Investigating the Effects of Flue Gas Injection, Hot Water Distribution, and Fill Distribution on Natural Draft Wet Cooling Tower Performance

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Acceptance of Senior Honors Thesis

This Senior Honors Thesis is accepted in partial fulfillment of the requirements for graduation from the Honors Program of Liberty University.

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Abstract

Natural draft wet cooling towers (NDWCT) are a common method of heat removal in powerplants. This study employs a numerical cooling tower model developed by Eldredge, Benton & Hodgson (1997) to examine whether utilizing nonuniform water, fill profiles, and flue gas injection can improve NDWCT efficiency. The results show that each of these variables can be optimized to lower outlet water temperature. Within the range tested for each parameter, the water profile had the most significant effect on outlet water temperature, followed by the flue gas temperature and then the fill profile. The optimum parameter combination reduced the predicted outlet water temperature by 0.5 °C which corresponds to annual fuel savings of up to 55 million dollars and 1.83 million metric tons of carbon dioxide for fossil plants in the United States.

Keywords: flue gas injection, outlet water temperature, fill profile, water profile

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Introduction

Cooling Tower Overview

Natural draft cooling towers are the most common method of heat removal utilized by powerplants to comply with thermal pollution regulations in the United States (Cooper, 2008). Natural draft wet cooling towers (NDWCT) are fundamentally large heat exchangers, in which heat transfer is driven by natural convection. The tower cools hot water from the power-plant's steam condenser and allows the water to be reused.

There are two broad categories of natural draft cooling towers: wet-cooling towers and dry-cooling towers. In natural draft wet cooling towers, the hot water is in direct contact with the atmospheric air flowing through the tower. For dry-cooling towers, the hot water is cooled using an air-water heat exchanger. Wet cooling towers have better heat transfer, since the water is in direct contact with the convective air. However, dry-cooling towers are advantageous where the water supply is limited (such as in desert environments), since dry-cooling towers eliminate the evaporation of the cooling water. Wet and dry cooling towers are both quite large, reaching up to 220 meters in height (Kilmanek, Cedzick & Bialecki, 2015). NDWCT utilize crossflow and counterflow configurations. The difference between the two types involves how the cooling fill is oriented with respect to the flowing air. In the US, all new NDWCT built after 1980 are counterflow towers due to superior performance and better thermal efficiency (Cooper, 2008). Consequently, this research focuses on counterflow cooling towers.

The general operation of a NDWCT is shown in Figure 1. Hot water from the steam condenser enters the tower and is distributed via a grid of spray nozzles. This distribution is intended to be uniform across the tower. However, in practice, depending on the nozzle spray pattern, nozzle locations, and nozzle functionality, many NDWCT operate with nonuniform water distributions. The water then passes through the fill region, which is composed of a porous material that further increases the surface area of the water droplets. As heat transfer occurs between the hot water and the air in the tower, the water is cooled and the air humidity and temperature increase while the density decreases (Kilmanek et al., 2015). This results in a buoyant force that drives the convective flow through the tower. Moist, warm air rises up the tower and exits the outlet, while ambient air is drawn into the tower from openings at the base. Approximately 1 to 2% of the hot water evaporates. The water is cooled and collected in a water basin at the tower base. The water is then piped out of the tower for re-use or disposal.

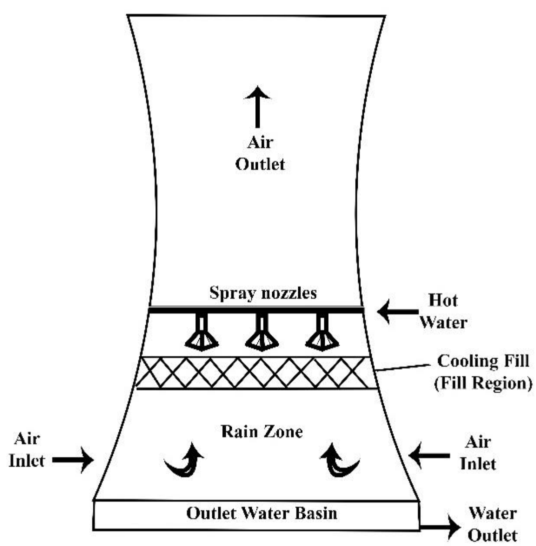

Figure 1. Natural draft wet cooling tower (NDWCT) schematic. This figure depicts the components of a NDWCT.

Cooling Towers and the Rankine Cycle

The power cycles in many powerplants can be modeled as a simple or modified Rankine cycle. Therein, the cooling tower functions as the heat sink, and the temperature of the outlet water corresponds to the temperature of the heat sink. It can be shown that decreasing this temperature improves the efficiency of the cycle. NDWCT are typically 100-200 meters tall and 50-150 meters in diameter at the base. Modest NDWCT circulate water at a rate of about 7,500 kg/s, while many towers circulate water at over twice this flow. At flowrates of 7,500 kg/s, cooling water by an additional 1°C requires additional 28 MJ/s of heat removal.

Because of the volume of typical operations, variations in outlet water temperature of several degrees can result in significant savings (or increased costs) due to the effect on the power cycle efficiency or heat rate. Ibrahim, Ibrahim, and Attia (2014) conducted a thermodynamic analysis of a nuclear power plant equipped with a cooling tower. They predicted that a reduction in outlet water temperature of 1°C results in a 0.16% improvement of the Rankine cycle efficiency. They also found that the change in cycle efficiency and the outlet water temperature had a linear relationship. Other literature supports their results (Durmayaz & Sogut, 2005). Using Ibrahim et al.'s results, the authors predict that a 0.5 °C reduction in outlet water temperature (the maximum reduction predicted in this study) could save $55,000,000 in fuel costs and lower carbon dioxide emissions by 1.83 million metric tons every year in the United States. This is without including the savings possible from eliminating flue gas reheating which is described below.

Flue Gas Injection

In many countries, increasingly stringent environmental regulations require that exhaust gases from coal fired power plants undergo deep desulphurization before introduction into the atmosphere (Kilmanek et al., 2015). This process also cools the flue gas to temperatures between 29-82 °C. At these temperatures, the flue gas is not hot enough to sustain sufficient flow using tall stacks. In such cases, the flue gas must be reheated. To avoid these costs, several dozen plants (mostly in Europe) have experimented with injecting the flue gas into the chimney of NDWCT. This alternative requires additional investment in construction and a corrosion resistant coating to the tower's inner surface but may be the most cost-effective option. Han, Liu, Chen, and Yang (2009) conducted an economic analysis of a 300MW fossil powerplant in China and predicted that flue gas injection could save the plant $600,000 per year simply by eliminating reheating cost. However, research suggests that flue gas injection also has the potential to alter cooling tower performance (i.e. outlet water temperature) (Eldredge, Benton & Hodgson, 1997; Kilmanek et al., 2015; Jahangiri & Golneshan, 2011).

