Performance Evaluation of an Indirect Evaporative Cooler

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ABSTRACT

Nowadays, buildings are responsible for 40% of energy consumption in the European Union, according to the International Energy Agency (IEA). To reach the European objective aiming for CO_2 neutrality of buildings, the IEA has developed the Energy in Buildings and Communities program (EBC), from which Annex 85 on Indirect Evaporative Cooling (IEC) is part of. This work contributes to assess the energy performance of indirect evaporative coolers in mixed-humid climates. A reference cooling system has been chosen as a base case to evaluate the performance of the IEC. The reference cooling system is composed of terminal units, a chiller, and a cooling tower as primary cooling system. In the upgraded cooling system, the cooling tower has been replaced by an indirect evaporative cooler. The latter is a modified version of the cooling tower that comprises an additional heat exchanger for air-pre-cooling. The energy consumption of both systems over a one-year period are then compared, including auxiliaries consumption. The possibility to perform free chilling has also been investigated. The models used to describe each system component are overviewed as well as their limitations. The annual energy consumption of the systems has been computed using three control methods: temperature control, flow rate control and optimized operation. While free chilling has a significant impact on the energy consumption (30% reduction), the IEC has a more moderate effect (up to 5% reduction). It is also shown that the advantage of the IEC over the cooling tower is directly dependent on the temperature difference between the dry bulb temperature and the dew point temperature.

1. INTRODUCTION

The building sector is one of the largest energy consumers in most countries worldwide. In Europe, it is responsible for 40% of the energy consumption and 36% of the CO_2 emissions. In most buildings, energy is mainly used for heating and cooling purpose (Birol, 2018). Climate change impacts the energy performance of buildings by increasing the energy demand for space cooling. The energy consumption of buildings affects global warming, which in turn leads to the increment of such consumption, resulting in a vicious circle that needs to be broken to tackle the crucial issues of our generation. The International Energy Agency (IEA), among others, is aiming to CO_2 neutral buildings through international cooperation in the Energy in Buildings and Communities program (EBC). Annex 85 is a project of the IEA-EBC program that studies indirect evaporative cooling to evaluate the performance of such systems and quantify its impact on the buildings energy consumption. The indirect evaporative coolers are expected to limit the energy consumption increase generated by global warming.

Evaporative cooling has been used worldwide for centuries. Its interest has been renewed since it could reduce the energy used for air-conditioning significantly (Anisimov et al., 2014). The most well-known practice of evaporative cooling is direct evaporative cooling (DEC). DEC consists in the adiabatic humidification of outdoor air, which can then be used to produce chilled water or be directly supplied in the building. With this technology, the minimum temperature limit for cooling is the wet bulb temperature of the outdoor air, which is mostly suitable for warm and dry regions (Boudjabi et al., 2021). Moreover, DEC can lead to a possible contamination of the supplied air by legionella.

The main advantage of indirect evaporative cooling (IEC) over DEC is that the air can be cooled without being humidified. The inlet air is cooled by so-called secondary air that has been cooled through evaporative cooling. With this method, the minimum temperature that can theoretically be reached by the air is the dew point temperature. Jiang & Xie (2010) identified three main types of evaporative cooling: direct evaporative cooling to produce cold air, direct evaporative cooling to produce chilled water and indirect evaporative cooling to produce cold air. To fill a knowledge gap, they proposed an indirect contact evaporative cooling technology to produce chilled water.

As a part of Annex 85, this work assesses the performance of the indirect evaporative cooler proposed by Jiang & Xie, extending their work. The electricity consumption of such systems is evaluated and compared to the energy consumption of the system with a regular indirect cooling tower. The possibility to perform free chilling is also investigated.

2. METHODOLOGY

To assess the performance of the indirect evaporative cooler and its impact on building energy consumption, its performance should be compared to a reference cooling system. The comparison between both systems has been done by simulation. Section 2.1 details the components of the reference cooling system and of the indirect evaporative cooler, as well as the models used to evaluate the energy consumption of the systems. As an input, the cooling system model needs a cooling load. The cooling load has been estimated based on the simulation of an office building that is described in section 2.2. The cooling system model could not be validated with empirical data so far, but this will be done in a further study. The building and the cooling system have been simulated using meteorological data of Nancy (France). Section 2.3 describes the climate of Central Europe.

