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Techno-economic analysis of air-to-water heat rejection systems

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Abstract

The challenge of an effective heat rejection is a key problem in air-conditioning design. Air-based heat rejection systems are the most widespread technologies in cooling applications, but even though these provide a mature technology, a better knowledge of products available on market is desirable in order to support design phases. In this paper a market survey on more than 1300 systems has been carried out within the activities of Task 48 of the International Energy Agency. The result has the form of a database and the collected data have been used for conducting a technical, energetic and economic comparison. A set of performance figures and dimensionless parameters has permitted to characterize heat transfer phenomena for different system technologies. A crosscheck with previous literature works has guaranteed the soundness of the outcomes.

Key-words: cooling, heat rejection, market analysis

1. Introduction

Higher indoor comfort requirements and warmer urban environment conditions have led in the last years a significant increase of the cooling energy demand in Europe also in those locations where there was hardly any before. Heat rejection is at the basis of any cooling process and particular attention should be paid to the consequent great electrical energy and water usage in both electrically-driven cooling (EDC) and in thermally-driven cooling (TDC) systems.

Different heat sinks (ground, water, sky or ambient air) can be used for rejecting the condenser heat (IEA, 2006; IEA, 2011). The choice of the heat rejection strategy is related to many reasons such as the local availability of a heat sink and the temperature gradient along the year, installation issues or investment costs. In most of the cases installation limits are dominant with respect to mere technical motivations. Because of this fact, air-based heat rejection systems have become the most successful and diffuse solution worldwide on the market, because of their flexibility and ease of installation.

Since the share of air-based cooling systems is becoming larger, the main aim of this is paper is to get a better insight of market available system variants. The work is further motivated by the fact that no similar activities have been carried out before. Within the IEA SHC Task 48, a market survey has been conducted on 1309 products. These have been selected from the catalogue of worldwide acknowledged manufactures of air-based heat rejection components (23 in all) when the technical documentation was available. The selection of manufacturers has been made by This analysis has led to the creation of a database of heat rejection components including dry coolers DCs (979) and wet cooling towers WCTs (330) classified according to following criteria:

- general characteristics (brand, series model, typology, sizes, weight);
- rated performance characteristics (chilling water volume flow rate, design capacity, chilling water temperature difference, rated air volume flow rate, rated water consumption);
- fan parameters (number of fan rows, number of fans, rotational speed, max fan electrical power, noise level);

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- coil parameters (heat transfer surface, tubes volume, internal/external tube diameter, fin and tube material, fin thickness and spacing, eventual accessories);
- cost (basic cost, accessories cost, spray water system cost).

The database comprises heat rejection components from small capacities typical of residential applications to great capacities adopted in industrial or tertiary application (Figure 1). In the next sections a synthesis of derived graphs has been presented. It has been additionally specified in the caption the amount of data used for deriving a given correlation.

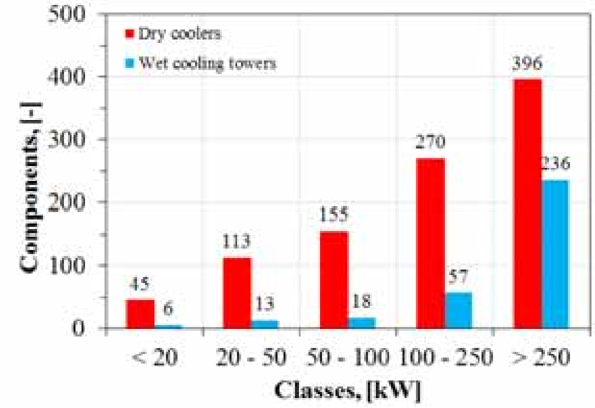

Figure 1. Cooling power classes of the market analysis.

2. Climatic suitability of air-based heat rejection devices

One of the key information listed in the technical documentation of market available DCs and WCTs is the chilling capacity under nominal conditions[1,2]. Along with these figures, many manufacturers also provide correction factors to calculate the performance under off-design conditions. In this way, it is possible to make a comparison between heat rejection devices whose nominal conditions are different and to analyze the influence on the performance of each variable, such as water and air temperatures and mass flow rates. The aim of this chapter is to evaluate the chilling capacity of DCs and WCTs under off-design conditions and to quantify the relative energy potential under different climatic conditions.

