Dominion Nuclear Connecticut, Inc.
Millstone Power Station
Rope Ferry Road, Waterford, CT 06385

AUG 1 2013

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U. S. Nuclear Regulatory Commission
Attention: Document Control Desk
Washington, DC 20555

Serial No.	13-450
NSSL/MLC	R0
Docket No.	50-336
License No.	DPR-65

DOMINION NUCLEAR CONNECTICUT, INC.
MILLSTONE POWER STATION UNIT 2
RESPONSE TO REQUEST FOR ADDITIONAL INFORMATION REGARDING LICENSE AMENDMENT REQUEST FOR CHANGES TO TECHNICAL SPECIFICATION 3/4.7.11, "ULTIMATE HEAT SINK"

By letter dated May 3, 2013, Dominion Nuclear Connecticut, Inc. (DNC) submitted a license amendment request (LAR) for Millstone Power Station Unit 2 (MPS2). The proposed amendment would modify Technical Specification (TS) 3/4.7.11, "Ultimate Heat Sink," to increase the current ultimate heat sink (UHS) water temperature limit from 75°F to 80°F and change the TS Action to state, "With the ultimate heat sink water temperature greater than 80°F, be in HOT STANDBY within 6 hours and in COLD SHUTDOWN within the following 30 hours."

In a letter dated June 26, 2013, the Nuclear Regulatory Commission (NRC) provided DNC an opportunity to supplement the LAR identified above. Supplemental information was provided to the NRC in a letter dated June 27, 2013. In a letter dated July 18, 2013, the NRC transmitted a request for additional information (RAI) related to the LAR. DNC responded to the RAI in a letter dated July 19, 2013. In an e-mail dated July 23, 2013, the NRC transmitted a second RAI to DNC. DNC responded to the second RAI in a letter dated July 30, 2013. In an e-mail dated July 26, 2013, the NRC transmitted a third RAI to DNC. Attachment 1 to this letter contains DNC's response to the third RAI.

If you have any questions or require additional information, please contact Wanda Craft at (804) 273-4687.

Sincerely,

S. E. Scace
Site Vice President – Millstone Power Station

WM. E. BROWN
NOTARY PUBLIC
MY COMMISSION EXPIRES MAR. 31, 2016

STATE OF CONNECTICUT)
)
COUNTY OF NEW LONDON)

The foregoing document was acknowledged before me, in and for the County and State aforesaid, today by S. E. Scace, who is Site Vice President of Millstone Power Station. He has affirmed before me that he is duly authorized to execute and file the foregoing document in behalf of that Company, and that the statements in the document are true to the best of his knowledge and belief.

Acknowledged before me this 1 day of August, 2013.

My Commission Expires: 3/31/16.

Notary Public

A001
NRR

Commitments made in this letter:

None

Attachments:

1. Response to Request for Additional Information Regarding License Amendment Request for Changes to Technical Specifications 3/4.7.11, "Ultimate Heat Sink"

2. Excerpt from Zachry Calculation 12-280.

3. Excerpt from Zachry Calculation 12-339.

4. Excerpt from Zachry Calculation 13-016.

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Attachment 1

Response to Request for Additional Information Regarding License Amendment Request for Changes to Technical Specifications 3/4.7.11, "Ultimate Heat Sink"

Dominion Nuclear Connecticut, Inc.
Millstone Power Station Unit 2

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Background
In RAI 1 of NRC letter dated July 18, 2013, the NRC staff asked the licensee to provide the acceptance criteria that proved that the increase in [Reactor Building Closed Cooling Water] RBCCW cooling water temperature is acceptable for each safety related load cooled by RBCCW cooling water and to explain how the acceptance criteria are met.

Issue
The licensee provided a response in a letter dated July 19, 2013. The NRC staff needs further clarification of the licensee's response.

Request

a) *FSAR Section 9.9.8 states that the Engineered Safety Features Room Air Recirculation System (ESFRARS) is designed to limit the maximum ambient temperature to 145°F except for a brief transient temperature excursion following an accident. Discuss the ability of the RBCCW system to meet this design requirement with the UHS temperature limit rising 5°F from 75°F to 80°F.*

DNC Response

The RBCCW system continues to be capable of limiting the ESFRARS maximum ambient temperature to 145°F even with the UHS temperature at 80°F, except for a brief transient temperature excursion following an accident,. Analyses were performed under two assumed conditions: 1) to minimize heat transfer (for determining worst case containment conditions) and, 2) to maximize heat transfer (for determining highest RBCCW temperature). The latter condition, for which a RBCCW temperature profile was developed, was then used as input to subsequent

calculations which evaluated the impact of the higher RBCCW temperature on critical components.

Specifically, the containment loss of coolant accident (LOCA) and main steam line break (MSLB) analyses modeled the RBCCW system and its interface with containment (via the containment air recirculation (CAR) fans in the injection mode and the CAR fans and shutdown cooling (SDC) heat exchangers in the sump recirculation mode). The RBCCW spent fuel pool (SFP) heat load was included in these analyses. For simplicity, other heat loads on the RBCCW system were included in the analyses as a single heat load.

The analysis for the ESFRARS used the RBCCW temperature profile, as well as heat loads in the room (running equipment, hot piping, etc.), to determine a room temperature profile which was then used as input to evaluate equipment environmental qualification (EEQ) components in those rooms. The results of the analysis concluded that with an increase in UHS temperature to 80°F, the ESF room equipment remains within its EEQ design limits.

b) *FSAR Section 6.5.2 states that each containment air recirculation (CAR) cooling unit is designed for removing 80 x 10^6 Btu/hr under Main Steam Line Break accident or LOCA conditions prior to recirculation with air flow of 34,800 cfm and a fouling factor of 0.0005 for the RBCCW side of the coil. Discuss the ability of the RBCCW system to meet this design requirement with the UHS temperature limit rising 5°F from 75°F to 80°F.*

DNC Response

As noted above, each CAR cooling unit is designed for removing 80 x 10^6 BTU/hr under MSLB accident or LOCA conditions prior to recirculation with air flow of 34,800 cfm and a fouling factor of 0.0005 for the RBCCW side of the coil. This 80 x 10^6 BTU/hr design point is based on a containment air inlet temperature of 289°F, a RBCCW inlet temperature of 130°F, and a RBCCW flow rate of 2000 gpm.

The Dominion GOTHIC containment analysis CAR cooling unit heat exchanger model was developed using the flow and temperature conditions identified in the previous paragraph. This model predicts that 80 x 10^6 BTU/hr will be removed from containment. The transient heat transfer rate from the containment atmosphere to the RBCCW system via the CAR cooling unit heat exchangers was determined using the GOTHIC heat exchanger model and the RBCCW and service water models with an 80°F service water inlet (ultimate heat sink) temperature to the RBCCW heat exchanger. The calculated transient CAR cooling unit heat transfer rate varies as a function of containment temperature and RBCCW inlet temperature. For a given containment temperature, the use of a service water inlet temperature of 80°F increases the RBCCW inlet temperature to the CAR cooling unit heat exchangers, resulting in a reduction in the CAR cooling unit heat exchanger heat transfer rate when compared to lower service water inlet

temperatures. This reduction in the heat transfer rate with an 80°F UHS is justified since the existing FSAR Section 14.8.2 for MSLB and LOCA containment analysis using the Dominion GOTHIC methodology, which includes the modeling of the transient CAR cooling unit heat transfer rate, continues to meet the established acceptance criteria.

c) *FSAR Section 6.3.2 states that the high pressure and low pressure safety injection pumps have mechanical seals. The seals are designed for operation with a pumped fluid temperature of 350°F. To permit extended operation under these conditions, a portion of the pump fluid is externally cooled by the RBCCW system and recirculated to the seals. Discuss the ability of the RBCCW system to meet this design requirement with the UHS temperature limit rising 5°F from 75°F to 80°F.*

DNC Response

The seals for the high pressure safety injection (HPSI) and low pressure safety injection (LPSI) pumps were previously evaluated based on an assumed RBCCW supply temperature of 147°F and a pumped fluid temperature of 300°F. This evaluation determined a resultant HPSI and LPSI pump seal temperature of 192°F and 232°F, respectively. For an increase in UHS temperature to 80°F, the maximum RBCCW supply temperature was increased to 149°F. This change increases the average temperature of the RBCCW through any heat exchanger by 2°F. Conservatively assuming 300°F for the pumped fluid (actual maximum calculated temperature is 234°F), increasing the RBCCW cooling water temperature to 149°F will result in an expected HPSI and LPSI seal outlet temperature of 194°F and 234°F, respectively. These temperatures are within the 250°F acceptance criteria for acceptable seal performance.

d) *The Containment Spray pumps also have mechanical seals cooled by RBCCW. Discuss the ability of the RBCCW system to meet this design requirement with the UHS temperature limit rising 5°F from 75°F to 80°F.*

DNC Response

The containment spray (CS) pump seals were previously evaluated at an RBCCW temperature of 147°F. Using vendor supplied information, it was concluded that, at an RBCCW temperature of 149°F, the CS pump seal temperature would be 225°F, which is within the design maximum seal temperature of 300°F.

e) *FSAR Table 9.3.1, "SHUTDOWN COOLING HEAT EXCHANGERS DESIGN BASIS PARAMETERS" lists design parameters for RBCCW and shutdown cooling. Discuss the ability of the RBCCW system to meet these design requirements with the UHS temperature limit rising 5°F from 75°F to 80°F.*

DNC Response

The referenced FSAR table provides tube side and shell side flow rates and heat transfer rates for the SDC heat exchangers at 27.5 hours after shutdown for a normal cooldown and for post LOCA sump recirculation. These values were used in the updated containment analyses for 80°F UHS temperature and can be found in the following locations in DNC's RAI response dated July 19. 2013.

- Shell side flow and heat transfer rate for cooldown is on Attachment 7, Page 4, Case III and, for sump recirculation, Case V.

- Tube side flow for cooldown is on Attachment 5, Page 6, Section 3.2.4. Tube side flow for sump recirculation is on Attachment 2, Page 5, Table 5.2.1, Item 2.b.

The results of the analyses concluded that the SDC heat exchangers can adequately perform their required functions for both normal cooldown and post LOCA sump recirculation.

Background

FSAR Section 9.9.16, "Vital Switchgear Ventilation System," specifies the maximum allowed room temperature limits for the vital AC and DC switchgear rooms. The upper and lower 4160/6190 volt switchgear rooms, the west 480 volt switchgear room, and the east and west vital DC switchgear rooms are the vital switchgear rooms cooled by the UHS. The service water systems from the UHS supply the cooling water to the ventilation systems and refrigerant system that cool these vital switchgear rooms. The current licensing basis uses a maximum UHS temperature of 75°F. FSAR Tables 9.7.5 and 9.9-21 and FSAR Section 9.9.16 provide the component description and room temperature limits of the associated cooling coils as stated below.

Switchgear Room	***FSAR Table***	***Cooling Coil***	***Thermal Performance (Btu/hr)***	***Room Temperature Limit (°F)***
West 480 Volt	*9.7-5*	*X-181 A/B*	*295,641*	*104*
Upper 4160/6900 Volt	*9.7-5*	*X-183*	*166,879*	*122*
Lower 4160/6900 Volt	*9.7-5*	*X-182*	*192,763*	*122*
East and West Vital DC	*9.9-21*	*X-169A/B*	*234,400*	*104*

Issue

For the proposed increase in maximum allowed UHS temperature to 80°F, the NRC staff wants the licensee to verify that the above listed performance requirements are met.

Request
Please answer the following questions separately for each of the switchgear rooms listed in the table above:

1) *With the proposed increase in UHS temperature to 80°F*

   a) *Does the UHS keep room temperature below the temperature limits listed above, and*
   b) *Do the cooling coils have the same or better thermal performance listed above?*

DNC Response
The values provided in the FSAR tables listed above (actually Table 9.9-19 for AC switchgear) are for the heat transfer capabilities of the cooling coils as designed. The required heat removal capacity is lower. The predicted flows provided below were determined in Calculation 12-001, provided as Attachment 8 in DNC's RAI response dated July 19, 2013. These calculations demonstrate that the vital switchgear rooms are maintained within their respective design temperature conditions with a UHS temperature of 80°F. Note that these values, along with the required heat loads, are provided in Enclosure 2 of DNC's supplement letter dated June 27, 2013.

West 480 Volt
Zachry Calculation 12-280 (excerpt provided in Attachment 2) contains the requested information for X-181A/B. It demonstrates that, with 80°F UHS temperature, the required flow is 90 gpm for X-181A and X-181B, combined. Predicted delivered flow is 145 gpm.

Upper 4160/6900 Volt
Zachry Calculation 12-280 contains the requested information for X-183. It demonstrates that, with 80°F UHS temperature, the required flow is 17 gpm for X-183. Predicted delivered flow is 23 gpm.

Lower 4160/6900 Volt
Zachry Calculation 12-339 (excerpt provided in Attachment 3) contains the requested information for X-182. It demonstrates that, with 80°F UHS temperature, the required flow is 15 gpm. Predicted delivered flow is 28 gpm.

East and West Vital DC
Zachry Calculation 13-016 (excerpt provided in Attachment 4) contains the requested information for X-169A/B. It demonstrates that the required flow is 26.9 gpm. Predicted delivered flow is 30 gpm.

2) *If service water flow is increased to compensate for the increase in UHS from 75°F to 80°F to satisfy the performance requirements listed above, do all the other safety related cooling loads supplied by service water have adequate flow to perform their safety functions.*

DNC Response

Service water flow to the components is calculated simultaneously for a number of different cases in Calculation 12-001 (using Proto-Flo), such that the delivered flow to the components must be greater than the required flow. The calculation demonstrates the safety related loads are supplied with adequate flow to perform their required safety function.

3) *Describe the methodology used to answer questions 1 and 2 above.*

DNC Response

To determine the required flow rates, a Proto-HX model for X-181, X-182 and X-183 and a Mathcad model for X169A/B were developed using design inputs (vendor data sheets, heat loading calculations, etc.). These models were benchmarked(1) and validated against the vendor data. The models were then used to determine the minimum required flow rates assuming overall fouling and tube plugging limits. These analyses typically involved iterations on service water flow until the correct heat rate was matched.

To determine the predicted flows, the MPS2 service water Proto-Flo model was used to analyze the five design basis scenarios (i.e., LOCA summer and winter, normal operations, seismic with and without loss of normal power, low tide cold shutdown, and cold shutdown assuming one service water train cools two RBCCW trains (refueling outage scenario)). The analysis used conservative design inputs and assumptions to predict flows. This model, originally developed in the mid-90s, was benchmarked against flow testing data to ensure accuracy(2). The most recent flow testing data was used to assess the flow model uncertainty that is assumed for each component. The predicted flows were reduced by 10% for the RBCCW, EDG, and X-181A/B heat exchangers; and by 15% for X-182, X-183 and X-169A/B heat exchangers (see DNC letter dated July 19, 2013, Attachment 8, Page 23, Assumption 9.2).

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(1) Analyses are benchmarked against the physical system by inputting values into the analytical model that match the field conditions and verifying the results of the model match the results in the field. Minor adjustments to the model may be required to obtain this match. For example, service water pump operating parameters (e.g. differential pressure) and system alignments are input into the model. Flows to each component in the field are accurately measured and compared to what the model predicts. If there is disagreement, minor adjustments are made to flow resistances in the model to get the model to match the field conditions. Model parameters that are adjusted are those that are most difficult to accurately model and are therefore the most likely to need adjustment.

