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Experimental investigation on a hybrid desalination and cooling unit using humidification-dehumidification technique

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ABSTRACT

Rapid growth in urbanization, industrialization and decrease in rainfall rates all over the world demands more fresh water for human use, industries and agriculture purpose. The novel desalination technologies for converting saline water into fresh water have more value and demand. Present work aim at evaluating the performance of an integrated air cooling and two stage desalination system using humidification dehumidification (HDH) technology experimentally by varying flow rate and temperature of saline water. The components of the proposed system are designed, developed and carried out experiments for different operating conditions to get optimum operating conditions. The plant consists of a solar water heater (SWH), two air preheaters, two humidifiers, two dehumidifiers with a normal and a vapour compression refrigeration water chiller. The influence of saline water temperature and flow rate on air temperature, desalination yield, recovery ratio, energy utilization factor (EUF), air cooling effect and coefficient of performance (COP) of the cooling system are studied. The experimental observations recommend higher saline water temperature with high flow rates to air preheaters and humidifiers, circulating and chilled water to dehumidifiers with lower possible temperatures. The integrated plant yields 240 W of cooling energy and a maximum fresh water of 2180 ml/h with 15 m^3/h air flow at saline water temperature of 54.5°C with a flow rate of 200 LPH. Experimental results show a maximum EUF of 0.5 for the hybrid plant and 0.45 for conventional plant single stage plant. A maximum COP of 2.2 with a recovery ratio of 0.78 is observed from the experimental results.

Keywords: Humidification; Dehumidification; Desalination; Cooling; Hybrid systems; Fresh water

1. Introduction

Fresh water is everything for all living beings, but its share is only 2–3% out of total water reserve on the earth. The alarming depletion of fresh water reserves across the globe is focusing more attention by scientists and technologists for novel low cost desalination technologies. Growing population, increased industrialization, urbanization is demanding more fresh water for their day to day needs, which can be met by sea water desalination. HDH desalination process is a simple technique uses air as the medium for humidification and dehumidification to get fresh water. HDH systems work by using low temperature heat for heating saline water or carrier gas and by consuming low parasitic power to drive the auxiliaries. The required heat energy can get from renewables such as solar, bio-mass, geothermal, ocean thermal energy or waste heat and are best suited for low capacity (1–100 m^3/d) desalination needs.

Many researchers developed thermodynamic simulation models to evaluate the performance of the HDH process by varying key process parameters. Narayanan et al. [1] carried out parametric optimization studies on different HDH desalination cycles to improve the performance. The HDH cycles proposed with an option of operating at different pressures, multiple extraction and vapour compression gives high gained output ratios (GOR) compared to

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conventional cycle. Sievers and Lienhard [2] developed a two-dimensional numerical model using segment by segment method to design a dehumidifier with plated fins for HDH desalination. The validated model with experimental data recommends a thicker fin dehumidifiers for an increased performance as the latent heat flow will be more in HDH desalination process compared to heating, ventilation and air conditioning dehumidifiers. Niroomand et al. [3] proposed direct contact dehumidification with cold water spray into the stream of hot humid air for an improved yield in fresh water. System performance is studied theoretically with varied air flow, hot and cold water temperatures, droplet size and diameter are studied. Study reveals the indirect contact dehumidifiers problems like component corrosion, leakage of cooling water and associated pressure drops can be avoided by opting direct contact dehumidifiers in HDH desalination. Hamed et al. [4] performed theoretical and experimental investigation on a HDH desalination unit using evacuated tube solar collector. Experimental observations gives 16 LPD with 8 hours continuous operation in a day from 9 am to 17 pm. Chiranjeevi and Srinivas [5–7] developed a thermodynamic simulation models for a conventional single stage HDH desalination, single stage HDH desalination with air cooling and hybrid air cooling and desalination system. Simulation studies recommend ground water cooling first stage dehumidifier and chilled water cooling in the second stage dehumidifier for an increased energy utilization.

Further, many researchers carried out studies by conducting experiments on pilot scale plants to evaluate the performance and sensitivity of HDH process. Abdel-Salam et al. [8] designed, developed and conducted experiments on HDH desalination system with a humidifier, heating equipment and dehumidifier. They evaluated plant performance by varying flow rates and temperatures of both air and saline water. Experimental study reveals that the size of the humidifier and dehumidifier can be reduced with a higher saline water and air temperatures, and by recycling the air coming from dehumidifier. Experimental studies by Dai and Zhang [9] on a HDH desalination system with a sheet paper packing material in humidifier, condenser and solar collector. Thermal efficiency of system is found to be 85% at saline water temperature of 65°C.Yamali and Solmus [10] studied HDH desalination process experimentally to evaluate the performance with and without preheating the air, and the influence of saline water and cooling water flow rates. Experimental observations reveals a higher fresh water yield with increased flow of hot saline water to humidifier and cold water to dehumidifier. A 15% increase in fresh water yield with preheating of air with same operating conditions. Soufari et al. [11] conducted experiments on a 10 LPH capacity plant and compared the results with the thermodynamic model. Experimental results shows productivity of the plant is 4% higher compared with the literature and are in agreeable limits with simulation model. Zhaniand Bacha [12] investigated the dynamic performance of water and air heated solar HDH desalination process in Tunisia. The experimental shows a maximum fresh water yield in the month of July and increases with solar radiation, air and saline water inlet temperatures. Antar and Sharqawy [13] carried out experiments on a single and two stage closed-air closed-water (CACW) HDH desalination system with air heating with solar evacuated tube collectors. The experimental observations shows fresh water yield of 3.5 LPD with single stage and 6 LPD with a two stage option. Li et al. [14] designed and constructed a 1000 LPD capacity HDH desalination system by using solar air heater with evacuated tube. The experimental results shows the outlet air relative humidity can increase to 97% with outlet air temperature of 42°C and saline water temperature of 27°C.

The integration energy systems will give multiple outputs with higher energy utilization compared to individual systems attracts the researchers to integrated HDH process with other systems to evaluate the performance. Many simulation models and experiments were carried out by researchers to check feasibility of possible integration of HDH plants with other systems. Design simulations with plant experimental data are carried out on the integrated HDH desalination and cooling system to evaluate the performance at different operating conditions by Chiranjeevi and Srinivas [15]. Marale et al. [16] studied the developed a simulation model for a dehumidifier using a CFD tool and validated it with experimental observations for a combined HDH desalination and cooling process. The simulation results from the model are in agreeable limits with the experimental data. Tabrizi et al. [17] evaluated the performance of a solar still integrated HDH desalination system experimentally. The overall fresh water yield increases to 141% by coupling solar still with HDH system with a thermal efficiency of 20%. An overall productivity of 5.4 kg/m^2-d is observed with a minimum flow of 40 ml/min. Giwa et al. [18] investigated an integrated HDH desalination system using thermal energy recovered from photovoltaic (PV). The study shows a daily average fresh water production of 2.28 L and 110 W of electricity per m^2 area of a PV and environmental impacts are decreased by 83.6% compared to a photovoltaic – reverse osmosis system. He et al. [19] proposed a cogeneration cycle by integrating HDH desalination technique with organic Rankine cycle (ORC) for fresh water and power. The integrated plant yields 25.46 kg/h of fresh water with an electricity production of 4.17 kW with an extended gained output ratio of 0.78. Ahmed et al. [20] conducted experiments on a single stage HDH desalination unit with corrugated aluminum sheet as packing in humidifier, finned tube dehumidifier, water and air heaters. The fresh water yield is highly influenced by air and mass flow rates and a maximum desalinated water of 15 kg/h at a cooling water inlet temperature of 17°C for an air flow of 0.15 kg/s.

