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Humidification Dehumidification Desalination Using Solar Collectors

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Abstract. The performance of a solar powered humidification-dehumidification system is studied theoretically for various constant parameters such as water flow rate, air flow rate, and cooling water temperature of dehumidifier etc. and varying solar intensity under climatological conditions of Surathkal, India. The primary components of the system are parabolic trough solar water heater, solar air heater, a humidifier, a dehumidifier and a storage tank. The mathematical model is developed by means of MATLAB software and governing equations are numerically solved using 4th order Runge-Kutta method. Daily and seasonal clean water productions are calculated for the system. Water is heated by using parabolic trough air heater whereas air is heated by solar air heater. The system used in this work is based on the idea of closed air and open water cycles.

1. Introduction

Water is available in abundance on the earth; however, there is a shortage of potable water in many countries in the world. The importance of ensuring availability of potable water of sufficient quality and quantity for today's needs and for future needs can hardly be overstressed. In the past, humans, plants, and animals have depended on lakes, rivers, and subsurface waters for their needs. However, continuing industrialization and population growth have resulted not only in increasing demands for fresh waters, but also in impairment of the limited sources. Even though several technologies are available for restoring impaired waters, they are often energy-intensive and not sustainable. With declining energy sources and the environmental impacts associated with energy production, current technologies are not sustainable solutions for meeting the water needs of the future.[8]

Desalination of saline waters is an option that has emerged as a feasible solution for producing potable water, able it at a high-energy cost. For example, in the oil-rich Arabian countries, where suitable fresh water sources are scarce and low-cost energy is abundant, desalination of seawater has become the process of choice. Installed capacity of traditional desalination units is estimated at 25 million m3/day world-wide, with an energy consumption of 0.63 million tons of oil/day (Kalogirou, 2001). Desalination processes can reduce the gap between supply and demand, if adequate sources of saline waters are available and if the energy requirements for desalination can be minimized [9].

A Humidification-dehumidification (HD) desalination process is regarded as a favourable technique for small water capacity production plants. The main feature of this process is its capability to operate

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at low temperatures, the use of low-level technical features, and the possible integration of sustainable energy sources.S. M. Soufari et al.[1], studied humidification-dehumidification(HD) desalination process and its performance optimization using mathematical programming. A nonlinear programing system model is solved for three objective functions like minimization of specific thermal energy consumption, maximization of productivity and maximization of condenser heat recovery. His results revealed that the humidifier inlet water temperature and air to water ratio are most important parameter.CemilYamal*I*,Ismail Solmus[2], theoretically investigated HD system using double pass solar air heater with two glass covers in the region of Ankara,Turkey.The energy equations are solved numerically for each component of the system using MATLAB software. The productivity of the system is increased up to 8% by using double pass solar air heater and decreased about 30% without double pass solar air heater under the same operating conditions. It is also found that the air and water mass flow rates have a significant influence on the system productivity.

Fahad A. Al-Sulaiman et al. [3] evaluated thermodynamic analysis to assess the performance of an HD system with an integrated parabolic trough solar collector (PTSC).Two different configurations were considered of the HD system. In the first configuration, the solar air heater was placed before the humidifier whereas in the second configuration the solar air heater was placed between the humidifier and dehumidifier. They concluded that PTSC is suitable for air heating for high intensity of solar radiation and second configuration was giving the better performance and a higher productivity.CihanYıldırım et al.[4]presented a theoretical model of to investigate the effect of design and operating parameters on clean water production rate for the climatically condition ofAntalya,Turkey .It is observed that water heating has a major importance on fresh water production. This is due to the fact that specific heat of water is higher than that of air.Farhad Nematollahi et al.[10] suggested the effect of temperature of air and water, humidity of air and dimensions of humidification tower on the overall exergy efficiency of the system. Their study aids in designing humidification tower for better solar desalination system.

