VOL. 19, NO. 7, APRIL 2024 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences


www.arpnjournals.com

# DESIGN AND DEVELOPMENT OF WATER PURIFIER USING HUMIDIFICATION AND DEHUMIDIFICATION PROCESS BY USING RENEWABLE ENERGY

Surekha Patil, Anand Bhatt, Tejas Raval, Ath Singhal and Shelin Mehta
Mechanical Engineering, Institute of Technology, Nirma University, Ahmedabad, India
E-Mail: anand.bhatt@nirmauni.ac.in

**ABSTRACT**

Drinking water, the basic requirement for life is deteriorating day by day and its sources are decreasing at a faster rate. Sea water is present in abundance on the Earth but converting it to drinkable form is expensive for domestic usage. The paper focuses on purifying the sea water using the humidification and dehumidification process and reducing its TDS so that it can be consumed. The system works on the principle of the natural water cycle in which the sea water evaporates and humidifies the air just above. As the temperature of air drops below its dew point, the vapour in the air condenses and falls on the earth as rain. Similarly, the experimental setup consists of a solar water heater of evacuated tube collector to heat the water, a double-pipe heat exchanger to heat the impure sea water from hot water, a humidifier where the air gets heated and humidified by the impure water, and a dehumidifier where the air gets cooled below the dew point and thus its condensation results in pure water. The paper presents the design of a humidifier, dehumidifier, and double-pipe heat exchanger and the results obtained from the setup built based on the design.

**Keywords:** humidification, dehumidification, evacuated tube collector, double-pipe heat exchanger.

Manuscript Received 10 February 2024; Revised 20 April 2024; Published 30 May 2024

## ABBREVIATION, NOTATIONS AND NOMENCLATURE

$T_a$ = $Air\ temperature\ in\ K$
$T_w$ = $Water\ temperature\ in\ K$
$\omega$ = $Humidity\ Ratio\ in\ kgkg\ of\ dry\ air$
$dX$ = $infinitisimal\ element\ of\ property$ X
$\frac{dX}{dZ}$= $rate\ of\ change\ of\ property$ X $with\ respect\ to\ dZ$
$m_a$, $m_w$ = $mass\ flow\ rates\ of\ air\ and\ water\ respectively\ in\ kg/s$
$h_{ma}$ = $Enthalpy\ of\ air\ in\ kJ/kg$
$Q_m$, $Q_c$ = $Energy\ transfer\ due\ to\ mass\ transfer\ and\ heat\ transfer\ resp.\ in\ kJ$
$h_v$ = $Latent\ heat\ of\ evaporation\ in\ kJ/kg$
$h_d$ = $Mass\ Transfer\ Coefficient\ in\ kg/m2s$
$h_{fgwO}$ = $Latent\ heat\ of\ vapourisation\ at\ 0°C\ in\ kJ/kg$
$C^a{}_{pa}$= $specific\ heat\ capacity\ of\ air\ at\ air\ temperature\ in\ kJ/kgK$
$C^a{}_{pa}$= $specific\ heat\ capacity\ of\ vapour\ at\ air\ temperature\ in\ kj/kgK$
$C^w{}_{pw}$ = $specific\ heat\ capacity\ of\ water\ at\ water\ temperature\ in\ kJ/kgK$
$C^w{}_{pv}$ = $specific\ heat\ capacity\ of\ vapour\ at\ water\ temperature\ in\ kJ/kgK$
$Le_f$ = $Lewis\ Factor$
$a_{Fi}$ = $Area\ Density\ in\ m2/m3$
$A_{Fr}$ = $Frontal\ Area\ in\ m2$

## 1. INTRODUCTION

Drinking water, the basic requirement for human survival, is getting very scarce and even many regions suffer from lack of water resources. Water is an essential human need. 97% of the water on Earth exists in seas and oceans, 2% is in ice caps and glaciers, and remaining 1% exists in rivers and lakes. On the other hand, rise of industries and agriculture, the need for pure water has increased. Sole dependence on natural resources for pure water supply for drinking purposes and industries cannot be satisfied. This has forced mankind to find alternatives to obtain low salinity water. Existing conventional water purification processes such as RO (Reverse Osmosis), MSF (Multi-stage Flash), MED (Multiple Effect Distillation), etc. use electricity for the desalination of water. The production of electricity requires pure water and fossil fuels in power plants. Using Renewable Energy Sources for purification of the water is the solution. Thermal Energy of the Solar systems can be used for the purification of the saline water.

