thus removing any accumulated non-condensable gases.

If two chambers containing water are connected together, water will distill from the higher vapor pressure side to the other. The vapor pressure of seawater is about 1.84% less than that of fresh water over the temperature range of 0–100°C. This means that if a saline water chamber (evaporator) and a fresh water chamber

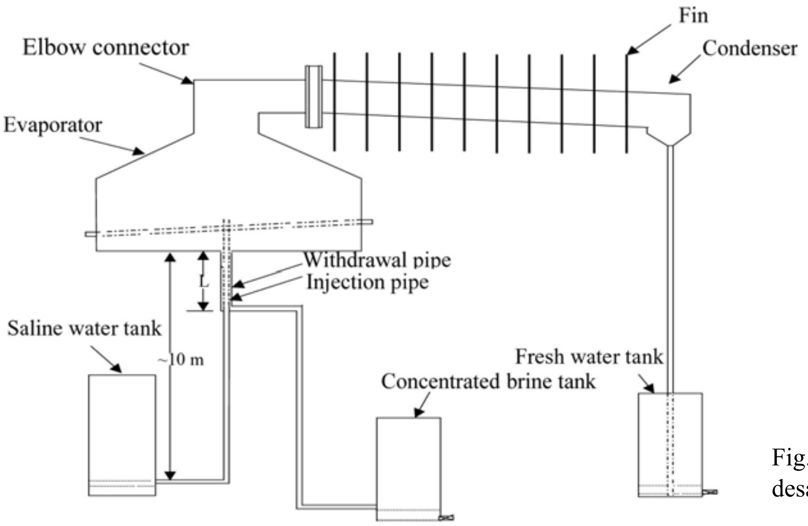

Fig. 1. Schematic of the experimental desalination system.

(condenser) are connected from the top while maintained at the same temperature, water will distill from the fresh water side to the saline water side. In order to maintain distillation of water from the saline water to the fresh water side, the vapor pressure of the saline water must be kept above that of the fresh water by maintaining it at a higher temperature. In the proposed process, this will be done by utilizing solar energy. Under vacuum conditions water can be evaporated at a low temperature level, thus requiring a smaller amount of thermal energy. This energy may be provided from simple flat- plate solar collectors, which will operate at a higher efficiency because of lower operating temperatures.

Evaporation from saline water increases its salinity, which tends to decrease the evaporation rate and increase the chances of scale formation. So it becomes necessary to withdraw the concentrated brine at a certain flow rate and inject saline water at a rate equivalent to the sum of the withdrawal and evaporation rates. The withdrawn brine will be at a high temperature (the evaporator temperature), so it is important to recover energy from it. A tube-in-tube heat exchanger may be used for this purpose, such that the supply water flows in the inner tube and the withdrawn water flows in the annulus in the opposite direction. The heat exchanger area should be sized to recover a major part of the energy. Under vacuum conditions at the saline water surface in the evaporator, the supply water can be injected by the effect of atmospheric pressure; hence no pumping power is required. This makes the proposed system of a continuous process type, unlike a flat-basin solar still, which is usually a batch process.

3. Theoretical analysis

To simulate the system performance, mass, energy and salt balances are needed. In applying those balances, it is assumed that no temperature stratification occurs in the system. Application of conservation of mass, solute concentration, and energy gives the following equations.

- Conservation of mass:

$$\rho_i \dot{V}_i = \rho_w \dot{V}_w + \rho_e \dot{V}_e \tag{1}$$

- Conservation of solute concentration:

$$\frac{d}{dt}(\rho C V)_s = (\rho C)_i \dot{V}_i - (\rho C)_s \dot{V}_w \tag{2}$$

- Conservation of energy:

$$\frac{d}{\mathrm{d}t}\left(\rho C_p VT\right)_s + \frac{d}{\mathrm{d}t}\left(\rho C_p VT\right)_{\text{evaporator}} = Q_{\text{input}} + \left(\rho C_p T\right)_i \dot{V}_i - \left(\rho C_p T\right)_s \dot{V}_w - Q_e - Q_{\text{loss}} \quad (3)$$

