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Performance evaluation of natural draft dry cooling towers and pre-cooled natural draft dry cooling towers in concentrated solar power plants

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ABSTRACT

This study examines the integration of Natural Draft Dry Cooling Towers (NDDCT) and pre-cooled NDDCT as innovative heat rejection systems for a 50 MW Concentrated Solar Power (CSP) plant, comparing their performance against conventional alternatives: Mechanical Draft Wet Cooling Towers (MDWCT), Air-Cooled Condensers (ACC), and hybrid pre-cooled ACC systems (media pads and spray nozzles). Analytical models for the cooling systems and their interaction with the CSP plant were developed and validated against results reported in the literature. Using hourly temporal resolution and real climatic data from Granada, Spain, the analysis evaluates the annual impact of these systems on energy generation and water consumption. The results demonstrate that pre-cooled NDDCT increases annual power generation by approximately 1400 MWh compared to NDDCT, with substantial performance improvements during peak summer conditions. Although MDWCT achieves the highest efficiency, it comes at the cost of significant water consumption. Pre-cooled NDDCT stands out as a promising hybrid solution, balancing improved condensation performance (only 2.9% less efficiency compared to MDWCT) with minimal water usage (76.7% less than MDWCT). This study provides valuable insights for optimising CSP plant performance in arid regions, advancing beyond previous efforts in the literature.

1. Introduction

A Natural Draft Dry Cooling Tower (NDDCT) is a heat dissipation system that works like a chimney to drive the ambient air across the bundles of the heat exchanger. This dry cooling system primarily relies on convective heat transfer, where heat from the working fluid is rejected to the environment via ambient air. The key advantage of NDDCTs over wet cooling systems is their elimination of water usage, making them an ideal choice for power plants located in arid regions, such as Concentrated Solar Power (CSP) plants, where water resources are scarce. However, NDDCTs suffer from lower efficiency when ambient air temperature is high (hot periods) because NDDCTs performance depend on dry-bulb temperature of the air. These hot periods often match with peak electricity demand and higher market prices. Several hybrid cooling approaches have been reported in the literature to offset the disadvantages related to the use of dry cooling during high temperature periods. The use of evaporative cooling (i.e. introducing a small amount of water to cool the entering air) has become very popular in the last decade for heat rejection with NDDCTs.

In wetted-media cooling, the air flows through a moist cooling pad and water evaporates. The latent heat of water evaporation is extracted from the air resulting in an adiabatic cooling and humidification of the air stream. For example, He et al. [1,2] experimentally investigated the behaviour of two film media, cellulose and PVC, in a low-speed wind tunnel. The authors concluded that cellulose media provided higher cooling efficiency and pressure drop than PVC media. They also observed that both media had severe water entrainment at large air velocities, that could be a problem to avoid water droplets reaching the heat exchanger bundles.

The same group of authors developed an analytical model of a NDDCT coupled with an evaporative cooling pad to evaluate the impact of air pre-cooling on the performance of a NDDCT. Three different cellulose pads with three different thicknesses each were studied in [3]. The effect of the supplied water flow rate to the media, and ambient conditions (temperature and humidity) was investigated in [4]. A comparative study of three different Natural Draft Cooling Towers (i.e., a NDDCT, a pre-cooled NDDCT and a Natural Draft Wet Cooling Tower (NDWCT)) was conducted in [5]. Finally, four wetted media were comparatively studied and the most promising type was proposed for air pre-cooling in NDDCTs in [6]. It can be concluded

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Nomenclature

A	Area (m^2)
C_c	Cooling tower cycles of concentration
c_p	Specific heat (J/(kg K))
D	Drift rate (%)
ΔT_{lm}	Logarithmic mean temperature difference
e_f	Effectiveness of the finned surface
F_T	Logarithmic mean temperature difference correction factor
g	Gravity constant (m/s^2)
H	Height (m)
h	Heat transfer coefficient ($W/(m^2\ K)$)
H_m	Manometric head (m)
K	Loss coefficient
k	Thermal conductivity coefficient (W/(m K))
$\dot{m}$	Mass flow rate (kg/s)
$\dot{m}_b$	Blowdown water mass flow rate (kg/s)
$\dot{m}_d$	Drift water mass flow rate (kg/s)
$\dot{m}$	Mass flow rate (kg/s)
Me	Merkel number ($= h_d a_V V / \dot{m}_w$)
Δp	Pressure loss (Pa)
Pr	Prandtl number
$\dot{Q}$	Heat rate (W)
T	Temperature (K)
U	Overall heat transfer coefficient ($W/(m^2\ K)$)
V	Volume (m^3)
v	Velocity (m/s)
W	Energy consumption/generation (Wh)
$\dot{W}$	Power consumption/generation (W)

Greek symbols

α	Pad constant
β	Pad constant
δ	Pad thickness (m)
ε	Cooling effectiveness
η	Efficiency
γ	Pad constant
μ	Dynamic viscosity (kg/(m s))
ω	Humidity ratio (kg/kg)
ϕ	Relative humidity
ρ	Density (kg/m^3)

Subscripts

a	Air
amb	Ambient
cond	Condenser
evap	Evaporated
ext	Exterior
fr	Frontal
gross	Gross
HE	Heat exchanger
int	Interior
net	Net
pre-cool	Outlet of evaporative cooling
w	Water
wb	Wet-bulb
1	Inlet
2	Outlet

Abbreviations

ACC	Air cooled condenser
CFD	Computational fluid dynamics
CSP	Concentrated solar power
ITD	Initial temperature difference
MDWCT	Mechanical draft wet cooling tower
NDDCT	Natural draft dry cooling tower
NDWCT	Natural draft wet cooling tower

from the literature review that the introduction of wetted media leads to extra pressure drop, which is significant for inlet air pre-cooling of the NDDCT because the pressure drop decreases the airflow through the tower. To avoid this extra pressure drop, Ma et al. [7] proposed a novel solution consisting of annular-arranged wetted media. The authors conducted an exploratory research and concluded that, with the assistance of the introduced moist media, air inflow field optimisation (minimising the effect of crosswind) and inflow air pre-cooling were achieved simultaneously.

In spray cooling applications, water is sprayed into the stream of air and it evaporates. Spray cooling of inlet air of NDDCTs presents several advantages compared to wetted media pre-cooling, including a simpler configuration, lower costs and easy operation and maintenance. The most important advantage, however, is the elimination of the pressure drop induced in the air flow, which is critical in NDDCT whose cooling capacity relies on the induced mass flow rate of air. Sun et al. [8] presented a review on the performance evaluation of natural draft dry cooling towers and possible improvements via inlet air spray cooling. Only a few experimental studies have been conducted to explore the effects of spray cooling on NDDCTs. Alkhedhair et al. [9] performed a detailed experimental investigation of inlet air cooling using water spray in a wind tunnel. The study utilised a representative test cross-section area of $1 \times 1\ m^2$ and a length of 5.2 m. Sun et al. [10,11] further advanced the experimental work by testing different spray cooling system designs on a full-scale natural draft dry cooling tower.

Due to the complexity and costs associated with full-scale experimental setups, most of the research on spray cooling in NDDCTs have been conducted through numerical methods, primarily using Computational Fluid Dynamics (CFD) models. Alkhedhair et al. [12,13] conducted a parametric nozzle design optimisation to maximise spray systems performance. Sun et al. [14,15] investigated the effects of nozzle height, injection direction, and multi-nozzle combination on the spray-cooling performance. Further building on this, Ma et al. [16] found that the pre-cooling reduces the buoyancy and ventilation of the tower while enhancing heat transfer as the reduced inflow air velocity promotes droplet evaporation. He et al. [17] analysed the control mechanisms of spray pre-cooling in NDDCTs, evaluating the impact of different nozzle arrangements, spray angles, and water flow rates on heat rejection rates. The study offered guidelines for designing effective spray systems that balance cooling efficiency with water consumption. Similarly, Pang et al. [18] conducted a detailed study on the feasible zones for nozzle arrangement in a spray pre-cooled NDDCT, focusing on optimising nozzle height, radius, and spray angle to maximise droplet evaporation and improve cooling efficiency, providing specific recommendations for nozzle placement under varying conditions. Additionally, the performance of a spray pre-cooled NDDCT under crosswind conditions has been explored in recent studies [19,20]. Liu et al. [19] examined different spray strategies, investigating how incomplete or complete saturation of the airflow impacts cooling performance. In contrast, Pang et al. [20] delved into how nozzle arrangement variables, such as height, angle, and radius, affect the system's overall effectiveness in crosswind scenarios.

Apart from direct evaporative cooling approaches, high-efficient indirect evaporative cooling methods, such as the Maisotsenko cycle (M-cycle), may also offer a promising solution for pre-cooling the inlet air to NDDCTs, as they are able of achieving greater cooling, below the wet-bulb temperature [21].