Water requirement for NITK Surathkal campus is about 14 lakh Liters per day, out of which Mangalore City Corporation supply 50 % and the balance is met by open well in the campus and due to growing population, campus is going to face a scarcity of water in the near future. Moreover, ground water level is getting recede as the days pass on which will further contribute to the above mentioned water crisis. No study had been conducted in Surathkal region regarding this problem and furthermore due to its location, it receives an appreciable amount of solar intensity which can be utilized to construct an efficient system to obtain fresh water from saline water. In the present study, humidification-dehumidification desalination system is made by using a combination of double-pass flat plate solar air heater with two glass covers and a Parabolic Trough Solar Collector (PTSC). The primary objective of this work is to observe the impact of various parameters on productivity of the configuration and to develop a theoretical model to cope up with the increasing demand of water in the region.

2. System Description and Mathematical Modeling

2.1 System Description

Humidification dehumidification desalination consists of a parabolic trough water heater, a double pass solar air heater, a humidifier, a dehumidifier, and a storage tank. The system is based on closed air and open water circuit. The schematic diagram is shown below fig.1Air at ambient the ambient condition is heated by the first pass of double pass solar air heater and further heated by second pass, then humidified with seawater water in the humidifier. Sea water is heated by parabolic trough solar collector (PTSC) and distributed to humidifier. Heated sea water and air are brought together in the humidifier. Hot humidified air is passes through the dehumidifier from which water vapor condenses and turns into to fresh water. Collected water at the bottom of the humidifier is stored in the insulated tank. The water at the tank is pumped into the solar water heater and recalculated to the humidifier. Cooling water is supplied to the dehumidifier from a constant temperature water bath.

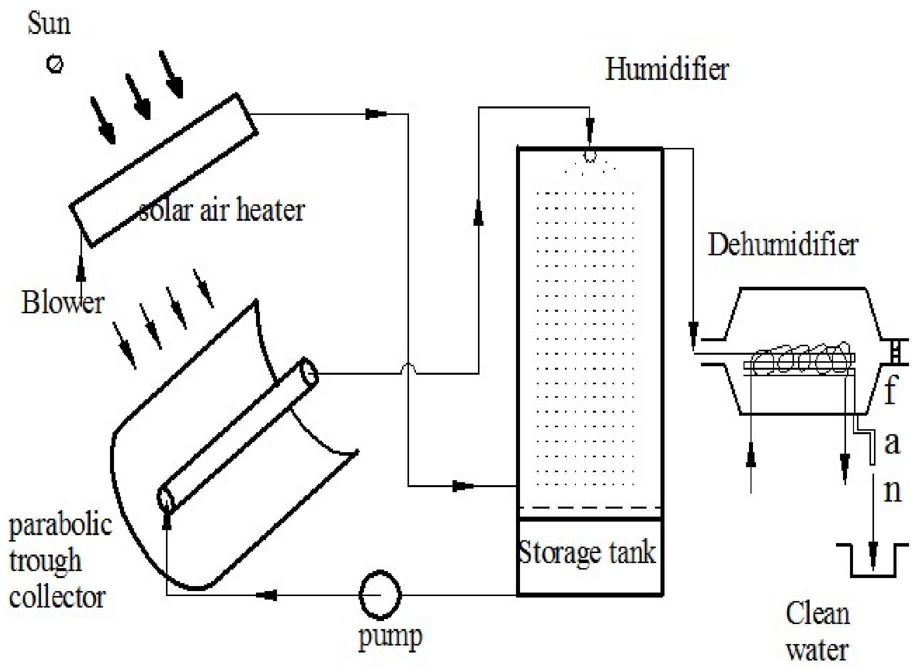

Figure 1. A humidification dehumidification process with solar and parabolic collector.

2.2 Mathematical Modeling

A theoretical mathematical model is proposed to simulate the system in question. The energy and mass balance equations are derived for each of the system components. The governing conservation equations were simultaneously solved by fourth order R-K method. The model proposed based on the following assumptions and simplifications:

Inlet water temperature to the humidifier is equal to the storage tank water temperature. Cooling water temperature is constant during the day. Air temperature varies linearly in the flow directions. The effectiveness of the humidifying tower is assumed to be equal to one which means that the air leaving the humidifier is at saturation condition and therefore, its wet-bulb and dry-bulb temperatures are identical. The dehumidification process lies on the saturation curve. Exit temperatures of the condensate water and cooling water from the parallel flow dehumidifying exchanger are the same as the dry-bulb temperature of the air leaving the dehumidifier.Temperature gradient inside the water storage tank is neglected. Heat losses (or gains) from the edges of the solar air heater, water storage tank, humidifier and dehumidifier to the ambient are neglected.Flow which is laminar or turbulent is fully developed.Temperature of the water leaving the humidifier is equal to the wet-bulb temperature of the air leaving the humidifier. In the both channels, radiant energy absorbed by the flowing air is neglected. There is no air leakage from the system, when air passes through the air heater, humidifier and dehumidifier in that sequence. Solar radiation, ambient temperature, wind speed, relative humidity of the ambient air are constant at each instant through 60 minutes.

