Theoretical Analysis of Desiccant Cooling System: A Case Study

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Abstract: This paper presents a theoretical analysis of desiccant cooling system based on second law analysis for the environment condition of Kurukshetra, Haryana, India. Desiccant cooling system worked on ventilation and recir-culation mode. Renewable energy and low grade or waste energy can be used in the system. In this analysis, coef-ficient of performance (COP) is to be calculated in both ventilation and recirculation mode and found the best mode for the system according to environment condition. A very few studies have done on the second law aspects of these system. The reversible COP depends on the operating parameters and the analysis is based on certain op-erating conditions for the desiccant dehumidifier. The thermal and reversible COPs of an open desiccant cooling system depend on operating conditions of the system. A second law analysis is performed and reversible COPs are 2.26 and 2.56 in ventilation and recirculation mode respectively at ambient condition for month of May 43.3◦C and 20%RH of Kurukshetra, Haryana. The results of the analysis provide an upper limit for system.

Keywords: Desiccant cooling, Ventilation mode, Recirculation mode, Second law analysis, Waste energy.

1. Introduction

Desiccants are those materials which adsorbed/absorbed (solid and liquid desiccants respectively) the moisture from humid air and hold it. These materials are used in many application like air conditioning etc. particularly when the latent load is large in comparison to the sensible load. Solid desiccants can be reused thousands of cycles. After regeneration they are ready to reuse but in case of liquid desiccants materials reuse cannot possible . The cost of energy to regenerate the desiccant is low. The cost of running a vapour compression cooling system is very high. A desiccant process may offer considerable advantages in energy .Desiccant cooling system is also a very good op-tion for air conditioning system because it is driven by low grade energy which is easily available.

For achieving comfort room condition different solid desiccant cooling cycles were evaluated for typical hot and humid climates in 16 cities. Psychometric evolution of cycles (Ventilation, Recirculation and Dunkle cycles) for 16 cities carried out for achieving comfort condition in room. It was found that among all cycles Dunkle was better for a wide range of outdoor condition (Jain S. and Dhar P.L.,1995). Desiccant wheel and enthalpy recovery wheel were also compared with each other (Zhang L.Z. et al.,2002). For open-cycle desiccant cooling system a procedure de-veloped for the analyses of energy and exergy. The analysis shows that an exergy analysis can provide useful in-formation with respect to the theoretical upper limit of the system performance, which cannot be obtained from an energy analysis alone (Kanoglu M.,2003). Exergy analysis had been carried out for the different design temperatures and relative humidity conditions from ASHRAE summer thermal comfort conditions (Alpuche G.Ma.et al. 2004).A second law analysis performed on open cycle desiccant cooling system for ventilation and recirculation modes. The results of the analysis provide an upper limit for the system performance at various ambient conditions and may serve as a model to which actual desiccant cooling systems may be compared (Kanoglu M.,2006).

The objective of this paper to find out the theoretical reversible COP according to different temperature and relative humidity of Kurukshetra, Haryana, India for desiccant cooling system in both ventilation and recirculation mode and compare results in both mode.

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2. Decant system

A desiccant cooling system consists of a desiccant wheel, a heat exchanger, two evaporative coolers and associated blowers for air movement. After the desiccant wheel adsorbs moisture from the process air, this become hot and dry after exist due to the heat of adsorption by desiccants materials. This hot and dry air goes into the rotary regenerator where air cools sensibly. Now this cool air pass through the evaporator where moisture to be added before going to conditioned space. On the regeneration side, air is fist reduced in temperature by passing it through an evaporative cooler. Now this air passes through the rotary heat exchanger where it becomes heat.

Before passes through the desiccant wheel it is routed to external heat source and when finally this air passes through the desiccant wheel and becomes hot and humid because it picks up the moisture from process air. The hottest air exiting the heat exchanger is used for regeneration of the desiccant wheel. Desiccant cooling systems can be operated either in a ventilation mode (**Figure 1**) or a recirculation mode (**Figure 2**).

