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Numerical and Experimental Study of a Passive Solar Still Integrated With an External Condenser

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***Abstract*— In this paper, a numerical simulation of a passive basin type solar still enhanced by an external condenser is proposed. A computer code has been developed to predict the transient thermal behaviour of the still. The mathematical model has been derived based on the basic energy balance equations for the different components of the still. Furthermore, the main heat and mass transfer phenomena taking place in the system are modeled. The model validation consists in comparing the main thermal-hydraulic parameters characterizing the still against the experimental data obtained during a typical summer day. The comparison shows that the calculation results are in good agreement with the experimental data of the solar still.**

***Keywords*— Solar desalination, Passive solar still, External condenser, Numerical and experimental study.**

I. INTRODUCTION

Conventional Solar Still (CSS) is the oldest, economical and simple technology used for water purification especially when used in remote and arid areas where sunshine is abundant and fresh water is scarce [1,2]. The CSS is selected due to its simplicity and passive nature, no need for hard maintenance or skilled persons, which leads to little operation and maintenance costs. However, the CSS suffer from some drawbacks, which sometimes limit the use of this system for large-scale production [3]. Some of these drawbacks are, large solar collection area requirement, system vulnerability to weather-related damage, less market demand of technology and low interest of the manufacturers [4,5]. The main limitation is the low productivity compared with modern desalination processes, where the daily yield from a single slope basin type solar still may vary from 0.5 to 2.5 kg/m² and its efficiency is usually about 5 to 40% [4,6].

Increasing solar still productivity has been the subject of intensive research efforts and remains a challenge to the scientists. However, this improvements can be achieved by a proper modifications in the still design and its operating parameters [7,8]. In fact, this improvements are resulted from enhancing evaporation, condensation, heat storage and reducing thermal losses [9,10]. Internal air-convection is another way to increase the still productivity. This effect has not received enough attention where a few attempts have been addressed. This effect was studied by Ali et al. [11,12] by placing an electrical fan inside the CSS where he found that the still productivity is increased by about 30%. However, operation under natural circulation mode has been proven to be more advantageous in terms of simplicity, reliability and cost effectiveness [13].

In a previous work [14] we are attempted experimentally to enhance the single slope solar still productivity by generating air-convection inside the still using the thermo-siphon effect. The proposed solar still is designed to operate as a Natural Circulation Loop (NCL) with the humid-air as working fluid. In which the still serves as a heater, a separate tubular condenser serves as a cooler and a vertical PVC tubes acting as hot and cold legs. The experimental tests of the proposed still show that a significant improvements in terms of water productivity and efficiency are achieved in comparison to the CSS and the NCL is found to have a good effect on the still performances using air-convection.

In the scope of this work, a transient computer program is developed in order to simulate the transient behaviour of the still during a typical summer day. This computer code will be a powerful tool used to investigate the thermal-hydraulic performances and for further improvements of the solar still. The modeling validation has been made by comparing the calculated results with experimental test for a typical day of the 10/07/2015 at the Faculty of Science and Applied Sciences, Oum-El-Bouaghi University, Algeria (Latitude: 35°79'N, Longitude 7°40'E). The comparison shows that there is reasonable agreement between the simulation results and the experimental data.

II. OVERVIEW OF THE SOLAR STILL

A schematic description of the solar still is shown in Figure 1. The system is a combination of a modified CSS with a passive natural circulation loop where the still serves as a heat source (heater), an integrated passive separate condenser serves as a heat sink (cooler). The link between the heater and the cooler is performed by a vertical insulated PVC tubes acting as hot and cold legs.

The condenser is a horizontal heat exchanger made of three parallel aluminium tubes in which the water steam is separated from the air and forms a condensate film. It is slightly inclined to facilitate the drainage of the condensate water. The condenser is shaded from sun rays by a plastic cover. The air-convection in this case is created by the buoyancy forces that evolve from the density gradients induced by the simultaneous effect of temperature and

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humidity between evaporator and condenser. More details about the still design and the experimental tests can be found in [14].

