PSEG Nuclear LLC
P.O. Box 236, Hancocks Bridge, New Jersey 08038-0236

PSEG
Nuclear LLC

NOV 1 6 2005

LR-N05-0446

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

ASME CODE RELIEF REQUEST
SALEM GENERATING STATION – UNIT 1 AND UNIT 2
DOCKET NOS. 50-272 AND 50-311
FACILITY OPERATING LICENSE NOS. DPR-70 AND DPR-75

Pursuant to 10 CFR 50.55a(a)(3)(i), PSEG Nuclear LLC (PSEG) requests relief from American Society of Mechanical Engineers (ASME) Section VIII, Division 1, UG-27. This section does not permit the use of plastic analysis; however, ASME Section VIII, Division 2 does permit the use of plastic analysis provided minimum wall thickness requirements are satisfied and any seam welds are fully radiographed.

This relief request is being submitted in an effort to align the lower design pressure of the Component Cooling (CC) heat exchanger with that of the remainder of the Salem Service Water System. Analysis has been performed which demonstrates that there is adequate safety margin of the CC Heat Exchanger under the design pressure requirements. PSEG is requesting for deviation from the normal approach of performing calculations. The proposed alternative is to use plastic analysis of the CC heat exchanger using the methodology provided in ASME Code, Section VIII Division 2, 2004 Edition, Appendix 4, Paragraph 4-136.4.

If you have any questions please contact Mr. Justin Wearne at 856-339-5081.

Sincerely,

Thomas P. Joyce

Thomas P. Joyce
Site Vice President
Salem Generating Station

Attachments

(1) Relief Request SC-RR-W03.

A047

C: Mr. S. Collins, Administrator – Region I
U. S. Nuclear Regulatory Commission
475 Allendale Road
King of Prussia, PA 19406

Mr. S. Bailey, Licensing Project Manager - Salem
U. S. Nuclear Regulatory Commission
Mail Stop 08B1
Washington, DC 20555

USNRC Senior Resident Inspector – Salem (X24)

Mr. K. Tosch, Manager IV
Bureau of Nuclear Engineering
PO Box 415
Trenton, New Jersey 08625

10 CFR 50.55a Request Number SC-RR-W03

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

Component Description

Component Cooling (CC) shell-and-tube heat exchangers (1CCE5, 2CCE5, 2CCE6)

Applicable ASME Code Edition and Addenda:

American Society of Mechanical Engineers Boiler and Pressure Vessel Code (ASME Code), Section VIII – Division 1, 1968 Edition, with No Addenda (Reference 1), Paragraph UG-27 provides the requirements for the minimum wall thickness.

Reason for Request:

Background

The current design pressure for the tube-side (head) of the heat exchangers is less than the Service Water (SW) System design pressure. The vessel design pressure is 150 psig versus a SW system design pressure of 200 psig. The SW system typically operates at a pressure under 150 psig. However, under certain system cold weather configurations, the operating pressure can increase above 150 psig. The maximum operating pressure would occur following a loss-of-offsite power (LOOP) event due to the combination of three pumps operating and decreased winter flow demand as the Containment Fan Cooler Units (CFCUs) and non-safety flow loads are automatically isolated. In this configuration, the system pressure is expected to be approximately 180 psig, with some components experiencing slightly higher pressures due to hydrostatic pressure.

In order to address this issue, an evaluation was performed of the SW side of the heat exchangers to demonstrate that the ASME Code margins are maintained at the worst-case operating conditions. The goal was to demonstrate acceptability of the SW side of the heat exchangers to 200 psig for compatibility with the system design pressure. This analysis is documented in S-C-SW-MEE-1882 (Enclosure 1). The evaluation concluded that the CC shell-and-tube heat exchangers are acceptable per the original Code to 154 psig. This pressure was limited by the minimum wall thickness requirements of the Code for the channel (i.e., the channel head minimum wall thickness criteria per Paragraph UG-27 are not met for higher pressures).

Specific Code Issues

This relief request is to permit the use of an alternative analysis that will demonstrate that the CC heat exchangers meet the intent of the ASME Code for a pressure of 191 psig. This pressure (191 psig) is high enough to satisfy all of the system pressure requirements for the CC heat exchanger.

10 CFR 50.55a Request Number SC-RR-W03

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

The standard ASME Code calculations identify three areas that do not meet the standard Code requirements at the higher SW pressure (191 psig). Specifically,

1. The existing channel head wall thickness is 0.625 inches versus a required wall thickness 0.756 inches.

2. The existing nozzle reinforcement area is 7.48 in^2 versus a required reinforcement area of 14.17 in^2.

3. The existing channel flange minimum thickness is 4.63 inches versus a required thickness of 5.05 inches, based on maintaining an allowable stress of 17.5 ksi.

Proposed Alternative and Basis for Use:

Proposed Alternative

The proposed alternative is to allow the use of plastic analysis of the CC shell-and-tube heat exchanger channel using the methodology provided in ASME Code, Section VIII - Division 2, 2004 Edition, Appendix 4, Paragraph 4-136.4.

Basis for Use

The issues related to the channel heads were evaluated using a finite element analysis (FEA). The FEA (enclosure 2) evaluated the inlet/outlet head using a three-dimensional model, which included the head, the nozzle and the flange. This permitted a single model to address all three design code compliance issues, as detailed in the previous section. The FEA included both elastic analysis and plastic analysis.

A scoping elastic analysis, using the allowable stresses provided in Section VIII - Division 1, concluded that the flanges and nozzle reinforcement are acceptable for a SW pressure of 191 psig. However, the additional support provided by the nozzles and the pass partition plate are not enough to limit the membrane stress in the channel head to less than the allowable stress. Hence, a plastic analysis was necessary to justify operation at higher pressures. Plastic analysis can support higher working pressures by accounting for such strengthening phenomenon as strain hardening, redundancies by load shedding to other locations and strengthening by changing the basic shape of the component (i.e., large deflections).

The original Code of Record (ASME Section VIII – Division 1) does not permit the use of plastic analysis. However, ASME Section VIII – Division 2 does permit the use of plastic analysis, provided the minimum wall thickness requirements are satisfied and any seam welds are fully radiographed. The CC shell-and-tube heat exchangers do not satisfy either of these conditions. The minimum wall thickness requirement is not met and the seam welds were only spot radiographed. Accordingly, the FEA results could not be used to support a formal Code evaluation of the heat exchangers. Regardless, plastic analysis was used to determine whether overall Code margins to failure (i.e., 2/3 factor of safety to plastic collapse) would be maintained.

The plastic analysis showed that the 2/3 factor of safety on plastic collapse was maintained for channel head elements subjected to internal pressure of at least 191 psig (even though the minimum wall thickness requirements are not satisfied). This was accomplished by accounting for such strengthening phenomenon as strain hardening, redundancies by load shedding and changing the basic shape of the component (i.e., large deflections). The head shell material is a 90-10 Cu-Ni material which is very ductile at 30% elongation. The flange is also constructed of a ductile material (carbon steel). The analysis approach is an adaptation of the Section VIII – Division 2 procedure.

- A 15% reduction of the stress-strain curve was made to philosophically account for the joint efficiency associated with spot radiography. This is not in accordance with the Code, but was done to maintain the overall Code approach of penalizing allowable stresses based on the level of inspections performed.

- Allowable stresses were based on Section VIII – Division 1 allowable stresses.

When plastic analysis is used, the Code also requires that fatigue and ratcheting be specifically considered, because of the potential for higher than typical strains. For fatigue, the number of pressure cycles experienced by the channel heads is very low and the fatigue is not considered significant. Ratcheting is not considered a concern because the plastic analysis included a 15% reduction factor on the yield strength to account for the joint efficiency. This reduction is analytical, but in reality, sections will not develop plastic hinges until the actual minimum yield strength is reached, which is at higher pressures. Therefore, ratcheting is not considered to occur.

Conclusion

Through the use of plastic analysis, it is demonstrated that the flanges and nozzle reinforcement are acceptable at a pressure of 191 psig, and that the inherent Code 2/3 margin to plastic collapse of the channel head (inherent in the ASME Code) is met for at least 191 psig. Based on this result, it is concluded that the heat exchanger provides margin that is consistent with the intent of the Original Code.

10 CFR 50.55a Request Number SC-RR-W03

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

Duration of Proposed Alternative

The proposed "one-time-only" relief request alternative is requested on a permanent basis for Salem Units 1 and 2.

Precedents

None

References:

1. ASME Code Section VIII – Division 1, 1968 Edition with No Addenda.

2. ASME Code, Section VIII - Division 2, 2004 Edition, Appendix 4, Paragraph 4-136.4.

Enclosures:

1. S-C-SW-MEE-1882, Salem SW Heat Exchangers—Suitability for Operation at Higher Pressures, Revision: 0, dated 1/27/05, Attachment C, MPR Calculation 0108-0309-jlh-1, "*Component Cooling Shell and Tube Heat Exchanger Service Water Pressure Rerate Evaluation – 1CCE5, 2CCE5 & 2CCE6,*" Revision 0.

2. S-C-SW-MEE-1882, Salem SW Heat Exchangers—Suitability for Operation at Higher Pressures, Revision: 0, dated 1/27/05, Attachment D, MPR Calculation 0108-0.309-jem-1, "*Component Cooling Water Shell and Tube Heat Exchanger Channel Analysis,*" Revision 0.

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

Enclosure 1

S-C-SW-MEE-1882, Salem SW Heat Exchangers - Suitability for Operation at Higher Pressures, Revision 0, **Attachment C**, MPR Calculation 0108-0309-jlh-1, "*Component Cooling Shell and Tube Heat Exchanger Service Water Pressure Rerate Evaluation – 1CCE5, 2CCE5 & 2CCE6,*" Revision 0, dated 1/27/05.

10 CFR 50.55a Request Number SC-RR-W03

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

Enclosure 2

S-C-SW-MEE-1882, Salem SW Heat Exchangers - Suitability for Operation at Higher Pressures, Revision: 0, **Attachment D**, MPR Calculation 0108-0.309-jem-1, "*Component Cooling Water Shell and Tube Heat Exchanger Channel Analysis,*" Revision dated 1/27/050.

RJH 2-2-05

Attachment C: MPR Calculation 0108-0309-jlh-1, "Component Cooling Shell and Tube Heat Exchanger Service Water Pressure Rerate Evaluation—1CCE5, 2CCE5, &2CCE6" Revision 0

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

CALCULATION TITLE PAGE

Client: Public Service Electric and Gas	Page 1 of 37
Project: Salem SW HX Re-rate	Task No. 0108-0418-0309-00
Title: Component Cooling Shell and Tube Heat Exchanger Service Water Pressure Rerate Evaluation—1CCE5, 2CCE5, & 2CCE6	Calculation No. 0108-0309-jlh-1

Preparer / Date	Checker / Date	Reviewer & Approver / Date	Rev. No.
J L Hibbard 12-23-04 J. L. Hibbard	FOR R. PARKERSON 12-23-04 R. Parkerson	12/23/04 R. B. Keating	0

QUALITY ASSURANCE DOCUMENT
This document has been prepared, checked, and reviewed/approved in accordance with the Quality Assurance requirements of 10CFR50 Appendix B, as specified in the MPR Quality Assurance Manual.

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

RECORD OF REVISIONS

Calculation No.	Prepared By	Checked By	Page: 2
0108-0309-jlh-1	JHubbard	[signature] FOR R. PARKERSON	

Revision	Affected Pages	Description
0	All	Initial Issue.

Note: The revision number found on each individual page of the calculation carries the revision level of the calculation in effect at the time that page was last revised.

MPR
MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 3
0108-0309-jlh-1	J Hubbard	[signature] FOR R. PARKERSON	Revision: 0

Table of Contents

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: JHilliard	Calculation No.: 0108-0309-jlh-1
	Checked By: [signature] FOR R. PARKERSON	Revision No.: 0 Page No.: 4

1.0 PURPOSE

This calculation is a Section VIII, Division 1, 1968 Edition (Reference 1) evaluation of the Salem Unit 1 Component Cooling (CC) Heat Exchanger Nos. 11, 21, and 22. The allowable stress for the cover bolting is from the 2001 edition of the ASME Code. The heat exchangers are evaluated for an increase in the tube side design pressure from 150 psig to 191 psig. Only the heat exchanger components that are affected by the design pressure increase are evaluated. As such, this is a partial code evaluation of the tube-side of the Component Cooling Heat Exchangers and is an addenda to the original code evaluation.

2.0 SUMMARY

Results of the code evaluation for the Component Cooling Heat Exchanger tube-side pressure increase are provided below.

