ALTERNATIVE DESIGN OF HYBRID DESICCANT COOLING SYSTEMS

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ABSTRACT

Traditional vapor-compression refrigeration systems handle the air cooling and dehumidification in a single process. Depending on the characteristics of the thermal load, it is not possible to simultaneously meet the cooling and dehumidification requirements, resulting in under dried or overcooled supply air. Recent studies indicate that a combination of adsorptive dehumidification and vapor-compression might result in an improved system, regarding both energy consumption and air-quality. Accordingly, the present study aims at an alternative design of a hybrid system. It is shown that such a system can provide air at the same supply condition as a vapor-compression system, requiring as much as 40% less cooling capacity. Also, the proposed system works with a 100% of air renovation, being compatible with any air quality standard.

INTRODUCTION

Modern air-conditioning design practice aims at systems with low ambient impact and operating costs, while providing a comfort and healthy indoor environment. Desiccant cooling systems emerged as promising alternative to traditional vapor compression systems, since it allows for the use of solar energy or low grade waste heat for the reactivation of the desiccant rotor, in addition to the use of water as refrigerant. Moreover, desiccant systems often work with 100% of fresh air, thus providing maximum indoor air quality.

Recent studies, however, changed the perspective from competition to integration. In conventional air-conditioning systems, air cooling is simultaneously carried out with dehumidification. In humid environments the air is often overcooled to achieve the adequate humidity level, and sub sequentially re-heated so as to bring temperature back to a satisfactory supply condition, in wasteful process energy wise [1]. Accordingly, the combination of the desiccants and vapor-compression in a hybrid cycle has been increasingly studied. The main advantage is the thermal load decomposition into sensible and latent components, which are respectively handled by the cooling coil (vapor-compression cycle) and the desiccant rotor. Since the cooling coil has been unburned of the latent load, its refrigeration capacity is significantly downsized, when compared to a standard vapor-compression cycle.

Moreover, the cooling coil will be able to work at a temperature higher than the air dew point, which in turn allows for a refrigerant expansion at a higher temperature inside the cooling coil. As a consequence, the cycle coefficient of performance (COP) increases [2]. Also, the condenser waste heat can be harvested so as to partially supply the reactivation heat required by the desiccant cycle, which could be supplemented by a gas fired or solar source.

The hybrid cycle is usually designed with the desiccant rotor and the evaporator in an in-line arrangement, in which the air is sub sequentially dried and cooled, so as to achieve the desired supply condition. The present work presents a alternative arrangement for the hybrid cycle, with the desiccant rotor and the evaporator working in parallel rather than in series. It is shown that the proposed cycle is more versatile than the current hybrid cycle design, allowing for the accommodation of thermal load or outside air condition fluctuations with a simple air flow rate control.

The combination of a vapor compression and a liquid desiccant system was reported to increase the COP by as much as 50%, when compared to the standard vapor compression cycle alone [3]. This very same increase has been reported by an experimental investigation of the cycle [4]. An experimental setup of a desiccant wheel assisting a 6 kW vapor-compression cycle exhibited energy savings of 37.5%, compared to the standard operation [4]. A similar result (30% of energy savings) was reported by a simulation of a hybrid cycle which considered the desiccant wheel and the evaporator working in series and in parallel [5]. A close figure (26.3%) was found in an experimental setup for a mobile air-conditioning system [6]. An even greater energy saving (70%) was reported by a hybrid cycle driven by geothermal power [7]. Figure (1) shows the schematic representation of a typical hybrid desiccant system.

