

# Performance of a Solar Humidification Dehumidification Desalination System on December 27th and 28th in North Cyprus

Uğur Atikol, Reza Enayatollahi, Seyed Saeed Madani

Mechanical Engineering department, Eastern Mediterranean University, NORTH CYPRUS

## ABSTRACT

The present study is concerned with desalinating the saline water, using the humidification dehumidification method. An open air open water cycle is chosen, whilst forced air circulation is applied. The system utilizes water and air heating technique. The energy required for heating the air is provided by solar radiation, while the required energy for heating the water is provided by electricity. The goals of this study are: a) To study different existing systems in the literature; b) To review the governing equations for each segment of the desalination unit; and c) To conduct and experiment on the above described humidification dehumidification desalination unit under North Cyprus winter conditions. The solar air heater placed to azimuth angle of  $-20^\circ$ , with a tilt angle of  $45^\circ$ . According to the experiment, the optimum flow rate for water stream is 4 Lt/min, while the air flow rate is  $1.5\text{m}^3/\text{min}$ . Besides, the inlet water temperature to the evaporator is kept at  $55^\circ\text{C}$ . The maximum productivity of  $1.55\text{ Lt/day.m}^2$  is achieved for this condition. The present experiment implies that the maximum productivity is achieved with higher temperature of both air and water streams into the evaporator, higher flow rate for water and lower flow rate for air.

**Keywords:** Humidification, Dehumidification, Desalination

## INTRODUCTION

Three quarter of the earth's surface is covered with water, but only 3% of this water is potable water Kalogirou, 1997, which human beings and animals are able drink. Shortage of potable water strongly motivated scientists to find a solution for converting brackish water to desalinated water. The humidification-dehumidification (HDH) desalination technique is a low temperature procedure to provide small amount of potable water. It consumes lower amount of energy than the other methods of desalination. Furthermore, in this process, coolant is the saline water and heating fluid is the moist air. It is common practice to use ammonia or other refrigerants as coolant. Using such working fluids needs additional components such as compressors and expansion devices which may consume energy. Besides, in case of any leakage they may contaminate the produced water. since in HDH desalination systems the cooling fluid is saline water that requires no extra

energy consuming devices, the process is simpler and there is no danger of contamination of the produced fresh water. Typical desalination units are producing potable water, but most of them are burning fossil fuels, directly or indirectly to provide the required energy for heating sea water. Burning fossil fuels has several undesirable impacts on the environment such as; polluting the air with toxic gases and increasing the quantity of green house gases in the atmosphere. Moreover, the fossil fuels themselves are quite expensive; furthermore, the amount of fossil fuels on earth is finite. Therefore, using the renewable and clean sources of energy is strongly suggested to prevent such drawbacks. The present study is performed in North Cyprus, which has approximately 300 sunny days per year, and suffers from the shortage of potable water. Hence, producing potable water from saline water by using the solar energy as the source of heat is the objective of the study. The present work is concerned with the setting up and conduction of an experiment on a HDH desalination system designed by Kraft et al. (2010) to find its suitability for the North Cyprus winter conditions. A review study, on the researches and experiments carried out before, is essential to find out the most up to date and effective desalination units. By a review study, one may be able to find a proper system for the location of study, according to availability of the requirements in there. A review of the governing equations may help understanding the thermodynamics of the system considered. This may also help the student who seek to carry out theoretical analysis. The inlet and outlet temperature difference of the moist air in the condenser, and the condenser inlet humidity of the air are the most important parameters affecting the potable water production. These parameters change with changing flow rates of both water and air streams. Besides, the temperature of both water and air streams are influential on the production. Therefore, optimizing the flow rates for both water and air flows, in addition to evaluate the impact of changing the water temperature, are aimed for the current experiment.

## GOVERNING EQUATIONS

### Energy Balance

In order to find the outlet temperature of the air from the solar air collector energy balance must be applied on the collector. Applying an energy balance on each part of collector, and combining the results will give us an equation, which is temperature distribution equation for air in the duct.

$$T = \left( \frac{H_a}{U_c} + T_a \right) - \frac{1}{U_c} [H_a - U_c (T_{A,in} - T_a)] \exp \left[ - \frac{U_c F'}{\dot{m} c_p} x \right] \quad (1)$$

Where

$H_a$  = Absorbed radiation, [W/m<sup>2</sup>]

$U_c$  = Collector overall heat loss coefficient, [W/m<sup>2</sup>.K]