Pressure Part Wall Thickness

T1 =

"Location"	"Required"	"Actual"	"Result"
""	"Thickness"	"Thickness"	""
""	"in."	"in."	""
"Tube"	0.013	0.035	"Ok"
"Channel A & B"	0.756	0.625	"Not Ok"
"Inlet & Out. Nozzle"	0.232	0.625	"Ok"
"Manway Nozzle"	0.263	0.313	"Ok"
"Channel Flange"	5.035	4.625	"Not Ok"
"Manway Flange"	1.437	2	"Ok"
"Channel A & B Cover"	3.991	6.188	"Ok"
"Manway Cover"	1.188	1.5	"Ok"
"Tubesheet"	1.966	3.5	"Ok"

Flange Bolt Tensile Stress Area

T2 =

"Location"	"Required"	"Actual"	"Result"
""	"Bolt Area"	"Bolt Area"	""
""	"in.^2"	"in.^2"	""
"Channel A & B"	30.1	30.2	"Ok"
"Manway"	5.4	16.6	"Ok"

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: JHolland

Checked By: [signature] FOR R. PARKERSON

Inlet & Outlet Nozzle Flange

$$T5 = \begin{pmatrix} \text{"Location"} & \text{"Pressure"} & \text{"Design"} & \text{"Result"} \\ \text{""} & \text{"Rating"} & \text{"Pressure"} & \text{""} \\ \text{""} & \text{"psi"} & \text{"psi"} & \text{""} \\ \text{"Inlet \& Outlet Nozzle Flange"} & 240 & 191 & \text{"Ok"} \end{pmatrix}$$

Reinforcement Area at Nozzles

$$T7 = \begin{pmatrix} \text{"Location"} & \text{"Required"} & \text{"Actual"} & \text{"Result"} \\ \text{""} & \text{"Area"} & \text{"Area"} & \text{""} \\ \text{""} & \text{"in.^2"} & \text{"in.^2"} & \text{""} \\ \text{"Inlet/Outlet Nozzle"} & 14.17 & 7.48 & \text{"Not Ok"} \\ \text{"Manway"} & 35.92 & 39.63 & \text{"Ok"} \end{pmatrix}$$

Seismic Evaluation

An evaluation of the pressure retaining components for seismic acceleration concludes that the stresses from seismic acceleration are negligible.

3.0 CALCULATION BASES

1. The joint efficiency for the tube is $E_{Tube} = 0.6$. This assumes the tube is a single sided butt weld and that no inspection was performed, which is conservative.

2. Joint efficiencies for the inlet nozzle and outlet nozzle are $E_{Inlet} = 0.8$. This conservatively assumes the weld is a single sided butt weld with spot radiography (spot radiography is specified on Reference 4).

3. The manway gasket is a full face gasket design. For the gasket seating load calculation, the gasket is assumed to have an OD equal to the bolt circle diameter minus the bolt hole diameter. This gives a smaller than actual gasket surface area, which is considered to give a reasonable load for the gasket seating load calculation.

4. The gasket area for the gasket seating load calculation is based on the gasket OD and ID; the area for the gasket seal to the pass partition plate is not included in the gasket area.

5. No credit is taken for the thickness of the 90-10 CuNi integral clad on the carbon steel manway cover and tubesheet.

6. The channel flange is evaluated as a loose type flange, i.e., no credit is taken for stiffening provided by the channel hub. This is a conservative approach that simplifies the analysis, since the flange ring and hub are different materials (carbon steel and 90-10 CuNi).

7. The manway flange thickness is evaluated with the approach in Reference 1, Paragraph UA-6(b)(2). This provides the required flange thickness for a spherically dished cover with a full face gasket. This evaluation approach is suggested in Reference 1, Paragraph UA-56, since the typical flange evaluation in Appendix A is not applicable to full face gaskets.

8. The corrosion allowance of the 90-10 CuNi materials is 0 inches based on Reference 13. A corrosion allowance of 0 inches was also used for the titanium tube.

4.0 CALCULATION

4.1 Data

Design Conditions

$P_t \equiv 191 \cdot psi$	Tube-side design pressure
$P_s \equiv 150 \cdot psi$	Shell-side design pressure; Ref. 2, Data Sheet
$T_t \equiv 200 \cdot F$	Tube-side design temperature; Ref. 2, Data Sheet

Operating Conditions

$T_{avg.salt} \equiv (90 + 99.3) \cdot F \div 2$	$T_{avg.salt} = 94.65\,F$	Average salt water temperature; Ref. 2, Data Sheet
$T_{avg.sw} \equiv (113 + 100) \cdot F \div 2$	$T_{avg.sw} = 106.5\,F$	Average service water temperature; Ref. 2, Data Sheet
$T_{avg.t} \equiv (T_{avg.salt} + T_{avg.sw}) \div 2$	$T_{avg.t} = 100.57\,F$	Average tube water temperature

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: FOR R. PARKERSON

Calculation No.: 0108-0309-jlh-1

Revision No.: 0

Page No.: 8

Materials

Component	Material	Reference
Tube	Titanium, Gr. 2	No. 2, Data Sheet
Channel A & B	SB-171 90-10 CuNi	No. 2, Data Sheet
Channel A & B Cover	SA-105, Gr. II with a 90-10 CuNi Liner	No. 2, Data Sheet
Channel A & B Flange	SA-105, Gr. II	No. 4, Grid D-2
Channel A & B Bolting	SA-193, Gr. B7	No. 4, Grid D-2
Inlet & Outlet Nozzle	SB-171 90-10 CuNi	No. 4, Grid C-2
Inlet & Outlet Nozzle Flange	SA-181, Gr. II	No. 4, Grid C-2
Manway Cylinder	SB-171 90-10 CuNi	No. 5, p. 3
Manway Cover	SA-516, Gr. 70	No. 5, p. 3
Manway Flange	SA-516, Gr. 70	No. 5, p. 3
Manway Flange Bolting	SA-193, Gr. B7	No. 5, p. 4
Tubesheet	SA-515, Gr. 70 with a 70-30 CuNi Liner	No. 2, Data Sheet
Shell	SA-515, Gr. 70	No. 4, Grid D-2
Nozzle Reinforcement Pad	SA-515, Gr. 70	No. 4, Grid C-1
Cover Gasket	1/8" thick, MONEL jacketed Asbestos	No. 5, p. 4
Manway Gasket	1/8" thick, Garlock 3400	No. 5, p. 5

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: JHubbard Checked By: [signature] FOR R. PARKERSON	Calculation No.: 0108-0309-jlh-1 Revision No.: 0 Page No.: 9

Data for the Tubes, Channels, Inlet/Outlet Nozzles, and the Manway Cylinder

$$t_a \equiv \begin{pmatrix} 0.035 \\ 0.625 \\ 0.625 \\ 5 \div 16 \end{pmatrix} \cdot in \qquad l_i := \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Actual wall thickness
-tube; Ref. 2, Data Sheet and Ref. 3, Table 9.2-3
-Channels A & B; Reference 4, Grid C-7
-Inlet/out. (N1/N2); Reference 4, Grid C-2
-Manway; Reference 5, p. 5

$$od \equiv \begin{pmatrix} 0.75 \\ 66.25 \\ 20 \\ 18.625 \end{pmatrix} \cdot in \qquad l_i = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Outside diameter
-tube; Ref. 2, Data Sheet
-Channels A & B; Reference 4, Grid C-7
-Inlet/out. (N1/N2); Reference 4, Grid C-2
-Manway; Reference 5, p. 5

$$id \equiv od - 2 \cdot t_a$$

$$id = \begin{pmatrix} 0.68 \\ 65 \\ 18.75 \\ 18 \end{pmatrix} in \qquad l_i = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Inside diameter

$$r_i \equiv id \div 2$$

$$r_i = \begin{pmatrix} 0.34 \\ 32.5 \\ 9.375 \\ 9 \end{pmatrix} in \qquad l_i = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Inside radius

$$S \equiv \begin{pmatrix} 8.4 \\ 9.8 \\ 9.8 \\ 9.5 \end{pmatrix} \cdot ksi \qquad l_i = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Allowable stress at design temperature
-tube; Ref. 1, Table UNF-23; assume welded tube
-Channels A & B; Reference 1, Table UNF-23
-Inlet/out.; Reference 1, Table UNF-23
-Manway; Reference 5, p. 3

$$E \equiv \begin{pmatrix} 0.6 \\ 0.85 \\ 0.80 \\ 0.7 \end{pmatrix} \qquad l_i = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Joint efficiency
-tube; Section 3.2
-Channels A & B; Ref. 13
-Inlet/out.; Reference 4, Grid B-2
-Manway; Reference 5, p. 8

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: R. PARKERSON FOR

Gaskets

$od_g \equiv \begin{pmatrix} 67.875 \\ 21.5 \end{pmatrix} \cdot in$ $\quad l_2 := \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Gasket effective OD
-Channels A & B; Reference 2, Dwg. No. G-15763
-Manway; Ref. 5, p. 5; assumed to be bc minus bolt hole diameter

$N_g \equiv \begin{pmatrix} 0.5 \\ 1.5 \end{pmatrix} \cdot in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Gasket width
-Channels A & B; Reference 2, Dwg. No. G-15763
-Manway; Ref. 5, p. 5; based on radial distance from od_g to gasket ID

$d \equiv od_g - N_g$

$d = \begin{pmatrix} 67.38 \\ 20 \end{pmatrix} in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Mean gasket diameter

$T_g \equiv \frac{1}{8} \cdot in$

Thickness of channel gasket;
Reference 2, Dwg. No. G-15763

$m_g \equiv \begin{pmatrix} 3.5 \\ 6.6 \end{pmatrix}$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Gasket factor
-Channels A & B; Ref. 5, p. 6
-Manway; Ref. 5, p. 5 and Ref. 6, p. 35

$y_g \equiv \begin{pmatrix} 8000 \\ 3000 \end{pmatrix} \cdot psi$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Gasket seating stress
-Channels A & B; Ref. 5, p. 6
-Manway; Ref. 5, p. 5 and Ref. 6, p. 35

Flange Bolting

$n_b \equiv \begin{pmatrix} 72 \\ 24 \end{pmatrix}$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Number of bolts
-Channels A & B; Ref. 4, Grid A-2
-Manway; Ref. 5, p. 5

$A_r \equiv \begin{pmatrix} 0.419 \\ 0.693 \end{pmatrix} \cdot in^2$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Bolt area at thread root diameter
-Reference 7, Table 8.2.2 for bolt area
-Channels A & B; Ref. 5, p. 5
-Manway; Ref. 5, p. 5

$S_b \equiv \begin{pmatrix} 25 \\ 25 \end{pmatrix} \cdot ksi$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Allowable bolt stress at design temperature
Reference 14 Table 3

$S_a \equiv S_b$

Allowable bolt stress at room temperature
Reference 14, Table 3

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: [signature] FOR R. PARKERSON

Calculation No.: 0108-0309-jlh-1
Revision No.: 0
Page No.: 11

Flanges

$A \equiv \begin{pmatrix} 71.125 \\ 25 \end{pmatrix} \cdot in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

OD of flange
-Channels A & B; Reference 4, Grid A-3
-Manway; Reference 5, p. 5

$B \equiv \begin{pmatrix} 66.25 \\ 18.625 \end{pmatrix} in$ $\quad B \equiv \begin{pmatrix} od_{Channel} \\ od_{Manway} \end{pmatrix}$

ID of flange based on pipe OD (see Ref. 1, Figure UA-48(8a))

$t_{fl} \equiv \begin{pmatrix} 4.625 \\ 2 \end{pmatrix} \cdot in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Thickness of flange
-Channels A & B; Reference 4, Grid A-3
-Manway; Ref. 5, p. 5 (not including clad)

$bc \equiv \begin{pmatrix} 69.25 \\ 22.75 \end{pmatrix} \cdot in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Bolt circle diameter
-Channels A & B; Reference 4, Grid A-3
-Manway; Ref. 5, p. 5

$w_g \equiv \frac{1}{2} \cdot in$

Width of raised face on channel flange; Reference 4, Grid A-3

$S_f \equiv \begin{pmatrix} 17.5 \\ 17.5 \end{pmatrix} \cdot ksi$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Allowable stress for flange at design temperature
-Channels A & B; Reference 1, Table UCS-23
-Manway; Reference 1, Table UCS-23

$P_{150.allow} := 240 \cdot psi$

Allowable pressure for a 150# class flange at the design temperature; Ref. 8, Table 2

Flat Plates

$t_{flat} \equiv \begin{pmatrix} 6.188 \\ 1.5 \end{pmatrix} \cdot in$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Flat head thickness (not including clad)
-Channels A & B; Reference 4, Grid D-8
-Manway; Reference 5, p. 5

$C \equiv \begin{pmatrix} 0.3 \\ 0.25 \end{pmatrix}$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Factor
-Channels A & B; Reference 1, Figure UG-34(j)
-Manway; Reference 1, Figure UG-34(p)

$S_{flat} \equiv \begin{pmatrix} 17.5 \\ 17.5 \end{pmatrix} \cdot ksi$ $\quad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$

Allowable stress for plate at design temperature
-Channels A & B; Reference 1, Table UCS-23
-Manway; Reference 1, Table UCS-23

