Energy and Buildings 45 (2012) 197–201

Contents lists available at SciVerse ScienceDirect

Energy and Buildings

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

Optimization of solar collector surface in solar desiccant wheel cycle

Zahra Hatami [a,*], Mohammad Hassan Saidi [b], Masoud Mohammadian [c], Cyrus Aghanajafi [d]

[a] Department of Mechanical Engineering, Science and Research Branch, Islamic Azad University, Tehran, Iran
[b] Mechanical Engineering Department, Sharif University of Technology, Tehran 11155-9567, Iran
[c] Department of Mechanical Engineering, South Tehran Branch, Islamic Azad University, Tehran, Iran
[d] Mechanical Engineering Department, K.N. Toosi University of Technology, Tehran, Iran

ARTICLE INFO

Article history:
Received 1 August 2011
Received in revised form 22 October 2011
Accepted 1 November 2011

Keywords:
Hybrid system
Desiccant wheel
Solar collector surface
Evaporative cooler
Solar irradiance

ABSTRACT

This work presents the optimization of a solar collector surface in solar desiccant wheel cycle which for cooling process with typical configuration naming desiccant wheel, heat exchanger and water spray evaporative cooler. In this cooling cycle the thermal solar energy has used to heat the regeneration air of desiccant wheel cycle.

The optimum solar collector surface has determined by taking into account of design parameters such as velocity of air, wheel speed, thickness of the desiccant and hydraulic diameter of the desiccant wheel and also operating conditions such as outside temperature, outside relative humidity, regeneration air temperature and total solar irradiance. For this purpose, effect of desiccant wheel parameters has investigated on solar collector surface. After that, optimum design parameters and minimum solar collector surface has calculated.

In this cooling process, a mathematical model has used that shows physical properties of air.

The calculated values for design condition show that necessary solar collector surface has decreased about 45% in comparison of an empirical model in equal operating conditions.

The results of this study show that necessary solar collector surface will be decreased by increasing inlet air temperature, inlet air humidity ratio and solar irradiance and will be increased by increasing the regeneration air temperature.

1. Introduction

The desiccants are natural or synthetic substances capable of absorbing or adsorbing water vapor due to the difference of water vapor pressure between the surrounding air and the desiccant surface. They exist in both liquid and solid forms.

Desiccant cooling processes can be driven with a low temperature heat around 60 °C obtained from lower level energy such as waste heat or solar heat instead of electricity. This system usually consists of a dehumidifier, sensible heat exchanger and evaporative water spray coolers. It is a promising technology that can be developed as a candidate for non-front and less electricity cooling from viewpoints of global environment and energy sources [1].

The effect of airflow rate and solar radiation intensity on the system regeneration and absorption processes are studied by Kabeel [2]. The obtained results show that the system is highly effective in the regeneration process.

Ahmed et al. [3] used a desiccant cooling system model to investigate the unglazed transpired solar collector for regeneration of the desiccant. They also compared its performance with an ordinary flat plate collector.

Nelson et al. [4] studied the feasibility of two configuration open cycle air conditioning systems using solid desiccants and solar energy as regeneration energy. The two configurations are the ventilation mode, in which ambient air is continually introduced into the room; and the re-circulation mode, in which room air is re-circulated. Vazirifard and Saidi [5] studied the importance of desorption and harm of disusing of it in Iran.

Ahmed and Kattab [6] developed numerical model to simulate the heat and mass transfer for the adsorption and regeneration processes. Also, a prototype of a solar desiccant wheel was constructed and tested under the actual climatic conditions. Saidi [7] studied on effect of desiccant wheel, heat exchanger and cooling coil on decreasing the wet bulb temperature of entering air to cooling tower and decreasing the outlet cold water temperature.

Several researchers have studied the desiccant wheels and hybrid desiccant wheel systems; however there is no specific survey on the optimal collector surfaces. Therefore, the main objective of this research is optimizing the solar collector surface of the hybrid system for cooling processes with typical configuration includes desiccant wheel, heat exchanger and water spray evaporative cooler and also a solar air heater.

* Corresponding author. Tel.: +98 912 141 3768; fax: +98 2188606537.
E-mail address: mona.hatami1@yahoo.com (Z. Hatami).

0378-7788/$ – see front matter
doi:10.1016/j.enbuild.2011.11.002

Nomenclature

A	necessary solar collector surface, m^2
d_t	thickness of the desiccant, mm
D_h	hydraulic diameter of the desiccant wheel channels, mm
f, g	desiccant wheel parameters functions
h	enthalpy, kJ/kg
m_p	process air mass flow, kg/h
m_r	regeneration air mass flow rate, kg/h
N	wheel speed, RPH
P	ambient pressure, Pa
P_v	partial pressure of water vapor, Pa
P_{vs}	saturation pressure of water vapor at T_{db}, Pa
P_w	saturation pressure of water vapor at T_{wb}, Pa
R_a	gas constant of air, J/kg K
T_{amb}	ambient temperature, °C
T_{in}	inlet process air temperature to desiccant wheel, °C
T_{out}	outlet process air temperature from desiccant wheel, °C
T_{db}	dry bulb temperature, °C
T_{dp}	dew point temperature, °C
T_{reg}	regeneration temperature, °C
T_{wb}	wet bulb temperature, °C
ω	ambient humidity ratio, kg/kg
ω_i	humidity ratio of process and regeneration air, kg/kg
ω_{in}	inlet humidity ratio to desiccant wheel, kg/kg
ω_{out}	inlet humidity ratio to desiccant wheel, kg/kg
Q_{abs}	absorbed heat by collector, kJ/h
Q_{reg}	necessary heat for heating regeneration air, kJ/h
I_t	total solar irradiance, kJ/m^2
U	velocity of air, m/s

Greek letters

ε	desiccant wheel efficiency
ε_h	heat exchanger efficiency
υ	specific volume, m^3/kg
η	collector efficiency
φ	relative humidity

2. Problem description

In this study, a system will be evaluated consists of desiccant wheel, sensible heat exchanger, water spray evaporative cooler and also air heater. Necessary surface of solar collector will be calculated for various ambient conditions by selecting the desiccant wheel design parameters in optimum quantities.