Saidia et al. [21] carried out performance study on a single stage HDH desalination unit with open-air open-water (OAOW) cycle, and found an optimum air velocity of 3.34 m/s for maximum fresh water yield. Elsafi [22] evaluated the performance of a cogeneration system with concentrated photovoltaic thermal collectors coupled with HDH desalination. The study reveal an estimated annual fresh water production of 12 m^3 and 960 kWh electrical power production. Anand and Srinivas [23] investigated theoretically on a photovoltaic thermal HDH desalination system, and found a maximum integrated plant efficiency of 43.15% with fresh water production of 0.82 L/h for cold water flow rate of 30 kg/h. Shafii et al. [24] studied the performance of heat pump integrated HDH desalination system experimentally by varying air and saline water flow rates. A maximum desalination yield of 2.79 kg/h with gained output ratio of 2.08 is observed for an air flow rate of 260 m^3/h at 22°C ambient air temperature. Kabeel et al. [25] proposed solar energy based hybrid HDH desalination system with indirect evaporative air cooling. Maximum daily fresh water yield of 38.65 LPD is observed with coefficient of performance of 0.214 to 3.49 is observed for 70 m^3/h air flow. Kabeel and El-said [26] evaluated the performance of an integrated solar still coupled humidification dehumidification desalination system. The maximum fresh water production of 18.25 L/m^2-d is resulted 0.03 kg/s flow of water and air with a GOR of 2.57 and maximum overall efficiency of 39%. Kabeel and Abdlgaied [27] performed experiments on a two stage solar dryer integrated with HDH desalination system with an objective of providing fresh water and product drying. Experimental results recommends a high flow rate of air and two stage drying for an increased moisture removal rate and fresh water production, an increase of 29% overall GOR of the system is observed compared to a single stage HDH desalination system.

Gao et al. [28] integrated a vapour compression heat pump with HDH desalination system utilising heat energy from condenser for air heating and cold energy from evaporator for dehumidification. A mathematical model of the proposed integrated plant is developed and is validated with experimental results. The studies on model reveals a fresh water production of 60 kg/d with electrical power consumption of 500 W. Srithar et al. [29] proposed an HDH desalination system coupled with energy recovery option from condenser and evaporator of a vapour compression refrigeration unit. The integrated facility is tested with different twisted tape inserts and the cone type turbulators in the dehumidifier resulting in refrigeration unit COP increase up to 7.6 and the fresh water yield of 0.4 L/m^2 h.Lawal et al. [30] developed heat and mass transfer based mathematical model for closed air open water HDH system with saline water/air heating in condenser and saline water cooling in evaporator of the heat pump. The simulation model results a maximum gained output ratio of 8.88 and 7.63 for a mass flow ratio of 0.63 and 1.3 respectively with component effectiveness of 0.8. Lawal et al. [31] further carried out exergo-economic studies on these two configurations and compared the same with electric water heated HDH system. The exergy efficiency of air heated, water heated HDH system with heat pump and electric water heater HDH system are found to be 1.097%, 0.06965% and 0.05795% respectively. The desalinated water cost varied from 4.61$/$m^3$ to 5.49$/$m^3$, 6.00$/m^3 to 7.14$/$m^3$ and 4.44$/$m^3$ to 14.95$/$m^3$ respectively for the three systems for a plant life of 15–30 years.

The literature reported concludes HDH desalination is a proven technology which consumes less high grade electrical power and can operate with a low grade heat energy. It is also best suited for remote, non-grid connected localities to meet fresh water requirement as it needs little electricity for its operation. The detailed literature review reveals that only single stage HDH cycle is discussed with little emphasis on cooling integration. Therefore the objective of the present study isto evaluate the performance of proposed integrated two stage HDH desalination and air cooling experimentally.

2. Experimental setup and system description

HDH desalination process using closed water open air (CWOA) option integrated with a vapour compression refrigeration (VCR) cooler in the dehumidifier for chilled water supply. Schematic representation of the proposed integrated two stage HDH desalination and cooling system with components is presented in Fig. 1. The process of humidification dehumidification desalination takes place in two stages with extra benefit of air cooling. The main components in each stage of HDH process are an air preheater, a humidifier and a dehumidifier. Concentric tube type heat exchangers are used as air preheaters to facilitate flow of hot water through the tube side and cold air in the annulus. Humidifiers are vertical cylindrical towers equipped with packing material for direct contact heat exchange between hot water and cold air. A tube with holes is provided at the top section of the humidifier to spray the hot saline water into cold air stream passing through the humidifier packing. Shell and tube type heat exchanger design procedure followed for dehumidifiers with baffles provided on shell side for better flow of humid air for heat rejection to cold water circulated on the tube side of the dehumidifier.

As represented in Fig. 1 ambient air at state 1 is supplied by the blower into the first stage air preheater (2–3) will gets heated up with hot water circulated from solar water heater. The heated air then enters at the bottom of first stage humidifier (3–4) flows through the packing and exits at the top. Air is further heated and humidified with sprayed hot saline water in the humidifier, the packing provides an increased direct contact surface area for air and water resulting in enhanced humidification. Heated humid air then enters the first stage dehumidifier (4–5) will undergo cooling and dehumidification by cold water flowing through the tubes of first dehumidifier tubes. The process of cooling and dehumidification results in condensation of evaporated water vapour and the condensate collected was removed at regular interval of time from the bottom of dehumidifier. The cooled air exiting the first stage dehumidifier then enters the second stage (7–11) HDH plant and undergo the similar processes in air preheater, humidifier and dehumidifiers. In second stage dehumidification takes place in second and third dehumidifier. The second dehumidifier is a plate finned heat exchanger rejects latent heat to the cold atmospheric air by natural convection will reduce cooling load on VCR chiller. The air exiting at state 10 then enters the third dehumidifier is further cooled with chilled water supplied from a VCR chiller. The chilled water supply in third dehumidifier cools the air to the temperatures below atmosphere yields high desalinated water. The air exiting dehumidifier at state 13 will further used for space cooling (13–14) as its temperature is below ambient. The primary benefit from the proposed system is enhanced fresh water and the secondary benefit of the system is space cooling. Solar water heaters are provided at the roof top for hot saline water supply to the first and second stage air preheaters and humidifiers at states 18, 22, 20, 24 respectively. The cooled hot water from air preheaters at states 19 and 23, rejected brine from humidifiers at states 21 and 25 are pumped back to the solar water heaters with a centrifugal pump. The cold water to the first dehumidifier is supplied from an overhead tank, and the VCR water cooler supplies chilled water to third dehumidifier.

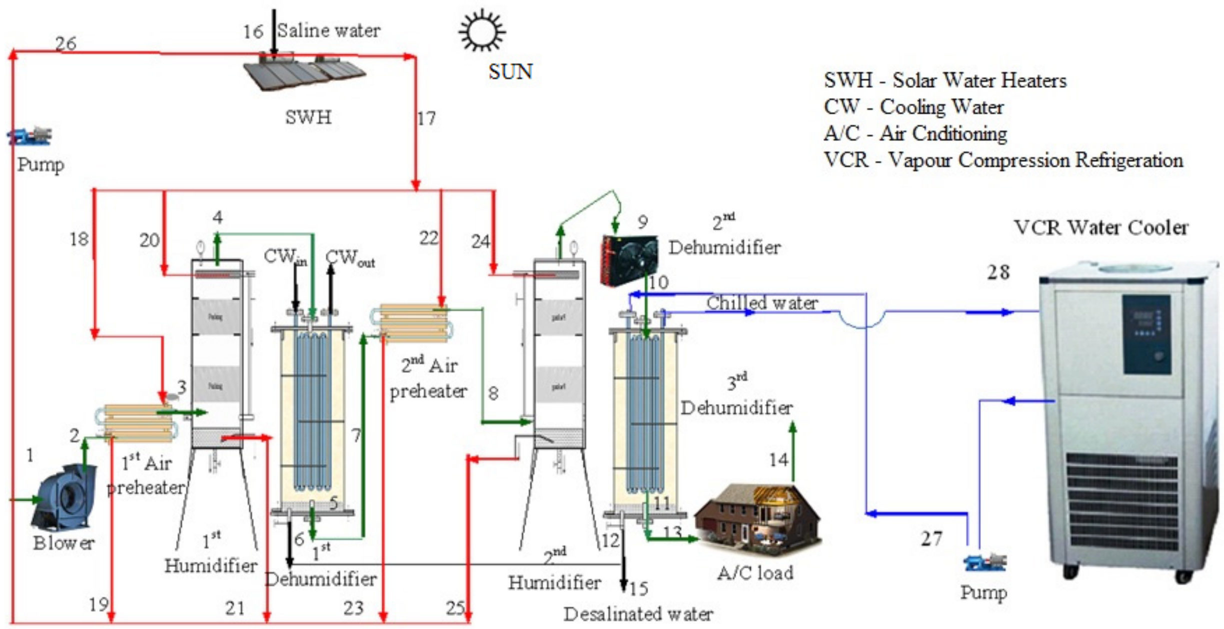

Fig. 1. Schematic representation of two stage desalination with cooling integrated system.