Tubes

$N_t \equiv 3400$ — Number of tubes; Ref. 2, Data Sheet

$p_t \equiv 1 \cdot in$ — Tube pitch; Ref. 2, Data Sheet

$\alpha_t \equiv 4.65 \cdot 10^{-6} \cdot \frac{in}{in \cdot F}$ — Mean coefficient of thermal expansion for tube at 100°F (approximate average tube temperature); Ref. 11, Table TE-5

$E_t \equiv linterp\left[\begin{pmatrix}70\\200\end{pmatrix} \cdot F, \begin{pmatrix}15.5\\15\end{pmatrix} \cdot 10^{6} \cdot psi, 100 \cdot F\right]$ — Modulus of elasticity for tube at 100°F (approximate average tube temperature); Ref. 11, Table TM-5

$E_t = 15.38 \times 10^{6}\, psi$

Shell

$id_s \equiv 65 \cdot in$ — Shell ID; Ref. 2, Data Sheet

$t_s \equiv 0.5 \cdot in$ — Shell wall thickness; Ref. 2, Data Sheet

$od_s \equiv id_s + 2 \cdot t_s \qquad od_s = 66\, in$ — Shell OD

$L \equiv 25 \cdot ft + 11.75 \cdot in - 2 \cdot 3.625 \cdot in$ — Length between tubesheets; Ref. 10, Grid D-6

$L = 304.50\, in$

$\alpha_s \equiv 5.73 \cdot 10^{-6} \cdot \frac{in}{in \cdot F}$ — Mean coefficient of thermal expansion for shell at 100°F (approximate average shell temperature); Ref. 11, Table TE-1, Material Group B

$E_s \equiv linterp\left[\begin{pmatrix}70\\200\end{pmatrix} \cdot F, \begin{pmatrix}29.5\\28.8\end{pmatrix} \cdot 10^{6} \cdot psi, 100 \cdot F\right]$ — Modulus of elasticity for shell at 100°F (approximate average shell temperature); Ref. 11, Table TM-1, C<0.3%

$E_s = 29.34 \times 10^{6}\, psi$

Tubesheet

$t_{ts} \equiv 3.5 \cdot in$ — Tubesheet thickness, not including clad; Ref. 10, Grid C-6

$S_{ts} \equiv 17.5 \cdot ksi$ — Allowable stress for tubesheet at design temperature; Reference 1, Table UCS-23

$$E_{ts} \equiv linterp\left[\begin{pmatrix}70\\200\end{pmatrix} \cdot F, \begin{pmatrix}29.5\\28.8\end{pmatrix} \cdot 10^{6} \cdot psi, 100 \cdot F\right]$$

$$E_{ts} = 29.34 \times 10^{6}\ psi$$

Modulus of elasticity for tubesheet at 100°F (approximate average temperature); Ref. 11, Table TM-1, C<0.3%; the tubesheet material specification for carbon is in the range of 0.3%; for this calculation, it is conservative to use the modulus for carbon <0.3% (based on a trial and error approach)

Inlet/outlet Nozzle Reinforcement Pad

$t_{pad} \equiv \frac{5}{8} \cdot in$ — Inlet and outlet nozzle reinforcement pad thickness; Ref. 4, Grid C-1

$od_{pad} \equiv 30 \cdot in$ — Inlet and outlet nozzle reinforcement pad OD; Ref. 4, Grid C-1

Seismic

$a_v \equiv 1.5 \cdot g$

$a_h \equiv 1.2 \cdot g$

Seismic static horizontal and vertical acceleration; Ref. 9, Attachment B, p. B-3

$P_{eq.ch} := 2.416 \cdot psi$ — Channel equivalent pressure for seismic acceleration; Ref. 5, Paragraph 7.B.1

$F_{eq.m} := 540 \cdot lbf$ — Manway equivalent load for seismic acceleration; Ref. 5, Paragraph 7.B.2

4.2 Minimum Thickness of Cylindrical Shell

Determine the minimum required wall thickness of cylindrical shells from Reference 1, UG-27(c)(1). UG-27(c)(1) requires that one of the following applicability criteria be met:

$$\overrightarrow{\left(t_a \le \frac{r_i}{2}\right)} = \begin{pmatrix} 1 \\ 1 \\ 1 \\ 1 \end{pmatrix} \qquad l_l = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

A value of 1 indicates the applicability criterion is met. A value of 0 indicates the applicability criterion is not met.

$$\overrightarrow{\left(P_t \le 0.385 \cdot S \cdot E\right)} = \begin{pmatrix} 1 \\ 1 \\ 1 \\ 1 \end{pmatrix}$$

where

$$t_a = \begin{pmatrix} 0.035 \\ 0.625 \\ 0.625 \\ 0.313 \end{pmatrix} in \qquad r_i = \begin{pmatrix} 0.34 \\ 32.5 \\ 9.375 \\ 9 \end{pmatrix} in \qquad S = \begin{pmatrix} 8.4 \\ 9.8 \\ 9.8 \\ 9.5 \end{pmatrix} ksi \qquad E = \begin{pmatrix} 0.6 \\ 0.85 \\ 0.8 \\ 0.7 \end{pmatrix}$$

$P_t = 191\ psi$

Determine the minimum required wall thickness.

$$t_{r.cyl} := \overrightarrow{\frac{P_t \cdot r_i}{S \cdot E - 0.6 \cdot P_t}}$$

where t_r = required wall thickness
P_t = design pressure
r_i = inside radius
S = maximum allowable stress
E = joint efficiency

$$t_{r.cyl} = \begin{pmatrix} 0.013 \\ 0.756 \\ 0.232 \\ 0.263 \end{pmatrix} in \qquad l_l = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

Summarize the results and compare to the actual wall thickness.

$$T1 := \left| \begin{array}{l} \text{for } i \in 1..\text{rows}(t_{r.cyl}) \\ \quad a_i \leftarrow \text{if}\left(t_{r.cyl_i} \le t_{a_i}, ok, nok\right) \\ \text{stack}\left(R1, \text{augment}\left(l_1, t_{r.cyl} \div in, t_a \div in, a\right)\right) \end{array} \right.$$

$$T1 = \begin{pmatrix} \text{"Location"} & \text{"Required"} & \text{"Actual"} & \text{"Result"} \\ \text{""} & \text{"Thickness"} & \text{"Thickness"} & \text{""} \\ \text{""} & \text{"in."} & \text{"in."} & \text{""} \\ \text{"Tube"} & 0.013 & 0.035 & \text{"Ok"} \\ \text{"Channel A \& B"} & 0.756 & 0.625 & \text{"Not Ok"} \\ \text{"Inlet \& Out. Nozzle"} & 0.232 & 0.625 & \text{"Ok"} \\ \text{"Manway Nozzle"} & 0.263 & 0.313 & \text{"Ok"} \end{pmatrix}$$

4.3 Flange Bolt Tensile Area

The flange design bolt load is calculated with Reference 1, Paragraph UA-49. The basic gasket seating width is calculated in accordance with Reference 1, Table UA-49.2. Use Sketch 1c, Column II for the channel gasket and Sketch 1a, Column II for the manway gasket.

Note: The inlet and outlet nozzle flange bolts are not included in the evaluation because the inlet and outlet nozzle flanges are standard ANSI B16.5 flanges for attached piping. See Section 4.6 for an evaluation of the inlet and outlet nozzle flanges.

The basic gasket seating width for the channel and manway gaskets are:

$$b_o := \begin{pmatrix} min\left(\frac{w_g + T_g}{2}, \frac{w_g + N_{g_1}}{4}\right) \\ \frac{N_{g_2}}{2} \end{pmatrix} \qquad b_o = \begin{pmatrix} 0.25 \\ 0.75 \end{pmatrix} in \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

where

$$\begin{pmatrix} \frac{w_g + T_g}{2} \\ \frac{w_g + N_{g_1}}{4} \end{pmatrix} = \begin{pmatrix} 0.313 \\ 0.25 \end{pmatrix} in \qquad w_g = 0.5\,in \qquad T_g = 0.125\,in \qquad N_{g_1} = 0.5\,in$$

The effective gasket seating width is calculated with Reference 1, Table UA-49.2.

$$b_i := if\left(b_{o_i} \le 0.25 \cdot in, b_{o_i}, \frac{\sqrt{b_{o_i} \cdot in}}{2} \cdot in\right) \qquad b = \begin{pmatrix} 0.25 \\ 0.433 \end{pmatrix} in \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

where $b_o = \begin{pmatrix} 0.25 \\ 0.75 \end{pmatrix} in$

Note: The gasket width for the manway is less than the actual width and so the gasket seating load calculated below is underestimated. The approach used is considered reasonable for a full face gasket for the purpose of sizing the flange bolts.

The location of the gasket load reaction is calculated with Reference 1, Table UA-49.2.

$$G_i := if\left(b_{o_i} \le 0.25 \cdot in, d_i, od_{g_i} - 2 \cdot b_i\right) \qquad G = \begin{pmatrix} 67.375 \\ 20.634 \end{pmatrix} in \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

where $d = \begin{pmatrix} 67.375 \\ 20 \end{pmatrix} in$ $\qquad od_g = \begin{pmatrix} 67.875 \\ 21.5 \end{pmatrix} in$

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: [signature] FOR R. PARKERSON

The flange design bolt load W_{m1} is (UA-49(a)(1) formula (1)):

$$H_i := \frac{\pi}{4}\cdot\left(G_i\right)^2\cdot P_t \qquad H = \begin{pmatrix}680959\\63869\end{pmatrix} lbf \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

$$Hp_i := 2\cdot b_i\cdot\pi\cdot G_i\cdot m_{g_i}\cdot P_t \qquad Hp = \begin{pmatrix}70749\\70769\end{pmatrix} lbf \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

$$Wm1_i := H_i + Hp_i \qquad Wm1 = \begin{pmatrix}751708\\134637\end{pmatrix} lbf \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

where $m_g = \begin{pmatrix}3.5\\6.6\end{pmatrix}$ $\qquad P_t = 191\ psi$

The flange design bolt load W_{m2} is (UA-49(a)(2) formula (2)):

$$Wm2_i := \pi\cdot b_i\cdot G_i\cdot y_{g_i} \qquad Wm2 = \begin{pmatrix}423330\\84208\end{pmatrix} lbf \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

where $y_g = \begin{pmatrix}8000\\3000\end{pmatrix} psi$

The actual bolt tensile area is:

$$Ab_i := nb_i\cdot A_{t_i} \qquad Ab = \begin{pmatrix}30.17\\16.63\end{pmatrix} in^2 \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

where $nb = \begin{pmatrix}72\\24\end{pmatrix}$ $\qquad A_t = \begin{pmatrix}0.419\\0.693\end{pmatrix} in^2$

The required bolt areas A_{m1} and A_{m2} are:

$$Am1_i := \frac{Wm1_i}{Sb_i} \qquad Am1 = \begin{pmatrix}30.07\\5.39\end{pmatrix} in^2 \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

$$Am2_i := \frac{Wm2_i}{Sa_i} \qquad Am2 = \begin{pmatrix}16.93\\3.37\end{pmatrix} in^2 \qquad l_2 = \begin{pmatrix}\text{"Channel A \& B"}\\\text{"Manway"}\end{pmatrix}$$

where $Sa = \begin{pmatrix}25\\25\end{pmatrix} ksi$ $\qquad Sb = \begin{pmatrix}25\\25\end{pmatrix} ksi$

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

The required bolt area A_m is:

$$A_{m_i} := max\left(A_{m1_i}, A_{m2_i}\right) \qquad A_m = \begin{pmatrix} 30.07 \\ 5.39 \end{pmatrix} in^2 \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

Summarize the results.

$$T2 := \left| \begin{array}{l} for\ \ i \in 1..rows(A_m) \\ \quad a_i \leftarrow if\left(A_{m_i} \le A_{b_i}, ok, nok\right) \\ stack\left(R2, augment\left(l_2, A_m \div in^2, A_b \div in^2, a\right)\right) \end{array} \right.$$

$$T2 = \begin{pmatrix} \text{"Location"} & \text{"Required"} & \text{"Actual"} & \text{"Result"} \\ \text{""} & \text{"Bolt Area"} & \text{"Bolt Area"} & \text{""} \\ \text{""} & \text{"in. ^ 2"} & \text{"in. ^ 2"} & \text{""} \\ \text{"Channel A \& B"} & 30.07 & 30.17 & \text{"Ok"} \\ \text{"Manway"} & 5.39 & 16.63 & \text{"Ok"} \end{pmatrix}$$

The flange design bolt load is the maximum of the load for operating and gasket seating conditions (Reference 1, Paragraph UA-49(c)). From Reference 1, UA-49(c), the bolt load for operating conditions is W_{m1}. The bolt load for gasket seating conditions is:

$$W_{seat_i} := \frac{\left(A_{m_i} + A_{b_i}\right) \cdot S_{a_i}}{2} \qquad W_{seat} = \begin{pmatrix} 752954 \\ 275219 \end{pmatrix} lbf \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

The flange design bolt load is the maximum of the load for operating and gasket seating conditions.

$$W_i := max\left(W_{m1_i}, W_{seat_i}\right) \qquad W = \begin{pmatrix} 752954 \\ 275219 \end{pmatrix} lbf \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

4.4 Channel Flange Evaluation

Flange moments for the operating and gasket seating conditions are calculated with Reference 1, Paragraph UA-50. The flange is evaluated as a loose type flange because the material strength of the channel is less than that of the flange ring and because the ASME Code allows this approach (Reference 1, Figure UA-48 and Paragraph UA-48(a)(3)). Evaluate the criteria for applying the loose flange evaluation.

