Applied Thermal Engineering 113 (2017) 1256–1265

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

Effect of operation conditions on the second law analysis of a desiccant cooling system

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HIGHLIGHTS

- The second law analysis of the desiccant cooling system is evaluated.
- Effect of various operating parameters on the system performance was studied.
- Effect of ambient air temperature and relative humidity was investigated.
- The Carnot COP is between 9 and 25.

ARTICLE INFO

Article history:
Received 26 July 2016
Revised 18 November 2016
Accepted 20 November 2016
Available online 22 November 2016

Keywords:
Air conditioning
Desiccant cooling system
Carnot COP
Second law analysis
Experimental data
Adsorption

ABSTRACT

In order to evaluate the potential of a desiccant based evaporative air conditioning system, a second law analysis of the systems was investigated. In this work, a wide range of working parameters that are regeneration temperature from 90 °C to 110 °C, volume flow rate from 2000 m^3 to 4000 m^3 and ambient air conditions which are relative humidity, ambient air temperature and wet-bulb temperature is carried out for performance calculation of the system. It is observed from results that the Carnot COP and the thermal COP of the system varies vice versa during the day. The highest value of Carnot COP and the thermal COP is about 25 and 0.62, respectively. Also, the cooling capacity has significant effect on the second law efficiency. The importance of this study shows that the second law analysis can provide useful information with respect to the theoretical upper limit of the system performance, which cannot be obtained from the first law of thermodynamic analysis alone.

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1. Introduction

Due to the negative impact of vapor-compression refrigeration systems, novel alternative cooling systems are being tried to develop recently. The negative effects of these systems have more difficulty due to energy spending and using of Chloro-Fluoro-Carbons caused greenhouse effect and creating ozone layer destruction.

One of the significant novel cooling systems is desiccant based cooling systems that are environmentally safe, affordable, energy-efficient. The operating range of these systems is quite extensive and they can work alone systems, or may work in conjunction with another chiller to increase indoor air quality of all types of buildings [1–4].

A general desiccant assisted cooling system consists of filters, fans, heat exchanger rotary desiccant dehumidifier, indirect/direct evaporative cooler, pumps, controls and ducts [5–9].

In these types of systems, the most used equipment is an evaporative cooling unit which is especially used for dry and hot ambient air conditions, but this component is not efficient for humid air conditions. To eliminate the disadvantages of humid air conditions, desiccants are used to apply evaporative cooling in such ambient air condition. One of the other important units for desiccant cooling system is a desiccant dehumidifier that is crucial component of these type systems [10–12].

Due to the formation of a low vapor pressure on the desiccant surface area, they draw moisture from the air. Because of the high partial pressure of the water in humid air, the water molecules move into the surface of the desiccant that has a low partial pressure. Therefore, the ambient air turns into dry air form with losing his moisture [13]. To understand the performance of air conditioning systems, one of significant thermodynamic analysis method is

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http://dx.doi.org/10.1016/j.applthermaleng.2016.11.143
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second law as is evaluated as an efficient method to achieve high efficiency of the system [14–17].

In order to show the real capacity of desiccant assisted cooling system, a number of thermodynamic analyses on the second law are performed in last years. Lavan et al. [18] evaluated a desiccant air conditioning system depend on the second law analysis and observed the performance of its individual components. According to investigation, they evaluated the reversible COP of desiccant system and present equal Carnot temperatures idea for evaluating reversible COP. As result, they found that the reversible COP and analyses are associated with the operating parameters.

After the concept of Lavan et al. [18], the thermodynamic properties of these type of cooling systems have been studied in more depth. Kodama et al. [19], developed a second law analysis and defined the entropy equilibrium of the system. The results of their study illustrate that the total of the entire take into accounted entropy generation completely expressed the difference between the Carnot COP and the actual COP of the unit. Van den Bulck et al. [20] presented a second law evaluation on rotary dehumidifiers. They developed the equations for entropy generation for the adiabatic flow of humid air over a solid desiccant dehumidifier. They found that the efficiency of second law of rotary desiccant wheel is a function of working parameters. Shen and Worek [21] especially focused on desiccant dehumidifier which is made of 13X molecular sieve. Using this designed rotary wheel, they investigated desiccant air conditioning system which is operated in recirculation mode based on the first and second law of thermodynamic.

Furthermore, it emphasized the importance of working conditions based on first and second law of thermodynamic in another study [22]. In this study, the researchers investigated an open desiccant air conditioning system which operates in two modes. In their study, they investigated thermal coefficient of performance (COP) and the cooling capacity according the temperature and relative humidity of ambient air. They also evaluated the reversible COP of the system with second law evaluation at ARI conditions and found that the reversible COP of the system in recirculation mode is higher than that of the ventilation mode with 3.04 and 2.63, respectively.

Kanoglu et al. [23] investigated a desiccant based cooling system which is operated in recirculation mode. They used natural zeolite material for solid rotary dehumidifier. Researchers investigated also energy and exergy approach for evaluating performance of this system. Based on this approach for their system, they derived exergy destruction and exergy efficiency for reversible COP.

La et al. [8] thoroughly investigated for every component of a desiccant cooling system. According to their investigation, they also derived convenient indicator according to thermodynamic laws. The COP concept is very important indicator, illustrating a cooling system performance and this concept mainly is the ratio of useful cooling capacity to needed energy.

In this study, we concentrated on the second law of thermodynamics, which put forward that processes occur in a certain direction and energy has quality as well as quantity. The first law of thermodynamics which assert that energy can be conserve during a process is evaluated also. According to thermodynamic rules, reversible cycles are the most efficient cycles and they consist entirely of reversible processes. The best known reversible cycle is the Carnot cycle and the Carnot COP define the upper limit of this cycle. But reversible cycles cannot be achieved in practice because the irreversibilities associated with each process cannot be eliminated. The thermal efficiency and the thermal coefficient of performance (COPth) for any device determine their performances and these parameters are specified on the basis of the first law only. A cycle that is composed entirely of reversible processes served as the model cycle to which the actual cycles can be compared. This idealized model cycle enabled us to determine the theoretical limits of performance for cyclic devices under specified conditions and to examine how the performance of actual devices suffered as a result of irreversibilities.

