DOI: 10.18462/iir.icr.2023.0471

Performance evaluation of a desiccant cooling system coupled with indirect evaporative cooling technologies under various climates

Alanis ZEOLI*[(a)], Samuel GENDEBIEN[(a)], Vincent LEMORT[(a)]

[(a)] Thermodynamics Laboratory of the University of Liège
Liège, 4000, Belgium
*Corresponding author: alanis.zeoli@uliege.be

ABSTRACT

This work contributes to assess the impact of coupling two indirect evaporative cooling technologies with a standard desiccant evaporative cooling system. In the first modified version, an indirect evaporative cooler (IEC) is added to the system. The secondary air of the IEC is the air extracted from the building. In the second investigated system, the process air is sensibly cooled in a dew-point indirect evaporative cooler (D-IEC). Part of the cooled process air is used as secondary air in the D-IEC. The performance of the three systems in terms of regenerative energy, electricity and water consumptions are compared over the cooling periods of seven climatic zones. It is shown that adding an IEC to the system reduces the overall consumption of the system, especially for warm, moderate and/or dry climate zones. In very hot and humid climate zones, the ambient specific humidity is too high and the system should be enhanced to provide more suitable supply conditions.

Keywords: Desiccant wheel, evaporative cooling, dehumidification, air conditioning

1. INTRODUCTION

In the past decades, demand for space cooling has been increasing continually, and with global warming this growth is not expected to slow down. Nowadays, cooling represents almost 20% of the total electricity consumption in buildings worldwide (Birol, 2018). Many researchers have focused on alternative air-conditioning to tackle the issues of vapor compression systems, namely their high-grade energy consumption rate and the use of HCFCs as working fluids. The main advantage of the desiccant cooling technology is that it can use low grade energy sources such as solar energy or waste heat from industrial processes. It can also take advantage of district heating networks in summer when the heating demand is lower. The desiccant evaporative cooling system (DECS) consists in coupling the desiccant technology with evaporative cooling technologies to cool down buildings.

Ali et al. (2015), investigated the possibility to modify the air source for both process and regeneration air flows, which can be either ambient air or recirculated air from the indoor environment. They evaluated the impact of the configuration on system cooling capacity and performance in five climate zones.

Elgendy et al. (2015) added an indirect evaporative cooler module to the standard system. They proposed three configurations and assessed their performance in terms of building cooling load, system cooling capacity and regeneration power consumption depending on ambient temperature and relative humidity.

Pandelidis et al. (2016) evaluated the possibility to couple the desiccant wheel with three types of indirect evaporative coolers. They showed that all the analysed systems can provide satisfactory supply air temperatures, even when the desiccant wheel is regenerated with low temperature.

Pacak et al. (2023) experimentally proved the positive impact of using a dew point evaporative cooler before air dehumidification in the desiccant wheel. They showed that pre-cooling the outdoor air allows to enhance the dehumidification rate of the system and to provide lower supply temperatures.

The goal of this paper is to evaluate the performance of enhanced desiccant evaporative cooling systems depending on the climate zone. The influence of the operating conditions and component performance of the cooling system on the control strategies were assessed through a sensitivity analysis. The first part of this paper presents the results of this analysis. In the second part, the performance of the system is studied under various climatic conditions to predict the energy savings related to the appropriate control strategy.

2. SYSTEM CONFIGURATIONS DESCRIPTION

Three DECS have been studied with variations to increase the performance of the system. The system configurations and their representation in the psychrometric diagram are shown in Fig. 1.

The reference system (System 0) consists of a desiccant wheel (DW), a regenerative heat exchanger (Reg.), a sensible heat exchanger (HEX) and two direct evaporative coolers (DEC). The air entering the system on process side is ambient fresh air. The process air is first dehumidified in the desiccant wheel, where it is also heated. The supply air is cooled down in a sensible heat exchanger and, if necessary, the specific humidity can be adjusted with a direct evaporative cooler. On regeneration side, air is extracted from the building and cooled down in a second DEC before entering the sensible heat exchanger. The temperature of regeneration air is further increased in the regenerative heat exchanger until the regeneration temperature is reached. It can therefore be used in the desiccant wheel to absorb moisture from the process air.

The first proposed modification (System 1) consists in replacing the DEC on regeneration side by an indirect evaporative cooler (IEC). An IEC is a heat exchanger in which the secondary air exchanges heat and mass with a water film while the primary fluid is cooled without being humidified. The limit temperature that could be reached in an IEC is thus the dew point of the primary air. The IEC allows to reach lower temperatures on process side compared to a standard counterflow heat exchanger.

In the second proposed modification (System 2), the indirect evaporative cooler is used on process side only. After being cooled in the IEC, part of the process air is diverted and used as secondary fluid in the IEC. This type of IEC is commonly called dew-point IEC (D-IEC). On the one hand, this configuration theoretically allows to reach lower supply air temperatures, hence decreasing the remaining sensible load provided by an additional cooling system. On the other hand, since part of the process air is used as secondary fluid in the D-IEC, System 2 should work with higher air flow rates to provide the hygienic flow rate to the building. It can also be noted that, in order to have balanced flows in the DW, the air that is extracted from the building should be mixed with air coming from the outdoor environment.

3. METHODOLOGY

3.1. Description of the case study and system sizing

The methodology used for the design of the HVAC installation is inspired from the work of Pillai and Desai (2018). The building is a one-storey 300 m² office building with 30 occupants and a North-South orientation. On nominal conditions, the latent load of the building is 2.5 kW and comes from the occupants. The hygienic ventilation rate is fixed to 1000 m³/h but it is possible to ventilate at higher air flow rates to ensure thermal comfort. The air handling unit should always be operated to meet the building latent load. The covered sensible load is maximised but the remaining sensible load is supposed to be delivered by an additional cooling system. The cold water used in the sensible cooling system is assumed to be produced by a chiller.

