



encit 2020



18th Brazilian Congress of Thermal Sciences and Engineering
November 16-20, 2020, (Online)

ENC-2020-0312

AN ALTERNATIVE HYBRID DESICCANT-COMPRESSION SYSTEM FOR AIR CONDITIONING APPLICATIONS WITH FULL AIR RENOVATION

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Abstract. *The quest for low energy buildings has brought the design acclimatization systems under special attention, since they can be accountable for as much as 70% of the overall energy consumption in tropical climates. At the same time, indoor air quality has been of increasing concern since the Legionnaire's disease outbreak in 70's. Traditional single zone systems heavily rely on air recirculation, as the cooling capacity prohibitively grows with the amount of fresh air admitted to the systems. Accordingly, the present work addresses an Hybrid Cycle for a single zone air conditioning system, which combines vapor compression cooling with air drying through a desiccant system. Results show the proposed system is able to provide the same comfort condition as the vapor compression system, with 100% of fresh air volume. This condition is achieved at the cost of a significant amount of low grade heat, which can be accomplished with a waste heat source or solar collectors.*

Keywords: *Desiccants, Vapor Compression, Hybrid Cycles*

1. INTRODUCTION

Evaporative cooling systems use water as refrigerant, thereby offering a harmless alternative to traditional vapor compression systems, which often use refrigerant substances associated to ozone layer depletion and global warming. Moreover, evaporative systems require the use of fans and pumps, which demand much less power than alternative or centrifugal vapor compressors, thereby providing an overall smaller ecological footprint. The cooling effect achieved by evaporative systems is, however, highly dependent on the environmental conditions, since it relies on a low relative humidity. Even for low relative humidity levels, the air temperature drop is not as significant as in vapor compression systems, which can be compensated (to a certain degree) by a greater air volume. Accordingly, the application evaporative system is usually confined to high air flowrate applications, such as sports arenas and in dry climates (McQuiston et al., 2004). One possible way to circumvent this limitation is by the use of solid and liquid desiccants, to artificially dry the air stream before its admission to the evaporative systems, increasing the evaporative cooling potential. Vapor compression systems, dehumidification and air cooling occur simultaneously, in a single process, which often leads to inadequate temperature or humidity levels. For instance, if the process is controlled by a thermostat, the compressor shuts as soon as the room design temperature is met, allowing for the air humidity to be oblivious to the fan coil, or even carrying out some extra humidity resulting from undrained condensate. Even if there is humidity control, sometimes the air stream needs to be overcooled to reach the adequate supply humidity level, and has to be subsequently re-heated in order to bring temperature back to the supply condition, in wasteful process energy wise (Threlkeld, 1998). Accordingly, modern engineering design has switched the perspective from competition to integration. The idea is to use both desiccant and vapor-compression on a single cycle, thereby allowing for the thermal load to be decomposed into its latent and sensible components. A COP increase of as much as 50% for a combination of a vapor compression and a liquid desiccant has been reported, when compared to the standard vapor compression cycle alone (Dai et al., 2001). The same figure has been obtained by an experimental set up of the cycle (Jia et al. 2003). An empirical work, consisting of a desiccant wheel assisting a 6 kW vapor-compression cycle showed a lesser energy demand of as much as 37.5%, when compared to the standard operation (Ling et al., 2010). A reduction within the same order of magnitude (26.3%) was measured in a prototype for a automotive air-conditioning system (Lee et al., 2016). An even greater energy saving (70%) was reported by a hybrid cycle driven by geothermal power (Casas et al., 2005).

2. THE STANDARD SINGLE ZONE CYCLE: DESIGN PROCEDURE AND ISSUES

Figure (1) shows the schematic representation of a standard single-zone air conditioning system. Fresh air is admitted at state OA to the system, and mixed with a given amount of recirculated air (state RA), resulting in air state EA. Figure (2) shows the psychrometric representation of the cycle.

