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Limits of feasibility and energy consumption of desiccant and evaporative cooling in temperate climates

ABSTRACT

Desiccant and evaporative cooling are alternatives to compressor driven air conditioning systems. In low load buildings, the indirect evaporative cooling system can be sufficient to reach comfort. In high load buildings, the desiccant cooling mode can take over from indirect evaporative cooling but in very high load buildings, it turns out to be inadequate.
The limits of feasibility of the indirect evaporative system and desiccant cooling systems have been assessed. The methodology is based on simulations which have been carried out for different temperate climatic zones, different thermal inertia levels, different internal load levels and different solar gain levels.
The annual energy consumption has been assessed. Since water consumption cannot be neglected when considering operating costs, it has been evaluated.

INTRODUCTION

Evaporative cooling and desiccant cooling are classified in low energy technologies which can be an alternative to conventional mechanical cooling [1]. These technologies consist in cooling the supply air by the evaporation of liquid water within this air.
If the supply air is cooled directly by water evaporation, the process is called direct evaporative cooling. This technology has proven to be efficient in arid climates. In Europe, where the climate is rather warm and quite humid, the indirect evaporative cooling systems are more appropriate than the direct evaporative cooling devices [2]. The desiccant cooling technique consists in dehumidifying the supply air in order to cool it then by direct evaporation. Both techniques do not require any refrigerant and can reduce energy consumption and electricity demand peaks. However, their cooling power is limited. This paper deals with the feasibility of desiccant and indirect evaporative cooling in temperate climates. The desiccant cooling system provides more cooling power than indirect evaporative cooling system, but it also uses more electricity and requires thermal energy to reactivate the desiccant wheel. A comparison of energy consumption between both system is carried out.

METHODOLOGY

Description of the systems

In a desiccant cooling system, the outside air is first dehumidified through a desiccant wheel (Figure 1). During the adsorption of vapour water in the wheel, the air dry-bulb temperature is increased. Then, this air is cooled in a heat exchanger by the return air which is previously cooled in a humidifier. Before to be supplied to the room, the process air is cooled further in a humidifier.

