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Parametric Sensitivity of Water Collection on Desiccant Coated Heat Exchanger

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Abstract

Atmospheric water harvesting (AWH) is used to address the current scarcity of fresh water. Sorption-based AWH technology using desiccant coated heat exchanger (DCHE) could enable a good water supply in arid regions since sorbents can grab moisture from arid environment and then release water once it is thermally generated. Recent developments focus on novel adsorbents with high sorption capacity at low humidity and low regeneration temperature. However, a thermodynamic analysis on overall system to predict the best operating parameters and how it affects system efficiency has not been studied. In this study, we investigated the energy and heat transfer properties of desiccant-coated heat exchanger. A one-dimensional mathematical model is developed to predict the performance of a DCHE system which uses super-porous hydrogel (SPH) made of N-isopropylacrylamide (NIPAM) as a desiccant material. The effects of inlet temperature, inlet humidity ratio, air mass flow rate, and cold side temperature brought by using thermo-electric cooling (TEC) on the amount of water collected, COP, and second law efficiency of the system were analyzed. The results suggest that the rate of water production and COP of the system increased as the temperature and humidity of the air increased. It emerged that increasing airflow increases water intake and COP due to the improved heat transfer through the system. However, rising cold side temperature of TEC decreases water production rate and COP while it improves second law efficiency because of reduced heat transfer and entropy generation. The obtained results show that water harvesting using the proposed system with thermoelectric cooling device and solar energy is feasible.

Keywords: *Cooling Load; Air-Conditioning; Desiccant coated heat exchanger; Atmospheric water collection*

1. Introduction

Water scarcity is a global risk, and it is expected to increase with time to the level that people have no access to sufficient quantities of safe and clean water [1], [2]. It is reported that there are about 783 million people facing severe water scarcity all year around [3]. It is predicted that the number of people living in regions with high water scarcity will increase to 3.2 billion by 2050 [2]. The increasing water scarcity is due to increasing demand by population growth and climate change, i.e. higher temperature and draughts [3]. Also, uneven distribution of water resources and population densities is contributing to increasing water scarcity [2], [4]. Water can be provided to these water-scarce areas by transportation of water from other locations where water is dispensable, desalination of saline water (ground and underground), and collection of water from atmospheric air. Water transportation is very expensive, and the availability of ground and underground water is rare in arid areas to be desalinated. There is however huge source of clean and good quality of water within the atmospheric air [1], [5]. This amount in natural global water cycle, is evaluated, at any given time, to be 12,900 km^3 [3]. This accounts only 0.001 % of the total water on earth and 0.04% from the available freshwater [3].

Surely, there are different methods of extraction water from the atmospheric air including cooling and desiccant technologies as the most popular ones, as shown in Figure 1. There are many factors affecting the choice of the best suitable method, including local climatic conditions, economic aspects and costs (i.e., capital, operating, and energy), and size and scale of being small unit for personal usage to larger urban deployment [1], [5].

Despite technology development for fog collection, there are certain challenges in fog harvesting manifested chiefly in the low efficiency. Collection is done by optimal entrapment of water droplets and microscopic mist from the flowing fogy air over either fine thickness wire mesh while avoid clogging, or

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over thicker wires that sacrifices the capturing of the microscopic mist.

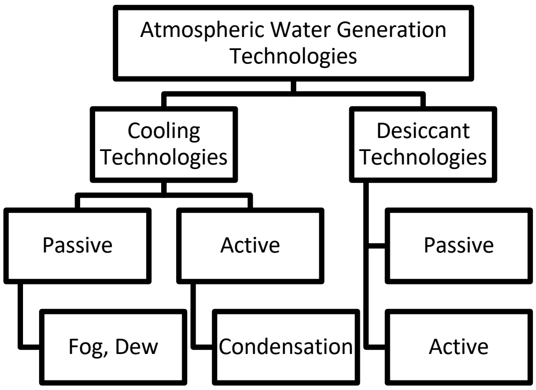

Fig. 1: Atmospheric water collection technologies

Dew collection using radiative collectors is limited by the rate of radiative heat exchange, weather conditions, and collector surface properties [1]. Using different condenser configurations and shapes would improve water collection capacity at the cost of adding more complexity to the common planer condensers. Also, scaling up of the water collector is not linear as it may suffer 40% decrease in efficiency, which does not allow for the collection of high volumes of dew water [6], [7].

Active water collectors, also known as atmospheric water generators, are based on either of the two processes: i) Using refrigeration circuit where the condenser surface is cooled below the dew temperature to initiate condensation; ii) Using regenerative desiccant that absorb water vapor. Water collection using this latter approach is economically viable due to lower initial, operating, and maintenance costs. However, it is deemed inefficient during periods of low air flow [6].

