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## COMPARISON OF PERFORMANCES OF DRYING TECHNIQUES OF SURFACES IN AGRO FOOD PREMISES

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### ABSTRACT

Cleaning and disinfection (C&D) are among the most important hazard control measures in ready-to-eat food plants. The C&D process requires a huge volume of water that wets the surfaces of the food processing premises. The water remaining on these surfaces are susceptible to be the source of a microbial reservoir as wet media are favorable to microbial growth. To face this problem, a rapid drying after C&D is to be sought. For this purpose, the humidity control of the air in the premises is to be obtained.

In this purpose, three technologies could be employed: (i) a classical one based on dew point air treatment, (ii) the use of liquid desiccant and (iii) the use of desiccant wheels.

This paper compares the energy consumption of these three techniques for typical premises which can be found in agrofood industry.

The energy consumptions of these techniques are assessed with simplified approaches which may sometimes be open to discussion. Nevertheless, the results indicate some major tendencies and allow supporting interesting conclusions.

### 1. INTRODUCTION

Cleaning and disinfection (C&D) are among the most important hazard control measures in ready-to-eat food plants. However, these procedures require large amounts of water and generate huge volumes of sewage with high loads of cleaning agents and biocides. More sustainable C&D strategies are therefore needed. Several food business operators have already noted the positive effects of adding an air-drying step after C&D to dry the surfaces and thus control the growth of microorganisms that are not detached from surfaces. However, air drying is applied empirically and no attempts have been made to define optimal air drying conditions, which would increase the efficiency of the lethal hydric stress.

The drying of the surfaces of food processing premises can be obtained with an adequate management of air humidity in the premises. A very classical technique is based on the dew point air treatment: the temperature

of air is lowered down to a dew point corresponding to the targeted absolute humidity and then re-heated in order to avoid blasting a too cold air in the room –at the risk of fogging and/or freezing the surfaces. The cooling is obtained with a classical compression / expansion refrigeration device, and the reheating is generally made with electric resistances (cheap and easy to implement, despite additional running expenses compared to other possible heat recovery devices). The energy consumption of this technique is high, especially when low humidity is expected.

Liquid desiccant air treatment is a quite old technology as the first experience on this technique dates back to 1930 – 1940 for air conditioning applications. Already at this time, the efficiency of this technology regarding the moisture treatment was recognized, as well as its ability to run mostly on low-medium temperature thermal energy. Another advantage of this technique is its low electric demand for air conditioning applications (Lowenstein, 2008). However, the application of liquid desiccant air treatment for moisture and temperature management in the food industry remains infrequent, despite the benefits regarding the food safety and the microbial growth on working surfaces.

Desiccant wheels technology, based on the use of a solid sorbent fixed onto the surface of an adequate device, is widely used when low humidity ratios are expected – as for instance in low temperature warehouses or processes in order to better manage the frosting of evaporators. This technique is reputed to be efficient, easy to implement, but quite energy consuming in reason of the temperature required to regenerate the sorbent.

A research project financed by the French National Research Agency aims at developing tools and know-how in order to “Reduce the environmental impact of hygiene procedures in refrigerated food processing plants through optimal use of air drying” (ECOSEC Project). A technologic WP of this project deals with comparison of the energy consumption of these three techniques. This comparison is made on a typical food processing premises. The calculations are based on simplified approaches which may sometimes be open to discussion. Especially, the fact of not taking into consideration the electricity consumption for blower to supply air and pump to supply liquid desiccant is an approximation. This approximation could be justified by two considerations : (i) for the three techniques, the consumption of fans could be assumed to be of the same order of magnitude and (ii) the consumption of pumps are usually quite small regarding to the consumption of the entire system. Nevertheless, these approximations have to be kept in mind for the interpretation of the results.

