

## Appendix

### 1. Power Plant Data

TP2M as described in the main text and detailed in this section requires power plant engineering design variables as inputs. This information, for the case of the U.S., can be obtained from readily available public sources. The Energy Information Agency (EIA) has downloadable datasets in which the data can be used to calculate an estimate of thermal efficiencies. It also contains information on fuel type and the capacity of the power plants.<sup>7, 13</sup> Recently, there have been other groups that have thoroughly researched typical withdrawal and consumption rates for thermoelectric plants (National Renewable Energy Laboratory) and more accurate locations of these plants and identifying their cooling technologies (Union of Concerned Scientists).<sup>4,14</sup> The combination of this information with standard thermodynamic equations can provide some currently unavailable variables, such as the temperature increase of the withdrawn cooling water after it passes through the condenser. Thus, there is currently sufficient literature and datasets that provide the necessary variables (within logical and reasonable ranges) required for this type of detailed dynamic modeling.

### 2. Model Description

TP2M computations were organized as a sequence of sub-models, the specific pathways of which determined by a series of 'checks' that are described here including equations and key assumptions (see Figure 2 in main text).

#### 2.1. Cooling Technology Check

First is the cooling technology check, which determines whether the power plant needs to use inputs and equations that relate to once-through or recirculating methods. The explanations of the checks and computations that follow are described below, separated into five sections based on technology type: once-through, cooling tower, dry-cooling, combined-cycle technologies and hybrid.

#### 2.2. Once-through: Inlet Temperature Check

Once the cooling technology is identified, the model checks the temperature of the cooling water at the inlet. In the case of the once-through cooling technology, this is the temperature of the river at intake. For the cooling tower it is defined as the temperature of the water in the cooling tower. Inlet temperatures of cooling water above optimal conditions raise the condensing temperature and pressure of the steam coming from the turbine, which results in a decrease in thermal efficiency.<sup>5</sup> Power plants may vary their intake water withdrawal rates to optimize the condensing pressure.<sup>40</sup> However, past a certain threshold inlet temperature, which is specific to each plant, the condensing temperature and pressure will increase. Because this specific information is not typically published, we assume that the ideal condensing temperature for power plants is a minimum of 30°C and that this is forced to increase once inlet

temperatures rise above a threshold limit -  $T_A$  °C. For this paper  $T_A$  °C is assumed to be 22°C, considered to be a conservative approach that assumes higher efficiency than typical. This threshold reflects studies that show that the electric output decreases from optimal when the inlet temperature is above ~10°C.<sup>19, 41</sup> (This threshold limit can change for testing different scenarios. The writers appreciate that this assumption may vary greatly between power plants, but a generalized assumption is necessary until more plant specifics become public information. This is an issue that may need to be addressed – basic details of plant performance should become publicly available – the same way solar panel manufacturers publicize the performance of their products.)

It has been stated that for an increase in 1kPa of condensing pressure there is a corresponding 1%-1.5% decrease in efficiency (cited in Laković et al. 2010).<sup>41</sup> Therefore, taking the saturated vapor temperature (state of steam after turbine) range of 30°C-70°C and the corresponding increase in pressure from standard property tables,<sup>12</sup> a general relationship between condensing temperature and decrease in thermal efficiency can be calculated (Figure 3a in main text).

Assuming a linear relationship between inlet temperature and condensing temperature,<sup>40</sup> using the gradient of Figure 3a and choosing an appropriate  $T_A$ , the decrease in thermal efficiency can be estimated as a function of the increase of inlet temperature  $T_{ri}$  above  $T_A$  (Figure 3b; also shown in equation (2) below).

