

Evaluation of cooling requirements of post-combustion CO₂ capture applied to coal-fired power plants

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Abstract

Whilst CO₂ capture and storage (CCS) technology is widely regarded as being an important tool in mitigating anthropogenic climate change, care must be taken that its extensive deployment does not substantially increase the water requirements of electricity generation. In this work, we present an evaluation of the cooling demand of an amine-based post-combustion CO₂ capture process integrated with a coal-fired power plant. It is found that the addition of a capture unit translates into an increase in the total cooling duty of $\approx 47\%$ (subcritical), $\approx 33\%$ (supercritical) and $\approx 31\%$ (ultra-supercritical) compared to a power plant without capture. However, as the temperature at which this cooling is required varies appreciably throughout the integrated power capture process, it is found that this increase in cooling duty (MW) does not necessarily lead to an increase in cooling water usage (kg_{H₂O}/MW). Via a heat integration approach, we demonstrate how astute cascading of cooling water can enable a reduction of cooling water requirements of a decarbonised power plant relative to an unmitigated facility. This is in contrast to previous suggestions that the addition of CCS would double the water footprint.

Keywords: Water-energy nexus, Carbon Capture and Sequestration, Carbon Capture Readiness (CCR)

1. Introduction

1.1. Background and Motivation

The link between climate change [1, 2, 3] and anthropogenic CO₂ emissions [4, 5, 6, 7] has been overwhelmingly acknowledged [8, 9]. The 2015 United Nations Climate Change Conference (COP21) saw a historic international agreement with 193 countries agreeing to work to limit the global temperature rise to no more than 2 °C above pre-industrial levels by the end of the century. Recently, this agreement has gained further momentum with an announcement that the two biggest emitters, the United States and China [10, 11, 12], will both ratify the Paris Agreement. In many countries, the power generation sector is responsible for the majority of CO₂ emissions - for example, thermal power generation from fossil fuels contributed to 25% of the United Kingdom's total CO₂ emissions in 2015 [13, 14, 15]. Thus, it is broadly
10 agreed that CO₂ capture and storage (CCS) will play an important role in climate change mitigation [16, 17, 18]. It has been shown that the UK's future energy mix should accommodate CCS [19, 20] in order to deliver secure energy supply and to meet UK's climate targets of reducing its CO₂ emissions by 80% from 1990 levels by 2050 at least cost [21].

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However, efforts to mitigate climate change must be considered in the context of an increasing population base and an associated increase in global energy demand. The link between water use and power generation is well known, and as CCS technologies will also incur a cooling duty, this has led to concerns that the deployment of CCS technology will unsustainably increase the water footprint of power generation. Cooling demands in fossil fuel-fired power plants are primarily associated with the condenser in the steam cycle. The integration of a carbon capture process with the power plant [22, 23] will serve to increase the number of points where cooling is required; specifically in the direct contact cooler in the flue gas inlet, lean solvent cooler in the solvent loop, the condenser in the solvent regeneration process and also in the intercoolers in the CO₂ compression train. Power plants' cooling systems predominantly [11, 24] utilise water as the cooling medium and consequently ready access to a reliable source of cooling water is an additional constraint on the siting of power plants equipped with CCS [25, 26, 27, 28, 29], and previous studies have anticipated that a CO₂ capture plant will increase [30, 31, 32, 33] the combined cooling requirement of fossil fuel-fired power plant and thus contribute to regional water stress [31, 34, 35].

In the context of post combustion capture from fossil fuel power plants, to date the primary focus has been on the development of new solvents and processes aiming to minimize the associated energy penalty with the capture unit. However, relatively little has been done in order to investigate the variations in individual and combined cooling duties and the corresponding water consumption associated with a post combustion capture unit. Previous studies [30, 36, 37] reporting on the cooling water demand have considered the power plant and capture processes as separate blocks, *i.e.*, not as integrated systems, and have failed to present any detailed breakdown in cooling duties and water consumption. There would appear to exist an important gap in the modelling and evaluation of cooling water requirements of the integrated process.

The remainder of this paper is laid out as follows; we first present the methodology used in this work. This comprises our modelling approach in calculating the cooling duties of the power plant and CO₂ capture and compression process in addition to evaluating cooling tower sizing. We then present our results which provide insight on the cooling duties associated with retrofitting a capture unit to sub-, super- and ultra-supercritical coal-fired power plants. This is followed by a discussion on the selection of the optimum cooling technology and an assessment of the impact of process intensification via the integration of cooling system design on cooling water utilisation and whole-system efficiency.

2. Methodology

2.1. Outline

A model has been developed to estimate the cooling requirements of an amine-based post combustion CO₂ capture unit integrated with sub-, super- and ultra-supercritical coal-fired power plants. In this study, we assume the use of an aqueous 30 wt% monoethanolamine (MEA) solvent, however in principle any solvent can be used.

2.2. Power Plant Modelling

The Integrated Environmental Control Model (IECM) [38] was used to relate the power generated by a pulverized coal-fired power station integrated with post-combustion CO₂ capture with the exhaust gas flowrate and composition produced and to specify the overall process efficiency. Given the steam cycle operating conditions, this enabled the calculation of

the associated steam flowrate thus the cooling duty in the condenser. By mass and energy balance, the solvent flowrate within the capture process was calculated, and thereafter the cumulative cooling duty and water demand of the integrated process. Lastly, the space required for the cooling tower was calculated. An overview of the proposed model is illustrated in Figure 1.

