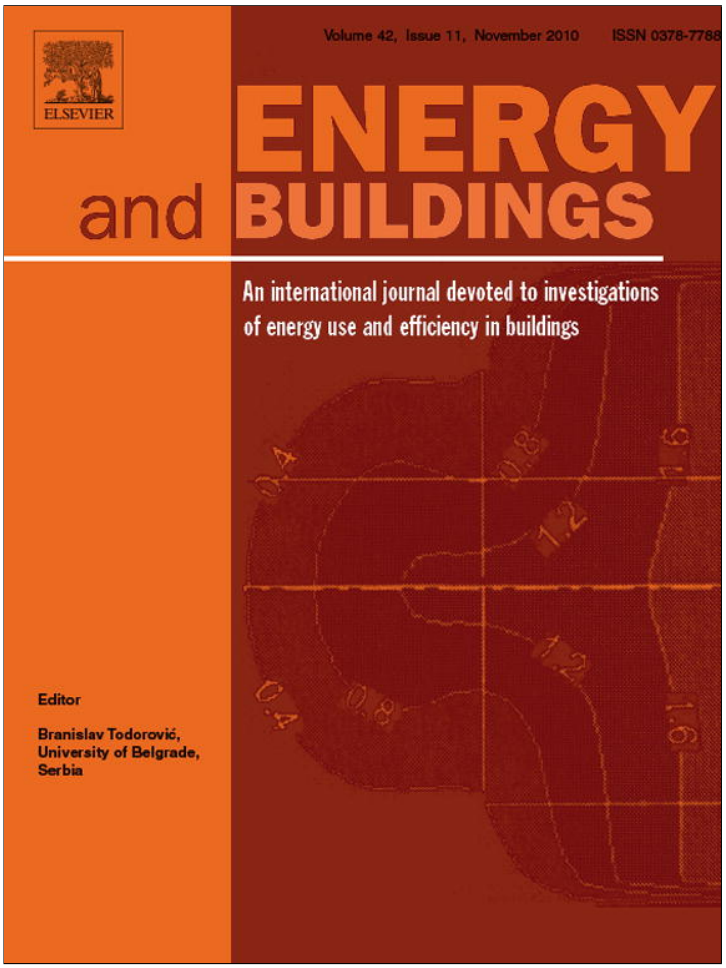
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Experimental investigation of a novel desiccant cooling system

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ABSTRACT

Desiccant based air conditioning systems are a suitable way to improve indoor air quality due to its superior humidity control. In this study, a novel desiccant based air conditioning system is designed and tested experimentally to improve the indoor air quality and reduce energy consumption. In the system studied, a heat exchanger, which is not used in this type of systems, for pre-heating the regeneration air with exhaust air is used. This paper reports results of initial operation and operational procedures. The performance of the system and its components is discussed.

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1. Introduction

Relative humidity at the indoor environment of buildings must be within a specified range for human comfort [1]. Since infections spread through air depend on humidity in some medical institutions like hospitals, humidity control in air conditioning is very important [2]. In conventional systems commonly used for summer air conditioning, the air sent to the place being air-conditioned comes through coils, cooled by a cooling unit generally by a vapor compressing cycle. While coming through coils, both the heat (sensible heat) and the humidity (latent heat) of air are reduced. Removing humidity from the air is realized by the condensation of the water steam in the air, which meets cold coil. This process creates a suitable atmosphere in the air sent to the place, for reproduction of bacteria that can cause infections [3].

Desiccant based air conditioning systems are a suitable way to improve indoor air quality due to its superior humidity control. These systems are also alternatives for air conditioning in health care facilities to reduce the airborne disease transmission. The substances used in these processes for removing moisture, remove the bacteria that can cause illnesses and various infections in the air sent to the place being air-conditioned [4–8]. In desiccant based air conditioning systems, a desiccant removes moisture from the air, which releases heat and increases the air temperature. A combination of heat exchange with ambient air and evaporative or conventional cooling coils then cools the dry air. Temperature and humidity loads are very effectively and efficiently met by separating them in this way. The desiccant is then dried out (regenerated) to complete the cycle using thermal energy.

Several review studies are performed on the description and operation of desiccant cooling systems by different researchers [9–12]. Systems that use rotary desiccant wheel to dehumidify the air are the most popular desiccant cooling systems and studied by different researchers [13–15]. They showed that desiccant cooling systems are viable alternative to vapor compression systems. Most solid desiccant cooling systems operate according to the Pennington cycle [16] also known as ventilation cycle. In this cycle, ambient air is taken into a rotating desiccant dehumidifier where moisture is adsorbed. The air is then sensibly and evaporatively cooled and introduced into the air-conditioned room. The air exiting room is first evaporatively cooled, then passed through the sensible heat exchanger where it recovers heat of adsorption from the supply air. Next, it is heated with low-grade thermal energy and the hot air is used to regenerate the desiccant.

In addition to the rotary desiccant cooling systems that operate according to the Pennington cycle, some different arrangements (cycles) were also studied in the literature. One early variation of the Pennington cycle was the recirculation cycle. In this cycle, return air is used as process air and ambient air is used as regeneration air. The main disadvantage of recirculation cycle is lacking in fresh air. Dunkle cycle [17], which in a sense is a cross between the Pennington and recirculation cycles, has an additional sensible heat exchanger for improving the cycle performance. Maclaine-Cross [18] proposed a simplified advanced solid desiccant cycle, namely SENS cycle. In this cycle, ambient air is first dehumidified in a rotary desiccant wheel, and then sensibly cooled in two heat exchanger in

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Nomenclature

c_p	Specific heat (kJ/kg K)
$\dot{C}$	Capacitance rate (kW/K)
COP	Coefficient of performance
$\dot{E}_{tot}$	Total energy input (kW)
h	Enthalpy (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Actual heat transfer rate (kW)
$\dot{Q}_{CC}$	Cooling capacity (kW)
$\dot{Q}_{max}$	Maximum possible heat transfer rate (kW)
$\dot{Q}_{reg}$	Energy consumption of regeneration heat (kW)
T	Temperature (°C)
W	Humidity ratio (kg/kg air)
$\dot{W}$	Power consumption (kW)
ΔW	Moisture removal (kg/kg air)

Greek letter

η	Effectiveness

Subscripts

1, 2, ..., 17	States described in Fig. 1
c	Cold air
com	Compressor
EC	Evaporative cooler
eh	Electric heaters
fan	Fans
f	Fresh air
h	Hot air
HE	Heat exchanger
i	Inlet
min	Minimum
o	Outlet
oth	Other electrical components
w	Wet bulb

tandem. It is then mixed with return air and cooled in a cooling coil by exchanging heat with cold water from a cooling tower. Some cooled air from the coil is supplied to the cooling tower from which it is discharged to the atmosphere and the rest is supplied to the conditioned space. On the regeneration side, ambient air is passed through the heat exchanger. Then, it is heated by heat source and finally drawn through the desiccant wheel to regenerate the desiccant. Jain et al. [19] investigated various solid desiccant cycles (the ventilation cycle, the recirculation cycle, the Dunkle cycle and the wet surface heat exchangers cycle (SENS cycle)) for various outdoor conditions of many cities in India.

Various researchers studied a system that regenerates the desiccant with double stage (staged regeneration) to improve thermal performance of rotary desiccant air conditioning systems. Ando et al. [20] investigated experimentally a four-rotor desiccant cooling process equipped with a double stage dehumidification. Their experimental results showed that the amount of dehumidified water at the process without evaporative cooler was larger than that of process with evaporative cooler. Performance of a desiccant cooling system with double stage dehumidification was also studied by Zhang and Niu [21] and Ge et al. [22]. They reported that the required regeneration temperature of the system is lower and coefficient of performance is higher than single stage desiccant cooling system. Ge et al. [23] investigated experimentally a one-rotor two stage rotary desiccant cooling system (OTSDC), in which two stage dehumidification process is realized by one desiccant wheel. They showed that OTSDC achieved the objective of reducing the size of two stage system with no reduction in system performance.

In addition to these cycles, some hybrid desiccant air conditioning systems are also studied. Tsay et al. [24] investigated the possibilities and methods of combining a desiccant cooling system with a CO_2 heat pump to improve overall efficiency and indoor air quality and to provide a steady heat source. They showed that using a relatively low reactivation temperature could satisfy the non-condensing premise and provide good indoor air quality conditions without mold pollution. It was also found that Coefficient of Performance (COP) of system based on energy consumption can be improved from 0.6 to 1.8.

Desiccant cooling systems are cost effective systems when a cheap heat source especially solar energy is used to regenerate the desiccant. For this purpose, various studies were performed [25–30].