NOMENCLATURE

COMPR	[-]	Compressor
COND	[-]	Condenser
DW	[-]	Desiccant wheel
EVAC	[-]	Evaporative Cooler
EVAP	[-]	Evaporator
EXH	[-]	Exhaust
f	[-]	Desiccant fraction
h	[W/m^{2}C]	Heat transfer coefficient
h_a	[J/kg m]	Air specific enthalpy
HW	[-]	Heat Wheel

k	[kg/m^{2}S]	Mass transfer coefficient
R	[-]	Separation factor
REH	[-]	Reheater
RH	[-]	Relative Humidity
RLH	[-]	Room latent heat
RSH	[-]	Room sensible heat
RSHR	[-]	Room sensible Heat Ratio
T	[°C]	Temperature
Q	[kJkg]	Adsorption heat
Y	[-]	Air absolute humidity
V	[m^3/s]	Volumetric air flow rate
W	[-]	Solid absolute humidity

Special characters

λ	[-]	Non-dimensional parameter
ρ	[kg/m^3]	Air density

Subscripts

a	air
1,2,3	Air States
DW	Desiccant Wheel
EVAC	Evaporative Cooler
f	Fresh
HW	Heat Wheel
OA	Outside air
OA2	Air state
REH	Reheat
SA	supply air
RA	Room air
EVAP	evaporatot
rec	recirculated
t	total
w	wall

The system consists of a desiccant wheel, a regenerator, a sensible heat cooler and an evaporative cooler arranged in series. The outside air (OA) is admitted to the desiccant wheel, undergoing an isenthalpic dehumidification process. The regenerator then brings the air temperature back to the ambient level. The air is then cooled under the ambient level at a sensible heat cooler, in a constant humidity process. The cooling is promoted by a vapor-compression cycle. Finally, the air is admitted to an evaporative cooler, through which it suffers an isenthalpic humidification process, until it reaches the supply air condition. The main advantage of this scheme is that it handles the latent load and the sensible load in two separate processes. The desiccant wheel handles the outside air humidity, whereas the regenerator and the heat exchanger handle the sensible heat.

Since the desiccant wheel can be driven by low grade thermal energy (or even solar energy), it implies a significant operation cost reduction when comparing to vapor compression dehumidification, which is typically driven by electric power. This allows for the cooling capacity of the evaporator to be downsized, since it now exclusively handles the sensible component of the thermal load. In addition, the evaporator will work at a higher temperature (and thus pressure) than that of the standard cycle, which has to be kept below the air dew point. This benefits the COP, implying in further energy savings. Moreover, there is also the possibility to harvest the condenser rejected heat to pre-heat the air at the thermal source inlet, thereby mitigating the fuel consumption at the thermal source.

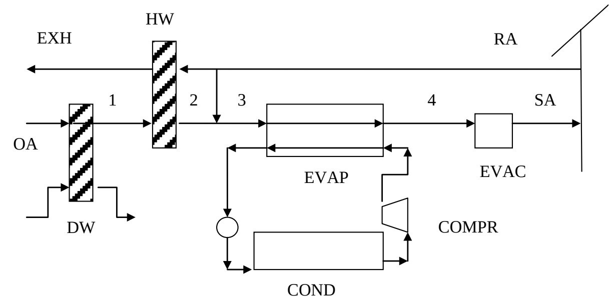

Figure 1 Typical Hybrid Desiccant Cycle

METHODOLOGY

Figure 2 show an alternative design of the hybrid desiccant system. In the proposed system, the outside air stream is split in two streams. One stream follows the route to the desiccant wheel, heat wheel and evaporator, whereas the other stream bypasses the desiccant wheel and is directed to a regenerative heat wheel. At the evaporator outlet the streams are mixed, and the air mix reaches the supply condition. The proposed design has two advantages over the typical hybrid cycle design. First, it lacks the evaporative cooler, allowing for diminished initial and operation costs. Second, it allows for the accommodation of any fluctuation of outside air conditions. For instance, should the outside air humidity be higher than usual, a greater air flow rate can be induced through the desiccant wheel. Conversely, for a higher outside air temperature, the bypass rate should be increased.