## 2. DESCRIPTION OF THE REFERENCE CASE STUDY

A typical food processing premises has been investigated by IRSTEA and presented by Guilpart et al (2014):

- Surface of the food processing premise: 100 m<sup>2</sup>
- Height of the premise: 3 m
- Air conditions: 6°C – 90% RH
- Estimation of residual water after C&D process: film thickness of 0.5 mm on floor and of 0.05 mm at half-height of the walls (ceiling is neglected). The water remaining inside the equipment is not taken into account (assumed to be trapped inside the equipment and therefore difficult to evaporate).
- Estimation of the amount of water to be evaporated in order to dry the premises: 51.2 liters
- Approximate drying time: between 1h30 and 2h.

According to the technical guidelines and recommendations applicable in refrigeration, the following parameters are estimated:

- Air change rate: 20V, corresponding to an air flow rate of 6 000 m<sup>3</sup> h<sup>-1</sup>
- Blast air conditions: 2.5 °C / 40% RH, corresponding to a dew point of -8.7°C

For the calculations, the ambient conditions in the premises are assumed to be constant and equal to 6°C / 100% RH during the whole drying period.

### 3. PERFORMANCE OF THE DEW POINT AIR TREATMENT TECHNIQUE

For the above mentioned case study, the performance of the dew point air treatment technique is assessed by Guilpart et al. (2014) as followed:

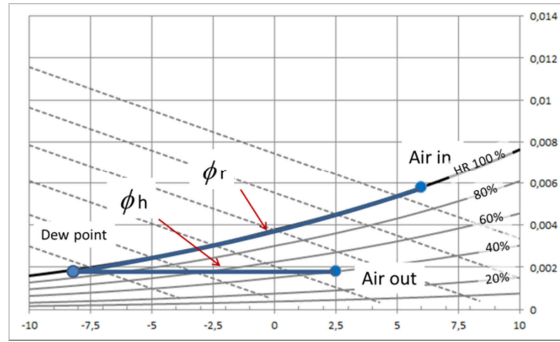


Figure 1 : Air evolution on Mollier diagramm

Table 1: key characteristics of air

		air in	dew point	air out
t	°C	6,0	-8,7	2,5
HR	%	100	100	40
w	Kg kg <sup>-1</sup>	5,79E-03	1,80E-03	1,80E-03
h	kJ kg <sup>-1</sup>	20,5	-4,2	7,1
ρ	Kg m <sup>-3</sup>	1,25	1,33	1,28

To ensure the cooling of air down to -8.7°C to reach the expected dew point, an evaporation temperature of -13.7°C has to be obtained (pinch = 5K), leading to a COP of 3.4.

The drying phase lasting 1h45', following consumptions are expected:

	Demand (kW)	Consumption (kWh)
Refrigeration	51.5 kW <sub>refrig.</sub> / 15.1 kW <sub>elec</sub>	26.5 kWh <sub>elec</sub>
Heating	23.4 kW <sub>elec</sub>	41.0 kWh <sub>elec</sub>
Defrosting		6.4 kWh <sub>elec</sub>
Total		73.9 kWh <sub>elec</sub>

The defrosting energy costs are roughly assessed on following assumptions:

- Mass of the evaporator : 300 kg
- Average heat capacity of the evaporator : 0.5 kJ kg<sup>-1</sup> K<sup>-1</sup> (in-between copper 0.385 kJ kg<sup>-1</sup> K<sup>-1</sup> and aluminum 0.897 kJ kg<sup>-1</sup> K<sup>-1</sup>)
- Frost mass to be melted : 51 kg – heat capacity : 2.09 kJ kg<sup>-1</sup> K<sup>-1</sup> – latent heat of fusion : 333 kJ kg<sup>-1</sup> at °C
- Initial temperature : -13.7 °C
- Final temperature of evaporator : +10°C

Leading to following energy (electric) costs:

Heating phase:

Heating of evaporator + frost from -13.7 to 0°C : 0.97 kWh

Melting of frost: 4.71 kWh

Heating of evaporator from 0°C up to 10°C : 0.42 kWh

Cooling phase:

Lowering down the evaporator's temperature from +10°C down to -13.7°C : 0.99 kWh<sub>refrigeration</sub>  
that is, with a COP of 3.4 leads to an electric consumption of : 0.29 kWh

Total electric consumption: 6.4 kWh

That represents roughly 10% of the energy required for the cold production assuming that the efficiency of the defrosting devices is 1.

