METHODOLOGY FOR ESTIMATING GENERATING CAPACITY LOSSES IN THERMOELECTRIC FACILITIES WITH RECIRCULATING COOLING SYSTEMS UNDER CLIMATE CHANGE

Updated Supplementary Information for:


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LETTERS

Impacts of climate change on electric power supply in the Western United States

Impacts of Climate Change on Electric Power Supply in the Western U.S.
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This report updates Supplementary Information section 2.1.2.2 (Recirculating Cooling) of Bartos and Chester (2015). Extraneous derivations have been removed and an error corrected.

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2.1.2.2 - Power Output of Recirculating Cooling Facilities

Recirculating cooling systems reject heat by evaporating water, rather than discharging it directly into a nearby water body[1,2,3]. Water that is not evaporated during the cooling process is re-used, meaning that much less water is withdrawn overall compared to open-loop cooling. For these facilities, cooling water requirements are determined primarily by atmospheric conditions—such as air temperature and humidity—while the intake temperature of water plays a smaller role[1]. Water withdrawals are required to make up for three types of losses in the cooling system: evaporation losses, blowdown losses, and drift losses.

$$\dot{W}_{mu} = \dot{W}_{evap} + \dot{W}_{bd} + \dot{W}_{d} \tag{1}$$

Where $\dot{W}_{mu}$ represents the makeup water requirement (m^3/s), $\dot{W}_{evap}$ represents evaporation losses (m^3/s), $\dot{W}_{bd}$ represents blowdown losses (m^3/s), and $\dot{W}_{d}$ represents drift losses (m^3/s). Evaporation losses comprise the majority of makeup water requirements. These losses are mainly a function of the heat load into the condenser[1]:

$$\dot{W}_{evap} = \frac{\dot{Q}_{load}(1 - k_{sens})}{\rho_w h_{fg}} \tag{2}$$

Where $\dot{Q}_{load}$ is the heat load to the condenser (MJ/s). The quantity k_{sens} represents the fraction of the heat load that is rejected through sensible heat transfer—that is, heat transfer from the liquid water to the air, and not from evaporation. The fraction of heat rejected through sensible heat transfer is mainly a function of the temperature of the incoming air, although it also depends on humidity and ambient air pressure[1]. The quantity $1 - k_{sens}$ represents the fraction of heat load rejected through latent heat transfer—that is, heat transfer involved in the phase change of water from a liquid to a gas (evaporation). The parameter h_{fg} represents the latent heat of vaporization of water, which is assumed to be constant at 2.45 MJ/kg[1]. The quantity ρ_w represents the density of water (1000 kg/m^3).

Blowdown losses consist of small releases of water required to prevent the buildup of contaminants (namely chlorides) in the cooling system. These losses can be expressed in terms of the evaporation rate[2]:

$$\dot{W}_{bd} = \frac{\dot{W}_{evap}}{n_{cc} - 1} \tag{3}$$

Where n_{cc} represents cycles of concentration—a measurement of the relative concentration of chlorides in the recirculating water:

$$n_{cc} = \frac{[C]_{rc}}{[C]_{mu}} \tag{4}$$

Here, the parameter $[C]_{rc}$ represents the chloride concentration in the circulating water, while $[C]_{mu}$ represents the concentration of chlorides in the makeup water. The number of cycles of concentration ranges from 3 to 6, with a typical value of 6[2]:

Drift losses consist of sprays of liquid water that escape the cooling tower. Given that drift losses are typically less than 0.005% of circulation flow, they can be considered negligible[2]. Thus, the amount of makeup water required is equal to evaporation losses plus blowdown losses:

$$\dot{W}_{mu} \approx \dot{W}_{evap} + \dot{W}_{bd} \tag{5}$$

Having developed the water inputs to the cooling tower, the heat load to the condenser can be determined by performing an energy balance around the cooling tower. Energy inputs to the tower include the heat load from the condenser, the makeup water, and the stream of air used to cool the hot process water. Outputs from the tower include the hot, humid air stream leaving the tower, and blowdown losses. Altogether, the energy balance can be expressed as follows:

$$\dot{Q}_{load} + \rho_w \dot{W}_{mu} h_{mu} = \rho_a \dot{G}\left(h_{a,out} - h_{a,in}\right) + \rho_w \dot{W}_{bd} h_{bd} \tag{6}$$