The heat input to the system is assumed to be in the form hot water from a flat-plate solar collector. Heat of evaporation is calculated as given by Bemporad [9] after accounting for the pressure drop between the evaporator and condenser. The heat of evaporation will mainly be dissipated to the environment via the condenser during the process of vapor condensation and the rest will be carried away by the fresh water produced. So the condenser will dissipate the amount of energy given by:

$$Q_c = \dot{m} h_{fg}^* \quad (4)$$

where h_{fg}^* is the modified latent heat of condensation [12],

$$h_{fg}^* = h_{fg} + 0.68\, C_{pf}\left(T_s - T_{ci}\right) \quad (5)$$

This amount of heat is conducted through the condenser wall, and eventually transferred to the environment by convection (if we neglect radiative transfer). For heat conduction through the condenser wall,

$$\dot{m} h_{fg}^* = \frac{2\pi l_c k_c \left(T_{ci} - T_{co}\right)}{\ln\left(r_{co}/r_{ci}\right)} \quad (6)$$

For convective heat transfer to the ambient, the condenser is assumed to be a horizontal tube with circular fins. The rate of heat transferred from the condenser (fins and prime surface) may be calculated as

$$Q_c = \left(h_{co,\text{tip}} N A_{f,\text{tip}}\, \eta_f + h_{co} N A_{f,\text{sides}}\, \eta_f + h_{co} A_b\right)\left(T_{co} - T_a\right) \quad (7)$$

where the heat transfer coefficients may be calculated from the relations given by Rohsenow et al. [13]. Heat loss to the environment is assumed to be due to natural convection only.

The operating pressure of the proposed unit may be taken as the sum of the pressure in the vapor space at the point of condensation and the pressure drop occurring in the column. This pressure drop can be calculated by applying continuity equation and the energy equation between the inlet and outlet of the column:

$$\left(\rho A v\right)_{\text{in}} = \left(\rho A v\right)_{\text{out}} \quad (8)$$

$$\frac{P_{\text{out}}}{\gamma} + \frac{v_{\text{out}}^2}{2g} + z_{\text{out}} = \frac{P_{\text{in}}}{\gamma} + \frac{v_{\text{in}}^2}{2g} + z_{\text{in}} - h_L \quad (9)$$

Because of a larger volume of vapor in vacuum distillation, the tubing should be as short and as wide as possible to avoid any significant impedance to vapor flow in the connecting pipes. It is a normal practice to use connecting tubes of diameter larger than (1/10) of the evaporator diameter, at least in the upper part of the evaporator, where the lowest pressure prevails and flooding is most likely to take place [14].

4. Results and discussion

For all simulations the system specifications and dimensions were assumed to be the same as the ones in the actual experimental system. The coil supplying heat to the saline water is a copper tube of 2.4 m length and 1.27 cm outside diameter. The evaporator is a cylinder of 0.2 m^2 cross sectional area, 0.2 m height, with a truncated cone attached on top of it. The evaporator has a provision for feed water supply, through a 1.27 cm copper tube, enclosed by a 2.54 cm PVC pipe that is used for withdrawing the concentrated brine. The two pipes form a tube-in-tube heat exchanger. The condenser is a 4" copper tube of 0.5 m length and 0.25 cm thickness. On its lateral surface, 10 fins of 25.4 cm diameter and 0.0635 cm thickness are soldered 4 cm apart. The other end of the condenser is connected to a condensate receiver via a 1.27 cm PVC pipe. For theoretical simulations, a reference state was fixed as 25°C reference temperature, 3.5% reference solute concentration and 1021 kg/m^3 reference density. A solar collector with an efficiency given by,

$$\eta = 0.72 - \frac{5.6\Delta T}{I_c} \qquad (10)$$

was used. A collector area equal to the evaporator area of 0.2 m^2 was assumed. For all calculations the ambient temperature was taken as 25°C. The solar radiation values were taken for a clear day in July for Gainesville, Florida (latitude 29.68°N, longitude 82.27°W) for a collector facing south, tilted at an angle equal to the latitude.