$g_o := t_{a_{Channel}}$ $\qquad g_o = 0.625\,in$

$$\begin{pmatrix} g_o \le \frac{5}{8}\cdot in \\ \frac{B_{channel}}{g_o} \le 300 \\ P_t \le 300\cdot psi \end{pmatrix} = \begin{pmatrix} 1 \\ 1 \\ 1 \end{pmatrix}$$

A value of 1 indicates the applicability criterion is met. A value of 0 indicates the applicability criterion is not met.

where $\quad B_{channel} = 66.25\,in \qquad \frac{B_{channel}}{g_o} = 106$

$P_t = 191\,psi$

Operating Condition

Calculate the moments M_D, M_T, and M_G with Reference 1, UA-47(b).

$H_D := \frac{\pi}{4}\cdot\left(B_{channel}\right)^2\cdot P_t \qquad H_D = 658408\,lbf$

$H_G := W_{m1_{channel}} - H_{channel} \qquad H_G = 70749\,lbf$ $\qquad$ See Reference 1, UA-47

$H_T := H_{channel} - H_D \qquad H_T = 22551\,lbf$

where $\quad W_{m1_{channel}} = 751708\,lbf \qquad H_{channel} = 680959\,lbf$

$$h_D := \frac{bc_{channel} - B_{channel}}{2} \qquad h_D = 1.5\,in$$

See Reference 1, Table UA-50; the flange is conservatively treated as a loose type flange, since the channel material is weaker than the flange material.

$$h_G := \frac{bc_{channel} - G_{channel}}{2} \qquad h_G = 0.94\,in$$

$$h_T := \frac{h_D + h_G}{2} \qquad h_T = 1.22\,in$$

where $bc_{channel} = 69.25\,in$ $\qquad G_{channel} = 67.375\,in$

$$M_D := H_D \cdot h_D \qquad M_D = 82301\,ft\cdot lbf$$

$$M_G := H_G \cdot h_G \qquad M_G = 5527\,ft\cdot lbf$$

$$M_T := H_T \cdot h_T \qquad M_T = 2290\,ft\cdot lbf$$

The total moment is:

$$M_{op} := M_D + M_G + M_T \qquad M_{op} = 90119\,ft\cdot lbf$$

Gasket Seating Condition

From Reference 1, Paragraph UA-50, Equation (5):

$$M_{gasket} := W_{seat_{channel}} \cdot \frac{\left(bc_{channel} - G_{channel}\right)}{2} \qquad M_{gasket} = 58825\,ft\cdot lbf$$

Flange Stresses

The maximum moment is:

$M := max(M_{op}, M_{gasket})$ $\quad$ $M = 90119\ ft \cdot lbf$

The flange stresses from Reference 1, Paragraph UA-51(2) are:

$K_{ch} := \dfrac{A_{channel}}{B_{channel}}$ $\quad$ $K_{ch} = 1.074$ $\quad$ Reference 1, Paragraph UA-47(b)

where $\quad A_{channel} = 71.125\ in \quad B_{channel} = 66.25\ in$

$$Y := \frac{1}{K_{ch} - 1} \cdot \left(0.66845 + 5.71690 \cdot \frac{K_{ch}^2 \cdot log(K_{ch})}{K_{ch}^2 - 1}\right)$$ Reference 1, Figure UA-51.1

$Y = 27.18$

$S_T = \dfrac{Y \cdot M}{t_{fl}^2 \cdot B}$

$S_R := 0 \cdot psi$

$S_H := 0 \cdot psi$

Rearrange the equation to calculate the minimum required thickness of the flange.

$t_{r.flange} := \sqrt{\dfrac{Y \cdot M}{S_{f_{channel}} \cdot B_{channel}}}$ $\quad$ $t_{r.flange} = 5.035\ in$

where $\quad S_{f_{channel}} = 17.5\ ksi$ $\quad$ The allowable stress for the tangential stress, ST, is from Reference 1, Paragraph UA-52(a)(3)

The actual flange thickness is $t_{fl_{channel}} = 4.63\ in.$ Add the results to the summary table T1.

$T1 := stack\left(T1, augment\left(\text{"Channel Flange"}, t_{r.flange} \div in, t_{fl_{channel}} \div in, if\left(t_{r.flange} \le t_{fl_{channel}}, ok, nok\right)\right)\right)$

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: [signature] FOR R. PARKERSON

4.5 Manway Flange Evaluation

The manway flange has a full face gasket, which cannot be evaluated with the rules of Reference 1, Appendix II, Part A. Reference 1, Paragraph UA-56 references Paragraph UA-6(b)(2). This provides the required flange thickness for a spherically dished cover with a full face gasket.

$$t_{r.manway} := 0.6 \cdot \sqrt{\frac{P_t}{Sf_{manway}} \left[\frac{B_{manway} \cdot (A_{manway} + B_{manway}) \cdot (bc_{manway} - B_{manway})}{A_{manway} - B_{manway}} \right]}$$

$$t_{r.manway} = 1.44\ in$$

where $P_t = 191\ psi$ $\quad A_{manway} = 25\ in$ $\quad bc_{manway} = 22.75\ in$

$B_{manway} = 18.625\ in$ $\quad Sf_{manway} = 17.5\ ksi$

The actual flange thickness is $tfl_{manway} = 2\ in$. Add the results to the summary table T1.

$$T1 := stack\left(T1, augment\left(\text{"Manway Flange"}, t_{r.manway} \div in, tfl_{manway} \div in, if\left(t_{r.manway} \le tfl_{manway}, ok, nok\right)\right)\right)$$

4.6 Inlet and Outlet Nozzle Flange Evaluation

Reference 1, Paragraph UG-44(a) recommends that bolted flanges to external piping conform to a recognized standard. Such flanges may be used in accordance with the pressure-temperature ratings of the standard.

The inlet and outlet nozzle flanges are ANSI B16.5 standard flanges with a 150# rating (Reference 4, Grid C-2). At a design temperature of 200°F, the allowable pressure is $P_{150.allow} = 240\ psi$ (Reference 8, Table 2).

Summarize the results.

$$T5 := \begin{pmatrix} \text{"Location"} & \text{"Pressure"} & \text{"Design"} & \text{"Result"} \\ \text{""} & \text{"Rating"} & \text{"Pressure"} & \text{""} \\ \text{""} & \text{"psi"} & \text{"psi"} & \text{""} \\ \text{"Inlet \& Outlet Nozzle Flange"} & P_{150.allow} \div psi & P_t \div psi & if(P_t \le P_{150.allow}, ok, nok) \end{pmatrix}$$

$$T5 = \begin{pmatrix} \text{"Location"} & \text{"Pressure"} & \text{"Design"} & \text{"Result"} \\ \text{""} & \text{"Rating"} & \text{"Pressure"} & \text{""} \\ \text{""} & \text{"psi"} & \text{"psi"} & \text{""} \\ \text{"Inlet \& Outlet Nozzle Flange"} & 240 & 191 & \text{"Ok"} \end{pmatrix}$$

4.7 Minimum Thickness of Flat Cover

Determine the minimum required wall thickness of flat heads from Reference 1, Paragraph UG-34, Equation 2. The minimum thickness for the channel cover is calculated for operating conditions. The minimum thickness for the manway cover is calculated with the formula for no edge moment, because the manway uses a full face gasket.

$$t_{r.flat} := \begin{bmatrix} d_{channel} \cdot \sqrt{\dfrac{C_{channel} \cdot P_t}{Sflat_{channel}} + \dfrac{1.78 \cdot Wm1_{channel} \cdot h_G}{Sflat_{channel} \cdot \left(d_{channel}\right)^3}} \\ bc_{manway} \cdot \sqrt{\dfrac{C_{manway} \cdot P_t}{Sflat_{manway}}} \end{bmatrix} \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

where	$t_{r.flat}$	=	required wall thickness
	d	=	flat plate diameter, for which the mean gasket diameter is used
	C	=	geometry dependent factor from ASME Code
	P_t	=	design pressure
	S_{flat}	=	maximum allowable stress
	W	=	bolt load
	h_G	=	gasket moment arm
	bc	=	flat plate diameter, for which the bolt circle diameter is used

$$t_{r.flat} = \begin{pmatrix} 3.991 \\ 1.188 \end{pmatrix} in \qquad l_2 = \begin{pmatrix} \text{"Channel A \& B"} \\ \text{"Manway"} \end{pmatrix}$$

where

$P_t = 191\ psi$ $\qquad Sflat = \begin{pmatrix} 17.5 \\ 17.5 \end{pmatrix} ksi$ $\qquad bc_{manway} = 22.75\ in$

$d_{channel} = 67.38\ in$ $\qquad Wm1_{channel} = 751708\ lbf$

$C = \begin{pmatrix} 0.3 \\ 0.25 \end{pmatrix}$ $\qquad h_G = 0.938\ in$

The actual cover thicknesses are $t_{flat} = \begin{pmatrix} 6.19 \\ 1.5 \end{pmatrix} in$. Add the results to the summary table T1.

$$T1 := \left| \begin{array}{l} \textit{for } i \in 1..\mathrm{rows}(t_{r.flat}) \\ \quad a_i \leftarrow \mathrm{if}\left(t_{r.flat_i} \le t_{flat_i}, ok, nok\right) \\ \mathrm{stack}\left(T1, \mathrm{augment}\left(I_3, t_{r.flat} \div in, t_{flat} \div in, a\right)\right) \end{array} \right.$$

The channel cover plate is grooved for the gasket as shown in Reference 1, Figure UG-34(k). The requirement for the cover plate thickness at the groove is provided in Reference 1, Paragraph UG-34(d).

$$t_{r.groove} = d_{channel} \cdot \sqrt{\frac{1.78 \cdot W_{m1_{channel}} \cdot h_G}{S_{flat_{channel}} \cdot \left(d_{channel}\right)^3}}$$

Reference 5, p. 5 shows that the gasket groove in the cover plate is in the clad material and that the groove does not infringe on the cover plate thickness of $t_{flat_{channel}} = 6.19\ in$ that is used in the above evaluation. It is concluded that the thickness of the cover at the gasket groove is acceptable without further evaluation.

4.8 Reinforcement Area

Evaluate the reinforcement area for the inlet/outlet nozzle opening in the channel and the manway opening in the channel cover. The reinforcement area requirements are in Reference 1, Paragraphs UG-37 (channel) and UG-39 (channel cover).