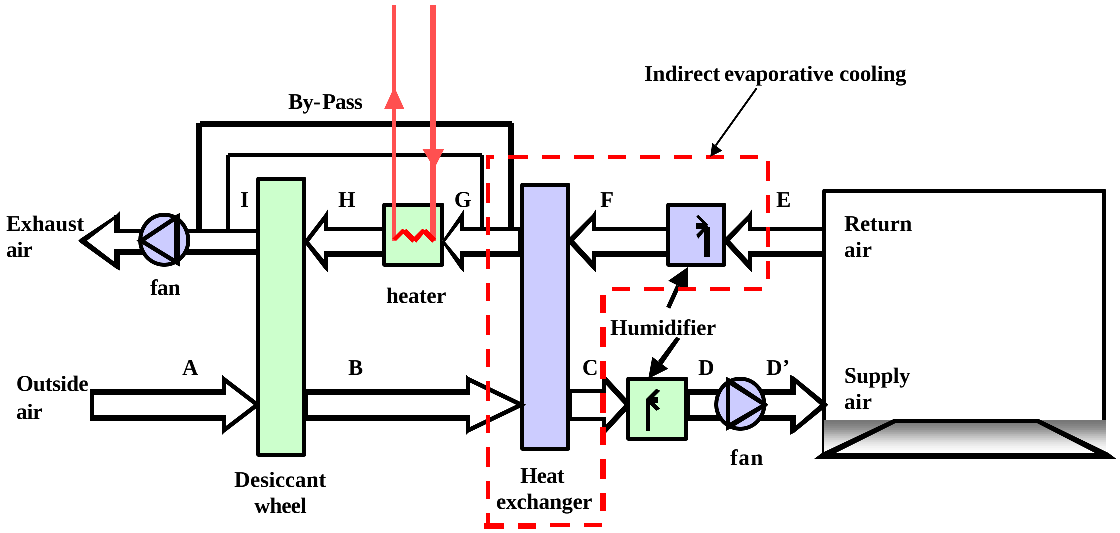

Figure 1: Schematic of a desiccant and an indirect evaporative plant

The return air passes through an evaporative cooler and is used to cool down the process air in the heat exchanger. Then, it is heated to regenerate the desiccant wheel. The regeneration temperature ranges from 50°C to 100°C. It is common to by-pass a fraction of return air (about 20%) from the desiccant wheel in order to reduce the regeneration heat consumption. In the case of the indirect evaporative cooling system, the cooling of the outside air is achieved only in a heat exchanger thanks to the heat transfer with the return air beforehand cooled in a humidifier (Figure 1).

Simulations

Since the cooling power of these systems is limited and very dependent on climatic conditions, anticipate control strategies are required. To assess the feasibility of both systems, simulations have been carried out by using a dynamic building simulation tool, called ConsoClim [3]. The systems have been modelled and have been implemented in a building simulation software.

Control strategy

The control strategy used for simulations is presented in Table 1. During occupancy, the supply air flow rate is the minimum required and the humidifier is off as long as the room temperature does not reach 23°C. When the room temperature is over 24°C, the supply air flow rate is maximum and the humidifier is on. In the range between 23° and 24°C, the humidifier is on and the supply air flow rate varies between the minimum and the maximum air ventilation rate proportionally to the room temperature. During non occupancy, the cooling plant stops when the indoor temperature is lower than 20°C. From 21°C, the air flow rate is maximum and the humidifier is on. Additional controls are added in order to avoid dysfunction of the plant. In case of the desiccant plant, the desiccant wheel and the direct humidifier operate as soon as the indoor temperature is higher than 25°C. During the non occupancy, the desiccant wheel and the direct humidifier relieve the indirect evaporative cooling mode as soon as the indoor temperature overtakes 26°C. This strategy, which limits the use of desiccant mode, is adapted in the case of a purchased thermal energy such as gas. If solar energy is used for desiccant wheel regeneration, the strategy would favor the use of desiccant mode.

Table 1: Control matrix for summer (indirect evaporative cooling and desiccant cooling)

	T_i (°C)	19	20	21	23	24	25	26	27
DURING OCCUPANCY									
Air flow rate	min = 1 air change/h max = 2 to 6 ac/h (desiccant plant) 4 to 8 ac/h (evaporative plant)	min	min	min	min	max	max	max	max
Indirect humidifier control		off	off	off	**on**	**on**	**on**	**on**	**on**
Desiccant wheel + direct humidifier		off	off	off	off	off	**on**	**on**	**on**
DURING NON OCCUPANCY									
Air flow rate	min = 1 air change/h max = 2 to 6 ac/h (desiccant plant) 4 to 8 ac/h (evaporative plant)	0	0	max	max	max	max	max	max
Indirect humidifier control		off	**on**	**on**	**on**	**on**	**on**	**on**	**on**
Desiccant wheel + direct humidifier		off	off	off	off	off	off	**on**	**on**

Sizing parameters

In the case of the indirect evaporative cooling plant, the parameters are:

- evaporative cooler efficiency : 0.9
- rotary heat exchanger efficiency : 0.75
- airflow rates : 4 ac/h, 6 ac/h, 8 ac/h.

In the case of the desiccant cooling plant, the parameters are:

- direct evaporative cooler efficiency : 0.85
- indirect evaporative cooler efficiency : 0.95
- rotary heat exchanger efficiency : 0.8
- regeneration temperature : 95°C
- desiccant wheel performance: absolute humidity depression of 4.9 g/kg at 34°C dry-bulb, 21°C wet-bulb and 95°C regeneration temperatures
- airflow rates : 2 ac/h, 4 ac/h, 6 ac/h.

Meteorological data

Three climatic zones are studied: Trappes (closed to Paris), Lyon and Nice. According to Ashrae Handbook [4], the dry- and wet-bulb temperatures in Table 2 represent values that have been equaled or exceeded by 2.5% of the total hours during the cooling season (June to September).