The desiccant system can be used in hybrid formations with vapor compression cycle (VCC) or as standalone system. The atmospheric air-cooling process in VCC takes into account both sensible and latent heat due to the presence of water vapor. This increases the total cooling load in proportion to the mass fraction of water vapor, thereby it requires higher cooling energy. Nevertheless, in a hybrid system the addition of a desiccant reduces the cooling load because the desiccant material absorbs moisture from the atmospheric air and consequently reduces latent heat fraction. Salmi et al. [8] indicated 60% increase in VCC performance when associated with a desiccant system. Another advantage of the hybrid system is that it avoids condensation of water at high relative humidity [9]. This reduces the power consumption and the VCC operating costs by adding a desiccant. Whether for refrigeration or other purposes, the vital role of water vapor absorption systems is that it has prominent role in dehumidification. This can be accomplished by two types of desiccants, i.e., the liquid and solid desiccants. Liquid desiccants are not chemically stable and are liable to conditions that lead to foaming, salt precipitation, and acidification [10]. They also tend to crystallize at low temperatures and corrode metals. On the other hand, the solid desiccants are more chemically stable, inexpensive, durable, and environmentally friendly [10]. Solid desiccants are currently used in three classes of system: Fixed-bed, rotary-bed, and desiccant-coated heat exchangers (DCHE) systems [9].

The DCHEs have three main usages, i.e., dehumidification, air conditioning, and adsorption chillers [8]. Until now, this type of desiccant system has not been commercially used in atmospheric water collection. There are many studies on the performance of DCHE for other applications like dehumidification and cooling using conventional desiccants such as zeolite and silica gel [11-16]. The DCHE system is superior to fixed and rotating beds because of its higher dehumidification performance [9]. Operationally, DCHE dissipates heat of adsorption and reduces the cooling load. As vapor is adsorbed heat is generated and temperature of the desiccant raises which reduces its dehumidifying capacity. By binding the desiccant to the heat exchanger, the heat exchanger dissipates the generated heat of adsorption and decreases the temperature of the desiccant. The DCHE metal parts provides high heat transfer efficiency and enough heat for the desiccant to regenerate. The fins of the heat exchanger provide a large surface area where the desiccant can be coated to improve the overall adsorption performance. Cengel and Boles [9] and Rouquerol et al. [17] indicated that at least two DCHEs are required for continuous dehumidification. This is mainly due to the periodic mode of operation. The cycle begins in a cooling/adsorption mode to dehumidify the air, and then in a heating/regeneration mode to dispose of the adsorbed vapor [18]. Furthermore, DCHE can be easily implemented using low-grade energy sources and renewable energy that renders the environmental friendliness of the system [19], [20].

Recent studies have focused on enhancing the sorption capacity of new sorbents and building small-scale prototypes of DCHE, but thermodynamic analysis of factors affecting water collection have not been investigated. Therefore, in this study, we develop a mathematical model for the DCHE based on the 1st and 2nd laws of thermodynamics to identify the optimal operating conditions for maximum water collection. We also investigate the effect of various parameters on system performance, including inlet temperature, inlet relative humidity, and air flow rate as well as the cold side temperature brought by using thermo-electric cooling (TEC).

2. Theoretical Formulation

The DCHE system consists of several components as shown in Fig. 2. It includes two heat-exchanger, i.e., one is coated with desiccant material and other one is kept plain (uncoated). The moist air is circulated in the side of the coated heat exchanger (HE) where the desiccant material adsorbs water vapor and causes the release of latent heat. The DCHE temperature is controlled by the TEC device, which supplies a constant and steady cooling load. The plain HE is used to dissipate the waste heat brought in for cooling of the DCHE. For desorption and heating the DCHE, the plain HE cooling is managed by the TEC device in order to increase the amount of heat on the hot side. A flow of air is also tunneled to the HE to reduce the amount of condensation. The needed power to operate the entire system's components and their processes is provided by a photovoltaic (PV) solar panel module.