Radial Water and Fill Distributions

One difficulty with optimizing NDWCT performance is that heat transfer across the tower is nonuniform in the radial direction. This occurs because air rising through the center of the tower travels further through the rain zone than the air rising near the tower walls (air cannot rise around the fill since the fill extends to the tower walls). Thus, the air at the center of the tower undergoes more heat and mass transfer. This air has acquired some moisture and is partially heated upon reaching the middle of the tower. Cooling effectiveness near the center of the tower is reduced as a result.

One possible method of countering this situation is to radially vary the water and/or fill distribution in the cooling tower such to bias some of the cooling load from the center of the tower to the periphery. A few numerical studies (Eldredge, et al., 1997; Kilmanek et al., 2015; Pierce, 2007; Reuter, 2010; Smrekar, Oman & Širok, 2006; Williamson, Armfield & Behnia, 2008) have examined nonuniform water and fill distributions, but experimental data from cooling towers employing such systems is lacking in the literature.

Literature Review

Given the scale of operations and potential economic savings from improved cooling tower performance, a substantial amount of research has been devoted to improving the efficiency of NDCT. This research generally falls into one of three categories:

1) Mathematically and numerically modeling cooling towers.
2) Investigating the performance of cooling tower components (fill type, entrance height, spray nozzle type, etc.).
3) Examining the effect of the environment on tower performance.

Despite the large amount of research on cooling towers, investigations into the effects of flue gas injection, fill distribution, and water distributions are limited. The existing literature straddles categories 1 and 2 by using computational fluid dynamics (CFD) or numerical models to simulate cooling tower performance with flue gas injection or variable fill and water distributions. To date, no data is available in the literature to rigorously validate the predictions of flue gas injection. Very little data has been published on the effects of water distribution (Mayur, 2014; Smrekar, et al., 2006).

Effects of Flue Gas Injection

Eldredge et al. (1997) considered the effects of flue gas flow rate, flue gas temperature, radial injection location, injection orientation and liquid entrainment on the performance of NDWCT with flue gas injection. They found that flue gas temperature had the strongest effect on tower performance. Higher flue gas temperatures increased overall buoyancy of air in the tower resulting in increased airflow and enhanced cooling.

Han et al. (2009) conducted an economic evaluation of a 300 MW power plant equipped with a cooling tower with flue gas injection. They predicted significant economic savings by using flue gas injection to eliminate reheating costs but did not examine the effect of flue gas injection on tower efficiency or operating cost. Kilmanek et al. (2015) developed a 3D ANSYS Fluent model of a NDWCT with flue gas injection. They found that the mass flow rate of flue gas in the cases they tested was too low to significantly alter outlet water temperature. Similarly, Jahangiri and Golneshan (2011) employed a 3D CFD model of a dry cooling tower and flue gas injection. They predicted only 0.07 °C reduction in outlet water temperature.

Effects of Varied Fill and Water Profiles

It has been well established that heat and mass transfer can be nonuniform across NDWCT. Yet minimal attention has been given to optimizing cooling tower performance by varying the water and fill distributions to correct for these nonuniformities. The few studies that examine this field are summarized below.

In an experimental study, Smrekar et al. (2006) measured a 9 °C increase in outlet water temperature between the outer edge and the center of a cooling tower (Smrekar et al., 2006). The air velocity profile was also measured and found to be nonuniform and varied from 2.2 m/s at the tower's center to 3.8 m/s on the periphery (almost double the centerline value). Their results suggest three regions of differing air velocity. The air velocity is strongly coupled to cooling via latent heat transfer. Consequently, each of the three regions has a different cooling load. The average outlet-water temperature leaving the tower was a mass-weighted-average of the water temperatures from these regions. Smrekar et al. (2006) used a numerical model to vary the mass flow rate of the water across the tower to achieve a constant water/air mass flow ratio [kg-H20-s^{-1} / kg-air-s^{-1}] across the tower. This resulted in a constant outlet water temperature which minimized entropy generation and exergy destruction from mixing. Smrekar et al. (2006) predicted a 1.6 °C improvement in outlet water temperature from these optimizations.

Williamson, Behnia, and Armfield (2008) used an axisymmetric tower model to predict a linear increase in outlet water temperature across the cross section of a cooling tower. They found a maximum radial variation of 6 °C in outlet water temperature. Their model also predicted that the outer 7 meters of the radial fill (with a tower radius of 46.8 meters) delivered over 30% of the cooling load, while the inner 15 meters delivered less than 10%. These trends agree well with Smrekar et al.'s (2006) measured values. However, Williamson, Behnia, and Armfield (2008) predicted a mostly uniform mass flow rate of air across the tower with a sharp decrease in mass flow rate of air in a narrow region against the tower wall. This profile is different from the velocity profile measured by Smrekar et al. (2006).

Williamson, Behnia, and Armfield (2008) also tested three uniform fill depths (0.6 m, 0.9 m, and 1.2 m). They found that once a sufficient fill depth was implemented, further increases in fill depth did not produce significant change in the non-uniformity of the air flow, the outlet water temperature, or the cooling load distribution. They suggested that due to these nonuniformities, future work should focus on optimizing tower performance by introducing annularly varying fill depth and water distribution profiles.

In subsequent research, Williamson, Armfield, and Behnia (2008) used a 1D zonal model (validated against the axisymmetric CFD model used in his previous work) to predict that the optimal fill and water distribution profiles vary significantly from the uniform profiles. However, their model showed a maximum improvement in outlet water temperature of only 0.04 °C. They suggested that the nonuniform water and fill profiles offset any potential improvement in outlet water temperature by increasing flow resistance in the regions with increased fill depth. They noted that detailed validation of their 1D zonal model was not possible due to the lack of published experimental data describing the performance of cooling towers with nonuniform water and fill depth profiles.

Reuter (2010), constructed an axisymmetric cooling tower model in Fluent as part of his dissertation. He varied fill and water profiles and concluded that the fill profile had a more influential effect on outlet water temperature. By dividing the fill profile into two annular regions and varying the height of each region while maintaining the same fill volume, he predicted a 0.5 °C drop in outlet water temperature (a 2.2% change in tower cooling). Reuter also varied the water profile but found negligible improvement (0.01 °C), although he found that the improvements from varied fill and water profiles were additive.

