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Performance studies of hybrid solid desiccant–vapor compression air-conditioning system for hot and humid climates

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ABSTRACT

Suitability of the solid desiccant cooling to supplement the conventional vapor compression refrigeration air conditioner has been studied for typical hot and humid climate. A solid desiccant and vapor compression hybrid air-conditioning system for a test room of cooling capacity 1.8 kW, has been modelled in simulation studio project and simulated in TRNSYS environment for the cooling season from March to September. The obtained results have been validated against experimental test data. The effects of ambient humidity ratio and temperature on coefficient of performance and supply air temperature have been evaluated. The variation in coefficient of performance has been obtained at various regeneration temperatures. The simulation results show the suitability of such systems for cooling of building in hot and humid climates.

1. Introduction

Increasing energy consumption in space cooling applications is one of the prime concerns of the present age all over the world. Environmental issues associated with the production of high grade electrical energy and increasing demand for space-cooling applications have compelled researchers to investigate newer technologies for air conditioning. Desiccant cooling is an approach for air conditioning towards resolving the environmental and economic issues of the conventional vapor compression cooling systems.

Comparison of various solid-desiccant based cooling cycles for air conditioning in hot and humid climates was carried out [1]. Solid desiccant dehumidification system with back up of traditional vapor compression air-conditioner has been simulated and performance was obtained for different climatic zones of U.S.A. [2]. Considerable energy saving was found using solid desiccant based hybrid air conditioning systems in hot and humid weather under high latent load conditions [3] compared to conventional vapor compression refrigeration based air-conditioning system. Desiccant cooling was suggested as supplement to conventional vapor compression cooling or evaporative cooling due to its energy and cost saving in hot and humid environment conditions [4]. Experimental tests on hybrid desiccant dehumidification and air conditioning system have been conducted [5] for Hong Kong, a high humid region. It has been evaluated that the hybrid solid desiccant cooling system achieves a higher part load performance in hot and humid climates. A hybrid solid desiccant air cooling system was designed and experimentally tested [6] for improving the air quality and to find out the reduction in energy consumption. Performance of a hybrid solid desiccant air conditioning system have been experimentally investigated [7] in terms of energy saving. Experimental investigation of solar assisted hybrid desiccant cooling system for hot and humid weather of Malaysia was also conducted [8]. Solid desiccant based hybrid air-conditioning systems can give substantial energy savings as compared to conventional vapor compression refrigeration based air-conditioning systems in hot and humid climatic conditions [9]. Performance study of solid desiccant cooling system has been done for different climatic zone in India [10]. Taweekun and Akvanich [11] have designed a solid desiccant dehumidification system suitable for tropical climate, to reduce the latent load of air-conditioning system and to improve the thermal comfort.

Kodama et al. [12] have carried out the optimization of rotational speed and performance of rotary desiccant wheel for simultaneous change in enthalpy and humidity. An analysis of the desiccant wheel for low humidity conditions was also carried out [13]. The effect of different parameters like air flow rate and speed of wheel was observed on the performance of the system. Further, they have added the parametric study on a desiccant assisted air-conditioner [14]. Solid desiccant air conditioning model was developed and experimentally validated for investigating the performance and for examining the influence of parameters like weather conditions,

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Nomenclature

COP	coefficient of performance
C_p	specific heat of air at constant pressure (kJ/kg/°C)
DBT	dry bulb temperature (°C)
EV	expansion valve
h	specific enthalpy (kJ/kg)
m_p	mass flow rate of process air (kg/s)
m_r	mass flow rate of reactivation air (kg/s)
Q	regeneration heat (kW)
RH	relative humidity
T	temperature (°C)
VCR	vapor compression refrigeration
W_c	compressor power (kW)

Greek letters

ε	effectiveness
ω	humidity ratio (g/kg)

Subscripts

DW	desiccant wheel
F_1, F_2	potential functions
HRW	heat recovery wheel
i	inlet
o	outlet
R	recirculation
REG	regeneration
1, 2, etc.	reference state points

cooling load, air flow rate and regeneration temperature [15]. Further, an experimental validation for a desiccant wheel model, based on the concept of the analogy method has also been presented by them [16]. Effect of change in ambient conditions as well as effectiveness of desiccant wheel and sensible regenerator, on overall performance of desiccant cooling system have been investigated for conventional and recirculation configurations [17]. A model of desiccant cooling system based on transient and coupled heat and mass transfer condition has been developed by [18]. Simulations have been carried out [19] to study the effects of variation in outdoor conditions and regeneration temperature on the performance of a desiccant cooling system for different system configuration. A numerical procedure have also been suggested [20] for designing solid desiccant cooling systems and for analyzing the impact of individual component characteristics on the overall system performance.

The viability and energy saving potential of desiccant cooling system was simulated and also experimentally determined [21] for the weather conditions of Pakistan. TRNSYS has been used for modelling and simulating the desiccant cooling system. Energy saving potential of a solar assisted solid desiccant cooling system for an institutional building was studied [22] using TRNSYS. Technical and economical parameters such as COP, solar fraction, life cycle analysis and payback periods were studied to check viability of the system. Life cycle analysis and economic assessment of the system were also presented. Khoukhi [23] has studied feasibility of desiccant cooling system as an alternative to conventional vapor compression cooling to achieve thermal comfort for cooling buildings in hot and humid climates.

A solid desiccant and vapor compression hybrid air-conditioning system was compared to traditional HVAC system in terms of performance, operating cost saving and payback period [24]. Experimental tests were carried out to evaluate performance of hybrid cooling system coupled to micro CHP in terms of primary energy saving [25]. Further, the same authors have regenerated the desiccant wheel of hybrid cooling system by using co-generated thermal power and obtained substantial energy saving as well as considerable CO_2 emission reduction [26].

