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Energy Saving of Hybrid Desiccant Air Conditioning System Powered by Solar Energy

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Abstract

A two-stage hybrid air dehumidification system referred as, system A is compared to a single stage one referred as, system B and tested in the present work. The governing equations of the theoretical model is solved and validated. The results showed a good agreement to study the effect of different operating parameters of the presented system. The desiccant material is regenerated by an air solar heater. The coefficient of performance of this system, cooling productivity and required regeneration temperature are studied at these variables temperature of process air and humidity ratio, the temperature and concentration of the desiccant solution, and air to desiccant mass flow rates ratios. The obtained results from system A are compared to a single stage cycle (system B). At the specific operating and design parameters, an average increase of 48.2 % in both cooling productivity and COP of system A is achieved compared to system A. This why, the presented system is recommended to be used in air conditioning applications in Saudi Arabia market as it saves energy of about 37.2%.

Keywords: Solar energy; Energy saving; Evaporative cooling; Desiccant solution; Hybrid systems.

INTRODUCTION

Dehumidifying air by desiccant equipment became an attractive and efficient method due to utilizing a low-grade source of energy for regeneration. Through this technology, an energy- efficient AC systems have been generated leading to an energy saving over the traditional VCS besides allowing independent control of both temperature and relative humidity especially in the hot and humid area [1]. Ali et al. [2] introduced a novel algebraic model to investigate the effect of critical operating parameters on the performance of desiccant wheels in arid climates, using wound silica gel and molecular sieve desiccants. These parameters are process inlet humidity ratio, inlet volume flow rate, R/P ratio regenerative temperature and speed of rotation. Liu et al. [3] revealed a two-stage desiccant wheel systems are an effective method enhancing the dehumidification performance. Demis et al. [4] introduced a numerical study of a novel, multi-stage desiccant AC system allocated for moderate climates. This system was based on multi-stage cooling process through the Maisotsenko Cycle(M–Cycle. The proposed system was able to attain a thermal COP of up to 4.0.

Rambhad et al. [5] presented a desiccant systems for moist and moderate climates based on the combination of a desiccant unit (solid or liquid) and an indirect evaporative air cooler. Xianhua et al. [6] investigated an energy efficient liquid desiccant cooling and dehumidification (LDCD) system. They indicated that the flow rate of desiccant solution and energy utilization of the presented system are lowered by 39.64% and 22.3% compared to the conventional LDDS; respectively.

Joon et al. [7] studied empirically the effect of the working-to-primary air flow ratio variation on the dehumidification efficiency of an evaporative cooling-assisted internally cooled liquid desiccant dehumidifier. They reported that the optimal dehumidification and cooling performance of the presented arrangement was likely when the working-to-primary air flow ratio is 0.5.

Ghulam et al. [8] designed and experimentally tested an integrated solar assisted desiccant cooling system on some different month' days in summer.

They found that, the effectiveness of dehumidification dropped while increasing the inlet temperature and positively affected by raising in the inlet humidity. It changed from 29% to 49%. Meanwhile, the effectiveness which based on the dew point changed from 50% to 78%. Some researchers [9,10] introduced an experimental analysis of a novel internally-cooled dehumidifier with self-cooled liquid desiccant.

The regeneration energy represents a great problem that hinder the spread of using the desiccant dehumidification systems. So, it is important to study the desiccant dehumidification system as whole integrated cycle (i.e. , dehumidifier and regenerator) in order to evaluate the actual energy performance of the whole system. In the present work the two-stage counter flow air dehumidifier with two feeding desiccant solution lines introduced by the author [11] is integrated with a desiccant regenerator and both are introduced in a cycle (system A). The coefficient of performance of this system, cooling productivity and required regeneration temperature are studied at these variables temperature of process air and humidity ratio, the temperature and concentration of the desiccant solution, and air to desiccant mass flow rates ratios. The obtained results from system A are compared to a single stage cycle (system B).