The meteorological data used in this study have been generated and validated in the framework of the IEA EBC Annex 80. Seven climatic zones have been studied in this work to evaluate the climate dependency of the systems, as well as the impact of the climate on control strategy and energy consumption. The climatic zones have been defined following the climate classification of ASHRAE (2013). For each climate, the cooling period is defined as the period during which sensible cooling should be provided to the building. The hourly indoor building conditions were obtained through an annual dynamic simulation performed using MATLAB.

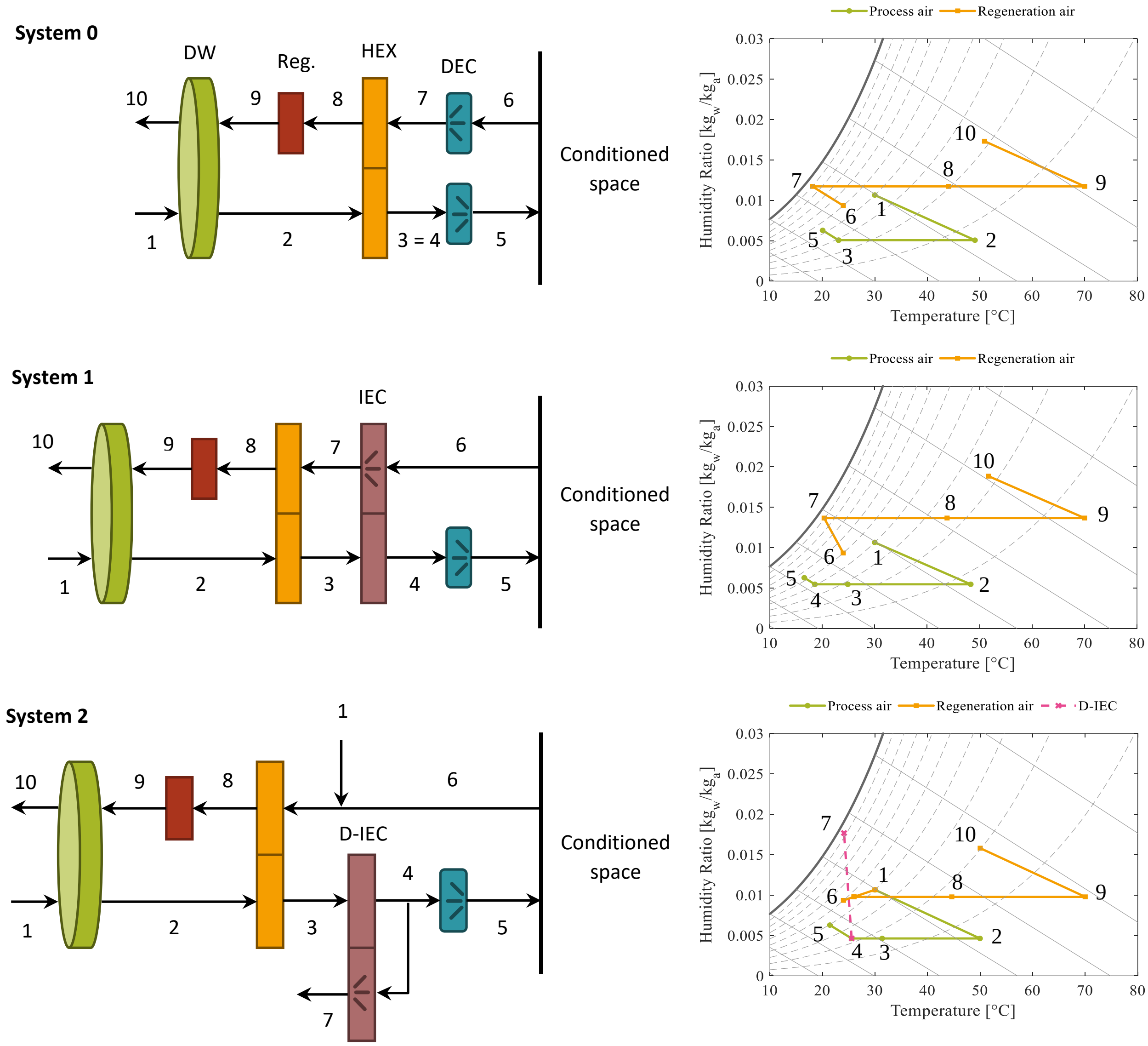

Figure 1 – Schematic of the conventional and proposed desiccant evaporative cooling system configurations with their representation in the psychrometric diagram.

The sensible heat exchanger model is based on the ε-NTU method. For the IEC modules, the model is based on the work of Chengqin and Hongxing (2006). The DW model is a simplified model based on two parameters η_{F1} and η_{F2} (Panaras et al., 2010). It has been validated and calibrated using the results presented by De Antonellis et al. (2015). The fan and chiller consumptions are evaluated based on the models proposed by Bertagnolio (2012). For each climate zone, the components of the systems have been sized to provide the required latent load to the building in nominal conditions. The nominal indoor conditions (25°C/60%) were based on WHO indoor comfort recommendation (Morawska and Thai, 2018). The nominal design outdoor conditions were determined based on the 2% hottest hours of the climatic data. Their representation on a psychrometric diagram is shown in Fig. 2. The key results of the sizing are reported in Tab. 1. It can be noted that the recirculation rate of the D-IEC has been set to 1/3. This is the result of a trade-off between energy consumption and system performance. The larger the recirculation rate, the larger the system cooling capacity and the fan electricity consumption.

