July 2010 Submittal

Admin. Rec. 793

RESPONSE TO ENVIRONMENTAL PROTECTION AGENCY'S INFORMATION REQUEST FOR NPDES PERMIT RE-ISSUANCE

PSNH MERRIMACK STATION UNITS 1 & 2 BOW, NEW HAMPSHIRE

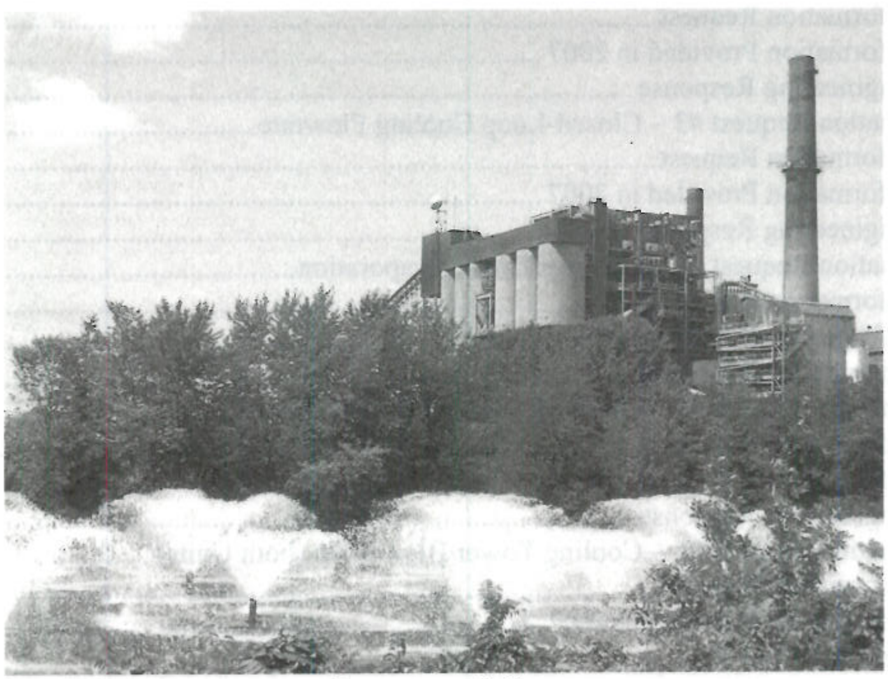

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TABLE OF CONTENTS

List of Attachments

List of Tables

1 Background, Introduction, and Scope

1.1 Background and Introduction

Public Service Company of New Hampshire's (PSNH's) Merrimack Station electrical generating facility in Bow, New Hampshire is seeking a renewal of its existing National Pollutant Discharge Elimination System (NPDES) permit. To this end, an engineering and biological assessment was prepared by Enercon Services, Inc. (ENERCON) and Normandeau Associates, Inc. and submitted by PSNH to the United States Environmental Protection Agency (EPA) in November 2007 that responded to EPA's request for certain technology and fisheries information to support development of the new permit for Merrimack Station.

The November 2007 Response to United States Environmental Protection Agency CWA § 308 Letter (2007 Response) reflects the information requested by EPA and contained the following:

- All fisheries data collected during entrainment and impingement sampling conducted from 2005 to 2007.
- A detailed description of Merrimack Station's cooling system.
- Response regarding projected retirement date for Merrimack Station's existing coal-fired operation.
- A description of the processes employed at Merrimack Station with regard to the operation of the boiler, condenser, cooling water intake structure (CWIS), and effluent treatment.
- A description of the engineering analysis involved with converting the Merrimack Station cooling system from the current once-through cooling to the following cooling scenarios:
    - Mechanical draft cooling towers for use in a recirculating (or "closed-cycle") cooling system for both generating units
    - Mechanical draft cooling towers for use in a recirculating (or "closed-cycle") cooling system for one generating unit
    - Mechanical draft cooling towers for use in a "helper tower" or "chiller" configuration that would be used to reduce thermal discharges by Merrimack Station.
- An analysis of alternate CWIS screening systems, including a discussion of the major components and major modifications that would be required to retrofit Merrimack Station with this technology.
- A discussion of the least expensive, most cost-effective means by which Merrimack Station could meet the evaluated scenario whereby the temperature differential between Stations N10 and S4 in the Hooksett Pool is limited to 5°F.

1.2 Scope

After review of PSNH's response, EPA determined that PSNH needed to further respond to the items below:

- An estimate of the most stringent thermal discharge limits that Merrimack Station would be able to comply with utilizing the cooling tower technologies in question.
- An estimate of the most stringent cooling water withdrawal flow and thermal load limits that the facility would be able to comply with utilizing the cooling tower technologies in question.

As a result, EPA submitted a request for information which in some cases explains items in previous EPA requests, and in other cases requests additional information not previously requested to ensure items are presented clearly. In addition, EPA also requested information regarding certain assumptions and/or calculations that were used as the basis for the information provided in the 2007 Response.