Fig. 2 shows the amount of incident solar radiation on the collector and the amount of useful heat supplied by the collector to the system. This is shown for different depths of water body inside the evaporator. The smaller the water depth, the lower the amount of useful energy gain. During the peak of incident solar radiation, the collector efficiency is about 60%.

Variation of water temperature inside the evaporator and the collector outlet temperature for different depths of water body are shown in Fig. 3. As is clear from the figure, the smaller the depth of water body, the higher the temperature during the peak solar radiation. At a water depth of 0.04 m, the maximum collector outlet

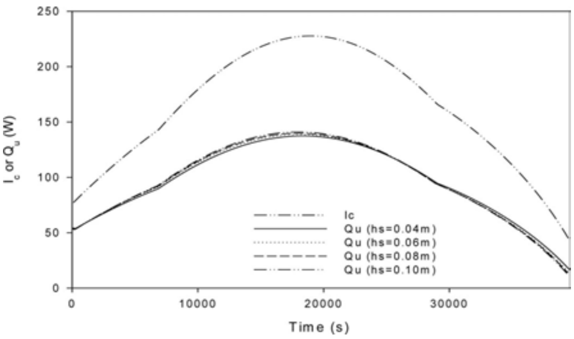

Fig. 2. Solar radiation and useful heat gain from the collector.

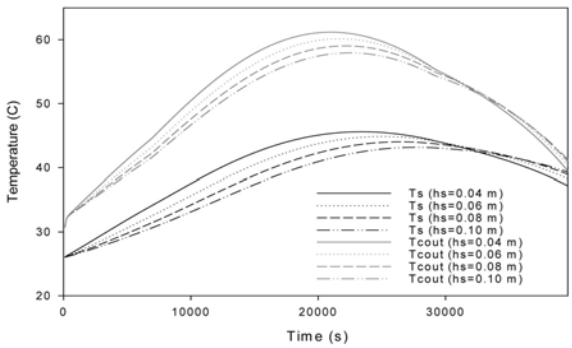

Fig. 3. Variation of saline water and collector outlet temperatures with time

temperature reaches about 61.2°C, and the water temperature is about 45.6°C. The corresponding values at a water depth of 0.1 m are 58 and 43.15 °C, respectively. The peak temperature is reached faster at a smaller depth of water.

The variation of output from the system with time for different amounts of water inside the evaporator is shown in Fig. 4. The highest amount is for the minimum amount of water. The maximum evaporation rate at a water depth of 0.04 m is about 5.1*10^{-5} kg/s; as water depth increases to 0.1 m, this rate is reduced to about 4.2*10^{-5} kg/s. The accumulated output for water depth of 0.04 m is about 1.3 kg, and this amount decreases to about 1.1 kg at a water depth of 0.1 m.

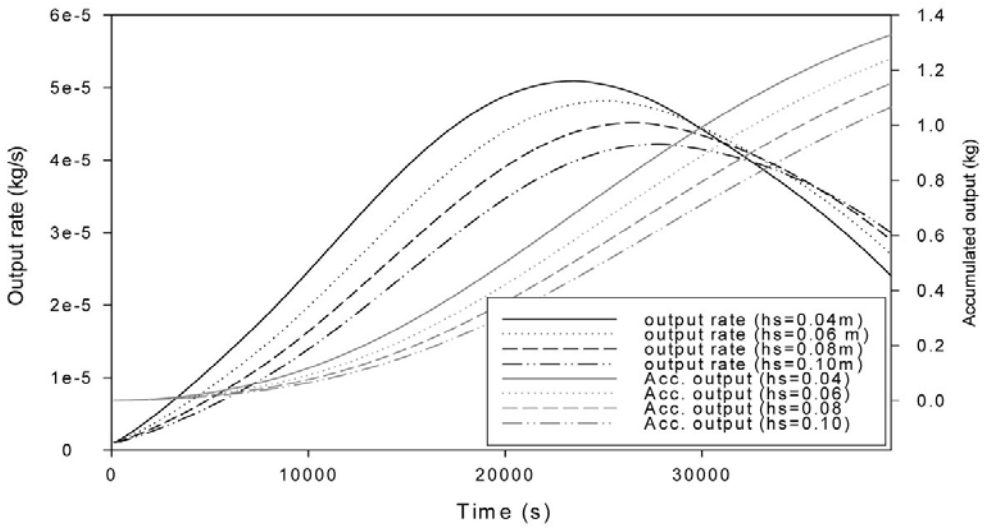

Fig. 4. System output at different depths of water with time.

