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Research paper

A DOUBLE EFFECT THERMAL DRIVEN AIR CONDITIONING SYSTEM USING AMMONIA/WATER ABSORPTION AND DESICCANT EVAPORATIVE COOLING

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ABSTRACT

A thermal driven air-conditioning system, worked out coupling two sorption cycles, is presented. The goal is to develop a thermal driven primary air conditioning system, which is energy efficient, compact, reliable and without an auxiliary heat rejection system. The motivation is to reduce the non renewable primary energy consumption of the air-conditioning sector and to increase the share of thermal driven system. The system integrates an absorption heat pump cycle and a solid desiccant evaporative cooling open cycle (DEC). The system main feature is the ability of the absorption heat pump to drive a DEC, creating a double effect thermal driven cycle. After a compatibility analysis on the temperature levels an ammonia/water absorption heat pump and a silica-gel desiccant wheel have been selected. The solid DEC cycle is implemented in a double duct air handling unit. In one duct the building supply air flows, while in the other the exhaust air from the building flows. In cooling and dehumidification operation mode, the heat pump has two useful effects: it delivers the heating power needed to the desiccant based dehumidification process and it provides cooling power for the supply air flow. Moreover, the system is able to work also in heating mode. The hydraulic connections between the DEC air handling unit and the heat pump is inverted, and the latter is used to heat up the supply air, recovering energy from the indoor exhaust air. In order to assess the energy performance a numerical model of the system has been created. It has been tuned with experimental results and completed with the definition and implementation of a control strategy.

The ratio between the net supply air energy exchange and the non-renewable primary energy used by the system has been used as the main performance figure (PER). The PER comparison with a conventional air-conditioning system (using an air source electric heat pump and a gas boiler) has shown a better energy performance of the presented system. It is particularly suitable in areas where the natural gas is cheap and/or the electric energy generation and distribution system is weak.

Keywords: Air-conditioning; sorption; adsorption; absorption; DEC; desiccant wheel; ammonia/water; GAX; modelling.

Introduction

The demand of buildings air-conditioning is growing, due to higher external temperature and people thermo hygrometric comfort requirement. Moreover, the current construction sector trend is to design buildings with reduced energy requirements for space heating and cooling using thermal insulation and passive design techniques (shadings, natural ventilation). At the same time there is the need for an indoor air quality control, through the use of heating ventilation air-conditioning (HVAC) systems. Thus, the ratio between the energy requirement for the treatment of the fresh air and the total energy requirement is increasing, particularly for end-users typologies such as: offices, light commercial, libraries and conference rooms. Furthermore, the cooling season for non-residential buildings is generally longer, due to higher internal heat gains (people and appliances).

Today, most of the air-conditioning systems are based on electric driven vapour compression technology. The concern for the increasing electricity peak demand due to air-conditioning as well as the need to reduce the cooling primary energy needs are pushing towards new technological solutions.

In the research project here presented a double effect thermal driven air conditioning system for primary air is defined. It aims to increase the energy efficiency of the dehumidification process and to reduce drastically the air-conditioning electric energy demand switching to thermal energy. It is based on the integration of two sorption cycles: an absorption heat pump and a solid desiccant evaporative cooling system (DEC). The absorption heat pump cycle is the first effect, at the highest pressure and temperature levels, which drives the second effect that is a solid DEC cycle.

The research object is based on two main technologies: desiccant evaporative cooling open cycle and the absorption cycle. Hereafter, a brief technological background of these two technologies is reported.

The so-called DEC (Desiccant Evaporative Cooling) is an air-conditioning system based on the use of a desiccant process combined with an evaporative cooling process (Daou et al., 2006)

A desiccant process consists in the absorption or adsorption of water vapour by a substance, which can be natural or synthetic. The driving force is the water vapour pressure difference between the surroundings moist air and the desiccant surface. Desiccant materials are primarily classified according to their states, liquid or solid, at operating condition.

In the air-conditioning sector the most common use of solid desiccant materials is as coating of rotating wheel, usually called desiccant wheel. The desiccant wheel rotates slowly (order of magnitude 10 RPH) in a special containment structure which is divided, at least, into two areas. The first area is used to adsorb the water vapour of process air, and the other it is used to ensure a continuously operating cycle regenerating the rotor by removing the water uptake through an auxiliary hotter air flow (regeneration air).

In the last few years many research works have dealt with solid DEC cycle for air-conditioning (Daou et al., 2006; Henning, 2007; Beccali et al., 2009; Frein et al., 2015). The reason is that a DEC cycle could be activated by low temperature heat like solar heat, heat from heat pump and waste heat. Under this condition a DEC cycle could become competitive and cost effective.

About absorption cycles, they are heat-activated thermodynamic cycles which can be used for refrigeration, air conditioning or heat pumping. The main advantages in comparison to other cycles are: no vapour compression, no mechanical compressor and low temperature sources can be used. Absorption machine useful power range from small units with an order of magnitude of 10 kW to huge units of few MW.

Among several available refrigerant/absorbent pairs the most used are water/ lithium bromide and ammonia/water which offer the best compromises of thermodynamic performance and have zero GWP (Global Warming Potential).

