Influence of the control strategy on the heat rejection potential and electricity consumption of air-to-water condensers for solar cooling systems

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1. Introduction

An efficient heat rejection is crucial for achieving a satisfactory primary energy balance in any cooling application. This is key in case of thermally activated cooling systems where the rejected heat rate is significantly larger than in electrically driven solutions working under the same boundary conditions. During the recent years, this topic has been extensively discussed among the scientific community and through national and international programs [1,2].

In this context, the management of an air-based heat rejection component has been identified as a key factor influencing the overall system effectiveness. Electrical consumption due to fans' operation is an objective function to be minimized. As shown in a recent market analysis [3], the average specific consumption of dry coolers (DCs) amounts to around 0.033 kW_{el}/kW_t, whereas for wet cooling towers (WCTs) is about 0.017 kW_{el}/kW_t. Despite the lower rated electric consumption, WCTs are characterized by higher operation and maintenance costs due to fresh water consumption and legionella growth risk. The respective disadvantages of DCs and WCTs can be tackled by adopting hybrid coolers (HCs), which are capable of wetting coil surface only when required, therefore contemporarily avoiding legionella propagation (as in WCTs) and reducing electricity consumption with respect to DCs.

This paper aims to demonstrate quantitatively the validity of these components through a set of dedicated numerical simulations. A hybrid cooler model validated with monitoring data has been used in this study. Spray water cycles and fan control strategies have been developed. With respect to fan operation, the following approaches are considered:

- constant fan speed (WM0);
- variable fan speed in order to maintain a fixed fluid temperature ΔT_w difference (WM1).

The analysis is conducted by calculating heat rejection potential for a number of climatic locations. For each of these, different control strategies have been compared and correlated to local weather conditions during cooling season. Simulation results show that the benefit of spraying water is a function of climatic parameters (air relative humidity, dry-bulb air temperature, length of the cooling season) and operation conditions (outlet condensing water temperature). The most relevant advantage of using HCs with respect to DCs is noticed when condensing outlet water temperature approaches dry-bulb air temperature. Conversely, significant water savings can be attained by hybrid coolers in comparison to WCTs for relatively high condensing temperatures.

2. Methodology

The performance analysis of air-based heat rejection technologies is a problem affected by both predictable (i.e. working fluid temperatures and mass flow rates) and unpredictable (i.e. weather conditions) variables. In particular, the type of cooling technology (i.e. electrically or thermally driven chiller), the system control strategy and building's load play a major role. Because of these reasons, an energy potential analysis has been chosen, since this approach allows to derive a figure of their overall energy potential performance, independent of a system layout or building load.

In this work, three different technologies are investigated:

- Dry Cooler (DC) of 27 kW [4] of heat rejection capacity under nominal conditions [5];
- Hybrid Cooler (HC) of the same capacity under dry conditions equipped with a spray water system. Water is sprayed periodically when dry-bulb ambient temperature is greater than 26 °C;
- closed Wet Cooling Tower (WCT) of the same capacity under dry conditions of the DC in which fully wet conditions on coil surface are present.

A set of transient simulations have been conducted by using a validated numerical TRNSYS model [6]. This model is capable of simulating a generic air-based heat rejection component thanks to a modular control volume definition and the possibility of spraying water upwards or downwards.

The seasonal behavior has been derived by operating the above mentioned technologies at constant inlet fluid temperature ($T_{w,in}$=40°C) and mass flow rate. Since the dependency to seasonal weather conditions are sought, 227 worldwide locations are considered for the analysis. For a given location, the Design Reference Year (DRY) has been used for representing mean weather conditions. For sake of homogeneity among simulations, the IWEC hourly weather format has been used for all cases. The heat rejection technologies are operated during the summer, defined as the period of the year with a 24-h average dry-bulb temperature greater than 18.3°C [7]. According to this definition, 10 locations are excluded since cooling season is null.

In Figure 1, the dependency of the cooling season duration is shown versus the average ambient dry-bulb temperature. As it can be noticed, for about 90% of locations (174, blue circles), the cooling season lasts less than 60% of the year and because of this fact the correlations presented in the next pages are filtered according to this criterion.

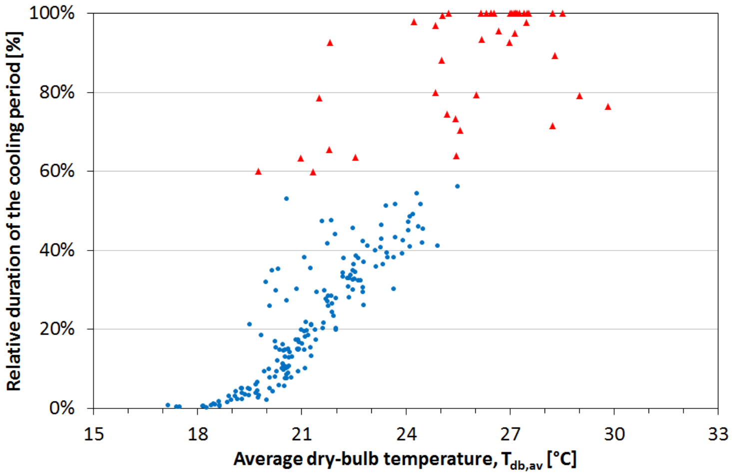

Figure 1. Dependency of the relative cooling period to the average dry-buld temperature.

