Analysis of Heat and Mass Exchange Performance of Enthalpy Recovery Wheel

Hong Fan and Liu Chen

Abstract Enthalpy recovery wheel is a high-efficiency heat recovery device with fresh air and return air, which can recycle both the obvious and latent heat from return air at the same time, and reduced energy consumption for handling fresh air. In this paper, a mathematical model of enthalpy recovery wheel was established, taking into account the air side and adsorbent side. The heat and mass transfer equation of the enthalpy recovery wheel through the coupled heat and mass transfer equations on the air side and the adsorbent side was established. The COMSOL Multiphysics which multiphysics coupling software is used to simulate the effect of fresh air inlet temperature, air inlet humidity, and face velocity on the enthalpy recovery wheel under summer conditions, revealing the heat and mass exchange performance of the enthalpy recovery wheel. This provides a reference for the optimal operation of the enthalpy recovery wheel.

Keywords Enthalpy recovery wheel · Fresh air · Heat recovery efficiency · Porous media

Nomenclature

c	Water vapor concentration, g/m^3
c_p	Constant pressure specific heat capacity, J/(kg k)
c_l	Liquid water concentration on the side of the adsorbent, g/m^3
C_{pa}	Specific heat of air, J/(kg K)
C_{cp}	Adsorption, liquid phase and gas combined with specific heat, J/(kg K)
d_z	Adsorbent layer thickness, m
d_e	Hydraulic diameter, m
G_p	Exhaust volume, kg/s

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h Heat transfer coefficient, W/(m^2 K)
K Mass transfer rate, 1/s
K_a Air–steam diffusion coefficient, m^2/s
K_{dv} Air–Steam effective diffusion coefficient of gas phase, m^2/s
K_{dl} Liquid water diffusion coefficient on the side of the adsorbent, m^2/s
M Dehumidification (parsing) volume, g/(m^3 s)
q_{st} Heat of sorption, J/kg
Q_t On apparent heat exchange, kw
Q_d Latent heat exchange, kw
W Extent of adsorption kg adsorbate/kg adsorbent
λ_{cp} Adsorbent, liquid phase and gas phase combined with thermal
t Time, s
T Temperature, K
u Face velocity, m/s
x Axial coordinate, m
y Axial coordinate, m
Y Moisture content, g/kg
T Temperature, K

Greek letters

γ Latent heat of vaporization, kJ/kg
λ Thermal Conductivity, W/(m K)
ρ_{da} Gas phase density on the adsorbent side, kg/m^3
ρ Density, kg/m^3
ρ_{cp} Adsorbent, liquid phase and gas phase bonding density, kg/m^3

Subscripts

a Air
d Adsorbent
f Fresh air
r Return air
0 Initial state
1 Import
2 Export

1 Introduction

Desiccant wheels have two major applications: air dehumidification [1, 2] and enthalpy recovery [3, 4]. For dehumidification wheels, process air is dried after it flows through the wheel, which rotates constantly between the process air and a hot regenerative air stream. However, enthalpy recovery wheels are used to recover energy by transferring sensible and latent heat between supply air and exhaust air. Due to different operating conditions, heat and moisture transfer behaves quite differently in the wheels [5]. The enthalpy recovery wheel handles a large volume of air, flexible in layout, high heat recovery efficiency, and easy to clean. It is widely used as a heat recovery component in large air handling units. Enthalpy recovery wheel uses the wheel as the only heat exchange core to maintain 10–25 r/min speed rotation, and its performance determines the efficiency of the entire heat recovery system [6].

Pan and Gang [7] analyzed the working characteristics and energy-saving effect of the enthalpy recovery wheel. Nóbrega and Brum [8] established a mathematical model for the enthalpy wheel, an effectiveness number of thermal units (NTU) analysis is carried out. La [9] compared the effects of the changes of wheel thickness, rotation speed, and face velocity on the enthalpy recovery wheel efficiency when silica gel and lithium chloride were used as moisture absorbents. Horton [10] established a one-dimensional transient heat and mass transfer model and analyzed the performance of enthalpy recovery wheels both with and without purge air.