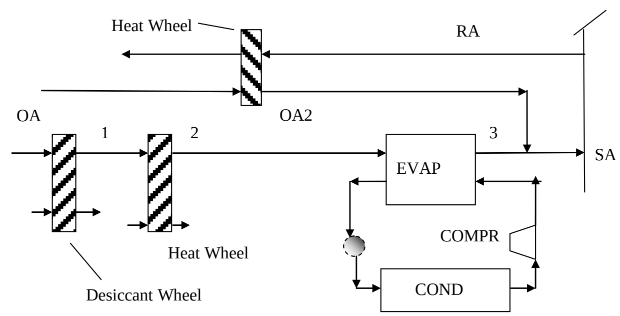

Figure 2 Alternative Hybrid Desiccant Cycle

The objective of the present work is to compare the energy demands of the proposed cycle with the traditional vapour compression cycle, shown in Fig (4). The comparison will be carried out for a case study, an application with characteristically high latent component of the thermal load. All calculations will be carried from the air side of the cycle, i.e., the vapor compression cycle simulation is out of the scope of the present work. Only the evaporator cooling capacity will be addressed, from the temperatures T_3 and T_2 and the air flow rate. In all cycles the exhaust stream is regenerated at a heat wheel (HW) so as to pre-cool the fresh air stream. The systems will be dimensioned for the case study depicted in Table (1) [8].

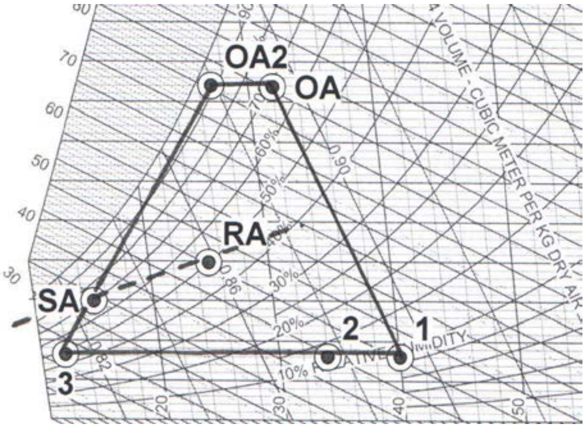

Figure 3 Psichrometric representation of the proposed hybrid cycle

The typical vapour-compression system will be dimensioned according to the usual air-conditioning design methodology [9,10]. Table (2) shows the air states, the total air flow rate, required reheat and cooling capacity for the evaporator.

Table (1): Case Study

Outside Air Conditions:	T_{OA}: 32.2°C RH_{OA}: 44.5%
Room Air Conditions:	T_{RA}: 25.9°C Y_{RA}: 50.0%
Fresh air flow rate:	V_{OA}: 1.18 m^3/S
Room Sensible Heat:	RSH: 35.16 kW
Room Latent Heat:	RLH: 19.05 kW

Table (2): Typical Vapour-compression cycle

	T(°C)	Y(Kg/Kg)	h(KJ/kg)
State OA	32.2	0.0136	67.50
State 1	26.9	0.0105	53.70
State 2	9.8	0.0073	28.14
State SA	15.1	0.0073	33.89
State RA	24.9	0.0093	49.13
V_t (m^3/s)	3.23		
Q_{reh} (kW)	21.00		
Q_{cool} (kW)	103.00		

