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LIQUID DESSICCANT TECHNOLOGY APPLIED TO TEMPERATURE AND MOISTURE MANAGEMENT IN THE FOOD INDUSTRY

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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.
This paper compares the energetic performances of two humidity control techniques: a classical one based on dew point air treatment, and a second one based on the use of liquid desiccant (LiCl and $CaCl_2$).
The results show that the liquid desiccant system based on LiCl solution is very competitive compared to the classical dew point treatment.

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. 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. 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, typically one-fourth that of a classical vapor-compression air conditioner (Lowenstein, 2008).

These specificities explain the renewed interest in this forgotten technique. Number of publications (e.g. Elsayed 1994, Hassan 2007, Condé-Petit 2009) highlight the interest of this technology, especially in the domain of air conditioning in dry and hot climates. 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.

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 the analysis of the potential of liquid desiccant system based on $CaCl_2$ or LiCl solutions in order to manage both temperature and moisture in food plants. It also aims to compare the energy consumption of this technique to the energy consumption of classical dew point treatment.

This paper compares the performances of these two techniques on a typical food processing premises.

2. DESCRIPTION OF THE REFERENCE CASE STUDY

A typical food processing premises has been investigated by IRSTEA:

- Surface of the food processing premise: 100 m²
- 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^3 h^{-1}
- Blast air conditions: 2.5 °C / 40% RH, corresponding to a dew point of -8.6°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

3.1. Air evolution on Mollier diagram

Figure 1 shows the evolution of air on Mollier diagram.

The refrigeration and heating needs are classically calculated by: $\phi_r = m_a\left(h_{a,i} - h_{a,d}\right)$ and $\phi_h = m_a\left(h_{a,d} - h_{a,o}\right)$. The amount of condensed water is calculated by $E = m_a\left(w_{a,i} - w_{a,d}\right)$

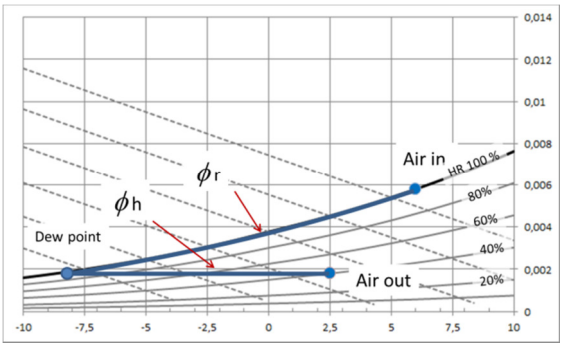

Figure 1: Air evolution on Mollier diagram

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^{-1}	5,79E-03	1,80E-03	1,80E-03
h	kJ kg^{-1}	20,5	-4,2	7,1
ρ	Kg m^{-3}	1,25	1,33	1,28

The use of the air characteristics of air presented Table 1 leads to following needs:

Refrigeration: $\phi_r =$ 51.5 kW
Heating: $\phi_h =$ 23.4 kW

A water mass balance leads to a water flux of 8.34 10^{-3} kg s^{-1}, that is 30 l h^{-1}, allowing the premise being dried in ~ 1h40, consistent with the observations of Irstea.

3.2. Energy consumption of the system

In order to ensure the cooling of air down to the dew point (-8.5°C), the evaporating temperature of the refrigeration device is assumed to be 5K below, that is t_0 = -13.5°C. The condensing temperature being fixed to t_k = 32°C, the Carnot efficiency can be evaluated at $COP_{th} = \frac{T_0}{T_k - T_0} = 5.7$.
Defining the cycle efficiency by $\eta_c = \frac{COP_r}{COP_{th}}$, the real coefficient of performance of the refrigerating machine can be calculated by $COP_r = \eta_c\, COP_{th}$. The use of a classical value of $\eta_c \approx 0.6$, leads to a real COP value of 3.4.

Hence the energy consumption of the system can be evaluated as:

Refrigeation: $\phi_r =$ 15.1 kW electric for the compressor
Heating: $\phi_h =$ 23.4 kW for the electric resistance
Total: 38.5 kW electric.

In this approach, it is assumed that heating is ensured by an electric resistance (this often the case) and not by heat recuperation on the refrigeration unit. This assumption will be resumed for the liquid desiccant systems. Note that the energy consumption related to defrosting is not taken into account, thus overstating the global energy consumption of the system.