In terms of performance figures, a cooling effectiveness ε_{ch} can be defined for DCs and WCTs (Eq. 1) as the ratio between the actual rejected power expressed by water temperature difference (ΔT_w) and the difference between the inlet water temperature $T_{w,in}$ and the sink temperature T_{sink}. In the case of DCs, sink temperature corresponds to the dry bulb ambient temperature whereas in the case of WCTs wet bulb temperature should be used.

$$\varepsilon_{ch} = \frac{\Delta T_w}{T_{w,in} - T_{sink}} \qquad \text{(eq. 1)}$$

Additionally to this the off-design cooling ratio f can be calculated (Eq. 2). It is defined as the ratio between the actual rejected power $Q_{ch,act}$ and the nominal condition $Q_{ch,nom}$.

$$f = \frac{Q_{ch,act}}{Q_{ch,nom}} \qquad \text{(eq. 2)}$$

By comparing the off-design performance of a DC and a WCT of similar capacities (about 28 kW_{ch}), the plots of Figure 2 have been derived. Here the off-design cooling ratio f has been computed for both DC and WCT by placing in the denominator the nominal rejected power of a DC in order to compare both technologies under the same conditions. From here it is evident that the performance of the WCT is better than that of the DC, both in terms of rejected thermal power and cooling effectiveness. In particular, the difference in terms of cooling effectiveness increases when the inlet dry bulb air temperature approaches the

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[1] Nominal conditions for dry coolers: $T_{w,in}$=40°C, $T_{w,out}$=35°C, T_{db}=25°C according to ENV1048:1995.
[2] Nominal conditions for wet cooling towers: $T_{w,in}$=95°F (35°C), $T_{w,out}$=85°F (29.4°C), T_{wb}=78°F (25.5°C) according to CTI STD-203:2005.

inlet water temperature (40°C in this case). It is important to underline that the effectiveness of the WCT is sensitive to a change of the relative humidity condition, in a way that the lower the relative humidity content, the higher the effectiveness. On the contrary when the relative humidity increases, the WCT effectiveness is very close to the dry cooler one. It can be further noticed as off-design cooling ratio and cooling effectiveness for a dry cooler become null before reaching the physical limit temperature (40°C). To conclude, the effectiveness trends found for the dry cooler are in agreement with the outcome of "Solarrück" project (Solarrück, 2013).

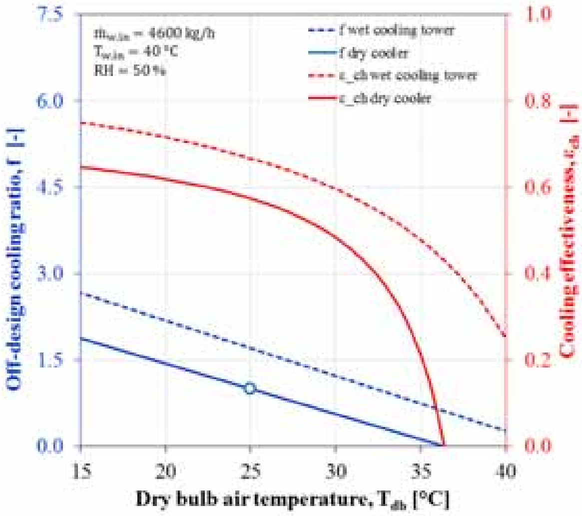

Figure 2. Comparison between dry and wet cooling towers in terms of off-design cooling ratio and cooling effectiveness.

An additional advantage of WCTs on DCs can be expressed in terms of fan's power savings. Because of the lack of free-available information of fan's electric power under off-design conditions, a validated numerical code has been used (Romeli D., 2014). Assuming the same operation ($T_{w,in}$=40°C) and environmental boundary conditions (T_{db}=25°C, RH=50%) for the two technologies, it can be appreciated from Figure 3 as in general fan electrical consumption of DCs is booming when the outlet water temperature is approaching dry bulb ambient temperature. For a given amount of rejected heat (left axis, blue line), WCTs are more effective if the outlet water temperature approaches the ambient temperature, whereas no significant improvement is noticed at higher temperature values. With this regard, the development of so-called "hybrid" heat rejection systems represent a smart and efficient technology to achieve water and electrical energy saving by spraying water only when needed.