System description and operation

The flow chart of the proposed system A is shown in Fig.1a. The detailed description of the system is introduced in [11]. The exit process air from deh_a at state 1 (T_{A1b} =T_{A2ac}, y_{A1b} = y_{A2ac}) is cooled in HX using an air path of 24°C temperature then routed to the deh_b for further dehumidification and exited at state 2 (T_{A2b}=T_{A2}, y_{A2b}=y_{A2}). The desiccant solution exit condition (m_{s2},T_{s2}, X_{s2},h_{s2}) is calculated by combination both solution exit conditions (m_{s2a}, T_{s2a}, X_{s2a}, h_{s2a} and m_{s2b}, T_{s2b}, X_{s2b}, h_{s2b}) from the dehumidifiers a and b; respectively at state 2. The diluted desiccant solution at state 2 is heated to state 3 by the heat rejected from the strong desiccant solution exited from the regenerator in a solution heat exchanger (SHX). In order to return the desiccant solution concentration to its initial conditions X_{s1}, the regeneration temperature is adjusted to state 4 in an auxiliary heater (AH) to achieve this purpose. Also, the desiccant solution temperature is adjusted to T_{s1} using a cooling water path through a water to solution heat exchanger (HX). By this way the cycle of air and desiccant is completed and system A and B parameters can be calculated.

The impact of different operating parameters are on both systems A and B coefficient of performance, cooling productivity and required regeneration temperature is investigated.

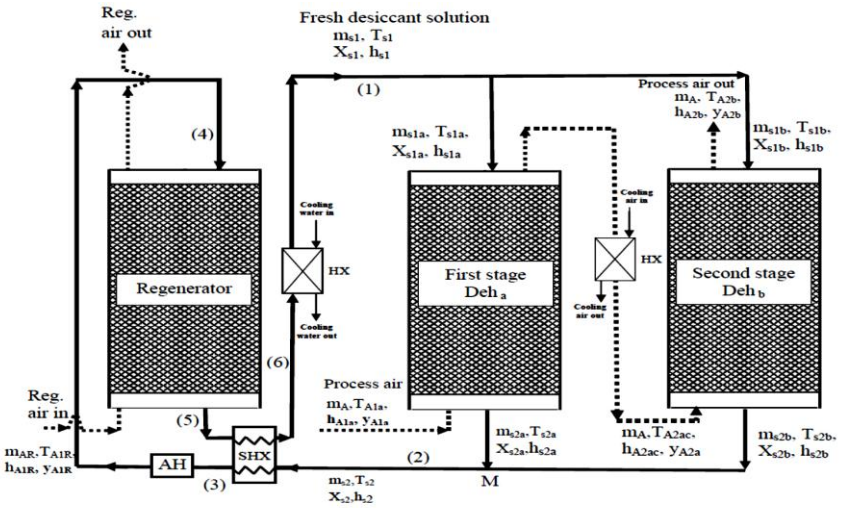

Fig.1a Schematic diagram for the proposed air dehumidification system A

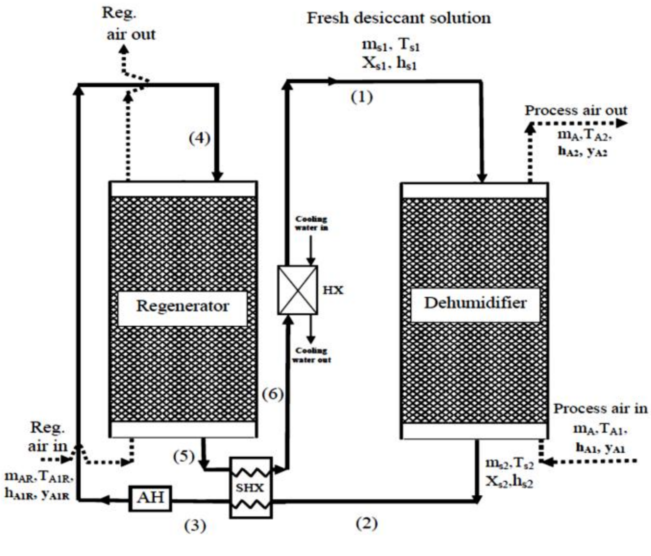

Fig.1b Schematic diagram for the proposed air dehumidification system B

Theoretical model validation and Numerical solution

The detailed model and solution methodology is introduced by M. Bassuoni [11]. The model is validated and shows good reliability to describe the proposed system performance indices.