4. PERFORMANCE OF THE LIQUID DESICCANT TREATMENT TECHNIQUE

4.1. Basic principle of liquid desiccant systems

Typical liquid desiccant systems have two primary components, an absorber and a regenerator (Cf fig 6). Cooled liquid desiccant flows down into the absorber, the process air flows flowing upward in the opposite direction. Heat and moisture is transferred from the air to the desiccant in the absorber. As the desiccant absorbs water from the air, it becomes diluted and heated. Upstream of the regenerator, the solution is heated again, and then flowed in the regenerator. The hotter solution is, the higher is the release of gathered moisture, transferred into the counterflowing outside air stream.

The basic principle is to ensure a contact between the treated air (hot and wet) and a liquid desiccant. The temperature and the solute concentration of the desiccant are adjusted to ensure at the same time:

(i) the air dehumidification – the water vapor pressure above the solution being dependent on its solute concentration and of its temperature, and

(ii) the air cooling – the temperature of the desiccant imposing the air temperature at the outlet of the exchanger

The contact between air and the desiccant solution can be ensured by different techniques, e.g. spraying systems, drainage on a packing or on Raschig rings, bubbling, membrane contractors.
Spraying and/or drainage on packing remains mainly used as they are quite easy to implement – looking a bit like a classical cooling tower.

Different solutes can be used as desiccant media: NaCl, $CaCl_2$, LiCl, LiBr and triethylene glycol are often cited. The literature mentions that $CaCl_2$ presents interesting characteristics: food-friendly, cheap and quite efficient. LiCl is recognized as being more efficient but slightly more expensive and less food-friendly. These two solutes are classically used in air conditioning systems. They are preferred here, because the presence of chloride is recognized as an inhibitor of the microbial growth.

4.2 Liquid – vapor equilibrium conditions

Calculation methods of thermophysical characteristics of $CaCl_2$ and LiCl solution can be found in the literature. One of the most complete and interesting work on this subject is proposed by Condé-Petit (2009). The calculation methods proposed in his work are fully implemented here.

The proposed equation allows the calculation of basic parameters (crystallization point, vapor pressure above the solution, heat capacity, density, enthalpy, enthalpy of dilution) as a function of the solute concentration and of the temperature. The developed calculation tool is coupled with a calculation tool of wet-air characteristics (mainly through water partial vapor pressure) in order to have a direct link between desiccant solution and wet air.

Figure 2 shows typical values one can expect with $CaCl_2$ and LiCl.

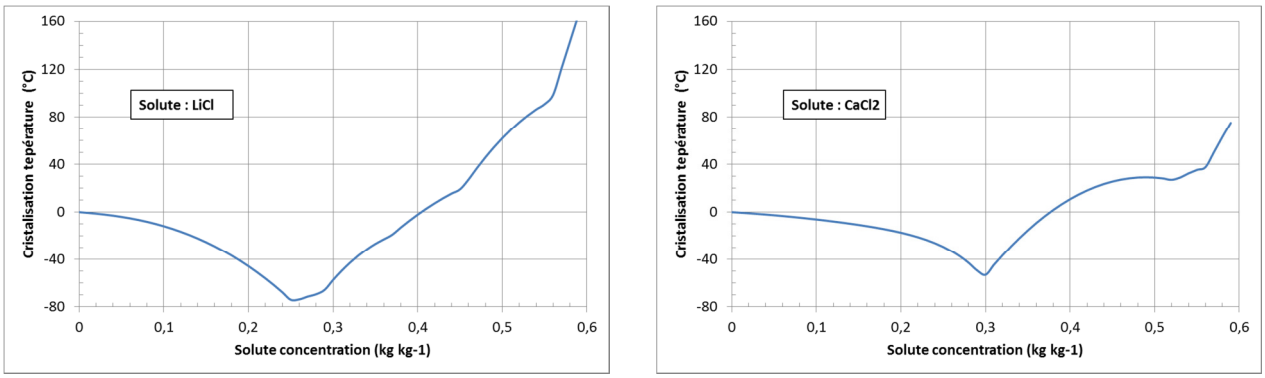

Figure 2: Crystallisation temperature of the desiccant solution depending on its concentration