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Modeling of PTSC:

Absorbed Radiation: [3]

$$S = I_b \rho (\Upsilon\tau\alpha)_n K_{\tau\alpha} \tag{1}$$

where I_b is the available beam radiation is the collector reflectivity, and τ_a is the transmittance absorptance product.

Convection and Radiation Heat loss:

$$Q_{loss} = \pi D_{co} L h_w (T_{co} - T_a) + \epsilon_c \pi D_{co} L \sigma (T_{co}^4 - T_{sky}^4) \tag{2}$$

where h_w is the outside heat transfer coefficient, a is the ambient temperature, and T_{sky} is the sky temperature.

Radiative heat transfer loss: The radiative heat transfer from the receiver to the covers inner surface is given by [7]

$$Q_{loss} = \frac{\pi D_o L \sigma (T_r^4 - T_{ci}^4)}{\frac{1}{\epsilon_c} + \frac{1-\epsilon_c}{\epsilon_c}\left(\frac{D_o}{D_{ci}}\right)} \tag{3}$$

Where D_o is the outer diameter of the receiver, D_{ci} is the cover inner diameter length of the collector, σ is Stefan-Boltzmann's constant, T_r is the receiver temperature, T_{ci} is the cover inner temperature and ϵ_c is the cover emissivity.

Conductive heat transfer loss:

The conductive heat loss through the cover thickness is given by

$$Q_{loss} = \frac{2\pi k_c L (T_{ci} - T_{co})}{\ln\left(\frac{D_{co}}{D_{ci}}\right)} \tag{4}$$

where k_c is the thermal conductivity , T_{co} is the cover outer temperature, and D_{co} is the outer diameter.

Total heat loss:

$$Q_{loss} = U_l A_r (T_r - T_a) \tag{5}$$

where U_L is the overall heat transfer coefficient and A_r is the receiver area. h_w is calculated using Nusselt number correlation for the flow of air along a tube in an outdoor environment as:

Nusselt Number for laminar flow:

$$Nu = 0.40 + .54 Re^{0.52} \quad \text{for } 0.1 < Re < 1000 \tag{6}$$

$$Nu = 0.30 Re^{0.6} \quad \text{for } 1000 < Re < 50000$$

where Re is the Reynolds's number.

Nusselt Number for turbulent flow:

$$Nu = \frac{\left(\frac{f}{8}\right)(Re - 1000) Pr}{1.07 + 12.7\sqrt{\frac{f}{8}}\left(Pr^{\frac{2}{3}} - 1\right)} \left(\frac{\mu}{\mu_w}\right)^n \tag{7}$$

n is equal to 0.11 for heating and equal to 0.25 for cooling.

Darcy friction factor:

$$f = (0.79 \ln Re - 1.64)^{-2} \tag{8}$$

Overall heat transfer coefficient between the environment and the fluid is given by

$$U_o = \left(\frac{1}{U_l} + \frac{D_o}{h_{fi} D_i} + \frac{D_o \ln\left(\frac{D_o}{D_i}\right)}{2k_c}\right)^{-1} \tag{9}$$

where h_{fi} is fluid heat transfer coefficient

Useful energy gain:

$Q_u = F_R A_a [S - \frac{A_r}{A_a} U_l (T_i - T_a)]$ (10)

where F_R is heat removal factor

Collector flow factor:

$F'' = \frac{F_R}{F'} = \frac{\dot{m}C_p}{A_r U_l F'} [1 - \exp(-\frac{A_r U_L F'}{\dot{m} C_p})]$ (11)

where F' is given by the following equation

Collector Efficiency factor:

$F' = \frac{U_o}{U_l}$ (12)

Useful energy gain in terms of collector inlet and outlet temperatures:

$Q_u = \dot{m} C_p (T_i - T_o)$ (13)

where T_o is the outlet temperature

Taking the variation in the transmissivity, absorptivity, and the reflectivity into consideration the incidence angle modifier is then given by

$K_{\tau\alpha} = 1 - 6.74 * 10^{-5} * \theta^2 + 1.64 * 10^{-6} * \theta^3 - 2.51 * 10^{-8} * \theta^4$ (14)

MODELING OF WATER STORAGE TANK:

$m_{w1} C_{pw} \frac{dT_{w1}}{dt} = M_{w1}(t) C_{pw} T_{w2}(t) + M_{mw}(t) C_{pw} T_{mw}(t) - M_{w1} C_{pw} T_{w1}(t) - q_{1w1amb}$ (15)

MODELING OF SOLAR AIR COLLECTOR:

The energy balance equations for the solar air heater are as follows:[2]

Second glass cover:

$m_g C_{pg} \frac{dT_{g2}}{dt} = I . \alpha_g . A_c + Q_{r,g1g2} - Q_{c,g2amb} - Q_{r,g2s} + Q_{c,g1g2}$ (16)

For the first glass cover:

$m_g . C_{pg} \frac{dT_{g1}}{dt} = I . \alpha_g . \tau_g . A_c - Q_{r,g1g2} - Q_{c,g2\,a1} + Q_{r,pg1} - Q_{c,g1g2}$

(17)

First air pass :

$m_a C_{pa} \frac{dT_{a1}}{dt} = Q_{c,pa1} + Q_{c,g1a1} - M_a C_{pa} (T_{a1e} - T_{ai})$ (18)

Absorber plate:

$m_p C_{pp} \frac{dT_p}{dt} = I . \alpha_p . \tau_g^2 . A_c - Q_{c,pa2} - Q_{c,pa1} - Q_{r,pg1} - Q_{r,pb}$

(19)

Second air pass:

$m_a C_{pa} \frac{dT_{a2}}{dt} = Q_{c,pa2} + Q_{c,ba2} - M_a C_{pa} (T_{a2e} - T_{a1e})$ (20)

Base plate:

$m_b C_{pb} \frac{dT_b}{dt} = Q_{r,pb} - Q_{c,ba2} - Q_{1,b\,amb}$ (21)

MODELING OF HUMIDIFIER:

$\dot{m}_a (h_{a3}(t) - h_{a2}(t)) = \dot{m}_{w1} C_{pw} T_{w2}(t) - \dot{m}_{w2} C_{pw} . T_{w3}(t)$

(22)

MODELING OF DEHUMIDIFIER:

$\dot{m}_a (h_{a3}(t) - h_{a4}(t)) = \dot{m}_{w3} C_{pw} (T_{w5}(t) - T_{w4}(t)) - \dot{m}_c(t) C_{pw} T_{w6}(t)$

(23)

The Heat Transfer Expressions:

The heat transfer terms in the equations above are given as below.

$Q_{r,g1g2} = A_c \,.h_{r,g1g2} \,.(T_{g1} - T_{g2})$ (24)

where Ac = w.L

The radiation heat transfer coefficient between the two glass cover can be found by

$h_{r,g1g2} = \frac{\sigma.(T_{g1}^2 + T_{g2}^2).(T_{g1}+T_{g2})}{(\frac{1}{\varepsilon_{g1}} + \frac{1}{\varepsilon_{g2}} - 1)}$ (25)

$Q_{c,g2amb} = A_c \,.h_{r,g2amb}.(T_{g2} - T_{amb})$ (26)

$h_{r,g2amb} = 2.8 + 3.V_{wind}$ (27)

$Q_{r,g2\,s} = A_c \,.h_{r,g2s}.(T_{g2} - T_s)$ (28)

where$T_{sky} = T_{amb} - 6$

And the radiation heat transfer coefficient from second glass cover to sky is

$h_{r,g2\,s} = \varepsilon_{g2}\,.\sigma.\left(T_{g2}^2 - T_s^2\right).(T_{g1} - T_s)$ (29)

