**Energy efficient air conditioning and fresh water production hybrid system for low-humidity buildings applications**

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**Abstract**

The air conditioning systems of low humidity applications consume huge amount of energy to maintain the required low temperature and humidity needed by the application. In the present work, proposed efficient systems that integrate the technologies of desiccant dehumidification, evaporative cooling, and fresh water production with the air conditioning system of low-humidity applications are presented. Thermodynamics analysis of the systems is conducted and the systems of equations are analytically solved and analyzed using EES and mat lab software. Parametric studies are conducted to investigate the effects of the different operating parameters on the systems performance. The power savings and the fresh water productions of the different proposed systems are evaluated and compared to select the optimum system configuration.

***Keywords***

Energy saving, Hybrid systems, Air Conditioning, Desiccant wheel, HDH system.

1. **Introduction**

A large part of the energy in the air-conditioning system of low-humidity applications like pharmaceutical applications is consumed for the removal of moisture and maintaining the required low humidity and temperature. A desiccant dehumidification is using desiccant wheel is typically used in this system to remove the moisture from the processing air before it is mechanically refrigerated. In the desiccant wheel, the process air to be dehumidified passes through the adsorption section of the wheel where the water vapor in the process air is directly removed from the air and holds in the wheel while rotating. As the desiccant section saturated with water vapor passes through the regeneration sector, the water vapor is transferred to a heated regenerative airstream, which is exhausted to the outside. A lot of researches were conducted to study the effects of the operating and geometric conditions on the desiccant wheel performance [1-5].

Combing the desiccant wheel system with the air conditioning (AC) system of some applications gained the interest of many designers and researchers. Zendehboudi et al. [[6]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CPerformance%20assessment%20of%20a%20Solid%20Desiccant%20Based%20on%20Dehumidifier.pdf) showed that using a desiccant wheel coupled with AC system leads to energy saving and higher system efficiency. Khoukhi [[7]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5C3-%20A%20Study%20of%20Desiccant-Based%20Cooling%20and%20Dehumidifying.pdf) reported that DCS combined with heat wheel and a heat source is a perfect solution for hot- humid climate. Srivastav [[8]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CIRJET-V3I480.pdf) reported that combined desiccant-cooling systems lead to the advantages of energy and cost savings for different climatic conditions. Camargo and Ebinuma, [[9]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CAn%20Evaporative%20and%20Desiccant%20cooling%20system.pdf) studied the effects of the different operating parameters of the evaporative-cooling and desiccant-dehumidification proposed hybrid system for humid climates. Yıldırım and Solmus [10] conducted theoretical investigation of the performance of a solar powered HDD-desiccant system at different operating and climatological conditions.

Recently, different hybrid systems were proposed for air conditioning and fresh water productions systems for different climate conditions. Huzayyin et al. [11] experimentally studied the performance of a cooling and dehumidifying coil under different wetting conditions of the coil. Nada et al. [[12]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CExper.%20hybrid%20water%20desinlation%20%26%20AC%2C%20Deslination%202015.pdf)conducted an experimental study of the performance of hybrid AC and HDH water desalination system using mechanical chillers and steam humidification. Nada et al. [[13]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CSolar%20Deslantion%20and%20Air%20conditioning-%20Deslanation.pdf) studied the effects of the operating parameters and conditions on the performance of proposed four arrangements of hybrid AC, HDH, and energy recovery systems for power and water savings. Elattar et al. [14] and Fouda et al. [15] presented analytical studies of the performance of solar assisted AC and HDH hybrid systems for power and water saving. Milani et al. [[16]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CExperimentally%20validated%20model%20for%20atmospheric%20water%20generation.pdf) conducted an experimental study using a prototype dehumidification system for freshwater production from the atmosphere air using a small-scale air-cooled desiccant wheel dehumidifier. He et al. [17] Studied the performance of a HDH desalination system integrated to plate-heat exchangers to produce fresh water utilizing the waste heat from the exhaust gas. Gang et al. [18] experimentally investigated the performance of a hybrid heat recovery with tandem HDH system using the principle of the multi-effect solar HDH system.