The operating thermal efficiency ( $\eta_{th}$ ) can be calculated as the optimal efficiency ( $\eta_{opt}$ ) less the loss of efficiency due to above-optimal inlet temperature ( $\eta_{loss}$ ):

$$\eta_{th} = \eta_{opt} - \eta_{loss} \quad (1)$$

$$\eta_{loss} = \chi_1(T_{ri} - T_A)^2 + \chi_2(T_{ri} - T_A) \quad (2)$$

Where  $\chi_1, \chi_2$  are constants with units  $K^{-2}$  and  $K^{-1}$ , respectively. When  $T_{ri} - T_A \leq 0$  there is no loss in efficiency. Once  $\eta_{th}$  is calculated, the waste heat rejected through the condenser,  $Q$  (MW), can be calculated using the input heat load  $Q_{in}$  (MW) and the fraction of waste heat released to heat sinks other than the condenser ( $q_{hs}$ ), which is assumed to be constant:<sup>33</sup>

$$Q = (1 - \eta_{th} - q_{hs})Q_{in} \quad (3)$$

For the case of a natural gas fired power plant with a combined cycle, the impact of warm water on the thermal efficiency is attributed to the steam part of the cycle only.<sup>8</sup>

### 2.3. Once-through: Water Availability Check

The next step is to determine the influent withdrawal rate required based on the heat load:

$$\dot{m} = \frac{Q}{c_p \Delta T} \quad (4)$$

Here,  $\dot{m}$  ( $\text{m}^3\text{s}^{-1}$ ) is the withdrawal rate of cooling water that moves through the condenser,  $c_p$  is the specific heat of water ( $\text{MJm}^{-3}\text{K}^{-1}$ ), and  $\Delta T$  (K) is the difference between inlet and outlet (where the water is discharged back to the river) temperature of the withdrawn water,  $T_o - T_{ri}$ . (Assumed constant for this step. The writers appreciate that  $\Delta T$  may not be constant. However, the model uses heat exchanges between the power plant and its surroundings to calculate thermal pollution and changes in efficiency. Therefore, although it is important to capture the behavior of  $\Delta T$  and  $\dot{m}$ , the waste heat load would still be the same.). As mentioned in the power plant data section, the optimal  $\Delta T$  of power plants is not readily available variable. However, by combining equations (3) and (4) with variables at their optimal state (i.e. unconstrained), a unique value of  $\Delta T$  can be obtained for each power plant.

Reviewing equations (1)-(4), if  $\eta_{loss} > 0$ , more water is needed to cool the system than during typical operations.

Next, a water check is conducted to assess if there is enough water available in the river to satisfy the water requirement set in the temperature check. There are two possible conditions:

$$\dot{m} < M_{ri} - M_{min} \quad (5)$$

$$\dot{m} \geq M_{ri} - M_{min} \quad (6)$$

Here,  $M_{min}$  ( $\text{m}^3\text{s}^{-1}$ ), is the minimum amount of flow a river must have at all times, which can be determined by the user and  $M_{ri}$  is the flow of the river ( $\text{m}^3\text{s}^{-1}$ ). The first possibility implies that there is enough water for the power plant and the process can continue to the next stage. The second possibility occurs when the calculated water allowance is not sufficient to satisfy the requirement set in the temperature check. This may happen in the summer during a drought or in winter when the water is frozen. In this case, the power plant is forced to decrease its withdrawal rate and it is assumed that the withdrawal is equal to the maximum allowance, which is  $M_{ri} - M_{min}$ . A reduction in withdrawal does not necessarily reduce output. Instead, the plant can compensate for a reduction in water by increasing  $\Delta T$ , as seen in equation (4). However, there are physical limits taken into account in the model for  $\Delta T$  that are based on typical plant operations.<sup>49</sup>

#### 2.4. Once-through: Regulatory (CWA) Check

The calculations completed up to this point establish the physically-allowable waste heat loads. However, the model can consider other constraints such as the CWA. If power plants do not to comply with regulations the only constraint on operations are the physical constraints of high temperatures and

insufficient river flow. However, compliance results in a forced reduction in heat load when river temperatures approach the regulation limit. To check whether the plant needs to reduce its thermal load, the following equation is applied:

$$T_{ro} = \frac{(M_r - \dot{m})T_{ri} + \dot{m}T_o}{M_r} \quad (7)$$

Here,  $T_{ro}$  (°C) is the temperature of the river at the outlet point assuming immediate thermal equilibrium, and  $T_o$  (°C) is the discharge temperature. The regulation states that  $T_{ro} \leq T_{lim}$ , where  $T_{lim}$  (°C) is the temperature limit set by the regulation. If  $T_{ro} \geq T_{lim}$  the waste heat load  $Q$  must be reduced until the regulatory condition is satisfied. But, a decrease in  $Q$  raises the internal heat within the system causing a higher condensing temperature and resulting in a lower net efficiency and power output.