(2) Actual flows to all service water cooled heat exchangers are measured during flow testing. The actual service water pump is not degraded (as it is in the model). After benchmarking the model to the flow test, the pump is degraded in the model and the results are further reduced by the uncertainty below and then reported as predicted flows. Predicted flows are therefore always below and less than measured flows.

Attachment 2

Excerpt from Zachry Calculation 12-280

Dominion Nuclear Connecticut, Inc.
Millstone Power Station Unit 2

1.0 PURPOSE

The purpose of this calculation is to: develop PROTO-HX models of Millstone Power Station Unit 2 Vital Switchgear Room Coolers X-181A/B and X-183 using Version 5.10 of the Air Coil Module; and use the PROTO-HX models to determine the minimum required SW flow rates necessary to remove the design basis cooling loads for the West 480V Load Center Room (X-181A/B) and the Upper 4160V/6900V Switchgear Room (X-183). The thermal performance of each cooler is evaluated with Service Water (SW) inlet temperatures of 75°F and 80°F at conditions of maximum overall fouling and maximum tube plugging.

2.0 EXECUTIVE SUMMARY

Millstone Power Station is evaluating the effects on Unit 2 operation of increasing the allowable Service Water (SW) inlet temperature to 80°F. This calculation determines the minimum required SW flow rate to each cooler to support operation with a SW inlet temperature of 80°F. In addition, the minimum required SW flow rate to each cooler is determined at the current maximum design basis SW temperature of 75°F. Cooler X-181A/B consists of three identical coils (Reference 9.2) with X-181A containing two coils and X-181B containing one coil (Reference 9.3); the discussion that follows is applicable to one coil, which is referred to as X-181.

The minimum required SW flow rates to cooler X-181 for seismic conditions are: 13 gpm per coil when operating with a SW inlet temperature of 80°F; and 9 gpm per coil when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 64,724 Btu/hr per coil with maximum overall fouling and maximum tube plugging.

The minimum required SW flow rates to cooler X-181 for post LOCA conditions are: 30 gpm per coil when operating with a SW inlet temperature of 80°F; and 16 gpm per coil when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 83,248 Btu/hr per coil with maximum overall fouling and maximum tube plugging.

It is noted that the results of this calculation for cooler X-181 are contingent upon revising the minimum acceptable fan flow rate stated in Reference 9.11 from 18,300 cfm to 19,800 cfm consistent with the direction of Reference 9.13.

The minimum required SW flow rates to cooler X-183 for seismic conditions are: 10 gpm when operating with a SW inlet temperature of 80°F; and 9 gpm when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 93,949 Btu/hr with maximum overall fouling and maximum tube plugging.

The minimum required SW flow rates to cooler X-183 for post LOCA conditions are: 17 gpm when operating with a SW inlet temperature of 80°F; and 13 gpm when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 121,536 Btu/hr with maximum overall fouling and maximum tube plugging.

3.0 APPROACH

This calculation evaluates the thermal performance of coolers X-181A/B and X-183 at limiting conditions with 75°F and 80°F SW using the following general approach. Each step listed below is discussed in greater detail in the sections that follow.

- A PROTO-HX model of each cooler is developed using the design inputs listed in Section 4. Cooler X-181A/B is an assembly of three identical coils in parallel; X-181A consists of two coils and X-181B consists of one coil (Reference 9.3). The PROTO-HX model of cooler X-181A/B represents an individual coil which is referred to as X-181.

- The PROTO-HX models are benchmarked and validated against the operating conditions specified on the vendor data sheets contained in Reference 9.2.

- The benchmarked PROTO-HX models are used to determine the minimum required SW flow rate for each coolers with maximum overall fouling and maximum tube plugging. The SW flow rate is adjusted in whole gpm increments until the required heat duty for the particular operating scenario is just met.

3.1 Model Development

The physical configuration and performance data for the coils presented in Design Inputs 4.1 and 4.2 are entered into PROTO-HX under the "HX Data Sheet" menu as described in Reference 9.4. An initial selection of air coil configuration is made from the PROTO-HX library based on similarities of coil layout and geometry. The initial coil configuration selection is the SQR 9.17-1/2T coil with the following Colburn j-factor correlation data points:

- $Re_1 = 300$ $j_1 = 0.013170$
- $Re_2 = 6{,}000$ $j_2 = 0.005000$

This is a continuous (plate) fin coil with staggered tube rows. The actual Colburn j-factor correlation to be applied to the X-181 and X-183 cooling coils are determined through the model benchmarking process discussed in Section 3.3.

The entering air and leaving air conditions provided in Attachment B of Reference 9.2 for each cooler represent dry air with no appreciable change in humidity ratios through the cooling process. Therefore, the performance data listed in Table 2 are entered on the "Dry Performance" tab in PROTO-HX with inlet and outlet relative humidity values of 0%. The models are benchmarked to these dry conditions (i.e., non-condensing) as discussed in Section 3.3.

Similarly, the entering air and leaving air conditions provided in Attachment C of Reference 9.2 for each cooler represent dry air with no appreciable change in humidity ratios through the cooling process. Accordingly, the performance data listed in Table 3 are entered as "User Specified Conditions" in the PROTO-HX models with inlet relative humidity values of 0%. Performance of the PROTO-HX models are compared to the vendor specified performance data after the models are benchmarked to further validate the models. Cooler performance under the conditions used for benchmarking and validation is expected to be similar since both sets of conditions represent non-condensing operation of the coolers.

It is noted that the performance parameters from Reference 9.2 used herein for model development and subsequent benchmarking are based on vendor data from a non-QA source. An analytical model of the coolers was developed in Reference 9.2 to provide independent validation of the non-QA vendor data. Despite differences between the predicted cooler performance from the Reference 9.2 model and the non-QA vendor data, Reference 9.2 concluded that the non-QA vendor data "is valid for use in all related design basis calculations". The differences in predicted cooler performance can be attributed to:

- The Reference 9.2 model was based on a Colburn j-factor versus air-side Reynolds number curve for a standard coil with a similar configuration to the coils installed at Millstone. However, the j-factor curve for the Reference 9.2 model was not adjusted to account for differences between the configurations of the standard coil and the coils installed at Millstone. Adjustment of the j-curve is accomplished through the benchmarking process used with present versions of PROTO-HX (see Section 3.3 for further discussion on model benchmarking).

- The air-side mass flow rates were incorrectly calculated in the Reference 9.2 model based on the vendor data sheet volumetric flow rate and the density of air at the coil inlet conditions which directly affects the calculated heat transfer rates. An evaluation is performed in Section 6.1 of this calculation to confirm that the volumetric flow rates from the vendor data sheet should be applied at the coil outlet conditions.

Based on the discussion above, use of the data from Attachments B and C of Reference 9.2 for model development and benchmarking is considered appropriate for this analysis. Benchmarking the PROTO-HX models to the vendor data, along with proper treatment of the air volumetric flow rate, will yield more accurate predictions of the cooler performance than the evaluations from Reference 9.2.

3.2 Air-Side Flow Rate Correction

PROTO-HX requires air-side volumetric flow rates to be given in units of actual cubic feet per minute (acfm) at either the inlet or outlet conditions. Fan capacities provided in Reference 9.2 are given in both acfm and standard cubic feet per minute (scfm). The following equation from Reference 9.5 is used to convert volumetric flow rates in scfm to acfm:

$$acfm = scfm \times \left(\frac{T_{act}}{T_{std}} \times \frac{p_{std}}{p_{act}}\right) \qquad \text{Equation (1)}$$

Where:

acfm	≡	volumetric flow rate at actual conditions
scfm	≡	volumetric flow rate at standard conditions
T_{act}	≡	actual temperature (°R)
T_{std}	≡	standard temperature (°R)
p_{act}	≡	actual pressure (psia)
p_{std}	≡	standard pressure (psia)

3.3 Model Benchmarking

Once all of the parameters have been entered into PROTO-HX, the model is benchmarked against the vendor performance data by selecting the "Dry Benchmark" option on the "Benchmark" tab under the "Data Sheet" menu. Model benchmarking adjusts the Colburn j-factor as a function of air-side Reynolds number in order to match the vendor's specified heat transfer rate (Reference 9.4). The Colburn j-factor, in turn, is used to calculate the outside film heat transfer coefficient. The resulting correlation for the Colburn j-factor is unique to the cooler and takes the following form:

$$j = e^{a} Re_{a}^{b} \qquad \text{Equation (2)}$$

The exponents "a" and "b" in Equation 2 are determined by the benchmarking process. Close correlation between the outlet temperatures and heat transfer rate indicate that the model accurately reflects the vendor's specified performance.

3.4 Overall Fouling Resistance Validation

The overall fouling resistance for the analysis herein is taken from Reference 9.8 which calculates an overall fouling resistance from test data collected for cooler X-183 in 1997 using Version 3.01 of PROTO-HX. Since the model used in Reference 9.8 is different than the model developed in this calculation, a comparison of output from both models is made in Section 6.3 to verify the appropriateness of the specified fouling limit. The nominal fouling results from Reference 9.8 (including extrapolation conditions) are applied to the model developed herein. Close agreement between the outlet temperatures and heat transfer rates will validate the use of the Reference 9.8 fouling resistance for the current analysis.

In addition, the PROTO-HX model used to calculate the overall fouling resistance in Reference 9.8 was benchmarked to non-QA vendor data from the same source as the data used for benchmarking the models developed herein. Use of the non-QA vendor data for benchmarking PROTO-HX models is considered appropriate as discussed in Section 3.1.

A similar evaluation of test data was performed in Reference 9.1 using the analytical model developed in Reference 9.2. The results of the Reference 9.8 calculation are considered more appropriate for use in this analysis than the results from the Reference 9.1 analysis based on the following:

- The Reference 9.8 model was benchmarked to match the vendor's predicted performance. This benchmarking will produce more accurate predictions of cooler performance as discussed in Section 3.1. The model used in Reference 9.1 was not benchmarked.

- The test evaluated in Reference 9.1 was performed on the coils just after they had been placed into service. The test evaluated in Reference 9.8 was performed on a coil that had been in service for approximately 24 months and, thus, provides a more accurate assessment of fouling accumulation over time.

- The Reference 9.8 analysis includes an analytical determination of test uncertainty. The Reference 9.1 analysis does not include any evaluation of test uncertainty.

- The evaluation of design conditions contained in Reference 9.1 is based on the clean fouling resistances calculated from test data plus an assumed additional fouling resistance of 0.001 hr-ft^2-°F/Btu. No basis is provided in Reference 9.1 for the assumed additional fouling resistance.

3.5 Tube Plugging and Fouling Analysis

Cooler performance is evaluated at conditions of minimum SW flow, maximum tube plugging, and maximum overall fouling by extrapolating cooler performance to "User Specified Conditions". The "User Specified Conditions" represent cooler operation at off-design conditions. For the purpose of this analysis, the potentially limiting scenarios of post LOCA operation and operation following a seismic event are evaluated. The required heat duty for the post LOCA scenario is limiting; however, the available SW flow rate for the seismic scenario is limiting. Therefore, both scenarios are evaluated to ensure the limiting operating conditions for each cooler are properly defined.

A brief description of each operating scenario follows.

Seismic Scenario

The required heat duty of each cooler during a seismic scenario is equivalent to the heat duty required for normal operations as determined in Reference 9.6 (X-181) and Reference 9.7 (X-183). However, the SW flow rate to each cooler is less than the normal SW flow rate due to line breaks that are assumed to occur in non-safety related branch connections in the SW System. Therefore, operation following a seismic event is more limiting than normal operations.

The minimum required SW flow rate is determined such that the resulting heat transfer capacity exceeds the required heat duty with the cooler operating at the limiting conditions identified in Design Input 4.15.

Post LOCA Scenario

Reference 9.6 (X-181) and Reference 9.7 (X-183) calculate the required heat duty of each cooler during an accident scenario with both offsite power available and with a loss of offsite power. For both coolers, the required heat duty during a post LOCA scenario is higher when offsite power is available. Therefore, post LOCA operation with offsite power available is more limiting than when offsite power is lost.

The minimum required SW flow rate is determined such that the resulting heat transfer capacity exceeds the required heat duty with the cooler operating at the limiting conditions identified in Design Input 4.15.

4.0 DESIGN INPUTS

4.1 Physical configuration data for coolers X-181 and X-183 are provided in Reference 9.2 as summarized in Table 1.

Table 1: Physical Configuration Data

Parameter	Cooler	
	X-181	X-183
Coils per Unit	3[a]	1
Coil Length (in)	60	51
Serpentines	1[b]	1[b]
Fin Pitch (fins per in)	9[c]	9[c]
Fin Conductivity (Btu/hr-ft-°F)	128.3	128.3
Fin Thickness (in)	0.0085	0.0085
Tube Rows	4	4
Tubes per Row	16	22
Tube Inner Diameter (in)	0.55	0.55
Tube Wall Thickness (in)	0.049	0.049
Tube Outside Diameter	0.648[d]	0.648[d]
Longitudinal Tube Pitch (in)	1.5	1.5
Transverse Tube Pitch (in)	1.5	1.5
Tube Conductivity (Btu/hr-ft-°F)	25.8	25.8

Table 1 Notes

(a) Room cooler X-181 consists of three identical coils. The PROTO-HX model is based on data for one coil. Heat transfer rates are multiplied by three to provide an equivalent cooler duty.

(b) The number of serpentines is the number of water circuits (16 for X-181 and 22 for X-183) divided by the number tubes per row (Reference 9.4).

(c) PROTO-HX requires the fin pitch to be specified in fins per inch. The vendor data sheet contained in Reference 9.2 states that the coils have 108 fins per foot which is equivalent to 9 fins per inch.

(d) The vendor data sheet contained in Reference 9.2 provides tube wall thickness and tube inner diameter. PROTO-HX requires tube wall thickness and tube outside diameter. The tube outside diameter is calculated as the tube inner diameter plus two times the tube wall thickness.

4.2 The conditions used to benchmark the models of room coolers X-181 and X-183 are provided in Attachment B of Reference 9.2 and are summarized in Table 2.

Table 2: Model Benchmark Conditions

Parameter	Cooler	
	X-181	X-183
Air Flow (scfm)	6,100[a]	6,000
Air Flow (acfm)	6,299[a]	6,197
Inlet Dry Bulb Temperature (°F)	104.0	104.0
Inlet Wet Bulb Temperature (°F)	58.36	58.36
Outlet Dry Bulb Temperature (°F)	86.51	86.57
Outlet Wet Bulb Temperature (°F)	51.29	51.32
Atmospheric Pressure (psia)	14.7	14.7
Coil DP (in WC)	0.6742[b]	0.5010[b]
Tube-Side Fluid	Seawater	Seawater
Tube Flow (gpm)	26.67[a]	25.00
Tube Inlet Temperature (°F)	75	75
Tube Outlet Temperature (°F)	83.86	84.26
Air-Side Fouling (hr-ft^2-°F/Btu)	0.0000	0.0000
Tube-Side Fouling (hr-ft^2-°F/Btu)	0.0005	0.0005
Coil Heat Duty (Btu/hr)	115,500[c]	113,200

Table 2 Notes

(a) Room cooler X-181 consists of three identical coils. Flow rates listed are per coil.

(b) PROTO-HX requires coil pressure drop to be specified in units of psi. The pressure drop specified in Reference 9.2 is converted to psi using the conversion factor given in Design Input 4.11.

(c) Cooler heat duty is per coil.

4.3 The conditions used to validate the benchmarked models of room coolers X-181 and X-183 are provided in Attachment C of Reference 9.2 and are summarized in Table 3.