The work aims to experimentally and numerically study a specifically designed Humidification and Dehumidification plant with a double pipe Heat-Exchanger using Solar Energy. This particular system will be designed, fabricated, and tested to find the various climatic conditions on the performance of the system.

## 2. LITERATURE REVIEW

0Al-Hallaj *et al.* [1] used a solar collector (tubeless flat plat type of 2 $m^2$ area) to heat the water to 50-70 ℃ and the air is circulated by both natural and forced convection to compare the performance of both these modes. They found that at low top temperatures forced circulation of air was advantageous and at higher top temperatures natural circulation gives better performance. Ben-Bacha *et al.* [2] used a solar collector ($6m^2$area) to heat water and thorn trees as packing material in a humidifier. They found that water temperature at the inlet to the humidifier, the air and water flow rate along

ARPN Journal of Engineering and Applied Sciences



with humidifier packing material play a vital role in the performance of the plant. Farid *et al*. [3] used a 1.9 $m^2$ solar collector to heat water, forced circulation for air, wooden shaving packing in the humidifier, and multiple pass shell and tube heat exchanger as a dehumidifier. They achieved water production of 12L/$m^2$ and found out that the effect of air velocity on production is complicated and can't be simply stated and the water flow rate was observed to have an optimum value. Nafey *et al*. [4] used separate heaters for air and water, canvas as the packing material in humidifiers and air-cooled dehumidifiers. They obtained water production of 1.2 L/h and about 9L/day and observed that a higher air mass flow rate gave less productivity as increasing air flow reduced the inlet temperature of the humidifier. Dai *et al*. [5] used honeycomb paper as humidifier packing, forced circulation for air flow, and fin-tube type condenser. It was observed that the performance of the system was strongly dependent on the temperature of inlet salt water to the humidifier, the mass flow rate of salt water, and process air. It was also observed that the top water temperature has a strong effect on the production of fresh water. Khedr [6] used a packed tower dehumidifier (with 50mm ceramic raschig rings). They obtained a GOR of 0.8 and based on economic analysis, they concluded that the HDH systems have significant potential for small capacity desalination plants as low as 10 $m^3$/day. Yamali *et al*. [7] used single stage double pass flat plate solar collector to heat water, pad humidifier, and finned tube dehumidifier. The plant produced 4 kg/day and it was observed that increasing the air flow rate has no effect on the performance but an increase in water flow rate increased the productivity. Houcine *et al*. [8] used 5 heating and humidification stages, corrugated cellulose material as packing in humidifier 3 and the air was forced circulated. The maximum water production was 516L/day and 37% of the cost is that of the solar collector field. Klausner [9] used a direct contact packed bed dehumidifier, and waste heat to heat water to 60°C, and part of the water produced was used as a coolant and recovers heat from this coolant in a separate heat exchanger. They obtained fresh water production efficiency of 8% with an energy consumption of 0.56 kWh/kg of fresh water production based on the feed water temperature of 60°C and the energy consumption doesn't include the solar energy consumed. Muller-Holst *et al*. [10] used natural circulation for air, a thermal storage tank of 2m3 size, 38m2 collector field to heat water to 80-90°C and latent heat recovered to heat the water to 75°C. They obtained a GOR of 3-4.5 and daily water production of 500 L for a pilot plant in Tunisia and a 50% reduction in cost because of continuous operation provided by the thermal storage device. Nawayseh *et al.* [11] used solar collectors to heat water to 70-80°C, both natural and forced circulation, humidifier with vertical/inclined wooden slates packing, and heat recovered in the condenser. They observed that the water flow rate has a major effect on the wetting area of the packing and natural circulation yields better results than forced circulation. Moreover, Singhal [12] reduced the top loss coefficient of solar flat plate collectors by providing a trapezoidal glass cover, and efficiency was improved from 48% to 77% by adding aluminum foil reflectors as well. Hence, the same type of modification can also be useful for heating the water. Bhatt *et al* [13][14][15] have developed a heat pipe with multiple heat sources and optimized its performance. Heat pipes have become more and more popular in solar collectors to harness the maximum energy from solar radiation.

## 3. MATHEMATICAL MODELING

### 3.1 Humidifier

In the humidifier, hot water coming out of the Solar Water Heat Exchanger will be sprinkled on the packing material.