In this paper, the physical model of the enthalpy recovery wheel was established, taking into account the air side and adsorbent side. A coupled heat and mass transfer equation for the air and adsorbent side was established. The porous media saturated heat and mass transfer model is applied to the adsorbent side. Numerical simulation was carried out by using COMSOL Multiphysics, investigated the effect of the temperature of process air, humidity of process air, and face velocity on the performance of enthalpy recovery wheel under typical summer conditions.

2 Model of Enthalpy Recovery Wheel

Simulation study on the enthalpy recovery wheel with silica gel as adsorption material and rotational speed as 10 r/min, and the shape of the honeycomb channel is sinusoidal. The enthalpy recovery wheel is composed of the air and the adsorbent side, in which the adsorbent side is a porous medium composed of solid skeleton and pores, and the pores contain liquid water and gaseous steam. Silica gel remains solid during adsorption and desorption, and the adsorption process is generally physical adsorption [11].

2.1 Teat and Mass Transfer Model

For the air side and the adsorbent side of the enthalpy recovery wheel, establish the two-dimensional geometric model shown in Fig. 1 and assume that:

(1) The diffusivity of steam and air are assumed to be constant.
(2) The inlet air conditions are uniform in space.

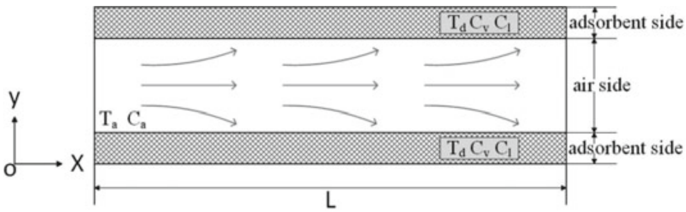


Fig. 1 Physical model of single-channel for enthalpy recovery wheel

(3) The adsorbent material is isotropic.
(4) All honeycombed channels in the wheel are identical and evenly distributed over the entire wheel.
(5) The analytical heat of silica gel is approximately equal to the adsorption heat.
(6) The adsorption potential energy of the pore surface of the adsorption material is negligible.
(7) The physical parameters of steam and liquid water are constant.
(8) The heat dissipation of the wheel shell is negligible.
(9) The effect of centrifugal force is negligible

Based on the above assumptions, the mathematical model of enthalpy recovery wheel is established:

(1) Energy balance differential equation on the air side:

$$\frac{\partial(\rho_a C_{pa} T_a)}{\partial t} - \frac{\partial}{\partial x}\left(\lambda_a \frac{\partial T_a}{\partial x}\right) - \frac{\partial}{\partial y}\left(\lambda_a \frac{\partial T_a}{\partial y}\right) = \frac{1}{d_e} h(T_d - T_a) \quad (1)$$

(2) Mass balance differential equation on the air side:

$$\frac{\partial c_a}{\partial t} - \frac{\partial}{\partial x}\left(K_a \frac{\partial c_a}{\partial x}\right) - \frac{\partial}{\partial y}\left(K_a \frac{\partial c_a}{\partial y}\right) = K(c_v - c_a) \quad (2)$$

(3) Energy balance differential equation on the adsorbent side:

$$\frac{\partial(\rho_{cp} C_{cp} T_d)}{\partial t} - \frac{\partial}{\partial x}\left(\lambda_{cp} \frac{\partial T_d}{\partial x}\right) - \frac{\partial}{\partial y}\left(\lambda_{cp} \frac{\partial T_d}{\partial y}\right) = \frac{1}{d_z} h(T_a - T_d) + Mq_{st} \quad (3)$$

(4) Gas phase mass balance differential equation on the adsorbent side:

$$\frac{\partial c_v}{\partial t} - \frac{\partial}{\partial x}\left(K_{dv} \frac{\partial c_v}{\partial x}\right) - \frac{\partial}{\partial y}\left(K_{dv} \frac{\partial c_v}{\partial y}\right) = K(c_v - c_a) + M \quad (4)$$

(5) Liquid phase mass balance differential equation on the adsorbent side:

$$\frac{\partial c_l}{\partial t} - \frac{\partial}{\partial x}\left(K_{dl}\frac{\partial c_l}{\partial x}\right) - \frac{\partial}{\partial y}\left(K_{dl}\frac{\partial c_l}{\partial y}\right) = M \tag{5}$$