In this work, an experimental desiccant based evaporative air conditioning system is investigated based on the second law of thermodynamic to determine upper limit. This system has a novel configuration and firstly operation of it is described. The objective of the study is to evaluate thermal COP, reversible COP, second law efficiency and cooling capacity based on operation and ambient air condition.

2. System overview

A schematic view of the experimentally investigated system is shown in Fig. 1. The configuration presented in this figure uses 100% fresh air. The evaluated system consists mainly of the following components: one solid rotary dehumidifier, three fans, two evaporative coolers that are used as direct, one electrical heater unit, three heat exchangers, two pumps and three air ducts. Further information about working of this system can be obtained in [24]. Three air ducts are used for fresh air regeneration air and waste air. The fresh air duct is used to take fresh air from outside environment and supply it into the investigated room. The waste air channel is designed to discharge the exhaust air from the room by a waste air fan. The main purpose of the third air duct is to remove moisture from fresh air.

As described above, one of the most important components is rotary desiccant wheel in this system, due to remove moisture from fresh air.

In Fig. 1, the moisture removal process performed by dehumidifier (DW) occurs between state 2 and state 3. For the outlet of desiccant dehumidifier, the air is dehumidified but the temperature of fresh air increases also. In this system, the main cooling process is carried out by evaporative cooler 1 (EC1). For further cooling of the fresh air, the heat exchangers are installed before EC1. According to this installation, heat exchangers have a very important task that is precooling process of the fresh air.

2.1. Design methodology

The following simplifying assumptions underline the design procedure:

- All the components are perfectly insulated.
- The evaporative cooling processes are accepted isenthalpic.
- The desiccant dehumidification processes are accepted isenthalpic.

The temperature of the dry and warm fresh air is decreased first by the regeneration air in the heat exchanger 1 (HEX1) (process 3–4). After that the second precooling process is fulfilled by heat exchanger 2 (HEX2) in process 4 and process 5 by using exhaust air. Finally, the fresh air is entering into the EC1 (process 5–6) where last cooling process occurs. The fresh air cooled in these processes is supplied into the air conditioning room (state 6). In this system, the evaporative cooler 2 (EC2) is used for precooling of the fresh air by using of HEX2 (process 7–8). In this process, the exhaust air is again cooled until reaches to the saturation point. The increasing of waste air humidity of the exhaust air is not considered a trouble because this humidity is not transmitted to the fresh air. It is only used for precooling fresh air as an indirect evaporative cooling process (process 8–9). After that, it is discharged to atmosphere (process 9–10).

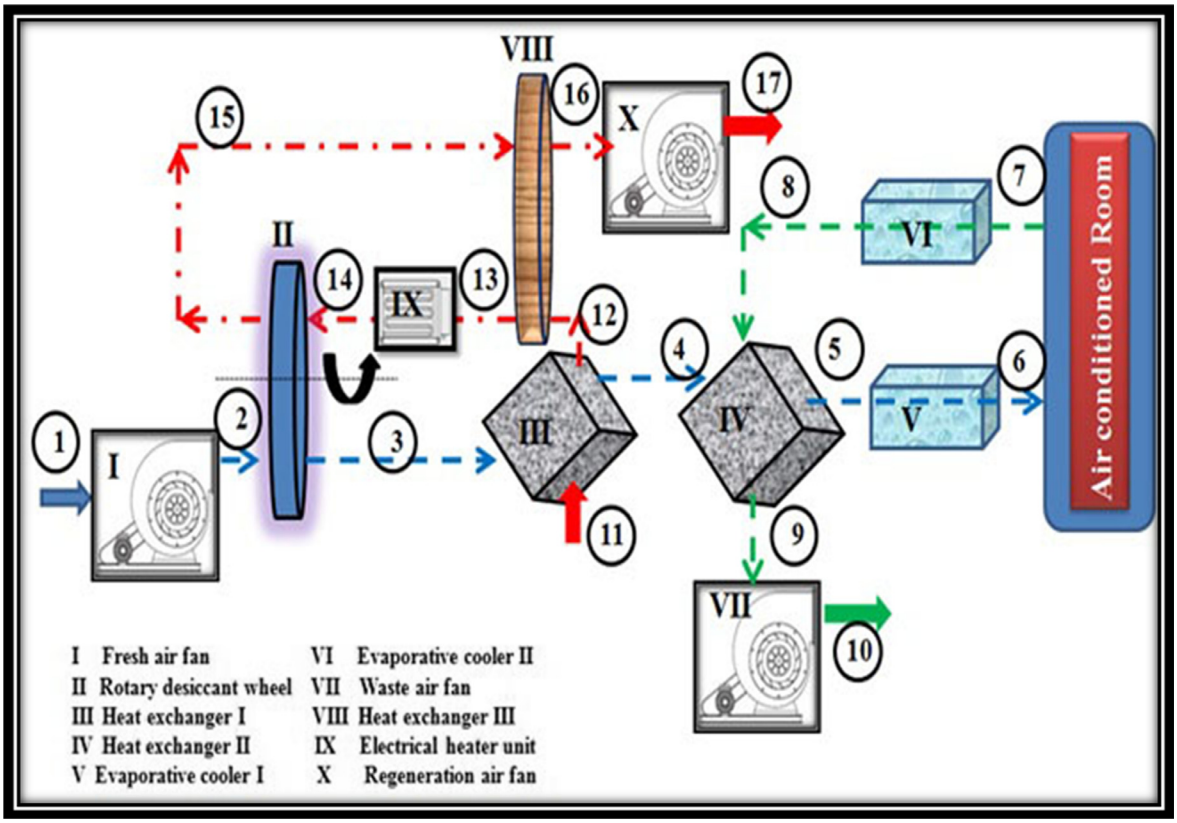


Fig. 1. Schematic view of the desiccant cooling system studied.

