Solid Desiccant Air Conditioning Systems Using Maisotsenko Cooling Cycle

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ABSTRACT

Air conditioning systems are responsible for a gigantic portion of energy consumption in the UAE. Specially, UAE's temperature might reach to extremely high temperatures of 50°C during summer seasons. Fortunately, hot summers are coincided with high solar insolation periods and high peak demands. Therefore, it is wise to utilize solar energy for air conditioning systems to reduce the electricity consumption associated with the needs of air conditioning operation. Solid desiccant air conditioning systems are capable of exploiting low grade sources of energy such as solar for their operation. Direct and indirect evaporative cooling are the most commonly used cooling methods implemented in solid desiccant air conditioning systems. The cooling best performance of these types of cooling methods is limited by the air wet bulb temperature and relative humidity. On the other hand, Maisotsenko cooling cycle is a recently commercialized cooling system capable of cooling air to its dew point temperature. Coupling Maisotsenko cooler with a solid desiccant enables the suggested cooling systems to be employed in humid conditions such as in UAE.

In this paper, two new air conditioning systems combining solid desiccant and Maisotsenko cooler are proposed and investigated for humid climates such as in UAE. Solar energy is selected as the primary source of energy to operate the desiccant air cooling systems. Detailed thermodynamic analysis is performed to illustrate and evaluate the proposed air conditioning systems performance in the UAE under different operating conditions. A comparative analysis is carried out to signify the advantages and disadvantages of integrating the Maisotsenko cooler with conventional air condition systems. Furthermore, a transient analysis is executed to examine the performance of the proposed systems throughout a day in summer for Abu Dhabi weather data. The results demonstrate the benefits of integrating Maisotsenko coolers with solid desiccants as a considerably efficient air conditioning system. Finally, Transient analysis illustrates that both MDVC and MDRC are capable of efficient operation in Abu Dhabi worst weather conditions without any difficulties.

Keywords: *Solar air conditioning systems; solid desiccant* cooling; *Maisotsenko cooling cycle; system optimization*

1. INTRODUCTION

World's energy demand is rising with an exceptional rate such that 35% increase is estimated from 2010 to 2035 [1]. Growth of energy consumption can only result in higher greenhouse gases emission which is the main cause of global warming. Concentrating on renewable energy is proposed as the most promising solution to satisfy the higher energy demand as well as reducing greenhouse gases emission. Air conditioning is responsible for a gigantic portion of energy consumption in the UAE. Specially, UAE's extremely high temperatures during summers can only amplify the energy consumption associated with air conditioning. Fortunately, hot summers are coincided with high solar insolation periods. Therefore, it is wise to utilize solar energy for air conditioning to dampen the electricity consumption associated with air conditioning.

Desiccant air conditioning low operating temperature enables it to be integrated with a solar collector as its main source of energy. Desiccant mainly functions on the vapor pressure basis. There are two different technologies for desiccant air condition including solid and liquid. Liquid desiccant consist of an absorber and a regenerator sections [1]. In the absorber, absorbent is sprayed into humid air mixture absorbing portion of the moisture. Because of the water addition, liquid desiccant is required to be regenerated. Thus, portion of the absorbent is constantly pumped into the regenerator to reduce its water content and be regenerated for further employment. Moreover, liquid desiccant's partial vapor pressure must remain greater than the air's to preserve its capability of water removal [1]. Lithium chloride and calcium chloride are the most commonly used adsorbents in Liquid desiccants. One drawback for liquid desiccant is that a portion of absorbent might be carried away by the air stream through absorption and regeneration processes [2]. On the other hand, liquid desiccants are better dehumidifiers and require lower regeneration temperature [2].

Solid desiccant air conditioning is a compelling alternative to vapor compression systems [3]. Solid desiccant functioning depends on partial vapor pressure difference between its surface and air passing through it.

Silica gel, alumina silicate, and zeolite are the most commonly employed materials for solid desiccants [4]. The main problem with solid desiccant is the air pressure drop in result of air dehumidification [4]. Nevertheless, solid desiccants are easier to operate compare to liquid desiccants [4]. Solid desiccants are available in forms of stationary or rotary wheels as well as fixed bed, cross flow bed and belt [4]. Rotary wheels can operate without any disruption which makes them more appropriate for air conditioning purposes.