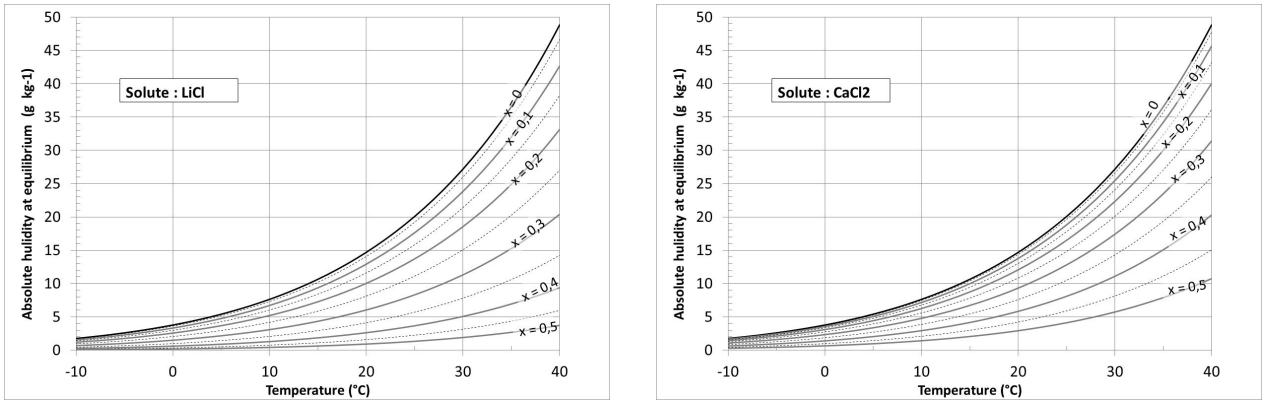

Figure 3: Absolute humidity above a desiccant solution depending on its concentration and temperature

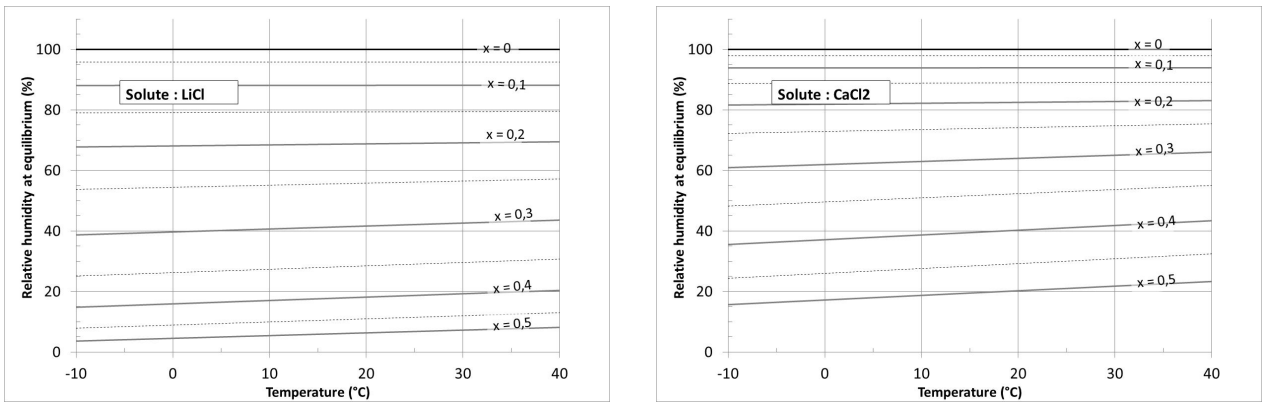

Figure 4: Relative humidity above a desiccant solution depending on its concentration and temperature

4.3. Air evolution on Mollier diagram

As the air is simultaneously treated in temperature and humidity, its evolution on Mollier diagram is assumed to follow the line defined by the slope $\frac{h_{a,o}-h_{a,i}}{w_{a,o}-w_{a,i}}$ as presented below.

