**Supporting Information**

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**S1.1 General Circulation Models**

While running the Integrated Environmental Control Model (IECM) to simulate power plant operations under a variety of climate conditions, we rely on downscaled climate data from five General Circulation Models: access1-0, bcc-csm1-1-m, canesm2, ccsm4 and cesm1-bgc. Such data include simulated air temperature, precipitation, and wind speed between 1950 and 2099 for the continental U.S. under Representative Concentration Pathway (RCP) 8.5. We compiled all the data into a list of meteorological input scenarios. These data are meant to incorporate the broadest possible range of meteorological conditions within the continental United States.

**S1.2 Backpressure Efficiency Relationships**

Backpressure within a turbine can be thought of as resistance pressure. Higher ambient air pressures improve the boiler efficiency, but higher turbine backpressure instead reduces plant efficiency. Previous studies considering capacity loss due to climate change have not incorporated this effect. Each class of cooling technology has different design condition backpressures, and when backpressures exceed design conditions, each class of cooling technology has different associated output reductions

(Electric Power Research Institute (EPRI), 2002; 2004; 2007).

Figure S1-Figure S3 show the temperature and backpressure relationship for turbines associated with once-through cooling systems, recirculating cooling systems, and dry-cooling systems. Figure S4 – Figure S6 show the relationship between backpressure and output reduction (or net heat rate increase) for these plants. For each cooling system, we combine the two relationships to derive a link between dry bulb air temperature and output reduction. Separately, we examine the link between steam cycle heat rate and net plant efficiency, to determine how much net plant heat rates are affected by changes to the steam cycle heat rate. The final equation we used is summarized in Equation S1, and the relevant coefficients are in Table S1. We found design conditions, temperature-backpressure relationships, and backpressure-output relationships in several National Renewable Energy Laboratory (NREL) and Electric Power Research Institute (EPRI) reports

(Ashwood & Bharathan, 2011; Electric Power Research Institute (EPRI), 2002; 2004; 2007).

For natural gas combined cycle plants, we do not incorporate this efficiency-backpressure-steam cycle heat rate relationship. Though the latest version of IECM allows users to modify the steam cycle heat rate in natural gas combined cycle plants, the steam cycle heat rate for these plant does not impact the net heat rate in a way that meaningfully affects net efficiency or net capacity. Since our analysis found this effect to be minimal in coal plants, where all generation comes from the steam cycle, in combined cycle plants it will be an even less significant effect, since approximately one third of generation capacity stems from the steam cycle.

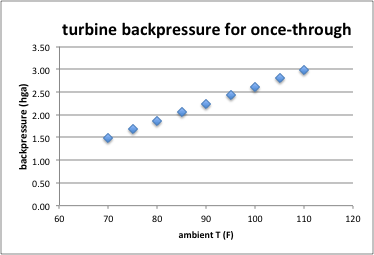


Figure S. Ambient temperature and turbine backpressure relationship for once-through systems

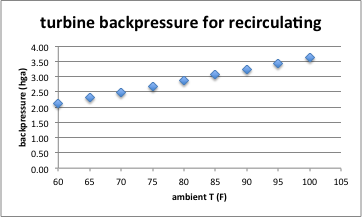


Figure S. Ambient temperature and turbine backpressure relationship for recirculating systems

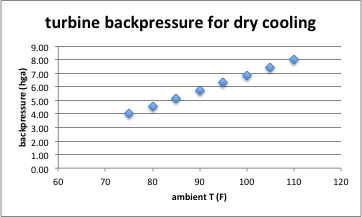


Figure S. Ambient temperature and turbine backpressure relationship for dry-cooling systems

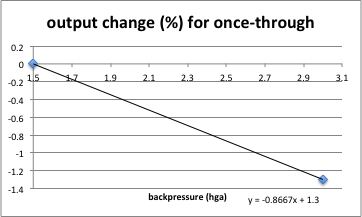


Figure S. Backpressure and output reduction relationship for once-through systems

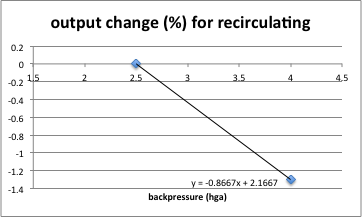


Figure S. Backpressure and output reduction relationship for recirculating systems

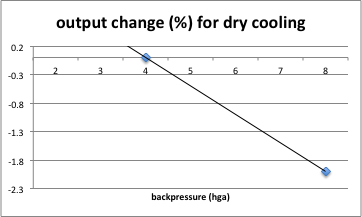


Figure S. Backpressure and output reduction relationship for dry-cooling systems

Equation S

where T is the ambient temperature in Fahrenheit, a and b are the slope and intercept relating temperature (°F) and backpressure (hga), c and d are the slope and intercept relating backpressure (hga) and output reduction (%), and e and f are the slope and intercept relating steam cycle heat rate (Btu/kWh) and percentage reduction.