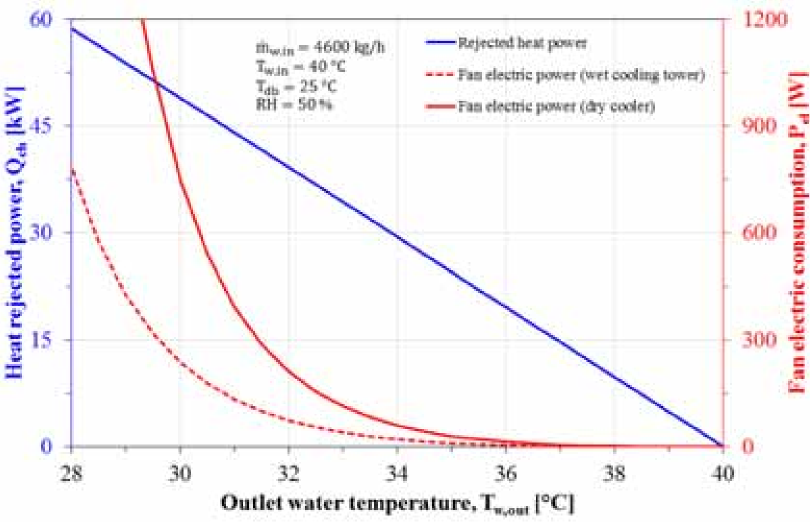

Figure 3. Comparison of fan's electric power of wet cooling towers and dry coolers under the same operation and environmental boundary conditions.

Regarding the application of DCs and WCTs in specific climatic conditions, an analysis in terms of rejected energy potential has been done. The energy potential represents the amount of heat that can be possibly rejected in different locations. Let's consider the three cities of Rome, Montreal and Singapore, as representative locations of the wide range of working conditions in which a DC or a WCT could operate. In order to evaluate a realistic operation of a cooling system, the operation period has been defined during the daytime (horizontal global solar radiation $I_{g,h}$ is greater than 100 W/m^2) and when the ambient temperature is higher than 20°C (see Table 1).

With inlet/outlet water temperature values of 32°C and 27°C respectively and comparing again a DC and a WCT of similar capacities, the energy potential is shown from Figure 4 to Figure 6. Along with the energy potential for the selected wet cooling tower (right axis, red dotted curve) and dry cooler (right axis, red solid curve), the distribution of cooling time defined as the number of hours where a dry bulb air temperature greater or equal to 20°C occurs, is reported (left axis, blue curve).

Table 1. Climatic parameters of reference locations.

Locations	Cooling period (filtered), [h]	Avg. relative humidity, [%]	Max dry bulb temperature, [°C]	Energy potential – Dry cooler, [MWh]	Energy potential – Wet cooling tower, [MWh]
Rome	1521 (17.4%)	59.7	37.6	9.16	47.59
Montreal	813 (9.3%)	64.7	31.7	7.48	28.43
Singapore	3463 (39.5%)	84.0	33.7	5.37	53.98

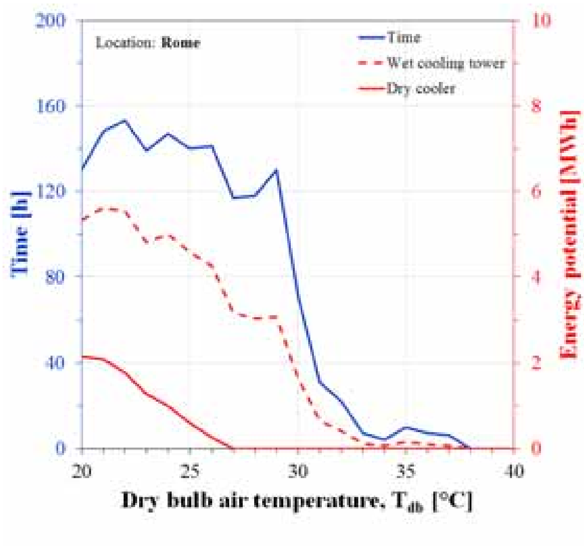

Figure 4. Energy potential of dry cooler and wet cooling tower for the location of Rome (inlet/outlet water temp.: 32/27 °C).