The literature search showed that different types of rotary desiccant cooling systems were studied both analytically and experimentally by several researchers. In this study, a novel desiccant based air conditioning system was designed and tested experimentally to improve the indoor air quality and reduce energy consumption. In the system studied, the moisture of the air sent to the place being air-conditioned was reduced passing it through a solid desiccant wheel and then its temperature was brought to the desired blowing temperature by the "dry coil" of a vapor compression cycle. To enhance the performance of the system, some techniques such as "pre-cooling with outdoor air", "waste cool recovery", "pre-cooling of waste air with evaporative cooling" and "pre-heating the regeneration air with exhaust air" were utilized.

2. System description and experimental setup

Fig. 1 illustrates a schematic view of desiccant cooling system studied. In the system, three separate air channels are used for fresh, waste and regeneration air streams. Fresh air channel is used to supply fresh air to the air-conditioned room. The waste air sucked from the air-conditioned room is sent to outside via waste air channel. Regeneration air channel is used to remove moisture of desiccant wheel.

Humidity of the fresh outside air (state 1) is absorbed by the desiccant material of the wheel. The fresh air leaves the process side of the desiccant wheel (state 2) hotter, and much drier than when entering the system. Sensible cooling of the fresh air is carried out (process 4–5) in a cooling coil, which is fed by chilled water from a chiller unit. However, the fresh air is passed through two recuperative type heat exchangers (heat exchangers 1 and 2) before coming to the cooling coil for cool recovery. In the recuperators, only sensible heat is removed from the fresh air, and therefore, humidity ratio of the air remains constant.

The desired blowing temperature of the fresh air (state 6) is obtained in the cooling coil. The surface temperature of the cooling coil is always kept higher than the dew point temperature of the fresh air entering into the cooling coil. Therefore, only sensible cooling is performed in the coil (dry coil application). Moisture from the fresh air (latent cooling) is removed only in the desiccant wheel and there is no change in the humidity ratio of the fresh air after the desiccant wheel.

The air sucked from the indoor (state 7) into the waste air duct is evaporatively cooled before entering into heat exchanger 2 in order to increase the cool recovery. In heat exchanger 2, due to the heat transfer from the fresh air to the waste air, temperature of the fresh air decreases and that of the waste air increases (process 9–10). The waste air leaving the heat exchanger is rejected to the outdoors.

Hot regeneration air is used for removing moisture at the desiccant wheel. In this study, fresh outdoor air is used for regeneration process. The reason of using ambient air is to eliminate the possibility of mixing of the fresh air with the waste air, which may contain

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harmful bacteria originating from the air-conditioned room. The regeneration air first comes to heat exchanger 1 in which heat transfer from the fresh air to the regeneration air takes place. Temperature of the regeneration air at the exit of the desiccant wheel (state 15) is generally higher than that of the regeneration air leaving heat exchanger 1 (state 12). Therefore, a rotary regenerator (heat exchanger 4) is used for heat recovery. The regeneration air leaving the desiccant wheel passes through the heat exchanger 4 (process 15–16), in which heat is transferred from the regeneration air that left the desiccant wheel (state 15) to the regeneration air left heat exchanger 1 (state 12). Although the temperature of the regeneration air is increased in heat exchangers 1 and 4 (process 11–13), it is not high enough for dehumidification of the desiccant wheel. Therefore, temperature of the regeneration air should be increased to the desired temperature before entering into the desiccant wheel. Temperature can be increased economically, if a cheap thermal energy source such as solar energy, waste heat or geothermal energy is available. Utilization of such cheap heating energy is very important for the economic operation of desiccant systems. If these systems can be backed up by a cheap thermal energy source, operating cost can be decreased dramatically. In this study, the final temperature of the regeneration air is achieved with the help of electric heaters (process 13–14) to simulate the cheap heat source. Electric heaters are used for adjusting the limits of regeneration temperature accurately, for simplifying the system and also reducing the size of the system. The air removes the humidity of the desiccant wheel (process 14–15) and flows through heat exchanger 4 (process 15–16) before discharged to the outdoors.

Fig. 2 shows the photographic view of the desiccant cooling system studied. The system mainly consists of a desiccant wheel, four sensible heat exchangers, three fans, refrigeration unit, an evaporative cooler and an electric heater unit. In this study, a room that has 58.7 m^2 floor area and a volume of 223 m^3 is air-conditioned.

The desiccant wheel with a diameter of 965 mm and a depth of 200 mm is coated with a silica gel as adsorbent material. The process and regeneration areas are apportioned 50–50% of the wheel face area. Rotational speed of the wheel is 12 rev h^{-1}.

Four sensible heat exchangers are used in the system. Two of them (heat exchangers 1 and 2) are recuperative type (cross-flow, air to air) with dimensions of 600 mm × 600 mm × 600 mm. The fin-tube cooling coil (heat exchanger 3), which is composed of aluminum fins and copper tubes, has 5 rows and 18 tubes in each row. The dimension of the coil is 600 mm × 600 mm × 140 mm. Heat exchanger 4 (rotary regenerator) is an aluminum honeycomb structure. The diameter and depth of the regenerator that are equal with desiccant wheel are 965 mm and 200 mm, respectively. It rotates at 12 rev min^{-1}.

Three centrifugal fans are used for the circulation of air in the fresh, waste and regeneration channels. Flow rates of the fresh, waste and regeneration air streams are adjusted precisely by the frequency controllers that drive the fan motors.

Water that is used to cool the process air down to blowing temperature is cooled by vapor compression (refrigeration) unit. In the refrigeration unit, a semi-hermetic type compressor, an air-cooled condenser with 120 m^2 area and a tube type evaporator with 50 m tube length and 25 mm tube diameter are used. Evaporator is located inside an insulated water tank with a volume of 315 l.

In the evaporative cooler, fogging technology is used to humidify the air. 15 spray nozzles with 8 μm droplet diameters are placed in 5 rows in the evaporative cooler.

Electric heater unit that is used to simulate the cheap heat source consists of 24 U-shaped heating elements (each one is 2 kW). The heating elements are placed in 3 rows to obtain a uniform temperature distribution inside the unit.

Temperature, relative humidity, flow rate, electric current and electrical potential difference are measured during the experiments. Dry bulb temperature of the air at states 1–17 (Fig. 1) and temperature of the cooled water are measured using K-type thermocouples with an accuracy of 0.1 °C. An ice-bath is employed as the reference junction for all the thermocouples used. Relative

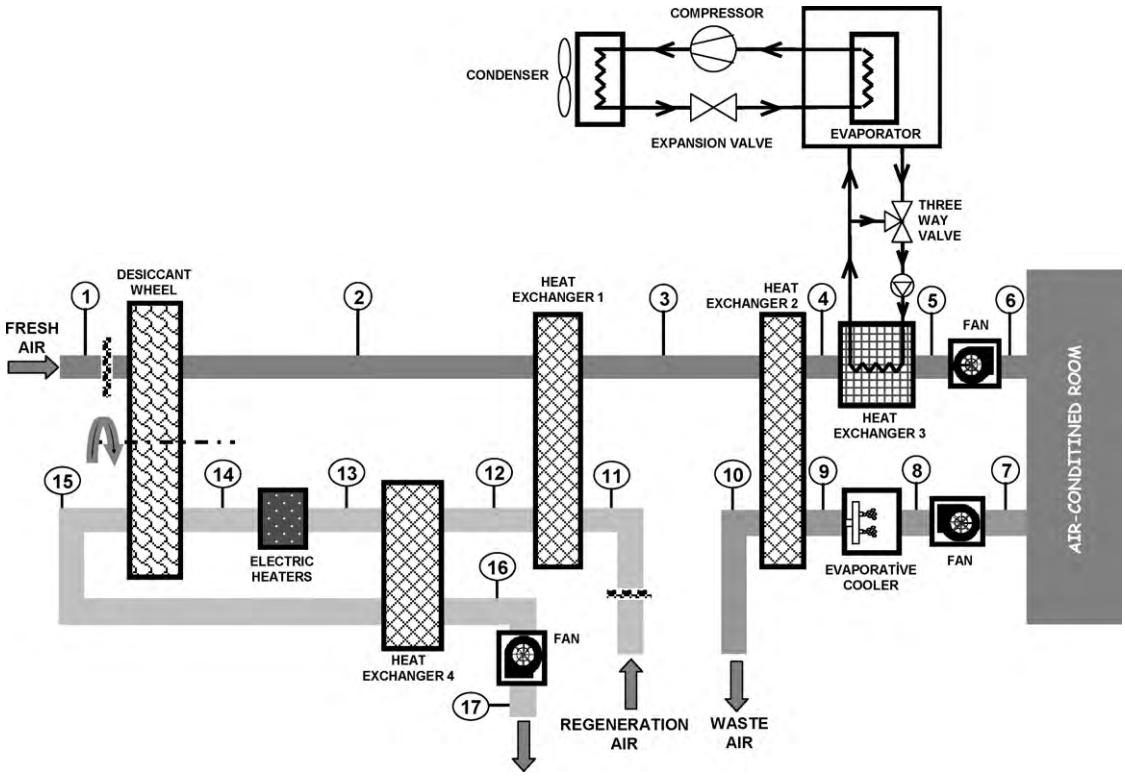


Fig. 1. Schematic view of desiccant cooling system studied.