Six different solar desiccant cooling systems are studied in [5] employing TRNSYS for the simulation requirements. All configurations integrated solid desiccant with a cooling technique for sensible cooling purposes. In another research work, artificial neural network is used in [6] to assess the performance of a desiccant cooling system. An experimental analysis of a two stage solid desiccant cooling coupled with a vapor compression cycle is accomplished in [7]. A detailed energy and economic assessment of different desiccant cooling systems is performed by TRNSYS and the results are presented in [8]. Solid desiccant integration with evaporative cooler, heat pump, and enthalpy wheel are investigated with solar air collector as the primary source of energy. The results indicate heat pump configuration better energy saving compared to the other configurations. A solid desiccant cooling system utilizing photovoltaic thermal collector coupled with a vapor compression cycle for Abu Dhabi is studied in [9]. Using TRNSYS, the simulation results showed that coefficient of performance (*COP)* for the proposed cycle was 0.68.

Direct and indirect evaporative cooling are the most commonly used cooling techniques implemented in solid desiccant air conditioning systems. These types of cooling technique best performance are limited to the air wet bulb temperature. Maisotsenko cooling cycle is a newly commercialized cooling technique. Coupling Maisotsenko cooler with a solid desiccant enables it to be employed in humid conditions such as UAE. The number of investigation conducted on Maisotsenko cooling cycles are insufficient at the time of writing; nonetheless, Maisotsenko coolers might be a breakthrough in desiccant air conditioning technology because of its capability to cool down air to its dew point temperature. Evaluation of Maisotsenko cooling cycle is presented in [10]. The author who is one of the prominent researchers in this cooling system also proposed the implementation of Maisotsenko cooling cycle in desiccant air conditioning. Energy and exergy analysis of Maisotsenko cooling cycle is performed in [11]. Maximum exergy efficiency is reported to be 19.14% for a reference temperature of 23.88℃ which corresponds to the optimum operating condition. In addition, calculated energy *COP* is greater than exergy *COP* for this cooling cycle.

In this paper, two new air conditioning systems combining solid desiccant and Maisotsenko cooler are proposed and investigated for humid climates such as UAE. Solar energy is selected as the primary source of energy for the air conditioning operation. Thermodynamic analysis is executed to illustrate the proposed air conditioning systems performance in the UAE under different operating conditions. A comparative analysis is performed to signify the advantages and disadvantages of Maisotsenko cooler integration by comparing the performance of the proposed cycle configurations with studied cycle in [3]. Furthermore, a transient analysis is executed to examine the performance of proposed systems throughout a day in summer for Abu Dhabi. It is important to bear in mind that the main objective of this research work is to investigate the potential of these air conditioning units. Developing a solar model is not in the scope of this work.

2. SYSTEM CONFIGURATION

There are two configurations proposed in this paper integrating solid desiccant air conditioning with Maisotsenko coolers. Referring to Figure 1, the schematic diagram and the psychometric chart for Maisotsenko desiccant ventilation cycle (MDVC) is presented. In the lower air stream, Exhaust air from the conditioned space at state 10 is mixed with ambient air at state 1 before entering the product channel of the Maisotsenko cooler at state 5. Ambient air goes through the working dry channel of the Maisotsenko cooler where it will be cooled down to its dew point temperature. Afterward, same air stream enters the working wet channel as it is heated up by the air stream in the dry channels and the product channel of the Maisotsenko cooler. Air at state 5 passes through the product channel of the Maisotsenko cooler and it is cooled to the dew point temperature of the air in the working channel to state 6. Air is heated up in an air to air heat exchanger to recover a portion of the heat from the upper air stream at state 7. Air is heated in a solar heater to state 8. Heated air goes through the desiccant wheel for its regeneration requirement at state 9 and it is released to the atmosphere. In the upper stream, ambient humid air goes through the desiccant wheel where its moisture content is reduced from state 1 to state 2. Next, hot and dehumidified air at state 2 enters the air to air heat exchanger where it is cooled down by exchanging heat with the lower air stream at state 3. In the last stage of MDVC, the flowing air is divided into two equal streams and enter the upper Maisotsenko cooler where it is cooled down to its dew point temperature at state 4 before arriving at the conditioned space.

Referring to Figure 2, the schematic diagram and the psychometric chart for Maisotsenko desiccant recirculation cycle (MDRC) is presented. The main alteration for this configuration compared to the previous one is the location of the mixer. In MDRC, mixer is located in the upper air stream whereas MDVC's mixer is in the lower air stream.