The regeneration air channel uses 100% fresh air, and is installed to regenerate desiccant wheel and to decrease regeneration heat by using two types of heat exchangers that are regenerative type heat exchanger and recuperative type heat exchanger.

The regeneration air stream firstly begins by the heat exchanger 1 (HEX1) in which preheating process occurs, to achieve the suitable temperature for the regeneration of the rotary dehumidifier (process 11–12). For further preheating of regeneration air a rotary heat exchanger is used (process 12–13). Moisture removal is fulfilled by heating of regeneration air. For the heating process (13–14), an electric heater is installed before the entrance of a rotary desiccant dehumidifier, which is used at the end of this air stream, removed moisture from the fresh air stream ~~is~~ and transferred to regeneration air stream (process 14–15) before, finally, being discharged to the environment (process16–17).

2.2. Second law analysis of the system

The second law of thermodynamics interest with the quality and quantity of energy and this law says that real processes continue in the direction to reduce energy [25]. To achieve the maximum COP of a desiccant air conditioning system, the total system must be fully reversible. The cooling system would be reversible if the heat from the heat source were transferred to a Carnot heat engine, and the work output of this engine is supplied to a Carnot refrigerator to remove heat from the cooled space. In analyzing real performance of these type systems, the actual COP value and Carnot COP(COP_c) must be calculated according to the second law of thermodynamics.

In this system, maximum energy consumption takes place in electrical heater unit. Consumed regeneration heat $\dot{Q}_{rh}$ is expresses by:

$$\dot{Q}_{reg} = \dot{m}_{ra} \cdot c_p \cdot (T_{14} - T_{13}) \quad (1)$$

Cooling capacity of the system $\dot{Q}_{cc}$ is defined by;

$$\dot{Q}_{cc} = \dot{m}_{fa} \cdot (h_1 - h_6) \quad (2)$$

The regeneration air is taken from outdoor that is fresh air and it is preheated by HEX1 and HEX3. After these processes, measured regeneration temperature is called T_{13} due to state 13. The last heating operation for regeneration temperature occurs in the electric heater unit. The temperature measured after this unit is called T_{14} due to state 14. Also; h_1 and h_6 are enthalpy of state 1 and state 6. These are used for calculating the cooling capacity of the system.

The thermal performance of the system is evaluated based on thermal COPth;

$$COP_{th} = \frac{\dot{Q}_{cc}}{\dot{E}_{total}} \quad (3)$$

$\dot{E}_{total}$ is the total amount of energy consummated by regeneration heat ($\dot{Q}_{reg}$), fresh, waste and regeneration fans ($\dot{W}_{fan}$), and other electrical components ($\dot{W}_{oth}$):

$$\dot{E}_{total} = \dot{Q}_{reg} + \dot{W}_{fan} + \dot{W}_{oth} \quad (4)$$

Carnot COP(COP_c) that is used for this system depends on ambient air temperature, air conditioned room and regeneration temperature. Also, it is calculated as different from vapor compression refrigeration systems [22]. The Carnot COP of our system is evaluated according to this researcher approach. They found that the COPc can be evaluated as:

$$COP_c = \left(\frac{T_{reg} - T_{ambient}}{T_{ambient} - T_{room}}\right) \cdot \left(\frac{T_{room}}{T_{reg}}\right) \quad (5)$$

where $T_{ambient}$, T_{room} and T_{reg} are the ambient air temperature, air conditioned room temperature, and the regeneration heat temperature, respectively.

According to our system shown in Fig. 1, T_1 and T_7 are ambient air temperature and cooled room temperature respectively, and the regeneration temperature T_{14} is accepted as the heat source temperature. Based on this approach, Eq. (4) can be derived as:

$$COP_c = \left(\frac{T_{14} - T_1}{T_1 - T_7}\right) \cdot \left(\frac{T_7}{T_{14}}\right) = \left(\frac{1 - T_1/T_{14}}{T_1/T_7 - 1}\right) \quad (6)$$

The second law efficiency η_{II} can be evaluated based on the ratio of the actual COP to the available maximum COP_c under the same conditions [25]:

$$\eta_{II} = \frac{COP}{COP_c} \quad (7)$$

3. Results and discussions

The desiccant based air conditioning designs need new configuration to improve performance of such system. The present work focuses on the second law equations that are applied to the new configuration of a desiccant based evaporative air conditioning system. The system has three air channels that work differently from each other. In these channel, three type air streams that are fresh, waste and regeneration are used to ensure constant volume flow rate with a value of 2000 $m^3 h^{-1}$, 3000 $m^3 h^{-1}$ and 4000 $m^3 h^{-1}$. The ratio of flow rate for all air streams is 1 for all experiments. Also, the effect of regional ambient air conditions that hot and humid ambient air is investigated according to the second law. The ratio of flow rate for all air streams is 1 for all experiments. The regeneration air temperature was regulated to 90 °C, 100 °C and 110 °C for every flow rate.

As part of the analysis, Carnot COP, cooling capacity, variations of COP, the second law efficiency with respect to the temperature, relative humidity of ambient air and relations with each other's was investigated in this study.

Different experiments were carried out on the system. Given experiment results are between the hours 09:00–18:00. Figs. 2–12 illustrate variation of the second law parameters with operation condition carried out during the day. For these figure; daily volume flow rate for fresh, waste and regeneration air streams have equal volume flow rates and is taken as 4000 $m^3 h^{-1}$. During carried out of experiments, the temperature of regeneration air is set to a constant temperature of 110 °C.