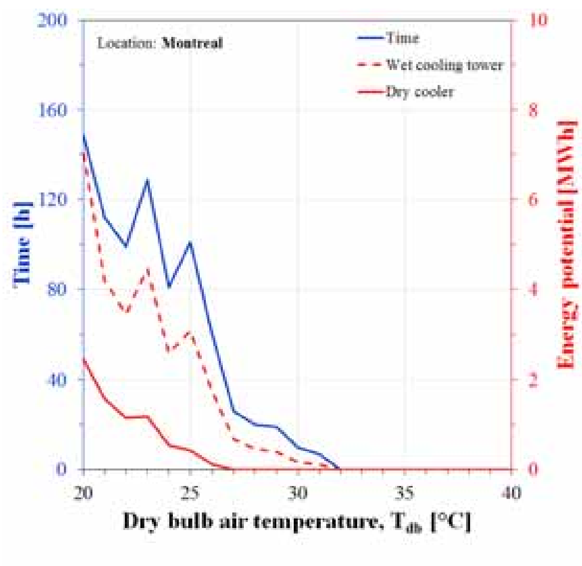

Figure 5. Energy potential of dry cooler and wet cooling tower for the location of Montreal (inlet/outlet water temp.: 32/27 °C).

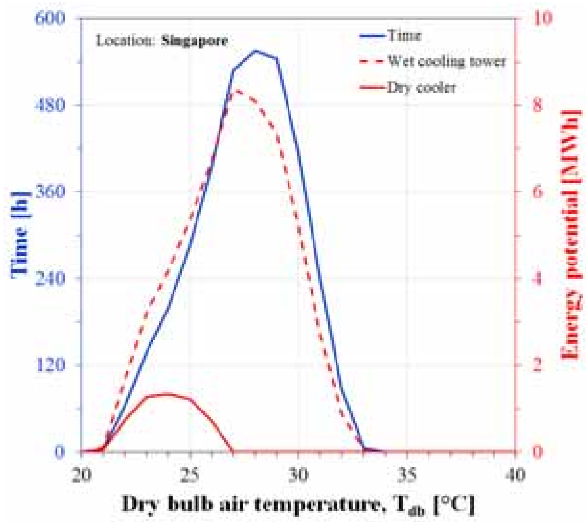

Figure 6. Energy potential of dry cooler and wet cooling tower for the location of Singapore (inlet/outlet water temp.: 32/27 °C).

The enhancement of the amount of rejected heat from a DC compared to a WCT increases significantly and in particular by a factor of 5 for Rome, 4 for Montreal and 10 for Singapore. The difference between the three locations is mainly affected by the length of the cooling period (i.e. Singapore has 39.5% of the year with potential cooling occurrence) and the relative humidity of ambient air (i.e. Rome is drier than Singapore).

3. Technical features

3.1. Sizes

An important feature of air-based heat rejection systems during installation phases are size issues. It has emerged that DCs and WCTs have almost the same weight-to-volume ratio (Figure 7) with a linear trend up to 10 tons. For DCs the average weight-to-volume ratio is between 45-126 kg/m^3, while for WCT is 41-101 kg/m^3. For volumes above ca. 70 m^3, only WCTs can be found. Figure 8 shows as, for a fixed base gross area, WCTs are heavier (208-376 kg/m^2) than dry coolers (97-185 kg/m^2). This is due to piping equipment and accessories additionally required in WCTs that make them higher than DCs.

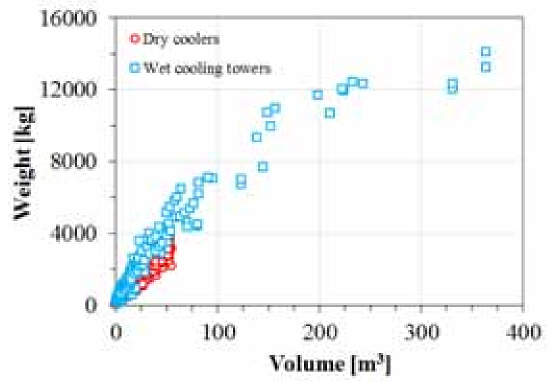

Figure 7. Relationship between volume and weight of air-based heat rejection components (derived from 82.7% of database data).

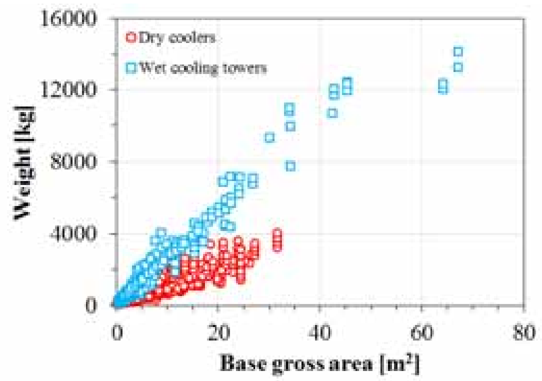

Figure 8. Relationship between coil surface and weight of air-based heat rejection components (derived from 82.7% of database data).

