A Mathematical Model for Predicting the Performance of the Solar Energy assisted Hybrid Air Conditioning System (SEAHACS) with One-Rotor Six-Stage Rotary Desiccant Cooling System.

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A mathematical model for predicting the performance of the solar energy assisted hybrid air conditioning system (SEAHACS) with one-rotor six-stage rotary desiccant cooling system

النموذج الرياضي للتنبؤ باداء نظام تكيف الهواء الثنائي المعضد بالطاقة الشمسية ذو عجلة الامتصاص الواحدة مجزءة الي ستة مراحل

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الملخص:

يقدم البحث نموذج رياضياً للتنبؤ باداء نظام تكيف الهواء الثنائي المعضد بالطاقة الشمسية . ذو عجلة الامتصاص المستخدمة في هذه الدراسة عبارة عن قرص يشبه خلايا النحل مشبع بالسيلكا جل، عجلة الامتصاص الواحدة مجزءة الي ستة اجزاء، منها مرحلتين ازالة رطوبة من هواء التغذية، و مرحلتين تبريد بيني، و مرحلتين اعادة تنشيط للمواد الماصة للرطوبة. حيث يوجد ثلاث مسارات للهواء خلال هذا النظام. حيث ان النموذج الرياضي تمت مطابقتة بنتائج عملية. وبة تمت دراسة تاثير متغيرات التصميم و التشغيل و تتضمن هذه المتغيرات ،درجة حرارة الدخول لهواء اعادة التنشيط للمواد الماصة تتغير في المدي من 65 °م – 140 °م، نسب المساحات ما بين هواء التغذية المراد ازالة الرطوبة منة الي الهواء المستخدم في اعادة التنشيط للمواد الماصة تتغير في المدي من 1-3,57، سرعة الدخول لهواء اعادة التنشيط تتغير في المدي من 1,5 م/ث – 5,5 م/ث هذا يتحقق بالكامل عند سرعة دوران مختلفة لعجلة الامتصاص تتغير في المدي من 6 – 20 لفة/ساعة. افضل اداء لهذة المتغيرات يتحقق من خلال معرفة سعة الرطوبة المزالة، نسبة سعة الرطوبة المزالة، معامل الاداء لازالة الرطوبة، معامل الاداء الحراري. وفي النهاية يتم مناقشة تاثير هذة المتغيرات علي افضل سرعة دوران.

Abstract.

A mathematical model for predicting the performance of solar energy assisted hybrid air conditioning system (SEAHACS) is presented. The honeycombed silica gel desiccant wheel was used in this study, one-rotor six-stage rotary desiccant cooling system, in which two-stage dehumidification process, two-stage pre-cooling process and two-stage regeneration process are realized by only one wheel. Three air streams are involved in the present system. The mathematical model has been validated with the experimental data. As the key operating and design parameters,

the range of regeneration air inlet temperature from 65 – 140 °C, area ratio of process air to regeneration air change from 1-3.57, regeneration air inlet velocity from 1.5 – 5.5 m/s have been examined for a range of rotation speed from 6 -20 rev/h. the optimization of this parameters is conducted based on the moisture removal capacity D, relative moisture removal capacity, dehumidification coefficient of performance and thermal coefficient of performance. At last, the influences of these main parameters on optimal rotation speed are discussed.

Nomenclature				
a	Channel height (m)	P	Perimeter of channel (m)
A	Cross section area (m^2)	P_s	The pressure of saturated water vapor (pa)
b	Channel width (m)	Q	Heat energy (J)
c	Layer thickness of desiccant material (m)	T	Temperature (K)
		Sh	Sherwood number
C_p	Constant pressure specific heat (J/kg.K)	u	Velocity (m/s)
COP$_{th}$	Thermal coefficient of performance	W	Water content of desiccant material (kg/kg)
D	Moisture removal capacity	Y	Humidity ratio (kg/kg)
DCOP	Dehumidification coefficient of performance	Φ	Relative humidity ratio
		ε	Effectiveness
D_h	Hydraulic diameter (m)	ρ	density (kg/m^3)
f_m	Mass friction of desiccant in the wheel.	z	Axial direction
		t	Time (s)
h	Convective heat transfer coefficient (W/m^2.K)	**Subscripts**	
		a	Air
H_{sor}	Heat of adsorption (J/kg)	in	Inlet
K_m	Mass transfer coefficient (kg/m^2.s)	l	Liquid
L	Wheel thickness (m)	out	Outlet
N_{uH}	Nusselt number at constant heat flux	p	Process
		r	Regeneration
N_{uT}	Nusselt number at constant heat temperature	v	Vapor
		w	Desiccant
N_u	Nusselt number	c	Cooling
$\dot{m}$	Mass flow rate (kg/s)		

1. Introduction

The traditional resource such as oil and gas cannot meet the rising energy demand. This has led to increasing interest in alternate-energy research such as solar energy, geothermal energy and world is facing a large-scale and potentially devastating global energy crisis. People have already realized wind energy. Besides, increasing occupant comfort demands are leading to growing requirement for air conditioning, whereas deteriorating global energy and environment crisis are starving for energy saving and environmental protection. Thus, how to provide thermal comfort for occupants with low grade thermal energy has attracted more and more attention and extensive types of technologies have been developed [1]. Among these technologies, rotary desiccant cooling, which is advantageous in controlling humidity independently, using low grade thermal energy and being free from CFCs, has been considered as a potential choice. Over a number of years, investigations on rotary desiccant cooling have been widely carried out on the basis of mathematical simulation, thermodynamic analysis, experimental investigation and practical application [2].

Due to the process air is usually heated by released adsorption of heat in dehumidification process and the standalone desiccant system could not handle the air to qualified conditions, auxiliary cooler must be adopted and it is right the main difference between systems.

Various aspects of the desiccant cooling systems have been intensively investigated by many researchers. The desiccant material studies [3-7], rotary desiccant dehumidification and air conditioning processes [8-15], performance modeling of desiccant wheels [16-20].

In this study, a solar energy assisted hybrid air conditioning system (SEAHACS) with a one-rotor six-stage rotary desiccant cooling system, in which two-stage dehumidification process, two-stage pre-cooling process and two-stage regeneration process are realized by only one wheel, is proposed and investigated. A numerical model is used to study and discuss the moisture removal capacity and relative moisture removal capacity, dehumidification coefficient of performance, and thermal coefficient of performance considering the operating and design parameters such as area ratio of process air to regeneration air, regeneration air inlet temperature, and inlet velocity of regeneration air.

2. Problem Formulation

Figure 1 illustrates the schematic diagram of the proposed system. The system consists of a desiccant wheel, two cross-flow heat exchangers, two air heaters and three direct evaporative cooler.

2.1. System Operation

There exists one process air path (1-2-3-4-5-6), two regeneration air paths (7-8-9-10-11 and 7-8-12-13-14), and two cooling paths (15-16).

The process air cycle is as follows: Process air at state point1 passes through section 1 of the desiccant wheel, where its moisture is removed and temperature is increased due to the adsorption heat. Then this hot dry air is sensibly cooled from state point 2 to 3 in a cross-flow heat exchanger. Afterwards, the process air passes through section 2 of the desiccant wheel, where its moisture is further removed. And then in another cross-flow heat exchanger, the process air (state 4) is sensibly cooled with a dry air output at state point 5. At state point 5, the process air passes through a direct evaporative cooler; the air stream is humidified and cooled to state point 6.

The regeneration air cycle is as follows: The regeneration air is first cooled and humidified in a direct evaporative cooler from state point 7-8. Then it is divided into two groups, which work in parallel (state 8-9-10-11; state 8-12-13-14). These two air streams are sensibly preheated to state points 9 and 12 in the cross-flow heat exchangers. The parts of the warm air streams are passing through the air heaters to state points 10 and 13, and exit at state points 11 and 14.

