Renewable and Sustainable Energy Reviews 60 (2016) 1451–1469

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

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

Solid desiccant air conditioning – A state of the art review

D.B. Jani *, Manish Mishra, P.K. Sahoo

Department of Mechanical & Industrial Engineering Indian Institute of Technology, Roorkee 247667, India

ARTICLE INFO

Article history:
Received 4 May 2015
Received in revised form
9 December 2015
Accepted 4 March 2016
Available online 22 March 2016

Keywords:
Desiccant wheel
COP
Regeneration temperature

ABSTRACT

Recently, solid desiccant air conditioning system has been proposed as an alternative to the conventional vapor compression refrigeration air conditioning systems for efficient control over humidity of conditioned air especially in hot and humid areas. The solid desiccant cooling can be more favorable over the traditional vapor compression refrigeration air conditioners, because it assures more accessible, economical and cleaner air conditioning. It is still more important when it is powered by free energy sources like solar energy and waste heat with temperatures of between 60°C and 80°C. In addition, it can significantly reduce the operating cost as well as save energy. In the present paper, principle of solid desiccant cooling system is recalled and its technological applications and advancements are discussed. Through a rigorous literature review, different configurations of desiccant cooling cycles, conventional and hybrid desiccant cooling cycles, different types of mathematical models of rotary desiccant dehumidifier, performance evaluation of desiccant cooling system, technological improvement and the advantage it can offer in terms of energy and cost savings are highlighted. This paper also gives a detailed account of the general features and performance of the solid desiccant cooling system when it is powered by solar energy or industrial waste heat for regenerating the desiccant. This review is useful for making opportunities to further research of solid desiccant cooling system and its feasibility which is becoming common in the coming days.

Contents

* Corresponding author.
E-mail addresses: dbjani@rediffmail.com (D.B. Jani), mmishfme@iitr.ac.in (M. Mishra), sahoofme@iitr.ac.in (P.K. Sahoo).

http://dx.doi.org/10.1016/j.rser.2016.03.031
1364-0321/

1. Introduction

One of the prime concerns of the present age is the day by day increase in energy consumption for space cooling applications. Nowadays, space conditioning in most parts of the world is done by conventional vapor compression refrigeration based air conditioners which require large amount high grade electrical energy for its running. The use of vapor compression refrigeration based air conditioners has led to increased CFC levels resulting to depletion of the ozone layer. Production of electricity in power stations has also been created many environmental issues, among them global warming. Further, the constant increase in the demand for space-cooling applications, due to development of many parts of the world, has compelled researchers to investigate alternate technologies for air conditioning to overcome the above mentioned issues. Desiccant cooling is a newer approach to air conditioning that helps to resolve environmental and economic issues that results from the use of conventional vapor compression air conditioning systems. The desiccant cooling system maintains required indoor comfort by optimal use of thermal energy with least electrical power. The desiccant cooling can be used either in a standalone system or coupled judiciously with a vapor compression refrigeration air conditioning system and free energy like solar or industrial waste heat to achieve high performance over wide range of operating conditions. In hybrid system, more efficient cooling process would result if a vapor compression refrigeration air conditioner and a desiccant cooling system were combined. This is because in hybrid system, first the desiccant dehumidifier efficiently removes the moisture from fresh ventilated air before it enters into the conditioned space while the vapor compression system removes only sensible heat then after. This type of arrangement rules out the requirement of low dew point temperature of evaporator cooling coil and subsequently reheating. It also avoids the condensation problem, occurs during excess humid ambient conditions. The use of hybrid desiccant cooling system not only controls the humidity but also reduce operating costs and electric power demand. Thus, the 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 by controlling temperature and humidity independently. Its operating costs can be reduced further by the use of low grade heat energy like solar, waste heat and natural gas. This is because the use of solar or waste thermal energy for heating the reactivation air which is used for desorption of desiccant wheel can help to alleviate the problem of high electricity consumption of regeneration heater. The peak cooling demand in summer is associated with high solar radiation which offers an excellent opportunity to exploit solar assisted desiccant cooling technology. Desiccant cooling can meet the current demands of occupant comfort, energy saving along with operational cost reduction and finally and the most important is environmental protection has taken it out from research niche to broader industrial applications like pharmaceutical, hospitals, supermarkets, restaurants, theater, schools and office buildings.

The main objective of this review is to find an optimal use of solid desiccant cooling system for hot and humid climate. It also summarizes recent research developments related to solid desiccant cooling system and to provide information for its potential application. It also contains a brief comparison related to performance of various systems having different environment conditions as well as operating parameters. The paper is categorised by type of cooling element, auxiliary regeneration heat source, operating mode used in the system configuration. An attempt has been made to gather up the conventional standalone solid desiccant cooling system and hybrid solid desiccant cooling system. Hybrid desiccant cooling combining additional application of solar/waste heat as well as vapor compression refrigeration system to conventional solid desiccant cooling system for the performance enhancement of overall system.

2. Overview of solid desiccant cooling

2.1. Brief history of solid desiccant cooling

Several studies have been carried out on desiccant cooling system by the different researchers. Pennington [1] proposed the earliest desiccant cooling cycle by coupling the rotary desiccant dehumidifier with heat source and evaporative cooler. Similar cycle was also proposed by Dunkle [2] using dehumidifier of molecular sieve with additional heat exchanger. Later on Munter [3] improved the performance of the desiccant cooling cycle by introducing parallel passages in dehumidifier and provided backup of vapor compression system to tackle cooling load if desiccant cooling cycle does not meet the cooling demand. Jurinak [4] has simulated desiccant system by incorporating the component models into TRNSYS software. He has proposed new combined potential technique for numerical solution of rotary dehumidifier. Later on, Pesaran [5] has outlined use of solar and waste heat to increase cycle efficiency [6]. Since then, a number of efforts have been made in order to ameliorate the performance of desiccant dehumidifiers and different configurations of the desiccant cooling systems.

2.2. Principle of solid desiccant cooling

A desiccant is hygroscopic substance having high affinity to water. Desiccants may be solid or liquid. Silica gel, LiCl, Molecular sieves etc. are commonly used as solid desiccants. Solid desiccant cooling system operates on principle of adsorption of water vapor from air. In solid desiccant cooling system, the moisture in ventilated/recirculated process air is first removed by a rotating desiccant wheel. The temperature of this dried process air is then lowered further to the desired room conditions by use of sensible heat exchangers and cooling coils. To make the system working continually, amount of water vapor adsorbed by the rotating desiccant wheel must be driven out of the desiccant material so that it can be dried enough (regenerated) to adsorb water vapor in the next cycle. This is done by heating the desiccant material to its temperature of regeneration which is dependent upon the type of the desiccant used. Energy required for regeneration of rotary desiccant wheel is supplied via regeneration heat source either by electrical heater or solar/waste heat. A desiccant cooling system, therefore, comprises principally four components, namely the regeneration heat source, the rotary dehumidifier, sensible heat exchanger and the cooling unit (Fig. 1). The possible configurations and the composition of each of the four components can vary largely according to the nature of the desiccant employed as described in the following.

2.3. Types of solid desiccant cooling cycles

First solid desiccant cooling cycle was introduced by Pennington as shown in Fig. 2. It has been also known as ventilation cycle [7]. On the process air side, ambient air at state point 1 passes through a desiccant wheel where its moisture is removed and temperature is increased up to state 2 due to the adsorption heat effect. It sensibly cooled during process 2–3 in air to air sensible heat exchanger. Then after, the process air is cooled by evaporation to supply air state 4 by passing through direct evaporative cooler. On the regeneration side, return air at state point 5 is cooled and

Nomenclature

A_{coll}	solar collector area (m^2)
COP	coefficient of performance
COP_H	thermal coefficient of performance
COP_T	total coefficient of performance
COP_{tf}	coefficient of performance of solar hybrid system
COP_w	total work coefficient of performance
d_e	hydraulic diameter (m)
C_p	specific heat (J/kg K)
DEC	direct evaporative cooling
E_{tot}	total energy consumption (kW)
EC	evaporative cooling
f	desiccant fraction in wall or in the matrix
h	air-side convective heat transfer coefficient (w/m^2 K)
h	enthalpy (kJ/kg)
k	thermal conductivity (W/m K)
K_y	gas-side mass transfer coefficient (kg/m^2 s)
L	actual process channel length (m)
m	mass flow rate of air stream (kg/s)
PVT	pipe vacuum tube
Q_c	cooling load (kW)
Q_{cl}	total cooling load (kW)
Q_{COMP}	energy consumption for compressor (kW)
Q_{EVAP}	evaporator cooling load (kW)
Q_s	cooling load in hybrid solar system (kW)
Q_{zs}	regeneration heat load (kW)
RHSF	room sensible heat factor
r	radial direction
T	temperature (K)
t	time (s)
u	air velocity (m/s)
VC	vapor compression
VCOP	coefficient of vapor compression
VCR	vapor compression refrigeration
VCS	vapor compression system
W	humidity ratio (g/kg)
Y	absolute humidity ratio (g/kg)
z	axial direction

Greek letters

η	efficiency
ε	effectiveness
δ	thickness of desiccant felt (m)
β	solar radiation (W/m^2)
ϕ	circumferential direction

Subscript

a	dry air
DW	desiccant wheel
d	desiccant
EC	evaporative cooler
HRW	heat recovery wheel
m	matrix
p	process air
r	regeneration air
wb	wet bulb

humidified in another direct evaporative cooler. This air is then sensibly heat exchanged with process air to pre cool the process air and preheats itself. The warm air stream is then heated up further to the required regeneration temperature of desiccants used in rotary dehumidifier via heater. After regenerating the desiccant wheel, this air is then exhausted to ambient at state point 9. A certain portion of the return air steam at the state point 7 bypasses the heater in order to reduce the reactivation heat consumption.

To elevate cooling capacity, recirculation cycle shown in Fig. 3 which is a modified form of Pennington cycle is designed to reuses the room return air as a dehumidifier process air inlet in hot and humid ambient conditions [7].

Dunkle cycle combines the merits of ventilation cycle, having supply of large amount of fresh air with relatively low temperature and recirculation cycle, having higher cooling capacity. As shown in Fig. 4, an additional sensible heat exchanger is incorporated which can provide colder process air with relatively low temperature for the heat exchanger [8].

Large amount of ventilated fresh air which has been provided in Pennington cycle for air conditioning was not only meant for comfort and health but also represents an additional cooling load. In some cases, it is not necessarily required that ambient air should be the source of air supply to the system. Hence, supply fresh air proportion should be maintained at the required level to ensure both the favorable system performance and better indoor air quality. In view of this, Maclaine-cross has proposed a simplified advanced solid desiccant cycle namely SENS cycle [9]. As shown in Fig. 5 ambient air is first dehumidified in desiccant wheel. Then, this air is sensibly cooled by passing through the two sensible heat exchangers which are connected in tandem. Afterwards, it is mixed with certain amount of room return air and cooled further in a cooling coil by exchanging heat with cold water from a cooling tower. Then supply air is divided into two parts, one part is redirected to cooling tower and exhausted to

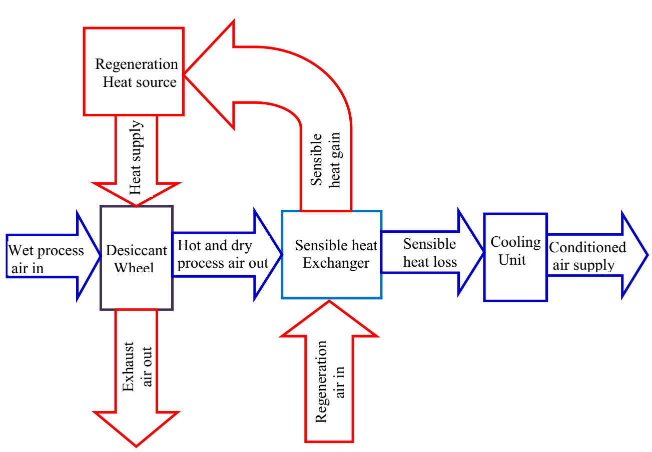

Fig. 1. Principle of solid desiccant cooling.

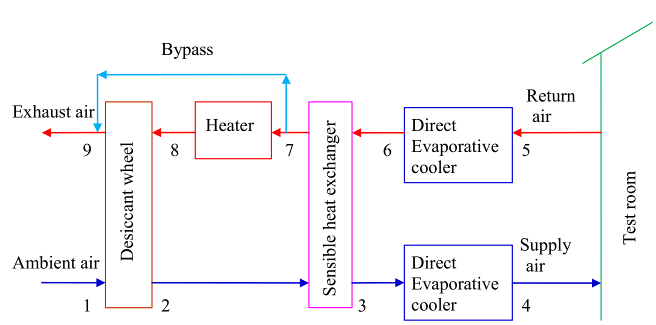

Fig. 2. Pennington cycle.

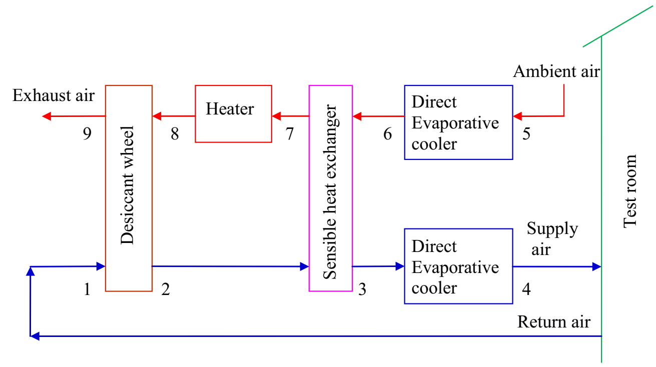

Fig. 3. Recirculation cycle.

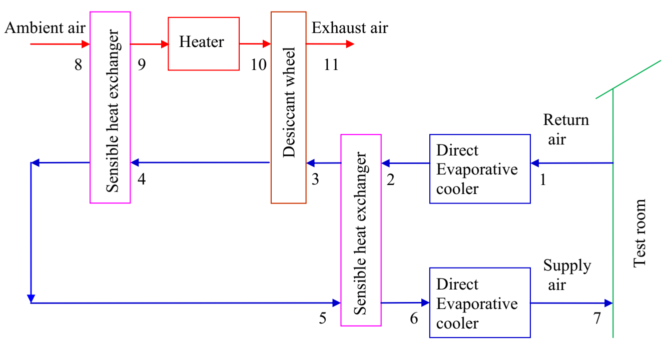

Fig. 4. Dunkle cycle.