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Fig. 2. The photographic view of desiccant cooling system studied.

humidity of the air at states 1, 3, 6, 7, 9 and 16 is measured using relative humidity transmitters with an accuracy of ±2% (0–90 RH), ±3% (90–100 RH). A dew point transmitter with an accuracy of ±0.2 °C is utilized to determine dew point temperature of the air at state 4. Flow rates of the fresh, waste and regeneration air streams at states 2, 7 and 15 are measured using averaging (blade type) pitot tubes and differential pressure transmitters with an accuracy of ±1%. Electric current and electrical potential difference are measured to determine the power consumption of each electrical component in the experimental system. Monitoring modules are used to measure the electric current and electrical potential difference with an accuracy of ±0.5% and ±1.5%, respectively. Electrical signal outputs of the thermocouples, transmitters and modules are monitored and recorded with an interval of 30 s using a computer-controlled data acquisition system with 16 bit resolution and 1 MHz maximum sample rate.

A Programmable Logic Controller (PLC) is used to control the operation of desiccant cooling system. Its functions included: (a) controlling the system and operation of its components (start-up and shut-down); (b) adjusting the dry bulb temperature of the air-conditioned room according to the set values by controlling the cooled water temperature with three way valve; (c) adjusting the relative humidity of the air-conditioned room according to the set values by controlling the regeneration temperature with electric heaters; (d) preventing the condensation at heat exchanger 3 by controlling the cooled water temperature with three way valve; (e) protecting the desiccant wheel from the high temperature with the control of electric heaters.

3. Calculations and uncertainty analysis

In the analysis of the experimental data, various calculations (effectiveness of the heat exchangers and humidifier, moisture removal, cooling capacity, COP) are carried out to determine performance of the system and its components.

The effectiveness of the heat exchangers (η_{HE}) is determined by:

$$\eta_{HE} = \frac{\dot{Q}}{\dot{Q}_{max}} \quad (1)$$

where $\dot{Q}$ is the amount of actual heat transfer and $\dot{Q}_{max}$ is the maximum possible heat transfer:

$$\dot{Q} = \dot{m} c_p (T_i - T_o) \quad (2)$$

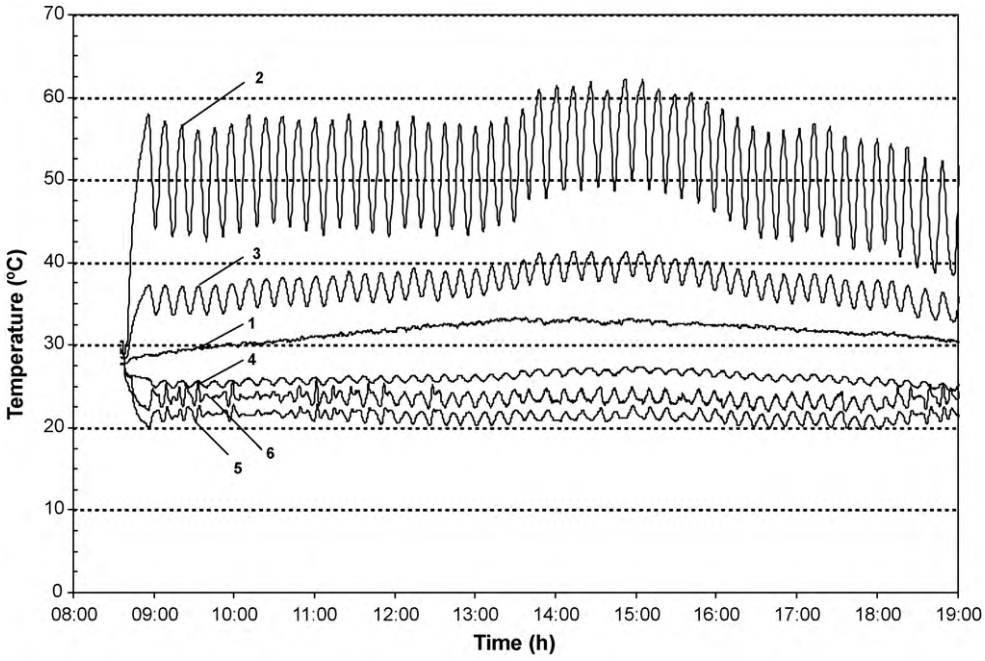


Fig. 3. Variation of dry bulb temperature during the day at different states for fresh air stream.

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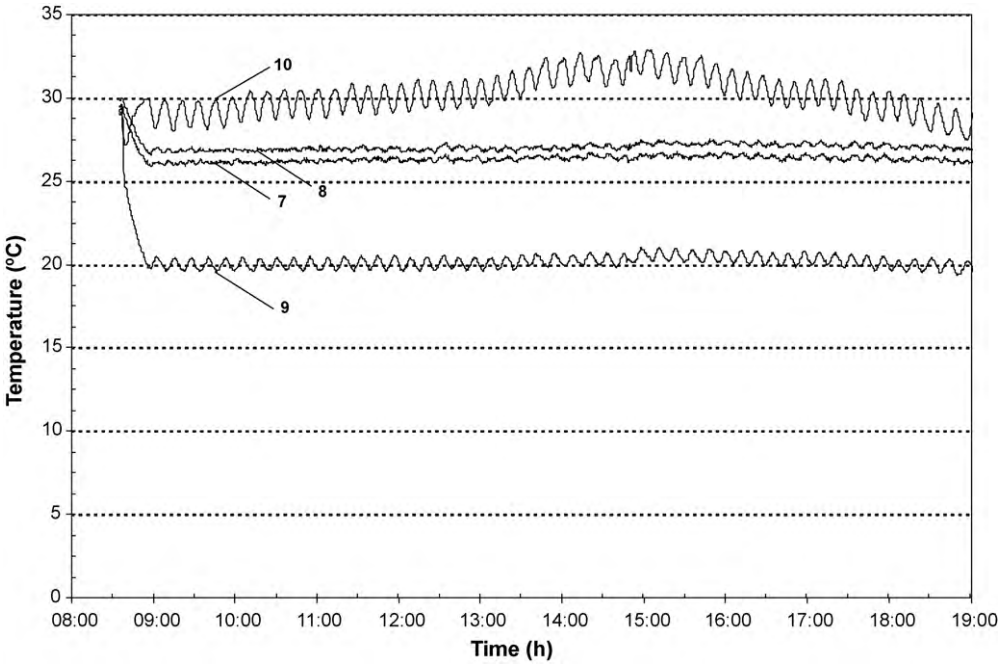


Fig. 4. Variation of dry bulb temperature during the day at different states for waste air stream.

$$\dot{Q}_{max} = \dot{C}_{min}(T_{hi} - T_{ci}) \quad (3)$$

$\dot{C}_{min}$ is the minimum of the capacitance rate of cold ($\dot{C}_c$) and hot ($\dot{C}_h$) air streams:

$$\dot{C}_h = \dot{m}_h c_{ph} \quad (4)$$

$$\dot{C}_c = \dot{m}_c c_{pc} \quad (5)$$

The effectiveness of evaporative cooler (η_{EC}) is determined by:

$$\eta_{EC} = \frac{T_8 - T_9}{T_8 - T_{8w}} \quad (6)$$

where T_{8w} is the wet bulb temperature of waste air at state 8.

Amount of the moisture removed from the process air (moisture removal, ΔW) is defined as the difference between the humidity ratio at the inlet (W_1) and outlet (W_2) of the desiccant wheel:

$$\Delta W = W_1 - W_2 \quad (7)$$

Cooling capacity of a desiccant cooling system $\dot{Q}_{CC}$ is obtained from:

$$\dot{Q}_{CC} = \dot{m}_f(h_6 - h_1) \quad (8)$$

where $\dot{m}_f$ is the mass flow rate of the fresh air stream and h is the enthalpy of the air.