Based on literature survey, it was found that very limited studies have been carried out on transient simulation of solid desiccant and vapor compression hybrid air-conditioning system for hot and humid climatic zone having relative humidity in range of 70–85% to the best of the authors' knowledge. This provided the prime motivation behind doing the present study. Solid desiccant and vapor compression hybrid air-conditioning system has been studied for hot and humid climate. The system has been modelled and simulated using TRNSYS software. In the present study, simulation of hybrid air-conditioning system, which is actually an integration of a rotary solid desiccant dehumidifier and a traditional vapor compression air-conditioning unit, has been carried out. The complete system has been modelled in simulation studio project for a test room (3 m × 3 m × 3 m) having 1.8 kW cooling load. The simulation studio project has been simulated using TRNSYS [27] to study the performance of the system during cooling season from March to September in a hot and humid climate of North India (Roorkee). Simulated results have been validated with the experimental data. The effects of variation in important parameters like regeneration temperature, ambient temperature and humidity ratio on system performance has also been discussed. The variation in regeneration temperature for the changes in potentials effectiveness ε_{F_1} and ε_{F_2} of dehumidifier model is also discussed.

2. Description of the system

In the system presented in Fig. 1, the supply air stream at state 1 is passed through rotary desiccant wheel. Its moisture is partly but significantly adsorbed by the desiccant wheel material and the heat of adsorption elevates its temperature. Thus, a warm and dry air stream exits at state 2. The air stream is then sensibly cooled in the heat recovery wheel from state 2 to state 3 and further by passing over a finned evaporator coil from state 3 to state 4.

In the regeneration side, ambient air is first passed though heat recovery wheel to recover heat (state 6 to state 7) from warm and dry air and it is further heated by an electric heater (state 7 to state 8) to a required regeneration temperature at state 8. In the last step, hot regeneration air extracts humidity from desiccant wheel (state 8 to state 9) and releases it to the ambient at state point 9.

3. System modelling

A test room with a sitting capacity of 5 persons has been selected for modelling. The sensible cooling load and the latent cooling load are obtained as 1.371 kW and 0.391 kW, respectively [28]. The inside room conditions are assumed as 50% RH and 26 °C DBT [29]. Outdoor conditions for the present case are selected as 35 °C DBT and 70% RH. The designed air conditioning system is a hybrid one, which combines desiccant dehumidification system with the conventional vapor compression air conditioning system. Operation in recirculation mode keeps away the unpleasant outdoor dust particles and improves the cooling output with better energy efficiency. The problem with recirculation mode is successive increase in the humidity level inside the room and of course regarding the purity of air. The hybrid system described here can effectively control the humidity separately using desiccant dehumidifier. The small amount of ventilation fresh air for the test room has been incorporated in terms of DOR (door opening rate) while calculating the infiltration cooling load by HLF method. However, this is the first phase of the study. In the next phase, ventilation mode will be considered and the comparison between ventilation and recirculation modes will be explored in terms of COP. The mass flow rates of the

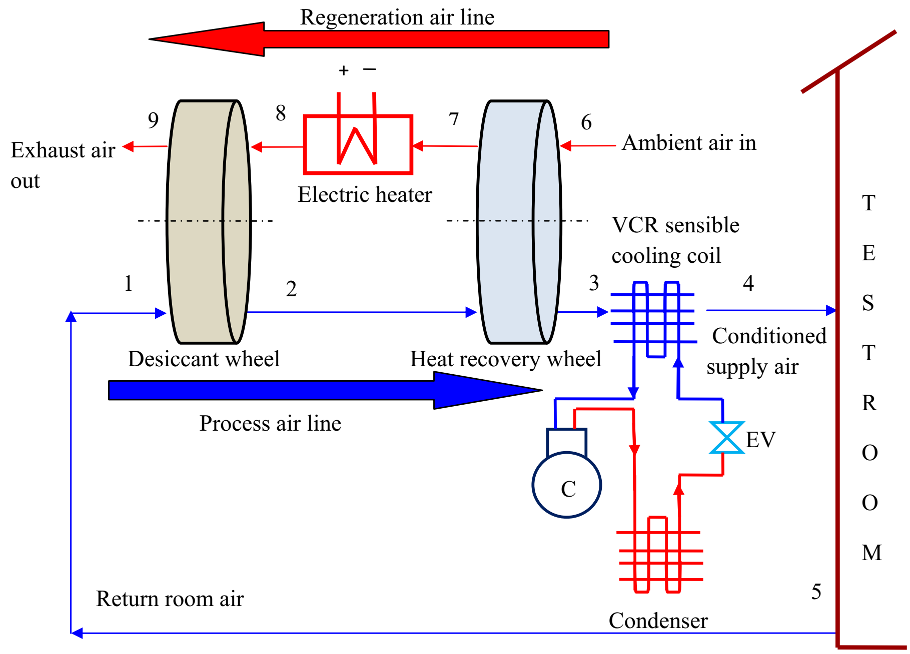

Fig. 1. Schematic diagram of hybrid solid desiccant–vapour compression air-conditioning system.

process as well as the regeneration air are 0.125 kg/s. Effectiveness of desiccant wheel and heat recovery wheel are assumed as 70% and 80% respectively. The effectiveness of desiccant wheel (DW) and heat recovery wheel (HRW) are as given in Eqs. (1) and (2) respectively [4]:

$$\varepsilon_{\mathrm{DW}} = \frac{\omega_1 - \omega_2}{\omega_1} \tag{1}$$

$$\varepsilon_{\mathrm{HRW}} = \frac{T_2 - T_3}{T_2 - T_6} \tag{2}$$

Processes 2–3 and 6–7 are sensible cooling and sensible heating respectively in HRW. Thus,

$$\omega_2 = \omega_3 \tag{3}$$

$$\omega_6 = \omega_7 \tag{4}$$

Supply air to the room is obtained by sensible cooling of air during 3–4 in cooling coil, which gives

$$\omega_3 = \omega_4 \tag{5}$$

Similarly, regeneration air is sensibly heated between 7 and 8 in an electric heater, giving

$$\omega_7 = \omega_8 \tag{6}$$

Energy balance between the two air streams in DW gives

$$m_{\mathrm{p}}(h_2 - h_1) = m_{\mathrm{r}}(h_8 - h_9) \tag{7}$$

Applying energy balance between the two air streams in HRW will give

$$m_{\mathrm{p}}(h_2 - h_3) = m_{\mathrm{r}}(h_7 - h_6) \tag{8}$$

Regeneration heat can be obtained from

$$Q = m_{\mathrm{r}}(h_8 - h_7) \tag{9}$$

Coefficient of performance (COP) of the system is defined as

$$\mathrm{COP_R} = \frac{m_{\mathrm{p}}(h_5 - h_4)}{m_{\mathrm{r}}(h_8 - h_7) + W_{\mathrm{c}}} \tag{10}$$