3. First law analysis ventilation mode

The whole system is taken as ideal or every component works as a ideally for the analysis of the system and mass flow rate in process line and regeneration are equal, which gives the upper limit of the system. (M.Kangolu et al.,2007). Ideal desiccant wheel fully adsorb the moisture and becomes hot and dry. So at the exit of desiccant wheel specific humidity is given by

$$W2=0 \quad (1)$$

The rotary regenerator (RR) is works as a counter flow heat exchanger and for achieve maximum heat transfer hot inlet air in process line is cooled to the temperature of inlet regeneration air

$$T_3=T_6 \quad (2)$$

An energy balance for ideal adiabatic RR (i.e. effectiveness is unit) is given by-

$$h_2-h_3=h_7-h_6. \quad (3)$$

Specific humidity across the RR for process line is given by -

$$W_3=W_2 \quad (4)$$

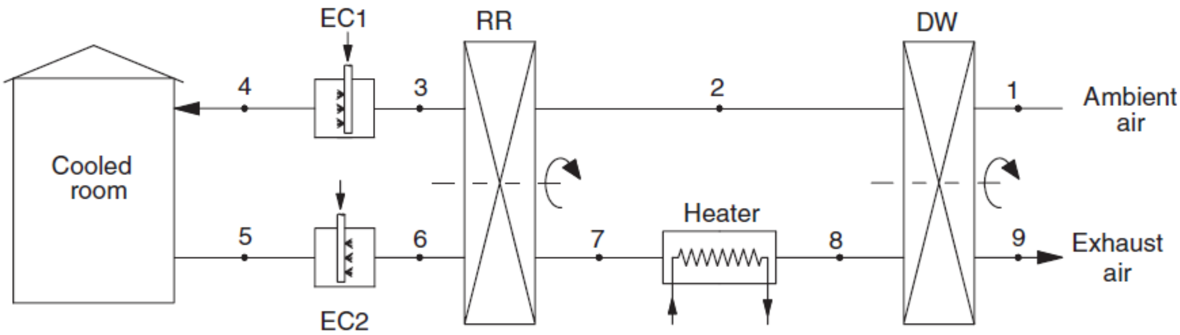

Figure 1. Schematic diagram of desiccant cooling system in ventilation mode (M.Kanoglu *et al*.,2007)

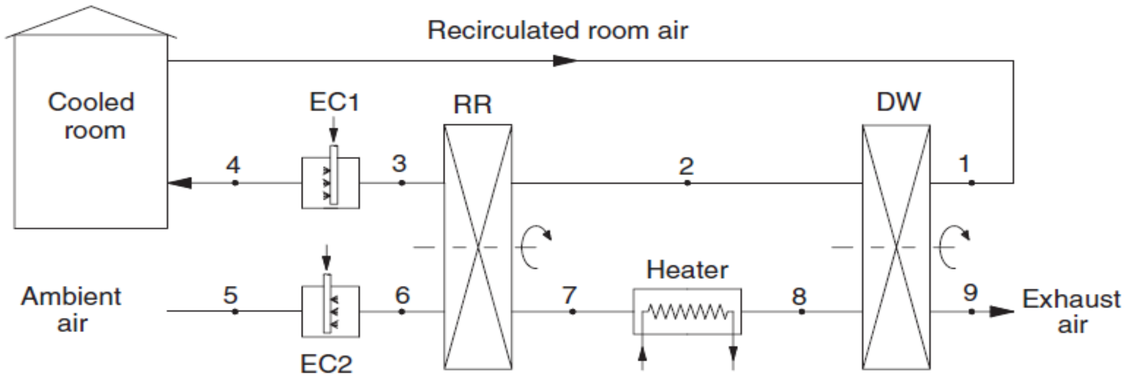

Figure 2. Schematic diagram of desiccant cooling system in regeneration mode (M.Kanoglu *et al*.,2007)