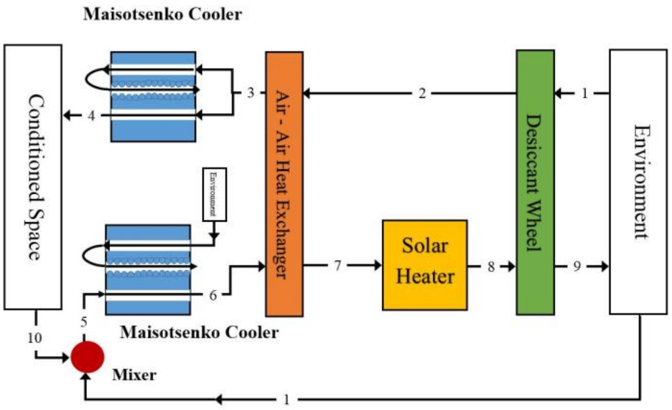

Figure 1: MDVC air conditioning unit configuration

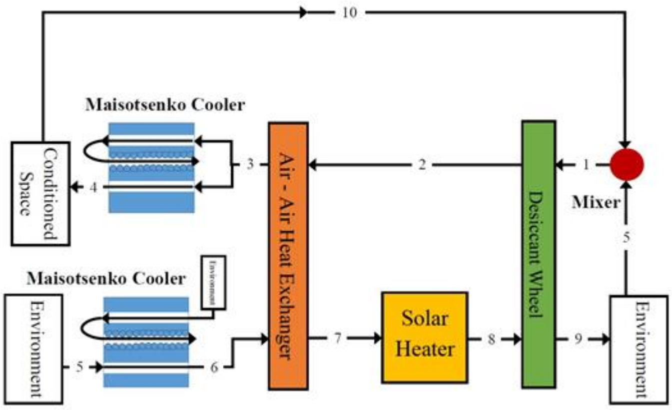

Figure 2: MDRC air conditioning unit configuration

3. MATHEMATICAL FORMULATION

The proposed desiccant wheel model in [3] is selected for simulation requirement of this research work. Energy and mass equations associated with desiccant wheel are converted into system of linear hyperbolic partial differential equations and they are solved by wave shock method for model development [12]. The model is experimentally validated in [13] and it implements the analogy theory and the combined potentials as follow:

$$f_{1,i} = \frac{-2865}{(t_i)^{1.49}} + 4.344(\omega_i)^{0.8624} \quad (1)$$

$$f_{2,i} = \frac{(t_i)^{1.49}}{6360} - 1.127(\omega_i)^{0.07969} \quad (2)$$

$$\eta_{F1} = \frac{f_{1,2}-f_{1,1}}{f_{1,8}-f_{1,1}} \quad (3)$$

$$\eta_{F2} = \frac{f_{2,2}-f_{2,1}}{f_{2,8}-f_{2,1}} \quad (4)$$

$$\dot{m}_u(t_2 - t_1) = \dot{m}_l(t_8 - t_9) \quad (5)$$

$$\dot{m}_u(\omega_2 - \omega_1) = \dot{m}_l(\omega_8 - \omega_9) \quad (6)$$

where ω_i and t_i are the humidity ratio in kg/kg and temperature at state *i* in K, $\dot{m}_u$ and $\dot{m}_l$ are the upper and lower air stream mass flow rates in kg/s, $f_{1,i}$ and $f_{2,i}$ are the combined potential for the desiccant wheel, η_{F1} and η_{F2} are the efficiency of the desiccant wheel. Based on the values presented in [3], 0.09 and 0.76 are selected experimentally for η_{F1} and η_{F2} respectively. The estimated values for desiccant wheel efficiency are associated with moderate level of operation which represent the worst case scenario. For high performance desiccant wheel, 0.05 and 0.95 are considered [3]. The air temperature at state 8 is referred to as the regeneration temperature for solid desiccant air conditioning. Advance in technology facilitates desiccants to exploit low grade energies results in low regeneration temperature in the range of 50 ℃ to 80 ℃ [3].

Mathematical formulation for the air to air heat exchanger is achieved by pinch point analysis with constant heat capacitance. Pinch temperature difference is located where the hot and cold air stream temperatures are closest to each other. In this research work, a counter flow heat exchanger is selected. In counter flow heat exchanger, pinch temperature difference might occur at each end of it. Because both hot and cold streams are air with same specific heat capacity, their mass flow rate plays a major role in identifying the location of the pinch temperature difference. If both air streams mass flow rate are equal, pinch temperature difference is located at both end of the heat exchanger. In case of upper air stream greater mass flow rate, air temperature at heat exchanger outlets are given as:

$$t_7 = t_2 - \Delta t_{pinch} \quad (7)$$

$$t_3 = t_2 - \frac{\dot{m}_l}{\dot{m}_u}(t_7 - t_6) \quad (8)$$

If the lower air mass flow rate is greater than the upper air mass flow rate, air temperature at heat exchanger outlets are given as:

$$t_3 = t_6 + \Delta t_{pinch} \quad (9)$$

$$t_7 = t_6 + \frac{\dot{m}_u}{\dot{m}_l}(t_2 - t_3) \quad (10)$$

In case of equal mass flow rates, air temperature at heat exchanger outlets are given as:

$$t_7 = t_2 - \Delta t_{pinch} \quad (11)$$

$$t_3 = t_6 + \Delta t_{pinch} \quad (12)$$

where Δt_{pinch} is the pinch point temperature difference for the air to air heat exchanger and its value is considered to be 10 ℃ in the analysis. From the presented formulas, it can be perceived that air to air heat exchanger best performance is achieved in case of equal upper and lower air stream mass flow rates. This is the main reason for choosing an identical air mass flow rate for the upper and lower streams in this research work simulation.

Maisotsenko cooler operates based on the combination of thermodynamic process of evaporative coolers and heat exchanger [11]. In an ideal case, product air will be cooled down to the working air dew point temperature. Nevertheless, product air temperature will be higher than the working air dew point temperature in reality. Product air temperature is estimated by dew point temperature effectiveness as follow [14]:

$$t_{o,m} = t_{i,m} - E(t_{i,m} - t_{dew}) \quad (13)$$

where $t_{i,m}$ and $t_{o,m}$ are the inlet and outlet air temperature in ℃ , t_{dew} is the Maisotsenko cooler working air dew point temperature in ℃ and E is the effectiveness of Maisotsenko cooler dew point temperature. It is reported in [15] that dew point temperature effectiveness of 0.8 is achieved experimentally. Consequently, this value is considered in this paper.

Desiccant wheel performance is analyzed based on the assumption of having equal air stream flow rate. Additionally, air to air heat exchanger best performance is achieved by having equal heat capacitance of hot and cold fluids. Since, both streams are air and their temperature difference does not have a significant effect on their specific heat capacity. Equal air flow rate in both streams will maximize the heat exchanger performance. Therefore, a mixer is required to match the upper and lower air streams at the desiccant wheel and the heat exchanger. This complexity is rises due to the presence of the Maisotsenko cooler and air stream division at its entrance. The mixture output temperature and humidity ratio are achieved as follow:

$$t_{o,mx} = P_f t_{amb} + P_r t_{cs} \tag{14}$$
$$\omega_{o,mx} = P_f \omega_{amb} + P_r \omega_{cs} \tag{15}$$

where P_f and P_r are the mass fraction of fresh air and recirculated air at the mixer, t_{amb} and t_{cs} are the ambient and conditioned space air temperature in ℃, ω_{amb} and ω_{cs} are the ambient and conditioned space humidity ratio in kg/kg, $t_{o,mx}$ and $\omega_{o,mx}$ are the mixer outlet air temperature in ℃ and humidity ratio in kg/kg, respectively. Since air flow rate is divided into two approximately equal streams at the Maisotsenko coolers, fresh air and recirculated air must have equal shares which implies P_f and P_r are both equal to 0.5.

The proposed system performance is evaluated with its COP as follow:

$$\dot{Q}_{reg} = \dot{m}_l (h_8 - h_7) \tag{16}$$
$$\dot{Q}_{cool} = 0.5\dot{m}_u (h_{cs} - h_4) \tag{17}$$
$$COP = \frac{\dot{Q}_{cool}}{\dot{Q}_{reg}} \tag{18}$$

where $\dot{Q}_{cool}$ and $\dot{Q}_{reg}$ are cooling capacity and regeneration heat requirement for their respective system in kW, h_i is the enthalpy of humid air at state *i* in kJ/kg, and COP is the air conditioning unit coefficient of performance.

In order to comprehend the advantages of Maisotsenko cooler implementation for desiccant air conditioning, same operating conditions as presented in [3] is considered. Consequently, sensible and latent heat for conditioned space is calculated as follow [15]:

$$\dot{Q}_{sen} = UA(t_{amb} - t_{cs}) \tag{19}$$
$$\dot{Q}_{lat} = LF\dot{Q}_{sen} \tag{20}$$

where $\dot{Q}_{sen}$ and $\dot{Q}_{lat}$ are the sensible and latent heat in kW, LF is the latent heat factor and UA is the multiplication of overall heat transfer coefficient and area in kW/K. The values for LF and UA are taken from [3] to be 0.1 and 0.35, respectively.