Fig. 2 shows the simulation studio modelling of hybrid solid desiccant – vapor compression air-conditioning system in recirculation mode using TRNSYS 16. In simulation studio project (Fig. 2), rotary desiccant dehumidifier is modelled as type 683, type 760b is heat recovery wheel, type 651-1 is sensible cooler, type 690 is room load, type 33e is psychrometrics, type 663-1 is unit heater, type 65d is plotter and type 109-TMY2 is weather data file, type 112a and type 112a-2 are process and regeneration air fans respectively.

4. Rotary desiccant dehumidifier model

Desiccant dehumidifier is a device which removes moisture from wet air without cooling the air below its dew point. Instead, it relies on the ability of hygroscopic adsorption of water on to its surface. In the process of adsorption, a thin layer of molecules of water vapor adheres to the substrate surface of desiccant material. Eventually, the desiccant material becomes saturated with water and has to be regenerated through a drying process. A schematic layout of the rotary desiccant dehumidifier modelled by type 683 in TRNSYS has been shown in Fig. 3. In the present configuration, the process and regeneration air streams flow simultaneously through a rotating desiccant dehumidifier. At any instant, a portion of the total dehumidifier area is regenerated by hot regeneration air stream while the remaining portion adsorbs water vapor from the process air stream. The regeneration temperature is an important parameter which influences the effectiveness of desiccant wheel in terms of moisture removal capacity. So, for the determination of reactivation temperature of desiccant wheel with variations in outdoor humidity ratio, the modelling of rotary desiccant dehumidifier has been carried out. Regeneration as well as process air stream outlet condition were determined by simulating the model.

Type 683 of the model uses an iterative process to locate the actual outlet condition (point D) of the dehumidifier as shown in Fig. 4 [27]. The dehumidifier does not have any limit on its capacity, e.g. the user can specify any humidity ratio of the process air outlet provided the dehumidifier is on. The model determines the required temperature of regeneration air stream to meet the desired humidity ratio. The user provides certain known values to

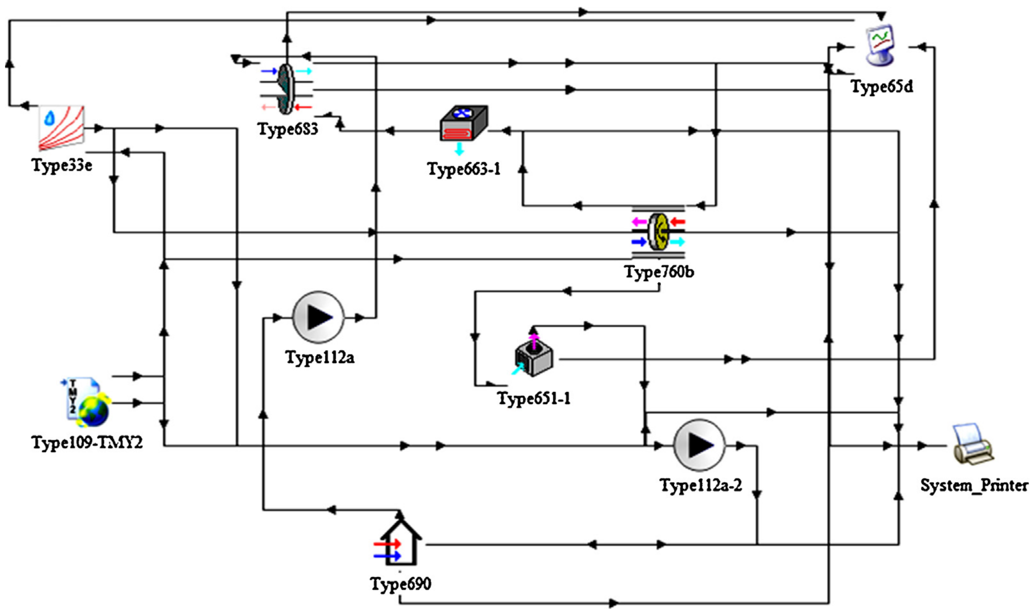

Fig. 2. TRNSYS simulation studio project.

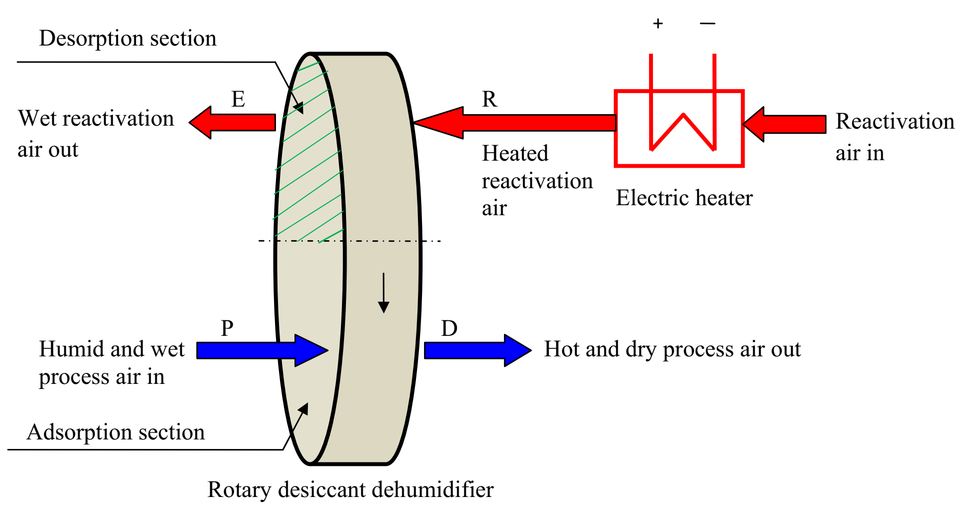

Fig. 3. Schematic layout of rotary desiccant dehumidifier.