The literature review conducted has highlighted the potential of pre-cooled NDDCT as heat rejection systems in areas where water is scarce. However, most studies focus on the performance of NDDCTs without considering their interaction with the power plant. Only few investigations have addressed the interaction between the power block of a CSP and the cooling system. Huang et al. [22] developed a numerical model of an NDDCT with water spray cooling and evaluated its performance under both windless and windy conditions. While they aimed to evaluate the interaction between NDDCTs and the power block, they did not directly couple them. Instead, the calculation of back pressure was performed through a numerical approach by iteratively matching the heat rejection of the cooling system to the assumed exhaust steam conditions, and determining the back pressure based on the final temperature and flow conditions of the circulating water. Miao et al. [23] conducted a comparative study on the cooling performance of NDDCTs using various evaporative pre-cooling systems, including normal wet-medium, A-frame wet-medium, and nozzle spray. Additionally, they coupled the NDDCTs to a supercritical power unit using the EBSILON software for two typical summer days. However, commercial software like EBSILON has notable limitations, as its plant component models are black boxes, limiting flexibility in modifying component selection or the model's internal equations and assumptions. Cutillas et al. [24] developed a numerical model of the entire CSP plant in EES, paying special attention to the Rankine cycle and condenser models. They compared three heat rejection systems: Mechanical Draft Wet Cooling Tower (MDWCT), Air-Cooled Condenser (ACC) and a pre-cooled ACC using media pad. Redelinghuys et al. [25] investigated the use of an adiabatic pre-cooling mechanical draft system for a CSP plant. They developed a numerical model of the condensation system in MATLAB, which was experimentally validated. The CSP plant was modelled in SAM software. However, as with other commercial software, SAM also poses disadvantages due to its black-box nature. To couple the adiabatic pre-cooling, the pre-cooled dry-bulb temperatures and humidity obtained from the MATLAB model were introduced to the ACC inlet. Finally, they compared this novel system, with an ACC and a MDWCT.

While these studies offer valuable insights, they also have notable limitations that this current work seeks to address. To the best of our knowledge, the following issues have not yet been explored in the existing literature:

1. The actual interaction between NDDCTs and the power unit either does not fully capture the dynamic interaction [22] or relies on commercial software with limited flexibility [23]. In this work, a more precise interaction between the NDDCT and the power unit is proposed. Both the CSP plant and the NDDCTs are numerically modelled and coupled through the tower outlet water temperature by fixing a constant Initial Temperature Difference (ITD).
2. While Ref. [22] focuses on back pressure and Ref. [23] examines the power generation of NDDCTs coupled to the power unit, water consumption is a variable that has not yet been investigated.
3. Refs. [22,23] only study NDDCTs as cooling systems for a power unit, and Refs. [24,25] compare different condensation systems but without including the NDDCT. Thus, no research has compared the performance of the power block of a CSP plant coupled to a NDDCT and a spray pre-cooled NDDCTs with other cooling technologies.
4. Refs. [24,25] conduct an hourly analysis throughout a year of operation for different condensation systems in a CSP plant, but do not include an NDDCT. Conversely, Ref. [23] simulates the interaction between NDDCTs and the power block only for two typical days: one hot and dry, and the other non-hot with high humidity. Therefore, there is no hourly analysis of the interaction between NDDCTs and the power block over a full year of operation.

In this study, the primary objective was to analyse the performance of a 50 MW CSP power plant with an NDDCT and a pre-cooled NDDCT as heat rejection systems. The results were compared with the performance of the same plant operating along with four conventional systems: MDWCT, ACC, and pre-cooled ACC using cooling pad and spray nozzle. This analysis was conducted on an hourly basis using real climatological data from the site (Granada, Spain), focusing on both electric generation and water use.

Therefore, the main novelties of this work are:

1. The simulation of NDDCT and pre-cooled NDDCT in a CSP plant with hourly temporal resolution over an entire year, using real meteorological data.
2. The comprehensive evaluation of CSP plant performance when coupled with NDDCT and pre-cooled NDDCT, considering both power generation and water consumption.
3. The comparison of the performance of a CSP plant coupled with an NDDCT and a pre-cooled NDDCT against other cooling technologies.

This paper is organised as follows: the mathematical models of the power cycle, the cooling systems, and the calculation procedure are described in Section 2. Next, the results obtained in the tests are presented and discussed in Section 3. Finally, the most important findings of the research are summarised in Section 4.

2. Methodology

2.1. Plant modelling

The Andasol I is used as a real reference case of a concentrated solar power station. A conventional, reheated Rankine cycle coupled with a parabolic trough solar field is used to generate a net power capacity of 50 MW (Fig. 1). A detailed description of the plant, the main operation parameters, and the plant modelling can be found in [24].

2.2. Cooling systems modelling

Six different cooling systems were considered in the study: natural draft dry cooling tower, pre-cooled natural draft cooling tower, mechanical draft wet cooling tower, air-cooled condenser and pre-cooled air-cooled condenser using cooling pad and spray nozzle. The mechanical draft wet cooling tower is classified as a wet system, the natural draft dry cooling tower and the air-cooled condenser are dry systems while the pre-cooled technologies are considered as hybrid systems.

2.2.1. Natural draft dry cooling tower and pre-cooled natural draft dry cooling tower

A NDDCT is a dry, indirect cooling system where the waste heat from the condenser is transferred to a water stream and then to the air, both times by convection. Hence, this kind of systems depend on the ambient air dry bulb temperature. Fig. 2 shows a schematic arrangement of a typical hyperbolic NDDCT. Here, the geometric parameters and the main locations for air properties evaluation are shown. Air is driven by buoyancy to pass through the heat exchanger bundles to cool the tube-side fluid. The heat exchanger bundles are laid out horizontally at the lower end of the tower (H_3). The operation of

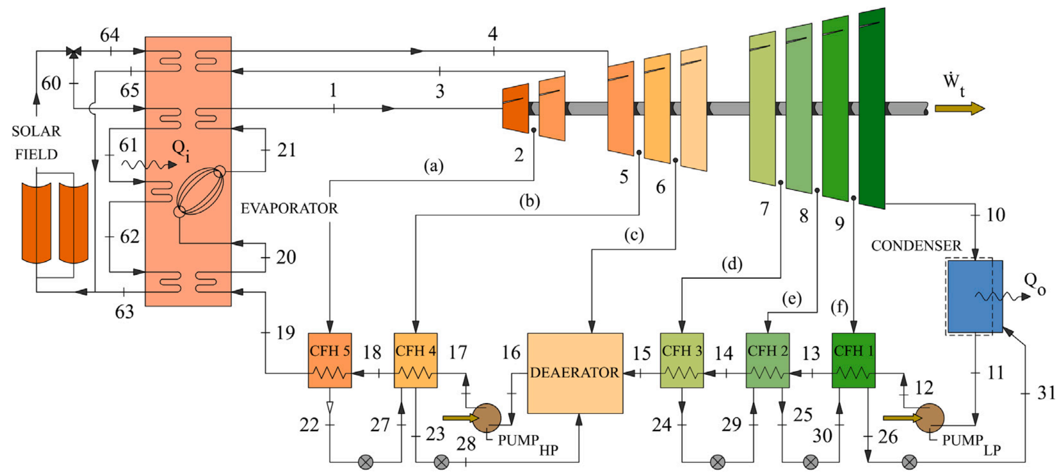


Fig. 1. Schematic arrangement of the Rankine cycle in Andasol I power plant.

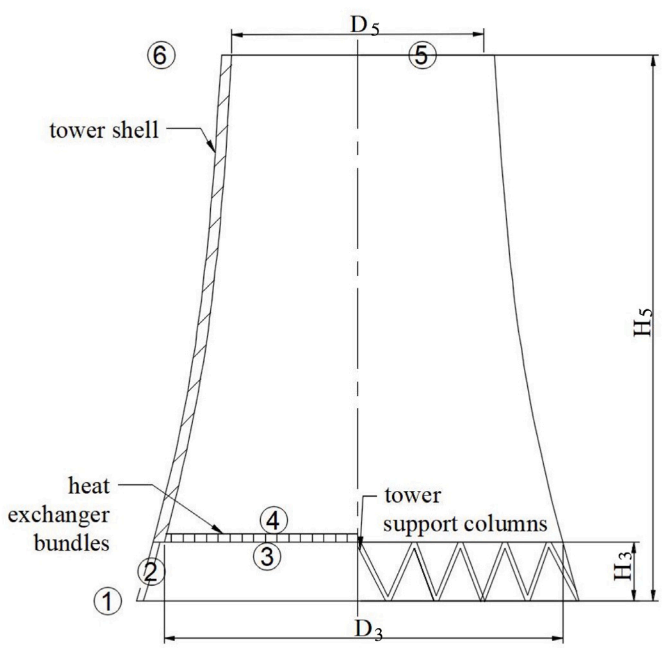


Fig. 2. Typical layout of a hyperbolic NDDCT.

a NDDCT is coupled by the energy and draft equations. The energy equation of the NDDCT can be written as,

$$\dot{Q}_{\text{cond}} = \dot{m}_w c_{p_w}\left(T_{w_1} - T_{w_2}\right) = \dot{m}_a c_{p_{34}}\left(T_4 - T_3\right) \tag{1}$$

$$\dot{Q}_{\text{cond}} = F_T \text{UA}\Delta T_{lm} \tag{2}$$

which can be interpreted as that the NDDCT dissipates the heat rate from the cycle (condenser) which is transferred from the steam to the water, and from the water to the air. Here, F_T is the logarithmic mean temperature difference correction factor to modify the simple counterflow LMTD to crossflow cases, which is defined in [26]. The overall thermal resistance is defined as,

$$\frac{1}{\text{UA}} = \frac{1}{h_{\text{ext}} e_f A_{\text{ext}}} + \frac{1}{h_{\text{int}} A_{\text{int}}} \tag{3}$$

The air-side heat transfer coefficient, h_{ext}, and the effectiveness of the finned surface, e_f, are strongly dependent on the heat exchanger type, geometry, and air velocity, [26].