When limitations on the available space are present (i.e. rooftops), it is important to consider the relationship between the minimal amount of space required and the cooling power (Figure 9 and Figure 10). The cooling power-to-volume ratio for dry coolers and wet cooling towers range between 10-40 kW_{ch}/m^3 and 8-47 kW_{ch}/m^3, respectively. An analogous trend is noticed when chilling cooling power is plotted as a function of the gross area and in particular for dry coolers 13-80 kW_{ch}/m^2 and for wet cooling towers 60-163 kW_{ch}/m^2.

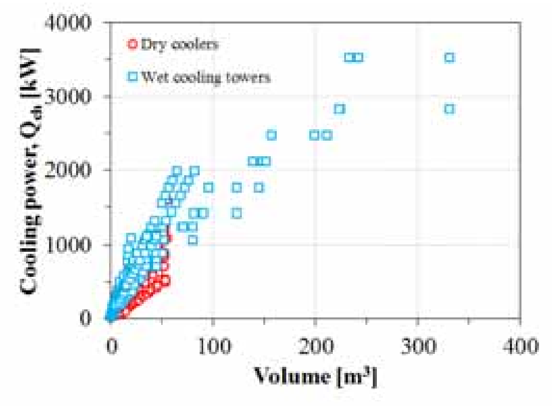

Figure 9. Relationship between volume and cooling output of air-based heat rejection components (derived from 82.7% of database data).

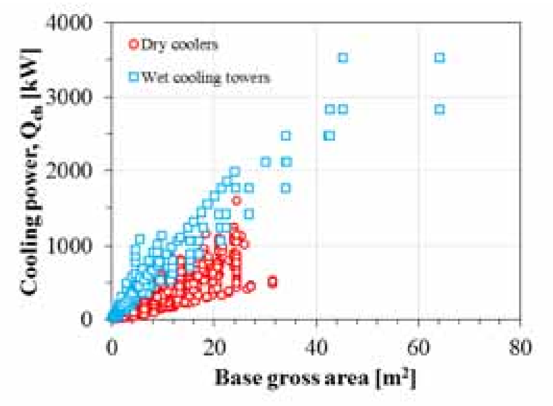

Figure 10. Relationship between coil surface and cooling output of air-based heat rejection components (derived from 82.7% of database data).

3.2. Thermal and electrical powers

Figure 11 shows that WCTs can reject a larger cooling power, under nominal conditions, than DCs components. These seem to have a good linear electric power-to-cooling power relationship. On the contrary, for wet cooling towers different linear trends can be found mostly due to the cooling tower type (open or closed) and the fan function (induced or forced draught towers). The specific consumption values (kW_{el}/kW_{ch}) of dry coolers are in general higher than those of wet cooling towers. In particular, the specific consumption for dry coolers ranges between 0.0125-0.091 kW_{el}/kW_{ch}, while for wet cooling tower is comprised between 0.005-0.060 kW_{el}/kW_{ch}. Finally, wet cooling towers have been divided in induced and forced draught towers (Figure 12). It has resulted that the consumption for the induced draught tower is 0.005-0.025 kW_{el}/kW_{ch} and for the forced draught tower is 0.010-0.060 kW_{el}/kW_{ch}. All these values are in good agreement with those given in the available literature (from Eicker et al, 2012; Saidi et al., 2011 the following average relationships have been gathered: dry coolers: 0.045 kW_{el}/kW_{ch}; wet cooling towers: 0.018 kW_{el}/kW_{ch}; forced: 0.02; induced. 0.007).

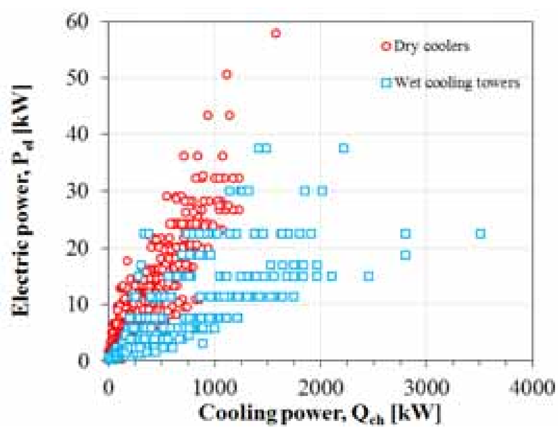

Figure 11. Relationship between cooling power and electric power of air-based heat rejection components (derived from 90.1% of database data).