The cooling air cycle is as flows: The cooling air from state point 15-16, is used to pre-cooling the desiccants wheel after the regeneration section and before the process section to increase the dehumidification performance of the desiccant wheel.
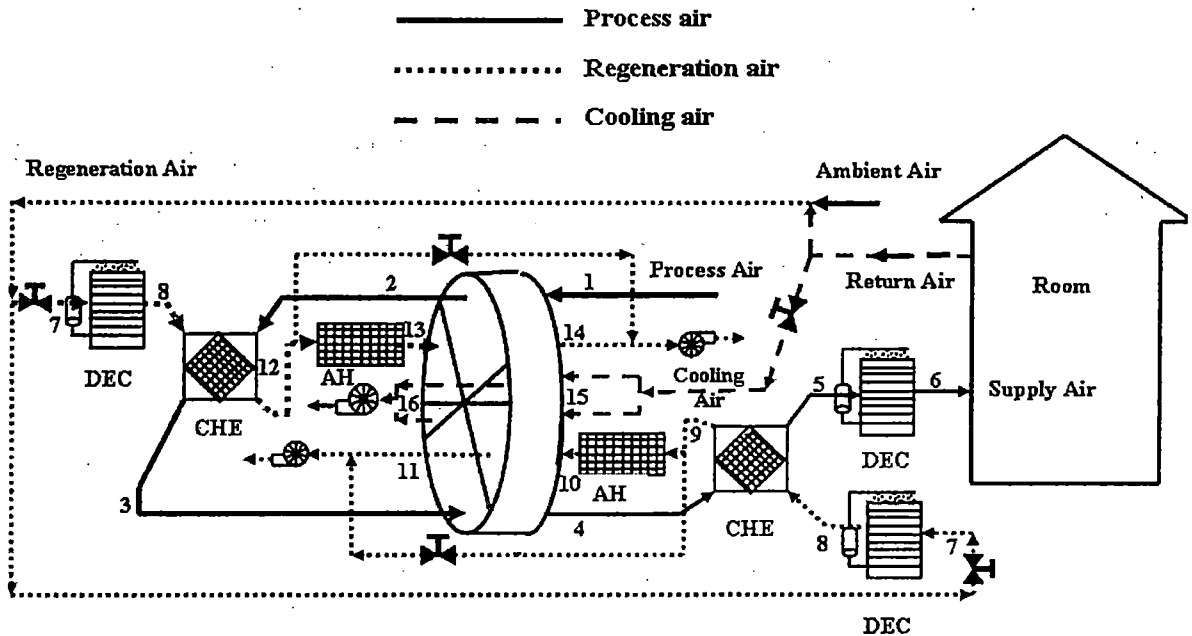
DW: Desiccant Wheel; DEC: Direct Evaporative Cooler; CHE: Cross- Flow Heat Exchanger;

AH: Air Heater

Fig.1 Schematic Diagram of Solar Energy Assisted Hybrid Air Conditioning System

3. Mathematical Model

3.1. Governing Equation

Figure 2 shows the schematics of a desiccant wheel. One honeycombed channel with a length of L and cross section area is shown in Fig.2 in this study, the unsteady one dimensional model (t,z) is chosen for the coupled heat and mass transfer process in the rotary desiccant wheel.
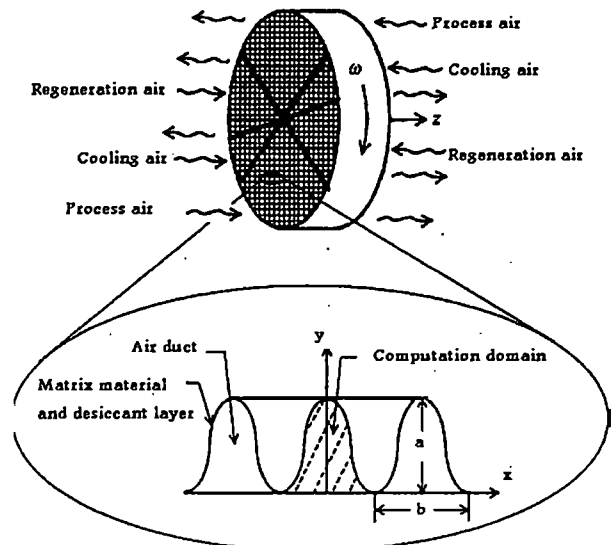

Fig. 2 The corrugated air duct and the computation domain.

The numerical analysis is based on the following assumption:

(1) The flow is one - dimensional.

(2) The axial heat conduction and the mass diffusion in the fluid are neglected.

(3) The axial water vapor and adsorbed water diffusion in the desiccant are neglected.

(4) There is no leakage of the fluid in the desiccant wheel.

(5) All ducts are impermeable and adiabatic.

(6) The thermodynamic properties are constant and uniform.

(7) The heat transfer coefficient between the air flow and desiccant wall is constant along the channel.

(8) The inlet conditions are uniform across the wheel surface, but they can vary with time.

(9) There is no heat or moisture storage in the wheel when it completes one rotation.

Based on the above assumptions, the energy and mass conservation equation can be obtained as follows [19, 21, 22].

Mass Conservation of Process Air:

$$\frac{\partial Y_a}{\partial Z} = \frac{K_m P_p}{u_a \rho_a A_p} (Y_w - Y_a) \qquad (1)$$

The term in the left hand side in equation (1) is the change of moisture content in control volume "sorbet influx by fluid flow". The term in the right hand side is the rate of moisture variation in the air caused by convective mass transfer between air and desiccant.

Conservation of water content for the absorbent:

$$\frac{\partial W}{\partial t} = \frac{K_m P_w}{\rho_w f_m A_p} (Y_a - Y_w) \qquad (2)$$

The term in the left hand side in equation (2) is the convective mass transfer between the air and desiccant. The term in the right hand side is the moisture storage term in the desiccant material.

Energy conservation for process air:

$$\left(C_{pa} + Y_a C_{pv}\right)\frac{\partial T_a}{\partial z} = \frac{h P_p}{u_a \rho_a A_p} (T_w - T_a) \qquad (3)$$

The term in the left hand side in equation (3) is the sum of the energy transferred by fluid flow and decreased by sorbet transfer to felt. The term in the right hand side is the conduction heat transfer to the felt.

Conservation of energy for the absorbent:

$$\left(C_{pw} + f_m W C_{pl}\right)\frac{\partial T_a}{\partial t} = \frac{h P_w}{\rho_w A_w} (T_a - T_w) + \frac{K_m H_{sor} P_w}{\rho_w A_w} (Y_a - Y_w) \qquad (4)$$

The term in the left hand side in equation (4) is the energy storage in desiccant material. The first term in the right hand side is the energy transfer by heat conduction, and the second term in the right hand side is the energy transfer by mass transfer.

3.2. Initial and boundary conditions:

The above equations are subject to the following boundary and initial conditions which are easily obtained considering periodic natural desiccant wheel.

For the process section	For the regeneration section
Initial condition	
$T_{ap}(t, 0.0) = T_{ap,in}$	Initial condition
$Y_{ap}(t, 0.0) = Y_{ap,in}$	$T_{ar}(t, 0.0) = T_{ar,in}$
Boundary codition	$Y_{ar}(t, 0.0) = Y_{ar,in}$
$T_{wp}(0.0, z) = T_{wc}(t_c, L-z)$	Boundary codition
$Y_{wp}(0.0, z) = Y_{wc}(t_c, L-z)$	$T_{wr}(0.0, z) = T_{wp}(t_p, L-z)$
	$Y_{wr}(0.0, z) = Y_{wp}(t_p, L-z)$

For the cooling section
Boundary codition
$T_{ac}(t, 0.0) = T_{ac,in}$
$Y_{ac}(t, 0.0) = Y_{ac,in}$
Boundary codition
$T_{wc}(0.0, z) = T_{wr}(t_r, L-z)$
$Y_{wc}(0.0, z) = Y_{wr}(t_r, L-z)$

3.3. Auxiliary conditions:

The governing equation Equations (1-4) have five unknowns T_a, T_w, Y_a, Y_w, and W. in order to solve this set of equations, i.e. to close the problem, it is necessary to relate the equilibrium composition Y_w to the water content W and the temperature of the adsorbent T_w. Here we employ silica gel as a desiccant material so the water content is covered by the following isotherm [19, 23, 24].

$$\emptyset_w = 0.0078 - 0.0576\,W + 24.2W^2 - 124W^3 + 204W^4 \quad (5)$$

Where $\emptyset_w$ is the equilibrium relative humidity over the desiccant with the water content W. the relationship between the humidity ratio and the relative humidity is expressed as.

$$Y_w = \frac{0.622\,\emptyset_w\,P_s}{P_{atm} - \emptyset_w\,P_s} \quad (6)$$

The pressure of saturated water vapor Ps was given by [18] as,

$$P_s = exp\left(23.196 - \frac{3816.44}{T_w - 46.13}\right) \quad (7)$$

and heat of adsorption H_{sor} was given by [18] as,

$$H_{sor} = \begin{cases} -12400\,W + 3500 & W \le 0.05 \quad \left(\frac{kJ}{kg\ water}\right) \\ -1400\,W + 2950 & W > 0.05 \quad \left(\frac{kJ}{kg\ water}\right) \end{cases} \quad (8)$$

The convective heat transfer coefficient and hydraulic diameter are calculated from the Nusselt number in the sinusoidal shipped channels [19].