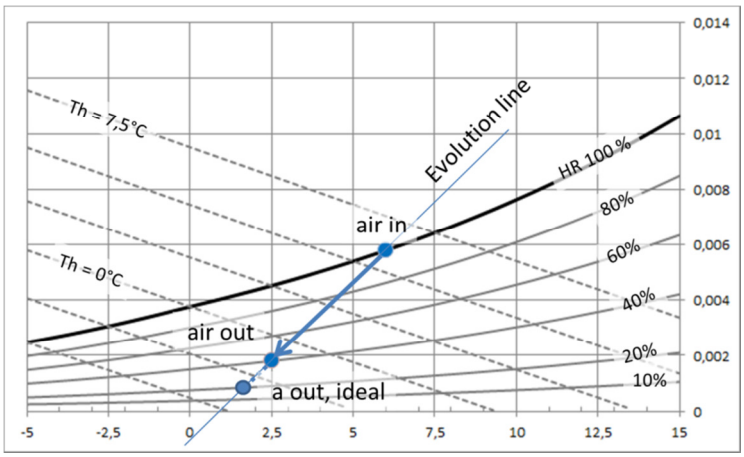

Figure 5: air evolution on the Mollier diagram (desiccant solution treatment)

If the heat and mass transfer were ideal, the air would come out at equilibrium with the desiccant solution (temperature and water vapor pressure).

Given the lack of experimental data on heat and mass transfer efficiencies, we propose to introduce an heat exchange efficiency by: $\eta_e = \frac{h_{a,o}-h_{a,i}}{h_{a,o,id}-h_{a,i}}$.
Moving along the evolution line defined above leads to the calculation of the ideal absolute humidity at the outlet of the exchanger by: $w_{a,o,id} = w_{a,i} + \frac{1}{\eta_e}\left(w_{a,o} - w_{a,i}\right)$

As far as the actual air outlet conditions are fixed by the process (blast conditions: 2.5°C / 40% RH, thus imposing a drying time equal to the one obtained for dew point treatment), the characteristics of the desiccant solution can be computed from this in terms of temperature and concentration.

4.4. Premise air treatment

The premise air treatment implies a dilution and a heating of the desiccant solution. This dilution is related to the air drying (direct condensation of vapor inside the desiccant). The heating is related to the sensible heat exchange (air is hotter than the desiccant), and mainly to the latent heat of vapor condensation.

The temperature drop on the desiccant side depends on the desiccant flow rate. Reasonably, we propose to adopt a desiccant flow rate delivering a temperature drop of 5K.

To ensure the required air treatment with a $CaCl_2$ desiccant, it would have been necessary to use a concentration of 0.39 kg kg^{-1} and a temperature lower than 2.5 °C. This point is below the crystallization curve of $CaCl_2$ solution: therefore, the use of this solute is not possible for this application.

Figure 6 (left side) shows the calculation results obtained with a LiCl desiccant.

4.5 Desiccant regeneration

The desiccant being heated and diluted, it is necessary to ensure its re-treatment in order to use it again in the loop. This regeneration is obtained by heating the desiccant and by contacting it with outside ambient air. The desiccant must be heated enough in order to ensure an equilibrium vapor pressure higher than the one of ambient air: in this condition, desiccant will lost water and concentrate.

A similar approach than the one described above is used. Figure 6 (right side) shows the results obtained for the regeneration.

4.6. The internal heat exchanger

Heating the desiccant from 7.5°C (outlet of the cold exchanger) up to 34°C (temperature necessary to ensure its regeneration) would consume a large amount of energy. Cooling the desiccant from 27.1°C (outlet of the regenerator) down to 2.5°C (temperature necessary to ensure the air treatment in the premise) would also consume a large amount of energy. The installation of an internal heat exchanger easily enhances the global performances of the system.
Nevertheless, the addition of a heater and a cooler remains necessary. Figure 6 (middle) shows the calculation results obtained for the whole system. It also shows the cooling and heating capacities consumed by the system.