Where $\dot{W}_{mu}$ and h_{mu} are the mass flow rate (kg/s) and enthalpy (MJ/kg) of the makeup water entering the cooling tower, respectively; ρ_a is the density of air (kg/m^3); $\dot{G}$ is the dry air mass flow rate of cool air entering the tower (m^3/s); $h_{a,out}$ and $h_{a,in}$ are the enthalpies (MJ/kg) of the hot air exiting the tower and cool air entering the tower, respectively; and $\dot{W}_{bd}$and h_{bd} are the mass flow rate (kg/s) and enthalpy (MJ/kg) of the blowdown water. To solve the energy balance around the tower, a mass balance must also be constructed. Performing a mass balance around the cooling tower yields the following expression:

$$\rho_w \dot{W}_{mu} = \rho_w \dot{W}_{evap} + \rho_w \dot{W}_{bd} = \rho_a \dot{G}(\omega_{out} - \omega_{in}) + \rho_w \dot{W}_{bd} \tag{7}$$

Where ω_{out} is the humidity ratio of air exiting the tower, and ω_{in} is the humidity ratio entering the tower. Substituting blowdown requirements with the expression developed in Equation 5 yields makeup water requirements as a function of the evaporation rate:

$$\dot{W}_{mu} = \dot{W}_{evap} + \frac{\dot{W}_{evap}}{n_{cc} - 1} = \frac{n_{cc}\dot{W}_{evap}}{n_{cc} - 1} \tag{8}$$

The energy and water balances developed in the preceding equations can be solved by generating an expression for the dry air mass flow rate. Because data on dry air mass flow rate is rarely available, the water-air mass flow ratio is specified instead:

$$\sigma = \frac{\rho_w \dot{W}_{circ}}{\rho_a \dot{G}} \tag{9}$$

$$\rho_a \dot{G} = \frac{\rho_w \dot{W}_{circ}}{\sigma} \tag{10}$$

Where σ is the water-air mass flow ratio, $\dot{W}_{circ}$ is the flow rate of water circulating through the condenser (m^3/s), and $\dot{G}$ is the dry air flow rate (m^3/s). The value of σ ranges between 0.5 and 1.5 with a typical value of 0.8[1].

The heat load into the condenser can now be expressed in terms of the circulating water rate and the enthalpy difference through the cooling system:

$$\dot{Q}_{load} = \frac{\rho_w \dot{W}_{circ}}{\sigma}\left[h_{a,out} - h_{a,in}\right] = c_{p.w} \rho_w \dot{W}_{circ}\, \Delta T \tag{11}$$

Where ΔT is the temperature difference across the condenser and $c_{p.w}$ is the heat capacity of water (MJ/kg-K). Recognizing that the available electricity generating capacity is a function of the heat load and the plant efficiency, Equation 11 can be rearranged to yield the generating capacity as a function of water circulation rate and meteorological parameters:

$$\boxed{P_{rc} = \frac{c_{p.w} \rho_w \dot{W}_{circ}\, \Delta T}{\frac{(1 - \eta_{net,i} - k_{os})}{\eta_{net,i}}}} \tag{12}$$

Where $\eta_{net,i}$ is the net plant efficiency for a given month, and k_{os} is the fraction of heat lost to sinks other than the condenser (such as flue stack losses). To account for constraints on water availability, an expression is developed to relate makeup water requirements to the total rate of water recirculating through the system. Equations 7-10 can be rearranged to yield the relationship between makeup water flow and circulating water flow for a given humidity differential:

$$\boxed{\frac{\min(\dot{W}_{mu}, \gamma Q_i)}{\dot{W}_{circ}} = \frac{(\omega_{out} - \omega_{in})}{\sigma} \cdot \left(1 + \frac{1}{n_{cc} + 1}\right)} \tag{13}$$

Thus, when the makeup water requirement ($\dot{W}_{mu}$) is greater than available streamflow (γQ_i), the volume of water passing through the condenser ($\dot{W}_{circ}$) decreases such that the ratio of makeup water to recirculating water remains the same for a given set of humidities. Accordingly, as $\dot{W}_{circ}$ decreases, the heat load (and thus, the available capacity) decreases in turn, as per Equation 13. For each facility, $\dot{W}_{circ}$ at 100% load is taken from EIA Form 860 data[4]. The temperature range is calculated using reported values from EIA Form 860 and 923[4,5]. The fraction of streamflow available for withdrawal is determined using the method of Tennant[6]. For the average case, this fraction is taken to be 0.7 for summer months and 0.9 for winter months.

Atmospheric parameters related to humidity can be calculated using VIC full-energy outputs, using the empirical methods of Kimball et al. and Thornton et al.[7,8]. The primary parameters of interest include the humidity ratios of incoming and outgoing air (ω_{in} and ω_{out}). These parameters are derived from the following full-energy outputs from the VIC model: dry-bulb air temperature (T_d), total ambient pressure (P_{tot}), vapor pressure (P_w), and relative humidity (RH).