$T_{A,in}$  = Inlet air temperature, [K]

$\dot{m}$  = Mass flow rate of air through the collector, [kg/sec]

$C_p$  = Specific heat of air, [J/kg.K]

The outlet temperature of air can be calculated by

$$T_{A,out} = T_{A,in} + \frac{1}{U_c} [H_a - U_c (T_{A,in} - T_a)] \left[ 1 - \exp \left( - \frac{A_c U_c F'}{\dot{m} c_p} \right) \right] \quad (2)$$

This is a general equation for all the collectors with the same configuration. In order to specify this equation for a particular collector, the overall heat loss coefficient of that collector must be substituted in the equation 2. The useful energy gain by the air stream is then

$$Q_u = \dot{m}c_p(T_{A,out} - T_{A,in}) \tag{3}$$

Where

$T_{A,out}$  = Outlet air temperature, [C]

$Q_u$  = Total rate of useful energy gain, [W]

It is highly desirable to express the total useful energy gain of the collector in terms of the inlet air temperature, which is known and equal to the ambient temperature. Therefore, it is necessary to define a factor, so called; collector heat removal factor ( $F_R$ ). Collector heat removal factor is expressed in equation 4-

$$F_R = \frac{\dot{m}c_p(T_{A,out} - T_{A,in})}{A_c[H_a - U_c(T_{A,in} - T_a)]} \tag{4}$$

Where

$A_c$  = Collector area

Substituting equation 3 in equation 4 results:

$$Q_u = F_R A_c [H_a - U_c(T_{A,in} - T_a)] \tag{5}$$

Where:

$A_c$  = Collector area, [m<sup>2</sup>]

$H_a$  = Absorbed radiation, [W/m<sup>2</sup>]

$U_c$  = Collector overall heat loss coefficient, [W/m<sup>2</sup>.K]

$T_{A,in}$  = Inlet air temperature, [K]

$T_a$  = Ambient temperature, [K]

$\dot{m}$  = Mass flow rate of air through the collector, [kg/sec]

**Absorbed Solar Radiation**

When solar radiation reaches to the collector surface, a major portion of it will transmit through transparent cover and falls on the absorber plate. The absorbed amount of solar radiation depends on the absorptivity of the absorber plate and the transmissivity of the glazing system. Incident radiation consists of three components; beam radiation, diffuse radiation and ground reflected radiation, and they are incident on the collector surface with different angles. Therefore, they must be taking into account separately. The total incident radiation on a tilted collector surface is:

$$H_t = H_B R_B + H_d \left( \frac{1 + \cos s}{2} \right) + H \rho_g \left( \frac{1 - \cos s}{2} \right) \tag{6}$$

$H_B$  = Beam radiation on a horizontal surface, [W/m<sup>2</sup>]

$R_B$  = The beam radiation tilted factor

$H_d$  = Diffuse sky radiation on a horizontal surface, [W/m<sup>2</sup>]

$H$  = Total radiation on a horizontal surface

$S$  = Tilted angle

$\rho_g$  = Ground diffuse reflectivity, [W/m<sup>2</sup>]

Therefore, treating for the three radiation components separately, the absorbed radiation by the collector can be expressed as:

$$H_t = H_B R_B (\tau\alpha)_B + H_d (\tau\alpha)_d \left( \frac{1+\cos s}{2} \right) + H \rho_g (\tau\alpha)_g \left( \frac{1-\cos s}{2} \right) \tag{7}$$

Where  $(\tau\alpha)_B$  and  $(\tau\alpha)_d$  and  $(\tau\alpha)_g$  are the transmissivity-absorptivity products for beam, diffuse sky and diffuse ground reflection radiation, respectively. Ground diffuse reflectivity depends on the ground surface covered by snow, and it is varying from 0.2 for no snow on the ground, to 0.7 for fully snowed ground (Mitsos et al, 2011).

**Condenser**

Condenser itself is a double pipe heat exchanger, where the cooling fluid is saline water, and heating fluid is the moist air from the evaporator. Water and air inlet temperatures, besides the outlet temperature of air are known. Outlet water temperature must be compute, which is equal to the inlet water temperature to the evaporator. In order to calculate the outlet, water temperature the equation 8, (Holman, 1986), is suggested.

$$q = \dot{m}_c C_c \Delta T_c = \dot{m}_h C_h \Delta T_h \tag{8}$$

Where

$\dot{m}_c, \dot{m}_h$  = Mass flow rate of cooling and heating fluid, respectively.