The draft equation balances the buoyancy against the total flow resistances (or the sum of the pressure drop) across various components of the tower, i.e.,

$$\Delta p = \left(\rho_1 - \rho_4\right) g \left[H_5 - \left(\frac{H_3 + H_4}{2}\right)\right] = \sum K\rho\frac{v^2}{2} \tag{4}$$

It is worth noting that the effect of the height on the outdoor air density in Eq. (4) was neglected in the buoyancy term.

The major component of the total flow resistance is the frictional loss due to the heat exchanger. The flow resistances also include the pressure drop loss at tower supports (Δp_{ts}), the loss due to the separation and redirection of flow at the lower edge of the tower shell (Δp_{ct}) and the contraction and expansion losses at the heat exchanger (Δp_{ctc} and Δp_{cte}). The loss at heat exchanger supports was neglected. Finally, the right side of the draft equation is completed by the loss in kinetic energy at the outlet of the tower (Δp_{to}).

$$\sum K\rho\frac{v^2}{2} = \left(K_{\text{ts}} + K_{\text{ct}} + K + K_{\text{ctc}} + K_{\text{cte}}\right)_{\text{HE}} \frac{\left(\dot{m}_a/A_{\text{fr}}\right)^2}{2\rho_{34}} + K_{\text{to}}\frac{\left(\dot{m}_a/A_5\right)^2}{2\rho_5} \tag{5}$$

The subscript HE in Eq. (5) represents that the corresponding loss coefficients are referred to the frontal area of heat exchanger and the mean density of air flowing through it (ρ_{34}).

Fig. 3 presents a schematic representation of the pre-cooled NDDCT. The spray pre-cooling system is the key component in the performance of a pre-cooled NDDCT. A small amount of water is sprayed by spray nozzles within the inlet air stream. The water evaporates, absorbing the heat from the air stream and cooling it. The pre-cooled air then flows through the heat exchanger bundles as previously described. The performance of the pre-cooling system, is typically characterised by the cooling efficiency, $\varepsilon_{\text{evap}}$, defined as:

$$\varepsilon_{\text{evap}} = \frac{T_{\text{amb}} - T_{\text{pre-cool}}}{T_{\text{amb}} - T_{wb}} = \frac{\omega_{\text{amb}} - \omega_{\text{pre-cool}}}{\omega_{\text{amb}} - \omega_s} \tag{6}$$

where ω_s refers to the humidity ratio of the air at the T_{wb} of the ambient conditions and relative humidity of 100%. As a result, the amount of evaporated water is,

$$\dot{m}_{\text{evap}} = \dot{m}_a\left(\omega_{\text{pre-cool}} - \omega_{\text{amb}}\right) \tag{7}$$

The NDDCT does not need any auxiliary elements to operate, but the pre-cooled NDDCT works with a pump that atomises the water into the air. The pump power consumption is estimated using Eq. (8), assuming a manometric head of 200 m and the pump efficiency equal to 80%, [27].

$$\dot{W}_{\text{pump}} = \frac{\dot{m}_{\text{evap}} g H_m}{\eta_{\text{pump}}} \tag{8}$$

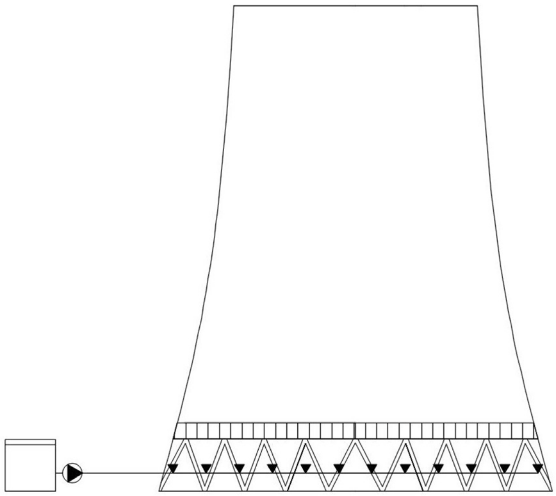


Fig. 3. Schematic representation of a pre-cooled NDDCT using spray cooling.

Table 1
NDDCT design conditions and parameters.

Variable	Value
Tower height, m	130
Tower inlet diameter, m	92
Tower outlet diameter, m	65
Finned tube heat exchanger, Apex angle of A-frame	$2\theta = 61.5°$
Total heat exchanger frontal area, m^2	4444.6
Heat exchanger tubes	4 rows, 2 passes
Designed ambient pressure, kPa	94.8
Designed ambient temperature, °C	37.4
Designed ambient humidity, %	20.0
Designed water mass flow, kg/s	7593.8
Designed heat rejection rate, MW	95.2

The NDDCT and the pre-cooled NDDCT have the designed conditions listed in Table 1. The designed ambient conditions correspond to the hottest hour in the year in Granada (37.4 °C and 20% relative humidity). The heat exchanger used in the NDDCT and the pre-cooled NDDCT is extruded bimetallic finned tubes. The NDDCT and the pre-cooled NDDCT have the same heat exchanger designs: the heat exchanger bundles are laid out horizontally at the lower end of the tower and are arranged in the form of A-frame placed in a radial pattern.

The heat exchanger characteristics (thermal performance and induced pressure drop) have been taken from [5,26], and are shown in Eq. (9).

$$
\begin{aligned}
(hA)_{\text{ext}} &= k\,\text{Pr}\left[383.61731\left(\frac{\dot m_a}{\mu A_{fr}}\right)^{0.523761}\right] \\
K &= 1383.94795\left(\frac{\dot m_a}{\mu A_{fr}}\right)^{-0.332458}
\end{aligned}
\tag{9}
$$

The water-side heat transfer coefficient, h_{int}, was determined by the empirical correlation proposed by Gnielinski. The rest of the geometric parameters included in the equations, including the loss coefficients, were evaluated using the empirical relations provided by [26]. A cooling efficiency of 50% was used in the calculations of the pre-cooled NDDCT performance, as reported by [11].

As a result, for a given ambient conditions and inlet water temperature, the developed models for the NDDCT and the pre-cooled NDDCT provide the air mass flow rate flowing through the NDDCT and the temperatures of the water and the air leaving the heat exchanger.

2.2.2. Mechanical draft wet cooling tower

The operating principle of the mechanical draft wet cooling tower is evaporative cooling, being characterised by the Merkel number, [28, 29]. As a wet system, it depends on the ambient air wet bulb temperature. The analytical modelling of this system is extensively described in [24]. It is modelled using the Poppe and Rögener theory for the thermal evaluation of cooling towers [30]. The set of equations derived by the authors allows to predict the air properties through the entire cooling process.

Table 2
MDWCT design and operating parameters.

Variable	Value
Designed ambient pressure, kPa	94.8
Designed ambient temperature, °C	37.4
Designed ambient humidity, %	20.0
Ratio between water and air mass flow rate	1
Designed water mass flow rate, kg/s	4160.7
Merkel number at designed ratio	1.617

Regarding the water consumption, the major consumption is due to the water evaporation, calculated with the Eq. (7) once the MDWCT outlet air properties are calculated. Additionally, losses by drift and blowdown have also been taken into account, disregarding the rest of losses, like process leaks or cleaning. Drift losses indicate the portion of the total water flow from the MDWCT that escapes into the air, and can be calculated as follows:

$$D = 100\frac{\dot m_d}{\dot m_w} \tag{10}$$

where D is the drift rate, $\dot m_d$ is the water escaping from the tower and $\dot m_w$ is the recirculated water. A 0.0005% drift rate is considered, according to [31]. Blowdown losses ($\dot m_b$) represent the dissolved solids left in the recirculating water as a result of evaporation from the MDWCT, and can be analytically estimated as follows:

$$\dot m_b = \frac{\dot m_{\text{evap}} + \dot m_d}{C_c - 1} \tag{11}$$

where C_c is the cooling tower cycles of concentration. It is assumed C_c = 6, as it is a typical design value [24].

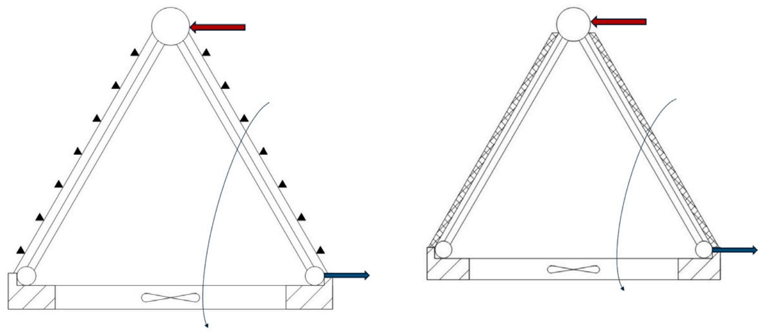
With regard to the system's auxiliary power consumption, the MDWCT requires both a fan and a pump to operate. The fan power consumption is taken as $\dot W_{\text{fan,MDWCT}}$ = 500 kW, as reported in [24]. Similarly, the pump power consumption, assuming a manometric head of 15 m and a pump efficiency of 80%, is calculated using Eq. (8), resulting in $\dot W_{\text{pump,MDWCT}}$ = 1020 kW.