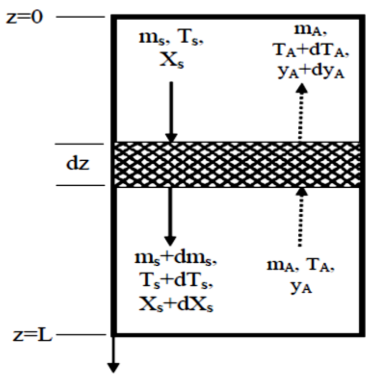

Fig.1c Elemental heat and mass transfer

Performance analysis

Air exit parameters from the second stage ($y_{A2}= y_{A2b}$, $T_{A2}=T_{A2b}$)which represents the process air exit conditions can be calculated. Also, all desiccant solution exit conditions from the regenerator is determined. So, the coefficient of performance of the system(COP) and cooling productivity (latent heat removal rate) ($\dot{Q}_{C.P}$) of both systems A and B at a constant inlet air and desiccant solution conditions can be given as follows:

Performance indices for system A:

The cooling productivity ($\dot{Q}_{C.P}$) can be calculated as follows:

$$\dot{Q}_{C.P} = \dot{m}_A(h_{A1a} - h_{A2b}) \qquad (1)$$

The cooling power ($\dot{Q}_{Cooling}$) required to cool the strong desiccant solution from state 6 to state 1 can be calculated as follows:

$$\dot{Q}_{C.P} = \dot{m}_s(h_{S6} - h_{S1}) \qquad (2)$$

The regeneration heat($\dot{Q}_{reg}$) required to regenerate the dilute desiccant solution from state 3 to state 4 can be calculated as follows:

$$\dot{Q}_{C.P} = \dot{m}_s(h_{S4} - h_{S3}) \qquad (3)$$

The system coefficient of performance (COP) can be calculated as follows:

$$COP = \frac{\dot{Q}_{C.P}}{\dot{Q}_{reg} + \dot{Q}_{Cooling}} \qquad (4)$$

The same equations 20-23can be rearranged for system B.

RESULTS AND DISCUSSION

Effects of inlet parameters on the system performance indices

A performance comparison between the two systems A and B is separately investigated as follows:

Impact of inlet air humidity ratio

The Impact of inlet air humidity ratio (y_{A1}) on exit air temperature (T_{A2}), exit air humidity ratio (y_{A2}) and required regeneration temperature (T_{reg}) for both systems A and B is shown in Fig. 2a. As y_{A1} is increased, T_{A2} is nearly constant while both T_{reg} and y_{A2} are increased. The exit y_{A2} from system A is lower than system B. For system A, y_{A2} decreases by about 47.8 % compared to system B at y_{A1} of 0.02 $kg_v kg_{d.a}^{-1}$. Although there is an average increase in the required T_{reg} for system A with about 7.6 % compared to system B, both COP and cooling productivity ($Q_{C.P}$) of system A is increased by about 50% as shown in Fig. 2b. This may be viewed as T_{A2} for system A decreases due to intermediate cooling between the two stages of the dehumidifier, the $Q_{C.P}$ is increased which in turn increases the COP for system A over System B.

Impact of inlet air temperature

The impact of inlet air temperature (T_{A1}) on exit air temperature (T_{A2}), exit air humidity ratio (y_{A2}) and required regeneration temperature (T_{reg}) for both systems A and B is shown in Fig. 3a. As T_{A1} is increased, T_{A2} is also increased while y_{A2} is slightly increased and T_{reg} is slightly decreased. This may be explained as follows: when T_{A1} increases, the temperature of desiccant solution is also increases during absorption process which in turn decreasing the ability of air to be dehumidified. The exit air humidity ratio y_{A2} from system A is lower than system B. For system A, y_{A2} is decreases by about 42.4 % compared to system B at T_{A1} equal to 35°C. Although there is an average increase in the required T_{reg} for system A with about 6.25% compared to system B, both COP and cooling productivity ($Q_{C.P}$) of system A is increased by about 57.3% and 50%; respectively as shown in Fig. 3b. When T_{A2} for system A decreases due to intermediate cooling between the two stages of the dehumidifier, the $Q_{C.P}$ is increased which in turn increases the COP for system A over System B.

Impact of inlet desiccant solution concentration

The effect of inlet desiccant solution concentration (X_{s1}) on exit air temperature (T_{A2}), exit air humidity ratio (y_{A2}) and required regeneration temperature (T_{reg}) for both systems A and B is shown in Fig. 4a. As X_{s1} is increased, y_{A2} is decreased while T_{A2} is nearly constant and T_{reg} is directly increases. From Fig. 4a, when X_{s1} is increased from 0.33 to 0.43 kg_d/kg_s, the regeneration temperature is increased by about 16.8% for both systems. On the other hand, although there is an average increase in the required T_{reg} for system A compared to system B, both COP and cooling productivity ($Q_{C.P}$) of system A achieve an average increase of about 51.1% and 53.2%; respectively as shown in Fig. 4b.