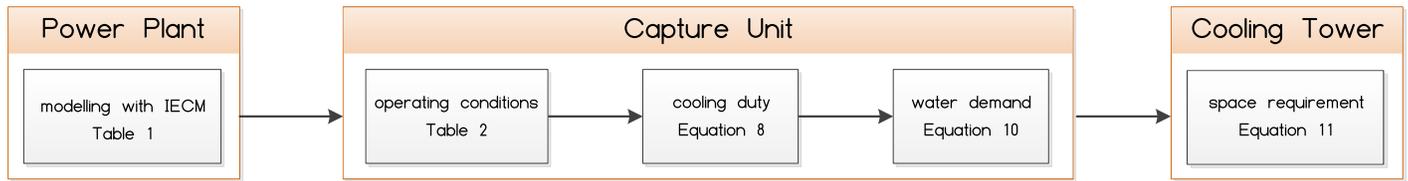


Figure 1: Overview of the proposed model used to calculate the cooling water- and space requirement

The block flow diagram of the integrated power-capture process is illustrated in Figure 2. The exhaust gas (dashed red line) arising from the boiler goes through several post-combustion pollution prevention and control processes: selective catalytic reduction (SCR), inlet air pre-heating (APH), electrostatic precipitator (ESP), flue gas desulphurisation (FGD) and finally CO₂ capture before remaining gas is released to the atmosphere. A portion of the steam from the power plant steam cycle (SC) is diverted to the capture process and the corresponding condensate is returned to the SC (dot-dashed blue line). Figure 2 also illustrates how the cooling water is circulated within the plant and the capture unit.

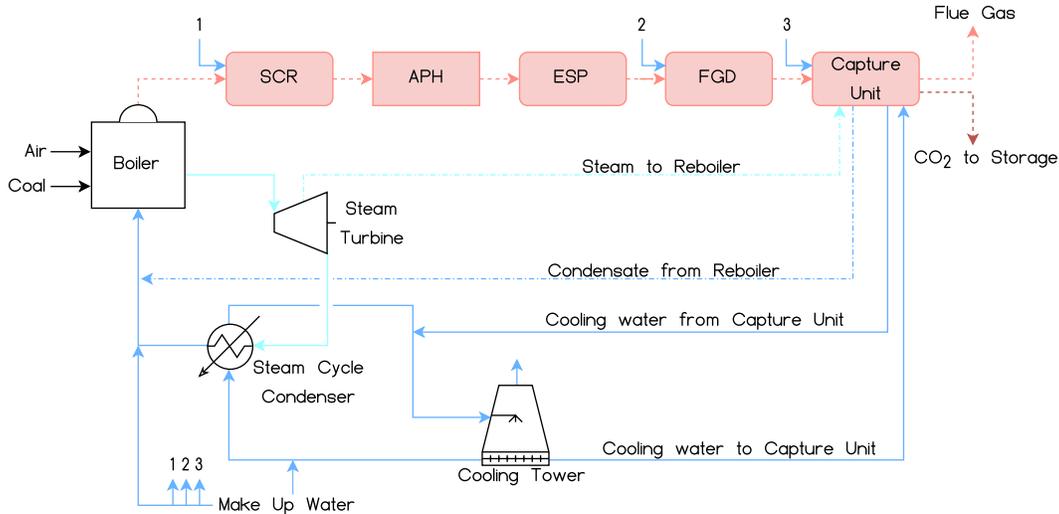


Figure 2: Block flow diagram of a pulverized coal-fired power plant with post-combustion CO₂ capture. Note that whilst a single "Steam Turbine" is shown here, a multi-stage turbine train was considered, with high pressure (HP), intermediate pressure (IP) and low pressure (LP) turbines included in the model. Steam for the solvent regeneration process is bled from the IP-LP cross-over point.

It is important to note that the diversion of steam from the power plant steam cycle has the effect of substantially reducing the cooling duty in the power cycle, and moves it to the capture process. This steam is then condensed in the solvent reboiler, and this heat is used to produce a mixture of stripping steam and CO₂, MEA being essentially non-volatile under these conditions. A sensitivity study with changing steam cycle type was carried out with the fixed parameters in Table 1, with ambient weather conditions are taken as the average values [39] for a typical summer day (16.5 °C dry-bulb, 77% relative humidity) near the location of Drax power station in the UK.

Table 1: IECM power and capture plant configurations: list of fixed parameters and their values [38, 30])

Parameter	Value
NO _x control	Hot-side SCR
Particulates control	Cold-side ESP
SO ₂ control	Wet FGD
CO ₂ capture	Amine System
Coal Type	Appalachian Medium Sulfur
Cooling System	Wet tower, forced draft
Ambient air pressure	0.1024 MPa
Ambient air temperature, dry bulb	16.5 °C
Ambient relative humidity	77 %
CO ₂ mass fraction in flue gas	18.4 wt%
CO ₂ removal efficiency	90 %
Sorbent concentration	30 wt%
Lean loading	0.2 mol mol ⁻¹
Regeneration heat requirement	4492 kJ/kg _{CO₂}

2.3. Cooling Demand of the CO₂ Capture Unit

After passing through the desulphurisation unit, the flue gas enters the direct contact cooler (DCC) in which it is brought into direct contact with a counter-current water flow. The flue gas enters an adiabatic absorber in which CO₂ is removed from the flue gas by the MEA solvent. In order to minimise solvent losses with the CO₂-lean flue gas, a water-wash section is placed after the absorber. The rich solvent is pumped through a heat exchanger into the stripper in which the CO₂ is stripped from the solvent by reheating the liquid, and subsequently passed through an overhead condenser. The high-purity CO₂ stream is then compressed and fed into a pipeline for subsequent transport and storage. The recycled solvent is then cooled prior to being returned to the absorber. This process is illustrated in Figure 3.