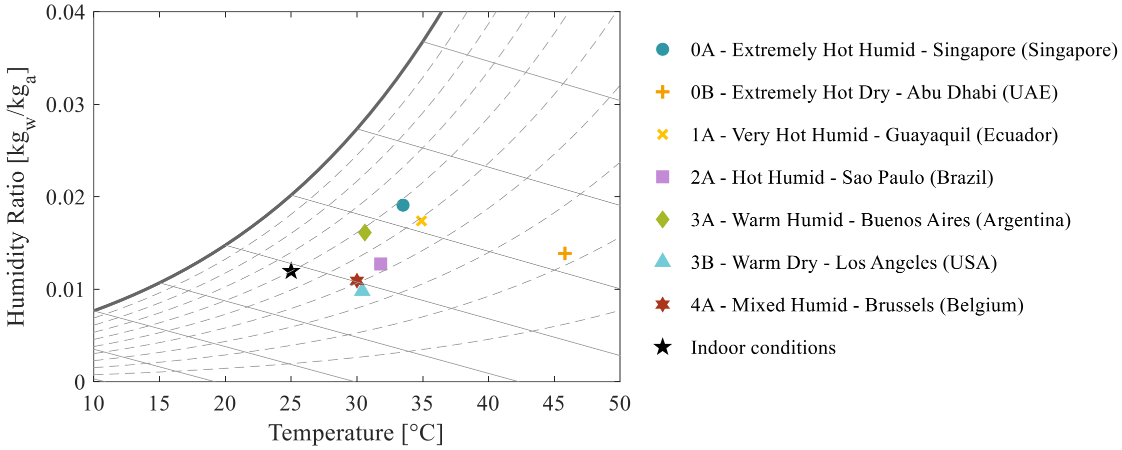

Figure 2 – Nominal design conditions of the system depending on the climate zone.

Table 1 – Components parameters.

Fan	
SFP	750 W/(m³/s)
Additional HEX	300 W/(m³/s)
$\dot{V}_{nom}$	2000 m³/h

Chiller	
$\dot{Q}_{nom}$	6.2 – 9.2 kW
COP	3

DEC	
ε	0.5 – 0.85

HEX	
NTU	0.9 – 4.4

IEC	
NTU	0.5 – 2.6

DW	
η_{F1}	0.07 – 0.1
η_{F2}	0.78 – 0.82
T_{reg}	65°C – 75°C

D-IEC	
NTU	0.5 – 2.6
Recirculation rate	1/3

3.2. Control strategy

For each system, different operating modes have been defined to reduce either the regenerative energy consumption, the chiller consumption or the fan consumption while maintaining optimal comfort conditions. The system should ideally be operated to always provide the building with the nominal latent load. The latent load is computed as shown in Eq. (1).

$$\dot{Q}_{lat} = \dot{M}_a \cdot h_{vap} \cdot (w_6 - w_5) \tag{1}$$

The indoor specific humidity w_6 being fixed by the building simulation, the only parameters that can be varied to provide the latent load are the supply air specific humidity w_5 and the supply mass flow rate $\dot{M}_a$. The building should always be ventilated at least with the hygienic air flow rate ($\dot{M}_{a,min}$) while the maximum air flow rate ($\dot{M}_{a,max}$) has been defined as twice the hygienic flow rate. If the building should be provided with the nominal latent load, a lower and a upper bound can be defined for the supply air specific humidity:

$$w_{su,min} = w_6 - \frac{\dot{Q}_{lat,nom}}{\dot{M}_{a,min} \cdot h_{vap}} \tag{2}$$

$$w_{su,max} = w_6 - \frac{\dot{Q}_{lat,nom}}{\dot{M}_{a,max} \cdot h_{vap}} \tag{3}$$

If the supply specific humidity is lower (resp. higher) than $w_{su,min}$ (resp. $w_{su,max}$), the installation works with the minimum supply air flow rate and the building is de-humidifed (resp. humidified).

The supply specific humidity can be adjusted by modifying the quantity of water that is re-injected in the process air through the DEC. The air humidification also allows to lower the supply air temperature, hence decreasing the sensible load that should be provided by the additional cooling system. The quantity of water to be re-injected in the process air is evaluated through a parameter called the DEC efficiency, expressed by:

$$\varepsilon_{DEC} = \frac{T_4 - T_5}{T_4 - T_{wb,4}} \tag{4}$$

In practice, the DEC efficiency is limited to 85%. The DEC efficiency *per se* is not a parameter that can physically be modified directly but it can give an estimation of the quantity of water that should be injected in the process air. Its value during system operation can thus vary between 0 and 0.85.

The second control strategy that has been investigated is the bypass of the desiccant wheel. Since the system operation is based on the control of the specific humidity of supply air, if the outdoor specific humidity is lower than the required threshold, it is assumed that the desiccant wheel can be bypassed. The bypass of the desiccant wheel allows to suppress the need for regeneration energy and decrease the water consumption of the system. The effect of bypassing the desiccant wheel on the cycle is shown in Fig. 3. In this case, the DW is not physically bypassed, the bypass is done by stopping the rotation of the DW. The general scheme of the control system is shown in Fig. 4.