2.1 Description of the cooling system and components modeling

The cooling system that is studied in this paper is illustrated in Figure 1. The building cooling load is covered by terminal units, in this case, radiant ceilings. In normal operating mode, chilled water is provided to the terminal units by a chiller, the condenser of which being cooled by an indirect wet cooling tower. The system can also work in free chilling mode. In this case, the chiller is bypassed, and the same water is used in the cooling tower and in the terminal units. If the water produced by the cooling tower is too cold for the terminal units, it is also possible to recirculate part of the water that leaves the radiant ceilings, as indicated by the dashed gray line on the schematic. Finally, the indirect evaporative cooler (IEC) consists in the coupling of the indirect cooling tower with an air-water quasi countercurrent heat exchanger. The chilled water produced by the IEC is used to both cool down the condenser and pre-cool the outdoor air entering the cooling tower.

All the components of the cooling system are reviewed in more details here under. The model used to describe each component is developed as well as the computation of their energy consumption. It has been decided to assess the performance of the cooling systems based on their energy consumption only. The energy consumption accounts for main components consumption (*i.e.,* compressor) and auxiliaries consumption (*i.e.,* fans and pumps).

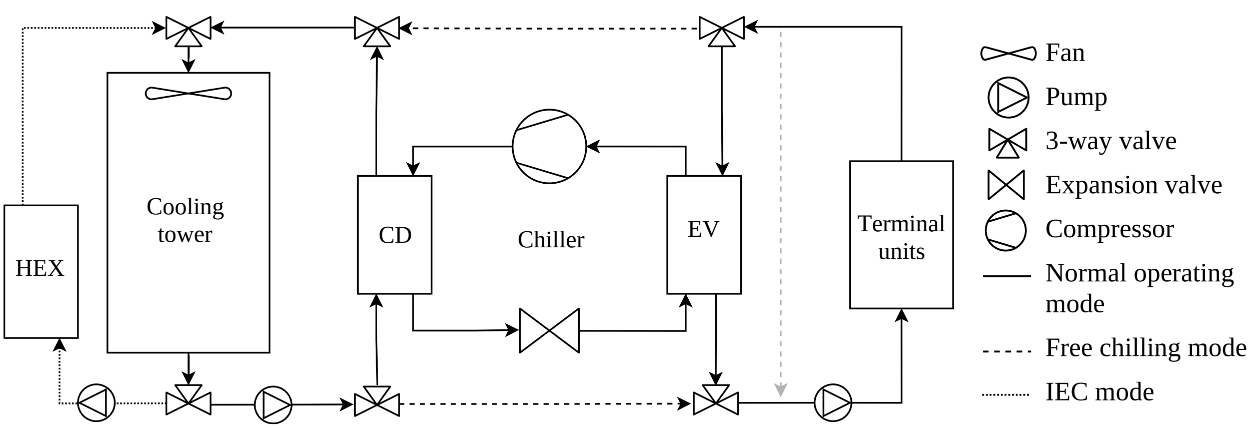

Figure 1: Schematic of the cooling system.

2.1.1 Indirect cooling tower: The indirect cooling tower is the primary cooling system of the reference case study. The model used in this paper is derived from Lebrun et al. (2004). It is assumed that the wet cooling tower is an air-water heat exchanger in which the humid air is replaced by a fictitious perfect gas, the temperature of which is the wet bulb temperature. It is thus possible to define a fictitious specific heat $c_{p,af}$:

$$c_{p,af} = \frac{h_{a,ex} - h_{a,su}}{T_{wb,a,ex} - T_{wb,a,su}} \tag{1}$$

To enhance the heat transfer in the cooling tower, the capacity flow rates between the two fluids should be balanced:

$$\dot{M}_a \cdot c_{p,af} = \dot{M}_w \cdot c_{p,w} \tag{2}$$

A balanced capacity flow rate allows to reach the minimum water temperature at the exhaust of the cooling tower while using a minimum air flow rate.

The ε-NTU method allows to define the efficiency (ε_f) and the global heat transfer coefficient (AU_f) of the cooling tower based on the fictitious perfect gas. For an indirect cooling tower, the global heat transfer is the sum of the resistances on the water side and on the air side.

$$AU_f = \frac{1}{R_f} \tag{3}$$

$$R_f = R_{af} + R_w \tag{4}$$

with R_{af}, the fictitious air resistance, that is given by:

$$R_{af} = R_a \cdot \frac{c_{p,a}}{c_{p,af}} \tag{5}$$

Variations in the air side resistance can happen due to variations in the air flow rate. The air side resistance is varied according to the following relationship:

$$R_a = R_{a,n} \cdot \left(\frac{\dot{M}_a}{\dot{M}_{a,n}}\right)^n \tag{6}$$

where the coefficient n is -0.6 (Lebrun et al., 2004).

The cooling tower can be controlled based on three variables: the air flow rate, the water flow rate, and the water exhaust temperature. The air and water flow rates are manipulated variables that can be directly adapted by varying the rotation speed of the fan and the pump, respectively. The water exhaust temperature is a controlled variable that can be only be changed indirectly. The fan and pump rotation speeds should be adapted until steady state is reached at the desired temperature set point.