1. System description

The proposed air-conditioning system is mainly composed by a desiccant wheel based DEC and a gas fired absorption heat pump cycles (GAHP) (Figure 1). The result is a double effect thermal driven air-conditioning system. The absorption heat pump cycle is the first effect, at the highest pressure and temperature levels, which drives the second effect that is the solid DEC.

The key aspect of this system is that, when air dehumidification is needed, both the side

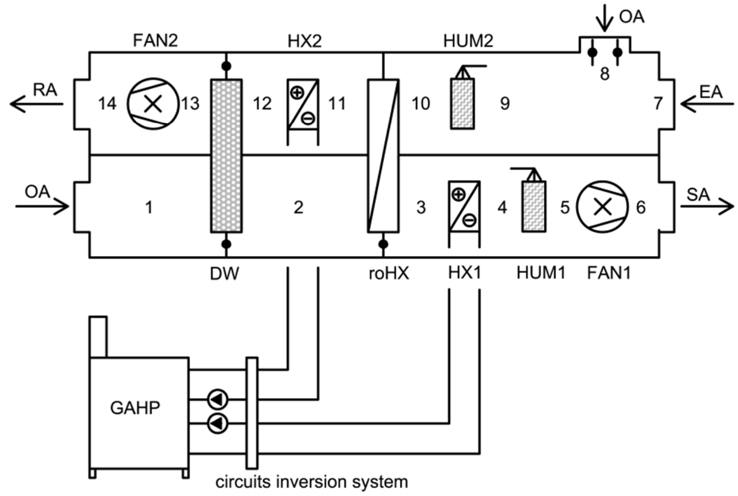

Figure 1 - System scheme.

of the heat pump are useful effect. Indeed, the heat pump provides both, the heating power needed by the dehumidification process (points 11-12, Figure 1), and the sensible cooling power needed to post-cool the supply air stream (points 4-5) at the exit of the DEC process.

For the choice of the thermal driven heat pump technologies among the available ones, the following constraints have been taken into account:

- deliver, at the same time, hot water at a temperature ≥65 °C (for DW regeneration) and cold water at a temperature of around 10 °C (for supply air post-cooling);
- needless of an auxiliary rejection heat at the previous point condition;
- output thermal power suitable for a light commercial application (around 35 kW);
- available on the market, in order to be able to perform experimental tests in a short time horizon.

Thus, one of the most spread thermal driven heating system in the world has been selected. It is a gas driven absorption heat pump, implementing a absorber generator heat exchange cycle (GAX) and using ammonia-water has working pair. It is optimized for traditional space heating and cooling applications. It has a minimum outlet temperature of -5 °C at the evaporator and a maximum outlet temperature of 65 °C at the condenser/absorber.

Starting from the working temperature range and power capacity of the selected absorption heat pump, a general layout of the system has been defined (Figure 1) taking into account several requirements like: useful cooling and heating power, technological feasibility, energy efficiency, initial cost, reliability, number of components minimization, low electric energy demand.

As the maximum available regeneration temperature is around 60 °C we have decided to use a silica gel based desiccant wheel which is able to work with relatively low regeneration temperatures.

From the thermodynamic point of view, in the standard solid DEC cycle a significantly low humidity by mass at the desiccant wheel process side outlet is the precondition for the use of direct evaporative cooling stage. In the system under analysis, the limited dehumidification capacity, due to the limited regeneration temperature, it does not allow the use of a direct evaporative cooling stage. However, the latter is replaced by a cooling coil fed by the cooling power made available by the heat pump, thus the limited dehumidification capacity of the wheel can be used only for covering the latent load.

Another peculiarity of the system is the availability of a mixing section for the exhaust air with external air (points 7 and 8, Figure 1). This is due to solve the problem linked to the imbalance between the supply and the exhaust air flow rate, which is needed for keep the indoor pressure higher than the outdoor, limiting air infiltration. Otherwi-

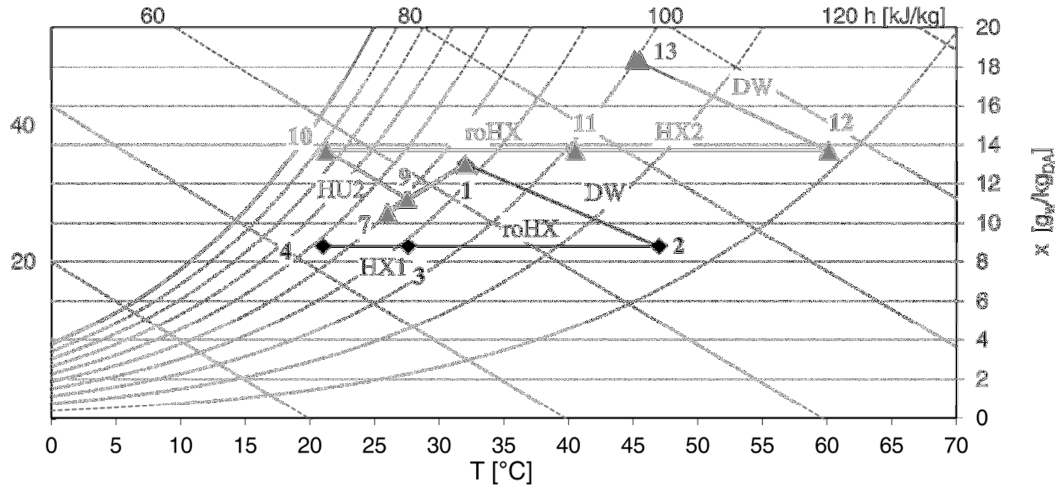

Figure 2 - System thermodynamic processes on psychrometric chart for cooling and dehumidification operation mode.