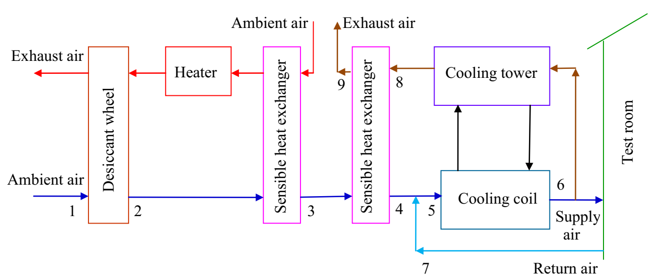

Fig. 5. SENS cycle.

ambient after exchanging heat with process air in a sensible heat exchanger, the other part is supplied to the conditioned space.

Fig. 6 depicts the direct–indirect evaporative cooling (DINC) cycle in which little modification over SENS cycle is done by replacing sensible heat exchanger, cooling tower and cooling coil with a pair of an indirect evaporative cooler and a direct evaporative cooler to avoid complexity and to simplify system configuration [9]. More ever, the thermal coefficient of performance of the DINC cycle has been obtained around 1.6.

Fig. 7 shows comparison between various solid desiccant cooling cycles on psychrometric chart. It has been observed from the psychrometric chart that the regeneration temperature required for desorption of desiccant wheel in case of recirculation cycle is highest due to higher reactivation air side humidity ratio. The temperature of supply room air is observed lowest in case of recirculation cycle because it reuses 100% building air as a process air inlet to the desiccant wheel. In case of Pennington cycle, maximum amount of fresh ventilated air supply is possible because it uses 100% outdoor air as a process air at inlet to the desiccant wheel. But, in case of ventilation cycle thermal coefficient of performance and specific cooling capacity would be reduced in comparison with other standard cycles because humidity ratio and temperature of

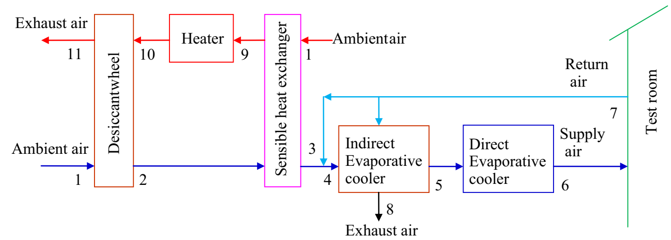

Fig. 6. DINC cycle.

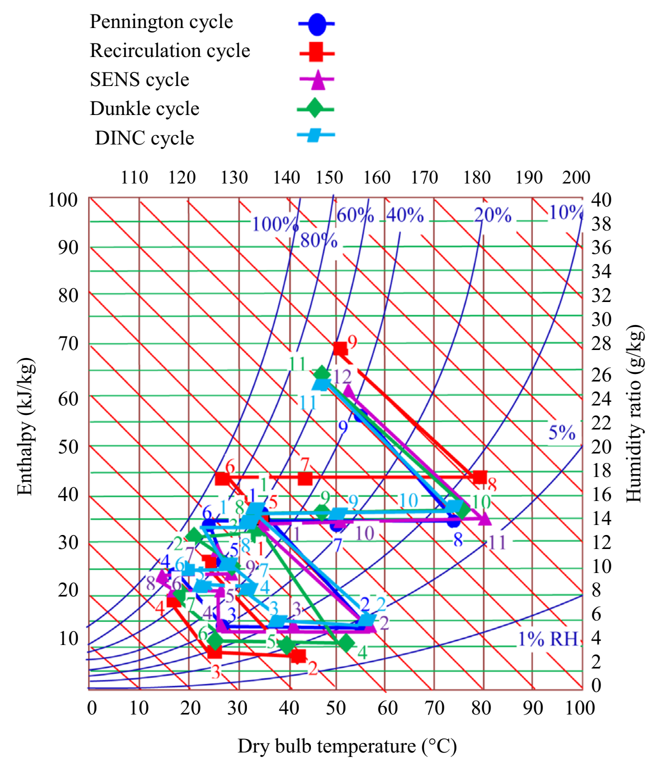

Fig. 7. Comparison between various solid desiccant cooling cycles on psychrometric chart.

outdoor air are usually higher than that of room recalculated air. SENS cycle can achieve highest thermal coefficient of performance because of tandem arrangement for two sensible heat exchangers. However, this cycle is blocked by its complexity [9].

2.4. Progress in solid desiccant cooling system configuration

To provide an overview of progress in solid desiccant cooling system configuration, the conventional solid desiccant cooling cycles operating in different modes namely ventilation, recirculation and mixed along with the recent advancement to basic cycles which is commonly known as hybrid cycles have been described in this section. Hybrid system has been designed to supplement the conventional stand alone solid desiccant cooling judiciously with conventional vapor compression cooling as well as or by solar heating to enhance the overall performance of the conventional system operating under varying ambient conditions.

2.4.1. Solid desiccant evaporative cooling system

In the solid desiccant cooling system, a rotary desiccant wheel is integrated with the sensible heat exchangers and evaporative coolers. The evaporative coolers used in the conventional system are either direct evaporative coolers or indirect evaporative coolers according the type of working climate i.e. dry or humid. The

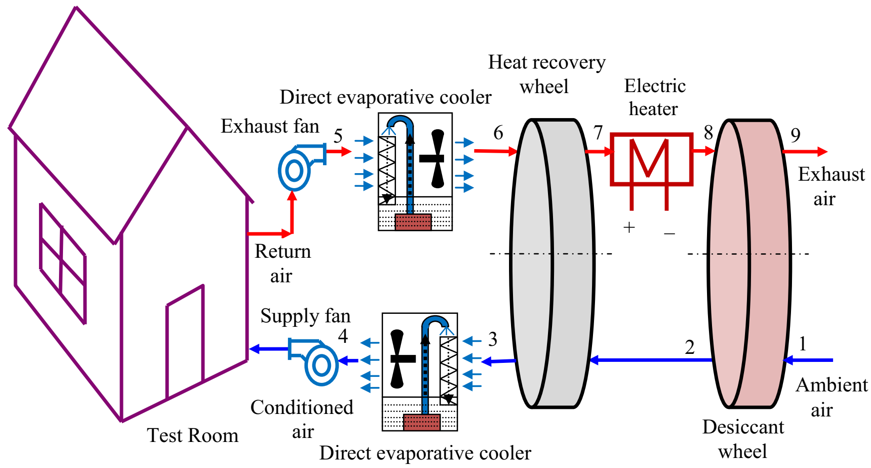

Fig. 8. Conventional solid desiccant evaporative cooling system (ventilation mode).

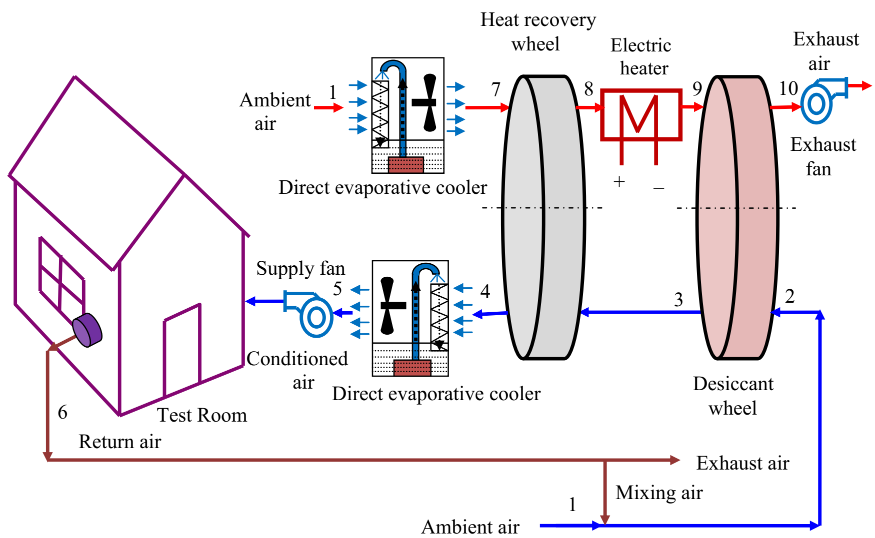

Fig. 9. Solid desiccant evaporative cooling system (recirculation mode).

system described here is operating in both the ventilation as well as the recirculation mode as shown in Figs. 8 and 9 was suggested by Bourdoukan et al. [10]. In ventilation mode, humidity of the fresh outside air (state 1) is adsorbed by the desiccant material of the wheel and becomes hotter and drier at exit (state 2).

Sensible cooling of this hot and dehumidified air steam is carried out first in air to air sensible heat recovery wheel from the state 2 to state 3. Between state 3 and state 4 air is cooled to required supply room condition with help of direct evaporative cooler and supplied to conditioned space then after. Another direct evaporative cooler is used to cool down the return air from state 5 to state 6 as this cold air stream serves as heat sink to cool supply air in the sensible heat exchanger. Consequently, its temperature is risen when exiting the sensible heat exchanger at the state point 7. This preheated air is then heated up to the state point 8 i.e. required regeneration temperature necessary for desorption of desiccant material of dehumidifier. After regenerating the desiccant wheel, the air is then exhausted to ambient at state point 9. In case of recirculation configuration, the process air stream is 80% of return air from the building and 20% of fresh outside air, while the regeneration air stream is 100% outside air as shown in Fig. 9.

Comparison between ventilation and recirculation configurations of evaporative solid desiccant cooling system has been depicted in psychrometric chart as shown in Fig. 10. It is observed from the behavior that the recirculation configuration requires more energy for regenerating the dehumidifier at higher ambient humidity, since having low temperature at the inlet of desiccant wheel and thus higher regeneration power. But this increase in reactivation power is not very significant as compared to the increase in cooling capacity of system. Because it supplies air to the room at lower temperature and the impact of decreasing the inlet humidity of the desiccant wheel is greater than the impact of increasing the inlet humidity of return humidifier. That is why the coefficient of performance remains higher in the case of recirculation configuration. Thus, the conventional configuration is more suitable for low outside humidity ratios while the recirculation configuration is more effective for high outside humidity ratios. So, the optimal configuration of the system should be selected with respected to the change in outside ambient conditions for better operational economy.

2.4.2. *Solar assisted solid desiccant evaporative cooling system*

Solar assisted conventional solid desiccant evaporative cooling system has been developed as one of the promising alternatives to the conventional solid desiccant evaporative cooling system which consumes lot of electrical power for operating the regenerative heater. Other construction is same as the configuration described above except the solar heating system have been incorporated in the present system in place of electric heater as a regeneration heat source. It includes solar collector, back up heater, storage tank, circulating pump and liquid to air heating coil. Generally, the solar collectors convert solar radiation into thermal energy in solar desiccant cooling systems. But in the case when insufficient thermal energy from solar collector in cloudy weather, the backup heater is used to achieve the required thermal energy for the total cooling system driving energy. The percentage ratio of the thermal energy produced by the solar collectors to the total cooling system driving energy is known as the solar fraction, which can be expressed [11] as follows

$$SF = \frac{Q_U}{Q_T} \tag{1}$$

where Q_T is the total cooling system driving energy produced by the solar collectors and the backup heater. Q_U is the thermal energy produced by the solar collectors that can be expressed [11] as

$$Q_U = A_{ST} \times \eta_{coll} \times I_\beta \tag{2}$$

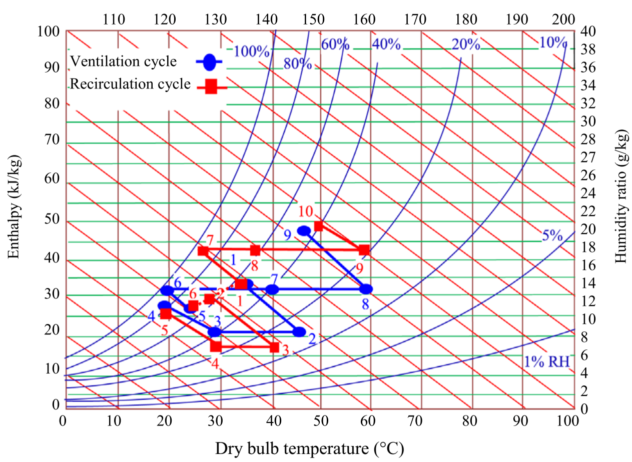

Fig. 10. Comparison between ventilation and recirculation configurations in solid desiccant evaporative cooling system.

Fig. 11 shows the ventilation configuration of the solar desiccant cooling is an open cycle that provides supply air to the room from ambient air and return room air is used as regeneration air. Fig. 12 shows the recirculation configuration of the solar desiccant cooling system in which the process air side is a closed loop while the regeneration air side is an open cycle. Both the cycles described above were designed by Dezfouli et al. [11].

Fig. 13 shows comparison between ventilation and recirculation configurations for the solar assisted solid desiccant cooling system on psychrometric chart. By comparing the temperature results of both the configuration, it is found that the reactivation temperature under the ventilation configuration is found to be considerably higher than that of the recirculation configuration. Although the cooling system driving energy in ventilation configuration was higher than the recirculation configuration, the rate of cooling capacity in ventilation configuration was found higher than the recirculation configuration. The temperatures of supply air under both the configurations are almost the same. In addition, the room temperatures under both the configurations are almost same.

Summary of performance of solar assisted solid desiccant evaporative desiccant cooling system has been tabulated below which includes types of solar collector, area of solar collector, storage tank volume and performance index. It is based on the previous research work carried out by many researchers according to various climatic zones existing in different parts of the world (Table 1).