When calculating the energy consumption of the cooling tower, only the fan consumption is considered. The fan consumption is related to the part load ratio of the fan.

$$PLR_{fan} = \frac{\dot{M}_a}{\dot{M}_{a,n}} \tag{7}$$

$$\dot{W}_{fan} = \dot{W}_{fan,n} \cdot \left(C_0 + C_1 PLR_{fan} + C_2 {PLR_{fan}}^2 + C_3 {PLR_{fan}}^3\right) \tag{8}$$

C_0, C_1, C_2 and C_3 are constants that have been determined based on the work of Bertagnolio (2012). The nominal electricity consumption of the fan is assumed to be 1.8% of the cooling tower nominal cooling capacity. The pump consumption is first neglected as it represents only 0.2% of the cooling tower nominal cooling capacity. It is thus not expected to significantly impact the results.

In a perfect cooling tower, the minimum water temperature that could be reached is the wet bulb temperature of the outdoor air. To decrease the chilled water temperature, the air should enter the cooling tower at a lower temperature. This is what is done in the indirect evaporative cooler that is presented in the next section.

2.1.2 Indirect evaporative cooler: the indirect evaporative cooler is the combination of two components, the indirect cooling tower, and an air-water heat exchanger. Part of the chilled water produced by the cooling tower is diverted and used to pre-cool the outdoor air that enters the cooling tower. The additional air-water heat exchanger is a fin and tube heat exchanger. It allows to reach lower water temperatures at the cooling tower exhaust. The complete cooling process of the indirect evaporative cooler is shown in Figure 2. The indirect evaporative cooler model combines the models of the cooling tower and the air-water heat exchanger. The cooling tower model is the same as in section 2.1.1

but the air entering the cooling tower is the air that leaves the air-water HEX, and the water flow rate is the sum of the water flow rate in the condenser of the chiller and in the air-water HEX. The water flow rate in the air-water HEX should be chosen so that the heat transfer between air and water is optimized:

$$\dot{M}_w c_{p,w} = \dot{M}_a c_{p,a} \tag{9}$$

The air flow rate is the same as in the cooling tower. The complete model is described in the work of Jiang & Xie (2010). So far, the IEC is assumed to have the same electricity consumption as the cooling tower if working with the same air flow conditions. However, the air-water heat exchanger induces a head loss on the air side that should be compensated by operating the fan at a higher rotation speed. Characterizing that head loss is a priority for the future of this work.

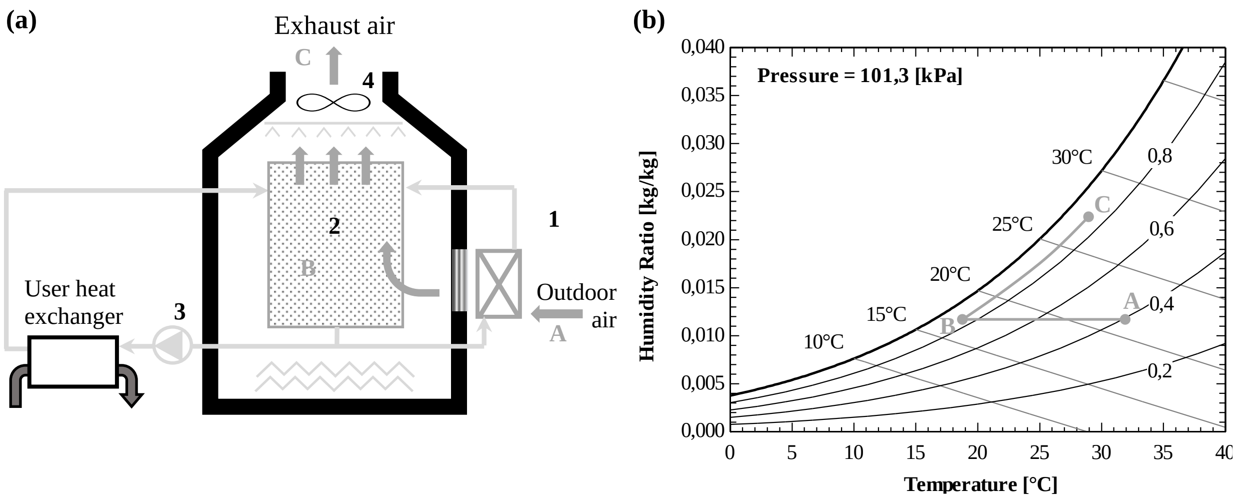

Figure 2: Principle of the evaporative indirect cooler. (a) Structure of the indirect evaporative cooler: 1. Air-water countercurrent heat exchanger (HEX); 2. Indirect wet cooling tower; 3. Water pump; 4. Fan. (b) Representation of the process of indirect evaporative cooling on a psychrometric chart.