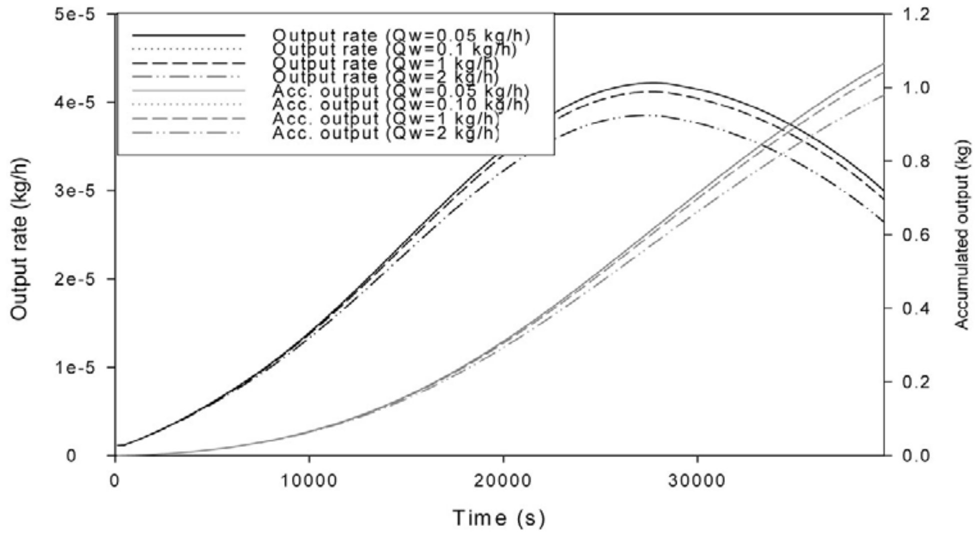

Fig. 5. System output at different withdrawal rate with time.

Fig. 5 shows the variation of the system output at different withdrawal rates with time. As the withdrawal rate increases, the system output decreases because the withdrawn water carries an amount of heat with it as it leaves the evaporator. The maximum evaporation rate at a withdrawal rate of 0.05 kg/h is about $4.2*10^{-5}$ kg/s; this amount remains almost constant as the withdrawal rate increases to 0.1 kg/h. The accumulated output at this withdrawal rate is about 1.1 kg. As the withdrawal rate increases to 2 kg/h, the evaporation rate reduces to $3.8*10^{-5}$ kg/s and the accumulated output reduces to about 0.97 kg. At a withdrawal rate of 0.1 kg/h, the loss is low and the possibility of scale for-mation is minimized.

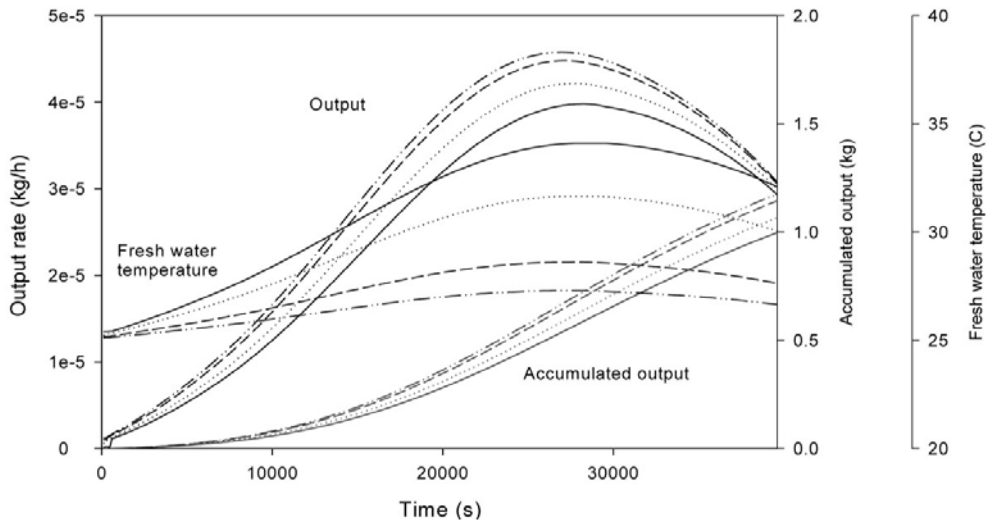

Fig. 6. Effect of fresh water temperature on the system performance.