4. RESULTS AND DISCUSSION

This paper introduces a new solid desiccant air conditioning system employing Maisotsenko coolers instead of evaporative coolers. In order to signify the advantages and disadvantages of this alteration, same operating parameters from [3] are considered for a comparative analysis between the proposed systems and studied systems in [3]. The system is considered for air conditioning requirements of a building with surface area of 48 m^2 and volume 150 m^3. Ambient air temperature and relative humidity are 34 ℃ and 42% with reference comfort condition of 24 ℃ and 50%. In [3], two different desiccant air conditioning systems with evaporative coolers are studied. In this paper, they will be referred to as desiccant evaporative ventilation (DEV) and desiccant evaporative recirculation (DER) air conditioning units.

Results for the recommended systems in this paper and studied systems in [3] are presented in Table 1. It can be seen form Table 1 that for regeneration temperature of 70 ℃ , high performance desiccant wheels are recommended due to its high COP. Desiccant air conditioning with Maisotsenko cooler generally presents lower COP as compared to evaporative coolers. The major reason is the Maisotsenko cooler requirement of two air streams for functioning. Consequently, half of the dehumidified air stream is not employed for air conditioning purposes by being exhausted into atmosphere at the upper Maisotsenko cooler. Considering that the air mass flow rate in the solar heater is approximately twice the air mass rate supplied to the conditioned space. On the other hand, MDVC and MDRC are capable of providing cooler air stream to the conditioned space which results in lower required air volumetric flow rate. Lower COP might degrade the economical attractiveness of the proposed systems. Nonetheless, lower air volumetric flow rate indicates the possibility of commercializing these systems and their competitiveness with conventional air conditioning units. Furthermore, the effect of increasing the regeneration temperature is more evident in MDVC and MDRC configurations compared to DEC and DEV air conditioning units. By keeping in mind that air volumetric flow rate through the system is twice the required volumetric flow rate supplied into the conditioned space, MDRC results are promising in particular with high regeneration temperatures. It is important to bear in mind that the presented data solely purpose is for comparison and it does not necessarily imply the superiority of one configuration over the others.

Table 1: Comparative analysis of proposed systems and investigated systems in [3]

Regeneration Temperature	MDVC air Volumetric flow rate (m^3/h)	MDVC COP	MDRC air Volumetric flow rate (m^3/h)	MDRC COP	DEV air Volumetric flow rate (m^3/h)	DEV COP	DEC air Volumetric flow rate (m^3/h)	DEC COP
70 ℃	875	0.29	695	0.27	950	0.85	1000	0.42
90 ℃	760	0.20	615	0.20	1200	0.29	1000	0.26

Sensitivity analysis is necessary to understand the system behaviors and the effect of different operating variables and conditions on the air conditioning units. It also provides a detailed picture for designers to select the most appropriate design parameters. Detailed sensitivity analysis is performed based on the weather data available for a typical day in August for Abu Dhabi with ambient dry bulb temperature of 41℃ and relative humidity of 29%. The desiccant wheels are simulated with high efficiency and regeneration temperature of 80℃.

Regeneration temperature is the only parameter that can be controlled to achieve the best performance of the investigated air conditioning units. The results concerning the effect of regeneration temperature on the minimum required air volumetric flow rate are depicted in Figure 3: Effect of regeneration temperature on required air volumetric flow rate. Expectedly, increasing regeneration temperature improves the desiccant wheel dehumidification capability which enhances the unit performance for MDVC and MDRC. In principle, air moisture content and dew point temperature drops at the upper Maisotsenko cooler and air is supplied into the conditioned space with lower temperature. Therefore, a smaller amount of air is required to satisfy the cooling load. In addition, the effect of increasing the regeneration temperature is more evident on the MDVC configuration required air volumetric flow rate. The results concerning the effect of regeneration temperature on the required solar heating are presented in Figure 4: Effect of regeneration temperature on required solar heating. It is fascinating that the required solar heating reduces as the regeneration temperature increases. One may conclude that the reduction in the required air volumetric flow rate impact is more significant than the solar collector higher outlet temperature. In consequence, required solar heating abated and the system becomes more compact. Additionally, by keeping the cooling load constant in this analysis, it can be noted that required solar heating is in indirect relationship with the *COP*. Hence, growth in regeneration temperature improves both suggested air conditioning units *COP*.

