MPR

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

CALCULATION TITLE PAGE

Client: Quad Cities 1 & 2	Page 1 of 8
Project: QC 2 ERV Root Cause Emergent Support	Task No. 1101-0604-0009-01
Title: ASME Code Evaluation for Vibration Testing of ASBs	Calculation No. 1101-0009-HDG-01

Preparer / Date	Checker / Date	Reviewer & Approver / Date	Rev. No.
Hans D. Giesecke 4-13-2006 H. Giesecke	4-13-2006 A. Limaye	4/13/2006 R. Coward	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
1101-0009-HDG-01	Hans D. Puesch	[illegible]	

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.*

Table of Contents

1.0 Background

At the Quad Cities nuclear power plant, acoustic modes are causing the branch lines (standpipes) that run from each main steam line to the Electromatic Relief Valves (ERV) and the Safety Relief Valves (SRV) to vibrate. The level of vibration increases significantly at high power levels associated with Extended Power Uprate (EPU) operating conditions. An Acoustic Side Branch (ASB) is being installed on each standpipe in an attempt to mitigate the vibration by suppressing the acoustic mode. The ASB is a 6" diameter pipe that is either 24" long or 30" long depending on the valve, closed off by a blind flange at the free end, and would branch off the vertical standpipe to the valve. The ASB contains a steel canister that is packed with circular stainless steel screens to provide resistance to steam flow. The purpose of the ASB is to mitigate the acoustic mode that exists in the particular SRV or ERV branch line when the flow rate in the main steam line is high.

2.0 Purpose

This purpose of this calculation is to determine the magnitude of vibration that would be considered acceptable during power operation in the plant based on the magnitude and duration of vibration testing conducted on the ASB at the Quanta Laboratories in Santa Clara, California. The calculation method is based on the test requirements provided in Reference 1, and the test vibration levels are provided in Reference 2.

3.0 Summary of Results

The acceptance criteria for vibration are based on the guidance provided in Reference 1 and can be summarized as follows:

- The vibration acceleration amplitude in the axial direction of the ASB should be limited to 1.6 g RMS if it is measured on the blind flange end of the ASB or 1.5 g RMS if it is measured on the pipe end of the ASB.

- The vibration acceleration amplitude in the transverse direction should be limited to 2.3 g RMS if measured on the pipe end of the ASB. The acceleration in the axes transverse to the ASB should be combined as the square root of the sum of the squares (SRSS) and the resultant should be less than 2.3 g RMS.

- The ASBs can be operated at the above limits for at least an operating cycle (24 months). Based on the fatigue calculations, there is very little difference with respect to operating 2 years and 20 years.

4.0 Analysis Approach

The ASBs were tested on vibration tables at the Quanta Laboratories in Santa Clara, California. Two separate test tables were used; one for vertical vibration and one for horizontal vibration. The ASBs were mounted in a vertical orientation so the vertical table provided an axial excitation whereas the horizontal table provided a transverse excitation. Since the ASB is symmetric about its axis, only one horizontal direction was tested.

The tests in Reference 2 included separate vibration tests in the transverse direction and in the axial direction for an ASB identical to those that will be installed in the plant. The prototype that is to be installed in the plant was tested in each of these directions for a total of 80 hours (Reference 2) and the magnitude of the vibration for the screens at the point of highest vibration was 4.26 g RMS at the base of the ASB for the horizontal test and 3.0 g RMS at the flanged end of the ASB for the vertical test (Reference 2). The screen meshes inside the ASB canisters are constructed of stainless steel and testing has demonstrated that the only failure mode that could be of consequence is the wire mesh failing in fatigue. The concern is that the wire mesh fragments could become lodged in critical components in the main steam system.

Representing the fatigue curve for stainless steel from Figure I-9.2.2, curve C (Reference 3), as $S_a(N)$, where S_a is the allowable stress as a function of the number of stress cycles, N, a test fatigue curve is obtained as $S_{at}(N)$.

$$S_{at}(N) = K_s S_a(N)$$

where: K_s is a factor applied to the design fatigue to obtain the test fatigue curve and is comprised as a product of the five individual factors described below.

- K_{sl} factor for the effect of size on fatigue life
- K_{sf} factor for the effect of surface finish
- K_{st} Factor for the effect of test temperature
- K_{ss} Factor for the statistical variation of test results = 1.470-0.044 x the number of replicate tests
- K_{sc} Factor for differences in design fatigue curves at various temperatures

Using the actual equipment for the test, the factors K_{sl}, K_{sf} and K_{sc} are equal to unity. The screen material is either 304 or 316 stainless steel per Reference 4. Since the fatigue curves are strain controlled, the stress is directly proportional to Young's moduls. Therefore K_{st} is calculated as the ratio of Young's modulus for stainless steel at room temperature divided by Young's modulus at operating temperature or $30x10^6/25x10^6$ or 1.2. Since only a single test is used, $K_{ss} = 1.426$. The result is:

$$K_s = 1.711$$

The resulting fatigue curves are shown in Figure 1, where the lower curve is the design curve from Reference 1 and the upper curve is the test curve.