As can be observed from Figure 3, heat must be removed from the DCC, the lean solvent cooler, the condenser and the compressor intercoolers. The cooling duty of the pre-cooler can be estimated with an energy balance around the DCC: Given the inlet temperature of the flue gas in the DCC ($T_{DCC,in}$) and the inlet temperature to the absorber ($T_{Absorber}$), the cooling duty of the pre-cooler is calculated as:

$$\dot{q}_{DCC} = \dot{m}_{Flue\ gas} \cdot c_{p,Flue\ gas} \cdot (T_{DCC,in} - T_{Absorber}) \quad (1)$$

A similar approach is employed for the lean solvent cooler (LSC):

$$\dot{q}_{LSC} = \dot{m}_{Solvent} \cdot c_{p,Solvent} \cdot (T_{LSC,in} - T_{Absorber}) \quad (2)$$

Approximately 26% of the reboiler regeneration energy in a 30 wt% MEA system is used for generating stripping stream [40]. As some of this steam will condense within the stripping column, this is an upper bound on the cooling duty in the overhead condenser, and therefore a conservative estimate of the condenser cooling duty is:

$$\dot{q}_{Condenser} = 26\% \cdot 4492\text{ kJ/kg}_{CO_2} \cdot \dot{m}_{CO_2} \quad (3)$$

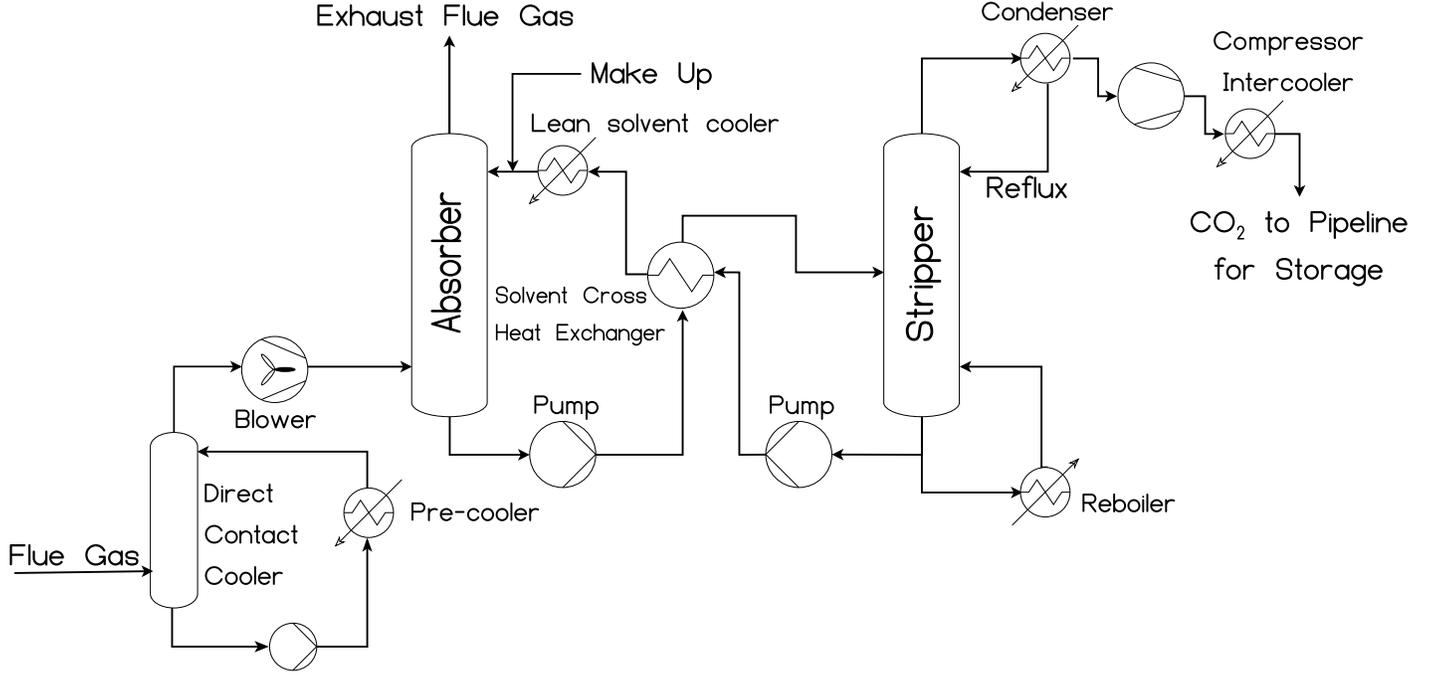


Figure 3: Simplified process flow diagram of a MEA post-combustion capture unit

Note that the value of 4492 kJ/kg_{CO₂} includes all contributions to the energy of regeneration, not simply the energy required to break the chemical bonds between the CO₂ and the MEA.

The cooling duty of the intercoolers in the compression train is dependent on the number of compression stages, their efficiency and pressure ratio. The temperature of the high-purity CO₂-stream at isentropic compressor's discharge is calculated by:

$$T_{\text{Comp,out}} = T_{\text{Comp,in}} \left(\frac{p_{\text{Comp,out}}}{p_{\text{Comp,in}}} \right)^{\frac{\kappa-1}{\kappa}} \quad (4)$$

90 where κ is the isentropic coefficient defined as the ratio between the isobaric and isochoric heat capacity (assumed to be 1.301 [41]) and $p_{\text{Comp,out}}$ is the compressor's discharge pressure. Assuming a neglecting pressure drop and fully-insulated pipes (*i.e.*, no heat loss; $p_{\text{Comp,in}} = p_{\text{Desorber}}$ and $T_{\text{Comp,in}} = T_{\text{Condenser}}$, where p_{Desorber} is the stripper's pressure and $T_{\text{Condenser}}$ is the condenser's temperature), the condenser's cooling duty is calculated as:

$$\dot{q}_{\text{Compression}} = \dot{m}_{\text{CO}_2} \cdot c_{p\text{CO}_2} \cdot (T_{\text{Condenser}} - T_{\text{Comp,out}}) \quad (5)$$

The cooling demand of a compressor with n stages is calculated by:

$$\dot{q}_{\text{Compression}} = n \cdot \dot{m}_{\text{CO}_2} \cdot c_{p\text{CO}_2} \cdot T_{\text{Condenser}} \cdot \left[1 - \left(\frac{p_{\text{Comp,out}}}{p_{\text{Comp,in}}} \right)^{\frac{\kappa-1}{n \cdot \kappa}} \right] \quad (6)$$

Equation 6 was evaluated assuming CO₂ available at 2 bar and compression to 110 bar. The normalised cooling demand is illustrated in Figure 4. The one-stage compressor shows the highest cooling demand and no significant reduction can be observed with more than six stages. For all ensuing calculations, we have used the cooling duty associated with a single stage compression process. This approximation was made in order to be as conservative as possible, *not* because single stage compression of CO₂ is considered to be a realistic option. In reality, a 4 - 6 stage compression train is likely to be employed.