The initial conditions for the adsorbent and air are:

$$\begin{cases} W = W_0 \\ T_d = T_{do} \end{cases} \tag{6}$$

$$\begin{cases} T_a = T_{a0} \\ Y_a = Y_{a0} \end{cases} \tag{7}$$

The temperature and humidity boundary conditions for the air are:

$$T_a = \begin{cases} T_f \\ T_r \end{cases} \tag{8}$$

$$Y_a = \begin{cases} Y_f \\ Y_r \end{cases} \tag{9}$$

The above governing Eqs. (1)–(5), initial conditions (6)–(7) and boundary conditions (8)–(9), constitute a complete mathematical model of the enthalpy recovery wheel.

2.2 *Performance Indexes*

(1) Apparent heat exchange:

$$Q_t = G_p c_p (T_1 - T_2) \tag{10}$$

(2) Latent heat exchange:

$$Q_d = G_p \gamma (Y_1 - Y_2) \tag{11}$$

3 Model Parameters

The predefined interface [12, 13] of COMSOL Multiphysics is used to simulate. The standard inlet parameters and parameter changes during simulation are shown in Table 1, and the defined parameters in the model are shown in Table 2.

Table 1 Standard inlet parameters and parameter changes in simulation

Inlet parameter	Standard condition	Variation range
Fresh air temperature °C	35	28–37
Fresh air relative humidity %	50	40–70
Face velocity m/s	2	1–3
Return air temperature °C	24	–
Return air relative humidity %	50	–

4 Results and Discussion

When the return air parameters are constant, the face velocity, inlet temperature, and relative humidity of fresh air are, respectively, changed to simulate the change of the performance of the wheel in one turn (6 s).

(1) Face velocity

When other conditions are under standard conditions, the face velocity is changed from 1 to 3 m/s, simulating the effect of face velocity on the performance of the enthalpy recovery wheel.

Figure 2a, b, c shows the single-channel temperature distribution of fresh air when the face velocity increases from 1 to 3 m/s, and it can be observed that the temperature range near the wall surface of the micro-channel is large. With the increase of face velocity, only the temperature near the wall of the micro-channel changes and the amplitude is small. The analysis shows that the increase of the face velocity shortens the time that the fresh air stays in the micro-channel and shortens the heat transfer time between fresh air and adsorbent.

As shown in Figs. 3 and 4, Q_t and Q_d decrease with an increasing face velocity. The analysis shows that the increase of face velocity makes the time of air stay in the wheel shorter, and the heat and moisture exchange is not sufficient. In addition, the hygroscopic ability of adsorbent decreases with the increase in face velocity, which deteriorates the performance of the adsorbent and weakens the hygroscopic ability to fresh air. Therefore, Q_t and Q_d are gradually reduced. In practical application, under the condition of ensuring airflow, a larger wheel should be chosen to increase the airflow channel area and reduce the face velocity.

(2) Fresh air temperature

Temperature of fresh air increases from 28 to 37 °C when other operating conditions are in standard conditions, simulating the influence of fresh air temperature on the performance of the enthalpy recovery wheel.

As shown in Fig. 5, Q_t increases with temperature of fresh air, and the rise of fresh air temperature increases the temperature difference between fresh air and adsorbent and strengthens the heat transfer of the fresh air through the wheel, and so, Q_t increases. As shown in Fig. 6, the increase of air temperature causes a slight reduction of Q_d, The surface temperature of absorbent rises with the fresh air

Table 2 Model parameters and constants

Description	Value
Channel length m	0.15
Thickness of the wheel m	0.05
Adsorbent porosity	0.7
Adsorbent permeability m^2	1.28×10^{-14}
Adsorbent thermal conductivity w/(m•k)	0.175
Adsorbent heat capacity J/(kg•k)	921
Adsorbent density kg/m^3	1201
Ambient pressure pa	101,325