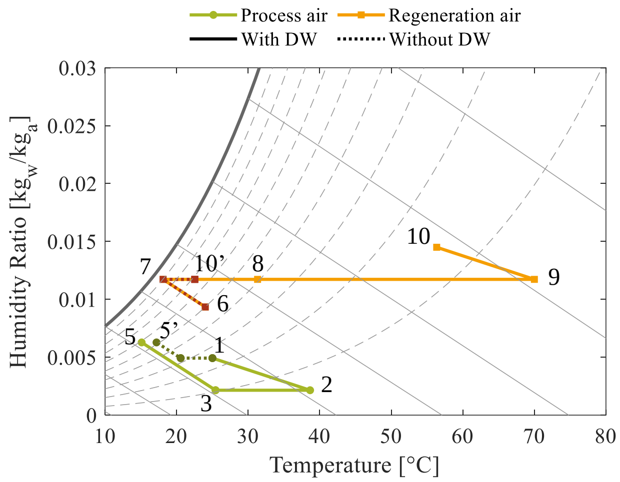

Figure 3 – Psychrometric representation of the cycle with and without DW.

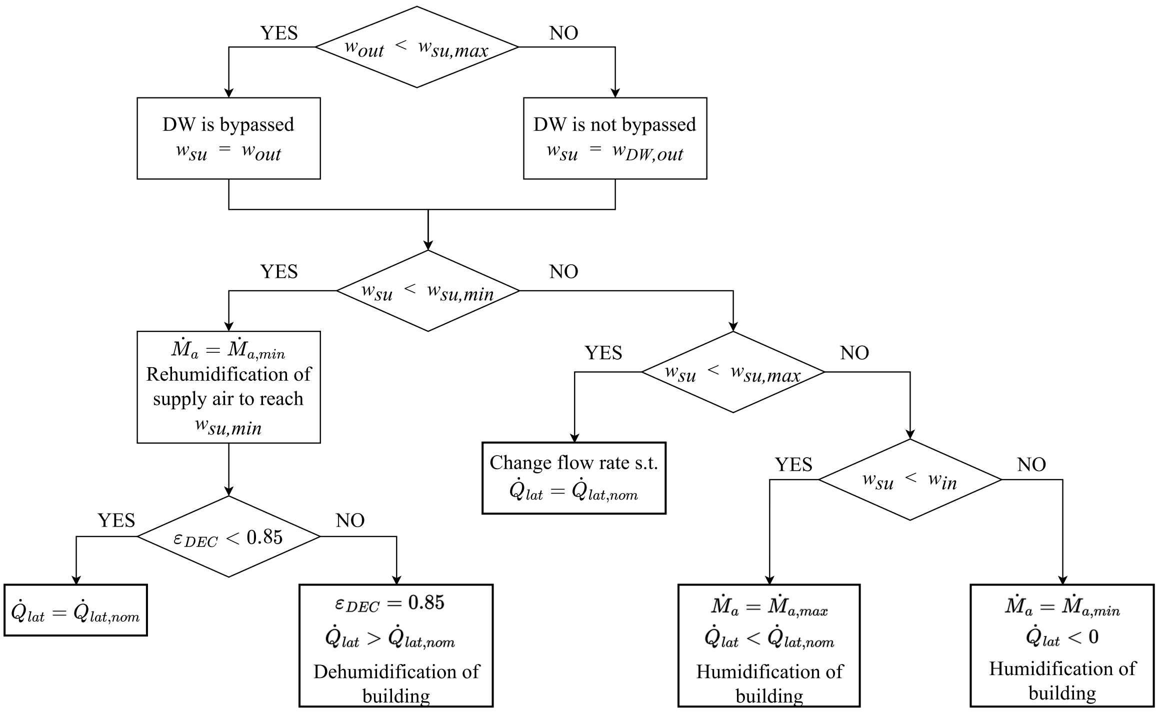

Figure 4 – Decision flow chart of the control system strategy.

3.3. Definition of criteria for performance evaluation

Once the air properties at each stage of the cycle are known, it is possible to study the performance of the system in terms of system cooling capacity ($\dot{Q}_{cool}$), regeneration energy consumption ($\dot{Q}_{reg}$), electricity consumption of fans ($\dot{W}_{fan}$) and chiller ($\dot{W}_{ch}$) and water consumption ($\dot{M}_w$). The system cooling capacity and the regeneration energy consumption can be determined as follows:

$$\dot{Q}_{cool} = \dot{M}_a \cdot (h_1 - h_5) \quad (5) \qquad \dot{Q}_{reg} = \dot{M}_a \cdot (h_9 - h_8) \quad (6)$$

The system cooling capacity refers to the heat that should be removed from the outdoor incoming air in order to be supplied to the building in acceptable conditions. It should be distinguished from the cooling load that is the cooling effect provided by the system to the building. The regeneration energy is the heat provided by the regenerator to increase the temperature of secondary air to the regeneration temperature.

The fan electricity consumption is proportional to the cube of the air flow rate while the chiller electricity consumption is proportional to the square of the required cooling load of the building (Bertagnolio, 2012). The water consumption of the system is obtained through a water mass balance on the humid air in components including evaporative cooling.

Based on those consumptions, it is possible to define indicators to evaluate the systems performance. For desiccant evaporative cooling systems, two main energy consumptions have to be accounted for, the regeneration energy consumption and the electricity consumption. Two coefficients of performance can then be defined, a thermal one and an electrical one. In this case, since the simulations are performed on the whole cooling season, the performance of the systems is evaluated based on the seasonal coefficient of performance (SCOP). The thermal and electrical SCOPs are defined as follows:

$$\mathrm{SCOP}_{th} = \frac{Q_{cool}}{Q_{reg}} = \frac{\sum_i \dot{Q}_{cool,i}}{\sum_i \dot{Q}_{reg,i}} \quad (7) \qquad \mathrm{SCOP}_{el} = \frac{Q_{cool}}{W_{fan}} = \frac{\sum_i \dot{Q}_{cool,i}}{\sum_i \dot{W}_{fan,i}} \quad (8)$$

4. RESULTS AND DISCUSSIONS

The systems 0, 1 and 2 have been simulated over the cooling period of each climate zone. For each system the possibility to bypass the DW has been considered if the outdoor conditions are suitable.