se, this imbalance in the air flow rate limits the maximum thermal power exchangeable by the energy recovery wheel, limiting the indirect evaporative cooling potential. This aspect does not affect the performance of the system in heating mode thanks to the contribution of the heat pump, so the mixing section is not used. On the other hand, when dehumidification is required, this limitation on the indirect evaporative cooling stage is quantitatively important and could not be compensated only with the heat pump operation. Therefore, in dehumidification mode, the process air flow rate is increased up to the regeneration air flow, adding an external air flow rate.

Moreover, the system is able to work also when the supply air has to be heated and humidified. For this reason, there is an adiabatic humidifier at the end of the process side air duct (points 4-5), which is used only in this case. In this operation mode, the heat pump is used to heat up the supply air, recovering energy from the indoor exhaust air. Practically, it is implemented inverting the hydraulic connections between the DEC's coils and the heat pump evaporator and condenser/absorber inlets and outlets (*circuits inversion system*, Figure 1).

Figure 2 shows, in a psychrometric chart, the complete sequence of the supply and return air thermodynamic processes which occur inside the DEC air handling unit when cooling and dehumidification are needed.

It follows, a brief description of the supply air process (point 1 to 4, Figure 2) and of the return air (point 7 to 13) for a cooling and dehumidification case (the humidifier at the supply side is off):

- the process 1-2 is linked to the process 12-13. They are the consequence of the air dehumidification by adsorption (Ruthven, 1984). Between point 1 and 2 the supply air is heated up by the exothermic water vapour adsorption process. The goal is to guarantee the supply air humidity by mass set-point;
- the process 2-3 is linked to the process 10-11. They are due to the heat exchange through the rotary heat exchanger between the supply air and the return air;
- the process 3-4 is due to sensible cooling through the cooling coil on the supply air flow (HX1). The cooling coil is driven by the evaporator side of the heat pump. The goal is to increase the cooling effect of the DEC cycle;
- the process 9-10 on the return side is an adiabatic evaporative cooling through a wetted media humidifier. It is part of the indirect evaporative cooling process, which is completed with the process 2-3;
- the process 11-12 is a sensible heating of the return air for the regeneration of the desiccant wheel. The heating coil (HX2) is linked to the condenser/absorber side of the heat pump.
- the process 12-13 is the regeneration of the desiccant wheel. The water content of the desiccant wheel is desorbed and released to the return air flow.

Table 1- Active components for each operation mode.

	FAN_1	FAN_2	ro_{HX}	BP_{RoHX}	HUM_1	HUM_2	DW	BP_{DW}	GAHP +aux
DEC+AHP	X	X	X	X		X	X		X
IEC+AHP	X	X	X	X		X		X	X
IEC	X	X	X	X		X		X	
VENT	X	X		X				X	
HR	X	X	X	X				X	
HR+AHP	X	X	X	X				X	X
HUM+AHP	X	X	X	X	X			X	X

2. Control strategy and operation modes

The air handling unit has to guarantee the supply air set-point conditions set by the end-user adapting the operation mode according to the indoor and outdoor conditions. The system can work in seven operation modes (**Errore. L'origine riferimento non è stata trovata.**):

- cooling and dehumidification (DEC+AHP);
- indirect evaporative cooling and absorption chiller (IEC+AHP);
- indirect evaporative cooling (IEC);
- ventilation (VENT);
- heat recovery (HR);
- heat recovery and absorption heat pump (HR+AHP);
- heating and humidification (HUM+AHP).

Most of the operation modes are characterized by proportional signals that must be assigned.

The most tricky operation modes are the ones using the heat pump, because it creates another link between the supply and return side of the AHU in addition to the links made by the rotary heat exchange and the desiccant wheel. Indeed, the heating power delivered to one side of the AHU is function of the cooling power delivered on the other side of the AHU. For these reasons, for example, for specific conditions is needed to reduce the effect of the heat recovery unit to ensure that the exhaust air temperature is compatible with the power which the GAHP needs on its cold side (evaporator).

