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# Computational Analysis of The Proposed Cooling Tower Installation At Indian Point Units 2 & 3 FINAL REPORT

Alden Report No:
413038-1R1

Submitted to:

Entergy®

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## Introduction

Alden Research Laboratory, Inc was contracted by Entergy to complete a computational fluid dynamic (CFD) model study of the proposed mechanical draft cooling tower installation at Entergy's Indian Point Energy Center, Units 2 and 3. The purpose of the CFD model is to evaluate if, and to what extent any of the high temperature, high humidity cooling tower exhaust is re-circulated into the cooling tower ambient air inlet.

The basic operating principle of a cooling tower is rejection of heat by evaporation. Because these units use ambient air, their performance is influenced by atmospheric conditions such as ambient temperature, ambient humidity, wind direction, and wind speed. The performance may also be influenced by the proximity of surrounding structures and topography.

Because of the proximity of the ambient air inlet to the exhaust, cross winds have been known to cause recirculation of hot, humid exhaust air back to the unit, which results in reduced cooling capability of the unit. The cooling capability of ambient air is generally measured using wet bulb temperature, which represents the lowest temperature that the condenser can possibly reach using only evaporation.

The following report documents the inputs and results of CFD model, particularly the increase in average ambient wet bulb temperature at the inlets to the proposed Unit 2 and Unit 3 cooling towers.

## Model Description

The model geometry was developed using the commercially available three-dimensional CAD and mesh generation software, GAMBIT V2.4.6. The computational domains generated for the model consisted of approximately 3.5 million tetrahedral cells.

Alden used the CFD software package ANSYS-Fluent v14.5 to calculate the full-scale, three-dimensional, incompressible, turbulent flow throughout the domain. A stochastic, two-equation realizable k-ε model was used to simulate the turbulence. Detailed descriptions of the physical models employed in each of the Fluent modules are available from ANSYS-Fluent. CFD solver information is presented in Table 1.

**Table 1 - CFD Solver Information**

| CFD Solver Information: | Units: | Value: |
|---|---|---|
| Mesh Name | | 413038_IndianPoint_Y_K.msh |
| Cell count | | 3,492,306 |
| Cell Shape | | Hexahedral / Tetrahedral |
| CFD Code | | ANSYS-Fluent v14.5 |
| Solver | | Pressure-based Segregated |
| Spacial Discritization | | 2nd Order Upwind |
| Density Formulation | | Incompressible Ideal Gas |
| Turbulence Model | | k-epsilon, realizable |
| Near-Wall Treatment | | Non-equalibrium Wall Functions |

## Model Geometric Parameters

The computational domain was created from topography data provided to Alden by ENERCON via email on 10/04/2013 (170641-00_Base_NAD27Feet-B.dwg). This data was augmented by satellite data taken from Google Earth on 10/07/2013 to fill in building dimensions in the vicinity of the proposed cooling towers.

The cooling towers were created based on dimensions provided in Tetra Tech Report No. 11431116100-REP-R0001-02, and outlined in Table 2. The only dimension that was required for the flow model that was not provided in the report is the ambient air inlet opening height, which was assumed to be 36-ft. The 36-ft opening assumption is from graphics of the ClearSky cooling tower showing three floors within the ambient air inlet, and an assumption of 12-ft of elevation per floor. The proposed cooling towers, required grading, and plant property are colored grey in Figure 1.

Due to the size of the model domain, a few simplifications to the cooling tower were necessary, including:

- Cooling water piping to each cell is not included.
- Railings and small support structure are not included.

- The exhaust stack is approximated as cylindrical, with the correct height, and a diameter equal to that of the fan (28-ft).

These simplifications are conservative, in that their omission is expected to create less hot gas re-entrainment than their inclusion.

The cooling towers were located based on Tetra Tech Report No. 11431116100-REP-R0001-02, Figure 3-3, which is reproduced here as Figure 2. This includes a 150-ft standoff on either side of both cooling towers. This does not interfere with any existing structures for the Unit 2 cooling tower; however there are a few buildings that interfere with the construction of the Unit 3 cooling tower. These buildings have been removed from the model domain, and have not been relocated.

**Table 2 - Cooling Tower Dimensions**

| Dimensions (per Cooling Tower): | Units: | Cooling Tower 2: | Cooling Tower 3: |
|---|---|---|---|
| Model | | F4811D-6.0-44B | F4811D-6.0-44B |
| Number of Cells | | 44 | 44 |
| Fan Diameter | (ft) | 28 | 28 |
| Tower Width | (ft) | 151 | 151 |
| Tower Length | (ft) | 1,408 | 1,408 |
| Tower Height | (ft) | 91 | 91 |
| Fan Deck Height | (ft) | 77 | 77 |
| Inlet Opening Height | (ft) | 36 | 36 |
| Basin Length | (ft) | 1,409 | 1,409 |
| Primary Orientation | (°) | 35 | 83 |
| Tree Height | (ft) | 45 | 45 |

There are many trees in the vicinity of the proposed cooling tower locations, which will have an impact on the air flow patterns, and ultimately the recirculation. Areas with trees were identified based on satellite pictures, and were elevated by 45-ft, representing the mixed coniferous and deciduous tree height outlined in section 4.3.2 of Tetra Tech Report No. 11431116100-REP-R0001-02, p.50. Areas that are raised to represent trees are colored green in Figure 1.

Areas that surround the plant that do not contain dense areas of trees are not raised, and are colored brown in Figure 1. The river is colored blue in Figure 1.

The model domain extends 2000-ft vertically above river level, and at least 2000-ft around the cooling towers to ensure that the flow field near the cooling tower is not unduly influenced by the model domain boundaries.

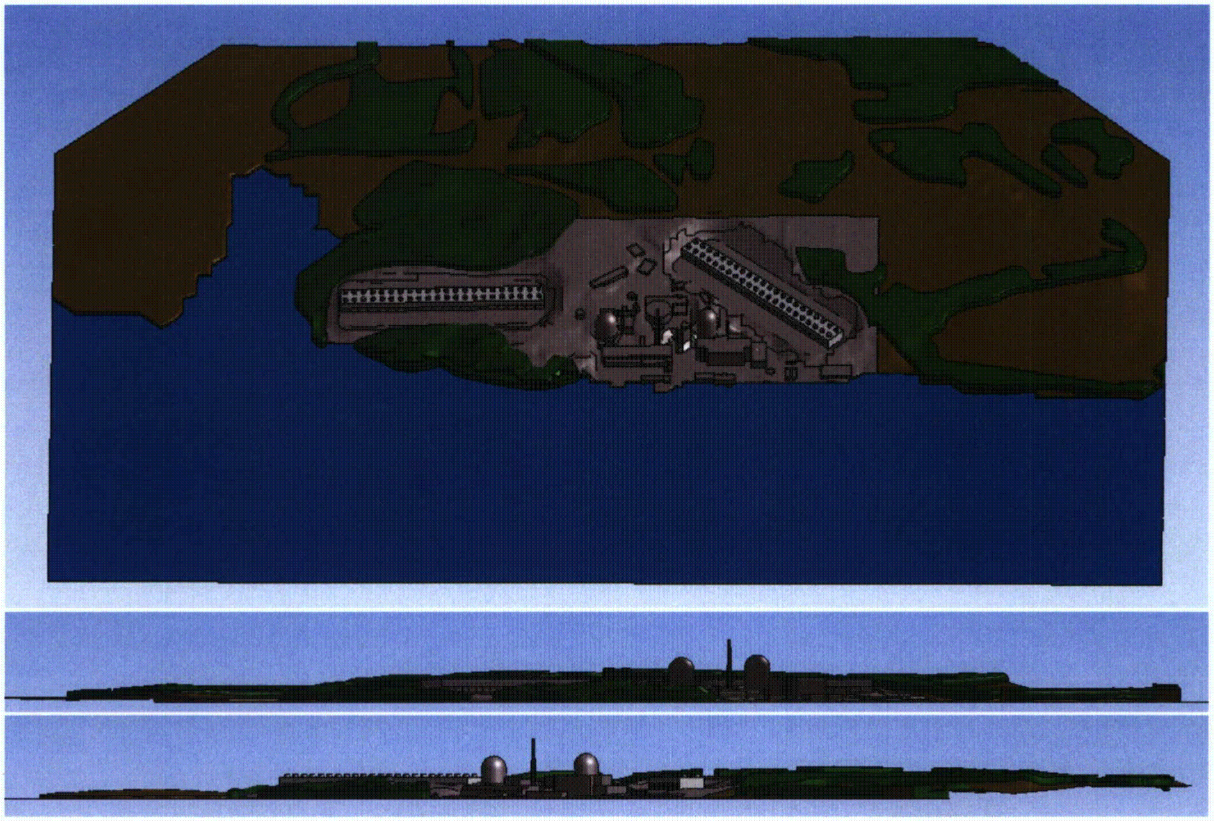

**Figure 1 - Model Domain as seen from above looking NE (top); river level looking East (middle); river level looking North (bottom)**


Figure 2 - Tetra Tech Report No. 11431116100-REP-R0001-02, Figure 3-3

## Atmospheric Conditions

The atmospheric conditions are made up of the ambient temperature, humidity, wind speed, and wind direction. The design condition for the cooling towers is an ambient wet bulb temperature of 77°F (Tetra Tech Report No. 11431116100-REP-R0001-02, Table 2-1, p.13), with no specification of the dry bulb temperature or relative humidity. Based on the expected water evaporation rate and estimated gas velocity at the fan of 30.4-ft/s (calculated by Ted Main, provided to Alden via email on 10/8/2013), an ambient dry bulb temperature of 92.6°F was calculated, with a corresponding relative humidity of 50.7%. More details on the ambient atmospheric conditions are provided in Table 3.

**Table 3 - Atmospheric Conditions**

| Atmospheric Conditions: | Units: | Direction 1: | Direction 2: |
|---|---|---|---|
| Wind Direction | (°) | 180 | 22.5 |
| Wind Speed (@ 10m height) | (m/s) | 2.12 | 2.12 |
| | (ft/s) | 6.96 | 6.96 |
| Dry Bulb Temperature | (°F) | 92.6 | 92.6 |
| Wet Bulb Temperature | (°F) | 77.0 | 77.0 |
| Relative Humidity | (%) | 50.7% | 50.7% |
| Humidity Ratio | (lbm/lbm) | 0.01663 | 0.01663 |
| Air Density | (lbm/ft3) | 0.071 | 0.071 |
| Inlet Enthalpy | (Btu/lbm) | 40.24 | 40.24 |

According to onsite meteorology measurement, there are two prevailing wind directions (from the NNE, and from the South), corresponding roughly to the orientation of the Hudson River valley in this location (Figure 4). One case was run from each prevailing wind direction, with the stated average wind speed of 2.12-m/s (6.96-ft/s). This is a relatively low wind speed for exhaust gas re-entrainment for this type of cooling tower, greater re-entrainment is expected at higher wind speeds.

The wind was applied with a specified magnitude and direction (depending on the case) to the surrounding boundaries of the model as a velocity inlet condition for the upwind boundaries, and a pressure outlet for the downwind boundaries. The wind speed measurement in the wind rose data was taken at an elevation of 10m, however due to the atmospheric boundary layer the wind is typically less than the measured wind speed closer to the ground, and greater than the measured wind speed at higher elevations. The generally accepted $1/7^{th}$ power law wind profile was used at the model boundaries (Figure 3).

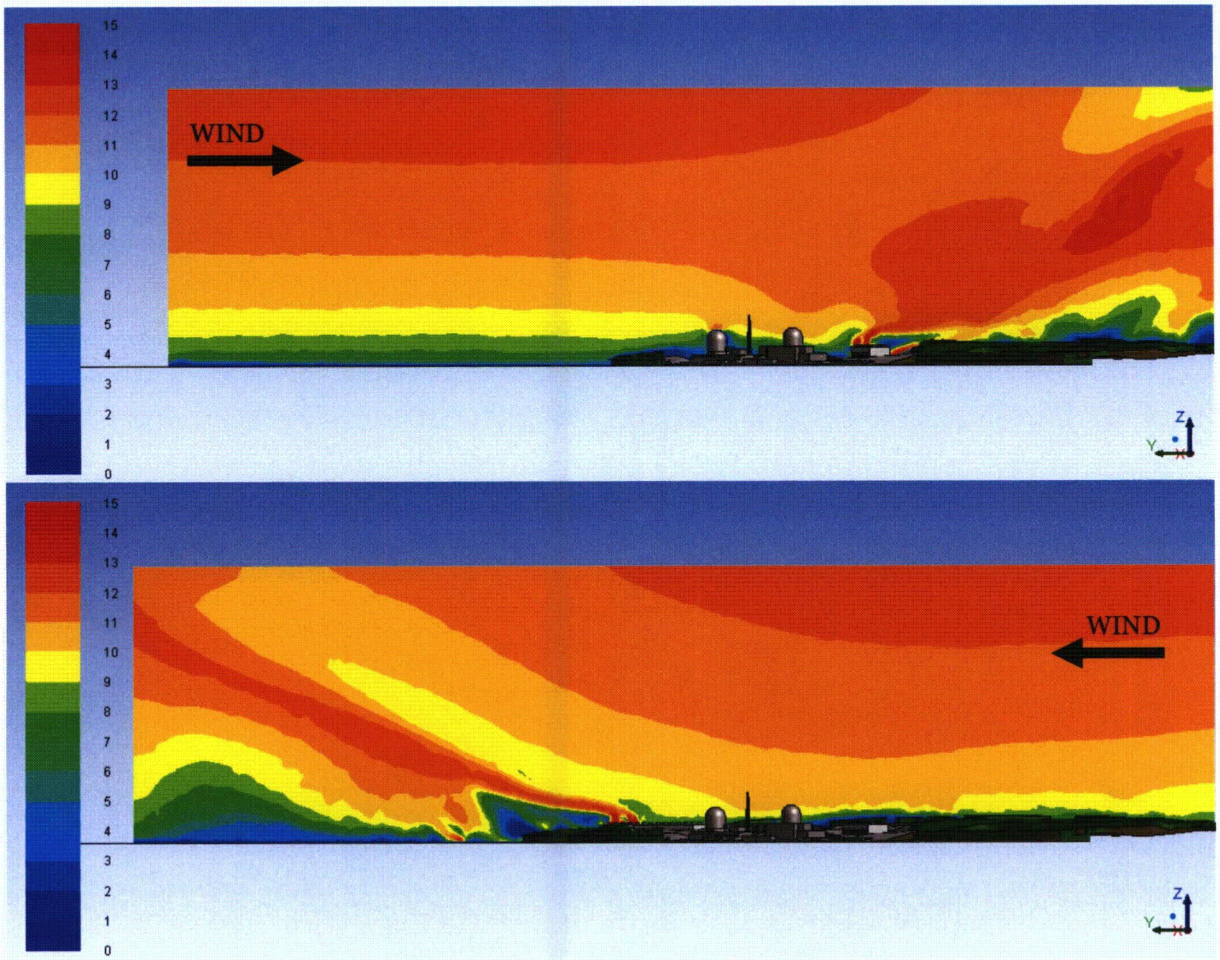


**Figure 3 - Contours of Velocity (ft/s) Showing Atmospheric Boundary Layer with a NNE Wind (top) and a South Wind (bottom)**

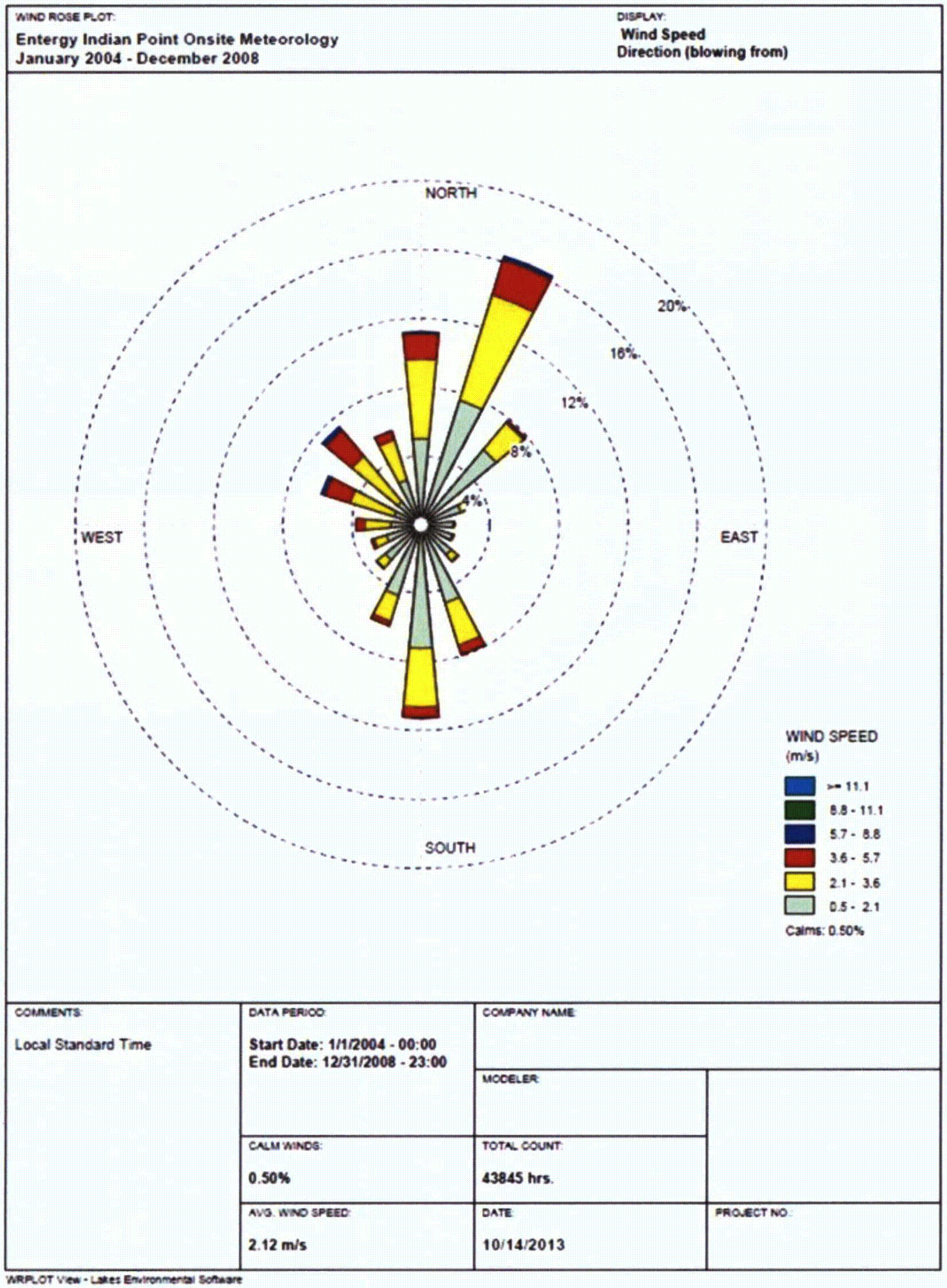


**Figure 4 - Entergy Indian Point Onsite Meteorology Wind Rose Data**

## Process Information

The heat rejection per cooling tower of 7,000-MMBtu/hr was taken from Tetra Tech Report No. 11431116100-REP-R0001-02, Table 2-1, p.13. The heat rejection rate was split evenly between the 44 cells in each tower, and that heat rejection rate per cell was assumed to be constant regardless of the extent of exhaust gas re-entrainment at individual cells. Plume abatement was assumed to be inactive.

The values listed in Table 4 represent the cooling tower performance and exhaust gas conditions if there is no exhaust gas re-entrainment. The outlet conditions are assumed to be saturated with enthalpy input into the gas stream equivalent to the cell heat duty divided by the gas mass flow rate.

