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# Application feasibility of low temperature cooling tower for high-temperature buildings to daytime ventilation

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**Abstract:**

An experimental study has been carried out to characterise a laboratory-scale cooling tower (CT) intended to provide water to a high temperature radiant cooling system during night-time. The present study evaluates the possibility of broadening its operation during daytime to precool the outdoor airflow required for ventilation during occupation periods. The performance of the CT combined with a water-to-air coil demonstrates that its operation within a water-economiser cycle would result into energy savings and better indoor air quality (IAQ) for most summer conditions studied, with the only exception of humid tropical climates. The few non-interesting locations can be easily identified regarding the corresponding Köppen-Geiger climate classification, and relative energy efficiency of the system at different climates is rather foreseeable through the local wet bulb depression (WBD) bin data. This leads to the conclusion that a CT designed to fed TABS during night-time can efficiently operate during diurnal occupation periods to remove ventilation loads of the required dedicated outdoor air system (DOAS) or support the additional mechanical cooling if required, with little initial cost.

**Keywords:**

Cooling Tower; Free cooling; Water-side economiser; Ventilation air cooling; ANOVA.

## 1. Introduction

Current research and policies being developed towards less energy-consuming buildings aim for energy efficient strategies to achieve indoor comfort. Improvements in cooling systems are key steps on this path, given the growing trends on energy consumption for space cooling due to global warming and urban heat islands, in addition to the quality of life expectations of a population that spends more and more time indoors.

Ventilation rates have such an impact on buildings energy requirements that it may result appealing to try to minimise them. However, it is essential to avoid compromising IAQ, and in this sense strategies such as heat recovery, demand control or free cooling are targeted as possible solutions (1).

Research conducted on Thermally Activated Building Systems (TABS) for space cooling is gaining interest due to higher cool water temperatures required, but a Dedicated Outdoor Air System (DOAS) would be always required to fulfil ventilation rates established in the standards (2).

Among the number of existing alternatives designed to simultaneous and efficiently achieve indoor thermal comfort and IAQ, those based in the free-cooling potential of outdoor conditions are particularly appealing. Low temperature cooling towers (CT) can efficiently exploit this free-cooling potential for either cool ventilation air or fed high temperature water cooling systems, though to date little research work has been focused on the characterisation of these CT (3,4). In the present paper it is aimed to maximise this free-cooling potential by enabling a CT designed to operate during favourable night-time outdoor conditions for TABS cooling, to extend its operating hours during daytime within a water-economiser cycle for ventilation air cooling.

### 1.1. Water-Economiser cycles

The simplest solution of directly use cool outdoor air as ventilation air is the only one that could simultaneously improve energy efficiency and IAQ when higher airflows are supplied (5,6). The implementation of this all-air free-cooling solution is called air-side economiser cycle, and in principle it is always interesting in ventilation

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systems because it can minimise or even avoid mechanical cooling requirements if outdoor conditions are favourable. However, it incurs in also higher fan requirements.

Some research work focused on this particular issue. Wang and Song (7) approached it considering a steady-state study in terms of operating conditions and auxiliary fan power consumption, to design the optimal control for an air handling unit (AHU) equipped with an economiser cycle. They determined the optimal supply temperature and airflow for three different operating zones of (I) 100% free-cooling, (II)partially assisted mechanical cooling and (III) minimum airflow supplied, obtaining energy savings of up to 90%. Rackes and Waring (8) also defined ranges of optimal temperature set-points and airflows for different building zones and periods along a representative day in different seasons. On contrary, from their analysis of the impact of changing the ventilation rate on the energy demand, Santos and Leal (1)obtained for the European climatology a general lowering of the yearly cooling demand if ventilation rates are increased. Hence, their results showed variations on the cooling demand from -0.01 to -0.16 kWh/($m^2$·year) when ventilation rate was changed in 1$m^3$/(h·person). Besides being favourable for every case studied, they concluded that, although results depended on the building type and use, climate is the determinant factor in the case of new buildings in general. They all nonetheless agreed in combining different systems with an adequate control to optimise energy savings at each operating conditions.

Control is the core in economiser cycles, which can be on a temperature or enthalpy basis. Although the latter can in principle be more energy-saving, its further complexity and relative precision make temperature control generally more interesting, provided that climate is not excessively humid. A fixed-dry bulb temperature control can then be interesting if set-temperatures are adequately defined according to the climate (5). Hence, limitations of these systems are intrinsically related to climate conditions. When outdoor conditions are beyond the psychrometric region of interest for air-side economisers applicability, they can still be interesting for water-side economisers operation. This is because water-side economisers exploit outdoor air free-cooling potential indirectly, by cooling water in a CT that later passes through a coil where ventilation air is cooled. However, additional electric power requirements due to fans and pumps would have considerable weight in this case, thus restricting the range of applicability more importantly.

Although free-cooling potential for different climate conditions and the related energy savings have been extensively studied for air-side economisers for different locations and cases (9), water-economisers have not been so widely approached.

Kim et al. (10) proposed a water-side free-cooling system to provide cooling water required in a liquid desiccant system assisted with evaporative cooling that operates with 100% outdoor air. A range of hot and humid conditions of dry bulb temperature (DBT) between about 20 and 32°C, and wet bulb temperature (WBT) from about 20 to 25°C were tested for the CT operation. With this combined system they obtained COPs of 3 to 20 times those expected for a conventional chiller. They also obtained important reductions in the chilling demand through a similar system proposed for a Data Centre (11).