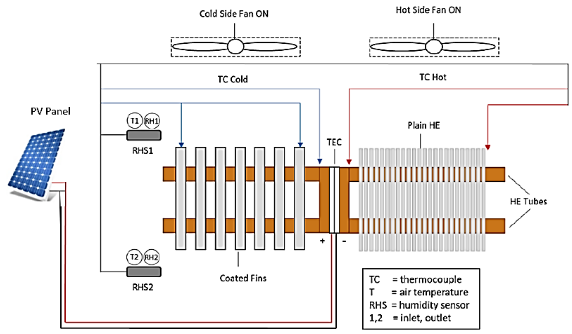

Fig. 2: Schematic of components of the designed system

To simplify the mathematical model and its thermodynamic analysis, the following assumptions are made:

(1) Energy transfer is a steady state process through the entire system.
(2) Heat transfer is constant through the system.
(3) Desiccant material is uniformly distributed over the heat exchanger surface.
(4) Heat transfer resistance of the fin is negligible, and its temperature is assumed to be equal to the cold side of TEC.
(5) Heat conduction in the air stream is negligible.
(6) The heat and mass transfer coefficients between the air stream and the desiccant is constant over the entire DCHE.
(7) Adsorbed heat is released to the desiccant layer only; however, it is transferred to the air by convective heat mode.
(8) Adsorbed heat is constant through the entire process.
(9) The inlet conditions of air are uniform in space but may vary with time.
(10) Thermodynamic properties of the air are assumed to follow ideal gas fluid.

2.1 Minimum thermal energy for atmospheric water harvesting system

To determine the minimum heat input for atmospheric water harvesting system, the total process of adsorption, regeneration, and water collecting considered as a black box is shown in Fig. 3 [21]. This system was analyzed using 1st and 2nd laws of thermodynamics. The minimum heat input can be calculated from the energy balance according to Eq. 1 which is written as:

$$\dot{Q}_{H,min} = [\dot{m}_a(e_o - e_i) + \dot{m}_w e_w](1 - \tfrac{T_o}{T_H})^{-1} \quad (1)$$

Where $\dot{m}_a$ and $\dot{m}_w$ is the mass flow rate of the inlet air and collected water, T_H and T_o are the heat source temperature and environmental temperature. Also, e_i, e_o, e_w are the specific exergy of air at inlet, outlet, and water collected states, respectively. These can be found from Eq. 2 and 3 as:

$$e_{ha} = (C_{p,a} + \omega C_{p,v})T_o\left[\frac{T}{T_o} - 1 - \ln\frac{T}{T_o}\right] + (1 + \tilde{\omega})R_a T_o \ln\left(\frac{P}{P_o}\right) + R_a T_o\left[\ln\left(\frac{1+\tilde{\omega}_o}{1+\tilde{\omega}}\right) + \tilde{\omega}\ \ln\left(\frac{\tilde{\omega}\,(1+\tilde{\omega}_o)}{\tilde{\omega}_o(1+\tilde{\omega})}\right)\right] \quad (2)$$

$$e_w = h_f(T) - h_g(T_o) - T_o(s_f(T) - s_g(T_o)) \quad (3)$$

Where $C_{p,a}$ and $C_{p,v}$ are dry air and vapor specific heat capacities, respectively, $\tilde{\omega}$ is a vapor mole fraction ratio defined as $\tilde{\omega}$= 1.608ω, R_a is the gas constant of air (0.287 kJ/kg-K), P is the pressure, and h_o and s_o are the vapor state enthalpy and entropy evaluated at the complete dead state [21].

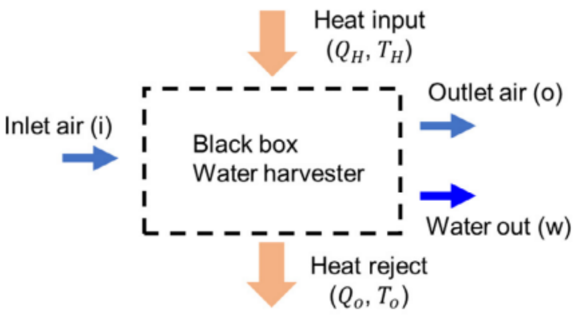

Fig. 3: Black box of water harvesting system [21]

2.2 Thermodynamic modeling of atmospheric water

To analyze the system, the energy equation for the coated heat exchanger during adsorption is invoked. The flow passages of the humid air as it passes through the heat exchanger is schematically illustrated in Fig. 4.

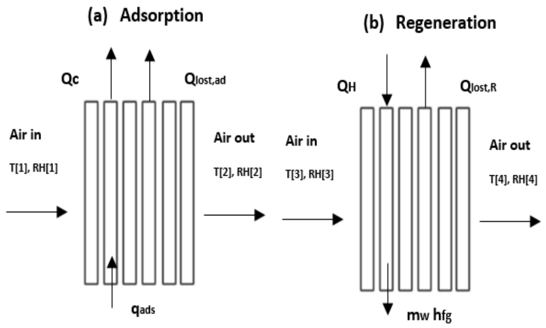

Fig. 4: Illustration of humid air passage through the HE during adsorption and regeneration