The picture of integrated two stage desalination with air cooling plant along with control panel indicating different components is shown in Fig. 2. The integrated plant was designed and sized based on the thermodynamic simulation [7] results. Air preheaters are fabricated with stainless steel tubes placed concentrically with a provision of water flow in the tubular section and air in annulus section of concentric heat exchanger. Humidifiers fabricated with 0.3 m vertical stainless steel pipes inserted with corrugated plastic packing with triangular slots. A header is provided at the humidifier top to spray the saline hot water on to the packing and a bend tube facilitated at humidifier bottom for brine exit. The constructional features of first and third dehumidifiers are same and uses acrylic sheet tube as the shell with stainless steel tube to circulate the cold and chilled water respectively. The tubes of dehumidifier are arranged in staggered arrangement and baffles are provided on shell side for better flow of humid air. Second humidifier is plate finned heat exchanger with an arrangement of warm humid air flows in the tube with plate fins exposed to ambient. An indirect contact solar water heaters with a 0.5 hp recirculation pump is used to supply hot saline water to air preheating in air preheaters and humidification in humidifiers. A VCR water cooler runs with a 0.5 hp capacity compressor for generating chilled water and a 0.5 hp capacity pump for chilled water circulation to third dehumidifier. An overhead tank supplies normal cooling water for first dehumidifier. All the components of the proposed plant are assembled properly with necessary instrumentation to measure air and water flow rates, temperatures and air relative humidity. Separate rotameters are fitted in respective pipe lines for regulating hot saline water flow to air preheater and humidifiers.

Experiments were conducted on the integrated plant by varying hot saline water flow and its temperature at inlet of humidifiers with constant air flow of 15 m^3/h. The hot saline water flow to humidifiers was varied between 100 LPH and 200 LPH with a constant 1150 LPH of hot water flow to air preheaters. A constant flow of 310 LPH and 125 LPH of cold water and chilled water are supplied to first and third dehumidifiers respectively. Circulation pump is switched on and the rotameters are adjusted to flow the specified quantities of hot saline water to first and second stage air preheaters and humidifiers. The valve is opened to allow the air to flow through first stage air preheater gets preheated with hot saline water. The warm air entered into humidifier will flow through the packing to the top against hot water spray. Heating and humidification of air will takes place due to direct contact of air with hot water and is enhanced with increased surface area with packing kept inside the humidifier. The warm air coming out from the humidifier then enters at the top of dehumidifier and flows over the cooling water coils will cool and dehumidify the air results in water vapour condensation completes the first stage HDH process. The cooled air then enters the second stage HDH plant and follows the same processes in air preheater, humidifier and dehumidifiers. The second stage dehumidification first takes place in a plate finned air cooled heat exchanger and then enters the chilled water cooled dehumidifier. The cooled air is then circulated to room for space cooling as the air temperature is below the atmosphere.

Once the system is allowed for flow of air and water into different components, after ensuring no water and air leakages in the HDH process circuit the plant. Calibrated thermocouples are used to measure the temperatures of hot saline water, cooling water, chilled water at entry and exit of respective components. Air relative humidity and temperatures are measured with digital relative humidity meters and thermocouples respectively provided at entry and exit points at a regular interval of time. A separated opening with a valve is provided to collect condensate in

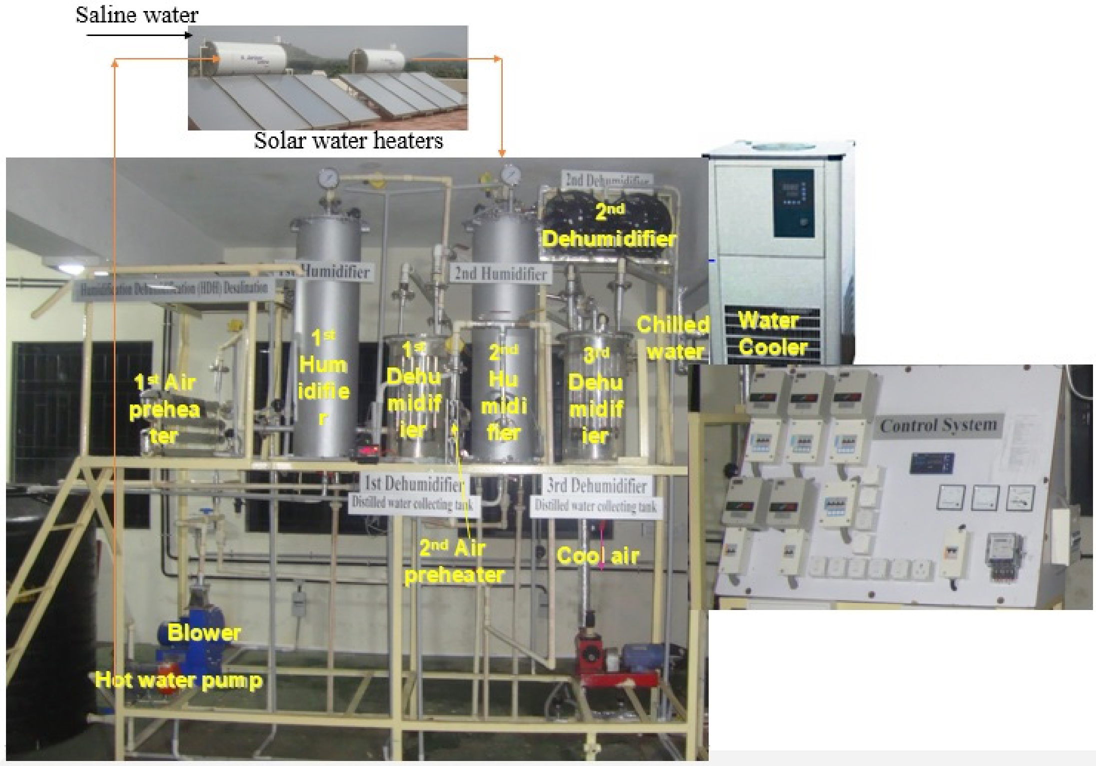

Fig. 2. Photograph of experimental set up of integrated desalination and cooling system.

the dehumidifiers periodically and is measured with a jar for assessing fresh water yield. The hot saline water flow to humidifiers is controlled with individual valves and is measured with rotameters kept in the hot water circuit line.

3. Plant performance and error analysis

The combined process performance is evaluated by varying flow rate and temperature of hot saline water to humidifier for a constant air flow of 15 m^3/h. The plant performance is assessed by evaluating the fresh water yield, cooling output and the EUF for single stage process and combined two stage process. Following formulations are used to estimate the performance parameters mentioned.