2.1.3 Chiller: The chilled water that is produced by the cooling tower is used to cool down the condenser of the chiller. The latter is used to produce water at 18°C or less to feed the terminal units of the building. In nominal conditions, the chiller works in a 30/18°C temperature regime. The nominal temperature glide (waterwise) of the evaporator and condenser is 5K. The chiller model has been developed based on the Ph.D. thesis of Bertagnolio (2012), in which the chiller power consumption is computed as a polynomial function.

$$\dot{Q}_{ch,FL} = \dot{Q}_{ch,n} \cdot \left(1 + D_1\left(T_{cd,su} - T_{cd,su,n}\right) + D_2\left(T_{ev,ex} - T_{ev,ex,n}\right)\right) \tag{10}$$

$$PLR_{ch} = \frac{\dot{Q}_{ev}}{\dot{Q}_{ch,FL}} \tag{11}$$

$$\dot{W}_{ch,FL} = \frac{\dot{Q}_{ch,FL}}{COP_n} \cdot (1 + C_1 \Delta T + C_2 \Delta T^2) \tag{12}$$

$$\dot{W}_{ch} = \dot{W}_{ch,FL}\left(K_1 + (K_1 - K_2) \cdot PLR_{ch} + (1 - K_2) \cdot PLR_{ch}{}^2\right) \tag{13}$$

with

$$\Delta T = \frac{T_{cd,su}}{T_{ev,ex}} - \frac{T_{cd,su,n}}{T_{ev,ex,n}} \tag{14}$$

All the model constants have been determined based on the thesis of Bertagnolio (2012).

2.1.4 Terminal Units: The terminal units are the link between the cooling system and the end-users. In this case, they consist in radiant ceilings. Those terminal units are known to work with temperature levels around 18°C. The thermal performance of the radiant ceiling has been determined based on Messana's radiant cooling ceiling ("Tech Specs," 2022). The terminal unit model used in this article is inspired from Rhee et al. (2017). The cooling load provided by the terminal unit can be written as

$$\dot{Q}_{tu} = K \cdot (T_i - T_w)^n \tag{15}$$

or

$$\dot{Q}_{tu} = \dot{m}_{tu} \cdot c_{p,w} \cdot \left(T_{w,tu,ex} - T_{w,tu,su}\right) \tag{16}$$

Where T_i is the indoor temperature, T_w is the mean water temperature inside the terminal units and K and n are constants to be determined using manufacturer data. The electricity consumption of the terminal units consists in the electricity consumption of the circulation pump. It can then be computed based on the part load ratio:

$$PLR_{tu} = \frac{\dot{M}_w}{\dot{M}_{w,n}} \tag{17}$$

$$\dot{W}_{tu} = \dot{W}_{tu,n} \cdot \left(C_0 + C_1 PLR_{tu} + C_2 {PLR_{tu}}^2 + C_3 {PLR_{tu}}^3\right) \tag{18}$$

C_0, C_1, C_2 and C_3 are constants that have been determined based on the work of Brandemuehl (1993). The nominal electricity consumption of the terminal units is assumed to be 1% of their nominal cooling capacity (Bertagnolio, 2012). The entire behavior of the terminal units can be determined either by fixing the supply water temperature or the water flow rate. The flow rate can be adapted depending on the pump rotation speed while the supply temperature depends on the evaporator temperature level. If the fed chilled water is colder than the minimum water temperature that can be accepted in the terminal units to provide the required cooling load, part of the flow rate leaving the terminal units can be recirculated.

The main characteristics of the studied systems are summarized in Table 1.

Table 1: Summary of the main characteristics of the studied systems.

	Cooling tower	Air-water HEX	Chiller	Terminal units
Nominal cooling capacity [kW]	460	36	400	400
Nominal consumption [kWe]	8.3	-	50.8	4
Nominal water flow rate [kg/s]	22.01	4.18	Condenser: 22 Evaporator: 19	22.24
Temperature regime in nominal conditions	Water: 30°C Air: 32°C	Water: 28.5°C Air: 32°C	Condenser: 30-35°C Evaporator: 23-18°C	Min: 8°C

2.2 Building model

The reference building used in this study is part of the US Department of Energy (DOE) reference building collection. The DOE developed 16 reference building models that represent most commercial buildings across 16 most typical climate locations in the United States (US Department of Energy, 2018). The building has been designed based on the medium office building and adapted to the climate of Central Europe. The main characteristics of the building are presented in Table 2.