Table 3: Model Validation Conditions

Parameter	Cooler	
	X-181	X-183
Air Flow (scfm)	6,100[a]	6,000
Air Flow (acfm)	6,315[a]	6,215
Inlet Dry Bulb Temperature (°F)	104.0	104.0
Inlet Wet Bulb Temperature (°F)	58.36	58.36
Outlet Dry Bulb Temperature (°F)	87.86	88.21
Outlet Wet Bulb Temperature (°F)	51.87	52.02
Atmospheric Pressure (psia)	14.7	14.7
Coil DP (in WC)	0.6742[b]	0.5010[b]
Tube-Side Fluid	Seawater	Seawater
Tube Flow (gpm)	20.20[a]	19.10
Tube Inlet Temperature (°F)	75	75
Tube Outlet Temperature (°F)	85.79	85.99
Air-Side Fouling (hr-ft^2-°F/Btu)	0.0000	0.0000
Tube-Side Fouling (hr-ft^2-°F/Btu)	0.0005	0.0005
Coil Heat Duty (Btu/hr)	106,600[c]	102,500

Table 3 Notes

(a) Room cooler X-181 consists of three identical coils. Flow rates listed are per coil.

(b) PROTO-HX requires coil pressure drop to be specified in units of psi. The pressure drop specified in Reference 9.2 is converted to psi using the conversion factor given in Design Input 4.11.

(c) Cooler heat duty is per coil.

4.4 The required cooling capacity of cooler X-181 (including fan heat) is 194,172 Btu/hr for normal operating conditions (Reference 9.6). The equivalent heat duty per coil is 64,724 Btu/hr. The heat duty for normal operating conditions is applied to a seismic scenario as discussed in Section 3.5.

4.5 The required cooling capacity of cooler X-181 (including fan heat) is 249,744 Btu/hr for post LOCA conditions (Reference 9.6). The equivalent heat duty per coil is 83,248 Btu/hr.

4.6 The required cooling capacity of cooler X-183 (including fan heat) is 93,949 Btu/hr for normal operating conditions (Reference 9.7). The heat duty for normal operating conditions is applied to a seismic scenario as discussed in Section 3.5.

4.7 The required cooling capacity of cooler X-183 (including fan heat) is 121,536 Btu/hr for post LOCA conditions (Reference 9.7).

4.8 The limiting conditions for tube plugging established in Reference 9.1 for coolers X-181 and X-183 are summarized in Table 4.

Table 4: Limiting Conditions for Tube Plugging

Parameter	Cooler	
	X-181	X-183
Tube Flow (gpm)	20.20[(a)]	19.10
Tube Plugging Limit (Plugged Tubes per Row)	2[(b)]	3[(b)]

Table 4 Notes

(a) Room cooler X-181 consists of three identical coils. Flow rate listed is per coil.

(b) The tube plugging limits established in Reference 9.1 are based on the flow rates listed above.

4.9 Reference 9.8 demonstrates that a cleaning frequency of at least once every 24 months results in acceptable cooler operation with a SW inlet temperature of 75°F. Calculated fouling resistances from Reference 9.8 are used in this calculation to evaluate cooler performance and are summarized in Table 5. The nominal fouling resistances were calculated in Reference 9.8 from nominal test data and the adjusted fouling resistances take into account test uncertainties. The thermal performance test was performed prior to cleaning cooler X-183 and was repeated after cleaning.

Table 5: X-183 Thermal Performance Test Results

Pre-Cleaning Test Results		Post-Cleaning Test Results	
Nominal Fouling ($hr\text{-}ft^2\text{-}°F/Btu$)	Adjusted Fouling ($hr\text{-}ft^2\text{-}°F/Btu$)	Nominal Fouling ($hr\text{-}ft^2\text{-}°F/Btu$)	Adjusted Fouling ($hr\text{-}ft^2\text{-}°F/Btu$)
0.010254	0.040102	0.007874	0.037548

4.10 The standard pressure for air of 29.921 in Hg (see Assumption 5.1) is equivalent to 14.696 psia (Reference 9.5). For the purpose of this calculation, 14.696 psia is rounded to 14.7 psia. Rounding to the third significant digit has no appreciable effect on the results of this calculation.

4.11 One inch water pressure is equivalent to 0.036091 psi (Reference 9.9).

4.12 The maximum indoor design temperature for the West 480V Switchgear Room (EQ-T04) is 104°F and the Lower 4160V Vital Switchgear Room (EQ-T07) is 122°F (Reference 9.10).

4.13 The minimum air flow rate for fan F-51 (X-181) is 19,800 cfm (Reference 9.13). This is equivalent to 6,600 cfm across each coil. Since the fan is at the coil outlet, this flow rate is applied at the outlet conditions for the analysis herein.

4.14 The minimum air flow rate for fan F-133 (X-183) is 6,000 cfm (Reference 9.12). Since the fan is at the coil outlet, this flow rate is applied at the outlet conditions for the analysis herein.

4.15 The limiting conditions for the analysis are summarized in Table 6. The limiting conditions are evaluated with 75°F SW and 80°F SW.

Table 6: Limiting Conditions

Parameter	Cooler	
	X-181	X-183
Air Flow (acfm)	6,600[(a)]	6,000
Inlet Dry Bulb Temperature (°F)	104.0	122.0
Inlet Relative Humidity (%)	22[(b)]	13[(b)]
Atmospheric Pressure (psia)	14.7	14.7
Coil DP (in WC)	0.6742[(c)]	0.5010[(c)]
Tube Plugging Limit (Plugged Tubes per Row)	2	3
Overall Fouling (hr-ft^2-°F/Btu)	0.040102	0.040102
Coil Heat Duty (Btu/hr)	Seismic – 64,724[(d)] Post LOCA – 83,248[(d)]	Seismic – 93,949 Post LOCA – 121,536

Table 6 Notes

(a) Room cooler X-181 consists of three identical coils. Flow rate listed is per coil.

(b) See Assumption 5.2.

(c) PROTO-HX requires coil pressure drop to be specified in units of psi. The pressure drop is converted to psi using the conversion factor given in Design Input 4.11.

(d) Cooler heat duty is per coil.

5.0 ASSUMPTIONS

5.1 It is assumed that standard conditions for air are 29.921 in Hg, 70°F, and 0% relative humidity (RH). Air at these conditions has a density of 0.075 lb_m/ft^3 (Reference 9.5).

5.2 For the analysis cases herein, it is assumed that the RH of the air in the West 480V Load Center Room (i.e., inlet air RH for X-181) is 22% and the RH of the air in the Upper 4160V/6900V Switchgear Room (i.e., inlet air RH for X-183) is 13%. These RH values correspond to air at dry bulb temperatures of 104°F and 122°F, respectively, with a humidity ratio of 0.010 lb_w/lb_{da}. This humidity ratio is a typical value used in air conditioning applications (Reference 9.9, p. 18.14).

6.0 ANALYSIS

6.1 Air-Side Flow Rate Correction

As discussed in Section 3.2, PROTO-HX requires the air-side volumetric flow rate to be specified in units of acfm at either the inlet or outlet conditions. By inspection, the acfm volumetric flow rates specified in Reference 9.2 appear to be at outlet conditions. To confirm this, Equation 1 is used with the standard conditions specified in Assumption 5.1 and the actual conditions at the coil outlet specified in Design Inputs 4.2 and 4.3. The pressure at the outlet of the coil, p_{act}, is equal to the pressure at the inlet minus the pressure drop across the coil.

X-181 Benchmarking Conditions

The pressure drop across the coil is converted from inches of water to psi as follows:

$$0.6742\ in\ water \times \left(\frac{0.036091\ psi}{1\ in\ water}\right) = 0.0243\ psi$$

The resulting pressure at the coil outlet is:

$$p_{act} = 14.7\ psi - 0.0243\ psi = 14.6757\ psi$$

The volumetric flow rate in acfm at outlet conditions is then calculated as:

$$acfm = 6{,}100\ scfm \times \left(\frac{(86.51 + 459.67)°R}{(70 + 459.67)°R} \times \frac{14.7\ psi}{14.6757\ psi}\right) = 6{,}300.6\ acfm \qquad \text{Equation (1)}$$

The percent error between the calculated value and the vendor specified value is 0.025%. Therefore, the volumetric flow rate used for benchmarking the cooler X-181 model is 6,299 acfm at the coil outlet.

X-181 Validation Conditions (used as "User Specified Conditions")

The pressure losses specified in Attachment B and Attachment C of Reference 9.2 are the same. Therefore, the volumetric flow rate in acfm is calculated as:

$$acfm = 6{,}100\ scfm \times \left(\frac{(87.86 + 459.67)°R}{(70 + 459.67)°R} \times \frac{14.7\ psi}{14.6757\ psi}\right) = 6{,}316.1\ acfm \qquad \text{Equation (1)}$$

The percent error between the calculated value and the vendor specified value is 0.017%. Therefore, the volumetric flow rate entered as "User Specified Conditions" for validation of the cooler X-181 model is 6,315 acfm at the coil outlet.

X-183 Benchmarking Conditions

The pressure drop across the coil is converted from inches of water to psi as follows:

$$0.5010\,in\,water \times \left(\frac{0.036091\,psi}{1\,in\,water}\right) = 0.0181\,psi$$

The resulting pressure at the coil outlet is:

$$p_{act} = 14.7\,psi - 0.0181\,psi = 14.6819\,psi$$

The volumetric flow rate in acfm at outlet conditions is then calculated as:

$$acfm = 6{,}000\,scfm \times \left(\frac{(86.57 + 459.67)°R}{(70 + 459.67)°R} \times \frac{14.7\,psi}{14.6819\,psi}\right) = 6{,}195.3\,acfm \qquad \text{Equation (1)}$$

The percent error between the calculated value and the vendor specified value is -0.027%. Therefore, the volumetric flow rate used for benchmarking the cooler X-183 model is 6,197 acfm at the coil outlet.

X-183 Validation Conditions (used as "User Specified Conditions")

The pressure losses specified in Attachment B and Attachment C of Reference 9.2 are the same. Therefore, the volumetric flow rate in acfm is calculated as:

$$acfm = 6{,}000\,scfm \times \left(\frac{(88.21 + 459.67)°R}{(70 + 459.67)°R} \times \frac{14.7\,psi}{14.6819\,psi}\right) = 6{,}213.9\,acfm \qquad \text{Equation (1)}$$

The percent error between the calculated value and the vendor specified value is -0.018%. Therefore, the volumetric flow rate entered as "User Specified Conditions" for validation of the cooler X-183 model is 6,215 acfm at the coil outlet.

6.2 Model Benchmarking

X-181

After all construction and performance data are entered in PROTO-HX, the model for cooler X-181 is benchmarked as discussed in Section 3.3. The resulting correlation for the Colburn j-factor as a function of air-side Reynolds number is:

$$j = e^{-2.2615} Re_a^{-0.3233} \qquad \text{Equation (2)}$$

The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the vendor performance data in Table 7. The PROTO-HX output report for cooler X-181 model benchmarking is contained in Attachment C.

Table 7: X-181 Model Benchmark Results

Parameter	PROTO-HX	Vendor Data	% Error
Air-side Outlet Temperature (°F)	86.49	86.51	-0.023%
Tube-side Outlet Temperature (°F)	83.86	83.86	0.000%
Heat Transfer Rate (Btu/hr)	115,402	115,500	-0.085%

To further validate the PROTO-HX model, cooler performance at the conditions listed in Table 3 is evaluated. The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the vendor performance data in Table 8. The PROTO-HX output report for X-181 model validation is contained in Attachment D.

Table 8: X-181 Model Validation Results

Parameter	PROTO-HX	Vendor Data	% Error
Air-side Outlet Temperature (°F)	87.86	87.86	0.000%
Tube-side Outlet Temperature (°F)	85.78	85.79	-0.012%
Heat Transfer Rate (Btu/hr)	106,349	106,600	-0.235%

X-183

After all construction and performance data are entered in PROTO-HX, the model for cooler X-183 is benchmarked as discussed in Section 3.3. The resulting correlation for the Colburn j-factor as a function of air-side Reynolds number is:

$$j = e^{-2.2580} Re_a^{-0.3233} \qquad \text{Equation (2)}$$

The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the vendor performance data in Table 9. The PROTO-HX output report for cooler X-183 model benchmarking is contained in Attachment E.

Table 9: X-183 Model Benchmark Results

Parameter	PROTO-HX	Vendor Data	% Error
Air-side Outlet Temperature (°F)	86.56	86.57	-0.012%
Tube-side Outlet Temperature (°F)	84.26	84.26	0.000%
Heat Transfer Rate (Btu/hr)	113,126	113,200	-0.065%

To further validate the PROTO-HX model, cooler performance at the conditions listed in Table 3 is evaluated. The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the vendor performance data in Table 10. The PROTO-HX output report for X-183 model validation is contained in Attachment F.

Table 10: X-183 Model Validation Results

Parameter	PROTO-HX	Vendor Data	% Error
Air-side Outlet Temperature (°F)	88.13	88.21	-0.091%
Tube-side Outlet Temperature (°F)	86.03	85.99	0.047%
Heat Transfer Rate (Btu/hr)	102,937	102,500	0.426%

As can be seen in Tables 7 through 10, the PROTO-HX models of coolers X-181 and X-183 accurately predict model performance as compared to the vendor data sheets contained in Attachment B and Attachment C of Reference 9.2.

6.3 Overall Fouling Resistance Validation

The overall fouling resistance for the analysis herein is taken from Reference 9.8 which calculates an overall fouling resistance from test data collected for cooler X-183 in 1997 using Version 3.01 of PROTO-HX. Since the model used in Reference 9.8 is different than the model developed in this calculation, a comparison of output from both models is made to verify the appropriateness of the specified fouling limit. The nominal fouling results from Reference 9.8 (including extrapolation conditions) are applied to the model developed herein and the results are shown in Table 11.

Table 11: Validation of Overall Fouling Resistance

Parameter	Present Results	Ref. 9.8 Results	% Difference
Air-side Outlet Temperature (°F)	97.40	96.68	0.742%
Tube-side Outlet Temperature (°F)	93.93	94.46	-0.563%
Heat Transfer Rate (Btu/hr)	176,600	181,583	-2.782%

As can be seen by the results in Table 11, there is close agreement between the outlet temperatures and heat transfer rates between the model developed herein for X-183 and the corresponding results from Reference 9.8. Therefore, it is concluded that use of the Reference 9.8 fouling resistance for the current analysis is appropriate.

The PROTO-HX output report for the model of X-183 developed herein as well as the corresponding output report from Reference 9.8 are contained in Attachment G.

6.4 Tube Plugging and Fouling Analysis

X-181

The benchmarked model of cooler X-181 is used to evaluate thermal performance of the cooler at the limiting conditions identified in Design Input 4.15 with SW inlet temperatures of 75°F SW and 80°F. The results are summarized in Table 12.

Table 12: X-181 Thermal Performance Results

Operating Scenario	SW Inlet Temperature (°F)	Minimum Required SW Flow Rate (gpm)[a]	Cooler Capacity (Btu/hr)[a]	Required Heat Duty (Btu/hr)[a]
Seismic	75	9	64,837	64,724
Post LOCA	75	16	85,009	83,248
Seismic	80	13	65,204	64,724
Post LOCA	80	30	83,299	83,248

Table 12 Notes

(a) Room cooler X-181 consists of three identical coils. Flow rates, cooler capacities, and required heat duties listed are per coil.