Generally, air handling unit controllers use sequencing logic in order to maintain the set-point conditions in the most economical way. One drawback of this approach is the risk of instability when several components are controlled at the same time (Seem et al., 1999). This problem can be partly addressed in the tuning phase of the control parameters. Sequencing logic applied to DEC systems may result in a complex tuning exercise due to the number of components involved. Therefore, alternative approaches have been investigated in which a subset of components are operated steadily according to some rules and only a limited number of components use feedback control simultaneously. The particular operating condition is associated to a well-defined operation mode (e.g., ventilation, indirect evaporative cooling, cooling and dehumidification). The transition from one operation mode to another one is managed by a selector which is based on simple rules. Actuators use can be reduced by adopting a finite state machine approach (Vitte et al., 2008), i.e. delaying the transition until the current operation mode has reached saturation for a predefined time interval. In the following, the adopted control logic is described.

The simplest operation mode is the ventilation mode (VENT). Only the supply and return fans are on. The return air flow rates is equal to the exhaust air from the building, i.e. it is not increased through the mixing with external air. The air by-pass around the desiccant wheel and the rotary heat exchanger are open (Figure 1).

If the external air is too warm, the indirect evaporative cooling mode is activated (IEC). The return air adiabatically cooled by the humidifier, thus capable to provide cooling to the supply air thanks to the rotary heat exchanger (RoHX). The RoHX air by-pass is proportionally controlled according to the supply air set-point temperature.

If the IEC cooling power is not enough, the system switches to the indirect evaporative cooling and absorption chiller mode (IEC+AHP). The absorption heat pump is turned on in cooling mode, along with the associated water pumps. The natural gas input to the heat pump is proportionally controlled in order to follow the

supply air temperature set-point.
When the outdoor air humidity by mass is higher than the maximum allowed, the system switches to the cooling and dehumidification mode (DEC+AHP). The desiccant wheel air by-pass is closed. The return air flow rate is increased up to the supply air flow rate through a mixing with external air. The natural gas input to the heat pump is proportionally controlled in order to primary follow the supply air humidity by mass set-point.
On the other hand, when the external temperature is lower than the set-point, the first mode to be activated is the heat recovery mode (HR). Thus, the rotary heat exchanger air by-pass is proportionally controlled in order to follow the supply air temperature set-point.
If the heat recovery mode heating capacity is not enough the heat pump is turned on (HR+AHP). The three-way valves system addresses the hot water to the air-water heat exchanger on the supply side (HX1). The air by-pass around the rotary heat exchanger is proportionally controlled in order to avoid ice formation at the air-water heat exchanger on the return side (HX2). Indeed, without this heat transfer limitation it could happen that the inlet water at (HX2) falls below zero in order to satisfy the heat pump evaporator energy demand. The natural gas input to the heat pump is proportionally controlled in order to follow the supply air temperature set-point.
When both the outdoor air humidity by mass and temperature are lower than the set-point, the system switches to the heating and humidification mode (HUM+AHP). The water feed to the supply side humidifier is turned on. The desiccant wheel air by-pass is opened. The natural gas input to the heat pump and the water feed to the humidifier are controlled in order to primary follow the supply air humidity by mass set-point.

3. Modeling

In order to assess the performance of the system using computer simulation, a numerical model has been developed and implemented in a Matlab code. The model is a steady-state model based on a node structure. Each node represents a sub-component of the whole system (e.g. fan, rotary air-air heat exchanger) and it is characterized by specific inputs, outputs and parameters.
The model is based on the use of ε-NTU correlations for the finned-tube heat exchanger and the rotary heat exchanger. The input parameters UA have been assessed through a tuning with the experimental results from a system prototype. The wetted media humidifiers have been modelled with constant adiabatic humidification efficiency. The desiccant wheel has been modelled using the model developed by Aprile and Motta (2013) and it has been tuned with the experimental data. Lastly, the gas absorption heat pump performance has been derived from the manufacturer's data (ROBUR company).

4. The reference system

The reference system used for comparative analysis (Figure 3) is composed of: two fans, a cross-flow heat recovery, a cooling coil (for sensible cooling and dehumidification), a heating coil, a wetted media humidifier. The cooling coil is fed by an electric air-source heat pump, while the heating coil is fed by a natural gas boiler.
The performance of this reference system has been assessed through a numerical model developed and implemented in a Matlab code. As for the presented system model, it is a steady-state model based on a node structure. The fans, humidifier, the air heat recovery device and the air-water finned tubular heat exchangers have been modelled like the presented system. The cross-flow heat exchanger ε has been fixed at 0,6. The gas natural boiler has been modelled considering an overall constant thermal efficiency of 0,9 respect to the natural gas gross calorific value and a fixed electric consumption (150 W). Lastly, the electric air-source heat pump has been modelled as a black-box mathematical model, starting from the performance data provided by a manufacturer. The inputs are the forward temperature of the working fluid to the AHU cooling coil and the temperature of the external air (T_{ext}).
The forward temperature of the working fluid to the AHU cooling coil has been assessed starting from the cooling coil apparatus dew point (T_{ADP}), linked to the set-point humidity by mass