Most research work on economisers applications focused on Data Centres, due to the large cooling demands involved in these applications, which lead to the existence and development of further numerous technologies to efficiently provide the required cooling (12–14). Air-side economisers are particularly appealing for higher energy savings are expected, as demonstrated by Cho et al. (15) for subtropical climates. However, some authors have directed their studies to specific implementations of water-side economisers because of the requirements on air cleanliness and humidity, besides the advantages on the system's integration within the central cooling system (16). Ham and Jeong (16) demonstrated through an annual simulation of a water-side economiser modelled in EnergyPlus, that this system would always result into energy savings with respect to cooling consumptions in Data Centres. If a proper design of the Data Centre is developed (that is, an aisle contained instead of uncontained Data Centre) higher supply temperatures could be set, thus enlarging the number of operating hours and consequently the energy savings. Agrawal et al. (17) compared various strategies to improve efficiency in air cooling for Data Centres at different locations of 17 climate types. Among them, they proposed configurations of 1 or 2 cooling towers to provide cooling water that is later used to precool water used in the AHU. Results showed that interest of water-side economisers increased for colder climates and that a two-cooling tower configuration would improve energy savings in some cases. Interest of water-side economiser over indirect evaporative cooling strategies focused on its less water consumptions.

As could be expected, efficiency of water-side economisers also depends on the season. Durand-Estebe et al. (18) obtained, through simulation in TRNSYS of different water-side economiser strategies in a Data Centre, optimal temperature set-points of 20°C during Winter, whereas it rises to 24°C for mid-season operation at a Mediterranean mild climate. However, these optimal set-points would differ for different case studies in terms of air-cooling production or control strategy. Hence, they proposed a control that adapts to the outdoor climate operating conditions.

On the overall, water-side economisers have been demonstrated to be effective. However, as also stated for air-side economisers, actual energy savings may not always be significant due to extra fan and pump power consumption. Durand-Estebe et al. (18) disregarded the power required by water pumps in water-side


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economiser cycles, though payed close attention to extra fan power required when air temperature set is raised. They in fact obtained an increase in the fan energy consumption of 43% when air temperature is modified from 16°C to 24°C to optimise energy savings by reducing heat pump energy requirements by 78%.

Besides, the installation is voluminous and costly, so it would be only reasonable if energy savings obtained balance out the initial investment, costs of water consumption and maintenance of the CT (17–19). Nonetheless, initial cost can be justified through the expected energy savings (10).

The aim of the present study is to assess the energy savings potential of expanding the operation of a CT designed for TABS, towards daytime operation in a water-economiser cycle, either to remove ventilation loads or to precool supply air when thermal comfort is not achieved by means of the TABS and additional cooling is required.

## 2. Methodology

A laboratory-scale device has been designed and constructed. It consists of a low temperature CT combined with an external finned coil as a water-to-air heat exchanger. This system would permit the pre-cooling of ventilation air through the coil with water previously cooled in the tower.

The target was to experimentally characterise the operation of this system in diurnal summer air conditions, to study the possibilities of expanding the operating period of low temperature cooling towers implemented for night-time water cooling for TABS, to daytime operation for cooling ventilation air.

### 2.1. Cooling tower

The experiments are conducted with the open, forced draft wet cooling tower shown in Figure 1.

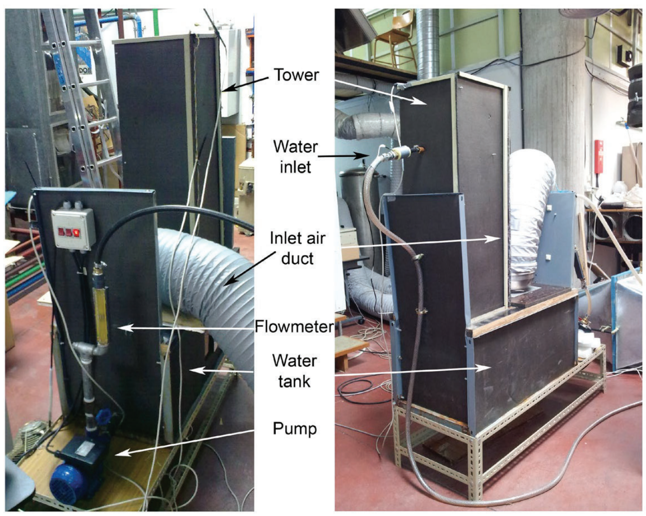


**Figure. 1.** Low temperature cooling tower used for the tests.

Water is sprayed from the upper part of the tower with the aid of a set of nozzles. The tower is equipped with a droplet separator at the air outlet to avoid generation of aerosols and unnecessary water losses. The packing consists of polycarbonate panels with a structure of hexagonal cells, offering a total exchange area of 13,5 $m^2$, which results in a surface density of 370 $m^2/m^3$. The water tank has a total capacity of 100 l; water level can be controlled with a level-viewer in the tank and is maintained at 60 l. Air enters the tower through a separate orifice in the same water tank, generating the desired counter-flow through the tower filling. Cooled water in the tank is then driven to the coil by a water pump placed at the water tank exit, and from the coil to the tower nozzles.

A copper-tube aluminum-fin coil is employed to pre-cool ventilation air with the aid of water cooled in the tower. It consists of staggered tubes of 3/8'' and 3.2mm separation between fins, arranged in 4 columns and 12 rows. Its total dimensions are 320x320x90mm. The coil is assembled within a structure to which the air ducts for ventilation air are connected.