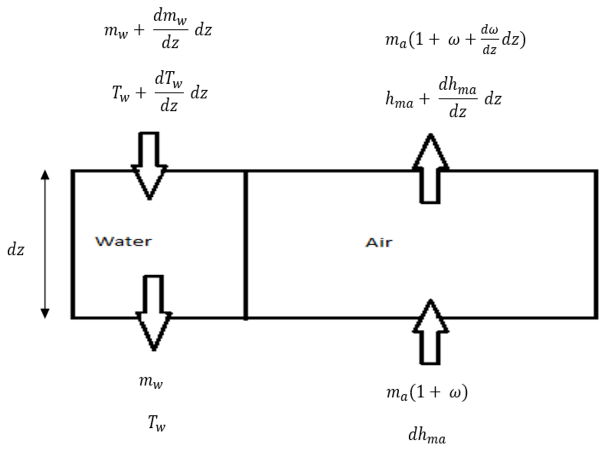



**Figure-1.** Mass flow balance along the humidifier.

Air at ambient conditions will be blown from the bottom which itself gets heated up and humidified and cools the water by evaporative cooling.

Considering a small control volume of the humidifier of height dz,

Applying mass balance to the control volume,

$$\frac{dm_w}{dz} = m_a \frac{d\omega}{dz} \tag{1}$$

Applying energy balance to the control volume,

$$m_a h_{ma} + \left(m_w + \frac{dm_w}{dz}dz\right) C_{pw}^{w} \left(T_w + \frac{dT_w}{dz}dz\right) = m_a \left(h_{ma} + \frac{dh_{ma}}{dz}dz\right) + m_w C_{pw}^{w} T_w$$

By rearrangement, we get

$$m_w C_{pw}^{w} \frac{dT_w}{dz} + C_{pw}^{w} T_w \frac{dm_w}{dz} = m_a \frac{dh_{ma}}{dz} \tag{2}$$

Where $h_{ma} = C_{pa}^{a} T_a + \omega \left( h_{fgw0} + C_{pv}^{a} T_a \right)$ (3)

Now,
Total Energy transfer
= Energy transfer due to mass transfer
+ Energy transfer due to heat transfer

$$dQ = dQ_m + dQ_c$$

Mass transfer,

$$\frac{dm_w}{dz} = h_d\,(\,\omega_{sw} - \omega\,)\frac{dA}{dz} \tag{4}$$

Mass transfer rate,

$$dQ_m = h_v\,\frac{dm_w}{dz} = h_v\,h_d\,(\omega_{sw} - \omega)\frac{dA}{dz}$$

Where $h_v = h_{fgw0} + C_{pv}^{w}\,T_w$ (5)

Heat transfer rate,

$$dQ_c = h\,(\,T_w - T_a\,)\frac{dA}{dz}$$

Total Energy transfer,

$$dQ = [h_v\,h_d\,(\omega_{sw} - \omega) + h\,(\,T_w - T_a\,)]\frac{dA}{dz}$$

The energy transferred from the water to the air due to evaporation of mass and heat convection is balanced with the increase of the enthalpy of the air.

$$m_a\frac{dh_{ma}}{dz} = [h_v\,h_d\,(\omega_{sw} - \omega) + h\,(\,T_w\,T_a\,)]\frac{dA}{dz} \tag{6}$$

Substituting (4) in (1), we get

$$\frac{d\omega}{dz} = \frac{h_d\,(\,\omega_{sw} - \omega\,)}{m_a}\frac{dA}{dz} \tag{7}$$

Differentiating equation (3) concerning z, we get

$$\frac{dh_{ma}}{dz} = (C_{pa}^{a} + \omega C_{pv}^{a})\frac{dT_a}{dz} + \left(h_{fgw0} + C_{pv}^{a}\,T_w\right)\frac{d\omega}{dz} \tag{8}$$

Combining (8), (7) and (6) yields,

$$\frac{dT_a}{dz} = \frac{h_d[Le_f(\,T_w - T_a\,)(C_{pa}^{a}+\omega C_{pv}^{a}) + (C_{pv}^{w}T_w - C_{pv}^{a}T_a)(\omega_{sw} - \omega)]}{m_a(C_{pa}^{a}+\omega C_{pv}^{a})}\frac{dA}{dz} \tag{9}$$

Combining (6), (4) and (2) yields,

$$\frac{dT_w}{dz} = \frac{h_d[Le_f(\,T_w - T_a\,)(C_{pa}^{a}+\omega C_{pv}^{a}) + (h_{fgw0} + C_{pv}^{w}T_w - C_{pw}^{w}T_a)(\omega_{sw} - \omega)]}{m_w\,C_{pw}^{w}}\frac{dA}{dz} \tag{10}$$

where, $Le_f = \frac{h}{h_d(C_{pa} + \omega C_{pv})}$, $\left(\,Le_f = Lewis\ Factor\right)$,