Hafiz and Fahad [19] studied the effect of the operating conditions on the performance of HDH desalination system based on a multistage stepped bubble column humidifier. The operating condition for the optimum performance was reported. Fahad et al. [20] presented a thermodynamic analysis of two different systems of a hybrid open-air open-water HDH system integrated with a parabolic-trough solar-collector. Sharshir et al. [21] experimentally investigated the performance of a continuous HDH designation system integrated with evacuated tube solar collector and solar still. [Ameri](https://www.sciencedirect.com/science/article/pii/S1359431116305336#!) and [Eshaghi](https://www.sciencedirect.com/science/article/pii/S1359431116305336#!) [22] presented thermodynamic modeling and energy analysis to study the performance of a reverse osmosis HDH system integrated with and flat plate solar collector.

The literature review revealed that the reduction of the huge energy consumption of AC system of low humidity application and the utilization of the vapor removed in fresh water production using hybrid AC HDH have not been proposed yet. In the current study, different configurations of hybrid AC, desiccant, and HDH systems are proposed for the reduction of electric consumption and the production of fresh water are proposed for the air conditioning and desiccant system of low-humidity applications. A parametric analysis and comparative study of the proposed systems are also presented for optimum selection.

**2. Systems description**

Air conditioning, desiccant and HDH hybrid systems are proposed for low humidity application for energy saving and fresh water production. The first system (named as system1) is a basic air conditioning system of low-humidity applications that can also be utilized for fresh water production. The other two systems, system 2 and system 3, are a modification of basic system with the integration of a heat recovery in the system downstream (system 2) and upstream (system 3) the desiccant wheel. The Schematic diagram of the three system are shown in Fig 1. For the sake of comparison, the three systems are assumed to serve a space of a 100 TR cooling capacity a room sensible heat factor (RSHF) of 0.7 and is required to be maintained at 22 °C temperature and 30% relative humidity.

The systems consists of air cooled chiller unit to serve the cooling coil to cool the air, desiccant wheel to keep the low humidity in the served space, heating coil to produce the hot air required to the regeneration process of the desiccant wheel, and HDH unit consists of humidifier and water cooling coil for fresh water production from the humid hot air that exit the desiccant wheel. In systems 2 and 3 a heat recover is used for energy recovery by cooling the process air by the cold air exhausted from the air condition space. The thermodynamic processes that occur on the air in the three systems are on the Psychometric chart in Fig. 2.

**3. Thermodynamic analysis and mathematical modeling**

The following assumptions are considered during driving and solution of the mathematical model:

* Thermal/regeneration efficiencies of desiccant wheel are taken as 20 % [[3]](file:///E%3A%5CDesiccant%20wheel%5CPaper%20File%5CIntroduction%20of%20a%20new%20definition%20for%20effectiveness%20of%20desiccant%20wheels.pdf).
* Heat recovery efficiency is taken 60 % as per typical values of suppliers [23].
* Air and water leakages in the system are neglected.
* The air leaves the humidifier at wet-bulb temperature equal to the leaving water temperature.
* The air humidification in the humidifier occurs until saturation of air
* The dehumidification process in the water coil occurs on the saturation curve.
* Any auxiliary power in the system is negligible as compared to the AC power.
* The indoor design conditioned of the production hall is taken to be TR = 22 °C and RHR = 30% as per the practical pharmaceutical applications.
* The supply cooling air to the production hall is at 12 °C.
* The cooling capacity and the room sensible heat factor (RSHF) are 100 TR and 0.7, respectively.

The amount of water condensate, cooling coil capacity, process and reactivation air flow rates are obtained from the thermodynamic analysis of mixing section, air and water cooling coil, conditioned space, and desiccant wheel, as follows:

|  |  |
| --- | --- |
| $$m\_{w1}^{°}= m\_{s}^{°}(ω\_{2}-ω\_{s})$$ | (1) |
| $$m\_{w2}^{°}= m\_{Reg}^{°}(ω\_{6}-ω\_{7})$$ | (2) |
| $$m\_{w}^{°}=m\_{w1}^{°}+m\_{w2}^{°}\_{}^{}$$ | (3) |
| $$m\_{s}^{°}=\frac{ Q\_{R}^{°}}{(h\_{R}-h\_{s})}$$ | (4) |
| $$RR=\frac{m\_{Reg}^{°}}{m\_{s}^{°}}$$ | (5) |
| $$Q\_{CC}^{o}= m\_{s}^{°}(h\_{2}-h\_{s})$$ | (6) |