the model. These are mass flow rates of process air and regeneration air streams and process stream inlet conditions (dry bulb temperature and absolute humidity ratio), on the basis of that model evaluates the outlet humidity ratio of process air stream. The model first computes the values of the potential functions F_1 and F_2 of the process air stream inlet. Because the value of the potential function

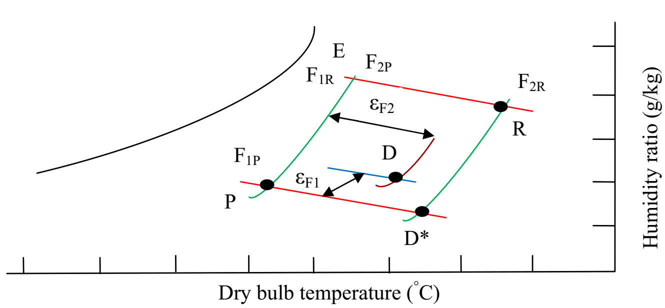

Fig. 4. Dehumidifier process paths.

F_1 is the same at points P and D* (Fig. 4), the ideal outlet temperature of the process air can then be determined.

The equations for F_1 and F_2 derived by Jurinak [30] for a silica gel desiccant are

$$F_1 = \frac{-2865}{T^{1.490}} + 4.344\omega^{0.8644} \tag{11}$$

$$F_2 = \frac{T^{1.490}}{6360} - 1.127\omega^{0.07969} \tag{12}$$

The isopotential lines for F_1 and F_2 are further modified for non idealities in the system by two effectiveness values ε_{F_1} and ε_{F_2} in terms of potential functions [31] and defined by

$$\varepsilon_{F_1} = \frac{F_{1D} - F_{1P}}{F_{1R} - F_{1P}} \tag{13}$$

$$\varepsilon_{F_2} = \frac{F_{2D} - F_{2P}}{F_{2R} - F_{2P}} \tag{14}$$

Once the temperature at point D* is known, the value of the potential function F_2 at D* can be computed. The required

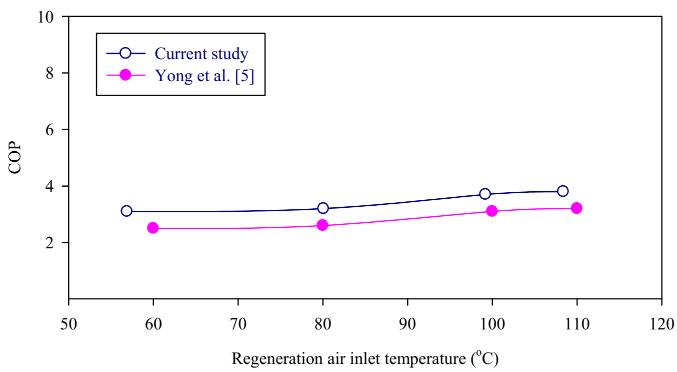

Fig. 5. Validation of the current results with the experimental results [5].

regeneration temperature R in Fig. 4 is evaluated when potential function F_2 has been calculated by model for known value of temperature at point D*. Modified values of the potential functions F_1 and F_2 at D* are computed, corresponding to the value of the potential function at D. Using the values of F_1 and F_2 at D, the model then iterates to find a corresponding value of humidity ratio at the process outlet. Once the assumed value for the calculation of humidity ratio falls within a certain tolerance, the corresponding temperature is computed. The effectivenesses ε_{F_1} and ε_{F_2} of the dehumidifier expresses the degree the process approximates the adiabatic one and the degree of dehumidification respectively. The state of the air stream exiting desiccant dehumidifier has been computed by the model using mass and energy balance on the entire dehumidifier using process and regeneration air stream inlet conditions and the effectiveness of the dehumidifier. Required values of ε_{F_1} and ε_{F_2} have been assumed as 0.05 and 0.95 which corresponds to a high efficiency wheel, as the condition $\varepsilon_{F_1}, \varepsilon_{F_2} \leq 1$ is always valid and the ideal wheel refers to $\varepsilon_{F_1} = 0$ and $\varepsilon_{F_2} = 1$.

5. Validation of the model

Fig. 5 shows the comparison between the simulated results obtained in the current study and the experimental results obtained by Yong et al. [5]. It has been observed that system performance in terms of COP increases with the increase in regeneration temperature. This is because during regeneration phase, at higher regeneration temperature air could remove more moisture; similarly during dehumidification process, the desiccant material adsorbs significant portion of water vapor from the process air, resulting in saving of cooling power. This is due to the substantial reduction in humidity ratio of process air flowing over VCR cooling coil. It saves the compressor power to cool process air below dew point to obtain the required dehumidification. Thus, when the regeneration temperature increases the temperature difference (which is the driving potential for convective heat transfer) as well as the absolute humidity difference (which is the driving potential for mass transfer) between the air and desiccant material increases respectively. So, as the temperature of regeneration air stream increases, simultaneously the adsorption rate in the process air side also increases. This in turn causes a successive reduction in the latent load met by the VCR cooling coil, ultimately leading to an increase in the system COP. However, with an increase in regeneration temperature ($T_{reg} > 100\,°C$), sensible heat transferred to the process air side is slightly raised, resulting in a higher outlet temperature of the process air; this in turn has to be cooled by the heat recovery wheel along with the evaporator cooling coil of conventional vapor compression air-conditioning unit, affects the COP.