Table S1. Variables to use in Equation S1 for determining steam cycle heat rate at given temperature

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| **Cooling System** | **Boiler Type** | **a** | **b** | **c** | **d** | **e** | **f** |
| **Once-Through** | **Sub-**  **critical** | .0375 | -1.13 | -0.866 | 1.3 | -0.01397 | 108.84 |
| **Super-critical** | .0375 | -1.13 | -0.866 | 1.3 | -0.01397 | 102.81 |
| **Ultra-supercritical** | .0375 | -1.13 | -0.866 | 1.3 | -0.01397 | 93.68 |
| **Recirc-ulating** | **Sub-**  **critical** | .0375 | -0.13 | -0.866 | 2.166 | -0.01395 | 108.68 |
| **Super-critical** | .0375 | -0.13 | -0.866 | 2.166 | -0.01395 | 102.67 |
| **Ultra-supercritical** | .0375 | -0.13 | -0.866 | 2.166 | -0.01395 | 93.55 |
| **Dry-Cooling** | **Sub-**  **critical** | .114 | -4.57 | -0.5 | 2 | -0.01359 | 109.04 |
| **Super-critical** | .114 | -4.57 | -0.5 | 2 | -0.01359 | 103 |
| **Ultra-supercritical** | .114 | -4.57 | -0.5 | 2 | -0.01359 | 93.86 |

**S1.3.1 Recirculating Cooling Towers**

Cooling towers circulate water that absorbs and discharges heat from the steam cycle. They are designed so that a specific temperature drop occurs between the water coming into the tower (inlet) and leaving the tower (outlet). Outlet water exits the cooling tower and returns to the condenser and auxiliary heat exchanger, ideally cooled as much as possible so that it can efficiently absorb heat again. The temperature difference between this outlet water and the ambient wet bulb temperature is called the approach.

A wet bulb temperature is the lowest air temperature that evaporation can achieve, so a very high ambient wet bulb temperature may limit the cooling capability of a cooling tower, since towers employ evaporative cooling. As temperature and humidity rise, so does the wet bulb temperature, potentially limiting the approach, which then limits the tower cooling capacity, and consequently the power plant’s net capacity. To avoid this scenario, cooling tower design is typically scaled to the an annual 1% wet bulb temperature, or the wet bulb temperature that is only exceeded 1% of the time, so as to maintain a minimum approach and preserve an intended cooling duty (Jiang, 2013). For this analysis, we include a range of inlet and outlet cooling water temperatures to represent different design conditions that could be observed in power plants in different climatic regions.

To maintain an approach, the flow rate of cooling water within a cooling system can shift according to need, but only to a certain extent. If the ambient wet bulb temperature exceeds the design outlet temperature, a larger flow rate can help maintain the designated cooling duty by increasing heat transfer from exhaust steam to cooling water. If the temperature change across a cooling tower or condenser is constrained by ambient conditions, increased flow can compensate for this constraint and absorb more heat than cooling water at standard flow. Beyond a given volume of increased flow, however, this limited cooling capacity will instead translate to net plant capacity reductions.

In our calculations, we assume that a cooling tower can increase its flow rate up to 30% with minimal impact on power plant capacity. However, for ambient conditions requiring water flow above that point (identified as scenarios in which IECM resized the cooling system above the baseline design parameters), there is a reduction in cooling capability and consequently a reduction in the net capacity of the power plant (ASHRAE, 2008). We determine the capacity penalty based on the following equation (Zhai, Rubin, & Versteeg, 2011):

where MWg is the plant’s gross capacity; is the total recirculation cooling water flow rate (tons/hr); HR, refers to the steam cycle heat rate; , is the auxiliary cooling load in a constant percentage of load on the primary steam cycle, representing additional heat load from auxiliary equipment that requires cooling; Cp is a constant that represents cooling capacity of water; and is the temperature drop in the cooling water that occurs across the cooling tower. In a cooling tower, high ambient temperatures and relative humidities will constrain **,** and can then increase to offset the limited temperature differential, while maintaining the same gross capacity. Therefore, in comparing the gross capacity of a constrained system to that of a system operating under design conditions, we have:

=

where and represent design condition flow rate, temperature differential, and gross capacity, respectively. We can cancel out constants, and since we demonstrated that heat rate varies minimally, we can drop this variable as well, leaving the equation:

Again, we are assuming that the cooling water flow rate can increase by 30% to accommodate constrained temperature differentials while maintaining a designed gross capacity. However, when IECM sizes a cooling system, it does not include the oversizing mentioned above. We are instead interested in the capacity loss incurred by a system of given size, with some oversizing, operating in sub-optimal conditions. In such a system, we take the percentage by which the cooling water flow rate exceeds the maximum flow rate to signify the resulting capacity loss. We assume that this represents the percentage by which the temperature differential would have to decrease, since in practice the flow rate cannot increase without limit. The percentage by which the temperature differential would have to decrease, then, also represents the capacity penalty. The equation we use for recirculating systems is:

where is the total recirculation cooling water flow rate (tons/hr) obtained from the IECM simulations; and is the default water flow rate for the cooling tower (tons/hr).

This equation defines the available capacity as a function of the water flow rate at which the power plant operates in extreme conditions and the design water flow rate into the cooling tower (both from IECM). We use this equation to estimate the available capacity when plants operate at a water flow rate that exceeds the design water flow rate into the tower by more than 30% (ASHRAE, 2008).

**S1.3.2 Once-Through Cooling Systems**

Like recirculating systems, once-through cooling systems operate with a cooling water flow rate according to their capacity. However, once-through cooling systems use open-loop water bodies instead of cooling towers to reject the exhaust steam heat. Two effects result in changing cooling water requirements in once through systems. In these systems, an increase in air temperature leads to an increase in the turbine backpressure, which in turn increases the temperature of the steam. Higher temperature steam can be condensed by higher temperature cooling water (Electric Power Research Institute (EPRI), 2012). At the same time, cooling systems are designed for a specific temperature differential across the condenser (which is the gap between steam condensing temperature and water intake temperature or the increase in cooling water temperature). If the actual temperature differential is lower than the design temperature differential, the plant could face capacity constraints.