$C_c, C_h$  = Specific heat of coolant and heating fluid, [J/K], respectively.

$\Delta T_c, \Delta T_h$  = Temperature difference of coolant and heating fluid, [C], respectively.

$q$  = The rate heat transfer, [W]

Temperature difference of water in the condenser can be calculated from the above equation. Consequently, the outlet temperature of dehumidifier is computable. The goal of humidification dehumidification desalination unit is to produce potable water, which it depends strongly on the condenser efficiency. Therefore the condenser is the most vital segment of the whole unit. The thermodynamic of the system is the same as the evaporator. But the cooling and heating fluid are not in contact with each other, thus there would be no mass transfer between them. Before doing any calculation the flow regime inside of the pipes must be defined, it means calculation of Reynolds number is necessary:

$$Re = \frac{\rho u D}{\mu} \tag{9}$$

Where

$\rho$  = Density of the fluid, [kg/m<sup>3</sup>]

$u$  = Velocity of the fluid, [m/s]

$D$  = Diameter of the pipe, [m]

$\mu$  = Dynamic viscosity of the fluid, [kg/m.sec]

Heat transfer through the humidifier is mostly conducting with convection heat transfer, but in order to define the heat transfer through the condenser, the overall heat transfer coefficient is required, and that can be calculated with equation 10.

$$\frac{1}{UA} = \sum \frac{1}{hA} + \sum R \tag{10}$$

Where in this equation parameter R stands on thermal resistance and it is computing by equation 11.

$U$  = Overall heat transfer coefficient, [W/m<sup>2</sup>.K]

$A$  = Heat transfer area, [m<sup>2</sup>]

$h$  = Convection heat transfer coefficient, [W/m<sup>2</sup>.K]

$$R = \frac{x}{k.A} \tag{11}$$

X= The wall thickness, [m]

K= The thermal conductivity of the substance, [W/m.K]

A= The overall area of the heat exchanger, [m<sup>2</sup>]

Accordingly the rate of heat transfer can be compute:

$$Q = U A \Delta T_{Lm} \tag{12}$$

$\Delta T_{Lm}$  represents the logarithmic mean temperature difference, which is expressed as:

$$\Delta T_{Lm} = \frac{\Delta T_A \Delta T_B}{\ln \left( \frac{\Delta T_A}{\Delta T_B} \right)} \tag{13}$$

Where  $\Delta T_A$ , is the temperature difference among the two streams at end A, and  $\Delta T_B$  is the temperature difference between the two streams at end B. There are also some fins assembled on the tubes, in order increase the rate of heat transfer, by increasing the overall heat transfer area. The effect of these fins must be considered in calculations. There are three different types of fins (Holman, 1986): first, the fin is too long and the temperature at the end of fin is equal to the ambient temperature. Second, the fin is finite and there is heat transfer at the end of it. Third, the end of fin is insulated, so it is not equal to ambient but there is no heat transfer at the end point. The fin that we are using in this experiment is the third type where insulated at the end point of fin. The rate of heat transfer is:

$$q = -kA\theta m \left( \frac{1}{1+e^{-2mL}} - \frac{1}{1+e^{2mL}} \right) = \sqrt{hP/kA}\theta_0 \tan h mL \tag{14}$$

Finally the heat transfer ratio for the case of with fin over without fin becomes:

$$\frac{q \text{ with fin}}{q \text{ without fin}} = \frac{\eta_f A_f h \theta_0}{h A_b \theta_0} \tag{15}$$

$A_f$ , is the fin area  $A_b$  is the base area and the fin efficiency  $\eta_f$  expressed as:

$$\eta_f = \frac{\tan h mL}{mL} \tag{16}$$

Thus:

$$\frac{q \text{ with fin}}{q \text{ without fin}} = \frac{\tan h mL}{\sqrt{hP/kA}} \tag{17}$$

Measuring the produced amount of potable water is conducted with the dehumidification calculation on the shell side of the condenser. Here we assume that the inlet air temperature is equal to the outlet temperature of water, and the inlet water temperature is equal to the outlet air temperature. Steady state condition is assumed. Therefore, the air flow rate is constant. Kinetic and potential energy changes are negligible. Besides, the air and water are accounted as the ideal gases. The outlet air is also assumed to be saturated, with 100 percent relative humidity. The enthalpy of saturated liquid water at inlet and outlet state of condenser should be read from water properties table. Also the air properties are available in both states, in psychrometric chart. Therefore, applying mass and energy balance yields:

$$\dot{m}_a = \text{constant} = \frac{\dot{V}}{v} \left[ \frac{m^3/\text{min}}{m^3/\text{kg dry air}} \right] \tag{18}$$