The required reinforcement area is:

$$A_r := \begin{pmatrix} id_{Inlet} \cdot t_{r.cyl_{Channel}} \cdot 1 \\ 0.5 \cdot id_{Manway} \cdot t_{r.flat_{channel}} \end{pmatrix} \qquad A_r = \begin{pmatrix} 14.17 \\ 35.92 \end{pmatrix} in^2 \qquad t_4 := \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$

where $id_{Inlet} = 18.75\,in$ $\qquad id_{Manway} = 18\,in$

$t_{r.cyl_{Channel}} = 0.756\,in$ $\qquad t_{r.flat_{channel}} = 3.991\,in$

Define functions to calculate the reinforcement area available in the vessel wall and in the nozzle (Reference 1, UG-40).

$$A1(E_1, t, F, t_r, d, t_n) := \left| \begin{array}{l} a1 \leftarrow (E_1 \cdot t - F \cdot t_r) \cdot d \\ a2 \leftarrow 2 \cdot (E_1 \cdot t - F \cdot t_r) \cdot (t + t_n) \\ max(a1, a2, 0) \end{array} \right.$$

$$A2(t_n, t_m, t, t_e) := \left| \begin{array}{l} a1 \leftarrow (t_n - t_m) \cdot 5 \cdot t \\ a2 \leftarrow (t_n - t_m) \cdot (5 \cdot t_n + 2 \cdot t_e) \\ max(0, min(a1, a2)) \end{array} \right.$$

where E_1 = joint efficiency for a longitudinal weld if it passes through nozzle; 1 otherwise
t = nominal vessel wall thickness
F = correction factor for variation in pressure stress; 1.0 for this evaluation
t_r = required thickness of vessel
d = diameter of finished opening
t_n = nominal nozzle thickness
t_m = required thickness of nozzle
t_e = thickness of reinforcement pad

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: JHubbard Checked By: [signature] FOR R. PARKERSON	Calculation No.: 0108-0309-jth-1 Revision No.: 0 Page No.: 27

Calculate the limit of reinforcement parallel and perpendicular to the vessel wall (Reference 1, Paragraph UG-40).

$$L_{par} := \begin{pmatrix} max\left(id_{Inlet}, \frac{id_{Inlet}}{2} + t_{a_{Channel}} + t_{a_{Inlet}}\right) \\ max\left(id_{Manway}, \frac{id_{Manway}}{2} + t_{flat_{channel}} + t_{a_{Manway}}\right) \end{pmatrix} \qquad \text{UG-40(b)(1)\&(2)}$$

$$L_{par} = \begin{pmatrix} 18.75 \\ 18 \end{pmatrix} in \qquad l_4 = \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$

$$L_{per} := \begin{pmatrix} min\left(2.5 \cdot t_{a_{Channel}}, 2.5 \cdot t_{a_{Inlet}} + t_{pad}\right) \\ min\left(2.5 \cdot t_{flat_{channel}}, 2.5 \cdot t_{a_{Manway}} + 0 \cdot in\right) \end{pmatrix} \qquad \text{UG-40(c)(1)\&(2)}$$

$$L_{per} = \begin{pmatrix} 1.56 \\ 0.78 \end{pmatrix} in \qquad l_4 = \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$

where

$$t_a = \begin{pmatrix} 0.035 \\ 0.625 \\ 0.625 \\ 0.313 \end{pmatrix} in \qquad id = \begin{pmatrix} 0.68 \\ 65 \\ 18.75 \\ 18 \end{pmatrix} in \qquad l_1 = \begin{pmatrix} \text{"Tube"} \\ \text{"Channel A \& B"} \\ \text{"Inlet \& Out. Nozzle"} \\ \text{"Manway Nozzle"} \end{pmatrix}$$

$$t_{flat_{channel}} = 6.19\, in \qquad t_{pad} = 0.625\, in$$

Calculate the reinforcement area A1.

$$A_1 := \begin{pmatrix} A1\left(1, t_{a_{Channel}}, 1, t_{r.cyl_{Channel}}, id_{Inlet}, t_{a_{Inlet}}\right) \\ A1\left(1, t_{flat_{channel}}, 1, t_{r.flat_{channel}}, id_{Manway}, t_{a_{Manway}}\right) \end{pmatrix} \qquad \text{UG-40(d)(1))}$$

$$A_1 = \begin{pmatrix} 0 \\ 39.55 \end{pmatrix} in^2 \qquad l_4 = \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$

where $t_{r.flat_{channel}} = 3.991\, in$ $\qquad t_{r.cyl_{Channel}} = 0.756\, in$

Calculate the reinforcement area A2.

$$A_2 := \begin{pmatrix} A2\left(t_{a_{Inlet}}, t_{r.cyl_{Inlet}}, t_{a_{Channel}}, t_{pad}\right) \\ A2\left(t_{a_{Manway}}, t_{r.cyl_{Manway}}, t_{flat_{channel}}, 0 \cdot in\right) \end{pmatrix}$$ UG-40(d)(2)

$$A_2 = \begin{pmatrix} 1.229 \\ 0.077 \end{pmatrix} in^2 \qquad l_4 = \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$

where $t_{r.cyl_{Inlet}} = 0.232\,in$ $\quad t_{r.cyl_{Manway}} = 0.263\,in$ $\quad t_{pad} = 0.625\,in$

Calculate the reinforcement area of the reinforcing pad, A5.

$$A_5 := \begin{bmatrix} t_{pad} \cdot \left(min\left(od_{pad}, 2 \cdot L_{par_1}\right) - od_{Inlet}\right) \\ 0 \cdot in^2 \end{bmatrix}$$

$$A_5 = \begin{pmatrix} 6.25 \\ 0 \end{pmatrix} in^2 \qquad l_4 = \begin{pmatrix} \text{"Inlet/Outlet Nozzle"} \\ \text{"Manway"} \end{pmatrix}$$ UG-40(d)(3)

where $od_{pad} = 30\,in$ $\quad L_{par_1} = 18.75\,in$ $\quad od_{Inlet} = 20\,in$

The total area of reinforcement is:

$$A_{tot} := A_1 + A_2 + A_5$$

$$A_{tot} = \begin{pmatrix} 7.48 \\ 39.63 \end{pmatrix} in^2$$

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: JHHubbard Checked By: [signature] FOR R. PARKERSON	Calculation No.: 0108-0309-jlh-1 Revision No.: 0 Page No.: 29

Summarize the results.

$$T7 := \left| \begin{array}{l} \text{for } i \in 1..\text{rows}(A_{tot}) \\ \quad a_i \leftarrow \text{if}\left(A_{r_i} \le A_{tot_i}, ok, nok\right) \\ \text{stack}\left(R3, \text{augment}\left(l_4, A_r \div in^2, A_{tot} \div in^2, a\right)\right) \end{array} \right.$$

$$T7 = \begin{pmatrix} \text{"Location"} & \text{"Required"} & \text{"Actual"} & \text{"Result"} \\ \text{""} & \text{"Area"} & \text{"Area"} & \text{""} \\ \text{""} & \text{"in.^2"} & \text{"in.^2"} & \text{""} \\ \text{"Inlet/Outlet Nozzle"} & 14.17 & 7.48 & \text{"Not Ok"} \\ \text{"Manway"} & 35.92 & 39.63 & \text{"Ok"} \end{pmatrix}$$

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hubbard

Checked By: [signature] FOR R. PARKERSON

Calculation No.: 0108-0309-jlh-1

Revision No.: 0

Page No.: 30

4.9 Tubesheet

The ASME Code (Reference 1) has no requirement for the thickness of the tubesheet. Reference 2, Data Sheet indicates the heat exchanger was designed to the TEMA R standard. Use the 1968 TEMA R standard (Reference 12) to determine the minimum required tubesheet thickness.

The minimum tubesheet thicknesses for bending and for shear are (Reference 12, Paragraphs R-7.122 and R-7.123):

$$t_{bend} = \frac{F_m \cdot G_c}{2} \cdot \sqrt{\frac{P_{ts}}{S_{ts}}}$$

$$t_{shear} = \frac{0.31 \cdot D_L}{1 - \frac{od_t}{p_t}} \cdot \frac{P_{ts}}{S_{ts}}$$

where
- t_{bend} = minimum required tubesheet thickness for bending
- t_{shear} = minimum required tubesheet thickness for shear
- F_m = thickness multiplier
- G_c = ID of pressure vessel
- P_{ts} = tubesheet design pressure
- S_{ts} = tubesheet allowable stress at design temperature
- D_L = equivalent diameter of tube bundle (4*area/circumference)
- od_t = tube OD
- p_t = tube pitch

The tubehseet is integral with the shell and the channel head, i.e., the tubesheet is welded to both structures and benefits from the stiffening they provide. In addition, the tubesheet is stationary (there is no shell expansion joint). Accordingly, the parameters G_c and F_m are determined with Paragraph R-7.141 of Reference 12.

G_c can be the ID of the channel head or the ID of the shell depending on the calculation. Since the two diameters are equal ($id_{Channel} = 65\ in$ and $id_s = 65\ in$), set G_c to the ID of the channel.

$G_c = id_{Channel}$ $\qquad\qquad$ $G_c = 65\ in$

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: [signature] Checked By: [signature] For R. PARKERSON	Calculation No.: 0108-0309-jlh-1 Revision No.: 0 Page No.: 31

Calculate the thickness multiplier, F_m, with Reference 12, Figure R-7.141 for integral stationary tubesheets. Calculate the ratio of wall thickness to ID for both the channel and the shell. Use the smaller of the two ratios to calculate F_m for conservatism.

$$ratios := \begin{pmatrix} \dfrac{t_{a_{Channel}}}{id_{Channel}} \\ \dfrac{t_s}{id_s} \end{pmatrix} \qquad ratios = \begin{pmatrix} 0.01 \\ 0.008 \end{pmatrix}$$

where $t_{a_{Channel}} = 0.625\,in$ $\quad id_{Channel} = 65\,in$ $\quad t_s = 0.5\,in$

$id_s = 65\,in$

$$r_{max} := min(ratios) \qquad r_{max} = 0.008$$

$$F_{7.141}(x) := \begin{cases} 1 & \text{if } x < 0.02 \\ linterp\left[\begin{pmatrix} .02 \\ .05 \end{pmatrix}, \begin{pmatrix} 1 \\ 0.8 \end{pmatrix}, x\right] & \text{if } 0.02 \le x \le 0.05 \\ 0.8 & \text{if } x > 0.05 \end{cases}$$

$$F_m := F_{7.141}(r_{max}) \qquad F_m = 1.000$$

The tubesheet design pressure, P_{ts}, is the the hydrostatic design pressure modified by Reference 12, Paragraphs R-7.153 and R-7.154. Calculate the modified tubesheet design pressure. This requires equivalent differential expansion pressure from R-7.151 and the equivalent bolting pressure from R-7.152.

Equivalent Differential Expansion Pressure—R-7.151

Calculate the equivalent differential thermal expansion pressure.

$J \equiv 1$ Shell has no expansion joint

The tube (sub t) and shell (sub s) differential temperatures for thermal growth are:

$\Theta_t = (T_{avg.t} - 70 \cdot F)$ $\qquad \Theta_t = 30.57\,F$

$\Theta_s = T_{avg.sw} - 70 \cdot F$ $\qquad \Theta_s = 36.5\,F$

where $\quad T_{avg.t} = 100.57\,F \qquad T_{avg.sw} = 106.5\,F$

Calculate the constant K.

$$K = \frac{E_s \cdot t_s \cdot (od_s - t_s)}{E_t \cdot ta_{Tube} \cdot N_t \cdot (od_{Tube} - ta_{Tube})} \qquad K = 0.734$$

where $\quad E_s = 2.934 \times 10^7\,psi \qquad od_s = 66\,in \qquad E_t = 1.538 \times 10^7\,psi$

$\quad ta_{Tube} = 0.035\,in \qquad N_t = 3400 \qquad od_{Tube} = 0.75\,in$

Guess that the required tubesheet thickness is $t_{r.ts} = 1.966\,in$. Calculate the constant F_q.

$$F_q := max\left[1, 0.25 + (F_m - 0.6) \cdot \left[\frac{300 \cdot t_s \cdot E_s}{K \cdot L \cdot E_{ts}} \cdot \left(\frac{G_c}{t_{r.ts}}\right)^3\right]^{\frac{1}{4}}\right] \qquad F_q = 5.243$$

where $\quad L = 304.5\,in \qquad E_{ts} = 2.934 \times 10^7\,psi \qquad G_c = 65\,in$

The differential thermal expansion equivalent pressure is:

$$P_d := \frac{4 \cdot J \cdot E_s \cdot t_s \cdot (\alpha_s \cdot \Theta_s - \alpha_t \cdot \Theta_t)}{(od_s - 3 \cdot t_s) \cdot (1 + J \cdot K \cdot F_q)} \qquad P_d = 12.57\,psi$$

where $\quad \alpha_s = 5.73 \times 10^{-6} \frac{in}{in \cdot F} \qquad \alpha_t = 4.65 \times 10^{-6} \frac{in}{in \cdot F}$

Equivalent Bolting Pressure—R-7.152

Since there is no bolting on the tubesheet periphery:

$$P_{Bt} \equiv 0 \cdot psi$$

$$P_{Bs} \equiv 0 \cdot psi$$

Effective Shell-side Design Pressure—R-7.153

$$f_s \equiv 1 - N_t \cdot \left(\frac{od_{Tube}}{G_c}\right)^2 \qquad f_s = 0.5473$$

$$D_J \equiv G_c$$ Shell has no expansion joint

$$P_{s'} := P_s \cdot \left[\frac{0.4 \cdot J \cdot \left[1.5 + K \cdot \left(1.5 + f_s\right)\right] - \left[\left(\frac{1-J}{2}\right) \cdot \left[\left(\frac{D_J}{G_c}\right)^2 - 1\right]\right]}{1 + J \cdot K \cdot F_q}\right] \qquad P_{s'} = 37.16\,psi$$

where $P_s = 150\,psi$

$$P_{al} := \begin{pmatrix} \left|\frac{P_{s'} - P_d}{2}\right| \\ \left|P_{s'}\right| \\ \left|P_{Bs}\right| \\ \left|\frac{P_{s'} - P_d - P_{Bs}}{2}\right| \\ \left|\frac{P_{Bs} + P_d}{2}\right| \\ \left|P_{s'} - P_{Bs}\right| \end{pmatrix} \qquad P_{al} = \begin{pmatrix} 12.3 \\ 37.16 \\ 0 \\ 12.3 \\ 6.28 \\ 37.16 \end{pmatrix} psi$$

$$P_{shell} := \max\left(P_{al}\right) \qquad P_{shell} = 37.16\,psi$$

Effective Tube-side Design Pressure—R-7.154

Note: The formulas in this section assume $P_{s'}$ is positive, i.e., there is no shell expansion joint and J=1.

$$f_t \equiv 1 - N_t \cdot \left(\frac{od_{Tube} - 2 \cdot t_{a_{Tube}}}{G_c}\right)^2 \qquad f_t = 0.6279$$

$$P_{t'} := P_t \cdot \left[\frac{1 + 0.4 \cdot J \cdot K \cdot (1.5 + f_t)}{1 + J \cdot K \cdot F_q}\right] \qquad P_{t'} = 64.01\ psi$$

$$P_{a2} := \begin{pmatrix} \left|\frac{P_{t'} + P_{Bt} + P_d}{2}\right| \\ \left|P_{t'} + P_{Bt}\right| \end{pmatrix} \qquad P_{a2} = \begin{pmatrix} 38.29 \\ 64.01 \end{pmatrix} psi$$

$$P_{tube} := max(P_{a2}) \qquad P_{tube} = 64.01\ psi$$

Design Pressure for Tubesheet (see definition of P in Reference 12, 7.122)

$$P_{ts} := max(P_{tube}, P_{shell}) \qquad P_{ts} = 64.01\ psi$$

Tubesheet Thickness

$$t_{bend} := \frac{F_m \cdot G_c}{2} \cdot \sqrt{\frac{P_{ts}}{S_{ts}}} \qquad t_{bend} = 1.97\ in$$

where $S_{ts} = 17.5\ ksi$

Drawings are not available to calculate the parameter D_L. Approximate D_L using the area and perimeter of a circle with diameter G_c. This is a conservative approach, since the ID of the channel head is larger than the tube bundle diameter. Also, using a circle with diameter G_c to calculate the perimeter is likely conservative since it is judged to be smaller than the tube bundle perimeter as shown in Reference 12, Figure R-7.123.