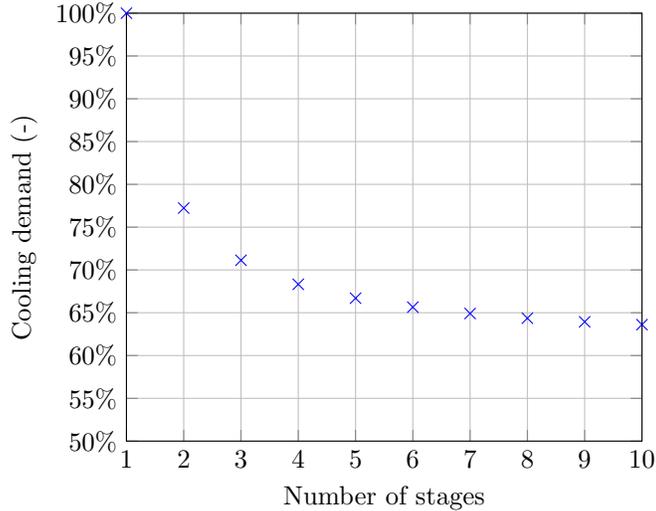


Figure 4: Impact of the number of stages on the cooling duty of the compression unit normalised to that of a one-stage process

The cooling duty of the steam cycle (SC) condenser is calculated using a similar approach. An energy balance around the condenser with the cooling water and a temperature rise of $\Delta T_{\text{SC}_{\text{cond}},\text{H}_2\text{O}}$ will lead to:

$$\dot{q}_{\text{SC}_{\text{cond}}} = \dot{m}_{\text{SC}_{\text{cond}},\text{H}_2\text{O}} \cdot c_{\text{P},\text{H}_2\text{O}} \cdot \Delta T_{\text{SC}_{\text{cond}},\text{H}_2\text{O}} \quad (7)$$

The combined total cooling requirement of the plant is therefore calculated:

$$\begin{aligned} \dot{q}_{\text{Total}} &= \dot{q}_{\text{SC}_{\text{cond}}} + \dot{q}_{\text{CCS}} \\ &= \dot{q}_{\text{SC}_{\text{cond}}} + \dot{q}_{\text{DCC}} + \dot{q}_{\text{LSC}} + \dot{q}_{\text{Condenser}} + \dot{q}_{\text{Compression}} \end{aligned} \quad (8)$$

Our model allows the user to set input parameters including the gross electrical output, $T_{\text{DCC},\text{in}}$, T_{Absorber} , $T_{\text{LSC},\text{in}}$, $T_{\text{Condenser}}$, p_{Desorber} and $p_{\text{Comp},\text{out}}$. The default operating conditions are listed in Table 2.

2.4. Cooling Tower and Retrofit Designs

There are three major cooling technologies generally used in power plants: Air cooling, once-through (or direct) cooling and wet-cooling. Water is the cooling medium in both once-through and wet cooling technologies; however, they differ significantly in water usage (withdrawal from natural sources *e.g.*, river) and water consumption (water losses, which are not returned to the original source due to evaporation). Once-through cooling is linked to a higher water usage compared to wet-cooling. However, it is associated with a smaller water consumption as it is designed to work with minimum evaporative losses [42]. Wet cooling, delivered via cooling towers, is the dominant choice for coal-fired power plants [31, 11]; hence, in this work, wet cooling technology is further investigated.

Natural draft cooling towers are gradually being phased out due to their lower social acceptance and efficiency in comparison to mechanical draft towers [43, 44]. There are two distinct types of mechanical draft cooling towers: forced and induced. In a forced mechanical draft cooling tower, air is blown through the packing; whereas in induced draft cooling towers, air is sucked through the towers with fans often placed at the top of the cooling module. In general, a lower electrical energy and a more uniform flow distribution throughout the tower are achieved by using induced draft cooling towers [42]. On the other hand, forced draft cooling towers can use cheaper blowers due to reduced humidity at the module's air inlet and exhibit superior drift (water entrainment) control. For this study a mechanical forced draft

Table 2: Default operating conditions of a 30 wt% MEA capture unit.

Parameter	Value
Gross electrical output	500 MW _g
$T_{\text{DCC,in}}$	53.5 °C
T_{Absorber}	40 °C
$T_{\text{LSC,in}}$	80 °C
$T_{\text{Condenser}}$	30 °C
$T_{\text{Desorber,bottom}}$	120 °C
$T_{\text{Desorber,head}}$	90 °C
p_{Absorber}	1 atm
p_{Desorber}	2 bar
$p_{\text{Comp,out}}$	11 MPa
ΔT_{Safety}	15 °C
$\Delta T_{\text{SC}_{\text{cond,H}_2\text{O}}}$	16.67 °C
$\Delta T_{\text{DCC,H}_2\text{O}}$	30 °C
$\Delta T_{\text{LSC,H}_2\text{O}}$	30 °C
$\Delta T_{\text{Condenser,H}_2\text{O}}$	30 °C
$\Delta T_{\text{Compression,H}_2\text{O}}$	30 °C
$\Delta T_{\text{Cascade,H}_2\text{O}}$	10 °C

cooling tower (see Table 1) has been chosen (with same configuration as described by Zhai et al. [30]).