**Table 4 - Cooling Tower Process Information**

| Process Flow Information: | Units: | Value: |
| --- | --- | --- |
| Heat Rejection per Unit | (MMBtu/hr) | 7,000 |
| Heat Rejection per Cell | (MMBtu/hr) | 159.1 |
| Exhaust Velocity | (ft/s) | 30.4 |
| Outlet Air Density | (lbm/ft3) | 0.0687 |
| Exhaust Mass Flow per Cell | (lbm/hr) | 4,629,550 |
| Outlet Enthalpy | (Btu/lbm) | 74.61 |
| Outlet Wet Bulb Temperature | (°F) | 102.5 |
| Outlet Relative Humidity | (%) | 100% |
| Outlet Humidity Ratio | (lbm/lbm) | 0.0467 |

The CFD model was initialized with a specified mass flow exiting through the fan of each cell, with the temperature and water vapor mass fraction consistent with no exhaust gas re-entrainment (Table 4). Similarly, a flow rate was specified for the ambient air inlet to each cell; however the temperature and humidity entering each cell is determined by the flow patterns surrounding the cooling tower. The inlet and outlet flow rates were assumed to remain constant regardless of the extent of re-entrainment.

The model was allowed to iterate until convergence under the assumption of no exhaust gas re-ingestion. Once converged, the inlet temperature and humidity of each cell was calculated. If there was no re-ingestion of exhaust gas, then the cell outlet temperature and humidity were at an appropriate level to reject the specified heat duty, and no update was necessary. However, if a cell did experience any exhaust gas re-entrainment, the enthalpy difference between the inlet and outlet gas flow streams would be less that the heat duty required of that cell.

If the cell was not meeting its heat duty, then the outlet temperature was increased (remaining saturated) until the enthalpy difference between the inlet and outlet gas streams multiplied by the mass flow rate of the gas once again met the heat duty of the cell.

The CFD model was run again with the new cooling tower exhaust gas conditions until converged, and then another iteration on cooling tower heat duty was completed. The model was run in this manner until the average inlet wet bulb temperature for each cooling tower changed less than 0.05°F per iteration.

## Results

The deliverable from the CFD model is the mass-weighted average increase in ambient wet bulb temperature at the inlet to each cooling tower for each wind condition. These results are presented in Table 5.

**Table 5 - Model Results**

| Results: | Units: | Cooling Tower 2: | Cooling Tower 3: |
|---|---|---|---|
| Average Wet Bulb Temperature Increase @ 10m Wind Speed = 2.12 m/s: | | | |
| Wind Direction 1 (S): | (°F) | 1.8 | 1.9 |
| Wind Direction 2 (NNE): | (°F) | 0.1 | 3.0 |

## Wind Blowing from the South

With the wind blowing from the south, the Unit 2 cooling tower experiences an average increase in wet bulb temperature of 1.8°F, and the Unit 3 cooling tower experiences an average increase in wet bulb temperature of 1.9°F. Both towers experience re-entrainment only on the downwind (northern) side (Figure 7).

With the wind at 2.12-m/s from the south, there is little influence of the Unit 3 cooling tower on the Unit 2 cooling tower. An iso-surface of wet bulb temperature equal to 79°F, or 2°F above ambient is shown in Figure 8 and Figure 9, which shows the plumes from the cooling towers from various angles. The plume from cooling tower 3 is not drawn into the inlet of cooling tower 2. It should be noted that this is not the visible plume.

A determining factor for the amount of entrainment a cooling tower is likely to encounter is the momentum ratio of the plume to wind. The momentum of each component is measured as the product of the density times the velocity squared. With a relatively greater plume momentum, the exhaust will penetrate upward into the wind a greater distance before turning downwind, and not much re-entrainment will occur. This is what happens at lower wind speeds, and can be seen in Figure 7 as a small number of pathlines being drawn back into the inlets on the downwind side.

At higher wind speeds, the plume has a lower momentum relative to the wind, and the plume turns downwind more quickly, enabling more exhaust to get caught in the recirculation zone on the downwind side of the cooling tower. At wind speeds higher than 2.12-m/s, it is expected that there would be more exhaust gas re-entrainment than is reported here.

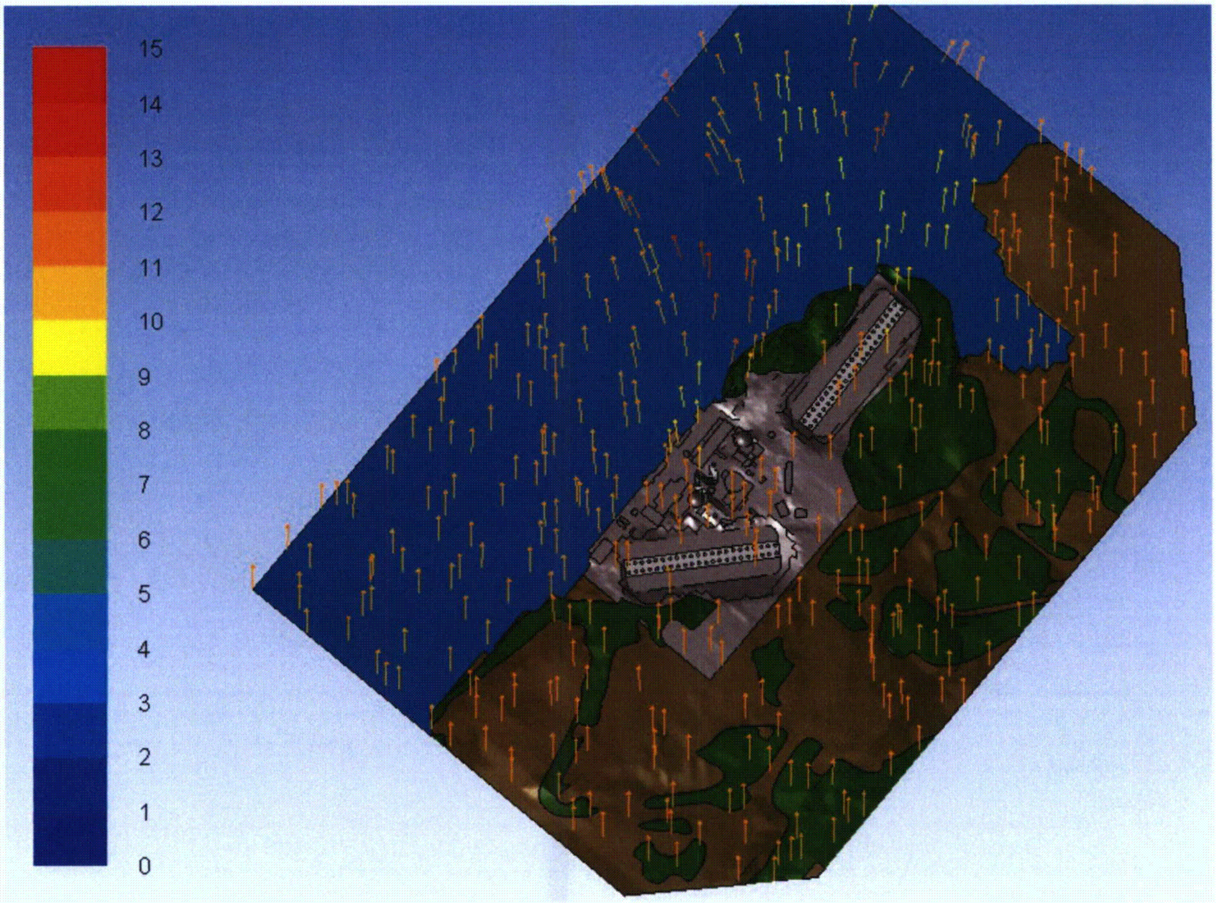


**Figure 5 - Wind Velocity Vectors (ft/s), Elevation = 500-ft, Wind Blowing from the South**


**Figure 6 - Contours of Velocity (ft/s) Showing Atmospheric Boundary Layer. South Wind.**

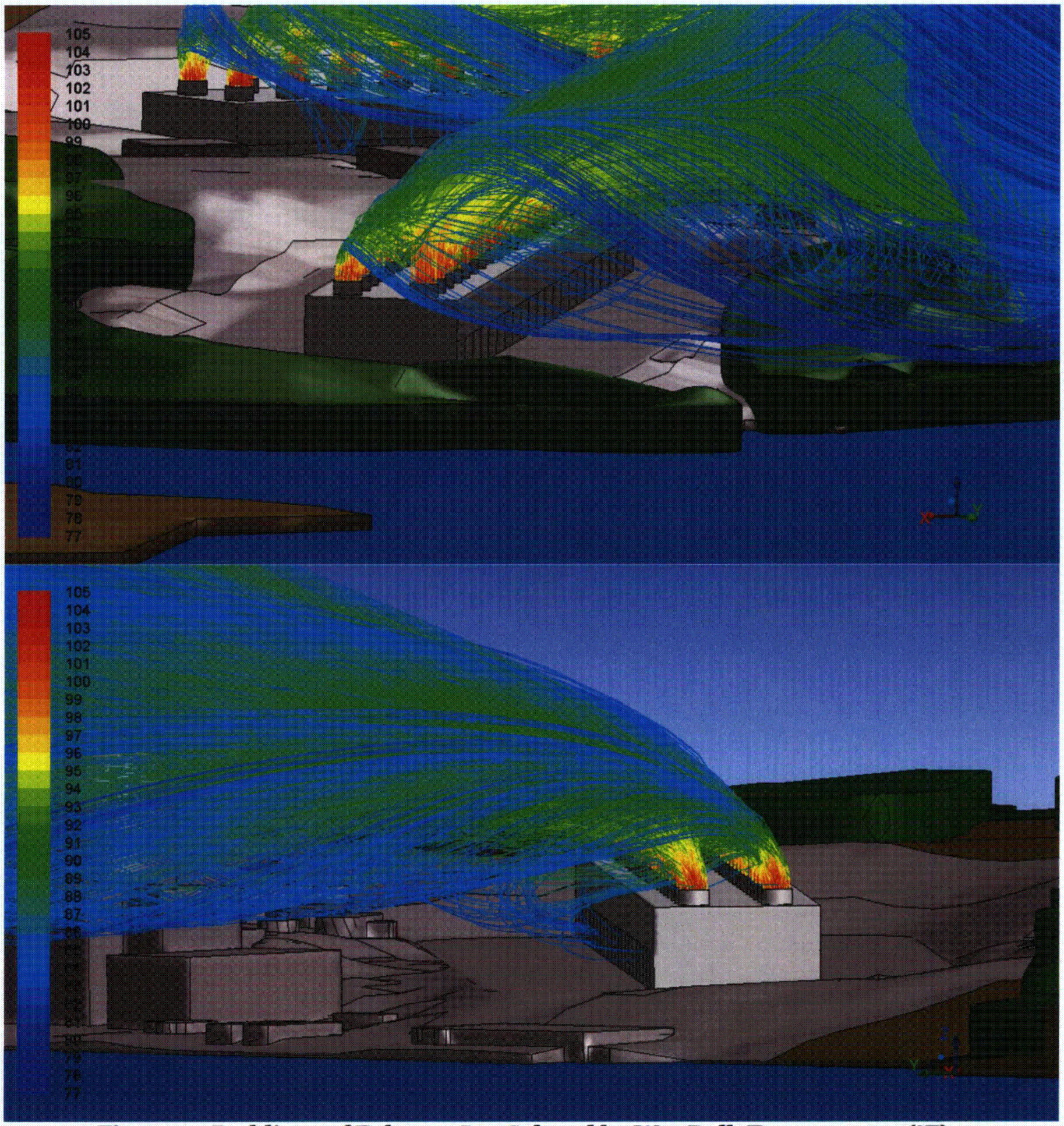


**Figure 7 - Pathlines of Exhaust Gas Colored by Wet Bulb Temperature (°F); Unit 2 (top), Unit 3 (bottom). Wind from the South.**

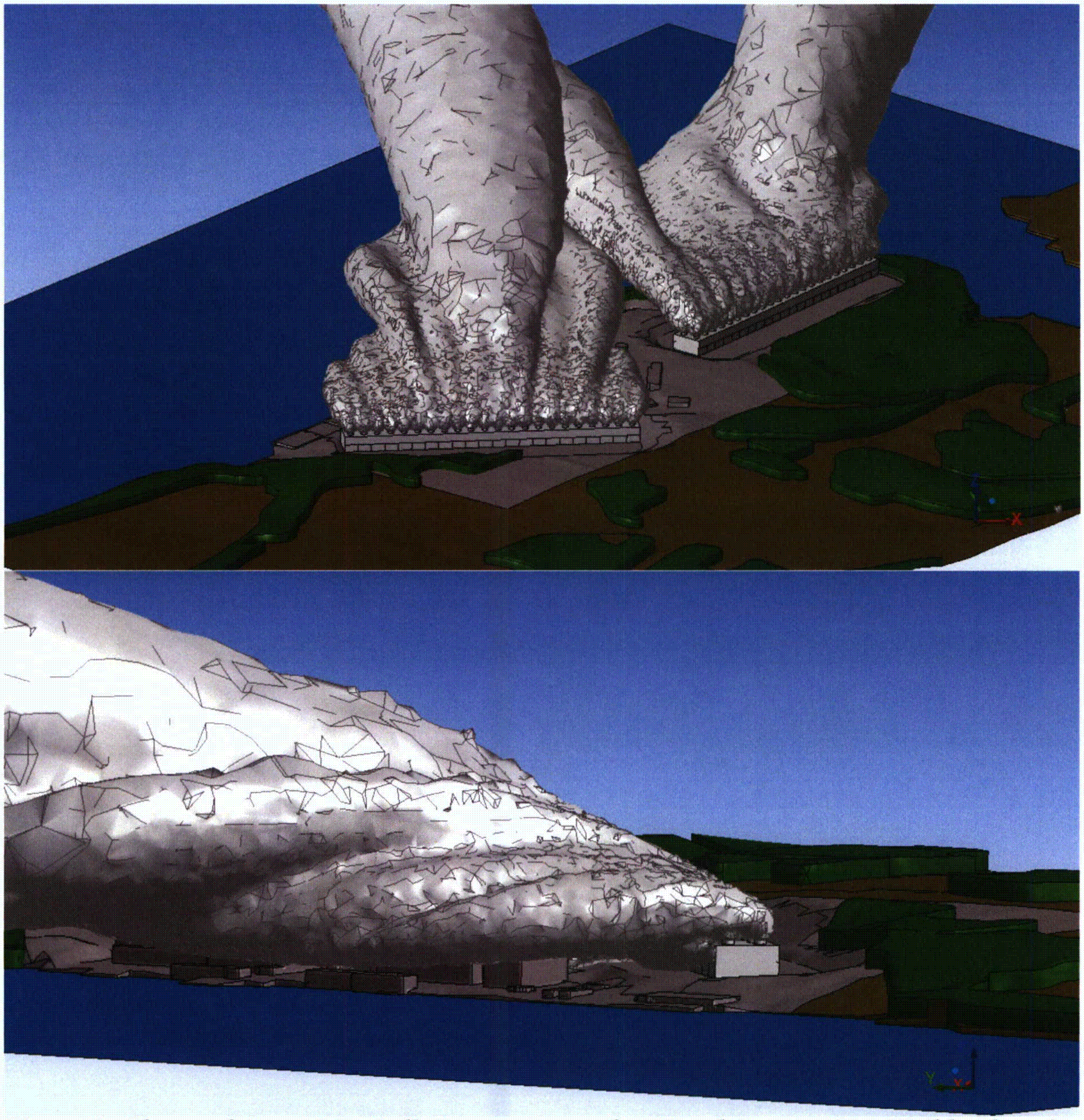

**Figure 8 - Plume of Gas at Wet Bulb Temperature of 79°F and Greater (Not Visible Plume); View from South (top), View from West (bottom). Wind from the South.**

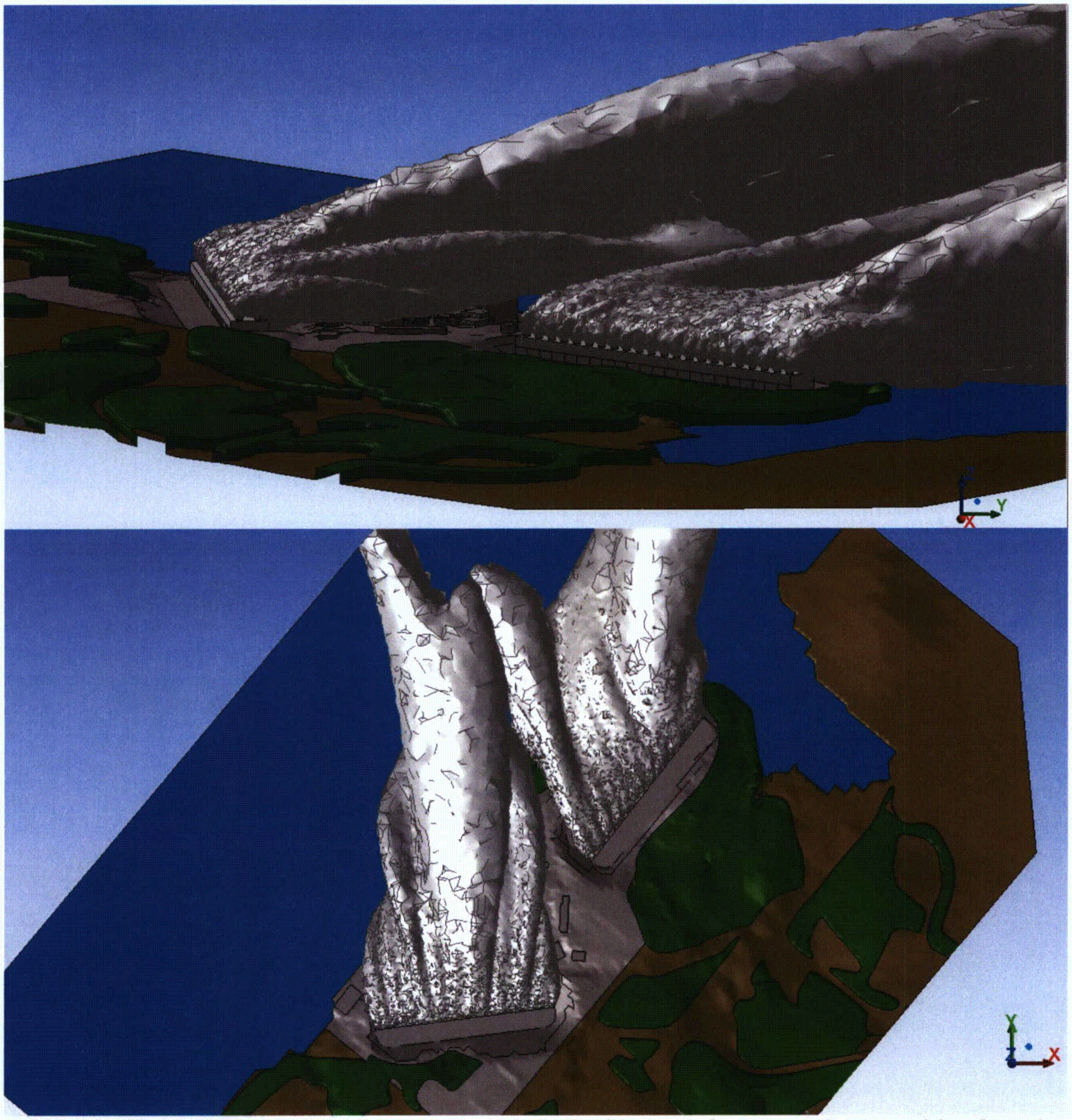

**Figure 9 - Plume of Gas at Wet Bulb Temperature of 79°F and Greater (Not Visible Plume); View from East (top), View from Above (bottom). Wind from the South.**

## Wind Blowing from the NNE

With the wind blowing from the NNE, the Unit 2 cooling tower experiences an average increase in wet bulb temperature of 0.1°F, and the Unit 3 cooling tower experiences an average increase in wet bulb temperature of 3.0°F. The Unit 3 cooling tower experiences re-entrainment only on the downwind (southern) side , and since it is aligned with the flow, the Unit 2 cooling tower experiences nearly no re-entrainment (Figure 12).