According to Reference 1, the minimum number of test cycles that are required to avoid fatigue within a specific number of service cycles is given by the formula:

$$N_{T\min} = 10^2\sqrt{N_D}$$

where: N_D are the number of service cycles required, which are assumed to be $1.0x10^{11}$.

$$N_{T\min} = 4.32x10^7$$

From the curves in Figure 1, S_{at} at N_T is 24.82 Ksi (point C) and S_a at N_D is 13.6 Ksi (point D). Therefore, K_{TS} will be:

$$K_{TS} = \frac{S_{at}(N_T)}{S_a(N_D)} = 1.825$$

The allowable stress for operation is the stress at which the test took place divided by K_{TS}. Since the stress in the wire mesh is proportional to the acceleration, the acceleration will scale similarly to the stress. The stress on a particular screen will depend on the local acceleration for that screen. Therefore, the acceleration used in the analysis is the peak acceleration for the ASB. For the transverse vibration, the highest acceleration occurred on the open end of the canister and since no screen fragments were found there, no significant fatigue failures occurred during the testing. For the axial vibration, the ASB is very stiff and there is little difference in the response at the two ends. The measured value from the flange end was 3.0 g RMS whereas the measured end from the pipe end was 2.73 g RMS. Therefore, the acceptable acceleration during operation of the plant will be as shown in Table 1.

Table 1. Calculation of Allowable Acceleration During Plant Operation

Direction	Location of Maximum Acceleration	Test Acceleration (g's)	K_{TS}	Allowable Acceleration (g's)
Transverse	Pipe End	4.26	1.825	2.3
Axial	Flange End	3.00	1.825	1.6
Axial	Pipe End	2.73	1.825	1.5

5.0 REFERENCES

1. 2004 ASME Boiler and Pressure Vessel Code , Section III, Division 1 – Mandatory Appendix II-1500, Cyclic Tests.

2. 06Q4568-DR-004, "Quad Cities Acoustic Side Branch (ASB) Vibration Test Report", Rev 2 dated April 9, 2006.

3. 2004 ASME Boiler and Pressure Vessel Code , Section III, Division 1 – Mandatory Appendix I, Figure I-9.2.2.

4. Exelon Nuclear drawing number SK-005, Sheet 1, Revision C, entitled "Fabrication Details ASB Canister 30" Effective Length Main Steam SRV Application".