MPR
MPR Associates, Inc.
320 King Street
Alexandria VA 22314

Prepared By: J Hibbard

Checked By: [signature] FOR R. PARKERSON

$$D_L = 4\cdot\frac{\frac{\pi}{4}\cdot G_c^2}{\pi\cdot G_c} \qquad D_L = 65\,in$$

$$t_{shear} := \frac{0.31\cdot D_L}{1-\frac{od_{Tube}}{p_t}}\cdot\frac{P_{ts}}{S_{ts}} \qquad t_{shear} = 0.29\,in$$

where $p_t = 1\,in$

$$t_{req.ts} := max(t_{bend}, t_{shear}) \qquad t_{req.ts} = 1.97\,in$$

Verify that the guessed value and the calculated value meet the criterion of Reference 12, Paragraph 7.151, variable T definition.

$$\left|\frac{t_{req.ts} - t_{r.ts}}{t_{req.ts}}\right| < 1.5\cdot\% = 1$$

A value of 1 indicates the criterion is met.
A value of 0 indicates the criterion is not met.

The actual tubesheet thickness is $t_{ts} = 3.5\,in$. Add the results to the summary table T1.

$$T1 := stack(T1, augment("Tubesheet", t_{req.ts} \div in, t_{ts} \div in, if(t_{req.ts} \le t_{ts}, ok, nok)))$$

4.10 Seismic Evaluation

Channel Cover, Cover Bolting, and Channel Flange

Reference 5, Paragraph 7.B.1 determined that the equivalent pressure load on the cover from seismic acceleration is $P_{eq.ch} = 2.4\ psi$. This is $P_{eq.ch} \div P_t = 1.3\,\%$ of the new design pressure of $P_t = 191\ psi$. It is concluded that the seismic loading on the channel cover, cover bolting, and channel flange is negligible and that no further evaluation is required.

Channel Head and Manway Cylinder

Section 4.2 of this calculation determines the channel and manway required wall thickness for the design pressure. The limiting stress in determining the minimum required wall thickness is the hoop stress. Seismic acceleration produces longitudinal membrane and bending stresses. It is concluded that the seismic acceleration does not affect the minimum wall thickness calculation of Section 4.2 and that no further evaluation of the channel or manway cylinder for seismic acceleration is required.

Manway Cover, Manway Bolting, and Manway Flange

Reference 5, Paragraph 7.B.2 determined that the equivalent force on the manway cover from seismic acceleration is $F_{eq.m} = 540\ lbf$. This is an equivalent pressure of:

$$P_{eq.m} := \frac{F_{eq.m}}{\frac{\pi}{4}\cdot\left(G_{manway}\right)^2} \qquad P_{eq.m} = 1.61\ psi$$

where $G_{manway} = 20.634\ in$

This equivalent seismic pressure is $P_{eq.m} \div P_t = 0.8\,\%$ of the new design pressure of $P_t = 191\ psi$. It is concluded that the seismic loading on the manway cover, manway cover bolting, and manway flange is negligible and that no further evaluation is required.

MPR MPR Associates, Inc. 320 King Street Alexandria VA 22314	Prepared By: J Hubbard Checked By: [signature] FOR R. PARKERSON	Calculation No.: 0108-0309-jth-1 Revision No.: 0 Page No.: 37

5.0 REFERENCES

1. 1968 ASME Code, Section VIII, Division 1.
2. PSE&G VTD No. 301110, 7/25/91, Auxiliary Heat Exchangers Westinghouse Instruction Manual.
3. Greenkorn and Kessler, "Transfer Operations," McGraw-Hill Book Company, 1972.
4. PSE&G VTD No. 111008-05, EFCO Drawing No. NEN-15763, "Shell, Chan. & Support Details."
5. PSE&G VTD No. 316898-01, "ASME Code Calculations for Component Cooling Heat Exchanger Modification No. 1 Unit," Original Issue.
6. Garlock Gasket Product Catalog, available at web site: http://www.garlock.net/upload/catalogs/GSK%203-1%20Gasket%20catalog.pdf.
7. E. Avallone & T. Baumeister, "Marks' Standard Handbook for Mechanical Engineers," McGraw-Hill Book Company, 9th Edition.
8. ANSI B16.5 (1961), "Steel Pipe Flanges and Flanged Fittings."
9. PSE&G VTD No. 320478, Equipment Specification No. G-676454, "Auxiliary Heat Exchangers," Revision 1.
10. PSE&G VTD No. 117638-04, EFCO Drawing No. NEN-B-15763, "Tubesheets.".
11. 1992 ASME Code, Section II, Materials, Part D, Properties.
12. TEMA, "Standards of Tubular Exchanger Manufacturers Association," 5th Edition, 1968.
13. Form U-1, Manufactured by Engineers and Fabricators, Vessel No. S-15763-A, National Board No. 1148.
14. 2001 ASME Code, Section II, Materials, Part D, Properties.

10 CFR 50.55a Request Number SC-RR-W03

Proposed Alternative
In Accordance with 10 CFR 50.55a(a)(3)(i)
Alternative Provides Acceptable Level of Quality and Safety

Enclosure 2

S-C-SW-MEE-1882, Salem SW Heat Exchangers - Suitability for Operation at Higher Pressures, Revision: 0, **Attachment D**, MPR Calculation 0108-0.309-jem-1, "*Component Cooling Water Shell and Tube Heat Exchanger Channel Analysis,*" Revision dated 1/27/050.

Attachment D: MPR Calculation 0108-0309-jem-1, "Component Cooling Water Shell and Tube Heat Exchanger Channel Analysis," Revision 0

RFH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

CALCULATION TITLE PAGE

Client: PSEG Nuclear	Page 1 of 15 (+Attachments)
Project: Salem SW HX Re-Rate	Task No. 0108-0418-0309-00
Title: Component Cooling Water Shell and Tube Heat Exchanger Channel Analysis	Calculation No. 0108-0309-JEM-1

Preparer / Date	Checker / Date	Reviewer & Approver / Date	Rev. No.
James Moroney 12/28/04	Teresa Tetlow 12/28/04	Robert Coward 12/28/04	0

QUALITY ASSURANCE DOCUMENT
This document has been prepared, checked, and reviewed/approved in accordance with the Quality Assurance requirements of 10CFR50 Appendix B, as specified in the MPR Quality Assurance Manual.

RSH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

RECORD OF REVISIONS

Calculation No.	Prepared By	Checked By	Page: 2
0108-0309-JEM-1	[signature]	J. K. Jetlow	

Revision	Affected Pages	Description
0	All	Initial Issue

Note: *The revision number found on each individual page of the calculation carries the revision level of the calculation in effect at the time that page was last revised.*

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 3
0108-0309-JEM-1		J. K. Tellow	Revision: 0

Table of Contents

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 4
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

1.0 Purpose

This calculation documents a finite element analysis of the Service Water side of the Component Cooling (CC) Heat Exchangers Nos. 11, 21, and 22 at Salem. PSEG Nuclear is currently re-evaluating the Salem Service Water (SW) system heat exchangers for an increase in design pressure from 150 psig to 200 psig. This analysis is being performed as part of that effort. This elastic-plastic finite element analysis calculates the collapse pressure of the heat exchanger channel.

2.0 Summary of Results

Using the methodology of Reference 7, Appendix 4, 4-136.5, the maximum allowable pressure for the SW side of CC Heat Exchangers Nos. 11, 21, and 22 is 267 psig. This is not a certification that the Heat Exchangers meet all applicable ASME B&PV code requirements.

3.0 Finite Element Model

A three-dimensional finite element model of the CC heat exchangers "A" channel has been prepared to calculate the collapse load of the heat exchanger head. Note that the "A" channel is larger than the "B" channel. Therefore the results of this analysis are bounding for both channels. While this calculation uses a methodology described in the ASME B&PV, it does not certify that the CC heat exchanger is in compliance with all ASME B&PV requirements.

The finite element analysis was performed using the ANSYS general purpose finite element computer program Version 8.1 on a Sun Microsystems 280R server running the Solaris 8 operating system. The ANSYS installation verification is documented in QA-81-1

3.1 Model Geometry

The model geometry is taken from References 1 through 3. Channel "A" is modeled along with approximately 2 feet of the heat exchanger shell. Note that this section of shell is not being evaluated, but is included for boundary conditions only. In addition, the cover and nozzle flanges are conservatively not included in the model. Scoping evaluations indicated that these components do not significantly impact the behavior of the channel shell. No credit is taken for the structural strength of the tubesheet clad.

The model is a three-dimensional half model, with the axis of symmetry at the along the heat exchanger axial length at the nozzle centerline. Figure 3-1 shows the model geometry. The model is meshed with 3-D solid elements with 10 nodes (ANSYS Element SOLID45). Figure 3-2 shows the model mesh.

RSH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 5
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