The system is evaluated using the energy balance between the air and desiccant for adsorption and regeneration process. The cooling load in adsorption process has two components, i.e., the sensible and latent energy that is caused by the phase change of the adsorbed moisture. Assuming steady state process and for the ideal gas/air, the energy rate for the adsorption process is described in Eq. 4 and 6 as:

$$\frac{dE}{dt} = -\dot{Q}_c + \dot{m}_1 h_1 - \dot{m}_2 h_2 + q_{ads}.\ \dot{m}_w - \dot{Q}_{lost,ad} = 0 \quad (4)$$

The specific enthalpy of moist air (h) is defined as the total enthalpy of dry air and water vapor mixture per every kg of dry air and is calculated by Eq. 5 [22] as:

$$h_a = C_{p,a}T + \omega(C_{p,v}T + 2501) \quad (5)$$

where $C_{p,a}$ and $C_{p,v}$ are dry air and vapor specific heat capacities, respectively, T is the air temperature, and ω is specific humidity of moisture in air.

The $\dot{Q}c$ is the cooling load provided by TEC and can be written per Eq. 6 as:

$$\dot{Q}c + \dot{Q}_{lost,ad} = \dot{m}a.(T[1] - T[2])[C_{p,a} + C_{p,v}(\omega_1 - \omega_2)] + q_{ads}.\ \dot{m}_w \quad (6)$$

Where $\dot{m}_{ax}$, h_{ax}, $T[x]$ are respectively mass flow rate, enthalpy, and temperature of the air at the inlet (x=1) and outlet (x=2) of heat exchanger, $\dot{m}_w$ *is* water collection capacity, while $Q_{lost,ad}$ and q_{ads} is the lost heat during adsorption and adsorption heat of desiccant materials. This adsorption heat to the NIPAM coating is evaluated using Eq. 7 and is expressed as [20]:

$$q_{ads} = 412.6\, X_e^{-0.18}.\exp(-2.55\, X_e) + 2407 \left(\frac{kJ}{kg_w}\right) \quad (7)$$

The energy rate for the regeneration process is described in Eq. 8 as:

$$\frac{dE}{dt} = \dot{Q}_H + \dot{m}_3 h_3 - \dot{m}_4 h_4 - \dot{Q}_{lost,R} - \dot{m}_w h_{fg} = 0 \quad (8)$$

Where $\dot{Q}_H$ is the heating load delivered by TEC, $\dot{Q}_{lost,R}$ is the lost heat during regeneration and is written per Eq. 9 as:

$$\dot{Q}_H - \dot{Q}_{lost,R} = \dot{m}a.(T[4] - T[3])[C_{p,a} + C_{p,v}(\omega_4 - \omega_3)] + \dot{m}_w h_{fg} \quad (9)$$

General exergy balance in relation to DCHE behavior under adsorption and regeneration processes is described by Eq. 10 and 11, respectively. Exergy balance analysis helps to evaluate the quality of the energy transfer within the system by combining both the 1st & 2nd thermodynamics laws. The destroyed exergy $\dot{E}x_{dest,ad}$ during adsorption is computed by Vivekh et al. [22] as per Eq. 10 as:

$$-Q_c\left(1 - \frac{T_o}{T_C}\right) + \dot{m}_a(e_1 - e_2) + q_{ads}.\,\dot{m}_w\left(1 - \frac{T_o}{T_{d,ad}}\right) - Q_{lost,ad}\left(1 - \frac{T_o}{T_{d,ad}}\right) = \dot{E}x_{dest,ad} \quad (10)$$

Where T_c and $T_{d,ad}$ are the temperatures of cold side of TEC and the desiccant during adsorption, respectively. The destroyed exergy $\dot{E}x_{dest,R}$ during regeneration is computed according to Eq. 11 as [22]:

$$Q_H\left(1 - \frac{T_o}{T_H}\right) + \dot{m}_a(e_1 - e_2) - Q_{lost,R}\left(1 - \frac{T_o}{T_{d,R}}\right) - \dot{m}_w e_w = \dot{E}x_{dest,R} \quad (11)$$

Where T_H and $T_{d,R}$ are the temperatures of hot side of TEC and the desiccant during regeneration, respectively. The absolute entropy generated during adsorption and regeneration process is defined by Eq. 12 as [22]:

$$\dot{S}_{gen} = \frac{\dot{E}x_{dest,ad} + \dot{E}x_{dest,R}}{T_o} \quad (12)$$

2.3 Heat transfer modeling of atmospheric water harvesting system

Next, heat transfer analysis is carried out and that is done following the schematics in Fig 5.