$Q_{c,g1g2} = A_c \,.h_{c,g1g2} \,.(T_{g1} - T_{g2})$ (30)

The natural convection heat transfer coefficient between the first and second glass covers is given as follow:

$h_{c,g1g2} = Nu_{g1g2} \,.\, \frac{K_a}{x}$ (31)

$$Nu_{g1g2} = 1 + 1.44 \times \left[1 - \frac{1708}{R_a \times CosB}\right]^{+} \left(1 - \frac{(Sin1.8 \times B)^{1.6} \times 1708}{R_a \times CosB}\right) + \left[\left(\frac{R_a \times CosB}{5830}\right)^{1/3} - 1\right]^{+} \quad (32)$$

The positive exponent means that the value of the term is equal to zero if the term is negative.

$R_a = \frac{g\,.\,\beta\,.(T_{g1} - T_{g2})\,.x^3}{\alpha\,.\vartheta}$ (33)

Thermal properties

$K = .0244 + .6773 \times 10^{-4} \times T$ (34)

$\alpha = 7.7255 \times 10^{-10} \times T^{1.83}$ (35)

$\vartheta = 0.1284 \times 10^{-4} + 0.00105 \times 10^{-4} \times T$ (36)

$Q_{c,g1\,a} = A_c \,.h_{c,g1a1}.(T_{g1} - T_{a1})$ (37)

The forced convective heat transfer coefficient inside the upper channel of the double pass flat plate solar air heater can be found by:

$h_{c,g1a1} = Nu_{g1a1} \,.\, \frac{K_a}{D_h}$ (38)

$$D_h = \frac{4\,A_{sec}}{(2.W+2.D)} \tag{39}$$

For laminar flow

$$Nu_{g1a1} = 4.9 + \frac{.0606\,.(Re_{a1}.\mathrm{Pr}.D_h/L.)^{1.2}}{1+0.0909\,.(Re_{a1}.\mathrm{Pr}.D_h/L.)^{.7}.Pr^{0.17}} \tag{40}$$

$$Nu_{g1\,a1} = \frac{(f_{a1}/8).(Re_{a1}-1000)\times Pr}{1+12.7\times(f_{a1}/8)^{.5}\times(Pr^{0.67}-1)} \tag{41}$$

where Re_{a1} is the Reynolds number
For turbulent flow

$$Nu_{g1\,a1} = \frac{(f_{a1}/8).(Re_{a1}-1000)\times Pr}{1+12.7\times(f_{a1}/8)^{.5}\times(Pr^{0.67}-1)} \tag{42}$$

where Re_{a1} is the Reynolds number

$$Re_{a1} = \frac{V_{a1}D_h}{\vartheta_{a1}} \tag{43}$$

$$f_{a1} = (0.79\times lnRe_{a1} - 1.64)^{-2} \tag{44}$$

$$Q_{r,pg1} = A_c\,.h_{c,pg1}.(T_p - T_{g1}) \tag{45}$$
$$Q_{c,pa1} = A_c\,.h_{r,pa1}.(T_p - T_{a1}) \tag{46}$$

$$\dot{m}_c\,(t) = \dot{m}_a\,.[w_3(t) - w_1(t)] \tag{47}$$

3. Solution Procedure

A computer simulation program has been developed using MATLAB based on the energy equations to study the effect of various parameters such as mass flow rate of sea water, air, temperature etc. In this simulation program, energy equations are solved simultaneously using the forth order Runge-Kutta method. The interval is chosen to be 10 seconds and the initial values of $T_{g2}.T_{g1}$, T_{a1}, T_{a2}, T_{w2} are assumed to be nearly equal to atmospheric temperature. The base plate temperature andabsorber temperature are assumed to be 5 and 10degrees above the ambient temperature. Then, the first order differential equations are solved numerically to obtain new temperatures which are used as initial conditions and the process above will be repeated for the next time step.
Mean parabolic trough water heater temperature is guessed initially and calculated iteratively. As a result, after knowing inlet water and air temperatures to the humidifier at every interval of time, the temperature of the air leaving from the humidifier is calculated by using Eq. (22) and the temperature of the air leaving from the dehumidifier is evaluated by Eq. (23).The quantity of the fresh water product from the dehumidifier is calculated by Eq.(47)