The most important parameters in designing a cooling tower are approach (*i.e.*, the difference between the cold inlet air and the cold outlet water) and range (*i.e.*, the temperature difference between the inlet and outlet water). The maximum cooling tower efficiency is limited by the wet bulb temperature of the air owing to the fact that water can not be cooled to temperatures below the wet bulb temperature. Therefore, the design of a cooling tower is always tied to site-specific weather conditions [45]. The lowest water temperature in the entire plant dictates the approach value of the cooling tower. A design margin (ΔT_{Safety}) between the lowest temperature throughout the plant, usually corresponding to the stripper's condenser temperature, and the cooling water discharge temperature is often considered.

It is best to estimate the size of the cooling tower in early design phases, for which the cooling water flow rate has to be known. Given the cooling duty in an arbitrary process unit i , the cooling water mass flow rate can be calculated by an energy balance around the corresponding cooling water side (see Figure 5):

$$\dot{m}_{i,\text{H}_2\text{O}} = \frac{\dot{q}_i}{c_{p,\text{H}_2\text{O}} \cdot \Delta T_{i,\text{H}_2\text{O}}}. \quad (9)$$

In our model, the cooling water temperature rise, $\Delta T_{i,\text{H}_2\text{O}}$, can be specified by the user for each unit within a set interval imposed by the pinch point at the hot end of the heat exchanger. With an increase in $\Delta T_{i,\text{H}_2\text{O}}$, the associated

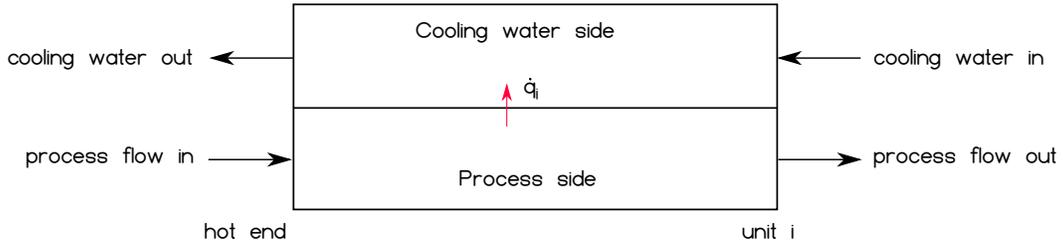


Figure 5: A schematic of a conceptual process unit i

cooling water mass flow rate ($\dot{m}_{i,\text{H}_2\text{O}}$) is reduced. The combined total cooling water flow rate is:

$$\dot{m}_{\text{Total,H}_2\text{O}} = \sum_i \dot{m}_{i,\text{H}_2\text{O}} \quad (10)$$

Hill [42] presents eq. 11 which approximates the base plant area (A_{CT}) of a mechanical forced draft cooling tower:

$$A_{\text{CT}} = \frac{\dot{m}_{\text{Total,H}_2\text{O}}}{\text{water holdup}} \cdot F, \quad (11)$$

The water holdup (waterloading $2.3 \text{ kg s}^{-1} \text{ m}^{-2}$) depends on the packing type *e.g.* splash fill. The factor $F = 0.57$ can be read from related design plots (see Figure A.3.5.1 in [42]). In order to integrate the cooling tower base area into our model, an iteration led to a regression based on the cooling water mass flow rate. Equation 11 does not directly account for the cooling water inlet temperature; however, the range value affects the cooling tower sizing through F . Therefore, the iteration was done by taking into account the most common operating conditions and is only valid for a certain range of values. The base plant areas calculated using this method were compared against the literature data as well as the designing tools provided by manufacturers. Our results were observed to be within 5% of the reported values.

One of the potential concerns of retrofitting CO_2 capture to an existing power plant is whether any additional cooling demand arising from the capture process can be accommodated within the existing cooling tower, or if additional cooling capacity will be required. As shown in Figure 2, a portion of the generated steam is bled from the steam cycle in order to meet the stripper's energy demand. This results in a reduction in total steam flow rate into the condenser and therefore a reduction in the steam cycle condenser's cooling duty. In other words, this means that there exists a trade off between the increased cooling duty associated with the CO_2 capture process and the reduction in cooling duty of the steam cycle condenser. It is assumed that the coal-fired plant operates with an existing cooling tower and an additional capture unit has been retrofitted. Both cases, *i.e.*, with and without CCS, have been analysed as presented and a balance value δ for the total cooling duty in addition to total cooling water mass- and volume flow has been calculated:

$$\delta_{\text{Cooling}} = \dot{q}_{\text{Total,H}_2\text{O}}^{w/o-CCS} - \dot{q}_{\text{Total,H}_2\text{O}}^{w-CCS} \quad (12)$$

$$\delta_{\text{Mass}} = \dot{m}_{\text{Total,H}_2\text{O}}^{w/o-CCS} - \dot{m}_{\text{Total,H}_2\text{O}}^{w-CCS} \quad (13)$$

$$\delta_{\text{Volume}} = \dot{V}_{\text{Total,H}_2\text{O}}^{w/o-CCS} - \dot{V}_{\text{Total,H}_2\text{O}}^{w-CCS} \quad (14)$$

2.5. Cooling Cascade

Energy integration puts the emphasis on reducing the energy penalty by re-using the waste energy throughout the process [46]. We have evaluated the option of a cooling cascade between the capture plant and the steam cycle condenser.

Instead of returning the used cooling water from the capture plant to the cooling tower, it is first used to partially condense

Table 3: Net plant efficiencies (HHV) as reported by IECM with and without CCS for different steam cycle.