Simulated TRNSYS model of hybrid solid desiccant vapor compression air-conditioning system was validated with the help of experimental data obtained by Yong et al. [5]. Fig. 5 depicts that both the simulation and test results show the same trend i.e. system performance slowly enhance with successive increment in regeneration temperature up to 100 °C. A reasonable agreement between the test and simulation results was observed from the behaviour. The error analysis had been done to find the error between simulated results and test results which shows a deviation of 5.2%. It was found that the experimental and theoretical simulation results show a similar trend, but there are discrepancies between them. The reasons causing the differences between the simulated results and experimental results are as follows:

(1) The pre-set conditions of the simulation were the average value of the fluctuating testing conditions, which is the key factor inducing the difference between the simulation and experimental results.
(2) The uncertainties involved in the measurement also have a major role behind the deviation.
(3) The limit accuracy of the simulation model was another probability causing the errors.
(4) It was found that the simulation results were higher than the experimental results. This is because of energy carried by the matrix of the dehumidifier from regeneration air, which is hotter than the process air. This results into sensible heat transfer to the process air, which was not considered in the simulation. Hence, the gap between the simulation and experimental measurement is mainly caused by un-considering this part energy (carried out by pre-cooled air) into the simulation. This extra energy transferred into the air stream from the desiccant matrix was ignored during simulation. This causes the energy consumption in simulation lower than the experiments.

The errors in individual module employed in the model, the environmental impact which causes energy loss during the testing, unavoidable measuring errors in the testing and reading data and theoretical calculation also limit the accuracy of the model. Because of the complexity of the simulation model of the whole system, the assumptions settled in the simulation and the unavoidable influences in the experiments cause errors. The errors obtained between the simulation and the experimental results are acceptable and the model can approximately predict the system conditions. Therefore, the overall system performance predicted by the simulated model remains closer to the experimental results within a good agreement. This is very useful when evaluating the feasibility of the hybrid solid desiccant vapor compression cooling system for the hot and humid climate.

Jia et al. [32] have also carried out experiments on a hybrid desiccant air-conditioning system which is integration of a rotary desiccant dehumidification and a vapor compression air-conditioning unit. Fig. 6 shows the comparison between current study and that of Jia et al. [32] on psychrometric chart. The results are in good agreements with the experimental measurements for process air side while for regeneration air side results are not in good agreement and the reasons should be further explored. The psychrometric processes involved in the operation of a hybrid solid desiccant vapor compression air-conditioning system are shown in Fig. 6.

As depicted on the psychrometric chart, process 1–2 which represents adsorption process while process 8–9 represents desorption process follows a path that diverges slightly from constant enthalpy. This is due to fact that a temperature at process air outlet arises during dehumidification process 1–2 because of thermal energy carried out slightly by the matrix of the wheel from the regeneration air which is hotter than the process air.

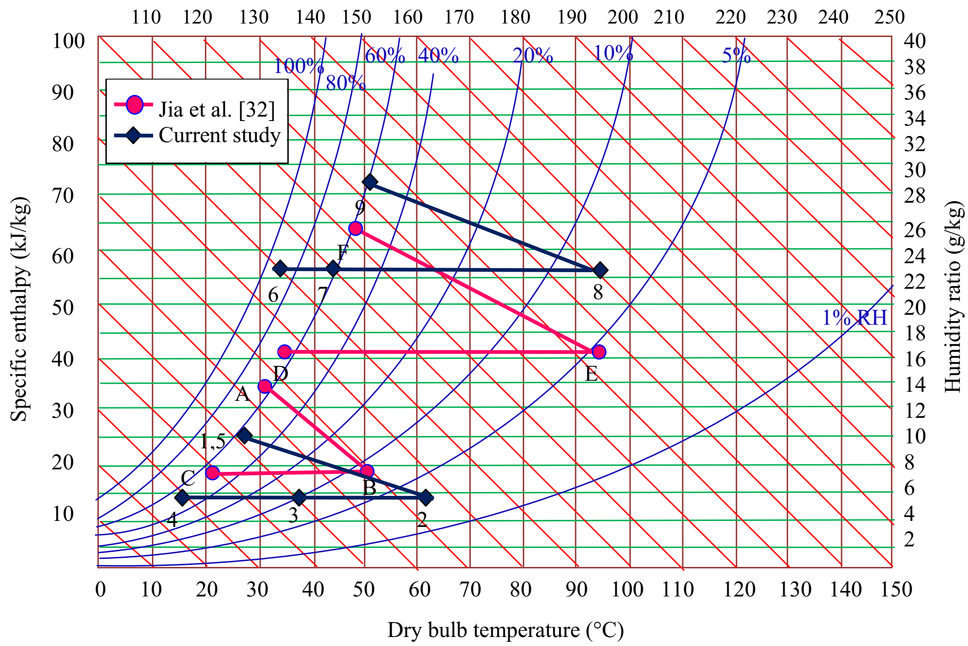

Fig. 6. Comparison between the results of the current study with the experimental measurements [32] on psychrometric chart.

Similarly, during regeneration process 8–9 regeneration air outlet temperature decreases slightly due to the sensible heat transfer to the colder process air in regeneration section.

6. Results and discussion

TRNSYS simulation studio project for hybrid solid desiccant vapor compression air-conditioning system is simulated in TRNSYS environment for typical north Indian hot and humid climate (Roorkee) during the period of March–September. Figs. 7 and 8 show the simulation runs for temperature and humidity ratio for the selected cooling season. The results of simulation are explained in detailed. These results are including temperature and humidity ratio of process as well as regeneration air streams against the time.

Fig. 7 shows the variation in important temperatures (°C) of hybrid solid desiccant vapor compression air conditioning system versus time. Ambient air temperature, room air temperature, process air outlet temperature, supply room air temperature, regeneration air inlet temperature and regeneration air outlet temperature are shown in Fig. 7. It is observed that the regeneration temperature of air at inlet is 94 °C almost. Regeneration temperature is one of the important temperatures that has main role in changes of the overall system performance.