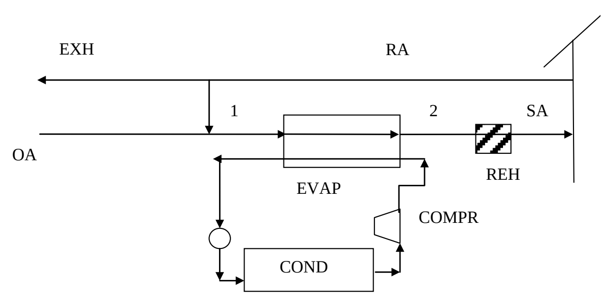

Figure 4 Typical vapour-compression cooling cycle

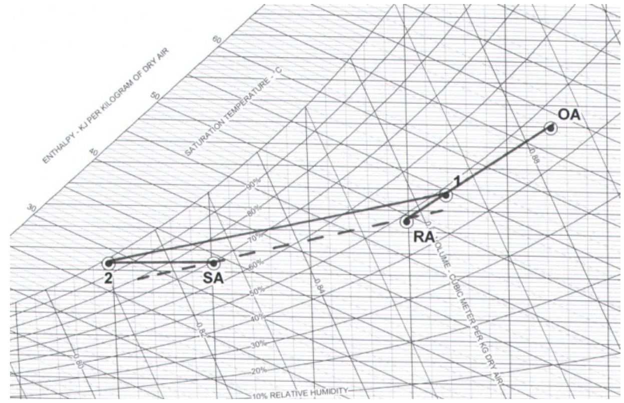

Figure 5 Psichrometric representation of the vapour-compression cooling cycle

While the methodology for sizing the standard compression air-conditioning system is straightforward and can be found in any HVAC handbook, the dimensioning of the hybrid cycle requires a graphical procedure, along with effectiveness values for the heat exchangers. The desiccant wheel outlet state can be obtained by running a desiccant wheel simulation:

1. Set the OA state and the heat wheel effectiveness, and use Eq.(1) to determine T_{OA2} . State OA2 is then defined, since $Y_{OA2}=Y_{OA}$.
2. State 1 is obtained as the output of the desiccant wheel simulation program, after setting the regeneration temperature.
3. Set 1 state and the heat wheel effectiveness and use Eq.(1) to determine T_{OA2} . State OA2 is then defined, since $Y_2=Y_1$.
4. Set T_{evap} and the evaporator effectiveness and use Eq.(3) obtain T_3, with $Y_3=Y_2$.
5. Determine the supply air state (SA) at the intersection of line 3-OA2 with the RSHR line.
6. With states SA and RA defined, calculate the total volume flow rate V_t according to Eq.(4).

Figure (3) shows the psichrometric representation of the cycle. The resulting air states, flow rates, required heat and cooling capacity are shown in Table (3). The following equations are used in the methodology:

$$\varepsilon_{HW} = \frac{T_{OA} - T_{OA2}}{T_{OA} - T_{RA}} \quad (1)$$

$$\varepsilon_{HW} = \frac{T_1 - T_2}{T_1 - T_{OA}} \quad (2)$$

$$\varepsilon_{EVAP} = \frac{T_2 - T_3}{T_2 - T_{EVAP}} \quad (3)$$

$$\frac{RSH + RLH}{\rho\left(h_{RA} - h_{SA}\right)} = V_t \quad (4)$$

Details of the mathematical model development and validation of the desiccant wheel simulation are widely available in the literature. [11-14]. In short, as shown in Fig. (6), energy and mass balances are applied to elementary control volumes enclosing the sorbent layer and the flow channel, resulting in equations (5) to (8), and an algebraic equation (9) which relates the adsorptive capacity as a function of the vapor pressure in the pore vicinity, known as isotherm. The parameter R is a measure of the strength of adsorption of a particular desiccant material.

$$\frac{\partial Y^*}{\partial x^*} = \lambda_3 \left(Y_w^* - Y^* \right) \tag{5}$$

$$\frac{\partial W}{\partial t^*} = \lambda_2 \left(Y^* - Y_w^* \right) \tag{6}$$

$$\frac{\partial T_a}{\partial x^*} = T_w - T_a \tag{7}$$

$$\frac{\partial T_w}{\partial t^*} = \left(T_a - T_w \right) + \lambda_1 \left(Y^* - Y_w^* \right) \tag{8}$$

$$\frac{W}{W_{\max}} = \frac{1}{(1 - R + \frac{R}{\phi_w})} \tag{9}$$

$$\lambda_1 = \frac{Q}{\left(\frac{\partial h_a}{\partial T_a} \right)} \tag{10}$$

$$\lambda_2 = \frac{C_{wr}}{f \left(\frac{\partial h_a}{\partial T_a} \right)} \tag{11}$$

$$\lambda_3 = \frac{k}{h} \left(\frac{\partial h_a}{\partial T_a} \right) \tag{12}$$