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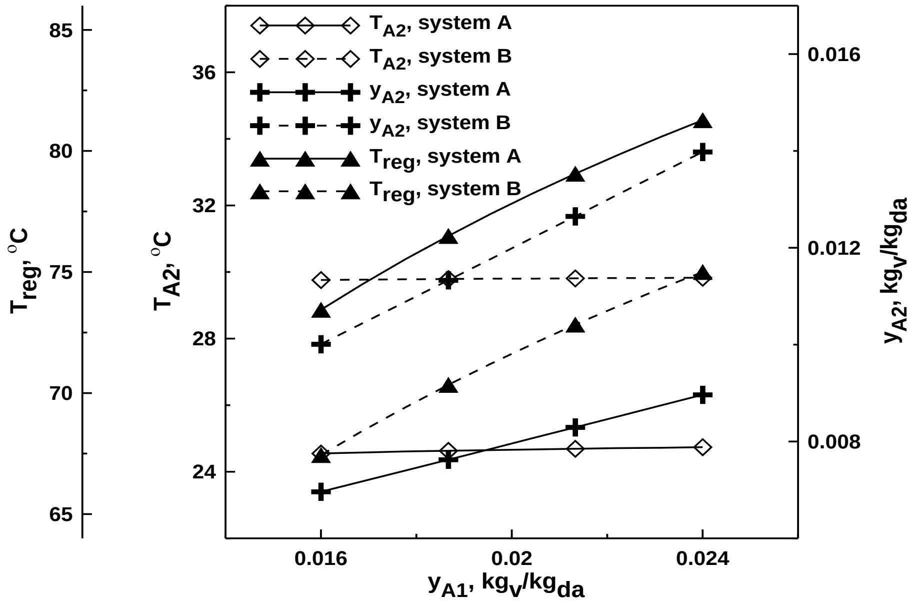

Fig.2a The effect of inlet air humidity ratio (y_{A1}) on T_{A2}, y_{A2} and T_{reg}.

(m_A=0. 3 kg/s, m_{s1}=0. 2 kg/s, T_{A1}=40 °C, T_{s1}=24 °C, X_{s1}=0.32 kg_d/kg_s)

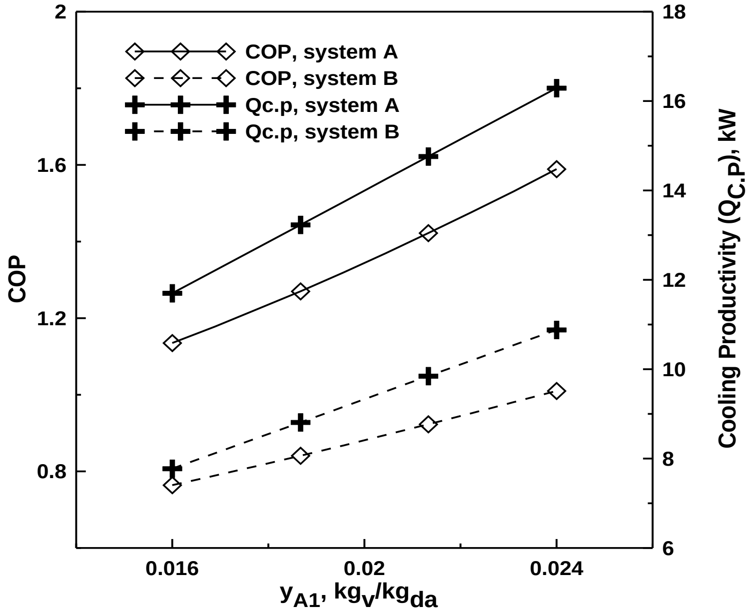

Fig.2b The effect of inlet air humidity ratio (y_{A1}) on COP and $Q_{C.P.}$

(m_A=0. 3 kg/s, m_{s1}=0.2 kg/s, T_{A1}=40 °C, T_{s1}=24 °C, X_{s1}=0.32 kg_d/kg_s)