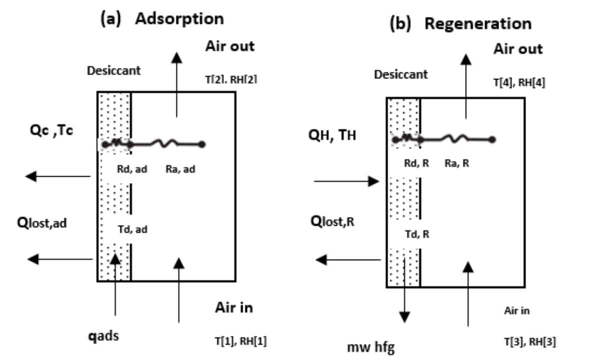

Fig. 5: Heat transfer passage through the heat exchange during adsorption and regeneration

For adsorption process, the heat transfer between the air and the desiccant is of two types, i.e., convection and latent, and is written per Eq. 13 as [23-24]:

$$\dot{Q}_{a,ad} = h_{a,ad}.A.\left(T[1] - T_{d,ad}\right) + q_{ads}.\dot{m}_w \quad (13)$$

Where $\dot{Q}_a$ is heat rate transfer between the inlet air and desiccant materials, h_a is heat transfer coefficient of air, A is the area of heat transfer and T_d is the temperature of desiccant materials. The heat transfer between the desiccant and the heat exchanger fins is of conductive type and is expressed in Eq. 14 as [23-24]:

$$\dot{Q}_{d,ad} = K_d.A.\frac{(T_{d,ad} - T_c)}{t_d} \quad (14)$$

Where T_c is the temperature of the cold side of the TEC, A is the heat transfer area, K_d is the desiccant conductivity, and t_d is the desiccant thickness. Assuming the heat transfer is constant through the entire system, i.e., $\dot{Q}_a = \dot{Q}_d$ this reduces the heat equation to Eq. 15 and is expressed as [23-24]:

$$\dot{Q}_{ad} = \frac{A.(T[1] - T_c) + \frac{q_{ads}.\dot{m}_w}{h_{a,ad}}}{\frac{1}{h_{a,ad}} + \frac{t_d}{K_d}} \quad (15)$$

Where A is the total area of heat transfer and comprises of $A = A_f + A_t$ and thus this heat is equated to the cooling load required, i.e., $\dot{Q}_{ad} = \dot{Q}c + \dot{Q}_{lost,ad}$ [23].

For generation process, the heat transfer between the air and the desiccant is per Eq. 16 as:

$$\dot{Q}_{a,R} = h_{a,R}.A.\left(T[3] - T_{d,R}\right) \quad (16)$$

The heat transfer between the desiccant and the heat exchanger fins is of conductive type and is expressed in Eq. 17 as:

$$\dot{Q}_{d,R} = K_d.A.\frac{(T_{d,R} - T_H)}{t_d} \quad (17)$$

Where T_H is the temperature of the cold side of the TEC, A is the heat transfer area, K_d is the desiccant conductivity, and t_d is the desiccant thickness. Assuming the heat transfer is constant through the entire system, so the heat transfer is expressed as:

$$\dot{Q}_R = \frac{A.(T[3] - T_H)}{\frac{1}{h_{a,R}} + \frac{t_d}{K_d}} \quad (18)$$

This heat is equated to the heating load for regeneration, i.e., $\dot{Q}_R = \dot{Q}_H - \dot{Q}_{lost,R}$.

2.4 Performance parameters

Moisture removal capacity (MRC) is the difference in humidity ratio at the inlet and outlet of the DCHE and this is expressed in Eq. 19 as:

$$MRC = \omega_1 - \omega_2 \quad (19)$$

Moisture removal rate (MRR) is the rate of water vapor removal from air which is induced by the desiccant coating and that is written per Eq. 20 as:

$$MRR = \dot{m}_a(\omega_1 - \omega_2) \quad (20)$$

The 1st law energy efficiency of DCHEs is calculated in terms of the thermal coefficient of performance (COP), which is defined as the ratio of the mass of water produced to the thermal energy input of TEC during adsorption and regeneration processes and is written per Eq. 21 as:

$$COP = \frac{\dot{m}_w h_{fg}}{Q_H + Q_C} \quad (21)$$

The 2nd law efficiency is defined by the ratio of the minimum heat input to the thermal energy input during adsorption and regeneration processes and expressed in Eq. 22 as:

$$\eta_{\Pi} = \frac{\dot{Q}_{H,min}}{Q_H + Q_C} \quad (22)$$

3. Results and Discussion

To setup the model, the inlet dry bulb temperature and relative humidity are considered fix and are set at 30ºC and 90%, respectively. Furthermore, water collection capacity is assumed at nominal value of 0.25L/h. Heat transfer coefficient (h_a) was calculated based on the pervious equations and dimensions of heat exchanger using commercial software Engineering Equation Solver (EES). The calculated value for adsorption and generation process was found to be $h_{a,ad}$ = 0.04804 kW/m^2.K and $h_{a,R}$ = 0.219 kW/m^2.K. The heat exchanger dimensions, and system input data are summarizing in Table 1.