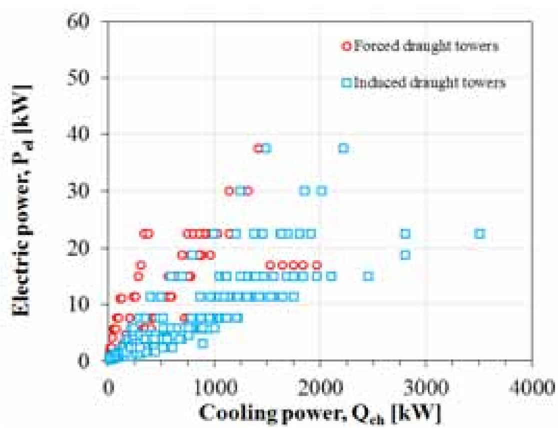

Figure 12. Relationship between cooling power and electric power of wet cooling towers (derived from 98.8% of wet cooling towers).

3.3. Mass flow rate

Another interesting point to investigate is the aspect ratio between the volumetric flow rates on the air and water side. As shown in Figure 13, two very clear linear trends for the two heat rejection technologies can be distinguished. The slope of trend curve is about 1.7×10^3 whereas for wet cooling towers is 0.49×10^3. Therefore for a given chilling water flow rate, the air flow rate elaborated by a dry cooler is about 3.5 times larger than for a wet cooling tower.

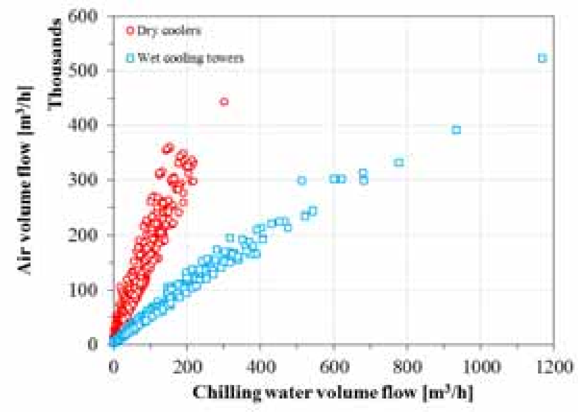

Figure 13. Relationship between air and water volume flow rates (derived from 95.3% of database data).

3.4. Noise level

Within this specific context, noise level is defined as the weighted average of the values measured at a distance of 10 meters. As it is shown in Figure 14, for a fixed fan electric power dry coolers produce less noise than wet cooling towers. For both component categories, the noise level increases with the fan electric power until a maximum noise level is reached: this is about 65 dB for dry coolers and 70 dB for wet cooling towers.

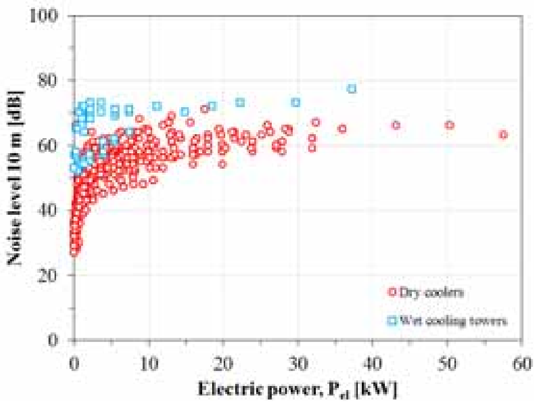

Figure 14. Relationship between noise pressure level and fan electric consumption of air-based heat rejection components (derived from 68.7% of database data).

4. Economic features

For the present analysis, a primary classification was draft by researching on the market the catalogue investment costs of heat rejection devices: starting from a base cost, each manufacturer can provide other components, some of them necessary, i.e. the electric kit, other ones facultative. In particular, four types of accessories have been considered for the cost analysis: wiring (including all the electric components except for the inverter), EC fans controller when the fan velocity is controlled by a highly efficient electronic device, water spray system when the cooler can work in the hybrid mode, and the inverter.