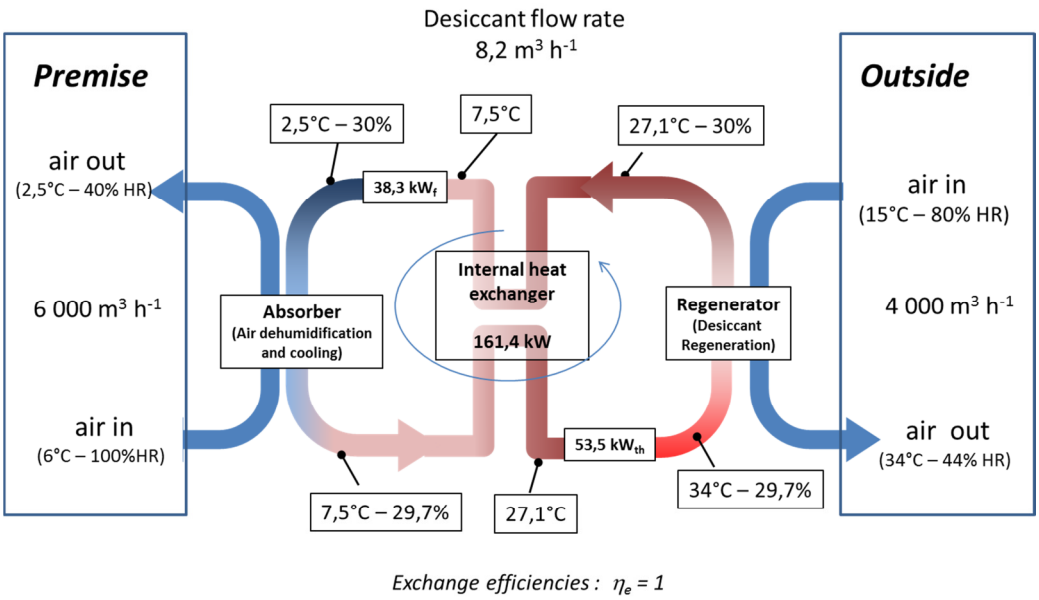

Figure 6: Key characteristics of air and LiCl desiccant solution necessary to ensure the premise air treatment and to regenerate the solution.

4.7. Energy consumption of the system

Figure 6 indicates that a cooling capacity of 38.3 kW is necessary to lower the temperature of the desiccant from 7.5°C (after heat recuperation) down to 2.5°C. This cooling can be ensured by a refrigeration facility working with an evaporating temperature of -2.5°C (pinch of 5 K) and a condensing temperature of 32°C.

As previously, we propose to adopt a cycle efficiency of η_c = 0.6, leading to a real COP of $\frac{273.15-2.5}{32-2.5}$ x 0.6 = 4.7 and to an electric consumption of 38.3 / 4.7 = 8.1 kW_{elec} at the compressors, which is half the electric consumption of the dew point air treatment.

Figure 6 also indicates that a heating capacity of 53.5 kW is necessary to heat the desiccant from 27.1 °C (after heat recuperation) up to 34°C. This heating is 2.3 times greater than the heating capacity required for the dew point air treatment.
According to the temperature levels (34°C on the desiccant and 32°C on condensing temperature) one can note that this additional heating capacity can easily (but not totally) be ensured by heat recuperation on the refrigerating unit. In this case, the energetic advantage of the desiccant solution would be largely enhanced.

5. CONCLUSIONS AND PERSPECTIVES

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 reduction of the electric consumption for the cold production can be expected while, the regeneration of the desiccant is highly heat consuming. However, this heating need can be largely provided by heat recovery on the refrigeration unit. In this case, the global energy consumption of desiccant systems would be very competitive compared to classical dew point air treatment systems.
The approach proposed in the present communication will benefit from being improved on some points: introduction of exchange efficiencies (heat and mass), coupling of heat and mass transfer (Chilton analogy), and accounting for the energy consumption of the annexes (e.g. fans, pumps, defrost). The problem of corrosion and of aerosols entrainment in the supply air has also to be addressed.
Nevertheless, it confirms the potential and the interest of liquid desiccant systems application in the food industry.

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NOMENCLATURE

Symbols

E	mass flux	kg s^{-1}
h	enthalpy	kJ kg^{-1}
RH	relative humidity	%
m	flow rate	kg s^{-1}
t	temperature	°C
T	temperature	K
w	absolute humidity	kg kg^{-1}
V	premise volume	m^3

Greek symbols

ϕ	heat flux	kW
η	efficiency	-

Subscripts

a	air
d	dew point
i	inlet
o	outlet
id	ideal
r	real
th	theoretical
c	cycle
e	exchanger

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