Simulation of Desiccant Cooling

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Abstract

Desiccant cooling system has been an attractive topic for study lately, due to its environmentally friendly nature. It also consume less electricity and capable to be operated without refrigerant. A simulation study was conducted using 1.5 m long ducting equipped with one desiccant wheel, one sensible heat exchanger wheel, one evaporative cooling chamber and two blowers and one electric heater. The simulation study used 8.16 m/s primary air, the drying coefficient from desiccant wheel, k_1=2.1 (1/s), mass transfer coefficient in evaporative cooling, k_2=1.2 kg vapor/s, heat transfer coefficient in desiccant wheel, h_1=4.5 W/m^2 ^{o}C, and heat transfer coefficient in sensible heat exchanger wheel h_2= 4.5 W/m^2 ^{o}C. The simulation results show that the final temperature before entering into the air conditioning room was 25 ^{o}C and RH of 65 %, were in accordance with the Indonesian comfort index.

Keywords: desiccant wheel, desiccant cooling, evaporative cooling, sensible heat exchanger wheel, silica gel

1. Introduction

Indonesia lies in the tropic where the average air temperature is around 30 ^{o}C and RH around 80% all year round. Under this weather condition it is not fit for people to work in the office or stay at home. Therefore, there is an urgent need for air conditioning facilities in order to be able to live in a better condition. There is also need for industry to increase their productivity by creating better working environment. As the current air conditioning system requires high electricity consumption which requires high fossil fuel input, there is a need to find alternative for the conventional air conditioning system. The best option would be a desiccant cooling system which does not need refrigerant for operating the system. Through the manipulation of air condition using silica gel it is possible to create a comfort condition of a room.

Research on this type of air conditioning system is new in Indonesia and very rare if any attempt to apply the system in Indonesia. Research by Chadi Maalouf. at al in France (2006) indicated that by using solar energy they were capable to construct adsorption cooling system for application in several city in France. Daou et al. (2004) and Jurinak (1982) have conducted research to determine the performance of an adsorption cooling machine using silica gel the Pennington cycle. Rajat Subhra Das et al. (1995) study the application of solar energy for liquid desiccant cooling system in India Two dimensionless parameters - enthalpy and moisture effectiveness are taken as performance indices of the absorber. The performance of the overall system is presented in terms of its cooling capacity, moisture removal rate and COP (coefficient of performance). Davangere et al. (1999) had applied a desiccant cooling system with capacity 10 kW (2.85 ton refrigeration) assisted by vapor compression machine. The resulting room temperature 26.7 ^{o}C with humidity ratio of W=0.01183 kg/kg dry air for the condition Florida which have outside air of 36 ^{o}C. They conducted analysis using Psychrometric chart and the result of their simulation works were applied to four cities in the USA. Bellia, et al (2000) had studied several hybrids cooling system using various desiccant wheels and using DesiCalc™ computer program and applied to four cities in Italy. They concluded that the maximum saving in cost was 22%, and for the theater the saving were greater from 23% to 38% with electricity saving of up to 55%.

The purpose of the study is to obtain mathematical model for the purpose of simulation of desiccant cooling system.

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2. The Working Principle of A Desiccant Cooling System

Fig. 1 shows the major component of a desiccant cooling system which comprises of a desiccant wheel containing silica-gel, sensible heat exchanger wheel, a hot water heater supplied from solar collector, blowers and evaporative cooler (Pons and Kodama, 2014). Outside air is introduced through point (1) passing the hot desiccant wheel where the humidity is reduced to point (2). The air will further passed through the sensible heat exchanger (point 3) where its temperature will be reduced while keeping its RH constant. From the sensible heat exchanger the air will be introduced into the evaporative cooling where its temperature will be reduced by its RH will be increased (point 4). When entering the room the temperature and RH will reach 26 ^{o}C and 55 %, respectively, a comfortable condition for air conditioning. The air condition in the desiccant cooling system can also be traced using the Psychometric chart in Fig.2. From the room under condition of point (5) the air will be passed again through the evaporative cooling unit which will reduced its temperature and increase its RH. After passing through the sensible heat exchanger its temperature will increase while its RH is kept constant as in point (6). After passing through the heater, the air temperature increase again heating the desiccant wheel to the condition as point (8). After passing the desiccant wheel the air will gain moisture due to evaporation from the desiccant wheel and it temperature will drop.