Fig. 2 illustrates the variations of the cooling capacity and relative humidity during the day. It is observed that the cooling capacity shows a reverse trend with relative humidity values of the ambient air taken from the outdoor. At the beginning of the experiment the relative humidity is higher than that of the mid-day with 75%, but the cooling capacity is lower than that of mid-day. In addition, the cooling capacity is calculated as 20 kW, but it increases to 26 kW in mid-day. It is illustrated that when relative humidity decreases toward mid-day, the cooling capacity has reached the highest value. It is also seen that when relative humidity is at 53%, the highest cooling capacity reaches approximately 26 kW, depending on operating conditions.

Fig. 3 shows variation of ambient air relative humidity and thermal COP values during the day. In above investigation, it is seen that the relative humidity affects the cooling capacity. We can also see same effect on COP values in this figure. When relative humidity value is 53% that is the lowest value of the relative humidity during the day, the COPth value are 0.62 that is the highest value of COP during the day and the COP values vary between about 0.42–0.62 that depends on operating conditions during the day.

Fig. 4 shows the variation of COPth and wet bulb temperature of ambient air ($T_{wb,ambient}$) during the day. As seen from this figure, ambient air wet bulb temperature varies between 26 and 32 °C and as it increases, the COP of the system increases at the same time. When the wet bulb temperature of the ambient air reaches to highest value, the COP also reaches the highest value with 0.63. The variation of COP_c and ambient air temperature with time during the day is shown in Fig. 5. Ambient air temperature is one of parameters effecting COP_c value. As seen from this figure, as ambient air temperature increases toward mid-day, the COP_c has reached the lowest value. It is also seen that when the ambient air temperature is 32 °C that is the highest ambient air temperature during the day, the lowest COP_c is approximately 9 which depends on operating conditions.

The variation of COP_c and relative humidity of ambient air with time during the day can be seen from Fig. 6. When we examine the equations that are given for COP_c, we see that there is not any direct relation between relative humidity and COP_c. However it can be seen from this figure that as relative humidity of ambient air decreases toward mid-day, the COP_c decreases to the lowest value. Note that we know ambient air relative humidity decreases toward mid-day while temperature of ambient air increases at the same time. We also know that temperature of ambient air has

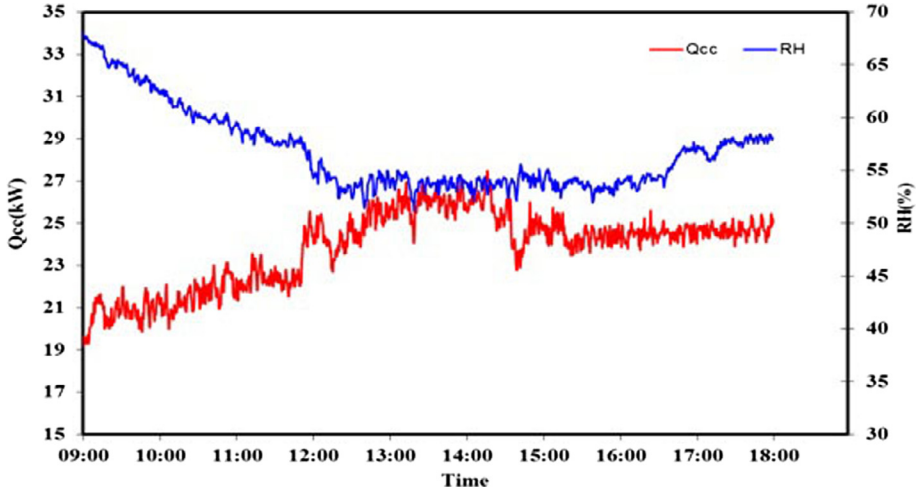


Fig. 2. Variation of cooling capacity and ambient air relative humidity during the day.

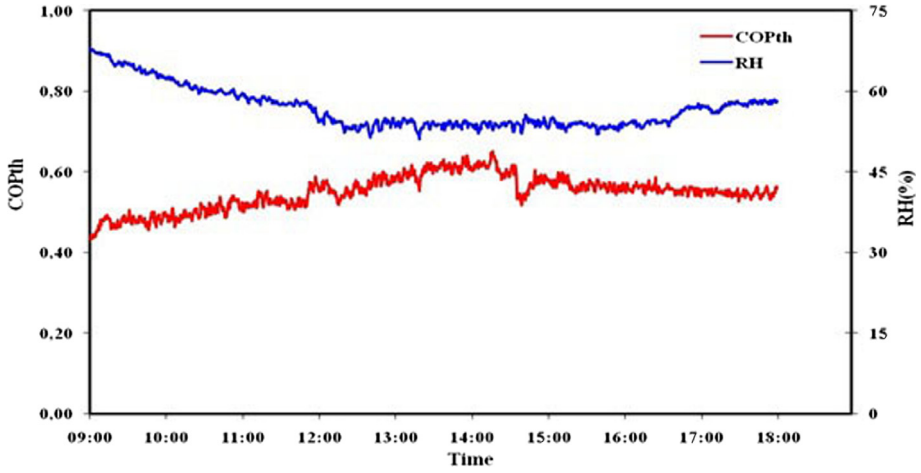


Fig. 3. Variation of COPth and ambient air relative humidity during the day.

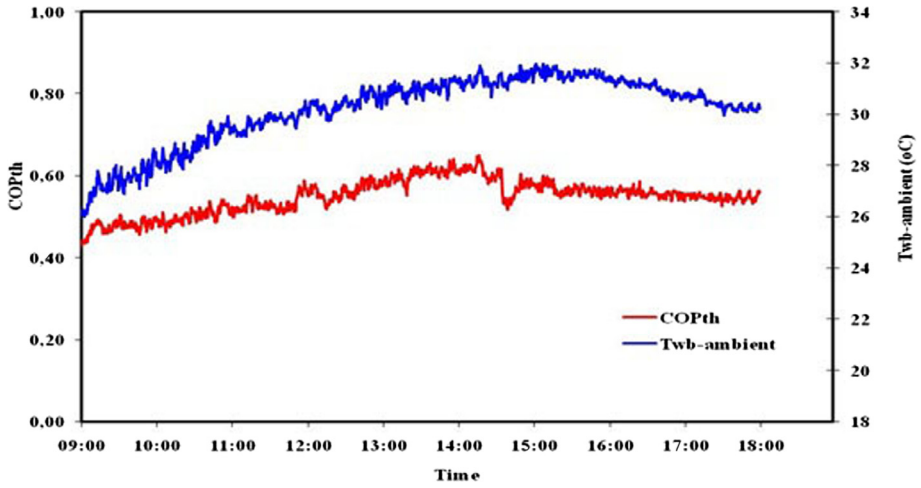


Fig. 4. Variation of COPth and wet bulb temperature of ambient air during the day.