4.1. Results of the annual simulations

The results of the annual simulations are summarised in Fig. 5. The comparison between the considered systems is done based on a consumption analysis, including the regeneration energy, chiller and fan electricity consumptions and water consumption.

In most climates, the regeneration energies required by Systems 0 and 1 have similar orders of magnitude while the regeneration energy required by System 2 can be up to 50% higher. This is due to the fact that the air flow rate in the DW is higher than the air flow rate that is supplied to the building, since air flows in the D-IEC are not balanced. To ensure the hygienic ventilation in the building, the system should work with higher flow rates, which also induces an increase of the fan electricity consumption of System 2.

The chiller electricity consumption reflects the ability of the system to provide sensible load. The higher the sensible load of the system, the lower the additional sensible load that should be provided by the chiller. System 1 provides the largest sensible load to the building, allowing to reduce the electricity consumed by the chiller.

Regarding the water consumption of the systems, System 0 has the lowest water consumption. Systems 1 and 2 have a larger water consumption (from 12 to 130% increase) because the IEC and D-IEC configurations enhance water evaporation on secondary side compared to a standard DEC.

The bypass of the desiccant wheel is more interesting as the climate becomes less humid (climate zones 0B, 3A, 3B and 4A). In those climate zones, the outdoor specific humidity can be lower than the required supply specific humidity. The bypass of the DW results in up to a 40% reduction in the annual regeneration energy and 20% reduction in the water consumption. However, it can also lead to a slight increase in the electricity consumption of the system (around 10%). If the air is not dehumidified in the DW, the amount of water that can be added in the process air is reduced, increasing the building supply temperature, which in turn, increases the cooling load that has to be provided by the chiller. Moreover, if the outdoor supply specific humidity is between $w_{su,min}$ and $w_{su,max}$, the building should be ventilated with larger flowrates than the hygienic one, hence resulting in an electric fan consumption increase.

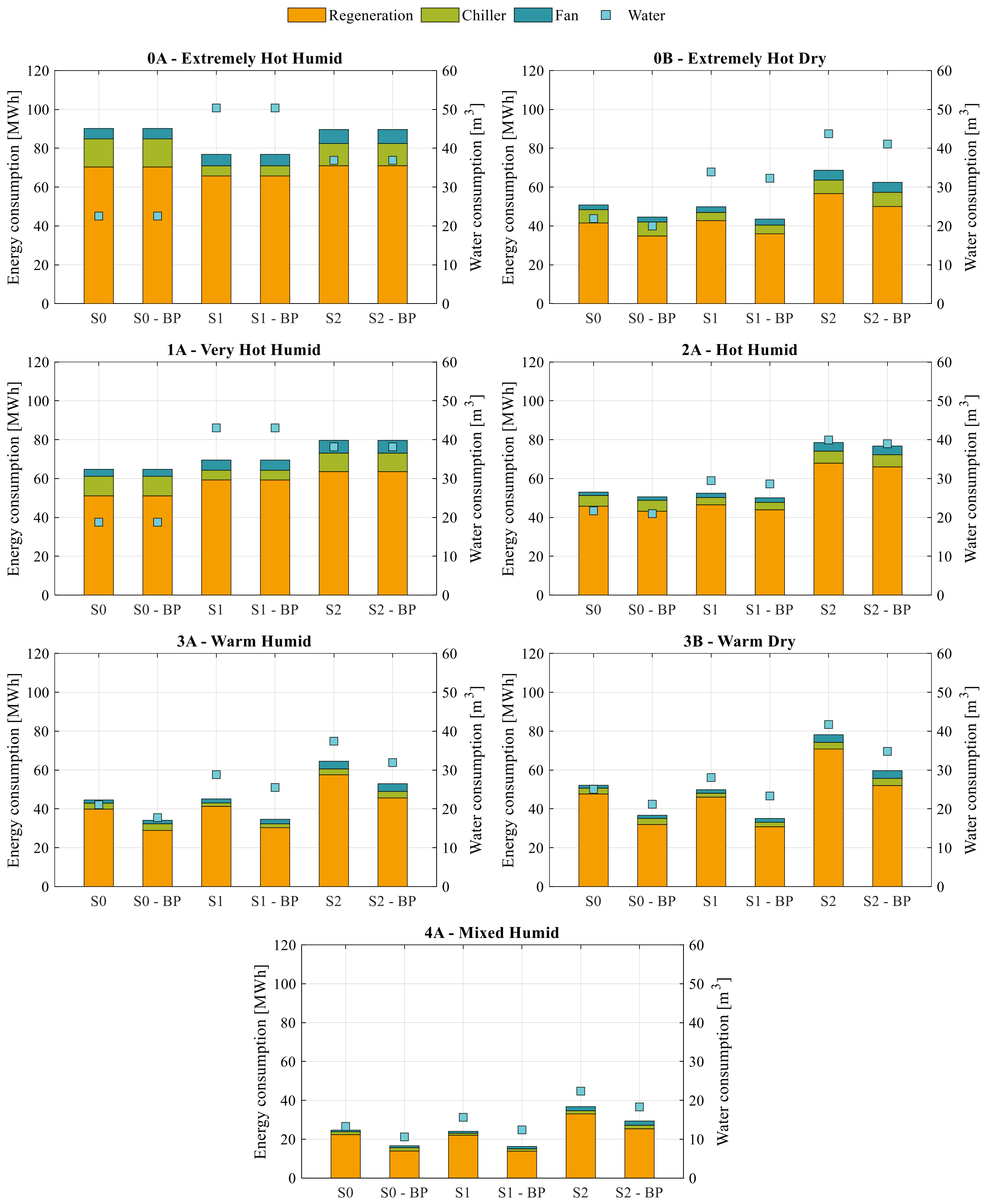

Figure 5 – Results of the annual system simulation depending on the climate zone.