The fresh water yield from the first stage can be estimated by,

$$m_{fw,1} = m_a\left(\omega_4 - \omega_5\right) \tag{1}$$

The fresh water yield from the second stage is computed by,

$$m_{fw,2} = m_a\left(\omega_9 - \omega_{11}\right) \tag{2}$$

The fresh water yield from the combined first and second stages is computed by,

$$m_{fw,ts} = m_{fw,1} + m_{fw,2} \tag{3}$$

The air cooling effect from the plant at dehumidifier exit is computed by,

$$Q_{cooling\ effect} = m_a \times \left(h_{ambient} - h_{13}\right) \tag{4}$$

Enthalpy (h) of the moist air in the above equation is determine by,

$$h = c_p T + \omega\left(h_{fg} + c_{p,wv} T\right) \tag{5}$$

Specific humidity (ω) of moist air is found from the following equation [32].

$$\omega = 0.622 \times \left(\frac{\varphi p_{sat}}{p - \varphi p_{sat}}\right) \tag{6}$$

Saturation pressure of water vapour (p_{sat}) present in the air is determined by the following equation [32],

$$\begin{aligned} p_{sat} = {} & \left(2.7\times10^{-9}\right)\times(T^5) + \left(2.8\times10^{-7}\right)\times(T^4) + \left(2.7\times10^{-5}\right)\times(T^3) \\ & + \left(0.0014\times T^2\right) + \left(0.044\times T\right) + 0.61 \end{aligned} \tag{7}$$

The effectiveness of air preheater heat defined as the actual heat transfer (Q_{act}) by the fluid to the maximum possible heat transfer (Q_{max}) in the air preheater, and determined by the following equation [39],

$$\varepsilon_{APH} = \frac{Q_{act}}{Q_{\max}} = \frac{m_a c_{p,a}\left(T_o - T_i\right)}{\left(m_a c_p\right)_{\min}\left(\Delta T_{\max}\right)} \tag{8}$$

The effectiveness of humidifier is defined as the ratio of actual enthalpy change (ΔH_{act}) of the either of the streams to the maximum possible enthalpy change (ΔH_{max}), the equation for humidifier effectiveness is given by [30],

$$\varepsilon_{HDF} = \max\left\{\frac{h_{a,o} - h_{a,i}}{h_{a,o,\max} - h_{a,i}}, \frac{h_{w,i} - h_{w,o}}{h_{w,i} - h_{w,o,\max}}\right\} \tag{9}$$

Similarly the effectiveness of dehumidifier is determined by [30],

$$\varepsilon_{DHDF} = \max\left\{\frac{h_{a,i} - h_{a,o}}{h_{a,i} - h_{a,o,\max}}, \frac{h_{w,o} - h_{w,i}}{h_{w,o,\max} - h_{w,i}}\right\} \tag{10}$$

In the above Eqns. (9) and (10) maximum air enthalpy at outlet ($h_{a,o,max}$) is calculated when the outlet air is fully saturated at the inlet water temperature. Similarly maximum water enthalpy at outlet ($h_{a,o,max}$) is calculated when water temperature is equal to the inlet air dry-bulb temperature.

The combined desalination and cooling unit performance is estimated by evaluating single stage EUF and two stage EUF. The blower work is not considered in the present study as it utilizes the available kinetic energy from exhaust air of air expander.

The EUF of the single stage desalination unit alone is determined by,

$$EUF_{ss\ cycle} = \frac{m_{fw,1} h_{fg}}{Q_{APH,1} + Q_{HDF,1} + W_{pump,hw}} \tag{11}$$

The EUF of integrated desalination unit with air cooling is determined by,

$$EUF_{ts\ cycle} = \frac{m_{fw,ts} h_{fg} + Q_{cooling\ yield}}{Q_{APH,1} + Q_{APH,2} + Q_{HDF,1} + Q_{HDF,2} + W_{pump,hw} + W_{VCR,chiller} + W_{pump,chw}} \tag{12}$$

The coefficient of performance is a performance parameter for assessing the cooling load and is defined as the ratio of refrigeration effect (i.e. heat gained by chilled water circulating in 3rd dehumidifier) to the work input to the VCR chiller [29] and is expressed by,

$$COP = \frac{Refrigeration\ effect}{Work\ input\ to\ VCR} = \frac{\left(mc_p \Delta T\right)_{chld}}{W_{VCR}} \tag{13}$$

The recovery ratio is another performance parameter for HDH desalination systems which is defined as the ratio of rate of fresh water produced to the quantity of saline water supplied in the humidifier for the HDH system [30] which is expressed in % as,

$$RR(\%) = \frac{m_{fw}}{m_{sw}} \times 100 \tag{14}$$

The performance of integrated HDH desalination unit with air cooling effect is evaluated by measuring various parameters. The relative humidity and dry bulb temperature of atmospheric air; the air flowing through air preheaters, humidifiers and dehumidifiers at inlet and exit are measured with thermocouples and humidity sensors. The entry and exit temperatures of hot saline water to air preheaters, humidifiers, the entry and exit temperatures of cooling water and chilled water to first and third dehumidifiers are measured with thermocouples. The flow rate of hot saline water to air preheaters and humidifiers; cooling water to first dehumidifier and chilled water to dehumidifier are measured with rotameters. The desalinated water from the dehumidifiers is measured with a calibrated flask. The velocity of air leaving the third dehumidifier is measured with an anemometer.

The details of the instruments used for measuring the data in the experimental work are mentioned in Table 1. The Type B uncertainty is calculated by using Eq. (1) specified in [33].

$$u = \frac{a}{\sqrt{3}} \tag{15}$$

4. Results and discussion

The performance evaluation of hybrid HDH desalination and air cooling unit is studied by carrying out experiments. The influence of hot saline water flow and its temperature at the inlet of humidifiers are studied on system performance (fresh water yield and cooling effect). Air flow is fixed at 15 m^3/h during the experimentation. The varied conditions are saline water flow from 100 LPH to 200 LPH and saline water temperature from 40°C to 54.5°C. The temperatures of hot water and air at entry and exit of each component, flow rate of hot water was noted down. Measured data of variation in air temperature at entry and exit in air preheaters, humidifiers and dehumidifiers are plotted against the saline water temperature at humidifier inlet for different flow rates. The computed values of single and integrated unit desalination yield, air cooling, and EUF of the unit are plotted for against saline water temperature at humidifier inlet and its flow rates.

Fig. 3 depicts the change in temperature of air at inlet and exit of air preheaters for different temperatures and flow rates of hot saline water temperature. Fig. 3a shows the change in air temperature before and after the first air preheater, the air temperature increase with increase hot water temperature and its flow rate. Increase in hot water temperature increases the heat transfer rate to cold fluid and hence the air temperature rise at exit. The higher flow rates hot water increases the fluid turbulence and hence an increase in heat transfer coefficients on water will prone for high air exit temperatures. Fig. 3b shows the variation of air temperature at entry and exit of second air preheater against hot water temperature for different hot water flow rates. The reason for increase air temperature in second air preheater is same as discussed in first air preheater case. The moisture presence in the air entering the second air

Table 1
Experimental uncertainty of measured quantities

Sl. No.	Instrument	Accuracy	Range	Uncertainty
1	Thermocouple	±0.5°C	0–200°C	±0.288°C
2	Digital humidity sensor	±3.5% RH	0–100% RH	±2.02 %
3	Anemometer	±0.2 m/s	0.4–30 m/s	±0.115 m/s
4	Rotameter	±0.5 LPH	0–400 LPH	±0.288 LPH
5	Measuring jar	±10 ml	1–1000 ml	±5.77 ml