We examine a range of typical design temperature differentials between 10°F and 30°F (Madden, Lewis, & Davis, 2013; US Environmental Protection Agency, 2005). In open-loop once-through systems, cooling water intake temperature does not necessarily shift according the exhaust steam temperature. In IECM, we cannot model input stream temperatures directly, but rather we can indicate the change in temperature that occurs across the condenser (the increase in cooling water temperature, or the gap between steam condensing temperature and water intake temperature). We run scenarios with design parameters of 10°F, 20°F, and 30°F differentials between the cooling water intake and discharge, so that the change in temperature is the minimum of: either 1) 10°, 20° or 30°, or 2) the difference between the steam condensing temperature and the water intake temperature.

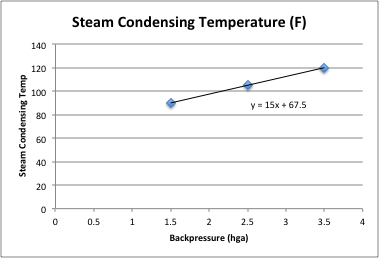


Figure S7. Steam condensing temperature vs turbine backpressure for once-through cooling

We use the same equation used for recirculating systems to examine the flow-based capacity constraint. Similar to cooling towers, we assume that the cooling water flow rate can increase to mitigate the constraint, but only to a certain extent – for once-through systems, we allow for a doubling of cooling water flow

(Macknick, Newmark, Heath, & Hallett, 2011). Therefore the equation becomes:

where is the total condenser cooling water flow rate (tons/hr) and is the default water flow rate for the condenser (tons/hr) (both from IECM).

Note that environmental regulations on discharge temperatures can also curtail capacity in thermoelectric power plants. In practice, however, power plant operators could obtain permits to temporarily exceed discharge limits to maintain power system reliability (Henry & Pratson, 2016). Therefore our analysis focuses on the thermodynamic principles governing the operational constraints described above.

**S1.3.3 Dry Cooling Systems**

Dry-cooling systems use no water and instead condense the steam via convection by routing the steam to an array of coils and blowing air directly across the array. Multiple coils comprise a cooling cell. A dry cooling system will have a certain number cells, according to design, and each cell takes up a given amount of area. The footprint area of the cooling system, rather than the total cycled cooling water, represents the cooling system size. The designated size of cooling systems varies by the difference between the anticipated peak air temperature (usually the summer average) and the exhaust steam temperature, or the initial temperature difference (ITD) (Jiang, 2013). The ITD is the primary design parameter for the size of a dry cooling system. Table S2 summarizes the footprint area of cooling systems with different ITDs under default IECM conditions.

Table S2. Baseline footprint areas of dry cooling systems for 650 MWg pulverized coal plant with supercritical boiler.

|  |  |
| --- | --- |
| ITD (°F) | Baseline Footprint Area of Cooling System (sq ft) |
| 25 | 67,220 |
| 35 | 47,460 |
| 45 | 36,590 |
| 55 | 29,730 |

While water flow in cooling towers and once-through systems can be somewhat modulated according to cooling need, above or below designed flow rates, dry-cooling systems are constrained by the number of constructed cooling cells (ASHRAE, 2008). Cooling loads exceeding this constraint impose a capacity penalty on the power plant. To account for such capacity constraint, we first run IECM model to identify baseline cooling sizes for different design ITD. For example, we first use IECM to estimate the cooling system size for a 650 MW dry-cooling plant expected to operate under an annual peak air temperature of 80°F. Due to climate change, such plant might operate in temperatures exceeding 80°F. To estimate the capacity penalty of such operations, we used Equation S2 to calculate parameter , which is the land surface area of the dry cooling system normalized by rejected heat (m2/MW). In this equation, an instantaneous ITD (ITDi) is calculated outside of IECM as the difference between the steam temperature (126.1°F) and the instantaneous air temperature we aim to represent from the GCM data. Pambient is the associated air pressure (in kPa). We then use this parameter to estimate a modified gross capacity (MWg) that maintains the designated number of cooling cells obtained from the initial IECM simulation (Zhai et al., 2009), as defined in Equation S3, where HRs is the steam cycle heat rate. Finally, we input this modified gross capacity into IECM to obtain a modified net capacity, which we then use to estimate the capacity penalty of the plant (defined as a percentage of the original nameplate capacity).

Equation S2

Equation S3

Dry cooling systems are expensive to build, about three to four times as much as cooling towers, so constructing excess cooling capacity is to be avoided if possible (Electric Power Research Institute (EPRI), 2002). A higher design peak temperature corresponds to smaller design ITDs and higher capital costs (Jiang, 2013). In our analysis, we include the range of typical design ITDs found in literature (Electric Power Research Institute (EPRI), 2002).

**S2.1 Tested Combinations**

We evaluated various technological combinations to determine whether meaningful differences existed in performance response to meteorological change. Table S3 below lists the plants we tested, where each number represents a different plant within the cooling technology category and asterisks in that row indicate technologies present. We ultimately found that cooling systems and their design parameters played a much greater role in determining how a power plant responded to meteorological change, but here we present the other configurations considered.