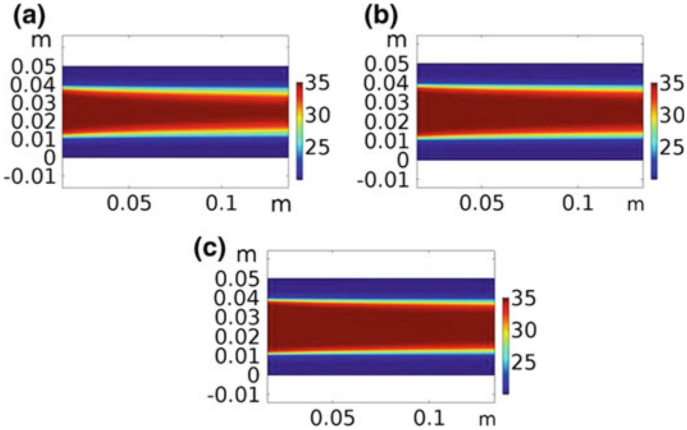


Fig. 2 Single-channel air temperature distribution **a** u = 1 m/s; **b** u = 2 m/s; **c** u = 3 m/s

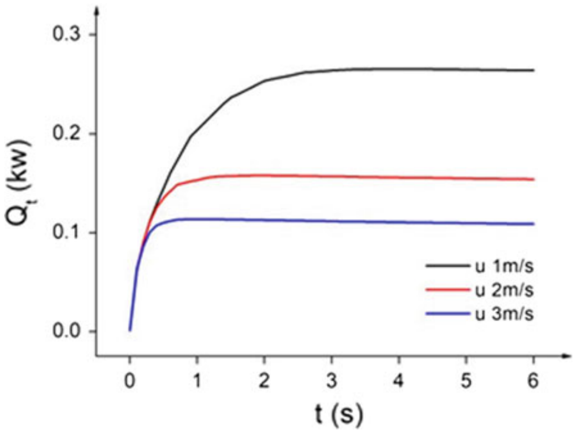


Fig. 3 Effect of face velocity on apparent heat exchange

Fig. 4 Effect of face velocity on latent heat exchange

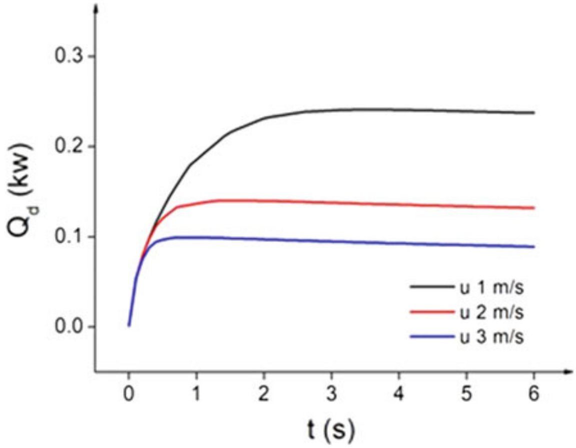
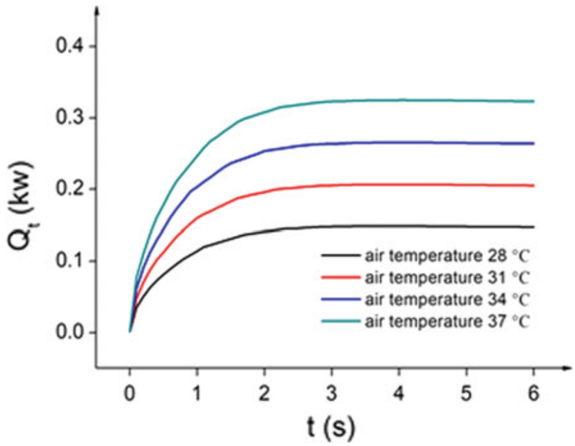


Fig. 5 Effect of fresh air temperature on apparent heat exchange



temperature, which causes the water vapor partial pressure difference between the fresh air and the adsorbent to decrease, and the mass transfer driving potential difference is reduced, which weakens the moisture transfer of the fresh air through the wheel, so Q_d decreases.

(3) Fresh air relative humidity

Relative humidity is increased by 65% from 50% (the corresponding moisture content increased from 17.7 to 23.21 g/kg) when other operating conditions are in standard conditions, simulating the influence of fresh air relative humidity on the performance of the enthalpy recovery wheel.