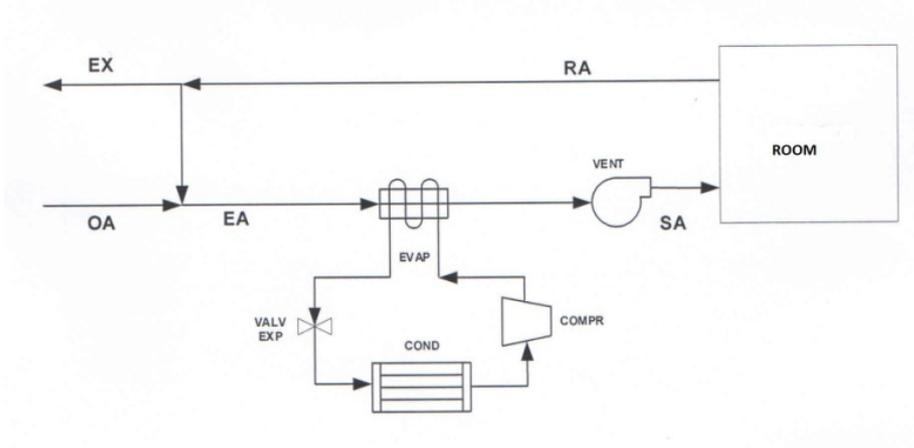


Figure 1: Schematic of a single zone air-conditioning system

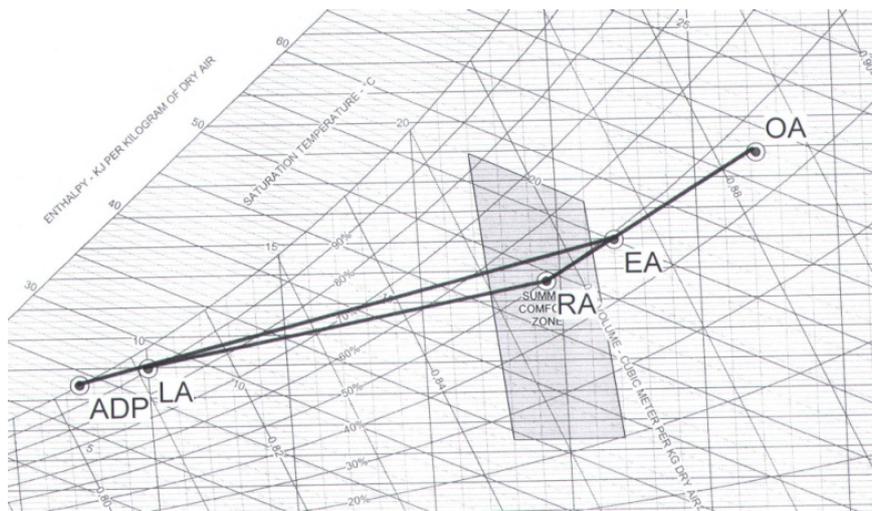


Figure 2 Psychrometric representation of of a single zone air-conditioning system

The air states subscripts refer to,

- ADP apparatus dew point
- LA leaving aparattus
- EA entering apparatus
- OA outside air
- RA room air

The design procedure of a single-zone air-conditioning system can be described with the aid of Fig.(2) (Pitta, 2002). States RA and OA are previously set, along with the inclination of the process line through RA (the process line represents the 'heat gain' of the air stream through the conditioned room). State ADP is chosen over the saturation curve, at the lowest temperature possible value (so as to increase compressor performance) and yet not so low as to avoid condensate freezing at the coil. State EA is chosen over the line RA-OA, the choice depending on the desired air

renovation rate. The closer EA lies to RA, the greater the air recirculation (and energy savings). Conversely, the closer EA lies to OA, the greater the air renovation and the required cooling power. Once state EA has been established, a the line EA-ADP intersects the process line at the LA states, which defines the room supply conditions. Should these lines do not intersect, the supply air will require some reheating, as described in the introduction. This process is represented in Fig. (3), by the line 2-SA. This often happens by one of two reasons. First, process line SA-RA might be too steep, as a indication of a thermal load with a dominant latent component (i.e., highly humid ambient). Second, increased air renovation rate would bring point 1 closer to state OA, thereby increasing line 1-2 steepness and preventing it to intersect process line SA-RA. The sensible heating indicated by line 2-SA is usually accomplished by an electrical heater