Table S3. Technology combinations for coal power plants

|  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| **Technology Combination Number** | | | Desulfurizer | | Coal Rank | | | Boiler Type | | |
| **Once-Through** | **Recirc-ulating** | **Dry** | Wet FGD | Lime Spray Dryer | Bit-uminous | Sub-bituminous | Lignite | Sub-critical | Super-critical | Ultra super-critical |
| 1 | 19 | 37 | \* |  | \* |  |  | \* |  |  |
| 2 | 20 | 38 | \* |  | \* |  |  |  | \* |  |
| 3 | 21 | 39 | \* |  | \* |  |  |  |  | \* |
| 4 | 22 | 40 | \* |  |  | \* |  | \* |  |  |
| 5 | 23 | 41 | \* |  |  | \* |  |  | \* |  |
| 6 | 24 | 42 | \* |  |  | \* |  |  |  | \* |
| 7 | 25 | 43 | \* |  |  |  | \* | \* |  |  |
| 8 | 26 | 44 | \* |  |  |  | \* |  | \* |  |
| 9 | 27 | 45 | \* |  |  |  | \* |  |  | \* |
| 10 | 28 | 46 |  | \* | \* |  |  | \* |  |  |
| 11 | 29 | 47 |  | \* | \* |  |  |  | \* |  |
| 12 | 30 | 48 |  | \* | \* |  |  |  |  | \* |
| 13 | 31 | 49 |  | \* |  | \* |  | \* |  |  |
| 14 | 32 | 50 |  | \* |  | \* |  |  | \* |  |
| 15 | 33 | 51 |  | \* |  | \* |  |  |  | \* |
| 16 | 34 | 52 |  | \* |  |  | \* | \* |  |  |
| 17 | 35 | 53 |  | \* |  |  | \* |  | \* |  |
| 18 | 36 | 54 |  | \* |  |  | \* |  |  | \* |

**S2.2 Baseline Net Capacities and Heat Rates**

Some of our dependent values are presented as absolute while others are relative reductions or increases. These decisions are based on what information is readily available and what we believe would be useful for other users of our simplified models. Net capacities are presented as relative percentage changes and water use intensities are presented as absolute values.

Power plant net capacities are generally found easily. For existing power plants in the U.S., net capacity and heat rates are available in public power plant databases, and for hypothetical plants, these characteristics and fuel types are often first considerations in planning. In contrast, water use intensities are not reliably reported or easily accessible. Recently EIA began collecting self-reported water use data but other studies have found it to be incongruent with literature values and to have numerous internal inconsistencies (Averyt et al., 2013). Water management practices differ substantially from the systems that govern electricity generation and distribution. At present, these water management practices do not result in precise records for water use like those that exist for electricity. Estimated water use intensities vary by technology, and by climate, instead of plant size. Therefore we believe it is most useful to present water use intensities as absolute values in our regressions.

For values that are presented as relative percentage changes, Tables S4 – S6 detail the baseline values for a default plant in IECM. Since we tested for changes in heat rate, we also include these baseline values along with net capacity, though we found that climate did not significantly alter the heat rate. The technology combination numbers correspond to those found in Table S3.

Table S4. Baseline Net Capacities and Heat Rates for Coal Plants with Once-Through Cooling Technology

|  |  |  |  |
| --- | --- | --- | --- |
| Once-Through Technology Numbers | Net Capacity (MW) | Gross Heat Rate (Btu/kWh) | Net Heat Rate (Btu/kWh) |
| 1 | 611.2 | 8715 | 9268 |
| 2 | 613.5 | 8233 | 8722 |
| 3 | 616.4 | 7501 | 7910 |
| 4 | 602.5 | 9119 | 9837 |
| 5 | 605.3 | 8614 | 9250 |
| 6 | 609 | 7849 | 8378 |
| 7 | 593.4 | 9415 | 10310 |
| 8 | 596.7 | 8894 | 9689 |
| 9 | 601.1 | 8104 | 8764 |
| 10 | 617.7 | 8715 | 9170 |
| 11 | 619.7 | 8233 | 8636 |
| 12 | 622 | 7501 | 7839 |
| 13 | 610 | 9119 | 9716 |
| 14 | 612.4 | 8614 | 9143 |
| 15 | 615.4 | 7849 | 8290 |
| 16 | 601.3 | 9415 | 10180 |
| 17 | 604.2 | 8894 | 9569 |
| 18 | 607.9 | 8104 | 8666 |

Table S5. Baseline Net Capacities and Heat Rates for Coal Plants with Recirculating Cooling Technology

|  |  |  |  |
| --- | --- | --- | --- |
| Recirculating Technology Combination Numbers | Net Capacity (MW) | Gross Heat Rate (Btu/kWh) | Net Heat Rate (Btu/kWh) |
| 19 | 604.7 | 8715 | 9368 |
| 20 | 608 | 8233 | 8801 |
| 21 | 611.6 | 7501 | 7972 |
| 22 | 596 | 9119 | 9945 |
| 23 | 599.8 | 8614 | 9335 |
| 24 | 604.2 | 7849 | 8444 |
| 25 | 586.9 | 9415 | 10430 |
| 26 | 591.2 | 8894 | 9779 |
| 27 | 596.3 | 8104 | 8834 |
| 28 | 611.2 | 8715 | 9269 |
| 29 | 614.1 | 8233 | 8713 |
| 30 | 617.2 | 7501 | 7900 |
| 31 | 603.5 | 9119 | 9822 |
| 32 | 606.9 | 8614 | 9226 |
| 33 | 610.6 | 7849 | 8355 |
| 34 | 594.8 | 9415 | 10290 |
| 35 | 598.6 | 8894 | 9658 |
| 36 | 603.1 | 8104 | 8735 |