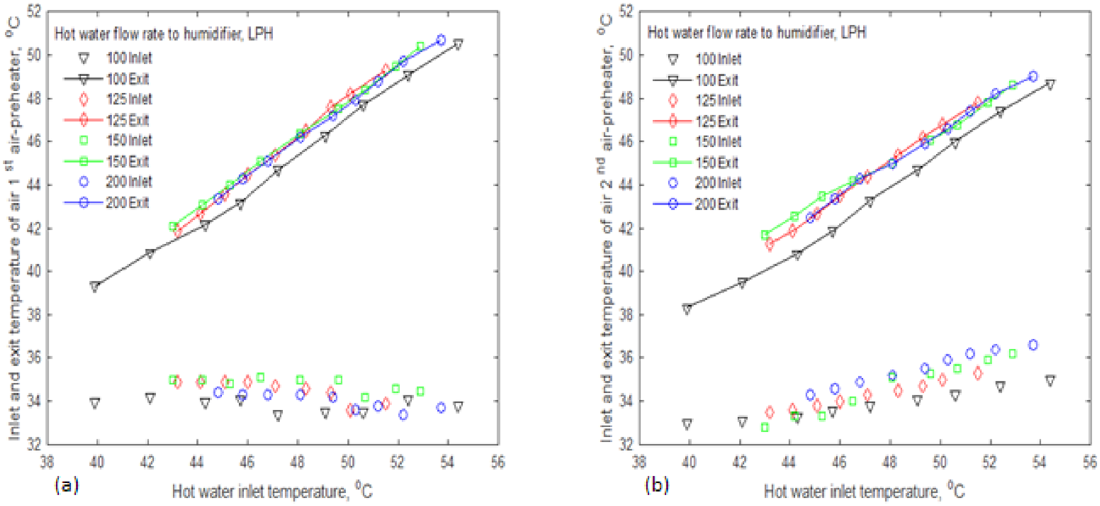

Fig. 3. Variation of air temperature at entry and exit of (a) 1stAPH and (b) 2ndAPH with hot water inlet temperature to humidifiers.

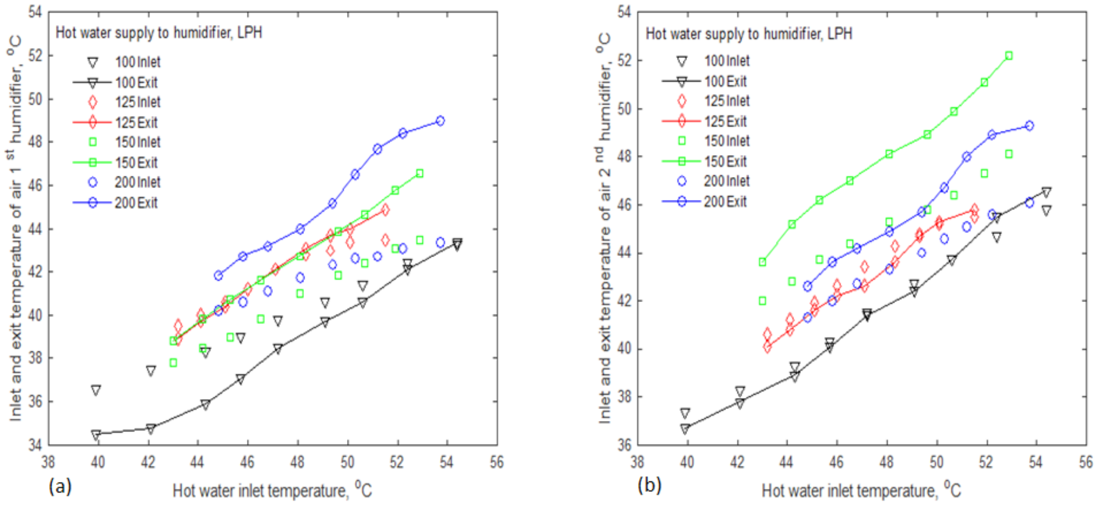

Fig. 4. Variation of air temperature at entry and exit of (a) 1st humidifier and (b) 2nd humidifier with hot water inlet temperature to humidifiers.

preheater is more, so it will absorb certain quantity of heat leads to decreased rise of air temperature in second air preheater as compared to first air preheater. The advantage of preheating the air in HDH process will decrease the relative humidity of air entering the humidifier will be an advantage for more moisture addition from hot water. The air preheaters is supplied with saline water at a maximum possible flow rate with high temperatures to achieve a high exit air temperature. The maximum air temperature observed is 51°C for 200 LPH hot water flow at 53.5°C in first air preheater and 49.5°C was observed for second air preheater for same hot water flow rate.

The influence of saline water temperature and its flow rate on rise in air temperature in humidifier is represented in Fig. 4. As shown in Fig. 4a the air entering the first air humidifier is already preheated in air preheater will cause a further rise temperature due to direct contact heat exchange. An increased air temperature rise was observed at higher flow rates of hot saline water at high temperatures compared to lower hot water flow rates. From Fig. 4b similar phenomena observed as discussed for first humidifier, less rise in air temperature for lower flow rates hot water at low temperatures was observed. In second humidifier the accumulated heat and moisture in the air leaving the first dehumidifier will cause low rise in air temperature at high hot water flow rates. To conclude in second humidifier the heat energy in hot water is more utilized for air humidification than its heating. It is observed that a maximum air temperature of 49°C in first humidifier for 200 LPH saline water flow at 53.5°C. In second humidifier a maximum air temperature of 52°C is observed for 150 LPH hot water flow at 53°C.

The influence of hot saline water flow and its temperature on the air temperatures before and after the dehumidifiers is plotted in Fig. 5. Cold water from an overhead tank is circulated in the first dehumidifier and the third dehumidifier is circulated with chilled water from VCR chiller. Fig. 5a shows the variation of air entry and exit temperatures in the first dehumidifier with respect to hot saline water temperature. Heat and moisture gained by air at lower saline water temperatures and its flow resulting in less heat and moisture rejection in dehumidifier. An increased air temperature drop at higher flows and temperatures of hot saline water is due to the presence of more heat and moisture in air. The lower limiting temperature for heat rejection by air in first dehumidifier is limited by the temperature of cooling water supplied. As shown in Fig. 5b similar trends of air at inlet and exit are observed in third humidifier circulated with chilled water as the cooling medium. The air temperature drop in third dehumidifier is more compared first dehumidifier because it is cooled by chilled water in the third dehumidifier to temperatures lower than atmosphere. Higher hot water flow rates at increased temperatures to humidifiers will demand more chilled water in third dehumidifier for increased air cooling and dehumidification. A maximum air temperature drop of 11°C with an exit air temperature of 36.5°C is observed from first dehumidifier for hot water flow of 200 LPH at 53.5°C. In third dehumidifier a maximum air temperature drop of 22.5°C is observed for 150 LPH flow of hot water at 53°C. A minimum air temperature of 17.5°C is observed from third dehumidifier for 100 LPH hot water flow at 40°C.

Fig.6 shows the variation of air preheaters effectiveness for different flow rates and temperatures of hot saline water to humidifiers. Fig. 6a represents the 1^{st} air preheater effectiveness and it ranges from 0.88 to 0.96 for all flow rates of saline water and the variation is very small for all the flow rates of hot water. The 1^{st} APH effectiveness is not varied much because the temperature and relative humidity of inlet air is almost same. Also the flow rate of hot water also not varied to 1^{st} APH, but it is varied for humidifiers. Fig. 6b represents effectiveness variation with saline water temperature and its flow rates to humidifiers. The dehumidified air from 1^{st} stage HDH enters the 2^{nd}APH will have more relative humidity and hence the amount heat gain will be less compared to a low humid air entering the 1^{st} APH. As the temperature of the air leaving the 1^{st} dehumidifier is lower for low hot water flows and hence the temperature rise of air in 2^{nd} APH is less at lower flows compared to higher flow rates of hot saline water.