The specific humilities and enthalpies in Eqs. (1)-(6) are obtained from air properties and desiccant wheel relations as follows:

|  |  |
| --- | --- |
| $$\frac{m\_{o}^{°}}{m\_{s}^{°}}=\frac{T\_{R}-T\_{1}}{T\_{1}-T\_{o}}$$ | (7) |
| $$ε\_{Dw1}=\frac{T\_{2}-T\_{1}}{T\_{4}-T\_{1}}$$ | (8) |
| $$ε\_{Dw2}=\frac{m\_{s}^{°}(ω\_{1}-ω\_{2})h\_{fg}}{m\_{Reg}^{°}(h\_{4}-h\_{3})}$$ | (9) |
| $$ε\_{HR}=\frac{m\_{s}^{°}(T\_{o.HR}-T\_{i,HR})}{m\_{o}^{°}(T\_{Room}-T\_{i,HR})}$$ | (10) |
| $$T\_{5}=T\_{4}-(\frac{T\_{2}-T\_{1}}{RR})$$ | (11) |
| $$ω\_{5}=ω\_{4}+(\frac{ω\_{1}-ω\_{2}}{RR})$$ | (12) |

The COP of the air conditioning system is calculated from:

|  |  |
| --- | --- |
| $$COP=\frac{ Q\_{cc}^{°}}{ W\_{c}^{°}}$$ | (13) |

It is well known that the COP (coefficient of performance) of the air cooled chillers is function in the ambient air temperature. The COP correlation in terms of the outdoor temperature was previously expressed by [24]:

COP=11.1-0.4To+7.2×10-3 To2 -5.18×10-5 T o3 (14)

The total power consumed in the system and the power saving and water saving percentages in system 2 and system 3 due to incorporating the energy recovery system can be found from

$E°= W\_{C}^{°}+E\_{pump}^{°}$ (15)

$Power saving \%=\frac{E\_{1}^{°}-E\_{PS 2,3}^{°}}{E\_{1}^{°}}$ 100 % (16)

$Fresh water saving \%=\frac{m\_{wS2,3}^{°}-m\_{ws1}^{°}}{m\_{ws1}^{°}}$ 100 % (17)

The above systems of equations are solved using EES and mat lab software to find the water production rate (m◦w), air condition cooling capacity and power consumption and percentage of power and water saving as a function of the operating and ambient conditions. The operating and ambient conditions are assumed to vary in the following given ranges:

* Fresh air ratio: 0-1
* Ambient air temperature: 30-50 ◦C
* Ambient air specific humidity:0.01-0.03Kgv/ Kga

**4. Results and discussions**

**4.1 The effects of outdoor air ratio on the systems performances**

Figure 3 shows the variation of m◦w , E◦, Q◦cc and percentage of power saving with the outdoor air ratio (m◦o/ m◦s). Fig 3-a shows that for all the studied systems, the rate of the obtained water from the systems m◦w increases with the increase of the outdoor air ratio for the entire ranges of the studied parameters. This may be attributed to the following reasons, (a) increasing the fresh air ratio makes state (1) on the Psychometric chart (see Fig. 2, system 1) moves along the line O-R to be more close to state O. Accordingly, the humidity of air at the inlet of the cooling coil rises as a result of moving state (2) upwards on the Psychometric chart. This increases the condensation rate on the cooling coil (m◦w1) as per Eq. (1), and (b) the increase of water vapor removed by the desiccant wheel $(ω\_{2}-ω\_{s})$ which leads to the increase of the humidity of the reactivation at desiccant wheels exit (state 5) and consequently rising the reactivation air humidity at the water cooling coil inlet (state 6) leading to more vapor condensation and water rate m◦w2 (see Eq. 2). It can be noted from Fig. 3-a that the rate of water obtained for system-1 and system-2 increase with the increase of the outdoor air ratio by the same ratio and this makes the trends of the two lines coincide with each other. However, for system-3, the rate of increase in the water rate is smaller than those of systems-1 and 2, and this is due to the decrease of $(ω\_{2}-ω\_{s})$ of system (3) as a result of the decrease of $(ω\_{2})$ due to the free cooling of the outdoor air in the energy recovery.

Figure 3-b shows that the cooling capacity of the air conditioning system Q◦cc increases with increasing the outdoor air ratio for the three systems. The reason of that is the increase of the temperature and humidity of the air (i.e. the increase of the air enthalpy) at cooling coil inlet with the increase of the outdoor air ratio (refeer to Eq. (6)). Figure 3-b also shows that using the heat recovery in the system reduces the coil cooling capacity where the coil capacity of systems-2 and 3 are lower than that of system-1 at any outdoor air ratio. This is attributed to the free cooling of the fresh air by cooling recovery from the exhaust air. Figure 3-b also reveals that the cooling capacity of system-2 is lower than that of system-3 for any outdoor air ratio. This proves that locating the heat recovery after the desiccant wheel as in system-2 is more effective than locating it at the entrance of the fresh air intake as in case of system-3.