2.4.3. Solid desiccant evaporative cooling with waste heat recovery system

The rotary desiccant dehumidifier used in solid desiccant cooling system may be regenerated through heat recovery carried out either from the exhaust of large internal combustion engine (as shown in Fig. 14) or from micro CHP unit. This resulted in enhancement of both the cooling performance of desiccant dehumidification system and better fuel utilization with higher thermal efficiency of engine. Moreover, in solid desiccant vapor compression hybrid air conditioning system, the regeneration heat needed by the desiccant wheel is supplied by the condenser

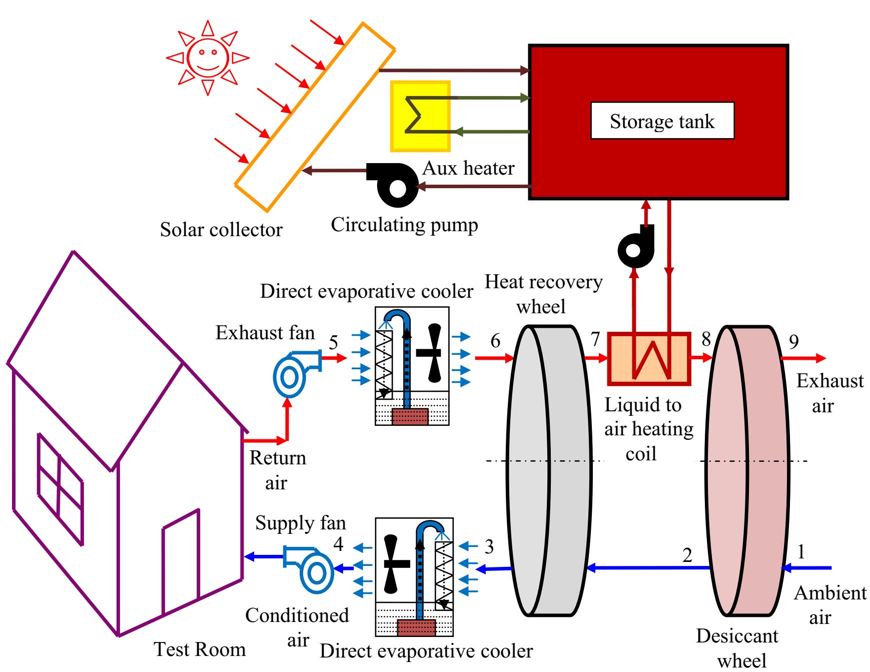

Fig. 11. Solar assisted solid desiccant evaporative cooling system (ventilation mode).

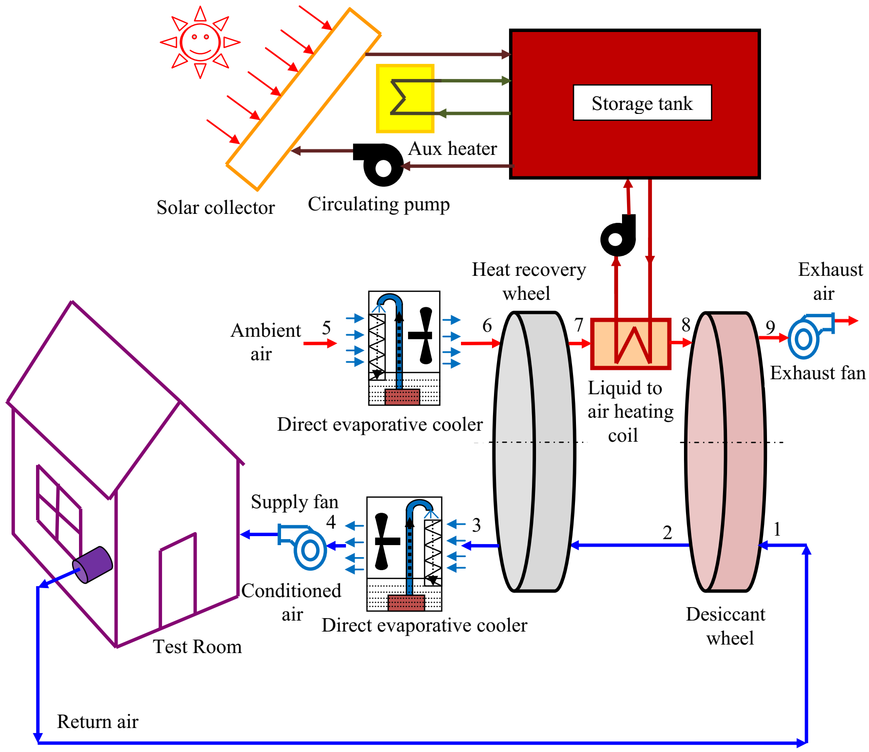

Fig. 12. Solar assisted solid desiccant evaporative cooling system (recirculation mode).

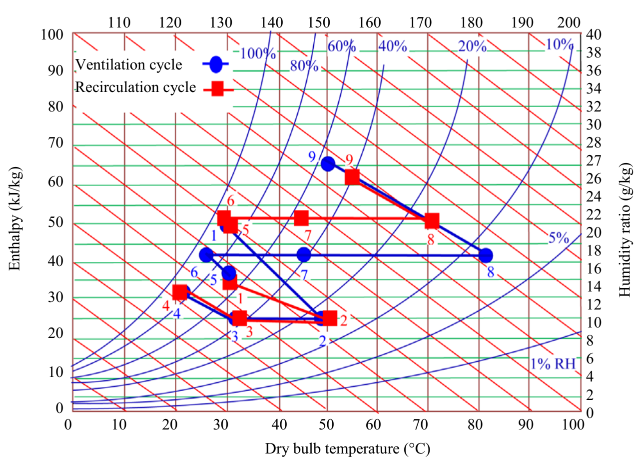

Fig. 13. Comparison between ventilation and recirculation configurations in solar assisted solid desiccant evaporative cooling system.

dissipated heat as shown in Fig. 15. Summary of different studies related to desiccant cooling system using waste heat for the regeneration of rotary dehumidifier tabulated below in Table 2. These results prove that the utilization of waste heat for the regeneration of desiccant wheel not only achieves great energy saving, but also marginal reduction in running cost of the system. Furthermore, the efforts are still needed to develop desiccants which are regenerated to ambient temperature to take account of the low temperature waste heat. This also offered benefit in system COP as compared to that of the traditional system.

2.4.4. Hybrid solid desiccant vapor-compression air-conditioning system

Conventional solid desiccant cooling system as described above can efficiently handle the sensible heat load of dehumidified the process air by carried out its sensible cooling in direct or indirect evaporative coolers. In case of hot and humid climates, the performance of this systems would be degrade due to possible dehumidification may not be high enough to enable the evaporative cooling of supply air. To quantify the output or quality supply air, other air conditioning technologies such as traditional vapor compression air conditioning unit should be incorporated to present configuration to constitute hybrid system as shown in Fig. 16. As we can see, the latent heat load and sensible heat load are removed separately or independently by desiccant wheel and evaporator cooling coil respectively. Therefore, both the performance and applicability would be improved significantly. Performance of such hybrid systems can further improve by making the use of rejected heat of the condenser for partial regeneration of the desiccant. It eliminates the use of sensible heat exchanger in the cycle. Figs. 16 and 17 shows hybrid solid desiccant vapor compression air conditioning systems studied by Yadav and Kaushik [35] in ventilation and recirculation configurations respectively. In recirculation configuration as shown in Fig. 14, the air from ambient space is mixed with some portion of return room air (state 1) and processed though desiccant dehumidifier while remaining return room air is exhausted to atmosphere. The desorption process in the dehumidifier causing hot and dry air to leave the dehumidifier at state 2. The air-stream is then sensibly cooled by an indirect evaporative cooler up to state 3. Then after, the sensible cooling is performed further according to conditioned space requirement by passing it over the VCR sensible cooling coil and conditioned air is then supplied to the room (state 4). To regenerate the desiccant material of dehumidifier, waste condenser heat is utilized to preheat the ambient air (state 7). This preheated air is then heated up to the state point 8 i.e. required regeneration temperature necessary for desorption of desiccant material of dehumidifier. After regenerating the desiccant wheel, the air is then exhausted to ambient at state point 9. Ventilation configuration is similar to recirculation one except that only the ambient air (state 1) is processed through regeneration line while the part of room return air is mixed with evaporative cooler

Table 1
Summary of performance of solar assisted solid desiccant evaporative cooling systems.

Author	Type of solar collector	Area of solar collector (m^2)	Volume of storage tank (m^3)	COP_{th}
Khalid and Nabeel [12]	Flat plate (water)	5.4	–	0.5
Li et al. [13]	PVT (air)	105	0.65	0.51
Kabeel [14]	Porous type solar air heater	1.2	0.3	0.9
Bourdoukan et al. [15]	Evacuated tube (water)	300	–	–
Bourdoukan et al. [16]	Evacuated tube (water)	40	2.5	0.4
White et al. [17]	Flat plate (water)	100	30	0.5
Enteria et al. [18]	Flat plate (water)	10	0.3	0.25
Fong et al. [19]	Flat plate (air)	100	–	1.38
Ge et al. [20]	Evacuated tube (water)	550	–	1.28
Li et al. [21]	Evacuated tube (air)	120	–	0.8
Preisler and Brychta [22]	Flat plate (water)	285	15	–
Li et al. [23]	Evacuated tube (air)	92.4	–	0.34

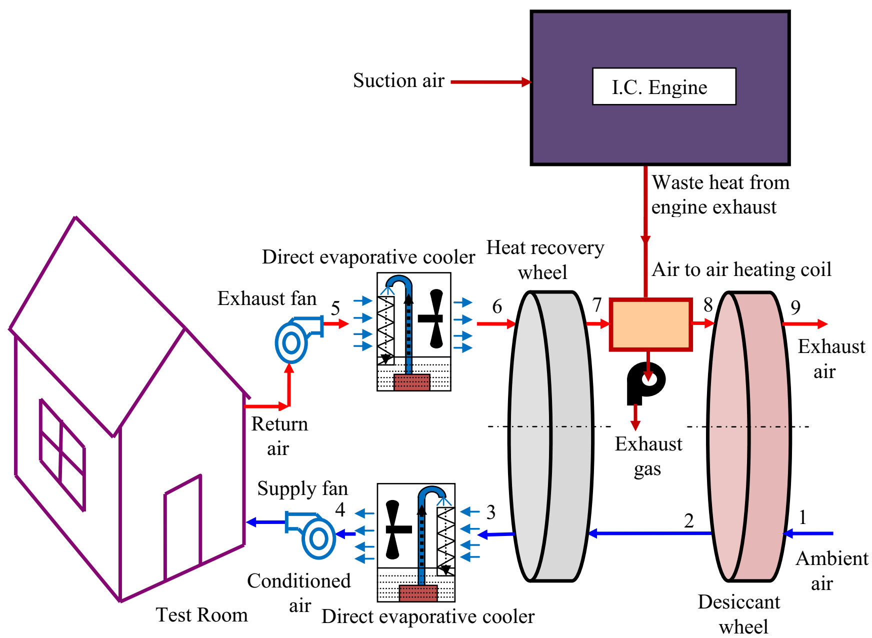

Fig. 14. Solid desiccant cooling system with waste heat recovery system.

process air outlet (state 4) and remaining return room air is exhausted to ambient. It can be seen in Fig. 16.

Fig. 18 shows comparison between ventilation and recirculation configuration for hybrid solid desiccant vapor compression air conditioning on psychrometric chart. In recirculation cycle, mixture of room return air as well as fresh ventilated air is passed through the dehumidifier. Consequently, the dehumidifier size needed is smaller since it has to handle smaller volumes of air and so, quantity of moisture to be removed per kg of air is slightly lower which in turn needed lesser regeneration temperature required to desorbed the desiccant material in desiccant wheel. In hot and humid outdoor conditions (RHSF=0.35) recirculation cycle also has lower weighted energy consumption as compared to ventilation configuration.

2.4.5. Solar assisted hybrid solid desiccant vapor-compression air-conditioning system

Regenerating the solid desiccant dehumidifiers with help of renewable heat sources like solar thermal energy not only reduce the electricity consumption but also achieve the substantial fossil energy saving. The solar air heating for regeneration is an interesting option to converge the cooling demands of conditioned space as the available intensity of solar energy and demand for cooling are greatest during the same period. It has the same construction as solar assisted solid desiccant evaporative cooling system as discussed earlier except in process air line direct evaporative cooler replaced by means of the auxiliary cooling coil powered by a vapor compression refrigerator as shown in Fig. 19 was described by Gagliano et al. [36]. The desiccant is regenerated by solar thermal energy produced by vacuum tube collectors.

Fig. 20 shows psychrometric chart representation of solar assisted hybrid solid desiccant vapor compression air conditioning cycle. It is observed that the required regeneration temperature for the desiccant dehumidifier is comparatively low and that can be efficiently maintained by the solar thermal collectors. It also depicted that there is no additional dehumidification is the required to reach the required humidity ratio as ambient humidity ratio of the outside air is comparatively low. Further, it neglects any post heating of supply air since the chiller works at higher evaporator temperatures. The final supply air temperature is set according to conditioned space requirement by means of the auxiliary cooling coil of vapor compression refrigerator.

Summary of previous work carried out by many researchers to evaluate the performance of solar assisted hybrid solid desiccant vapor compression air cooling system based on experimental work has been tabulated below which includes types of solar collector, area of solar collector, storage tank volume and coefficient of performance (Table 3).

In addition to the various configurations of solid desiccant cooling cycles described above, it is vital to compare these various

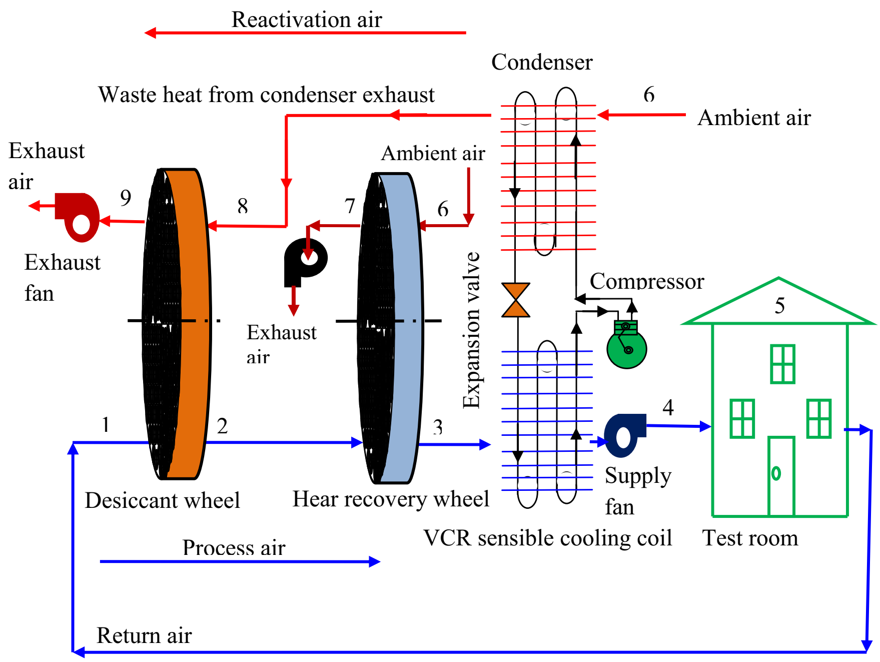

Fig. 15. Hybrid solid desiccant cooling system with VCR condenser heat recovery system.