With the wind at 2.12-m/s from the south, there is little influence of the Unit 2 cooling tower on the Unit 3 cooling tower. An iso-surface of wet bulb temperature equal to 79°F, or 2°F above ambient is shown in Figure 13 and Figure 14, which shows the plumes from the cooling towers from various angles. The plume from cooling tower 2 is not drawn into the inlet of cooling tower 3. It should be noted that this is not the visible plume.

A determining factor for the amount of entrainment a cooling tower is likely to encounter is the momentum ratio of the plume to wind. The momentum of each component is measured as the product of the density times the velocity squared. With a relatively greater plume momentum, the exhaust will penetrate upward into the wind a greater distance before turning downwind, and not much re-entrainment will occur. This is what happens at lower wind speeds, and can be seen in Figure 12 as a small number of pathlines being drawn back into the inlets on the downwind side of Unit 3.

At higher wind speeds, the plume has a lower momentum relative to the wind, and the plume turns downwind more quickly, enabling more exhaust to get caught in the recirculation zone on the downwind side of the cooling tower. At wind speeds higher than 2.12-m/s, it is expected that there would be more exhaust gas re-entrainment than is reported here.

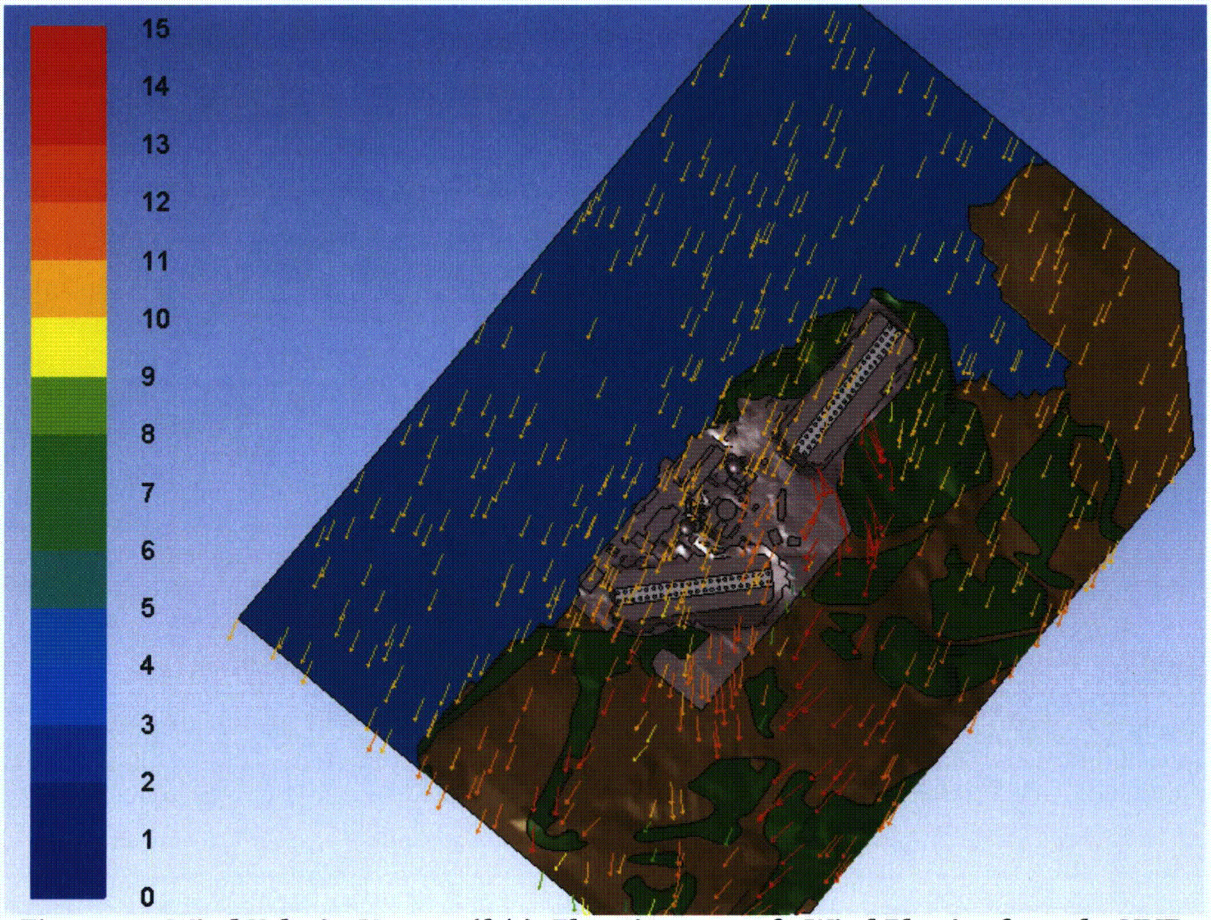


Figure 10 - Wind Velocity Vectors (ft/s), Elevation = 500-ft, Wind Blowing from the NNE


Figure 11 - Contours of Velocity (ft/s) Showing Atmospheric Boundary Layer. NNE Wind.

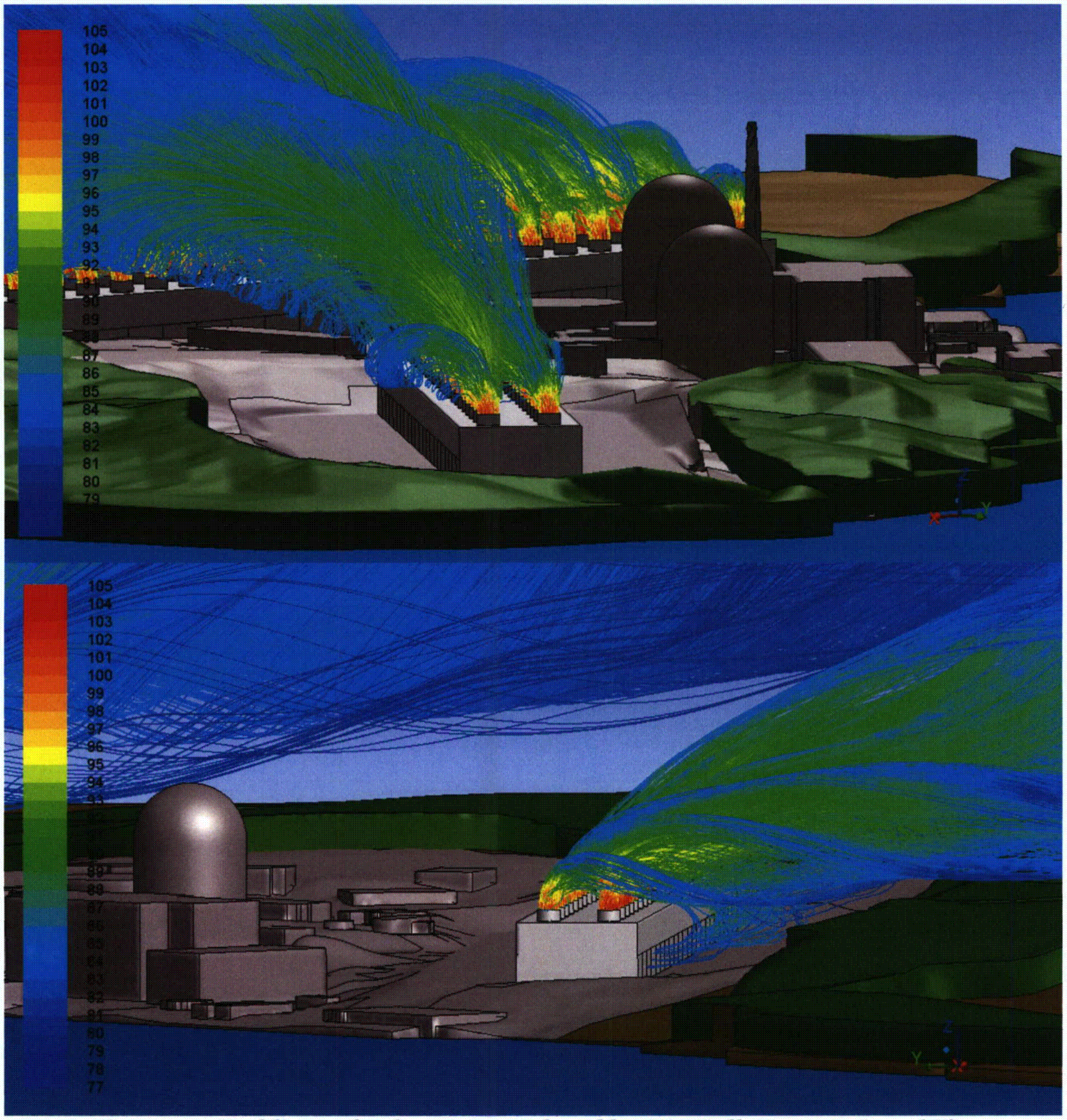


**Figure 12 - Pathlines of Exhaust Gas Colored by Wet Bulb Temperature (°F); Unit 2 (top), Unit 3 (bottom). Wind from the NNE.**

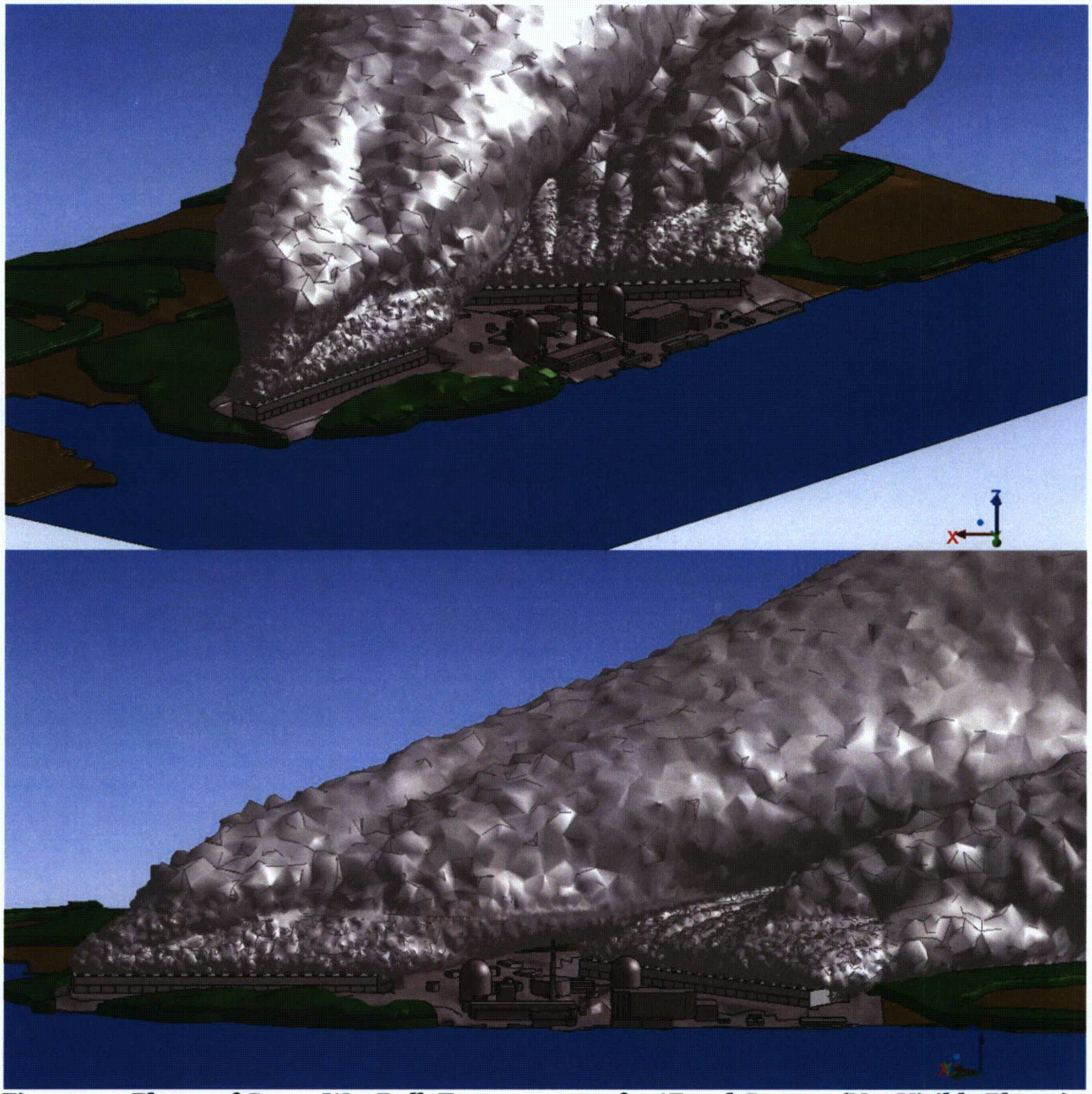

**Figure 13 - Plume of Gas at Wet Bulb Temperature of 79°F and Greater (Not Visible Plume); View from North (top), View from NW (bottom). Wind from the NNE.**

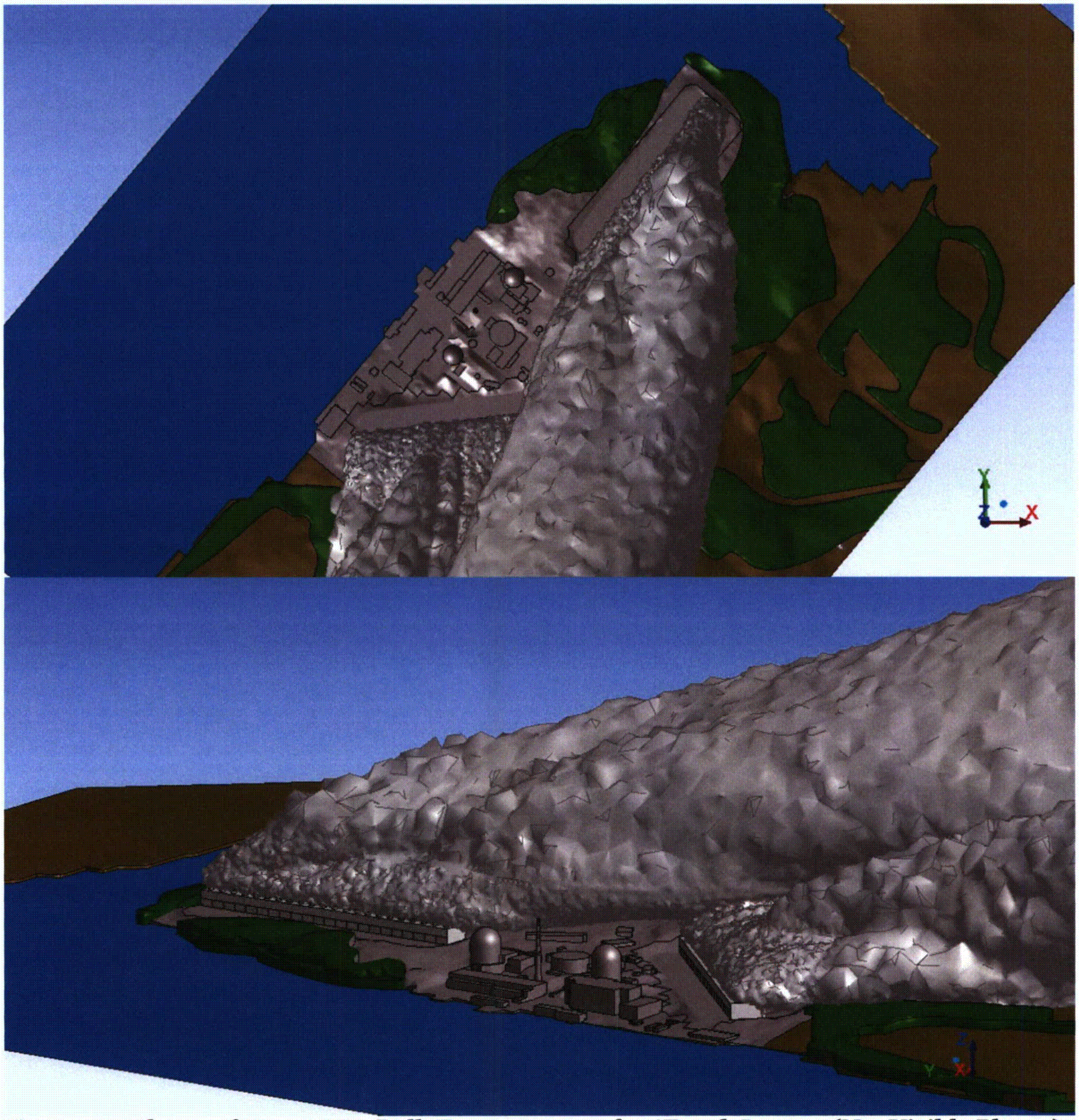


**Figure 14 - Plume of Gas at Wet Bulb Temperature of 79°F and Greater (Not Visible Plume); View from Above (top), View from West (bottom). Wind from the NNE.**

**ATTACHMENT 3**

BREI Independent Review of the Proposed ClearSky© Cooling Towers Impact on the Indian Point Energy Center Main Condensers

# Independent Review of the Proposed ClearSky© Cooling Towers Impact on the Indian Point Energy Center Main Condensers

# Independent Review of the Proposed ClearSky© Cooling Towers at the Indian Point Energy Center

Prepared by Burns and Roe Enterprises, Inc.
Nuclear Services and Advanced Technologies
for Entergy Nuclear

**Revision Record**

| Revision | Date | Signature Block |
|---|---|---|
| 0 | 10 December 2013 | **Prepared by:** **Garrett Goldfinger**, Nuclear Engineer |
| **Description** | | **Reviewed by:** **Sam Petrosi,** Chief Nuclear Officer |
| Final Report | | **Approved by:** **Yan Kishinevsky**, Director – Nuclear Services and Advanced Technologies |

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# I. EXECUTIVE SUMMARY

## A. Introduction

The New York State Department of Environmental Conservation (NYSDEC) Staff submitted the "Notice of DEC Staff's Alternative BTA Proposal regarding the Matter of Entergy Indian Point Units 2 and 3 SPDES permit proceeding" on May 21, 2010, identifying an alternative to the closed-loop cooling system evaluated by Enercon in feasibility studies conducted in 2003 and 2010.

The NYSDEC staff expects to advance another closed-cycle cooling system consisting, primarily, at this time, of plume-abated cooling towers such as the Marley SPX ClearSky© wet-dry hybrid system that allows for a more compact arrangement of cooling tower cells.

## B. Scope of Report

Burns and Roe Enterprises, Inc. (BREI) has been retained by Entergy to perform an independent review of certain aspects of the proposed ClearSky© Cooling Towers as presented in the Tetra Tech, Inc. Indian Point Cooling Tower Alternative Assessment, Report 11431116100-REP-R0001-02.

Specifically, BREI has been directed by Entergy to examine the following:

(1) Condenser Structural Impact
- (a) New tube flow, pressure and temperature
- (b) Tube minimum wall thickness
- (c) Tube vibration and support plate spacing
- (d) Tubesheet joints (4 per shell) torqueing requirements to seal
- (e) Waterbox flow, pressure and temperature (P and T)
- (f) Tubesheet to waterbox joint torqueing requirements to seal
- (g) Circulating water piping to waterbox expansion joint stress?