The approach of the tower results to be about 5°C, which falls within the expected achievable range in practice (20). This high value is due to the also high outdoor air operating temperatures tested in the study, which has an important influence on the CT performance owing to results in existing literature (3,10). The outdoor DBT is also decisive for the approaching temperature, according to the Analysis of Variance (ANOVA) performed from the experimental results obtained in the present work (figure 2.a). However, the approaching temperature could be reduced by increasing the air flow rate (3,4), as observed in figure 2.b.

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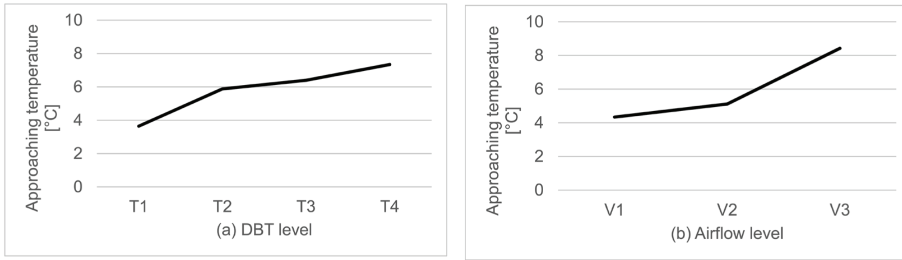


**Figure. 2.** Average approaching temperatures in the CT for increasing (a) outdoor air DBT; and (b) air flow rate.

This high approach justifies the limited effectiveness obtained in the tower (figure 3), as the effectiveness of the tower is calculated from the water temperature drop achieved in the tower to the maximum temperature drop achievable (1):

$$\varepsilon_T = (T_{Tw\,i} - T_{Tw\,o})/(T_{Tw\,i} - \mathrm{T}_{wb\,Ta\,i}) \tag{1}$$

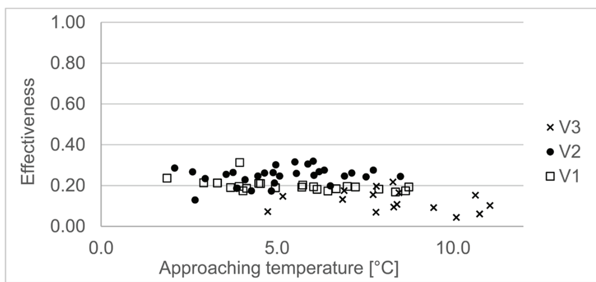


**Figure. 3.** CT effectiveness to approaching temperatures obtained for different air flows.

## 2.2. Test bench

The cooling tower and the coil are connected together as illustrated in figure 4. Air supplied to both the CT and the coil is conditioned by an Air Handling Unit that enables the system to operate at the different outdoor conditions defined in the design of experiments. Water cooled in the tower would be automatically driven to the TABS during night-time, and to the coil for diurnal operation, switched when the building becomes occupied. Ventilation air pre-cooled in the coil would be supplied to the conditioned space.

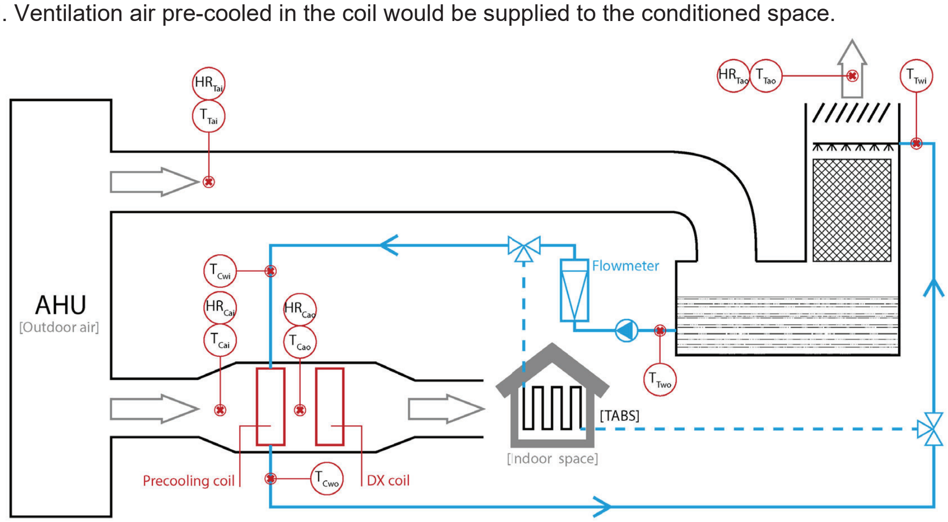


**Figure. 4.** Configuration of the combined system.

Measurements of air DBT and RH are acquired at the air inlet and outlet of the CT and the coil, as indicated in figure 4. Also, water temperatures are measured in the water circuit at the inlet and outlet of both the coil


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and the tower. Water flow is measured with a variable area flowmeter at the outlet of the water tank, with ±2% accuracy and range from 1.5 to 10 l/min. Air flow is measured through the pressure drop obtained in orifice plates placed in the air ducts downstream the AHU plenum for air distribution.

Temperature sensors are 4-wired Pt-100 with accuracy of ±0.1°C and range from 50 to 250°C. They have been calibrated before the tests with a calibration oven.