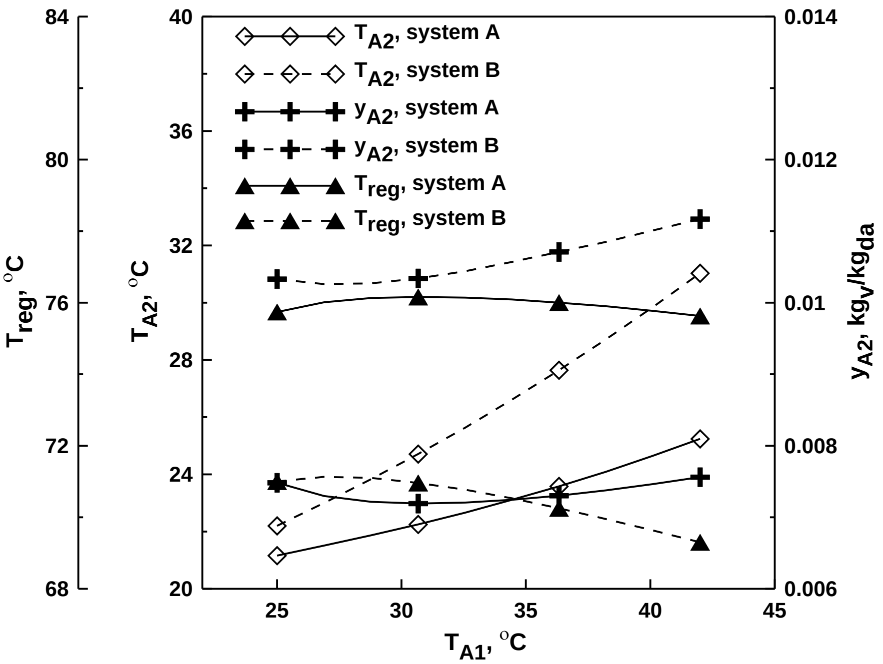

Fig.3a The effect of inlet air temperature (T_{A1}) on T_{A2}, y_{A2} and T_{reg}.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, X_{s1}=0.32kg$_d$/kg$_s$, T_{s1}=24 °C, y_{A1}=0.018 kg$_v$/kg$_{da}$)

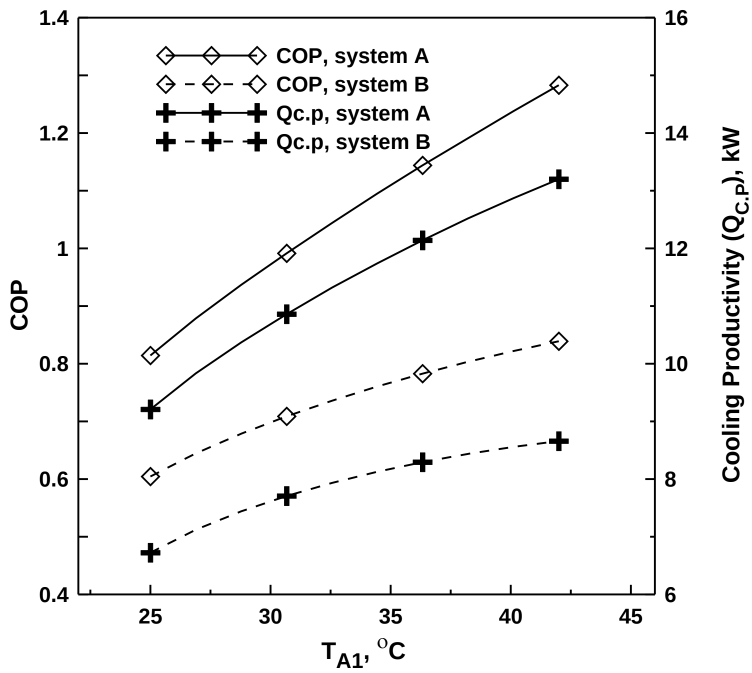

Fig.3b The effect of inlet air temperature (T_{A1}) on COP and $Q_{C.P}$.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, X_{s1}=0.32kg$_d$/kg$_s$, T_{s1}=24 °C, y_{A1}=0.018 kg$_v$/kg$_{da}$)