Table. 1: Heat exchanger dimensions and input data

Parameter	Value
Height of the DCHE-H (m)	0.22
Width of the DCHE-W (m)	0.105
Length of the DCHE-L (m)	0.11
Inner diameter of tube-din (m)	0.00598
Thickness of tube-dt (m)	0.0004
Number of tubes Nt	16
Thickness of fin-df (m)	0.0004
Length of fin – Lf (m)	0.0058
Thickness of desiccant layer-dd (m)	0.002
T[1] (°C)	25-30
Rh [1] (%)	0.3-0.9
P (kPa)	100
T[3], T_o (°C)	25
Rh [3], Rh_o (%)	0.3
Qc (kW)	0.3
Tc (°C)	15-20
QH (kW)	0.4
TH (°C)	40
Kd (kW/m.K)	0.0006
Ra (kJ/kg.K)	0.287
Ma (kg/s)	0.005-0.05

3.1 Effect of inlet air temperature:

The effect of inlet air temperature (T[1]) on the thermodynamic performance of water collection system is studied in this section. Fig. 6 and 7 illustrate that MRC and MRR are increased when T[1] is regulated from 25°C to 30°C. This improved performance is attributed to the increased sorption capacity of the desiccant at higher air temperature. A higher RH at higher air temperature increases the sorption capacity of the desiccant and the moisture transfer rate. A higher MRC and MMR translate to a higher COP according to the Eq. 21 as shown in Fig. 8. A higher temperature gradient between air and TEC cold side promotes heat transfer rates and yields improved MRC, MRR, and COP more than 70% at relative humidity of 90%. However, additional heat transfer through the DCHE system at higher T[1] the entropy generation rates are increased, thereby lowering the second law efficiency of the system by more than 70% at relative humidity of 90% as shown in Fig. 9 and 10.

3.2 Effect of air flow rate:

Fig. 11 and 12 illustrate the performance of water collection system under varying air flow rate (m_a). Increasing m_a from 0.005 to 0.05 kg/s reduces the air-desiccant interaction/residence time. As a result, MRC drops by 88% indicating that lower air flow rates increased sorption capacity of the desiccant and produce drier air as shown in Fig. 11. However, the rate of moisture removal rate (MRR) by the system increases by 18% at higher m_a as evaluated by Eq. 20.

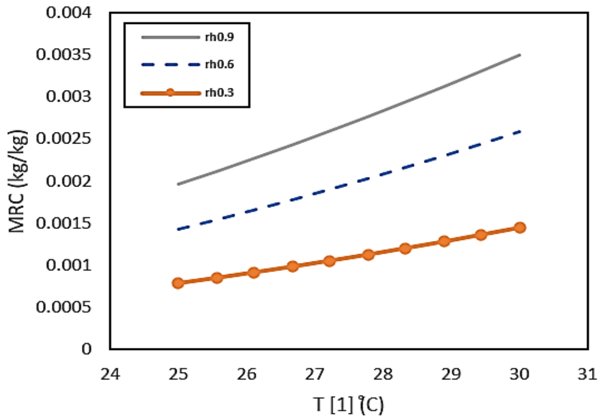

Fig. 6: Effect of varying inlet temperature on MRC at different relative humidity

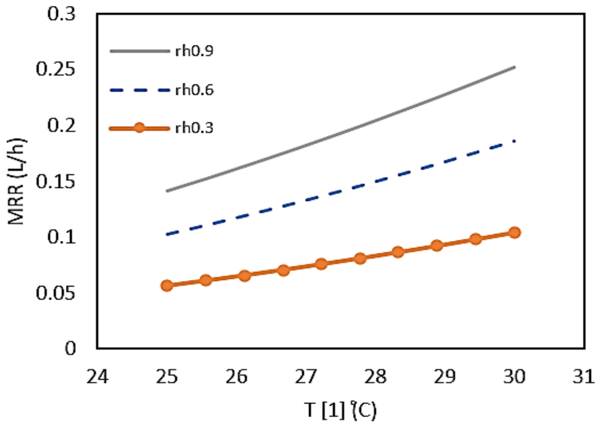

Fig. 7: Effect of varying inlet temperature on MRR at different relative humidity