A power plant can reduce its thermal load by reducing the amount of withdrawn water,  $\dot{m}$ , or reducing the temperature difference between inlet and outlet,  $\Delta T$ . However,  $\dot{m}$  and  $\Delta T$  are dependent on each other and an increase in one implies a decrease in the other and visa versa when the heat load ( $Q$ ) varies. Therefore, a relationship between  $\Delta T$  and  $\dot{m}$  needs to be established that agrees with a decrease in  $Q$ , which is required to meet regulation. For this paper a decrease in  $\Delta T$  will be through heat load reduction and its relationship with  $\dot{m}$  is given by:

$$\dot{m}_{CWA} = \beta \dot{m} \quad (8)$$

Where  $\dot{m}_{CWA}$  ( $\text{m}^3\text{s}^{-1}$ ) is the new withdrawal rate that complies with the CWA and  $\beta$  is the increased withdrawal coefficient given by:

$$\beta = \left( \frac{\Delta T}{\Delta T_{CWA}} \right)^\gamma \quad (9)$$

Here,  $\gamma$  is a constant based on the results in Hamanaka et al. (2009) and  $\Delta T_{CWA}$  is the new decreased temperature difference that satisfies the regulatory condition.<sup>40</sup> Thus, the heat load becomes:

$$Q_{CWA} = \dot{m}_{CWA} C_p \Delta T_{CWA} \quad (10)$$

This new heat load is sufficiently low to satisfy equation (7) and the relationship described in equation (9) can be seen in figure 3. Once the waste heat load is calculated, the output of the plant can be estimated (discussed later).

## 2.5. Cooling Tower – Inlet Temperature Check

Variables needed to calculate the waste heat load for the cooling tower differ slightly from the once-through technology but the methodology follows the same checks and logic. For a power plant using a cooling tower, the river temperature does not determine the inlet temperature  $T_i$  (°C) of cooling water for the condenser. Instead it is dependent on the approach (°C) (an indicator of cooling tower performance – the lower the approach, the cooler the intake cooling water and the more efficient the cooling process) and wet bulb temperature  $T_{WB}$  (°C) given by:

$$T_i = T_{WB} + A \quad (11)$$

The minimum approach (A) is typically considered to be a few °C.<sup>5,50</sup> The ability to determine  $T_i$  enables the use of equations (1)-(4), with the assumption that the condensing temperature rises, and the waste heat load increases, when  $T_i \geq T_A$  (replacing  $T_{r1}$  with  $T_i$ ). For this paper, the wet-bulb temperature was assumed to equal the initial river temperature, which is an appropriate assumption for areas with comparatively high relative humidity (~80%) as in the U.S. northeast. This was concluded by calculating wet-bulb and river temperatures as a functions of air temperature between 1-30°C using equations from Stull (2011) and Morrill et al. (2005).<sup>44-45</sup>

## 2.6. Cooling Tower: Water Availability Check

The relationship between the increase in waste heat (reduction in efficiency) load and make-up water required to be withdrawn into the cooling tower ( $\dot{m}_{mu}$ ) is given by:<sup>16</sup>

$$\dot{m}_{mu} = \frac{Q \left( \frac{f_{lat}}{h_{fg}} \right)}{\left( 1 - \frac{1}{n} \right)} \quad (12)$$

Here,  $f_{lat}$  (unitless) is the fraction of heat rejected by latent heat transfer and  $h_{fg}$  (MJm<sup>-3</sup>) is the latent heat of vaporization. The value  $n$  represents the number of cycles of concentration of impurities allowed in the circulating water, ranging from 2-10.<sup>33</sup> The evaporated water and the blowdown  $\dot{m}_{BD}$  (the water that is discharged back to the water body) are calculated as functions of  $\dot{m}_{mu}$ ,  $\dot{m}$ , and  $n$ .<sup>44</sup> Similarly to the once-through computations; once the amount of make-up water required is calculated the plant needs to check if it is available.