of the supply air and to the cooling coil air-by

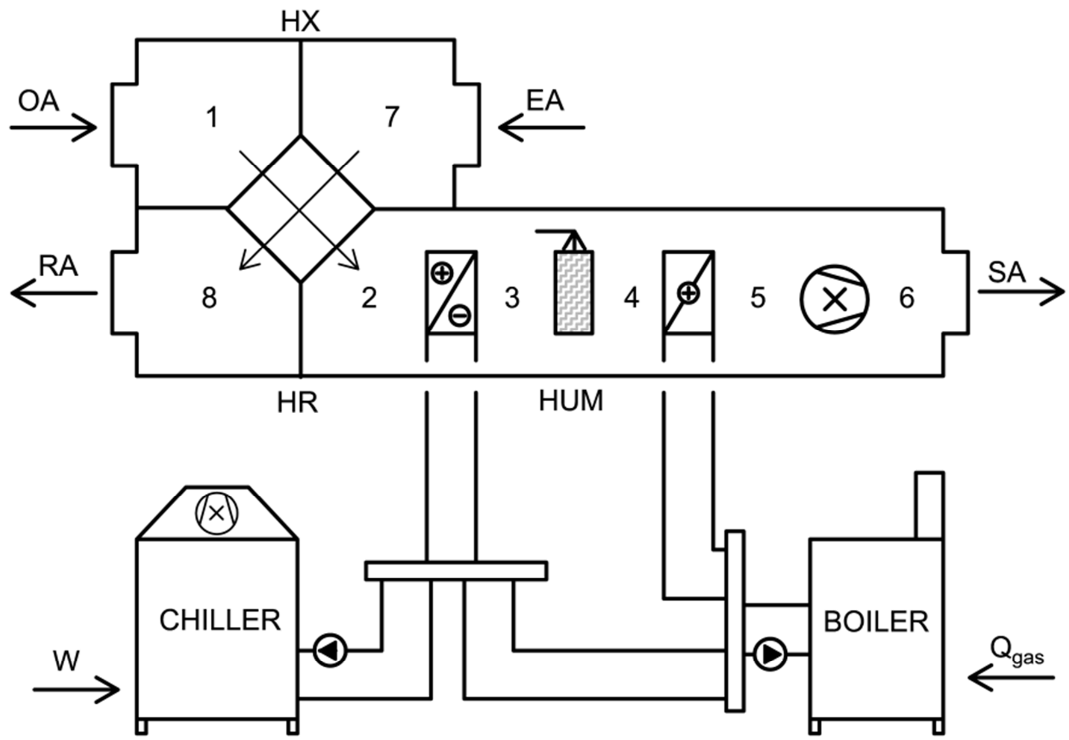

Figure 3 - Reference system scheme.

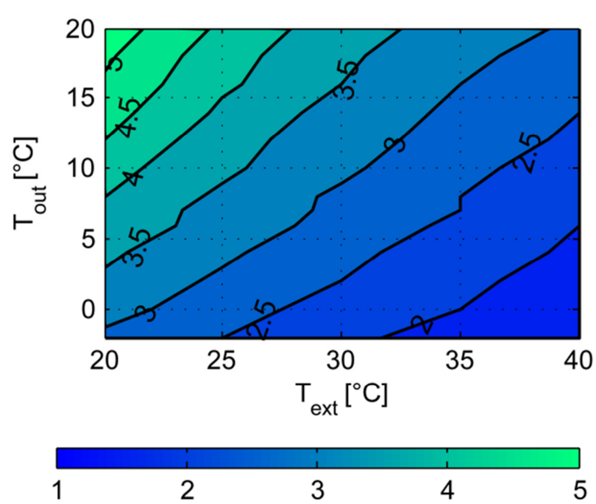

Figure 4 - Electric heat pump energy efficiency ratio (EER).

pass factor, which has been fixed at 0,15. The main output of air-source heat pump model is the energy efficiency ratio EER (Figure 4), which is predicted using a polynomial equation. The model is valid only for the water outlet temperature (T_{out}) and external air temperature (T_{ext}) shown in Figure 4.

5. Results and discussion

In this section the simulation results considering stationary boundary conditions and a supply volumetric flow rate of 5000 m^3/h are presented. Among the seven operation mode only the two mode with the higher energy

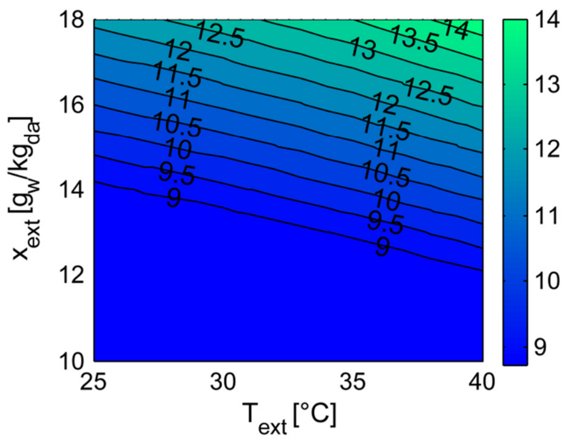

Figure 5 - DEC+AHP operation mode - supply air humidity by mass x_{sa} [g_w/kg_{da}].