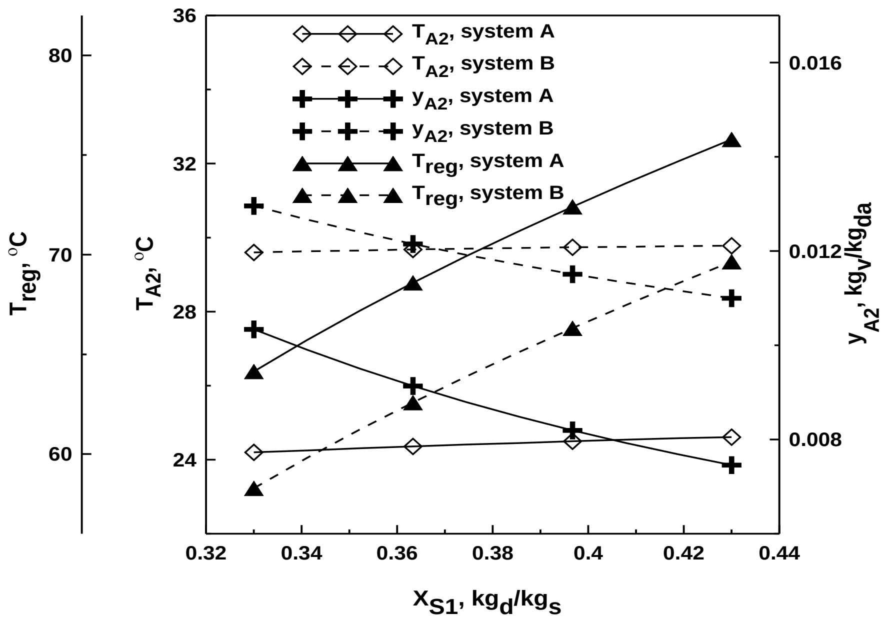

Fig.4a The effect of inlet desiccant solution concentration (X_{s1}) on T_{A2}, y_{A2} and T_{reg}.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, T_{A1}=40 °C, T_{s1}=24 °C, y_{A1}=0.018 kg_v/kg_{da})

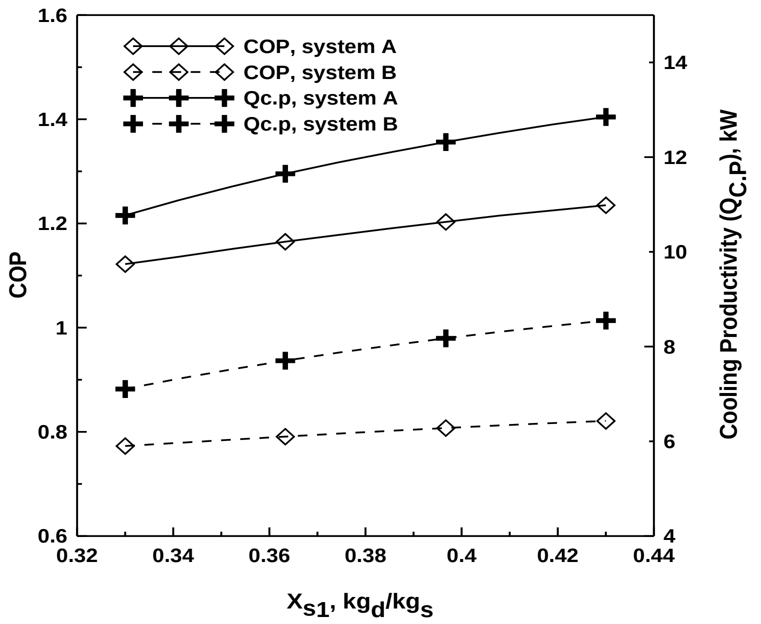

Fig.4b The effect of inlet desiccant solution concentration (X_{s1}) on COP and $Q_{C.P}$.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, T_{A1}=40 °C, T_{s1}=24 °C, y_{A1}=0.018 kg_v/kg_{da})

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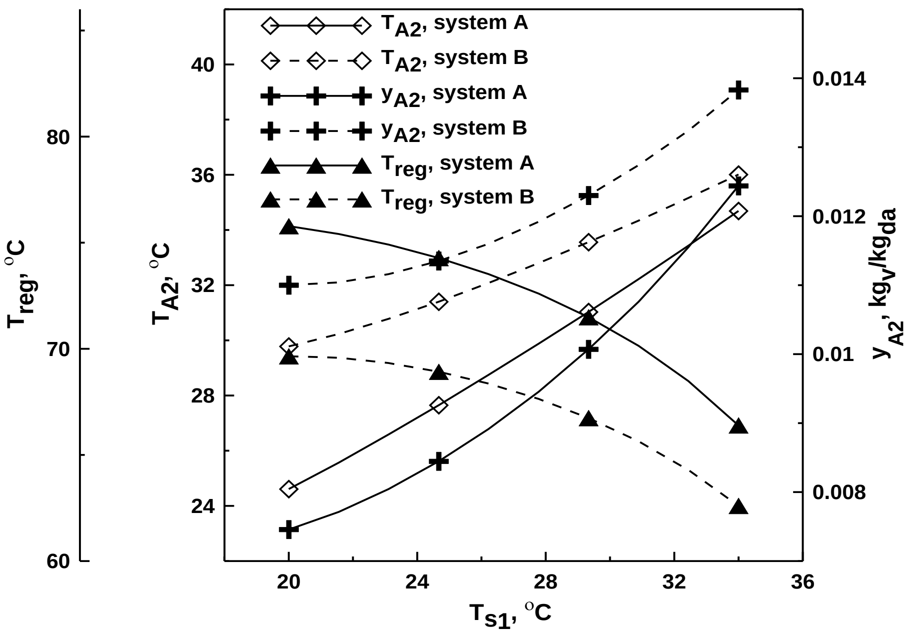