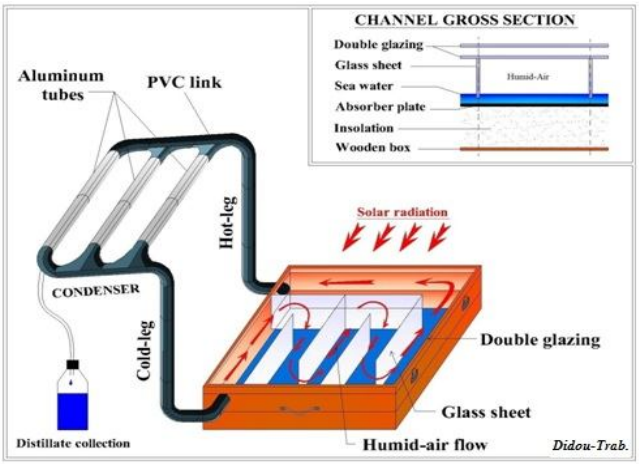

Fig. 1. Schematic illustration of the solar still [14].

III. Mathematical model

The energy balance equations describing the different components of the solar still are obtained by the application of the first law of thermodynamics as:

a) Basin (absorber)

$$M_b C_{Pb} \frac{dT_b}{dt} = G_b A_b - Q_{b-w} - Q_{b-a} \quad (1)$$

Where,

$$Q_{b-a} = U_b A_b (T_b - T_a)$$

$$Q_{b-w} = h_{cb-w} A_b (T_b - T_w)$$

$$G_b = I.(1-\alpha_{gi})(1-R_{gi})(1-\alpha_{go})(1-R_{gop})(1-R_w)(1-\alpha_w)\alpha_b$$

The convective heat transfer coefficient from the basin to seawater, $h_{cb\text{-}w}$, is calculated according to the following correlations [15]:

$$h_{cb-w} = Nu \lambda_w / L_C$$

$$for \;\; Gr < 10^5, Nu = 1$$

$$for \;\; 10^5 < Gr < 2.10^7, Nu = 0.54(GrPr)^{0.25}$$

$$for \;\; Gr > 2.10^7, Nu = 0.14(GrPr)^{0.33}$$

P_r is the Prandtl number and Gr is the Grashof number for the simultaneous heat and mass transfer.

b) Saline water

$$M_{bw} C_{Pw} \frac{dT_w}{dt} = G_w A_w + Q_{b-w} - Q_{w-f} - Q_{w-gi} \quad (2)$$

Where,

$$G_w = I.(1-\alpha_{gi})(1-R_{gi})(1-\alpha_{go})(1-R_{gop})(1-R_w)\alpha_w$$

Heat transfer from water to the fluid is carried out by evaporation, convection and radiation (Eq. 3) and from the water to the inner glass cover, the rate of heat transfer is given by Eq. (4):

$$Q_{w-f} = Q_{ew-f} + Q_{cw-f} + Q_{rw-f} \quad (3)$$

Where,

$$Q_{ew-f} = h_{ew-f} A_w (T_w - T_f)$$

$$Q_{cw-f} = h_{cw-f} A_w (T_w - T_f)$$

$$Q_{rw-f} = \sigma \varepsilon_{eff} A_w [T_w{}^4 - T_f{}^4]$$

$$Q_{w-gi} = Q_{ew-gi} + Q_{cw-gi} + \mathrm{Q}_{\mathrm{rw-gi}} \quad (4)$$

Where,

$$Q_{ew-gi} = h_{ew-f} A_w (T_w - T_{gi})$$

$$Q_{cw-gi} = h_{cw-f} A_w (T_w - T_{gi})$$

$$Q_{rw-gi} = \sigma \varepsilon_{eff} A_w [T_w{}^4 - T_{gi}{}^4]$$

Evaporation heat transfer is related to the convective heat transfer coefficient [16] by Eq. (5). Where, P_w and P_f are the partial pressure of internal glass and the flowing humid-air.