Desiccant cooling systems can dehumidify inlet air stream by forcing it through a desiccant material and drying the air to achieve desired indoor temperature. To make the system working continually, adsorbed water in desiccant wheel must be driven out of the desiccant material in regeneration process so it can adsorb water vapor in the next cycle [8]. This process can be done by heating the desiccant material to its regeneration temperature about 60–90 °C, which was used by Ahmed [6].

The schematic of the present system is illustrated in Fig. 1. The aim of this system is changing the ambient air from point 1 to make supply air with a temperature about 25 °C. Solar energy will be used in this system because of using the solar air heater.

3. Governing equations

The most important section of present study is calculating outlet air conditions from desiccant wheel. In this section, temperature

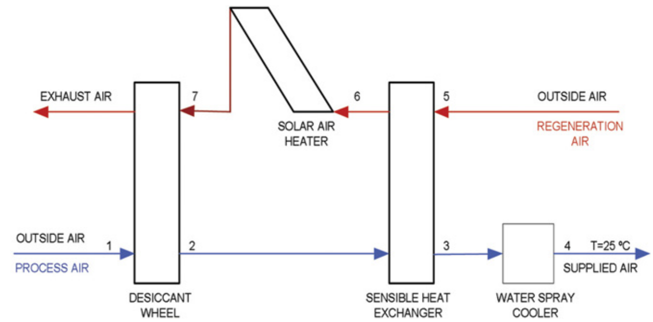

Fig. 1. Schematic of process air and regeneration air streams through hybrid system.

and humidity of outlet air from desiccant wheel will be calculated by using a mathematical model. This model is achieved by solving heat and mass transfer equations as follows [9]:

$$\omega_{out} = \omega_{in} - \varepsilon\omega_{in} \tag{1}$$

$$\varepsilon = f_1(N)f_2(T_{in})f_3(d_t)f_4(T_{reg})f_5(\omega_{in})f_6(D_h)f_7(U) \tag{2}$$

$$T_{out} = g_1(N)g_2(T_{in})g_3(d_t)g_4(T_{reg})g_5(\omega_{in})g_6(D_h)g_7(U) \tag{3}$$

In Eqs. (2) and (3), and g are functions of desiccant wheel parameters that are as follows: f

$$\begin{aligned} f_1(N) &= -0.0001N^2 + 0.0042N + 0.4474 \\ f_2(T_{in}) &= -0.0001T_{in}^2 - 0.0031T_{in} + 0.8353 \\ f_3(d_t) &= -21.67d_t + 6.93d_t + 1.34 \\ f_4(T_{reg}) &= -0.0001T_{reg}^2 + 0.0355T_{reg} - 0.4924 \\ f_5(\omega_{in}) &= 592.77\omega_{in}^2 - 41.23\omega_{in} + 1.283 \end{aligned} \tag{4}$$

$$\begin{aligned} g_1(N) &= -0.0002N^2 + 0.0112N + 0.4201 \\ g_2(T_{in}) &= -0.0001T_{in} + 0.0275T_{in} + 0.7993 \\ g_3(d_t) &= -18.79d_t + 7.92d_t + 1.75 \\ g_4(T_{reg}) &= -0.0004T_{reg}^2 + 0.1255T_{reg} + 0.6757 \\ g_5(\omega_{in}) &= 594.48\omega_{in}^2 + 26.76\omega_{in} + 3.79 \end{aligned} \tag{5}$$

By using Eqs. (1)–(5), outlet temperature and humidity from desiccant wheel, (point 2) will be calculated. After that effect of heat exchanger on the outlet air of desiccant wheel, must be calculated (point 3). For this purpose a heat exchanger with effectiveness equal to 90% can be used. The effectiveness of heat exchanger can be written as:

$$\varepsilon_h = \frac{T_{db2} - T_{db3}}{T_{db2} - T_{db5}} \tag{6}$$

After cooling the process air by passing it through heat exchanger, process air should pass through water spray evaporative cooler, in order to have supply air with a temperature about 25 °C. As it is shown in Fig. 2, we suppose enthalpy remains constant in this process, so $h_3 = h_4$. h_3 can be calculated from model that has been proposed by AL-Nimr [10]. In this study, the mentioned model will be used and psychrometric conditions of air will be investigated by means of trial and error method. This model is given by:

$$\nu = \frac{R_a(T_{db} + 273.15)}{P - P_v} \tag{7}$$

$$w = 0.622\frac{P_v}{P - P_v} \tag{8}$$

$$\phi = \frac{\omega(P - P_v)}{0.622P_{vs}} \tag{9}$$

$$P_v = P_\omega - \frac{(P - P_\omega)(T_{db} - T_{\omega b})}{1532 - 1.3(T_{\omega b} + 273.15)} \tag{10}$$