The minimum required flow rates, coil capacities, and required heat duties provided in Table 12 are for one coil. Multiply these values by 3 to obtain the total for the cooler.

The supporting PROTO-HX output reports for the thermal performance evaluation of X-181 are included in Attachment H.

X-183

The benchmarked model of cooler X-183 is used to evaluate thermal performance of the cooler at the limiting conditions identified in Design Input 4.15 with SW inlet temperatures of 75°F SW and 80°F. The results are summarized in Table 13.

Table 13: X-183 Thermal Performance Results

Operating Scenario	SW Inlet Temperature (°F)	Minimum Required SW Flow Rate (gpm)	Cooler Capacity (Btu/hr)	Required Heat Duty (Btu/hr)
Seismic	75	9	99,414	93,949
Post LOCA	75	13	122,184	121,536
Seismic	80	10	96,033	93,949
Post LOCA	80	17	122,762	121,536

The supporting PROTO-HX output reports for the thermal performance evaluation of X-183 are included in Attachment I.

7.0 CONCLUSION

The minimum required SW flow rates to cooler X-181 for seismic conditions are: 13 gpm per coil when operating with a SW inlet temperature of 80°F; and 9 gpm per coil when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 64,724 Btu/hr per coil with maximum overall fouling and maximum tube plugging.

The minimum required SW flow rates to cooler X-181 for post LOCA conditions are: 30 gpm per coil when operating with a SW inlet temperature of 80°F; and 16 gpm per coil when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 83,248 Btu/hr per coil with maximum overall fouling and maximum tube plugging.

It is noted that the results of this calculation for cooler X-181 are contingent upon revising the minimum acceptable fan flow rate stated in Reference 9.11 from 18,300 cfm to 19,800 cfm consistent with the direction of Reference 9.13.

The minimum required SW flow rates to cooler X-183 for seismic conditions are: 10 gpm when operating with a SW inlet temperature of 80°F; and 9 gpm when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 93,949 Btu/hr with maximum overall fouling and maximum tube plugging.

The minimum required SW flow rates to cooler X-183 for post LOCA conditions are: 17 gpm when operating with a SW inlet temperature of 80°F; and 13 gpm when operating with a SW inlet temperature of 75°F. These flow rates are based on a required heat duty of 121,536 Btu/hr with maximum overall fouling and maximum tube plugging.

The PROTO-HX models, M2X181.phxac (of file size 736 KB and last saved on 10/10/12 at 9:03 AM EDT) and M2X183.phxac (of file size 736 KB and last saved on 10/10/12 at 9:09 AM EDT), are included with this calculation as Attachment J. These PROTO-HX models have been verified in accordance with Zachry procedure N0301 (Reference 9.14) and the resulting output is suitable for use in Safety Related calculations.

8.0 PRECAUTIONS AND LIMITATIONS

This calculation has been prepared in support of DC MP2-12-01205 (Millstone Unit 2: Increase in Ultimate Heat Sink Temperature Limit from 75°F to 80°F). Plant operation at an UHS temperature of 80°F is not permitted until such time as the associated UHS technical specification has been revised. However, the results of this calculation are bounding of plant operation at UHS temperatures less than 80°F.

The results of this calculation are contingent upon revising the minimum acceptable fan flow rate stated in Reference 9.11 from 18,300 cfm to 19,800 cfm consistent with the direction of Reference 9.13.

Attachment 3

Excerpt from Zachry Calculation 12-339

Dominion Nuclear Connecticut, Inc.
Millstone Power Station Unit 2

1.0 PURPOSE

This calculation documents the development of a PROTO-HX model of Millstone Power Station Unit 2 Vital Switchgear Room Cooler X-182 using Version 5.10 of the Air Coil Module. The model is used to evaluate the thermal performance of the cooler with a Service Water (SW) inlet temperature of 80°F. The following results are reported:

- Heat transfer capacity for cooler operation with the design SW flow rate, maximum tube plugging, and design fouling.
- Heat transfer capacity for cooler operation with the design SW flow rate, maximum tube plugging, and maximum fouling.
- The minimum required SW flow rate for cooler operation with maximum tube plugging and design fouling.
- The minimum required SW flow rate for cooler operation with maximum tube plugging and maximum fouling.

2.0 EXECUTIVE SUMMARY

Millstone Power Station is evaluating the effects on Unit 2 operation of increasing the allowable Service Water (SW) inlet temperature to 80°F. This calculation evaluates the thermal performance of Vital Switchgear Room Cooler X-182 with a SW inlet temperature of 80°F.

The heat transfer capacity of cooler X-182 is 161,792 Btu/hr when operating with the design SW flow rate (22.9 gpm) at an inlet temperature of 80°F, maximum tube plugging (3 tubes per row plugged), and design fouling (0.0016 hr-ft^2-°F/Btu applied to the tube-side). At these conditions, the cooler has 30,539 Btu/hr of heat transfer margin above the required heat transfer rate of 131,253 Btu/hr.

The heat transfer capacity of cooler X-182 is 157,452 Btu/hr when operating with the design SW flow rate of 22.9 gpm at an inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and maximum overall fouling (0.040102 hr-ft^2-°F/Btu). At these conditions, the cooler has 26,199 Btu/hr of margin above the required heat transfer rate of 131,253 Btu/hr.

The minimum required SW flow rate to cooler X-182 is 14 gpm when operating with a SW inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and design fouling (0.0016 hr-ft^2-°F/Btu applied to the tube-side). At these conditions, the cooler heat transfer capacity is 132,514 Btu/hr.

The minimum required SW flow rate to cooler X-182 is 15 gpm when operating with a SW inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and maximum overall fouling (0.040102 hr-ft^2-°F/Btu). At these conditions, the cooler heat transfer capacity is 133,905 Btu/hr.

3.0 APPROACH

This calculation evaluates the thermal performance of cooler X-182 using the following general approach. Each step listed below is discussed in greater detail in the sections that follow.

- A PROTO-HX model of the cooler is developed using the coil construction and performance details discussed in Sections 4.0, 5.0, and 6.1.
- The PROTO-HX model is benchmarked against the Aerofin performance verification calculation for Case 1 contained in Reference 9.1.
- The benchmarked PROTO-HX model is used to evaluate the thermal performance of the cooler with 80°F SW.

3.1 Model Development and Benchmarking

The physical configuration and performance data for the coil presented in Sections 4.0, 5.0, and 6.1 are entered into PROTO-HX under the "HX Data Sheet" menu as described in Reference 9.2. An initial selection of air coil configuration is made from the PROTO-HX library based on similarities of coil layout and geometry. The initial coil configuration selection is the CF-9.05-3/4J A coil with the following Colburn j-factor correlation data points:

- $Re_1 = 1{,}400$ $j_1 = 0.008016$
- $Re_2 = 5{,}500$ $j_2 = 0.005006$

Preliminary analyses indicate that the air-side Reynolds number is below the minimum value of 1,400 for all cases considered herein. Therefore, a different configuration is chosen based on the next closest representation of the coil layout and geometry with a Reynolds number range that encompasses the values considered in this analysis. The final coil configuration selection is the CF-8.72(c) coil with the following Colburn j-factor correlation data points:

- $Re_1 = 500$ $j_1 = 0.018176$
- $Re_2 = 7{,}800$ $j_2 = 0.005565$

This is a circular fin coil with staggered tube rows. As noted in Reference 9.2, the precise selection of library configuration is not critical because the actual Colburn j-factor correlation applied to the X-182 cooling coil is determined through the model benchmarking process discussed below.

The original design conditions for the IHT coil documented in Reference 9.3 represent dry air with no appreciable change in humidity ratios through the cooling process. Consistent with the original design conditions, the performance data listed in Table 2 are entered on the "Dry Performance" tab in PROTO-HX with inlet and outlet relative humidity values of 0%. The model is benchmarked to these dry conditions (i.e., non-condensing).

The industry standard practice for evaluating air coils is to correct the tube-side heat transfer resistances (i.e., tube-side fouling resistance and tube-side film resistance) to the outside heat transfer surface area by multiplying these resistances by the ratio of outside to inside heat transfer surface areas. To ensure that PROTO-HX corrects these resistances to the outside area in a manner which is consistent with the Reference 9.1 calculation, the "Vendor Area" term is used for model benchmarking. The "Vendor Area" for X-182 is determined by multiplying the inside heat transfer surface area calculated by PROTO-HX by the area ratio from Reference 9.1 as discussed in Section 6.2. The area factor is then calculated as discussed in Reference 9.2.

Once all of the parameters have been entered into PROTO-HX and the area factor has been calculated, the model is benchmarked against the Aerofin performance calculation by selecting the "Dry Benchmark" option on the "Benchmark" tab under the "Data Sheet" menu. Model benchmarking adjusts the Colburn j-factor as a function of air-side Reynolds number in order to match the vendor's specified heat transfer rate (Reference 9.2). The Colburn j-factor, in turn, is used to calculate the outside film heat transfer coefficient. The resulting correlation for the Colburn j-factor is unique to the cooler and takes the following form:

$$j = e^{a} Re_{a}^{b}$$

The exponents (a and b) in the above equation are determined by the benchmarking process. Close correlation between the outlet temperatures and heat transfer rate indicate that the model accurately reflects the vendor's specified performance.

3.2 Thermal Performance Analysis

The benchmarked model is used to evaluate thermal performance of cooler X-182 with the design SW flow rate at an inlet temperature of 80°F and maximum tube plugging. Design fouling and maximum fouling resistances are evaluated. The heat transfer capacity of the cooler is determined for each case.

In addition, minimum required SW flow rates are determined for each level of fouling by manually iterating the SW flow rate until there is no margin on the heat transfer rate. As with the design flow rate cases, maximum tube plugging is used for the minimum SW flow rate cases.

The thermal performance analysis is performed using the same area factor calculated for model benchmarking in order to keep the model in a configuration as close to the benchmark configuration as possible and to maintain the area ratio at a value that is consistent with Reference 9.1.

4.0 DESIGN INPUTS

4.1 The design basis temperature for the Lower 4160V Switchgear Room (EQ Zone EQ-T07) is 122°F (Reference 9.4).

4.2 The required cooling capacity of cooler X-182 is 131,253 Btu/hr (Reference 9.5). This capacity is based on a room temperature of 122°F and includes the heat from fan F-134 which is integral to the cooler.

4.3 Physical configuration data for cooler X-182 is provided in References 9.1 and 9.6 as summarized in Table 1. Additional physical parameters which are required to model the coil in PROTO-HX are defined elsewhere throughout this calculation.

Table 1: Physical Configuration Data

Parameter	Value
Coils per Unit	1
Coil Length (in)	59
Fin Pitch (fins per in)	12
Fin Thickness (in)	0.012
Fin Height (in)	0.375
Tube Rows	5
Tubes per Row	22[(a)]
Tubes Fed	28
Tube Wall Thickness (in)	0.049
Tube Outside Diameter (in)	0.625
Transverse Tube Pitch (in)	1.39

Table 1 Notes

(a) The coil consists of three rows with 23 tubes and two rows with 22 tubes (Reference 9.6). PROTO-HX requires an equal number of tubes per row. Therefore, this analysis uses 22 tubes per row consistent with the Aerofin performance verification calculations documented in Reference 9.1.

4.4 The tube plugging limiting established in Reference 9.3 for cooler X-182 is 3 tubes per row.

4.5 The minimum flow capacity for fan F-134 is 7,200 cfm (Reference 9.7).

4.6 The X-182 design conditions used by Aerofin in Case 1 of Reference 9.1 are summarized in Table 2.

Table 2: Aerofin Case 1 Conditions

Parameter	Value
Air Flow at Coil Outlet (acfm)	7,200
Inlet Dry Bulb Temperature (°F)	122
Outlet Dry Bulb Temperature (°F)	97.4
Atmospheric Pressure (psia)	14.696
Coil DP (in WC)	0.50[(a)]
Tube-Side Fluid	Salt Water
Tube Flow (gpm)	22.9
Tube Inlet Temperature (°F)	77
Tube Outlet Temperature (°F)	93.5
Air-Side Fouling (hr-ft^2-°F/Btu)	0.0000
Tube-Side Fouling (hr-ft^2-°F/Btu)	0.0016
Coil Heat Duty (Btu/hr)	182,073

Table 2 Notes

(a) PROTO-HX requires coil pressure drop to be specified in units of psi. The pressure drop specified in Reference 9.1 is converted to psi using the conversion factor given in Design Input 4.7.

4.7 One inch water pressure is equivalent to 0.036091 psi (Reference 9.8).

5.0 ASSUMPTIONS

5.1 Dry air with 0% relative humidity is assumed for the analysis herein. This assumption is consistent with the design conditions listed in Reference 9.3 for the original IHT coil.

5.2 It is assumed that fin thickness is the same at the root and tip. This assumption is based on the Reference 9.6 description of the fins as helical "L" and the Reference 9.2 guidance for spiral wound fins.

5.3 It is assumed that the thermal conductivity of the fin material (aluminum) is 128 Btu/hr-ft-°F. This assumption is based on the 1060 Aluminum thermal conductivity available in the standard PROTO-HX material library.

5.4 It is assumed that the tube rows are symmetrically spaced along the depth of the coil (i.e., equal space between adjacent tube rows) which is typical of standard coil construction.

5.5 It is assumed that the thermal conductivity of the tube material (AL6XN) is 7.5 Btu/hr-ft-°F. This assumption is based on vendor literature for tube material meeting the specification requirements listed on Reference 9.6 for the tubes. The vendor literature is included in Attachment B.

5.6 It is assumed that the maximum overall fouling resistance for cooler X-182 is 0.040102 hr-ft^2-°F/Btu. This assumption is based on the test data analysis of Reference 9.9 which calculated the demonstrated fouling resistance (i.e., nominal result plus uncertainty) for cooler X-183 based on a 24-month service cycle. The demonstrated fouling resistance from Reference 9.9 is considered applicable to X-182 based on the following:

- The coils are similar in construction with the most significant difference being that X-182 has circular fins and X-183 has plate fins (Reference 9.3) which potentially affects the air-side fouling. Air-side fouling is typically negligible compared to tube-side fouling and the ratio of tube inside surface area to outside surface area is similar for both coils making the effect of tube-side versus air-side fouling essentially the same.

- Both coolers are exposed to SW from the same source (Reference 9.10). Therefore, both coolers will receive SW at the same temperature. In addition, the design limit SW temperature is the same for both coolers.

- Both coolers draw air from separate electrical switchgear rooms and recirculate cooled air back to the same room (Reference 9.11). The air in these rooms is relatively dry and clean and is filtered prior to entry into the coils. The air-side fouling potential, while small, is identical for both coils.

- Both coolers have continuous SW flow to prevent stagnant water conditions and to ensure proper distribution of the sodium hypochlorite injected into the SW System for microfouling and macrofouling control. The SW flow rate to both coolers is similar.

- Both coolers are normally in service to provide room cooling when the unit is at power. The air-side flow rate to both coolers is similar.

- Both coolers have the same size tubing (5/8" OD x 0.049" wall per References 9.6 and 9.13).

- The tube-side velocities under consideration for the present analysis are similar to the tube-side velocities experienced during the performance testing of X-183.

6.0 ANALYSIS

6.1 Model Development

Complete definition of a PROTO-HX model requires more information than what is explicitly defined in References 9.1 and 9.6. The following terms are calculated in support of model development.