Ambient conditions are the other important factors in analyzing the potentials of an air conditioning unit to obtain the inside door conditions. The effect of ambient air dry bulb temperature on the minimum required air volumetric flow rate is illustrated in Figure 5. Noting that the air humidity ratio is kept constant, there is no surprise that higher ambient air temperature requires more air to achieve the required comfort condition.

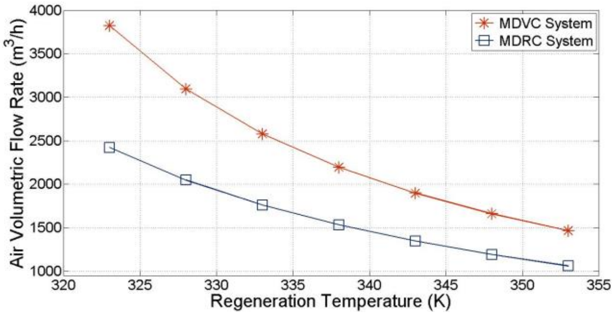

Figure 3: Effect of regeneration temperature on required air volumetric flow rate

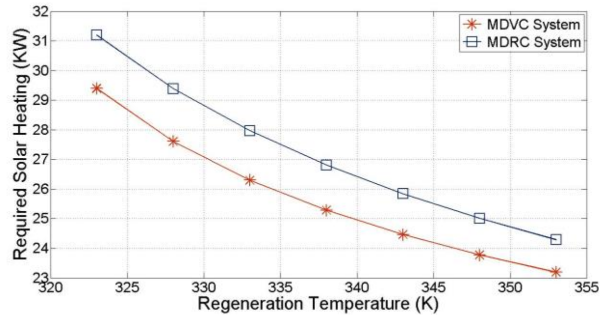

Figure 4: Effect of regeneration temperature on required solar heating

There are two main reasons for this growth. First, higher ambient air temperature increases the cooling load by increasing both sensible and latent heat. Second, air temperature supplied into the conditioned space has higher temperature. Moreover, the effect of ambient air rise is more evident on MDVC performance compared to MDRC. Since the MDVC is a ventilation air conditioning unit that supplies fresh air only to the conditioned space, ambient air temperature more considerable impact on its performance is predictable. Thus, MDRC is more appropriate for hot climates.

The main purpose of integrating a desiccant wheel with Maisotsenko cooler is to improve its performance in humid condition. It is only wise to investigate the effect of relative humidity on the air conditioning units. It is important to bear in mind that the proposed systems can only satisfy the assigned cooling load and conditioned space comfort conditions if the supplied air temperature is considerably lower than the conditioned space comfort condition temperature (24℃).

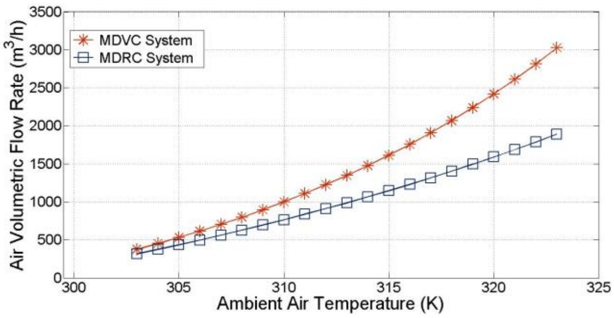

Figure 5: Effect of ambient air temperature on required air volumetric flow rate

The effect of ambient air relative humidity on the conditioned space supplied air temperature is illuminated in Figure 6: Effect of ambient air relative humidity on conditioned space supplied air temperature. With ambient air temperature of 41℃ and regeneration temperature of 80℃, the maximum possible relative air humidity for MDVC proper functioning is almost 43%. Higher relative humidity prevents MDVC to achieve the required constraints. On the other hand, MDRC is capable to perform for higher relative humidity values up to 59%. Based on the ambient temperature analysis, MDRC is a more promising configuration for hot and humid climates such as UAE. The results concerning the effect of ambient air relative humidity on the required air volumetric flow rate is presented in Figure 7. Humid climates increase the required supplied air to the conditioned space drastically. Increase in relative humidity results in higher air dew point temperature and higher supplied air temperature at the conditioned space entrance. Humidity has a more significant influence on MDVC performance as compared to MDRC.

The results concerning the effect of relative humidity on the required solar heating is demonstrated in Figure 8. The rise in the required solar heating is severe such that at relative humidity of 60%, the required solar heating is five times the required solar heating at relative humidity of 25%. Undoubtedly, the key factor in such a substantial growth is the reported increase in the required air volumetric flow rate. Considering that the air flow rate through the solar heater is twice the required air flow rate at the conditioned space, the impact of increasing relative humidity is massive.