(2) Plant Safety/Operational Function Impact
- (a) High pressure steam dump impact on turbine trip setpoints

# C. Conclusions and Observations

**Conclusions**

Conversion of IPEC Units 2 and 3 to closed-cycle cooling as proposed in the Tetra Tech report (Reference 1) will not impose any structural challenges (as evaluated in the above scope of work) to the existing condensers.

At Unit 3, the condenser seal water system will require a modification to relocate the seal water head tank to a higher elevation.

Rechecking the tubesheet joint closure torque, especially when there is vacuum on the unit and no circulating water in the waterboxes, is necessary.

The calculated limiting maximum absolute pressure is 3.95 in-Hg in Condenser No. 21 for a 50% load rejection condition given a total flow rate of 700,000 gpm and cooling water inlet temperature of 89 °F. This pressure is below the alarm and trip setpoints and is, therefore, acceptable.

The calculated average absolute pressure for Unit 3 is 5.07 in-Hg for a 50% load rejection condition given a total flow rate of 700,000 gpm and cooling water inlet temperature of 89 °F. This pressure is below the turbine trip set point (12 in-Hga), but higher than the low vacuum alarm set point of 4.0 in• Hg. Site Procedure 3-SOP-TG-4 requires a manual trip of the turbine at 4.5 in-Hga and the steam dump valves are prohibited from opening at pressure greater than 5 in-Hga. Therefore, the absolute condenser pressure of 5.07 in-Hg is unacceptable and measures such as increasing the heat removal capacity of the cooling towers should be taken to reduce it below 4.0 in• Hg or, at least, 4.5 in-Hga.

**Observations**

Referenced sources for the Condenser Guarantee Conditions, presented in the table below, appear to conflict with calculations performed in this report. The heat transfer parameters of the Unit 3 condenser vary significantly from that for the Unit 2 condenser. Given the similar design and flow rates, BREI suggests that values for Unit 3 identified in the table below should be reviewed and revalidated.

| Condenser Guarantee Conditions | | | |
|---|---|---|---|
| | | U2 | U3 |
| Steam Loading (lbm/hr) | | 2414657 | 2,452,630 |
| Heat Rejection Rate (Btu/lb) | | | 950 |
| Heat Rejected (Btu/hr) | | 2315 x 10E6/shell | 2330 x 10E6/Shell |
| Absolute Pressure (in. Hg) | | 2.05 | 1.6 & 2.04 |
| Condensate Temperature (°F) | | 101.96 | 95.6 & 101.9 |
| Inlet Water Temperature (°F) | | 78 | 50 & 75 |
| Tube Cleanliness (%) | | 85 | 85 |
| Quantity of Circulating Water (gpm) | | 274000 | 168,000 & 280,000 |
| Velocity in Tubes (ft/sec) | | 6.03 | 4.1 MAX |
| Circulating Water Rise (°F) | | 16.9 | 29 |
| Heat Transfer Rate (Btu/hrxsq-ftx°F) | | 529 | 296 |
| Free Oxygen in Condensate (ppb) | | 5 | 7 |
| Friction Loss (ft H20) | | 10.71 | 5.4 & 13.5 |

# II. TECHNICAL REVIEW

BREI researched the various referenced documents and compiled Table 1 through Table 5, which were then used in developing the technical reviews in the following sections.

**Table 1 IPEC Condenser Guarantee Conditions (Reference 6 & 7)**

| Condenser Guarantee Conditions | | | |
|---|---|---|---|
| | | **U2** | **U3** |
| **Steam Loading (lbm/hr)** | | 2414657 | 2,452,630 |
| **Heat Rejection Rate (Btu/lb)** | | | 950 |
| **Heat Rejected (Btu/hr)** | | 2315 x 10E6/shell | 2330 x 10E6/Shell |
| **Absolute Pressure (in. Hg)** | | 2.05 | 1.6 & 2.04 |
| **Condensate Temperature (°F)** | | 101.96 | 95.6 & 101.9 |
| **Inlet Water Temperature (°F)** | | 78 | 50 & 75 |
| **Tube Cleanliness (%)** | | 85 | 85 |
| **Quantity of Circulating Water (gpm)** | | 274000 | 168,000 & 280,000 |
| **Velocity in Tubes (ft/sec)** | | 6.03 | 4.1 MAX |
| **Circulating Water Rise (°F)** | | 16.9 | 29 |
| **Heat Transfer Rate (Btu/hrxsq-ftx°F)** | | 529 | 296 |
| **Free Oxygen in Condensate (ppb)** | | 5 | 7 |
| **Friction Loss (ft H20)** | | 10.71 | 5.4 & 13.5 |

Table 2 IPEC Condenser Specifications (Reference 6 & 7)

| Condenser Specifications | | | | | |
|---|---|---|---|---|---|
| | | U2 | U3 | | Notes |
| Surface Area (sq ft) | | 947100 | 858375 | | |
| Surface Area/Shell (sq ft) | | 315700 | 286125 | | |
| Tubesheet Material | | Titanium ASTM B265 GRII | Titanium ASTM B265 GRII | | |
| Tubesheet Thickness (in) | | 1 | 1 | | |
| Support Plates Material | | SA-285-C | SA-285-C | | |
| Support Plates Thickness (in) | | 0.5 | 0.625 | | |
| | | 0.75 | | | |
| Baffles Material | | | SA-36 | | |
| Baffle Thickness | | 0.25 | | | |
| Support Plates #/bundle | | 25 | 21 | | U2 12 support plates .75", 13 support plates 0.5" |
| Tube Material | | Titanium ASTM B338 GRII | Titanium ASTM B338 GRII | | Tubes are rolled and welded to tubesheet |
| Tube OD | | 0.875 | 0.875 | | |
| # Tubes condensing section | | 27702 | 22920 | | |
| | | 22 BWG | 22 BWG | | |
| # Tubes Pherephry | | | 566 | | |
| | | | 15 BWG | | |
| # Tubes Air Cooler Section | | | 1620 | | |
| | | | 22 BWG | | |
| # Tubes Total | | 27702 | 25106 | | |
| Effective Tube Length (ft, in ) | | 49, 9 | 49, 9 | | |
| Overall Tube Length (ft, in ) | | 49, 11.25 | 49, 11.75 | | |
| Lowest Tube Elevation (ft, in) | | | 19, 10.6 | | |
| Highest Tube Elevation (ft, in) | | | 33, 7 | | |
| | | | | | |
| Waterboxes Material | | SA-516-70 | SA-516-70 | | Inside coated with 0.25" neoprene |
| Waterbox Bonnet | | 0.5 | 0.5 | | |
| Waterbox Walls | | 0.625 | 0.625 | | |
| Waterbox Design Pressure (psig) | | 25 | 30 | | |
| Waterbox Test Pressure (psig) | | 37.5 | 45 | | |

Table 3 ClearSky© Design Basis Specifications (Reference 1)

| ClearSky Design Basis Specifications | | | | | |
|---|---|---|---|---|---|
| | | | | | |
| | | U2 | U3 | | Notes |
| Model | | F4811D-6.0-44B | F4811D-6.0-44B | | Counter flow mechanical draft cooling tower with plume abatement |
| Number of Cells | | 44 | 44 | | |
| | | | | | |
| Thermal Load (MMBTU/hr) | | 7000 | 7000 | | |
| Circulating Flow (gpm) | | 700000 | 700000 | | |
| Tower Footprint (acres) | | ~5 | ~5 | | |
| Tower Length (ft) | | 1408 | 1408 | | Individual cells are 64' x 75.5' and rise 91' above grade |
| Tower Width (ft) | | 133 | 133 | | |
| Tower Height (ft) | | 91 | 91 | | |
| Basin Width (ft) | | 151 | 151 | | |
| Basin Length (ft) | | 1409 | 1409 | | |
| Basin Elevation (ft) | | 52.5 | 52.5 | | |
| Normal Basin Water Depth (ft) | | 4 | 4 | | |
| | | | | | |
| Entering Wet Bulb (°F) | | 77 | 77 | | |
| Approach Temperature (°F) | | 12 | 12 | | |
| | | | | | |
| Circulating Water Hot (°F) | | 109 | 109 | | |
| Circulating Water Cold (°F) | | 89 | 89 | | |
| Circulating Water Temperature Rise (°F) | | 20 | 20 | | |

Table 4 Circulating Water Piping to Condenser Waterbox Expansion Joint Specifications (Reference 6 & 7)

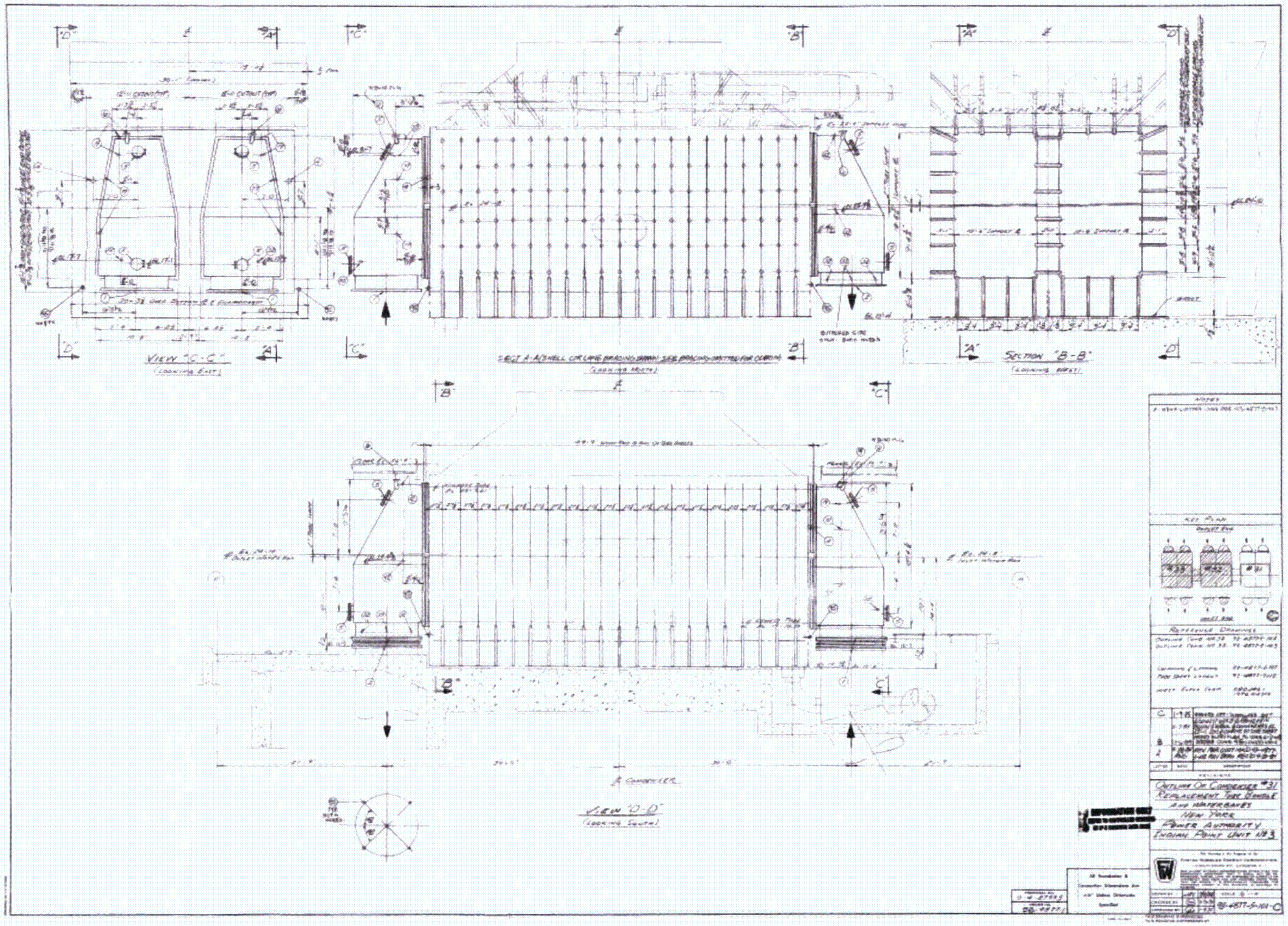
| Expansion Joint | | | |
|---|---|---|---|
| | | U2 | U3 |
| Manufacturer | | Mercer Rubber Company | Mercer Rubber Company |
| Model | | 501RN – 96" ID | 700N – 96" ID |
| Max Allowable Working Pressure (psig) | | 50 | 40 |
| Hydrostatic Test Pressure (psig) | | | 45 |
| Max Allowable Temperature (°F) | | 225/180 | 225 |
| Installed Elevation (ft, in) | | | 12, 7 |

Table 5 IPEC Circulating Water Pumps Specifications (Reference 13 & 15)

| Circulating Water Pumps | | | | | |
|---|---|---|---|---|---|
| | | U2 | U3 | | Notes |
| **High Speed (RPM)** | | 254 | 360 | | U2 - 2 speed pumps |
| **High Speed Flow (gpm)** | | 140000 | 140000 | | U3 - variable speed pumps |
| **High Speed Total Dynamic Head (ft)** | | 21 | 29 | | |
| **Low Speed (RPM)** | | 187 | 210 | | |
| **Low Speed Flow (gpm)** | | 84000 | 63000 | | |
| **Low Speed Total Dynamic Head (ft)** | | 15 | | | |

Outline of Condenser Tube Bundle and Waterboxes are depicted on Figure 1 below.

Figure 1 Outline of Condenser Tube Bundle and Waterboxes (Reference 7)



## 1. New tube flow, Pressure and Temperature

### *a) Flow*

Per the Foster Wheeler Energy Corporation (FWEC) Condenser Data Sheet for Unit 2 (Reference 6) the major parameters of condenser tubes are as follows:

| | |
|---|---|
| **Tube Size, O.D. (in)** | 0.875 |
| **Tube Material (Condensing Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |
| **Tube Material (Air Cooler Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |
| **Total Number of Tubes per Shell** | 27,702 |

Per the FWEC Condenser Data Sheet for Unit 3 (Reference 7) the major parameters of condenser tubes are as follows:

| | |
|---|---|
| **Tube Size, O.D. (in)** | 0.875 |
| **Tube Material (Condensing Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG with peripheral tubes 15 BWG |
| **Tube Material (Air Cooler Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |
| **Total Number of Tubes per Shell** | 25,106<br>(22 BWG: 24,540 and 15BWG: 566) |

The proposed Clearsky© Cooling Towers and associated piping are sized to handle 700,000 gpm per tower. The flow to each of 3 condensers is 233,333 gpm. Based upon the information provided above, the tube velocities are calculated as follows:

$$v = \frac{0.408Q}{d^2}$$

where: $v$ = velocity, ft/sec

$Q$ = flow, gpm

$d$ = internal diameter of tube, in. [The I.D of a 7/8" 22BWG tube is 0.819"]

Flow per tube in Unit 2 is 233,333 gpm divided by 27,702 tubes, or 8.4 gpm/tube. The velocity ($v$) in the Unit 2 tubes is 5.1 ft/sec. Likewise for Unit 3, the flow is 9.3 gpm/tube and the tube velocity is 5.7 ft/sec. These tube velocities are within industry recommended velocities (Reference 16) for flow of water through tubes not to cause additional concerns regarding erosion or vibration concerns.

### b) Pressure

The condenser tubes have a design pressure of 30 psig. The expected normal operating water level in the basin of the proposed ClearSky© Cooling Towers will be at approximately the 56' 6" el. This translates to a static head of approximately 42' or approximately 18.2 psig acting on the waterboxes. Per the results of the tube minimum wall thickness calculations in Item 2 below, the actual tube thicknesses have margins, which are greater than 25 times the required minimum wall thickness for a design pressure of 30 psig. Therefore, the expected maximum operating tube pressure of 18.2 psig in the condensers with the Clearsky© Cooling Towers is acceptable.

### c) Temperature

The maximum condenser outlet temperatures are well below 150°F and will have no effect on the tubes.

## 2. Tube minimum wall thickness

Per the FWEC Condenser Data Sheet for Unit 2 (Reference 6) the condenser tubes are as follows:

| | |
|---|---|
| **Tube Size, O.D. (in)** | 0.875 |
| **Tube Material (Condensing Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |
| **Tube Material (Air Cooler Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |

Per the FWEC Condenser Data Sheet for Unit 3 (Reference 7) the condenser tubes are as follows:

| | |
|---|---|
| **Tube Size, O.D. (in)** | 0.875 |
| **Tube Material (Condensing Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG with peripheral tubes 15 BWG |
| **Tube Material (Air Cooler Zone)** | Titanium ASTM B338 Gr. 2, 22 BWG |

Per ANSI/ASME B31.1, Section 104, the minimum wall thickness is be determined as follows:

$$t_m = \frac{PD_o}{2(S_E + Py)} + A$$

where:

$t_m$ = Minimum wall thickness, in.

$P$ = Internal design pressure, psig (45 psia) Use 45 psi since pressure on outside of tubes is near vacuum

$D_o$ = Outside diameter of tube, in. (0.875 inches)

$S_E$ = maximum allowable stress in material at design temperature, psi (13,700 psi at 150 °F)

$y$ = coefficient (coefficient is 0 for nonferrous materials)

$A$ = additional material thickness for erosion/corrosion, in.

Substituting the above values into the equation, the minimum wall thickness is 0.0014 inches.

The tube wall thicknesses for 22 Birmingham Wire Gauge (BWG) and 15 BWG are 0.028 inches and 0.072 inches, respectively. Therefore, the actual tube thicknesses have margins, which are greater than 20 times the required minimum wall thickness for a design pressure of 30 psig (see Section 5.b below for expected maximum operating pressure of 18.2 psig in the condenser with the ClearSky© Cooling Tower elevation differences).

## 3. Tube vibration and support plate spacing

The support plates provide support for the condenser tubes and act as internal bracing for the shell. They are symmetrically spaced and of sufficient number to minimize tube vibration induced by resonant frequencies and by high velocity steam passing around the tubes.

Per the FWEC Condenser Manual for Unit 2 (Reference 6) and the FWEC Condenser Manual for Unit 3 (Reference 7) the condenser tubes have an effective tube length of 49ft - 9 in.

In Unit 2 each upper and lower bundle has 25 support plates whereas in Unit 3 each upper and lower bundle has 21 support plates. Therefore, the spacing between support plates is greater in Unit 3 with a spacing of 27.125" between support plates and 27.25" between support plate and tube sheet. (vs. Unit 2 which has a maximum spacing of 22.96" between support plates).