Table 2: Climatic data as described in [4]

	Dry bulb temperature (2,5%)	Wet bulb temperature (2,5%)
Trappes (closed to Paris)	30°C	20°C
Lyon	32°C	22°C
Nice	29°C	23°C

The simulations for feasibility study have been carried out on a succession of reference hot days, based on a set of hourly climatic data collected on several years [5].

Case studies

The test cases are office building zones with the following characteristics:

- floor area A_{room} : 3 m × 5 m = 15 m^2
- front wall : 3 m × 2.7 m including a window area, A_{glaz}, of 3 m × 1 m
- room height : 2.7 m

- two thermal inertia are taken into account, medium and very high.
- double glazing window: solar factor F_{glaz} = 0.25 and 0,5 for the room with medium inertia and F_{glaz} = 0.25 and 0,75 for the room with very high inertia.

The other parameters are:

- occupation : 9 h to 18 h weekdays
- internal sensitive gains (convection part : 50%): 10 W/m^2 and 30 W/m^2 during occupancy
- internal moisture gains: 110 g/h during occupancy
- minimum ventilation: 1 ac/h of fresh air during occupancy
- orientation: East

Feasibility of indirect evaporative and desiccant cooling systems

The simulations on the reference hot days for each case provide the domain of feasibility of both systems (Figures 2 to 7). The maximum building loads have been calculated with the dynamic simulation tool [3]. One can notice that with a static calculation method, the building loads would be about 30% higher in medium inertia buildings and about 40% higher in very high inertia buildings. The maximum operative temperature is the average operative temperature on the three hottest hours of occupancy. A limit of feasibility at 27°C has been arbitrarily defined.

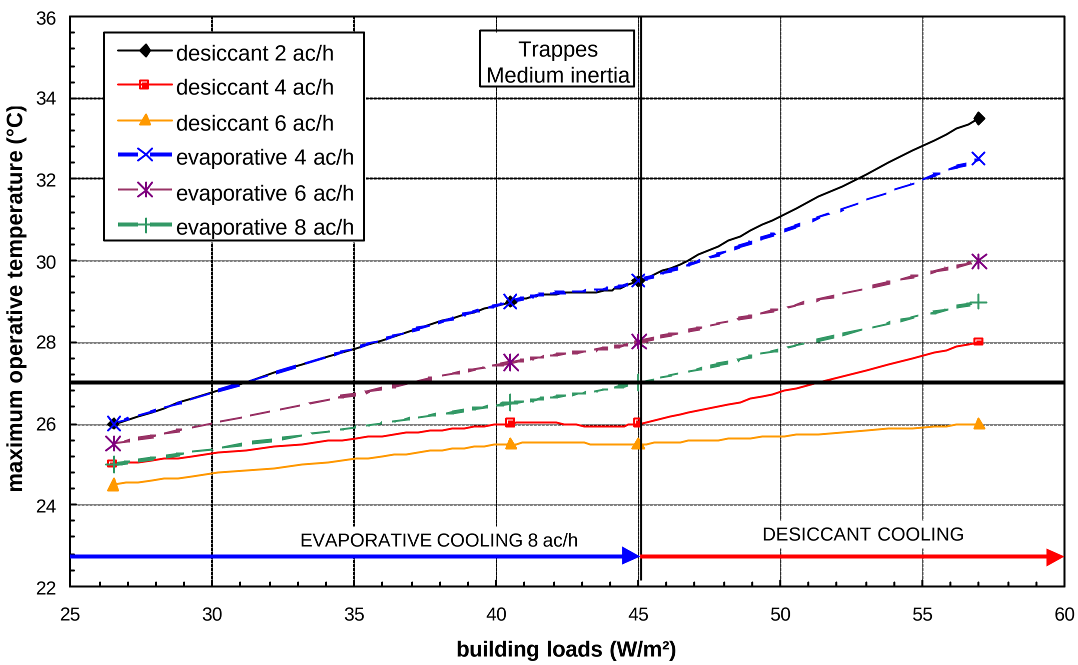

Figure 2: Maximum operative temperature on a reference hot day versus the maximum building loads for a medium inertia room in Trappes