Table 2
Summary of different studies related to use of waste heat for solid desiccant cooling.

Study	System description	Method	System performance
Huan and Jianlei [24]	Two stage desiccant cooling system	Simulation	COP is 0.964. The cooling capacity of two stage increases by 11.5%.
Ando and Kodama [25]	Evaporative desiccant cooling system with double stage dehumidification	Experimental	COP varies from 0.45 to 0.5
Azar et al. [26]	CHP integrated desiccant cooling system	Experimental	The electrical COP of the system is around 5.
Qin and Schmitz [27]	Engine driven hybrid air conditioning system	Experimental	30% saving in operation cost of the system by use of engine waste heat.
Kohlenbach et al. [28]	Evaporative desiccant cooling system driven by micro-turbine waste heat	TRNSYS simulation	COP is around 3.0 and emission saving of 58.6 t CO_2/yr.
Angrisani et al. [29]	Desiccant HVAC system coupled to a small size cogenerator	Experimental	35% reduction in CO_2 emission and 18% reduction in electrical power consumption
Ge et al. [30]	Air source heat pump (ASHP) system having use of condenser dissipated heat for desiccant regeneration	Simulation and experimental	8% primary energy saving in cooling mode while 14% energy saving in heating mode
El-Agouz and Kabeel [31]	Desiccant air conditioning system with geothermal energy	Experimental and simulation	COP based on primary energy consumption varies from 1.03 to 0.15
Ying et al. [32]	Desiccant wheel integrated heat pump system	Experimental	COP varies from 0.56 to 2.5
O'Kelly et al. [33]	Desiccant assisted hybrid air/water conditioning system/ HAWC heat pump system	Simulation	System used 5.61 kwh/(m^2 year) as compared to 9.5 kw h/(m^2 year) for traditional HVAC system
Angrisani et al. [34]	Geothermal heat source integrated desiccant wheel and air handling unit	TRNSYS simulation	Primary energy saving increases from 77% to 95% and payback period decreases from 14 to 1.2 years

system configurations on a common platform of overall system performance on basis of energy consumption is discussed below. Hong et al. [47] have compared various performance parameter of air cooling systems like first composed of conventional vapor compression (VC), the second composed of vapor compressor and desiccant (VC+D) cooling system and the third was composed of vapor compressor, desiccant and direct evaporative cooler (VC+D+EC) cooling system as tabulated in Table 4.

2.5. Mathematical models for rotary desiccant dehumidifier

In solid desiccant cooling system, the performance of the rotary desiccant dehumidifier is critical to the capability, size and operating cost of the whole system. In generalized desiccant wheel model system is considered to be one dimensional and Euler cylindrical coordinate system is usually employed to describe the wheel. Explanations of main term in governing equations are presented below. The physical model described by Ge et al. [48] is shown in Fig. 21.

The moisture conservation in the air can be described as

$$d_e\rho_a\left(\frac{\partial Y_a}{\partial t}+u\frac{\partial Y_a}{\partial z}\right)=K_y(Y_d-Y_a) \tag{3}$$

The energy conservation for air can be expressed as

$$d_e c_{pa}\rho_a\left(\frac{\partial T_a}{\partial t}+u\frac{\partial T_a}{\partial z}-\frac{k_a}{c_{pa}\rho_a}\frac{\partial^2 T}{\partial z^2}\right)=h(T_d-T_a)+c_{pv}K_y(Y_d-Y_a)(T_d-T_a) \tag{4}$$

The moisture conservation in desiccant can be written as

$$\rho_d\delta\left(\frac{\partial W}{\partial t}-D_e\frac{\partial^2 W}{\partial^2 z}\right)=K_y(Y_d-Y_a) \tag{5}$$

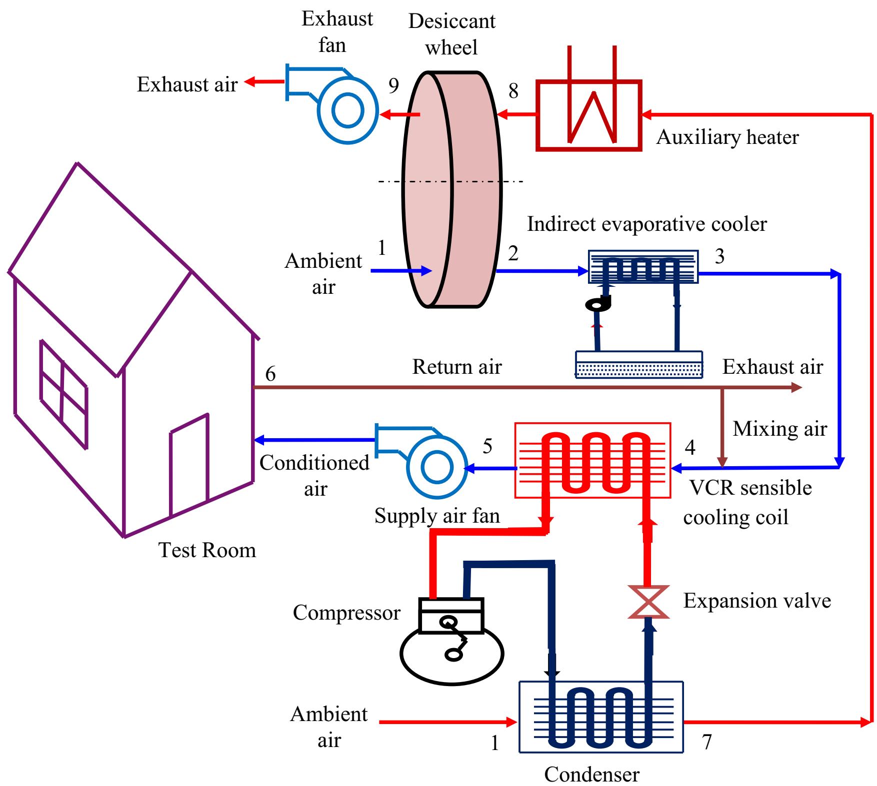

Fig. 16. Hybrid solid desiccant vapor compression air-conditioning system (ventilation mode).

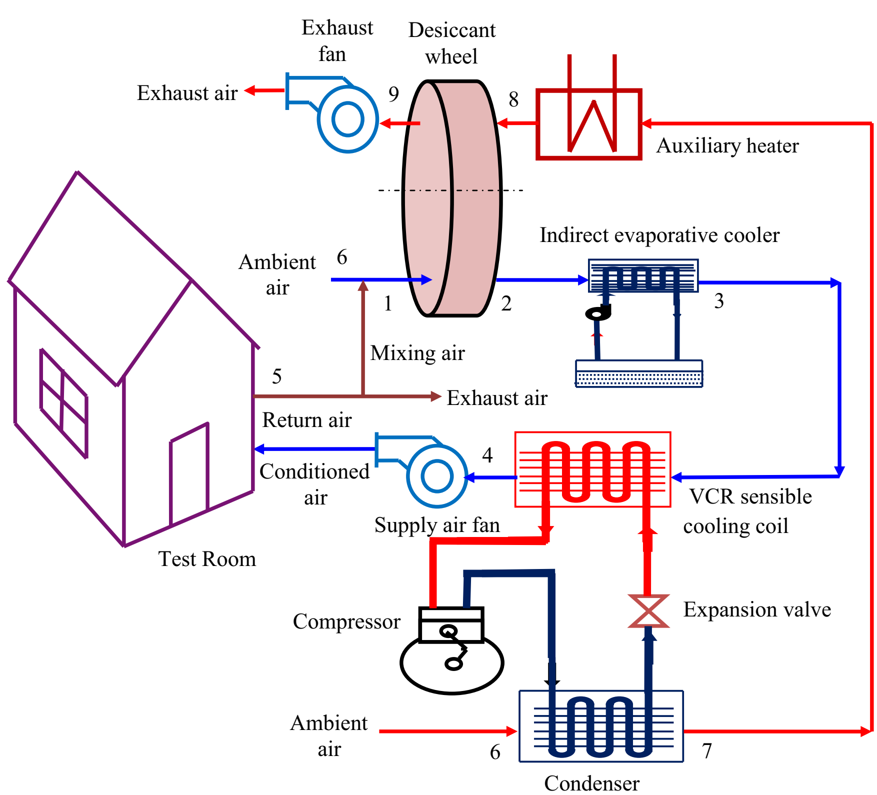

Fig. 17. Hybrid solid desiccant vapor compression air-conditioning system (recirculation mode).

The energy conservation in desiccant can be given as

$$\rho_d c_{pd} \delta \left(\frac{\partial T_d}{\partial t} - \frac{k_d}{c_{pd} \rho_d} \frac{\partial^2 T}{\partial z^2} \right) = h(T_a - T_d) + c_{pv} K_y (Y_a - Y_d) q_{st}$$

$$+ c_{pv} K_y (Y_a - Y_d)(T_a - T_d) \quad (6)$$

Boundary conditions are given by the normal derivative of parameters on a surface, such as the process/reactivation air ($i = 1$

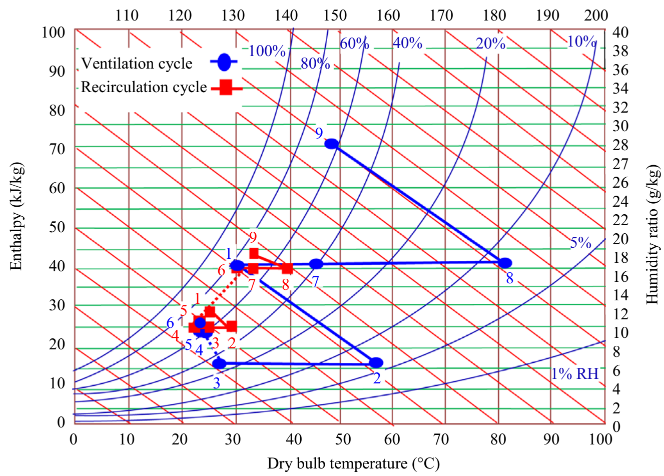

Fig. 18. Comparison between ventilation and recirculation configurations in hybrid solid desiccant vapor compression air conditioning system.

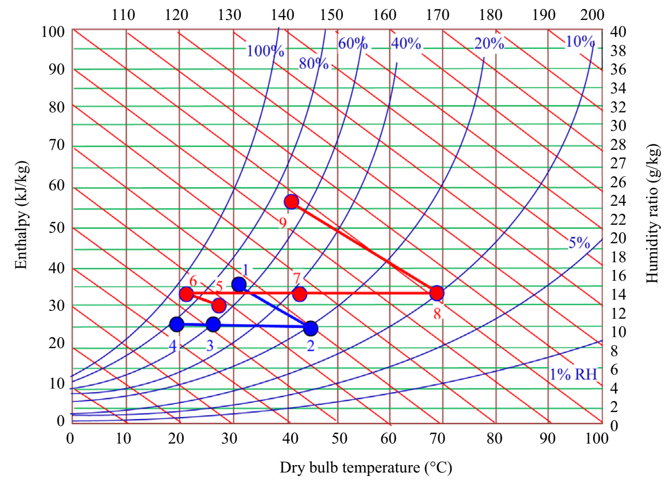

Fig. 20. Psychrometric chart representation of solar assisted hybrid solid desiccant vapor compression air conditioning.

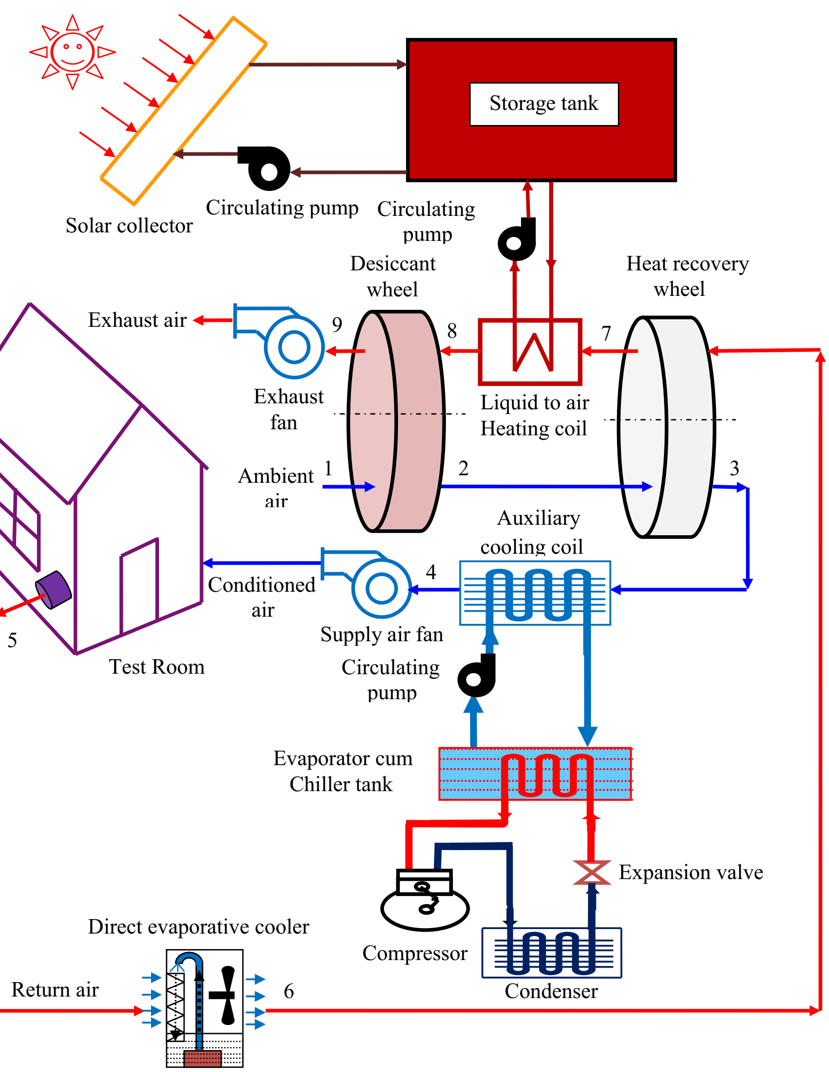

Fig. 19. Solar assisted hybrid solid desiccant vapor-compression air-conditioning system.