Fig. 7 represents the variation of humidifiers effectiveness with humidifier hot saline water temperature and flow rates different flow rates hot saline water. Fig. 7a shows the 1^{st} humidifier effectiveness and it ranges from 0.5 to 0.8 for all flow rates. It is observed that the heating and humidification of air in 1^{st} humidifier will be less at lower hot water flows compared to higher hot water flow rates and hence the effectiveness of the humidifier will be more for higher flow rates and lower at low flow rates of hot water. Fig.7b shows the 2^{nd} humidifier effectiveness ranging from 0.65 to 0.85. Higher effectiveness in 2^{nd} stage humidifier is due to the accumulated heat carried by air from first stage is further heated in second stage results in relatively higher temperatures and humidity of the air at 2^{nd} humidifier exit. Also higher hot water flow rates with higher temperatures will result in high heating and humidification of air in humidifier and hence an increased 2^{nd} humidifier effectiveness.

Fig. 8 represents the variation of dehumidifiers effectiveness with humidifier hot saline water temperature and flow rates different flow rates. Fig. 8a shows the 1^{st} humidifier effectiveness and it varies from 0.75 to 0.84 for all flow rates of hot saline water. The 3rd dehumidifier effectiveness ranging from 0.85 to 0.94 and it is more than the first stage as the temperature drop of the humid air will be more due to chilled water circulation in 3^{rd} dehumidifier. Also in 3^{rd} dehumidifier it is observed that the effectiveness is not much varied with respect to hot water flow rate, it is due to the chilled water helps in attaining a huge temperature drop of humid air compared to normal circulating water.

The effect of hot saline water temperature and its flow rate on the desalinated water yield in ml/h is represented in Fig. 9. The variation of fresh water yield from the unit for different temperatures and flows of saline water at humidifier inlet for the first stage HDH plant is shown in Fig. 9a. It is observed that at lower flowrates the desalinated water yield is less and it increase with increase in humidifier inlet

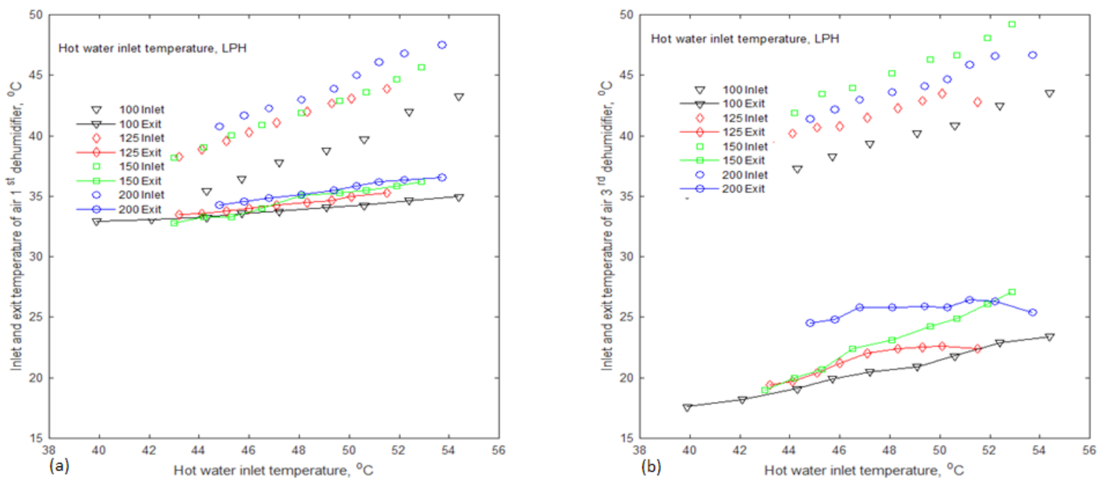

Fig. 5. Variation of air temperature at entry and exit of (a) 1^{st} dehumidifier and (b) 3^{rd}dehumidifierwith hot water inlet temperature to humidifiers.

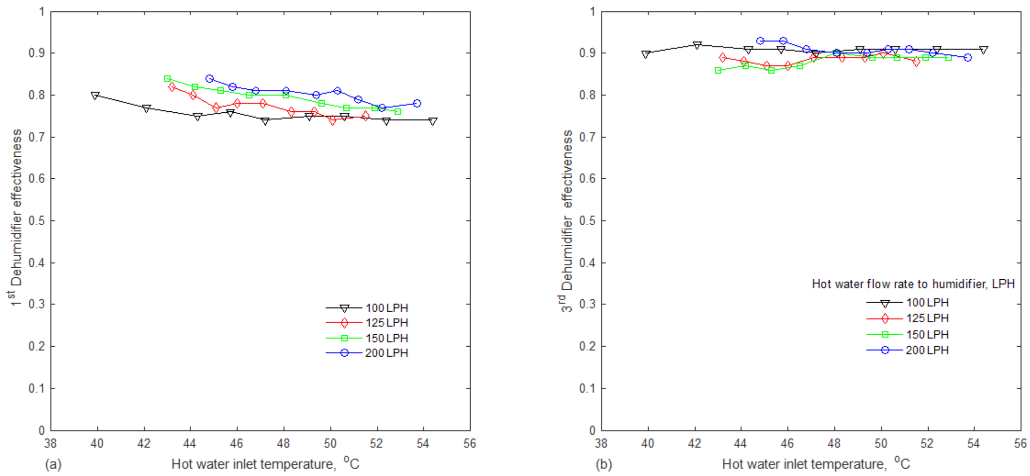

Fig. 6. Influence of humidifier hot water temperature on air preheaters effectiveness (a) 1st air preheater and (b) 2nd air preheater.

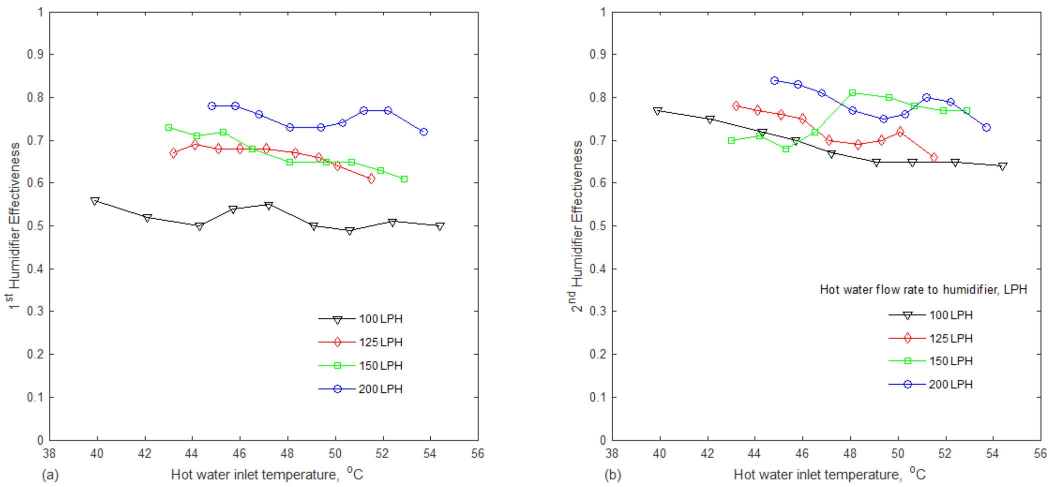

Fig. 7. Influence of humidifier hot water temperature on humidifiers effectiveness (a) 1st humidifier and (b) 2nd humidifier.