Hydraulic diameter:

$$\frac{D_h}{b} = (1.0542 - 0.4660\,\alpha^* - 0.1180\,\alpha^{*2} + 0.1794\,\alpha^{*3} - 0.0436\,\alpha^{*4})\alpha^* \quad (9)$$

Nusselt number:

$$N_{uT} = 1.1791 \times (1 + 2.7701\,\alpha^* - 3.1901\,\alpha^{*2} - 1.9975\,\alpha^{*3} - 0.4966\,\alpha^{*4})$$

$$N_{uH} = 1.903 \times (1 + 0.4556\, \alpha^* +$$
$$1.2111\, \alpha^{*2} - 1.6805\, \alpha^{*3} + 0.7724\, \alpha^{*4} -$$
$$0.1228 \alpha^{*5})$$

$$N_u = \frac{N_{uT} + N_{uH}}{2} \qquad (10)$$

Where N_{uH} is the Nusselt number at constant heat flux, N_{uT} is the Nusselt number at constant heat temperature, $\alpha^* = \frac{a}{b}$, a=channel height and b= channel width.

Convective heat transfer coefficient h:

$$h = \frac{N_u\, K_a\, P_p}{4\, A_p} = \frac{N_u\, K_a}{D_h} \qquad (11)$$

Where K_a is the thermal conductivity W/m.K

Mass transfer coefficient K_m:

Mass transfer coefficient K_m [25] is given by,

$$K_m = \rho_a \frac{Sh\, D_a\, P_p}{4\, A_p} \cong$$

$$\rho_a \frac{Sh\, D_a}{D_h} \quad \frac{kg}{m^2.s}, \quad Sh \cong N_u \qquad (12)$$

Where D_a is the mass diffusion diffusivity of the water vapor in the air

$$D_a = 2.302 * 10^{-5} \frac{P_{atm}}{P_a} \left(\frac{T_a}{T_{atm}}\right)^{1.81}$$

$$(13)$$

$P_{atm} = 101.13 * 10^3$ Pa, $T_{atm} = 308$ K

4. Model of other elements:

4.1. Cross-flow heat exchanger:

The heat transfer characteristics of the two cross-flow heat exchangers are the same.

Their heat transfer process can be represented by [26; 27 as,

$$C_{pa,hot} \cdot \dot{m}_{hot} \left(T_{hot,in} - T_{hot,out}\right)$$
$$= C_{pa,cold} \cdot \dot{m}_{cold} \left(T_{cold,out} - T_{cold,in}\right)$$
$$(14)$$

Effectiveness of cross-flow heat exchanger:

$$\varepsilon_{CHE} =$$
$$\frac{\dot{m}_{hot} \cdot C_{pa,hot} \left(T_{hot,in} - T_{hot,out}\right)}{min(C_{pa,hot} \cdot \dot{m}_{hot} \, , \, C_{pa,cold} \cdot m_{cold}) \times (T_{hot,in} - T_{cold,in})}$$
$$(15)$$

Where hot; hot stream, cold; cold stream.

4.2. Direct evaporative cooler:

The heat and mass transfer of the direct evaporative cooler occurs between the falling film and the air flow through it. The evaporative cooler is divided into micro-segments along both the air and water flow directions. Additionally, since the basic heat and mass transfer principle of the direct evaporative cooler used for supply air, the regeneration air is of the same as the cross-flow evaporative cooler, it can be simulated with the model of the cross-flow evaporative cooler too. However, in view of the air in the direct evaporative cooler experiencing an throttling procedure, it can be simulated by the following simplified expressions with few influences on simulation accuracy [26, 27].

Effectiveness of direct evaporative cooler:

$$\varepsilon_{DEC} = \frac{(T_{in} - T_{out})}{(T_{in} - T_{wet})} \qquad (16)$$

$$h_{in}(T_{in}, Y_{in}) = h_{out}(T_{out}, Y_{out}) \qquad (17)$$

Where T_{wet} is the wet bulb temperature, h the enthalpy.

4.3. Air heater:

Based on the energy balance equation, the thermal energy used to drive the system is equal to regeneration heat consumed in the heaters:

$$Q_{reg} = \dot{m}_{a,hot} \, C_{pa,hot}(T_{a\,hot,out} - T_{a\,hot,in}) = \dot{m}_w \, C_{pw}(T_{wi} - T_{wo}) \qquad (18)$$

Where $\dot{m}_w$ is the mass flow rate of heating water, C_{pw} is the water specific heat, T_{wi} inlet temperature of heating water, and T_{wo} out temperature of heating water.

5. Solution method

The governing equation set is solved by a mixed numerical scheme. The two governing equations on the desiccant side (Eqs. 2 and 4) are solved using a fully-implicit finite difference scheme and the other two equations on the gas side (Eqs. 1 and 3) are solved using forward finite difference scheme. The space step is 0.002 m and the time step depends on the air inlet velocity and equal to

$$dt = \frac{Thickness \; of \; desiccant \; wheel \; (m)}{Inlet \; Velocity \; (^m/_s)}$$

6. Validation

Experimental results related to a one-rotor two –stage rotary desiccant cooling system [12], are adopted to validate the present model. Under a wide range of operating conditions (the regeneration air temperature changes from 50 °C to 90 °C, the rotation speed change from 4 r/h to 20 rev/h, the process air inlet temperature is constant at 35 °C, the process air inlet humidity ratio is constant at 14.3 g/kg), the predicted results in the terms of dehumidification coefficient of performance (DCOP) as well as moisture removal agree well with the experimental data as shown in Fig. 3. The discrepancy between the two results is less than 6.5%. Compared with the results of a model of compound desiccant wheel where the discrepancy between the experimental results and simulated data is about 8–9% [20], this is a large improvement.

7. Numerical Results and Discussion

The structural and operating parameters on the desiccant wheel vary a lot with different climate condition and the application areas. The model is used to investigate the influences of main different parameters on the system performance.
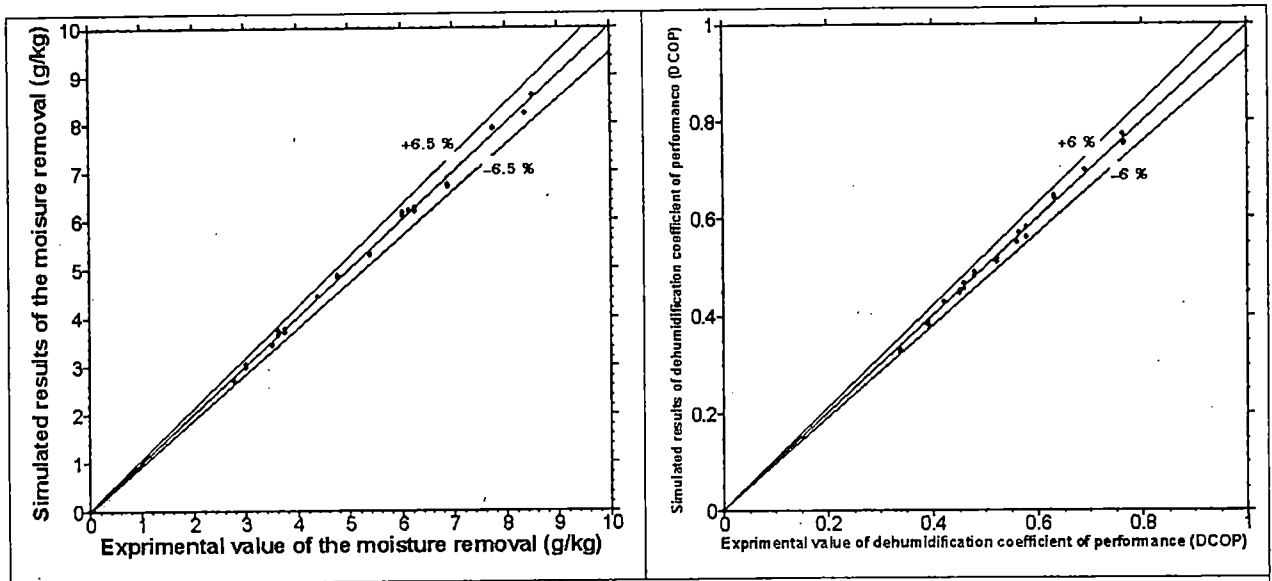
Fig.3 Comparison between the simulated and the experimental results [12].