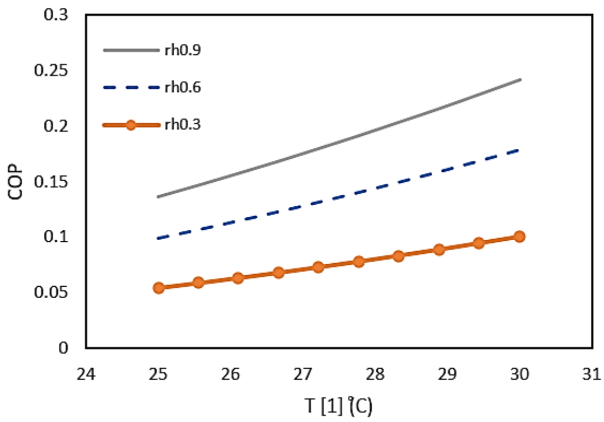

Fig. 8: Effect of varying inlet temperature on COP at different relative humidity

Fig. 11 shows that the COP of the system increases almost linearly at 18% rate as the air mass flow increases. The increase in the 1st law efficiency is attributed to the improved moisture removal rate and collected water at higher air flow rates. Further, as observed in Fig. 12, exergy destruction rates measured in terms of entropy (S_{gen}) raised by about 118%. This is due to the directly proportional relationship between the exergy destruction rate and the air flow rate, as observed in Eq. 10 and

11. Additionally, a higher moisture transfer rate further contributes to higher entropy production rates. The increase in entropy generation lowers the 2nd law efficiency of the system by as much as 90%.

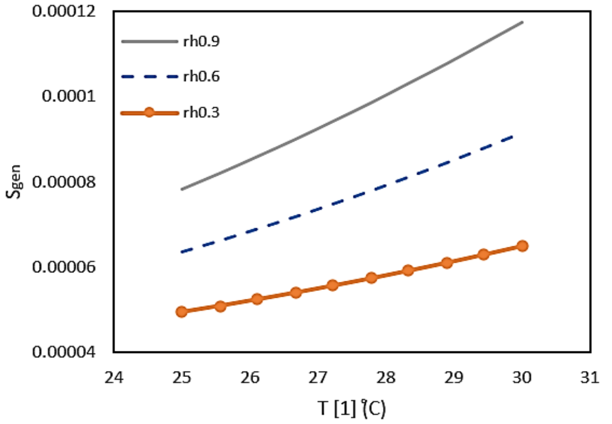

Fig. 9: Effect of varying inlet temperature on Sgen at different relative humidity

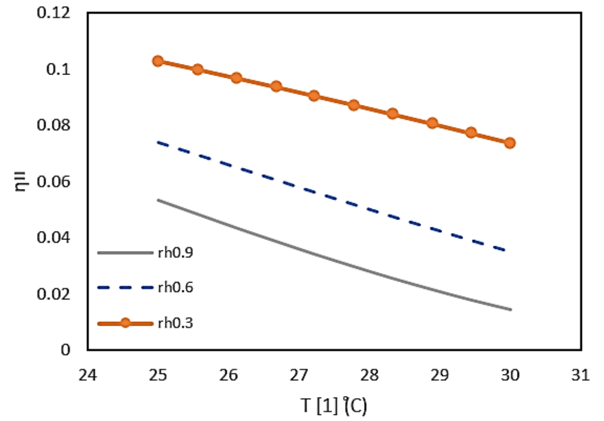

Fig. 10: Effect of varying inlet temperature on ηII at different relative humidity

3.3 Effect of cold side temperature of TEC:

The performance of water collection system based on varying cold side temperature of the TEC (T_c), is shown in Fig. 13 and 14. At 15°C, MRC, MRR, and COP are observed to be the highest. When Tc is systematically regulated to 20 °C the MRC, MRR, and COP are observed to linearly depreciate by 48%. A lower T effectively means more heat transfer during the sorption process and facilitates improved sorption capacity of the desiccant and 1st law efficiency. On the other hand, S_{gen} markedly decreases by almost 23% when T_c is raised from 15°C to 20°C; indicating a decrease in energy losses and destruction due to lowering the temperature difference between T_c and inlet temperature of air (T[1]). Subsequently, lower S_{gen} promotes 2nd law efficiency (ηII) that increases by almost 42%.