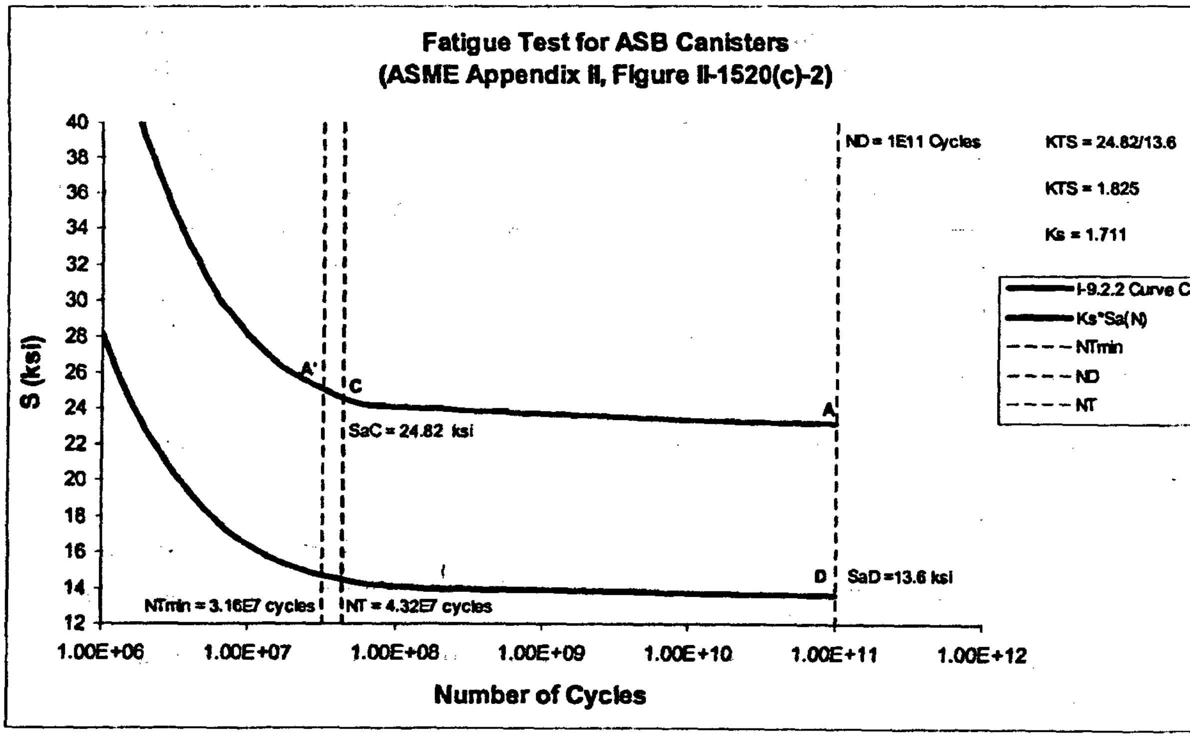

Figure 1. Construction of Testing Parameters Ratio Diagram

Attachment 9

Request 9(b) – GENE 0000-0053-2954, Revision 1

Request for Additional Information: Quad Cities Unit 2 Dryer Inspection, Start-up and Power Ascension Plan

RAI 9 (b): For the large dryer crack in the skirt base metal at 135-degree location, provide the following information and documentation supporting your responses.

(b) What may be the corresponding reduction in the fatigue stress limits?

Response: As discussed in earlier documents, the 135-140 degree location was subjected to significant deformation during the steam dryer's installation in 2005. Finite element analysis using a solid model was able to show that the cutout location was plastically deformed at least to ~4% strain levels equivalent to ~60 ksi or higher stress when actual nominal L-grade stainless steel properties were used in the analysis (Reference 1)[1]. This permanently strained material would be the equivalent to highly strained material with a corresponding high residual mean stress. These stress levels exceed the mean stress level that was considered in developing the high cycle regime fatigue Curve C of the ASME Code. The higher mean stress level is expected to translate into lower level of cyclic stress amplitude needed to produce crack initiation. The classic understanding regarding the effect of mean stress on the stress amplitude that will lead to fatigue cracking has led to the use of different formulations to calculate effect. Goodman, Soderberg and others have developed these relationships, all of which predict a reduction in the allowable alternating stress amplitude with increasing tensile mean stress (Reference 2).

For the steam dryer, a design value of 13600 psi was used for non-critical locations based on the most conservative ASME Code Curve C. The Code established this curve based on a nominal level of mean stress that may be present from sources such as the weld residual or fit up. The Curve C of the ASME Code used a mean stress of 44 ksi (at room temperature), based on a cyclic stress-strain curve for stainless steel. This calculation is described in the technical basis paper by Manjoine and Tome (Reference 3) that was used by the ASME Code in developing/extending the stainless steel fatigue curves into the very high cycle range (up to 10^{11} cycles). This design curve was derived by applying a factor of 2 reduction to the average fatigue initiation stress determined from laboratory data.

However, given the higher plastic strain and complementary increase in strength of the deformed base ring location, the expected fatigue endurance properties would be significantly reduced due to mean stress effects. This effect can be calculated directly from the equations used by Manjoine, et al. Although the region of interest was cold worked by the installation event, the evaluation of the mean stress effect was performed based on the fatigue properties of annealed

[1] The higher range of the properties seem to be applicable to the base ring material which had a room temperature yield strength > 40 ksi, a value well above the typical L-grade stainless steel value.

material. Therefore, the evaluation should be viewed in qualitative rather than quantitative terms. For conservatism, the loading was considered as stress controlled in the determination of the mean stress effect, i.e. the range of Pl + Pb +Q was assumed to exceed 27.2 ksi. The impact of an assumed residual (mean) stress of 60 ksi would be a 30% reduction in the allowable while the assumption of a 70 ksi yield strength to represent the local mean stress would reduce the allowable by 50%. These levels of reduction in fatigue properties are very likely given the deformation and the constraint imposed by the several intersecting welds present at the base ring cut out corner-solid gusset-skirt region where crack initiation occurred.

In summary, the plastic deformation would be expected to lead to a high residual mean stress. Consistent with the understanding of fatigue behavior in the presence of high mean stresses, the fatigue endurance limit would be reduced. Based on the conservative evaluation, the reduction in endurance limit would be expected to be a maximum of 50%.

References:

1. GENE 0000-0053-2910,"Quad Cities Units 2 Replacement Steam Dryer Analysis Detailed Stress Analysis of Skirt Base Plate Cutout and Gussets," April 2006.
2. Comparison of the ASME Approach for Determining the Influence of Mean Stress, With Data Obtained in High Temperature Water H.D. Solomon, R.E. DeLair and E. Tolksdorf, 2000CRD064, July 2000.
3. Manjoine, M.J. and Tome, R.E., "Proposed Design Criteria for High Cycle Fatigue of Austenitic Stainless Steels," International Conference on Advances in Life Prediction Methods, ASME, 1983, pp. 51-57.