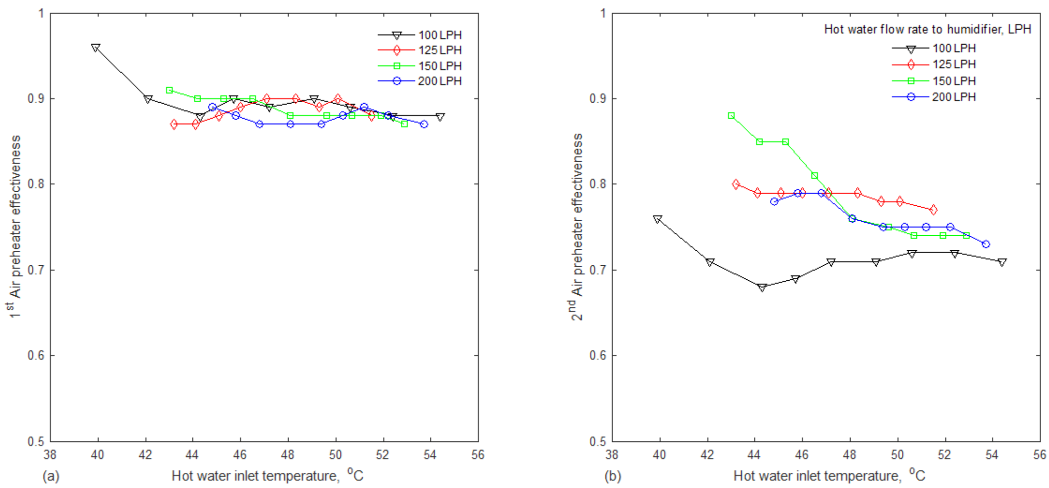

Fig. 8. Influence of humidifier hot water temperature on dehumidifiers effectiveness (a) 1st dehumidifier and (b) 2nd dehumidifier.

water temperature. The higher hot water temperature will increase the water vapour absorption capacity of air in first humidifier which will be dehumidified in first dehumidifier will result in proportionate increase in desalinated water. The increase of hot water flow rate to humidifier also enhances the humidification rated which will result in an increased fresh water yield after dehumidification in first dehumidifier. It is observed that a maximum of 1150 ml/h of desalinated water resulted for 200 LPH flow of hot water supply at 53.5°C. Fig. 9b represents the fresh water yield variation with inlet hot water temperature to humidifier in second stage HDH desalination process. As discussed for the first stage HDH process, a higher hot water flow rate and its temperatures favors an improved air humidification resulting in higher desalinated water during second stage dehumidification. The fresh water yield in second stage is more compared to first stage is due to the accumulated water vapour from first stage is rejected along with second stage water vapour to chilled water having below atmospheric temperature. Maximum desolation water resulted from second stage is 1000 ml/h with 200 LPH hot water supply at 53.5°C. An average of 650 ml/h of fresh water is resulted from the second stage for flow rates ranging from 100 to 200 LPH with a temperature variation of 40–54.5°C. Fig. 9c represents the combined first and second stage desalination yield from the plant. The hybrid desalination and air cooling unit will yield a fresh water for 2150 ml/h for saline water supplied at 53.5°C with a flow of 200 LPH. The combined plant recommends higher hot water flows with relatively high temperatures for enhanced desalination. A maximum of 2180 ml/h of desalinated water is resulted from combined two stage plant. It is observed that the chilled water cooling results more fresh water compared to normal cold water dehumidification cooling.

Fig. 10 represents influence of hot saline water inlet temperature and its flow rate on the performance of integrated two stage HDH desalination and cooling plant. Fig. 10a shows the air cooling rate from the third dehumidifier, which represents amount of cooling energy by air below the ambient temperature. At lower temperatures of hot water for a given flow rate the heat rejected by humid air will be less due to less moisture content will give more cooling effect. The moisture removal rate will increase with increase in hot water temperature demands more air cooling will result in less cooling effect. An increase in hot water flow rate to humidifier increases the humidification rate in humidifier which will demand more cooling in dehumidifier resulting in high desalination yield and decreased cooling effect. Approximately a cooling output of 100–240 W is resulted from the plant at third dehumidifier exit. Fig. 10b shows the variation of energy utilization factor of the single stage HDH desalination process with hot water temperature at inlet of humidifier. The EUF is directly proportional to the useful energy (desalinated water and air cooling) gained from the cycle. The fresh water yield from the single stage plant is less for low hot water flow rates results in a decreased EUF. At higher hot water flow rates the desalination yield increases and hence an increase in cycle EUF, it'll further increases with increased hot water temperatures. Fig. 10c shows the energy utilization of the combined two stage HDH desalination and cooling process. The total desalinated water yield is increased in the combined plant and the air cooling benefit will enhance the energy utilization. Though the plant uses an extra parasitic power for running VCR cooler and chilled water pump the overall energy gained from the plant will be more and hence an increase in cycle EUF. In single stage the cycle EUF is varied from 0.05 to 0.45 and in integrated plant 0.25 to 0.5 for different flow rates and inlet temperature of hot water at humidifier inlet.

Fig.11 shows the influence of hot water temperature and its flow rate on COP of the vapour compression refrigeration system and the recovery ratio (RR) of the 1st and 2nd

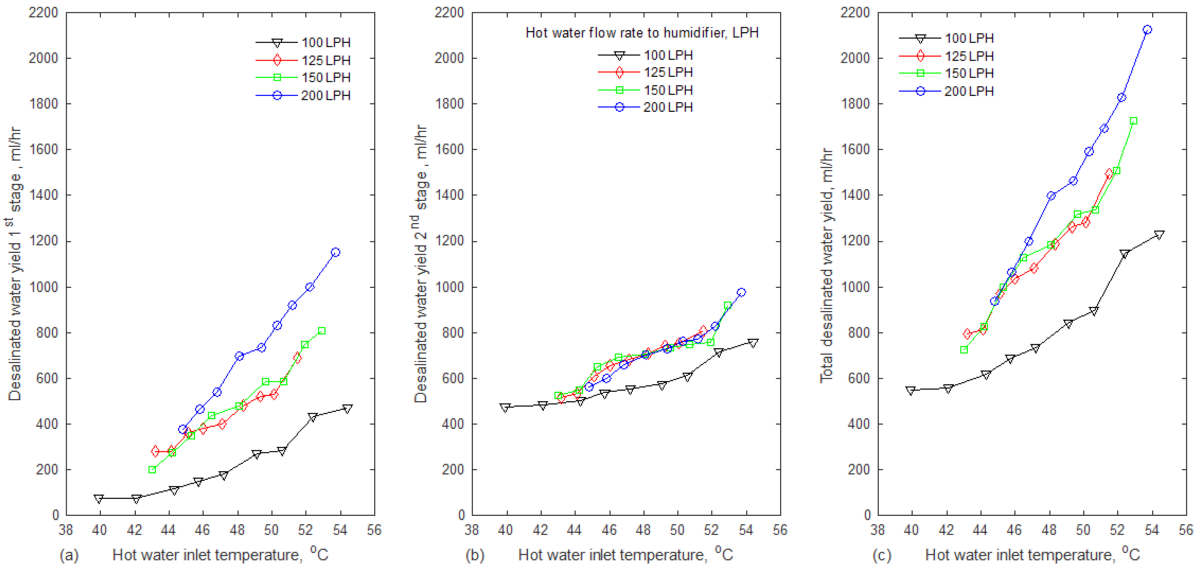

Fig. 9. Variation of fresh water yield in (a) first stage, (b) second stage and (c) combined two stage unit for different hot water inlet temperature to humidifiers.