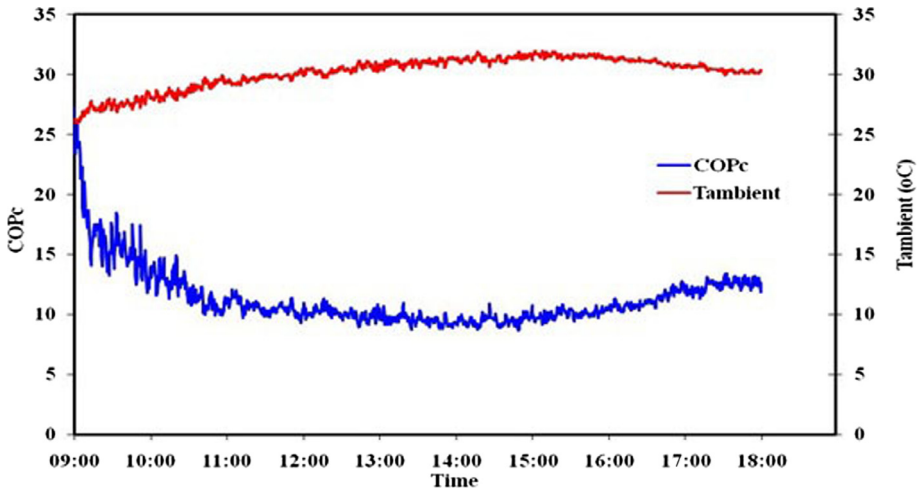


Fig. 5. Variation of COP_c and ambient air temperature during the day.

adverse effect on COP_c from Fig. 5. According to these explanation and Fig. 6, we can say that when relative humidity of ambient air decreases, the COP_c decreases at the same time.

The variation of the COP_c and the COPth during the day is shown in Fig. 7. It is seen that from this figure the COPth and COP_c tend to be vice versa to each other. As seen from this figure, the COPth values vary between 0.50and 0.62 and the COP_c values are

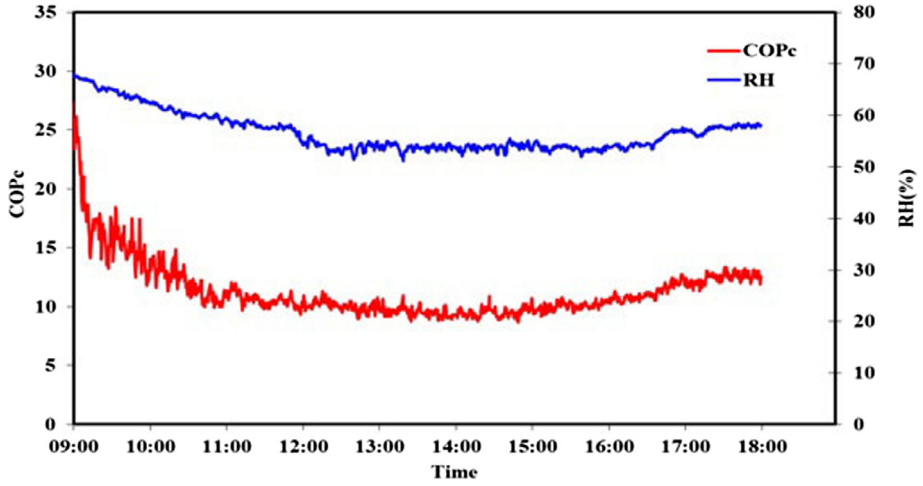


Fig. 6. Variation of COP_c and ambient air relative humidity during the day.

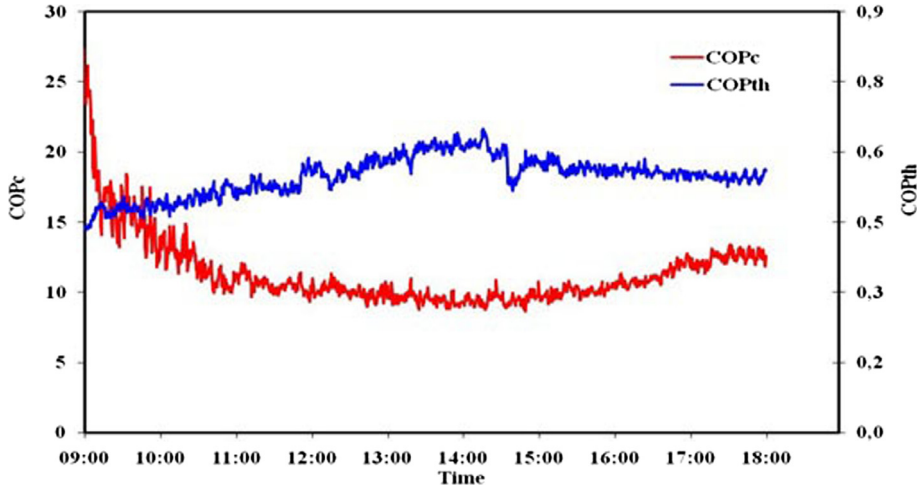


Fig. 7. Variation of the COP_c and COPth during the day.