4.2. Performance analysis

The performance of the systems are evaluated with the SCOPs defined in section 3.3. The thermal and electrical SCOPs of the systems in the different climate zones are presented in Fig. 6. Warmer climates (i.e. climate zones 0A, 0B and 1A) have a $SCOP_{th}$ between 1.2-1.4 while those of climate zones 3B and 4A are between 0.2-0.4 (see Fig. 6a). Since the climate is colder and less humid, the system has to provide a lower cooling capacity to the oudoor air to reach acceptable supply conditions. However, the regeneration temperature is not significantly lower than in warmer climates. The decrease in the regeneration energy is thus less significant than the required system coling capacity.

For all climates, the $SCOP_{th}$ of System 1 is larger than for the other systems, because it is able to provide a larger cooling capacity. System 0 and System 2 have comparable $SCOP_{th}$. It can also be noted that, as expected, the $SCOP_{th}$ increases when the DW is bypassed.

The $SCOP_{el}$ (Fig. 6b) is larger for System 0, since it is the installation with the lowest head losses, hence the lowest fan electricity consumption. Contrary to the $SCOP_{th}$, the $SCOP_{el}$ decreases when the DW is bypassed. First, in this work, it has been considered that bypassing the DW does not decrease the head losses because it is assumed that the DW simply stops rotating, preventing the air dehumidification. Second, as it has already been mentioned in the previous section, when in bypass mode, the installation could be working with larger flowrates, hence increasing the fan consumption.

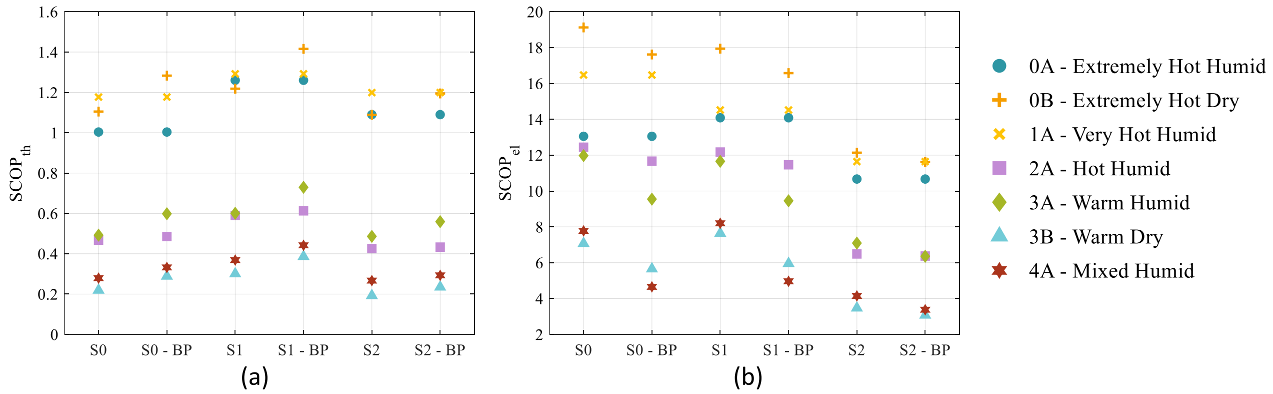

Figure 6 – (a) Thermal and (b) electrical SCOPs of the studied systems under various climate conditions.

In the previous definition of the $SCOP_{el}$, only the fan consumption has been taken into account. However, the fan electric consumption can be divided in two parts: the consumption due to the mechanical ventilation for hygienic purpose and the consumption due to the addition of a DECS. As it has been shown in the previous section, the presence of the DECS in the building also has an effect on the sensible load that should be provided by the chiller, hence on its electric consumption. Therefore, even though the main purpose of the DECS is to provide the nominal latent load, it is interesting to study its impact regarding the sensible load. To complete the analysis related to the electricity consumption, the DECS have also been compared to the case where all the sensible cooling load is supplied by the chiller and that the ventilation system is only used for hygienic purpose. In this fictitious case, the latent load is not handled anymore.

Fig. 7a shows the ratio between the electricity consumption of the fan with a DECS and the electricity consumption of the fan when the mechanical ventilation is used for hygienic purposes only. The consumption increases when a DECS is added on the ventilation because the added heat exchangers generate some additional head losses and the fans should have a higher specific fan power. For a ventilation used only for hygienic purpose, the SFP is supposed to be 750 W/(m³/s). For System 0 the SFP is 1350 W/(m³/s) and 1650 W/(m³/s) for Systems 1 and 2. In most climates, for Systems 0 and 1, the ratio between the electricity consumptions is close to the ratio between the SFPs. For System 2, however, the fan electricity consumption can be multiplied by a factor 5 to 8 due to the fact that the fan air flowrate should be increased to satisfy the hygienic ventilation requirements. In very hot and humid climates (0A and 1A), the fan consumption increases

2 to 3 times more than in other climates because the building should be ventilated with higher flowrates to provide the nominal latent load.