$dA = a_{Fi}\,A_{Fr}\,dz$,

where $a_{Fi} = Area\ Density,\ A_{Fr} = Frontal\ Area$

$$Le_f = \frac{0.866^{0.667}\,\left(\frac{\omega_{sw} + 0.622}{\omega + 0.622} - 1\right)}{ln\left(\frac{\omega_{sw} + 0.622}{\omega + 0.622} - 1\right)}$$ [Bosnjakovic Equation]

### 3.2 Dehumidifier

Salt water will be passed through tubes in the dehumidifier. The hot humidified air coming from the humidifier will be passed over the tubes. Due to heat exchange between hot humidified air and salt water, air will get dehumidified. Considering a small control volume of dehumidifier of height $dz$,

Applying mass balance to the control volume,

$$m_a\left(1 + \omega + \frac{d\omega}{dz}dz\right) + m_w = m_a(1 + \omega) + m_w + m_a\left(\frac{d\omega}{dz}dz\right)$$

Applying energy balance to the control volume,

$$m_a\left(\,h_{ma} + \frac{dh_{ma}}{dz}dz\,\right) + m_w C_{pw}^{w} T_w = m_a h_{ma} + m_w C_{pw}^{w}(\,T_w + \frac{dT_w}{dz}dz\,)$$

By rearranging the above equation,

$$m_a\frac{dh_{ma}}{dz} = m_w C_{pw}^{w}\frac{dT_w}{dz} \tag{11}$$

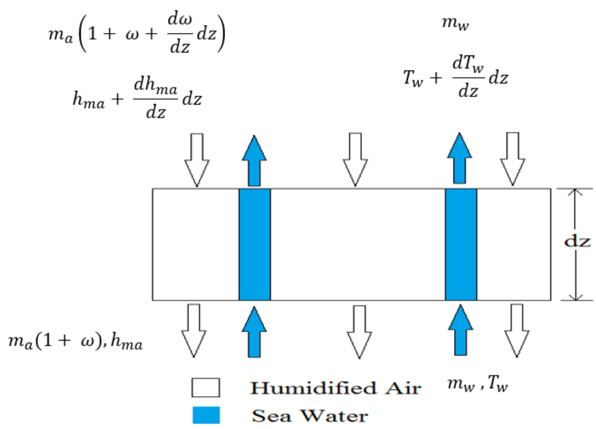



**Figure-2.** Mass flow balance along the dehumidifier.

where, $h_{ma} = C_{pa}^{a}T_a + \omega(h_{fgw0} + C_{pv}^{a}\,T_a)$

Total Enthalpy transfer,

$$m_a\frac{dh_{ma}}{dz} = U\,(\,T_a - T_w\,)\frac{dA}{dz} \tag{12}$$

Substituting (12) in (11), we get

$$\frac{dT_w}{dz} = \frac{U\,(\,T_a - T_w\,)}{m_w\,C_{pw}^{w}}\frac{dA}{dz} \tag{13}$$

Combining (8) and (12) yields,

$$\frac{d\omega}{dz} = \frac{m_w C_{pw}^{w}\frac{dT_w}{dz} - m_a(C_{pa}^{a} + \omega C_{pv}^{a})\frac{dT_a}{dz}}{m_a(h_{fgw0} + C_{pv}^{a}\,T_a)} \tag{14}$$

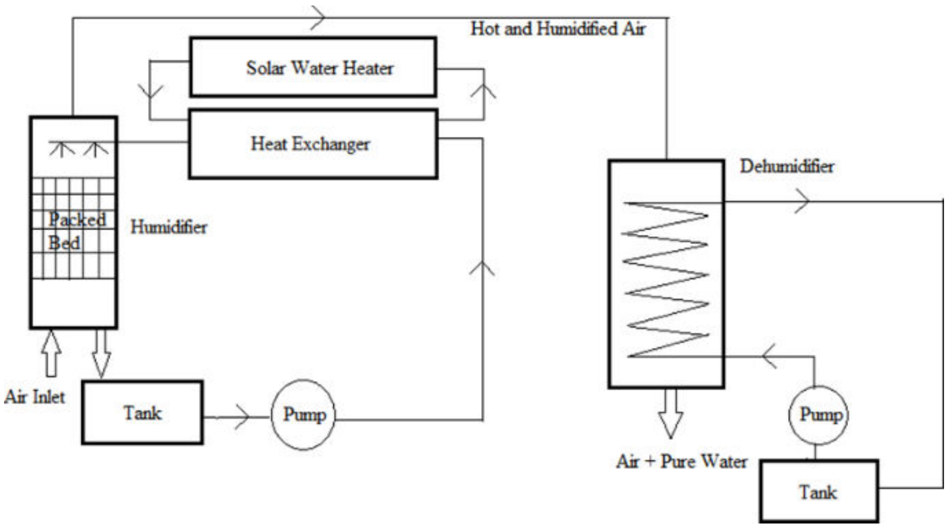
## 4. EXPERIMENTAL SET-UP

The proposed solar humidification and dehumidification unit comprises four main components

ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences



namely, a humidifier (evaporator), dehumidifier (condenser), solar water heater (evacuated tube collector), and a heat exchanger. The system incorporates an open cycle for both air and water. Forced circulation mode is used to circulate air. The impure water is stored in a tank from which it is passed through the heat exchanger where it gets heated up by the water from the solar water heater. The heated water is spread on the packing in the humidifier and the air is blown from the bottom. The air passing through the wetted packing gets heated up which increases its water holding capacity and thus gets humidified as well. The hot humidifier air is passed through the dehumidifier where the air is cooled below its dew point by water flowing through the tubes. On cooling below its dew point, the water holding capacity of air decreases and thus the air gets dehumidified and water condenses to give fresh water.




(a)


(b)

(c)

**Figure-3.** (a) Schematic diagram of experimental set-up (b) Photograph of experimental set-up (c) Packing material for a humidifier.

**5. RESULTS AND DISCUSSIONS**

The ordinary differential equations obtained for humidifier and dehumidifier have been solved for various boundary conditions using Runge-Kutta method in C++ and the results obtained are shown in terms of graphs as below followed by the experimental results.

**5.1 Humidifier Modeling**

The optimum length of the humidifier to achieve maximum humidification of the inlet air is obtained by several simulations. To find out the optimum length of the humidifier following variables have been considered and their graphs are plotted to understand the effect of different parameters.

Humidity ratio vs Length of the humidifier by keeping the air inlet temperature at 310 K and varying the water inlet temperature from 340 K to 350 K. The above graph indicates that the humidity ratio of air increases with increase in the length of the humidifier and the inlet water temperature. The humidity ratio almost becomes constant after a height of 1.75m, so there is no need to keep the humidifier height greater than 1.75m. Humidity ratio vs Length of the humidifier by keeping the water inlet temperature at 340 K and varying the air inlet temperature from 305 K to 315 K. The above graph indicates that the

ARPN Journal of Engineering and Applied Sciences



humidity ratio increases with increase in the length of the humidifier but is independent of the inlet air temperature. Humidity ratio vs $m_w/m_a$ at an air inlet temperature of 310 K and water inlet temperature of 350 K. The above graph indicates that the humidity ratio of air increases with increase in the ratio of the mass flow rate of water to that of air. Thus, a higher water flow rate and lower air flow rate are preferred for better performance. Humidification efficiency vs Length of the humidifier at an air inlet temperature of 310K and water inlet temperature of 350K.