Specific humidity across the RR for regeneration line is given by

$$W_7=W_6 \quad (5)$$

In the evaporative cooler (EC) adiabatic humidification takes place. Ideally process air leaves the EC at 100% relative humidity corresponding to unity effectiveness for evaporative cooler. Mathematically it is given by-

$$T_4=Twb_3 \quad (6)$$

$$\Phi_4=1 \quad (7)$$

Now for regeneration line, for the ideal EC, the equation is given by-

$$T_6=Twb_5 \quad (8)$$

$$\phi_6=1 \quad (9)$$

For heating the regeneration air, the heat supplied is given by-

$$q_{in}=h_8-h_7 \quad (10)$$

When hot regenerated air passes through the desiccant wheel it will become regenerate. Ideally regenerated heat is giv-en by

$$q_{ren}=(W_1-W_2)h_{fg}=(W_9-W_8)h_{fg} \quad (11)$$

where, hfg is the latent heat of vaporization for water.

In terms of enthalpy, regeneration heat is given by-

$$q_{ren}=h_8-h_9=h_2-h_1 \quad (12)$$

Between state '5' and state '4' there is a conditioned space. The cooling capacity of the system is given by-

$$q_{cool}=h_5-h_4. \quad (13)$$

The thermal COP of this system is given by

$$COP=(q_{cool}/q_{regen})=(h_5-h_4)/(h_8-h_7) \quad (14)$$

3.1 Recirculation mode

As shown in **Figure 2**, the process air is recirculated while to regenerate the desiccant wheel separately ambient air drained into the regeneration line. In recirculation mode state 1 is the room condition and state 5 is the ambient in-let condition.

For calculating the COP & q_{cool} in recirculation mode equations from 1-14 will be used.

3.2 Second law analysis

For the second law analysis and calculation for maximum COP the whole system can be used as combined carnot en-gine and carnot refrigerator. Carnot cycle gives the upper limit of COP. The work produced from carnot engine fed into the carnot refrigerator.

The expressions for the work output from the carnot heat engine, the cooling load of the carnot refrigerator, and the carnot COP of the whole system are

$$W_{out}=\eta_{th,c}q_{in} \quad (15)$$

$$q_{cool}=COP_{RC}W_{out} \quad (16)$$

$$COP_C=q_{cool}/q_{in}=\eta_{th,c} \qquad COP_{RC}=(1-T_{amb}/T_{source})(T_{space}/T_{amb}-T_{space}) \quad (17)$$

where, η_{th},c is the thermal efficiency of the carnot heat engine , $COP_{R,C}$ is the COP of Carnot refrigerator, and $T_{ambient}$, (i.e.T_1) Tspace,(i.e. T_5) and T_{source}, (i.e.T_8)are the temperatures of ambient, cooled space, and heat source, respectively.

Eq. (17) gives the upper limit for any heat-driven cooling system,. The reversible COP of the open desiccant cooling systems is given by in terms of temperature-

$COP_{rev}=(1-T_{amb}/T_s)(T_e/T_c-T_e)$ (18)

(Lavan et al. ,1982 and Kodama P.,2000) defined the equivalent Carnot temperature is

$T_{equiv}=(\ \Sigma\ m_ih_i\ -\ \Sigma\ m_eh_e)/(\ \Sigma\ m_is_i-\ \Sigma\ m_es_e)$ (19)

where 's' is expressed using the equivalent Carnot temperatures approaches the entropy of the fluid stream, and the subscripts 'i' and 'e' stand for inlet and exit states. Both the air and water stream satisfying the mass balances must be used in this equation. Apply these equation in ventilation mode and obtain the expressions for the equiva-lent temperatures to be

$T_S=(h_7–h_8)/(s_7-s_8)$ (20)

(23)

$T_c=(h_9-h_5)-(\Delta\omega_2+\Delta\omega_4)h_w/s_9-s_5-(\Delta\omega_2+\Delta\omega_4)s_w$ (24)

Where Δw_4 is the specific humidity exchange in the desiccant wheel.