7.1. Performance indices

The main function of a desiccant wheel is to remove the water vapor from a process air. Therefore, the first group of performance indices represents the dehumidification capacity of the desiccant wheel. Moisture removal capacity D is defined as the absolute dehumidification capacity of the desiccant wheel:

$$D = Y_{p,in} - Y_{p,out}$$

Where $Y_{p,in}$ and $Y_{p,out}$ are the humidity ratio of process air at inlet and outlet of desiccant wheel respectively.

Relative moisture removal capacity, show the ratio between moisture removal capacity D and inlet humidity ratio of process air $Y_{p,in}$. this value change between 0.0 and 1.

Relative moisture removal capacity increases with increasing moisture removal capacity.

Relative moisture removal capacity

$$= \frac{D}{Y_{p,in}} = \frac{Y_{p,in} - Y_{p,out}}{Y_{p,in}}$$

The dehumidification coefficient of performance DCOP, which reflects the dehumidification capacity and energy utilization performance at the same time is defined as the ratio of the latent heat which is contained in the adsorbed moisture and the regeneration heat used for regeneration air:

$$DCOP$$
$$= \frac{\dot{m}_{pa} \times D \times h_{fg}}{\{[\dot{m}_{ra} \times C_p \times (T_{10} - T_9)] + [\dot{m}_{ra} \times C_p \times (T_{13} - T_{12})]\}}$$

A higher DCOP indicates a better system performance because the energy input to the regeneration air is utilized in a better way or

less heat is being used to heat up the desiccant wheel.

Thermal coefficient of performance COP_{th} was used as another indicator to represent the overall performance and the energy efficiency of the desiccant cooling system. It was defined as the ratio between the cooling power Q_c and the regeneration heat Q_{reg}. The electrical power input of the fans was neglected as the thermal energy is the primary energy source.

$$COP_{th} = \frac{Q_c}{Q_{reg}}$$

$$= \frac{\dot{m}_{pa} \times C_p \times (T_1 - T_6)}{\{[\dot{m}_{ra} \times C_p \times (T_{10} - T_9)] + [\dot{m}_{ra} \times C_p \times (T_{13} - T_{12})]\}}$$

The constant thermodynamic properties and geometrical parameters of the desiccant wheel used in the simulation process are listed in the Table 1.

Table 1 Input data used in simulation

Channel shape	Sinusoidal
Height, a	1.75 mm
Width, b	3.5 mm
Wall thickness, c	0.15 mm
Rotary Length, L	0.1 m
Rotary diameter, D	0.2 m
Desiccant material	Silica gel
Mass friction of sorbent, f_m	0.7
Specific heat of the silica gel, C_{pw} (J/Kg.k)	921

Density of the silica gel, ρ_w (kg/m^3)	720
Air density, ρ_a (kg/m^3)	1.1614
Specific heat of the air, C_{pa} (J/Kg.k)	1005
Thermal conductivity of air, k_a (w/m.k)	0.0263
Specific heat of water vapor C_{pv} (J/Kg.k)	1872
Specific heat of water liquid C_{pl} (J/Kg.k)	4186

7.2. Effect of structural and operating parameters on system performance

The structure and operating data used in analysis are listed in the Table 2. One parameter is changed at a time to investigate its effect while all other parameters are fixed. There is no pressure loss calculation.

Table 2 Structural and operating parameters

Parameters	Parametric variations
Rotation speed	6 – 20 rev/h
Inlet temperature of process air, $T_{p,in}$	35 °C
Inlet humidity ratio of process air, $Y_{p,in}$	15 g/kg
Process air inlet velocity	2.5 m/s

Inlet   temperature   of 65 – 140 °C regeneration air, $T_{r,in}$

Inlet   humidity   ratio   of   15 g/kg regeneration air, $Y_{r,in}$

Regeneration air inlet velocity   1.5 – 5.5 m/s

Thickness   of   desiccant   0.15 mm material

Thickness of desiccant wheel,   0.1 m L

Area   ratio   of   process   to   1 – 3.57 regeneration air side, A_p/A_r

7.2.1. Effect of Regeneration Air Inlet Temperature on the Dehumidifying Performance and the Optimal Rotation Speed

In this section, effect of regeneration air inlet temperature on the dehumidifying performance and the optimal rotation speed is simulated. The process and regeneration humidity ratio are constant and equal to 15 g/kg and the process air inlet velocity and regeneration air inlet velocity are constant and equal to 2.5 m/s and process air inlet temperature is 35 °C and desiccant wheel thickness equal to 0.1 m.

Figure 4 show the impact of the regeneration air inlet temperature on

dehumidification performance for the regeneration air inlet temperature changes from 65 °C to 140 °C and all other operation parameters are held constant. It is found that: for increasing the inlet temperatures of the regeneration air from 65 °C to 140 °C, the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase increases from 1135 Pa to 1351 Pa, the moisture removal capacity D increase about 2.228 g/kg, relative moisture removal capacity increase: about 0.136, dehumidification coefficient of performance DCOP decrease about 0.517 and thermal coefficient of performance COP_{th} decrease about 1.24. Therefore, the moisture removal capacity increases with the regeneration air inlet temperature increase due to the desiccant being better regenerated and the dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th} decreases with increasing the regeneration air inlet temperature.

Figure 5 shows the effect of regeneration air inlet temperature and rotation speed on the difference in the vapor pressure between the inlet process air and air layer in equilibrium
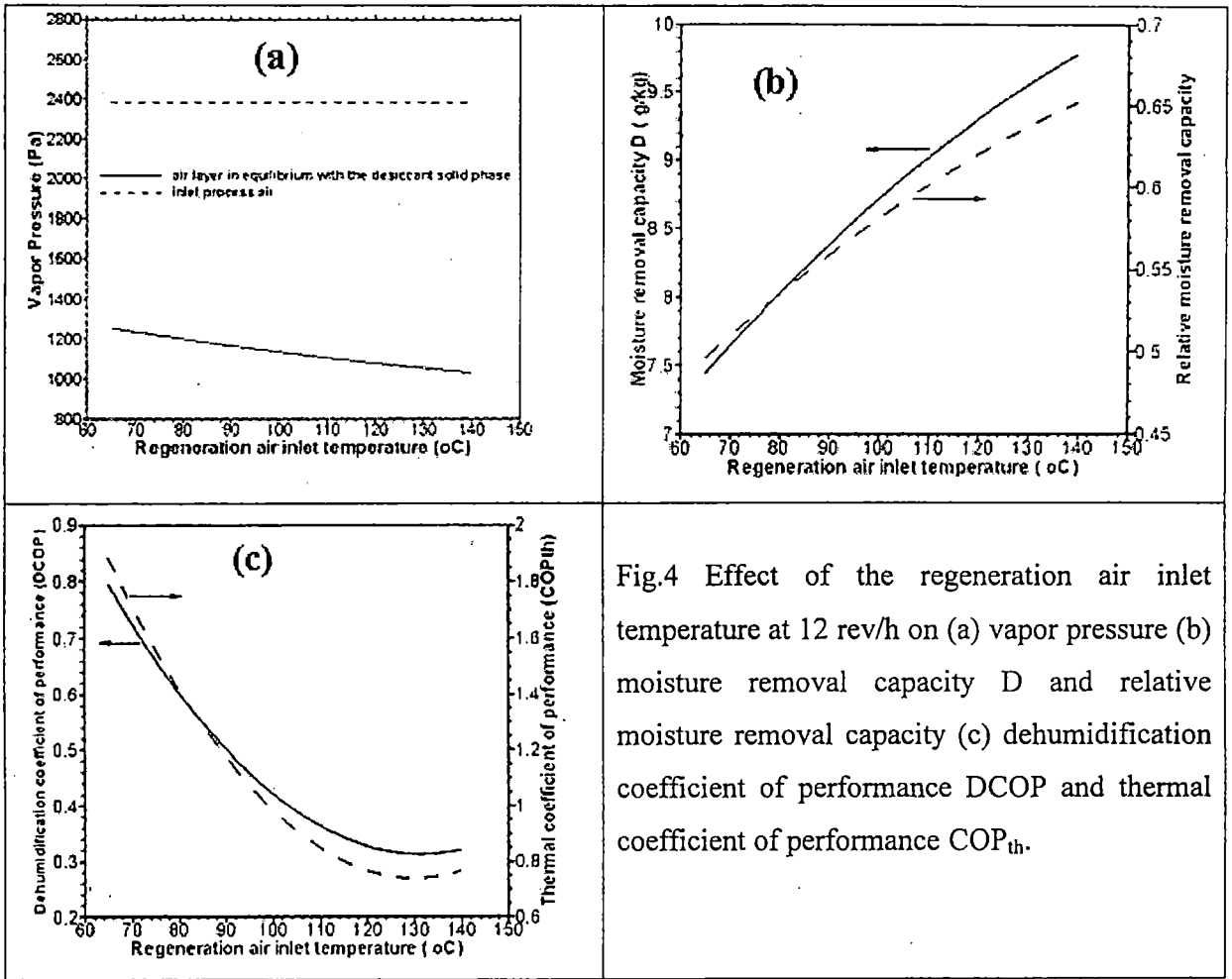
Fig.4 Effect of the regeneration air inlet temperature at 12 rev/h on (a) vapor pressure (b) moisture removal capacity D and relative moisture removal capacity (c) dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th}.