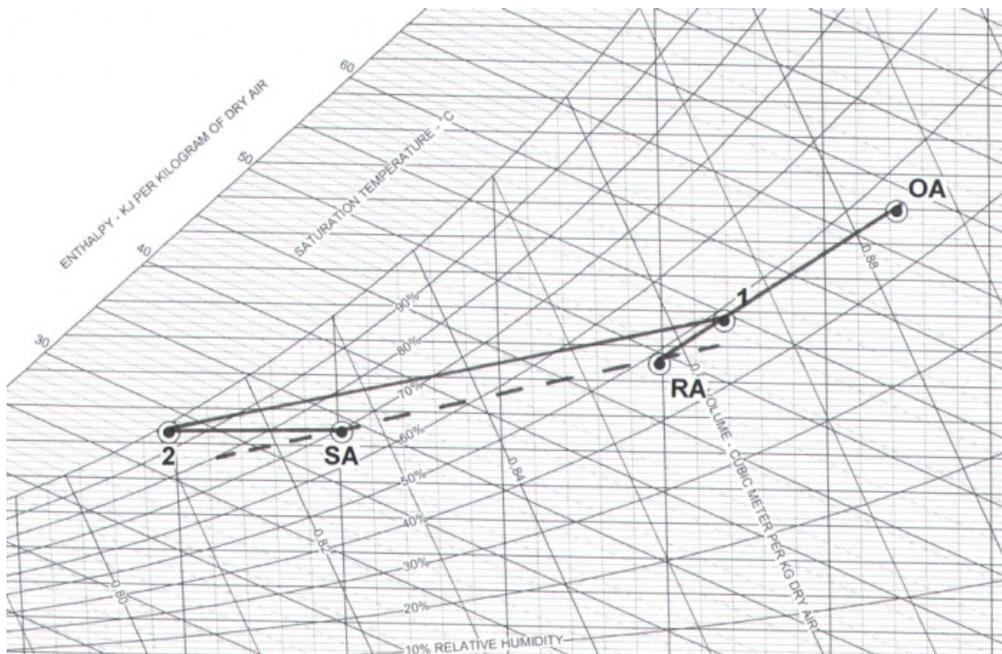


Figure 3 Psychrometric representation of of a single zone air-conditioning system with reheat

Although the increase concern over the last 40 years with indoor quality, the air recirculation rate in commercial business still lies in the range of 75 to 85% in most constructions standards, relying on active and passive air filtering techniques to mitigate the action of indoor pollutants and chemical emissions from cleaning products or printing machines, for instance. Increased air renovation rates are typical of intense human activities, such as dance halls or sports arenas, and can reach as much as 100% for hazardous environments such as industrial and medical facilities. The cooling capacity is proportional to the enthalpy difference between states 1 and 2. As illustrated, as point 1 approaches the fixed point OA, the greater will be the air renovation at the cost of an increased cooling capacity and reheat power. Thereby, a 100% air renovation would require prohibitively high cooling and reheat capacities. Accordingly, energy savings and air quality exhibit conflicting demands, challenging the design to meet both regulation guidelines.

3. THE PROPOSED HYBRID CYCLE

Figure (4) show an schematic of the proposed hybrid desiccant system. The outside air stream is split in two streams. One stream is directed to to the desiccant wheel, heat wheel and evaporator, whereas the other stream is diverted to a regenerative heat wheel. At the evaporator outlet the streams are reunited, and the air mix reaches the supply condition. The purpose of this bypass is to take advantage of outside air state fluctuations, allowing for the use of outside air to its maximum allowance. For a better understanding of the cycle, one should refer to its psychrometric representation, depicted in Fig.(5). The cycle starts by harvesting outside air at state OA, which is split into two streams. One stream is diverted to a heat wheel, through which is cooled to state OA2. The other stream is diverted to a desiccant

wheel, through which is heated and dehumidified, process OA-1. Sub sequentially; the air stream is regenerated through a heat wheel to state 2, close to its original outside temperature (but less humid). Then the air stream is sensibly cooled in process 2-3. Important to notice that in this case, the air through the evaporator does not need further dehumidification. Accordingly, it would require much less cooling capacity (a smaller compressor), since the evaporator would be unburdened from the latent load. Moreover, the compression cycle does not need to operate below the air dew point, allowing for a greater coefficient of performance COP. Once it has reached state 3, the air stream can be finally mixed to state OA2, resulting in state SA (supply air), which is supplied to the room, at the very same condition of the standard cycle.