Figure 3-c shows that the variation of the total electrical power consumption E◦ with the fresh air has the same trend of the variation of the cooling coil capacity. This is attributed to the linear relation between the electric power consumption and the cooling coil capacity as given by Eqs (13-15).

Figure 3-d gives the dependence of the power saving due to using heat recovery in system-2 and system-3 on the fresh air ratio. The figure shows the increase in the percentage of the power saving with the increase of the outdoor air ratio. The reason of this is the decrease of the hot stream air enthalpy at the inlet to the heat recovery and the increase of the free cooling that can be recovered. Figure 3-d also shows that the power saving of system-2 is always higher than that of system-3 for any outdoor air ratio indicating that locating the heat recovery after the desiccant wheel increases its capacity for the cooling recovery.

**4.2** **The effects of ambient air temperature on the systems performances**

Figure 4 gives the variation of m◦w , E◦, Q◦cc and power saving with the ambient air temperature To. Figure 4-a illustrates the decrease of m◦w with the increase of the ambient air temperature for the three systems. The reason of that is the decrease of the process air humidity at the entrance of the cooling coil (ω2) and the increase of the wet bulb temperature of the outdoor air with increasing its dry blub temperature leading to the increase of ω7 . Decreasing ω2 and increasing ω7 cause the decrease of the water production rates as given by Eq. 1-3, respectively.

Figure 4-b shows the increase of Q◦cc with the increase of To which may be attributed to the increase of the enthalpy of the air at the inlet of the cooling coil (h2) as indicated by the Psychometric charts that are shown in Fig. 2. Increasing the cooling coil inlet air enthalpy causes the increase of the cooling coil capacity as per Eq. 6. Fig. 4-b shows that the coil cooling capacity of system 2 is lower than that of systems 1 and 3 and the that of system 3 is lower than that of system-1.

Figure 4-c shows the variation E◦ with the ambient air temperature To. As shown in the figure, E◦ dramatically increases with increasing To. The reason of that is the increase of the cooling coil capacity Q◦cc, and the decrease of the COP of the air cooled chiller with increasing the outdoor air temperature (see Eqs. 13-15).

Figure 4-d shows the variation of power saving with ambient air temperature. As shown in the figure, the power saving in system-2 and 3 are approximately constants with To. This indicates that the energy consumptions of the three systems increase with the same rates with increasing the outdoor air temperature. This is confirmed by the same slope of the three lines in Fig. 4-c. Figure 4-d, confirms that system-2 has higher power saving compared with system-3 for all outdoor air temperature and outdoor air ratio.

**4.3** **Effects of ambient specific humidity ratio**

Figure 5 shows the variation of m◦w, E◦, Q◦cc and the percentage of power saving against ambient specific humidity (ωo). Figure 5-a shows the increase of the water production rate with increasing the specific humidity of the outdoor air for the three systems. The reasons of that is the increase both of the humidity of the process air at the entrance of the cooling coil (ω2) and the air at inlet of the water cooled dehumidifier ω2. This leads to the increase of the water production rate (see Eq. 1-3.).

Figure 5-b shows the increase of Q◦cc with increasing the ambient humidity ωo . The reason of that is the increase of the enthalpy of the air (latent part) at the inlet of the cooling coil (h2) with increasing the outdoor specific humidity as shown on the Psychometric charts given in Fig. 2. Increasing the enthalpy of air that enters the cooling coil leads to the increase of the cooling capacity as given by Eq. 6.

Figure 5-c shows the increase of E◦ with increasing ωo. The reason of that is the increase of the compressor power due to the increase of Q◦cc .

Figure 5-d shows that the power saving in system-2 and 3 slightly decrease with increasing the specific humidity of the ambient air ωo. The figure also shows that system-2 has higher power saving percentage comparable with system-3.