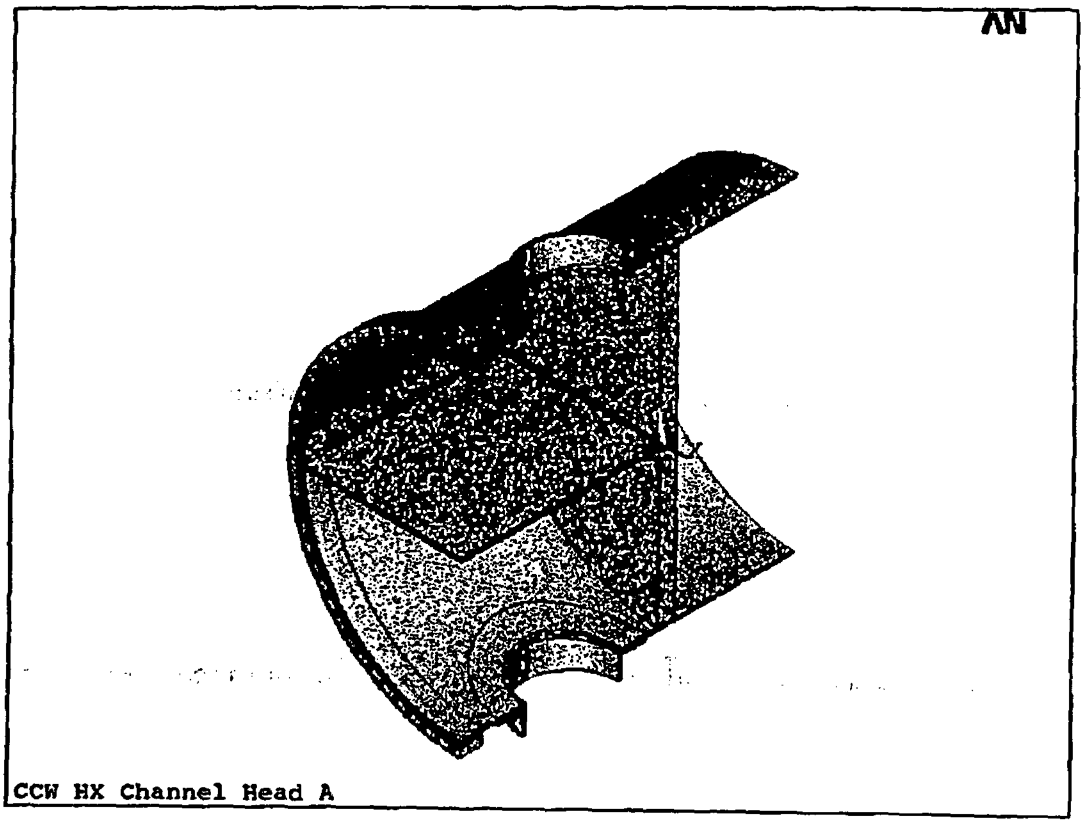

Figure 3-1. Model Geometry

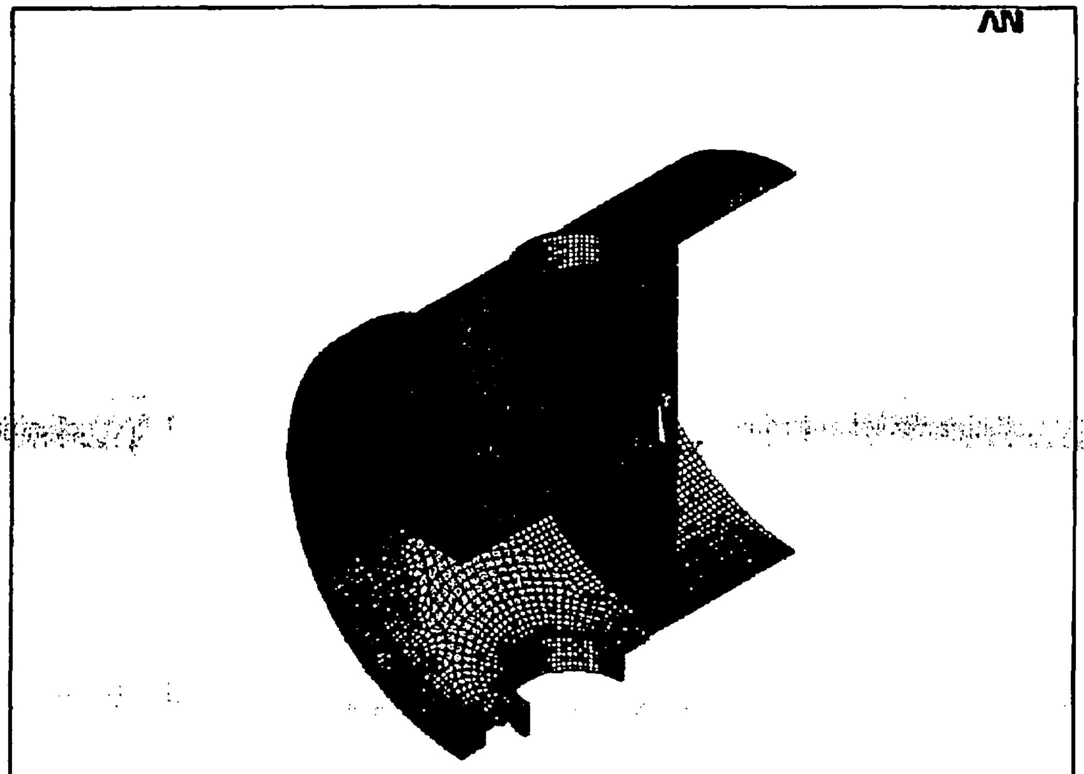

Figure 3-2. Model Mesh

RSA -2-2-05

MPR
MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 7
0108-0309-JEM-1	JEM	J. K. Titlow	Revision: 0

3.2 *Material Properties*

Table 3-1 shows the component material properties used in this analysis. All component materials are taken from References 1 and 3.

Table 3-1. Model Materials

Component	Material
Channel	SB-171 90-10 Cu-Ni
Nozzles	
Partition Plate	
Channel Flange	SA-105-II
Tubesheet	SA-515 Grade 70
Heat Exchanger Shell	
Nozzle Repad	

To properly account for the redistribution of load and potential concentration of strain for loads that produce stresses beyond the elastic limit, elastic-plastic material properties were included in this analysis. The behavior of the channel shell is of particular interest. Therefore, a stress-strain curve was developed for SB-171 90-10 Cu-Ni. Bi-linear stress-strain curves were used for the carbon steel portions of the model (SA-150-II and SA-515 Gr. 70). Key material properties used in the stress-strain curves are shown in Table 3-2 (Reference 4).

Table 3-2. Key Material Properties

Material	Yield Strength (ksi)	Strain at Yield Strength (%)	Ultimate Tensile Strength (ksi)	Maximum Elongation (%)
SB-171 90-10	15	0.5	40	30
SA-105-II	36	0.5	70	30
SA-515 Gr. 70	38	0.5	70	21

The stress-strain curve developed for SB-171 is based a standard stress-strain curve for SB-171 90-10 Cu-Ni (Reference 5). The following adjustments were made to this curve:

- The yield strength point was set at 15 ksi and 0.5% strain.
- The ultimate tensile strength (UTS) point was set at 40 ksi and 30% strain. All points on the curve between the yield strength and UTS points were scaled down based on the ratio

of the curve UTS (49 ksi) and the minimum UTS (40 ksi) specified in Reference 4. The scaling equation used is:

$$S_{adj} = S_y + (S_{curve} - S_y) * \left(\frac{S_{u\text{-}min} - S_y}{S_{u\text{-}curve} - S_y} \right)$$

- All points on the curve were adjusted for temperature. The material properties shown in Table 3-2 and the stress-strain curve from Reference 5 are based on room temperature, while the CC heat exchanger design temperature is 200 °F (Reference 6). The adjustment factor used is 0.98, which is based on the allowable stress intensity values for SB-171 90-10 Cu-Ni at 100 °F (10 ksi) and 200 °F (9.8 ksi), taken from Reference 4, Section VIII, Table UNF-23.

- All points on the curve were adjusted to account for a channel shell joint efficiency of 0.85 (Reference 1).

The digitized version of the initial stress-strain curve taken from Reference 5 and the three scaled curves discussed above are shown in Figure 3-3. The stress-strain curve for SB-171 90-10 Cu-Ni used in this analysis is shown in Figure 3-4.

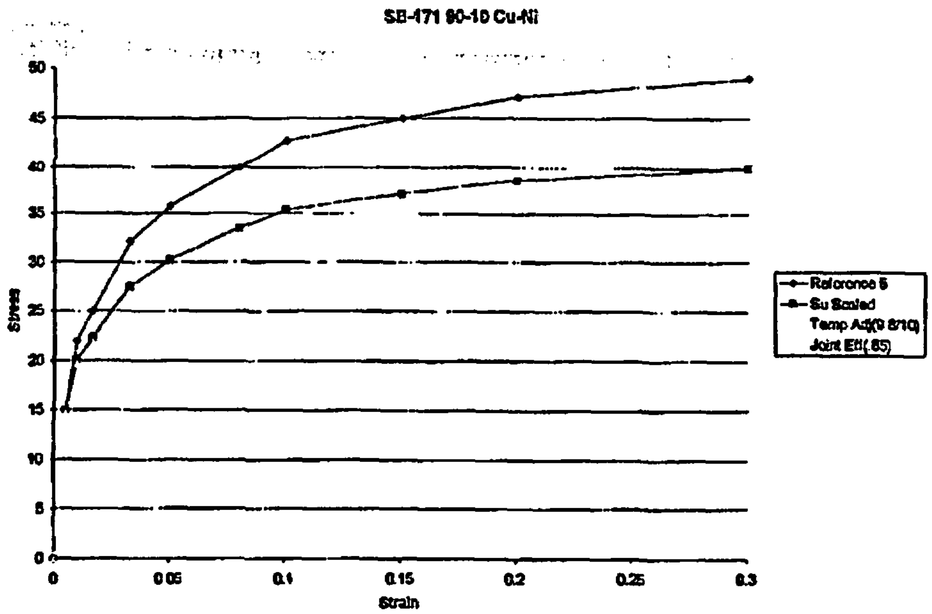

Figure 3-3. Stress-Strain Curve Scaling

RFI 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 9
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

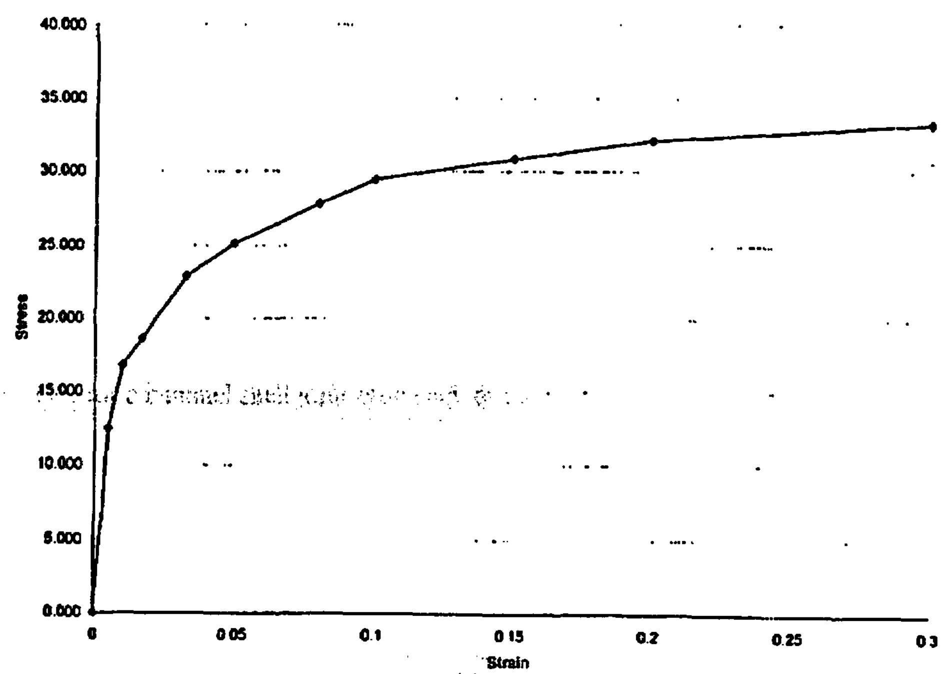

Figure 3-4. Stress-Strain Curve for SB-171 90-10 Cu-Ni

3.3 Boundary Conditions

Figure 3-5 shows the model boundary conditions. The model is constrained at the following locations:

- Transverse (X-axis) motion is constrained at the model plane of symmetry.
- Longitudinal (Z-axis) motion is constrained at the heat exchanger shell.
- The edge of the partition plate engaged with the cover is constrained against vertical (Y-axis) motion.
- The nozzle end nodes are coupled vertically (Y-axis) to ensure they remain in-plane vertically.

RJH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 10
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

Hydrostatic end loads are applied to the nozzles and the channel flange. The hydrostatic end load is calculated as follows:

$$F_{noz} = P_{des} \pi r_i^2$$

where:

P_{des} = Heat Exchanger Internal Pressure

r_i = Nozzle inside radius (Reference 1)

One half of this load (due to half-model) is applied as a pressure to the end of each nozzle and the channel flange.

The internal heat exchanger pressure is applied to all of the internal surfaces of the channel. The internal pressure applied to the tubesheet is modified to account for the missing volume of the tube penetrations as well as the stiffness of the tubes. Reference 8 calculates that the effective tubesheet pressure based on an internal pressure of 200 psi is 62 psi. This objective of this analysis is to calculate the collapse load of the heat exchanger channels. Therefore, a maximum pressure of 400 psi, which exceeds the re-rate design pressure of 200 psi, is applied. At an internal pressure of 400 psi, an effective pressure of 62*(400/200)=124 psi will be applied to the tubesheet.