with the desiccant solid phase, the moisture removal capacity D, relative moisture removal capacity, dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th}. It is found that at the different regeneration air inlet temperature the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase decreases with increasing the rotation speed as well as the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed.
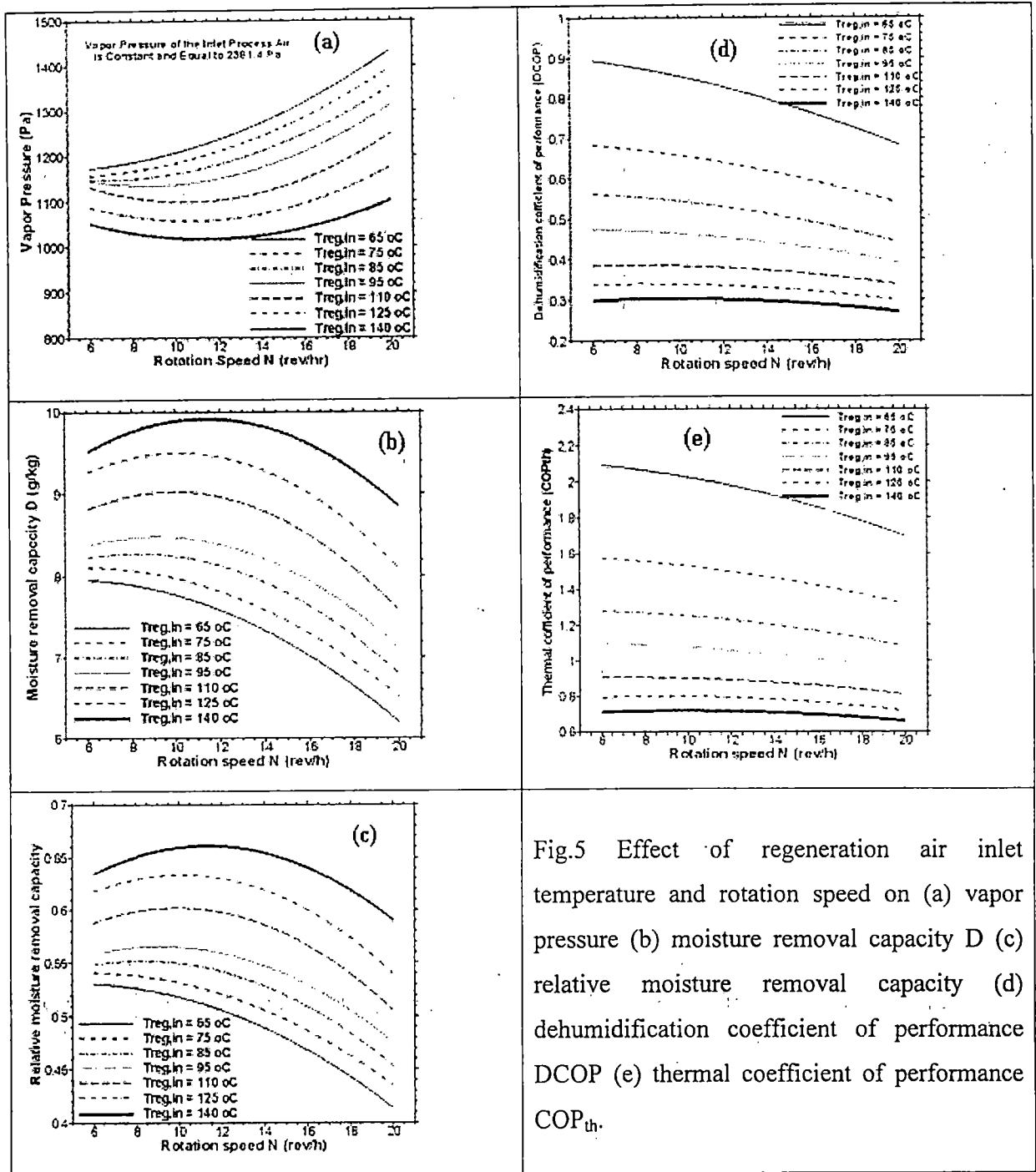
Fig.5 Effect of regeneration air inlet temperature and rotation speed on (a) vapor pressure (b) moisture removal capacity D (c) relative moisture removal capacity (d) dehumidification coefficient of performance DCOP (e) thermal coefficient of performance COP_{th}.

This is mainly because, for wheel rotates too fast, the desiccant material in the process air side as well as in the regeneration air side does not have enough time to remove the moisture. On the other hand, for increasing the inlet temperature of regeneration air the temperature of the desiccant material increase, which results in the higher relative humidity of the moisture air in equilibrium with the desiccant solid phase from the

isotherms, and the smaller dehumidification potential between the desiccant and the process air reducing the dehumidification if the wheel rotates too slowly. For high regeneration inlet air temperature no effect of rotation speeds on DCOP and COP_{th}. This is mainly because when the regeneration temperature is higher than 95 °C, the increasing of the energy needed to produce the high inlet temperature of the regeneration air is more significant than the increasing the moisture removal.

Figure 6 shows the relation between the optimal rotation speed of desiccant wheel and the regeneration air inlet temperature. It is found that the optimal rotation speed increases with increasing the inlet temperature of the regeneration air. This is mainly because, the higher of the regeneration air inlet temperature, these cause increase the temperature of the desiccant material, which results in the higher relative humidity of the moisture air in equilibrium with the desiccant solid phase from the isotherms, and the smaller dehumidification potential between the desiccant and the process air reducing the dehumidification if the wheel rotates too slowly. To increase the moisture removal capacity it is required decreasing the contact

time between the regeneration air and desiccant material by increasing the rotation speed of the desiccant wheel for increasing the process air inlet temperature.
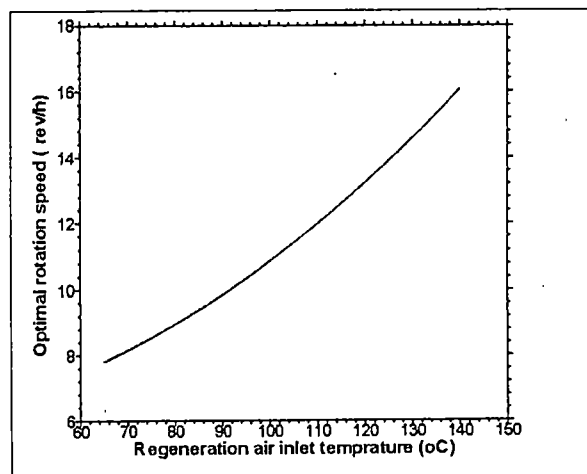

Fig. 6 Relation between the optimal rotation speed and the regeneration air inlet temperature.