4. Results And Discusion:

To simulate the humidification-dehumidification system and to obtain the distilled water quantity, the variations of direct solar radiation through the day over Surathkal, India region have been used as input values. Fig.2 represents the direct solar radiation intensity for the most effective3 months of the year.
The main goal of the present work is to construct the mathematical model, simulate, and predict the behavior and the fresh water productivity from the solar driven humidificationdehumidification system.The quantity of the fresh water depends on many parameters, such as sea water mass flow rate to air mass flow rate and the temperature of the sprayed water and air. In this analysis all the factors affect the fresh water productivity from these system are assumed to be constant.The study covered the

three seasons of the year. Calculations have been carried out to get the effect of parameters on the production of fresh water.

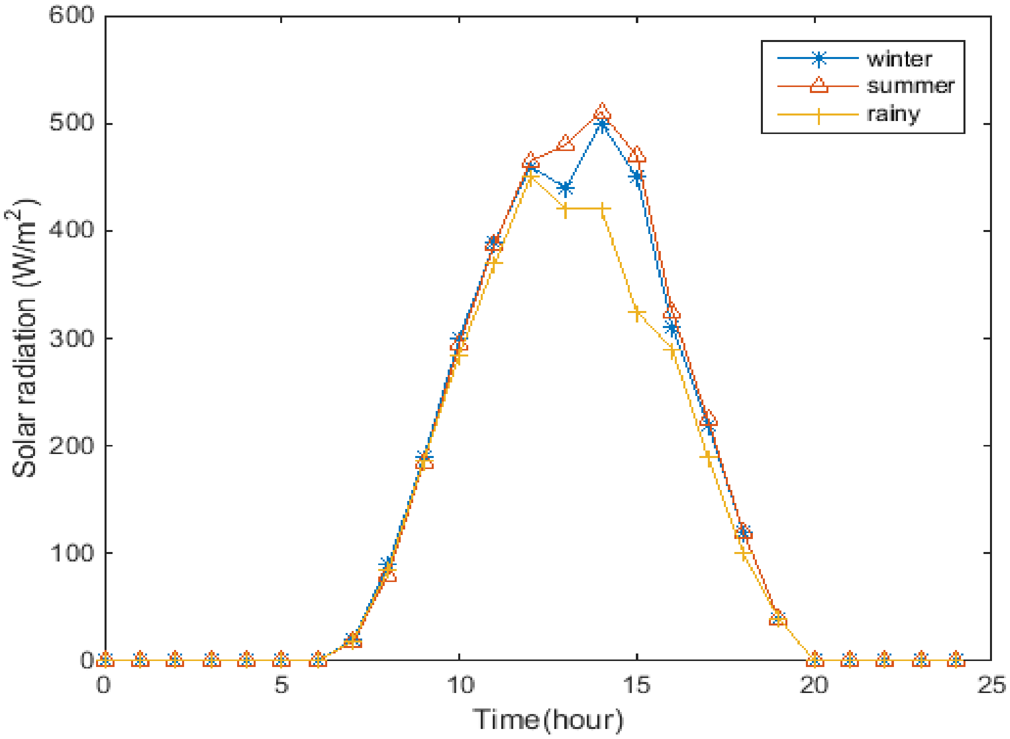

Figure 2. Variation of solar radiation with time.