In this approach, it is assumed that the defrosting efficiency is equal to one and that heating of air is ensured by an electric resistance (this often the case) and not by heat recuperation on the refrigeration unit.

#### 4. PERFORMANCE OF THE LIQUID DESICCANT TREATMENT TECHNIQUE

Guilpart et al. (2014) described the basics of this technique where direct contact – counter flow heat exchangers are used. In this reference and in the present work, the calculations are based on the thermophysical properties proposed by Condé-Petit (2003) for LiCl and CaCl<sub>2</sub> liquid desiccants. Guilpart et al. (2014) showed that for the studied application, the use of CaCl<sub>2</sub> is not possible according to the temperature and humidity values expected for the air blown in the premise (2.5°C – 40% RH).

In this work, it was assumed that the air evolves along the line defined by the slope  $\frac{h_{a,o}-h_{a,i}}{w_{a,o}-w_{a,i}}$  as presented Figure 2.

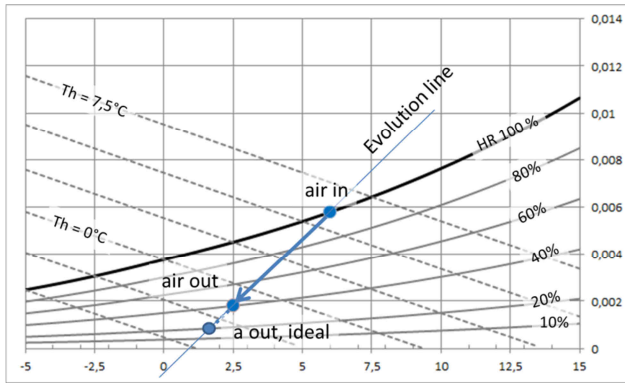


Figure 2 : simplified air evolution on the Mollier diagram according to Guilpart et al. (2014)

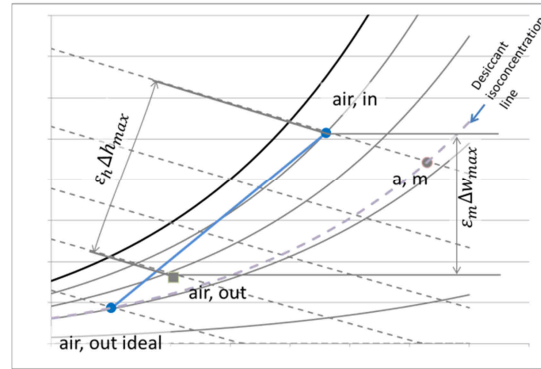


Figure 3 : air evolution on the Mollier diagram, taking into account heat and mass transfer efficiencies.

In Figure 2, the point  $a_{out, ideal}$  corresponds to an air in equilibrium with the desiccant solution entering in the exchanger (vapour pressure and temperature). This approach assumes that mass transfer efficiency is defined by heat transfer efficiency, which is an approximation acceptable for high heat and mass transfer efficiencies. For lower efficiencies, coupling of heat and mass transfer is necessary.

Xiaohua & al (2007) proposes analytical solutions of coupled heat and mass transfer processes in liquid desiccant air dehumidifier / regenerator. In this reference, the heat transfer efficiency for counter flows exchanges is calculated from :  $\varepsilon_h = \frac{1 - \exp\{NTU(1-m^*)\}}{m^* - \exp\{NTU(1-m^*)\}}$

and the mass transfer efficiency from  $\varepsilon_m = \varepsilon_h + K \frac{m^*(1 - \exp\{-NTU\}) - (1 - \exp\{-m^*NTU\})}{m^* - \exp\{NTU(1-m^*)\}}$

where :  $m^* = \frac{m_a c_{p_a}}{m_s c_{p_s}}$ ,  $K = \frac{w_{air,in} - w_{a,m}}{w_{air,in} - w_{air,out,ideal}}$  and  $NTU = \frac{h_d S}{m_a}$

The NTU expression based on the mass transfer coefficient  $h_d$  (kg s<sup>-1</sup> m<sup>-2</sup>) comes out from the coupling of heat and mass transfer through the Lewis number  $Le = \frac{h_c}{h_d c_p} = 1$  where  $h_c$  is the heat transfer (kW m<sup>-2</sup> K<sup>-1</sup>).