7.2.2. Effect of the Regeneration Air Inlet Velocity on the Dehumidifying Performance and the Optimal Rotation Speed

In this section, effect of the regeneration air inlet velocity on the dehumidifying performance and the optimal rotation speed is simulated. The process and regeneration humidity ratio are constant and equal to 15 g/kg and the process air inlet velocity is

constant and equal to 2.5 m/s and regeneration air inlet temperature is 95 °C and process air inlet temperature is 35 °C and desiccant wheel thickness equal to 0.1m.

Figure 7 show the impact of the regeneration air inlet velocity on dehumidification performance for the regeneration air inlet velocity changes from 1.5 m/s to 5.5 m/s and all other operation parameters are held constant. It is found that: the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase increases with increasing the regeneration air inlet velocity and the moisture removal capacity D, relative moisture removal capacity increase with increasing the flow rate of regeneration air from 1.5 m/s to 3.5 m/s. when the regeneration air inlet velocity changes further from 3.5 m/s to 5.5 m/s, the change of the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase is very small and the increase of

moisture removal is not significant. At high regeneration air inlet velocity the adsorbed moisture is almost in equilibrium with the regeneration air. Therefore with the further increasing of the air velocity, D does not change a lot. DCOP, COP_{th} achieves an optimal value when the regeneration air inlet velocity about 2.5 m/s. at higher velocities; on the one hand, the increasing of the energy needed to heat the regeneration air increases gradually, on the other hand, again the wheel is heated without significant additional adsorption capacity due to reheating the equilibrium.

Figure 8 shows the effect of the regeneration air inlet velocity and rotation speed on the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase, the moisture removal capacity D, relative moisture removal capacity, dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th}.
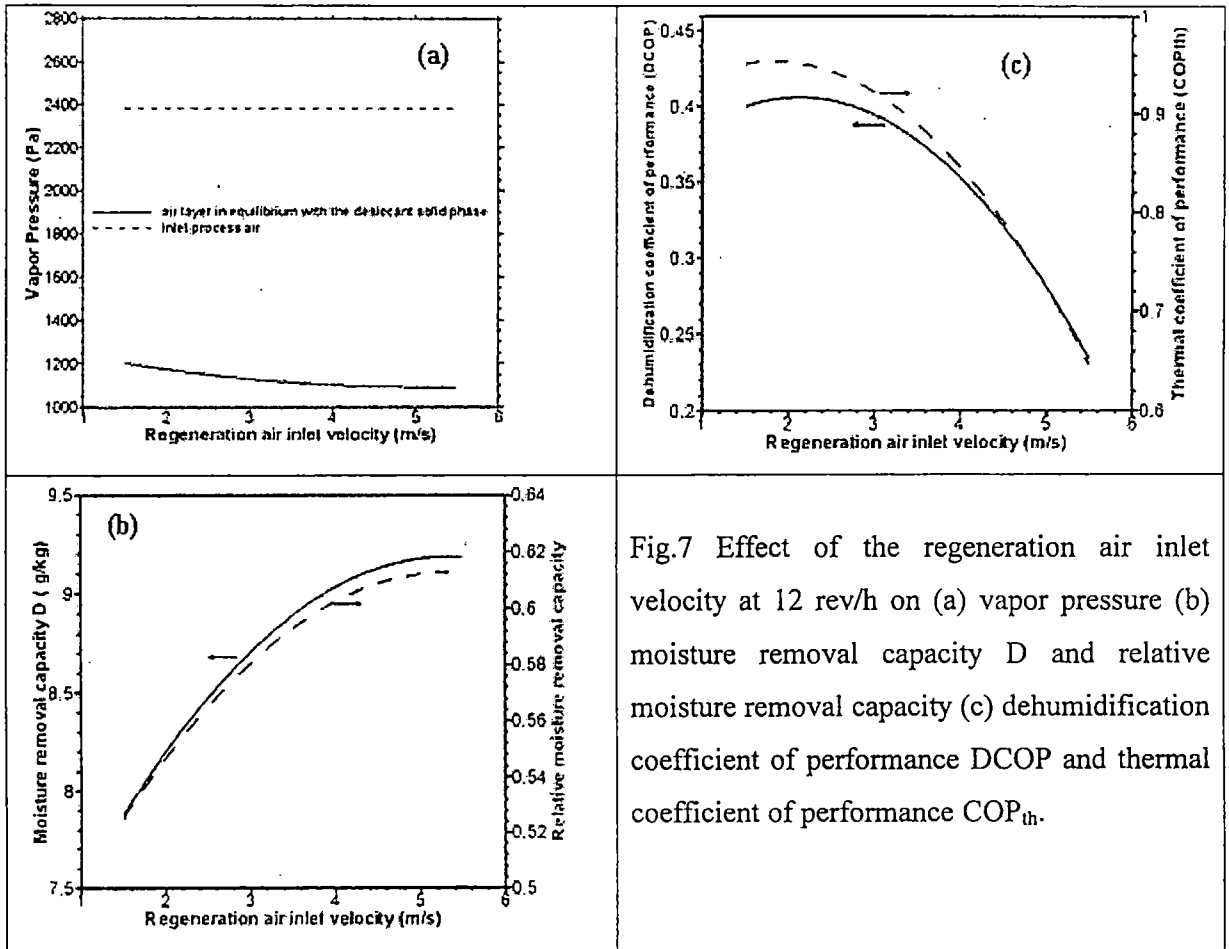
Fig.7 Effect of the regeneration air inlet velocity at 12 rev/h on (a) vapor pressure (b) moisture removal capacity D and relative moisture removal capacity (c) dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th}.

From Fig.7 it is found that at the different regeneration air inlet velocity the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase decreases with increasing the rotation speed as well as the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed.

This is mainly because, for wheel rotates too fast, the desiccant material in the process air side does not have enough time to remove the moisture. On the other hand, equilibrium is reached and the saturated desiccant material is still in the process section if the wheel rotates too slowly.
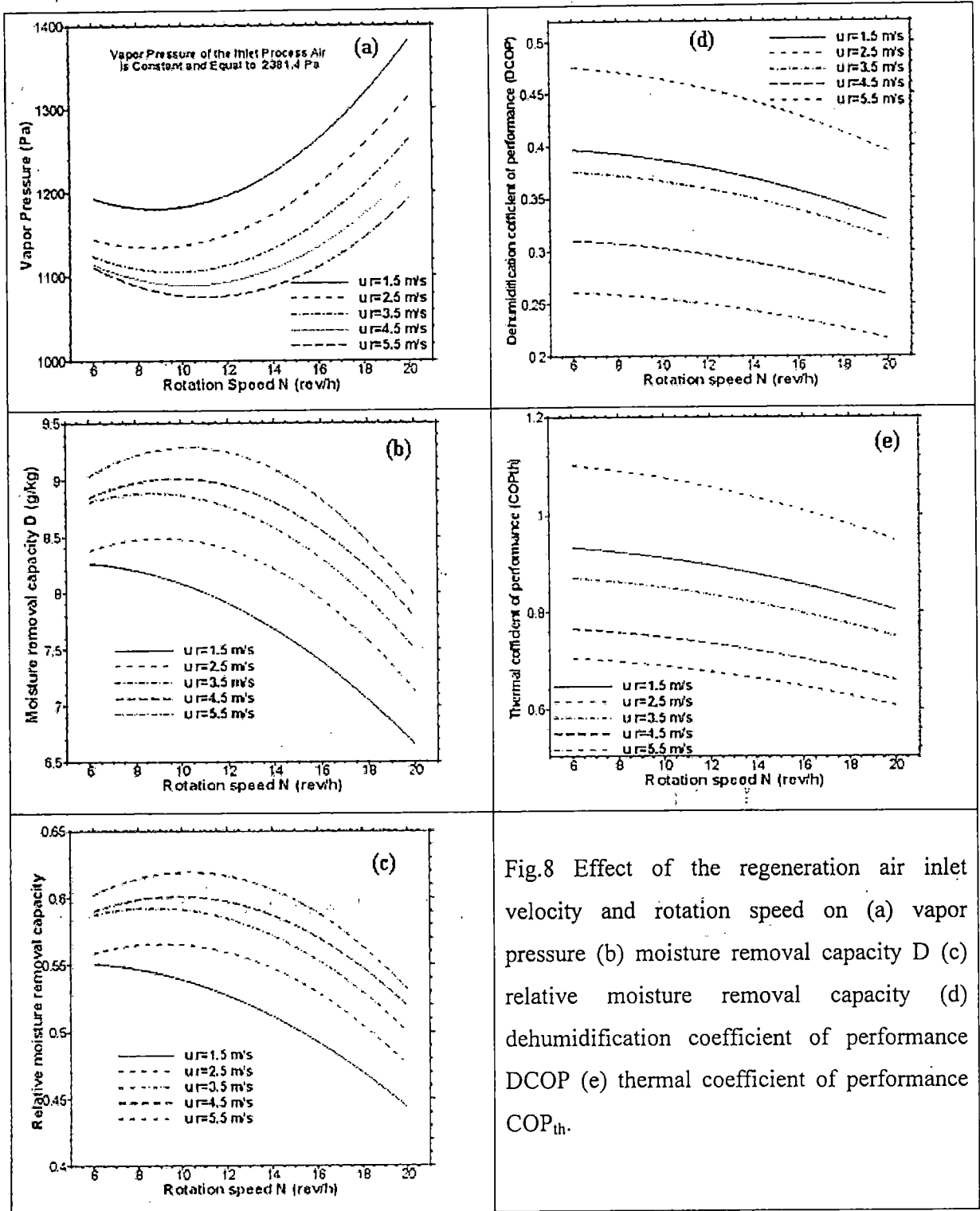
Fig.8 Effect of the regeneration air inlet velocity and rotation speed on (a) vapor pressure (b) moisture removal capacity D (c) relative moisture removal capacity (d) dehumidification coefficient of performance DCOP (e) thermal coefficient of performance COP_{th}.