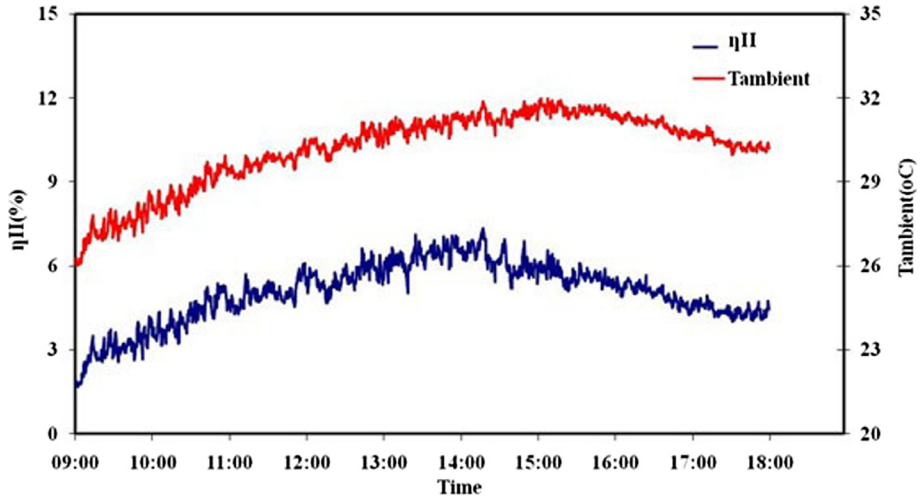


Fig. 8. Variation of the second law efficiency (η_{II}) and the ambient air temperature during the day.

between approximately 9–25. It is seen that COP_c is higher than the COPth. These differences occur due to irreversibility. If irreversibility can be decreased, then the COPth of desiccant cooling system may achieve high values. These irreversibilities occur in the desiccant wheel, heating system, rotary regenerator and evaporative cooling process.

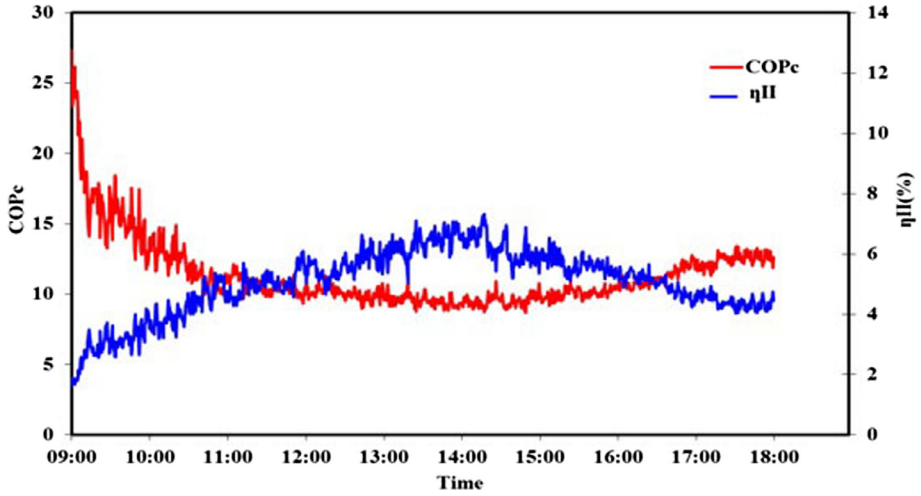


Fig. 9. Variation of COP_c and η_{II} during the day.

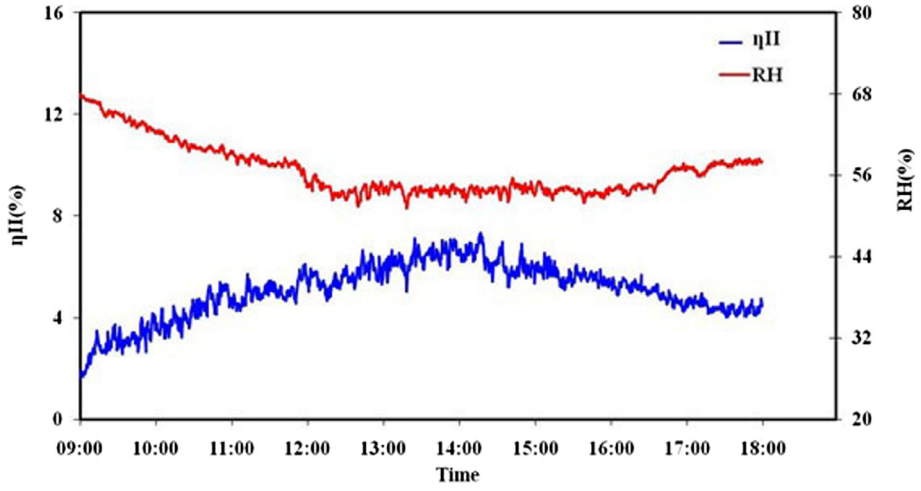


Fig. 10. The variation of the second law efficiency and the ambient air relative humidity during the day.

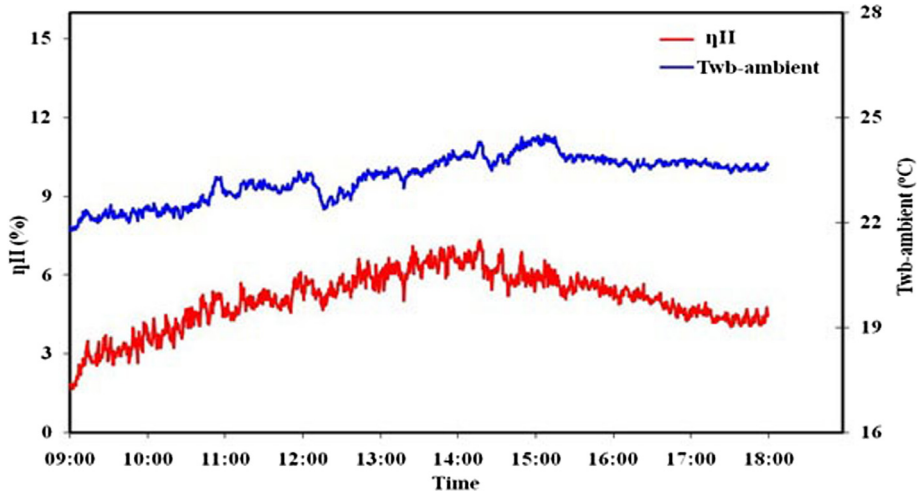


Fig. 11. Variation of the second law efficiency and wet bulb temperature of ambient air during the day.