To date, there are very few published experimental results available in the literature describing the effects of nonuniform water and fill profiles in NDWCT. One paper has been published by Mayur (2014) which experimentally examined nonuniform water profiles. The paper has a few numerical inconsistencies that may be typographical errors. The English is also poor, but Mayur's work still merits examining as it is one of the only experimental studies in the literature. Mayur's study (2014) is similar to Smrekar et al.'s (2006) work. He experimentally measured air velocity and cold-water temperature across a tower and then divided the tower cross section into three annular sections. He then varied the water distribution in these three regions to obtain a uniform water/air mass flow rate ratio. He modified the spray nozzle diameters in these three regions in a physical tower to approximate the ideal water distribution calculated. Over a four-hour testing period, the tower's outlet water temperature decreased by 1.0 °C and improved the tower efficiency by 7.33% (Efficiency = Approach / [(Range + Approach]). This is a significantly larger reduction in outlet water temperature than predicted by Williamson, Armfield, and Behnia (2008) or Reuter (2010), but still less than that predicted by Smrekar et al. (2006). A summary of results from the relevant literature is shown in Table 1.

Table 1

Summary of relevant literature examining the effects of varied fill and water profiles on outlet water temperature

Author	Method	Outlet water improvement from varying fill distribution [°C]	Outlet water improvement from varying water distribution [°C]	Conclusions
Smrekar et al. (2006)	Axisymmetric numerical model with a uniform water/air mass flow ratio.	N/A	1.6	Nonuniform water distribution has a significant effect on cooling
Williamson, Armfield, and Behnia (2008)	Axisymmetric numerical model coupled with a 1D zonal model.	0.04 combined		Optimal water and fill profiles are nonuniform but result in negligible improvement due to increased flow resistance.
Reuter (2010)	Axisymmetric Fluent Model.	0.5	0.01	Varied fill distribution has a significant effect on cooling. The effect of the varied water distribution is negligible but additive with improvements from the varied fill profile.
Mayur (2014)	Experimental study. Measured air temperature and velocity across tower then optimized the water/air mass flow ratio using three discrete annular regions.	N/A	1.0	Varied the water distribution across the tower with minimal expenditure and labor. Produced a significant improvement in cooling.

Summary

Overall, the scarcity of literature examining the effects of non-uniform fill profiles and non-uniform water distributions on NDWCT performance, and the conflicting results of existing studies demonstrates a need for additional research to quantify their effects. Furthermore, little literature exists examining the relationship between flue gas injection and water and fill distributions on NDWCT performance. This analysis employs the axisymmetric cooling tower model developed by Eldredge et al. (1997) to examine the dependency of the outlet water temperature on these variables. The possible savings that can be achieved by optimizing these parameters is also quantified.

Procedure

Five sets of cases were run as described below; see Table 2 for a summary:

1. Cases were run with a varied water distribution and no flue gas injection. The maximum improvement in outlet water temperature was quantified.
2. Cases were run in which the water and fill distribution were varied simultaneously, and flue gas injection was not included. The maximum improvement in outlet water temperature was quantified.
3. Cases were run with uniform water and fill distributions and flue gas injection. The flue gas temperature was varied within a reasonable range (32-66 °C) and the maximum outlet water temperature quantified.
4. Cases were run in which the water and fill distribution were varied simultaneously and hot (relative to the air temperature) flue gas was injected into the tower. The maximum improvement in outlet water temperature was quantified.
5. Cases were run in which the water and fill distribution were varied simultaneously and cool (relative to the air temperature) flue gas was injected into the tower. The maximum improvement in outlet water temperature was quantified.

Table 2

Procedure summary

Case	Varied Water Distribution	Varied Fill Distribution	Hot Flue Gas Injection	Cool Flue Gas Injection	Varied Flue Gas Injection
1	Yes	No	No	No	No
2	Yes	Yes	No	No	No
3	No	No	No	No	Yes
4	Yes	Yes	Yes	No	No
5	Yes	Yes	Yes	Yes	No

Nomenclature

C	Mass fraction of water vapor in the gas phase
C_w	Liquid water specific heat
$\vec{F}$	Drag force resulting from various flow resistances in the cooling tower
$\vec{g}$	Gravity vector
$\dot{M}_{ev}$	Volumetric evaporation rate of liquid water
S_p	Volumetric rate of energy exchange between the two phases
T_w	Liquid water temperature
$\vec{V}$	Velocity vector for the gas phase mixture
$\vec{V}_w$	Velocity vector for the liquid stage
Γ_{1e}	Effective diff Armfield, and Behnia (2008) usion coefficient for water vapor
Γ_{2e}	Effective diffusion coefficient for energy in gas phase (incorporates molecular and turbulent diffusion)
ρ	Gas phase mixture density using ideal gas assumptions
ρ_a	Air density computed using ideal gas equation of state (the R value was computed using the air-water vapor mixture molecular weight)
ρ_w	Liquid water density
τ_{ij}	Stress tensor for the gas phase momentum equation
σ_d	Surface tension ignoring functionality with temperature
ϕ	Enthalpy of the gas phase mixture

Model Description

The numerical model employed in this study is an axisymmetric model developed by Eldredge et al. (1997). An axisymmetric assumption was used because ideal NDWCT designs are symmetric (assuming no crosswind outside the tower). A decoupled 2D Lagrangian model was run for the droplets, using an approximate gas velocity field in the rain zone. From this Lagrangian model, a relation for mass transfer coefficient was developed for the rain zone and then used in the gas phase CFD model to predict evaporation rate in the rain zone. Somewhat similarly, the Lagangian model was used to compute momentum sink terms caused by the drops which were used in the gas phase momentum equations. The model employs an orthogonal boundary-fitted grid as shown in Figure 2.

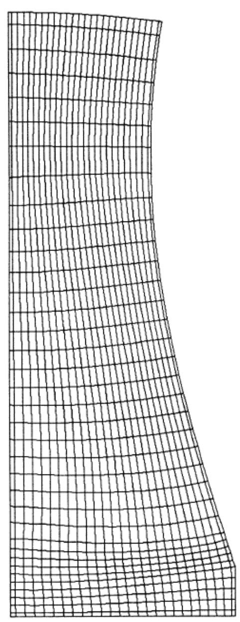

Figure 2. Orthogonal boundary-fitted cooling tower grid. This figure shows the axisymmetric grid used for the cooling tower model. Eldredge, T. V., & Stapleton, J. S. (2019, July 18-21). *Investigating the effects of flue gas injection and hot water distribution and their interaction on natural draft wet cooling tower performance* [Conference session]. Power Conference and Nuclear Forum, Salt Lake City, Utah. Reprinted with permission.