From the collected data, the relationship between the total investment cost and the rejected cooling power has been derived (as shown in Figure 15). For a given cooling power, the investment cost for dry coolers is typically higher than that of wet cooling towers. In particular, the average cost per unit of rejected heat power ranges for between 49 and 107 €/kW$_{ch}$ for dry coolers and between 22 and 27 €/kW$_{ch}$ for wet cooling towers.

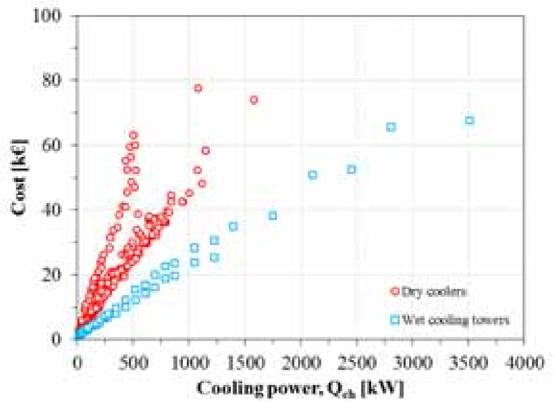

Figure 15. Trend of the investment cost with the provided cooling power (derived from 20.3% of database data).

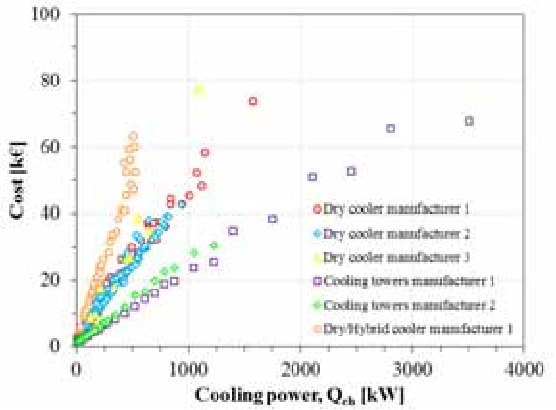

Figure 16. Investment cost–to-cooling power for different manufacturers (derived from 20.3% of database data).

From Figure 15 it is evident that, while wet cooling towers present a good linear cost-to-cooling power relationship, two different trends can be distinguished for dry coolers. This fact can be explained by highlighting the data of Figure 15 for single manufacturers (Figure 16) and noticing that the higher trend costs of dry coolers refer to hybrid components in which spraying water equipment is accounted for too.

5. Conclusions

A market survey on more than 1300 heat rejection components has been carried out by using free-available documentation of worldwide manufacturers within the activities of Task 48 of the International Energy Agency. The work has investigated technical, economical and energy related features of dry coolers (DCs) and wet cooling towers (WCTs). The climatic suitability of both technologies has been further quantified in terms of rejected energy potential by using off-design correction factors provided by the manufacturers and by a validated numerical model developed by the authors.

In some cases, collected data are fragmented or even missing because of the heterogeneity of the technical documentation of different manufacturers and therefore in some cases a clear statement has not been possible. More information would be necessary in particular on investment costs, water usage and rated and off-design fan electrical power.

In this work it has demonstrated how WCTs can reject higher cooling rates than DCs under the same climatic and operation boundary conditions. The average specific consumption value for DCs amounts to 0.033 kW_{el}/kW_{ch} whereas for WCTs is 0.017 kW_{el}/kW_{ch}. On the contrary, WCTs are characterized by higher costs during operation due to fresh water consumption (quantifiable between 4.3 and 7 l of water per kWh of cooling power according to Yik et al., 2001) and during maintenance due to the control of legionella growth. Also because of these reasons the use of WCTs has been prohibited or discouraged from local legislations (i.e. Hong Kong or Middle-East countries).

In general humid locations are not favorable for a WCT installation since the benefit represented by latent heat exchange diminishes when relative humidity increases. On the other hand, DCs should be carefully considered when the outlet water temperature has to be in the range of the dry bulb ambient temperature because of the huge fan electrical power. Therefore the use of so-called "hybrid systems" represents a smart and efficient technology to achieve water and electrical energy savings.

Additionally because of the high weight-to-base gross area ratio and the consequent bearing and static issues, WCTs should be carefully considered during design phases. The average value of the specific weight amounts for WCTs to 244 kg/m^2 and for DCs to 106 kg/m^2.

6. Acknowledgments

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