Fig. 8 shows variation of the second law efficiency (η_{II}) and the ambient air temperature during the day. As it can be seen from this figure, with the increasing of ambient air temperature, the second law efficiency also increases at the same time. When ambient air temperature has the highest value, the second law efficiency reaches to also highest value. As seen from this figure, η_{II} values

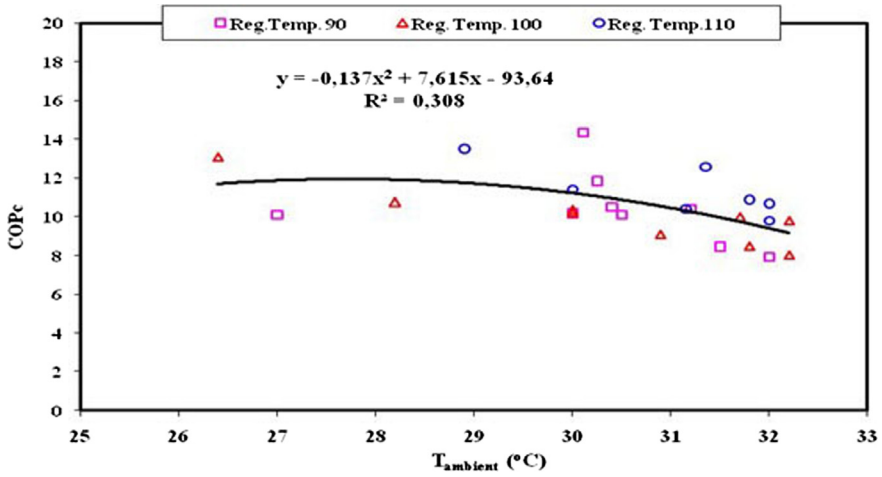


Fig. 12. Effect of ambient air temperature on COP_c for different regeneration temperature at 4000 $m^3\ h^{-1}$.

vary between 3% and 7% according to operating conditions. We can say that the second law efficiency of such systems is very low, because there is a lot of working unit that has high irreversibilities.

The variation of COP_c and the second law efficiency (η_{II}) during the day is shown in Fig. 9. The second law efficiency increases toward mid-day with 7% maximum value. When we examine this figure that are given for COP_c and second law efficiency, we see that there is an opposite relation between of them. This is due to increasing of ambient air temperature toward mid-day.

Fig. 10 shows the variation of the second law efficiency and the ambient air relative humidity with time during the day. We previous examined relation between the relative humidity and COPth in Fig. 3. It is seen that the ambient air relative humidity has adverse effect on COPth. After that we examined the COP_c and ambient air relative humidity in Fig. 6, we see that from this investigation the COP_c has same trend with relative humidity. According to this explanation given above and Fig. 10, we can say that when relative humidity decreases, then the second law efficiency increases at the same time.

In Figs. 8 and 10, it is observed that relative humidity and the temperature of ambient air varies vice versa with the second law efficiency of this system. So, the effect of the wet bulb temperature should be investigated to clearly understand with the both parameters. The variation of this parameter with the second law efficiency is illustrated in Fig. 11. It is seen from this figure that slightly increasing of temperature of the wet bulb toward midday increases also the second law efficiency at the same time. This is due to the fact the increase of the wet bulb temperature of ambient air decreases the COP, so the second law efficiency increases.

The effect of ambient air temperature ($T_{ambient}$) on daily average COP_c for different regeneration temperature set values and different flow rate of air that is equal for all air streams are shown in Fig. 12. As seen from this figure, the daily average COP_c decreases when ambient air temperature increases. It is also observed that the regeneration temperature of air has not considerable effect on COP_c. According to operating parameters in this system the daily average COP_c varies between approximately 8–14 kW. Dispersion of results with correlation equation and with R^2 (0.308) is also illustrated on this figure. The best equation is found as polynomial.

Fig. 13 is designed to clearly illustrate the effect of flow rate on COP_c at 110 °C regeneration temperature. It is seen from these figures that when the flow rate of air is increased at the same regeneration temperature, the daily average of COP_c value decreases ~~at~~ simultaneously. According to these operating parameters the maximum COP_c is obtained at the flow rate of 2000 $m^3\ h^{-1}$.

Fig. 14 shows the variation of daily average COP_c with daily average cooling capacity (Q_{cc}) for different flow rate. As seen from these figures, when the cooling capacity of the system increases, the daily average COP_c value decreases at the same time. For illustrating correlation between COP_c and Q_{cc}, an equation is founded with R^2. According to this equation R^2 value is 0.705 and this value significant for these parameters.

The variation of the daily cooling capacity with the daily average the second law efficiency (η_{II}) for different flow rates is shown in Fig. 15. As seen from this figure as Q_{cc} increases, the second law efficiency (η_{II}) increases at the same time. Also; it is observed that flow rate of air is the main operating parameters affecting this process. The slope of correlation line in this figure increases with increasing of flow rate. The best results for this correlation for R^2 value is found as 0.636 by polynomial equation.