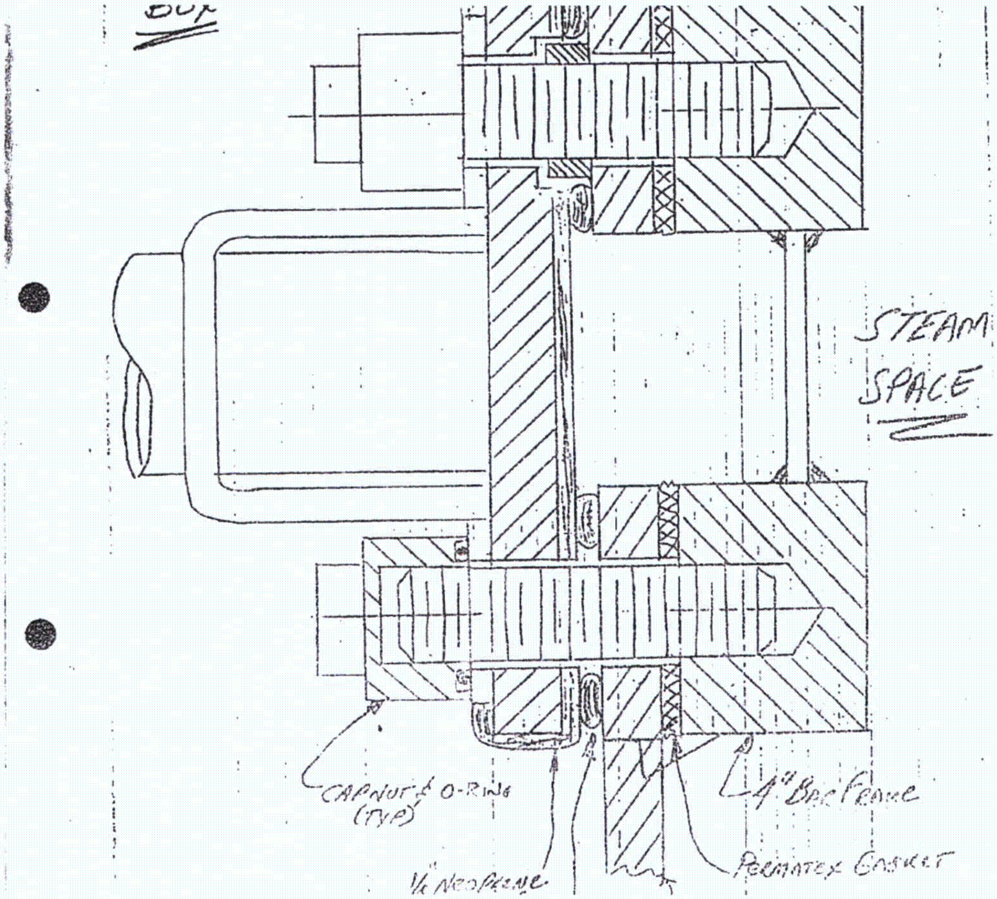
As seen in Section 1 above, the tube flow velocities are between 5 and 6 ft/sec, which are within industry recommended velocities for flow of water through tubes.

The support plate spacing for limiting bending stresses in unsupported spans of tubes is a function of entering steam flow, geometry of condenser neck, saturation pressure in the condenser shell, tube O.D., tube pitch, tube gauge, and tube material. None of these conditions are adversely affected by the Clearsky© Towers. Therefore, there will be no effect on the support plate spacing.

## 4. Tubesheet Joints (4 per shell) Torqueing Requirements to seal

Condenser tubesheet joint closure is depicted on Figure 2 below.

Figure 2 Condenser Tubesheet Joint Closure (Reference 10)




The documentation obtained from both Unit 2 and Unit 3 has not provided a sufficient level of detail regarding this joint closure. BREI has assumed that the tubesheet joint closure is similar for both units. BREI opines that the if the existing closure "bridge bar" and associated components have been performing as intended in the current configuration, then the additional waterbox pressure resulting from the proposed ClearSky© closed-cycle cooling system should not degrade the performance, as the additional pressure should act to compress and seal the gasketed joints. Nevertheless, BREI recommends that if the ClearSky© closed-cycle cooling system is implemented, rechecking the torque especially when there is vacuum on the unit and no circulating water in the waterboxes is necessary.

## 5. Waterbox Flow, Pressure and Temperature

### a) *Flow*

Currently, each of 3 condensers at Unit 2 are rated for a circulating water design flow of 274,000 gpm (Reference 6), resulting in a total circulating water flow of 822,000 gpm. Similarly, the Unit 3 condensers are rated for a circulating water design flow of 280,000 gpm each (Reference 7), resulting in a total circulating water flow of 840,000 gpm. For comparison, the proposed ClearSky© alternative closed-cycle cooling towers have been designed to provide a total of 700,000 gpm. Assuming that the flow is evenly distributed among all condensers, the resulting flow to each condenser (2 waterboxes) will be approximately 233,333 gpm, which is well below the design rated flow and should not negatively impact the structural integrity of the condenser waterboxes.

### b) *Pressure*

The condenser waterboxes are fabricated of welded steel plate and are of the trapezoid and cylindrical bonnet type. The Unit 2 waterboxes are designed to withstand an operating pressure of 25 psig and a test pressure of 37.7 psig (Reference 6). The Unit 3 waterboxes are designed to withstand of a pressure of 30 psig and a test pressure of 45 psig (Reference 7). Referring to FWEC drawing 93-4877-5-102-C, the lowest portion of the waterbox is approximately at the elevation 14' 6". BREI is assuming that Unit 2 installation is similar. From the Tetra Tech report (Reference 1), the proposed ClearSky© cooling tower basins will be at the approximate 52' 6" elevation. The Tetra Tech report also establishes that the normal operating basin level will be approximately 4' elevation. Therefore, the normal operating level will be at approximately the 56' 6" elevation. This translates to a static head on the waterboxes of approximately 42' or 18.2 psig. This is well below the design rated pressure for both Unit 2 and Unit 3 waterboxes and should not pose additional challenge to the structural integrity of the condenser waterboxes.

### c) *Temperature*

The Unit 2 condensers are rated for an inlet temperature of 78°F and a temperature rise of 16.9°F for an outlet temperature of 94.9°F. The Unit 3 condensers are rated for an inlet temperature of 75°F and a temperature rise of 29°F for an outlet temperature of 104°F. The proposed ClearSky© cooling tower system will supply circulating water at 89°F with an outlet temperature of 109°F.

Referring to the Stretch Power Uprate (SPU) calculation performed for Unit 2 and Unit 3 (References 9 & 8), the circulating water inlet temperature is assumed to be 95°F when a 50% load rejection (steam dump actuation) occurs, resulting in a circulating water outlet temperature of approximately 117°F at Unit 2. The Unit 3 SPU calculation has not included a PEPSE model. However, assuming a minimum of 29°F temperature rise, the resulting outlet circulating water temperature would be approximately 124°F. Further, the SPU calculation for Unit 3 has reference to an actual high river water temperature of 88.5°F. This would result in circulating water outlet temperatures of approximately 105.4°F for Unit 2 and 117.5°F for Unit 3.

Based on the above, the circulating water temperatures for the ClearSky© closed-cycle cooling system appear reasonably close to temperatures assumed in previous calculations and by actual river conditions that have been experienced. As such, they should not pose additional challenges to the structural integrity of the condenser waterboxes.

## 6. Tubesheet to Waterbox Joint Torqueing Requirements to Seal

For details on the tube sheet to shell flange joint, see Reference 7 (Unit 3 only).

Main condenser flange seal system was installed at Unit 3 to ensure that any in-leakage to the condenser via the waterbox/tubesheet/shell flange is pure condensate (Reference 4 & 5). The head tank for this system is located on the 33' elevation, which was chosen to provide adequate head to prevent circulating water system in-leakage to the condenser. The proposed 52.5' el. basin height will impose static head in excess of the current head provided by the seal head tank. Therefore, the seal tank and associated piping should be relocated to a higher elevation.

### a) Torqueing Requirement:

In order to evaluate the torqueing requirements for the waterboxes due to an internal operating pressure of 18.2 psig inside the waterboxes caused by the elevation difference between the ClearSky© Cooling Tower and the Condensers, BREI used the information from the Unit 3 Condensers, since the required information is readily available in the Unit 3 Condenser Manual.

The equations for the minimum bolt load for a bolted flange connection of circular design are as follows:

$$W_{m1} = (hydrostatic\ end\ force) + (residual\ gasket\ load) \quad \text{(Eqn. 1)}$$

$$W_{m1} = \left(\frac{1}{4}\pi G^2 P\right) + (2\pi GbmP)$$

$$W_{m2} = \pi Gby \quad \text{(Eqn. 2)}$$

Where:

$W_{m1}$ = Minimum required bolt load for operating conditions, $lb_F$

$W_{m2}$ = Minimum required bolt load for gasket seating, $lb_F$

$G$ = Diameter of location of Gasket Load Reaction, inches

$P$ = Internal Design Pressure, psig (30 psig)

$m$ = Gasket Factor (value is 0.5 for elastomers with durometer rating of 75 or less, such as neoprene))

$y$ = minimum gasket seating stress, psi (value is 0 for neoprene with durometer rating less than 75. However, use 200 psi.

Note: Generally speaking, gaskets stresses in elastomers should be kept low, between 400 and 900 psi).

$b$ = effective sealing width, inches

$b_0$ = basic gasket seating width, inches (4 inches)

$b$ = $b_o$ for $b_o \leq$ ¼" and b = $0.5\sqrt{b_0}$ for $b_o \geq$ ¼"

**Determination of $W_{m1}$**

Equation 1 is the sum of the hydrostatic end force $\left(\frac{1}{4}\pi G^2 P\right)$ plus a residual gasket load on the contact area $(2\pi GbmP)$. This equation requires that the minimum bolt load be such that it will maintain a residual unit compressive load on the gasket area that is greater than internal pressure, when the total load is reduced by the hydrostatic end force (i.e. de-pressurized condition). Equation 2 determines the minimum bolt load to seat the gasket regardless of the internal pressure.

Since these equations are based upon a circular flanged design, we will adjust them based the actual configuration of the waterboxes as follows:

A. Area upon which hydrostatic end force acts, $A = \frac{1}{4}\pi G^2$ inches$^2$ (circular flange)

Per drawings 93-4377-5-101-C and 93-4377-5-107-C of FWEC Condenser Manual for Unit 3 (FW Contract 4-87-4877) the waterbox flanged dimensions (including flanges) are as follows:

| Trapezoidal Top Portion | Dimension |
|---|---|
| $Base_1$ (in.) | 79.625 |
| $Base_2$ (in.) | 128.0 |
| $Height_1$ (in.) | 125.75 |
| **Rectangular Bottom Portion** | **Dimension** |
| $Base_3$ (in.) | 128.0 |
| $Height_2$ (in.) | 97.0 |

These dimensions yield an area of 25,470 in$^2$ (which conservatively includes flanged area) upon which the internal pressure (30 psi) exerts a force. The resultant force is 764,112 $lb_F$, which is the first component of Equation 1.

B. Area of effective gasket sealing , inches = $2\pi Gb$

The perimeter ($P$) of the waterbox replaces the $2\pi G$ term as follows:

$$P = \sum l$$
$$P = 79 + 128 + 97 + 128 + 97 + 128$$
$$P = 657\ inches$$

The width of the waterbox effective gasket sealing area under the flange is:

$$width = 2b$$
$$width = 2\left(\frac{\sqrt{4}}{2}\right)$$
$$width = 2\ inches$$

Therefore the effective gasket sealing area is:

$$Area = (2)(657)$$
$$Area = 1314\ in^2$$

This results in the second component of equation 1 being:

$$2b(3.14G)mP$$
$$(1314)(0.5)(30)$$
$$19{,}710\ lb_F.$$

$$W_{m1} = 764{,}112 + 19{,}710 = 783{,}822\ lb_F$$

**Determination of $W_{m2}$**

Equation 2 determines the minimum bolt load to seat the gasket regardless of the internal pressure.

$$W_{m2} = \pi Gby$$

As seen in item B above, $2\pi Gb$ was calculated to be 1314 $in^2$. So $\pi Gb$ would be half that, which is 657$lb_F$. Since y is being taken as 200 psi:

$$W_{m2} = (\pi Gb)y$$
$$W_{m2} = (657)(200)$$
$$W_{m2} = 131{,}400\ lb_F$$

Per drawing 93-4377-5-107-C of FWEC Condenser Manual for Unit 3 (FW Contract 4-87-4877), there are 178 one-inch bolts for each waterbox.

After $W_{m1}$ and $W_{m2}$ are determined, the minimum required bolt area is determined as follows:
$A_{m1} = W_{m1}/S_b$ where $S_b$ is the allowable bolt stress at operating temperature; and
$A_{m2} = W_{m2}/S_a$ where $S_a$ is the allowable bolt stress at atmospheris temperature.

Assuming medium strength bolting material such was as A307, Grade B, the allowable stress is 7 Ksi at atmospheric and operating temperatures. Therefore, $A_{m1}$ dictates the bolt area/size.

$A_{m1} = W_{m1}/S_b = $ 783,822 $lb_F$/7000 psi = 112 $in^2$ total area or 0.63 $in^2$/bolt. Actual bolts are 1" diameter with an area of 0.79 $in^2$/bolt.

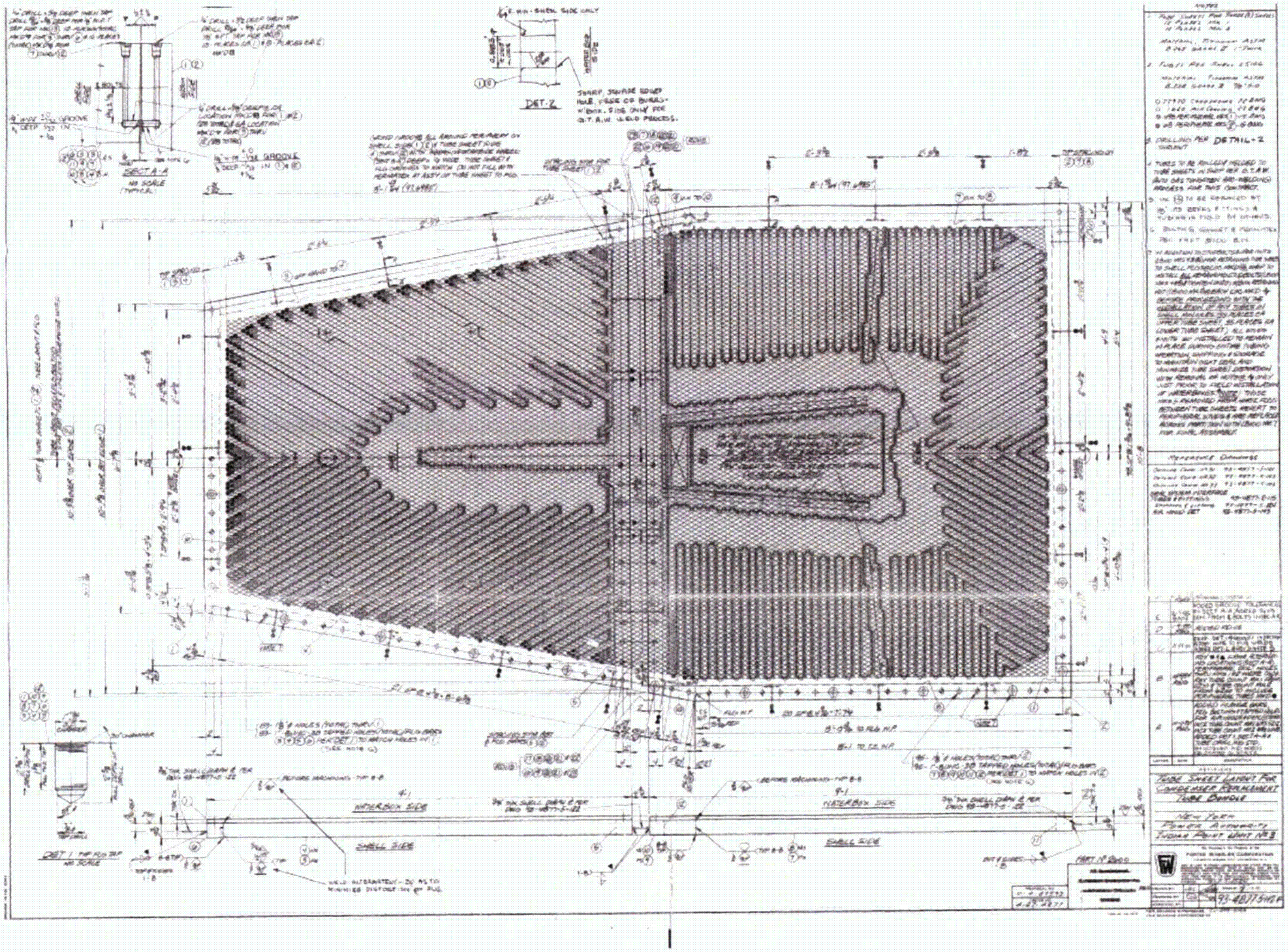
Torque = $F_P$ (KD)/12 where $F_P$ is 4403 $lb_F$ (i.e. 783,822 $lb_F$/178 bolts), K is the nut factor which ranges from .0158 (min torque) to 0.32 (max torque), and D is the bolt diameter = 1 inch.

$$Torque_{min} = 4403(0.158 * 1)/12 = 58 \text{ Ft-Lbs}$$

$$Torque_{max} = 4403(0.32 * 1)/12 = 117 \text{ Ft-Lbs}$$

The condenser tube bundle flange and bolt configuration is depicted on Figure3 below.

Figure 3 Condenser Tube Bundles (Reference 7)



## 7. Circulating Water Piping to Waterbox Expansion Joint Stresses

### a) *Working Pressure*

The Unit 2 circulating water to waterbox expansion joints are designed to withstand of an operating pressure of 50 psig (Reference 6). The Unit 3 circulation water to waterbox expansion joints are designed to withstand of a pressure of 40 psig and a test pressure of 45 psig (Reference 7). Referring to FWEC drawing 93-4877-5-102-C, the expansion joint is at approximately 12' 7" elevation. BREI is assuming that Unit 2 installation is similar. From the Tetra Tech report (Reference 1) the proposed ClearSky© cooling tower basins will be at the approximate 52' 6" elevation. The Tetra Tech report also establishes that the normal operating basin level will be approximately 4'. Therefore, the normal operating level will be at approximately the 56' 6" elevation. This translates to a static head of approximately 44' or approximately 19 psig acting on the expansion joint. This is well below the design rated pressure for both Unit 2 and 3 expansion joints and should not pose additional challenges to the structural integrity of the circulating water to waterbox expansion joints.

### b) *Axial and Lateral Deflection*

After reviewing the operating parameters for the proposed ClearSky© closed-cooling system, BREI opines that the resulting axial and lateral deflections will not exceed those induced by the existing open-cycle cooling system currently in use at U2 and U3.