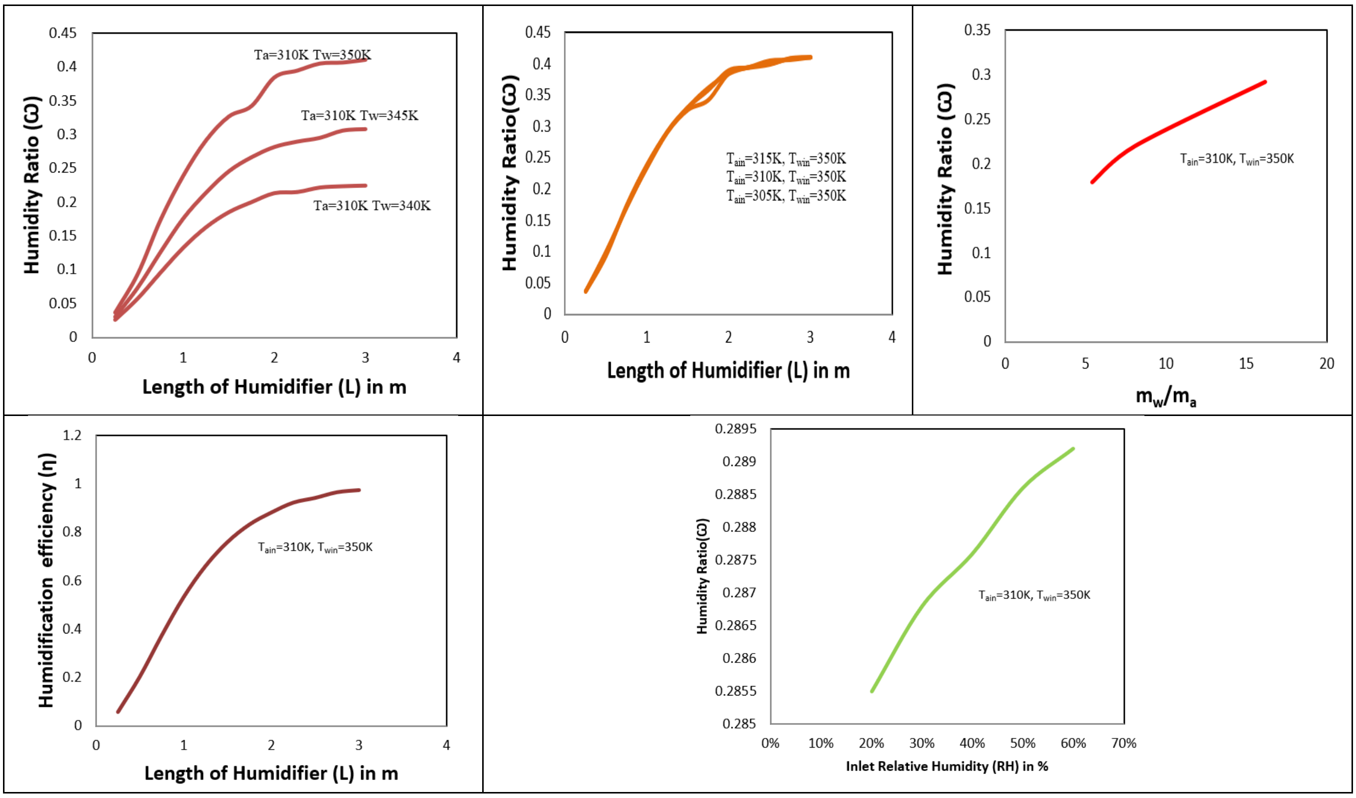



**Figure-4.** Simulation Results of variation of (a) Water temperatures (b) Length of humidifier (c) Humidity ratio (d) Humidification efficiency (e) Inlet air relative humidity.

**5.2 Dehumidifier Modeling**

Comparison of shell and coil type and shell and tube type dehumidifiers have been shown below based on water production and it appears that shell and coil type dehumidifiers give higher fresh water than shell and tube type for the same length of dehumidifier. Air Inlet Temperature of 310 K and Water Inlet Temperature of 340 K are considered as a boundary condition of humidifiers while analyzing the dehumidification modeling.

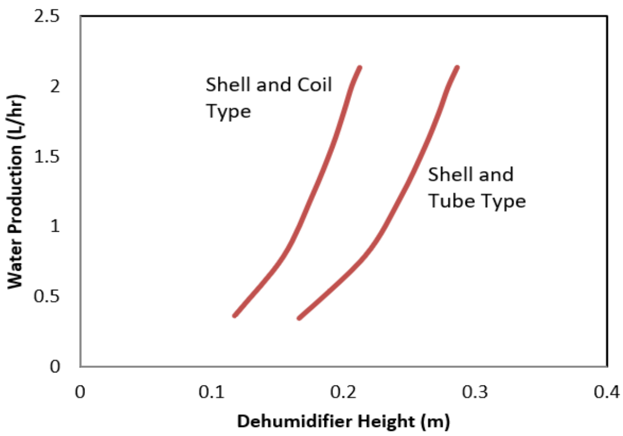



**Figure-5.** Water condensation quantity for various types of heat exchangers at different heights.

**5.3 Experimental Results**

Several tests were performed covering various combinations of operating conditions. The results are presented in this section. We have studied the influence of different parameters on the desalination unit performances. The quantities studied are feed water mass flow rate of

VOL. 19, NO. 7, APRIL 2024 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

HDH, m = 2-3 lpm, and air velocity m = 15 m/s to 27 m/s. The temperature in the heat exchanger is measured with the help of a thermocouple and Relative humidity is measured by thermo-hygrometer.