Where

$$\begin{aligned} \dot{V}_i &= \text{Volumetric flow rate, [m}^3/\text{min]} \\ v &= \text{Total mass of moisture in 1 Kg of dry air, [m}^3/\text{kg]} \\ \dot{m}_a \omega_1 &= \dot{m}_a \omega_2 + \dot{m}_w \rightarrow \dot{m}_w = \dot{m}_a (\omega_1 - \omega_2) \end{aligned} \quad (19)$$

Where

$\dot{m}_a$  = Mass flow rate of air, [kg/sec]

$\dot{m}_w$  = Mass flow rate of water, [kg/sec]

$\omega$  = Specific humidity

$$\sum_{in} \dot{m} h = \dot{Q}_{out} + \sum_{out} \dot{m} h \rightarrow \dot{Q}_{out} = \dot{m}(h_1 - h_2) - \dot{m}_w h_w \quad (20)$$

$$\omega = \frac{0.622 P_v}{P - P_v} \quad (21)$$

$$P_v = \phi P_g = \phi P_{sat@T} \quad (22)$$

## RESULTS AND DISCUSSION

Most of the energy gained by the solar air collector is consumed to heat up the water in the evaporator before saturation. Dramatic decrease in temperature of the moist air causes decrease in the potable water production. The thermal performance of the system without using auxiliary water heater in a typical day of December with the variation of irradiance illustrated in figure 1. it can be seen that the temperature difference between the cooling water into the condenser and the moist air into the condenser is too low leading to failure in water production. To solve this problem we need to increase the moist air temperature. In order to increase the moist air temperature the temperature of inlet water into the evaporator must be increased. Therefore, an auxiliary heater is required to heat up the water before it enters to the evaporator. For this goal, an electric water heater with 7250 W, power has used. This water heater is working just when the water flow rate is 4 Lt/min or more, and the flow rate required for the evaporator is 1 Lt/min. Therefore we are running this heater just 15 minutes in an hour. The variation of radiation on December 27<sup>th</sup>, during the day time is indicated in the Figure 2. In Figure 3 variation of water production during the day time is shown. In Figure 4 the hourly water production versus the total rate of energy gained by the unit is indicated. For December 28<sup>th</sup>, the flow rate of water into the evaporator was adjusted on 1 Lt/min, and the air flow rate was 2 m<sup>3</sup>/min. The flow rate of entering water into the condenser decreased to be 4 Lt/min. The evaporator inlet water temperature was adjusted on 55 . This change results with decreasing the water production to 1.11 Lt for the whole day. although the average radiation increased on December 28<sup>th</sup>, but the production is decreased. Decreasing in production from 1.66 Lt on December 27<sup>th</sup> to 1.11 Lt on December 28<sup>th</sup> is because of decreasing in water flow rate into the condenser. The air temperature difference in the condenser reduced to 18.2 . Therefore, while we are decreasing the water flow rate, air flow rate must be decreased proportionally to keep the production in an acceptable level. The variation of total radiation on the collector surface is presented in Figure 5. Besides, in Figure 6, hourly changing of the water production is indicated. The variation of water production versus the total rate of energy gained by the whole unit is illustrated in Figure 7As it mentioned earlier, while the water flow rate is decreasing, the air flow rate must be decreased as well in order to maintain the water production at a tolerable level. The experiment during December 29<sup>th</sup> is conducted

with the same condition as December 28<sup>th</sup>, but the air flow rate is dropped to 1.5 m<sup>3</sup>/min. reducing in air flow rate will result in increasing the moist air temperature difference in the condenser, consequently increases the potable water production.