## 8. 50% High Pressure Steam Dump (40% condenser steam flow) - impact on turbine trip setpoints.

This calculation determines the saturation temperature and, therefore, saturation pressure in the condenser given for Unit 2 and Unit 3. Burns and Roe follows the methodology presented in the *"Indian Point Cooling Tower Alternative Assessment" Report No. 11431116100-Rep-R0001-02 Appendix F: Turbine Backpressure Calculations* (Reference 1). Reference 1 provides the following assumed inputs:

| Parameter | Unit 2 | Unit 3 |
|---|---|---|
| Condenser Heat Load (BTU/hr) | $3.035\ (10)^9$ | $3.371(10)^9$ |
| Heat Transfer Surface Area ($ft^2$) | 315 700 | 286 125 |

### *a) Unit 2*

The cooling water flow rate through the condensers is calculated using Equation 1 as follows:

$$\dot{Q}_o = \dot{Q}_i - \dot{Q}_e - \dot{Q}_d - \dot{Q}_b \qquad \text{(Eqn. 1)}$$

where:

$\dot{Q}_o$ = Cooling water flow rate
$\dot{Q}_i$ = Inlet flow rate
$\dot{Q}_e$ = Evaporation flow rate
$\dot{Q}_d$ = Drift flow rate
$\dot{Q}_b$ = Blowdown flow rate

Inputs for flow rates are sourced from Reference 1. Using these inputs and solving for cooling water flow rate through the condenser:

$$\dot{Q_o} = \frac{(700\ 000) - (12\ 650) - (3.5) - (6\ 322)}{3}$$

$$\boxed{\dot{Q_o} = 227\ 008\ gpm}$$

The water velocity in the condenser tubes is calculated in §1 and is corrected for the reduced flow rate considering evaporation, drift, and blowdown.

| Parameter | Unit 2 | Unit 3 |
|---|---|---|
| Tube Flow Velocity (ft/s) | 4.96 | 5.50 |

Equation 2 is used in order to solve for the log-mean temperature difference given the known heat transfer rate, heat transfer surface area, and calculated heat flux.

$$Q = U \cdot A \cdot LMTD \qquad \text{(Eqn. 2)}$$

where:

$Q$ = Heat transfer rate

$U$ = Heat flux

$A$ = Heat transfer surface area

$LMTD$ = Log-mean temperature difference

Following the methodology in Reference 1 sourced from the "Heat Exchange Institute: Standard for Steam Surface Condensers, 9th Edition" (Reference 16), heat flux is calculated in Equation 3 as follows:

$$U = U_1 \cdot F_W \cdot F_M \cdot F_c \qquad \text{(Eqn. 3)}$$

Using inputs from Reference 1:

$$U = (588.1)(1.072)(0.91)(0.85)$$

$$U = 487.6 \frac{BTU}{hr \cdot ft^2 °\text{F}}$$

Substituting values into Equation 2 yields:

$$(3.035(10)^9) = (487.6)(315\,700)LMTD$$

$$LMTD = 19.72\ °\text{F}$$

Cooling water outlet temperature ($T_2$) is calculated using Equation 4:

$$T_2 = \frac{Q}{\dot{Q}_o} + T_1 \qquad \text{(Eqn. 4)}$$

where:

$$T_2 = \frac{3.035(10)^9}{(227\,008)(500)(1)} + 89$$

$$\boxed{T_2 = 115.7\ °\text{F}}$$

Finally, LMTD, cooling water inlet temperature ($T_1$), and cooling water outlet temperature ($T_2$) are used to calculate the saturation temperature ($T_s$) in Equation 5:

where:

$$LMTD = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_S - T_2}\right)} \qquad \text{(Eqn. 5)}$$

Substituting known values yields:

$$LMTD = \frac{115.7 - 89}{\ln\left(\frac{T_s - 89}{T_S - 115.7}\right)}$$

Solving for $T_S$:

$$19.72 = \frac{26.7}{\ln\left(\frac{T_s - 89}{T_S - 115.7}\right)}$$

$$\ln\left(\frac{T_s - 89}{T_S - 115.7}\right) = 1.354$$

$$\frac{T_s - 89}{T_S - 115.7} = 3.873$$

$$\boxed{T_s = 125.0\ \text{°F}}$$

At a $T_S$ of 125.0 °F, pressure corresponds to 1.94 psia (3.95 in-Hga). Converting to gauge pressure yields -12.75 psig (-25.96 in-Hg).

$$\boxed{\textbf{\textit{Unit 2 Condenser Backpressure}} = \mathbf{3.95}\ \boldsymbol{in \cdot Hg}}$$

The calculated limiting maximum pressure is 3.95 in-Hga in Condenser No. 21 for a 50% load rejection condition given a total flow rate of 700,000 gpm and cooling water inlet temperature of 89 °F. This pressure is below the alarm and trip setpoints and is, therefore, acceptable.

### b) Unit 3

Using Equation 3 and the inputs from Reference 1:

$$U = (616.8)(1.072)(0.91)(0.85)$$

$$U = 511.45\ \frac{BTU}{hr \cdot ft^2 \cdot \text{°F}}$$

Substituting values into Equation 2 and solving for the log-mean temperature difference (LMTD):

$$(3.726(10)^9) = (511.45)(286\ 125)LMTD$$

$$\boxed{LMTD = 25.46\ \text{°F}}$$

Substituting values into Equation 4:

$$T_2 = \frac{3.726(10)^9}{(227\,008)(500)(1)} + 89$$

$$\boxed{T_2 = 121.8\ °\mathrm{F}}$$

Substituting values into Equation 5 and solving for saturation temperature ($T_s$):

$$25.46 = \frac{121.8 - 89}{\ln\left(\frac{T_s - 89}{T_s - 121.8}\right)}$$

$$\ln\left(\frac{T_s - 89}{T_s - 121.8}\right) = 1.288$$

$$\frac{T_s - 89}{T_s - 121.8} = 3.627$$

$$\boxed{T_s = 134.3\ °\mathrm{F}}$$

At a $T_s$ of 134.3 °F, pressure is equal to 2.49 psia (5.07 in-Hga). Converting to gauge pressure yields -12.20 psig (-24.84 in-Hg).

$$\boxed{\boldsymbol{Unit\ 3\ Condenser\ Backpressure = 5.07\ in \cdot Hg}}$$

The calculated average pressure is 5.07 in-Hga for a 50% load rejection condition given a total flow rate of 700,000 gpm and cooling water inlet temperature of 89 °F. This pressure is below the turbine trip set point (12 in-Hga), but higher than the low vacuum alarm set point of 4.0 in-Hga. Site Procedure SOP-TG-4 requires a manual trip of the turbine at 4.5 in-Hga and the steam dump valves are prohibited from opening at pressure greater than 5 in-Hga. Therefore, the condenser pressure of 5.07 in-Hga is unacceptable and measures such as installing additional heat removal capacity should be taken to reduce it below 4.0 in-Hga or, at least, 4.5 in-Hga.

# III. References

1 Tetra Tech, Indian Point Closed-Cycle Cooling System Retrofit Evaluation, 114311161000-REP-R0001-02, prepared for New York State Department of Conservation, June 2013.

2 Bid specification for Installation of Main Condenser Tube Bundle Modules, Project No. 06958-42, Indian Point Generating Station – Unit No. 2, Doc I.D. MP-89-042.

3 Consolidated Edison Co., Indian Point Station, Modification MBX-94-01003-M, Replace No. 23 Condenser, Close Out Package

4 New York Power Authority Indian Point No. 3, MOD-85-03-069 COND, Condenser Tube Bundle Replacement.

5 New York Power Authority Indian Point No. 3, MOD-85-03-070 COND, Condenser Flange Seal System.

6 Foster Wheeler, Instructions for the Care and Operation of Surface Condenser Tubed Modules with Waterboxes, Installed for Consolidated Edison Co., Indian Point Unit 2, September 1992.

7 Foster Wheeler, Instructions for the Care and Operation of Surface Condenser Tubed Modules with Waterboxes, Installed for New York Power Authority, Indian Point Unit 3, May 1985.

8 Entergy Calculation IP-CALC-05-00600 (SWEC #59379-MU(S)-007 Rev. 1), IP-3, Main Condenser Evaluation at Power Uprate Conditions, 8/6/04.

9 Entergy Calculation FMX-00339-00 (SWEC #58030-MU(S)-007 Rev. 0), IP-2, Main Condenser Evaluation at Power Uprate Conditions, 10/23/03.

10 Con Edison Calculation No. MM-00015-00, I.P. Condenser Torque Value Required for Adequate Gasket Compression.

11 Con Edison Calculation No. MM-00003-00, I.P. Condenser Tube Failure Analysis

12 Indian Point Energy Center Unit No. 2, System Description No. 20, Revision 12, Condensate System.

13 Indian Point Energy Center Unit No. 2, System Description No. 23, Revision 7, Circulating Water System.

14 Indian Point Energy Center Unit No. 3, System Description No. 20, Revision 12, Condensate System.

15 Indian Point Energy Center Unit No. 3, System Description No. 23, Revision 5, Circulating Water System.

16 Heat Exchange Institute, Inc., Standards for Steam Surface Condensers, 9th Edition, Copyright 1995.

# ATTACHMENT 4

Review of Powers Engineering Closed-Cycle Cooling Report and Riverkeeper Selected Closed-Cycle Cooling Configuration

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# LIST OF FIGURES

# 1 Powers Engineering Closed-Cycle Cooling Report Overview

## 1.1 Statement of Issue

The Powers Engineering Revised Closed-Cycle Cooling Feasibility Assessment for Indian Point Energy Center Unit 2 and Unit 3 for Best Technology Available Report, dated October 24, 2012 [Ref. 3.1], and supplemental information submitted in September 2013 [Refs. 3.2 and 3.3], is not detailed enough to provide a meaningful engineering review and technology evaluation. The potential construction and operation of closed-cycle cooling is discussed in general, and various details are provided for a number of closed-cycle cooling configurations; however, the Powers Engineering report lacks the level of engineering design necessary to establish feasibility, or to conduct a meaningful engineering review of the closed-cycle cooling configuration selected by Riverkeeper as a best technology available (BTA) [Ref. 3.4].

For example, Powers Engineering's September 11, 2013 supplement provides a 17°F approach to wet-bulb temperature at the design point of 76°F for the SPX and GEA[1] plume-abated cooling towers; however, cooling tower performance curves (i.e., the curve necessary to determine the approach to wet-bulb temperature over the range of anticipated wet-bulb temperatures) are not provided. The only cooling tower performance curves provided are in Attachment B of the Powers Engineering report and are for a much smaller SPX cooling tower configuration operating at 110,000 gpm, at a 13°F approach to wet-bulb at a 76°F design point, and without a listed plume point (i.e., presumably a cooling tower performance curve for a non-plume abated cooling tower) [Ref. 3.1]. Without a cooling tower performance curve for each selected tower, it is not possible to calculate the operational power losses except for the rare instance when the wet-bulb temperature is 76°F. Without knowing these operational losses, it is impossible to determine if plant operation with the proposed towers is economically feasible or even possible during peak wet-bulb temperature conditions.

For comparison purposes, operational power losses at a wet-bulb temperature of 76°F were calculated for the Tetra Tech and Riverkeeper selected closed-cycle cooling configurations following the methodology outlined in Section 9.3 of ENERCON's Response to the Tetra Tech report. The Riverkeeper closed-cycle cooling configuration resulted in an additional operational power loss of 41.6 MWe above that of the Tetra Tech configuration. This comparison is limited to a wet-bulb temperature of 76°F as Powers Engineering does not provide a cooling tower performance curve for the Riverkeeper selected closed-cycle cooling configuration.

[1] Page 1 of the Powers Engineering September 11, 2013 supplement lists a 17°F approach temperature at a 78°F wet-bulb design temperature; however, a 17°F approach temperature at a 76°F wet-bulb design temperature is assumed in accordance with the values listed by GEA in Attachment 2 of the supplement.

An assessment of the difference in operational power losses between the Tetra Tech and Riverkeeper configurations at a wet-bulb temperature of 76°F and a discussion of the engineering design required for technology selection is discussed in the following section.

## 1.2 Operational Effects of Riverkeeper's Selected Closed-Cycle Cooling Configuration

Riverkeeper's selected cooling tower configuration for Indian Point utilizes a 28-cell in-line, plume-abated, mechanical draft cooling tower for each Unit. The Riverkeeper selected cooling tower would operate with a flow rate of 600,000 gpm, a design wet-bulb temperature of 76°F, and an approach to wet-bulb temperature of 17°F [Ref. 3.2]. For comparison, Tetra Tech's cooling tower configuration utilizes a 44-cell back-to-back ClearSky cooling tower for each Unit, operating at a flow rate of 700,000 gpm, a design wet-bulb temperature of 77°F, and an approach to wet-bulb temperature of 12°F.

The cooling tower configuration selected by Riverkeeper utilizes a significantly higher approach to wet-bulb temperature than does Tetra Tech. The approach to wet-bulb temperature is used to determine the temperature the cooling tower is capable of cooling the condenser inlet water temperature to at the design wet-bulb temperature. For example, the Riverkeeper selected cooling tower configuration would be capable of cooling the closed-cycle cooling water to 93°F (17°F + 76°F) when the ambient wet-bulb temperature at the cooling towers is 76°F. Increasing the approach to wet-bulb temperature decreases the cooling tower size, but results in greater circulating water temperatures. The figure below provides a graphical representation of how cooling tower size is affected by the approach to wet-bulb temperature selected, assuming a fixed heat load, flow rate, and design wet-bulb temperature.

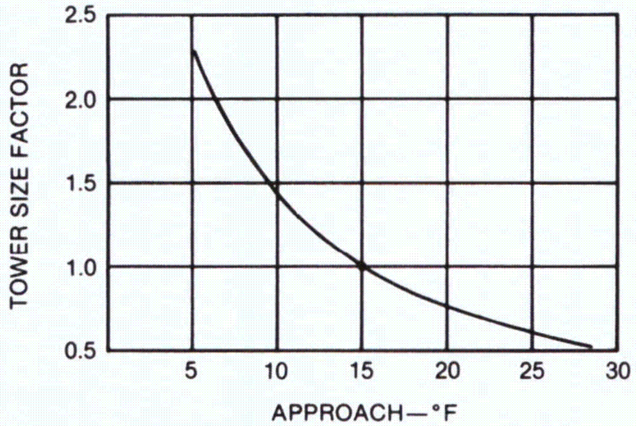


Figure 1: Effect of Approach to Wet-Bulb Temperature on Cooling Tower Sizing

The increased circulating water temperature and reduced flow rate specified by Riverkeeper's cooling tower configuration impacts Indian Point's ability to generate electricity, may result in Unit down-powering during times in excess of the condenser operational limits, and does not take into account transient or accident analysis. The balance of plant systems at each Unit relies on the condenser's ability to remove the design heat load from the system. Increasing the circulating water temperature decreases the condenser vacuum, resulting in reduced power generation at the turbines. Similarly, decreasing the circulating water flow rate from the current design of 840,000 gpm to 600,000 gpm (or a decrease of approximately 29%) results in further power reduction.

It should be noted that Consolidated Edison (ConEd) completed closed-cycle cooling evaluations for Indian Points Units 2 and 3 in 1974 and 1976, respectively [Refs. 3.6 & 3.7]. These reports utilized a circulating water flow rate of 600,000 gpm and either natural draft cooling towers with an approach to wet-bulb temperature of 16°F or mechanical draft cooling towers with an approach to wet-bulb temperature of 17°F (both at a design wet-bulb temperature of 74°F). Significant modifications to Indian Point Units 2 and 3 have occurred since the mid-1970s, including several power uprates. These modifications are accounted for in the Performance Evaluation of Power System Efficiency (PEPSE) model used in ENERCON's analysis. While the ConEd reports provide general background information on closed-cycle cooling technologies, they do not reflect the current configuration of Indian Point Units 2 and 3. As shown in the discussion below for the current plant configuration, closed-loop cooling utilizing a 600,000 gpm circulating water flow rate and a cooling tower with an approach to wet-bulb temperature of 17°F would impact Indian Point's ability to generate electricity and may exceed condenser operational limits during both normal and transient conditions.

Using the PEPSE model as described in Section 9.3 of ENERCON's Response to the Tetra Tech Report, the operational power losses were calculated for both the Riverkeeper and Tetra Tech cooling tower configurations at a 76°F wet-bulb temperature. For ClearSky cooling towers, the effect of recirculation for the most common wind direction (NNE) at the average site wind speed is included. Therefore, the entering wet-bulb temperature is 76.1°F for Unit 2 and 79.0°F for Unit 3 (Attachment 2). Operational power losses are calculated using a flow rate of 700,000 gpm and the same cooling tower performance curve as used in Section 9.3. For Riverkeeper's cooling tower configuration, the effect of recirculation is not included, representing a best case scenario for cooling tower performance. Operational power losses for the Riverkeeper cooling tower configuration are calculated using a flow rate of 600,000 gpm and a cooling tower cold water temperature of 93°F.

Based on this analysis, the Riverkeeper selected cooling tower configuration would result in an operational power loss of 23.0 MWe and 18.6 MWe greater than the operational power losses using ClearSky cooling towers for Units 2 and 3, respectively. Again, this comparison is limited to a wet-bulb temperature of 76°F as Powers Engineering does not provide a cooling tower performance curve for the Riverkeeper selected closed-cycle cooling configuration.

Without a cooling tower performance curve, it is not possible to determine operational parameters across the range of wet-bulb temperatures expected. Given the significant increase in circulating water temperature and reduction in flow rate, it is possible that there will be times when plant operation would be in excess of condenser operational limits. This would result in a Unit down-power and would occur during high ambient temperatures when power demand is at its highest. Once a cooling tower performance curve is provided, additional analysis will be necessary to determine the frequency of these potential down-powers by Unit.

As discussed in ENERCON's Response to the Tetra Tech report, a review of the transient and accident analysis is necessary for any significant plant modification to ensure that there are no potential impacts to the operation of Indian Point. BREI reviewed one transient condition for the Tetra Tech closed-cycle cooling configuration, identifying that the increased condenser backpressure would be higher than the low vacuum alarm setpoint during a high pressure steam dump at Unit 3 and would trip the Unit. While BREI's analysis was for the Tetra Tech's closed-cycle cooling configuration, the effects would be magnified under the Riverkeeper selected cooling tower configuration at increased circulating water temperature and reduced flow rate. The BREI analysis was limited to one transient condition and emphasizes the need to identify and resolve any impacts to the transient or accident analysis before concluding the feasibility of such a significant plant modification. Based on the appreciably smaller capacity of the Riverkeeper selected cooling tower configuration, there is every reason to conclude that condenser operation would be negatively impacted during both normal and transient conditions.

## 1.3 Riverkeeper Closed-Cycle Cooling Configuration Selection

Riverkeeper selected plume abated mechanical draft in-line 28-cell closed-cycle cooling towers for Indian Point Units 2 and 3; however, the Powers Engineering report lacks the level of engineering design necessary to conduct a meaningful engineering review of the Riverkeeper selected closed-cycle cooling configuration. The following sections provide a listing of the type of design information required, but not included in the Powers Engineering report or supplemental information.

### 1.3.1 Cooling Tower Elevation

Additional design and schedule information is needed to evaluate the structural improvements that would be necessary to the existing condenser to support the additional pressure from a cooling tower elevation of approximately 100 ft [Ref. 3.1]. As the elevation of the Riverkeeper selected closed-cycle cooling configuration is nearly double that used by Tetra Tech (52.5 ft) in their closed-cycle cooling configuration, impacts from the increased pressure are expected to be substantially increased. Identification, cost estimate, and installation schedule for the necessary improvements is needed.

### 1.3.2 Air Quality Analysis

Discussion is needed on what impact drift particulate is expected to have on plant equipment and the health effect of particulate matter (PM2.5) emissions from the cooling towers.

### 1.3.3 Blasting Considerations/Groundwater, Soil and Blast Spoil Contamination

The location and amount of blasting necessary for localized grading activities and piping trenches needs to be defined.

Additional analysis (charge size, blasting techniques, schedule, etc.) is needed to identify any potential impact blasting would have on plant operations. Vibrations from blasting need to be evaluated to ensure they are below plant equipment limitations, particularly sensitive electronic equipment. Once a blasting plan is completed, the availability and cost for insurance to provide the necessary coverage to conduct blasting with Indian Point Units 2 and 3 online must be determined and included in the cost assessment

Project risk and potential construction delays associated with groundwater, soil and blast spoil contamination removal, including the cost, need to be addressed and added to the project schedule.

### 1.3.4 Closed-Cycle Cooling Water Treatment

A cost and construction schedule for supporting chemical treatment required for cooling towers needs to be provided for each proposed closed-cycle cooling tower option for Indian Point Units 2 and 3. These chemical treatment facilities should also be located on a site layout drawing to determine any plant interferences.

### 1.3.5 Parasitic and Operational Power Losses

Structural changes to the condenser to accommodate the additional pressure from cooling towers located at a higher elevation may dramatically reduce the condensers ability to reject heat. Operational power losses need to be provided for a condenser post structural improvement.

### 1.3.6 Siting/Space Constraints

Layout drawings are not to scale. Scale drawings are needed including the depth and width of construction to allow for a review of the impact to existing plant equipment. Discussion on where existing buildings/equipment/structures that are impacted by construction (including laydown areas) would be relocated is needed (and at what cost).

### 1.3.7 Cooling Tower Recirculation and Interference Impacts

Calculation of cooling tower recirculation and interference is necessary (along with the cooling tower performance curve) to determine the operational impacts of the Riverkeeper selected closed-cycle cooling configuration.