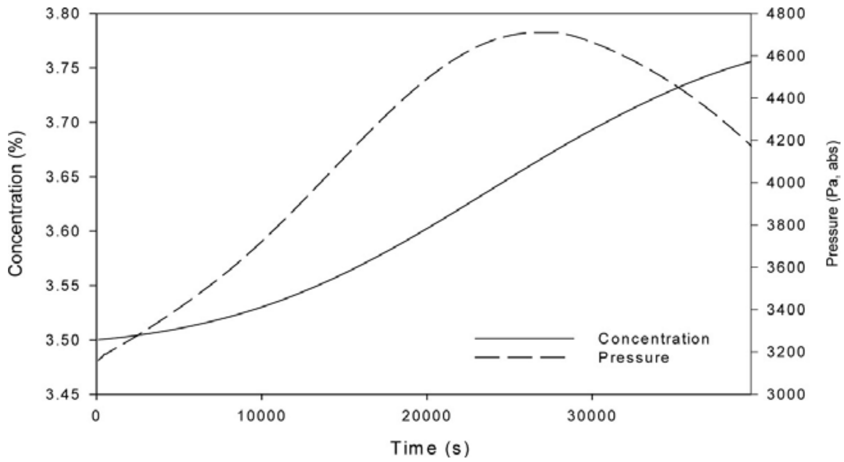

Fig. 7. Variation of concentration and pressure with time.

The effect of fresh water temperature on the system performance is shown in Fig. 6. Different values for fresh water temperature are obtained by varying the condenser heat transfer area. As the area increases, the fresh water temperature decreases, hence more output. For example, if the number of condenser fins is increased from 10 to 20, the evaporation rate and the daily accumulated output will increase from $4.2*10^{-5}$ kg/s and 1.07 kg to $4.6*10^{-5}$ kg/s and 1.17 kg, respectively. However, a compromise must be reached between the increase in the output and the increase in the condenser cost. Based on the above simulation for Gainesville, Florida, the daily output from a system of 1 m^2 evaporator area with a 1 m^2 solar collector area could reach 6.5 kg, which is almost double the amount from flat basin solar still, that is about 3-4 kg/day.m^2 [15].

Fig. 7 shows the variation of pressure and concentration with time. The pressure reaches a maximum value of about 4.7 kPa absolute (the

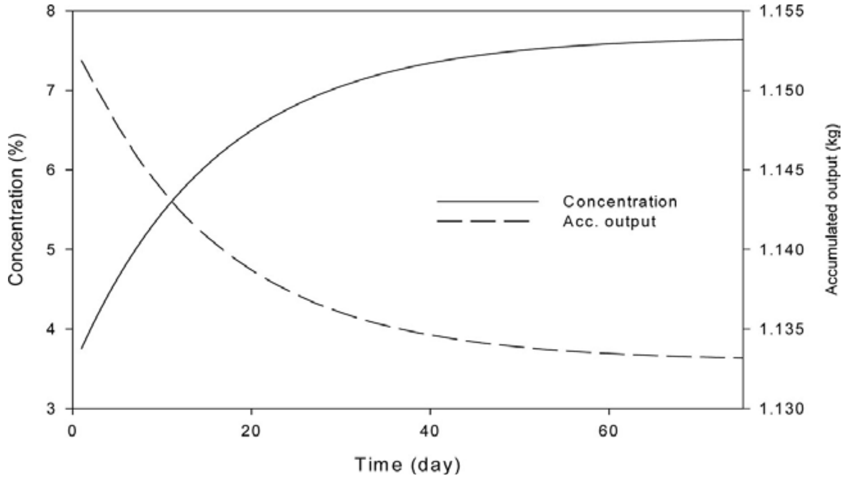

Fig. 8. Variation of concentration and accumulated output with time.