EPA's information request was submitted in January 2010 [Ref. 10.2]. This report individually reviews each information request, provides clarification of the information provided in the 2007 Response, and, where necessary, conducts new analysis to respond to EPA's information request.

2 Information Request #1 – Station Heat Load

2.1 Information Request

The EPA letter requests additional information on the Station heat loads and the ability of the defined cooling towers to reject those heat loads [Ref. 10.2]:

> *On pages 34 and 35 of PSNH's November 2007 Response, the Company states that the range and gpm is fixed by the heat load, and that it selected cooling towers with a design approach of 8°F because PSNH determined that a cooling tower designed with an 8°F approach provided the optimum trade-off between total capacity and performance, and size, initial cost, and operation costs.*
>
> *Please confirm that the cooling tower design PSNH put forth is capable of removing a heat load of 9,337,930 British thermal units (Btu) per minute for Unit 1, and 26,356,120 Btu/min from Unit 2; and that the total heat load that the cooling towers must eject, at full station power output, is 35,694,050 Btu/min. If PSNH does not agree that the specified heat loads are correct, please provide the heat load, in Btu/min, that PSNH contends must be removed in order to condense the exhaust steam from both Merrimack Station's generating unit turbines at full power.*

2.2 Information Provided in 2007

As noted in EPA's request, the design range and flow rate are listed in the SPX Cooling Technologies (SPX) specifications on page 35 of the 2007 Response. The listed design range and flow rate can be used to calculate EPA's nominal heat loads of 9,337,930 Btu/min for Unit 1, and 26,356,120 Btu/min from Unit 2, and 35,694,050 Btu/min for Units 1 and 2 combined.

2.3 Engineering Response

The heat load rejected by the cooling towers is dependent on several variables, and is affected both by the heat load discharged from the Station and the ability of the cooling towers to reject the heat load under ambient environmental conditions. Reviewing the heat balance diagrams, the heat discharged by the Station is based on the input/output parameters of several major components, each with their own operating design range. As a result of this operational flexibility, it would be more accurate to state that the Station operates with a heat load discharge range and not with a fixed heat load discharge design point.

Similar to Station discharge heat load, the cooling tower heat load rejection capacity varies as a function of the ambient environmental conditions (i.e., wet-bulb temperature). A cooling tower transfers heat from the cooling water stream to the surrounding atmosphere, and, as such, the capacity of the cooling tower heat transfer increases when there is a large difference between the water being cooled and the air providing the cooling.

As stated in the 2007 Response, SPX, a leading cooling tower manufacture, was consulted to design the optimum tower design approach and tower sizing. In optimizing the cooling tower design, SPX considered cooling towers of varying size and performance to optimize the cooling towers for the ambient environmental conditions and typical Station performance;

however, due to the Station's design, and the relatively high wet-bulb temperatures present in the summer months, the cooling tower configuration selected by SPX as optimum would still result in reduced plant performance under worst-case ambient conditions (i.e., the cooling towers are not capable of rejecting all of the Station's discharge heat load when the temperature difference between the water being cooled and the air providing the cooling is low). As a result, the cooling tower heat rejection capacity, as calculated from the designed information on page 35 of the 2007 Response, does not reflect the heat load discharged by the Station at full power.

To determine the heat load discharged by the Station at full power, Station data measured during July and August of 2003 through 2006 (i.e., the months typically containing the highest electricity production) was reviewed to determine the increase in cooling water temperature due to Station operation. Using Equation 1 below, and an intake flow rate of 59,000 gpm for Unit 1 and 140,000 gpm for Unit 2, the heat load discharged by the Station at full power was estimated to be approximately 16,220,000 Btu/min for Unit 1, approximately 33,820,000 Btu/min for Unit 2, and approximately 50,040,000 Btu/min for Units 1 and 2 combined.

$$\text{Heat Load} = \dot{Q} = \dot{m} c_p \left(T_{in} - T_{out}\right) \tag{1}$$

where,

m = mass flow rate (lbs/min) = flow (gpm) × density (lbs/gal)

c_p = specific heat of water (Btu / °F × lb)

$T_{in} - T_{out}$ = temperature rise across station (°F)

3 Information Request #2 – Maximum Ambient Conditions

3.1 Information Request

The EPA letter requests clarification of the term "maximum ambient conditions" [Ref. 10.2]:

On page 34 of its 2007 Response, PSNH states that "Since the 84°F condenser inlet water would only occur at maximum ambient conditions,..."