$$h_{ew} = 0.016273 h_{cw-f} (P_w - P_f)/(T_w - T_f) \quad (5)$$

According to Dunkle [17] the convective heat transfer coefficient, $h_{cw\text{-}f,}$ is given by:

$$h_{cw-f} = 0.884 \left[(T_w - T_f) + \frac{(P_w - P_f)\, T_w}{268.9\mathrm{x}10^3 - p_w} \right]^{1/3} \quad (6)$$

c) Fluid flow (humid-air)

Fluid temperature is evaluated by the arithmetic mean value between hot-leg and cold-leg temperatures, assuming that both legs are adiabatic. The fluid circulating over the loop is considered as a single phase and the physical properties of humid-air as function of temperature as presented in [16]. Temperature distribution in heating and cooling sections is linear. The friction factor in the region of laminar flow can be neglected and the effects of the bends can be simulated by a proper friction coefficients. Axial conduction effects are neglected and the Bossinesq approximation is assumed. Applying these assumptions to the system balance equations and integrating the momentum and energy equations through the entire loop (*N=1: heater, N=2: hot-leg, N=3: cooler and N=4: cold-leg*), yielded the following equations:

- *Loop Momentum Balance Equation:*

$$\sum_{i=1}^{N} \left(\frac{L_i}{a_i} \right) \frac{dW}{dt} = \frac{\beta \rho g Q_H H}{W \bar{c}_{pf}} \frac{W^2}{\rho_i a_c{}^2} \sum_{i=1}^{N} \left[\frac{1}{2} \left(\frac{fL}{d_h} + K \right)_i \left(\frac{a_c}{a_i} \right)^2 \right] \quad (7)$$

- *Loop Energy Balance Equation:*

$$M_f c_f \frac{dT_f}{dt} = Q_{w-f} + Q_{gi-f} - Q_{f-s} \quad (8)$$

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The heat transferred rate from the working fluid to the condenser wall is carried out by convection, radiation and condensation mechanisms. Assuming that the loop legs are adiabatic, the condensation heat transferred rate, Q_{cond}, can be equal to Q_{evap} expressed in Eq. (3).

$$Q_{f-s} = Q_{cond.} + Q_{c.f-s} + Q_{r.f-s} \quad (9)$$

$$Q_{cond.} = Q_{ew-f}$$

$$Q_{c.f-s} = h_{cf-s} A_{si} (T_f - T_s)$$

$$Q_{r.f-s} = \sigma \varepsilon_{eff} A_{si} [T_f{}^4 - T_s{}^4]$$

The governing momentum equation describing the fluid flow behaviour inside a closed loop is given by Eq. (7). The present loop shows quasi steady-state condition with laminar flow for the entire range of solar power input which varies slowly versus time. Therefore, the steady-state solution for the momentum equation along the loop can be written as the Vijayan correlation [17]:

$$W = \sqrt{2\rho_o^2 \beta g (T_H - T_C) H / R} \quad (10)$$

Where, W is the fluid flowrate, H is the centre line elevation difference between the cooler and the heater and R is the total hydraulic resistance of the loop given by:

$$R = \sum_{i=1}^{N} \left(\frac{fL}{D} + k\right)_i \frac{1}{A_i^2} \quad (11)$$

d) Internal glass cover

Due to the difference in temperature between the water and the glass cover, water condensation can take place at the inner glass. This difference is enhanced by the heat transfer to the flowing humid-air (relatively at low temperature) and heat losses through the outer glass cover by radiation and conduction, respectively.

$$M_{gi} C_{Pgi} \frac{dT_{gi}}{dt} = G_{gi} A_{gi} + Q_{w-gi} - Q_{cgi-f} - Q_{gi-go} \quad (12)$$

Where,

$$G_{gi} = I.(1 - \alpha_{gi})(1 - R_{gi})(1 - R_{gi})\alpha_{gi}$$

$$Q_{cgi-f} = h_{cgi-f} A_g (T_{gi} - T_f)$$

In order to estimate the convection and evaporation heat transfer coefficients, from the water to the internal glass cover, the same correlations (5 and 6) are used replacing P_f by P_{gi} and T_f by T_{gi}. The heat transferred rate from the inner to the outer glass cover is carried out by radiation and conduction through the air gap between the glasses as expressed by the following equation:

$$Q_{gi-go} = Q_{rgi-go} + A_g (T_{gi} - T_{go}) / \left(\frac{L}{k}\right)_{gap} \quad (13)$$