The “intermediate” point  $a, m$  is located at the intersection of the inlet desiccant isoconcentration line and the inlet air isenthalpic line. Figure 3 shows the air evolution in the Mollier diagram :

For the considered application, and assuming heat and mass exchanges efficiencies equal to 1, Guilpart et al. (2014) calculated following inlet parameters:

For  $t_{air,in} = 6^\circ\text{C} - HR_{air,in} = 100\%$ ,

the LiCl desiccant solution has to have following characteristics:  $t_{s,in} = 2.5^\circ\text{C} - x_{s,in} = 0.3$   
to provide an air at 2.5°C – 40% RH blown in the premises

Coupling heat and mass transfer around these inlet parameters leads to following efficiencies, as a function of the temperature pinch between the air at the outlet of the exchanger and the desiccant at the inlet of the exchanger:

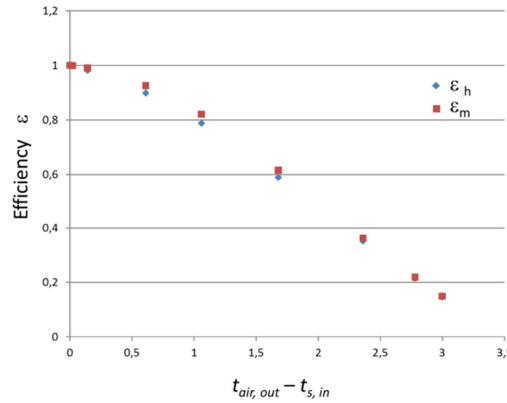


Figure 4: heat and mass transfer efficiencies vs observed temperature pinch, around operating conditions for present dehumidification application.

Figure 4 points out that around the expected running conditions in the premise, the mass transfer efficiency  $\epsilon_m$  and the heat transfer efficiency  $\epsilon_h$  have similar behavior and values.

In practice, the observed temperature pinch depends on many parameters, including the exchange surface between air and desiccant, the velocity of air and the type of desiccant flow (spraying of droplets, trickling along a surface, ...). A conservative approach led to adopt an efficiency of  $\epsilon_h = 0.95$ , due to the high developed exchange surface permitted with this type of exchangers.

Dehumidification side, the heat transfer efficiency  $\epsilon_h = 0.95$  permits to calculate a mass transfer efficiency of  $\epsilon_m = 0.965$ . Thus, an air outlet temperature of 2.5°C could be obtained with an inlet desiccant temperature of 2.2°C, and an air outlet humidity of 40% could be obtained with a desiccant concentration of 0.308

Regeneration side, the heat transfer efficiency  $\epsilon_h = 0.95$  permits to calculate a mass transfer efficiency of  $\epsilon_m = 0.944$ . Thus, the solution slightly diluted at the outlet of the dehumidification device ( $x = 0.305$ ) has to be heated up to 35.3 °C in order to be wetted with the outside air (assumed here to be at 15°C - 80% RH)

The efficiency of the internal heat exchanger is fixed to 0.95.