Table S6. Baseline Net Capacities and Heat Rates for Coal Plants with Dry Cooling Technology

|  |  |  |  |
| --- | --- | --- | --- |
| Dry-Cooling Technology Combination Numbers | Net Capacity (MW) | Gross Heat Rate (Btu/kWh) | Net Heat Rate (Btu/kWh) |
| 37 | 598.6 | 8976 | 9747 |
| 38 | 602.3 | 8479 | 9150 |
| 39 | 607.5 | 7726 | 8267 |
| 40 | 589.9 | 9392 | 10350 |
| 41 | 594.1 | 8872 | 9707 |
| 42 | 599.9 | 8084 | 8758 |
| 43 | 580.7 | 9697 | 10850 |
| 44 | 585.4 | 9161 | 10140 |
| 45 | 592 | 8347 | 9164 |
| 46 | 605.3 | 8976 | 9639 |
| 47 | 608.7 | 8479 | 9055 |
| 48 | 613.2 | 7726 | 8189 |
| 49 | 597.6 | 9392 | 10220 |
| 50 | 601.4 | 8872 | 9589 |
| 51 | 606.6 | 8084 | 8662 |
| 52 | 588.8 | 9697 | 10700 |
| 53 | 593.1 | 9161 | 10040 |
| 54 | 599 | 8347 | 9057 |

**S2.3 Regression Tables**

Table S7 details the format of our regression equations and Tables S8 – S17 provide information on the coefficients and intercepts, for each equation with different cooling system design parameters. In Table S7, y is either available capacity (% of installed capacity) or water withdrawal intensity (gal/MWh); and T are the coefficient and value for air temperature (°F); and P are the coefficient and value for air pressure (psia); and RH are the coefficient and value for relative humidity (%); and WT are the coefficient and value for intake water temperature (°F); and is the intercept. It should be noted that coefficients are unique for each dependent variable, plant configuration, and design condition – for example, considering the relative capacity reduction equation, the coefficient on air temperature at a once-through system with a designed water intake temperature rise of 10°F is different than the air temperature coefficient in estimating water withdrawal intensity for the same plant. Likewise, the coefficient differs if the plant is designed with a 20°F water temperature rise, or if the plant adopts a recirculating system. Capacity reduction equations yielding a dependent variable below 0 or above 1 indicate full curtailment or no curtailment, respectively. Withdrawal intensity equations similarly have limits beyond which the dependent variable output is no longer applicable, and these limits are found by the captions of those tables. For each table, the stars indicate p values (p.val < .001 ="\*\*\*", p.val < .01 = "\*\*", p.val < .05 = \*) and the values beneath the coefficients are t values. We also include the root mean squared error and mean absolute error, to better clarify the distribution and variance of model error.

The columns of the regression tables indicate design parameters described in Table 2 of the main text. For once-through systems, this is the designed increase in cooling water temperature rise. For recirculating systems, the temperature pairings refer to the designed cooling tower inlet-outlet temperatures. In the case of dry cooling systems, the initial temperature difference or ITD is the designed temperature difference between exhaust steam and ambient air temperature.

In Tables S18-S23, we present the results of linear regressions on the percentage change in net plant heat rate. We take the baseline net plant heat rates as those outlined in Tables S4-S6. A dependent variable of 0.01, for example, would correspond to a 1% increase in heat rate, or minor loss in efficiency. However, we did not find that meteorology meaningfully altered efficiency. Even for coal plants with dry-cooling, the technology configuration where we detected the greatest increase in plant heat rate with rising temperature, our model predicts that an extreme 110°F day would cause about a 2% increase in heat rate, or less than a 1% decrease in efficiency. Therefore, we do not present these results in the main paper.

Table S7. Format of regression equations for key variables influenced by meteorological change

|  |  |  |
| --- | --- | --- |
| **Cooling Type** | **Variable** | **Equation** |
| Once-Through | Capacity Available (%) |  |
| Withdrawal Intensity (gal/MWh) |  |
| Recirculating | Capacity Available  (%) |  |
| Withdrawal Intensity  (gal/MWh) |  |
| Dry-Cooling | Capacity Available  (%) |  |

Table S8 Capacity available for coal plants with once-through cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | 0.09 \*\*\* | 0.0991 \*\*\* | 0.071 \*\*\* |
|  | -193 | -125 | -190 |
| Stream Temperature (F) | -0.159 \*\*\* | -0.179 \*\*\* | -0.128 \*\*\* |
|  | (-181) | (-140) | (-192) |
| Intercept | 8.14 \*\*\* | 8.42 \*\*\* | 5.72 \*\*\* |
|  | -190 | -155 | -178 |
| R Squared | 0.991 | 0.966 | 0.972 |
| RMSE | 0.0146 | 0.0489 | 0.0461 |
| MAE | 0.0133 | 0.0408 | 0.0412 |

Table S9. Withdrawal intensity (gal/MWh) for coal plants with once-through cooling. For delta 10, values below 55,000 and above 120,000 should be replaced with those respective values. For delta 20, values below 26,000 and above 100,000 should be replaced with those respective values. For delta 30, values below 21,000 and above 72,000 should be replaced with those respective values.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | -9150 \*\*\* | -5110 \*\*\* | -2410 \*\*\* |
|  | (-189) | (-124) | (-180) |
| Stream Temperature (F) | 16100 \*\*\* | 9210 \*\*\* | 4340 \*\*\* |
|  | -177 | -138 | -182 |
| Intercept | -624000 \*\*\* | -331000 \*\*\* | -126000 \*\*\* |
|  | (-140) | (-117) | (-109) |
| R Squared | 0.99 | 0.965 | 0.969 |
| RMSE | 1520 | 2550 | 1650 |
| MAE | 1370 | 2130 | 1470 |