Serpentines

The number of serpentines is defined by Reference 9.2 as the number of water circuits (tubes fed in the Aerofin design data) divided by the number of tubes per row. The number of serpentines is calculated as:

$$\frac{28}{22} = 1.272727272727$$

It is noted that cooler X-182 has three rows containing 23 tubes and two rows containing 22 tubes (Reference 9.6). With this configuration, the number of serpentines is approximately 1.24. PROTO-HX models are based on an equal number of tubes per row; therefore, this calculation conservatively uses 22 tubes per row. The number of serpentines calculated above ensures that the tube-side velocities are calculated correctly based on 28 tubes per pass.

Circular Fin Height

The circular fin height as illustrated in Figure 18 of Reference 9.2 is calculated as follows:

$$CFH = d_o + 2 \times FH = 0.625\ in + 2 \times 0.375\ in = 1.375\ in$$

Where:

CFH	≡	circular fin height (in)
d_o	≡	tube outside diameter (0.625 in)
FH	≡	fin height (0.375 in)

Longitudinal Tube Pitch

The depth of the coil casing is 9.5 in (Reference 9.6). By inspection of Reference 9.6, it is seen that the distance from the edge of the casing to the centerline of the two outer-most tube rows is approximately the same as the distance from the edge of the casing to the centerline of the SW inlet and outlet nozzles which is 2.19 in. The resulting longitudinal tube pitch is:

$$\frac{(9.5 - 2 \times 2.19)in}{4} = 1.28\ in$$

It is noted that the longitudinal tube pitch calculated above is in agreement with the nominal tube spacing value listed on page 41 of Reference 9.1.

6.2 Model Benchmark

After all construction and performance data are entered in PROTO-HX, the model for cooler X-182 is benchmarked as discussed in Section 3.1. First, an area factor is calculated from the "Vendor Area" and then the "Dry Benchmark" is performed to match model performance to the Aerofin performance verification calculation.

Vendor Area

To ensure that PROTO-HX corrects the tube-side heat transfer resistances to the outside heat transfer surface area in a manner consistent with the Reference 9.1 calculation, the "Vendor Area" term is used for model benchmarking. The "Vendor Area" is calculated such that the ratio of outside heat transfer surface area to inside heat transfer surface area used by PROTO-HX is the same as the area ratio used in the Reference 9.1 calculation. The following equation is used by PROTO-HX to calculate the inside heat transfer surface area (Reference 9.2):

$$A_i = \pi \times N_{AT} \times N_L \times d_i \times L_C$$

Where:

A_i	≡	inside surface area (ft^2)
N_{AT}	≡	number of active (unplugged) tubes per row (22)
N_L	≡	number of tube rows (5)
d_i	≡	tube inside diameter (0.527 in/12-in/ft)
L_C	≡	coil effective tube length (59 in/12-in/ft)

The resulting inside surface area is:

$$A_i = \pi \times 22 \times 5 \times 0.527\ in \times 59\ in = 10{,}745\ in^2 = 74.618\ ft^2$$

Although not explicitly defined in Reference 9.1, it is apparent by its application to the inside film resistance and inside fouling resistance that the area ratio used by Aerofin is 19.45 (refer to the term "B" on page 18 of Reference 9.1).

Next, the "Vendor Area" is calculated as follows:

$$Vendor\ Area = 19.45 \times 74.618\ ft^2 = 1{,}451.3\ ft^2$$

The area factor is then calculated by pressing the "Area Factor" button.

Dry Benchmark

After the area factor is calculated, the "Dry Benchmark" button is pressed to adjust the Colburn j-factor as a function of Reynolds number as discussed in Section 3.1. The resulting correlation for the Colburn j-factor as a function of air-side Reynolds number is:

$$j = e^{-1.8972} Re_a^{-0.4306}$$

The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the Aerofin verification calculation in Table 3. The PROTO-HX output report for cooler X-182 model benchmarking is contained in Attachment C.

Table 3: X-182 Benchmark Results to Aerofin Case 1

Parameter	PROTO-HX	Aerofin Case 1 Results	% Error
Air-side Outlet Temperature (°F)	97.32	97.4	-0.082%
Tube-side Outlet Temperature (°F)	93.31	93.5	-0.203%
Heat Transfer Rate (Btu/hr)	182,418	182,073	0.189%

As can be seen in Table 3, the benchmarking of the PROTO-HX model provides close agreement with the Aerofin calculation contained in Reference 9.1.

6.3 Model Performance Validation

To further validate the PROTO-HX model, predicted coil performance is compared to the Case 5 calculation from Reference 9.1 which calculates the heat transfer rate of the coil with the same flow rates and inlet temperatures from Case 1, but with fresh water and zero fouling. Therefore, the tube-side fluid is changed from "Salt Water" to "Fresh Water" on the data sheet and the calculation is performed with an input fouling factor of 0 hr-ft^2-°F/Btu.

The PROTO-HX calculated outlet temperatures and heat transfer rate are compared to the Aerofin verification calculation results in Table 4. The PROTO-HX output report for cooler evaluation at Aerofin Case 5 conditions is contained in Attachment D.

Table 4: X-182 Performance at Aerofin Case 5 Conditions

Parameter	PROTO-HX	Aerofin Case 5 Results	% Error
Air-side Outlet Temperature (°F)	94.77	95.2	-0.452%
Tube-side Outlet Temperature (°F)	94.69	94.6	0.095%
Heat Transfer Rate (Btu/hr)	202,166	198,974	1.604%

As can be seen in Table 4, the PROTO-HX model predictions of cooler X-182 performance provides close agreement with the Aerofin calculation contained in Reference 9.1.

6.4 Thermal Performance Analysis

The benchmarked model of cooler X-182 is used to evaluate thermal performance as a function of SW flow rate and fouling as discussed in Section 3.2. Maximum tube plugging is used for all cases. The results are as follows:

Cooler Performance at Design SW Flow Rate and Fouling

The heat transfer capacity of cooler X-182 is 161,792 Btu/hr when operating with the design SW flow rate (22.9 gpm) at an inlet temperature of 80°F, maximum tube plugging (3 tubes per row plugged), and design fouling (0.0016 hr-ft^2-°F/Btu applied to the tube-side). At these conditions, the cooler has 30,539 Btu/hr of heat transfer margin above the required heat transfer rate 131,253 Btu/hr.

The PROTO-HX output report for this case is contained on pages 2 and 3 of Attachment E.

Cooler Performance at Design SW Flow Rate and Maximum Fouling

The heat transfer capacity of cooler X-182 is 157,452 Btu/hr when operating with the design SW flow rate of 22.9 gpm at an inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and maximum overall fouling (0.040102 hr-ft^2-°F/Btu). At these conditions, the cooler has 26,199 Btu/hr of margin over the required heat transfer rate of 131,253 Btu/hr.

The PROTO-HX output report for this case is contained on pages 4 and 5 of Attachment E.

Minimum Required SW Flow Rate for Design Fouling

The minimum required SW flow rate to cooler X-182 is 14 gpm when operating with a SW inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and design fouling (0.0016 hr-ft^2-°F/Btu applied to the tube-side). At these conditions, the cooler heat transfer capacity is 132,514 Btu/hr.

The PROTO-HX output report for this case is contained on pages 2 and 3 of Attachment F.

Minimum Required SW Flow Rate for Maximum Fouling

The minimum required SW flow rate to cooler X-182 is 15 gpm when operating with a SW inlet temperature of 80°F, maximum tube plugging (3 tubes per row), and maximum overall fouling (0.040102 hr-ft^2-°F/Btu). At these conditions, the cooler heat transfer capacity is 133,905 Btu/hr.

The PROTO-HX output report for this case is contained on pages 4 and 5 of Attachment F.

7.0 CONCLUSION

An evaluation of the thermal performance of Vital Switchgear Room Cooler X-182 is made using a benchmarked PROTO-HX model of the cooler. The purpose of this evaluation is to determine if the cooler has sufficient heat transfer capacity to operate with an 80°F Service Water inlet temperature.

Vital Switchgear Room Cooler X-182 has sufficient capacity to operate with 80°F SW at the design SW flow rate (22.9 gpm), maximum overall fouling (0.040102 hr-ft^2-°F/Btu), and maximum tube plugging (3 tubes per row). At these conditions, the cooler has a heat transfer capacity of 157,452 Btu/hr which provides 26,199 Btu/hr of margin over the required heat transfer rate of 131,253 Btu/hr.

The minimum required SW flow rate to cooler X-182 is 15 gpm at 80°F, maximum overall fouling (0.040102 hr-ft^2-°F/Btu), and maximum tube plugging (3 tubes per row). At these conditions, the cooler heat transfer capacity is 133,905 Btu/hr.

The PROTO-HX model, M2X182.phxac (of file size 736 KB and last saved on 1/31/13 at 7:37 PM EST) is included with this calculation as Attachment G. The PROTO-HX model has been verified in accordance with Zachry procedure N0301 (Reference 9.12) and the resulting output is suitable for use in Safety Related calculations. The model benchmark output shows close agreement with the Aerofin performance verification calculation documented in Reference 9.1.

8.0 PRECAUTIONS AND LIMITATIONS

The maximum SW inlet temperature used for the analysis herein (80°F) does not represent the current Design Basis for Millstone Power Station Unit 2. As such, the results of this calculation do not represent the Design Basis analysis of cooler X-182 until such time that the increased Ultimate Heat Sink temperature is fully implemented in the Unit 2 Licensing and Design Bases.

9.0 REFERENCES

9.1 Calculation 11-ENG-X182-04341M2; Millstone Unit 2 X182 Replacement Aerofin Cooling Coil Thermal Performance; Revision 0

9.2 PROTO-HX Air Coil Version 5.10 User Documentation

9.3 Calculation 95-DES-1242D2; Performance Evaluation of Installed Replacement Room Coolers (X-181, X-182, X-183) From Test Data; Revision 0

9.4 Calculation 99HVAC-02792M2; Plant Indoor Design Temperatures During Normal Plant Operation; Revision 1 (including CCN 1)

9.5 Calculation 92-FFP-932ES; MP2 Lower Switchgear Room 4.16 and 6.9 kV Heat Gains and Maximum Room Temperature; Revision 3

9.6 Drawing 25203-29136 Sheet 98A; Type "R" Coil; Revision 1

9.7 Procedure EN 21063F-002; Lower 4160V Switchgear Room Ventilation System Test; Revision 006-01

9.8 2009 ASHRAE Handbook; Fundamentals

9.9 Calculation 97-211; MP2 Vital AC Switchgear Room Cooler X-183 Thermal Performance Test Analysis; Revision 0

9.10 Drawing 25203-26008 Sheet 3; Millstone Power Station Unit – 2, Piping & Instrumentation Diagram, Service Water to Vital AC Switchgear Cooling Coil and AC Chillers; Revision 32

9.11 Drawing 25203-26027 Sheet 1; Millstone Power Station – Unit 2, Piping & Instrumentation Diagram, HVAC System Turb. Bldg., Intake Str, Whse, & Diesel Gen. Rms.; Revision 48

9.12 Zachry Nuclear, Inc. Procedure N0301; Engineering Calculations; Revision 0

9.13 Drawing 25203-29136 Sheet 99; Coil Assembly X-183 33 x 51 4-Row R. H.; Revision 4

Attachment 4

Excerpt from Zachry Calculation 13-016

Dominion Nuclear Connecticut, Inc.
Millstone Power Station Unit 2

1.0 PURPOSE

The purpose of this calculation is to evaluate the performance of the Millstone Unit 2 Vital Chiller Condensers (X-169A/B) when the Service Water System (SW) inlet temperature is 80°F. Specifically, this calculation determines the condensing temperature and pressure of the refrigerant when operating at rated load conditions and 80°F SW. Performance with 80°F SW is considered acceptable if X-169A/B can remove the required condenser heat duty without exceeding the high pressure cutout setting for the associated compressor.

2.0 BACKGROUND

Millstone Power Station is evaluating the effects on Unit 2 operation of increasing the maximum allowable Ultimate Heat Sink (UHS) temperature to 80°F. The Vital Chiller Condensers (X-169A/B) are cooled by the Service Water System (SW) and reject the cooling load from the DC Switchgear Room A/C Units (X-84A/B) to the UHS via evaporators X-169C/D during accident conditions (References 9.1 and 9.2). As such, an increase in SW temperature results in a higher condensing temperature at X-169A/B which, in turn, results in higher pressures in the refrigerant cycle. If the attendant increase in condensing pressure is too high, the compressor will trip on a high pressure cutout signal.

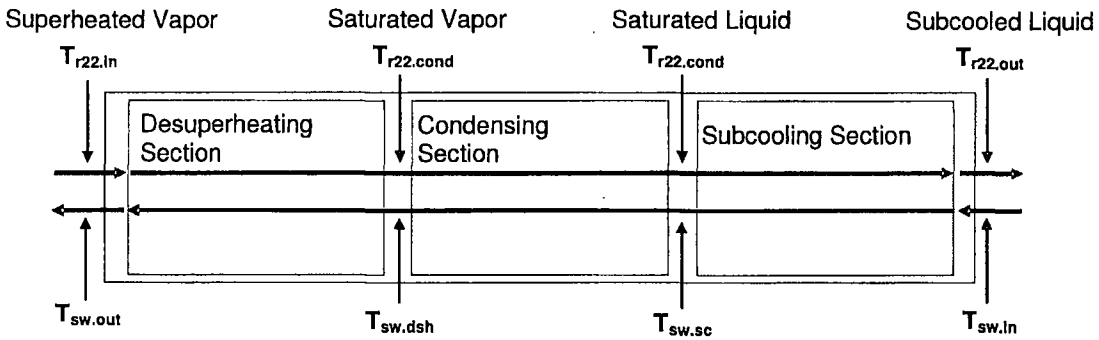
This calculation evaluates X-169A/B performance with a SW supply temperature of 80°F at the limiting conditions of minimum acceptable SW flow, design heat load, maximum tube plugging, and design fouling.

3.0 APPROACH

3.1 Model Development

Mathcad Version 15 is used to develop a model of the Vital Chiller Condensers based on the physical characteristics of the heat exchangers provided in Section 4.0 and Section 5.0. The model consists of three unique areas of the heat exchanger as shown in Figure 1.

Figure 1: Condenser Model



The superheated vapor entering the condenser is first de-superheated as it flows over the outside surface of the bank of tubes. Once saturation has been reached, the condensation process occurs as the latent heat of vaporization is removed from the refrigerant stream. The saturated liquid then becomes subcooled with continued contact with the tube surfaces. There is no defined barrier between these sections of the condenser so that the actual area utilized is a function of the incoming vapor superheat, the overall heat load, the cooling water flow rate, and the cooling water supply temperature.

Each section illustrated in Figure 1 is analyzed as a separate and distinct heat transfer process (i.e., sensible heat transfer in the desuperheating section, latent heat transfer in the condensing section, and sensible heat transfer in the subcooling section). The outlet conditions from each section are linked with the inlet conditions of the following section as shown. Equation 1 through Equation 36 are used in the analysis of each section with the overall solution obtained with multiple iterations until convergence is achieved. The solution process is described in greater detail in Section 6.0. The model is benchmarked to match the performance parameters specified in the vendor's TEMA data sheet which is included in Attachment D.

3.1.1 Desuperheating Section

The heat transfer rate for the desuperheating section is defined using Equation 2 and the enthalpy change across the desuperheating section. The inlet enthalpy is found from the given condensing temperature (and associated saturation pressure) in conjunction with the specified refrigerant inlet temperature. The outlet enthalpy of the desuperheating section is taken as the saturated vapor enthalpy at the condensing pressure/temperature.

The cooling water temperature entering the desuperheating section is back-calculated by setting Equation 1 equal to Equation 2 and re-arranging terms.