To replace the water consumed by the cooling tower, the power plant relies on the constant availability of water in the river to withdraw from. As with the once through technology, there are two possible conditions for the withdrawal rate - optimal and sub-optimal (equations 5 and 6). Under optimal conditions, when there is enough water to satisfy the withdrawal rate (make-up water), the variables (i.e. evaporation rate, efficiency etc.) calculated in the previous step remain unchanged. However, if the condition is sub-optimal

a constraint is imposed on  $\dot{m}_{mu}$  and thus the flow of recirculating water decreases. However, as with once-through the power plant can compensate by increasing  $\Delta T$ .

### 2.7. Cooling Tower: Regulatory (CWA) Check

After checking the physical settings for constraints, the CWA is considered. The equation for the river temperature now becomes:

$$T_{ro} = \frac{(M_r - \dot{m}_{mu})T_{ri} + \dot{m}_{BD}T_i}{M_r} \quad (13)$$

The discharge volume is considerably less and the discharge temperature is lower than the once-through method. This is because the make-up is at least 17 times less than the once-through system and water is discharged into the river after it is cooled in the tower (rather than directly after it passes through the condenser). Thus, cooling towers have a lower thermal impact but their higher rate of consumption, a few times higher than once-through, reduces the natural volume flow of the river and other users downstream may be affected. Theoretically, under certain climate conditions and providing a power plant has a sufficiently low approach, cooling towers may lower the temperature of the river at the outlet point.

### 2.8. Dry cooling

As in the cooling tower method, the dry-cooling method does not depend on river temperature to determine the temperature of the water at the inlet point in the condenser. Instead, it is dependent on air temperature:

$$T_i = T_{air} + ITD \quad (14)$$

Here,  $T_{air}$  is the ambient air temperature in °C and ITD is the Initial Temperature Difference in °C and is typically between 10°C and 30°C.<sup>40</sup> The efficiency of the power plant decreases by using equation (2) and simply substituting  $T_i$  for  $T_{ri}$ . The loss of efficiency at this point is final –there is no water check or regulatory check needed because the power plant does not withdraw any water from the river.

### 2.9. Combined Cycle

Combined cycle power plants employ two cycles – gas and steam. To accurately estimate efficiency losses in combined cycle plants it is important to calculate the optimal efficiencies both steam and gas cycles,  $\eta_{opt\ steam}$  and  $\eta_{opt\ gas}$ , respectively, and their efficiency losses due to physical or regulatory constraints. However, such detail is not typically available for the existing fleet of combined cycle plants. Nevertheless, to achieve a higher level of accuracy, in this paper, the combined cycle efficiencies for gas and steam were split into the same ratio as in the example in Fay and Golomb (2012), shown below<sup>8</sup>:

$$\eta_{opt\ steam} = Y\eta_{th} \quad (15)$$

$$\eta_{opt\ gas} = \frac{\eta_{th} - \eta_{steam}}{1 - \eta_{steam}} \quad (16)$$

Here,  $\eta_{th}$  is the constant that corresponds to the steam cycle portion of  $\eta_{th}$ , the total efficiency for the combined cycle plant.

Combined cycle power plants employ one of the three cooling technologies mentioned above, once-through, recirculating (cooling tower) or dry cooling, to cool the steam part of the cycle. The loss of efficiency on the steam side is calculated in the same way as described in the above sections: progressing from the inlet temperature check to the regulatory check. However, there is an additional potential physical constraint that needs to be taken into account for combined cycle power plants: warm ambient air at the compressor stage of the gas cycle. It has been estimated that for a rise in 10°F (5.56°C), the efficiency of the gas cycle and output decreases by 3-3.5%.<sup>21</sup> Therefore, an assumption is made that efficiency decreases above  $T_b$ , in the model taken to be 20°C, a conservative assumption considering literature stating efficiency losses starting above 5°C.<sup>22</sup> Thus, the following equations calculate the efficiency loss of the gas cycle,  $\eta_{g\ loss}$ , and the efficiency of the gas cycle,  $\eta_{gas}$ :