RSA 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 11
0108-0309-JEM-1	JcE	J. K. Jitlow	Revision: 0

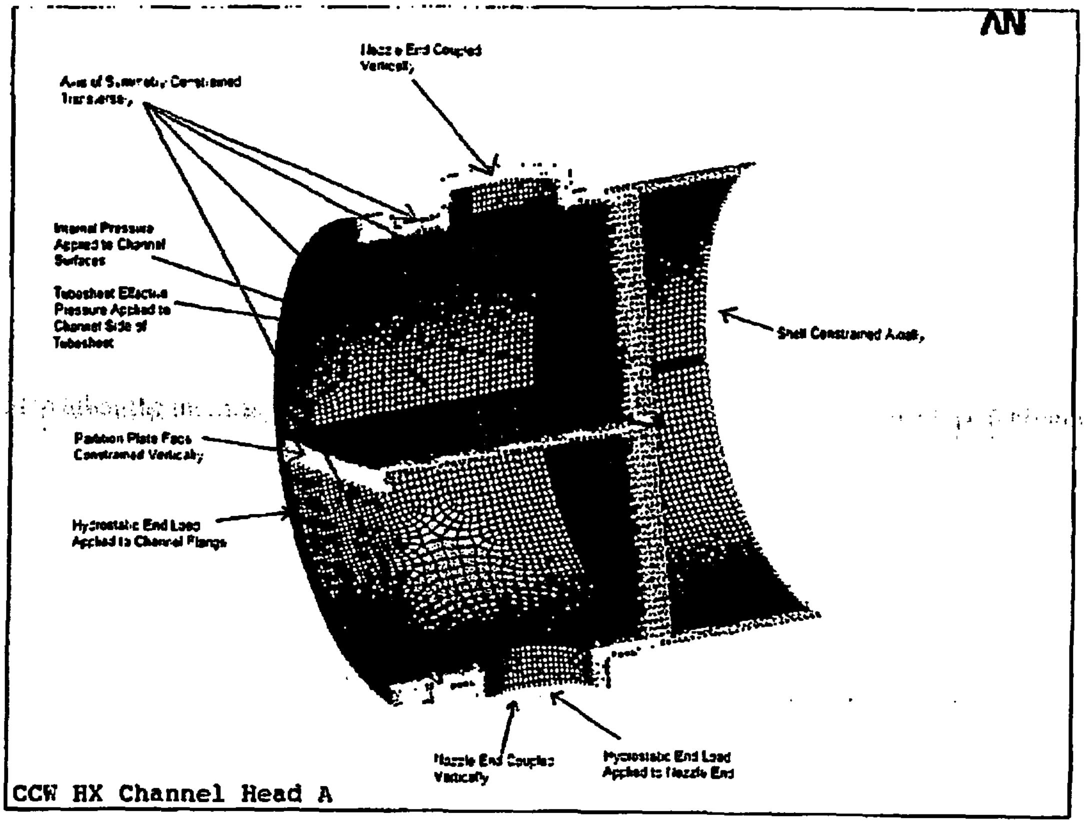

Figure 3-5. Model Boundary Conditions

Calculation No.	Prepared By	Checked By	Page: 12
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

4.0 Analysis

This calculation performs a Plastic Analysis of the CC heat exchanger channel using the methodology specified in Reference 7, Appendix 4, 4-136.4. This analysis does not attempt to certify that the heat exchanger meets the requirements of the ASME B&PV. Using the full stress-strain curve discussed above, a collapse load is calculated using the methodology described Reference 7, Appendix 6, 6-153. As the internal pressure is ramped up to a final arbitrary load of 400 psi, the deflection of the channel shell at a location remote from discontinuities is plotted. From this deflection vs. load data, a collapse limit line is plotted. The intersection of the deflection curve and collapse limit line is the collapse pressure. Per Reference 7, 4-136.4, the maximum loading of the component shall not exceed two-thirds of the plastic collapse load.

With the deflection vs. load plotted with the load on the ordinate, the collapse limit line is determined by first calculating the angle between the elastic portion of the deflection curve and the ordinate (θ). The angle between the collapse limit load and the ordinate (Φ) is calculated as follows:

$$\Phi = \tan^{-1}(2\tan\theta)$$

4.1 Analysis Results

Figures 4-1 and 4-2 show the stress intensity distribution of the channel at 200 psi and 400 psi, respectively. Review of these stress distributions indicates that the channel shell stress is primarily membrane stress and that no plastic hinges have been formed.

Figure 4-3 shows the deflection vs. load curve and the collapse limit line calculated in accordance with Reference 7, Appendix 6, 6-153 (See Appendix A for data). This data is taken at a point approximately midway between the tubesheet and flange (axially) and midway between the partition plate and upper nozzle repad. This is where the maximum shell displacement occurs. The deflection vs. load curve on Figure 4-3 had not yet intersected with the collapse limit line at a pressure of 400 psi. Based on this, it is conservative to assume that the plastic collapse load is at least 400 psi. Given a plastic collapse load of 400 psi, the maximum heat exchanger channel load is 2/3 * (400 psi) = 267 psi. The displacement and strain at other locations, including the highest stress region, were also evaluated using this methodology. In all cases, the displacement data and collapse limit lines were essentially identical, supporting the 267 psi maximum heat exchanger channel load.

RSB 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 13
0108-0309-JEM-1	J. E. M.	J. K. Jettow	Revision: 0

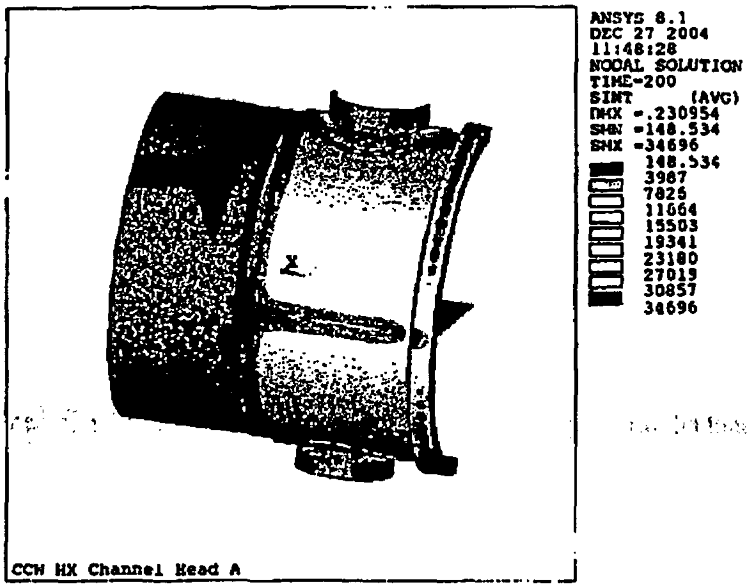

Figure 4-1. Stress Intensity Distribution at 200 psi

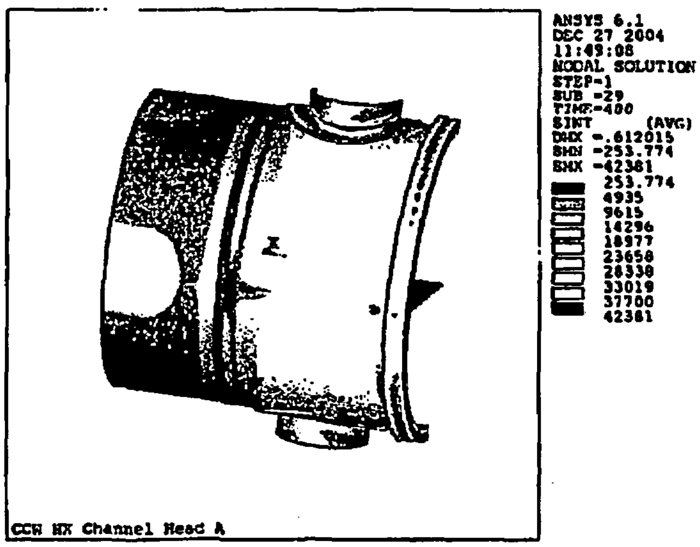

Figure 4-2. Stress Intensity Distribution at 400 psi

RSH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 14
0108-0309-JEM-1	[signature]	J. K. Tetlow	Revision: 0

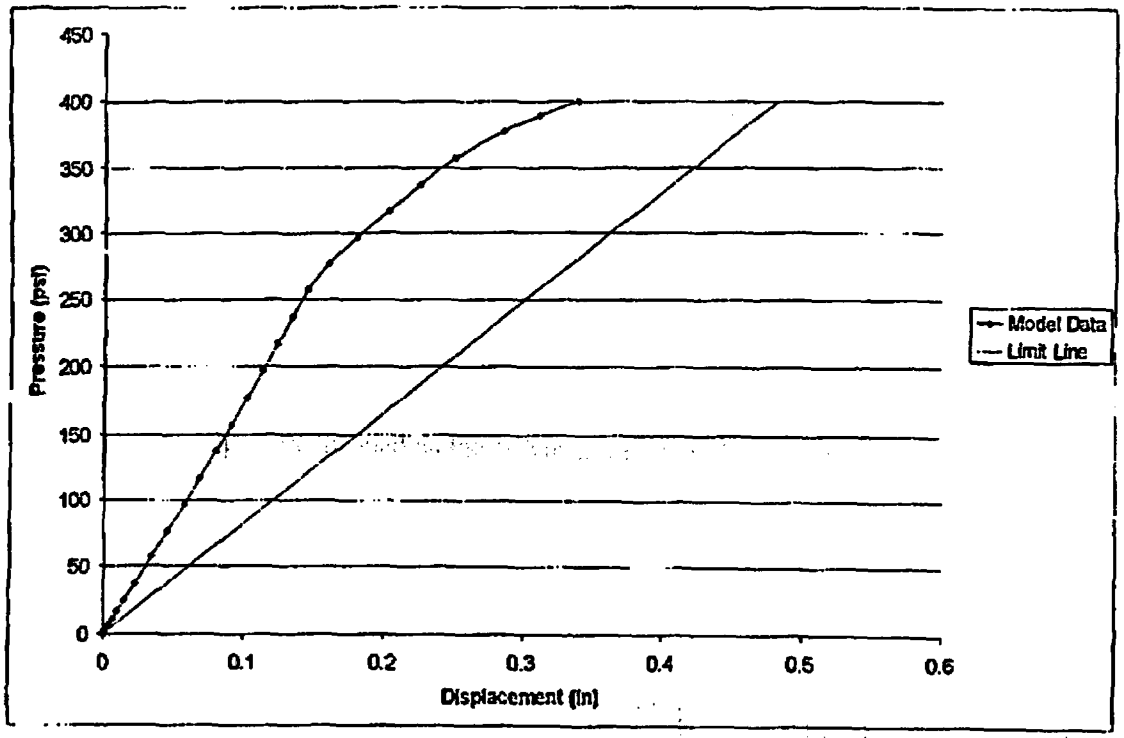

Figure 4-3. Deflection vs. Pressure Data (Including Collapse Limit Line)

RSH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: 15
0108-0309-JEM-1	[signature]	J. K. Jettlows	Revision: 0

5.0 References

1. PSE&G VTD No. 111008-05, EFCO Drawing NEN-15763, "Shell, Chan & Support Details," Revision K.

2. PSE&G VTD No. 108724-08, EFCO Drawing CD-15763, "Component Cooling Heat Exchangers," Revision G.

3. PSE&G VTD No. 117638-04, EFCO Drawing NEN-B-15763, "Tube Sheets," Revision B.

4. ASME Boiler and Pressure Vessel Code, 1968 Edition.

5. Boyer, Howard E., Atlas of Stress-Strain Curves, Figure 16-71, ASM International, 1987.

6. PSE&G VTD No. 30110, 7/25/91, Auxiliary Heat Exchangers Westinghouse Instruction Manual.

7. ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 2004 Edition.

8. MPR Calculation 0108-0309-JLH-1, "Component Cooling Shell and Tube Heat Exchanger Service Water Pressure Rerate Evaluation – 1CCE5, 2CCE5, & 2CCE6," Revision 0.

RFH 2-2-05

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: A-1
0108-0309-JEM-1	[signature]	J. K. Jetlow	Revision: 0

A

Plastic Analysis Deflection Data

Table A-1 shows the deflection vs. load data taken from the ANSYS collapse load analysis.

Pressure (psi)	Displacement (in)
0.50	0.0003
1.00	0.0006
1.75	0.0011
2.88	0.0018
4.56	0.0028
7.09	0.0043
10.89	0.0066
16.59	0.0101
25.13	0.0152
37.94	0.0228
57.17	0.0342
77.17	0.0459
97.17	0.0574
117.17	0.0688
137.17	0.0800
157.17	0.0911
177.17	0.1021
197.17	0.1129
217.17	0.1235
237.17	0.1344
257.17	0.1452
277.17	0.1599
297.17	0.1807
317.17	0.2028
337.17	0.2259
357.17	0.2499
377.17	0.2851
388.58	0.3109
400.00	0.3386

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: A-2
0108-0309-JEM-1	[signature]	J. K. Tetlow	Revision: 0

Using this data, the slope of the elastic portion of the deflection line is calculated:

$$m_{data} = \frac{P}{\delta} = \frac{37.94\,psi}{0.0226\,in} = 1675.2\,lb/in$$

The angle (θ) between the ordinate and the elastic portion of the deflection line is:

$$\theta = 90 - \tan^{-1}(m_{data}) = 0.034^{\circ}$$

The angle between the ordinate and the collapse limit line is:

$$\Phi = \tan^{-1}(2\tan(\theta)) = 0.068^{\circ}$$

MPR

MPR Associates, Inc.
320 King Street
Alexandria, VA 22314

Calculation No.	Prepared By	Checked By	Page: B-1
0108-0309-JEM-1	[signature]	J. K. Tetlow	Revision: 0

B

ANSYS Output Files

Filename	Date	Time
Head.out	11/17/2004	12:57
Mmshcoarseep.out	12/20/2004	18:25
Headep.out	12/22/2004	19:07
Pst.out	12/27/2004	11:43