Confirm that by the phrase "maximum ambient conditions," PSNH is referring to the wet bulb inlet temperature of 76°F and that 84°F would represent the maximum temperature of the discharge (blowdown) from Merrimack Station using the closed-cycle cooling tower design provided on page 35 of the Company's 2007 response. If PSNH cannot confirm this, then please explain the meaning of the term "maximum ambient conditions" as used by the Company in the 2007 response, and provide the maximum temperature that the discharge will reach using the cooling tower design provided on page 35 of the Company's 2007 response.

3.2 Information Provided in 2007

As noted in response to Information Request #1, Section 2, SPX was consulted for the 2007 Response to design the optimum tower design approach and tower sizing. As an input parameter to this design, ENERCON reviewed climatic design information from the 2001 American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) Fundamentals Handbook [Ref. 10.1], and provided the 74°F 0.4% wet-bulb exceedance temperature listed for the nearest meteorological monitoring station (Concord, NH). SPX designed the optimum tower configuration using an inlet wet-bulb temperature of 76°F (74°F plus a 2°F allowance for recirculation[1]) with an 8°F cooling approach and a resulting cooling water discharge temperature of 84°F.

3.3 Engineering Response

The meteorological conditions SPX used to design the optimum tower design are representative of maximum conditions listed by ASHRAE, and are likely to occur at the Station only 0.4% of the time. Cooling towers are typically designed to an exceedance value of 0.4% or higher to avoid the considerable overdesign necessary to meet infrequent maximum ambient conditions[2]. However, since 0.4% exceedance translates into approximately 35 hours per year, the SPX design conditions do not represent an absolute maximum design point. Additionally, and as noted in response to Information Request #1, Section 2, since the cooling tower configuration selected by SPX would not be capable of

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[1] Recirculation occurs when some portion of the saturated air leaving the tower is drawn back into the cooling tower inlets. The potential for recirculation is primarily related to wind force and direction; however, cooling tower design and orientation assist in both reducing and controlling recirculation [Ref. 10.4].

[2] Cooling towers are typically designed to a 2%, 1%, or 0.4% exceedance value, as listed in the ASHRAE Fundamentals Handbook [Ref. 18.1].

rejecting all of the Station's discharge heat load under worst-case ambient conditions, there are times at which the cooling water discharge temperature would be expected to exceed 84°F. Therefore, 84°F does not reflect the maximum cooling water discharge temperature, but instead a cooling water discharge temperature that is expected to be exceeded infrequently (i.e., 0.4 % of the time). The accurate calculation of an absolute maximum discharge temperature would require a final detailed closed-loop cooling design, and the analysis of long-term local meteorological data.

4 Information Request #3 – Closed-Loop Cooling Flowrate

4.1 Information Request

The EPA letter requests clarification on why the closed-loop cooling flowrates differ from the operating discharge flowrates [Ref. 10.2]:

> *On page 18 of PSNH's November 2007 Response, the Company states that "[t]he Station's normal operating mode is to operate both units at or near full power. When both units are operating, the maximum operating discharge flow rate is as follows: Unit 1: 48,000 gpm; Unit 2: 130,000 gpm ... This value is shown on the Merrimack Station Water Distribution Diagram ... and is also reported on the Discharge Monitoring Reports (DMR) under normal CWIS conditions. It is also the value that will be used to size the thermal discharge canal cooling tower requested to be evaluated by the EPA."*
>
> *On page 35 of PSNH's November 2007 Response, it states that "Based on a load/capacity assessment provided by SPX Cooling Technologies, the following tower configuration and size was evaluated to support a closed cycle cooling configuration for the Merrimack Station site: ... Unit 1 Flow = 59,000 gpm ... Unit 2 Flow = 140,000 gpm ..."*
>
> *Please explain why PSNH chose to evaluate cooling towers designed with the higher flows (59,000 gpm versus 48,000 gpm for Unit 1; 140,000 gpm versus 130,000 gpm for Unit 2).*

4.2 Information Provided in 2007

From Section 3.3.2.1 of the 2007 Response:

"The north (Unit 1) CWIS has two circulating water intake pumps. Each circulating water pump has a design intake capacity of 29,500 gpm (42.5 MGD, 65.7 cfs). The two flows join in a common header resulting in a combined design intake capacity for both pumps at Unit 1 of 59,000 gpm (85.0 MGD, 131.5 cfs). The Unit 1 circulating water pumps supply water to the following:

- 1806 gpm (2.6 MGD, 4.0 cfs) is supplied for the Slag Sluice. This value is an average daily flowrate since this flow is an intermittent demand. Slag sluice is typically run 9 hrs/day from mid-March through mid-December. During the winter, the sluice runs continuously to protect the system from freezing. The slag sluice discharges into the Slag Pond and ultimately into the discharge canal.
- 5556 gpm (8.0 MGD, 12.4 cfs) is supplied for De-Icing Recirculation. This system is only used during periods where the temperature is below freezing. In essence, hot water from either the condenser or equipment cooling water heat exchanger is recirculated back into the intake via 6" spray nozzles at the bar racks. Since 5556 gpm of water is being added to the intake, the demand for water from the River is decreased by the same amount.
- The traveling screen wash system draws suction from the discharge of the circulating water pumps. Each traveling screen has a single-pressure spray header

to wash fish and debris off of the traveling screens and then flush them back to the River. The Unit 1 traveling screen spray wash system draws a total of 560 gpm (0.8 MGD, 1.2 cfs).