The coefficient of performance (COP) of the system is defined as the ratio between the cooling capacity obtained and total energy

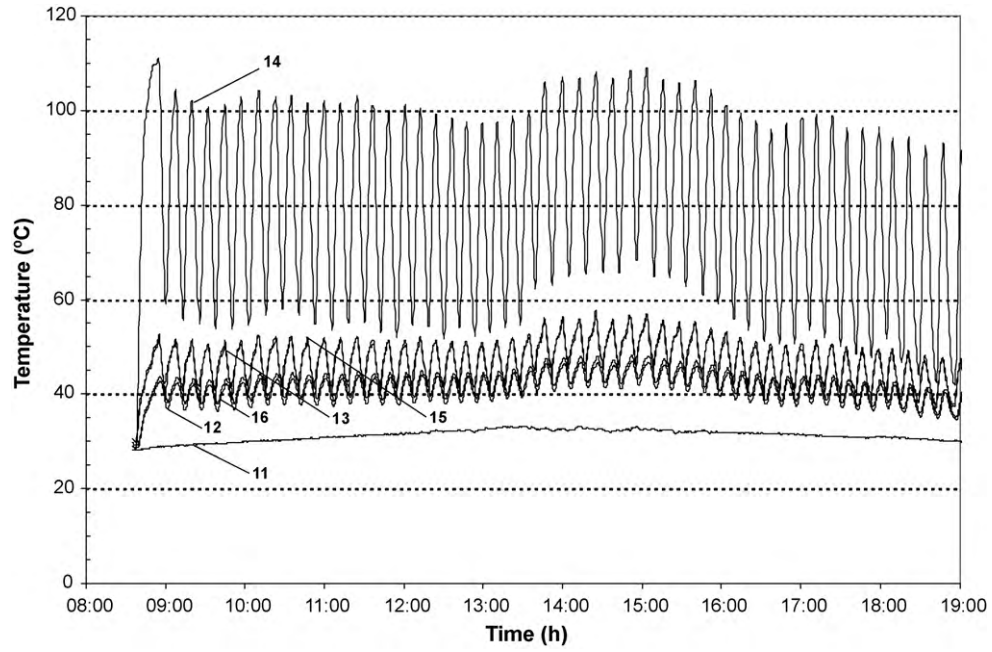


Fig. 5. Variation of dry bulb temperature during the day at different states for regeneration air stream.

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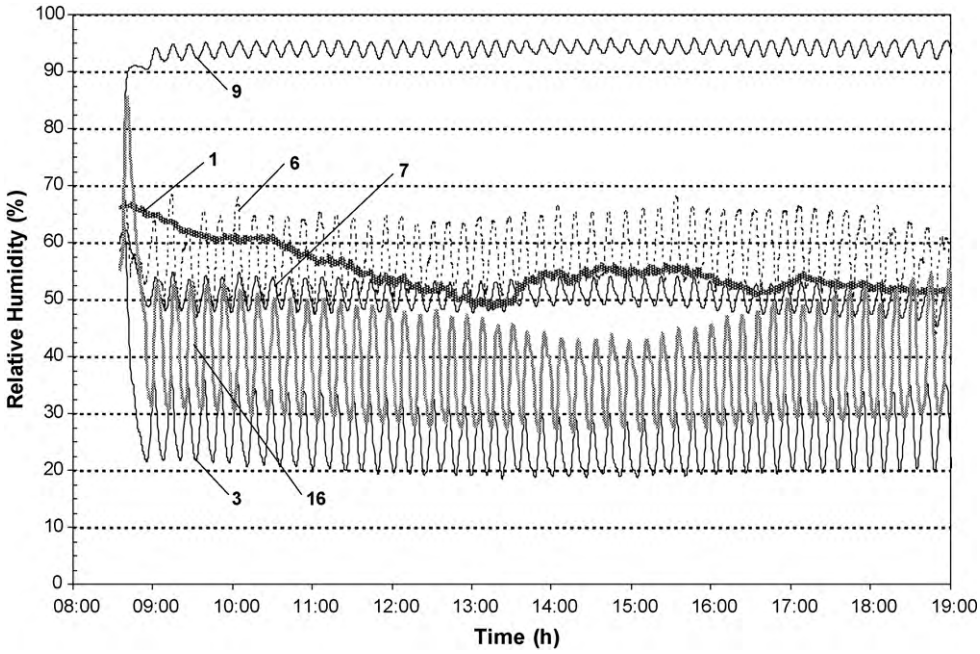


Fig. 6. Variation of relative humidity during the day at different states.

input ($\dot{E}_{tot}$) to the system:

$$\mathrm{COP} = \frac{\dot{Q}_{CC}}{\dot{E}_{tot}} \tag{9}$$

$\dot{E}_{tot}$ is the sum of energy consumption of regeneration heat ($\dot{Q}_{reg}$), fans ($\dot{W}_{fan}$), compressor of the vapor compression refrigeration cycle ($\dot{W}_{com}$) and other electrical components ($\dot{W}_{oth}$):

$$\dot{E}_{tot} = \dot{Q}_{reg} + \dot{W}_{fan} + \dot{W}_{com} + \dot{W}_{oth} \tag{10}$$

In this study, since electric heaters are used to simulate regeneration heat source, the heat input for the regeneration is almost same as power consumption of electric heaters ($\dot{W}_{eh}$) when the daily total values are considered ($\dot{Q}_{reg} = \dot{W}_{eh}$).

Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis is performed using the method described by Holman [31]. In the present study, the temperature, relative humidity, flow rate, electric current and electrical potential difference are measured with appropriate instruments explained previously. The total uncertainties of the measurements are estimated to be ±0.3 °C for the air and water temperatures, ±2.51% for the relative humidities, ±1.59% for power inputs to the electrical heaters, compressor and motor of the fans, dehumidifier, regenerator and circulating pump. The total uncertainty associated

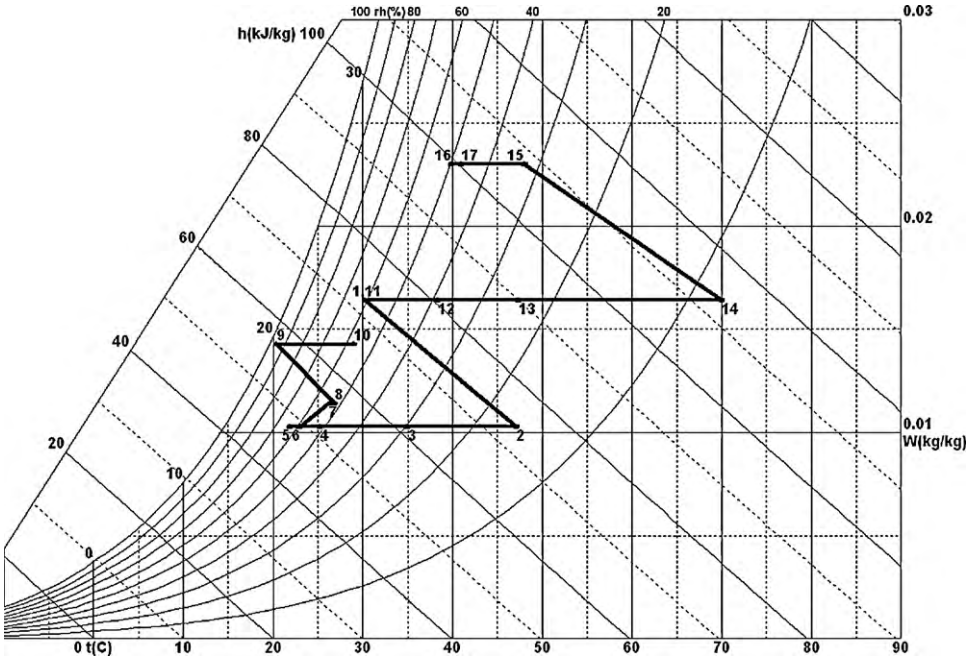


Fig. 7. Psychrometric diagram showing all processes at 10:30 am.