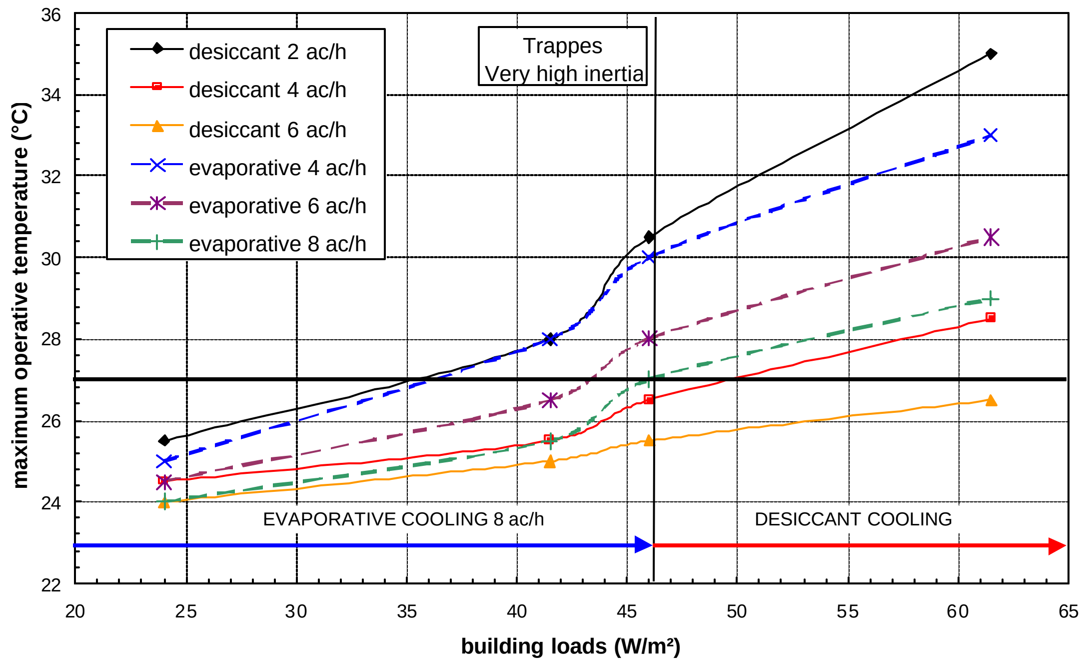

Figure 3: Maximum operative temperature on a reference hot day versus the maximum building loads for a very high inertia room in Trappes