RH sensors are capacitive type, ±2% stability and 0 to 100% range, also previously calibrated to a reference sensor.

Pressure drop is measured with differential pressure sensors placed upstream and downstream enough of the two orifice plates. These orifice plates have been previously calibrated with a reference nozzle.

### 2.3. Design of experiments

Table 1 presents the design of experiments. Three levels of airflow are studied, while the water flow is fixed at 4 l/min (and thus not reflected in table 1). This is because only water-to-air flow ratios are of interest (3). The consequent operating water-to-air ratios are of about 1.5, 0.9 and 0.8 for levels V1, V2 and V3, respectively. Four levels of outdoor air DBT and another four levels of wet bulb temperature (WBT) are proposed. These levels are selected to cover the widest range of probable diurnal outdoor air conditions during summer period. Conditions for the set T1-W4 belong to the fog region and are thus disregarded; consequently, a total of 90 tests have been developed.

**Table 1**. Design of experiments.

| FACTORS | | | | | |
|---|---|---|---|---|---|
| V | ($m^3$/h) | DBT | (°C) | WBT | (°C) |
| **V1** | 150 | T1 | 25 | W1 | 18 |
| **V2** | 250 | T2 | 30 | W2 | 21 |
| **V3** | 300 | T3 | 35 | W3 | 24 |
| | | T4 | 40 | W4 | 27 |

An AHU is used to reproduce the desired outdoor conditions, as shown in figure 4. Because the variable controlled in the AHU was the relative humidity, for each pair of DBT – WBT the corresponding RH was calculated, then established as the set condition in the AHU together with the DBT. These operating conditions are tested in an arbitrary order and twice to check repeatability. Those tests where repeatability was not observed, were developed again and the less reliable tests were disregarded.

## 3. Results and discussion

### 3.1. Ventilation and cooling potential

#### 3.1.1. Air temperature drop in the coil

Figures 5.a) to c) show the air temperature drop obtained in the coil for the different inlet airflow rate, DBT and WBT. It can be observed that ventilation air cooling in the coil is influenced by its initial DBT ($T_{Cai}$) and WBT ($T_{wb\ Cai}$). Higher $T_{Cai}$ result into larger temperature drops because the air DBT at the inlet determines the logarithmic mean temperature difference (LMTD). But, to understand how $T_{wb\ Ca\ i}$ affects air cooling achieved in the coil, it must be noticed that outdoor air is used both as operating air in the tower and as ventilation air in the coil ($T_{Cai}$=$T_{Tai}$; $T_{wb\ Cai}$=$T_{wbTai}$). Because the WBT determines the water temperature achieved at the outlet of the cooling tower ($T_{Two}$), then driven to the coil, $T_{wb\ Ca\ i}$ also influences the LMTD, though indirectly. Increasing $T_{ai}$ yields larger air-to-water temperature differences in the coil. Likewise, both higher $T_{ai}$ and lower $T_{wb\ a\ i}$ foresee larger water cooling in the tower, hence larger air-to-water temperature differences and then air temperature drops in the coil.

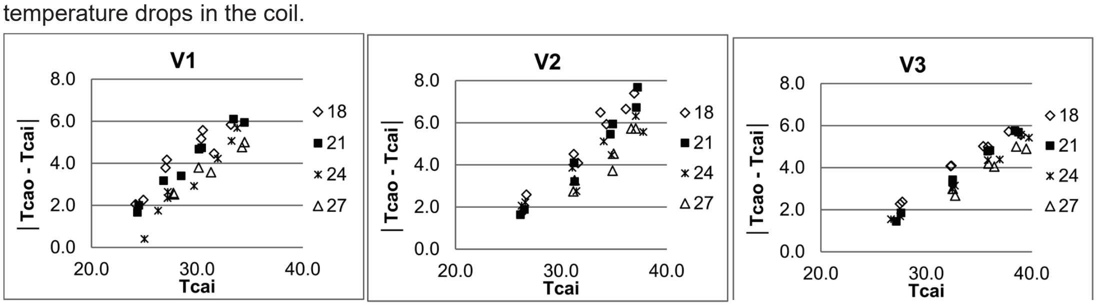


**Figure. 5.** Experimental air temperature drop through the coil for airflows (a) V1, (b) V2, and (c) V3.

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The ANOVA demonstrate a main contribution of $T_{Cai}$, of 65% on the temperature drop, and a contribution of $T_{WB\ C\ ai}$ of 15%. Effect of airflow is limited to 8%, and the remaining effect comprehends the effect of dual and three factors interactions, as well as a remaining non-determined contribution.

### 3.1.2. Ventilation air cooling capacity

Because only sensible cooling is performed through the coil, $x_{Ca}$ would be constant and ventilation air cooling can be obtained as expressed in equation (2):

$$\dot{Q}_{Ca} = \dot{m}_{Ca} \cdot (Cp_a + Cp_v \cdot x_{Ca}) \cdot \Delta T_{Ca} \tag{2}$$

Figure 6 represents air cooling achieved in the coil in terms of the the wet bulb depression (WBD), which is defined as the difference between the outdoor air DBT and its WBT. For the same reason derived from previous figures 5.a) to c), it can be observed that increasing the outdoor air WBD result in larger heat transferred from the air cooled in the coil. This is due to better evaporative cooling potential in the tower under larger WBD, hence larger water temperature range, which is afterwards used to cool the ventilation airstream in the coil. In this case, there is greater influence of the airflow rate, due to the same definition of equation (2).