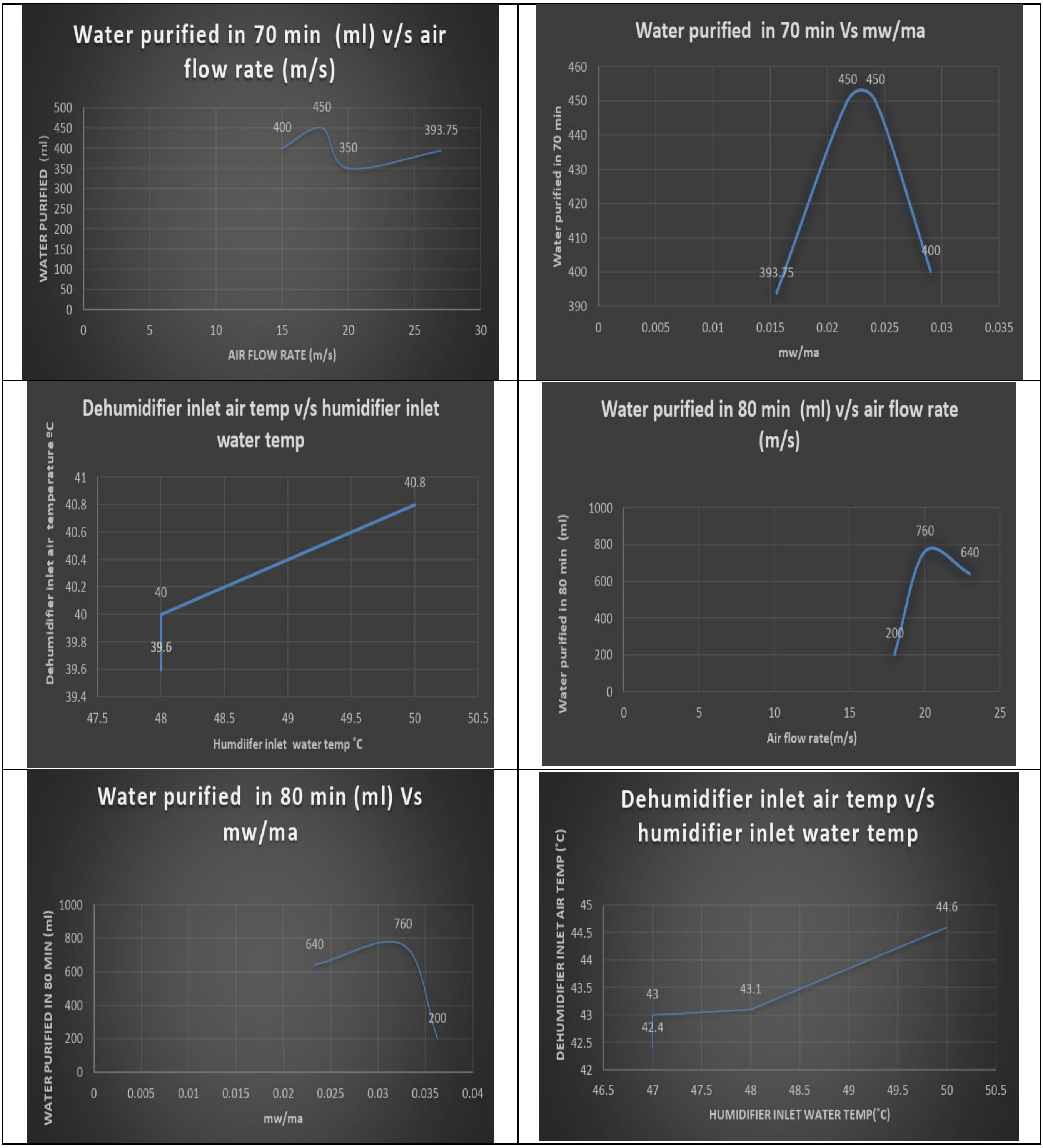



**Figure-6.** Experimental results at different variations in (a) Air flow (b) water to air mass flow ratio ($m_w/m_a$) (c) Inlet water temperature (d) (e) and (f) results of yield after 80 minutes for respective variables.

Figure-6 (a), (b), and (c) show the experimental data with a water flow rate of 2 lpm by varying the air flow rate and dehumidifier cooling temperature for 70 minutes. Figure-6 (d), (e), and (f) show the respective variations for 80 minutes of time duration. From the graph, we can conclude that as air flow increases, water purified yield also increases, reaches maximum, and further decreases. The above results could be due to vapours not getting sufficient time to condense at a higher rate. The other factors on which the yield depends are the temperature of inlet air at the dehumidifier and the dehumidifier cooling water temperature. From the graph, we can conclude that as the mw/ma ratio increases, the amount of water purified also increases to a certain point after that it decreases. So, for the optimum output, the ratio of water mass to air mass (mw/ma) should be optimum. From Figure-6 (c), we can conclude that as humidifier inlet water temperature increases, dehumidifier inlet air temperature also increases sharply. At air flow velocity 18 m/s output is much less along with the respective humidifier inlet temperature, hence, we can conclude that the more the humidifier inlet temperature more the output. Moreover, we can also see that the output of water purified is almost doubled as we increased the water flow rate from 2 lpm to 3 lpm.