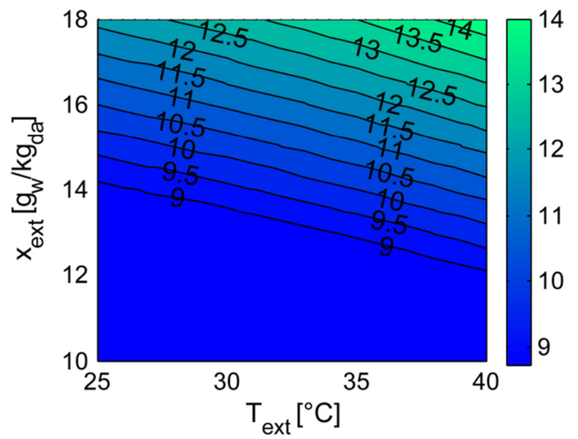

Figure 6 - DEC+AHP operation mode - supply air temperature T_{sa} [°C].

consumption (DEC+AHP and HUM+AHP operation mode) are presented hereafter.

Cooling and dehumidification operation mode

It follows the analysis of the cooling and dehumidification operation mode changing the external conditions.

The return air from the indoor space has been set at 26 ° C / 50%, while, for the supply air, only the humidity by mass set-point has been set; specifically at 8,8 g_w/kg_{da} equal to a relative humidity of 41,2 % at 26 °C.

Figure 5 shows that the system is able to guarantee the humidity by mass set-point if the external air is in the range 25-40 °C and 10-12 g_w/kg_{da}. For higher external humidity by mass the set-point is still guaranteed if the external temperature decreases, up to the limit 25 °C / 14 g_w/kg_{da}. This is due to a better performance of the desiccant wheel for higher relative humidity of the external air. Outside this range, due to the constraint on the maximum desiccant wheel regeneration temperature (due to the heat pump), the system cannot exceed a dehumidification rate of around 5 g_w/kg_{da}.

Figure 6 shows what happen to the supply air temperature. As all the air by-pass are closed, it shows the minimum achievable supply air temperature, which could be reduced limiting the indirect evaporative cooling effect.

Figure 7 and Figure 8 show the Primary Energy Ratio (PER) achieved by the system and the reference system. The PER is the ratio between the net supply air energy exchange and the non-renewable primary energy used by the system (Eq. 1).

$$\mathrm{PER} = \frac{\text{useful effect}}{\text{non renewable primary energy}} = \\ = \frac{\left|\left(h(T_{sa}, x_{sa}) - h(T_{ext}, x_{ext})\right)\dot{m}_{sa}\right|}{\sum P_{ele}\mathrm{PEF}_{ele} + \dot{Q}_{gas,GCV}\,\mathrm{PEF}_f} \tag{1}$$

The system PER (Figure 7) is higher than 1 for most of the conditions. The maximum value is at the maximum external temperature (40 °C) and minimum external humidity by mass (10 g_w/kg_{da}), achieving a PER of roughly 1,8. The minimum value is 0,8 at T_{ext} 25°C, x_{ext} 13,5 g_w/kg_{da}.

Going in more detail, Figure 9 and Figure 10 show the numerator and the denominator of the PER ratio. The PER equation numerator is the useful cooling effect, which get the maximum value of around 37 kW at the highest external temperature and humidity by mass.

On the other hand, Figure 10 shows the PER denominator, which is the primary energy consumption of the system (PE_{cons}). The PE_{cons} maximum value is at T_{ext} 25 °C, w_{ext} 14 g_w/kg_{da}, this is due to the highest energy demand at the regeneration coil (HX2) plus the minimum air inlet temperature at the cooling coil (HX1), i.e. the maximum heat pump heating demand at the minimum GUEh. The highest energy demand at HX2 is due to the minimum heat recovery through the rotary heat exchanger at the maximum achievable dehumidification rate (Figure 11).

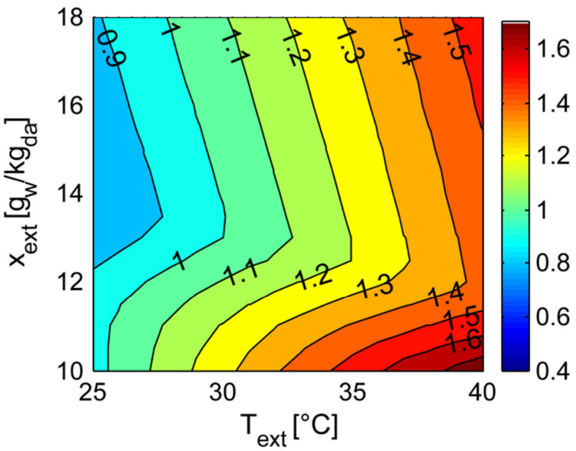

Figure 7 - DEC+AHP operation mode PER [-] - presented system.