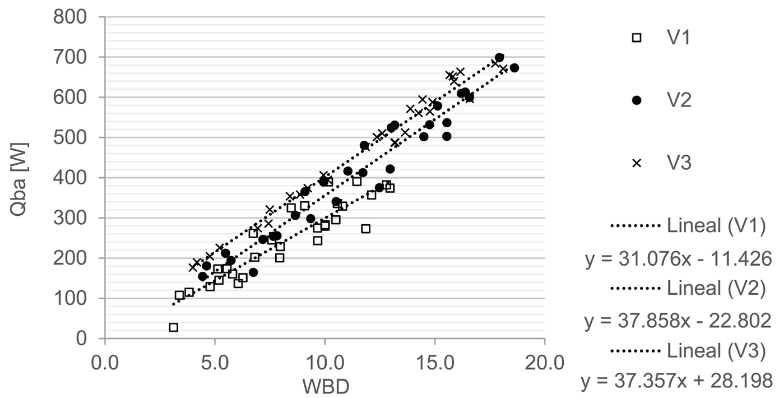


**Figure. 6.** Air cooling achieved in the coil at different outdoor air conditions and ventilation airflows.

The ANOVA developed on this parameter showed a main dependency on airflow, of 50%, being the contribution of $T_{Cai}$ a 35% whereas $T_{wb\ Ca\ i}$ accounts for only 8%. The remaining effect is similarly distributed between the effect of the three possible pairs of factors interaction and a non-determined error.

Consequently, although harsher outdoor air conditions entrain larger thermal loads, they also yield better performance. This point agrees with the idea stated by Ma et al. (21) that the external environment at the same time hinders the building autonomy and benefits the building homeostasis.

### 3.1.3. Coil thermal effectiveness.

Another interesting parameter to be considered concerning the coil operation is its thermal effectiveness. According to the ε-NTU method, thermal effectiveness of the coil can be defined as (3):

$$\varepsilon_c = (T_{Cao} - T_{Cai})/(T_{Cao} - T_{Cwi}) \tag{3}$$

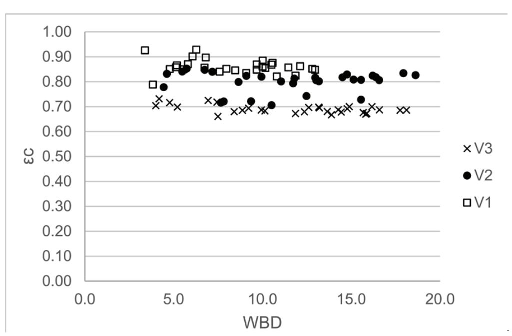


**Figure. 7.** Thermal effectiveness in the coil.


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Figure 7 shows that the difference between $T_{Cai}$ and the $T_{wb\ Ca\ i}$ has no effect. However, higher airflows imply shorter residence times and thus lower air temperature drops, hence smaller thermal efficiencies. This idea agrees with the results of the ANOVA on this parameter, as the contribution of the airflow rate is 82%.

## 3.2. Total cooling application

In this section, daytime operation of the cooling tower is studied as a mean to remove the ventilation loads generated by the required DOAS for TABS. The TABS are cooled with the night-time operation of the CT as the only cooling source and no further mechanical cooling is required during occupation times. The adaptive thermal comfort criteria can be fairly considered under these conditions.

The adaptive thermal comfort criteria is regarded in the existing European standards on indoor environment ([23] EN 16798-1:2019 Standard: Energy Performance of Buildings. Ventilation for Buildings. Part 1: Indoor Environmental Parameters for Design and Assessment of Energy Performance of Buildings Addressing Indoor Air Quality, Thermal Environment, Lighting and Acoustics. Module M1-6., n.d.) as suitable when there is no mechanical cooling and occupants can adapt to the thermal conditions by operating windows or modifying their clothing.

Some authors conceive adaptive comfort for buildings where TABS are implemented for thermal mass cooling, if water cooling demand is supported through low energy means. Whether considering adaptive or conventional thermal comfort models for TABS is thoroughly studied by Sourbron and Helsen (23). According to these authors every criterion would be valid in these cases, but they highlight that the key of the decision among existing models lays on the conflict between energy savings and strict comfort requirements. Pfafferot et al. (24) state that buildings with TABS fed by ground cooling should be considered as ‘‘mixed-mode buildings’’, and decide to consider the adaptive models. They nonetheless concede that further research on their applicability for these cases would be needed. The suitability of adaptive thermal comfort models for radiant cooling panels with water cooled in a CT is evaluated by Memon et al. (25), obtaining adequate indoor conditions in naturally ventilated spaces.

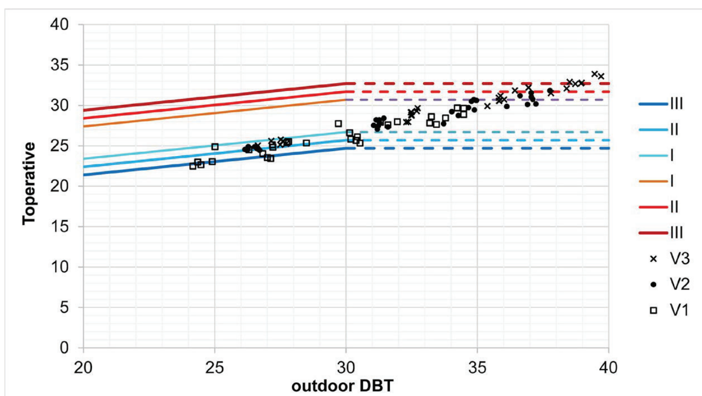


**Figure. 8.** Temperatures achieved in the ventilation air precooled in the coil and adaptive comfort ranges to be expected.