Fig. 7b shows the ratio between the chiller electricity consumptions when the mechanical ventilation is equipped with a DECS and when the sensible cooling load is provided only with a chiller. It can be seen that for all systems and for most climates, the presence of a DECS allows to reduce the sensible load that has to be provided with the chiller. System 1 is the one that decreases the most the electricity consumption of the chiller. Depending on the warmness and dryness of the climate, the chiller electricity consumption can reach between 30 and 60% of its original consumption. The colder and drier the climate the larger the chiller consumption reduction. In climate zone 0A, for Systems 0 and 2, the chiller consumption is increased. The climate is too humid for the cooling system to provide acceptable supply air conditions. The air supplied to the building is most of the time at a higher temperature than the indoor temperature and the sensible cooling load that has to be provided by the chiller is increased. System 1 is the only one that is able to provide suitable supply conditions

Fig. 7c shows the ratio between the total electricity consumption of the cooling systems with and without a DECS. System 0 allows to decrease the overall electricity consumption related to sensible cooling, except in climate zones 0A and 1A corresponding to very hot and humid climates. In other climates, the consumption increase due to the higher fan SFPs is counterbalanced by the savings related to the chiller. The use of System 2 leads to an increase in the electricity consumption of the cooling system independently of the climate zone. In zone 0A, the consumption is even doubled. In terms of electricity consumption, the most efficient system is System 1. For almost all climate zones, the electricity consumption is reduced at 50-90% of its original value.

For very hot and humid climate zones such as zones 0A and 1A, the desiccant evaporative cooling systems should be further enhanced to increase the system capacity to provide a sensible cooling load.

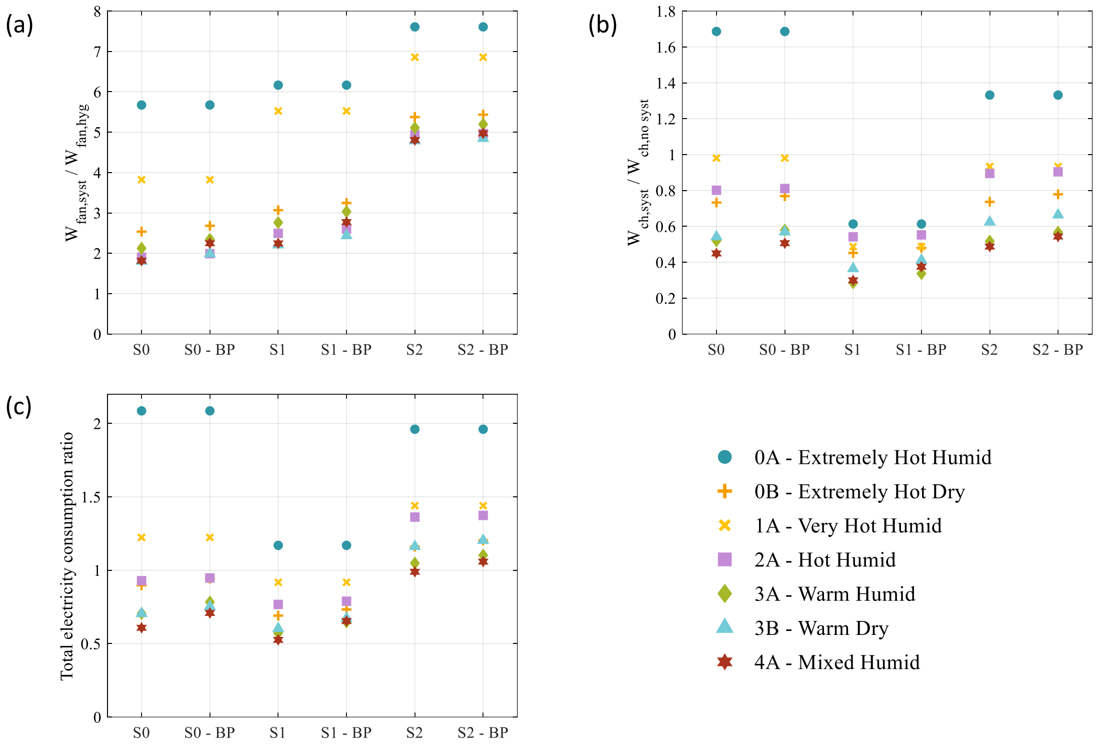

Figure 7 – (a) Ratio between fan electricity consumption with DECS and for hygienic ventilation only.
(b) Ratio between chiller electricity consumption with and without DECS.
(c) Ratio between total electricity consumption with and without DECS.

5. CONCLUSIONS

Three systems have been studied in this work. System 0 is a classic desiccant evaporative cooling system, System 1 uses a simple IEC after the sensible HEX to further decrease the supply air temperature and in System 2, the IEC is replaced by a D-IEC, which uses only process air as primary and secondary fluids. Part of the air flow at the exhaust of the primary side of the D-IEC is directed to the secondary side while the rest of the air is supplied to the building.

System 1 allows to increase the provided sensible load compared to System 0 and System 2. It generally shows better performance in terms of thermal and electric consumptions. On the contrary, the air provided by System 2 is usually at a higher temperature than with other systems because the D-IEC is not as efficient as the simple IEC. The D-IEC should preferably be used in systems in which there is no alternative source to outdoor air to be used as secondary fluid in the heat exchangers, such as in supply-only mechanical ventilation systems. If the outdoor temperature becomes too high, the D-IEC could become more efficient than a standard IEC and provide supply air at a lower temperature. A deeper analysis could also be carried out on the recirculation rate of the D-IEC. It could be interesting to implement a ventilation system with a variable recirculation rate depending on outdoor conditions.