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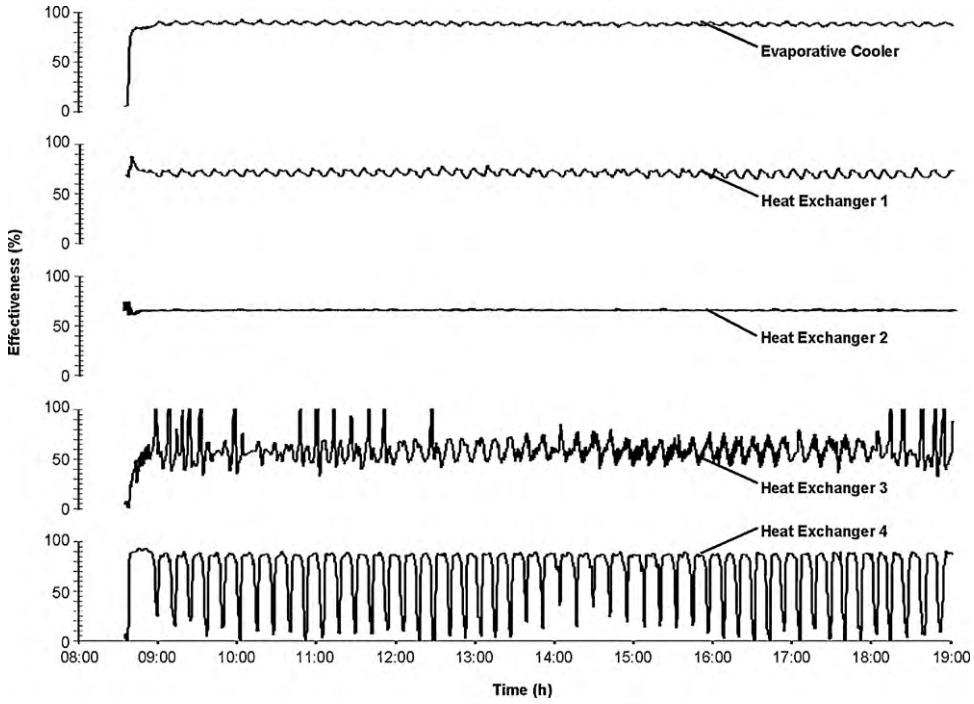


Fig. 8. Variation of effectiveness of evaporative cooler and heat exchangers during the day.

with mass flow rates of the air is found to be ±2.89%. The uncertainties associated with the calculated parameters moisture removal, cooling capacity and COP are found to be ±4.2%, ±6.3% and ±7.1%, respectively.

4. Results and discussion

A series of experiments are performed during the cooling season of 2008 to determine the performance characteristics of the system investigated. In the present study, the results obtained from the experiment on 29 July 2008, which are typical, are given and discussed. During the experiment, flow rate of the air streams (fresh, waste and regeneration) are kept constant at 4000 m^3/h. These very high flow rate values are selected to simulate very high-standard hygienic air conditioning [32] which in turn leads to very high cooling loads. Temperature and relative humidity of the air-conditioned room are adjusted 26 °C and 50%, respectively according to ASHRAE comfort zone [1].

Figs. 3–6 present dry bulb temperatures and relative humidities of air measured at different states shown in Fig. 1 during the experiment selected. At the beginning of the test, regeneration temperature is set to 120 °C. Figs. 3–5 show variation of dry bulb

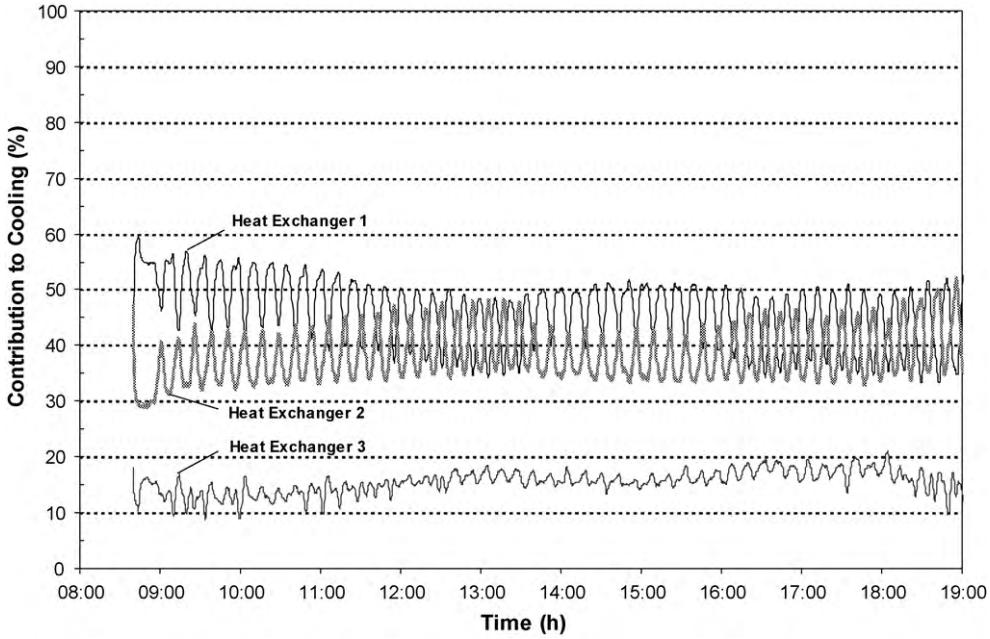


Fig. 9. Contribution of heat exchangers 1–3 to total cooling.

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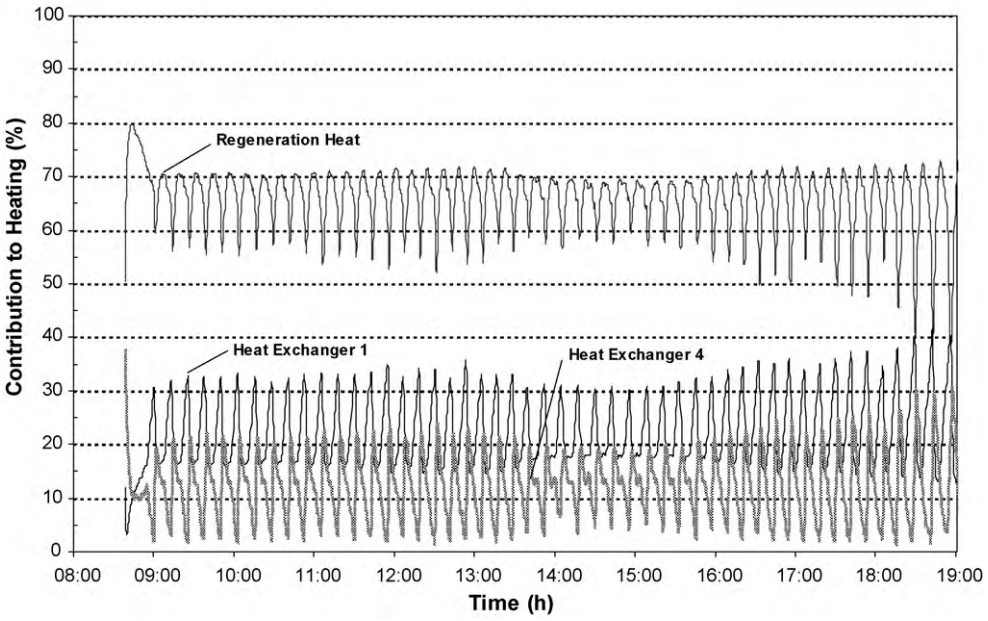


Fig. 10. Contribution of heat exchangers 1 and 4, and regeneration heat to the total heating.

temperature during the day at different states for fresh, waste and regeneration air streams, respectively. As can be seen from the figures, all the air temperatures except the ambient (T_1 and T_{11}) and the air-conditioned room (T_7) fluctuate with time. The fluctuations originate from the start-up and shut-down of the electric heaters, which is needed for the control of humidity in the air-conditioned room.

As can be seen from Fig. 3, dry bulb temperature of fresh outside air that enters the fresh air channel at 28 °C (T_1), increases to 55–60 °C (T_2) after the dehumidification process and then its temperature decreases down to blowing temperature (approximately 22 °C) by passing through the heat exchangers ($T_2 \rightarrow T_6$).

Fig. 4 shows dry bulb temperature of waste air stream at different states. It can be seen that after a short time (30 min), dry bulb temperature of the air-conditioned room (T_7) decreases down to set value 26 °C and remains constant rest of the day. It can also be seen from the figure that temperature of the air sucked from the room (T_7) is decreased down to 20 °C (T_9) with the help of the evaporative cooler before entering into heat exchanger 2.