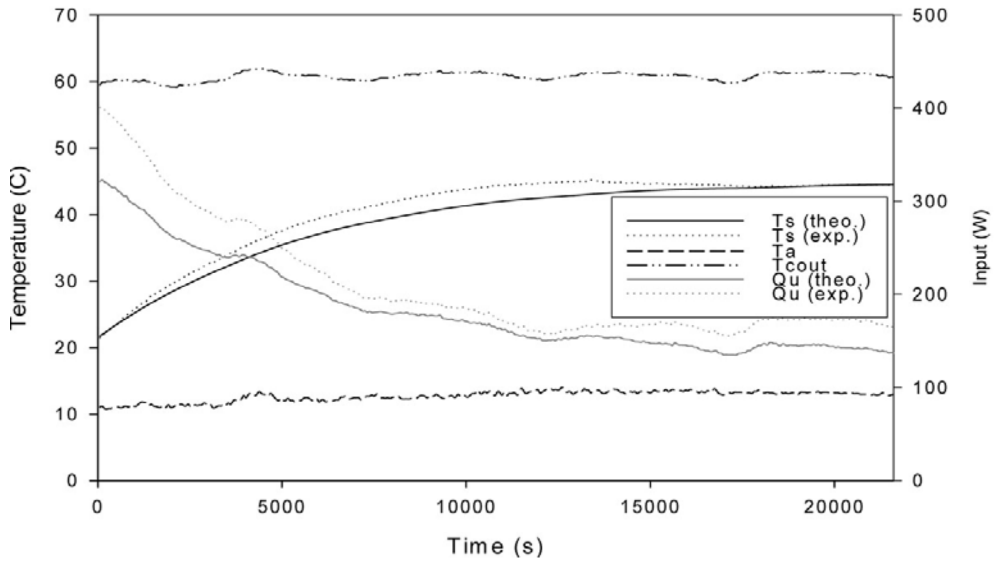

Fig. 9. Experimental and theoretical temperature and heat input with time.

effect of non-condensable gases is neglected). The concentration continues to increase till the end of the test day. Considering the value at the end of a day as the initial concentration for the next day, a steady-state value of 7.53% will be reached after 53 days, as shown in Fig. 8. The steady-state condition is assumed to be achieved if the increase in the concentration for a day is less than 0.01%. Also shown in the figure is the accumulated output, which decreases as the concentration increases. This result agrees with the result of Keren et al. [16], who found that the evaporation rate decreases by about 1% for each 1% increase in the salinity.

Fig. 9 shows the simulated and experimental variation of water temperature and heat input with time. The experimental results were ob-tained for the system operating under batch process, i.e., no withdrawal or injection, and with an initial water body depth of 0.08 m. The heat source for this test was hot water at 60°C supplied by an electric water heater. Simulation was also carried out for the same experimental conditions. The temperature of the water inside the evaporator reaches almost the same value of 44.5°C by the end of the test day for both the experimental and simulation results. The maximum difference between the two is about 2.5°C around the mid-point of the test period. The heat input starts from a high value where the system begins to heat up and a major part of the energy is used to raise the temperature of the water and the evaporator material, i.e., stored as sensible heat. This amount decreases as the system heads towards steady-state conditions. The steady-state energy input reaches about 158 W for the experimental test and 150.3 W for simulation. The experimental accumulated output for the 6-h test was 0.598 kg as compared to a simulated value of 0.675 kg. Towards the end of the test, when the system reaches steady-state conditions, the hourly output is 0.124 kg as compared to the simulated value of 0.15 kg. The difference between experimental and simulation results is in part due to the fact that some of the water vapor condenses back in the evaporator, whereas in simulation it was assumed that all vapor will reach the condenser. Another reason is that in simulation the heat loss from the system was assumed to be due only to natural convection; however, the actual heat loss in the outdoor test is due to a combination of convection (natural and forced) and radiation. Besides an uncertainty analysis showed that the errors in measurements result in an uncertainty of about 6% for the hourly distillate output.