Table S10. Capacity available for coal plants with recirculating cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively. The interaction term is the product of the independent air temperature and relative humidity variables.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | 0.0224 \*\*\* | 0.0107 \*\*\* | 0.00127 \*\*\* |
|  | -21.4 | -15.9 | -9.3 |
| Relative Humidity (%) | 0.07 \*\*\* | 0.0287 \*\*\* | 0.00311 \*\*\* |
|  | -36.6 | -23.2 | -12.5 |
| Interaction Term | -0.000941 \*\*\* | -0.000378 \*\*\* | -4.05e-05 \*\*\* |
|  | (-40.4) | (-25.1) | (-13.3) |
| Intercept | -0.61 \*\*\* | 0.195 \*\*\* | 0.901 \*\*\* |
|  | (-6.93) | -3.42 | -78.4 |
| R Squared | 0.689 | 0.391 | 0.133 |
| RMSE | 0.141 | 0.091 | 0.0184 |
| MAE | 0.107 | 0.053 | 0.00696 |

Table S11. Withdrawal intensity (gal/MWh) for coal plants with recirculating cooling. For 90-70, values below 320 and above 1050 should be replaced with those respective values. For 95-75, values below 645 and above 935 should be replaced with those respective values. For 100-80, values below 650 and above 905 should be replaced with those respective values.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | 7.46 \*\*\* | 6.03 \*\*\* | 4.92 \*\*\* |
|  | -58.7 | -162 | -180 |
| Relative Humidity (%) | -1.02 \*\*\* | -0.876 \*\*\* | -0.763 \*\*\* |
|  | (-17) | (-49.4) | (-58.8) |
| Intercept | 187 \*\*\* | 284 \*\*\* | 357 \*\*\* |
|  | -15.8 | -81.9 | -141 |
| R Squared | 0.754 | 0.954 | 0.963 |
| RMSE | 40.9 | 12.8 | 9.39 |
| MAE | 18 | 9.14 | 7.36 |

Table S12. Capacity available for coal plants with dry cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | ITD 25 | ITD 35 | ITD 45 | ITD 55 |
| Air Temperature (F) | -0.00772 \*\*\* | -0.0159 \*\*\* | -0.0192 \*\*\* | -0.0191 \*\*\* |
|  | (-17.1) | (-40.7) | (-101) | (-61.2) |
| Air Pressure (psia) | 0.0134 \*\*\* | 0.0226 \*\*\* | 0.0315 \*\*\* | 0.0271 \*\*\* |
|  | -3.43 | -6.67 | -19.1 | -9.99 |
| Intercept | 1.45 \*\*\* | 1.96 \*\*\* | 2.02 \*\*\* | 1.96 \*\*\* |
|  | -22 | -34.3 | -72.9 | -43.1 |
| R Squared | 0.562 | 0.856 | 0.982 | 0.976 |
| RMSE | 0.0796 | 0.0755 | 0.0298 | 0.0234 |
| MAE | 0.0663 | 0.0652 | 0.0207 | 0.0234 |

Table S13. Capacity available for natural gas combined cycle plants with once-through cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | 0.0875 \*\*\* | 0.0991 \*\*\* | 0.0655 \*\*\* |
|  | -181 | -120 | -140 |
| Stream Temperature (F) | -0.155 \*\*\* | -0.181 \*\*\* | -0.12 \*\*\* |
|  | (-171) | (-125) | (-148) |
| Intercept | 8.03 \*\*\* | 8.67 \*\*\* | 5.48 \*\*\* |
|  | -181 | -128 | -142 |
| R Squared | 0.991 | 0.96 | 0.953 |
| RMSE | 0.0133 | 0.0491 | 0.0572 |
| MAE | 0.0113 | 0.0376 | 0.0456 |

Table S14. Withdrawal intensity (gal/MWh) for natural gas combined cycle plants with once-through cooling. For delta 10, values below 10,400 and above 26,000 should be replaced with those respective values. For delta 20, values below 5000 and above 23,000 should be replaced with those respective values. For delta 30, values below 4000 and above 15,000 should be replaced with those respective values.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | -2170 \*\*\* | -1350 \*\*\* | -614 \*\*\* |
|  | (-48.1) | (-56.8) | (-73.8) |
| Stream Temperature (F) | 3860 \*\*\* | 2420 \*\*\* | 1110 \*\*\* |
|  | -45.5 | -58.6 | -77.1 |
| Intercept | -154000 \*\*\* | -90200 \*\*\* | -34800 \*\*\* |
|  | (-37.1) | (-47.7) | (-50.2) |
| R Squared | 0.886 | 0.843 | 0.849 |
| RMSE | 1240 | 1340 | 1000 |
| MAE | 866 | 968 | 760 |