### 1.3.8 Safety-Related Implications

Discussion is needed on what safety-related systems/equipment could be impacted by construction or operation of closed-cycle cooling retrofit at Indian Point Energy Center.

If any safety-related systems are impacted by the proposed closed-cycle cooling towers, a review should be done on whether or not a license amendment is necessary. In addition a discussion on whether the NRC would issue such a license amendment is also required.

### 1.3.9 Schedule for Implementation

Activity sequencing and duration information is needed for the Riverkeeper selected closed-cycle cooling configuration to illustrate the expected scheduling of design, permitting, procurement, construction, and start-up of the closed-cycle cooing system at each Unit. A schedule for implementation specific to the Riverkeeper selected closed-cycle cooling configuration is necessary to determine the cumulative biological effectiveness of the technology and must have sufficient background for support.

# 2 Conclusion

The Powers Engineering report is not detailed enough to provide a meaningful engineering review and technology evaluation. The potential construction and operation of closed-cycle cooling is discussed in general, and various details are provided for a number of closed-cycle cooling configurations in the report. The Powers Engineering report lacks the level of engineering design necessary to conduct a meaningful engineering review of the closed-cycle cooling configuration selected by Riverkeeper as BTA.

For comparison purposes, operational power losses at a wet-bulb temperature of 76°F were calculated for the Tetra Tech and Riverkeeper selected closed-cycle cooling configurations. The Riverkeeper closed-cycle cooling configuration resulted in an additional combined operational power loss of 41.6 MWe above that of the Tetra Tech configuration. This comparison is limited to a wet-bulb temperature of 76°F as Powers Engineering does not provide a cooling tower performance curve for the Riverkeeper selected closed-cycle cooling configuration.

The increased circulating water temperature and reduced flow rate specified by Riverkeeper's cooling tower configuration impacts Indian Point's ability to generate electricity, may result in Unit down-powering during times in excess of the condenser operational limits, and does not take into account transient or accident analysis.

# 3 References

**3.1** Powers Engineering, "Revised Closed-Cycle Cooling Feasibility Assessment for Indian Point Energy Center Unit 2 and Unit 3 for Best Technology Available Report," for Riverkeeper, Inc., October 24, 2012.

**3.2** Powers Engineering, "Powers Engineering Cost Estimate - Indian Point Units 2 and 3 Cooling Towers," September 6, 2013.

**3.3** Powers Engineering, "Supplement to September 6, 2013 Powers Engineering Cost Estimate for Indian Point Units 2 and 3 Cooling Towers," September 11, 2013.

**3.4** Riverkeeper, Inc., Letter "Re: Entergy Indian Point Nuclear Units 2 and 3, CWA § 401 Water Quality Certification Appeal (DEC Nos.: 3-5522-00011/0030 (IP2) and 3-5522-00105/0031 (IP3)) and SPDES Permit Renewal Proceeding (DEC No. 3-5522-0011/00004; SPDES No. NY-00004472) – Riverkeeper BTA Selection Offer of Proof/Reservation of Rights with Respect to Alternative Options," November 22, 2013.

**3.5** SPX Cooling Technologies, Inc., "Cooling Tower Fundamentals," 2nd Edition, 2009, Overland Park, Kansas.

**3.6** Consolidated Edison Company of New York, Inc., "Economic and Environmental Impacts of Alternative Closed-Cycle Cooling Systems for Indian Point Unit No. 2," December 1, 1974.

**3.7** Consolidated Edison Company of New York, Inc., "Economic and Environmental Impacts of Alternative Closed-Cycle Cooling Systems for Indian Point Unit No. 3," January 1976.

# ANALYSIS OF CLOSED-LOOP COOLING SALINITY LEVELS INDIAN POINT UNITS 2 & 3

**Prepared for**
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**November 2010**

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# Executive Summary

This report describes supplemental analyses and closed-loop operational scenarios for compliance with regulatory requirements for air emissions. The updated make-up flow rates to reduce closed-loop cooling salinity are presented along with the corresponding emissions of particulate matter, based on the recent Applied Science Associates, Inc. (ASA) salinity analysis.

In the 2003 Report "Economic and Environmental Impacts Associated with Conversion of Indian Point Units 2 and 3 to a Closed-Loop Condenser Cooling Water Configuration" (2003 Closed-Loop Cooling Report), the salinity of a closed-loop system for Indian Point Energy Center (IPEC) was determined to be 7.2 practical salinity units (psu), employing a constant (average) factor for Hudson River salinity of 1.8 psu. This salinity level was the basis for the air quality analyses of cooling tower particulate emissions performed by TRC Companies, Inc. (TRC) in 2009. The recent ASA salinity analysis has indicated that, although the 1.8 psu average is correct, the Hudson River salinity is highly variable and often significantly greater than 1.8 psu for extended periods of time. As a result, if installed, a closed-loop system at IPEC would not be able to maintain 7.2 psu, as previously evaluated by TRC.

This report evaluates how a closed-loop system would need to operate, given the recent salinity information provided by ASA and the associated air quality analyses performed by TRC. As detailed below, and summarized in the results section of this report, there is an essential trade-off between closed–loop cooling operation and air quality, given prevailing salinity conditions of the Hudson River. According to TRC, the closed-loop cooling system cannot reasonably be expected to comply with air quality standards if operated for substantial periods of time (including most of the summer months) given the expected Hudson River salinity values. As a result, previous assumptions about closed-loop cooling operations and configurations (contained in both the 2003 Closed-Loop Cooling Report and the "2010 Engineering Feasibility and Costs of Conversion of Indian Point Units 2 and 3 to a Closed-Loop Condenser Cooling Water Configuration" (2010 Closed-Loop Cooling Report)) require updating.

Closed-loop cooling requires make-up water to replace water lost in evaporation and drift from the cooling towers, and to allow blowdown from the closed-loop system to maintain water quality within the system. As defined in the 2010 Closed-Loop Cooling Report, the IPEC closed-loop cooling system would draw its make-up water from the service water (SW) discharged from each Unit, which reflects the salinity of the Hudson River. According to TRC, Hudson River water salinity is the primary contributing factor to emissions. The evaluated mechanism for controlling air emissions is to limit salinity in the system through alteration of the cooling tower operations, specifically cycles of concentration, or reverting to once-through cooling (bypassing the cooling towers). Theoretically, if the River salinity is sufficiently low, it can be used for closed-loop cooling; however, as the ASA salinity analysis shows, River salinity is high for extended periods of time. This salinity effectively constrains cooling tower operations, requiring the closed-loop system to revert to once-through cooling in order to avoid exceeding the $PM_{2.5}$ national ambient air quality standards (NAAQS) and $PM_{2.5}$ Significant Impact Levels (SIL).

TRC evaluated the exceedance of the $PM_{10}$ and $PM_{2.5}$ NAAQS and SIL that would result from operation of closed-loop cooling at IPEC. TRC determined that to avoid exceeding the $PM_{2.5}$ NAAQS with 1.5 cycles of concentration, the Hudson River salinity would have to be 0.846 psu or less. The limiting ground level concentration in the Westchester County $PM_{2.5}$ non-attainment area is the SIL; to avoid exceeding the $PM_{2.5}$ SIL with 1.5 cycles of concentration, TRC determined that the Hudson River dissolved solids would have to be 0.175 psu or less. These values represent make-up water salinity (i.e., Hudson River water salinity), which is the primary contributing factor to emissions.

TRC's analysis provided the basis for determining the operating profiles for closed-loop cooling based upon the Hudson River salinities (i.e., cooling tower make-up water salinities). In order to avoid exceeding the $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL under any meteorological condition, a "$PM_{2.5}$ NAAQS No Exceedance" and a "$PM_{2.5}$ SIL No Exceedance" scenario was run to determine how often IPEC would be forced to revert from closed-loop to once-through operation. While no detailed design work on a system that would allow switching from closed-loop to once-through operation at IPEC has been performed, operating constraints would likely limit the switch to a seasonal basis; however, this Report conservatively assumes the switch between closed-loop and once-through operation would be determined on a weekly basis (although impractical for actual Station operation). In addition, the closed-loop cooling configuration described in the 2003 and 2010 Closed-Loop Cooling Reports would have to be revised to accommodate switching between closed-loop cooling and once-through cooling (bypassing the cooling towers). The need to switch between once-through and closed-loop cooling may have substantial design, construction, operational, and cost ramifications.

In order to avoid exceeding the $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL, operation of closed-loop cooling would be expected to occur no more than 43% and 13% of the year, respectively. Operation of closed-loop cooling 43% of the time would result in reductions in entrainment, entrainment losses, and equivalent age 1 losses of 57.4%, 63.8%, and 56.6%, respectively; moreover, the $PM_{2.5}$ SIL would still be exceeded. Operation of closed-loop cooling 13% of the year would result in reductions in entrainment, entrainment losses, and equivalent age 1 losses of 26.7%, 41.4%, and 38.5%, respectively. For comparison, the reductions in equivalent age 1 losses for cylindrical wedgewire screens would be approximately 89.8%, as presented in Attachment 6 of the 2010 "Evaluation of Alternative Intake Technologies at IPEC Units 2 and 3" (2010 Alternative Technologies Report). Likewise, the reductions in equivalent age 1 losses associated with the existing technology and operational suite employed by Entergy (i.e., Ristroph screens and fish handling and return systems, as well as flow reductions due to variable and dual speed pumps and maintenance outages) are approximately 33.8%.

# 1 Introduction

## 1.1 Purpose

In the 2003 Report "Economic and Environmental Impacts Associated with Conversion of Indian Point Units 2 and 3 to a Closed-Loop Condenser Cooling Water Configuration" (2003 Closed-Loop Cooling Report) [Ref. 6.1], the salinity[1] of a closed-loop system for Indian Point Energy Center (IPEC) was determined to be 7.2 practical salinity units (psu), employing a constant (average) factor for Hudson River salinity of 1.8 psu. This salinity level was the basis for the air quality analyses of cooling tower particulate emissions performed by TRC Companies, Inc. (TRC) in 2009. The recent Applied Science Associates, Inc. (ASA) salinity analysis has indicated that, although the 1.8 psu average is correct, the Hudson River salinity is highly variable and often significantly greater than 1.8 psu for extended periods of time. As a result, if installed, a closed-loop system at IPEC would not be able to maintain 7.2 psu, as previously determined and evaluated by TRC.

This report describes supplemental analyses and closed-loop operational scenarios for compliance with regulatory requirements for air emissions. The updated make-up flow rates to reduce closed-loop cooling salinity are presented along with the corresponding emissions of particulate matter, based on the recent ASA salinity analysis.

## 1.2 Scope

This report evaluates how a closed-loop system would need to operate, given the recent salinity information provided by ASA (Appendix F) and the associated air quality analysis performed by TRC (Appendix C). As detailed below, and summarized in the conclusions section of this report, there is an essential trade-off between closed–loop cooling operation and air quality, given prevailing salinity conditions of the Hudson River. According to TRC, the closed-loop cooling system cannot reasonably be expected to comply with $PM_{2.5}$ national ambient air quality standards (NAAQS) and $PM_{2.5}$ Significant Impact Levels (SIL) if operated for substantial periods of time (including most of the summer months) given the Hudson River salinity values. As a result, previous closed-loop cooling operations and configurations (contained in both the 2003 Closed-Loop Cooling Report [Ref. 6.1] and the 2010 Engineering Feasibility and Costs of Conversion of Indian Point Units 2 and 3 to a Closed-Loop Condenser Cooling Water Configuration (2010 Closed-Loop Cooling Report) [Ref. 6.1]) require updating.

[1] For the purposes of this report, the term "salinity" is used to conservatively represent the sum of total dissolved solids (TDS) and total suspended solids (TSS), which, when measured may yield values greater than simply measuring salinity alone.

# 2 Salinity Analysis Inputs

## 2.1 Salinity Data

### 2.1.1 2003 Closed-Loop Cooling Report Salinity Data

Attachment 5 of the 2003 Closed-Loop Cooling Report [Ref. 6.1], reflected a closed-loop salinity of 7.2 psu (7200 ppm), based on an assumed average salinity level of 1.8 psu (1800 ppm) obtained from the 1974 Economic and Environmental Impacts of Alternative Closed-Cycle Cooling Systems for Indian Point Unit 2 [Ref. 6.5]. Closed-loop salinity was used as a design consideration for cooling tower component selection, and was used to evaluate the salt deposition around the two round hybrid cooling towers [Ref. 6.4].

### 2.1.2 ASA Hudson River Salinity Data

A long-term data set of Hudson River salinity in the vicinity of Indian Point was determined and provided by ASA, as documented in Appendix F. The data set consisted of 10 years of modeled Hudson River salinity data for the period 2000 – 2009 in 1-hr increments. Table 2.1 shows the average and maximum continuous Hudson River salinity in psu for the interpolated 10-yr data (Table 5.8 of Appendix F). Appendix F further describes ASA's analysis of the Hudson River salinity data. The average data recovery rate (i.e., percentage of data that is measure over a given period of time) for the ten year period analyzed (2000-2009) was 97.2% as shown in Appendix F, and represents an extremely robust data set.

**Table 2.1 Continuous 10-Year Hudson River Salinity Data (2000 – 2009)**

| Month | 10-Year Data *Average (psu)* | 10-Year Data *Maximum (psu)* |
|---|---|---|
| January | 1.11 | 6.77 |
| February | 1.59 | 6.96 |
| March | 1.08 | 5.84 |
| April | 0.51 | 4.51 |
| May | 0.75 | 6.60 |
| June | 1.17 | 6.07 |
| July | 2.45 | 7.27 |
| August | 3.14 | 7.55 |
| September | 3.90 | 7.67 |
| October | 3.14 | 7.66 |
| November | 1.76 | 7.63 |
| December | 1.06 | 7.26 |
| Average Annual | 1.81 | 6.82* |

*Average of the monthly maxima.

## 2.2 Service Water Flow Description

For this analysis and consistent with 2010 Closed-Loop Cooling Report, Service Water (SW) flows were utilized as make-up flow for the closed-loop cooling system. IPEC supplied seven years (2001-2007) of measured SW intake flow data to ASA Analysis & Communication, Inc. (ASAAC) in millions of gallons per day (MGD); the Unit 2 data includes Unit 2 service water (SW) and Unit 1 river water (RW) flow, and the Unit 3 data includes Unit 3 SW flow. This data was initially supplied for the Biological Assessment included in Attachment 6 of the 2010 Evaluation of Alternative Intake Technologies at IPEC Units 2 and 3 (2010 Alternative Technologies Report) [Ref. 6.3].

Table 2.2 shows the monthly and annual average historic flows for the Stations in gallons per minute (gpm). The monthly and average historic SW flows were used because coincident data (2000 – 2009) was not available.

**Table 2.2 Average Historic SW Flow Rates (2001-2007)**

| Month | Unit 2[1] *(gpm)* | Unit 3[2] *(gpm)* | Total *(gpm)* |
|---|---|---|---|
| January | 27,947 | 18,000 | 45,947 |
| February | 28,668 | 18,000 | 46,668 |
| March | 28,507 | 16,524 | 45,031 |
| April | 28,924 | 16,443 | 45,367 |
| May | 29,123 | 17,774 | 46,897 |
| June | 29,757 | 18,471 | 48,228 |
| July | 32,201 | 20,868 | 53,069 |
| August | 34,304 | 22,561 | 56,865 |
| September | 33,644 | 20,675 | 54,319 |
| October | 31,239 | 18,685 | 49,924 |
| November | 28,932 | 17,913 | 46,845 |
| December | 29,628 | 18,000 | 47,628 |
| Average Annual[3] | 30,251 | 18,668 | 48,919 |

[1] Unit 2 flow includes Unit 2 SW flow and Unit 1 RW flow.

[2] Unit 3 flow includes Unit 3 SW flow.

[3] The average annual historic (2001-2007) SW flow rate is a weighted average determined using the number of days in each month with respect to the number of days in one year.

## 2.3 Meteorological Data

Site wet-bulb temperature[2] governs the amount of evaporation from the cooling towers during operation. Since closed-loop salinity is concentrated by evaporation, it is necessary to accurately define monthly variations in evaporation for the closed-loop cooling salinity level analysis. Although wet-bulb temperature is not measured directly by site meteorological instruments, wet-bulb temperature was calculated using the measured dry bulb temperature and dew point temperature data obtained from IPEC.

The eight years of IPEC meteorological data (2001-2008) utilized in the 2010 Closed-Loop Cooling Report [Ref. 6.2] was also utilized for this analysis. A thorough review was conducted to ensure that the data set was uniform with no erroneous values. The average data

[2] Wet-bulb temperature is a meteorological measurement that incorporates both moisture content and temperature of the ambient air.

recovery rate for the eight year period analyzed (2001-2008) was 97.2% as shown in Attachment 4, Table 4-1 of the 2010 Closed-Loop Cooling Report [Ref. 6.2], and represents an extremely robust data set.

## 2.4 Closed-Loop Design

As discussed in the 2010 Closed-Loop Cooling Report [Ref. 6.2], conversion of both Units 2 and 3 to closed-loop cooling would necessitate the installation of two 100% capacity round hybrid cooling towers and the associated piping and equipment. Under the identified configuration, the new circulating water pumps (CW) for each Unit would draw suction from a modified discharge canal to provide water to cooling tower supply pipelines. In its modified configuration, the discharge canal would no longer serve its once-through cooling function to return circulating water to the Hudson River, but instead would become a new circulating water reservoir / pump pit. The new Unit 2 pump house would be located on the discharge canal between the Unit 1 and Unit 3 turbine generator buildings. The new Unit 3 pump house would be located on the discharge canal along the Hudson River bank. Although the existing CW pumps would no longer be required for closed-loop operation, SW flow would still be maintained through the existing intake structures. The discharge from the SW systems would be used after a conversion to closed-loop cooling for make-up water to the cooling towers.

In short, in order to convert to closed-loop cooling, multiple modifications to the discharge canal would be required. The existing discharge canal would need to be modified to serve as a reservoir/pump pit for the new circulating water pumps that would supply the cooling towers. The new reservoir would communicate between Units 2 and 3 and provide some operational flexibility, whereby the reserve volume would act as a buffer against flow disruptions and equipment failure.

Additional make-up flow for the closed-loop cooling system could be required to provide additional dilution during periods of high closed-loop salinity. One or more make-up pump(s) could be designed to supply the required flow to the cooling tower reservoir. The necessity for additional pumping capacity and resultant flow is discussed in Section 3.1.

# 3 Method of Analysis

## 3.1 Additional Make-Up Cases

Closed-loop cooling requires make-up water to replace water lost in evaporation and drift from the cooling towers, and to allow blowdown from the closed-loop system to maintain the water quality in the closed-loop system. As defined in the 2010 Closed-Loop Cooling Report [Ref. 6.2], the IPEC closed-loop cooling system would draw its make-up water from the SW discharged from each Unit, which reflects the salinity of the Hudson River. The mechanism for controlling air emissions is to limit salinity in the system through alteration of the cooling tower operations, specifically cycles of concentration.

The make-up flow provided by historic SW discharge is substantial (see Section 2.2); however, based upon the salinity analysis performed by ASA, SW discharge alone would not adequately reduce the closed-loop salinity in times of increased Hudson River salinity. In an attempt to limit closed-loop salinity, a control logic was chosen using SW discharge and additional make-up water used in instances of high closed-loop cooling salinity. The control logic analyzed is as follows:

1) If closed-loop salinity is less than the selected setpoint[3], then utilize the SW discharge flow rate only as closed-loop make-up.

2) If closed-loop salinity is greater than the selected setpoint, then utilize the SW discharge and additional make-up flow as closed-loop make-up.