Table 3
Summary of performance data for solar assisted solid desiccant hybrid vapor compression air conditioning systems.

Author	Type of solar collector	Area of solar collector (m^2)	Volume of storage tank (m^3)	COP_{VCR}
Khalid et al. [37]	Solar (air)	12	–	0.5
Marco et al. [38]	PVT (air)	50	–	–
Fong et al. [39]	Evacuated tube (water)	100	5	1.05
Fong et al. [40]	Evacuated tube (water)	100	5	3.39
Fong et al. [41]	Flat plate (water)	500	–	0.86
La et al. [42]	Flat plate (water)	72	–	3.28
Finocchiaro et al. [43]	Flat plate (water)	22.6	0.65	1.8
Hurdogan et al. [44]	Flat plate (water)	–	–	1.0
Marco et al. [45]	Flat plate (water)	22.5	0.65	1.94
Jose et al. [46]	Flat plate (water)	500	–	–

Table 4
Performance comparison of VC, VC+D and VC+D+EC systems.

Sr. no.	Performance parameter	VC	VC+D	VC+D+EC
1	Evaporative temperature (°C)	3.67	6.78	10.8
2	Regeneration temperature (°C)	–	68	70
3	Cooling load (Q_{EVAP}/kW)	26.50	12.83	7.25
5	Regeneration heat load (Q_{ZS}/kW)	–	8.17	8.65
6	Energy consumption of compressor (Q_{COMP}/kW)	8.36	3.48	1.78
7	Energy consumption of whole loads for conventional hybrid system (Q_C/kW)	13.69	12.08	10.80
8	Energy consumption of whole loads for solar hybrid system (Q_S/kW)	30.14	18.62	13.98
9	VCOP	3.17	3.68	4.08
10	COP of Solar hybrid system COP_{tf}	–	1.40	0.92

for process air; $i=2$ for reactivation air)

$$T_{a,in}=T_{i,in} \tag{7}$$

$$Y_{a,in}=Y_{i,in} \tag{8}$$

The insulated and impermeable boundaries result in

$$\left.\frac{\partial T_m}{\partial r}\right|_{r=0}=\left.\frac{\partial T_m}{\partial z}\right|_{z=0}=\left.\frac{\partial T_m}{\partial z}\right|_{z=L}=0 \tag{9}$$

$$\left.\frac{\partial Y_m}{\partial r}\right|_{r=0}=\left.\frac{\partial Y_m}{\partial z}\right|_{z=0}=\left.\frac{\partial Y_m}{\partial z}\right|_{z=L}=0 \tag{10}$$

The periodic boundary conditions at $\phi=0$

$$\begin{aligned}Y_a(0,z,t)&=Y_a(2\pi,z,t)\\ T_a(0,z,t)&=T_a(2\pi,z,t)\\ W(0,z,t)&=W(2\pi,z,t)\\ T_d(0,z,t)&=T_d(2\pi,z,t)\end{aligned} \tag{11}$$

Initial conditions give the value of parameter at a given starting time ($t=0$). The initial condition for Y is expressed as follows

$$Y(t=0)=Y_0 \tag{12}$$

Detailed analysis is carried out by further study of energy and mass exchange between air and desiccant matrix which is complex diffusion phenomena in the following types.

2.5.1. Gas side resistance (GSR) model

In a desiccant dehumidifier, gas side resistance come from diffusion and heat conduction within the air and convective heat and mass transfer between air and desiccant. It influences the heat and mass transfer from the air stream to the surface of the desiccant felt. Due to diffusion and heat conduction within the air is small in comparison with convective heat and mass transfer, they are always neglected. One dimensional model has been given by Charoensupaya and Worek [49] by assuming channel adiabatic and impermeable as shown in Fig. 22.

The conservation of moisture and energy between the process air stream and matrix cane were

$$\rho_a a_a\frac{\partial Y_a}{\partial t}+\frac{m_a}{nX_m}\frac{\partial Y_a}{\partial z}+f\rho_m 2\delta\frac{\partial W}{\partial t}=0 \tag{13}$$

$$\frac{m_g}{X_m}\left(\frac{1}{u}\frac{\partial H_g}{\partial t}+\frac{\partial H_g}{\partial z}\right)+\frac{m_m}{X_m L}\frac{\partial H_m}{\partial t}=0 \tag{14}$$

Conservation of moisture in the matrix was

$$\frac{fm_m}{nX_m L}\frac{\partial W}{\partial t}=2K_y(Y_a-Y_m) \tag{15}$$

where m_m is the total mass of the matrix in the dehumidifier

Conservation of energy in the air was calculated by

$$\frac{m_g}{nX_m}\left(\frac{1}{u}\frac{\partial H_g}{\partial t}+\frac{\partial H_g}{\partial z}\right)=2K_y(Y_m-Y_a)\frac{\partial H_g}{\partial Y_a}+2h(T_m-T_a) \tag{16}$$

Boundary conditions are given by the process/reactivation air ($i=1$ for process air; $i=2$ for reactivation air)

$$T_{a,in}=T_{i,in} \tag{17}$$

$$Y_{a,in}=Y_{i,in} \tag{18}$$

$$T_{w,in}=T_{i,in} \tag{19}$$

$$Y_{w,in}=Y_{i,in} \tag{20}$$

2.5.2. Gas and solid side resistance (GSSR) model

The previously mentioned GSR models do not consider the heat conduction and mass diffusion within solid side, thus they do not reflect actual mass transfer process occurring in rotary desiccant dehumidifier. GSSR model is more preferable as it taken into account heat conduction and mass diffusion within solid desiccant felt including micro pore and macro pore diffusion and adsorption within the desiccant particles. According to type of diffusion phenomena in porous medium they can further be divided into following categories.

2.5.2.1. Pseudo-gas-side (PGS) model. It includes heat and mass transfer in adsorption/desorption phenomena using lumped heat and mass transfer coefficients obtained by experimental data which includes heat and mass transfer resistances in gas side as well as solid side. The control volume is depicted in Fig. 23 was the model described by Maclaine-Cross and Banks [50]. Mass and energy conservation equations in PGS models were given by

$$m_a\frac{\partial Y_a}{\partial z}+\frac{fm_m}{L}\frac{\partial W}{\partial t}=0 \tag{21}$$

$$m_a\frac{\partial H_a}{\partial z}+\left(\frac{m_m}{L}\frac{\partial H_m}{\partial t}-kA_m\frac{\partial^2 T_m}{\partial z^2}\right)=0 \tag{22}$$

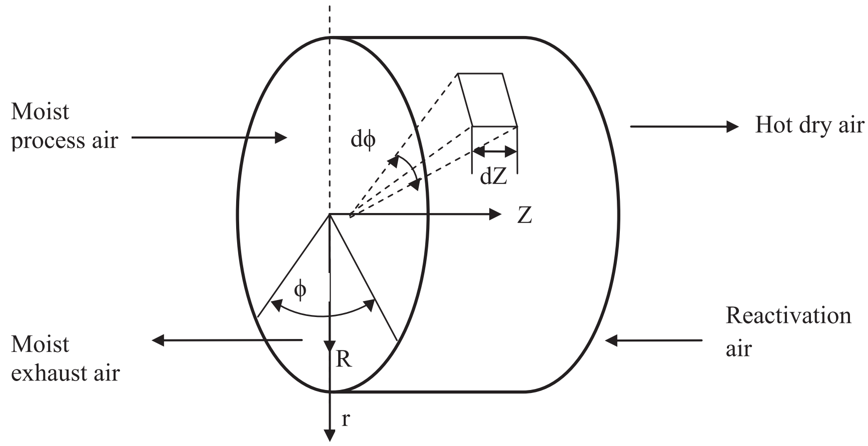

Fig. 21. Model of control volume of rotary dehumidifier in Euler cylindrical coordinate system.

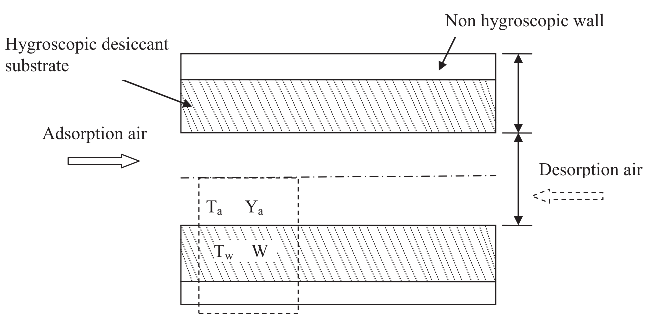

Fig. 22. GSR model of rotary desiccant dehumidifier.

Mass and energy transfer equations were described by

$$m_a\frac{\partial Y_a}{\partial z}=\frac{K'm_m}{L}(Y_m-Y_a) \tag{23}$$

$$m_a\frac{\partial H_a}{\partial z}=\frac{K'_yA}{L}\left[\frac{h'}{K'_y}(T_m-T_a)+h_{fg}(Y_m-Y_a)\right] \tag{24}$$

2.5.2.2. Gas and solid side (GSS) model. With the increasing knowledge of diffusion mechanism inside desiccant, GSS models are proposed in which heat and mass transfer diffusion terms in solid side are directly added. A two dimensional GSS model has been established by Charoensupaya and Worek [51] in which the heat conduction and mass diffusion including gas phase diffusion and surface diffusion in the radial direction within solid desiccant were considered. The control volume is illustrated in Fig. 24.

Conservation of moisture in the process air was expressed as

$$\frac{m_a}{X_m}\left(\frac{1}{u}\frac{\partial Y_a}{\partial t}+\frac{\partial Y_a}{\partial z}\right)=2K_y(Y_d-Y_a) \tag{25}$$

Conservation of moisture in desiccant felt was given by

$$\varepsilon_t\rho_{ad}\frac{\partial Y_d}{\partial t}+(1-\varepsilon_t)\rho_d\frac{\partial W}{\partial t}-D_G\rho_{ad}\frac{\partial^2 Y_d}{\partial r^2}-D_s\rho_d\frac{\partial^2 W}{\partial r^2}=0 \tag{26}$$

where D_G and D_S represent the effective gas phase diffusivity and surface diffusivity, respectively.

Conservation of energy within the desiccant felt gave

$$\varepsilon_t\rho_{ad}\frac{\partial H_{ad}}{\partial t}+(1-\varepsilon_t)\rho_d c_{pd}\frac{\partial T_d}{\partial t}-k_d\frac{\partial^2 T_d}{\partial r^2}=(1-\varepsilon_t)\rho_d q_{st}\frac{\partial W}{\partial t} \tag{27}$$

where subscript "*ad*" means the air in desiccant pore.

The rate of energy transfer between process air and desiccant felt yielded

$$\frac{\partial H_g}{\partial t}+u\frac{\partial H_g}{\partial z}=\frac{2K_y}{\rho_g\alpha_a}(Y_d(t,x,\alpha_d)-Y_a)\frac{\partial H_g}{\partial Y_a}+\frac{2h}{\rho_g\alpha_a}(T_d(t,x,\alpha_d)-T_a) \tag{28}$$

The boundary conditions are given as follows by assuming that the temperature of the wall was same as the desiccant temperature at $r=0$

$$T_w(t,z)=T_d(t,z,0) \tag{29}$$

$$\left.\frac{\partial T_m}{\partial r}\right|_{r=0}=\left.\frac{\partial T_m}{\partial z}\right|_{z=0}=\left.\frac{\partial T_m}{\partial z}\right|_{z=L}=0 \tag{30}$$

The models which have been discussed above can further be used to guide system operation, interpret experimental results and assist in system design and optimization.

2.6. Important literature survey on solid desiccant cooling

Dhanes and Wiliam [52] numerically modeled open cycle solid desiccant cooling system for ventilation mode to determine the effect of non-dimensional parameters such as dehumidifier channel length, desiccant mass fraction etc. on isotherm shape of desiccants. Pesaran et al. [53] illustrated the aspects of low grade heat application in desiccant cooling cycle for the purpose regeneration. It is also shown economic feasibility of desiccant cooling for residential and commercial building air conditioning. Shelpuk [54] demonstrated hybrid dehumidifier systems having advanced desiccant materials and compact components to replace large and expensive solid desiccant cooling equipments available. It is shown that the technology development and cost reduction can be increased further by widely market acceptance of advanced desiccant systems. Jain et al. [55] made comparison for various desiccant cooling cycles for air conditioning in hot and humid climate of various parts of the country like India. Psychrometric evaluation is also carried out to simulate exact room conditions. Influence of variation in outdoor condition on the effectiveness of coolers has also been investigated by them. It is found that Dunkle cycle give better performance for wide range of outdoor condition. Parametric study on influence of ambient temperature and humidity change on the evaporative desiccant cooling system performance for several cities of tropical climate has been carried out by Camargo et al. [56]. In addition, application of the system in different climatic condition of several tropical and equatorial cities has also been presented. An experimental investigation was conducted for an open cycle desiccant cooling system operating

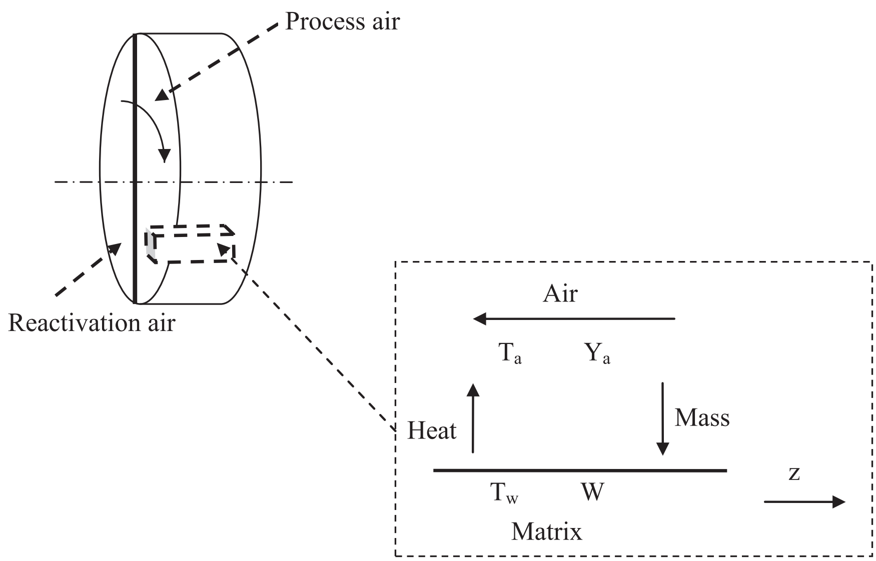

Fig. 23. Control volume and side view of one of ducts for PGS model.