Net plant efficiency (HHV (%))	Subcritical	Supercritical	Ultra-Supercritical
Without CCS	36.4	38.7	42.7
With CCS (30 wt% MEA)	22.0	24.8	28.8

and super-critical is not large, the relative difference between their associated cooling water flow rates (\dot{m}_{SC_{cond},H_2O}) is considerable: 29 % (with capture unit) and 20 % (without capture unit). These results are in good agreement with the earlier reported values [30, 37].

Table 4: IECM output with CCS for different types of steam cycle

Parameter	With CCS			Without CCS		
	Subcritical	Supercritical	Ultra-Supercritical	Subcritical	Supercritical	Ultra-Supercritical
Net electrical output (MW/MW _g)	0.743	0.767	0.791	0.930	0.935	0.941
$\dot{m}_{Flue\ gas}$ (kg/(s · MW _g))	1.526	1.408	1.259	1.083	1.023	0.932
$\dot{m}_{SC_{cond},Steam}$ (kg/(s · MW _g))	1.069	1.029	0.751	0.809	0.805	0.602
\dot{m}_{SC_{cond},H_2O} (kg/(s · MW _g))	13.71	9.68	6.99	18.64	16.80	14.02
$\dot{m}_{Solvent}$ (kg/(s · MW _g))	3.874	3.572	3.195	-	-	-
\dot{m}_{CO_2} (kg/(s · MW _g))	0.252	0.233	0.208	-	-	-

3.2. Cooling Duties Associated with the Carbon Capture Unit

The normalised cooling duties of four power plants are plotted for both cases (*i.e.*, without carbon capture (*a*) and with carbon capture (*b*)) in Figure 7. A value of 1 MW/MW_g means that for one megawatt of power, a cooling duty of one megawatt is required. Bearing in mind the thermal efficiency of power plants, values in the range of 1-2 MW are unsurprising. The cooling duty associated with the steam cycle condenser (green) is observed to decrease in proportion to an improvement in steam cycle efficiency, as shown in Table 4. The reduction in \dot{m}_{SC_{cond},H_2O} with CCS relative to that without CCS is $\approx 26\%$ (subcritical), $\approx 42\%$ (supercritical) and $\approx 50\%$ (ultra-supercritical). This is caused by the reduction in the steam flow rate in the SC. The flow rate of the extracted steam used to supply the heating duty to the stripper’s reboiler also changes but not to the same extent as $\dot{m}_{SC_{cond},Steam}$. Nevertheless, despite the reduction in steam cycle cooling duty, the total cooling duty of the integrated power-capture process increases with the addition of the capture unit, with the relative increase being greatest for the sub-critical (least efficient) power plant.

The respective share of the cooling duties associated with a carbon capture unit and the steam cycle condenser is presented in Figure 8. It is observed that the second largest cooling demand is associated with the lean solvent cooler (blue) in the capture unit and for the ultra-supercritical cycle, it is nearly as big as the SC condenser’s cooling duty. This is due to the substantial solvent mass flow rate and a relatively large ΔT_{LSC} . The third largest cooling duty is associated with the condenser.

The pre-cooler’s cooling duty contributes to $\approx 1\%$ of the total cooling demand. As it can be seen in eq. 1, the relatively small mass flow rate and heat capacity of the flue gas lead to a small cooling duty. The same is true for the compression

intercooling. The temperature after one isentropic compression stage is $\approx 490^\circ\text{C}$ which leads to a much higher temperature difference compared to the pre-cooler and therefore, the compression intercooling has the fourth largest share.

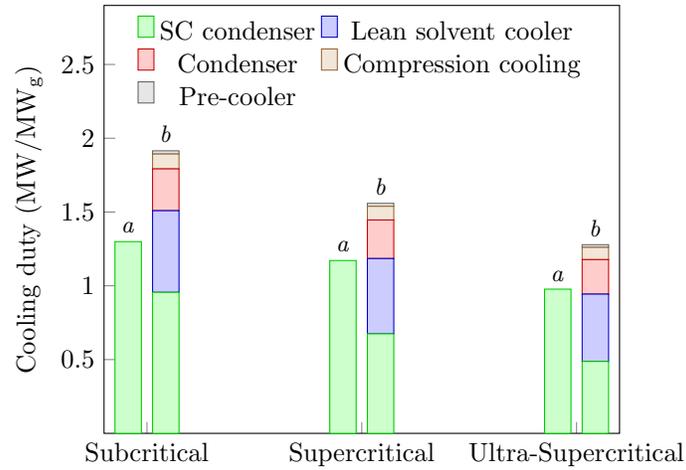


Figure 7: The breakdown of various cooling duties associated with a carbon capture unit retrofitted to a pulverized coal-fired power plant (a) without CCS and (b) with CCS, (summer weather conditions).