Note that if the River salinity is low enough, it can be used for closed-loop cooling; however, as the ASA salinity analysis shows, River salinity is high for extended periods of time. This salinity effectively constrains cooling tower operations, requiring the closed-loop system to revert to once-through cooling in order to avoid exceeding the $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL.

Figure 3.1 illustrates the closed-loop cycle for one Unit.

[3] The salinity setpoint is a selected point at which additional make-up flow is initiated to counteract high closed-loop salinity levels. The setpoints are selected to minimize make-up flow requirements at the given salinity level, based on the trended analysis discussed in Appendix A. The selected setpoints are documented in Table A.1 of Appendix A.

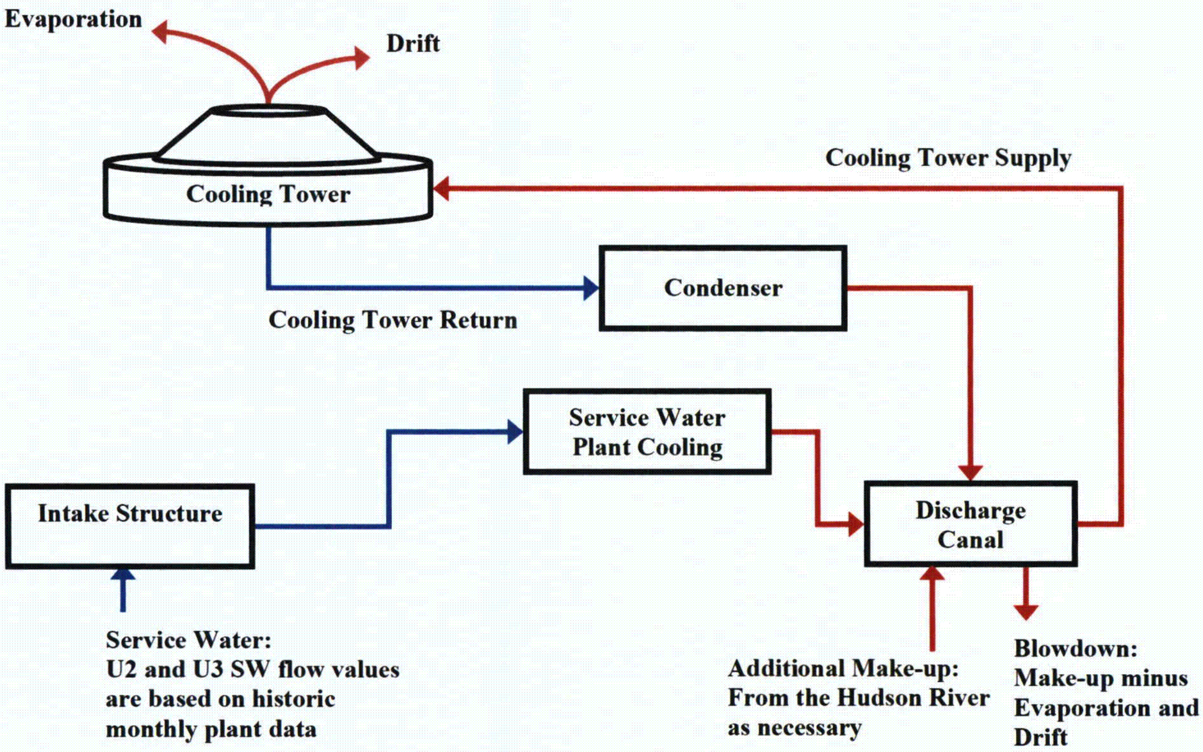


**Figure 3.1 Closed-Loop Design**

The river water salinity is added to the closed-loop system through the SW flow and additional make-up flow, as required. Salinity is removed from the closed-loop system through blowdown and drift[4] although the salinity lost from the closed-loop system from drift is negligible[5]. Salinity is concentrated in the closed-loop system through evaporation.

Hybrid cooling-tower operation was selected to minimize evaporation, and thereby reduce closed-loop salinity and make-up water flow requirements. SPX provided two data points for evaporation rates for hybrid cooling tower operation (Appendix E). These data points were used to create a correlation between the evaporation rates of a round hybrid cooling tower and the ambient wet-bulb temperature. Appendix E includes a chart from SPX illustrating the linear nature of the relationship between evaporation and wet-bulb temperature. The meteorological data described in Section 2.3 was used to determine the monthly average wet-

[4] Drift is liquid water that is carried away from the cooling towers through the exhaust air stream. Drift droplets have the same concentration of solids as the water flowing through the cooling tower.

[5] The amount of flow and salt lost to drift is only 14 gpm or approximately 0.04% of the make-up flow for the closed-loop system. Therefore, the salinity lost from the closed-loop system through drift is not included in this analysis.

bulb temperature at IPEC because coincident data (2000 – 2009) was not available. This monthly average wet-bulb temperature was input into the correlation derived from the SPX data to determine the monthly and annual average evaporation rates. Table 3.1 shows the monthly and annual average evaporation rates used in the salinity analysis.

**Table 3.1 Monthly Average Evaporation**

| Month | Evaporation Rate *Average (%)* | Evaporation Rate *Average (gpm)* |
|---|---|---|
| January | 0.80 | 11,144 |
| February | 0.80 | 11,161 |
| March | 0.87 | 12,187 |
| April | 1.01 | 14,114 |
| May | 1.12 | 15,738 |
| June | 1.25 | 17,566 |
| July | 1.30 | 18,218 |
| August | 1.29 | 18,058 |
| September | 1.22 | 17,055 |
| October | 1.08 | 15,174 |
| November | 0.98 | 13,774 |
| December | 0.86 | 11,976 |
| Average Annual[1] | 1.05 | 14,696 |

[1] The average annual evaporation rate is a weighted average determined using the number of days in each month with respect to the number of days in one year.

## 3.2 1.5 Cycles of Concentration

As discussed in the 2010 Closed-Loop Cooling Report, Hudson River water currently used in the Stations' circulating water systems must also be used for the circulating water in a closed-loop system[6]. Evaporation in the cooling tower would increase the concentration of dissolved solids in the circulating water, as compared to the Hudson River water. The number of times the dissolved minerals in the circulating water are concentrated, versus the level in the Hudson River water (i.e., the cycles of concentration), is an important parameter for cooling tower operation. Since the intake salinity at IPEC varies dramatically based on freshwater discharge to the Hudson River as well as other meteorological and oceanographic influences, the number of cycles of concentration would be dependent on the current intake salinity. The higher the salt content of the makeup water, the fewer cycles of concentration that can be employed to maintain the amount of dissolved solids in the circulating water below the design value.

When designing cooling towers, SPX prefers to limit the closed-loop TDS concentration (i.e., salinity) to 5000 ppm (5 psu) or less (Appendix E). Based on ASA's updated Hudson River

[6] As a result of the considerable unknowns, costs, and the numerous permits required, using recycled wastewater is considered infeasible, as discussed in Section 7.1.2 of the 2010 Cooling Tower Report.

salinity analysis (Appendix F), salinity in the vicinity of IPEC peaks as high as 7.67 psu, thus requiring additional make-up flow to moderate the effects of increased Hudson River salinity. The most practical flow scenario would utilize 1.5 cycles of concentration for the closed-loop system, meaning that the concentration of TDS in the circulating water would be 1.5 times that of the incoming Hudson River water. This make-up flowrate was selected based on the recommendation of SPX for saltwater towers[7]. The flowrate required to achieve 1.5 cycles of concentration is equivalent to the historic Unit 2 and Unit 3 service water flowrates. The evaporation and drift flow rates would be determined as described in Section 3.1.

[7] The water quality in saltwater cooling towers is typically 1.5 cycles of concentration, meaning the concentration of TDS in the circulating water would be 1.5 times that of the incoming water. Saltwater/brackish cooling towers are limited by material and thermal performance degradation at levels above 1.5 cycles of concentration and the biological impact of increased water usage at levels below 1.5 cycles of concentration.

# 4 Updated Salinity Calculation

The updated salinity analysis provides the monthly and annual closed-loop salinity levels based on an updated make-up flow operational scenario to reduce closed-loop cooling salinity in accordance with air quality requirements. The need to decrease the closed-loop salinity is balanced against the goal of not increasing the flow to a value that would significantly diminish closed-loop flow reductions. Table A.1 provides the salinity setpoint (i.e., the selected setpoint at which additional make-up flow is initiated to counteract high closed-loop salinity levels) selections, based on the trended analysis discussed in Appendix A. These setpoint values were selected in an attempt to minimize both the salinity and make-up flow required.

The closed-loop flow and salinity loop is illustrated in Figure 3.1. The initial salinity level within the closed-loop system ($T_1$) is based on an assumed initial salinity value[8] and the volume of water within the closed-loop system for both Units, shown in Equation 1.

$$T_1 = V \times S_{C1} \quad (1)$$

where,

$T_1$ = Initial salt content in the closed-loop system (psu × gallons)

$V$ = Volume of water in the closed-loop system (gallons)

$S_{C1}$ = Initial salinity of the water in the closed-loop system (psu)

The second, and subsequent closed-loop salinity values, are calculated using Equation 2.

$$T = T_L - S_C \times B + S_N \times M \quad (2)$$

where,

$T$ = Salt content in closed-loop system (psu × gallons)

$T_L$ = Previous salt content in closed-loop system (psu × gallons)

$S_C$ = Previous salinity of the water in the closed-loop system (psu)

$B$ = Blowdown volume (gallons)

$S_N$ = Salinity of the Hudson River water (psu)

$M$ = Make-up volume (gallons)

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[8] Using an iterative process, the starting closed-loop cooling salinity is assumed to be the average closed-loop salinity calculated for each setpoint and make-up flowrate; the average closed-loop salinity is used as a representative value and has a negligible impact on the overall calculation.

The closed-loop salinity values calculated using Equation 2 were reviewed and, if during a given week the closed-loop salinity would result in a value exceeding the $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL limits (Appendix C), the system was switched to once-through operation. For the purposes of this analysis, the switch from closed-loop to once-through cooling was conservatively determined on a weekly basis (i.e., if the closed-loop salinity value would exceed the $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL limits at any time in a given week, once-through operation was utilized instead of closed-loop operation). However, switching between closed-loop and once-through cooling may only be feasible (if at all practicable) on an infrequent period (such as a seasonal basis).

# 5 Results

The updated salinity analysis on the 10-year Hudson River data provided by ASA returned values greater than the 7.2 psu defined in the 2003 and 2010 Closed-Loop Cooling Reports. Analyses were done over a range of make-up pump flowrates as well as 1.5 cycles of concentration to determine if make-up pumps would be able to eliminate exceedance of the $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL. Each of these analyses was calculated in the manner described in Section 4 and was then utilized by TRC to determine the potential exceedance for each scenario.

TRC evaluated the exceedance of the $PM_{10}$ and $PM_{2.5}$ NAAQS and SIL that would result from operation of closed-loop cooling at IPEC. As discussed in Appendix C, the $PM_{2.5}$ NAAQS is 5.8 micrograms per cubic meter above the ambient background levels; to avoid exceeding the $PM_{2.5}$ NAAQS, the Hudson River dissolved solids would have to be 0.846 psu or less. When this value is concentrated 1.5 times, the maximum cooling tower salinity would be approximately 1.269 psu. The limiting ground level concentration in the Westchester County $PM_{2.5}$ non-attainment area is the SIL of 1.2 micrograms per cubic meter; to avoid exceeding the $PM_{2.5}$ SIL, the Hudson River dissolved solids would have to be 0.175 psu or less. When this value is concentrated 1.5 times, the maximum cooling tower salinity would be approximately 0.263 psu. Limiting the cooling tower salinity to below 0.263 psu theoretically would allow the closed-loop cooling system to operate at IPEC without exceeding the $PM_{2.5}$ SIL limit under any meteorological condition.

In order to avoid exceeding the $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL under any meteorological condition, a "$PM_{2.5}$ NAAQS No Exceedance" and a "$PM_{2.5}$ SIL No Exceedance" scenario was run to determine how often IPEC would be forced to revert from closed-loop to once-through operation. While a conceptual design has been created for a fully closed-loop system (2003 and 2010 Reports), the detailed design for a system that would allow switching from closed-loop to once-through operation at IPEC has not been performed. The consistent circulating water flow to the main condenser is necessary to serve as a heat sink (i.e., a mechanism for heat removal) for turbine exhaust steam, turbine bypass steam, and other flow. Switching between closed-loop and once-through cooling would be complicated by the start-up and realignment of components necessary for each cooling system and the operational need to maintain a consistent circulating water flow to the main condensers with the Stations in service. This would likely require a shutdown of each Unit to accomplish the switchover. Based on these engineering considerations, and operational considerations input from IPEC personnel, switching between closed-loop and once-through cooling for any potential system may only be feasible (if at all practicable) on an infrequent period (such as a seasonal basis). Limited to a seasonal switch between closed-loop and once-through cooling, IPEC would be forced to operate entirely in once-through cooling mode over the 10-year period analyzed by ASA (Appendix F) to avoid exceeding $PM_{2.5}$ SIL (based on a maximum basin salinity of 0.263 psu determined by TRC).

In order to calculate a theoretical best case scenario (i.e., maximize closed-loop operation time while avoiding exceeding $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL), although impractical for actual Station operation, this report conservatively assumes the switch between closed-loop and once-through operation could be accomplished on a weekly basis. The 10-year Hudson River salinity data was reviewed and if during a given week the closed-loop salinity would exceed the $PM_{2.5}$ NAAQS or $PM_{2.5}$ SIL, the system was switched to once-through operation. Appendix B includes the average annual percentage of once-through run time (bypassing the cooling tower) that would be required to avoid exceeding the air quality standards[9]. As shown in Appendix B, in order to avoid exceeding the $PM_{2.5}$ NAAQS, operation of closed-loop cooling would be expected to occur no more than 43% of the time; in order to avoid exceeding the $PM_{2.5}$ SIL, operation of closed-loop cooling would be expected to occur no more than 13% of the time.

The data in Appendix A and Appendix B was utilized by ASAAC to determine reductions in entrainment[10], entrainment losses[11], and equivalent age 1 losses[12] for each scenario that did not exceed $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL. Table 5.1 summarizes the results provided by TRC and ASAAC in Appendix C and Appendix D, respectively by presenting the potential exceedance of $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL for each closed-loop cooling make-up scenario and the associated percent reduction in entrainment, entrainment losses, and equivalent age 1 losses.

[9] The cooling tower make-up flow would be equal to the historic SW flowrates and the once-through flow would be equal to the historic SW and CW flowrates for both Units 2 and 3 as used by ASAAC in the 2010 Alternative Technologies Report.

[10] Entrainment refers to the eggs, larvae, and older life stages of fish that are drawn through a cooling water system.

[11] Entrainment loss refers to the eggs, larvae, and older life stages of fish that do not survive being drawn through a cooling water system.

[12] Equivalent age 1 refers to the number of fish at different ages that are equivalent one-year-old fish using estimates of the probabilities that fish entrained at various ages would survive to age 1. Equivalent age 1 loss refers to the equivalent age 1 fish that do not survive being drawn through a cooling water system.

**Table 5.1 IPEC Salinity Analysis**
**Air Quality Exceedance and Entrainment Reductions**

| Case (Make-Up Capacity[1]) | Air Quality Exceedance[2] | | Entrainment Reductions[3] | | |
|---|---|---|---|---|---|
| | $PM_{2.5}$ SIL (Exceedance) | $PM_{2.5}$ NAAQS (Exceedance) | Entrainment (Average % Reduction) | Entrainment Loss (Average % Reduction) | Equivalent Age 1 Loss (Average % Reduction) |
| SW Only (1.5 Cycles[4]) | YES | YES | N/A | N/A | N/A |
| SW + 10,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 25,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 50,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 100,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 152,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 304,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 456,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 608,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 760,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 912,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 1,064,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 1,216,000 gpm | YES | YES | N/A | N/A | N/A |
| SW + 1,367,000 gpm[5] | YES | YES | N/A | N/A | N/A |
| $PM_{2.5}$ NAAQS No Exceedance[6] | YES[7] | NO | 57.4 | 63.8 | 56.6 |
| $PM_{2.5}$ SIL No Exceedance[6] | NO | NO | 26.7 | 41.4 | 38.5 |

[1] Make-up flowrate based on closed-loop system logic and historic Unit 2 SW, Unit 1 RW, and Unit 3 SW flowrates used in Case 15 of Attachment 6 to the 2010 Alternative Technologies Report.
[2] The Air Quality Exceedance data is provided by TRC in Appendix C.
[3] The Entrainment Reduction data is provided by ASAAC in Appendix D.
[4] The flowrate required to achieve 1.5 Cycles of Concentration is equivalent to the historic Unit 2 SW, Unit 1 RW, and Unit 3 SW flowrates.
[5] Maximum make-up flowrate determined using minimum SW flowrate (33,000 gpm) and sufficient make-up capacity to produce 700,000 gpm per Unit.
[6] The "No Exceedance" case reverts from closed-loop operation to once-through operation, bypassing the cooling tower, on a weekly basis in order to avoid exceeding the $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL, as described in Appendix B.
[7] Although the "$PM_{2.5}$ NAAQS No Exceedance" case would not exceed the $PM_{2.5}$ NAAQS, the $PM_{2.5}$ SIL would be exceeded.

As discussed above, in order to avoid exceeding the $PM_{2.5}$ NAAQS and $PM_{2.5}$ SIL, operation of closed-loop cooling would be expected to occur no more than 43% and 13% of the year, respectively (see Appendix B). Table 5.1 shows that operation of closed-loop cooling to avoid exceeding the $PM_{2.5}$ NAAQS would result in reductions in entrainment, entrainment losses, and equivalent age 1 losses of 57.4%, 63.8%, and 56.6%, respectively; moreover, the $PM_{2.5}$ SIL would still be exceeded. Table 5.1 also shows that operation of closed-loop cooling to avoid $PM_{2.5}$ SIL would result in reductions in entrainment, entrainment losses, and equivalent age 1 losses of 26.7%, 41.4%, and 38.5%, respectively. For comparison, the reductions in equivalent age 1 losses for cylindrical wedgewire screens would be approximately 89.8%, as presented in Attachment 6 of the 2010 Alternative Technologies Report [Ref. 6.3]. Likewise, the reductions in equivalent age 1 losses associated with the existing technology and operational suite employed by Entergy (i.e., Ristroph screens and fish handling and return systems, as well as flow reductions due to variable and dual speed pumps and maintenance outages) are approximately 33.8% [Ref. 6.3].

In order to accommodate switching between closed-loop cooling and once-through cooling (bypassing the cooling towers), the closed-loop cooling configuration discussed in Section 2.4 would have to be revised. The need to move between once-through and closed-loop cooling may have substantial design, construction, operational, and cost ramifications.

# 6 References

**6.1** Enercon Services, Inc. Economic and Environmental Impacts Associated with Conversion of Indian Point Units 2 and 3 to A Closed-Loop Condenser Cooling Water Configuration. June 2003.

**6.2** Enercon Services, Inc. Engineering Feasibility and Costs of Conversion of Indian Point Units 2 and 3 to a Closed-Loop Condenser Cooling Water Configuration. February 2010.

**6.3** Enercon Services, Inc. Evaluation of Alternative Intake Technologies at Indian Point Units 2 & 3. February 2010.

**6.4** TRC Environmental Corporation. Cooling Tower Impact Analysis for the Entergy Indian Point Energy Center Westchester County, New York. Lyndhurst, NJ. September 2009.

**6.5** Consolidated Edison Company of New York. Economic and Environmental Impacts of Alternative Closed-Cycle Cooling Systems for Indian Point Unit 2. December, 1974.