3.3 Mathematical analysis

Kurukshetra ambient conditions (indiaweather site) :

DBT = 43.3 ℃ RH = 20%

Indoor condition:

DBT = 25 ℃ RH = 50%

Ventilation mode

By taking theses condition calculate the values of DBT ,WBT, RH,W,H and S at each and every state by using above equations. The summary of calculation is shown in **Table 1**.

3.4 Recirculation mode

Similarly calculation can be done for each state. The summary of calculation is shown in **Table 2**.

$T_e=h_4-h_5+\Delta\omega3h_w/s_4-s_5+\Delta\omega3s_w$ (21)

$T_C=h_9-h_1-(\Delta w_1+\Delta w_2+\Delta w_3)hw/s_9-s_1-(\Delta w_1+\Delta w_2+\Delta w_3)S_W$ (22)

where, Δω1, Δω2, and Δω3 are the moisture added per kg of dry air in the ECs in process and regeneration lines and in the room, respectively. In another words, they are simply the increases in the specific humidity of the air as air flows through them. Also, hw and sw are the enthalpy and entropy of liquid water at ambient state.

In recirculation mode, the equivalent temperatures are given by

$T_e=(h_4-h_1+\Delta w_3h_w)/(s_4–s_1+\Delta w_3s_w)$

Using below table now calculate the qcool ,COP,COPC and COPrev in both mode.

Ventilation mode:

q_{cool} = (h_5-h_4) = 32.335 KJ/Kgeq-13

$COP=(q_{cool}/q_{regen})=(h_5-h_4)/(h_8-h_7)=1.000123$ eq-14

$COPC= (1-T_{amb} / T_{source}) (T_{space} / T_{amb}-T_{space})$

$= (1-T_1/ T_8) (T_5 / T_1-T_5)$

= 3.36 eq-17

$T_S= (h_7 –h_8)/(s_7-s_8)$

= 382.84K (or 109.84 ℃)

eq-20

$T_e=h4-h_5+\Delta\omega_3h_w/s_4-s_5+\Delta\omega_3s_w$
= 285.834 K (or, 12.82 ℃)

eq-21

STATE	DBT(℃)	WBT(℃)	RH	W(Kg/Kg dry air)	H(KJ/Kg)	S(KJ/Kg K)
1	43.3	24.11	20	.0110	72.055	.2517
2	96.191	30.07	0.01	0	96.94	.30400
3	17.88	4.72	0.01	0	17.989	.06379
4	4.72	4.72	100	.005320	18.094	.067710
5	25	17.88	50	.009927	50.429	.180748
6	17.88	17.88	100	.01289	50.628	.18195
7	94.361	35.577	2.48	.01289	129.589	.42279

8	125.550	40.166	.870	.01289	161.92	.50724
9	73.45	36.45	10.296	.02389	136.97	.4599

Table 1

STATE	DBT(℃)	WBT(℃)	RH	W(Kg/Kg dry air)	H(KJ/Kg)	S(KJ/Kg K)
1.	25	17.88	50	.009927	50.42	.180748
2.	72.317	24.604	0.01	0	72.83	.23652
3.	24.11	7.826	0.01	0	24.259	.0851108
4.	7.826	7.826	100	.0066048	24.476	.090546
5.	43.3	24.11	20	.0110	72.055	.25176
6.	24.11	24.11	100	.01909	72.837	.2574
7.	70.67	34.100	9.3338	.01909	121.408	.4092
8.	95.489	38.078	3.49	.01909	147.361	.48213
9.	49.4500	34.453	37.37	.029017	124.96	.42831

Table 2

$T_C=h_9-h_1-(\Delta w_1+\Delta w_2+\Delta w_3)hw/s_9-s_1-(\Delta w_1+\Delta w_2+\Delta w_3$

$= 307.81$ K(or, 34.81 ℃)

eq-22

h_w and S_w are enthalpy and entropy condition of ambient at liquid state (taken by steam table)

Similarly,calculation can be done for recirculation mode.