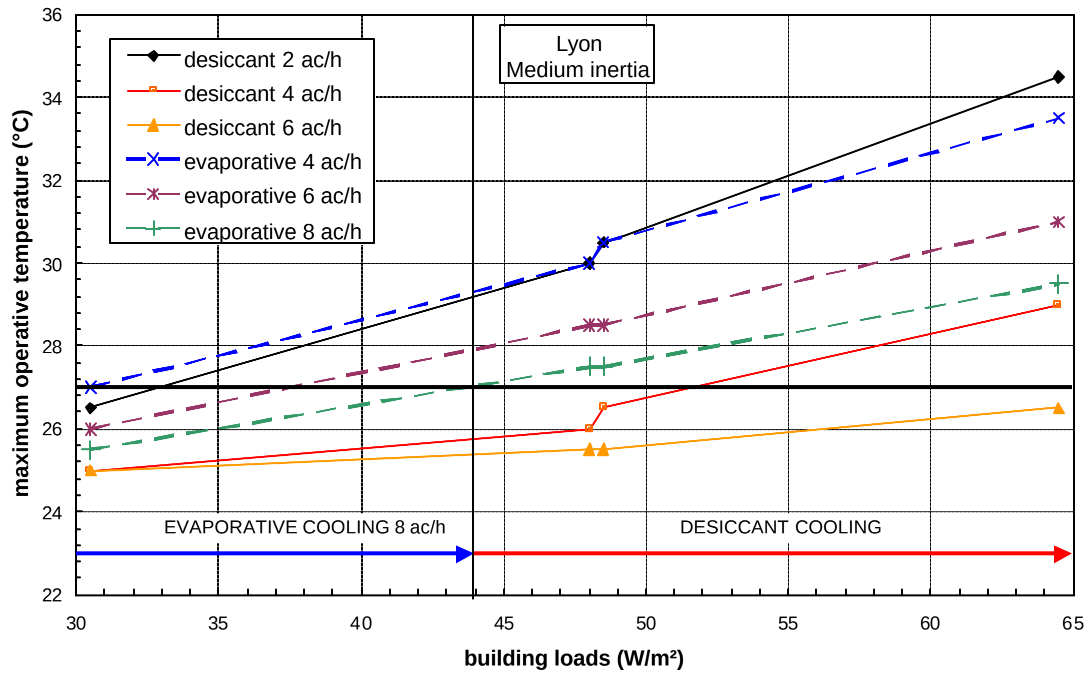

Figure 4: Maximum operative temperature on a reference hot day versus the maximum building loads for a medium inertia room in Lyon

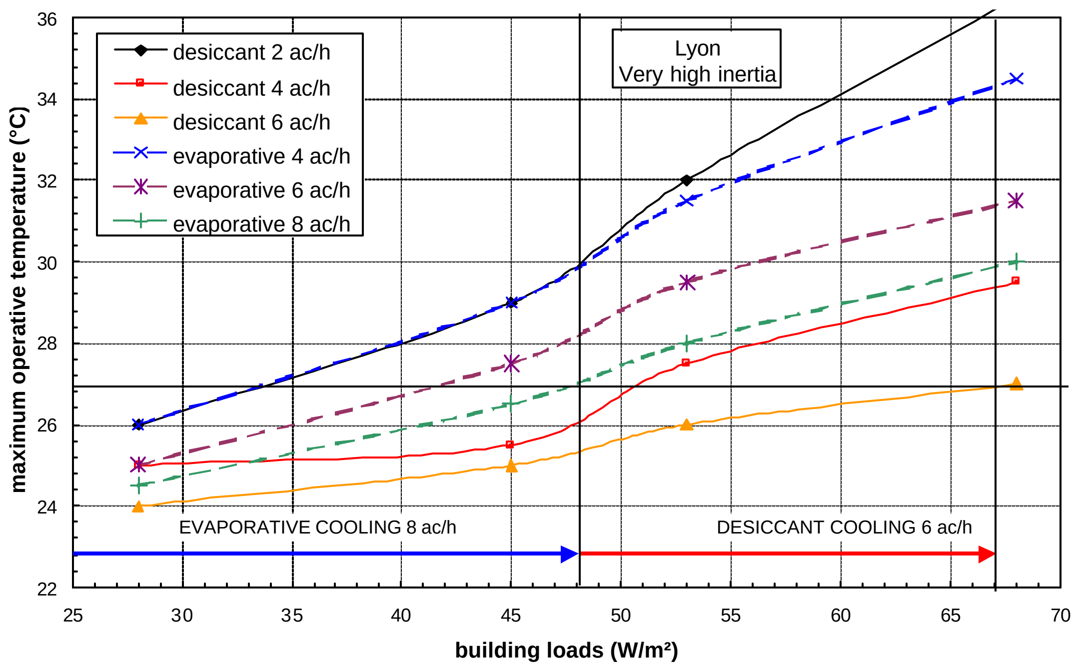

Figure 5: Maximum operative temperature on a reference hot day versus the maximum building loads for a very high inertia room in Lyon