The criterion adopted for comparing DC, HC and WCT consists of imposing to HC and WCT the same outlet water temperature of the DC. This condition is obtained by varying air volumetric flow rate through a fan speed controller. Hence, the energy savings of HCs and WCTs are evaluated for the same amount of rejected heat as for the DCs.

3. Simulation results

As stated above, an advantage of WCTs is their significant electrical savings and the consequent higher values of Seasonal Performance Factor (SPF, defined as the ratio between the rejected heat Q_{th} upon the fan electrical energy consumption W_{el}) in comparison with DCs and HCs. Figure 2 shows the dependency of fan electrical consumption to the rejected heat for the three technologies when DC is operated at constant maximum fan speed of 100% (working mode WM0). This relationship is well-fitted by a linear correlation, where the slope of the curve is the reciprocal value of the SPF. With respect to the simulation boundary conditions, DCs achieve an average SPF of 30.5. Thanks to the presence of water on coil surface, HCs improve their SPF value of about 21% (37.0), whereas WCTs of about 6.3 times (192.3).

WCTs wet continuously the coil surface during the cooling season, which results in an increase of water consumption and maintenance costs (i.e. for the mitigation of the legionella growth risk). With respect to this point, HCs permit larger water savings thanks to shorter spray activation periods limited to critical working conditions which amount to 44% of the cooling seasons at maximum (Figure 3).

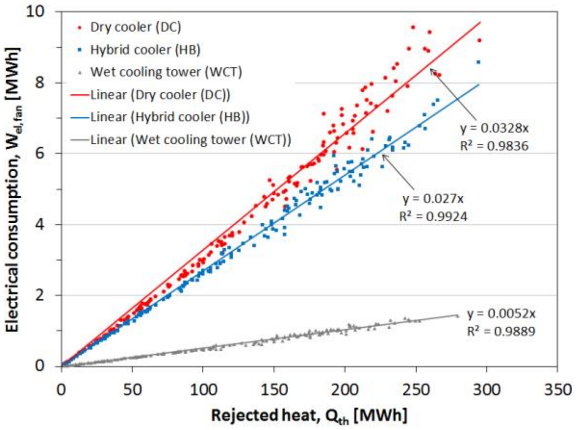

Figure 2. Electrical consumption as a function the rejected heat for DCs, HCs and WCTs (working mode WM0).

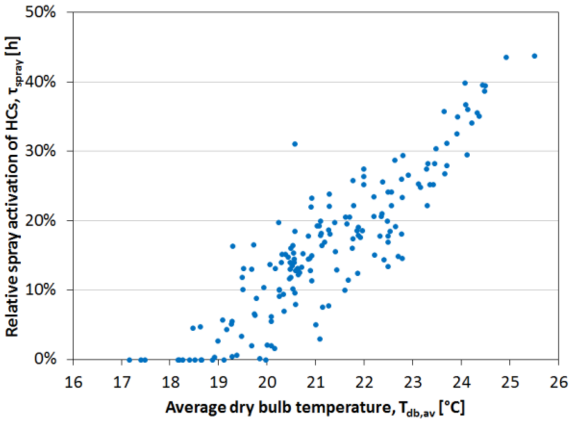

Figure 3. Relative activation period of HCs spray nozzles during the cooling season as a function of the average dry-bulb temperature (working mode WM0).

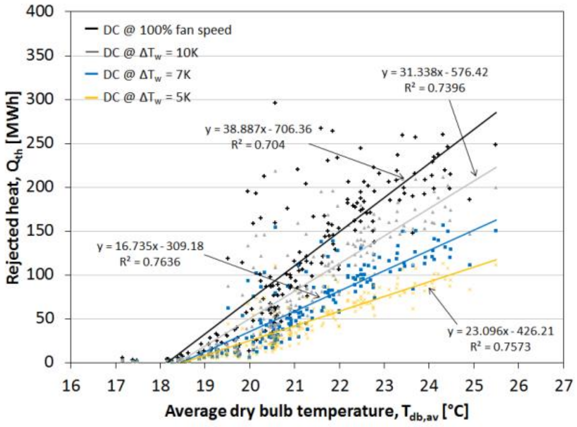

Figure 4. Dependency of rejected heat to average dry-bulb temperature for different setpoint water temperature different ΔT_w for DCs (WM0).