- Flow also is supplied to equipment cooling.
- The remainder of the flow supplies the condenser.

The fire pump draws water from the Unit 1 screenwell. It has a design intake capacity of 486 gpm (0.7 MGD, 1.1 cfs) and runs intermittently.

The south (Unit 2) CWIS also has two intake pumps, each with a design intake capacity of 70,000 gpm (100.8 MGD, 156 cfs). The two flows are combined in a common header with a combined design intake capacity for both at Unit 2 circulating water pumps of 140,000 gpm (201.6 MGD, 312 cfs). The Unit 2 circulating water pumps supply water to the following:

- 2780 gpm (4.0 MGD, 6.2 cfs) is supplied for Slag Sluice. This flow is a constant demand, which discharges into the Slag Pond and ultimately into the discharge canal.
- 9028 gpm (13.0 MGD, 20.1 cfs) is supplied for De-Icing Recirculation. This system is only used during periods where the temperature is below freezing. In essence, hot water from the condenser or equipment cooling water heat exchanger is recirculated back into the intake via 6" spray nozzles at the bar racks. Since 9028 gpm of water is being added to the intake, the demand for water from the River is decreased by the same amount.
- The Unit 2 traveling screen spray wash system works the same as the Unit 1 system. However, it draws a total of 588 gpm (0.9 MGD, 1.4 cfs).
- Flow also is supplied to equipment cooling.
- The remainder of the flow supplies the condenser."

4.3 Engineering Response

A closed-loop cooling design utilizing the maximum design intake flow for each unit (59,000 gpm at Unit 1 and 140,000 gpm at Unit 2) was determined to be the most appropriate for the evaluation of closed-loop cooling provided in the 2007 Response based on the flow requirements of the plant.

The 2007 Response notes that, when both units are operating, the maximum operating discharge flowrates are 48,000 gpm at Unit 1 and 130,000 gpm at Unit 2. These values are from the Merrimack Station Water Distribution Diagram (Figure D, Attachment 5, 2007 Response) and are also reported on the Discharge Monitoring Reports (DMR) under normal CWIS conditions. However, these maximum operating discharge flowrates are only for the condensers and do not include the flow required for de-icing recirculation, the traveling screen spray wash systems, and other equipment cooling that discharges to the Slag Pond. As noted in the 2007 Response, the De-Icing Recirculation system is only used during periods where the temperature is below freezing. Since the existing circulating water pumps are single-speed pumps, the flow is utilized as additional flow through the condenser, for additional equipment cooling, or additional flow provided to the slag sluice.

In addition, one or more pumps would be required to provide the necessary make-up flow to the cooling towers even after the conversion to closed-loop cooling. The make-up pump(s) would likely require traveling water screens and spray wash systems in addition to de-icing recirculation. Therefore, it is anticipated that the flow requirements, while possibly reduced, would still be necessary even after conversion to closed-loop cooling.

5 Information Request #4 – Cooling Tower Evaporation

5.1 Information Request

The EPA letter requests explanation of the evaporation coefficient, and, in particular, the term "Evaporation Wet Summer" [Ref. 10.2]:

> *On page 40 of PSNH's November 2007 Response, the Company states that "Evaporation $_{Wet\ Summer}$ can be approximated as Water Flow $_{Total}$ x 0.0167"*
>
> *Please define the term "Evaporation $_{Wet\ Summer}$" as used in PSNH's November 2007 Response. Please also provide the corresponding wet bulb temperature(s) during "Evaporation $_{Wet\ Summer}$"*
>
> *Please explain the basis of the 0.0167 multiplying factor. Please also explain why the factor of 0.0008 was not applied separately for each cooling tower, using the different tower flow rates and different tower range values in order to approximate the evaporation rate.*

5.2 Information Provided in 2007

The term "Evaporation $_{Wet\ Summer}$" from the 2007 Response is used to describe the estimated daily water loss from the Merrimack River due to evaporation by the evaluated cooling tower. The evaporation rate was used as the design point for determining the required make-up flow, and as such it was important to use the maximum evaporation rate (i.e., the evaporation rate from wet mode operation during the warmer summer months). The evaporation rate was approximated as 1.67% of the total water flowrate; this rate was estimated based on information provided by SPX for wet mode operation of a hybrid cooling tower at a wet bulb temperature of 77°F. The approximation was used to produce an evaporation rate of 3323 gpm, as calculated on page 54 of the 2007 Response.