5. Conclusions

An innovative water desalination system using low-grade solar heat was studied and tested. The system can be operated as a continuous or batch process type. The results show that the output from the proposed system can reach 6.5 kg/d.m^2 evaporator area, as compared to 3–4 kg/d.m^2 from a conventional flat-basin solar still.

6. Symbols

A — Area, m^2
C — Solute concentration, %
C_p — Specific heat, J/kg.°C
g — Gravitational acceleration, m/s^2
h — Convection heat transfer coefficient, W/m^2.K; height, m
h_{fg} — Latent heat of vaporization, J/kg
I — Incident solar radiation, W/m^2
k — Thermal conductivity, W/m.K
l — Length, m
$\dot{m}$ — Mass flow rate, kg/s
P — Pressure, Pa
Q — Heat transfer rate, W
T — Temperature, °C
V — Volume, m^3
$\dot{V}$ — Volumetric flow rate, m^3/s
v — Velocity, m/s
z — Elevation, m

Greek

ρ — Density, kg/m^3
γ — Kinematic viscosity, m^2/s; specific weight, N/m^3
η — Efficiency

Subscripts

a — Ambient
c — Cold, condenser, collector
e — Evaporation
i — Of the injection pipe, inside
L — Loss
o — Outside, outlet
s — Of seawater chamber or the evaporator; surface
w — Of the withdrawal pipe

References

[1] E. Delyannis and V. Belessiotis, Solar energy and desalination, in: Advances in Solar Energy, An Annual Review of Research and Development, D.Y. Goswami, ed., Vol. 14, American Solar Energy Society, Boulder, Colorado, 2001, pp. 287–330.
[2] S. Kumar and G.N. Tiwari, Performance evaluation of an active solar distillation system, Energy, 21 (1996) 805–808.
[3] S.A. Lawrence and G.N. Tiwari, Theoretical evaluation of solar distillation under natural circulation with heat exchanger, Energy Conv. Mgmt., 30 (1990) 205–213.
[4] Y.P. Yadav and L.K. Jha, A double-basin solar still coupled to collector and operating in the thermosiphon mode, Solar Energy, 14 (1989) 653–659.
[5] H. Kunze, A new approach to solar desalination for small-and medium- size use in remote areas, Desalination, 139 (2001) 35–41.
[6] K. Schwarzer, M.E. Vieira, C. Farber and C. Muller, Solar thermal desalination system with heat recovery, Desalination, 137 (2001) 23–29.
[7] J.L. Fernandez and N. Chargoy, Multi-stage, indirectly heated solar still, Solar Energy, 44 (1990) 215–223.
[8] S. Toyama, T. Aragaki, H.M. Salah, K. Murase and M. Sando, Simulation of a multi-effect solar still and the static characteristics, J. Chem. Engineering Jpn., 20 (1987) 473–478.
[9] G.A. Bemporad, Basic hydrodynamic aspects of a solar energy based desalination process, Desalination, 54 (1995) 125–134.
[10] B.A. Jubran, M.I. Ahmed, A.F. Ismail and Y.A. Abakar, Numerical modeling of a multi-stage solar still, Energy Conv. Mgmt., 41 (2002) 1107–1121.
[11] S.K. Sharma and D.Y. Goswami, Low temperature energy conversion system, unpublished document, University of Florida, 1994.
[12] F.P. Incropera and D.P. DeWitt, Fundamentals of Heat and Mass Transfer, 4th ed., Wiley, New York, 1996, p. 559.
[13] W.M. Rohsenow, J.P. Hartnett and Y.I. Cho, Handbook of Heat Transfer, 3rd ed., McGraw-Hill, New York, 1998, pp. 6.31–6.41.
[14] E. Krell, Handbook of Laboratory Distillation, Elsevier Scientific, Amsterdam, 1982, p.159.
[15] S. Kalogirou, Survey of solar desalination systems and system selection, Energy, 22(1) (1997) 69–81.
[16] Y. Keren, H. Rubin, J. Atkinson, M. Priven and A. Bemporad, Theoretical and experimental comparison of conventional and advanced solar pond performance, Solar Energy, 51(4) (1993) 255–270.