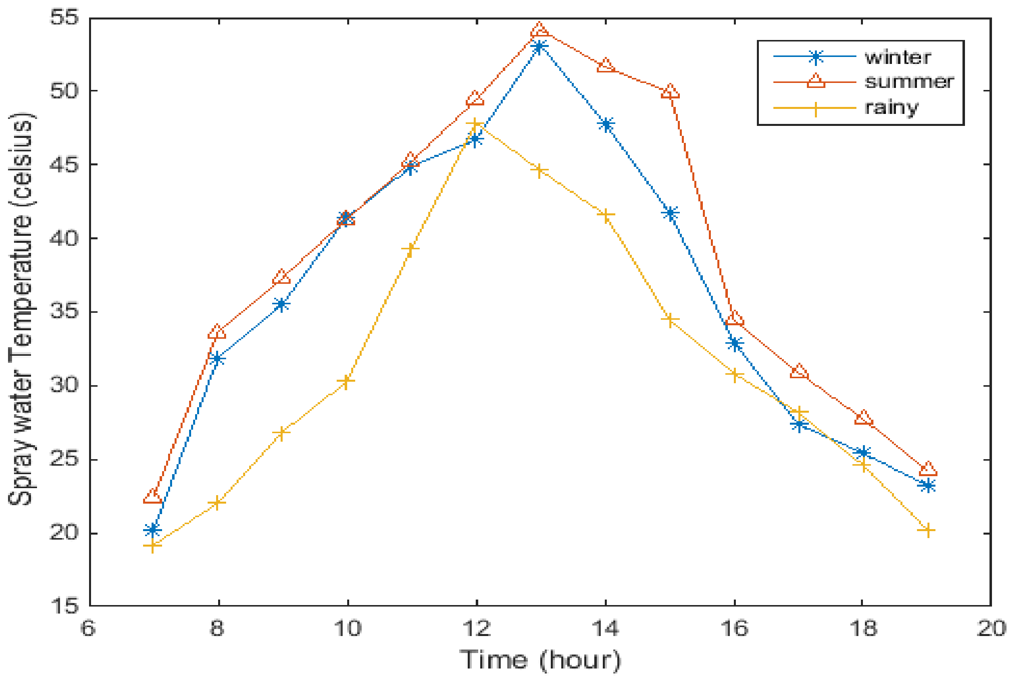

Figure 3. Variation of spray water temperature of ptsc with time.

A comparison had been made between the various values of PTSC outlet water temperature for different seasons of the year and obtained results are plotted in the Fig.3. It can be observed that the temperature is maximum at the summer season and it is due to the fact that incident solar radiation on the collectors is high in the summer season. Minimum values of the outlet temperature from the collectors occurred in the winter season.

Figure 4.shows the comparison between the quantities of fresh water per kg of air for the system related to direct solar radiation for three seasons of the year. Furthermore, a comparison had been made for different months of the year to show the seasonally effect on the proposed system productivity. It is clear from the graph that maximum quantity of fresh water is obtained in the summer season, while minimum quantity of fresh water in winter compared to summer season. However, the lowest quantity is obtained in the rainy season. Figure 5.and figure 6. shows the maximum fresh water per kg air values and the daily fresh water quantity of the day for different seasons.

The main output from the proposed system is the quantity of fresh water produced from the dehumidification process of the considered system. The combination of PTSC and solar air heater increase the temperature of water and air respectively. Due to rise in temperature, the heat capacity of both air and water increases which will further lead to generation of high moisture content and hence increment the quantity of fresh water obtained.