The log mean temperature difference is then calculated for the desuperheating section using the four terminal temperatures associated with that section and Equation 6. The corresponding correction factor is found using Equation 7 through Equation 11.

Attention is now focused on the calculation of the thermal resistances that make up the overall heat transfer coefficient. The overall fouling resistance is corrected to the outside area using Equation 30 and the data sheet values for shell-side and tube-side fouling resistances. This value is held as constant throughout the condenser. The wall resistance is calculated using Equation 31 and is also held constant throughout the condenser.

Fluid properties for both the shell and tube sides are calculated based on the average temperature on each side.

The tube-side and shell-side Reynolds numbers are calculated using Equation 26 and Equation 32, respectively. An initial guess of the tube-side and shell-side wall temperatures is made and the inside film coefficient is calculated using Equation 23, Equation 24, or Equation 25 depending on the flow regime and the outside film coefficient is calculated using Equation 34. The guess of the two wall temperatures is updated using Equation 29 and Equation 35 along with the calculated film coefficients and surface area involved in the heat transfer process. Both the tube-side and shell-side film coefficients are then recalculated based on revised wall temperatures. This process is repeated until successive iterations produces no significant change in the calculated wall temperatures compared to the previous values. These iterations are integrated with the calculation of the affected surface area via the overall heat transfer coefficient as discussed below.

The overall heat transfer coefficient is calculated using Equation 22 and the four thermal resistances just calculated. The outside film correction factor (X_c) starts at 1.0 but is manually adjusted in concert with the solutions for the other two sections of the heat exchanger (condensing and subcooling) until the overall thermal performance matches that specified by the heat exchanger vendor using the specified heat transfer surface area.

The surface area required to accommodate the heat transfer rate specified for the desuperheating section is back-calculated using Equation 3 after each iterative calculation of the overall heat transfer coefficient. This area is used to calculate wall temperatures and ultimately yields a convergent solution.

3.1.2 Condensing Section

The analytical method described for the desuperheating section is employed in the condensing section with two major differences:

- The outside film coefficient is calculated using Equation 36;
- There is no log mean temperature difference correction factor for the case where there is a change of state on the shell-side.

3.1.3 Subcooling Section

The analytical method described in 3.1.1 for the desuperheating section is employed in the subcooling section as well. The heat transfer rate is based on the enthalpy change from the saturated liquid at the condensing temperature/pressure to the specified subcooled temperature of the R22 outlet. The enthalpy of the subcooled liquid is approximated by the enthalpy of saturated liquid at the condenser outlet temperature. This approximation is reasonable as there is very little subcooling (3°F) and the difference in enthalpy between subcooled enthalpy and saturated enthalpy at the same conditions of temperature and pressure is minimal.

3.1.4 Model Benchmarking

The model is benchmarked by applying a correction factor (X_c) to the outside film coefficients of the desuperheating, condensing, and subcooling sections and manually adjusting the correction factor until the total heat transfer area required is equal to the total heat transfer area supplied. It is noted that the total surface area listed on the data sheet is used as opposed to the effective area. This approach results in a lower outside film correction factor (X_c) which, in turn, yields a higher condensing temperature for the analysis case.

In addition, fresh water is used as the tube-side fluid for model benchmarking because the specified heat transfer rate is consistent with fresh water at the stated flow rate and terminal temperatures. The heat transfer rate achieved with salt water for the stated flow and terminal temperatures is approximately 232,000 Btu/hr.

The mass flow rate of R22 on the shell-side is determined by a heat balance with the specified tube-side heat transfer rate.

3.2 Condenser Analysis for 80°F Service Water

As the SW supply temperature to the condenser increases, the condensing temperature and pressure also rise causing an increase in refrigerant temperature and pressure throughout the cycle. The thermostatic expansion valve (TXV) modulates in response to the increase in evaporator pressure in order to maintain the amount of refrigerant superheat at the compressor suction. The compressor suction conditions determine the mass flow of refrigerant through the system since the compressor is a constant volume machine. The TXV continues to modulate until a new equilibrium point is reached. It is this point that determines the condenser conditions at steady-state.

Therefore, the conditions at the compressor suction must be known in order to determine how the condensing temperature responds to higher tube-side temperatures. The benchmarked condenser model is modified to include an analysis of the compressor. The compressor analysis consists of calculating the mass flow rate and compressor discharge temperature based on the refrigerant conditions at the compressor suction as discussed in Section 6.6.

The compressor suction conditions are determined by the evaporator pressure/temperature as well as the amount of superheat. The compressor discharge temperature is back-calculated from the suction temperature, mass flow rate, total energy input to the refrigerant (i.e., the power consumption of the compressor motor), and the average specific heat of the refrigerant. Since the discharge temperature is not initially known, manual iteration is required until the change in calculated outlet temperature produces no further change in the corresponding specific heat.

For the analysis cases, the evaporator pressure and condensing temperature are iterated manually until the required heat load is met with the available heat transfer area. Although both parameters affect the entire process, changes in evaporator pressure have a larger impact on the heat transfer rate (due to changes in mass flow) and changes to the condensing temperature have a larger impact on the required heat transfer area (due to changes in outside film coefficient in the condensing section of X-169A/B). There is one unique combination of evaporator pressure and condensing temperature which minimizes the error in heat transfer rate (i.e., difference between calculated capacity and required heat transfer rate) and heat transfer surface area (i.e., difference between calculated heat transfer surface area and supplied heat transfer surface area).

Note on the use of Mathcad:

Mathcad will perform unit conversions automatically if the user specifies the desired units in the calculation of the various parameters. For instance, it is desired that heat transfer surface areas be calculated in units of ft^2; however, the parameters used to calculate these areas are often given in units of inches. Mathcad will automatically convert the units from in^2 to ft^2 if the user specifies that the output is to be given in ft^2. This built-in feature of Mathcad is used throughout the calculation.

4.0 DESIGN INPUTS

4.1 Physical configuration data for Vital Chiller Condensers (X-169A/B) are provided in Reference 9.4 as summarized in Table 1.

Table 1: Physical Configuration Data

Parameter	Value
Shell-Side Fluid	R22
Shell-Side Inlet Temperature	160°F
Condensing Temperature	105°F
Shell-Side Outlet Temperature	102°F
Total Area	110.5 ft^2
Effective Area	102.9 ft^2
Shell OD	8.625 in
Shell Thickness	0.322 in
Tube-Side Fluid	Salt Water(1)
Tube-Side Flow Rate	41 gpm
Tube-Side Inlet Temperature	75°F
Tube-Side Outlet Temperature	86.6°F
Tube Material	90/10 Cu-Ni
Number of Tubes	40
Number of Tube Passes	4
Fin Pitch	26 fpi
Tube Length	54 in(2)
Tube Pitch	15/16 in TRI
Shell-Side Fouling	0.0005 hr-ft^2-°F/Btu
Tube-Side Fouling	0.001 hr-ft^2-°F/Btu
Service Heat Transfer Rate	97.3 Btu/hr-ft^2-°F
Clean Heat Transfer Rate	180.3 Btu/hr-ft^2-°F
Design Heat Transfer Rate	237,600 Btu/hr

Table 1 Notes
(1) The working fluid is salt water; however, the specified heat transfer is based on fresh water properties. See Section 3.1.4.
(2) The tube length specified on the vendor data sheet is a total tube length including the thickness of the tube sheets. The effective tube length is the total tube length less the thickness of the tube sheets.

4.2 By inspection of the condenser outline drawing provided in Reference 9.3 (page A2), it is apparent that the condenser is a TEMA J shell with fixed tube sheets. The tube support plate acts as a single baffle located at the approximate midpoint of the shell.

4.3 The overall shell length is 54-inches (Reference 9.3, page A2).

4.4 The tube sheets are 1-inch thick (Reference 9.3, page A3).

4.5 The outer tube limit diameter is 7.593-inches (Reference 9.3, page A3).

4.6 The thermal conductivity of 90/10 Cu-Ni is 26 Btu/hr-ft-°F (Reference 9.5).

4.7 The design heat transfer capacity of cooling coils X-84A/B is 162,000 Btu/hr (Reference 9.13). This value represents a conservative coil load for the analysis herein as the actual load on the coil during accident conditions is 138,140 Btu/hr (References 9.2 and 9.6).

4.8 The maximum power consumption for the compressor motor is 24.06 kW when operating at Emergency Diesel Generator (EDG) over-frequency conditions (Reference 9.7). Since the compressor is semi-hermetic (Reference 9.13), the total motor power consumption will be transferred to the refrigerant.

4.9 The high pressure cutout set-point for the compressor is 275 psig (Reference 9.13).

4.10 The thermostatic expansion valve (TXV) is set to maintain 10°F ± 2°F of superheat leaving the evaporator (Reference 9.13). For the purpose of this analysis, a constant 10°F of superheat is used.

4.11 Salt water properties are calculated using the correlations developed in Reference 9.8.

4.12 Fresh water properties are calculated using the correlations developed in Reference 9.9. Fresh water is used as the tube-side fluid for model benchmarking only as discussed in Section 3.1.4.

4.13 Properties of R22 at saturation conditions are calculated using the correlations developed in Reference 9.10.

4.14 The entropy and enthalpy of superheated vapor R22 are calculated using the correlations developed in Reference 9.10.

4.15 The density, specific heat, thermal conductivity, and viscosity of superheated R22 are from Reference 9.11 using the ASHRAE reference state of 0 Btu/lbm for enthalpy and 0 Btu/lbm-°F for entropy for saturated liquid at -40°F (Reference 9.12). Use of Reference 9.11 is required because Reference 9.10 does not contain sufficient data to develop correlations for these properties. The reference state for both References 9.10 and 9.11 are consistent.

Additional information regarding Reference 9.11 as well as all output from Reference 9.11 used herein is included in Attachment E.

4.16 Compressor data are provided in Reference 9.13 as summarized in Table 2.

Table 2: Compressor Data

Parameter	Value
Number of Cylinders	3
Cylinder Bore	2-11/16 in
Cylinder Stroke	2-1/4 in
Rotating Speed	1,750 rpm

4.17 The normal operating range for the compressor suction pressure (evaporator pressure) is 55 psig to 75 psig (69.7 psia to 89.7 psia) and the normal operating range for the compressor discharge pressure (condensing pressure) is 170 psig to 260 psig (184.7 psia to 274.7 psia) (Reference 9.13).

5.0 ASSUMPTIONS

5.1 It is assumed that the condenser tubes are Wolverine Type S/T Trufin low finned tubes. The technical data contained in Reference 9.14 is consistent with that presented for the tubes in Reference 9.3 (page A4) with the exception of the surface area per foot of tube length. The discrepancy contained on page A4 of the Reference 9.3 calculation is addressed on page 4 of same. The physical characteristics of Type S/T Trufin tubes are provided in Reference 9.14 as summarized in Table 3.

Table 3: Tube Physical Characteristics

Parameter	Value
Fin Height	0.057 in
Root Diameter	0.625 in
Tube Wall Thickness (finned portion)	0.035 in
Fin Thickness	0.012 in
Surface Area per Foot of Length	0.640 ft^2/ft
Area Ratio	4.38

5.2 It is assumed that the spacing between fins is 0.026 in based on information contained in Reference 9.14 for Type S/T Trufin tubes with 26 fpi. This assumption is verified by the fact that the calculated heat transfer areas are in close agreement with Wolverine data for the subject tubes. The calculated area ratio and heat transfer area per linear foot of tube length can be found on page 6 of Attachment B and the corresponding Wolverine data can be found on page 8 of Attachment G.

5.3 It is assumed that a constant 3°F of subcooling is provided for the analysis herein. Heat transfer occurring in the subcooling section of the condensers is minimal compared to the condensing and desuperheating heat transfer; therefore, the results of this calculation are not expected to be very sensitive to this assumption. In addition, other conservative approaches taken in the calculation will compensate for any affect this assumption may have on the results.

5.4 It is assumed that the compressor is fully loaded for the analysis contained herein. This assumption is considered reasonable as the analyzed heat load on condensers X-169A/B is greater than the data sheet heat transfer rate.

5.5 It is assumed the fin resistance is constant over the range of conditions evaluated herein based on the discussion in Reference 9.14. This assumption is applied to the calculation of the overall heat transfer coefficients for each section of the condenser and is verified to be appropriate in each instance by comparison to Figure 1.52b of Reference 9.14.

5.6 It is assumed that the average outside film coefficient for the tube bank in the condensing section is the same as for one tube based on the guidance in Reference 9.14 (Section 3, page 29).

5.7 It is assumed that the correction factor F_2 in Equation 34 is equal to 0.85 based on Figure 2.46 of Reference 9.15. The correction factor is used in the calculation of the outside film coefficient in the desuperheating and subcooling sections of the condensers and is a function of the number of tube rows for each section. Each of these sections is small in terms of heat transfer area compared to the total heat transfer area and, therefore, the number of tube rows is also small resulting in a lower correction factor. The outside film coefficients for these sections are small and the effects of this assumption on the analysis are negligible.

5.8 It is assumed that the temperature at the outside of the tube wall in the desuperheating section is the same as the bulk average temperature of the fluid. This assumption causes the ratio of Prandtl number at the bulk fluid temperature to the Prandtl number at the wall temperature in Equation 34 to be equal to unity. This assumption is reasonable because small variations in temperature (i.e., 10°F or less) at constant pressure result in negligible changes in the Prandtl number. Therefore, this assumption has a negligible effect on the results.

5.9 It is assumed that the design fouling for condensers X-169A/B is 0.0005 hr-ft^2-°F/Btu on the shell-side and 0.0005 hr-ft^2-°F/Btu on the tube-side. The shell-side fouling is based on the data sheet fouling resistance from Reference 9.4. The tube-side fouling resistance is based on the guidance from Reference 9.19 for seawater service below 125°F. This assumption is consistent with the Reference 9.3 analysis of X-169A/B using 75°F SW (see Attachment F, Case 3 of Reference 9.3).

5.10 It is assumed that the minimum acceptable flow rate to X-169A/B is 26.9 gpm (approximately 13,427 lbm/hr). This assumption is based on the Reference 9.3 analysis of X-169A/B using 75°F SW (see Attachment F, Case 3 of Reference 9.3).

5.11 It is assumed that the tube plugging limit for condensers X-169A/B is 10% or 4 tubes plugged. This assumption is based on the Reference 9.3 analysis of X-169A/B using 75°F SW (see Attachment F, Case 3 of Reference 9.3).