Table 2: General description of the building case study.

General information		**Envelope**		**Operation**	
Location	Nancy (France)	U-value	(W/m² K)	Nb workers	977
Type	Office building	*External wall*	0.097	Nb hours/day	8
Nb floors	10	*Internal wall*	0.508	Occupation gain	100 W/pers
Orientation	South	*Floor*	0.132	Appliances	10 W/m²
Total surface	16 607 m²	*Window*	0.6	Lighting	6 W/m²
WWR	0.33			Ventilation rate	3.6 m³/h m²
Infiltration rate	0.2 ach	Window solar factor	0.5	Temperature set point	22-26°C

2.3 Climate zone

The climatic data used is the simulations correspond to the climatic zone of Nancy (France). According to the climate classification of the ASHRAE (2013), Nancy, is part of zone 4A, meaning that the climate is mixed humid. A mixed-humid climate is generally defined as a region that receives more than 50 cm of annual precipitation, has approximately 3000 heating degree days (18°C basis) or fewer, and where the average monthly outdoor temperature drops below 7°C during the winter months. Figure 3 shows the evolution of the average dry bulb and dew point temperatures in Nancy during an average day in the cooling season. As well as the frequency distribution of the temperature difference between dry bulb and dew point. The temperature difference between dry bulb and dew point is the highest during the cooling hours, which suggests a good match with the hours during which the IEC is most performant.

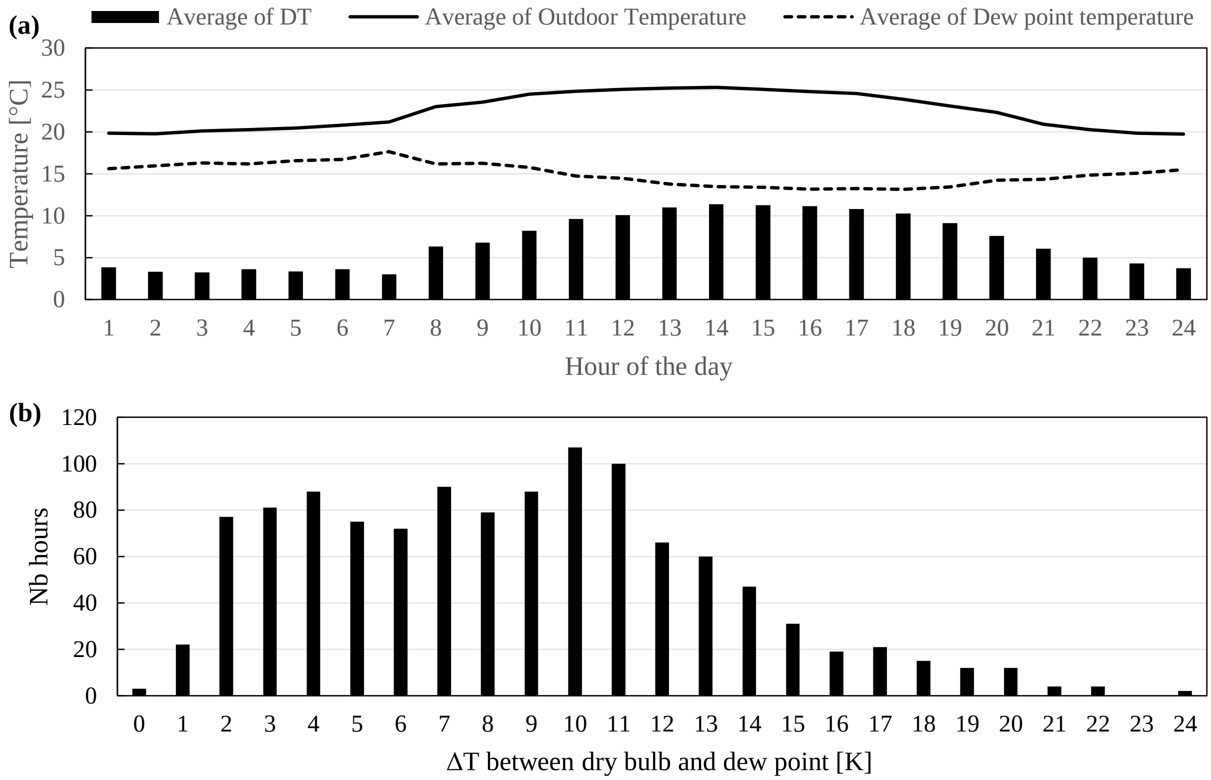

Figure 3: (a) Evolution of the dry bulb and dew point temperatures during an average day of cooling season. **(b)** Frequency distribution of the temperature difference between dry bulb and dew point during cooling hours.