Therefore, for given ambient conditions, inlet water temperature, and MDWCT operating parameters (Merkel number and water-to-air mass flow ratio), the developed models provide the evaporated water mass flow rate as well as the outlet temperatures of the water and air streams.

The MDWCT was designed according to the conditions listed in Table 2. The ambient design conditions correspond to the hottest hour of the year in Granada (37.4 °C and 20% relative humidity). The thermal performance characteristics, specifically the Merkel number, were taken from [32], where a wet cooling tower for CSP applications was experimentally analysed.

2.2.3. Air-cooled condenser and pre-cooled air-cooled condenser

An ACC is a direct, dry cooling system in which water vapour is condensed in a finned-tube heat exchanger, and the waste heat is therefore transferred by convection. Consequently, the performance of this kind of system depends on the ambient air dry-bulb temperature. In contrast to the NDDCT where the flow is buoyancy driven, an ACC uses fans to force ambient air over the heat exchanger surfaces, making it a mechanically driven system. The analytical modelling of this system is extensively described in [24]. The fan consumption is taken from the same reference, being $\dot W_{\text{fan,ACC}}$ = 4066.8 kW.



(a) Spray cooling. (b) Wetted-media pad cooling.

Fig. 4. Schematic diagram pre-cooled ACCs.

Fig. 4 presents a schematic diagram of the pre-cooled air-cooled condensers. As stated in Section 2.2.1, the performance of the pre-cooling system is characterised by the cooling efficiency, defined in Eq. (6).

As in the case of the pre-cooled NDDCT, one of the pre-cooling strategies used for the ACC is spray cooling (Fig. 4(a)). However, the nozzle arrangement is necessary different, resulting in a different value for the cooling efficiency included in Eq. (6). A constant value of 60% is assumed, [23]. The pump consumption is estimated using Eq. (8), assuming a manometric head of 200 m and a pump efficiency of 80%.

The other pre-cooling system is the wetted-media pad (Fig. 4(b)). In this case, the modelling is the one described in [24]. The cooling efficiency and the pressure drop in the media pad can be expressed as follows:

$$\varepsilon = 1 - e^{\left(-\frac{\beta\delta}{v^{\alpha}}\right)} \tag{12}$$

$$\Delta p = \gamma v^2 \tag{13}$$

Here, δ represents the thickness of the pad, v is the measured frontal velocity of the air stream, and α, β, and γ are empirical constants. In this study, the media pad used is CELdek 7090-15 with a thickness of 0.2 m, as it was identified through an optimisation analysis as the most suitable for this application. For this pad, the constants in Eq. (13) are $\alpha = 0.26$, $\beta = 13.2$, and $\gamma = 17.84$. The air velocity was set to 1 m/s, resulting in a cooling efficiency of $\varepsilon = 0.93$. The higher cooling efficiency of wetted-media pads compared to spray systems can be attributed to the increased contact time and more uniform distribution between the water and air streams [23].

Regarding the power consumption, the total fan consumption is estimated adding to the fan consumption of the dry air-cooled condenser the additional pressure drop induced by the media pad. It is important to note that the change in air density due to pre-cooling was neglected when calculating the fan power consumption. This simplification was adopted because the impact under design conditions was not significant (approximately 1.7% difference).

The water consumption for both hybrid systems is calculated with Eq. (7). The ACC and the pre-cooled ACCs have the designed conditions listed in Table 3, where again, the designed ambient conditions correspond to the hottest hour in the year in Granada.

2.3. Interaction between models

Fig. 5 shows an schematic arrangement of each cooling technology interacting with the power block via the plant condenser. The indirect cooling systems (MDWCT, pre-cooled NDDCT and NDDCT) and the power block models are coupled through the tower outlet water temperature (T_{w_2}), while for the direct cooling systems (ACC and pre-cooled ACC), the steam is condensed directly by ambient air or pre-cooled air, without an intermediate water loop.

Table 3
Air-cooled condenser design conditions and parameters.

Variable	Value
Designed ambient pressure, kPa	94.8
Designed ambient temperature, °C	37.4
Designed ambient humidity, %	20.0
Overall heat transfer coefficient, kW/K	7160.8
Designed air mass flow rate, kg/s	6207.4

The performance of all the cooling technologies is evaluated using the Initial Temperature Difference (ITD), which represents the difference between the condensing temperature and the lowest achievable temperature for each specific system. The ITD values considered for each technology are shown in Table 4, according to [33]. The other parameter defined to couple the cooling systems and the power block is the condenser range, which refers to the difference between the inlet and outlet cooling fluid temperature. Table 4 also summarises the considered range values defined for the design conditions.

For the indirect systems, the model calculates the pair of cooling water temperatures (T_{w_1}/T_{w_2}) that simultaneously satisfy the model equations: energy and draft equations for the NDDCT, the Merkel number and the energy equation for the MDWCT, and the heat rejection rate-condensing temperature relationship for the power block.

The results provided by the models cover the heat rejection rate in the condenser, the cycle and plant efficiencies (Eq. (15)), and the net and gross power generation. The net power production includes the consumption of the auxiliary elements of the cooling system (Eq. (16)). The solar field was considered in the analysis as a constant rate of heat input. A 7-h daily schedule (10–17 h) was assumed in the simulations.

$$\eta_{\text{cycle}} = \frac{\dot{W}_{\text{turbines}} - \dot{W}_{\text{pumps}}}{\dot{Q}_{\text{in}}} = \frac{\dot{W}_{\text{gross}}}{\dot{Q}_{\text{in}}} \tag{14}$$

$$\eta_{\text{plant}} = \frac{\dot{W}_{\text{gross}} - \dot{W}_{\text{ancillary}}}{\dot{Q}_{\text{in}}} = \frac{\dot{W}_{\text{net}}}{\dot{Q}_{\text{in}}} \tag{15}$$

$$\dot{W}_{\text{net}} = \dot{W}_{\text{gross}} - \dot{W}_{\text{ancillary}} \tag{16}$$

3. Results and discussion

Model validation

To validate the NDDCT model, the predicted results were compared with those reported in [5,23], where the performance of NDDCTs located in Birdsville, Australia, and Naomao Lake in Xinjiang Province, China, respectively, was evaluated over the course of a year. Fig. 6 illustrates the comparison between the predicted results and the

Table 4
ITD and range for the cooling systems analysed.

Cooling system	Condenser fluid	ITD (°C)	ITD expression	Range (°C)	Range expression
NDDCT and pre-cooled NDDCT	Water	15	T_{cond}-T_{w_2}	3	T_{w_2}-T_{w_1}
MDWCT	Water	15	T_{cond}-T_{wb}	5	T_{w_2}-T_{w_1}
ACC	Air	22	T_{cond}-T_{amb}	15	T_{a_2}-T_{a_1}
Pre-cooled ACCs	Air	22	T_{cond}-$T_{pre\text{-}cool}$	15	T_{a_2}-$T_{pre\text{-}cool}$

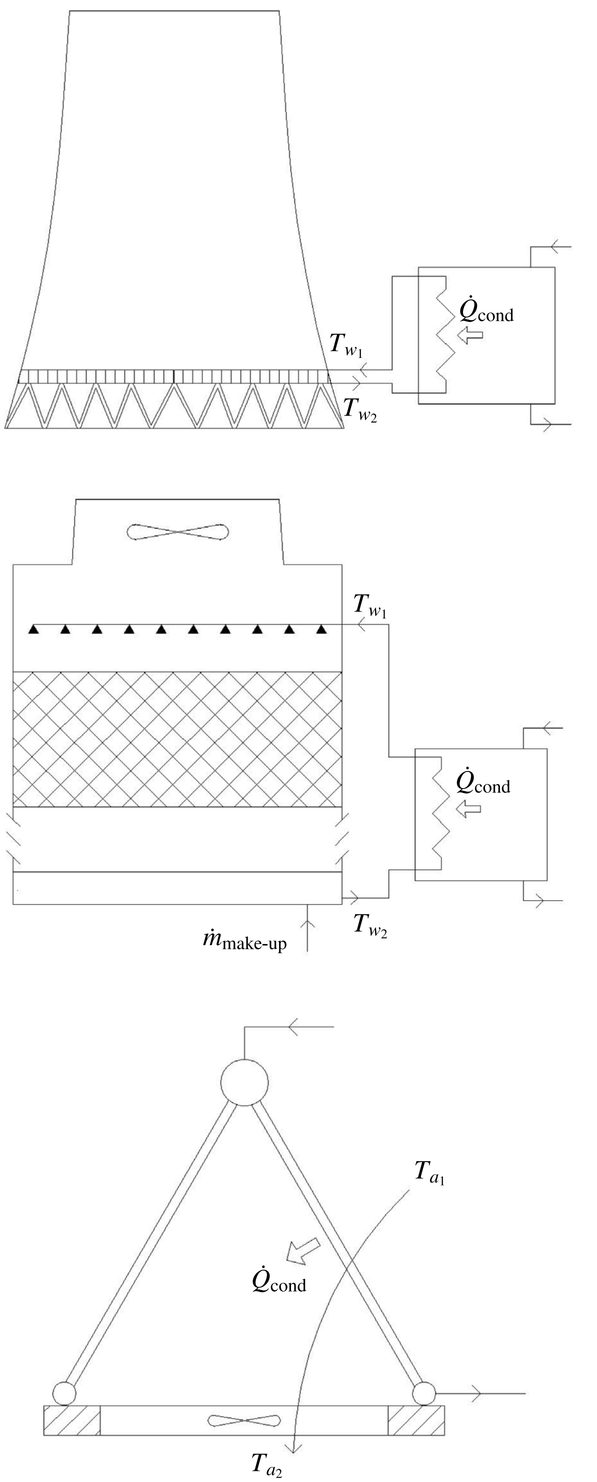


Fig. 5. Schematic arrangement of cooling technologies and their interaction with the power block: (a) NDDCT, (b) MDWCT, and (c) ACC.