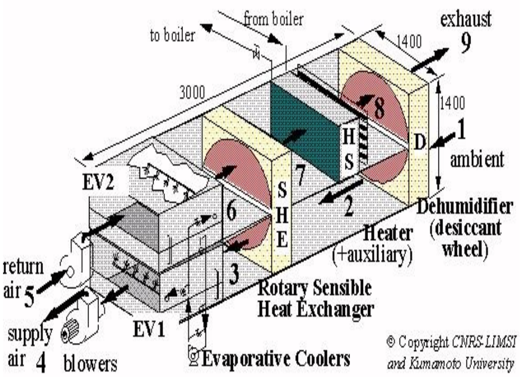

Fig. 1 Main component of desiccant cooling system (CNRS-LIMSI and Kumamoto University, 2014)

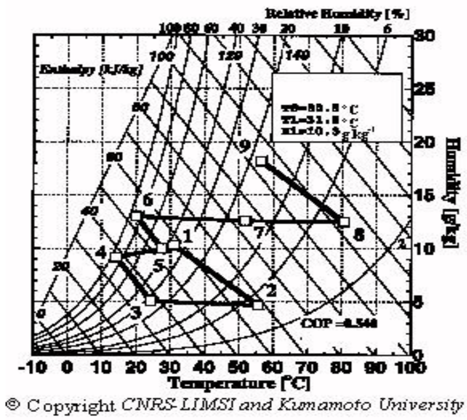

Fig. 2 Air condition in a Psychometric chart (CNRS-LIMSI and Kumamoto University, 2014)

3. Mathematical Modelling

If m is the total mass of air in the duct and x is the humidity ratio then the humidity ratio change along the z axis of the total length of 1.5 m of the duct can be calculated using the following mass balance equations

$$\frac{dx}{dz} = -k_1(M - Me)/V \quad for\ 0 < z < 0.5\,\mathrm{m} \tag{1}$$

$$\frac{dx}{dz} = c(x - x_s)/Vm \quad for\ 1.1 > z > 1.5\,\mathrm{m} \tag{2}$$

To calculate the change in temperature along the duct from the inlet to the duct out let and energy balance will be used.

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = h_1 Ax_1(Tx_1 - T) \tag{3}$$

for $0.0 < z < 0.5\,\mathrm{m}$

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = -h_2 Ax_2(Tx_2 - T) \tag{4}$$

for $0.5 < z < 1.1\,\mathrm{m}$

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = -h_{ev}(T - T_s) \tag{5}$$

for $1.1 < z < 1.5\mathrm{m}$

For the condition of the return air from the air condition room, the following mass balance equation will be used.

$$\frac{dx}{dz} = c(x - x_s)/Vm \quad \text{for} \; 1.5 > z > 1.1\,\text{m} \tag{6}$$

$$\frac{dx}{dz} = +k_1(M - Me)/V \tag{7}$$

$for \quad 0.5 < z < 0.0\,\text{m}$

From the energy balance the following relation can be obtained

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = -h_{ev}(T - T_s) \tag{8}$$

$\text{for} \; 1.1 < z < 1.5\text{m}$

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = h_2Ax_2(Tx_2 - T) \tag{9}$$

$\text{for} \; 1.1 < z < 0.8\,\text{m}$

$$m_{x_2}Cp_{x_2}\frac{dTx_2}{dt} = h_2Ax_2(Tx_2 - T) \tag{10}$$

$$mCp\frac{dT}{dz} = h_{hx}A_{hx}(T_{hx} - T) \tag{11}$$

$\text{for} \; 0.8 < z < 0.5m$

$$mCp\frac{dT}{dz}x\frac{dz}{dt} = h_1Ax_1(Tx_1 - T) \tag{12}$$

$\text{for} \; 0.5 < z < 0.0\,\text{m}$

$$m_{x1}Cp_{x1}\frac{dT_{x1}}{dt} = h_1A_{x1}(T_{x1} - T) \tag{13}$$

The amount of heat supplied from solar collector can be calculated using the following equations.