**4.4. Systems comparison and cost analysis**

The total saving in cost TSC ($/h) due to power saving and fresh-water productions of systems-2 and 3 can be estimated from:

 (17)

Typical values of potable water and electricity unit rate 0.02 $/ kWh and 2.5 $/m3 in Gulf cities [21] are considered in this study. The TSC for systems -2 and 3 at different operating parameters are shown in Fig. 6. Figure 6-a shows the increase of TSC of systems-2 with the increase in the outdoor air ratio. The reason of that is clear from Figure 3 where both of the fresh water production rate and the power saving for system-2 and 3 increases with the increase of fresh air ratio. Figure 6-b shows that for systems 2 and 3, the total saving in costs slightly decreases with the increase of the outdoor temperature. The results show that the trend is the same at all operating parameters. Decreasing TSC with the outdoor air temperature reveals that the decrease of the cost saving due to decrease of the fresh water production rate with increasing of the outdoor temperature (see Fig. 4-a) overcomes on the increase of the cost saving due to the increase of the power saving with the outdoor temperature (See Fig. 4-d). Figure 6-c shows the increase of the total cost saving with increasing the outdoor air humidity. The reason of that is the increase of the fresh water production rate and the power saving with the increase of the humidity ratio as shown in Fig. 5.

Comparing between the total cost saving of systems-2 and 3 that are shown in Fig. 8 reports that for all the operating parameters the cost saving of system 2 is higher than that of system 3. This can be attributed to that system 2 always has higher water production rate and power saving as compared to system-3 as shown in Fig. 3-5.

**5. Conclusion**

Three proposed systems for integrative air conditioning (air cooled chiller unit) and desiccant cooling system (DCS) and fresh water production for low humidity applications are presented, investigated, evaluated and compared. The effects of the fresh air ratio, outdoor air temperature and humidity on the performance of the proposed systems are investigated. The results reveal that (i) for the proposed systems, the fresh water production rate and the cooling coil capacity of the air conditioning system increase with increasing the fresh air mixing ratio and outdoor humidity and decrease with increasing the ambient air temperature and (ii) the saving in power consumption due to using energy recovery in system 2 and 3 remarkably noticed with increasing the outdoor air ratio, and (iii) the system with energy recovery located after the desiccant wheel is operating more efficiently and economically compared to the other systems.

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**List of Figure Captions**

Fig. 1: Schematic diagrams of the proposed systems

Fig. 2: Psychometric diagrams of the process air in the proposed systems

Fig. 3: Effect of fresh air ratio on (a) m◦w, (b) Q◦cc, (c) E◦, and (d) percentage of power saving

Fig. 4: Effect of ambient air temperature on: (a) m◦w, (b) Q◦cc, (c) E◦, and (d) percentage of power saving

Fig. 5: Effect of ambient specific humidity on (a) m◦w, (b) Q◦cc, (c) E◦, and (d) %power saving

Fig. 6: Effect of the operating parameters on the total cost savings of systems 2 and 3

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| **Nomenclature** |
| $$m^{o}$$ | Mass flow rate, kg/sec |  | $$Q\_{CC}^{o}$$ | Cooling Coil Capacity, kW |
| **T** | Temperature, oC |  | $$Q\_{HC}^{o}$$ | Heating Coil Capacity, kW |
| **h** | Specific enthalpy, kj/ kg |  | $$E^{o}$$ | Electric power consumption, kW |
| **ω** | Air specific humidity, kgv/ kga |  | $$W\_{C}^{o}$$ | Compressor power, kW |
| **RR** | Reactivation Air Ratio |  | **Hfg**  | Heat of water evaporation, kj/kg |
| **TR** | Tone of refrigeration |  | **Cp**  | Specific heat, kJ/kg·k |
| **RSHF** | Room Sensible Heat Factor  |  | **AC** | **Air conditioning**  |
| **DW** | Desiccant Wheel |  | **DSC** | **Desiccant** |
| **RH** | Relative Humidity |  | **CC** | **Cooling coil** |
| **HR** | Heat Recovery |  | **COP** | **Coefficient of performance** |
| **Greek symbols** | **Subscript** |
| $$ε\_{Dw1}$$ | Desiccant wheel thermal efficiency |  | **I** | Inlet  |
| $$ε\_{Dw2}$$ | Desiccant wheel Regeneration efficiency |  | **P** | Process air |
| $$ε\_{HR}$$ | Heat recovery efficiency |  | **DW** | Desiccant wheel  |
|  |  |  | **O** | Outlet/outdoor |
|  |  |  | **Reg** | Regeneration  |
|  |  |  | **HR** | Heat recovery |
|  |  |  | **R** | Room condition |
|  |  |  | **S** | Supply |
|  |  |  | **w** | Water |
|  |  |  | **c** | Compressor |