Fig.5a The effect of inlet desiccant solution temperature (T_{s1}) on T_{A2}, y_{A2} and T_{reg}.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, T_{A1}=40 °C, X_{s1}=0.32kg_d/kg_s, y_{A1}=0.018 kg_v/kg_{da})

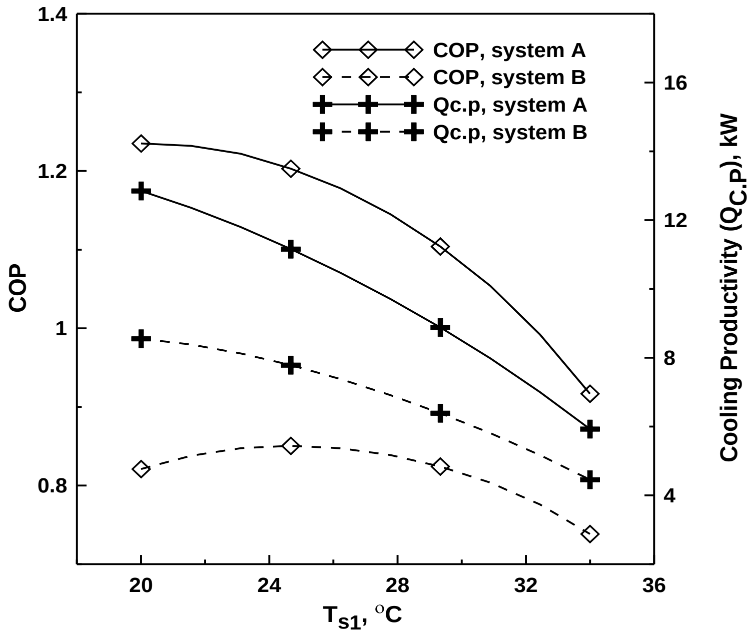

Fig.5b The effect of inlet desiccant solution temperature (T_{s1}) on COP and $Q_{C.P}$.

(m_A=0.3 kg/s, m_{s1}=0.2 kg/s, T_{A1}=40 °C, X_{s1}=0.32kg_d/kg_s, y_{A1}=0.018 kg_v/kg_{da})

Impact of inlet desiccant solution temperature

The effect of inlet desiccant solution temperature (T_{s1}) on exit air temperature (T_{A2}), exit air humidity ratio (y_{A2}) and required regeneration temperature (T_{reg}) for both systems A and B is shown in Fig. 5a. As T_{s1} increases, both T_{A2} and y_{A2} are increased while T_{reg} is sharply decreased. The dehumidification system A achieves an average increase in $Q_{C.P}$ and COP over the dehumidification system B by an amount of nearly 50% for both of them as shown in Fig. 5b. On the other hand, when T_{s1} is increased by 70%, both $Q_{C.P}$ and COP are decreased by about 90% and 34%; respectively for both systems. This system achieves an energy saving of about 37.2% compared to the conventional system.

Impact of air mass flow rate

The effect of air to desiccant solution mass flow rate ratio (m_A/m_{s1}) on exit air temperature (T_{A2}), exit air humidity ratio (y_{A2}) and required regeneration temperature (T_{reg}) for both systems A and B is shown in Fig. 6a. When the ratio of m_A/m_{s1} is increased, both T_{A2} and y_{A2} are slightly increased while T_{reg} increases. The $Q_{C.P}$ and COP for system A are increased by an average of 69.7 % and 61.2 % ; respectively compared to those of system B at m_A/m_{s1} is equal to 2.5 as shown in Fig 6.b.