To study whether the cooling tower could support ventilation loads during its daytime operation, it would be enough to pay attention to supply air temperatures reached in the coil. These are represented in figure 8 within the ranges of adaptive comfort defined in the European standard ([23] EN 16798-1:2019 Standard: Energy Performance of Buildings. Ventilation for Buildings. Part 1: Indoor Environmental Parameters for Design and Assessment of Energy Performance of Buildings Addressing Indoor Air Quality, Thermal Environment, Lighting and Acoustics. Module M1-6., n.d.). Because comfort ranges for the adaptive criterion are not defined for outdoor air DBT over 30°C, trends have not been extrapolated, and the corresponding limits for 30°C have been considered for temperatures over this value. Besides, cases between 25°C and 30°C are scarce, being anyhow limited the validity of its application.

Figure 8 shows that supply air temperatures are maintained within the comfort ranges, except from the cases when the harshest outdoor conditions occur working with the highest airflow rate. For these cases, as well as for those when TABS and the night-time operation cooling tower as the only cooling source are insufficient, additional cooling would be needed during daytime occupation, and adaptive comfort criteria would not be acceptable.

## 3.3. Air precooling application

If additional mechanical cooling is required to achieve comfort, then the adaptive thermal comfort criteria would no longer be applicable, and the water-economiser cycle would just reduce the energy demand required by the conventional chilling system.

The energy savings achievable through the water-economiser cycle are calculated for various locations at different summer climate types. Four cities Worldwide and then seven Spanish cities have been selected and compared to assess the climate applicability of the CT. The considered cooling season spans from May to September for the Northern Hemisphere and from November to March for the Southern Hemisphere.

Worldwide cities selected correspond to the following Köppen-Geiger climate classification (26): (a) London, UK (Cfb: Temperate climate without dry season, warm summer); (b) Sydney, Australia (Cfa: Temperate climate without dry season, hot summer); (c) Las Vegas, US (Bwh: Arid, desert, hot climate); and (d) Singapore (Af: Tropical, rainforest). Climate data used in the study is obtained from Meteonorm databases (27).

Results obtained for the three airflow levels tested are gathered in table 2. Operating periods are calculated for a base temperature of 25°C, as an upper limit set-temperature for indoor thermal comfort in offices in summer ([23] EN 16798-1:2019 Standard: Energy Performance of Buildings. Ventilation for Buildings. Part 1: Indoor Environmental Parameters for Design and Assessment of Energy Performance of Buildings Addressing Indoor Air Quality, Thermal Environment, Lighting and Acoustics. Module M1-6., n.d.). Consequently, number of operating hours corresponds to the summer occupation periods (May to September between 8 a.m. and 8 p.m.) when outdoor DBT is over 25°C. If outdoor air DBT falls below 25°C, the system would operate under an air-economiser cycle, while for temperatures above 25°C, the water-economiser cycle would start operating. This temperature control is considered over enthalpy control due to its simplicity and less cost; however, enthalpy control would be recommendable in humid climates (S. K. Wang, 2001).

Energy savings are calculated through the correlations obtained for the total air-cooling capacity (Figure 6) applied to the climate data at each location on an hourly basis. The corresponding electric energy savings in the conventional water chilling system are estimated for a seasonal COP of 2.5.

Finally, actual energy savings are calculated deducting the electric energy consumption of the circulation devices: the water pump and the additional fan power required to support the pressure drop in the cooling tower. No additional fan requirements are considered through the coil because it is part of the existing ventilation system.

The power of the water pump ($W_{pump}$) used in the tower is of 46W. To obtain the additional fan power requirements ($W_{fan}$), equation (4) is used:

$$W_{fan} = \Delta P_T \cdot \dot{V}_T \tag{4}$$

The pressure drop in the cooling tower has been experimentally characterised through equation (5) for different air volume flows in the tower. A constant of $K_T$=200 has been obtained, for the pressure drop $\Delta P_T$ in [hPa]:

$$\dot{V}_T = K_T \cdot \sqrt{\Delta P_T} \tag{5}$$

**Table 2.** Operating periods and expected energy savings (World)

| CITY | Water-economiser cycle operating period No. hours / % summer | Air cooling [kWh] | Energy savings (no aux.)* [kWh$_e$] | Energy savings [kWh$_e$] | Airflow level |
|---|---|---|---|---|---|
| LONDON (Cfb) | 58 / 2.9% | 11.6 | 4.6 | 1.8 | V1 |
| | | 13.6 | 5.4 | 2.1 | V2 |
| | | 16.3 | 6.5 | 2.8 | V3 |
| SYDNEY (Cfa) | 540 / 27.5% | 91.6 | 36.6 | 10.5 | V1 |
| | | 106.8 | 42.7 | 12.0 | V2 |
| | | 132.7 | 53.1 | 18.0 | V3 |
| LAS VEGAS (BWh) | 1772 / 89.1% | 935.8 | 374.3 | 288.6 | V1 |
| | | 1124.7 | 449.9 | 348.9 | V2 |
| | | 1199.7 | 479.9 | 364.8 | V3 |
| SINGAPORE (Af) | 1846 / 92.8% | 159.1 | 63.6 | -25.6 | V1 |
| | | 177.8 | 71.1 | -34.0 | V2 |
| | | 268.6 | 107.4 | -12.4 | V3 |

*Without considering additional power consumption of the auxiliary devices.