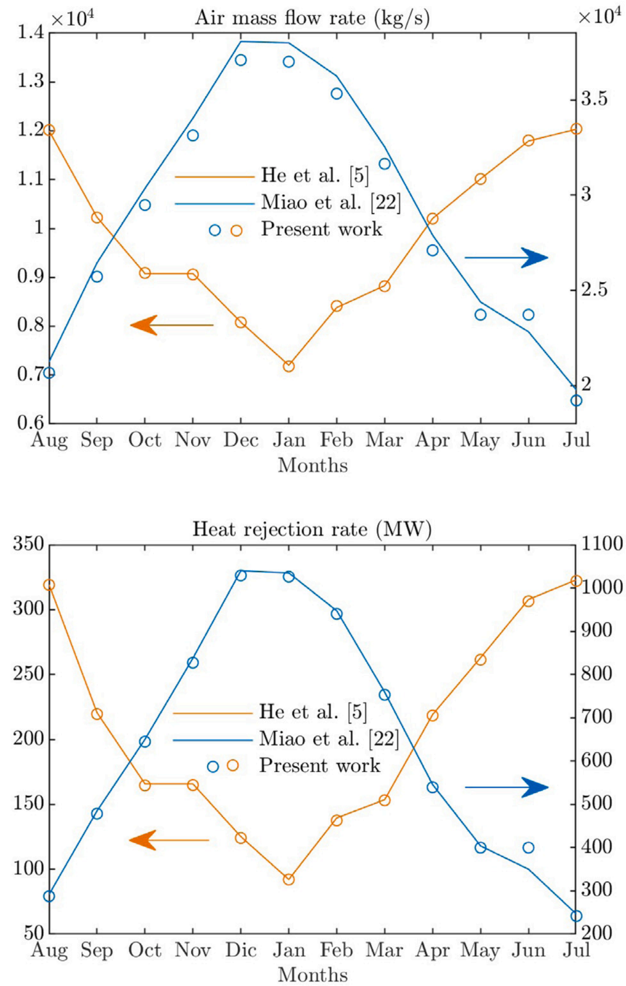
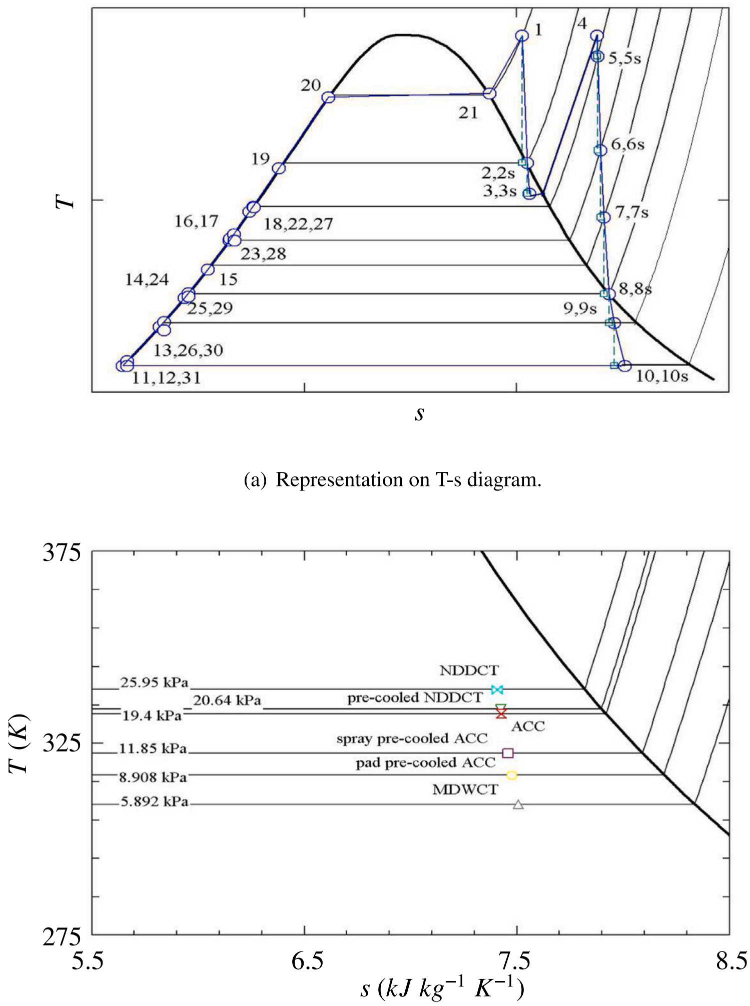


Fig. 6. Comparison between predicted and bibliographic results in [5,23]: (a) air mass flow rate and (b) heat rejection rate.

bibliographic data for the variation in air mass flow rate and the heat rejection rate of the tower. It is important to note that the shapes of the graphs differ because one tower is located in the Northern Hemisphere and the other in the Southern Hemisphere, meaning that the summer season occurs in different months. The results provided by the model show average deviations of 0.14% and 2.8% for air mass flow rate, and 0.5% and 2.2% for heat rejection, respectively, compared to the values published in the literature. Therefore, the model was considered validated.

Regarding the remaining cooling technologies, the models used for the MDWCT and the ACC were those developed by [24]. According to the authors, these models were verified using technical data from the plant under nominal operating conditions. Finally, for the power block, the EES model reported by [24] was also used. As stated by the authors, this model was validated using technical data from the plant at nominal conditions.



(a) Representation on T-s diagram.

(b) Condensing conditions for each cooling technology at 16 h of the 21^{st} of July.

Fig. 7. Rankine cycle operating within the CSP plant.

Energy analysis

Fig. 7(a) depicts the $T-s$ diagram of the Rankine cycle operating within the CSP plant. Point 10 represents the condensation conditions, with the enclosed area indicating the gross power generated. Consequently, this point is the variable influenced by the refrigeration system employed: the lower condensation temperature, the higher gross power generated. Fig. 7(b) focuses on point 10 for each cooling technology analysed at 16 h on the 21^{st} of July, the hottest hour of the day. These trends are further illustrated in Fig. 8, which shows the daily evolution of ambient conditions and condensing temperatures for each cooling technology on the same day. While Fig. 7(b) highlights the performance at a single critical hour, Fig. 8 provides a broader perspective by showing how the cooling systems interact with ambient conditions throughout the day. It is noteworthy that with low humidity, the condensation temperature of the MDWCT is lower than the ambient air dry-bulb temperature, because it works on the evaporative cooling principle, hence it relies on the ambient air wet-bulb temperature. Conversely, dry systems depend on the dry-bulb as heat transfer occurs via convection. Therefore, the trends of the condensation temperature are similar to the dry-bulb ambient air temperature variations. The ACC outperforms the NDDCT because of the increased heat transfer due to the forced convection. In the case of the pre-cooled systems, their condensation temperature is lower than the dry system, in between the air dry-bulb and wet-bulb ambient temperature. Comparing pre-cooled ACC systems, using cooling pads achieves the lowest condensing pressure due to superior cooling efficiency.

This analysis can be extrapolated to the entire year. Table 5 presents the globally integrated results for the most relevant parameters, comparing the performance of the six cooling systems over the 8760 h of a year. The condensing temperature shown in the table represents the annual average, while all other variables (gross energy, net energy and water consumption) are annual totals.