Figure 9 shows the relation between the optimal rotation speed of desiccant wheel and the regeneration air inlet velocity. It is found that the optimal rotation speed increases with increasing the regeneration air

inlet velocity. This is mainly because, the adsorption rate increases with increasing the regeneration air inlet velocity, and then the optimal rotation speed increases.
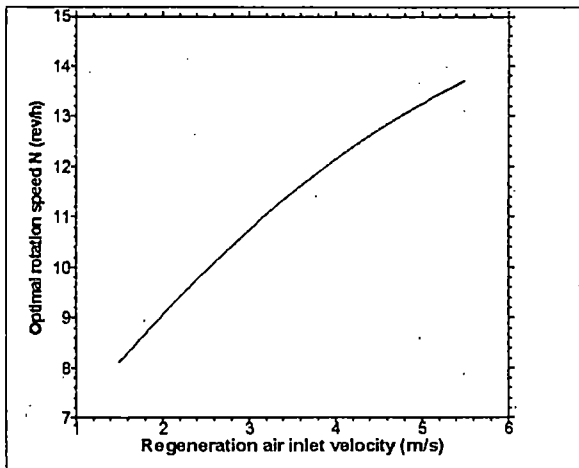

Fig. 9 Relation between the optimal rotation speed and the regeneration air inlet velocity.

7.2.3. Effect of Area Ratio of Process to Regeneration Air on the Dehumidifying performance and the Optimal Rotation Speed

In this section, effect of area ratio of process to regeneration air on the dehumidifying performance and the optimal rotation speed is simulated. The process and regeneration humidity ratio are constant and equal to 15 g/kg and the process air inlet velocity and regeneration air inlet velocity are constant and equal to 2.5 m/s and process air inlet temperature are constant and equal

to 35 °C and the regeneration air inlet temperature is 95 °C and desiccant wheel thickness equal to 10 cm.

Figure 10 show effect of area ratio of process to regeneration air on the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase, the moisture removal capacity D, relative moisture removal capacity, dehumidification coefficient of performance DCOP, and thermal coefficient of performance COP_{th}, for fixed air flow velocity, the mass flow rate of process air increase with increasing the area ratio of process to regeneration air, whereas the mass flow rate of regeneration air decreases with increasing the area ratio of process to regeneration air. This means that less regeneration air flow is adopted to remove the moisture of more process air flow for increase the area ratio of process to regeneration air. It is found that the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase decreases from 1354 Pa and 1154 Pa, the moisture removal capacity D decrease about 1.7184 g/kg and relative moisture removal capacity decrease about 0.1146 with increasing the area ratio of process to regeneration air. This is mainly because, the contact times between

the regeneration air flow and the desiccant side is not enough to removal the moisture from desiccant material. For decreasing the area ratio of process to regeneration air, the flux of process air decreases but the regeneration becomes more effective.

However, at the same time the thermal energy needed to heat the regeneration air decrease with increase the area ratio of process to regeneration air, and then DCOP and COP_{th} increases.

Fig.10 Effect of area ratio of process to regeneration air at 12 rev/h on (a) vapor pressure (b) moisture removal capacity D and relative moisture removal capacity (c) dehumidification coefficient of performance DCOP and thermal coefficient of performance COP_{th}.
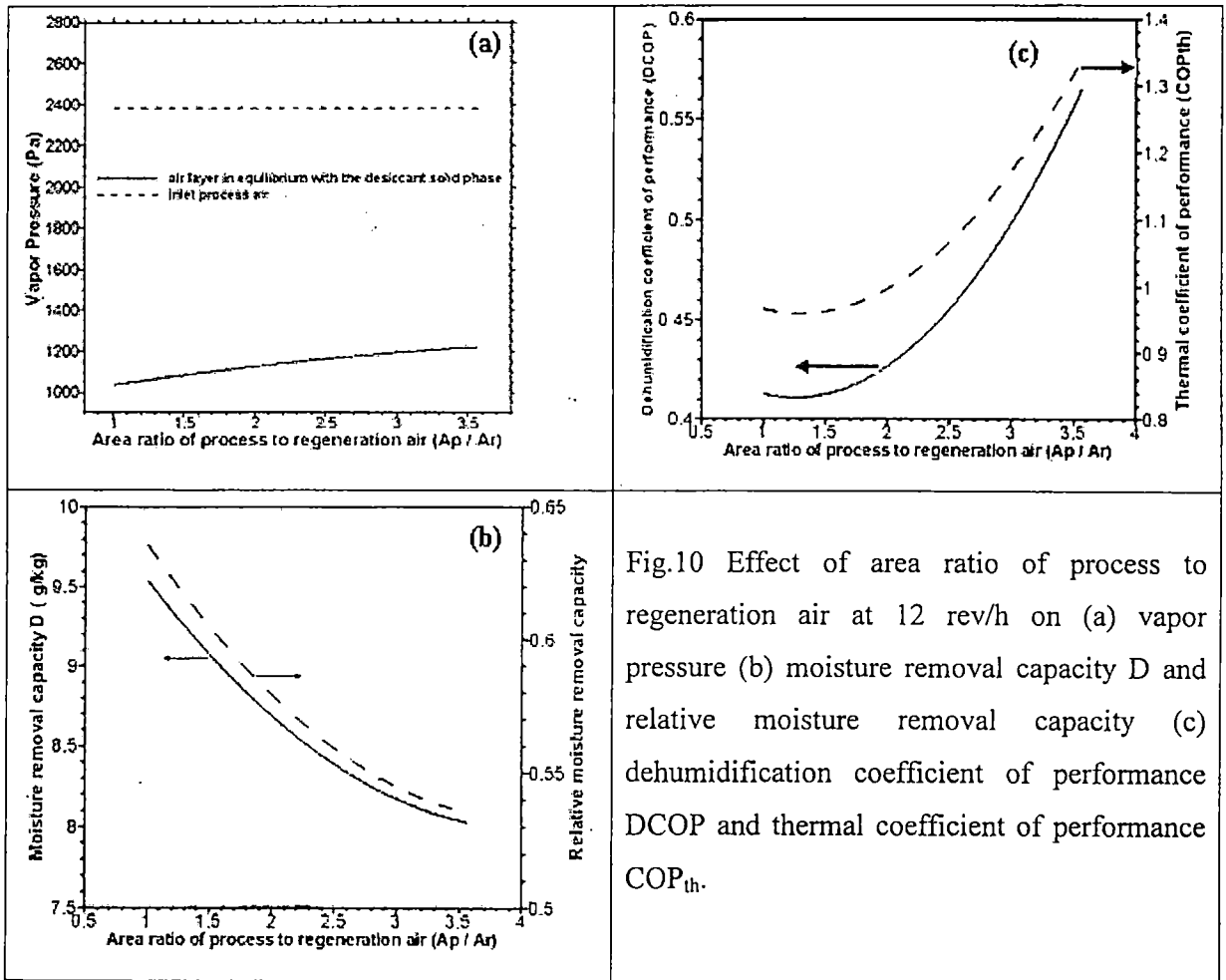
Figure 11 shows the effect of area ratio of process to regeneration air and rotation speed on the difference in the vapor pressure between the inlet process air and air layer in equilibrium with the desiccant solid phase, the moisture removal capacity D, relative moisture removal capacity, dehumidification

coefficient of performance DCOP and thermal coefficient of performance COP_{th}. It is found that at the different area ratio of process to regeneration air the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed. This is mainly

because, for wheel rotates too fast, the desiccant material in the process air side as well as in the regeneration air side does not have enough time to remove the moisture.