As shown in Fig. 7, Q_t remains basically unchanged with the increase of the relative humidity, and the fresh air humidity has no influence on the sensible heat exchange. As shown in Fig. 8, Q_d increase with inlet humidity, When fresh air is

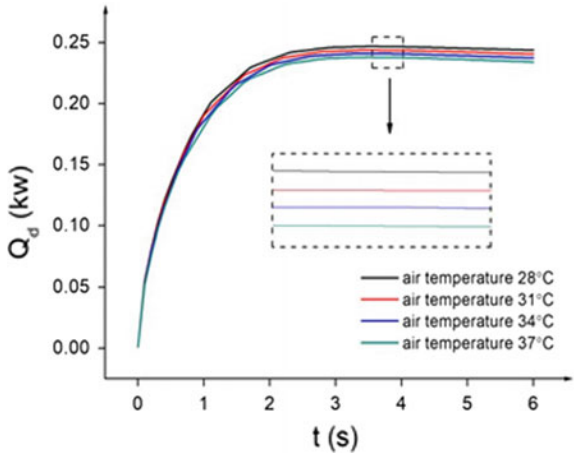


Fig. 6 Effect of fresh air temperature on latent heat exchange

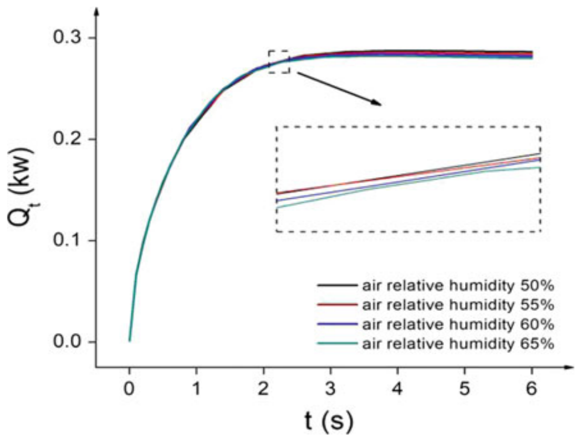


Fig. 7 Effect of relative humidity on apparent heat exchange

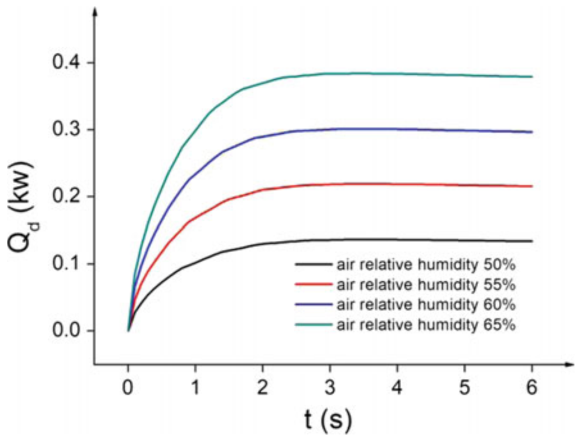


Fig. 8 Effect of relative humidity on latent heat exchange

more humid, a higher difference of vapor partial pressure between fresh air and the surface of the adsorbent side, which enhances the moisture transfer, so Q_d increases.

5 Conclusion

In this paper, the heat and mass transfer equations are established for the air side and the adsorbent side of the single channel of the enthalpy recovery wheel, and the numerical simulation is solved using COMSOL Multiphysics. The temperature distribution characteristics of the enthalpy recovery wheel are studied, and the influence of face velocity, fresh air temperature, and fresh air humidity on the performance parameters of the enthalpy recovery wheel is analyzed. By simulation results, it obtains the following conclusions: (1) The temperature distribution on the fresh air side of the enthalpy recovery wheel indicates that the temperature range near the wall is large, and the heat exchange mainly occurs near the wall surface. (2) Q_t and Q_d gradually decrease with an increasing face velocity. Therefore, in practical applications, the face velocity should not be too large in the case of ensuring airflow. (3) Q_t gradually increases and Q_d decreases slightly with the increase of fresh air temperature. (4) remains basically unchanged and the Q_d increases with the increase of relative humidity of fresh air.

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