Fig. 9(a) shows the predicted monthly gross energy generation for every technology analysed. As expected, gross energy generation decreases for the hottest months (June-August) because the ambient dry-bulb temperature is higher. The small value observed for February is due the reduced hours of operation. Additionally, the pre-cooling effect is higher for the hottest months of the year due to the higher wet-bulb depression. As mentioned before, the MDWCT demonstrates the highest performance, while the NDDCT performs the worst (7.2% less annually gross energy generated compared with the MDWCT). The difference between both systems increases in summer. By implementing the pre-cooled NDDCT, annual energy generation increases by 1% in comparison with the NDDCT, showcasing its potential, particularly during summer conditions (resulting in 217 MWh more energy generated

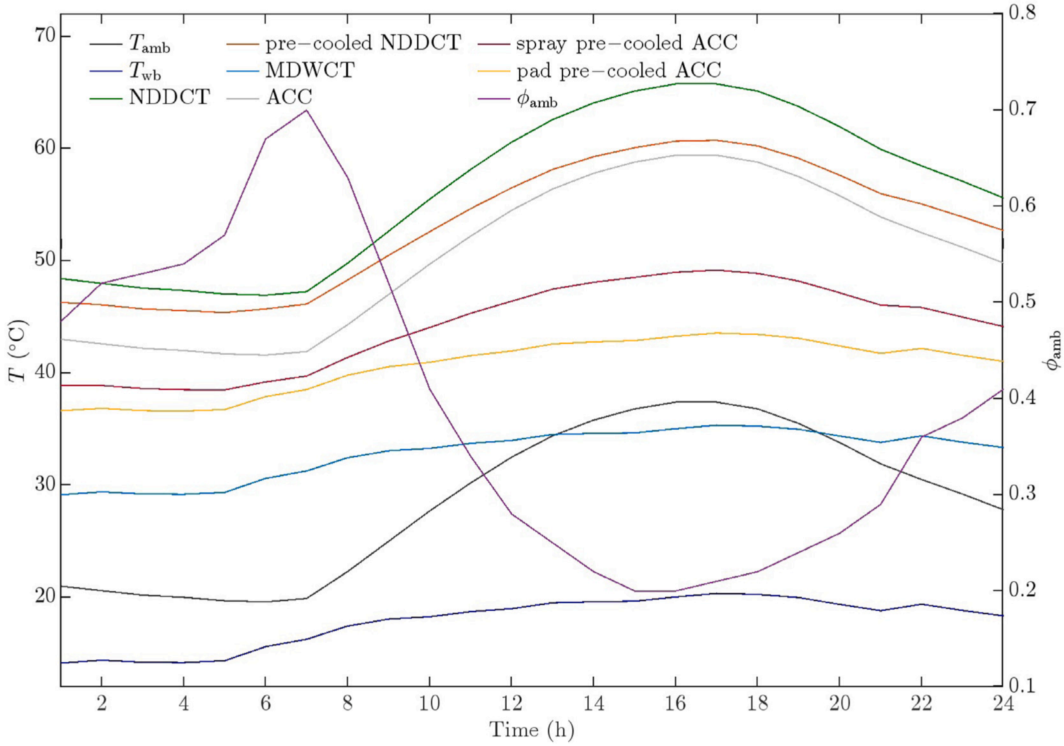


Fig. 8. Daily evolution of the ambient conditions and condensing temperatures for each cooling temperature for the 21st of July.

Table 5
Global integrated results for the six systems compared.

System	Condensation temperature (°C)	Gross energy (MWh)	Net energy (MWh)	Water use (m^3)
Air-cooled condenser	35.55	165 895	154 020	0
Pre-cooled pad ACC	31.03	170 181	157 674	176 470.2
Pre-cooled spray ACC	32.63	168 664	156 711	113 845.4
MDWCT	23.69	175 082	169 943	416 922.6
NDDCT	40.56	162 490	162 490	0
Pre-cooled NDDCT	39.1	163 892	163 825	97 093.9

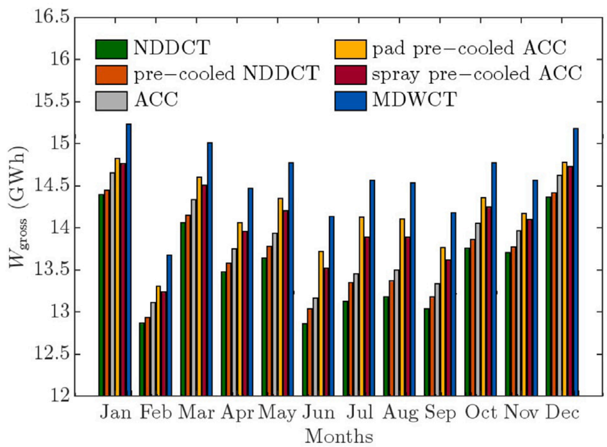
in July). Fig. 9(b) shows the monthly average cycle efficiency, which directly correlates with gross energy generation. Hence, it follows similar trends: MDWCT achieves the highest efficiency (0.413 annual average), followed by pre-cooled ACCs (0.401 and 0.399 for pad and spray), the ACC (0.395), and finally the pre-cooled NDDCT and NDDCT (0.389 and 0.387, respectively).

In terms of ancillary equipment for the plant, Fig. 10 displays the comparison between the annual gross and net energy generation for each technology. It can be observed that the energetic consumption required to drive the fans in the ACC strongly penalise their performance, resulting in a 7.7% difference between gross and net energy generation. This disparity can be attributed to the lower specific heat of air compared to water, requiring a greater airflow to remove heat. On the contrary, the NDDCT does not require any energy consumption for the ancillary equipment, so the net energy generation matches the gross energy generation. As for the pre-cooled technologies using the spray nozzle, the energy required to drive the spray pump is negligible compared to the total energy generation, given the small water flow rate. Consequently, the difference between the gross and net energy generation for the pre-cooled NDDCT is only 0.04%. However, in the case of pre-cooled ACC systems using cooling pads, the pressure loss induced by the cooling pad requires additional energy to drive the fan. As a result, the difference between gross and net energy generation is lower when employing spray nozzles (7.63%) compared to cooling pads (7.93%). Regarding MDWCTs, the variation between gross and net generation is 3% due to the energy needed to drive the fan and the pump.

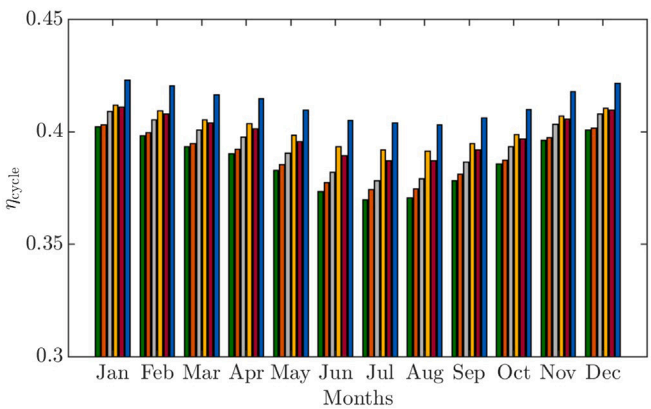
Fig. 11(a) shows the predicted monthly net energy generation and Fig. 11(b) depicts the monthly average efficiency of the plant for each technology. Similar to the previous discussion concerning cycle efficiency and gross energy, plant efficiency is also directly correlated with net energy generation and follows similar trends. Notably, despite having lower cycle efficiency, NDDCTs outperform ACCs in terms of net energy generation and plant efficiency. When compared to MDWCTs, the plant efficiency of pre-cooled NDDCTs is only 2.9% lower, whereas for ACCs, this difference is at least 7.2%.

Water consumption

Fig. 12 depicts the monthly water consumption. It is worth noting that dry systems, which do not require water for condensation, exhibit no water consumption in this process. The data reveals a notable increase in water consumption during summer (dry months) and a decrease in winter (humid months). This fluctuation can be attributed to higher wet bulb depression in drier ambient conditions. The MDWCT is, by far, the system that uses the largest amount of water since water evaporation constitutes the operating principle of MDWCTs. Not only the evaporated water must be taken into account when evaluating



(a) Total monthly gross energy generation.



(b) Average monthly cycle efficiency.

Fig. 9. Comparison between technologies: (a) gross energy generation, (b) cycle efficiency.

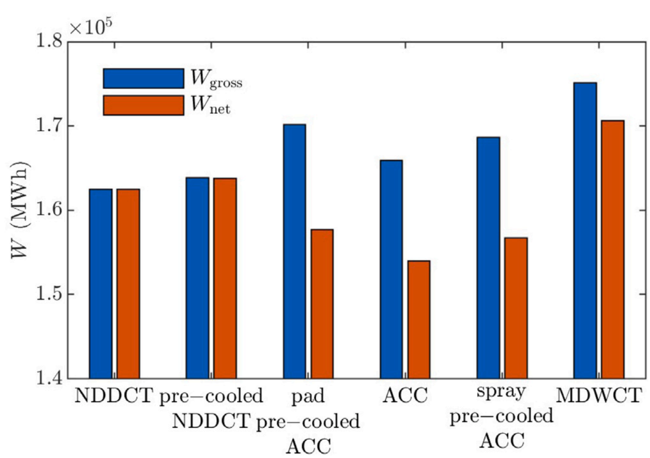
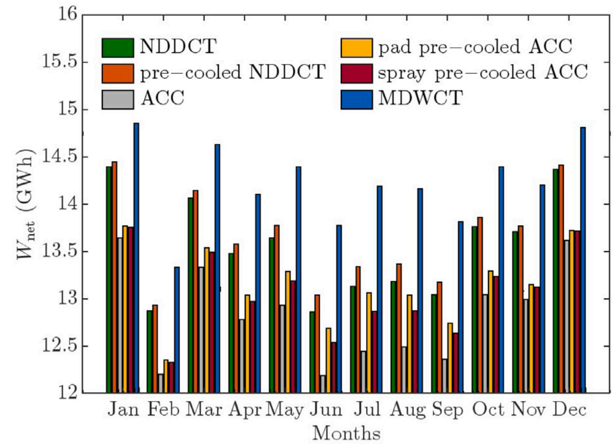
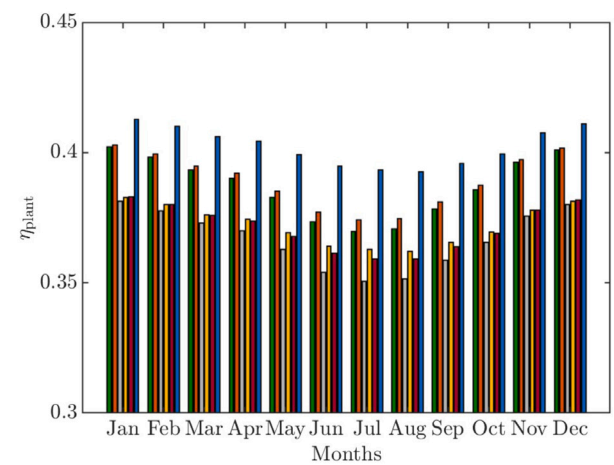


Fig. 10. Comparison between the annual gross and net energy generation.

the water usage of MDWCTs, but also the water used in maintenance and purging tasks to uphold circulation quality and the incidental water drift carried by the air stream because of its operating system.