$$qu = I_{rad}Ac - U_LAc(Tc - Ta) \tag{14}$$

$$qu = \dot{m}_wCp_w(Tc - T_{hx}) \tag{15}$$

4. System Simulation

With the use of parameters listed in Table 1, a simulation study was conducted. The results are as shown in Fig. 4 for humidity ratio change along the duct and in Fig. 5 showing the temperature change. Table 2 shows simulation data for solar collector hot water supply.

Table 1 Simulation data

Quantity	Quantity	Quantity
m=0.06kg	Ts=23 ^oC	h_{ev}=3.7 (W)
V= 8.16 m/s	A_{x1}=4.5 m^2	m_{x2}=1.25 kg
k_1=0.5 (1/s)	A_{x2}=1.5 m^2	Cp_{x2}=0.897 kJ/kg ^oC
k_2= 0.5 (kg vapor/s)	h_1=4.5 W/m^2 C	A_{hx}=1.5 m^2
Xs=0.007	h_2=4.5 W/m^2 C	h_{hx}=3.5 W/m^2 ^oC
Me=7(%db)	T_{x3}=68 ^oC	T_{hx}=68 ^oC
m_{x1}=2.3 kg	Cp_{x1}=0.921 kJ/kg ^oC	

Table 2 Data for Solar Collector Heating

m_w (kg/s)	Cpw(kJ/kg^{o}C)	I_{rad} (W/m^2 C)
15	4.19	600
Ac (m^2)	U_L (W/m^2 C)	
1.64	4.5	

Change in absolute humidity of incoming outside air is presented in Fig. 3 while the change in the incoming air temperature is shown in Fig. 5 below.

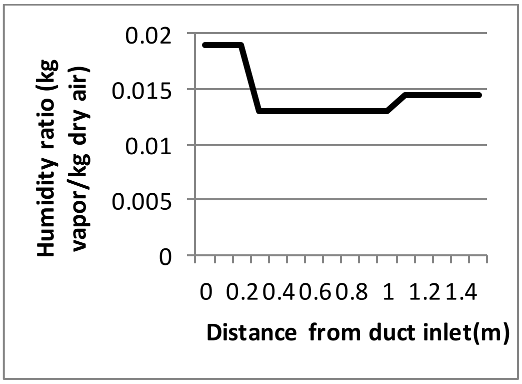

Fig. 3 Change of entering air humidity ratio across the duct

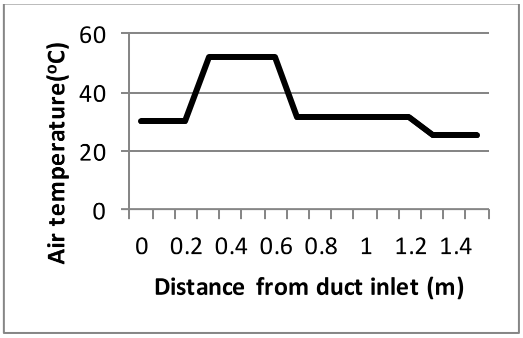

Fig. 4 Change of entering air temperature from duct inlet

For the returning air from the room its humidity ratio and temperature change are as shown in Fig. 6 and 7.