Figure 5 shows the calculation results obtained for the entire air treatment loop

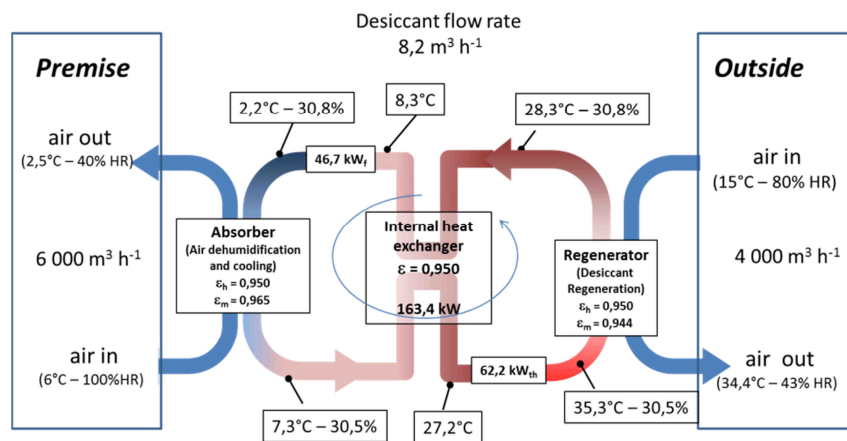


Figure 5 : key values for the dehumidification / regeneration loop (desiccant = LiCl).

For this application, the desiccant cooling down to 2.2°C could be obtained with an evaporation temperature of -2.8°C (pinch = 5K), leading to a COP of 4.7. As far as the air blown into the premise has the same characteristics (2.5°C – 40% RH), the drying duration remains the same (1h45'). According to these assessments, following energy consumption are expected:

	Demand (kW)	Consumption (kWh)
Refrigeration	46.7 kW <sub>refrig.</sub> / 9.9 kW <sub>elec</sub>	17.4 kWh <sub>elec</sub>
Heating	62.2 kW <sub>elec</sub>	108.9 kWh <sub>elec</sub>
Defrosting	-	-
Total		126.3 kWh <sub>elec</sub>

These values are based on an electric heating of the desiccant before regeneration. Some significant energy savings can be achieved by recovering heat on the chiller's condensation. Indeed, 46.7 + 9.9 = 56.6 kW of heat could be recovered at the condensing temperature (around 35°C), ensuring 90% of the heating needs.

Taking this heat recovery into account leads to following consumption:

	Demand (kW)	Consumption (kWh)
Refrigeration	46.7 kW <sub>refrig.</sub> / 9.9 kW <sub>elec</sub>	17.4 kWh <sub>elec</sub>
Total heating need	62.2 kW <sub>th</sub>	
Heat recovery	56.6 kW <sub>th</sub>	
Complementary heating need	5.6 kW <sub>elec</sub>	9.8 kWh <sub>elec</sub>
Defrosting	-	-
Sub Total		27.2 kWh <sub>elec</sub>

## 5. PERFORMANCE OF THE DESICCANT WHEEL TECHNIQUE

Desiccant wheels is a dehumidification technique based on the use of a solid sorbent fixed onto the surface of an adequate device. The basic principle of this technique is reminded Figure 6. For the studied desiccant wheel, one can note the absence of a heat recovery for preheating the regeneration air. Some new equipment are equipped with such devices, that would significantly reduce the global consumption of the system. Unfortunately, the studied system does not dispose of this device.

As far as the moisture condensation onto the desiccant is an exothermic process, the dry air has to be cooled down in order to be blown at 2.5°C in the premise. At the opposite, the regeneration of desiccant needs hot air, classically round 110 – 120°C.

The performance of the system depends on many parameters, including the sorption – desorption curves of the desiccant, the rotation speed of the wheel, the heat capacities of the sorbent and of the support, the depth of the wheel, the sorption heat, ... as highlighted by Zhai (2008) and many other references.