The performance of the systems have been studied in terms of regeneration energy, fan and chiller electricity consumption and water consumption for 7 different climate zones. It has been demonstrated that the use of System 2 always results in an increase of energy consumption compared to System 0. To keep the supply air flow at the hygienic rate, the air flow rate in the DW should be increased, resulting in higher fan electric consumption (+170%) and regeneration energy consumption (+50%).

In most climates, System 1 is the most efficient one because it also allows to increase the sensible load of the system, hence decreasing the chiller electricity consumption (-40%). In some cases, it also allows to decrease the regeneration energy consumption if outdoor conditions are favourable. Its main drawback is that it comes with an increased fan consumption (20-40%) due to additional head losses in the system and with an increased water consumption (12-130%).

For colder or drier climates, the bypass of the desiccant wheel makes sense, resulting in a 35% reduction in regeneration energy use and a 20% reduction in water use. However, it can also lead to increased fan and chiller electricity consumptions (around 10%). An economic and environmental analysis could highlight the trade-off between electricity, heat and water and establish the benefits of the bypass. It could also be possible to investigate other types of bypass control strategies. For example, under some outdoor conditions, it could be interesting to completely bypass the cooling system to work in ventilation mode only. Some guidelines about the bypassing conditions could be defined.

Finally, it has been determined that for very hot and humid climate zones (zones 0A and 1A), the desiccant evaporative cooling systems should be further enhanced to increase the system capacity to provide a sensible cooling load. The cooling system could be enhanced by introducing post-cooling coils to ensure acceptable supply air conditions. Some other configurations could also be introduced to enhance the energy use in the system. The use of other evaporative cooling technologies such as Maisotsenko cycles could be investigated.

NOMENCLATURE

	Abbreviations		Symbols
DEC	Direct evaporative cooler	$\dot{M}$	Mass flow rate (kg/s)
DECS	Desiccant evaporative cooling system	h_{vap}	Vaporisation enthalpy (J/kg)
D-IEC	Dew-point indirect evaporative cooler	$\dot{Q}$	Heat transfer rate (W)
DW	Desiccant wheel	T	Dry bulb temperature (°C)
HEX	Sensible heat exchanger	T_{wb}	Wet bulb temperature (°C)
IEC	Indirect evaporative cooler	$\dot{V}$	Volumetric flow rate (m³/s)
Reg.	Regenerative heat exchanger	w	Specific humidity (kg/kg)
SFP	Specific Fan Power	η_{F1}	Desiccant wheel parameter
		η_{F2}	Desiccant wheel parameter

	Subscripts		
a	Air	out	Outdoor environment
lat	Latent	pro	Process air
in	Indoor environment	reg	Regeneration air
nom	Nominal	su	Supply air

AKNOWLEDGEMENT

This study is a part of the International Energy Agency (IEA) EBC Annexes 80 – "Resilient cooling of buildings" and 85 – "Indirect Evaporative Cooling" project activities to define resilient cooling in buildings with a focus on indirect evaporative cooling.

REFERENCES

Ali, M., Vukovic, V., Sheikh, N.A., Ali, H.M., 2015. Performance investigation of solid desiccant evaporative cooling system configurations in different climatic zones. Energy Conversion and Management 97, 323–339. https://doi.org/10.1016/j.enconman.2015.03.025

Bertagnolio, S., 2012. Efficient-based model calibration for efficient building energy services. Université de Liège, Belgium.

Birol, D.F., 2018. The Future of Cooling - Opportunities for energy efficient air conditioning. International Energy Agency.

Chengqin, R., Hongxing, Y., 2006. An analytical model for the heat and mass transfer processes in indirect evaporative cooling with parallel/counter flow configurations. International Journal of Heat and Mass Transfer 49, 617–627. https://doi.org/10.1016/j.ijheatmasstransfer.2005.08.019

De Antonellis, S., Intini, M., Joppolo, C.M., 2015. Desiccant wheels effectiveness parameters: Correlations based on experimental data. Energy and Buildings 103, 296–306. https://doi.org/10.1016/j.enbuild.2015.06.041

Elgendy, E., Mostafa, A., Fatouh, M., 2015. Performance enhancement of a desiccant evaporative cooling system using direct/indirect evaporative cooler. International Journal of Refrigeration 51, 77–87. https://doi.org/10.1016/j.ijrefrig.2014.12.009

Morawska, L., Thai, P., 2018. Estimation of minimal risk and acceptable temperatures for selected cities, in: WHO Housing and Health Guidelines.

Pacak, A., Jurga, A., Kaźmierczak, B., Pandelidis, D., 2023. Experimental verification of the effect of air pre-cooling in dew point evaporative cooler on the performance of a solid desiccant dehumidifier. International Communications in Heat and Mass Transfer 142, 106651. https://doi.org/10.1016/j.icheatmasstransfer.2023.106651

Panaras, G., Mathioulakis, E., Belessiotis, V., Kyriakis, N., 2010. Experimental validation of a simplified approach for a desiccant wheel model. Energy and Buildings 42, 1719–1725. https://doi.org/10.1016/j.enbuild.2010.05.006

Pandelidis, D., Anisimov, S., Worek, W.M., Drąg, P., 2016. Comparison of desiccant air conditioning systems with different indirect evaporative air coolers. Energy Conversion and Management 117, 375–392. https://doi.org/10.1016/j.enconman.2016.02.085

Pillai, J., Desai, R., 2018. Dehumidification strategies and their applicability based on climate and building typology.