(a) Total monthly net energy generation.



(b) Average monthly plant efficiency.

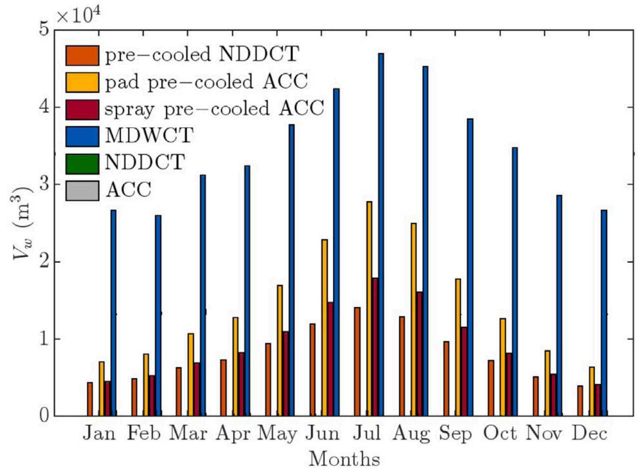
Fig. 11. Comparison between technologies: (a) net energy generation, (b) plant efficiency.

Though quantitatively minor, this drift holds significant environmental implications. In fact, MDWCTs are classified as high-risk systems due to their susceptibility to Legionella proliferation and dispersion issues. As for the pre-cooled systems, they have different water use requirements depending on the air mass flow rate in the system and the cooling efficiency: higher efficiency or air mass flow rate correlates with greater water consumption. Fig. 12(b) shows the annual water consumption per unit of energy generated. The MDWCTs requires 2.45 m^3 per MWh, consistent with the data provided by other Refs. [34] (2.3–3.4 m^3 per MWh). On the contrary, the pre-cooled NDDCT only needs 0.59 m^3 per MWh. Meanwhile, pre-cooled ACC systems require 0.726 m^3 per MWh with nozzle spray, or 1.12 m^3 per MWh with cooling pads.

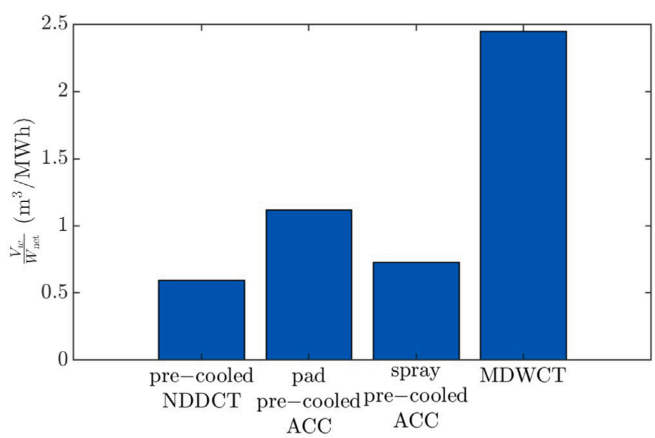
4. Conclusions

This study provides a comparative analysis of various condensation systems utilised in a 50 MW solar thermal power plant located in Granada (Spain), outlining the key findings and conclusions as follows:

- The MDWCT is by far the most efficient option, with a 7.78% higher net energy generation compared to the pre-cooled ACC. However, due to its high water consumption (416923 m^3 annually), may not be a viable option for arid regions with water scarcity.



(a) Monthly water usage for each cooling system.



(b) Annual water consumption in relation with the energy obtained.

Fig. 12. Water consumption.

- Comparing dry systems, it is concluded that the dry tower is a better option than the ACC because it yields higher net energy (5.5% more), despite a 2% lower gross energy generation. This is attributed to the high auxiliary consumption of the air-cooled condenser.
- The major disadvantage of the NDDCT compared to wet systems is its low efficiency, especially in summer months. Specifically, in July, a 7% lower net energy is obtained compared to the MDWCT.
- To mitigate this limitation, a hybrid system using evaporative pre-cooling can be considered. By pre-cooling the air using spray, gross energy generation is increased at the expense of a small water consumption (0.593 m^3 per MWh). In particular, in July, a 1.6% increase in net energy is achieved by consuming 14044 m^3 of water.
- Evaporative pre-cooling depends on ambient conditions, being more significant in hot and dry climates. For instance, in January, the difference in net energy generated between pre-cooled and non-pre-cooled NDDCT is less than 0.4%, while in July, it rises to 1.6%.
- Among the two technologies analysed for evaporative pre-cooling of air in the air-cooled condenser, it is concluded that using spray is preferable. This is because the same net power is obtained (difference less than 1%) while consuming 35% less water.

In conclusion, the pre-cooled NDDCT emerges as a viable option for implementation as a condensation system in CSP plants because it experiences minimal efficiency penalties compared to wet systems: 2.9% lower over the course of the year, with the greatest difference observed during summer months (4.5% in July) and the lowest in winter (1.95% in January). Simultaneously, the water consumption required to operate the plant is substantially lower, a 76.7% less. As a result, environmental performance of the plant is enhanced, with minimal impact on its energy efficiency. It is important to note that the analysis was conducted for a specific location using real climatological data; therefore, the results may vary under different climate conditions.

CRediT authorship contribution statement

J. Ruiz: Writing – review & editing, Writing – original draft, Supervision, Investigation, Funding acquisition, Formal analysis, Conceptualization. **C. Gascó:** Writing – review & editing, Writing – original draft, Investigation, Formal analysis. **M. Opolot:** Writing – review & editing, Writing – original draft. **K. Hooman:** Writing – review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

References

[1] He S, Guan Z, Gurgenci H, Hooman K, Lu Y, Alkhedhair AM. Experimental study of film media used for evaporative pre-cooling of air. Energy Convers Manage 2014;87:874–84. http://dx.doi.org/10.1016/j.enconman.2014.07.084, URL https://www.sciencedirect.com/science/article/pii/S0196890414007237.

[2] He S, Guan Z, Gurgenci H, Hooman K, Lu Y, Alkhedhair AM. Experimental study of the application of two trickle media for inlet air pre-cooling of natural draft dry cooling towers. Energy Convers Manage 2015;89:644–54. http://dx.doi.org/10.1016/j.enconman.2014.10.031, URL https://www.sciencedirect.com/science/article/pii/S0196890414009121.

[3] He S, Gurgenci H, Guan Z, Alkhedhair AM. Pre-cooling with Munters media to improve the performance of Natural Draft Dry Cooling Towers. Appl Therm Eng 2013;53(1):67–77. http://dx.doi.org/10.1016/j.applthermaleng.2012.12.033, URL https://www.sciencedirect.com/science/article/pii/S1359431113000070.

[4] He S, Guan Z, Gurgenci H, Jahn I, Lu Y, Alkhedhair AM. Influence of ambient conditions and water flow on the performance of pre-cooled natural draft dry cooling towers. Appl Therm Eng 2014;66(1):621–31. http://dx.doi.org/10.1016/j.applthermaleng.2014.02.070, URL https://www.sciencedirect.com/science/article/pii/S1359431114001628.

[5] He S, Gurgenci H, Guan Z, Hooman K, Zou Z, Sun F. Comparative study on the performance of natural draft dry, pre-cooled and wet cooling towers. Appl Therm Eng 2016;99:103–13. http://dx.doi.org/10.1016/j.applthermaleng.2016.01.060, URL https://www.sciencedirect.com/science/article/pii/S1359431116300102.

[6] He S, Xu Y, Zhang G, Hooman K, Gao M. Selection of wetted media for pre-cooling of air entering natural draft dry cooling towers. Appl Therm Eng 2017;114:857–63. http://dx.doi.org/10.1016/j.applthermaleng.2016.11.179, URL https://www.sciencedirect.com/science/article/pii/S135943111632018X.

[7] Ma H, Cai L, Si F, Wang J. Exploratory research on annular-arranged moist media to improve cooling capacity of natural draft dry cooling tower and thermo-flow characteristics of its radiators. Int J Heat Mass Transfer 2021;172:121123. http://dx.doi.org/10.1016/j.ijheatmasstransfer.2021.121123, URL https://www.sciencedirect.com/science/article/pii/S001793102100226X.