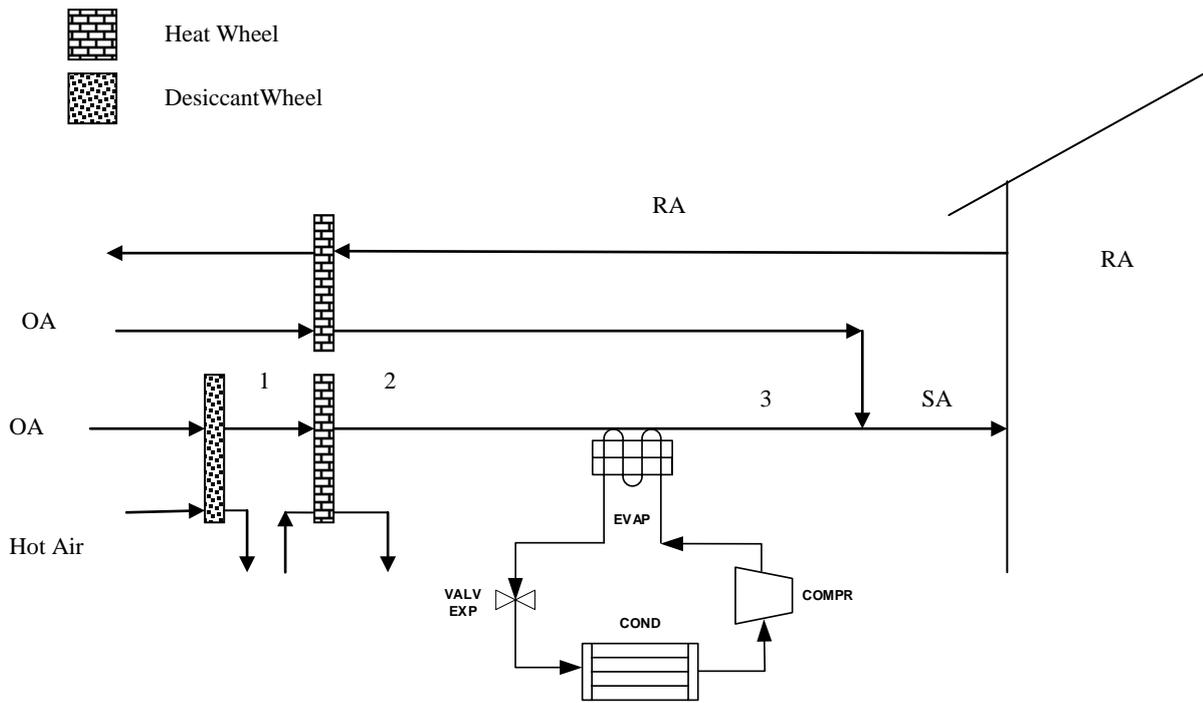


Figure 4 : Schematic of the proposed Hybrid Cycle for of a single zone air-conditioning system

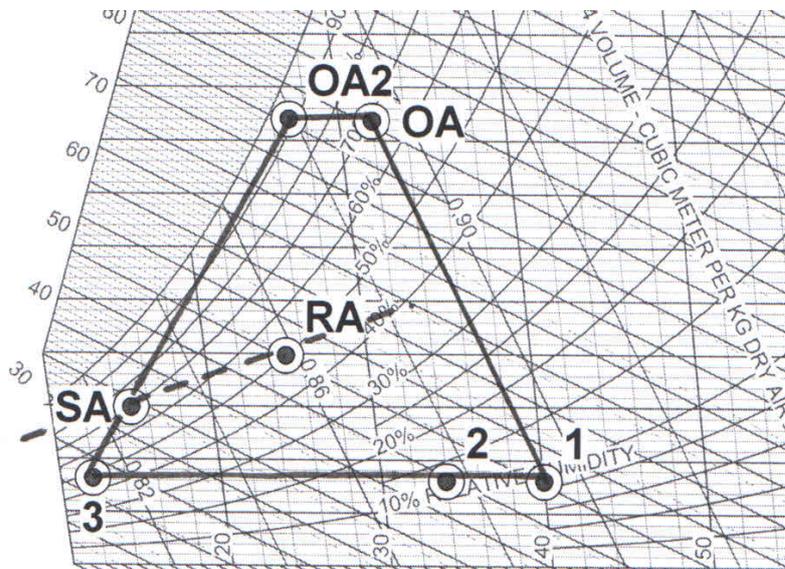


Figure 5: Psychrometric representation of the proposed Hybrid Cycle

In order to establish an unbiased comparison, a case study will be submitted to both design procedures, for the standard vapour compression and the proposed hybrid cycle. Room condition, thermal load and outside air condition OA will be the same for both cases. The analysis aims at comparing both cycles in terms of the energy demands and air quality.