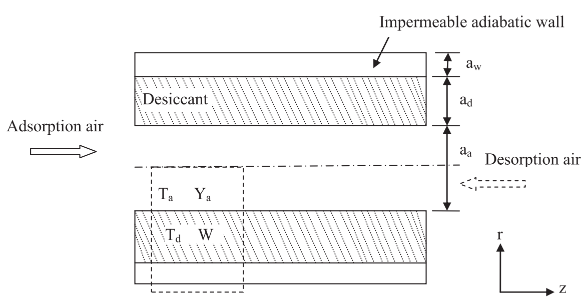

Fig. 24. The model of control volume for a two dimensional GSS model.

in the ventilation mode by Carpinlioglu et al. [57]. The interactive influence of the operational parameters like rotational speeds of dehumidifier and heat wheel, mass flow rate, regeneration temperature etc. on overall system performance has also been determined. Evaporative cooling technique is not much effective when ambient humidity is markedly high (Daou et al. [7]). Desiccant cooling when supplement to evaporative cooling can applicable under a diversity of climatic conditions. Desiccant cooling have feasibility in different climates proven in terms of energy and cost savings. Its energy saving potential can be increased further by their regeneration using low grade thermal energy such as solar or waste heat. Panaras et al. [58] proposed a desiccant evaporative cooling system having a greater potential for covering the space requirements, design improvement and flexible control strategies. Ge et al. [59] evaluated performance of two stage rotary desiccant cooling system (TSRDC). Compared with one-stage system, required regeneration temperature of TSRDC system is much lower hence low-grade energy such as solar energy and waste heat can be efficiently utilized and operating cost can be markedly reduced. It is also shown that the high COP_{th} of the system can be achieved under the lower regeneration temperature. Experimental tests on desiccant evaporative cooling system were performed by Ouazia et al. [60] to achieve better humidity control and acceptable comfort conditions using TRNSYS simulations. Medeiros et al. [61] conducted experiments and simulation study of desiccant air conditioning system to evaluate the overall performance of the system under various operational conditions. The dynamic response of complete system was observed by imposing the real conditions. Numerical and experimental validation has proven that the overall performance of the system depends on the individual performance of each component of the system. Theoretical model for a desiccant air conditioning system has been developed (Panaras et al. [62]). The proposed model is used for examining the influence of operating parameters such as weather conditions, cooling load, air flow rate and regeneration temperature on overall system performance. Potential of simple desiccant evaporative cooling cycle was examined by Parmar and Hindoliya [63] in Warm and Humid climatic zone of India. The coefficient of performance has been computed for different climatic zones and compared by them. It is also concluded that the coefficient of performance of the system is highly influenced by changes in outdoor humidity ratio. Higher is the ambient air humidity ratio, lower is the COP. Heidarinejad and Pasdarshahri [64] developed and applied mathematical model based on transient coupled heat and mass transfer to predict performance of the system under various design and operation conditions like ventilation, recirculation, make up and mixed mode. Mittal and Khan [65] evaluated the performance and energy saving capacity of a desiccant air-conditioning system composed of silica gel bed. Compared to conventional air conditioner where indoor air is recycled totally, the electricity savings is about 19%. Economic analysis shows that for a small cooling capacity this system is not a suitable option because the extra cost is quite significant compared with the operational cost savings. Panaras et al. [66] experimentally investigated the effect of operational parameters like air flow rate, regeneration temperature, temperature and humidity of supply as well as exhaust air stream, rotational speed of desiccant wheel and heat wheel on overall system performance. The effect of ambient condition on the coefficient of performance and exit air temperature of desiccant cooling cycle for ventilation and makeup modes have been investigated by Heidarinejad and Pasdarshahri [67]. The coefficient of performance of makeup mode is obtained higher than the ventilation mode for higher ambient air temperature due to lower energy needed in heater to obtain required regeneration air temperature for dehumidifier. These analyses can be very useful for determining the performance of desiccant cooling system for a multi-climate country where wide range of outdoor condition is available. Goldsworthy and White [68] experimentally investigated that the desiccant cooling cycle performance is governed by supply air flow rate i.e. larger flow rate of cool air or low flow rate of cold air. Multi-variable optimization is performed to determine both the preferred size and operating points for the key component like desiccant wheel and evaporative cooler as well as flow rate of each air stream in the system. Panaras et al. [69] proposed newer control strategy for the operation of desiccant air conditioning system to achieve comfort condition in space to be cooled. Hygrostat and thermostat regulate humidifier and indirect evaporative cooler operation on “on/off” concept. From energy consumption point of view the “on/off” humidifiers operation concept seems to be more efficient, especially in the case of mild weather conditions. Graphical methodology has been proposed by Nobrega and Brum [70] for the design of desiccant cooling cycles. Chung and Lee [71] analyzed the effect of various operational parameters like flow rate, humidity, temperature, speed etc. on the performance of a desiccant cooling cycle under two different system configurations. Out of them, the regenerative temperature is found to be the most dominant parameter that can contribute to operational cost saving. Sphaier and Nobrega [72] developed numerical procedure for analyzing the impact of individual component effectiveness on overall cycle performance. Their analysis shows that 20–30% decrease in dehumidifier performance can lead to 30–40% reduction in the overall cycle performance. A neural network model has been developed to predict the dehumidification capacity and the outlet desiccant conditions for the silica gel desiccant cooling system under different climatic conditions (Koronaki et al. [73]). Uckan et al. [74] conducted experiments to investigate the overall performance of the system and the performance of its components in hot and humid weather. The impact of the effectiveness of key components like dehumidifier, heat wheel and evaporative coolers on the overall system performance has been investigated.

Sheridan and Mitchell [75] developed a model of hybrid desiccant cooling system using TRNSYS and simulate it for two different climatic conditions. It is shown that for hot and humid climate, the hybrid desiccant cooling cycle uses 25–40% less energy than the conventional vapor compression unit. Dhar and Singh [76] analyzed the performance of four types of hybrid cycles for hot–dry and hot–humid weather based on analogy method. It gives substantial energy saving as compared to conventional VCR systems. The influence of room sensible heat factor, mixing ratio, regeneration temperature on over all cycle performance have also been studied. Henning et al. [77] presented parametric study to compare performance of different air conditioning system for various climatic zones. It is to be concluded that the hybrid system can save primary energy up to 50% along with reduction in its operational costs. Dai et al. [78] conducted a comparative study of a standalone VCS, the desiccant-associated VCS and the desiccant and evaporative cooling associated VCS. They have found an increase of cold production by 38.8–76% and that of COP by 20–30%. Desiccant wheel integrated vapor compression system is experimentally studied by Subramanyam et al. [79] for low humidity cooling. Effect of various parameters like air flow rate, compressor capacity and wheel speed etc. on system performance has been evaluated by them. It is shown that the desiccant wheel speed of 17.5 rph is optimum for maximum COP and high humidity removal. Yong et al. [80] experimented with LiCl dehumidifier for high humid regions. It is observed that regeneration temperature and air flow rate have major impact on the overall cycle performance. Hybrid system can achieve a greater part load performance with a flexibility to work at different operational modes and can assure satisfactory performance during year round in high humid regions. Experiment on hybrid system consist of solid desiccant cooling system followed by conventional VCR unit has been carried out by Jia et al. [81] to determine role of operating parameters to save electrical power. Hybrid system can also increase energy saving potential by decreasing consumption of high grade electrical energy. Energy consumption model of hybrid DOAS is established by Liu et al. [82] dedicated outdoor air system (DOAS) with rotary desiccant wheel is the combination of a desiccant dehumidification system and a conventional vapor compression refrigeration system. Significance of ventilation air flow rate, temperature and humidity of outdoor air, *R*/*P* ratio on the energy consumption and the COP of the hybrid DOAS has been determined. Sayegh et al. [83] carried out comparison between two methods for improving the overall performance of dehumidification in an air conditioning system. One is the hybrid system consists of desiccant cycle with back up of conventional VCR cooling coil and the second is heat exchanger cycle. The role of desiccant dehumidifier on the performance of hybrid system and heat exchanger efficiency on the performance of heat exchanger cycle were studied. When the relative humidity is low, hybrid system can perform better to achieve lower sensible heat ratio and dew point temperature than the heat exchanger cycle but with raise in the relative humidity the difference between the two systems reduces. Hong et al. [84] investigated two types of hybrid systems. One was composed of conventional vapor compressor air conditioner with desiccant cooling system and the other was composed of vapor compressor, desiccant and direct evaporative cooler cooling system. Mathematical model of dehumidifier and the physical model and numerical model of the hybrid system were studied. It was found that under the same operating condition, compared with conventional vapor-compression cooling system, coefficient of performance and energy saving of two vapor compression sub system for hybrid systems found increased.

Khalid and Madhi [85] experimentally investigated a solar assisted open adsorption cooling system. Numbers of tests have been conducted at constant regeneration temperature on silica gel dehumidifier. Performance of solar air heater has been compared experimentally as well as analytically. It is shown that the increase in regeneration temperature and mass flow rate can improve the overall system performance. Halliday et al. [86] discussed the feasibility of desiccant cooling system using solar energy to evaluate installations located at various places at UK. Solar energy supplied 72% of the thermal energy required operate the desiccant system. Solar heating coils in summer time can save energy up to 39%. Evaluation and optimization of solar assisted desiccant wheel done by Ahmed et al [87]. Effect of operating parameters like air flow rate, humidity ratio, regeneration temperature and design parameters like wheel thickness, speed, porosity etc. are evaluated by simulation of the heat and mass transfer for the adsorption and regeneration processes on developed numerical model of solar desiccant wheel. Maalouf et al. [88] tested LiCl dehumidifier coupled with solar liquid collectors to optimize heat regeneration source. Influence of various operating parameters has been studied to decrease primary energy consumption and increase the overall system performance. Air flow rate can determine the required regeneration hours. The reduction in air change hours during in occupation period decreases the required regeneration hours of about 40%. Solar assisted desiccant cooling system consists of direct flow vacuum tube collectors is simulated and validated with experimental results by Bourdoukan et al [89]. It is shown that for raising the regeneration temperature up to 12 °C decreases the global efficiency by 0.45. Kim and Ferreira [90] reviewed different technology that uses the solar energy for producing air conditioning such as thermo-mechanical, absorption, adsorption and desiccant solutions. Comparison is made between different systems on basis of energy efficiency and economic feasibility. Bourdoukan et al. [16] experimentally evaluated desiccant air handling unit powered by vacuum-tube solar collectors. The overall system performance has also been evaluated by them. Coefficient of performance of the system for moderately humid day is calculated 0.4 while on electrical energy consumption basis it is calculated 4.3 over the same day. The overall efficiency of the solar installation is 0.55 indicating high potential of vacuum collectors in desiccant cooling. Khalid et al. [37] simulated solar assisted pre-cooled hybrid desiccant cooling (PHDCS) system for air conditioning applications in Pakistan. TRNSYS model is validated with measured data sets. Life cycle assessment of solar air collector has been performed and payback period has been found out by them. Yadav [91] simulated a hybrid desiccant cooling system comprising the conventional vapor compression refrigeration air conditioning system coupled with a liquid desiccant dehumidifier system which was regenerated by solar heat. The study concluded that, when the latent load constitutes 90% of the total cooling load, the system can generate up to 80% of energy savings. Simulation and optimization of solar-assisted desiccant cooling system (SADCS) was developed and its performance was evaluated by Fong et al. [40]. SADCS increases supply air flow rate to improve indoor air quality. Required auxiliary heater for the regeneration of dehumidifier replaced by the optimal design and control scheme of the SADCS. La et al. [92] experimentally evaluated solar powered hybrid desiccant air conditioning system integrated with two-stage desiccant cooling (TSDC) and vapor compression air-conditioning (VAC) together. System model has been created in TRNSYS to evaluate energy saving potential by them. It is shown that the thermal COP of 0.95 with a solar fraction of 33.3% can be achieved under hot and humid conditions with electric power saving rates are about 31%. La et al. [93] modeled rotary desiccant wheel integrated with evacuated tube solar air collectors to design direct solar heating mode in winter season and solar heating with desiccant humidification mode during summer season using TRNSYS simulation studio. It is shown that the solar heating system has converted about 50% of the received solar radiation for space heating on a sunny day in winter to raise indoor room air temperature about 10 °C. While auxiliary heater improves indoor thermal comfort up to 30% for direct solar heating mode and 60% for solar heating with desiccant humidification mode during summer comfort cooling. Koronaki et al. [94] experimentally evaluated the thermodynamic performance of a solar assisted desiccant cooling system coupled to LiCl dehumidifier in Mediterranean areas with higher solar fractions. It is observed that higher process air inlet humidity increases dehumidification and ultimately the COP_{th}. While lower inlet temperature and humidity ratio of regeneration air leads to better system performance. A mathematical model has been proposed by Lafuenti et al. [95] to predict thermodynamic behaviour of novel solar assisted desiccant cooling cycle for hot and humid climates. Efficiency is estimated in the range of 0.17–0.76. Solar assisted cooling system evaluated by Baniyounes et al. [96] using TRNSYS software. Technical, economical and environmental performance of system was determined for total annual cooling load 6428 kW h. Almost 60% energy saving is achieved by use of solar energy for the purpose of regeneration of desiccant dehumidifier. Summary of above discussion has been tabulated in Table 5.

2.7. Performance definitions

The performance of the solid desiccant cooling system can be evaluated using the expressions defined below:

The coefficient of performance for the solid desiccant evaporative cooling system as shown in Fig. 8 can be expressed by [61]

$$COP = \frac{Q_{cool}}{Q_{reg}} = \frac{m_p(h_5 - h_4)}{m_r(h_8 - h_7)} \tag{31}$$

where m_p and m_r designates the mass flow rate of process air as well as regeneration air respectively.