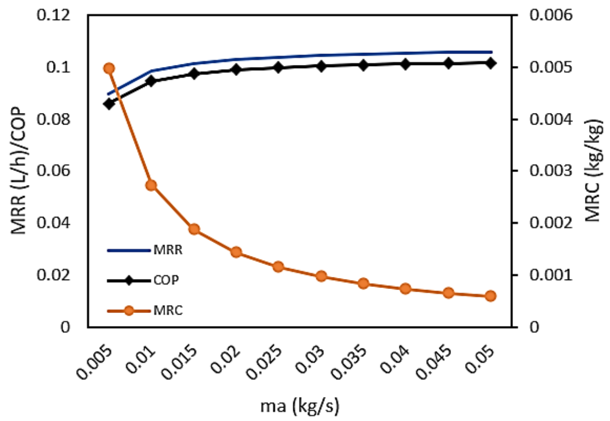

Fig. 11: Effect of varying air mass flow rate (ma) on MRC, MRR, and COP

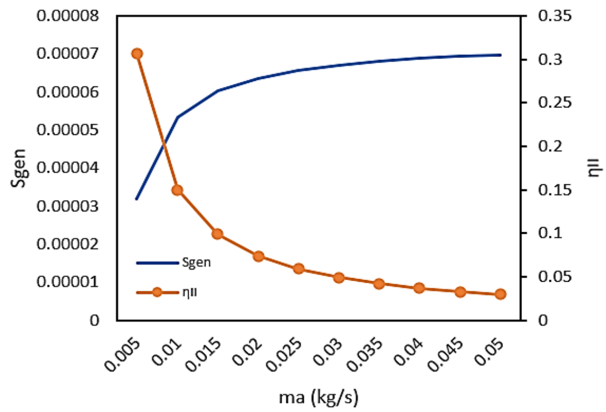

Fig. 12: Effect of varying air mass flow rate (ma) on Sgen and ηII

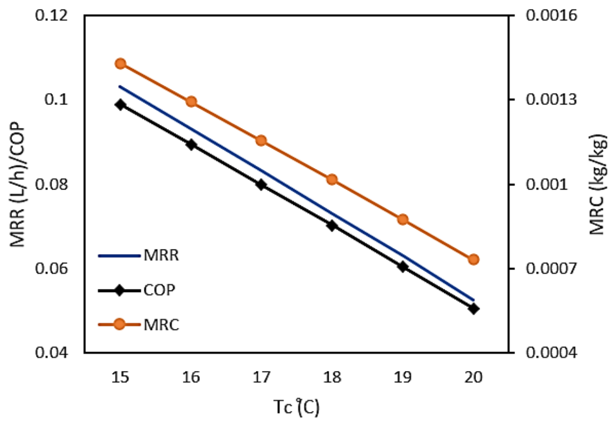

Fig. 13: Effect of varying cold side temperature of TEC on MRC, MRR, and COP

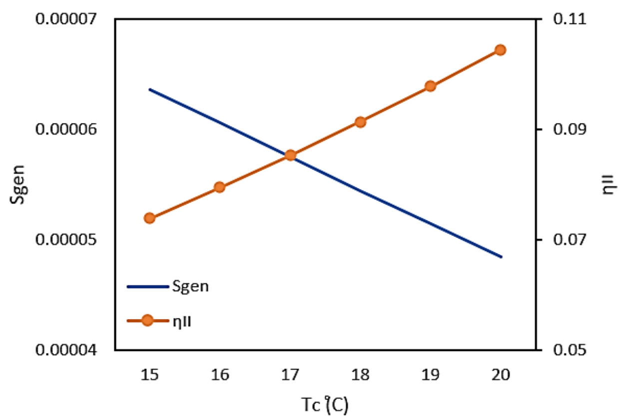

Fig. 14: Effect of varying cold side temperature of TEC on S_{gen} and η_{II}

4. Conclusion

In this work, water harvester system was analyzed using the 1st and 2nd laws of thermodynamics. A steady-state evaluation methodology is carried out to compute the energy transfer and heat losses associated with adsorption and regeneration processes. Moreover, heat transfer analysis through desiccant coated heat exchanger (DCHE) was employed for both processes. At each thermodynamic state of adsorption and generation process, the specific flow exergy was computed to determine the entropy generated by the system. Key results and observations that emerged from this study include:

- A higher relative humidity (RH) at higher air temperature increases the sorption capacity of the desiccant and the moisture transfer rate which translate to a higher moisture removal capacity (MRC), moisture removal rate (MRR), and coefficient of performance (COP). However, additional heat transfer through the DCHE system lowers the 2nd law efficiency due to the rise in entropy generation rates.
- 1st law efficiency and moisture transfer rate significantly improved with increasing flow rate. However, a lower air flow rate lowers the entropy production rate and improves 2nd law efficiency by 90%.
- Cold side temperature of TEC should be maintained as low as possible to achieve higher MRC and COP.

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