[8] Sun Y, Guan Z, Hooman K. A review on the performance evaluation of natural draft dry cooling towers and possible improvements via inlet air spray cooling. Renew Sustain Energy Rev 2017;79:618–37. http://dx.doi.org/10.1016/j.rser.2017.05.151, URL https://www.sciencedirect.com/science/article/pii/S1364032117307943.

[9] Alkhedhair A, Guan Z, Jahn I, Gurgenci H, He S. Water spray for pre-cooling of inlet air for Natural Draft Dry Cooling Towers – Experimental study. Int J Therm Sci 2015;90:70–8. http://dx.doi.org/10.1016/j.ijthermalsci.2014.11.029, URL https://www.sciencedirect.com/science/article/pii/S1290072914003421.

[10] Sun Y, Guan Z, Gurgenci H, Hooman K, Li X, Xia L. Investigation on the influence of injection direction on the spray cooling performance in natural draft dry cooling tower. Int J Heat Mass Transfer 2017;110:113–31. http://dx.doi.org/10.1016/j.ijheatmasstransfer.2017.02.069, URL https://www.sciencedirect.com/science/article/pii/S001793101633383X.

[11] Sun Y, Guan Z, Gurgenci H, Li X, Hooman K. A study on multi-nozzle arrangement for spray cooling system in natural draft dry cooling tower. Appl Therm Eng 2017;124:795–814. http://dx.doi.org/10.1016/j.applthermaleng.2017.05.157, URL https://www.sciencedirect.com/science/article/pii/S1359431117313339.

[12] Alkhedhair A, Gurgenci H, Jahn I, Guan Z, He S. Numerical simulation of water spray for pre-cooling of inlet air in natural draft dry cooling towers. Appl Therm Eng 2013;61(2):416–24. http://dx.doi.org/10.1016/j.applthermaleng.2013.08.012, URL https://www.sciencedirect.com/science/article/pii/S135943111300584X.

[13] Alkhedhair A, Jahn I, Gurgenci H, Guan Z, He S. Parametric study on spray cooling system for optimising nozzle design with pre-cooling application in natural draft dry cooling towers. Int J Therm Sci 2016;104:448–60. http://dx.doi.org/10.1016/j.ijthermalsci.2016.02.004, URL https://www.sciencedirect.com/science/article/pii/S1290072916000521.

[14] Sun Y, Guan Z, Gurgenci H, Wang J, Dong P, Hooman K. Spray cooling system design and optimization for cooling performance enhancement of natural draft dry cooling tower in concentrated solar power plants. Energy 2019;168:273–84. http://dx.doi.org/10.1016/j.energy.2018.11.111, URL https://www.sciencedirect.com/science/article/pii/S0360544218323223.

[15] Sun Y, Duniam S, Guan Z, Gurgenci H, Dong P, Wang J, Hooman K. Coupling supercritical carbon dioxide Brayton cycle with spray-assisted dry cooling technology for concentrated solar power. Appl Energy 2019;251:113328. http://dx.doi.org/10.1016/j.apenergy.2019.113328, URL https://www.sciencedirect.com/science/article/pii/S0306261919310025.

[16] Ma H, Si F, Zhu K, Wang J. Quantitative research of spray cooling effects on thermo-flow performance of the large-scale dry cooling tower with an integrated numerical model. Int J Heat Mass Transfer 2019;141:799–817. http://dx.doi.org/10.1016/j.ijheatmasstransfer.2019.06.085, URL https://www.sciencedirect.com/science/article/pii/S0017931019308774.

[17] He S, Li Y, Wang M, Wang A, Alkhedhair AM, Guan H, Wu X, Shi Y, Gao M. Investigation on the control mechanism of spray pre-cooling the inlet air of natural draft dry cooling tower. Appl Therm Eng 2022;217:119186. http://dx.doi.org/10.1016/j.applthermaleng.2022.119186, URL https://www.sciencedirect.com/science/article/pii/S1359431122011164.

[18] Pang H, Zhao C, Cheng S, Wang M, Gao Q, He S, Sun G, Jiang G, Sun Q, Liu Z, Geng Z, Zhang S, Gao M. Investigation on feasible zone of nozzle spray for pre-cooling the inlet air of natural draft dry cooling tower. Therm Sci Eng Prog 2023;38:101650. http://dx.doi.org/10.1016/j.tsep.2023.101650, URL https://www.sciencedirect.com/science/article/pii/S2451904923000033.

[19] Liu T, Wang J, Song T, He S, Zhan J, Geng Z, Zhang S, Huang X, Qi J, Gao M, Shi Y. Investigation on the spray strategy for the inlet air pre-cooling of natural draft dry cooling tower under typical crosswind. J Wind Eng Ind Aerodyn 2023;239:105465. http://dx.doi.org/10.1016/j.jweia.2023.105465, URL https://www.sciencedirect.com/science/article/pii/S016761052300168X.

[20] Pang H, Wang M, Gao Q, Cheng S, Liu Z, Geng Z, Zhang S, He S, Liu T, Zhao C, Liu W, Gao M. Nozzle arrangement study for pre-cooling the inlet air of natural draft dry cooling tower under crosswinds. Int J Refrig 2023;148:1–12. http://dx.doi.org/10.1016/j.ijrefrig.2023.01.011, URL https://www.sciencedirect.com/science/article/pii/S0140700723000117.

[21] Overview of the Maisotsenko cycle – A way towards dew point evaporative cooling. Renew Sustain Energy Rev 2016;66:537–55. http://dx.doi.org/10.1016/j.rser.2016.08.022.

[22] Huang X, Chen L, Yang L, Du X, Yang Y. Evaporation aided improvement for cooling performance of large scale natural draft dry cooling system. Appl Therm Eng 2019;163:114350. http://dx.doi.org/10.1016/j.applthermaleng.2019.114350, URL https://www.sciencedirect.com/science/article/pii/S1359431119311597.

[23] Miao J, He S, Lu Y, Wu Y, Wu X, Zhang G, Gao M, Geng Z, Zhang S. Comparison on cooling performance of pre-cooled natural draft dry cooling towers using nozzles spray and wet medium. Case Stud Therm Eng 2021;27:101274. http://dx.doi.org/10.1016/j.csite.2021.101274, URL https://www.sciencedirect.com/science/article/pii/S2214157X21004378.

[24] Cutillas C, Ruiz J, Asfand F, Patchigolla K, Lucas M. Energetic, exergetic and environmental (3E) analyses of different cooling technologies (wet, dry and hybrid) in a CSP thermal power plant. Case Stud Therm Eng 2021;28:101545. http://dx.doi.org/10.1016/j.csite.2021.101545, URL https://www.sciencedirect.com/science/article/pii/S2214157X21007085.

[25] Redelinghuys L, Tshamala M, Hans T. Performance of an adiabatic pre-cooling system for concentrating solar power plants in arid areas. Appl Therm Eng 2023;231:120819. http://dx.doi.org/10.1016/j.applthermaleng.2023.120819, URL https://www.sciencedirect.com/science/article/pii/S1359431123008487.

[26] Kröger D. Air-cooled heat exchangers and cooling towers: thermal-flow performance evaluation and design. Vol. 2, PennWell Books; 2004.

[27] Ruiz J, Martínez P, Aguilar F, Lucas M. Analytical modelling and optimisation of a solar-driven cooling system enhanced with a photovoltaic evaporative chimney. Appl Therm Eng 2024;245:122878. http://dx.doi.org/10.1016/j.applthermaleng.2024.122878, URL https://www.sciencedirect.com/science/article/pii/S1359431124005465.

[28] Ruiz J, Navarro P, Hernández M, Lucas M, Kaiser A. Thermal performance and emissions analysis of a new cooling tower prototype. Appl Therm Eng 2022;118065. http://dx.doi.org/10.1016/j.applthermaleng.2022.118065, URL https://www.sciencedirect.com/science/article/pii/S135943112200031X.

[29] Navarro P, Ruiz J, Hernández M, Kaiser A, Lucas M. Critical evaluation of the thermal performance analysis of a new cooling tower prototype. Appl Therm Eng 2022;213:118719. http://dx.doi.org/10.1016/j.applthermaleng.2022.118719, URL https://www.sciencedirect.com/science/article/pii/S1359431122006639.

[30] Poppe M, Rögener H. Berechnung von rückkühlwerken. VDI Wärmeatlas 1991;Mi 1.

[31] Ruiz J, Kaiser A, Lucas M. Experimental determination of drift and PM_{10} cooling tower emissions: Influence of components and operating conditions. Environ Pollut 2017;230:422–31. http://dx.doi.org/10.1016/j.envpol.2017.06.073, URL https://www.sciencedirect.com/science/article/pii/S026974911731120X.

[32] Navarro P, Serrano J, Roca L, Palenzuela P, Lucas M, Ruiz J. A comparative study on predicting wet cooling tower performance in combined cooling systems for heat rejection in CSP plants. Appl Therm Eng 2024;253:123718. http://dx.doi.org/10.1016/j.applthermaleng.2024.123718, URL https://www.sciencedirect.com/science/article/pii/S1359431124013863.

[33] EPRI. Comparison of alternate cooling technologies for california power plants: economic, environmental, and other tradeoffs. 2002.

[34] Cooperman A, Dieckmann J, Brodrick J. Power plant water use. ASHRAE J 2012;54:65–8.