Six equations govern the heat and mass transfer in/through the tower. These equations are:

Continuity equation for the gas phase mixture

$$\vec{\nabla} \cdot \left(\rho \vec{V}\right) = \dot{M}_{ev} \quad \text{(Eq 1.)}$$

Momentum equation for the gas phase mixture with 1-way coupling for momentum

$$\vec{\nabla} \cdot \left(\rho \vec{V} \vec{V}\right) = \vec{\nabla} \cdot \tau_{ij} + \rho \vec{g} + \vec{F} \quad \text{(Eq. 2)}$$

Continuity equation for water vapor (moisture transport equation)

$$\vec{\nabla} \cdot \left(\rho \vec{V} \mathrm{C} - \Gamma_{1\mathrm{e}} \vec{\nabla} C\right) = \dot{M}_{ev} \quad \text{(Eq. 3)}$$

Energy equation with 2-way coupling for the gas phase mixture

$$\vec{\nabla} \cdot \left(\rho \vec{V} \phi - \Gamma_{2\mathrm{e}} \vec{\nabla} \phi\right) = S_p \quad \text{(Eq. 4)}$$

Continuity equation for liquid water (neglecting the loss of liquid water)

$$\vec{\nabla} \cdot \vec{V}_w = 0 \quad \text{(Eq. 5)}$$

Energy equation for liquid water (neglecting diffusion)

$$\vec{\nabla} \cdot \left(\rho_w C_w T_w \vec{V}_{\mathrm{w}}\right) = -S_p \quad \text{(Eq. 6)}$$

These equations are solved simultaneously in two dimensions to provide the desired outputs. This model was validated on two NDWCT with the data provided by the Tennessee Valley Authority. The towers do not have flue gas injection, so the flue gas injection modeling could not be vigorously validated.

All the cases tested in this study used a water mass flow rate of 18,100 kg/s. To represent the radial water distribution in the cooling tower a bias parameter (β) was used, where β is defined as follows:

$$\beta = \frac{\int r\, dm}{\left[\int r\, dm\right]_{\mathrm{uniform}}}$$

where *r* is the radial distance from the center of each region, and *dm* is the fraction of the total water flow rate distributed in that region. $\beta = 1$ represents a uniform radial water distribution. For $\beta < 1$ the water is distributed more towards the center of the tower, while for $\beta > 1$ the water is distributed more toward the periphery. The model developed by Eldredge et al. (1997) and employed in this study uses 5 equal regions for varying water and fill distributions. These regions are referred to as region 1, region 2, etc., where region 1 is at the center of the tower and region 5 is at the tower wall.

The variable λ was also developed and is analogous to β but corresponds to the weighting of the fill volume. We define λ as follows:

$$\lambda = \frac{\int r\, dV}{\left[\int r\, dV\right]_{\mathrm{uniform}}}$$

dV is the fraction of the total fill volume distribution in that region. $\lambda = 1$ represents a uniform fill profile. CFD setup and convergence information may be found in the Appendix.

Results and Analysis

Varied Water Distribution with No Flue Gas Injection

First, a base case (Base.1) was run with uniform water and fill distributions and no flue gas injection. Eleven additional cases were run in which the water distribution was varied to find the most effective water distribution (characterized by the lowest average cold-water temperature). The results are listed in Table 3. The lowest outlet water temperature was obtained in case A.6 and resulted in a temperature decrease of 0.32 °C from the base case.

Table 3

Results from case set 1

Case	Region 1	Region 2	Region 3	Region 4	Region 5	β	Temp. Decrease From Uniform Dist. [°C]
Base.1	0.2	0.2	0.2	0.2	0.2	1.00	N/A
A.1	0.2	0.25	0.25	0.15	0.15	0.93	-1.83
A.2	0.15	0.15	0.2	0.25	0.25	1.12	0.14
A.3	0.12	0.15	0.23	0.24	0.26	1.16	-0.11
A.4	0.15	0.175	0.2	0.225	0.25	1.10	0.29
A.5	0.15	0.19	0.21	0.22	0.23	1.08	0.30
A.6	**0.16**	**0.18**	**0.2**	**0.22**	**0.24**	1.08	**0.32**
A.7	0.17	0.18	0.2	0.22	0.23	1.06	0.31
A.8	0.18	0.19	0.2	0.21	0.22	1.04	0.24
A.9	0.185	0.19	0.2	0.21	0.215	1.03	0.21
A.10	0.22	0.23	0.21	0.18	0.16	0.94	-0.95
A.11	0.2	0.22	0.21	0.19	0.18	0.97	-0.37

Outlet water temperature was found to be relatively sensitive to changes in water distribution, but fractional water distributions in the regions were not modified beyond a tenth of a percent for the sake of realism. Improvements to the base case could be obtained by changing the fraction of the water distribution in each of the regions within the range 0.15-0.25. Modifications outside of this range generally provided worse cooling tower performance due to increased impedance to air flow created by the higher water density. As expected, for $\beta > 1$ the outlet water temperature decreased while for $\beta < 1$ the outlet water temperature increased. The results are parabolic and have a clear minimum outlet water temperature at $\beta = 1.08$ as shown in Figure 3.

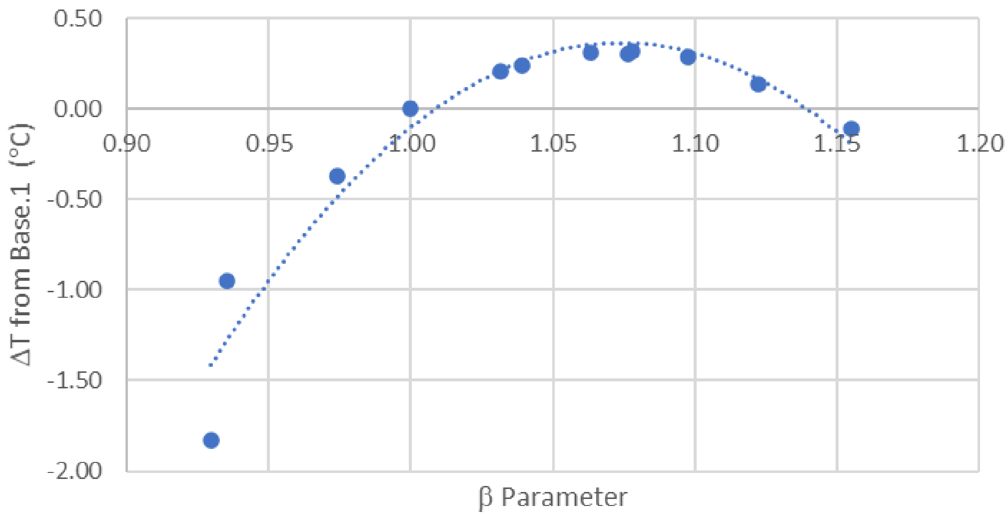

Figure 3. Outlet water temperature with respect to β. This figure shows the relationship between the outlet water temperature and the β parameter for case set 1.