Desiccant wheel effectiveness can also be expressed by [7]

$$\varepsilon_{DW} = \frac{w_1 - w_2}{w_1 - w_{2,ideal}} \tag{32}$$

where $w_{2,ideal}$ is the ideal specific humidity of the air stream at exit

Table 5
Summary of parameters studied by previous experimental work.

Author	Desiccant material	Desiccant wheel dimension (diameter/thickness) (m)	Speed (rph)	Regeneration temperature (°C)	Volume Flow rate (m^3/h)		Process air inlet conditions		COP
					Process air	Reg. air	Temp. (°C)	Humidity ratio (g/kg)	
Yadav and Kaushik [35]	–	–	–	80.80	5400	5400	30	16.67	2.7
Neti and Wolfe [97]	Silica gel	0.32/0.4	4–16	70–160	32.5	32.5	30	14.2	–
Dai et al. [78]	Silica gel	0.25/0.2	5	120	–	–	35	14.0	–
Qin et al. [98]	LiCl	0.65/0.45	20	70	750	750	28	14	–
Zhang and Niu [99]	Silica gel	1.0/0.1	–	90	1440	1440	30	21	–
Kanoglu et al. [100]	Natural Zeolite	0.5/0.25	–	60.8	400	400	31.5	9.5	0.34
Subramanyam et al. [79]	Metal silicate	0.45/0.2	24.5	–	250-750	250-750	32	21	1.0–2.5
Jia et al. [81]	LiCl	0.23/0.2	7	100	–	–	30.5	14.9	4.8
Yong et al. [80]	LiCl	0.32/0.2	7	60–120	400–1000	300	30	19	2.5–3
Bourdoukan et al. [10]	–	–	–	60–80	–	–	30–35	10–15	0.2–0.5
La et al. [9]	Silica gel LiCl	2.8/0.1	–	85	–	–	34	22	3.2
Panaras et al. [62]	Silica gel	0.63/0.2	6	50–80	600–1500	600–1500	32–36	7–10.2	0.4–0.8
Fong et al. [39]	Silica gel	0.6/0.2	13	80	900	800	32.8	23.4	0.8
Hurdogan et al. [44]	Silica gel	0.96/0.2	–	68.95	4000	4000	31.9	16.9	–
La et al. [42]	Silica gel	0.44/0.1	8	50–90	2600	1000	34	22	1.0
Baniyoues et al. [96]	–	–	42	–	1450	1450	28–36	17–32	07–1.2
Eicker et al. [101]	Silica gel	0.26/0.1	8.24	60–120	600–1000	400	32	24.5	–
Yadav and Bajpai [102]	Silica gel Alumina Charcoal	–		54.7–68.3	80–120	80–120	31.5–39.8	19.7–24.9	–
Angrisani et al. [103]	Silica gel	–	7–10	65	800	0–800	25.6–34.3	8.63–11.5	0.3–0.9
Khoukhi [104]	–	–	–	82	100–200	100–200	36	28.2	–
Yadav and Bajpai [105]	Silica gel	0.37/0.1	4–32	50–80	–	–	24–36	16–24	–
Mandegari et al. [106]	–	0.37/0.2	5–25	60–140	–	–	30	10	–
Yamaguchi and Saito [107]	Silica gel	0.35/01	0–200	50–80	–	–	32.3	19.5	–

of the rotary desiccant dehumidifier, it is to be taken zero by assuming that the air is completely dehumidified at this point.

Effectiveness of heat wheel can be determined by [7]

$$\varepsilon_{HW} = \frac{T_2 - T_3}{T_2 - T_6} \tag{33}$$

Effectiveness of evaporative coolers can be given by [7]

$$\varepsilon_{EC1} = \frac{T_3 - T_4}{T_3 - T_{wb,3}} \quad \text{and} \quad \varepsilon_{EC2} = \frac{T_5 - T_6}{T_5 - T_{wb,5}} \tag{34}$$

where T_{wb} is the wet bulb temperature of moist air.

The rates of moisture added to air by the evaporative coolers in the process and return lines are obtained by following [7]

$$m_{w1} = m_p(w_4 - w_3) \quad \text{and} \quad m_{w2} = m_r(w_6 - w_5) \tag{35}$$

where m_{w1} and m_{w2} are the rates of moisture added to air in the evaporative coolers in supply and return lines respectively.

Similarly, the coefficient of performance for the hybrid solid desiccant vapor compression air conditioning as shown in Fig. 16 can be given as follows [35]

$$\mathrm{COP}_T = \frac{Q_{cc}}{E_{total}} \tag{36}$$

where Q_{cc} is the cooling capacity is defined as following [108]

$$Q_{cc} = m_p(h_1 - h_5) \tag{37}$$

where E_{total} is the total energy consumption of the system expressed by [74,109]

$$E_{total} = Q_{reg} + W_{com} + W_{other} \tag{38}$$

W_{other} shows energy consumption [110,111] of other equipments are fans, desiccant wheel and heat wheel motor.

where Q_{reg} is the heat supply to regeneration air stream by an electrical heater is given as [80]

$$Q_{reg} = m_r(h_8 - h_7) \tag{39}$$

COP_w and COP_H can also be defined by [74]

$$\mathrm{COP}_w = \frac{Q_{cc}}{W} \quad \text{and} \quad \mathrm{COP}_H = \frac{Q_{cc}}{Q_{reg}} \tag{40}$$

From last equation we obtain:

$$\frac{1}{\mathrm{COP}_T} = \frac{1}{\mathrm{COP}_W} + \frac{1}{\mathrm{COP}_H} \tag{41}$$

where the subscripts correspond to the state point in the solid desiccant cooling system.

Similarly, the coefficient of performance for the solar assisted hybrid solid desiccant vapor compression air conditioning as shown in Fig. 19 can be given as follows [36]

$$\mathrm{COP}_{DEC} = \frac{Q_{DEC}}{Q_{HC}} \tag{42}$$

where solar cooling coefficient of performance is COP_{DEC} is the ratio between useful cooling output of the desiccant cooling cycle (Q_{DEC}) and the regeneration heat delivered by the solar heating coil (Q_{HC}).

So, $$Q_{DEC} = m_p(h_3 - h_2) \text{ and } Q_{HC} = m_r(h_8 - h_7) \tag{43}$$

while coefficient of performance of vapor compression refrigerators (COP_{VCR}) is expressed as follows

$$\mathrm{COP}_{VCR} = \frac{Q_L}{W_i} \tag{44}$$

where Q_L is cooling effect in kW and W_i is electric power for VCR compressor in kW.

Efficiency of solar collectors (η_{coll}) can be calculated from following relation

$$\eta_{coll} = \frac{Q_{HC}}{I_\beta \mathrm{COP}_{DEC} A_{ST}} \tag{45}$$

where I_β is solar radiation in kW/m^2 and AST is solar collector area in m^2.

3. Conclusion

Through this state of the art review it has been concluded that solid desiccant cooling is energy saving and environment friendly approach for building air conditioning. Numerous researchers have conducted its feasibility study by using simulation as well as experimental methodologies to make it energy efficient and cost effective. Ongoing research and development works suggest that the advanced desiccant materials and novel system configurations have significant potential for improving performance and reliability. Thus, improvement in the performance can play a key role in order to approach economic feasibility. Hybrid cycles can achieve significant energy savings by use of freely available solar energy or waste heat from industrial processes for regeneration of desiccant material can make system more cost effective. It can also help to alleviate the peak electricity demand caused by conventional vapor compression air conditioning system required during the hot sunny days. However, the system performance in terms of solar fraction and thermal coefficient of performance varies greatly with respected to different operational condition. By making the direction of future research towards desiccant with low cost material, low reactivation temperature, higher moisture removal rate and stability after long period of service augmenting the contribution of desiccant cooling which can bring to the amelioration of comfort, energy and cost savings.

Further improvement in energy utilization rate, reduction in cost and size, competitive design and production are the key issues faced by solid desiccant air conditioning techniques for obtaining more extensive acceptability in the field of space cooling.

References

[1] Pennington NA. Humidity changer for air conditioning. USA patent no. 2, 700, 537; 1955.
[2] Dunkle RV. A method of solar air conditioning. Mech Chem Eng Trans Inst Eng 1965;73:73–8.
[3] Munters CG. Inorganic, fibrous, gas-conditioning packing for heat and moisture transfer. U. S. patent 3, 377, 225; 1968.
[4] Jurinak JJ. Open cycle solid desiccant cooling-component models and system simulation [Ph.D. thesis]. Madison: University of Wisconsin; 1982.
[5] Pesaran AA, Mills AF. Modelling of solid-side mass transfer in desiccant particle beds. Sol Eng 1984:177–85.
[6] Davanagere BS, Sherif SA, Goswami DY. A feasibility study of solar desiccant air conditioning system- Part I: psychrometrics and analysis of the conditioned zone. Int J Energy Res 1999;23:7–21.
[7] Daou K, Wang RZ, Xia ZZ. Desiccant cooling air conditioning: a review. Renew Sustain Energy Rev 2006;10:55–77.
[8] Jain S, Dhar PL, Kaushik SC. Evaluation of solid based evaporative cooling cycles for typical hot and humid climates. Int J Refrig 1995;18 287-29.
[9] La D, Dai YJ, Li Y, Wang RZ, Ge TS. Technical development of rotary desiccant dehumidification and air conditioning: a review. Renew Sustain Energy Rev 2010;14:130–47.
[10] Bourdoukan P, Wurtz E, Joubert P. Comparison between the conventional and recirculation modes in desiccant cooling cycles and deriving critical efficiencies of components. Energy 2010;35:1057–67.
[11] Dezfouli MMS, Mat S, Pirasteh G, Sahari KSM, Sopian K, Ruslan MH. Simulation analysis of the four configurations of solar desiccant system using evaporative cooling in tropical weather in Malaysia. Int J Photo Energy 2014;2014:1–14.
[12] Khalid AJ, Nabeel SD. Application of solar assisted heating and desiccant cooling systems for a domestic building. Energy Convers Manag 2001;42 995–22.