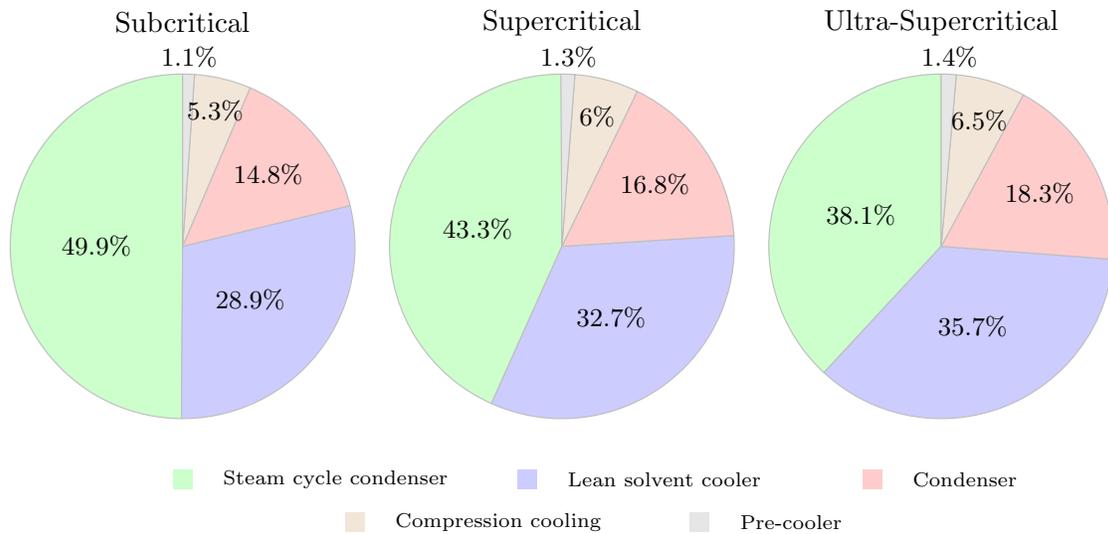


Figure 8: Relative share of various cooling duties associated with a carbon capture unit retrofitted to a pulverized coal-fired power plant with CCS, (summer weather conditions).

3.3. Cooling Tower and Retrofit Design

The normalised cooling water mass flow rates for the various power plants with and without CCS are shown in Figure 9. It can be observed that with an increase in steam cycle efficiency, the required mass flow of cooling water is reduced. Interestingly, the addition of CCS to the power plant actually *reduces* the quantity of cooling water required for the super- and ultra-supercritical power plants; only for the subcritical steam cycle the water flow rate with carbon capture is higher than that without the capture unit.

The total cooling duty is not directly proportional to the total cooling water mass flow. Instead, each unit's cooling duty, \dot{q}_i , is calculated taking into account its own temperature increase $\Delta T_{i,\text{H}_2\text{O}}$ (see eq. 9). These cooling duties are then

summed up to give the total cooling duty. This is justified by the different temperature levels: a condensation process is ideally operated at constant temperature limiting the temperature range of cooling water, while the lean solvent cooler can be operated at a much wider temperature range for the cooling water side. With an increase in $\Delta T_{i,\text{H}_2\text{O}}$, $\dot{m}_{i,\text{H}_2\text{O}}$ is reduced. Similarly, a smaller temperature difference (*i.e.*, a smaller driving force for heat transfer) from the process to the cooling side (see Figure 5) translates into a larger heat transfer area and therefore, a larger heat exchanger. Therefore, there exists a trade off between CAPEX (heat transfer area) and OPEX ($\dot{m}_{i,\text{H}_2\text{O}}$) with $\Delta T_{i,\text{H}_2\text{O}}$ as an additional design variable.

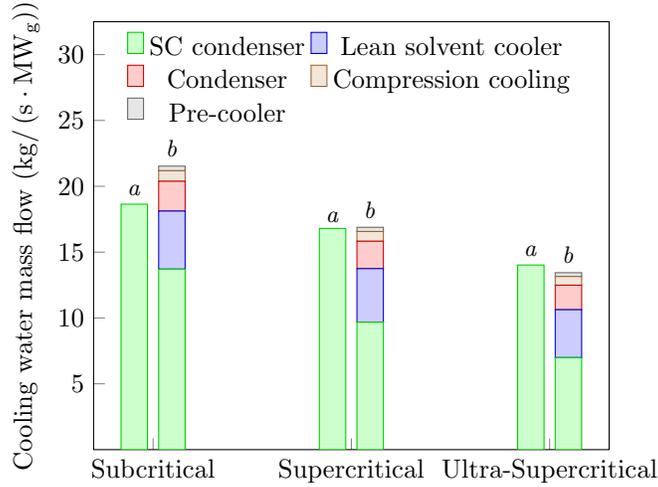


Figure 9: Impact of steam cycle (SC) types on the cooling water mass flows for a pulverized coal-fired power plant (a) without CCS and (b) with CCS

210 These results are summarised in Table 5. The cooling tower plot area decreases with increasing SC efficiency suggesting a potential reduction in CAPEX. These results are consistent with reported values in the literature [11].

Table 5: Normalised requirements for a mechanical forced draft cooling tower system for different types of steam cycle with capture unit.

Parameter	Subcritical	Supercritical	Ultra-Supercritical
\dot{m}_{Total} (kg/(s · MW _g))	21.5	16.9	13.4
\dot{m}_{Total} (kg/kg _{Flue gas})	14.1	12.0	10.7
\dot{V}_{Total} (l/(s · MW _g))	21.7	17.0	13.6
\dot{V}_{Total} (l/kg _{Flue gas})	14.2	12.1	10.8
A_{CT} (m ² /MW _g)	7.28	5.71	4.55
A_{CT} (m ² /(kg/s) _{Flue gas})	4.77	4.06	3.61

The main impacts of an additional capture unit are described in Figure 7 and Figure 9. The balance value, δ , first introduced in eq. 12, enables a quantitative analysis between with- and without-capture unit scenarios. For example, for a 500MW power station δ is calculated by eq. 12 as $\delta_{\text{Volume}} \approx 871\text{s}^{-1}$. That means \dot{V}_{Total} is $\approx 871\text{s}^{-1}$ higher for a plant with a capture unit compared to a plant without one. This can be accommodated in an existing cooling tower [42, 47], implying that an existing cooling system could be used, therefore significantly reducing the cost of retrofitting the capture unit.

3.4. Selection of the Cooling Technology

In order to select the most viable cooling technology, both CAPEX and OPEX must be simultaneously considered, explicitly accounting for the plant's location. The CAPEX is closely linked to the space considerations and to the cooling equipment (*e.g.*, heat exchangers, fans, *etc.*). In terms of OPEX, an air-cooled system might need more space and electricity compared to a wet-cooled system; however, it is advantageous in that it does not require any cooling water with its pre- and post-treatment units [44, 42]. Site-specific considerations are local constraints such as water or land scarcity. If a wet cooling system is the most economical option, regardless of potential for future issues of water stress, it is likely to be used [11, 43].