6.0 ANALYSIS

6.1 Heat Transfer Rate

The heat transfer process consists of cooling water flowing through the condenser tubes removing heat from the refrigerant flowing through the condenser shell. The balancing of the following three heat transfer processes is represented as follows:

$$Q_{sw} = \dot{m}_{sw}\, c_{p.sw}\, (T_{sw.o} - T_{sw.i}) \qquad \text{Equation 1}$$

$$Q_{r22} = \dot{m}_{r22}\left[\left(h_{r22.sh} - h_{r22.g}\right) + \left(h_{r22.fg}\right) + \left(h_{r22.f} - h_{r22.sc}\right)\right] \qquad \text{Equation 2}$$

$$Q = U_o \cdot A_{out} \cdot F \cdot \Delta T_{LM} \qquad \text{Equation 3}$$

$$Q = Q_{sw} = Q_{r22} \qquad \text{Equation 4}$$

Where:

Q_{sw}	≡	heat transfer to SW (Btu/hr)
$\dot{m}_{sw}$	≡	mass flow rate of SW (lbm/hr)
$c_{p.sw}$	≡	specific heat of SW (Btu/lbm-°F)
$T_{sw.o}$	≡	outlet temperature of SW (°F)
$T_{sw.i}$	≡	inlet temperature of SW (°F)
Q_{r22}	≡	heat transfer from the R22 (Btu/hr)
$\dot{m}_{r22}$	≡	mass flow rate of R22 (lbm/hr)
$h_{r22.sh}$	≡	enthalpy of superheated R22 entering condenser (Btu/lbm)
$h_{r22.g}$	≡	enthalpy of saturated vapor R22 at condensing temperature (Btu/lbm)
$h_{r22.fg}$	≡	heat of vaporization of R22 at condensing temperature (Btu/lbm)
$h_{r22.f}$	≡	enthalpy of saturated liquid R22 at condensing temperature (Btu/lbm)
$h_{r22.sc}$	≡	enthalpy of subcooled R22 exiting condenser (Btu/lbm)
U_o	≡	overall heat transfer coefficient (Btu/hr-ft^2-°F)
A_{out}	≡	heat transfer surface area (ft^2)
F	≡	the log mean temperature difference correction factor
ΔT_{LM}	≡	log mean temperature difference (°F)

Tube-side mass flow rate is calculated as follows:

$$\dot{m}_{sw} = vfr_{sw} \cdot \rho_{sw} \qquad \text{Equation 5}$$

Where:

vfr_{sw}	≡	tube-side volumetric flow rate (gpm)
ρ_{sw}	≡	tube-side fluid density at inlet temperature (lbm/ft^3)

6.2 Log Mean Temperature

The log mean temperature difference term (ΔT_{LM}) included in Equation 3 is expressed in terms of the terminal temperatures (T) with shell (s), tube (t), inlet (i), and outlet (o) subscript designations as follows:

$$\Delta T_{LM} = \frac{(T_{s,i} - T_{t,o}) - (T_{s,o} - T_{t,i})}{ln\left[\frac{(T_{s,i} - T_{t,o})}{(T_{s,o} - T_{t,i})}\right]} \qquad \text{Equation 6}$$

The ΔT_{LM} represents the driving force for heat transfer between the shell and tube sides of the heat exchanger.

6.3 Log Mean Temperature Correction Factor

The log mean temperature difference correction factor (F) included in Equation 3 is a function of the shell configuration. For pure counter flow heat exchangers (including a single tube pass TEMA-E shell), the correction factor is unity. For a fully condensing heat exchanger, the correction factor is also taken as unity. For all other configurations, the correction factor is dependent on the heat exchanger effectiveness (P) and the heat capacity ratio (R) which are defined by Equation 7 and Equation 8, respectively.

$$P = \frac{T_{t,o} - T_{t,i}}{T_{s,i} - T_{t,i}} = heat\ exchanger\ effectiveness \qquad \text{Equation 7}$$

$$R = \frac{T_{s,i} - T_{s,o}}{T_{t,o} - T_{t,i}} = heat\ capacity\ ratio \qquad \text{Equation 8}$$

For convenience in writing the formulas, the term δ is defined using Equation 9 as follows:

$$\delta = \frac{R - 1}{ln\left[\frac{1 - P}{1 - P \cdot R}\right]}\ for\ R \neq 1\ or\ \frac{1 - P}{P}\ for\ R = 1 \qquad \text{Equation 9}$$

An iterative calculation is required to determine the log mean temperature correction factor for a TEMA-J shell with an even number of tube passes (four tube passes in the Vital Chiller Condensers) using the following equations:

$$\phi = e^{(F \cdot \delta)^{-1}} \qquad \text{Equation 10}$$

$$P = \left[\frac{R \cdot \phi^R}{\phi^R - 1} + \frac{\phi}{\phi - 1} - \frac{1}{ln(\phi)}\right]^{-1} \qquad \text{Equation 11}$$

The correction factor (F) in Equation 10 is adjusted until the heat exchanger effectiveness (P) calculated in Equation 11 is equal to the heat exchanger effectiveness (P) defined by Equation 7.

6.4 Heat Exchanger Areas

The heat transfer area term (A_{out}) included in Equation 3 represents the surface area available for heat transfer and is a function of the heat exchanger geometry. The total surface area for Type S/T Trufin tubes consists of the surface area of the fins (A_{fin}) and the bare tube between the fins (A_{root}).

The heat transfer surface area of the bare tube between the fins (A_{root}) is calculated using the following formula from Reference 9.14 (Equation 2.8, adjusted for the number of tubes):

$$A_{root} = \pi \cdot d_r \cdot L_T \cdot N_f \cdot fin_s \cdot N_T \qquad \text{Equation 12}$$

Where:

d_r	≡	root diameter of finned tube (in)
L_T	≡	effective tube length (in)
N_f	≡	fin pitch (fpi)
fin_s	≡	fin spacing (in)
N_T	≡	number of tubes

The length of the shell is 54-inches and the tube sheets are 1-inch thick (Design Inputs 4.3 and 4.4, respectively). Therefore, the resulting effective tube length is 52-inches.

The heat transfer surface area of the fins (A_{fin}) is calculated using the following formula from Reference 9.14 (Equation 2.9, adjusted for the number of tubes):

$$A_{fin} = \left[\frac{\pi}{2} \cdot L_T \cdot \left(d_f^2 - d_r^2\right) \cdot N_f\right] \cdot N_T \qquad \text{Equation 13}$$

The diameter over the fins (d_f) is calculated as follows:

$$d_f = d_r + 2 \cdot fin_H \qquad \text{Equation 14}$$

Where:

fin_H	≡	fin height (in)

The resulting heat transfer surface area (A_{out}) is:

$$A_{out} = A_{root} + A_{fin} \qquad \text{Equation 15}$$

The tube inside surface area (A_{in}) for the tubes is calculated as follows:

$$A_{in} = \pi \cdot d_{in} \cdot L_T \cdot N_T \qquad \text{Equation 16}$$

The inside tube diameter (d_{in}) is found from the specified outside tube diameter and the specified tube wall thickness:

$$d_{in} = d_r - 2 \cdot t_w \qquad \text{Equation 17}$$

Where:

t_w ≡ tube wall thickness for the finned section (in)

The minimum shell-side free flow area (A_{min}) is calculated using the following formula from Reference 9.14 (Equation 2.43):

$$A_{min} = L_{bc} \cdot \left[\frac{d_{s.in} - d_{otl} + \left(\frac{d_{otl} - d_f}{L_{tp}}\right) \cdot}{\left[(L_{tp} - d_f) + 2 \cdot fin_H \left(\frac{fin_s}{fin_s + fin_Y}\right)\right]} \right] \qquad \text{Equation 18}$$

Where:

L_{bc} ≡ face-to-face baffle spacing (in)
$d_{s,in}$ ≡ inside diameter of shell (in)
d_{otl} ≡ shell outer tube limit (diameter, in)
L_{tp} ≡ tube pitch (in)
fin_s ≡ fin spacing (in)
fin_Y ≡ fin thickness (in)

The face-to-face baffle spacing is taken to be approximately 1/2 of the distance between the tube sheets based on the shell geometry shown on page A2 of Reference 9.3 (see also Design Inputs 4.3 and 4.4). This resulting baffle spacing is:

$$L_{bc} = \frac{1}{2} \cdot (54 - 2)in = 26\ in \qquad \text{Equation 19}$$

The shell inside diameter ($d_{s.in}$) is calculated as follows:

$$d_{s,in} = d_{shell} - 2 \cdot t_s \qquad \text{Equation 20}$$

Where:

d_{shell} ≡ outside diameter of shell (in)

t_s ≡ shell thickness (in)

The cross-sectional flow area of the tube bank (A_t) is needed to define tube-side fluid velocities and is calculated as follows:

$$A_t = \pi \cdot \frac{d_{in}^2}{4} \cdot N_t \qquad \text{Equation 21}$$

6.5 Overall Heat Transfer Coefficient

The overall heat transfer coefficient (U_o) included in Equation 3 represents the inverse of the sum of the thermal resistances in the one-dimensional heat transfer across the tube wall. Taking the outside surface of the tubes as the reference area included in Equation 3 results in the following simplified expression for U_o:

$$\frac{1}{U_o} = \sum R = \frac{1}{h_i}\left(\frac{A_{out}}{A_{in}}\right) + R_{f,i}\left(\frac{A_{out}}{A_{in}}\right) + R_w + R_{f,o} + R_{fin} + \frac{1}{X_c h_o} \qquad \text{Equation 22}$$

Where:

R	≡	net thermal resistance (hr-ft^2-°F/Btu)
h_i	≡	inside film coefficient (Btu/hr-ft^2-°F)
$R_{f,i}$	≡	thermal resistance due to inside fouling film (hr-ft^2-°F/Btu)
R_w	≡	thermal resistance due to tube wall (hr-ft^2-°F/Btu)
$R_{f,o}$	≡	thermal resistance due to outside fouling film (hr-ft^2-°F/Btu)
R_{fin}	≡	thermal resistance due to fin (hr-ft^2-°F/Btu)
X_c	≡	outside film correction factor
h_o	≡	outside film coefficient (Btu/hr-ft^2-°F)

Each of the thermal resistances that make up the overall heat transfer coefficient can be derived separately for a given set of heat exchanger conditions as discussed in the following sections.

The outside film correction factor (X_c) is used to benchmark the results of the condenser analysis to the data sheet specified performance as discussed in Section 3.1.4.

6.5.1 Inside Film Coefficient

The inside film coefficient (h_i) is calculated using Equation 23 for fully developed turbulent flow where $Re_{sw} \geq 10{,}000$ (Reference 9.16, Equation 8.59), Equation 24 for fully developed laminar flow where $Re_{sw} \leq 2{,}100$ (Reference 9.16, Equation 8.55), or Equation 25 for transition flow where $2{,}100 < Re_{sw} < 10{,}000$ (Reference 9.17).

$$h_i = 0.027 \cdot \frac{k_{sw}}{d_{in}} \cdot Re_{sw}^{0.8} \cdot Pr_{sw}^{1/3} \cdot \left(\frac{\mu_{sw}}{\mu_{s,i}}\right)^{0.14} \qquad \text{Equation 23}$$

$$h_i = 1.86 \cdot \frac{k_{sw}}{d_{in}} \cdot Re_{sw}^{1/3} \cdot Pr_{sw}^{1/3} \cdot \left(\frac{\mu_{sw}}{\mu_{s,i}}\right)^{0.14} \cdot \left(\frac{L_t}{d_{in}}\right)^{-1/3}$$ Equation 24

$$h_i = \frac{k_{sw}}{d_{in}} \cdot Pr_{sw}^{1/3} \cdot \left(\frac{\mu_{sw}}{\mu_{s,i}}\right)^{0.14} \cdot \left[0.027 \cdot Re_{sw}^{0.8} - \left[\varphi - \chi \cdot \left(\frac{L_t}{d_{in}}\right)^{-1/3}\right] \cdot \left(\frac{10^4 - Re_{sw}}{7900}\right)^2 \cdot e^{\left[-\varphi \cdot \frac{(10^4 - Re_{sw})}{7900}\right]} \right]$$ Equation 25

Where:

k_{sw}	≡	tube-side fluid conductivity at bulk average temperature (Btu/hr-ft-°F)
Re_{sw}	≡	tube-side Reynolds number
Pr_{sw}	≡	tube-side Prandtl number
μ_{sw}	≡	tube-side fluid viscosity at bulk average temperature (lbm/ft-hr)
$\mu_{s,i}$	≡	tube-side fluid viscosity at inside tube wall temperature (lbm/ft-hr)
φ	≡	38.25734951 (coefficient from Reference 9.17)
χ	≡	74.21557464 (coefficient from Reference 9.17)
ψ	≡	1.1365 (coefficient from Reference 9.17)

The tube-side Reynolds number is defined as:

$$Re_{sw} = \frac{\rho_{sw} \cdot V_{sw} \cdot d_{in}}{\mu_{sw}}$$ Equation 26

Where:

ρ_{sw}	≡	tube-side fluid density at bulk average temperature (lbm/ft^3)
V_{sw}	≡	tube-side velocity (ft/s)

Tube-side velocity is calculated as follows:

$$V_{sw} = \frac{\dot{m}_{sw} \cdot N_p}{\rho_{sw} \cdot A_t}$$ Equation 27

Where:

N_p	≡	number of tube passes

The Prandtl Number (Pr_{sw}) is defined directly in terms of input fluid properties:

$$Pr_{sw} = \frac{c_{p,sw} \cdot \mu_{sw}}{k_{sw}}$$ Equation 28

Where:

$c_{p.sw}$ ≡ tube-side fluid specific heat at bulk average temperature (Btu/lbm-°F)

The correction for variable fluid properties across the convection film is accounted for in the ratio of viscosities in Equation 23 through Equation 25. Calculating this ratio requires determination of the temperature of the fluid that is in contact with the tube wall ($T_{w.i}$). An initial guess of this temperature allows calculation of the inside film convection coefficient. The inside tube wall temperature is then re-calculated from the average tube-side fluid temperature and the first calculation of the inside film heat transfer coefficient using the following formula (Reference 9.14, Equation 2.22):

$$T_{w.i} = T_{sw.avg} + \frac{U_{dsh} \cdot A_{dsh}}{h_{i.dsh} \cdot A_i} \cdot \left(T_r - T_{sw.avg}\right) \qquad \text{Equation 29}$$

Where:

$T_{sw.avg}$ ≡ average tube-side temperature for the applicable section (°F)
U_{dsh} ≡ overall heat transfer coefficient for the applicable section (Btu/hr-ft^2-°F)
A_{dsh} ≡ outside heat transfer area for the applicable section (ft^2)
$h_{i.dsh}$ ≡ inside film coefficient for the applicable section (Btu/hr-ft^2-°F)
A_i ≡ inside heat transfer area for the applicable section (ft^2)
T_r ≡ average shell-side temperature for the applicable section (°F)

The subscripts in Equation 29 apply to the calculation of the inside tube wall temperature for the desuperheating section ("dsh"). However, the formula also applies to the condensing and subcooling sections of the condenser.

With the wall temperature calculated, the viscosity of the fluid at the wall temperature is calculated and the inside film coefficient is re-calculated using Equation 23, Equation 24, or Equation 25 depending on the flow regime. The wall temperature is then re-calculated using Equation 29. This process is iterated until a successive re-calculation of the wall temperature produces no significant change.

6.5.2 Fouling Resistance

Two fouling resistance terms (R_{fi} and R_{fo}) are included in Equation 22. A simplification of the contribution of tube-side and shell-side fouling resistances is to combine them into an overall fouling resistance (R_f). Per industry convention, the overall fouling is referenced to the outside tube surface area (A_{out}) as follows:

$$R_f = R_{f.i} \cdot \left(\frac{A_{out}}{A_{in}}\right) + R_{f.o} \qquad \text{Equation 30}$$

6.5.3 Wall Resistance

The term for the thermal resistance of the tube wall (R_w) included in Equation 22 is a function of the tube geometry and the conductivity of the tube material and is calculated using the following formula (Reference 9.14, Equations 1.44 and 1.45 adjusted for the number of tubes):

$$R_w = \frac{t_w \cdot A_{out}}{k_w \cdot \pi \cdot (d_r - t_w) \cdot N_t \cdot L_t} \qquad \text{Equation 31}$$

Where:

k_w ≡ thermal conductivity of tube wall material (Btu/hr-ft-°F)

6.5.4 Fin Resistance

Per Assumption 5.5, the fin resistance is assumed to be constant over the range of conditions evaluated in this analysis. By inspection of Figure 1.52b of 9.14, the fin resistance (R_{fin}) for 90/10 Cu-Ni tubes is 0.0006 hr-ft^2-°F/Btu for values of $(1/h_o + R_{f,o})^{-1}$ approaching 1,000. The term $(1/h_o + R_{f,o})^{-1}$ is calculated for each section and compared to Figure 1.52b to verify that the assumption is valid.