4. Result and discussion

The summary of result shown in below **Table 3**.

From **Figure 3** and **Figure 4** we observe that COP decreases as ambient temperature increases in both ventilation and recirculation mode and also cooling load increases in ventilation mode and decrease in recirculation mode. This is because first we take whole system work ideally and desiccant wheel completely adsorb moisture so because of this in ventilation mode minimum temperature obtained at point 4 and when we increased ambient temperature with respect to same relative humidity which means higher specific humidity at the inlet and so require higher re-generation heat. But in case of recirculation mode , regenerated air pass separately and remains constant and pro-cess air recirculated again and again so at state 4 temperature is higher and cooling load decrease.

Parameter	Ventilation Mode	Recirculation Mode
T_8(℃)	125.550	95.489
T_S(℃)	109.84	82.86
T_C(℃)	34.81	27.015
T_e(℃)	12.83	14.55
q_{cool}(KJ/Kg)	32.335	25.953
COP	1.00	1.12
COPc	3.36	2.306
COP_{rev}	2.260	2.56

Table 3

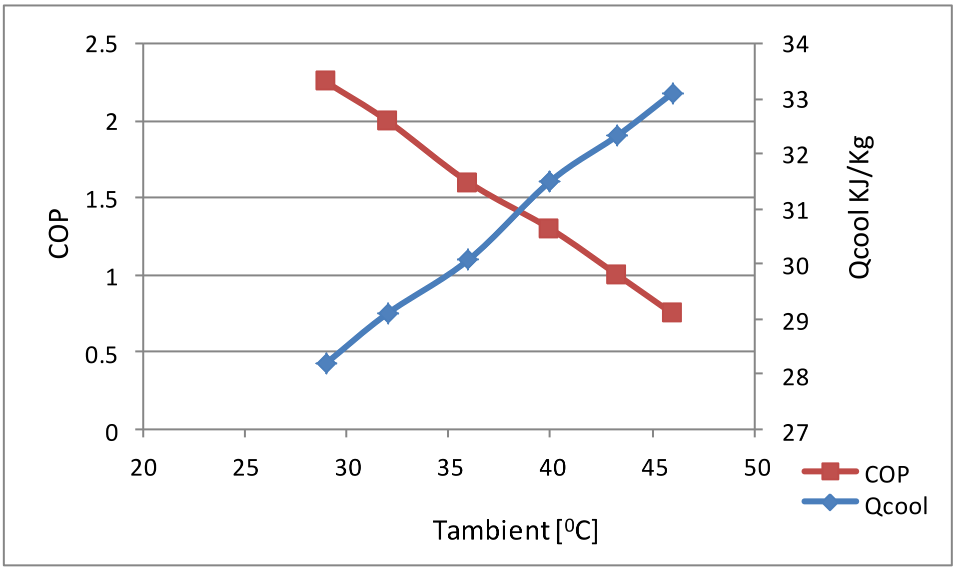

Figure 3. Ventilation mode (COPrev vs Tamb vs COPc)