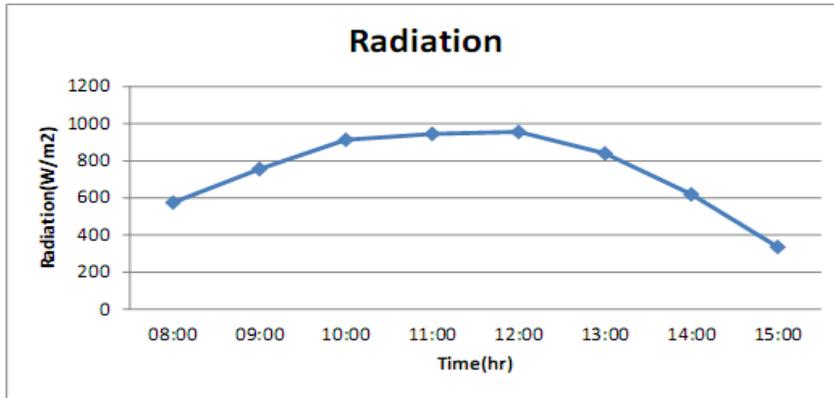


Figure 1: Insolation on the Collector Surface during December 21<sup>st</sup>

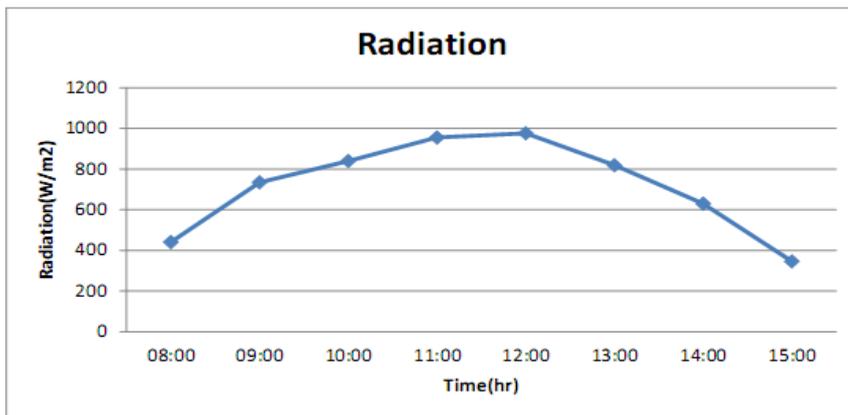


Figure 2: Total Radiation on December 27<sup>th</sup>

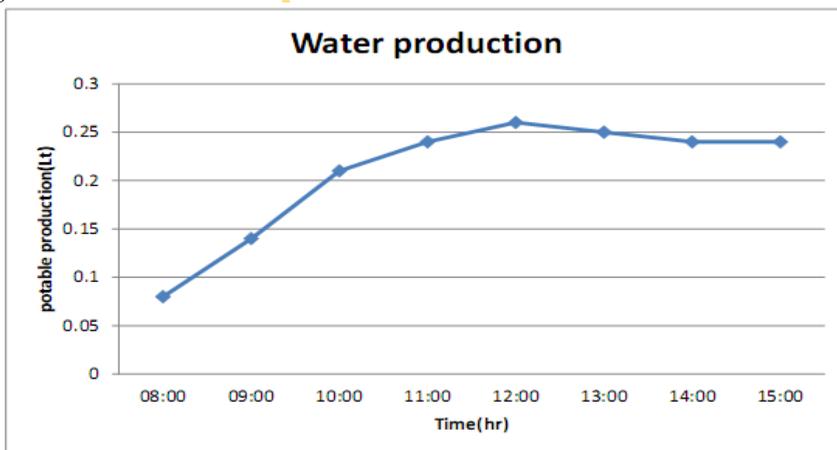


Figure 3: Water Production versus Time on December 27<sup>th</sup>

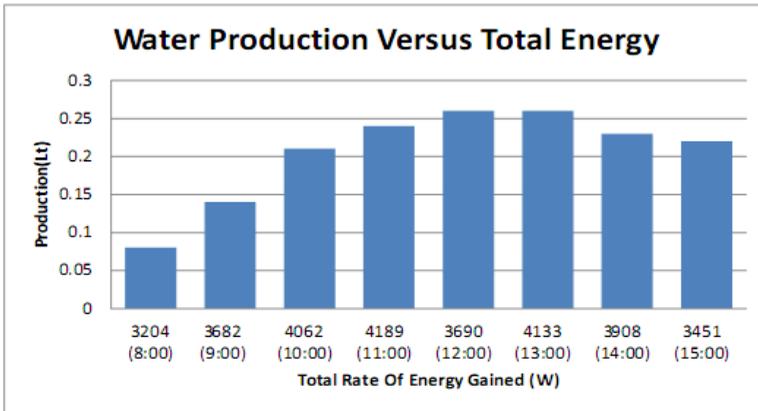


Figure 4: Water Production versus Total Rate of Energy Gained On December 27<sup>th</sup>, (Total Energy = Solar Energy + Electrical Energy)