5.3 Engineering Response

From SPX's Cooling Tower Fundamentals [Ref. 10.4], if not accurately known, evaporation can be approximated by multiplying total water flowrate in gpm times the cooling range (°F) times 0.0008. As shown on page 35 of the 2007 Response, a cooling range of 19°F was used for Unit 1 and a cooling range of 22.6°F was used for Unit 2. These cooling ranges can be averaged to obtain a combined range of approximately 20.8°F. This combined range is multiplied by 0.0008 to obtain a factor of 0.0167 and is equal to the approximation described above, thereby resulting in an evaporation rate of approximately 3323 gpm.

A separate way to determine the overall evaporation rate for the cooling tower would be to use the factor of 0.0008 listed in the Cooling Tower Fundamentals [Ref. 10.4] applied separately for each cooling tower at the different cooling tower flowrates and cooling range values. The evaporation rate would, therefore, be calculated as shown in Equation 2.

$$\text{Flow}_{\text{evap}} = (59{,}000\ \text{gpm} \times 19°\text{F} \times 0.0008) + (140{,}000\ \text{gpm} \times 22.6°\text{F} \times 0.0008) = 3428\ \text{gpm} \qquad (2)$$

This method of approximating the evaporation rate would result in slightly increased blowdown and make-up flowrates. Note that more evaporation would require larger make-up flowrates, resulting in larger make-up pumps and reduced effectiveness of closed-loop cooling. As such, the approximation of the evaporation rate used in the 2007 Response would be conservative.

6 Information Request #5 – Closed-Loop Cooling Makeup

6.1 Information Request

The EPA letter requests clarification of the closed-loop cooling makeup flowrates, and in particular the term "wet mode tower operation" [Ref. 10.2]:

> *On page 41 of PSNH's November 2007 Response, the Company states that the "Plant makeup from the River, wet mode tower operation would hence equal Unit 1 M_{wet} = 1232 gpm, and Unit 2 $_{Wet}$ = 2923 gpm.*
>
> *Please explain and/or define the term "wet mode tower operation" as used in PSNH's November 2007 Response. Also, please confirm whether the total value of 4155 gpm represents the maximum value of make-up water necessary.*

6.2 Information Provided in 2007

The term "wet mode tower operation" refers to operation of a hybrid (i.e., plume abated) cooling tower in non-plume abated mode. This operational mode would typically result in higher evaporation rates, which would in-turn result in higher make-up flowrates. It is important to design equipment based on the maximum make-up flowrates required to ensure that the cooling towers would be able to operate under all design conditions.

6.3 Engineering Response

Make-up water requirements are the sum of evaporation, drift, and blowdown from each Unit's closed-loop cooling system. As stated in the response to Information Request #4 (Section 5.3), the evaporation rates presented in the 2007 Response were estimated based on information provided by SPX for wet mode operation of a hybrid cooling tower. Likewise, the drift rate was estimated using information provided for the same hybrid cooling tower; however, since drift rate are based by on the physical geometry of the drift eliminators, one constant drift rate was provided for both hybrid and wet mode operation. The blowdown requirements are a function of evaporation, drift, and the cycles of concentration; whereby, the 2007 Response calculated blowdown at 5 cycles of concentration to account for worst case intake water quality. Using each of these inputs, the expected maximum make-up flowrates were calculated in the 2007 Response, and are provided in Equations 3 and 4.

$$\text{Make-Up} = \text{Evaporation} + \text{Drift} + \text{Blowdown} \quad (3)$$

$$\text{Blowdown} = \frac{\text{Evaporation} - \left[(\text{Cycles of Concentraion} - 1) \times \text{Drift}\right]}{\text{Drift}} \quad (4)$$

Unit 1 Water Flow = 59,000 gpm

Evaporation_{wet} = 0.0167 x 59,000 gpm = 985.3 gpm

Drift = Water Flow x 0.00001 gpm = 0.6 gpm

Blowdown_{wet} = 245.7 gpm

Make-Up_{wet} = 1231.6 gpm

Unit 2 Water Flow = 140,000 gpm

$Evaporation_{wet}$ = 0.0167 x 140,000 gpm = 2338.0 gpm

Drift = Water Flow x 0.00001 gpm = 1.4 gpm

$Blowdown_{wet}$ = 583.1 gpm

$Make\text{-}Up_{wet}$ = 2922.5 gpm

Summing the expected maximum make-up requirements, Merrimack Station would require a make-up flowrate of approximately 4155 gpm. It should be noted that a higher rate of evaporation, drift, or blowdown is possible dependent on the final design of the cooling towers and further investigation in river water quality and absolute maximum ambient wet-bulb temperature; however, any increase to the make-up water flowrates would result in larger make-up pumps and reduced effectiveness of closed-loop cooling. As such, the approximation of make-up flowrates used in the 2007 Response would be conservative.