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Table 2 shows how harsh climate conditions may enable the cooling tower to operate during longer daytime periods with larger energy savings. Indeed, the system would operate during around 90% of the summer - daytime occupation- period at Las Vegas. However, particularly humid climates like Singapore would have long operating periods but low effectiveness, yielding negative energy savings. It can also be observed that increasing airflows always allow larger energy savings, despite also larger pressure drops in the CT and the consequent extra fan power requirements.

A second study has been conducted for seven Spanish climates according to the Spanish building code (28) and the Köppen-Geiger classification (26): (i) Avila – E1/Csb (Temperate climate, dry warm summer); (ii) Barcelona – C2/ Csa (Temperate climate, dry hot summer); (iii) Bilbao – C1/ Cfb (Temperate climate without dry season, warm summer); (iv) Madrid – D3/ Csa (Temperate climate, dry hot summer); (v) Sevilla – B4 / Csa (Temperate climate, dry hot summer); (vi) Valencia – B3/ Csa (Temperate climate, dry hot summer); (vii) Valladolid – D2 / Csb (Temperate climate, dry warm summer). Concerning the Spanish building code climate index (composed by a letter and a number), only the number is of interest here, which refers to the summer climate harshness. Climate data used in this study has been obtained from the Spanish building code. Results are gathered in Table 3.

**Table 3.** Operating periods and expected energy savings (Spain)

| CITY | Water-economiser cycle operating period<br>No. hours / % summer | Air cooling<br>[kWh] | Energy savings (no aux.)*<br>[kWh$_e$] | Energy savings<br>[kWh$_e$] | Airflow level |
|---|---|---|---|---|---|
| Avila<br>(E1/ Csb) | 323<br>16.2% | 133.1 | 53.3 | 37.6 | V1 |
| | | 159.4 | 63.7 | 45.4 | V2 |
| | | 173.6 | 69.5 | 48.5 | V3 |
| Bilbao<br>(C1/ Cfb) | 288<br>14.5% | 65.0 | 26.0 | 12.1 | V1 |
| | | 76.6 | 30.6 | 14.2 | V2 |
| | | 90.2 | 36.1 | 17.4 | V3 |
| Barcelona<br>(C2/ Csa) | 579<br>29.1% | 120.5 | 48.2 | 20.2 | V1 |
| | | 141.7 | 56.7 | 23.7 | V2 |
| | | 169.2 | 67.7 | 30.1 | V3 |
| Valladolid<br>(D2/ Csb) | 715<br>35.9% | 280.9 | 112.4 | 77.8 | V1 |
| | | 336.0 | 134.4 | 93.7 | V2 |
| | | 367.8 | 147.1 | 100.7 | V3 |
| Valencia<br>(B3/ Csa) | 924<br>46.5% | 205.5 | 82.2 | 37.5 | V1 |
| | | 242.2 | 96.9 | 44.3 | V2 |
| | | 285.8 | 114.3 | 54.4 | V3 |
| Madrid<br>(D3/ Csa) | 873<br>43.9% | 353.7 | 141.5 | 99.3 | V1 |
| | | 423.3 | 169.3 | 119.6 | V2 |
| | | 462.0 | 184.8 | 128.1 | V3 |
| Sevilla<br>(B4/ Csa) | 1405<br>70.6% | 519.1 | 207.7 | 139.7 | V1 |
| | | 620.1 | 248.1 | 168.0 | V2 |
| | | 683.2 | 273.3 | 182.1 | V3 |

*Without considering additional power consumption of the auxiliary devices.

Again, increasing airflow levels entrain larger energy savings, despite additional consumption of the fan. On the other hand, longer operating periods do not necessarily involve larger savings. This is because a simple fixed DBT control is performed whereas it has been demonstrated that the system performance is highly dependent on the WBD. However, despite an enthalpy control of the cooling tower operating periods would provide more coherent results concerning number of operating hours and energy savings achieved, it would not be interesting due to its greater complexity and given that energy savings are anyway positive.

Hence, the achievable energy savings cannot be foreseen in detail if only the climate harshness index of the Spanish building code and the operating hours are considered. More humid climates incur into worse results than those expected regarding the operating hours. The cases of Madrid and Valencia, Avila and Bilbao, and Valladolid and Barcelona have the same climate classification in summer, but Valencia, Bilbao, and Barcelona are more humid climates. Hence, energy savings are of about 2.5, 3, and near 4 times higher for Madrid, Avila, and Valladolid, respectively. This lack of direct connection among operating hours and energy savings highlights the importance of carefully considering the climate of the location, to avoid excessive energy

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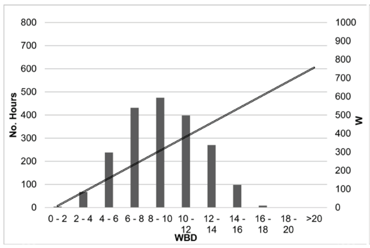
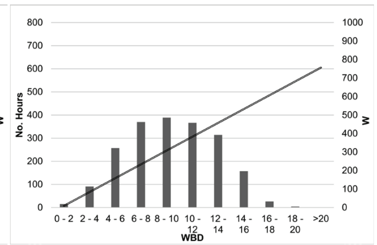
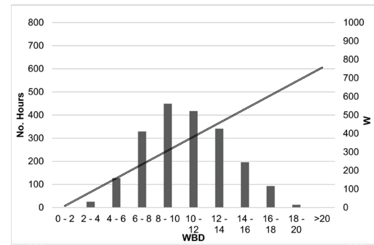
consumptions of the circulation devices. For this reason, summer climate classification in the Spanish building code would be insufficient to foresee the applicability of this system. Consequently, to better study the interest of the cooling tower diurnal operation, the WBD bin-data of the target location rises as a key point of study, if faced to the experimental results of the system (9). This comparison is illustrated in figures 9 (a) to (g).