Fig. 8 shows the variation in humidity ratio at different places in hybrid solid desiccant vapor compression air conditioning system versus time. Ambient air humidity ratio, room air humidity ratio, process air outlet humidity ratio, supply room air humidity ratio and regeneration air humidity ratio are shown in Fig. 8. There is almost 50% reduction in humidity ratio of the process air by passing through desiccant dehumidifier is observed from the behaviour. Humidity ratio is the main parameter indicating removal of moisture from room air in terms of latent heat to obtain desired comfort conditions inside the room.

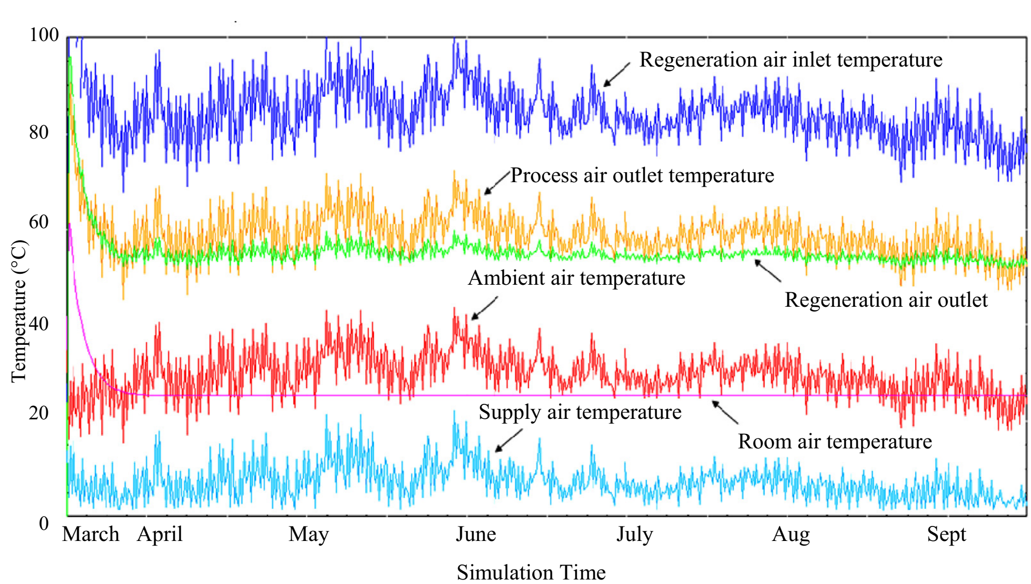

Fig. 7. Variations of temperature with time at important state points during cooling season.

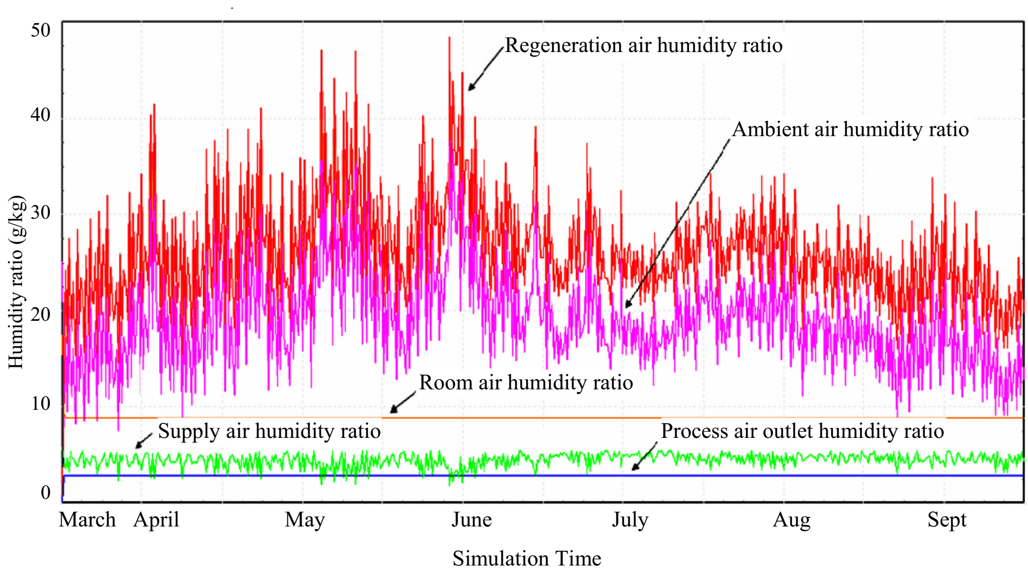

Fig. 8. Variations of specific humidity ratios with time at important state points during cooling season.

Table 1
Temperature and humidity ratio of air at different state points in the cycle.

State point	Temperature (°C)	Humidity ratio (g/kg)
1	26	10.5
2	62	5.0
3	38	5.0
4	16	5.0
5	26	10.5
6	35	23
7	45	23
8	94	23
9	52	29

The overall behaviour of the system in terms of dry bulb temperature and humidity ratio of air at different state points in the cycle has been presented in Table 1.

Figs. 9 and 10 show the variation in regeneration temperature with change in humidity ratio of process air inlet at different values of the effectiveness of the potential functions ε_{F_1} and ε_{F_2}. Here, ε_{F_1} expresses the degree by which the process approximates the adiabatic one, while the ε_{F_2} expresses the degree of dehumidification. $\varepsilon_{F_1} = 0$ and $\varepsilon_{F_2} = 1$ corresponds to an ideal process, which is adiabatic having the maximum dehumidification level for the respective geometry and flow conditions. The temperature and humidity of the air stream at the outlet of desiccant wheel could be calculated for the given values of the efficiency indices of the wheel ε_{F_1} and ε_{F_2}. The combined potential functions are dependent on the temperature and humidity ratio of the air steam as well as on the thermo physical properties of the processed air and of the wheel geometry context including the desiccant material.