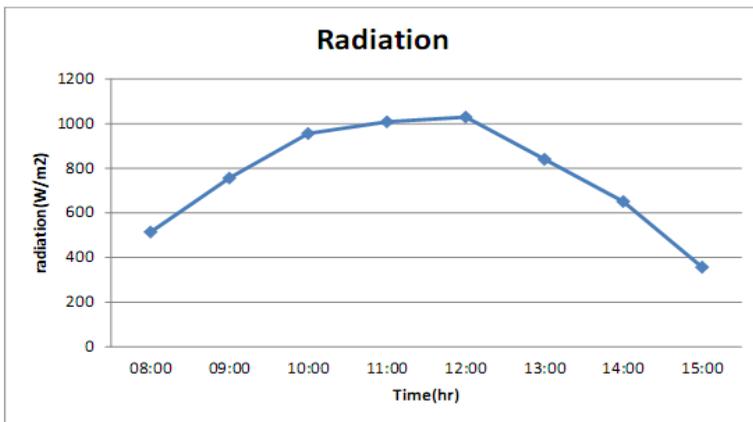


Figure 5: Variation of Insolation during the Day of December 28<sup>th</sup>

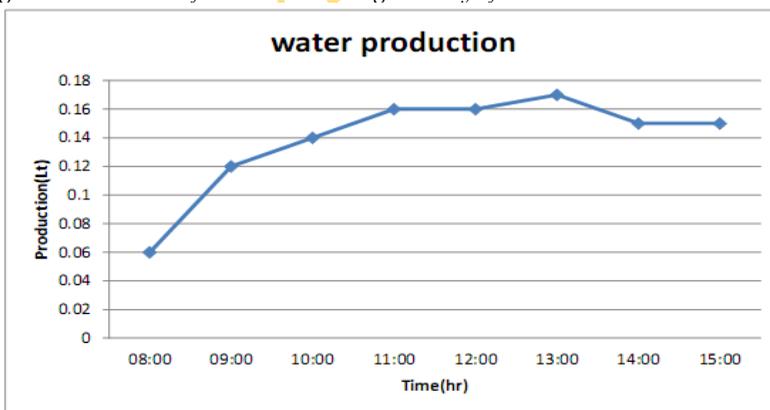


Figure 6: Variation of Water Production during the Day of December 28<sup>th</sup>

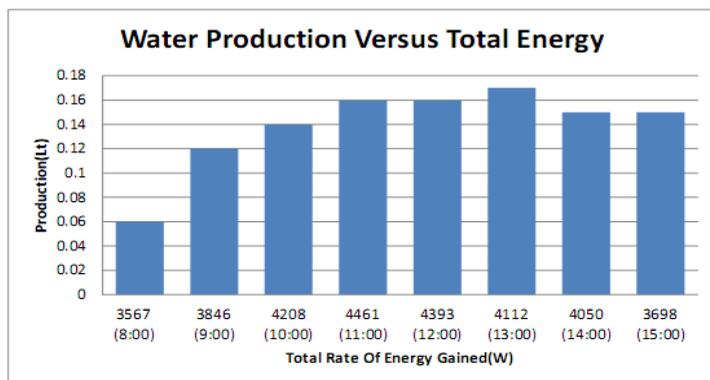


Figure 7: Water Production versus Total Rate of Energy Gained On December 28<sup>th</sup>, (Total Energy = Solar Energy + Electrical Energy)

### CONCLUSIONS

The present work is concerned with the setting up and conduction of an experiment on a humidification-dehumidification desalination system that was designed by Kraft et al. (Kraft, 2010). The experiment is carried out under North Cyprus winter condition. Additionally the governing equations, for each part of the unit are introduced and discussed. At the beginning of the study only air heating technique is employed, but the water production using only air heating method was almost zero. Therefore, water heating technique as well as air heating is applied. Finding the optimum flow rates for both water and the air was the intention of the experiment. The maximum production of potable water was achieved to be 3.1 Lt for the air flow rate of 1.5 m<sup>3</sup>/min, and water flow rate of 4 Lt/min for the condenser, and 1 Lt/min for the evaporator, whilst the water temperature was kept at 55 . The hourly average rate of total energy gained by the whole unit is 3775 W. In the present work it is observed that with the lower flow rate of air, higher flow rate of water and higher vaporator inlet temperatures for both water and air, the productivity of potable water will increase.

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