Table S15. Capacity available for natural gas combined cycle plants with recirculating cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively. The interaction term is the product of the independent air temperature and relative humidity variables.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | 0.0233 \*\*\* | 0.0113 \*\*\* | 0.00317 \*\*\* |
|  | -22 | -15.9 | -7.72 |
| Relative Humidity (%) | 0.0661 \*\*\* | 0.0266 \*\*\* | 0.00493 \*\*\* |
|  | -34.1 | -20.5 | -6.56 |
| Interaction Term | -0.000873 \*\*\* | -0.000336 \*\*\* | -6.03e-05 \*\*\* |
|  | (-37) | (-21.2) | (-5.49) |
| Intercept | -0.842 \*\*\* | 0.120 | 0.770 \*\*\* |
|  | (-9.46) | -0.0367 | -17.6 |
| R Squared | 0.606 | 0.239 | 0.0853 |
| RMSE | 0.142 | 0.0955 | 0.0552 |
| MAE | 0.109 | 0.0677 | 0.0439 |

Table S16. Withdrawal intensity (gal/MWh) for natural gas combined cycle plants with recirculating cooling. For 90-70, values below 260 and above 475 should be replaced with those respective values. For 95-75, values below 260 and above 400 should be replaced with those respective values. For 100-80, values below 260 and above 380 should be replaced with those respective values.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | 3.99 \*\*\* | 3.17 \*\*\* | 2.62 \*\*\* |
|  | -116 | -163 | -178 |
| Relative Humidity (%) | -0.48 \*\*\* | -0.446 \*\*\* | -0.39 \*\*\* |
|  | (-29.6) | (-48.3) | (-55.9) |
| Intercept | 5.36 | 62.3 \*\*\* | 98.7 \*\*\* |
|  | -1.67 | -34.5 | -72.2 |
| R Squared | 0.92 | 0.954 | 0.962 |
| RMSE | 11.1 | 6.68 | 5.06 |
| MAE | 8.05 | 4.87 | 3.98 |

Table S17. Capacity available for natural gas combined cycle plants with dry cooling. The dependent variable is a percentage from 0 to 1 (0% to 100%) of the designed net capacity available. Values below 0 or above 1 should be treated as 0 or 1, respectively.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | ITD 25 | ITD 35 | ITD 45 | ITD 55 |
| Air Temperature (F) | -0.000512 \*\*\* | -0.00399 \*\*\* | -0.0107 \*\*\* | -0.0185 \*\*\* |
|  | (-4.03) | (-9.93) | (-14.4) | (-35.5) |
| Air Pressure (psia) | 0.00262 \* | 0.00693 \* | 0.0161 \* | 0.0272 \*\*\* |
|  | -2.37 | -1.99 | -2.48 | -6 |
| Intercept | 1.01 \*\*\* | 1.24 \*\*\* | 1.66 \*\*\* | 2.09 \*\*\* |
|  | -54.4 | -21.1 | -15.2 | -27.5 |
| R Squared | 0.0845 | 0.264 | 0.531 | 0.933 |
| RMSE | 0.0224 | 0.0777 | 0.117 | 0.0474 |
| MAE | 0.0129 | 0.0531 | 0.0923 | 0.0474 |

Table S18. Percent increase in net plant heat rate for coal plants with once-through cooling. The baseline heat rate for this configuration is 8722 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to the baseline heat rate.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | 7.45E-05 | 4.01E-06 | 1.55E-05 |
|  | -1.53 | -0.167 | -1.28 |
| Stream Temperature (F) | -0.000114 | 1.39E-05 | -1.02E-05 |
|  | (-1.23) | -0.359 | (-0.474) |
| Intercept | 0.00516 | -0.00058 | 0.000603 |
|  | -1.15 | (-0.352) | -0.579 |
| R Squared | 0.0111 | 0.00482 | 0.00381 |
| RMSE | 0.00153 | 0.00148 | 0.00149 |
| MAE | 0.0012 | 0.00115 | 0.00115 |

Table S19. Percent increase in net plant heat rate for coal plants with recirculating cooling. The baseline heat rate for this configuration is 8801 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to baseline heat rates.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | 9.20E-07 | 9.20E-07 | 9.20E-07 |
|  | -0.205 | -0.205 | -0.205 |
| Relative Humidity (%) | 8.92e-06 \*\*\* | 8.92e-06 \*\*\* | 8.92e-06 \*\*\* |
|  | -4.19 | -4.19 | -4.19 |
| Intercept | 0.00122 \*\* | 0.00122 \*\* | 0.00122 \*\* |
|  | -2.93 | -2.93 | -2.93 |
| R Squared | 0.0105 | 0.0105 | 0.0105 |
| RMSE | 0.00154 | 0.00154 | 0.00154 |
| MAE | 0.00122 | 0.00122 | 0.00122 |

Table S20. Percent increase in net plant heat rate for coal plants with dry cooling. The baseline heat rate for this configuration is 9150 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to baseline heat rates.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | ITD 25 | ITD 35 | ITD 45 | ITD 55 |
| Air Temperature (F) | 0.000583 \*\*\* | 0.000616 \*\*\* | 0.000644 \*\*\* | 0.000669 \*\*\* |
|  | -95.1 | -105 | -138 | -108 |
| Air Pressure (psia) | -3.97E-05 | -8.76E-05 | -0.000144 \*\*\* | -0.000184 \*\*\* |
|  | (-0.745) | (-1.72) | (-3.54) | (-3.41) |
| Intercept | -0.0427 \*\*\* | -0.0447 \*\*\* | -0.0466 \*\*\* | -0.048 \*\*\* |
|  | (-47.6) | (-52.3) | (-68.4) | (-53) |
| R Squared | 0.974 | 0.975 | 0.99 | 0.992 |
| RMSE | 0.00108 | 0.00113 | 0.000735 | 0.000574 |
| MAE | 0.000904 | 0.00095 | 0.000608 | 0.000574 |