Varied Fill Distribution and Varied Water Distribution with No Flue Gas Injection

Cases were run varying the water and fill distribution simultaneously with no flue gas injection. The water and fill distribution profiles were the same for each case, since increased water flow requires increased fill volume to increase the surface area of the water and improve heat transfer. Results are shown in Table 4.

Table 4

Results from case set 2

Case	Region 1	Region 2	Region 3	Region 4	Region 5	β and λ	Temp. Decrease From Uniform Dist. [°C]
Base.1	0.2	0.2	0.2	0.2	0.2	1.00	N/A
C.1	0.2	0.25	0.25	0.15	0.15	0.93	-2.19
C.2	0.15	0.15	0.2	0.25	0.25	1.12	0.00
C.3	0.12	0.15	0.23	0.24	0.26	1.16	-0.28
C.4	0.15	0.175	0.2	0.225	0.25	1.10	0.28
C.5	**0.15**	**0.19**	**0.21**	**0.22**	**0.23**	1.08	**0.36**
C.6	**0.16**	**0.18**	**0.2**	**0.22**	**0.24**	1.08	**0.36**
C.7	**0.17**	**0.18**	**0.2**	**0.22**	**0.23**	1.06	**0.36**
C.8	0.18	0.19	0.2	0.21	0.22	1.04	0.30
C.9	0.185	0.19	0.2	0.21	0.215	1.03	0.25
C.10	0.22	0.23	0.21	0.18	0.16	0.94	-1.23
C.11	0.2	0.22	0.21	0.19	0.18	0.97	-0.45

The maximum outlet water temperature decrease from Base.1 was 0.36 °C and was obtained in cases C.5, C.6, and C.7. The β and λ values for the best cases were in the range 1.06-1.08 and the relationship to the change in outlet water temperature is parabolic as shown in Figure 4.

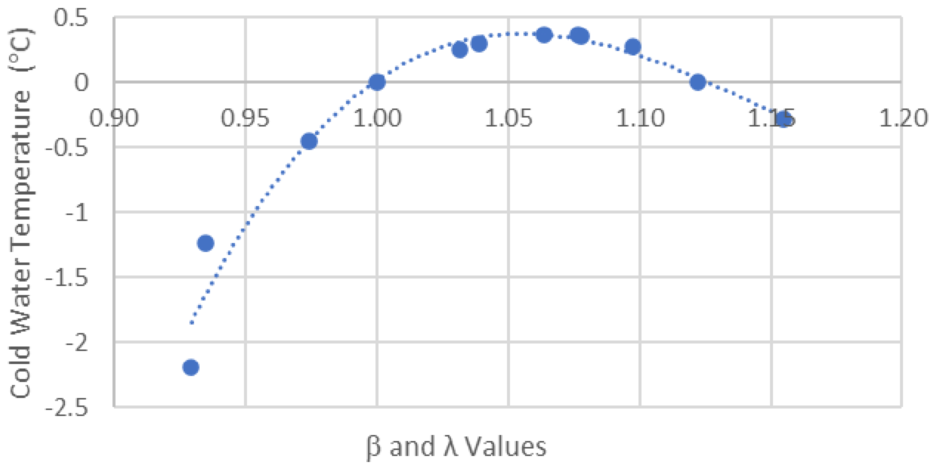

Figure 4. Outlet water temperature with respect to β and λ. This figure shows the relationship between the outlet water temperature and the β and λ parameters for case set 2.

Varying the water and fill profiles together cooled the water 0.04 °C lower than when only the water distribution was varied. In practice, varying the fill distribution in addition to the water distribution may not be cost effective due to very incremental improvement. However, varying the fill in addition to the water distribution lowers the model's sensitivity to the β and λ parameters, which would be favorable in implementation, since this would reduce the risk of accidently implementing a detrimental water distribution profile.

Effects of Flue Gas Temperature with Uniform Water and Fill

Flue gas temperatures from about 32 to 66 °C are typically encountered in industry (Eldredge et al., 1997). Seven cases were run in which flue gas temperature was varied independently of the other variables. The injected flue gas flow rate was approximately 18% of the air flow rate through the tower. Water and fill distributions were uniform. The results are listed in Table 5. The case with the lowest outlet water temperature is shown in bold.

Table 5

Results from case set 3

Temperature [°C]			
Flue Gas Temp.	Outlet water Temp.	Base Case Water Temp. (no flue gas)	Decrease in Water Temp.
32.2	32.6	31.9	-0.66
37.8	32.3	31.9	-0.34
43.3	32.1	31.9	-0.15
48.9	31.9	31.9	-0.03
54.4	31.9	31.9	0.04
60.0	31.8	31.9	0.10
65.6	**31.6**	**31.9**	**0.16**

Predictions show that outlet water temperature is inversely related to the flue gas temperature. The warm flue gas increases the overall buoyancy of air in the tower which increases convective airflow and cooling in the tower (Eldredge et al., 1997). There is a break-even flue gas temperature (approximately 50 °C), where the flue gas does not increase or decrease the outlet water temperature. This is depicted in Figure 5, which compares outlet water temperature from the uniform case with no flue gas (Base.1) to the cases with flue gas injection. Figure 5 also shows that lowering the flue gas temperature has a decreasing marginal rate of return.

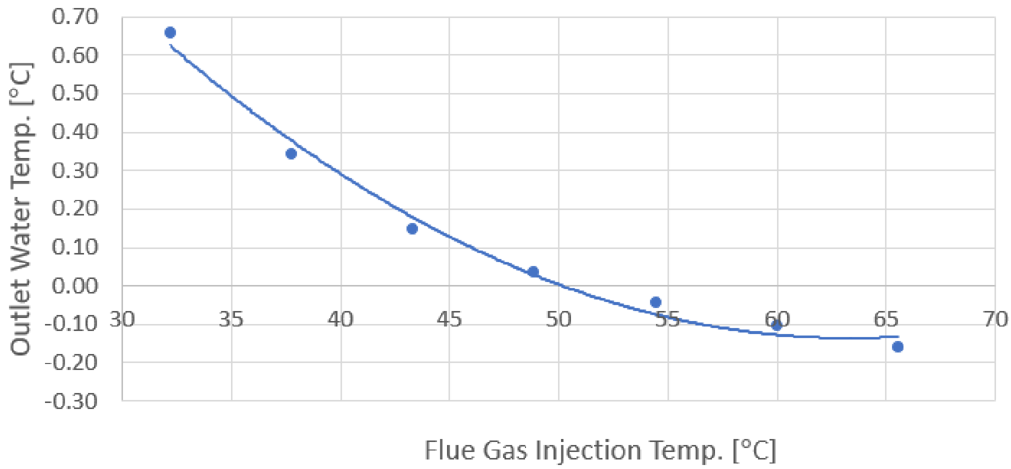

Figure 5. Variation of outlet water temperature with flue gas temperature. This figure shows the relationship between the outlet water temperature and the flue gas injection temperature.