6.5.5 Outside Film Coefficient

The shell-side film coefficient term (h_o) included in Equation 22 is calculated using the correlations for flow over a bank of tubes. Two expressions are involved to differentiate between sections that involve sensible heat transfer (i.e., desuperheating and subcooling sections) and the condensing section which involves latent heat transfer.

<u>Sensible Heat Transfer</u>

When there is no phase change (desuperheating the vapor and subcooling the condensed liquid), the outside film coefficient is a function of the fluid Reynolds number, tube geometry, and the vapor thermal conductivity and Prandtl number.

The maximum outside Reynolds Number is calculated using the following formula:

$$Re_{max} = \frac{\frac{mfr_{r22}}{2} \cdot d_r}{\mu_v \cdot A_{min}} \qquad \text{Equation 32}$$

Where:

mfr_{r22} ≡ shell-side mass flow rate (lbm/hr)

μ_v ≡ shell-side fluid viscosity at bulk average temperature (lbm/ft-hr)

The Prandtl Number (Pr_v) is defined directly in terms of input fluid properties:

$$Pr_v = \frac{c_{p.v} \cdot \mu_v}{k_v} \qquad \text{Equation 33}$$

Where:

$c_{p.v}$ ≡ shell-side fluid specific heat at bulk average temperature (Btu/lbm-°F)

k_v ≡ shell-side fluid thermal conductivity at bulk average temperature (Btu/hr-ft-°F)

The outside film coefficient (h_o) is calculated using the following formula for low finned tubes (Reference 9.15, Equation 2.240):

$$h_o = \left(\frac{k_v}{d_r}\right) \cdot 0.183 \cdot Re_{max}^{0.7} \cdot \left(\frac{fin_s}{fin_H}\right)^{0.36} \cdot \left(\frac{L_{tp}}{d_f}\right)^{0.06} \cdot \left(\frac{fin_H}{d_f}\right)^{0.11} \cdot Pr_v^{0.36} \cdot \left(\frac{Pr_v}{Pr_{v.o}}\right)^{0.26} \cdot F_2 \cdot F_3 \qquad \text{Equation 34}$$

Where:

$Pr_{v.o}$ ≡ shell-side fluid Prandtl number at outside tube wall temperature

F_2 ≡ correction factor for number of tube rows

F_3 ≡ correction factor for tube layout angle

Per Assumption 5.7, F_2 is equal to 0.85 and, per Reference 9.15, F_3 is equal to unity for heat exchangers with a triangular tube pitch.

For the subcooling section, the correction for variable fluid properties across the convection film is accounted for in the ratio of Prandtl numbers in Equation 34. Calculating this ratio requires determination of the temperature of the fluid that is in contact with the tube wall ($T_{w.o}$). An initial guess of this temperature allows calculation of the outside film convection coefficient. The outside tube wall temperature is then re-calculated from the average shell-side fluid temperature and the first calculation of the outside film heat transfer coefficient using the following formula (Reference 9.15, Equation 2.242):

$$T_{w.o} = T_r - \frac{Q_{dsh}}{\left(X_c \cdot h_{o.dsh} + \frac{1}{R_{fin}}\right) \cdot A_{dsh}} \qquad \text{Equation 35}$$

Where:

Q_{dsh} ≡ heat transfer rate for the applicable section (Btu/hr)

A_{dsh} ≡ outside heat transfer area for the applicable section (ft^2)

With the wall temperature calculated, the Prandtl number of the fluid at the outside wall temperature is calculated and the outside film coefficient is re-calculated using Equation 34. The wall temperature is then re-calculated using Equation 35. This process is iterated until a successive re-calculation of the wall temperature produces no significant change.

Because Reference 9.10 does not provide sufficient property data to develop the necessary correlations for superheated vapor Prandtl number, the Prandtl number of the refrigerant at the outside wall temperature in the desuperheating section is held constant at the value for the bulk average fluid temperature according to Assumption 5.8. This assumption simplifies the analysis for the desuperheating section and does affect the results.

The subscripts "v" and "dsh" in the formulas above apply to the vapor in the desuperheating section of the condenser. The same formulas apply to the subcooling section of the condenser, except the refrigerant properties are evaluated at liquid conditions which are annotated with the subscript "l" and are referenced to subcooling specific parameters which are annotated with the subscript "sc".

Latent Heat Transfer

The average film coefficient for filmwise condensation over a vertical tier of Trufin tubes is calculated using the following formula (Reference 9.14, Equation 3.79):

$$h_o = 0.725 \cdot \left[\frac{k_l^3 \cdot \rho_l \cdot (\rho_l - \rho_v) \cdot g \cdot h_{r22.fg}}{\mu_l \cdot (T_{r22.cond} - T_{w.o}) \cdot d_r}\right]^{1/4} \qquad \text{Equation 36}$$

Where:

k_l	≡	shell-side fluid conductivity for saturated liquid at condensing temperature (Btu/hr-ft-°F)
ρ_l	≡	shell-side fluid density for saturated liquid at condensing temperature (lbm/ft^3)
ρ_v	≡	shell-side fluid density for saturated vapor at condensing temperature (lbm/ft^3)
g	≡	acceleration due to gravity (32.2 ft/s^2)
μ_l	≡	shell-side fluid viscosity for saturated liquid at the condensing temperature (lbm/ft-hr)
$T_{r22.cond}$	≡	condensing temperature (°F)

The process of iterating the outside wall temperature and outside film coefficient in the condensing section is identical to the process used in the subcooling section.

6.6 Compressor Analysis

As the SW temperature increases, there is a corresponding rise in condensing temperature in response to the hotter tube-side conditions. An initial estimate of the condensing temperature (T_{sat}) is made from the following formula (Reference 9.14, Section 3, page 25) using the heat flux value in the condensing section from the benchmark results:

$$T_{sat} = \frac{Q_{cond}}{A_{cond} \cdot U_{cond}} + T_{sw.avg} \qquad \text{Equation 37}$$

Where:

Q_{cond}	≡	heat transfer rate occurring in condensing section (Btu/hr)
A_{cond}	≡	heat transfer surface in condensing section (ft^2)
U_{cond}	≡	overall heat transfer coefficient for the condensing section (Btu/hr-ft^2-°F)
$T_{sw.avg}$	≡	average SW temperature in the condensing section (°F)

The average SW temperature used in Equation 37 is back-calculated from the required heat load, the flow rate, and fluid properties (density and specific heat) at the inlet temperature.

As temperatures and pressures through the refrigerant cycle increase, the TXV modulates in order to maintain the desired superheat at the compressor suction. This, in turn, results in a change in mass flow rate through the refrigerant cycle. Since the Vital Chiller Compressor is a reciprocating compressor, it provides a constant volumetric flow rate of gas (vfr_{comp}) which is calculated using the following equation (Reference 9.18, Equation 11.5):

$$vfr_{comp} = \frac{\pi \cdot D_{cy}^2 \cdot L_{st} \cdot N_{cy} \cdot n}{4} \qquad \text{Equation 38}$$

Where:

D_{cy}	≡	cylinder bore (in)
L_{st}	≡	cylinder stroke (in)
N_{cy}	≡	number of cylinders
n	≡	rotating speed of the compressor (rpm)

The corresponding mass flow rate (mfr_{comp}) through the compressor is calculated using the following formula (Reference 9.18, Equation 9.72):

$$mfr_{comp} = vfr_{comp} \cdot \eta_v \cdot \rho_{suct} \qquad \text{Equation 39}$$

Where:

η_v	≡	volumetric efficiency

ρ_{suct} ≡ refrigerant density at suction conditions (lbm/ft^3)

To obtain the suction conditions, an initial estimate is made for the evaporator pressure (P_{evap}) and temperature (T_{suct}). Since the TXV is set to maintain 10°F of superheat at the compressor suction, the suction temperature is the saturation temperature corresponding to the evaporator pressure +10°F. The refrigerant vapor density at compressor suction (ρ_{suct}) is determined at these conditions by using the Reference 9.11 refrigerant properties database.

The volumetric efficiency is obtained from Figure 11.18a of Reference 9.18 which is a plot of volumetric efficiency versus compression ratio of a typical reciprocating compressor. By inspection of the figure, it is clear that the curve is linear. The following equation for a straight line is developed from the data points (1, 0.93) and (3, 0.82):

$$\eta_v = -0.055 \cdot P_{cr} + 0.9850 \qquad \text{Equation 40}$$

Where:

P_{cr} ≡ compression ratio (P_{cond}/P_{evap})

The condensing pressure (P_{cond}) is the corresponding saturation pressure for the condensing temperature. The compressor discharge temperature (T_{disch}) is now calculated using the following formula:

$$T_{disch} = T_{suct} + \frac{2 \cdot Q_{comp}}{mfr_{comp} \cdot (c_{p.suct} + c_{p.disch})} \qquad \text{Equation 41}$$

Where:

T_{suct} ≡ refrigerant temperature at the compressor suction (°F)
Q_{comp} ≡ power consumption of the compressor motor (Btu/hr)
$c_{p.suct}$ ≡ refrigerant specific heat at the compressor suction (Btu/lbm-°F)
$c_{p.disch}$ ≡ refrigerant specific heat at the compressor discharge (Btu/lbm-°F)

The refrigerant specific heats at the compressor suction ($c_{p.suct}$) and the compressor discharge ($c_{p.disch}$) are determined at the applicable conditions by using the Reference 9.11 refrigerant properties database. An iterative calculation is required since the specific heat at the compressor discharge is dependent on the discharge temperature. Therefore, Equation 41 is iterated with updated discharge specific heat values until the resulting change in discharge temperature results in no further change to the specific heat.

Once the compressor discharge temperature is calculated, all the required parameters to perform an evaluation of the condenser are known. At this point, the condensing temperature and evaporator pressure are iterated manually and corresponding fluid properties updated until the error in the heat load (condenser capacity less the required heat load) and the heat transfer area (heat transfer area required less heat transfer area available) are minimized. Adjusting the evaporator pressure has a larger effect on the heat load and adjusting the condensing temperature has a larger effect on the heat transfer area as discussed in Section 3.2.

The resulting condensing temperature is 114.5°F with a corresponding condensing pressure of 255.9 psia (241.2 psig). The evaporator pressure is 82 psia (67.3 psig). These pressures are within the normal operating range of the Vital Chillers.

7.0 CONCLUSION

A Mathcad model of the Vital Chiller Condensers (X-169A/B) is developed for the purpose of evaluating condenser performance with a Service Water supply temperature of 80°F. The model is benchmarked to match the heat transfer capacity specified on the vendor data sheet. The benchmarked model predicts condenser performance within 0.00022% based on heat transfer, -0.0003% based on overall surface area, and -0.013% based on SW inlet temperature. A printout of the Mathcad file is included in Attachment B. Based on a heat balance with the SW heat transfer rate, the mass flow rate of R22 through the shell for the data sheet conditions is 2,822.14 lbm/hr.

The benchmarked model is used to determine condensing conditions (i.e., temperature and pressure) with a SW supply temperature of 80°F. The resulting condensing temperature is 114.5°F at limiting conditions of heat load (244,096 Btu/hr), SW flow (26.9 gpm), tube plugging (10%), and fouling (0.0005 hr-ft^2-°F/Btu on the shell-side and 0.0005 hr-ft^2-°F/Btu on the tube-side). The corresponding condensing and evaporating pressures are 255.9 psia (241.2 psig) and 82 psia (67.3 psig), respectively. Both of these pressures are within the normal operating ranges of 170 psig to 260 psig (condensing pressure) and 55 psig to 75 psig (evaporating pressure). The resulting condensing pressure is below the compressor high pressure cutout of 260 psig. Therefore, it is concluded that the Vital Chiller Condensers (X-169A/B) have sufficient capacity to perform their intended function with a SW supply temperature of 80°F at the limiting conditions of minimum acceptable SW flow, design heat load, maximum tube plugging, and design fouling. A printout of the Mathcad file is included in Attachment C.

8.0 PRECAUTIONS AND LIMITATIONS

This calculation has been prepared in support of DC MP2-12-01205 (Millstone Unit 2: Increase in Ultimate Heat Sink Temperature Limit from 75°F to 80°F). Plant operation at an UHS temperature of 80°F is not permitted until such time as the associated UHS technical specification has been revised. However, the results of this calculation are bounding of plant operation at UHS temperatures less than 80°F.

9.0 REFERENCES

9.1 Drawing 25203-26027, Sheet 2 of 4; Millstone Nuclear Power Station Unit 2 Piping and Instrumentation Diagram – Turbine Building Intake Structure, Warehouse and DC Rooms Chilled Water System; Revision 49

9.2 Calculation 03-ENG-04035M2; MP2 Service Water System Design Basis Summary Calculation; Revision 00 (including CCNs 01 through 04)

9.3 Calculation 99-ENG-02819-M2; Thermal Performance Evaluation of Millstone Unit 2 – DC Switchgear Room Cooler; Revision 02

9.4 Material Receipt Inspection Report (MRIR) 287-229; approved 6/22/88 (excerpt provided in Attachment D)

9.5 The Century Heat Exchanger Tube Manual; Century Brass, Waterbury, CT, 1977

9.6 Calculation 92-FFP-00830-ES; MP2 East DC Vital Swgr Room Heat Gains; Revision 03 (including CCN 01)

9.7 Calculation PA-079-126-01027E2; MP2 EDG Loading Calculation; Revision 03 (including Addenda A and B)

9.8 Calculation 93-049; Fluid Properties – Salt Water – Range 32°F to 320°F – Salinity 35 g/kg; Revision A

9.9 Calculation 93-048; Fluid Properties – Fresh Water – Range 32°F to 600°F; Revision A

9.10 Calculation 96-009; Refrigerant Chiller Properties; Revision C

9.11 National Institute of Standards and Technology Standard Reference Database 23; Reference Fluid Thermodynamic and Transport Properties; Version 9.0 (see Attachment E)

9.12 2009 ASHRAE Handbook; Fundamentals

9.13 Vendor Technical Manual (VTM) 25203-741-004; Installation, Operation, and Maintenance of DC Switchgear Rooms Air Conditioning Units; Revision 9 (excerpt provided in Attachment F)

9.14 Wolverine Engineering Data Book II; Wolverine Division of UOP, Inc., 1984 (excerpt provided in Attachment G)

9.15 Process Heat Transfer, G. F. Hewitt, G. L. Shires, and T. R. Bott, CRC Press, 1994

9.16 Fundamentals of Heat and Mass Transfer, 2nd Ed., F. P. Incropera and D. P. Dewitt, John Wiley and Sons, 1985

9.17 Calculation 96-016; Calculating the Tube Inside Heat Transfer Coefficient, h_i, for Reynolds Numbers Corresponding to the Transition from Laminar to Turbulent Flow for Shell-and-Tube Heat Exchangers; Revision –

9.18 Handbook of Air Conditioning and Refrigeration, 2nd Ed., S. K. Wang, McGraw-Hill, 2001 (excerpt provided in Attachment H)

9.19 Standards of the Tubular Exchanger Manufacturers Association, 7th Ed., 1988