On the other hand, equilibrium is reached and the saturated desiccant material is still in the process section if the wheel rotates too slowly.
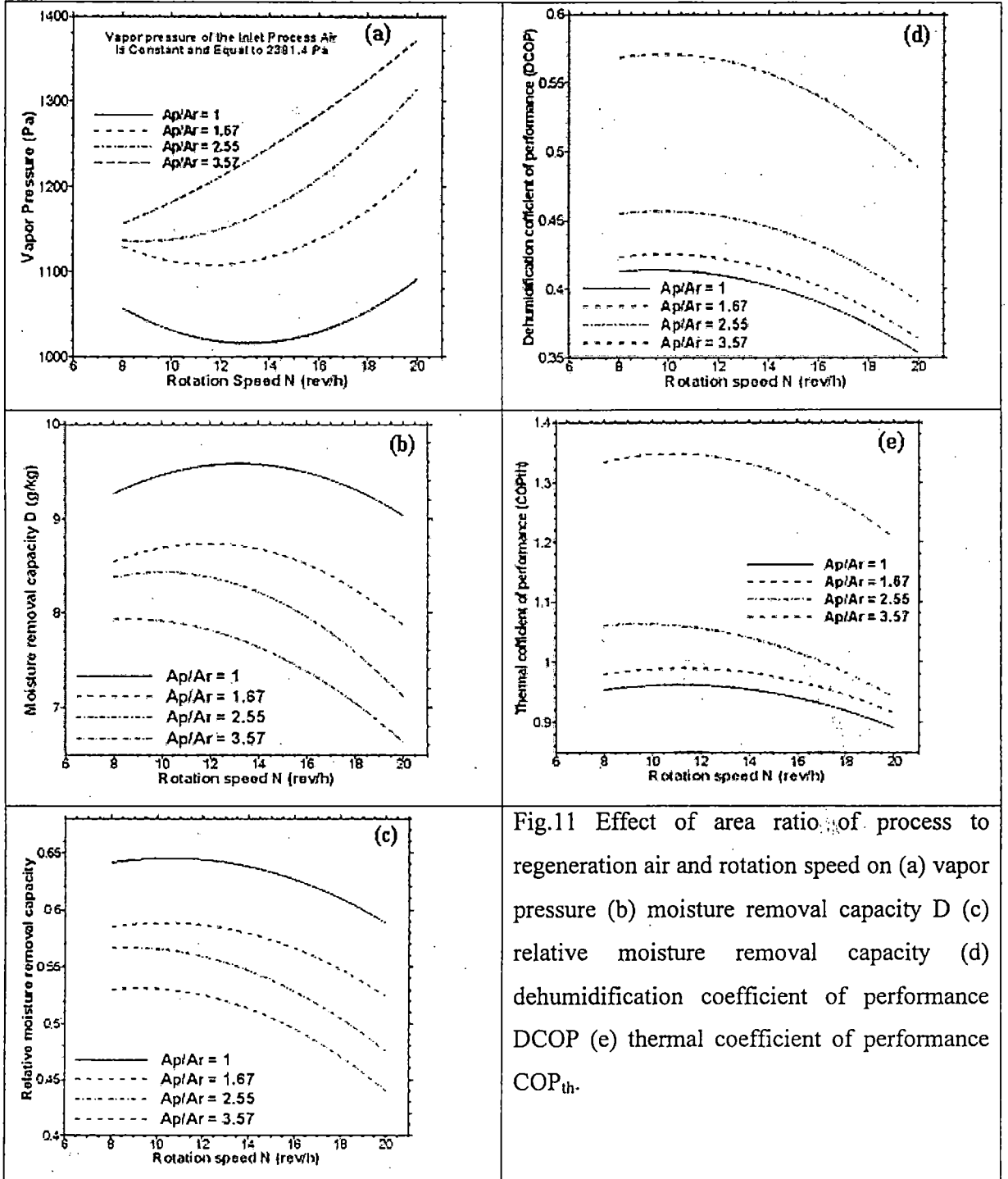

Fig.11 Effect of area ratio of process to regeneration air and rotation speed on (a) vapor pressure (b) moisture removal capacity D (c) relative moisture removal capacity (d) dehumidification coefficient of performance DCOP (e) thermal coefficient of performance COP_{th}.

Figure 12 shows the relation between the optimal rotation speed of desiccant wheel and the area ratio of process to regeneration air. The optimal rotation speed is obtained at biggest moisture removal capacity D. the value of optimal rotation speed decreases when the area ratio of process air to regeneration air increases. The results show that the desiccant wheel needs smaller area ratio between the process and regeneration air to complete the adsorption process if it rotates faster.
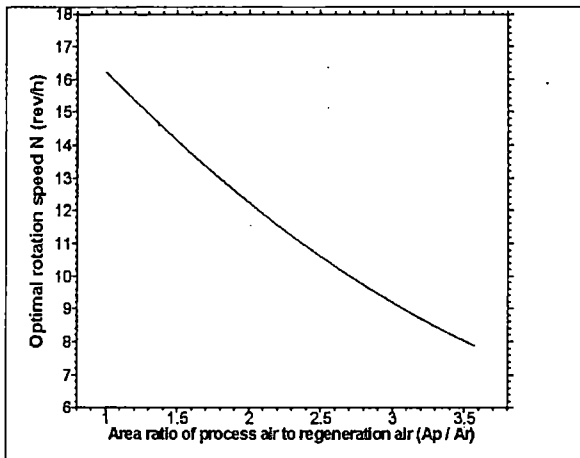

Fig. 12 Relation between the optimal rotation speed and the area ratio of process to regeneration air.

8. Conclusions

A mathematical model for predicting the performance of solar energy-assisted hybrid air conditioning system (SEAHACS). The desiccant wheels used honeycombed silica gel–haloids composite desiccant wheel, one-

rotor six-stage rotary desiccant cooling system, in which two-stage dehumidification process, two-stage pre-cooling process and two-stage regeneration process are realized by only one wheel. Three air streams are involved in the present system. The model is adopted to investigate the influence of the main parameters on system performance and optimal rotation speed. The conclusions of the main results from this paper are summarized below:

- For increasing the regeneration air inlet temperature, the moisture removal capacity, DCOP, and COP_{th} increases. Moreover, at the different regeneration air inlet temperature the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed.

- The optimal rotation speed increases with increasing the inlet temperature of the regeneration air.

- Moisture removal capacity D and relative moisture removal capacity increase with increasing the regeneration air inlet velocity from 1.5 m/s to 3.5 m/s. when the regeneration air inlet velocity changes further from 3.5 m/s to 5.5 m/s, the increase of moisture removal is not significant. At high regeneration air inlet

velocity the adsorbed moisture is almost in equilibrium with the regeneration air. Therefore with the further increasing of the air velocity, D does not change a lot.

- At the different regeneration air inlet velocity the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed. Moreover, the optimal rotation speed increases with increasing the regeneration air inlet velocity.

- For increase the area ratio of process to regeneration air the moisture removal capacity decreases as well as the DCOP and COP_{th} increasing with increase the area ratio of process to regeneration air.

- At the different area ratio of process to regeneration air the dehumidification desiccant wheel performance decreasing with increasing the wheel rotation speed. Moreover, the optimal rotation speed decreases when the area ratio of process air to regeneration air increases.

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