To calculate ω_{in} and ω_{out}, the saturated vapor pressure for each time step must first be determined. Saturated vapor pressure is calculated using relative humidity and vapor pressure at each time step, using the definition of relative humidity[9]:

$$RH = \frac{P_w}{P_{ws}} \cdot 100\% \tag{14}$$

$$P_{ws} = \frac{P_w}{RH} \cdot 100\% \tag{15}$$

Where RH is the relative humidity, P_w is the vapor pressure and P_{ws} is the saturated vapor pressure. Relative humidity and vapor pressure are provided from VIC full-energy outputs. Humidity ratios (sometimes called mixing ratios) can be calculated through the following formulae, assuming that the outgoing air is fully saturated[9]:

$$\omega_{in} = \frac{B \cdot P_w}{P_{tot} - P_w} \tag{16}$$

$$\omega_{out} = \frac{B \cdot P_{ws}}{P_{tot} - P_{ws}} \tag{17}$$

$$B = \frac{M(H_2O)}{M(Air)} \cdot 1000 = 621.9907\ g/kg \tag{18}$$

Where $M(H_2O)$ represents the molecular weight of water, and $M(Air)$ represents the molecular weight of air. Outputs from Equations 16 and 17 are combined with Equations 12 and 13 to calculate power output from vulnerable facilities employing recirculating cooling.

For steam turbine facilities employing recirculating cooling, we do not consider discharge temperature regulations as a constraint on generating capacity. First, for recirculating cooling facilities, it is difficult to determine how blowdown is discharged. Blowdown can be discharged into a holding pond, or into a nearby stream[1]. If blowdown is discharged into a holding pond, it is considered to be a consumptive use of water, and may not be subject to environmental regulations regarding maximum discharge temperatures[1]. Currently, EIA form 860 does not contain sufficient data to accurately assess how blowdown is disposed. Second, many of the facilities listed in EIA form 923 report discharge temperatures much higher than the 32 ℃ maximum required by most states (often 10 to 20 ℃ higher). Thus, it is unclear whether reported maximum discharge temperatures refer to the temperature of the blowdown water, or to the temperature of the hot water exiting the generator. Finally, because the cooling water is cooled through latent heat transfer, the temperature of the blowdown water will generally only be 4-8 °C hotter than the ambient wet-bulb air temperature, which is lower than the ambient air temperature for unsaturated conditions. Given the relatively small amount of blowdown released by the cooling system, and the relatively low temperature of the blowdown, it is unclear whether blowdown discharge temperatures warrant reductions in plant capacity for many of the plants included in this study. For these reasons, we do not include blowdown discharge temperatures as a constraint on generating capacity for recirculating cooling facilities.

2.1.2.2.1 - Selecting Vulnerable Recirculating-Cooling Steam-Turbine Facilities

Steam turbine facilities utilizing recirculating cooling are selected in two steps. First, the following prime mover types are selected from EIA forms 860 and 923: (a) steam turbine, (b) combined cycle steam turbine, and (c) binary cycle turbine. These prime movers are shown in Table 1, along with their EIA/eGRID field codes:

Table 1. Prime movers for steam turbine facilities utilizing recirculating cooling systems

Field Code	Prime Mover Type
ST	Steam turbine
CA	Combined cycle steam turbine
BT	Binary cycle turbine

From this set of steam-condensing plants, plants relying primarily on recirculating cooling are selected. A plant is considered to use recirculating cooling if its primary cooling system type is one of the technologies shown in Table 2:

Table 2. Recirculating cooling systems for steam turbine facilities

Field Code	Cooling System Type
RC	Recirculating with cooling pond(s) or canal(s)
RF	Recirculating with forced draft cooling tower(s)
RI	Recirculating with induced draft cooling tower(s)
RN	Recirculating with natural draft cooling tower(s)

Next, vulnerable facilities are isolated by determining the water source of the cooling system. Cooling systems relying on surface water are considered to be vulnerable, while cooling systems relying on groundwater, ocean water or municipal water are not considered to be vulnerable. Water sources of power facilities are identified using data from the Union of Concerned Scientists' EW3 database[10]. Facilities are considered "at-risk" if the reported water source is "Surface Water", "Unknown Freshwater", or "GW/Surface Water".

Many plants contain a steam turbine generator in addition to another prime mover. To account for the partial contribution of steam turbines to some plants, EIA forms 860 and 923 are used to determine the capacity contributed by each generator at each plant. First, the total nameplate capacity of each plant in the WECC region is determined; next the capacity contributed by steam turbine generators (field codes

ST, CA, BT) is determined for the same set of plants. Dividing the steam turbine contribution by the total capacity yields the fraction of capacity contributed by steam turbine generators to each plant that utilizes recirculating cooling.

2.1.2.2.2 - References

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