Tables (2) and (3) show the same SA states and same total air flow rates for both cycles, which imply the same condition of thermal comfort. However, the proposed cycle works with a 100% of fresh air, whereas table (1) shows that the vapor compression cycle runs with nearly 60% of recirculated air, resulting in a poorer indoor environment quality wise. Also, it can be seen that the required cooling capacity (chiller size) on the Hybrid cycle is 40% less when compared to that of the vapour compression cycle. Even though it handles a greater amount of outside air.

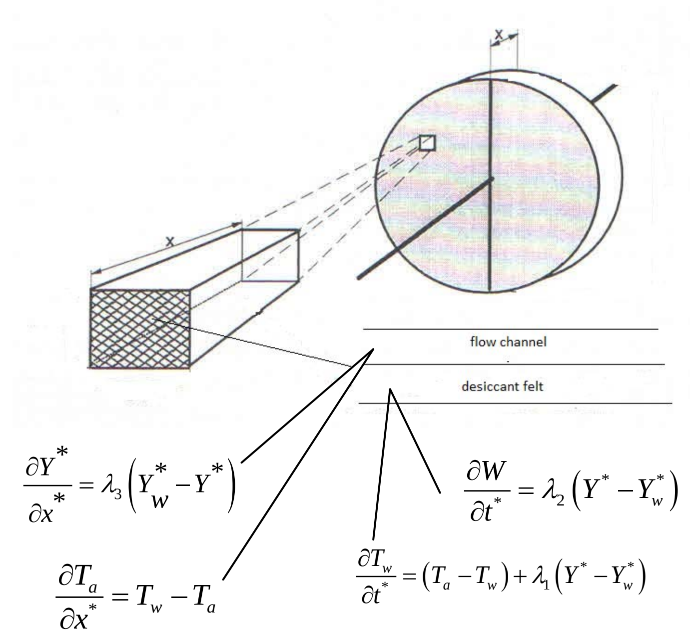

Figure 6 Computational domain (desiccant wheel)

Table (3): Proposed Hybrid cycle

	T(°C)	Y(Kg/Kg)	h(KJ/kg)
State OA	32.2	0.0136	67.50
State OA2	27.0	0.0136	61.80
State 1	55.0	0.0047	67.50
State 2	34.0	0.0047	44.42
State 3	9.0	0.0047	20.90
State SA	15.1	0.0073	33.89
State RA	24.9	0.0093	49.13
V_t (m^3/s)	3.23		
Q_{DW} (kW)	140.00		
Q_{cool} (kW)	60.50		

Caution should be taken when comparing the heat requirements. Table (2) indicates a reheat demand of 21kW, while Table (3) indicates a heat demand of 140kW on the desiccant wheel reactivation. Although the heat requirement of the vapor compression is much lower, one has to consider that the reheat is accomplished using electrical heaters, whereas the desiccant wheel may be reactivated using solar power or waste heat. Even compressor waste heat can be use to pre-heat the reactivation air, which can be supplementary heated with gas so as to reach the required reactivation temperature.

A meaningful comparison on the energy consumption and operating costs of each cycle would require a simulation of the vapor compression cycle, so as to account for the compression work in each case. This matter is however beyond the scope of the present study and shall be addressed in a future effort.

CONCLUSION

Modern HVAC design aims at low-energy systems which are able to provide indoor ambient with improved air quality. An alternative design for a Hybrid vapor-compression system has been presented, the performance of which has been compared to a traditional vapor-compression cycle. It has been shown that the proposed design can provide the same thermal comfort as the vapor-compression cycle, using less electrical power with total air renovation, in contrast with standard systems which recirculates as much as 2/3 of the total air flow. Also, the proposed system lacks an evaporative cooler, in opposition to the current Hybrid system design.

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