The dehumidification performance increases as ε_{F_1} is smaller and ε_{F_2} is larger (i.e. required regeneration temperature for desorption is lesser with higher ε_{F_2} and smaller ε_{F_1} as shown in Figs. 9 and 10).

Fig. 11 depicts the variations in coefficient of performance with change in humidity ratio at different ambient temperatures. As the humidity ratio at process air inlet increases, in the presence of higher water vapor content in the process air stream, there is a major difference of vapor partial pressure between process

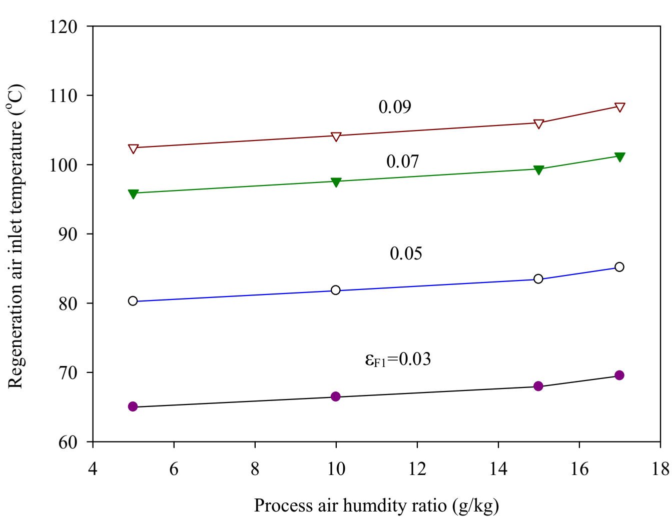

Fig. 9. Variations in regeneration temperature with process air inlet humidity ratio at different ε_{F_1}.

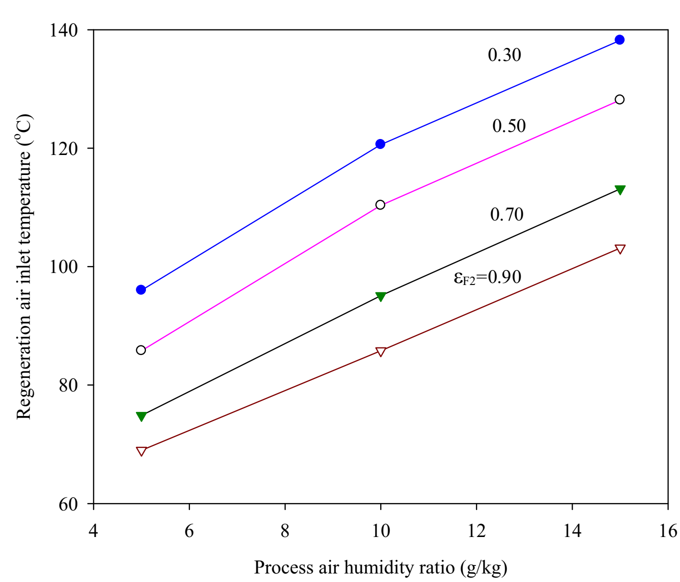

Fig. 10. Variations in regeneration temperature with process air inlet humidity ratio at different ε_{F_2}.

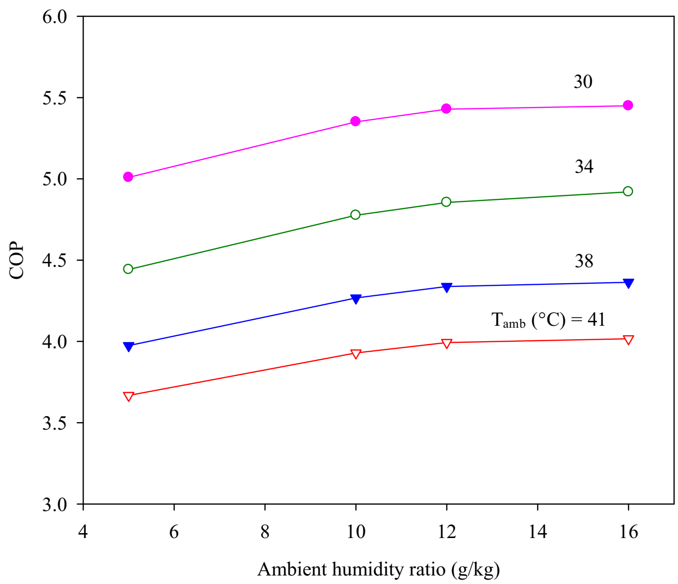

Fig. 11. Variations in C.O.P with ambient air humidity ratio at different ambient temperature.

air stream and desiccant felt leads to a higher diffusion of the water vapor droplets from the process air to the desiccant material surface. So, the reactivation air is heated to a slightly higher temperature to bring out the moisture extracted by the sorbent in the desiccant wheel. Thus, with rise in humidity ratio of process air at inlet, the desiccant wheel removes a bigger amount of water vapor, which results in the rise of process air temperature at outlet and finally results in an increase of the total cooling capacity. This causes the coefficient of performance of the system to increase at lower flow rates. This means that produced cooling effect is larger than the increases of energy input for heating the regeneration air.

Fig. 12 shows the effect of change in ambient temperature and humidity ratio on supply air temperature. As outdoor temperature and humidity ratio increases, the temperature of the process air stream flowing over evaporator coil is also increases. This is because as ambient temperature and humidity ratio increases, regeneration air temperature at inlet also increases (as ambient air is taken into

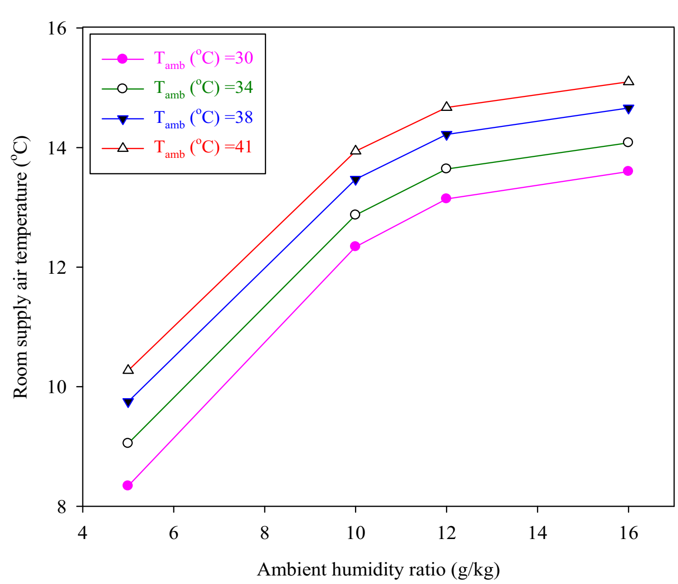

Fig. 12. Variations in room supply temperature with ambient air humidity ratio at different ambient temperature.