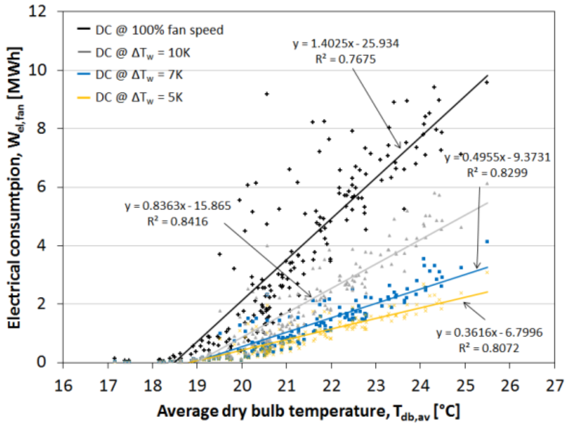

Figure 5. Dependency of electrical fans consumption to average dry-bulb temperature for different setpoint water temperature different ΔT_w for DCs (WM0).

Most of the times, heat rejection components are asked to regulate the outlet water temperature by varying the relative fan speed. Among other things, the selection of the appropriate control strategy is influenced by the magnitude of building loads, the severity of weather conditions and the chiller technology (i.e. electrically or thermally driven). In the case of a DC when a constant ΔT_w is sought (WM1), the amount of rejected heat Q_{th} increases proportionally as the target ΔT_w increases and its dependency to average seasonal dry-bulb temperature is shown in Figure 4. A similar relationship exists for the electrical consumption (Figure 5), but in this case, a stronger relationship to the magnitude of the outlet set-point temperature occurs.

Table 1 and Table 2 show the dependency of the SPF for different values of set-point water temperature drop for hot and dry (H&D) and hot and humid (H&H) climates. H&D locations are defined as those locations where the seasonal average dry-bulb $T_{db,av}$ and wet-bulb $T_{wb,av}$ temperature are comprised between 24-27°C and 15-18°C, respectively. H&H climates are characterized by both seasonal average dry-bulb and wet-bulb temperatures between 24-27°C. By comparing the performance in H&D and H&H conditions, the amount of rejected heat Q_{th} in H&H climates is in average about 2.27 times higher than respective values for H&D conditions. The electrical consumption of DC, HC and WCT is also higher about 2.32, 1.46 and 1.46, respectively.

Table 1. Average rejected heat Q_{th}, fan electrical consumption W_{el} and average SPF for DC, HC and WCT for hot and dry climate ($24<T_{db,av}\leq 27$; $15<T_{wb,av}\leq 18$).

		Water temperature drop ΔT_w [K]				
		3	5	7	10	13
	Q_{th} [MWh]	55.5	90.7	123.0	163.9	192.3
Dry cooler (DC)	W_{el} [MWh]	1.60	2.42	3.24	4.73	6.32
	SPF [-]	34.9	38.0	38.3	34.6	30.4
Hybrid cooler (HC)	W_{el} [MWh]	1.37	2.02	2.47	3.96	6.37
	SPF [-]	62.9	71.1	80.7	73.3	50.4
Wet cooling tower (WCT)	W_{el} [MWh]	0.40	0.68	0.98	1.38	1.55
	SPF [-]	241.5	239.4	235.0	223.4	215.1

Table 2. Average rejected heat Q_{th}, fan electrical consumption W_{el} and average SPF for DC, HC and WCT for hot and humid climate ($24<T_{db,av}\leq 27$; $24<T_{wb,av}\leq 27$).

		Water temperature drop ΔT_w [K]				
		3	5	7	10	13
	Q_{th} [MWh]	125.2	208.5	289.1	381.5	408.6
Dry cooler (DC)	W_{el} [MWh]	3.25	4.33	6.95	13.72	17.32
	SPF [-]	38.5	48.1	41.7	27.9	23.6
Hybrid cooler (HC)	W_{el} [MWh]	1.99	2.93	3.59	5.89	9.13
	SPF [-]	58.1	65.3	72.9	53.9	38.4
Wet cooling tower (WCT)	W_{el} [MWh]	0.59	1.00	1.44	2.01	2.23
	SPF [-]	185.4	180.5	170.4	156.6	153.5

In both climates, SPF values decrease as the ΔT_w increases, being this ratio mostly driven by the increase of fan electrical consumption. In H&D climates, DC and HC reach a maximum value at ΔT_w=7K (38.3 and 80.7, respectively). Similar performance for DC and HC are achieved in H&H regions. HC and WCT suffer high values of relative humidity which translates in a relative reduction of SPF between 25% and 30%. Basically, this occurs since fans are operated at higher speed than in H&D conditions in order to reach the desired outlet water temperature.

4. Conclusions

The heat rejection potential and the relative seasonal performance for air-based heat rejection technologies have been investigated. This methodological approach aims to compare reference dry cooler (DC), hybrid cooler (HC) and wet cooling tower (WCT) under the same working conditions for 227 different geographical locations.

As foreseen, the heat rejection potential is a primary function of the length of the cooling season. At maximum speed the DC reaches an average SPF of 30.5, whereas HC and WCT a value of 37.0 and 192.3, respectively. HC permits to reduce considerably the wetting of the cooling coil up and therefore their water consumption. When the water temperature drop is imposed, it emerges that WCTs are negatively affected by humid weather conditions, with maximum SPF reductions between 25-30% compared to dry locations.

Literature reference

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