7 Information Request #6 – Cooling Tower Blowdown (both Units)

7.1 Information Request

The EPA letter requests additional analysis to calculate the monthly thermal discharge from the cooling tower blowdown of both units [Ref. 10.2]:

> *On page 100 of PSNH's November 2007 Response, the Company states that "Complete closed loop conversion, as described in Section 6, would effectively eliminate all thermal discharges to the Merrimack River and is therefore assumed to represent a complete thermal reduction (i.e., river water temperature unaltered by the Station operation)."*
>
> *Please explain how it would be possible to "effectively eliminate all thermal discharges to the Merrimack River" or to achieve "a complete thermal reduction" using wet cooling towers, given that using that technology, the Station will still have a thermal discharge from the towers in the form of cooling tower "blowdown", based on 5 cycles of concentration.*
>
> *Please explain what assumptions or analyses went into the above-referenced statement that a complete thermal reduction could be achieved by converting to closed-loop cooling using wet cooling towers.*
>
> *Please confirm whether it is PSNH's position that a NPDES thermal limit derived from closed cycle cooling for Merrimack Bow would properly be zero (0.0) Btus. Given that any thermal limit would be monitored by determining the temperature difference between the intake water and the temperature of the blowdown (delta T), multiplied by the mass of blowdown, using the standard value heat capacity of water of 1.0 Btu/°F· lb, please explain whether, and how, PSNH concludes that Merrimack Station would be able to comply such an NPDES thermal permit limit.*
>
> *If PSNH determines that the thermal discharge would not actually be completely eliminated through the use of wet cooling towers, then please provide an accurate, estimated monthly thermal discharge (in Btu/month) that would be discharged from Merrimack Station as a result of conversion to closed-loop cooling using wet cooling towers as discussed and evaluated in PSNH's November 2007 Response (sum of daily: blowdown flow rate x 8.33 x (intake temperature - discharge temperature). Please also provide a separate estimated Btu discharge for each month of the year.*

7.2 Information Provided in 2007

The 2007 Response assumed that a determination of the thermal discharge to the Merrimack River from cooling tower blowdown was unnecessary. This assumption was based on the following:

1) The blowdown flow rate from any cooling tower(s) would be extremely limited as compared to the current, once-through operation (i.e., the amount of flow discharged as cooling tower blowdown would make up less than 1% of the current maximum combined discharge flow rate).

2) The blowdown from any cooling tower(s) would be cooled by the cooling tower(s) prior to discharge.

3) The current discharge canal would remove additional residual heat from the cooling tower blowdown.

While no specific determination was made, the overall effect of each of these factors on discharge temperature to the Merrimack River was estimated while preparing the 2007 Response. This initial estimation was based on five years (2002-2006) of meteorological data and river water temperatures used in conjunction with the cooling tower performance (defined by SPX Cooling Technologies) and the current Discharge Canal performance [Ref. 10.3, Section 3.4.3]. The estimate focused on the aggregate discharge temperature from a complete closed-loop conversion, making the assumption that the volume of the discharge canal was adequate to support thermal mixing of warmer blowdown during daylight hours with cooler blowdown during nighttime hours. This, coupled with the assumption that the Discharge Canal would provide the necessary additional cooling, lead to a non-conservative determination that, on average, no thermal discharge would be associated with a complete closed-loop conversion.

7.3 Updated Analysis

In updating the 2007 estimate, the thermal discharge from a complete closed-loop conversion was calculated making the following conservative assumptions,

1) The maximum combined cooling tower blowdown of approximately 830 gpm is assumed constant throughout the year (see Section 6),

2) Thermal discharge is measured directly as the difference between the temperature of make-up water drawn from the Merrimack River at N0 and the temperature discharged from the cooling tower blowdown, and does not account for any potential cooling in the discharge canal or in the river between S0 and S4.

Since cooling tower blowdown is drawn from the cooling tower basin, the temperature of the blowdown is identical to that cooled by the tower for recirculation back to the condenser. Thus, the blowdown thermal analysis uses a cooling tower performance calculation similar to that of the closed-loop cooling performance analysis [Ref. 10.3, Section 6.2.2]. Five years (2002-2006) of meteorological data and river water temperatures were coupled with the cooling tower performance calculation to determine the average and maximum monthly blowdown temperatures, as shown in Table 1.