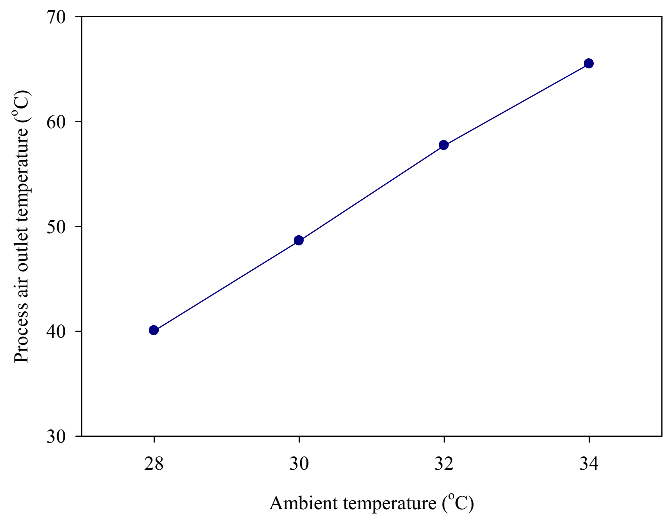

Fig. 13. Variation in process air outlet temperature with changes in ambient temperature.

regeneration air at inlet i.e. state point 6 as shown in Fig. 1). This also rises the dehumidifier process air outlet temperature by conveying that excess heat to the process air side. This is finally resulted into rise in room supply air temperature.

Figs. 13 and 14 show the effect of variation in ambient temperature on process air outlet temperature as well as regeneration air outlet temperature respectively. In both the cases, increase in ambient air temperature leads to increase in process air outlet temperature and regeneration air outlet temperature. This is due to increment in ambient temperature leads to increase in total cooling load of recirculation air leads to increase in required regeneration temperature needed for the adsorption and desorption in desiccant wheel. This can be explained in detail as increase in ambient air temperature induces increase in temperature of sorbent in the process air section, as a result of which less water content in the sorbent is required to keep humidity difference between the sorbent surface and the process air unchanged. Similarly, in the regeneration area, when the water content in the sorbent becomes less, the temperature of the sorbent must increase to keep the humidity difference constant. This means the required regeneration temperature is needed to be increased. Thus, when outdoor conditions

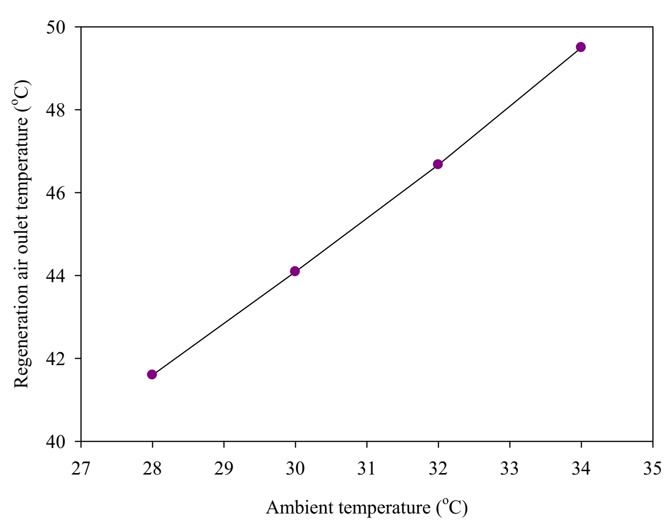

Fig. 14. Variation in regeneration air outlet temperature with changes in ambient temperature.

change, regeneration temperature must be adjusted to keep steady indoor conditions.

Due to high dependence of the performance of rotary desiccant dehumidifier on variations in ambient air conditions, further investigations are needed for a complete assessment of its feasibility in different climatic zones. Also, other possible system configurations (operating modes like ventilation or mixed, building types, locations and orientations, etc.) are suggested for consideration. Further research about desiccant materials that can be regenerated at lower temperature, sources from free energy such as solar or waster heat are the key of augmenting the contribution of the solid desiccant cooling system in the field of space cooling which can bring the amelioration of comfort, cost and energy savings.

7. Conclusion

Simulated results of hybrid solid desiccant – vapour compression air-conditioning system highlights good performance in hot and humid climates. The latent part of cooling load has been significantly reduced by the introduction of desiccant dehumidifier which increases the performance of the cycle. The analysis of variation in supply air condition for variation in ambient humidity ratio shows that the system is able to provide thermal comfort in hot and humid weather conditions and it can be an alternative for the conventional air conditioning systems. It was found that a marginal increase in coefficient of performance is obtained up to the regeneration temperature 100 °C. Further increase in the regeneration temperature does not positively influence the COP as compared to the amount of regeneration heat supply. Results also show that the system coefficient of performance is quite sensitive to changes in the regeneration temperature and humidity ratio. The change in the temperature of ambient air has great influence on the system performance. The performance of the hybrid solid desiccant vapor air-conditioning system can further be enhanced by coupling it with the solar energy or waste heat source for the supply of regeneration heat necessary for the desiccant wheel desorption. The variation in required regeneration temperature for the changes in potentials effectiveness ε_{F_1} and ε_{F_2} of dehumidifier model has been discussed.

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