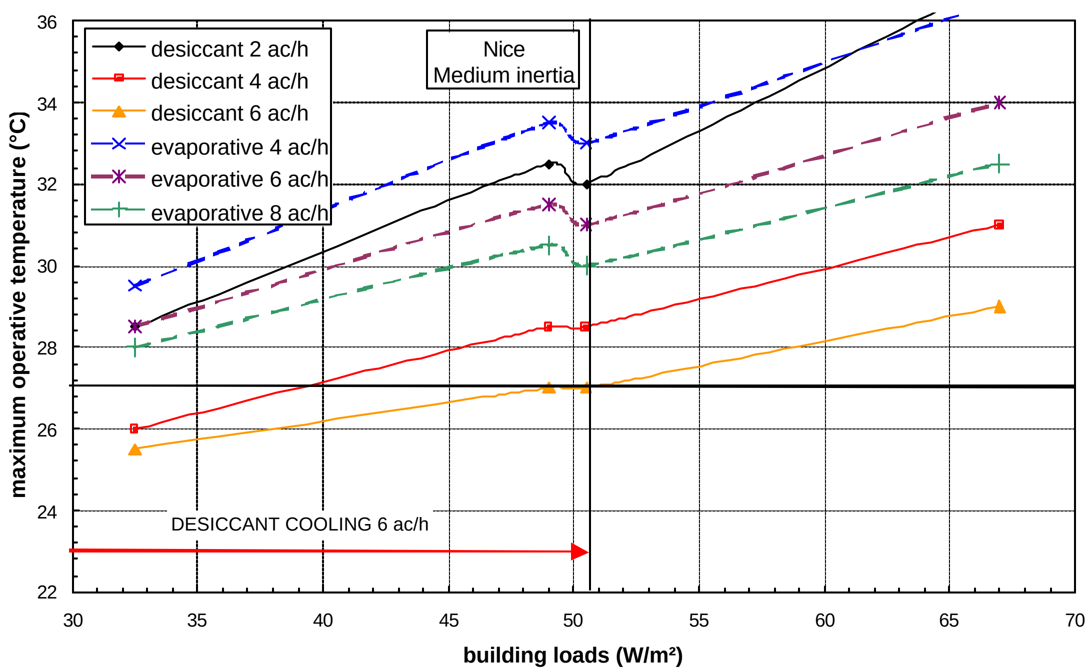

Figure 6: Maximum operative temperature on a reference hot day versus the maximum building loads for a medium inertia room in Nice

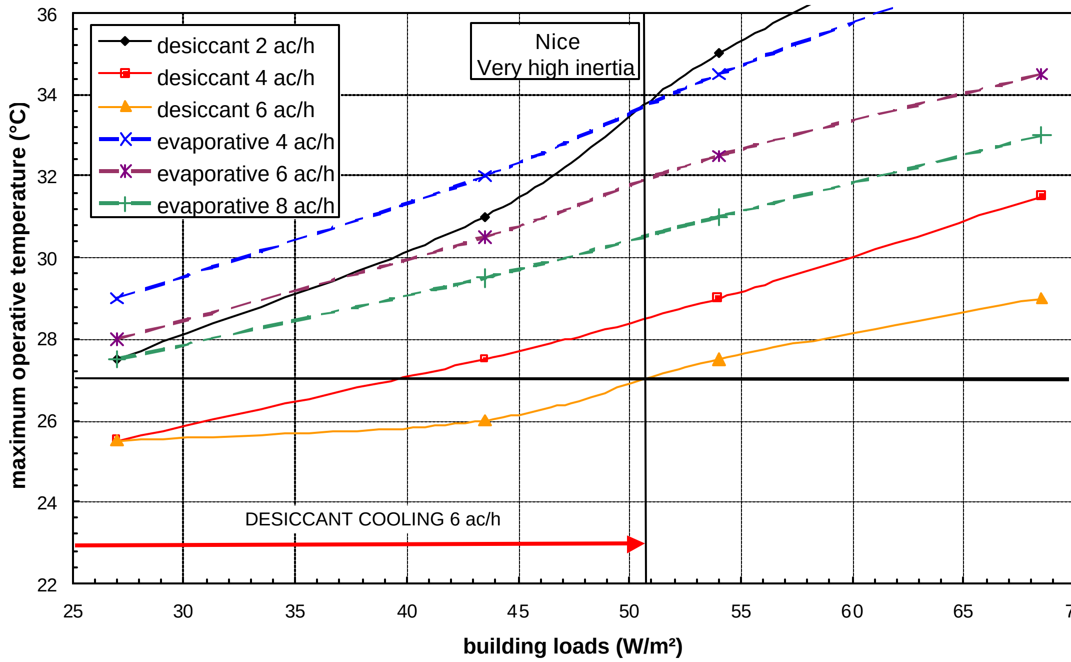

Figure 7: Maximum operative temperature on a reference hot day versus the maximum building loads for a very high inertia room in Nice