3. RESULTS AND ANALYSIS

In this section, the results of the simulations are discussed and analyzed. Section 3.1 explains the different control strategies in normal operation mode and in free chilling mode. The results are presented in section 3.2 and analyzed in section 3.3.

3.1 System operation

The building behavior has been simulated based on climatic data and building characteristics over one year. Then, the hourly cooling load required to maintain the indoor conditions around the temperature set point has been computed. The outputs of the building simulation have then been used as inputs for the cooling system model to compute the total energy consumption of the building for cooling purpose. At each hour, the response of the system to a specific cooling load, indoor temperature and external conditions is simulated. The simulation is quasi steady-state as only the operation point in steady-state conditions is calculated for every hour.

The cooling system can be operated with two control strategies: *temperature control* or *flow control*. In the temperature control strategy, the supply temperature of the terminal units and the water exhaust temperature of the cooling tower are fixed. It allows to study the fan consumption of the IEC compared to the reference system because if the water temperature at the evaporator and condenser are fixed, the chiller and terminal unit energy consumption remain unchanged. The temperature at the evaporator exhaust has been chosen to 15°C and the temperature at the condenser supply at 25°C. In the flow control strategy, the air flow rate inside the cooling tower and the water flow rate in the terminal units are kept constant at 90% of their nominal value. This strategy allows to keep the consumption of the fan and terminal units constant, which allows to study the impact of the IEC on the chiller energy consumption due to the decrease in condenser supply temperature.

Instead of using a temperature or flow control strategy, it is also possible to optimize the system operation. As can be seen in Figure 4, there exists an optimum configuration of cooling tower air flow rate and terminal units water flow rate that minimizes the energy consumption of the cooling system. Decreasing (resp. increasing) the water temperature at the condenser (resp. evaporator) leads to an energy consumption decrease of the chiller. However, it requires a higher air flow rate (resp. water flow rate) in the cooling tower (resp. terminal units), hence, a higher energy consumption of the auxiliaries.

The system can also work in free chilling mode by bypassing the chiller. The chilled water produced by the cooling tower is directly used in the terminal units. Free chilling allows a large energy consumption reduction as the main energy consumer of the system is the chiller compressor. In free chilling mode, none of the control techniques developed earlier can be applied. As shown in Figure 5a, the operating point is defined by the intersection between the cooling tower and terminal unit operating curves. If the curves have an intersection, it is possible to do free chilling, but if the minimum temperature of the water produced by the cooling tower is higher than the maximum admissible temperature in the terminal units, the chiller must be used.

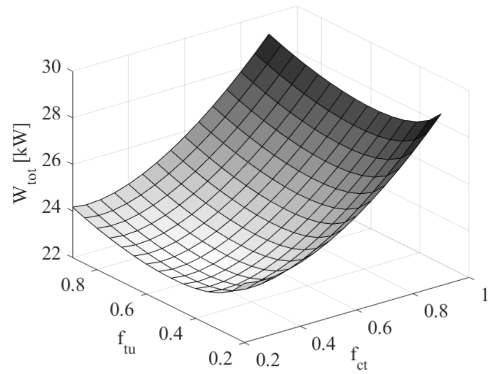

Figure 4: Evolution of the global consumption of the system depending on the load factors of the cooling tower (f_{ct}) and terminal units (f_{tu}).

Figure 5 also shows that the indirect evaporative chiller allows to increase the percentage of time during which the free chilling mode can be turned on. Since the chilled water produced by the IEC is at a lower temperature than the water produced by the cooling tower alone in the same conditions, there can exist an operating point for the IEC in free chilling mode while there is none for the cooling tower. However, it can also be seen that using the IEC to do free chilling can result in an energy consumption increase as the fan energy consumption rises more rapidly. For an identical water flow rate in the terminal units, the water flow rate in the cooling tower, hence the air flow rate, are higher in the IEC, which leads to a higher fan consumption.