Figure-6 (d), (e), and (f) show similar variations for 10 more minutes (i.e., 80 minutes) and the trends are similar. However, the yield values are large in this case due to more time being available for effective humidification and dehumidification. We can conclude that as the mw/ma ratio increases water purified amount also increases to a certain point after that it decreases.

## 6. CONCLUSIONS

A conceptual design was developed for a burning problem of water purification of sea water in the presented investigation. The results are analyzed mathematically and experimentally and found motivating on a large scale. Moreover, important performance parameters are also investigated and performance is optimized in terms of water mass to air mass flow ratio, air flow rate, and inlet water temperature. Further, the TDS, pH, and conductivity tests of the purified water were also checked to find the purity of the water collected. Test values of collected water were found as TDS-60 ppm, pH - 7.2 Conductivity - 0.39 ms/cm, which proved that water is pure for drinking purposes. Further, results concluded that at a higher flowrate of water, more pure water can be available to develop the system at a large scale to make it more economical.

## ACKNOWLEDGEMENTS

The financial and all technical support for this project was provided by the Department of Mechanical Engineering, Institute of Technology, Nirma University, Ahmadabad, India.

## REFERENCES

[1] Al-Hallaj S., Farid M. M., Tamimi A. R. 1998. Solar desalination with humidification dehumidification cycle: performance of the unit. Desalination. 120: 273-80.

[2] Ben-Bacha H., Damak T., Bouzguenda M. 2003. Experimental validation of the distillation module of a desalination station using the SMCEC principle. Renewable Energy. 28: 2335-54.

[3] Farid M. M., Parekh S., Selman J. R., Al-Hallaj S. 2002. Solar desalination with humidification-dehumidification cycle: mathematical modeling of the unit. Desalination. 151: 153-64.

[4] Nafey A. S., Fath H. E. S., El-Helaby S. O., Soliman A. M. 2004. Solar desalination using humidification-dehumidification processes. Part II. An experimental investigation. Energy Conversion Management. 45(7-8): 1263-77.

[5] Dai Y. J., Zhang H. F. Experimental investigation of a solar desalination unit with humidification and dehumidification. Desalination.

[6] Khedr M. 1993. Techno-Economic Investigation of an Air humidification-dehumidification Desalination Process. Chemical Engineering Technology. 16: 270-4.

[7] Yamali C., Solmus I. 2008. A solar desalination system using humidification-dehumidification process: experimental study and comparison with the theoretical results. Desalination. 220: 538-51.

[8] Houcine I., Amara M. B., Guizani A., Maalej M. 2006. Pilot plant testing of a new solar desalination process by a multiple-effect-humidification technique. Desalination. 196: 105-24.

[9] Klausner J. F., Mei R. 2005. Diffusion driven desalination apparatus and process. US Patent No. 6, 919, 000.

[10] Muller-Holst H. 2007. Solar thermal desalination using the multiple effect humidification (MEH) method. In: Book chapter, solar desalination for the 21st century. pp. 215-25.

[11] Nawayseh N. K., Farid M. M., Al-Hallaj S., Tamimi A. R. 1990 Solar desalination based on humidification process. Part I. Evaluating the heat and mass transfer coefficients. Energy Conversion Management. 40: 1423-39.

[12] Singhal A., Bhatt. A., Raval T. 2019. Performance enhancements of solar flat plate collector with aluminum foil reflectors and trapezoidal glass cover. ARPN journal of engineering and applied sciences. 14(17): 2948-2956.

[13] Bhatt A. A., Jain S. V., Patel R. N. 2022. Experimental investigation on performance analysis on wickless thermosiphon heat pipe with two heat

sources and multiple branches. Journal of thermal sciences and Engineering applications, 2022. doi: 10.1115/1.4054163.

[14] A. Bhatt, S. Jain, R. N. Patel, and D. Vaghela. 2023. Parametric Study on Axially Grooved Heat Pipe with Two Heat Sources and One Heat Sink with Multiple Branches. *J Therm Sci Eng Appl*, pp. 1-27, doi: 10.1115/1.4062155. 2000; 130: 169-75.

[15] A. Bhatt, S. Jain, R. N. Patel, and D. Vaghela. Experimental Investigations on Novel Orientation Study on Axially Grooved Heat Pipe with Two Evaporators and one condenser with Multiple Branches. Heat and mass transfer, Springer, DOI: 10.1007/s00231-023-03441-0