Four points are used for drawing each curve, which correspond to distinct levels of internal gains and solar gains giving the following couples (30 W/m² and F_s= 0.25), (10 W/m² and F_s = 0.5 or 0.75), (30 W/m² and F_s = 0.25) and (30 W/m² and F_s = 0.5 or 0.75). This is the reason why the curves are not straight lines.

The performance of a 2 ac/h desiccant plant is quite equivalent to a 4 ac/h evaporative plant for Trappes and Lyon climates but is better in Nice where the climate is more humid.

In the case of low building loads, the indirect evaporative cooling system can maintain comfort conditions. Since the initial and operating costs of evaporative cooling are lower than desiccant cooling, it should be preferred.

In the case of medium building loads, the indirect evaporative system can not strictly maintain 27°C operative temperature unlike the desiccant cooling system. In case of high building loads, the desiccant cooling system is much more adapted than the evaporative cooling system since its cooling power is higher. However, the desiccant at 6 ac/h is insufficient in Nice to control indoor temperature for buildings with loads above 50 W/m². Indeed, the cooling capacity is lower in Nice since the climate is humid.

Energy and water comparison

The office room in Trappes with a medium inertia, 30 W/m² of internal gains, and a solar factor of 0.25 is chosen. The dynamic building loads are of 45 W/m². Both systems can maintain 27°C operative temperature on a reference hot day. The feasibility study (Figure 2) shows that a 8 ac/h evaporative cooling plant or a 4 ac/h desiccant plant is enough. Simulations have been carried out on the whole cooling season for both systems.

The electric consumption of the evaporative plant is due to the two fans, the pump of the evaporative cooler and the motor of the rotary exchanger. The consumption of the auxiliaries can be neglected as compared to the fan consumption. So, the consumption of the plant will be proportional to the pressure drop and inversely proportional to the fan efficiency. An seasonal COP (from June to September) of 4.6 has been found for a plant with limited pressure drops (300 Pa has been taken for the simulation) and with a fan efficiency of 0,7.

In the case of the desiccant cooling plant, since there are additional equipment (desiccant wheel and direct humidifier), the pressure drop is higher (600 Pa). Thus, the energy consumption in evaporative mode will be higher than for the evaporative plant. The electric consumption of the auxiliaries are also negligible as compared to the fan consumption. The thermal energy consumption for the wheel regeneration should be added. In order to compare with the evaporative plant, the COP has been calculated as:

$$COP = \frac{\text{total cooling energy}}{(\text{thermal energy}/2.58 + \text{electric energy})}$$

The factor 2.58 accounts for the transformation of primary energy in electricity in France.
The seasonal COP is 2,2 (without consideration of thermal energy, the COP is 3,3). In both cases, a more optimised control strategy could reduce a bit more the energy consumption.
The water consumption is 36 L/m²/year for the desiccant plant and 42 L/m² for the evaporative plant. Despite the desiccant uses two evaporative coolers, its water consumption is lower. This is due to the fact that the maximum airflow rate is twice lower.

Conclusions

The indirect evaporative cooling system can be used in temperate climates provided that the climate is not too humid and the building loads are limited. The desiccant cooling plant can be used in buildings with much more thermal loads but appears not to be adapted for very high thermal load buildings or high thermal load buildings in humid climates such as Nice.
The energy performance of the evaporative cooling system appears to be interesting provided that pressure drops are limited and fan efficiency is high. When it can maintain comfort conditions, this system should be preferred to desiccant cooling since initial and operating costs are lower. The desiccant cooling can be considered since evaporative cooling is insufficient but the energy performance is quite low. The system can be really advantageous as compared to classical cooling solutions only if it is used with a free thermal energy source such as solar, waste heat…

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