As a first approach, the adsorption capacity of the wheel can be extracted from technical datasheets provided by an equipment provider, for a device adapted to the present application (Dessica dehumidifier DT 6000). In complement, a basic heat and mass balance calculation can be used in order to calculate missing parameters as follow :

Treated air :

Air inlet (from premise) :  $t_{a,in} = 6^\circ\text{C} - 100\% \text{ RH}$ ,  $w_{a,in} = 5.8 \text{ g kg}^{-1}$ ,  $m_{a,in} = 6\,000 \text{ m}^3 \text{ h}^{-1} = 2.09 \text{ kg s}^{-1}$

Dehumidified air at desiccant wheel outlet :  $t_{a,l} = 21^\circ\text{C}$ ,  $w_{a,l} = 1.4 \text{ g kg}^{-1}$  (manufacturer data),  $\text{RH} = 9\%$

Dehumidification capacity :  $E = m_{a,in} (w_{a,in} - w_{a,l}) = 8.89 \cdot 10^{-3} \text{ kg s}^{-1} = 32.02 \text{ kg h}^{-1}$

Refrigeration capacity needed :  $\phi_r = m_{a,in} (h_{a,in} - h_{a,l}) = 39.0 \text{ kW}$

Regeneration air :

$t_{a,r,in} = 15^\circ\text{C} - 80\% \text{ RH}$ , needed to be heated up to  $t_{a,2} = 115^\circ\text{C}$  (manufacturer data)

Flow rate :  $1\,700 \text{ m}^3 \text{ h}^{-1}$  (manufacturer data)  $\Rightarrow m_{a,r} = 0.57 \text{ kg s}^{-1}$

Heating capacity needed :  $\phi_h = m_{a,r} (h_{a,2} - h_{a,r,in}) = 58.3 \text{ kW}$

All these values are represented Figure 6

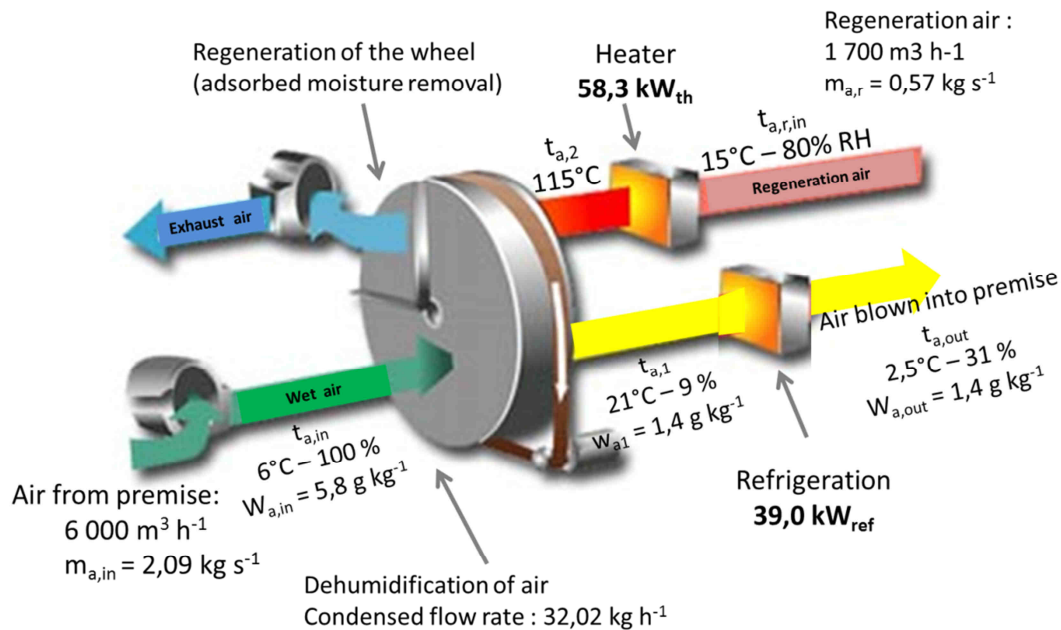


Figure 6: key values for the desiccant wheel technique (Dessica DT 6000 device)

The blown air being a bit more dry than for other techniques (31% RH i/o 40%), the drying duration is shortened of approximately 10 minutes, in respect of the dehumidification capacity of the wheel.