Where, $Q_{rgi-go} = \sigma \varepsilon_g A_g (T_{gi}^4 - T_{go}^4)$

e) External glass cover

$$M_{go} C_{Pgo} \frac{dT_{go}}{dt} = I.\alpha_{go} A_g + Q_{gi-go} - Q_{cgo-a} - Q_{rgo-a} \quad (14)$$

Where,

$$Q_{rgo-sky} = \sigma \varepsilon_g A_g (T_{go}^4 - T_{sky}^4)$$

$$Q_{cgo-a} = h_{cgo-a} A_g (T_{go} - T_a)$$

From the internal glass to the external one, the heat transfer is carried out by both radiation and conduction due to the small gap separate the two glasses. The convective heat transfer coefficient at the external glass cover, under the wind speed effects, is expressed by: $h_{cgo-a} = 2.8 + 3V_V$ [18].

f) Condenser (Sink)

$$M_s C_{Ps} \frac{dT_s}{dt} = Q_{f-s} - Q_{s-a} \quad (15)$$

Where,

$$Q_{s-a} = h_{cs-a} A_{so} (T_s - T_a) - \sigma \varepsilon_s A_{so} (T_s^4 - T_{sky}^4)$$

The convective heat transfer coefficient of the working fluid flow in the condenser, h_{cf_s}, is calculated according to Sider-tate correlation [19].

$$h_{cf-s} = 1.86 \frac{\lambda}{D} \left(\frac{RePr}{L/D}\right) \left(\frac{\mu}{\mu_c}\right)^{0.14} \quad (16)$$

From the condenser to the ambient, the heat transfer is carried out by radiation, natural and forced convection due to wind effect. So, the global convection heat transfer coefficient can be considered as:

$$h_{cs\_a} = h_{NC} + h_{FC} \quad (17)$$

$$Nu_{NC} = 0.3 + \sqrt{Nu_{lam} + Nu_{tur}} \quad (18)$$

Where,

$$Nu_{lam} = 0.664 Re^{1/2} Pr^{1/3}$$

$$Nu_{tur} = \frac{0.037 Re^{0.8} Pr}{1 + 2.443 Re^{-1} (Pr^{2/3} - 1)}$$

Depending on the wind velocity, the average heat transfer coefficients for turbulent flows, over horizontal cylinder have been proposed by Churchill and Bernstein [19].

$$Nu_{FC} = 0.3 + \frac{0.62 Re^{1/2} Pr^{1/3}}{\left[1 + (0.4/Pr^{2/3})\right]^{1/4}} \quad (19)$$

The sky temperature Tsky is calculated The sky temperature is calculated according to Ref. [20]:

$$T_{sky} = Ta - 6 \quad (20)$$

IV. Numerical Computation

The transient behaviour of the solar still is governed by a set of differential equations. for this reason, a computer program have been established and developed in Matlab 6.1 using ODE23 function for solving the heat balance equations and predicting the transient thermal behaviour of the still during a typical day. The input parameters for the computer code include the atmospheric conditions describing the test day

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(solar radiation, ambient temperature and wind velocity) given in Figure 2-3. In order to simulate the time variation of these parameters, they are properly fitted as polynomial functions.

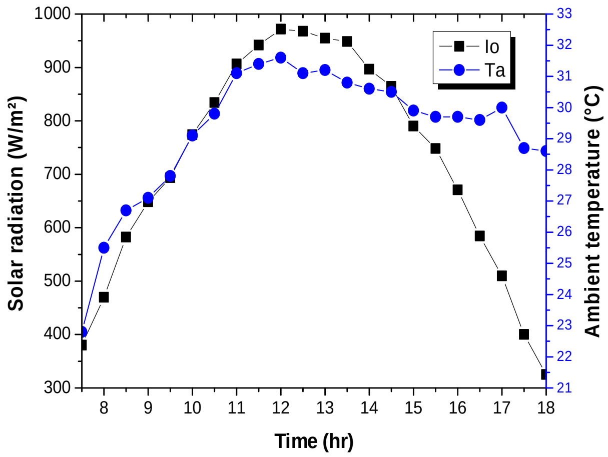

Fig. 2. Variations of solar radiation & ambient temperature.