Due to a large cooling duty, an air-cooled system is often not the most viable, although it can be relatively cost effective in comparison [48, 31]. Air-cooled systems are also considered to be more flexible than cooling towers (which are typically designed for a specific cooling load) [42, 49]. This is an important factor with regard to the desired [20, 50] flexible operation of power stations due to fluctuations in electricity demand. Hybrid cooling towers combine wet and dry cooling and thus, significantly reduce the water consumption in comparison to the regular cooling towers. Modular hybrid cooling towers are considered a preferred option for high cooling demands [48]. However, the most feasible cooling technology is closely linked to the plant's location.

3.5. Cooling Cascade

The total cooling water demand after integrating a cooling cascade (*c*) is plotted in Figure 10. It can be observed that the cascade configuration acts to reduce the cooling water flow rate associated with the steam cycle condenser. There is a relative reduction in cooling water flow rate by 21.7% (subcritical) up to 28.6% (ultra-supercritical) compared to the original configuration with no cascade (*b*). The impact of the reduced cooling water mass flow rate outweighs the higher cooling tower inlet temperature. Consequently, the cooling tower plot area decreases by 11.0% for sub- and 18.4% for ultra-supercritical steam cycles (see Table 6). Therefore, a cooling cascade demands less water and also a smaller cooling tower.

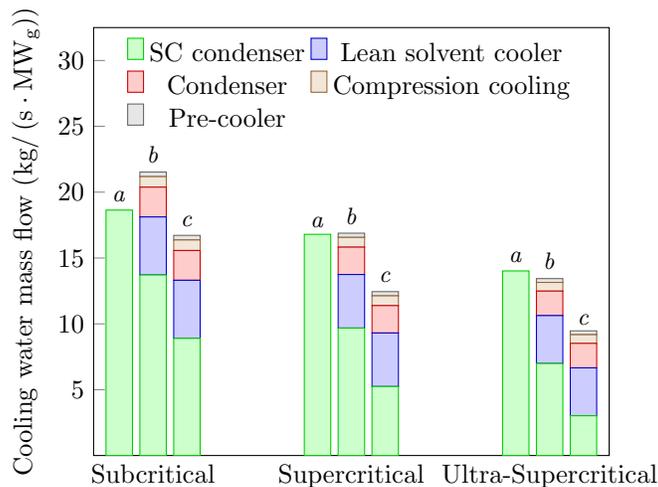


Figure 10: Effect of steam cycle (SC) type on cooling water flow rate for a pulverized coal-fired power plant (*a*) without CCS, (*b*) with CCS and (*c*) with CCS with an integrated cooling cascade

Table 6: Requirements and impact of cooling cascade on cooling system.

Parameter	Subcritical	Supercritical	Ultra-Supercritical
$\dot{m}_{\text{Total, cascade}}$ (kg/(s · MW _g))	16.8	12.6	9.6
$\dot{m}_{\text{Total, cascade}}$ (kg/kg _{Flue gas})	11.0	8.9	7.6
$\dot{V}_{\text{Total, cascade}}$ (l/(s · MW _g))	17.0	12.7	9.7
$\dot{V}_{\text{Total, cascade}}$ (l/kg _{Flue gas})	11.2	9.0	7.7
$A_{\text{CT, cascade}}$ (m ² /MW _g)	6.48	4.85	3.71
$A_{\text{CT, cascade}}$ (m ² /(kg/s) _{Flue gas})	4.24	3.44	2.95

In order to avoid fouling within the cooling tower, the maximum cooling water inlet temperature is limited to 50 °C [42, 51]. A typical pressure in a steam cycle condenser is 0.05 bar, with a dew point of ≈ 33 °C. The mixed cooling water coming from the capture unit is already at ≈ 44 °C and therefore, if the cooling cascade configuration is to be used, the pressure in the steam cycle condenser has to be increased to approximately 0.2 bar to 0.25 bar. This will significantly reduce the electricity generated by the LP turbine. The disadvantages (higher OPEX and less electricity generated) may well outweigh the advantages (lower CAPEX and smaller water demand), making the proposed cooling cascade an unfavourable option, except in scenarios of extreme water scarcity.

4. Conclusions and Recommendations

This study presents a detailed breakdown of the required cooling duty resulted from the addition of a carbon capture unit to a coal-fired power plant complemented with an estimation of the corresponding space requirement. A key conclusion of this current study is that the integration of CO₂ capture with a coal-fired power station will not lead to significant increases in water utilisation, unless the base power plant is itself very inefficient, *e.g.*, a sub-critical power plant. For modern, state-of-the-art power plants, water use for cooling is likely to remain approximately constant or potentially be reduced. Key to this is the understanding that the addition of CO₂ capture to a power plant will remove a significant fraction of the steam from the power plant to the solvent regeneration process. This will reduce the cooling duty of the steam cycle, and displace this to the capture plant. When the steam is condensed in the capture plant reboiler, this generates a mixture of CO₂ and stripping steam which then travels up the desorption column to the overhead condenser. Along the way, the stripping steam is partially condensed, and therefore the cooling duty of the condenser will be less than the energy required to generate the stripping steam in the first place. Similarly, the heat required to recover the CO₂ from the solvent will not be removed in the condenser, further reducing the cooling duty. These phenomena outweigh the additional cooling duties associated with the lean solvent cooler, compression intercooling and the exhaust gas direct contact cooler.

It was found that the cooling duty of the pre-cooler is small, supporting the concept of replacing the DCC with intercoolers in the absorber as suggested in other contributions [52, 53]. It was also shown that the proposed cooling cascade (Figure 10) reduces the total water usage, despite the increase in cooling duty. Alternative cascade configurations, such as using the cooling water coming from the steam cycle condenser in the lean solvent cooler, promise significant reductions in water demand without adjusting the steam turbine’s discharge pressure. The suggestion that adding a capture unit nearly doubles the water demand has been shown to be worth reconsidering.

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