Varied Water and Fill Distribution with Hot Flue Gas Injection

Ten cases were run with hot (65.6 °C) flue gas injection and varied water and fill distributions. The β and λ were equal in each case. The outlet water temperature was compared to the results from a base case (Base.2) with uniform water distribution and hot flue gas injection. Results are tabulated in Table 6.

Table 6

Results from case set 4

Case	Region 1	Region 2	Region 3	Region 4	Region 5	β and λ	Temp. Decrease From Base.2 [°C]
Base.2	0.2	0.2	0.2	0.2	0.2	1.00	N/A
D.1	0.15	0.15	0.2	0.25	0.25	1.12	0.37
D.2	0.15	0.175	0.2	0.225	0.25	1.10	0.49
D.3	0.15	0.19	0.21	0.22	0.23	1.08	0.46
D.4	**0.16**	**0.18**	**0.2**	**0.22**	**0.24**	**1.08**	**0.50**
D.5	0.17	0.18	0.2	0.22	0.23	1.06	0.46
D.6	0.18	0.19	0.2	0.21	0.22	1.04	0.35
D.7	0.185	0.19	0.2	0.21	0.215	1.03	0.30
D.8	0.22	0.23	0.21	0.18	0.16	0.94	-0.28

The maximum reduction in outlet water temperature (0.50 °C) was obtained in case D.4. Case D.4 has the same distribution as case C.6 which was the best distribution for the cases without flue gas injection. Thus, the same water and fill distribution optimized the cooling tower with and without hot flue gas injection. Furthermore, comparing Table 6 and Table 4 reveals that the outlet water temperature is more sensitive to the water and fill distribution when hot flue gas is included than in cases without hot flue gas. The relationship between β and λ and the outlet water temperature is shown in Figure 6. The relationship is parabolic.

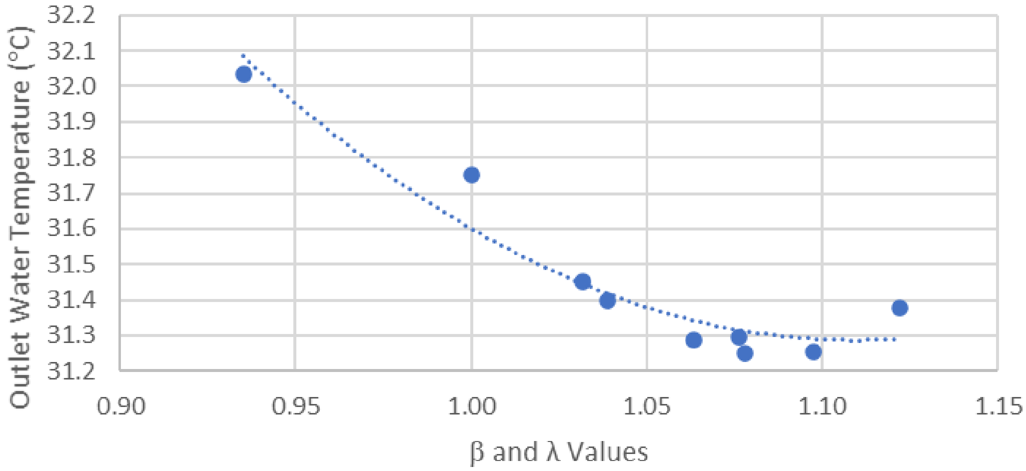

Figure 6. Outlet water temperature with hot flue gas injection. This figure illustrates the relationship between β and λ and the outlet water temperature with hot flue gas injection.

Varied Water and Fill Distribution with Cool Flue Gas Injection

Ten cases were run with cool (32.2 °C) flue gas injection. The water and fill distributions were varied together. Outlet water temperature was compared to base case (Base.3) in which cool flue gas was injected with uniform water and fill distributions. Results are listed in Table 7.

Table 7

Results from case set 5

Case	Region 1	Region 2	Region 3	Region 4	Region 5	β and λ	Temp. Decrease From Base.3 [°C]
Base.3	0.2	0.2	0.2	0.2	0.2	1.00	N/A
E.1	0.15	0.15	0.2	0.25	0.25	1.12	-0.18
E.2	0.12	0.15	0.23	0.24	0.26	1.16	-0.37
E.3	0.15	0.175	0.2	0.225	0.25	1.10	0.13
E.4	**0.15**	**0.19**	**0.21**	**0.22**	**0.23**	1.08	**0.26**
E.5	0.16	0.18	0.2	0.22	0.24	1.08	0.23
E.6	0.17	0.18	0.2	0.22	0.23	1.06	0.24
E.7	**0.18**	**0.19**	**0.2**	**0.21**	**0.22**	1.04	**0.26**
E.8	0.185	0.19	0.2	0.21	0.215	1.03	0.19
E.9	0.22	0.23	0.21	0.18	0.16	0.94	-0.93
E.10	0.2	0.22	0.21	0.19	0.18	0.97	-0.36

The maximum reduction in outlet water temperature from Base.3 was 0.26 °C and was obtained in cases E.4 and E7. The distribution in E.4 is the same as the distribution in C.5 which was one of the optimum distributions for the cases with no flue gas injection (see Table 4). Comparing Table 4 and Table 7 reveals that the outlet water temperature was less sensitive to the water and fill distributions when cool flue gas injection was included than in the cases without flue gas. The outlet water temperature was also less sensitive to the water and fill distributions than in the cases with hot flue gas injection. The relationship between β, λ, and outlet water temperature for the cases with cool flue gas injection is shown in Figure 7. Although the ideal β and λ values are identical (1.08) for cases with hot and cool flue gas injection.