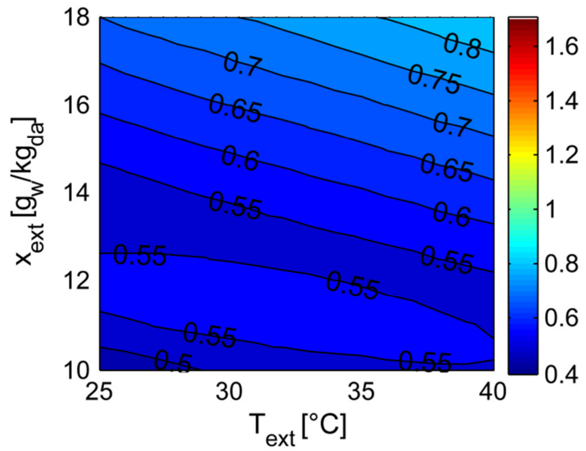

Figure 8 - DEC+AHP operation mode PER [-] - (b) Reference system.

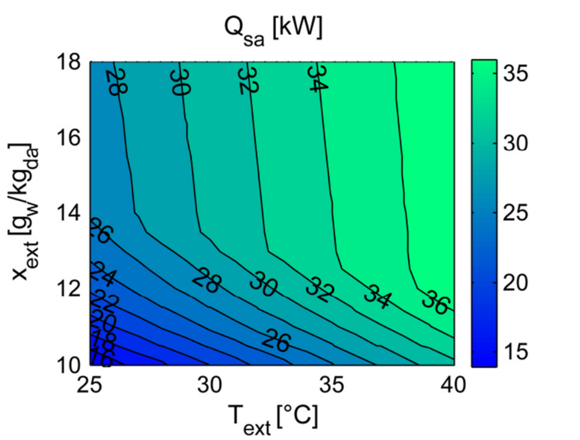

Figure 9 - DEC+AHP operation mode - Supply air total cooling power [kW].

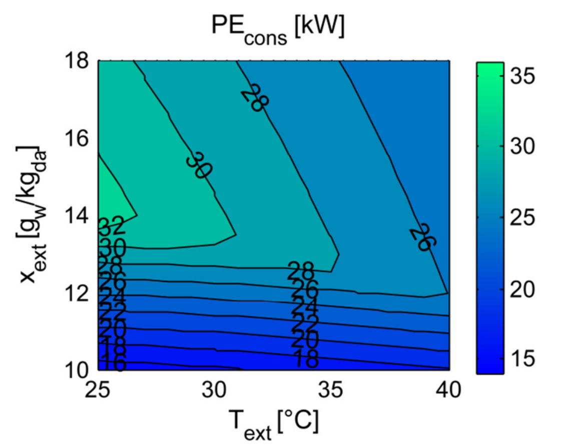

Figure 10 - DEC+AHP operation mode - Primary energy consumption [kW_{PE}].

Heating and humidification mode

The set-point condition for the supply air has been fixed at a temperature of 20 °C and a relative humidity of 37,5% (5,5 g_w/kg_{da}), while for the return air from the indoor space at 20 °C and 50%.

For the heating and humidification operation mode the desiccant wheel is not needed, thus only the heat pump absorption cycle is on, using the exhaust air as evaporator energy source.

According to the minimum water temperature at the cooling coil linked to the heat pump evaporator, a constraint on the outlet temperature of the rotary heat exchanger supply side has been set, in order to guarantee the heat pump evaporator energy demand.

Figure 12 shows that the system is able to guarantee the supply air humidity by mass set-point for most the case, except when the external air is colder than 2 °C, since the heating capacity of the selected heat pump is not enough.

Figure 13 and Figure 14 show the comparison between the PER (Eq.1) achieved by the presented system and the reference system for the heating and humidification mode.

The maximum PER value is at the highest external humidity by mass (3,5 [g_w/kg_{da}]) and an external temperature of 7 °C, achieving a PER of 1,7.

The highest PER values are around $T_{ext} \approx 7$ °C because it is the range where is maximum the use of the heat recovery from the exhaust air without compromise the availability of energy

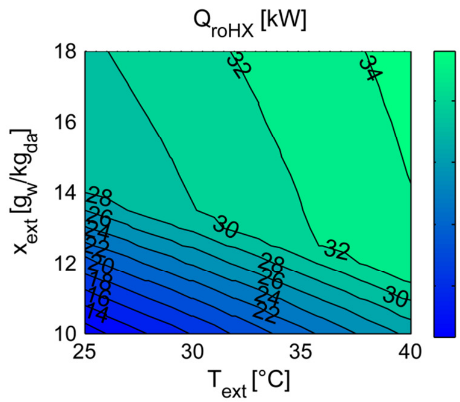

Figure 11 - Indirect evaporative cooling for DEC+AHP operation mode.

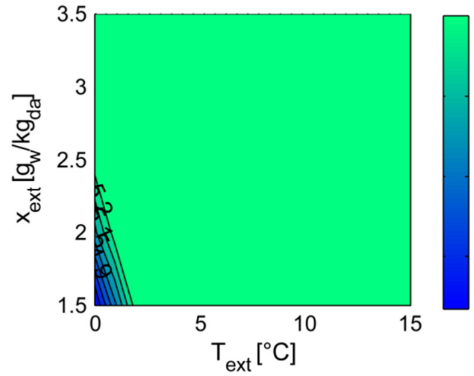

Figure 12 - HUM+AHP operation mode. Supply air humidity by mass x_{sa} [g_w/kg_{da}].