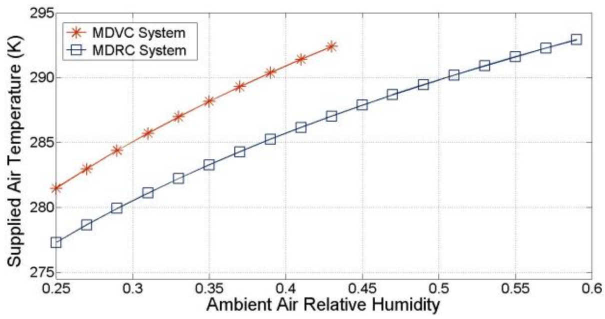

Figure 6: Effect of ambient air relative humidity on conditioned space supplied air temperature

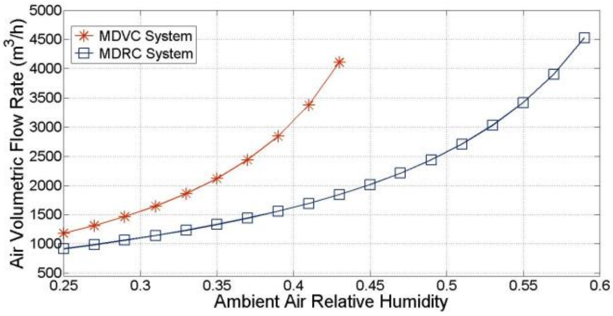

Figure 7: Effect of ambient air relative humidity on required air volumetric flow rate

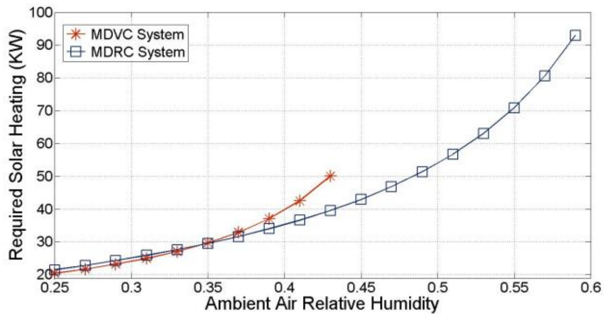

Figure 8: Effect of ambient air relative humidity on required solar heating

These air conditioning configurations are proposed to be utilized in hot and humid climates such as UAE. Therefore, a transient analysis is performed based on the weather data available for a typical day in July in Abu Dhabi. The ambient air dry bulb temperature and relative humidity during a day in July is presented in Figure 9. Night time condition is generally indicates lower air temperature and higher relative humidity. During sunlight, the air temperature exceeds 40℃ and relative humidity drops to around 30%. The required air volumetric flow rate throughout the day is presented in Figure 10. Both air conditioning units are capable of satisfying the cooling load and comfort conditions in the worst case scenario for Abu Dhabi with regeneration temperature of 80 ℃ and highly efficient desiccant wheels. Required air is at its minimum at dawn whereas the maximum required air volumetric flow rate occurs during noon for both MDVC and MDRC.

The required solar heating during a day is shown in Figure 11. These results can be beneficial for designing a thermal energy storage based on the available solar insolation and required solar heating throughout the day. Required solar heating for MDVC is lower for most of the day though difference between the required solar heating for MDVC and MDRC is very small. Fortunately, required solar heating rises during day light when the solar insolation is in abundant which indicates the high potential of this technology for UAE.

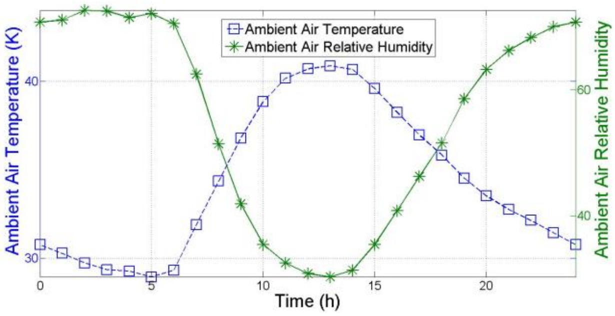

Figure 9: Abu Dhabi ambient air dry bulb temperature and relative humidity

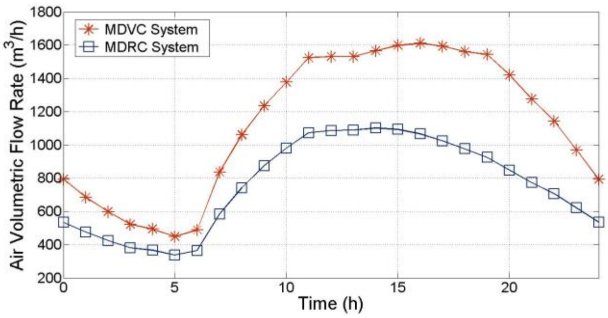

Figure 10: Minimum Required air volumetric flow rate for Abu Dhabi throughout a day