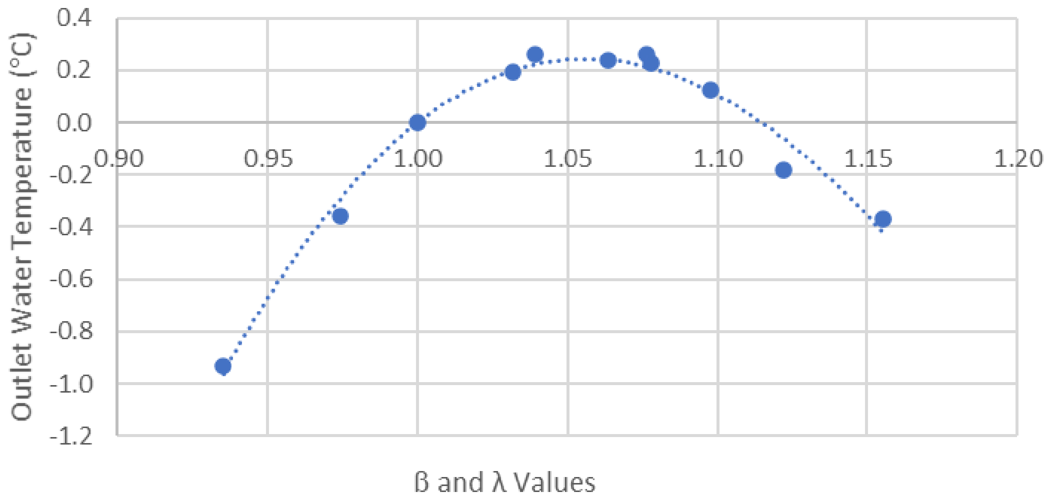

Figure 7. Outlet water temperature with cool flue gas injection. This figure illustrates the relationship between β and λ and the outlet water temperature with cool flue gas injection.

Conclusions

This thesis examined the interactions of flue gas injection and varied water and fill distributions on the outlet water temperature in NDWCT. The following are the primary conclusions from the cooling tower simulation predictions:

- The single parameter which had the dominant effect on outlet water temperature was the water distribution. When varied independently, the ideal water distribution reduced the outlet water temperature by 0.32 °C without flue gas injection.
- The results suggest that the positive effects of injecting hot flue gas (65.6 °C) on outlet water temperature can be enhanced by radially biasing the water and fill together (increasing water concentration with increasing radius). Hot flue gas with varied water and fill distributions resulted in the greatest drop in outlet water temperature (0.50 °C).

- By radially biasing the water and fill together, such that the water and fill increased with increasing radius, a maximum decrease of approximately 0.36 °C was predicted in outlet water temperature with respect to uniform water and fill distributions. Therefore, biasing the fill with the water increased the reduction in outlet water temperature from 0.32 to 0.36 °C. These calculations were performed without flue gas injection. See Table 3.
- The results suggest that the negative effects of injecting lower temperature flue gas (32.2 °C) on outlet water temperature can be partially mitigated by radially biasing the water and fill together (increasing distributions with increasing radius). Optimizing the fill and water distributions resulted in 0.26 °C reduction in outlet water temperature.
- The effects of flue gas injection and radially biasing the water and fill distributions are somewhat additive.

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Appendix: Cooling Tower Model Validation, Mesh Independence, and Convergence Criteria

Model Validation

The cooling tower model was validated using two NDWCT from the Tennessee Valley Authority. The validation data is shown in Table A.1.

Table A1

Model validation data

	Cooling Water Flow Rate	Predicted Outlet Water Temperature	Measured Outlet Water Temperature
Tower 1	18100 kg/s	31.9 °C	31.7 °C
Tower 2	26600 kg/s	34.2 °C	34.9 °C

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The CFD model assumed axisymmetric air flow, water distribution, and fill distribution. In reality, many towers are asymmetric due to plugged or missing nozzles, damaged fill, and other unknown factors. These effects influence the measure outlet water temperature in Table A1. Additionally, perfect sealing between the fill and the tower shell was assumed, although in an actual tower, poor installation can result in gaps at this interface.

Also, there is some uncertainty involved in the temperature measurement for the outlet water in Table A1. The International Electrotechnical Commission provides equations for the uncertainty of Class A and Class B RTDs (ICE Standard 751). Assuming the more accurate Class A RTDs The uncertainty of the more accurate Class A RTDs is given by the following relation:

$$\text{RTD Uncertainty}[\pm°\text{C}] = \pm(0.15 + 0.002T_w)$$

According to this equation, the RTD measurement for Tower 1 from Table 1 would be at least ± 0.21°C). Considering the uncertainties involved, Eldredge et al. (1997) concluded that being able to predict outlet water temperature to within 0.7 °C for two different cooling towers under different operating conditions provides strong evidence for validation of the cooling tower model used in this analysis.

The ASME V&V20-2009 standard recommends that the simulation error (the difference between the predicted value and the actual value) and the comparison error (the difference between the predicted value and the measured value) be used as a metrics for validation. For Tower 1 the comparison error is 0.2 °C (31.9 – 31.7) and the simulation error is estimated to be ± 0.7 °C. The simulation error (δ_s) is composed of three components as shown:

$$\delta_{\text{s}} = \delta_{\text{model}} + \delta_{\text{num}} + \delta_{\text{input}}$$

The modeling error results from modeling assumptions and approximations, i.e. which equations we choose to solve. The numerical error is associated with error in the numerical recipes, i.e. how we choose to solve those equations. The input error corresponds to inaccuracies in the simulation input parameters. The magnitude of the modeling, numerical, and simulation errors is shown in Table A2.

Table A2

Estimates of simulation error contributions

Error Contribution	Estimated Value
δ_{model}	±0.28 °C
δ_{num}	±0.02 °C
δ_{input}	±0.4 °C
δ_{S}	±0.7 °C

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The modeling error was estimated to be 0.28 °C and is a result of the selected turbulence model, ignoring the effect of water loss due to evaporation in the liquid water velocity field, and the error associated with accurately representing a fill volume with a curvilinear mesh. Each of these errors was quantified in earlier work by Eldredge et al. (1997). The numerical error was estimated to be 0.02 °C and is further explained under the model convergence criteria section. The input error was estimated to be 0.4 °C based on estimates of the possible error in the inputs of the water flowrate and the inlet hot water temperature.

Previous work by Eldredge et al. (1997) demonstrated that the effect of simulation error was shifting the trendlines up or down by some amount. Thus, even though the magnitude of the predicted changes in outlet water temperature are comparable in magnitude to the simulation error, they believe that the predicted trends and magnitudes are reasonably accurate.

Mesh Independence

Additionally, a mesh independence study was conducted for Tower 1 from Table 1. Results are tabulated in Table A3. The cases in this paper utilized the mesh with 999 nodes. While this mesh may seem rough, heat transfer within the tower is dominated by bulk convection. Thus, refining the mesh to account for thermal and velocity boundary layer effects is less important for obtaining an outlet temperature. The mesh selected provides a reasonable balance between computational efficiency and results accuracy.