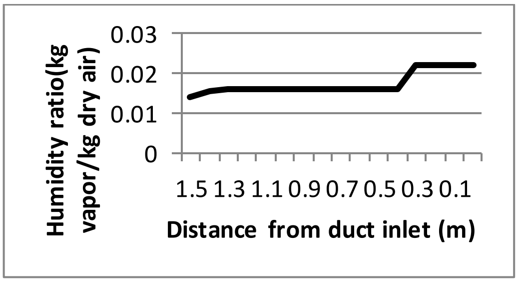

Fig. 5 Change in returning humidity ratio along the duct

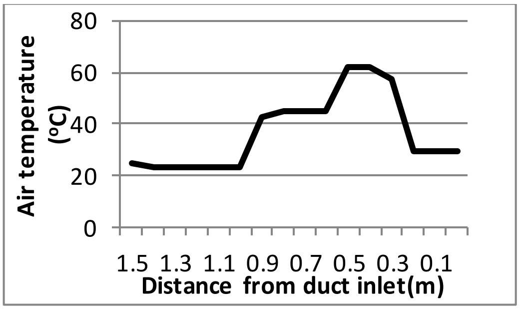

Fig. 6 Change in air temperature leaving the conditioned room

To achieve solar collector temperature of 75 ^{o}C there is a need to supply 651.9 Watt of energy and if this heat is supplied to the heater in the form of heat exchanger so that heat exchanger temperature can reach 65 ^{o}C the rate of water flow should be kept at 15 kg/s This temperature will be used to heat the desiccant wheel to drive the moisture out from the desiccant.

If the results of humidity ratio change and the air temperature change along the duct are plotted in the psychrometric chart the results is shown as in Fig. 8.

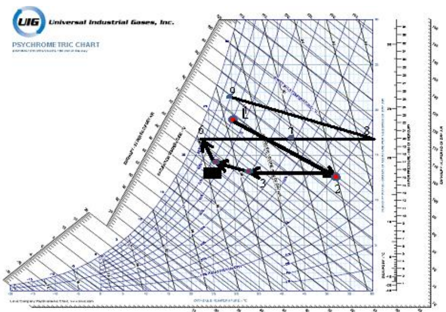

Fig. 7 Change in air condition along the duct as plotted in the psychrometric chart (Davanagere dkk, 1999)

As shown here the air condition entering the room has achieved the comfort condition of 25 ^{o}C and RH of 65% (see point 4).

5. Conclusions

1) It was possible to develop mathematical model for desiccant cooling.
2) Simulation results using 0.35 x 0.35 m cross section and length of 1.5 m, with air flow rate of 0.5kg/m^{2}s, and with heater temperature of 68 ^{o}C and using desiccant wheel, a sensible heat exchanger wheel and evaporative cooling it was possible to create comfort air condition of 25 ^{o}C and RH 65%.
3) It was necessary to supply heat from solar collector having area of 1.64 m^{2} with average solar irradiation of 600 W/m^{2} and water flow rate 15 kg/s in order to produce the necessary heating temperature of 68 ^{o}C.

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Nomenclature

Ac = area of solar collector (m^{2})

Ax_1 = heat transfer surface of the desiccant (m^{2})

Ax_2 = surface heat transfer of the sensible heat exchanger (m^{2})

c = mass transfer coefficient in the evaporative cooler (kg/s.)

Cp = specific heat of the air (kJ/kg ^{o}C)

Cp_{x1} = specific heat of desiccant (kJ/kg ^{o}C)

Cp_{x2} = specific heat of sensible heat exchanger (kJ/kg ^{o}C)

h_1 = heat transfer coefficient of the desiccant wheel (W/m^{2} ^{o}C)

h_2 = heat transfer coefficient of the sensible heat exchanger (W/m^{2} ^{o}C)

h_{hx} = heat transfer coefficient of the heater (W/m^{2} ^{o}C)

h_{ev} = rate of heat transfer with the evaporative cooling (W)

I_{rad} = solar irradiation (W/m^2)

k_1 = desorption constant (1/det.)

m = mass of air (kg)

m_{x1} = mass of desiccant wheel (kg)

m_{x2} = mass of sensible heat exchanger wheel (kg)

M = moisture content of desiccant (%db)

Me = equilibrium moisture content of desiccant (%db)

qu = usefull energy from the sun (Watt)

Ta = ambient temperature (^oC)

Tc = collector temperature (^oC)

T_{hx} = heat exchanger temperature (^oC)

Ts = temperature of evaporative cooling (^oC)

Tx_1 = desiccant temperature (^oC)

Tx_2 = temperature of sensible heat exchanger (^oC)

U_L = overall loss coefficient of the collector (W/m^2 C)

V = air flow rate (m/det.)

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