Variation of regeneration temperature (T_{14}) during the day is shown in Fig. 5. Although the regeneration temperature is set to 120 °C in this test, maximum temperature remains around 100 °C. This is because, when the relative humidity at the air-conditioned room reaches to the set value of 50%, the electric heaters shut-

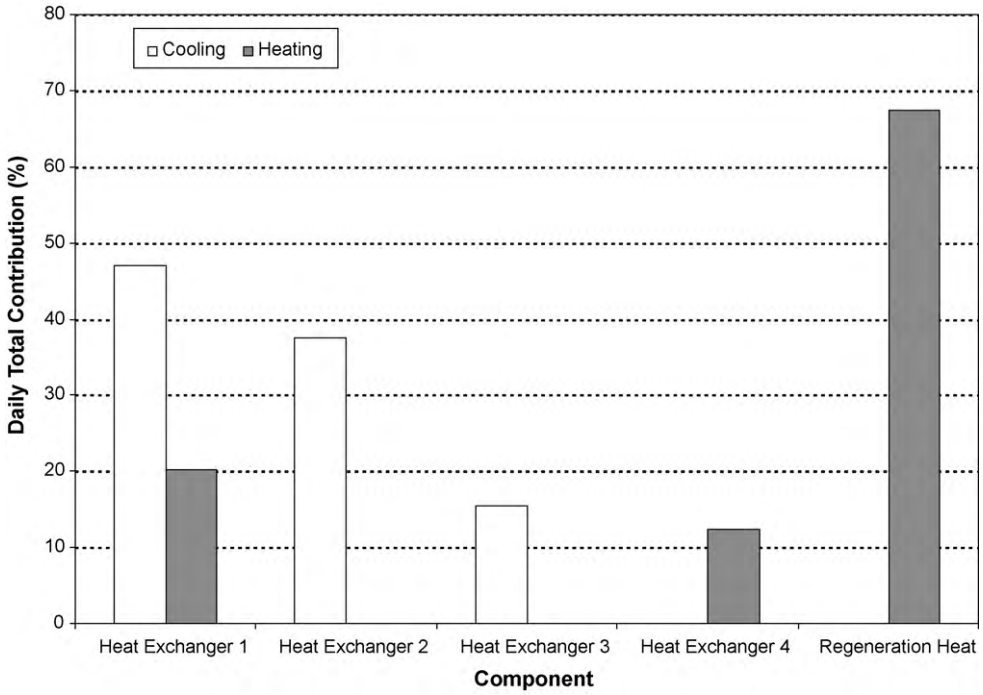


Fig. 11. Daily total contribution of heat exchangers 1–4 and regeneration heat to the heating and cooling requirement.

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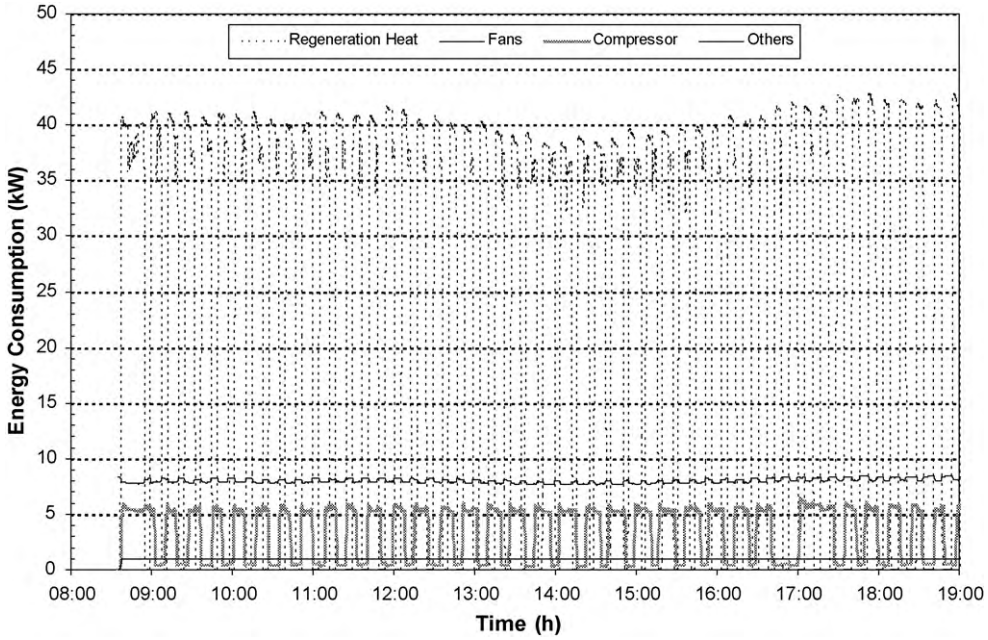


Fig. 12. Variation of energy consumption of regeneration heat and each electrical component during the day.

down. Therefore, without the need of regeneration temperature to reach 120 °C, the desired humidity value is obtained at the room.

Fig. 6 shows the relative humidities measured during the day at different states. As can be seen from the figure, relative humidity of the air-conditioned room changes around the set value 50% within a band of ±3%. Similar fluctuations with different amplitudes are also evident in the humidities at states 3, 6, 9 and 16. In this study, control of the humidity inside the air-conditioned room is achieved with the control of the temperature of the regeneration air. Due to the thermal inertia of the electric heaters, the dehumidifier and the air mass inside the channels, it is not possible to keep the humidity inside the air-conditioned room steady. However, amplitude of the fluctuations can be minimized by feeding the influence of the parameters above into the PLC. It can also be seen from the figure that humidity of the waste air is increased from 50% to approximately 95% by the evaporative cooler.

Fig. 7 shows all the processes on a psychrometric diagram at 10:30 am.

Variation of effectiveness of the evaporative cooler and the heat exchangers during the day is shown in Fig. 8. Effectiveness is approximately 65–75% for heat exchangers 1 and 2, 60% for heat exchanger 3, and 90% for heat exchanger 4 and the evaporative cooler. It can be seen from the figure that fluctuations are evident

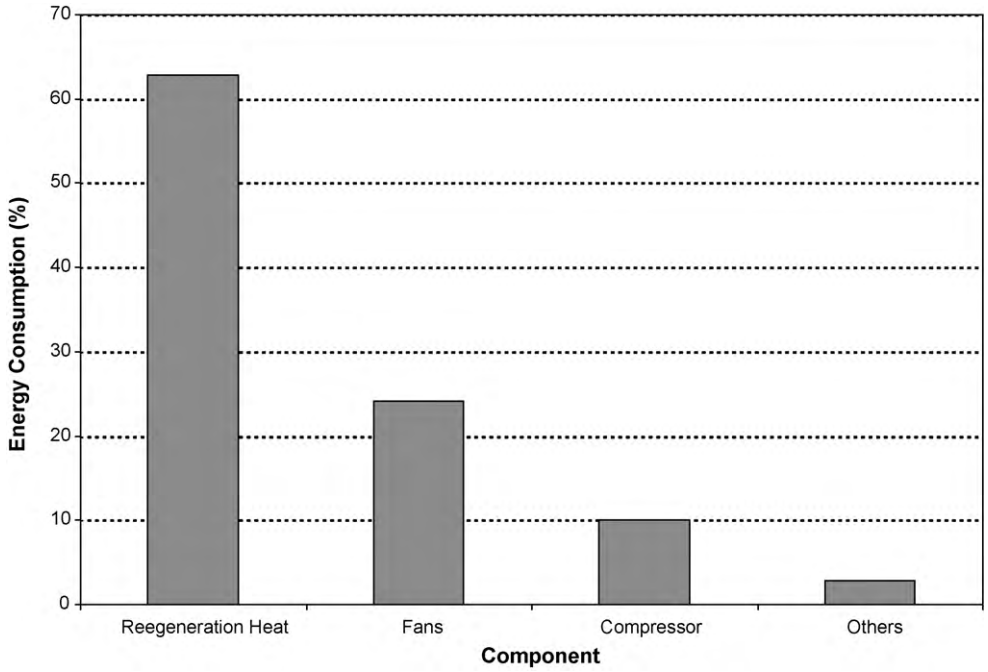


Fig. 13. Daily total energy consumption of regeneration heat and each electrical component.

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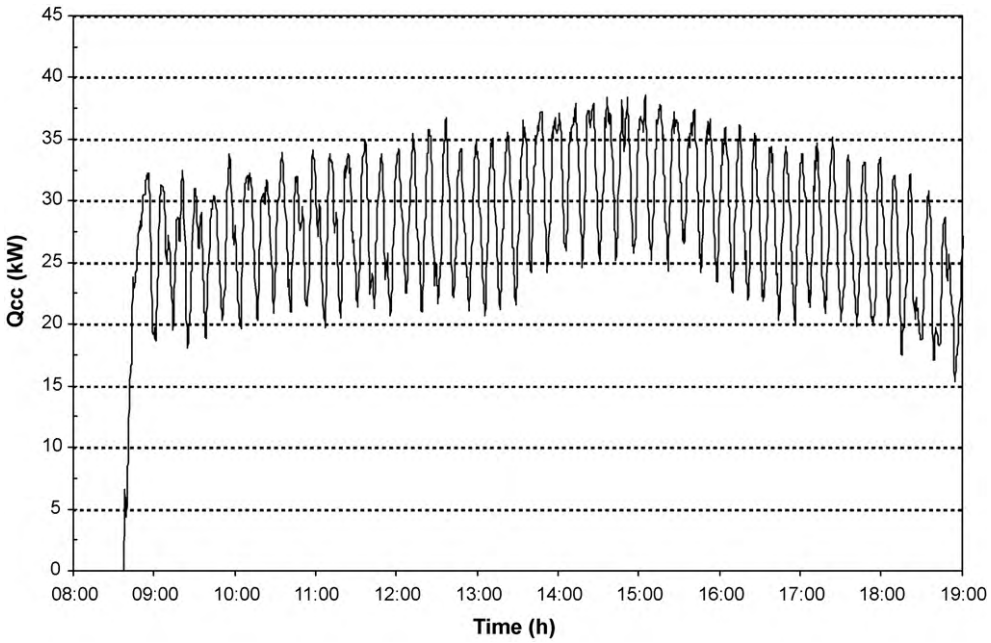


Fig. 14. Variation of cooling capacity during the day.

in the effectiveness values, again due to start-up and shut-down of the electric heaters to control the humidity in the air-conditioned room.