$$\eta_{g\ loss} = \alpha(T_{air} - T_b) \quad (17)$$

$$\eta_{gas} = \eta_{opt\ gas}(1 - \eta_{g\ loss}) \quad (18)$$

Here,  $\eta_{opt\ gas}$  is the optimal efficiency of the gas cycle and  $\alpha$  is the constant that accounts for the loss in efficiency per degree Celsius increase above  $T_b$ . When  $T_{air} < T_b$ , there is no loss in efficiency. After the efficiency loss is calculated, the efficiencies of the gas and steam cycle,  $\eta_{steam}$ , are used to calculate the overall efficiency for the combined cycle and taken as<sup>8</sup>:

$$\eta_{th} = \eta_{gas} + \eta_{steam} - (\eta_{steam}\eta_{gas}) \quad (19)$$

## 2.10. Hybrid

Hybrid plants can combine more than one type of cooling technology for their operations. In this paper, it is assumed that below warm temperatures, considered to be less than 20°C, the dry-cooling technology is used for 100% of the waste heat load. However, above this temperature, the hybrid cooling kicks-in and the power plant is split into 1/3 dry-cooling and 2/3 recirculating (cooling tower) as in the example in Hamanaka (2009).<sup>40</sup> Any efficiency losses due to physical and regulatory constraints apply in the same way as described above for dry-cooling and cooling towers.

### 3. Outputs

Model Outputs include thermal pollution, water withdrawal and consumption and electric production. To measure thermal pollution an equilibrium temperature model is used (Dingman 1972).<sup>42</sup> The relevant equation paper to determine the downstream impact of thermal pollution is:

$$T_d = (T_{ro} - T_e) \exp\left(-\frac{qx}{\rho C_p} h v_w\right) + T_e \quad (20)$$

Here,  $T_d$  (°C) is the downstream river temperature a distance of  $x$  meters from the discharge point of the power plant,  $T_e$  (°C) is the equilibrium temperature,  $\rho$  ( $\text{kgm}^{-3}$ ) the density of water (assumed = 1),  $q$  ( $\text{Jm}^{-2}\text{s}^{-1}$ ) is the energy exchange coefficient,  $h$  is the depth of the river in meters, and  $v_w$  ( $\text{ms}^{-1}$ ) is the velocity of the river. For the next power plant that is a distance of  $x$  meters from the upstream power plant, the inlet temperature would be  $T_d$ . Therefore, if there is a high thermal polluting plant upstream and the distance between two plants is short so that the river does not reach its equilibrium temperature before it reaches the next plant downstream, it will affect the downstream plant's output.

The output of the plant is calculated using the following equations:

$$W_e = Q_{in} \eta_{th} = \eta_{th} (W_e + Q + Q_{hs}) \quad (21)$$

Therefore:

$$W_e = \frac{-(Q+Q_{hs})}{1-\frac{1}{\eta_{th}}} \quad (22)$$

Where  $W_e$  (MW) is the electric output of the plant,  $Q_{hs}$  (MW) is the waste heat released to sinks other than the condenser,  $Q$  (MW) is the waste heat load calculated in previous steps, and  $\eta_{th}$  is the thermal efficiency, assumed to be constant. The model assumes that the input heat load,  $Q_{in}$  (MW), is optimal (highest) at all times (However, different scenarios could be taken into account based on results from economic models that change the operating hours or input heat load. For example, a reduction in the availability of coal would reduce the amount of heat input available, certainly in the short run, until a suitable replacement were found.). Thus, the electric output is based on the conditions of the surrounding environment and the CWA (if the CWA is turned on  $Q = Q_{CWA}$ ).

### 4. Computational Framework of the Model

TP2M was designed in MATLAB and run in MATLAB for this study. It can be run on a laptop and typically takes less than few seconds per scenario. Input files that contain power plant and physical variables (such as those described in the settings section of the methodology) are required and were formed in Excel. However, these inputs can be coded into the program and adjusted as necessary by the user. Outputs are generated in MATLAB and then moved into Excel for further analysis. The intention was to



transfer the framework into a water balance model. Indeed, this is already being done under the National Science Foundation project cited in the Acknowledgements – ‘Building a Northeast Regional Earth-System Model’.<sup>11</sup> In this effort, TP2M is being coupled within the Framework for Aquatic Modeling in the Earth System (FrAMES) to model the relationship between power plants and water resources in real world settings.<sup>35,51,52</sup>

**Figure 7. Relationship between withdrawal increase and reduced temperature difference for overall reducing Q to comply with the CWA regulations**