Table (1): Case Study

Outside air condition (OA)	$T_{OA}:32.2^{\circ}\text{C}$, $RH_{OA}:44.5\%$
Room air condition (RA)	$T_{RA}:25.9^{\circ}\text{C}$, $RH_{RA}:50.0\%$
Minimum Ventilatio Rate	$V_{OA}=1.18\text{m}^3/\text{s}$
Room Sensible Heat (RSH)	35.2kW
Room Lensible Heat (RLH)	19.1kW

Table (2): Typical Vapour-compression cycle air states

	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		

While the methodology for sizing the standard compression air-conditioning system has been described and can be found in further detail any HVAC handbook, the dimensioning of the hybrid cycle requires a graphical procedure. This can be accomplished by pre-setting typical effectiveness figures for the heat, which are readily available in the literature. Ideally, the air state at the desiccant wheel outlet would require the numerical simulation of heat and mass transfer at the desiccant channels. However, it has been shown (Nobrega and Brum. 2011) that the adiabatic adsorption is essential an isenthalpic process, thereby allowing for the air state outlet to be determined over the isenthalpic line between states OA and 1, provided some pre-set typical value for the dehumidification effectiveness is provided. Such values are available from the technical literature and manufacturer catalogs

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 determined by the desiccant wheel effectiveness, over the isenthalpic line, using Eq.(2)
3. Set 1 state and the heat wheel effectiveness and use Eq.(3) to determine T_2 . State 2 is then defined, since $Y_2=Y_1$.
4. Set an adequate value for 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 room process line.
6. With states SA and RA defined, calculate the total volume flow rate V_t (m³/s) according to Eq.(5), where h stands for air enthalpy (kJ/kg) and ρ for air density (kg/m³).

Figure (3) shows the psychrometric 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:

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

$$\varepsilon_{DW} = \frac{Y_{OA} - Y_1}{Y_{OA}} \quad (2)$$

$$\varepsilon_{HW} = \frac{T_1 - T_2}{T_1 - T_{OA}} \quad (3)$$

$$\varepsilon_{EVAP} = \frac{T_2 - T_3}{T_2 - T_{EVAP}} \quad (4)$$

$$\frac{RSH + RLH}{\rho(h_{RA} - h_{SA})} = V_t \quad (5)$$

Table (3): Proposed Hybrid cycle air states and energy demands

	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		

The values for the heat wheels effectiveness were set to 75%, whereas the desiccant wheel dehumidification was set to 23%. Both figures are easily achieved by commercial equipment. 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 (2) 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 vapor compression cycle, even though it handles a greater amount of outside air. 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 for the desiccant wheel regeneration. Although the heat requirement of the vapor compression is much lower, it has to be considered that the reheat is promoted by electrical resistances, whereas the desiccant wheel may be reactivated using low grade heat. Depending on the chosen desiccant, temperatures as low as 60°C is admissible for regeneration ((Nóbrega and Brum, 2014). Accordingly, a variety of sources such as solar power or engine waste heat are to be considered. Even condenser 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 thorough comparison on the energy requirements and economical performance between cycles would require a simulation of the vapor compression cycle, so as to account for the compression work in each case. An important feature of the hybrid cycle is the dry fan coil (i.e., the refrigerant evaporating above the air dew point), allowing for even further power consumption. This subject is, however, beyond the scope of the present study and is to be addressed in a future effort

4. CONCLUSION

The demands for low energy and good air quality exhibit some conflict of interest, as far as the amount of air renewal is of concern. While increased ventilation rates benefit the air quality, the extra amount of air has to be brought to supply room conditions, thereby increasing the required cooling power and energy consumption. The proposed hybrid desiccant-vapor compression cycle offers the opportunity of decoupling the thermal load in its sensible and latent components, thereby allowing for a more precise control of supply air temperature and humidity. Moreover, it has been shown to work with as much as 40% required less cooling power than the standard cycle, while providing 100% of air renewal at the same comfort conditions, under the same thermal load. The extra low grade thermal energy required for desiccant wheel regeneration is somewhat significant, but can be provided by solar energy or waste heat harvest.

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