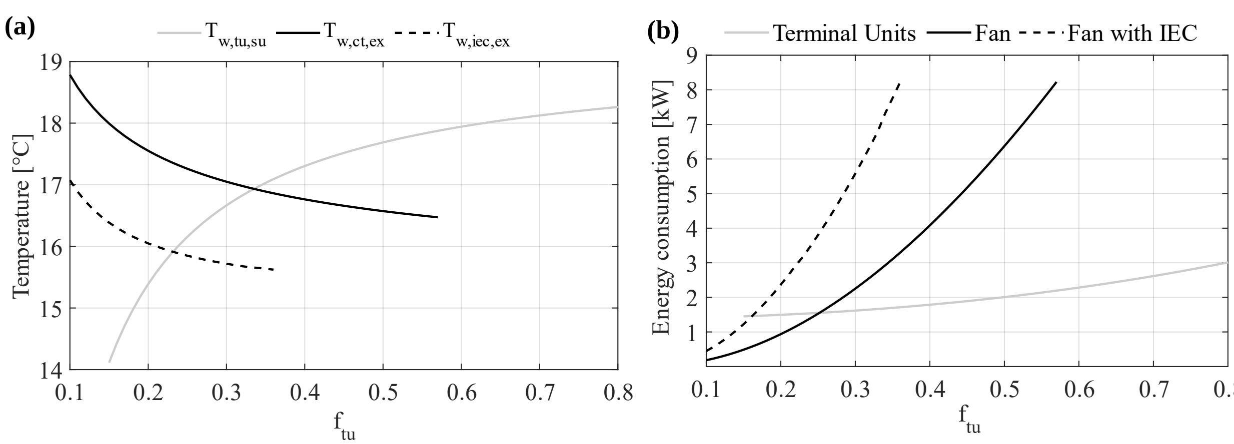

Figure 5: Impact of the water flow rate in the terminal units on **(a)** the temperature in the cycle components and **(b)** the energy consumption of the auxiliaries, in free chilling mode.

3.2 Results

The annual energy consumption of the system has been computed in four different cases. In the reference case, the primary cooling system is the cooling tower alone and the chiller cannot be bypassed. The first improvement is to activate the free chilling mode whenever it is possible. Then, the additional HEX has been added to use the IEC as primary cooling system. In the last improvement, it has been considered that the additional HEX can be bypassed if the system consumption with the cooling tower alone is lower than with the IEC in the same conditions. The simulation results are reported in Figure 6. The activation of the free chilling mode allows to decrease the global energy consumption by 30% on average. The impact of the IEC on the energy consumption is less significant (less than 5% decrease). It allows to increase the number of hours doing free chilling by about 10%, saving up to 500kWh/year, which is about 4% of the annual consumption. Most of the savings attributable to the use of the IEC are due to these additional hours doing free chilling since there is no significative difference between the third and fourth scenarios, meaning that the energy consumption of the system with IEC or with cooling tower alone is almost unchanged. This is further investigated in the next section.

The optimization of the system operation has a very positive impact on the global energy consumption. To find the trade-off between auxiliaries consumption and chiller consumption, the fans and pumps should be equipped with a frequency inverter that is regulated by a controller. However, the controller performance will strongly depend on the dynamic effects of the system and the time needed to reach steady state. Plus, it would also require having some data collection of the system consumption in real time.

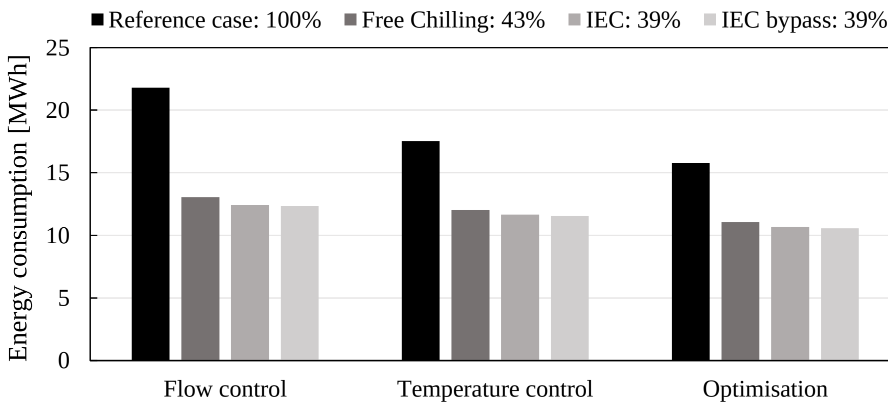

Figure 6: Comparison of the annual energy consumption of the cooling system depending on the control strategy and system operation. The percentage indicates the percentage of time during which the chiller is used in each situation.

3.3 Analysis

As previously said, the IEC performance is related to the difference between the dry bulb and dew point temperatures of the outside air, while the cooling tower performance depends on the difference between the dry bulb and wet bulb temperatures. Figure 7 shows the consumption reduction due to the IEC as a function of the temperature difference between dry bulb and dew point for each hour of operation. When the chiller is not bypassed, the energy consumption reduction is strongly related to this temperature difference. The higher the difference between the dry bulb and dew point, the lower the energy consumption of the system. It can also be seen that there is a temperature difference threshold under which the IEC is not beneficial compared to the cooling tower alone. The auxiliaries consumption increase is higher than the chiller consumption reduction due to the water temperature decrease at the condenser.