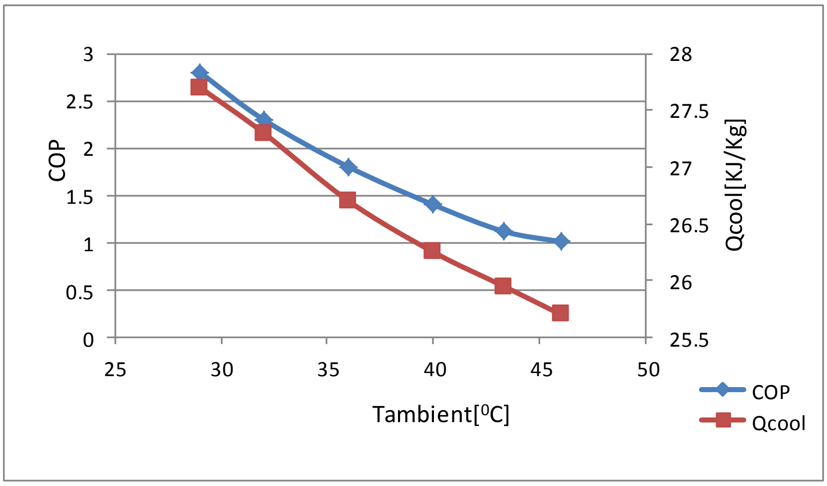

Figure 4. Recirculation mode (COP vs T_{amb} vs Q_{cool})

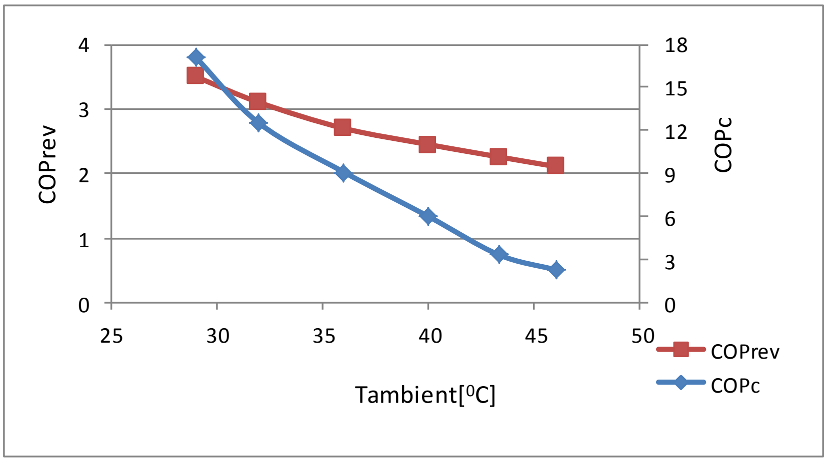

Figure 5. Ventilation mode (COP_{rev} vs T_{amb} vs COP_c)

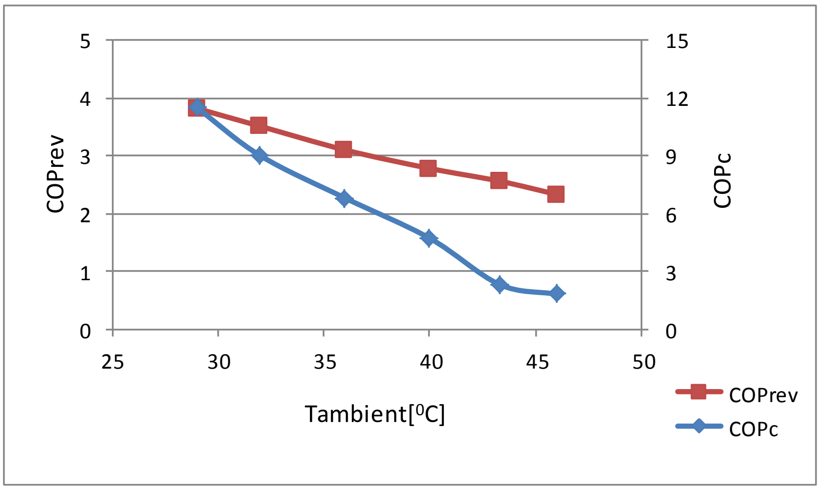

Figure 6. Recirculation mode (COP_{rev} vs T_{amb} vs COP_c)

Figure 5 and **Figure 6** we observe that both Carnot COP and reversible COP decreases as increase in ambient temperature. These trends can be understand by seeing trend of eq.17 and eq.18.

5. Conclusion

The thermal COP of the system gives the practical limit of system performance also the reversible COP represents the upper limit of the system and Carnot COP depends on temperature only and it represents the upper performance of closed system. The performance of system calculated when system's components worked as ideally but in future performance will be compare with experiment result.

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