Temperature of the fresh outside air increases significantly during the dehumidification process. It is pre-cooled in heat exchangers 1 and 2 before brought to the desired blowing temperature in heat exchanger 3. Fig. 9 presents the contribution of each heat exchanger to the total cooling requirement after dehumidification process (states 2–5) in percentage. As can be seen from the figure, approximately 45% of cooling requirement is met by heat exchanger 1, 40% by heat exchanger 2 and the remaining 15% by heat exchanger 3 (cooling coil). It is clear that use of heat exchangers 1 and 2 reduces significantly the cooling requirement. Approximately 85% of cooling is met without any external energy consumption.

Temperature of the regeneration air is a very important parameter for the moisture removal from the supply air. Regeneration air is pre-heated with the help of heat exchangers 1 and 4 before brought to the final regeneration temperature by the external regeneration heat source. Contribution of heat exchangers 1 and 4, and the regeneration heat to the total heating in percentage is given in Fig. 10. Approximately 65% of the heating requirement is met by the regen-

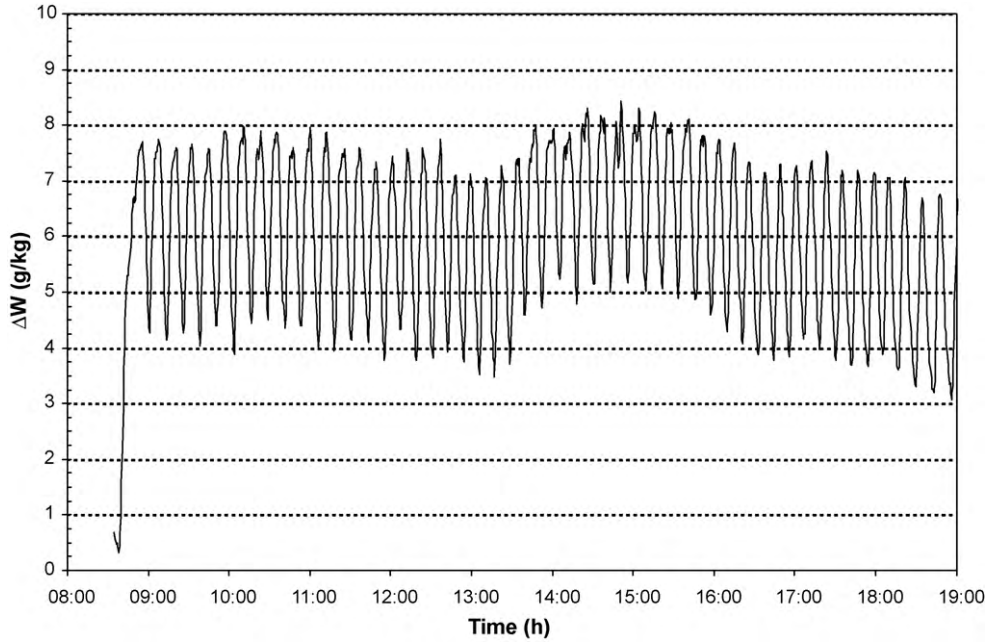


Fig. 15. Variation of moisture removal during the day.

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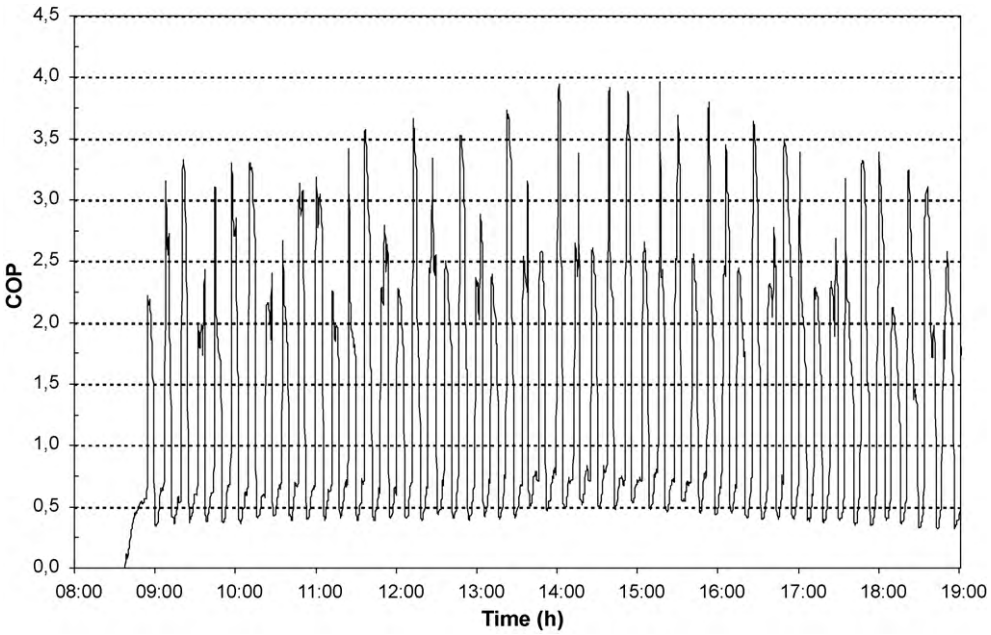


Fig. 16. Variation of coefficient of performance during the day.

eration heat, 25% by heat exchanger 1 and 10% by heat exchanger 4. Use of heat exchangers 1 and 4 reduces the heating requirement approximately 35%.

Daily total contribution of heat exchangers 1–4 and the regeneration heat to the heating and cooling requirement is presented in Fig. 11. The contribution of heat exchangers 1 and 2 can clearly be seen from the figure.

To provide comfort conditions in the air-conditioned room, in addition to the regeneration heat, which is produced by the electric heaters in this study, the fans, the refrigeration unit and other auxiliary electrical components (motors of the dehumidifier, the regenerator and the water pumps) consume energy. Sum of all these constitute total energy consumption ($\dot{E}_{tot}$). Fig. 12 shows the energy consumption of these units. Although power consumption of the fans and the auxiliary electrical components is constant during the day, variations in the regeneration heat and the refrigeration unit are considerable.

The highest energy consumption occurs in regeneration heat (Fig. 13). It is followed by the fans, the refrigeration unit and other electrical components. This shows the importance of using cheap heat source in desiccant cooling systems.

Fig. 14 shows variation of cooling capacity during the day. Cooling capacity exhibits fluctuations with a magnitude of approximately ±5 kW. An average 30 kW of cooling effect is produced by the system to provide comfort conditions in the air-conditioned room.

Variation of moisture removal during the day is shown in Fig. 15. As can be seen from the figure on an average 6 g/kg moisture is

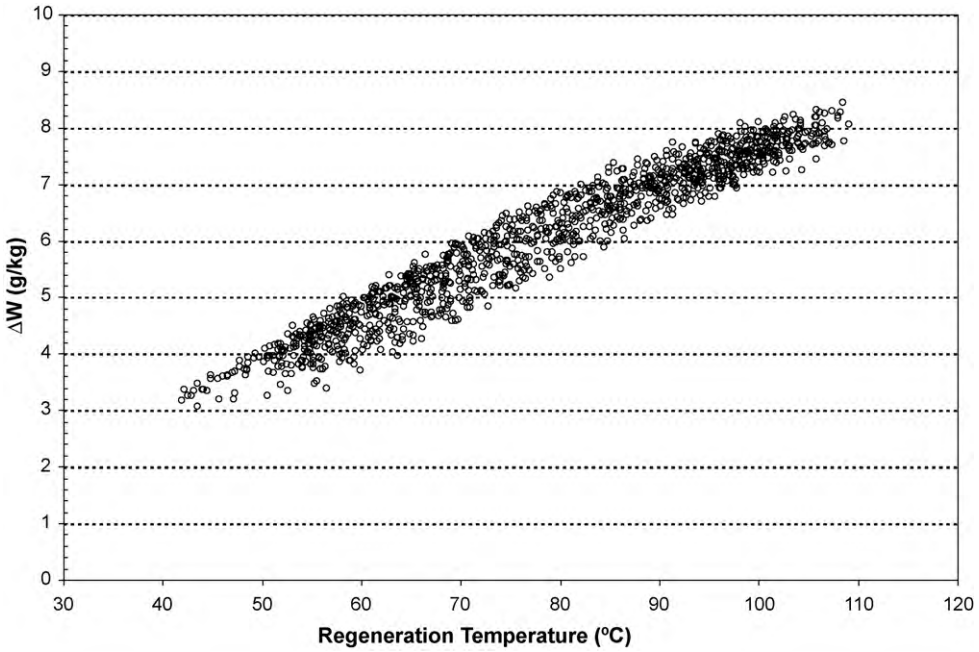


Fig. 17. Effect of regeneration temperature on moisture removal.