Table S21. Percent increase in net plant heat rate for natural gas combined cycle plants with once-through cooling. The baseline heat rate for this configuration is 6777 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to baseline heat rates.

|  |  |  |  |
| --- | --- | --- | --- |
|  | Delta 10 | Delta 20 | Delta 30 |
| Air Temperature (F) | -0.00024 \*\*\* | -0.000145 \*\*\* | -8.56e-05 \*\*\* |
|  | (-5.17) | (-5.68) | (-6.47) |
| Stream Temperature (F) | 0.000515 \*\*\* | 0.000258 \*\*\* | 0.000191 \*\*\* |
|  | -5.88 | -5.76 | -8.38 |
| Intercept | -0.0254 \*\*\* | -0.00895 \*\*\* | -0.00718 \*\*\* |
|  | (-5.92) | (-4.29) | (-6.57) |
| R Squared | 0.129 | 0.0488 | 0.066 |
| RMSE | 0.00128 | 0.00152 | 0.00161 |
| MAE | 0.00087 | 0.00115 | 0.00119 |

Table S22. Percent increase in net plant heat rate for natural gas combined cycle plants with recirculating cooling. The baseline heat rate for this configuration is 6819 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to baseline heat rates.

|  |  |  |  |
| --- | --- | --- | --- |
|  | 90-70 | 95-75 | 100-80 |
| Air Temperature (F) | -1.77e-05 \*\*\* | -1.8e-05 \*\*\* | -1.8e-05 \*\*\* |
|  | (-3.47) | (-3.52) | (-3.52) |
| Relative Humidity (%) | -3.15e-05 \*\*\* | -3.17e-05 \*\*\* | -3.17e-05 \*\*\* |
|  | (-13) | (-13.1) | (-13.1) |
| Intercept | 0.00613 \*\*\* | 0.00614 \*\*\* | 0.00614 \*\*\* |
|  | -12.9 | -13 | -13 |
| R Squared | 0.089 | 0.0899 | 0.0899 |
| RMSE | 0.00175 | 0.00175 | 0.00175 |
| MAE | 0.00136 | 0.00136 | 0.00136 |

Table S23. Percent increase in net plant heat rate for natural gas combined cycle plants with dry cooling. The baseline heat rate for this configuration is 6919 Btu/kWh. The dependent variable is a percentage from 0 to 1 (0 to 100%) that indicates the increase in heat rate due to meteorology, which becomes an efficiency loss. Any value below 0 should be replaced with 0, indicating no change to baseline heat rates.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | ITD 25 | ITD 35 | ITD 45 | ITD 55 |
| Air Temperature (F) | 2.20E-20 | -2.16E-19 | -1.31E-19 | 6.28E-21 |
|  | -1.99E-15 | (-1.88e-14) | (-1.6e-14) | -1.37E-15 |
| Air Pressure (psia) | 1.10E-18 | 1.59E-18 | -7.13E-20 | 7.14E-20 |
|  | -1.15E-14 | -1.59E-14 | (-9.97e-16) | -1.79E-15 |
| Intercept | -0.000145 | -0.000747 | -0.0021 | -0.00325 \*\*\* |
|  | (-0.0897) | (-0.444) | (-1.75) | (-4.86) |
| R Squared | 1.93E-29 | 1.67E-30 | 6.46E-29 | 1.74E-29 |
| RMSE | 0.00195 | 0.00223 | 0.00129 | 0.000506 |
| MAE | 0.00168 | 0.00195 | 0.00116 | 0.000506 |

**S3 Validation with Empirical Data**

To verify our model, we gathered gross plant heat rate as an output from IECM, along with hourly historical data. The empirical data is observational weather data and power plant operational data, for baseload coal plants with a recirculating cooling system on an hourly basis. The National Oceanic and Atmospheric Administration and Continuous Emissions Monitoring Program have publicly available data that allows us to perform this analysis for select power plants within SERC

(National Climatic Data Center, 2016; US Environmental Protection Agency (EPA), 2016). Using weather stations nearest to the power plants, we perform a fixed effects linear regression, with meteorological variables as independent variables (temperature, relative humidity, and air pressure) and gross plant heat rate as the dependent variable. The fixed effects model controls for plant-specific operational characteristics, so it detects changes in gross heat rate relative to a generator’s existing trend. The temperature coefficients are presented in Table S24, with the marginal changes indicated by our model in the final column.

Table S24. Coefficients on temperature in linear regression where meteorological conditions are independent variables and gross heat rate is dependent variable. All plants are coal-fueled with recirculating cooling systems. 2012 is omitted due to data gaps.

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
|  | 2015 | 2014 | 2013 | 2011 | 2010 | IECM-based regression model  (Percentage-based) |
| Temperature Coefficient | -1.47 | 5.99 | 4.08 | 5.28 | 6.38 | 0.00000092 |
| R-squared | 0.010 | 0.0084 | 0.0091 | 0.0063 | 0.0026 | 0.0105 |

The coefficients for empirical data demonstrate absolute heat rate changes, rather than relative, so the numbers above suggest minimal heat rate change, as does our regression model. Likewise, the R-squared values are well below satisfactory. There are numerous complications that may contribute to a poor model fit. Weather stations may be too far from power plants to precisely match ambient conditions at the location of the cooling tower. The weather data and power plant data may not perfectly align in timing either, if one measurement refers to the beginning of an hour and another measurement refers to the end of an hour. Furthermore, data gaps may be biasing the samples.

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