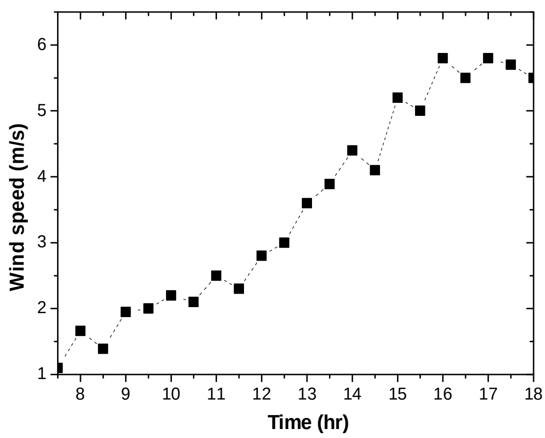

Fig. 3. Variations of the wind velocity during the test day.

V. Program Validation

The obtained simulation results are compared with the experimental data carried out in the 10/07/2015 at the Faculty of Science and Applied Sciences, Oum-El-Bouaghi University. Comparative analysis with the main temperatures describing the still behavior namely the absorber, seawater, glass covers, humid-air and the condenser wall is reported in Figures 4 to 8. In addition, the hourly productivity is also concerned by the comparison (Fig.9).

As could be seen, good agreement with the experimental data is obtained in the dynamic validation of the model. The maximum error in estimating the main thermal-hydraulic parameters of the still was found to be acceptable and the deviations are mainly due to simplifications introduced in the model.

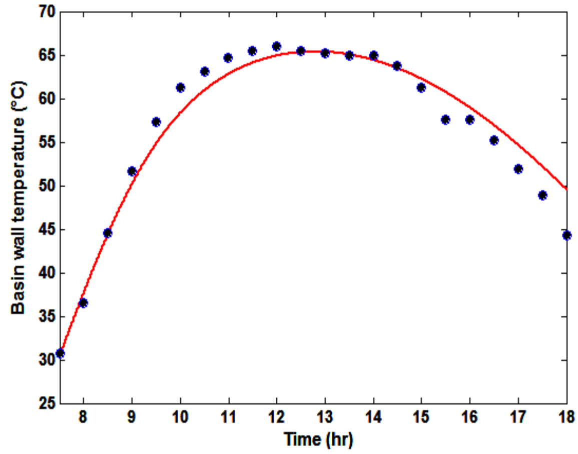

Fig. 4. Basin wall temperature (Experimental (●) & calculated (-)).

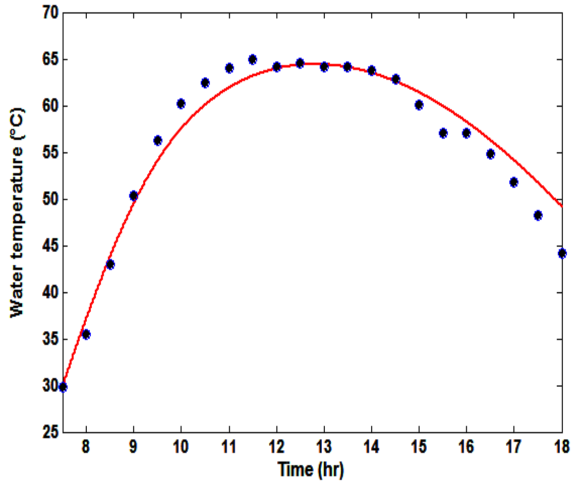

Fig. 5. Water temperature (Experimental (●) & calculated (-)).