4. Conclusion

A solid desiccant based evaporative air conditioning system has been investigated experimentally, based on the second law analysis of thermodynamic. This system includes different three air ducts. The first duct is used for fresh air to give it from outdoor and supply it into the cooling room. The waste air duct is used to recover of energy for precooling of supply air and after that it is used to discharge of the indoor air to outdoor. The third air duct is used for regeneration. In this study, the ambient air condition and operation condition on the thermal coefficient of performance (COPth), the Carnot COP, the second law efficiency and the cooling capacity is evaluated based on the second law performance of the system. The main results can be listed as follows:

- The thermal COP (COPth) increases with increasing ambient air temperature. According to operating parameters, the maximum COPth occurs at the highest ambient air temperature. The COPth of the system is almost lower than 1 under the recommended regeneration temperature.
- Daily average COP_c decreases when ambient air temperature increases. It is observed that flow rate of air has considerable effect on COP_c. When the flow rate of air is increased, COP_c decreases at the same time. According to operating parameters in this system, the daily average COP_c varies between 8 and 25 with flow rate of 4000 $m^3\ h^{-1}$ and 2000 $m^3\ h^{-1}$, respectively.

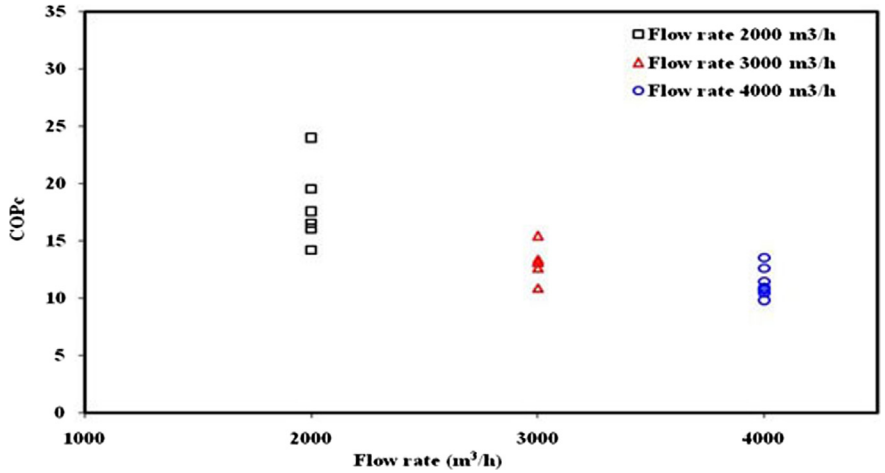


Fig. 13. Effect of different flow rate on daily average COP_c at 110 °C regeneration temperature.

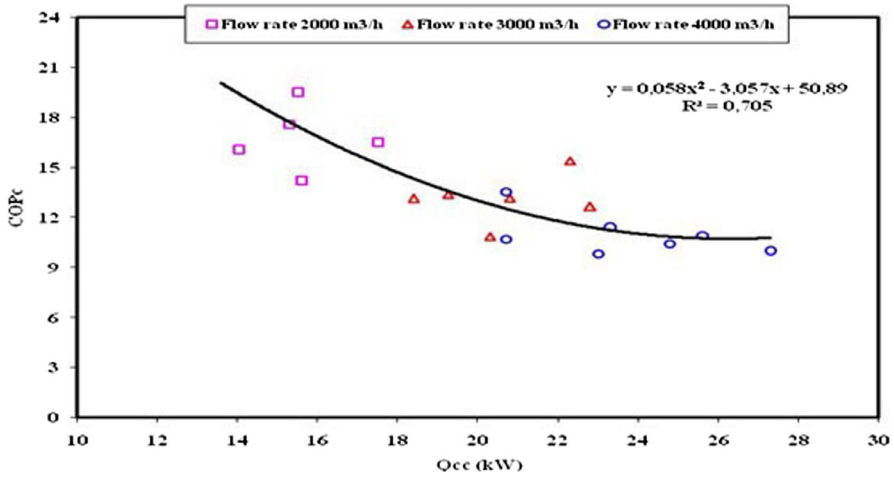


Fig. 14. Variation of daily average COP_c with daily average cooling capacity (Q_{cc}) for different flow rate at 110 °C regeneration temperature.

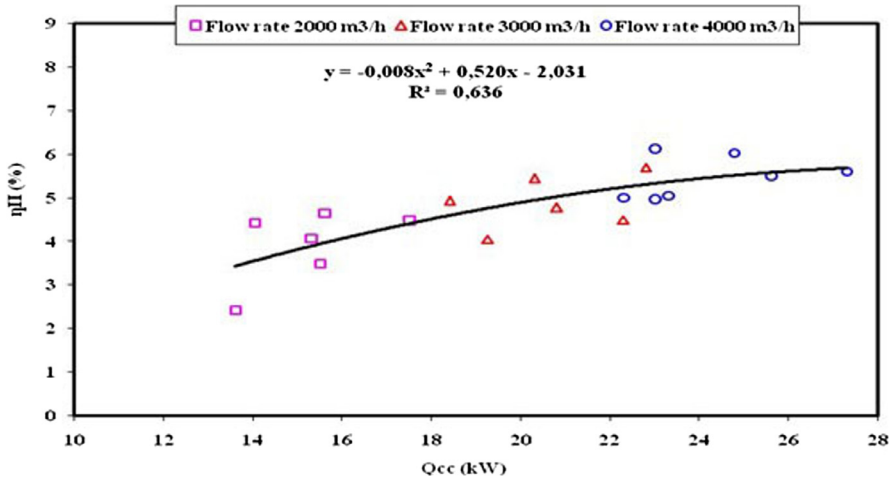


Fig. 15. Variation daily average the second law efficiency (η_{II}) with daily average cooling capacity for different flow rate at 110 °C regeneration temperature.

- According to investigated parameters, the temperature of ambient air has significant effect on the second law efficiency. Increasing of it also increases the second law efficiency (η_{II}).
- It is observed that there is direct relation between the cooling capacity and the second law efficiency. When the cooling capacity increased, the second law efficiency increases at the same time.

- One of important operation condition is flow rate of air that affects the second law efficiency. It is seen that increasing of flow rate, increases also the second law efficiency.

Calculations show that the thermal coefficient of performance COPth, the COPc, the second law efficiency and the cooling capacity strongly depends on the ambient air conditions.

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