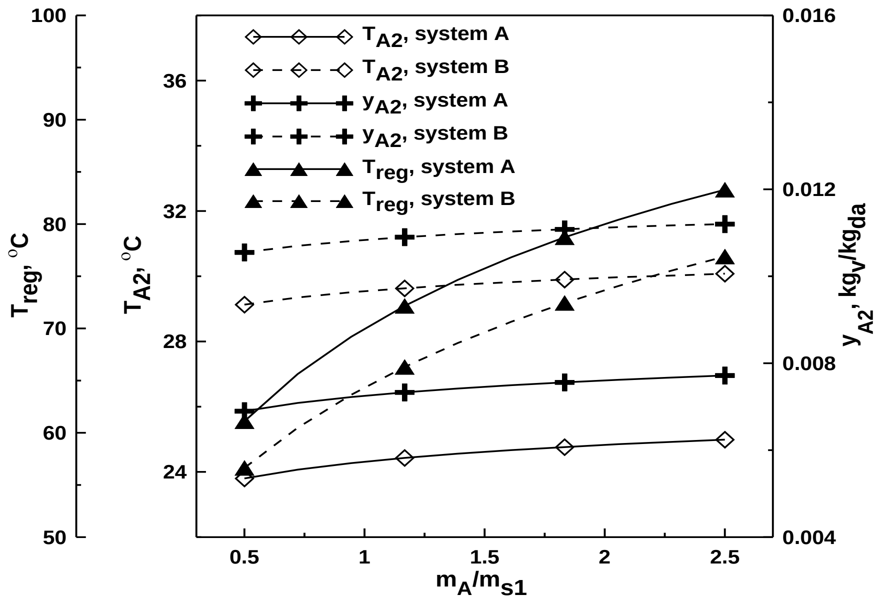

Fig.6a The effect of air to desiccant solution mass flow rate ratio (m_A/m_{s1}) on T_{A2}, y_{A2} and T_{reg}.

(X_{s1}=0.32kg_d/kg_s, T_{s1}=28°C, T_{A1}=38°C, y_{A1}=0.018 kg_v/kg_{da})

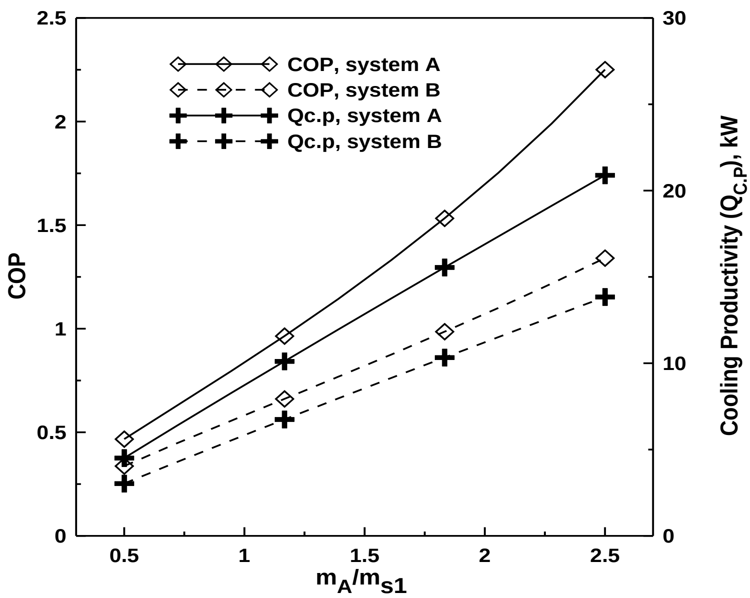

Fig.6b The effect of air to desiccant solution mass flow rate ratio (m_A/m_{s1}) on COP and $Q_{C.P}$.

(X_{s1}=0.32kg_d/kg_s, T_{s1}=28°C, T_{A1}=38°C, y_{A1}=0.018 kg_v/kg_{da})

CONCLUSION

A two-stage hybrid air dehumidification system referred as, system A is compared to a single stage one referred as, system B and tested in the present work. The governing equations of the theoretical model is solved and validated. The results showed a good agreement to study the effect of different operating parameters of the presented system. After different runs, the following conclusions are summarized:

- For system A, y_{A2} decreases by about 47.8 % compared to system B at y_{A1} of 0.02 $kg_v kg_{d.a}^{-1}$.
- The required T_{reg} for system A increased by about 7.6 % compared to system B, but both COP and $Q_{C.P}$ of system A are increased by about 50%.
- At different studied operating parameters, the performance of system A is higher than system B.
- The proposed system saves an energy consumption of about 37.2%.
- When T_{s1} is increased by 70%, both $Q_{C.P}$ and COP are decreased by about 90% and 34%; respectively for both systems.
- The $Q_{C.P}$ and COP for system A are increased by an average of 69.7 % and 61.2 % ; respectively.

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