Table 1 Complete Closed-Loop Cooling Blowdown Temperatures

Month	Closed-Loop Cooling Temperature Increase*	
	Hourly Average (°F)	*Hourly Maximum (°F)*
January	11.6**	22.2**
February	12.1**	20.1**
March	13.7**	24.1**
April	12.1**	24.1**
May	9.8	20.0
June	6.8	13.6
July	3.3	10.6
August	2.9	11.0
September	5.4	14.7
October	8.9	19.3
November	12.7**	26.1**
December	12.7**	22.4**
Annual	9.3**	26.1**

*Calculated using five years (2002-2006) of meteorological data and river water temperatures
**Calculated using a monthly data set where 32°F was used when river water temperatures were not measured

Using Equation 5 (as listed in EPA's Information Request) and the monthly average and maximum blowdown temperatures calculated above, the monthly thermal discharges resulting from the blowdown of a complete closed-loop conversion are calculated in Table 2.

$$\text{Thermal Discharge} = \dot{V}_{\text{Flow Rate}}\left(T_{\text{make-up}} - T_{blowdown}\right)\left(8.33\frac{\text{lb}}{\text{gpm}}\right)\left(1\frac{\text{Btu}}{{}^{\circ}\text{F}\times\text{lb}}\right) \quad (5)$$

Table 2 Complete Closed-Loop Cooling Thermal Discharge

Month	Closed-Loop Cooling Thermal Discharge*	
	Average (MBtu/Month)	*Maximum (MBtu/Month)*
January	3,590**	6,856**
February	3,371**	5,613**
March	4,244**	7,428**
April	3,610**	7,210**
May	3,028	6,164
June	2,041	4,064
July	1,022	3,264
August	898	3,393
September	1,609	4,396
October	2,749	5,950
November	3,801**	7,795**
December	3,905**	6,920**
Annual	33,860**	94,840**

*Calculated using five years (2002-2006) of meteorological data and river water temperatures

**Calculated using a monthly data set where 32°F was used when river water temperatures were not measured

7.4 Engineering Response

The 2007 Response used non-conservative assumptions to conclude that a determination of the thermal discharge to the Merrimack River from cooling tower blowdown was unnecessary. Using a methodology identical to the closed-loop cooling performance analysis, Table 2 was created to list the monthly thermal discharge of a complete closed-loop system at Merrimack Station assuming both an average temperature difference and a maximum temperature difference condition. In summary, an annual average and maximum thermal discharge of approximately 33,860 MBtus and 94,840 MBtus, respectively, would occur under a complete closed-loop conversion. As shown in Table 2, on average March is the most impactive month with a total monthly thermal discharge of approximately 4,244 MBtus, while November has the highest maximum with a thermal discharge of 7,795 MBtus.

For comparison, the annual maximum thermal discharge of approximately 94,840 MBtus is less than 0.4% of the combined heat load discharged by the Station at full power[3].

[3] The combined heat load discharged by the Station at full power is calculated in Section 2.3 as approximately 50,040,000 Btu/min, or 26,301,024 MBtu/year.

Conservatively assuming that no cooling is provided by the discharge canal or from the river surface between S0 and S4, the annual maximum thermal discharge would increase the river water temperature by approximately 0.01°F at average river flow rate conditions (approximately 4,500 cfs[4]).

[4] Average river flow rate of approximately 4,500 cfs based on 21 years (1984-2004) of Merrimack River flow measurements.

8 Information Request #11 – Power Spray Module Performance

8.1 Information Request

The EPA letter requests confirmation that PSM performance was included in the results of the Unit 1 only and Unit 2 only analyses [Ref. 10.2]:

> *The heading to the tables on pages 101 and 102 of PSNH's November 2007 Response contain the phrase "Merrimack Station Current PSM and Discharge Canal Performance"*
>
> *Please explain why the tables are labeled this way and confirm whether or not the performance of the Power Spray Module (PSM) is reflected in the results presented in the tables.*

8.2 Engineering Response

The tables on pages 101 and 102 of the 2007 Responses are labeled as "Merrimack Station Current PSM and Discharge Canal Performance" because the tables present a comparative evaluation which includes PSM operation under the current operational scheme. An empirical analysis of the discharge canal temperature data provided by Merrimack Station, which included PSM operation, was conducted and input over five years of meteorological data and river water temperatures. As described in the 2007 Response, this data was compared against 21 years of daily average measured river flowrate values (1984 - 2004).

9 Information Request #12 – Equipment Cooling

9.1 Information Request

The EPA letter requests additional analysis to determine the heat load added by the Station's equipment cooling system, including a listing of equipment and each component's cooling flowrate, cooling flow temperature rise, and the incremental heat load added to the equipment cooling system [Ref. 10.2]:

> *Figure D in Attachment 5 of PSNH's November 2007 Response shows that some Unit 1 and Unit 2 intake water is for equipment cooling, and that that water is ultimately discharged to the cooling canal and out outfall 003 to the River.*
>
> *Please provide the amount of heat, in Btu/hr, that must be transferred to the cooling water from such equipment. Please also list the equipment, the flow rate used to provide cooling for each piece of equipment, and the resulting increase in temperature of the cooling water after it has cooled the equipment.*
>
> *Please also clarify if the heat load from this equipment was considered when PSNH evaluated closed-cycle cooling for entire plant, closed-cycle cooling for Unit 1, closed-cycle cooling for Unit 2, the 10-cell helper tower, and/or the 14-cell helper tower.*