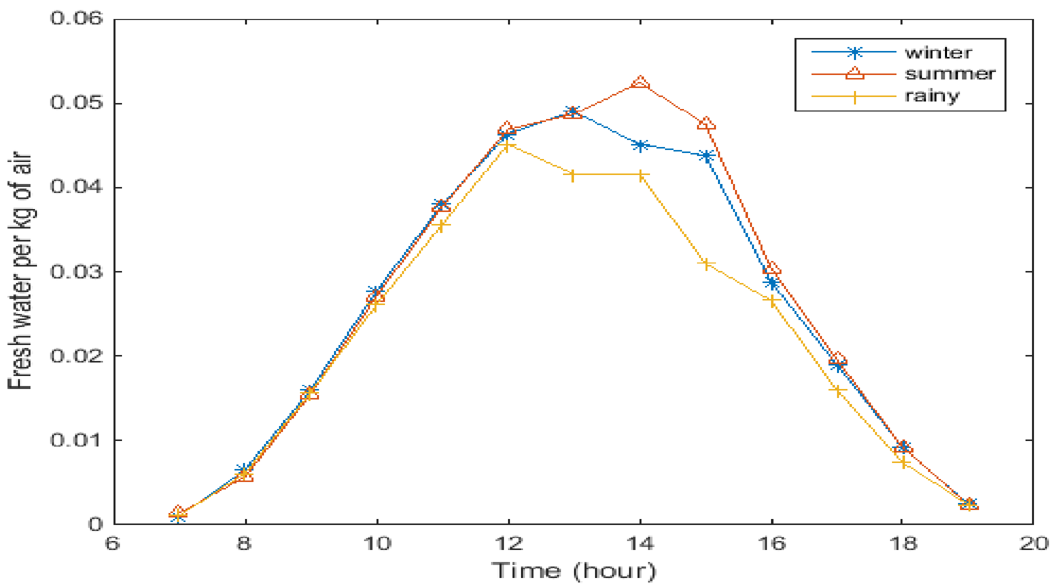

Figure 4. Comparison between the quantities of fresh water per kg of air for different seasons of the year.

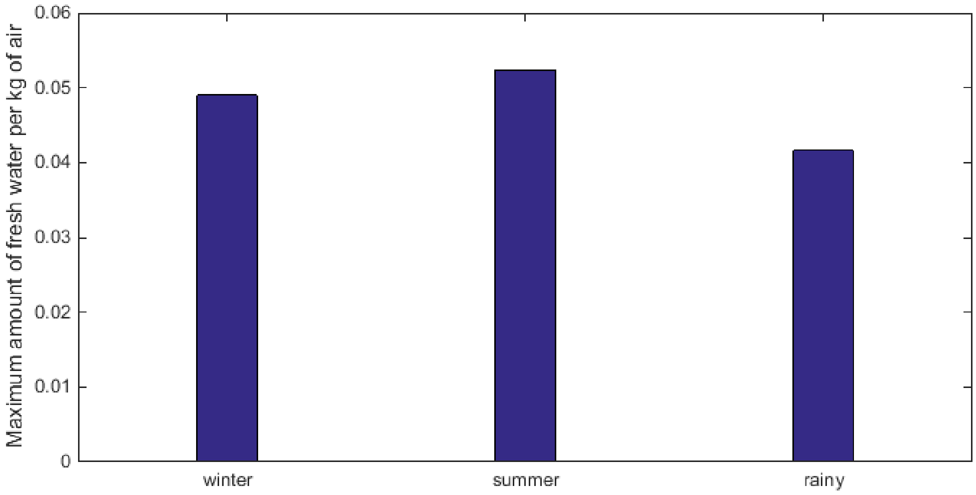

Figure 5. Comparison between the maximum quantities of fresh water per kg of air for different seasons of the year.

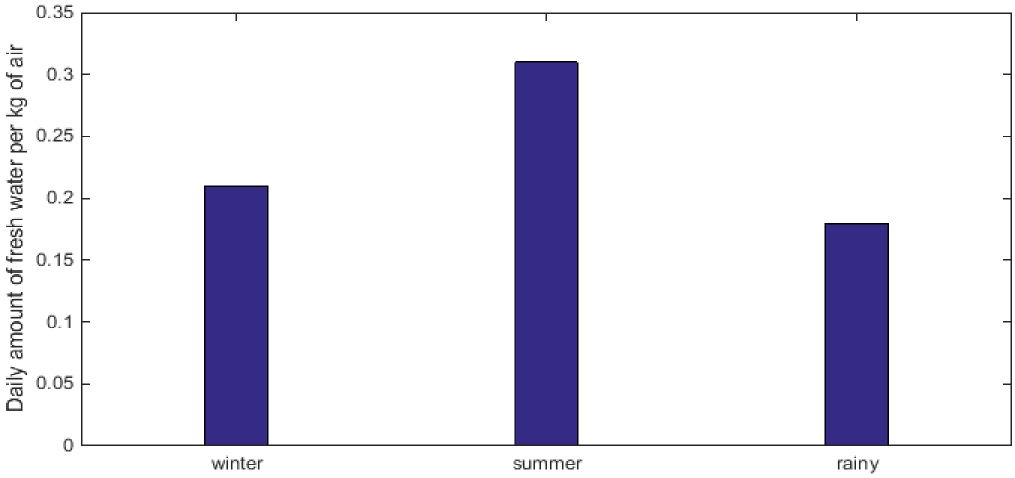

Figure 6. Comparison between the daily quantities of fresh water per kg of air for different seasons of the year.

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