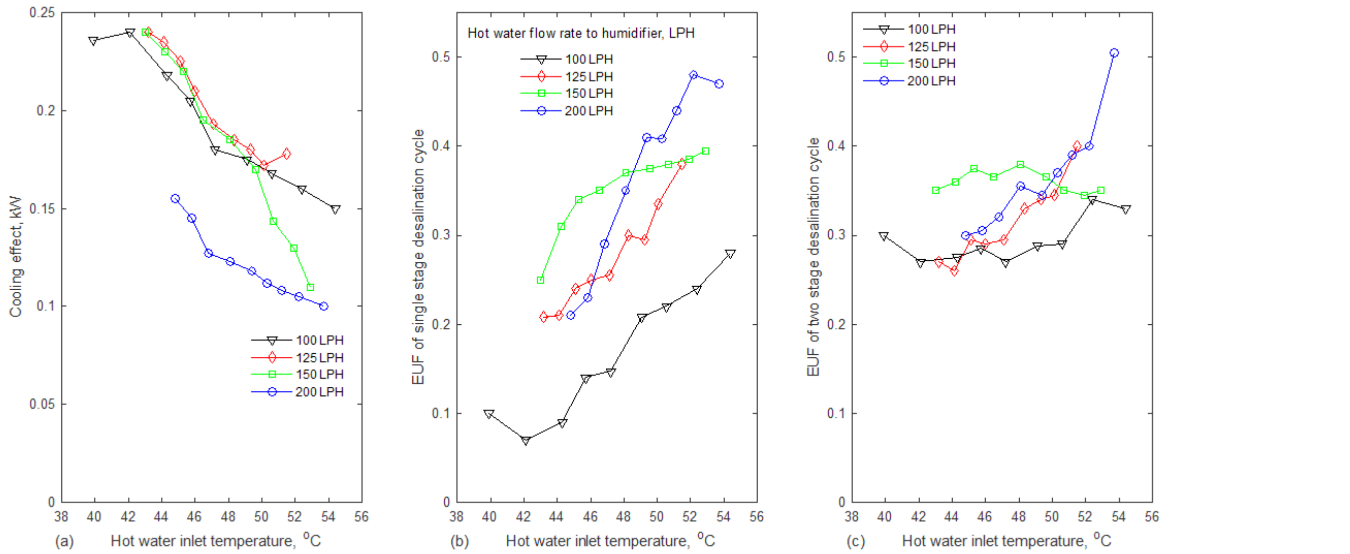

Fig. 10. Influence of inlet hot saline water to humidifiers on (a) cooling effect, (b) single stage cycle EUF and (c) two stage integrated cycle EUF.

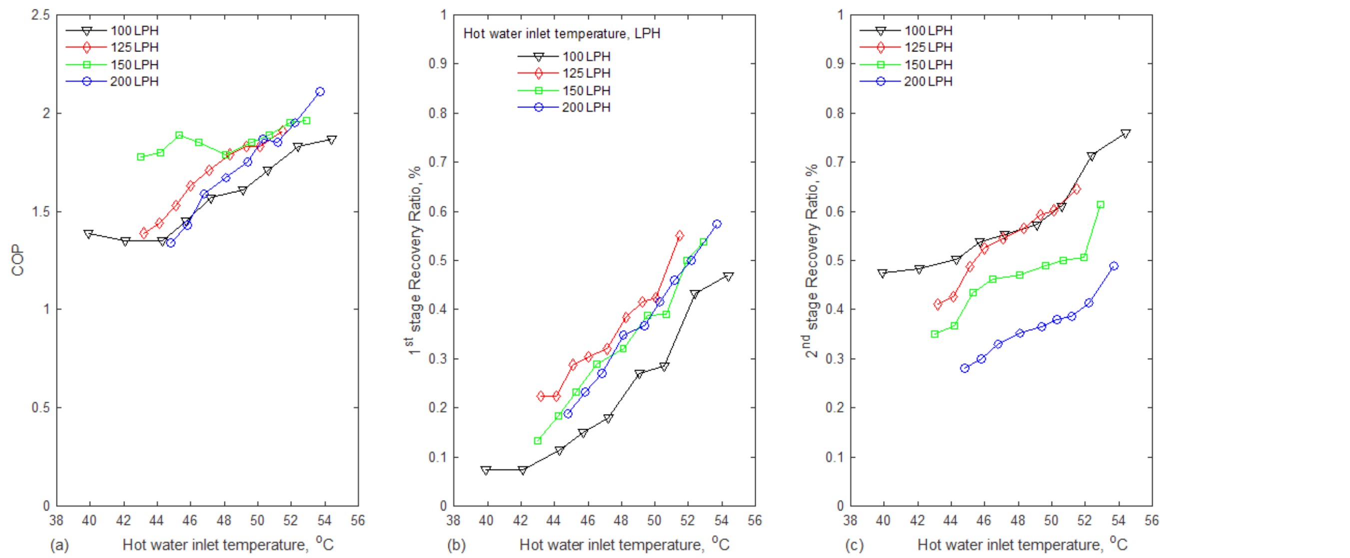

Fig. 11. Influence of humidifier hot water temperature on (a) COP, (b) recovery ratio in 1st stage , and (c) recovery ratio in 2nd stage.

stage of proposed HDH system. Fig. 11a represents variation of COP with respect to hot water inlet temperature, it varies from 1.4 to 2.2. At higher hot water flow rates and temperatures air will absorb more water vapour which will be condensed in 3rd dehumidifier by rejecting heat to chilled water will result in increased COP of the system. Fig. 11b shows the recovery ratio in 1st stage HDH desalination plant, and it reveals recovery ratio increases with increase in hot water inlet temperature and flow rate and it ranges from 0.08 to 0.58. High flow rates of hot water and temperature will increases the air humidification and the increased rate of water vapour rejection in 1st dehumidifier and hence an increase in recovery ratio. Fig. 11c represents the variation of recovery ratio of 2nd stage HDH desalination plant, it increases with increase in hot water temperature due to the same phenomena explained for 1st stage desalination and ranging from 0.29 to 0.78. But, decreased recovery ratio observes with an increase in flow rate, due to the decreased temperature drop of the humid air in 3rd dehumidifier.

Table 2 compares the experimental results from the proposed integrated plant with previous works reported in the literature. Desalinated water yield in the present work is shows a significant improvement in the performance of the plant with chilled water cooling in third dehumidifier. The proposed unit has a higher desalination yield as compared to the literature reported with an extra benefit of air cooling which can be used for room air conditioning.

Table 2
Comparison of present work with previous works reported in literature

Sl. No.	Author(s)	Hot water temperature (°C)	Air flow rate (kg/h)	Maximum fresh water production (kg/h)
1	Nafey et al. [34]	42.13	17.64	0.61
2	Amer et al. [35]	60.00	2160.00	3.00
3	Hermosillo et al. [36]	68.90–44.60	–	1.45
4	Zamen et al. [37]	60.00	–	30.00
5	Hamed et al. [4]	57.00	60.00	2.28
6	Sharshir et al. [38]	84.00–87.00	64.80	2.76
7	Present work	54.50	15.00	2.18

5. Conclusions

Experimental were conducted on a hybrid two stage HDH desalination with cooling plant. The effect of saline water flow and temperature are studied on the integrated plant performance for fixed air, cooling water and chilled water flow rates. The experimental investigation reveals following conclusions. A maximum of 145 ml of fresh water is obtained for 1 m^3 of air (2.18 LPH for 15 m^3/h air flow) from the integrated unit. Approximately 100–240 W of air cooling energy is available at the exit of the integrated unit for an air flor of 15 m^3/h. It was observed that in first stage low fresh water yield at lower saline water temperature, where as in second stage fresh water yield is considerably increased due to the air cooling by chilled water in third dehumidifier. The EUF for a conventional single stage HDH desalination unit varied between 0.05 and 0.45, whereas in the proposed integrated unit EUF varied between 0.25 and 0.50. A maximum COP of 2.2 and highest fresh water recovery rate of 0.78 is observed in 2nd stage desalination with chilled water cooling. Experimental observations recommends saline water flow of 150–200 LPH with possible higher temperatures for an increased fresh water yield. Increased cooling water and chilled water flow rates can further improve the fresh water yield from the hybrid plant.

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Symbols

h — Specific enthalpy, kJ/kg
m — Mass flow rate, kg/s
p — Pressure
RR — Recovery ratio, %
ϕ — Relative humidity, %
ω — Specific humidity, kg of water vapour/kg dry air
ε — Effectiveness

Subscripts

a — Air
APH — Air preheater
chw — Chilled water
fw — Ffresh water
HDF — Humidifier
hw — Hot water
i — In
o — Out
sat — Saturated
ss — Single stage
ts — Two stage
w — Water
wv — Water vapour

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