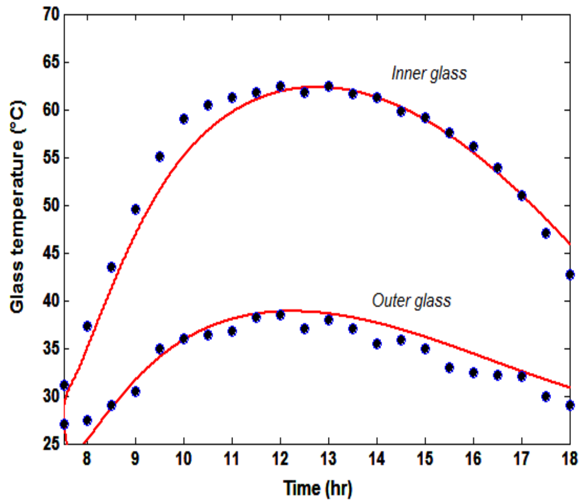

Fig. 6. Inner /outer glass temperature (Experimental (●) & calculated (-)).

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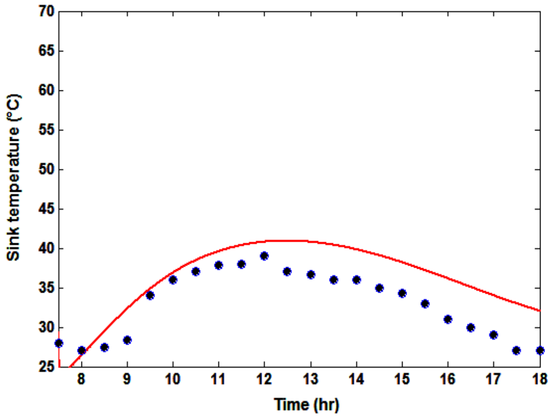

Fig. 7. Sink temperature (Experimental (●) & calculated (-)).

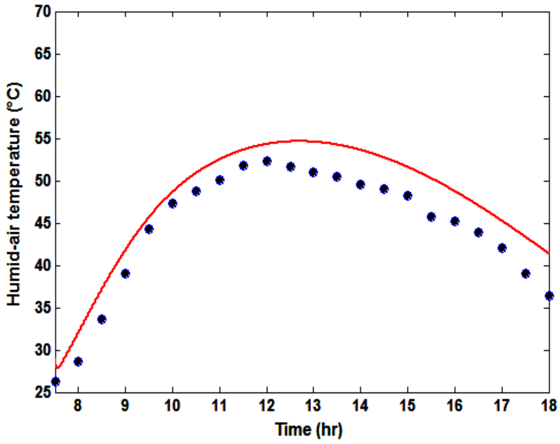

Fig. 8. Fluid (humid-air) temperature (Experimental (●) & calculated (-)) .

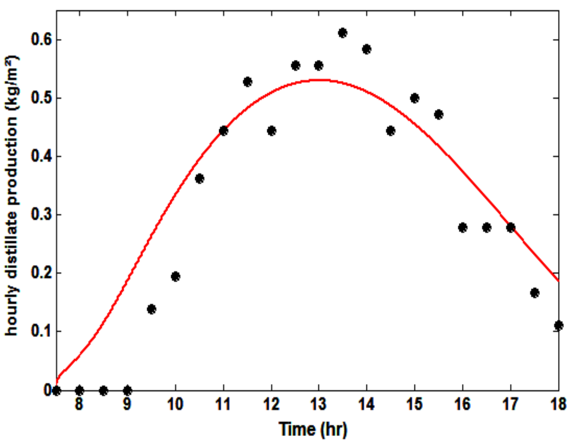

Fig. 9. Hourly distillated yield (Experimental (●) & calculated (-)) .

VI. CONCLUSION

This study related to the numerical simulation of a passive basin type solar still improved by an integrated external condenser. A transient computer code was developed and programmed using Matlab 6.1 software. The main heat and mass transfer aspects describing the thermal-hydraulic behaviour of the whole system are modelled. The code validation strategy is carried out through a comparative study between the obtained numerical results and outdoor experimental data. The comparison shows that a good agreement was found. The maximum error in estimating the gas temperature was found to be acceptable and the deviations are mainly due to simplifications introduced in the model.

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