[13] Li M, David I, Ursula E, Dennis L, Volker F. Cooling potential of ventilated PV facade and solar air heaters combined with a desiccant cooling machine. Renew Energy 2006;31:1265–78.
[14] Kabeel AE. Solar powered air conditioning system using rotary honeycomb desiccant wheel. Renew Energy 2007;32:1842–57.
[15] Bourdoukan P, Wurtz E, Joubert P, Spérandio M. Potential of solar heat pipe vacuum collectors in the desiccant cooling process: modelling and experimental results. Sol Energy 2008;82:1209–19.
[16] Bourdoukan P, Wurtz E, Joubert P. Experimental investigation of a solar desiccant cooling installation. Sol Energy 2009;83:2059–73.
[17] White SD, Kohlenbach P, Bongs C. Indoor temperature variations resulting from solar desiccant cooling in a building without thermal backup. Int J Refrig 2009;32 695–04.
[18] Enteria N, Yoshino H, Satake A, Mochida A, Takaki R, Yoshie R, et al. Development and construction of the novel solar thermal desiccant cooling system incorporating hot water production. Appl Energy 2010;87:478–86.
[19] Fong KF, Chow TT, Lin Z, Chan LS. Simulation optimization of solar-assisted desiccant cooling system for subtropical Hong Kong. Appl Thermal Eng 2010;30:220–8.
[20] Ge TS, Ziegler F, Wang RZ, Wang H. Performance comparison between desiccant cooling system and vapor compression system. Appl Thermal Eng 2010;30:724–31.
[21] Li H, Dai YJ, Li Y, La D, Wang RZ. Case study of a two-stage rotary desiccant cooling/heating system driven by evacuated glass tube solar air collectors. Energy Build 2012;47:107–12.
[22] Preisler A, Brychta M. High potential of full year operation with solar driven desiccant evaporative cooling system. Energy Procedia 2012;30:668–75.
[23] Li H, Dai YJ, Kohler M, Wang RZ. Simulation and parameter analysis of a two-stage desiccant cooing/heating system driven by solar air collectors. Energy Convers Manag 2013;67:309–17.
[24] Huan Z, Jianlei N. A two-stage desiccant cooling system using low temperature heat. Build Serv Eng Res Technol 1999;20:51–5.
[25] Ando K, Kodama A. Experimental study on a process design for adsorption desiccant cooling driven with a low-temperature heat. Adsorption 2005;11:631–6.
[26] Azar AAJ, Slayzak S, Judkoff R, Schaffhauser T, DeBlasio R. Performance assessment of a desiccant cooling system in a CHP application incorporating an IC engine. Int J Distrib Energy Resources 2005;1:163–84.
[27] Qin CK, Schmitz G. Engine-driven desiccant-assisted hybrid air-conditioning system. In: Proceedings of the 23rd world gas conference. Amsterdam; 2006.
[28] Kohlenbach P, Bongs C, White S, Ward J. Performance modeling of a desiccant-evaporative sorption air conditioning system driven by micro-turbine waste heat in tropical climates. AIRAH sustainability for tropical and sub-tropical climate conference; 2006.
[29] Angrisani G, Minichiello F, Roselli C, Sasso M. Experimental investigation to optimise a desiccant HVAC system coupled to a small size cogenerator. Appl Thermal Eng 2011;31:506–12.
[30] Ge F, Guo X, Hu Z, Chu Y. Energy savings potential of a desiccant assisted hybrid air source heat pump system for residential building in hot summer and cold winter zone in China. Energy Build 2011;43:3521–7.
[31] El-Agouz SA, Kabeel AE. Performance of desiccant air conditioning system with geothermal energy under different climatic conditions. Energy Convers manag 2014;88:464–75.
[32] Ying S, Yufeng Z, Lei F, Jinshe N, Lijun M. Optimization analysis of high temperature heat pump coupling to desiccant wheel air conditioning system. Trans Tianjin Uni 2014;20:182–8.
[33] Kelly MO, Walter ME, Rowland JR. Simulated hygrothermal performance of a desiccant- assisted hybrid air/water conditioning system in a mixed humid climate under dynamic load. Energy Build 2015;86:45–57.
[34] Angrisani G, Diglio G, Sasso M, Calise F, Dentice d'Accadia M. Design of a novel geothermal heating and cooling system: Energy and economic analysis. Energy Convers Manag 2016;108:144–59.
[35] Yadav YK, Kaushik SC. Psychrometric technoeconmic assessment and parametric studies of vapor – compression and solid/liquid desiccant hybrid solar space conditioning systems. Heat Rev Sys 1991;11:563–72.
[36] Gagliano A, Patania F, Nocera F, Galesi A. Performance assessment of a solar assisted desiccant cooling system. Thermal Sci 2014;18:563–76.
[37] Khalid A, Mahmood M, Asif M, Muneer T. Solar assisted, pre-cooled hybrid desiccant cooling system for Pakistan. Renew Energy 2009;34:151–7.
[38] Marco B, Pietro F, Bettina N. Energy and economic assessment of desiccant cooling systems coupled with single glazed air and hybrid PV/thermal solar collectors for applications in hot and humid climate. Sol Energy 2009;83:1828–46.
[39] Fong KF, Chow TT, Lee CK, Lin Z, Chan LS. Advancement of solar desiccant cooling system for building use in subtropical Hong Kong. Energy Build 2010;42:2386–99.
[40] Fong KF, Lee CK, Chow TT, Fong AML. Investigation on solar hybrid desiccant cooling system for commercial premises with high latent cooling load in subtropical Hong Kong. Appl Thermal Eng 2011;31 3393–01.
[41] Fong KF, Chow TT, Lee CK, Lin Z, Chan LS. Solar hybrid cooling system for high-tech offices in subtropical climate: radiant cooling by absorption refrigeration and desiccant dehumidification. Renew Energy 2011;52:2883–94.
[42] La D, Dai YJ, Li Y, Ge TS, Wang RZ. Case study and theoretical analysis of a solar driven two-stage rotary desiccant cooling system assisted by vapor compression air-conditioning. Sol Energy 2011;85 2997–09.
[43] Finocchiaro P, Beccali M, Nocke B. Advanced solar assisted desiccant and evaporative cooling system equipped with wet heat exchangers. Sol Energy 2012;86:608–18.
[44] Hurdogan E, Buyukalaca O, Yılmaz T, Hepbasli A, Uckan R. Investigation of solar energy utilization in a novel desiccant based air conditioning system. Energy Build 2012;55:757–64.
[45] Marco B, Pietro F, Bettina N. Energy performance evaluation of a demo solar desiccant cooling system with heat recovery for the regeneration of the adsorption material. Renew Energy 2012;44:40–52.
[46] Joss MCL, Francisco FH, Fernando DM, Antonio CA. The optimization of the operation of a solar desiccant air handling unit coupled with a radiant floor. Energy Build 2013;62:427–35.
[47] Hong H, Feng G, Wang H. Performance research of solar hybrid desiccant cooling systems. Procedia Environ Sc 2012;12:57–64.
[48] Ge TS, Li Y, Wang RZ, Dai YJ. A review of the mathematical models for predicting rotary desiccant wheel. Renew Sustain Energy Rev 2008;12 1485-28.
[49] Charoensupaya D, Worek WM. Parametric study of an open-cycle adiabatic, solid, desiccant cooling system. Energy 1988;13:739–47.
[50] Maclaine-Cross IL, Banks PJ. Coupled heat and mass transfer in regenerators-prediction using an analogy with heat transfer. Int J Heat Mass Trans 1972;15:1225–42.
[51] Charoensupaya D, Worek WM. Effect of adsorbent heat and mass transfer resistances on performance of an open cycle adiabatic desiccant cooling system. Heat Recovery System CHP 1988;8:537–48.
[52] Dhanes C, Wiliam MW. Parametric study of an open cycle adiabatic solid desiccant cooling system. Energy 1988;9:739–47.
[53] Pesaran AA, Penney TR, Czanderna AW. Desiccant cooling-state of the art-assessment. Colorado: National Renewable Energy Laboratory; 1992 DOE-EERE Research Reports.
[54] Shelpuk B. The technical challenges for solid desiccant cooling. Heat Recov Sys CHP 1993;13:321–8.
[55] Jain S, Dhar PL, Kaushik SC. Evaluation of solid based evaporative cooling cycles for typical hot and humid climates. Int J Refrig 1995;18 287-29.
[56] Camargo JR, Godoy E, Ebinuma CD. An evaporative and desiccant cooling system for air conditioning in humid climates. J Braz Soc Mech Sci Eng 2005;3:243–7.
[57] Carpinlioglu MO, Yildirim M. A methodology for the performance evaluation of an experimental desiccant cooling system. Int Commu Heat Mass Transf 2005;32:1400–10.
[58] Panaras G, Mathioulakis E, Belessiotis V. Achievable working range for solid all-desiccant air-conditioning systems under specific space comfort requirements. Energy Build 2007;39:1055–60.
[59] Ge TS, Li Y, Wang RZ, Dai YJ. Experimental study on a two-stage rotary desiccant cooling system. Int J Refrig 2009;32 498-08.
[60] Ouazia B, Barhoun H, Daddad K, Armstrong M, Szadkwski F. Desiccant evaporative cooling system for residential building. In: Proceedings of the 12th Canadian conference on building science and technology. Montreal, Quebec; 2009. p. 570–90.
[61] Medeiros J, Silva M, Santos J, Gurgel J. Experimental investigation and simulation study on a desiccant air-conditioning system in brazil. In: Proceedings of the 20th international congress of mechanical engineering. Gramado, RS, Brazil; 2009:. p.1–7.
[62] Panaras G, Mathioulakis E, Belessiotis V, Kyriakis N. Theoretical experimental investigation of the performance of a desiccant air-conditioning system. Renew Energy 2010;35:1368–75.
[63] Parmar H, Hindoliya DA. Performance of solid desiccant based evaporative cooling system in warm and humid climatic zone of India. Int J Eng Sci Tech 2010;2:5504–8.
[64] Heidarinejad G, Pasdarshahri H. The effects of operational conditions of the desiccant wheel on the performance of desiccant cooling cycles. Energy Build 2010;42:2416–23.
[65] Mittal V, Khan BK. Experimental investigation on desiccant air-conditioning system in India. Front Energy Power Eng 2010;4:161–5.
[66] Panaras G, Mathioulakis E, Belessiotis V. Solid desiccant air-conditioning systems-Design parameters. Energy 2011;36 2399-06.
[67] Heidarinejad G, Pasdarshahri H. Potential of a desiccant-evaporative cooling system performance in a multi-climate country. Int J Refrig 2011;34:1251–61.
[68] Goldsworthy M, White S. Optimization of a desiccant cooling system design with indirect evaporative cooler. Int J Refrig 2011;34:148–58.
[69] Panaras G, Mathioulakis E, Belessiotis V. Proposal of a control strategy for desiccant air-conditioning systems. Energy 2011;36:5666–76.
[70] Nobrega CEL, Brum NCL. A graphical procedure for desiccant cooling cycle design. Energy 2011;36:1564–70.
[71] Chung JD, Lee DY. Contributions of system components and operating conditions to the performance of desiccant cooling systems. Int J Refrig 2011;34:922–7.
[72] Sphaier LA, Nobrega CEL. Parametric analysis of component effectiveness on desiccant cooling system performance. Energy 2012;38:157–66.
[73] Koronaki IP, Rogdakis E, Kakatsiou T. Thermodynamic analysis of an open cycle solid desiccant cooling system using artificial neural network. Energy Convers Manag 2012;60:152–60.
[74] Uckan I, Yilmaz T, Hurdogan E, Buyukalaca O. Experimental investigation of a novel configuration of desiccant based evaporative air conditioning system. Energy Convers Manag 2013;65:606–15.

[75] Sheridan JC, Mitchell JW. A hybrid solar desiccant cooling system. Sol Energy 1985;34:187–93.
[76] Dhar PL, Singh SK. Studies on solid desiccant based hybrid air-conditioning. Appl Thermal Eng 2001;21:119–34.
[77] Henning HM, Erpenbeck T, Hindenburg C, Santamaria IS. Potential of solar energy use in desiccant cooling cycles. Int J Refrig 2001;24:220–9.
[78] Dai YJ, Wang RZ, Zhang HF, Yu JD. Use of desiccant cooling to improve the performance of vapor compression air conditioning. Appl Thermal Eng 2001;21 1185–05.
[79] Subramanyam N, Maiya MP, Murthy SS. Application of desiccant wheel to control humidity in air-conditioning systems. Appl Thermal Eng 2004;24:2777–88.
[80] Li Y, Sumathy K, Dai YJ, Zhong JH, Wang RZ. Experimental study on a hybrid desiccant dehumidification and air conditioning system. J Sol Energy Eng 2006;128:77–82.
[81] Jia CX, Dai YJ, Wu JY, Wang RZ. Analysis on hybrid desiccant air-conditioning system. Appl Thermal Eng 2006;26 2393-00.
[82] Liu W, Lian Z, Radermacher R, Yao A. Energy consumption analysis on a dedicated outdoor air system with rotary desiccant wheel. Energy 2007;32:1749–60.
[83] Sayegh MA, Hammad M, Faraa Z. Comparison of two methods of improving dehumidification in air conditioning systems: hybrid system (refrigeration cycle –rotary desiccant) and heat exchanger cycle. Energy Procedia 2011;6:759–68.
[84] Hong H, Feng G, Wang H. Performance research of solar hybrid desiccant cooling systems. Procedia Environ Sci 2012;12:57–64.
[85] Khalid AD, Madhi SM. An experimental investigation into a solar assisted desiccant-evaporative air-conditioning system. Sol Energy 1987;39 97-07.
[86] Halliday SP, Beggs CB, Sleigh PA. The use of solar desiccant cooling in the UK: a feasibility study. Appl Thermal Eng 2002;22:1327–38.
[87] Ahmed MH, Kattab NM, Fouad M. Evaluation and optimization of solar desiccant wheel performance. Renew Energy 2005;30:305–25.
[88] Maalouf C, Wurtz E, Mora L, Allard F. Optimization and study of an autonomous solar desiccant cooling system. International conference passive and low energy cooling for the built environment. Santorini, Greece; 2005. p. 663–8.
[89] Bourdoukan P., Wurtz E., Sperandio M., Joubert P. Global efficiency of direct flow vacuum collectors in autonomous solar desiccant cooling: simulation and experimental results. In: Proceedings of the building simulation. 2007. p. 342–7.
[90] Kim DS, Ferreira CAI. Solar refrigeration options-a state of the art review. Int J Refrig 2008;31:3–15.
[91] Yadav YK. Vapor-compression and liquid-desiccant hybrid solar space-conditioning system for energy conservation. Renew Energy 1995;7:719–23.
[92] La D, Dai Y, Li Y, Ge T, Wang R. Case study and theoretical analysis of a solar driven two-stage rotary desiccant cooling system assisted by vapor compression air-conditioning. Sol Energy 2011;85 2997–09.
[93] La D, Dai Y, Li H, Li Y, Kiplagat JK, Wang R. Experimental investigation and theoretical analysis of solar heating and humidification system with desiccant rotor. Energy Build 2011;43:1113–22.
[94] Koronaki IP, Rogdakis E, Tinia Kakatsiou T. Experimental assessment and thermodynamic analysis of a solar desiccant cooling system. Int J Sustain Energy 2012:1–16.
[95] Lafuenti I, Colangelo G, Milanese M, Risi AD. New solutions for the use of solar cooling in hot and humid weather conditions. In: Proceedings of the International Conference on Renewable Energies and Power Quality (ICREPQ-12). Santiago de Compostela (Spain); 2012.
[96] Baniyounes AM, Liu G, Rasul MG, Khan MMK. Analysis of solar desiccant cooling system for an institutional building in subtropical Queensland, Australia. Renew Sustain Energy Review 2012;16:6423–31.
[97] Neti S, Wolfe EI. Measurement of effectiveness in a silica gel rotary exchanger. Appl Thermal Eng 2000;20:309–22.
[98] Qin CK, Schmitz G. Performance prediction of LiCl rotor. Int J Archit Sc 2002;3:20–9.
[99] Zhang LZ, Niu JL. Performance comparisons of desiccant wheels for air dehumidification and enthalpy recovery. Appl Thermal Eng 2002;22:1347–67.
[100] Kanoglu M, Melda OC, Murtaza Y. Energy and exergy analysis of an experimental open cycle desiccant cooling system. Appl Thermal Eng 2004;24:919–32.
[101] Eicker U, Schurger U, Kohler M, Ge T, Dai Y, Li H, Wang R. Experimental investigations on desiccant wheels. Appl Thermal Eng 2012;42:71–80.
[102] Yadav A, Bajpai VK. Experimental comparison of various solid desiccants for regeneration by evacuated solar air collector and air dehumidification. Dry Technol 2012;30:516–25.
[103] Angrisani G, Roselli C, Sasso M. Effect of rotational speed on the performance of a desiccant wheel. Appl Energy 2013;104:268–75.
[104] Khoukhi M. A study of desiccant based cooling and dehumidifying system in hot-humid climate. Int J Mater Mech Manuf 2013;1:191–4.
[105] Yadav A, Bajpai VK. Numerical and experimental investigation of operating parameters of solar-powered desiccant wheel in India. Heat Transf 2013;42:1–30.
[106] Mandegari MA, Pahlavanzadeh H, Farzad S. Energy approach analysis of desiccant wheel operation. Energy Syst 2014;5:551–69.
[107] Yamaguchi S, Saito K. Numerical and experimental performance analysis of rotary desiccant wheels. Int J Heat Mass Transf 2013;60:51–60.
[108] Shen CM, Worek WM. The second law analysis of a recirculation cycle desiccant cooling system: cosorption of water vapor and carbon dioxide. Atmos Environ 1996;30:1429–35.
[109] Jani DB, Mishra Manish, Sahoo PK. Performance studies of hybrid solid desiccant-vapor compression air-conditioning system for hot and humid climates. Energy Build 2015;102:284–92.
[110] Jani DB, Mishra M, Sahoo PK. Performance prediction of rotary solid desiccant dehumidifier in hybrid air-conditioning system using artificial neural network. Appl Thermal Eng 2016;98:1091–103.
[111] Jani DB, Mishra M, Sahoo PK. Performance analysis of hybrid solid desiccant-vapor compression air conditioning system in hot and humid weather of India. Build Ser Eng Res Tech, http://dx.doi.org/10.1177/0143624416633605.