In free chilling mode, however, the effects of the IEC are more scattered and there is no correlation between the consumption reduction and the temperature difference between dry bulb and dew point. Generally, the use of the IEC in free chilling mode even leads to an overconsumption of the system compared to the cooling tower alone.

4. LIMITATIONS AND IMPROVEMENTS

So far, the cooling system is modeled as an “ideal” system. It can be operated at the best operating point at every hour. However, there are several factors that should be considered. First, the system has some inertia and the new operating point is not reached immediately, and the delay induced before reaching steady state could impact the indoor conditions.

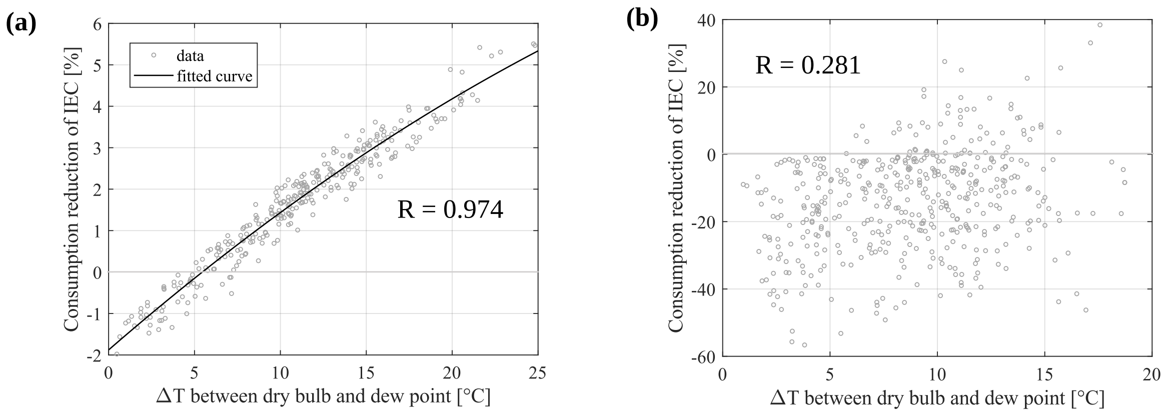

Figure 7: Energy consumption reduction due to the usage of the IEC as a function of the temperature difference between the dry bulb and the dew point. (a) is the consumption reduction only for the hours during which the chiller is not bypassed and (b) is the consumption reduction for hours during which the free chilling mode is active.

The coupling between the cooling system and building models must be improved. Both models are currently run separately and the results of the building simulation, *i.e.*, cooling load and indoor temperature, are used as direct input in the cooling system model. The next step would be to create a co-simulation model for both systems. Co-simulation will generate interactions between models that will have to be studied to verify if the present conclusions hold.

The co-simulation could also allow to see the building behavior when the cooling system is not able to provide the required cooling load. For example, the temperatures at the evaporator and condenser should always remain within the operating temperature range. For the other components, such as fans and pumps, there is also an operating range for the flow rate.

Finally, one crucial element that has not been modeled yet is the impact of the head losses due to the presence of an additional heat exchanger on the fan electric consumption. The air-water heat exchanger changes the operating curve of the circuit, hence the fan speed should be adapted. A higher rotational speed leads to a higher fan electricity consumption, which could reduce the interest for indirect evaporative chiller. Moreover, the water consumption of the system has not been accounted for yet.

5. CONCLUSIONS

The conclusions that can be withdrawn from this work are the following:

- Free chilling allows a consumption decrease of 30% on average.
- The number of hours doing free chilling can be increased by 10% using an indirect evaporative cooler.
- When doing free chilling, the indirect evaporative cooler does not significantly decrease the energy consumption compared to the wet cooling tower alone.
- The consumption decrease of the indirect evaporative cooler compared to the regular cooling tower with chiller is directly linked to the temperature difference between the dry bulb temperature and the dew point temperature.
- There exists a minimal temperature difference $T_{db} - T_{dp}$ below which there is no advantage in using the indirect evaporative cooler instead of the regular cooling tower.

In the future, the studied cooling system will be simulated in other types of climates and buildings to see if those conclusions still apply.

NOMENCLATURE

c_p	specific heat capacity	(J/kg K)
f	load factor	(-)
h	enthalpy	(J/kg)
$\dot{M}$	mass flow rate	(kg/s)
PLR	part load ratio	(-)
R	correlation coefficient	(-)
T	temperature	(K)

Subscript

a	air
af	fictitious air
cd	condenser
ch	chiller
ct	cooling tower
db	dry bulb
dp	dew point
ev	evaporator
ex	exhaust
f	fictitious
i	indoor conditions
iec	indirect evaporative cooler
n	nominal conditions
su	supply
tu	terminal units
w	water
wb	wet bulb

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