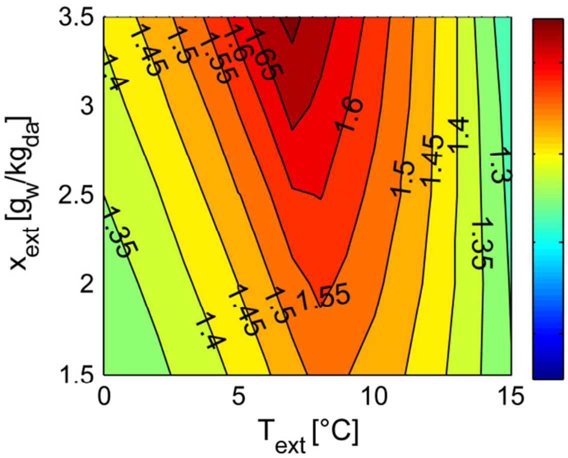

Figure 13 - HUM+AHP operation mode Primary Energy Ratio PER [-] - presented system.

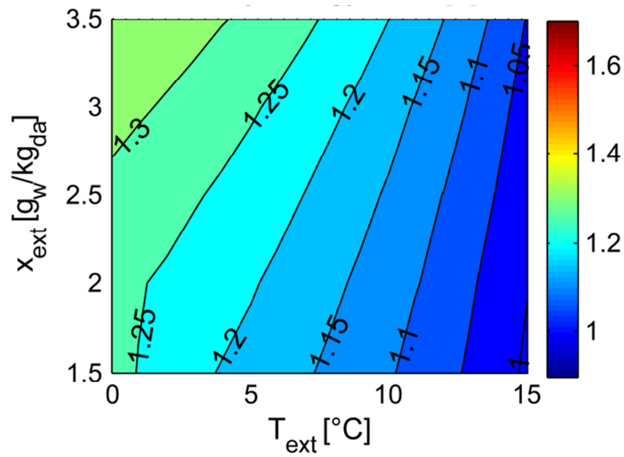

Figure 14- HUM+AHP operation mode Primary Energy Ratio PER [-] - reference system.

for the heat pump cold side at temperature higher than 0 °C.

6. Conclusions

A thermal driven air-conditioning system, worked out coupling two sorption cycles, has been presented.
The system integrates an absorption heat pump cycle and a solid desiccant evaporative cooling open cycle (DEC).

Its main feature is the ability of the absorption heat pump cycle to drive the desiccant evaporative cooling cycle, creating a double effect cycle.
After a compatibility analysis on the temperature levels a ammonia/water absorption heat pump and a silica-gel desiccant wheel have been selected.
The solid DEC cycle is implemented in a double duct air handling unit. In one duct the building supply air flows, while in the other the exhaust air from the building flows.

The system is able to work in seven operation modes according to the supply air set-point and the boundary conditions. The main operation modes are the cooling and dehumidification mode and the heating and humidification mode.
In cooling and dehumidification mode, the heat pump has two useful effects: it delivers the heating power needed to the desiccant based dehumidification process and it provides cooling power for the supply air flow.
In heating and humidification mode the hydraulic connections between the DEC air handling unit and the heat pump is inverted, and the latter is used to heat up the supply air, recovering energy from the indoor exhaust air.
The ratio between the net supply air energy exchange and the non-renewable primary energy used by the system (PER) has been assessed as the main performance figure.
As reference system for PER comparison an air-conditioning system composed by an air source electric heat pump, a gas boiler and a cross-flow air heat recovery has been considered.
Thus, the PER comparison between the presented system model and the reference system has shown a better energy performance of the presented system for both the heating humidification mode and the cooling dehumidification mode, but it cannot handle dehumidification rate higher than around 5 g_w/kg_{da}.
The presented thermal driven air conditioning system is particularly suitable in areas where the natural gas is cheap and/or the electric energy generation and distribution system is weak.

Acknowledgements

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Symbols

h	moist air specific enthalpy (kJ/kg_{da})
$\dot{m}$	mass flow rate (kg/s)
P	power (kW)
$\dot{Q}$	thermal power (kW)
RH	relative humidity [-]
T	temperature (°C)
x	humidity by mass (g_w/kg_{da})
ε	effectiveness (-)

Subscript

c	cold temperature or cooling mode
DW	Desiccant wheel
da	dry air
ext	outdoor
fw	forward
h	hot temperature or heating mode
hum	humidifier
in	inlet
int	indoor
lat	latent
ma	moist air
oa	outdoor air
out	outlet
reg	regeneration side
ret	return side
sa	supply air
sens	sensible
supply	supply side
w	water

ACRONYMS

ADP	Apparatus Dew Point
AHU	Air Handling Unit
BP	air By-Pass
COP	COefficient of Performance [-]
CC	Cooling Coil
DEC	Desiccant Evaporative Cooling
EER	electric heat pump Energy Efficiency Ratio [-]
GAHP	Gas Absorption Heat Pump
GAX	Generator Absorber heat eXchange
GCV	Gross Calorific Value
HC	Heating Coil
HX	heat exchanger
RoHX	Rotary Heat eXchanger

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