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removed from the ambient air in the desiccant wheel during the experiment.

Coefficient of performance (COP) of the system is shown in Fig. 16. Although COP changes between 0.4 and 4 according to the electric heaters switching on or off, the daily mean value is 1.35. This value is in the range of the COP values reported in the literature for these kinds of systems [9,12].

In this study, control of the humidity inside the air-conditioned room is achieved with the control of the temperature of the regeneration air. Therefore, regeneration temperature changes during the experiment according to the humidity inside the air-conditioned room. Effect of regeneration temperature on moisture removal is shown in Fig. 17, which contains 1250 data points representing the experiment selected. As can be seen from the figure, the moisture removed from the fresh air increases almost linearly with the increase of regeneration temperature.

Similar results are obtained from the other experiments performed during the cooling season of 2008.

5. Conclusion

A novel desiccant based air conditioning system is developed and initially tested. In the system studied, the moisture of the fresh air is reduced passing it through a solid desiccant wheel and then its temperature decreased by the "dry coil" of a vapor compression cycle. The following main conclusions emerged from this study:

1. Using a heat exchanger (heat exchanger 4), which is not used in this type of systems, for pre-heating the regeneration air with exhaust air is feasible to install. Moreover, heat pollution to the environment can be decreased with this heat exchanger.
2. Approximately 45% of cooling requirement is met by heat exchanger 1, 40% by heat exchanger 2 and the remaining 15% by heat exchanger 3 (cooling coil). It is clear that use of heat exchangers 1 and 2 reduces significantly the cooling requirement.
3. Approximately 65% of the heating requirement is met by the regeneration heat source, 25% by heat exchanger 1 and 10% by heat exchanger 4. Use of heat exchangers 1 and 4 reduces the heating requirement approximately 35%.
4. The highest energy consumption occurs in regeneration heat. This shows the importance of using a cheap heat source in desiccant cooling systems.
5. Although COP changes between 0.4 and 4 according to the electric heaters switching on or off, the daily mean value is 1.35.

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References

[1] ASHRAE, Handbook of Fundamental, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc, Atlanta, USA, 2001.
[2] A.V. Arundel, E.M. Sterling, J.H. Bigging, T.D. Sterling, Indirect health effects of relative humidity in indoor environments, Environmental Health Perspectives 65 (1986) 351–361.
[3] K. Abe, Evaluation of fungal growth at the air outlet and inlet of air conditioners, Indoor Air 3 (1996) 185–190.
[4] A.L. Hines, T.K. Ghosh, S.K. Loyalka, R.C. Warder, Investigation of co-sorption of gases and vapors as a means to enhance indoor air quality-phase 2: removal of particulates and airborne micro-organisms by solid adsorbents and liquid desiccants. Report of Gas Research Institute, 1992, GRI-92/0157.5.
[5] J.A. Phillips, M.B. Wagner, Antiseptic Effects of Desiccant Based HVAC Systems, Lehigh University, Bioprocessing Institute, 1994.
[6] B. Kovak, P.R. Heimann, The sanitizing effects of desiccant-based cooling, ASHRAE Journal 39 (1997) 60–64.
[7] M.D. Larranaga, The capability of a solid sorbent desiccant unit at removing selected indoor air quality-related microorganisms from the air, PhD Thesis, Graduate Faculty of Texas Tech University, 2004.
[8] L. Fang, G. Zhang, A. Wisthaler, Desiccant wheels as gas-phase absorption (GPA) air cleaners: evaluation by PTR-MS and sensory assessment, Indoor Air 18 (2008) 375–385.
[9] D.G. Waugaman, A. Kini, C.F. Kettleborough, A review of desiccant cooling systems, Journal of Energy Resources Technology 115 (1) (1993) 1–8.
[10] P. Mazzei, F. Minichiello, D. Palma, HVAC dehumidification systems for thermal comfort: a critical review, Applied Thermal Engineering 25 (2004) 677–707.
[11] K. Daou, R.Z. Wang, Z.Z. Xia, Desiccant cooling air conditioning: a review, Renewable and Sustainable Energy Reviews 10 (2006) 55–77.
[12] D. La, Y.J. Dai, Y. Li, R.Z. Wang, T.S. Ge, Technical development of rotary desiccant dehumidification and air conditioning: a review, Renewable and Sustainable Energy Reviews 14 (2010) 130–147.
[13] J.J. Jurinak, J.W. Mitchell, W.A. Beckman, Open-cycle desiccant air conditioning as an alternative to vapor compression cooling in residential applications, Journal of Solar Energy Engineering 106 (1984) 252–260.
[14] P.L. Dhar, S.K. Singh, Studies on solid desiccant based hybrid air-conditioning systems, Applied Thermal Engineering 21 (2001) 119–134.
[15] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang, Analysis on a hybrid desiccant air-conditioning system, Applied Thermal Engineering 26 (2006) 2393–2400.
[16] N.A. Pennington, Humidity changer for air conditioning, US Patent No. 2,700,537 (1955).
[17] R.V. Dunkle, A method of solar air conditioning, Mechanical and Chemical Engineering Transactions of the Institute of Engineers 73 (1965) 73–78.
[18] I.L. Maclaine-Cross, A theory of combined heat and mass transfer in regenerations, PhD Thesis, Monash University, 1974.
[19] S. Jain, P.L. Dhar, S.C. Kaushik, Evaluation of solid desiccant based evaporative cooling cycles for typical hot and humid climates, International Journal of Refrigeration 18 (5) (1995) 287–296.
[20] K. Ando, A. Kodama, T. Hırose, M. Goto, H. Okano, Experimental study on a process design for adsorption desiccant cooling driven with a low-temperature heat, Adsorption 11 (2005) 631–636.
[21] H. Zhang, J. Niu, A two-stage desiccant cooling system using low-temperature heat, Building Services Engineering Research Technology 20 (2) (1999) 51–55.
[22] T.S. Ge, Y. Li, R.Z. Wang, Y.J. Dai, Experimental study on a two-stage rotary desiccant cooling system, International Journal of Refrigeration 32 (2009) 498–508.
[23] T.S. Ge, Y.J. Dai, R.Z. Wang, Y. Li, Experimental investigation on a one-rotor two-stage rotary desiccant cooling system, Energy 33 (2008) 1807–1815.
[24] Y.S. Tsay, S. Kato, R. Ooka, M. Koganei, K. Nishida, K. Kawamoto, Study on non condensing air conditioning system performance when combining a desiccant cooling system with a CO_2 heat pump, HVAC&R Research (2006) 917–933, Special Issue 12 (3c).
[25] S.P. Halliday, C.B. Beggs, P.A. Sleigh, The use of solar desiccant cooling in the UK: a feasibility study, Applied Thermal Engineering 22 (2002) 1327–1338.
[26] A. Kodama, M. Ohkura, T. Hirose, M. Goto, An energy flow analysis of a solar desiccant cooling equipped with a honeycomb adsorber, Adsorption 11 (2005) 597–602.
[27] A.E. Kabeel, Solar powered air conditioning system using rotary honeycomb desiccant wheel, Renewable Energy 32 (11) (2007) 1842–1857.
[28] N. Enteria, H. Yoshino, A. Mochida, R. Takaki, A. Satake, R. Yoshie, T. Mitamura, S. Baba, Construction and initial operation of the combined solar thermal and electric desiccant cooling system, Solar Energy 83 (2009) 1300–1311.
[29] P. Bourdoukan, E. Wurtz, P. Joubert, Experimental investigation of a solar desiccant cooling installation, Solar Energy 83 (2009) 2059–2073.
[30] Y. Sukamongkol, S. Chungpaibulpatana, B. Limmeechokchai, P. Sripadungtham, Condenser heat recovery with a PV/T air heating collector to regenerate desiccant for reducing energy use of an air conditioning room, Energy and Buildings 42 (2010) 315–325.
[31] J.P. Holman, Experimental Methods for Engineers, fifth edn, McGraw Hill, NY, 1989.
[32] ASHRAE, Handbook of HVAC Applications, American Society of Heating, Refrigerating and Air Conditioning Engineers, Inc, Atlanta, USA, 2003.