9.2 Updated Analysis

As shown in Drawings MK1-M-186 / -187 / -191 and MK2-M-215 / -216 in Attachment 1, components that are cooled by the equipment cooling flow at each Unit include the following:

Unit 1 Equipment Cooling

- Cyclone Vertex Cooling Jackets
- Air Conditioning and Admin Bldg
- Seal Oil Cooler
- High Pressure Drip Pump
- Hydrogen Coolers
- Turbine Lube Oil Coolers
- Boiler Feed Pumps
- Control Air Compressor
- Forced Draft Fan Bearings
- Gas Recirculation Fan
- Sampling Coils

Unit 2 Equipment Cooling

- Exciter Air Coolers
- H2 Seal Oil Coolers
- Generator Hydrogen Coolers
- Deaerator Pumps
- Start Up B.F. Pump Oil Cooler
- Sample Cooling Coils
- Turbine Lube Oil Coolers
- Coupling Oil Coolers
- Air Compressor
- F.D. Fans
- Gas Recirculation Fans
- Cyclone Door Cooling Jackets

Each piece of equipment rejects heat to a secondary cooling water loop, which in turn rejects heat through the Cooling Water Heat Exchangers to the equipment cooling flow drawn from the River. Cooling flow rates and temperature increases have not historically been recorded at each piece of equipment. Temperature and pressure measurements were recorded at the Cooling Water Heat Exchangers, on the secondary cooling water loop side, at full load conditions on March 26th, 2010. These measurements were used in conjunction with the rated cooling water pump capacity to calculate the heat load rejected by the Cooling Water Heat Exchangers. This calculation provides a complete evaluation of the total equipment cooling heat load because the total heat load is transferred through the Cooling Water Heat Exchangers to the equipment cooling flow drawn from the River. As shown in Attachment 1, the calculated equipment cooling heat loads at Units 1 and 2 are 11,702,854 BTU/hr and 58,629,610 BTU/hr, respectively. These heat loads are equal to less than 3% of the condenser heat loads at each Unit, discussed in response to Information Request #1, Section 2.

The outlet temperature of the equipment cooling flow supplied to the Cooling Water Heat Exchanger is bounded by the hot inlet temperature of the secondary loop cooling water (i.e., the river water side of the Cooling Water Heat Exchanger cannot heat to a temperature above the hot inlet temperature of the secondary loop). The average temperature difference between the hot inlet temperatures of the secondary loop cooling water and the cold inlet temperatures of the equipment cooling flow is 17.9°F at Unit 1 and 22.4°F at Unit 2[5]. Using the calculated Cooling Water Heat Exchangers heat loads and the bounding average ranges, the equipment cooling flow rate can be estimated to be 1310 gpm at Unit 1 and 5235 gpm at Unit 2. These estimated flow rates result in flow velocities typical of heat exchanger supply piping (Unit 1: 3.7 ft/sec; Unit 2: 5.3 ft/sec).

9.3 Engineering Response

As discussed in Sections 6.1 of the 2007 Response, the closed-loop cooling tower configurations were sized to accommodate the maximum design intake flow rate and condenser cooling range of Units 1 and 2, individually and combined. The maximum design intake flow rate includes the equipment cooling flow rate.

As discussed in Sections 10.2 of the 2007 Response, the thermal discharge cooling tower configurations were sized to accommodate the maximum discharge flow rate and condenser outlet temperature of Units 1 and 2, combined. The maximum discharge flow rate includes the equipment cooling flow rate.

A comparison of the flow rates and cooling ranges of the condensers and Cooling Water Heat Exchangers is provided in Table 3.

[5] Cooling Water Heat Exchanger average ranges are based on historical operational data taken during July and August of 2003-2006.

Table 3 Comparison of Equipment Cooling and Condenser Cooling

Unit	Parameter	Condenser	Cooling Water Heat Exchanger	Comparison
1	Range [°F]	19	17.9	Equipment cooling range is slightly smaller than condenser cooling range
	Discharge Flow Rate [gpm]	48,000	1309	Equipment cooling flow is equal to approximately 2.7% of condenser flow
2	Range [°F]	22.6	22.4	Equipment cooling range is approximately equal to condenser cooling range
	Discharge Flow Rate [gpm]	130,000	5235	Equipment cooling flow is equal to approximately 4.0% of condenser flow

The temperature increases in the equipment cooling flow are less than the temperature increases in the condenser cooling flow; further, due to the relatively low flow rates of the equipment cooling flow compared to the condenser cooling flow, any abnormal differences in temperature increase would not have a significant impact on the cooling tower performance. In summary, the closed-loop and thermal discharge cooling towers were adequately sized to accommodate equipment cooling heat load at the Stations, both individually and combined.