COP of the air conditioning unit is another important factor in assessing the proposed cycles' potential. COP of MDVC and MDRC configurations for its implementation during July in Abu Dhabi for a complete day is demonstrated in Figure 12. COP for both configurations mostly rises throughout the daylight which indicates that the ratio of cooling load is increases compared to the required solar heating.

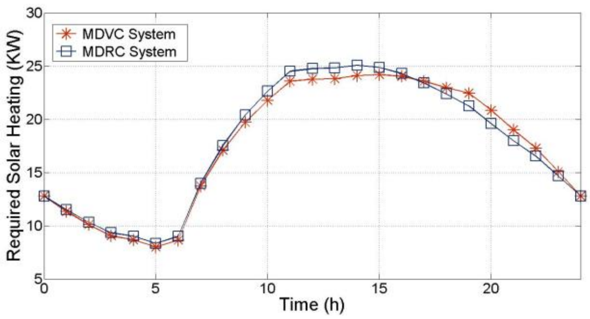

Figure 11: Required solar heating for Abu Dhabi throughout a day

While *COP* for both configurations mostly rises throughout the daylight which indicates that the ratio of cooling load is increases compared to the required solar heating. *COP* of MDVC is mainly higher than *COP* of MDRC since air with higher temperature enters the solar heater in MDVC configuration in consequence lower solar heating is required. There is one unexpected reduction in the *COP* curves for both configurations around 7:00 a.m. It is important to bear in mind that *COP* is affected by ambient air temperature as well as relative humidity. Air temperature increases the conditioned space cooling load and reduces the required solar heating at the same time. On the other hand, relative humidity affects the air temperature supplied to the conditioned space. In early morning, air temperature increases whereas the relative humidity decreases. Nevertheless, reduction in relative humidity cannot necessarily be translated into lower humidity ratio. Taking all these factors into account, at 7:00 a.m. their effects result in a sudden reduction of *COP* before it constantly rises till the noon.

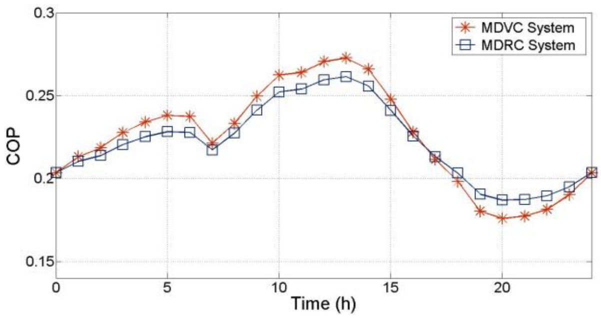

Figure 12: Air conditioning units *COP* for Abu Dhabi throughout a day

5. CONCLSUION

In this paper, two new air conditioning systems combining solid desiccant and Maisotsenko cooler are proposed and investigated for humid climates such as in UAE. Solar energy is selected as the primary source of energy for the air conditioning operation of these systems. Detailed thermodynamic analysis is executed to illustrate the performance of the proposed air conditioning systems in UAE under different operating conditions.

Comparative analysis indicates that the potential of using these desiccant air conditioning units utilizing Maisotsenko coolers are growing up fast as compared to conventional desiccant air conditioning systems. MDRC configurations are compared to DEC and DEV air conditioning units. By keeping in mind that air volumetric flow rate through the system is twice the required volumetric flow rate supplied into the conditioned space; MDRC results are very promising in particular with high regeneration temperatures. Transient analysis illustrates that both MDVC and Finally, MDRC systems are capable of operating in Abu Dhabi worst condition without any difficulties.

Results of transient analysis indicates that the maximum required air volumetric flow rate for a typical summer day in Abu Dhabi during noon is about 1600 m^3/h and 1100 m^3/h for MDVC MDRC, respectively. Additionally, the required solar heating for both configurations are approximately 25 kW at its maximum operation. Finally, it is important to note that the *COP* for both configurations increases during noon times.

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