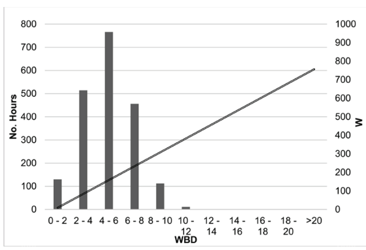

(a) Avila



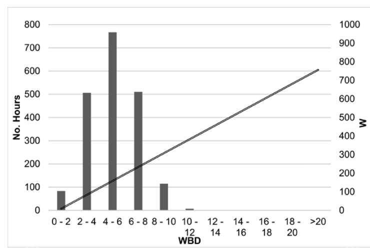

(c) Valladolid



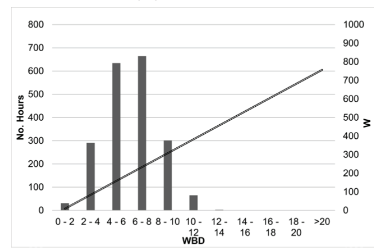

(e) Madrid



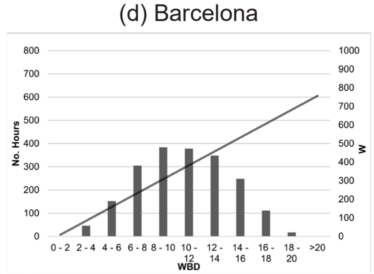

(b) Bilbao




(d) Barcelona




(f) Valencia




(g) Sevilla

**Figure. 9.** WBD-bin data of the selected Spanish cities compared to the system's cooling capacity.

In these figures, WBD of up to 20°C are considered, for being these the ones experimentally characterised. The cooling capacity of the system at V3 is represented, deducting the additional fan and pump power requirements. It can be seen how climates with larger WBD ranges would enable more energy savings, thereby justifying the values obtained in table 3. Therefore, from figure 9 Avila, Valladolid, Madrid and Sevilla would appear to be the most interesting locations for day-time operation of the cooling tower to pre-cool ventilation air, and actually are the ones with better results (table 3).

# 4. Conclusions and future work

A low temperature cooling tower has been experimentally characterised at a laboratory scale. Water cooled in the CT is used for ventilation air cooling in a water-to-air coil through a water-economiser cycle.

The performance of the CT is studied for three water-to-airflow rates of 0.8, 0.9 and 1.5, keeping the water flow constant, four DBT and four WBT. An ANOVA demonstrates that the airflow factor has the largest contribution to the water-economiser cycle thermal effectiveness and cooling capacity. Increased airflow yields larger energy savings, together with better IAQ expectations, despite higher fan power requirements.

Energy savings have been calculated for 4 cities Worldwide (London, Sydney, Las Vegas and Singapore) and seven Spanish cities. It was seen that, although the best results were obtained for the driest climates, broadening the operation period of the CT would result into positive energy savings for all cases studied but for Singapore.

Non-interesting humid climates can be easily identified regarding the WBD-bin data. On contrary, the Spanish building code climate classification for summer is not detailed enough to foresee the locations where the system proposed would have better potential, as it does not consider the humidity of the summer season.

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At those climates where the water-economiser cycle is effective, the use of fixed DBT control is enough to switch between an air or a water-economiser cycle. This option is preferable over an enthalpy control, due to its higher simplicity and the positive energy savings obtained at these climates.

Thanks to positive energy savings and no significant investment required, it is interesting in most cases to expand to daytime operation of an existing CT for night-time TABS cooling, either to simply supply ventilation loads or to pre-cool air if auxiliary mechanical cooling is needed.

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## Nomenclature

$\dot{Q}$ heat transfer rate, W

$\dot{m}$ mass flow, kg/s

$\dot{V}$ volume flow, $m^3/s$

T dry bulb temperature, °C

$T_{wb}$ wet bulb temperature, °C

x absolute humidity, $kg_{vapor}/kg_{dry\ air}$

$c_p$ heat capacity, $kJ/(kg_{dry\ air} \cdot K)$

RH Relative Humidity, %

P fan/pump power, W

ΔP Pressure drop, Pa

$K_T$ pressure drop constant in the tower

ε Thermal effectiveness

**Subscripts and superscripts**

o outlet

i inlet

T tower

C coil

a air

w water

v vapor

**Abbreviations:**

AHU=Air handling unit

ANOVA= Analysis of Variance

COP=Coefficient of Performance

CT=cooling tower

DBT=Dry bulb temperature

DOAS=Dedicated Outdoor Air System

IAQ= Indoor air quality

RH=Relative Humidity

TABS=Thermally Activated Building Systems

WBD=Wet bulb depression

WBT= Wet bulb temperature

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