As far as the evaporation temperature required to cool the air down to  $2,5^\circ\text{C}$  could be  $-2,5^\circ\text{C}$  (pinch = 5K), the COP of the refrigeration system is similar to the one retained for the liquid desiccant technique, that is 4.7. According to these assessments, following energy consumption are expected:

	Demand (kW)	Consumption (kWh)
Refrigeration	$39,0 \text{ kW}_{refrig.} / 8,3 \text{ kW}_{elec}$	$13,1 \text{ kWh}_{elec}$
Heating	$58,3 \text{ kW}_{elec}$	$92,3 \text{ kWh}_{elec}$
Defrosting <sup>(*)</sup>	<i>neglected</i>	<i>neglected</i>
Total		$105,4 \text{ kWh}_{elec}$

(\*) the low humidity level permitted with this technique leads to propose to neglect the defrosting consumption, that remains an approximation to be discussed.

## 6. CONCLUSION

Following table summarises the electric energy consumptions for the different techniques:

	Cooling (kWh)	Heating (kWh)	Total (kWh)	Obs.
Dew point	26.5	47.4	73.9	Defrosting needs (6.4 kWh) included in heating needs
Liquid desiccant	17.4	108.9	126.3	Without condensation heat recovery
Liquid desiccant	17.4	9.8	27.2	With condensation heat recovery
Desiccant wheel	13.1	92.3	105.4	Without heat recovery on regeneration air

These results are based on assumptions that have to be kept in mind : (i) the electricity consumption of fans is assumed to be the same for the three techniques, and (ii) the electricity consumption of pumps for the liquid desiccant technology is not taken into account.



Up to now, the advantages of liquid desiccant systems were underlined for classical air conditioning systems. The present work points out the potential of this technique in air treatment for food process premises in food industry. It shows that a significant reduction of the electric consumption can be expected if an efficient heat recovery system on the condenser of the chiller is used. Otherwise, the liquid desiccant technique is out of range. According to the complexity and to the size of the required equipment, this technique is to be reserved for high treated air flow rates, typically above  $10\,000\text{ m}^3\text{ h}^{-1}$  according to Dessica experience. Another advantage of this technique is related to the “air disinfection effect” of chlorine ions contained in the desiccant. This effect mentioned by different authors could be a real added value for food industry premises. In any case, the problem of corrosion and of aerosols entrainment in the supply air has also to be addressed.

Even if the energy consumption of the desiccant wheel technique remains higher than the energy consumption expected with a classical dew point system, the absence of frosting and the easiness of implementation of this technique explain its frequent use, especially for air treatment in small premises where low flow rates are expected. An optimization of the internal layout of the device (for instance internal plate heat exchanger between exhaust wet air and regeneration air) and of the driving parameters of the wheel (rotation speed, regeneration temperature) could lead to substantial energy savings, which is a continuation of this work.

## 7. ACKNOWLEDGEMENTS

The authors acknowledge the French National Research Agency for his support to the Ecosec project ANR-12- ALID-005.

## 8. NOMENCLATURE

t	temperature ( $^{\circ}\text{C}$ )	Subscripts	
RH	relative humidity (%)		
w	air humidity ration ( $\text{kg kg}^{-1}$ )		
h	enthalpy ( $\text{kJ kg}^{-1}$ )	r	refrigeration
$h_c$	heat transfer coefficient ( $\text{kW m}^{-2}\text{ K}^{-1}$ )	h	heat, heating
$h_m$	mass transfer coefficient ( $\text{kg m}^{-2}\text{ s}^{-1}$ )	m	mass
m	mass flow rate ( $\text{kg s}^{-1}$ )	a, air	air
Le	Lewis number	s	desiccant
x	concentration ( $\text{kg kg}^{-1}$ )	ideal	related to ideal exchanges
$\varepsilon$	exchange efficiency (-)	i, in	inlet
E	water flow rate ( $\text{kg s}^{-1}$ )	o, out	outlet
$\phi$	heat flux ( $\text{kW}$ )	r	regeneration
COP	$\Phi_0 / W$ (-)		

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