Table A3

Mesh independence study

Number of Mesh Nodes	Predicted Outlet Water Temperature	Error in Predicted Outlet Water Temperature
713	32.1 °C	1.26%
999	31.9 °C	0.63%
1715	31.8 °C	0.32%
2255	31.9 °C	0.63%
Measured outlet water temperature was 31.7 °C		

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Convergence Criteria

A global mass balance and energy balance across the model boundaries were used to determine convergence. For converged solutions, ANSYS Fluent documentation (ANSYS, 2009) recommends that the normalized mass and energy residuals be less than 0.5%. However, there are better ways to assess convergence. Specifically, point values of temperature or shear rate can be monitored throughout the domain to ensure they are no longer changing. These values typically take longer than the normalized residuals to converge.

For the model used in this analysis, cooled water temperature and air flow rate were monitored, as shown in Figure A2. The normalized residuals for the tower model were used to assess convergence. These values are significantly lower than 0.5% as shown in Figure A1. The mass air flow rate through the tower and the outlet temperature are also seen to be converged in Figure A2. The mass residual is normalized by the mass flow rate of air entering the tower. The energy residual is normalized by the average of the rate of energy acquired by the moist air and the rate of energy given up by the liquid water.

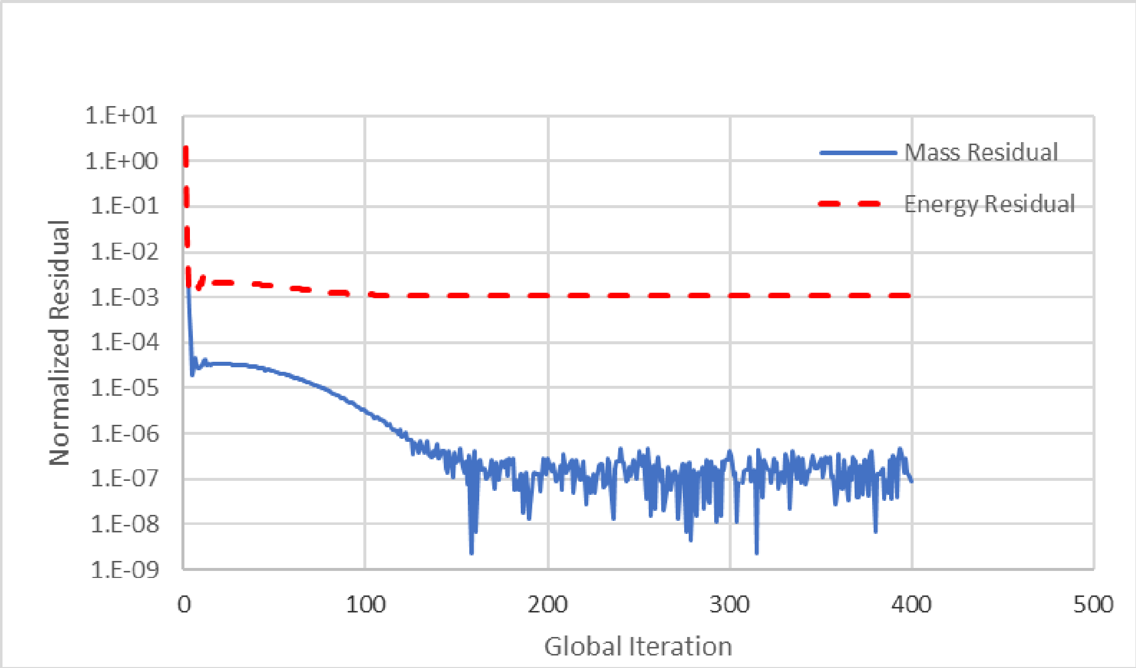

Figure A1. Normalized residuals. This figure shows the normalized mass and energy residuals for the base case. Eldredge, T. V., & Stapleton, J. S. (2019, July 18-21). *Investigating the effects of flue gas injection and hot water distribution and their interaction on natural draft wet cooling tower performance* [Conference session]. Power Conference and Nuclear Forum, Salt Lake City, Utah. Reprinted with permission.

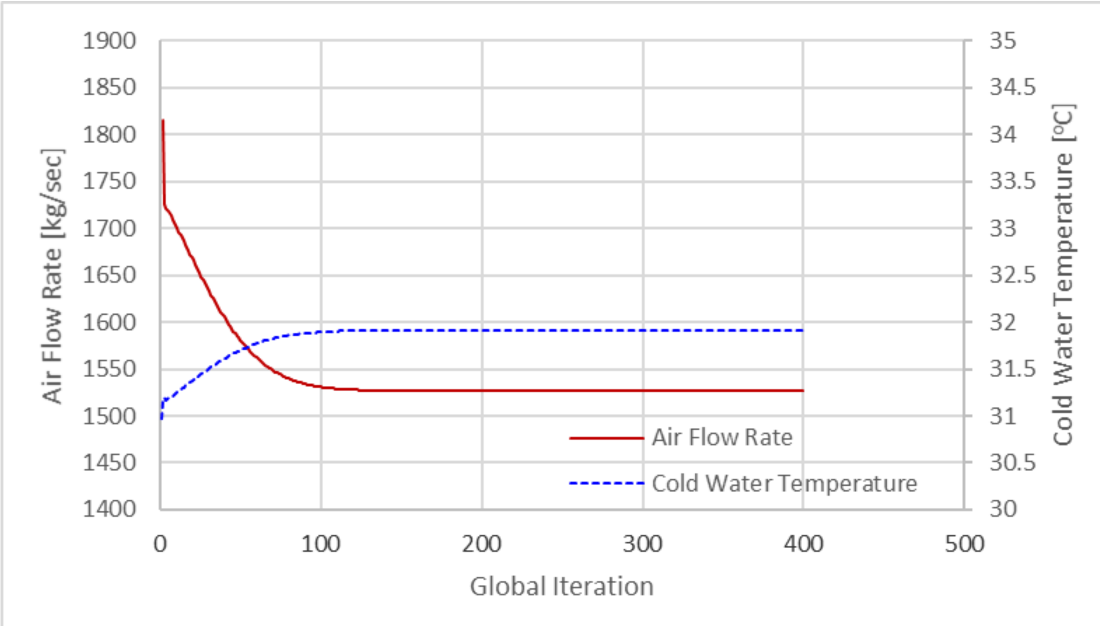

Figure A2. Convergence data. This figure shows the convergence of the predicted air mass flow rate and outlet temperature for the base case. Eldredge, T. V., & Stapleton, J. S. (2019, July 18-21). *Investigating the effects of flue gas injection and hot water distribution and their interaction on natural draft wet cooling tower performance* [Conference session]. Power Conference and Nuclear Forum, Salt Lake City, Utah. Reprinted with permission.