



## HIGH FREQUENCY CUTOFF ANALYSIS FOR EVALUATING VALVES SUBJECTED TO HYDRODYNAMIC LOADS

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### ABSTRACT

In-line and skid mounted components of piping systems such as valves are qualified for static as well as dynamic loads. Seismic loads are generally characterized by frequency content less than about 33 Hz. Hydrodynamic (HD) loads in pressure piping, on the other hand, due to various source mechanisms collectively exhibit a broad range of frequencies much higher than 33 Hz. HD loads may also result in accelerations in excess of those due to the design basis seismic events and allowable limits. Current regulatory guidelines do not specifically address the evaluation of equipment response to high frequency loading.

This paper investigates the response of skid and line mounted valves of piping systems under high frequency (HF) HD loads by using rigorous finite element (FE) analysis solutions with a representative piping system segment. This paper develops a rationale for issues such as frequency cutoff relevant to HD loading and presents an approach for the evaluation of the response of valves to HF HD loads.

### INTRODUCTION

This paper presents an approach for the evaluation of the response of safety-related in-line valves of the AP1000<sup>®</sup> plant standard design for HD loads resulting from valve closure, jet impingement, depressurization, and pipe whip. The HD loads are characterized by high frequency content resulting in relatively small displacements. The maximum response accelerations due to HD force time histories on the other hand could be relatively high at some locations of the system. However, predicted accelerations at valve bodies are associated with very small displacements, high frequencies, short duration, and a low impulse.

The paper develops the dynamic response to the HD loads of the valves located on a selected single representative segment. Figure 1 shows the FE model of the piping segment, including the location of in-line valves. The HD loads stem from a single event and consist of 23 separate force time histories applied simultaneously. HD loads are generally located close to elbows or intersection points with sub-segments. Most HD load time histories exhibit a few short duration pulses with a total duration less than a second within which the HD loads are essentially damped out.

Figure 2-i shows for example one of the 23 HD load time histories designated as THL004 and the corresponding Fourier Spectrum of the time history. As shown in Figure 2-ii, the HD time history is characterized by high frequency content well in excess of seismic motions. In some cases, the time histories exhibit a single peak which damps out in two milliseconds. In some others, HD load time history is defined with a step function which lasts one second.

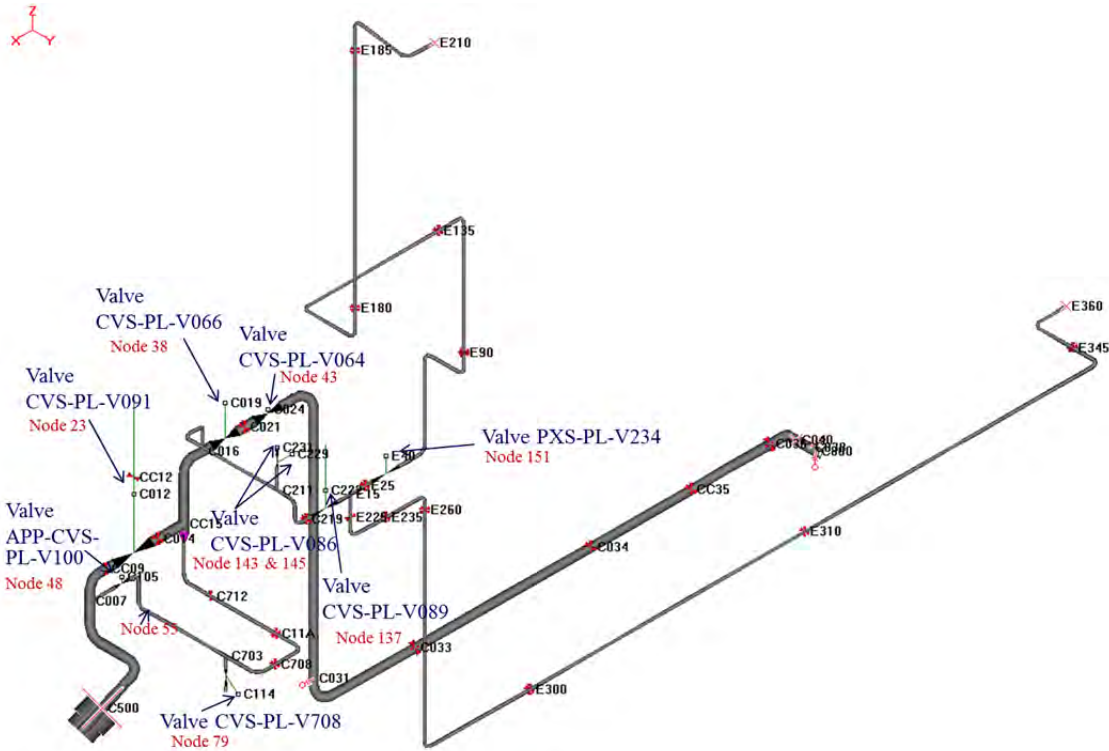


Figure 1. Finite element model of the piping segment and specified valve locations.

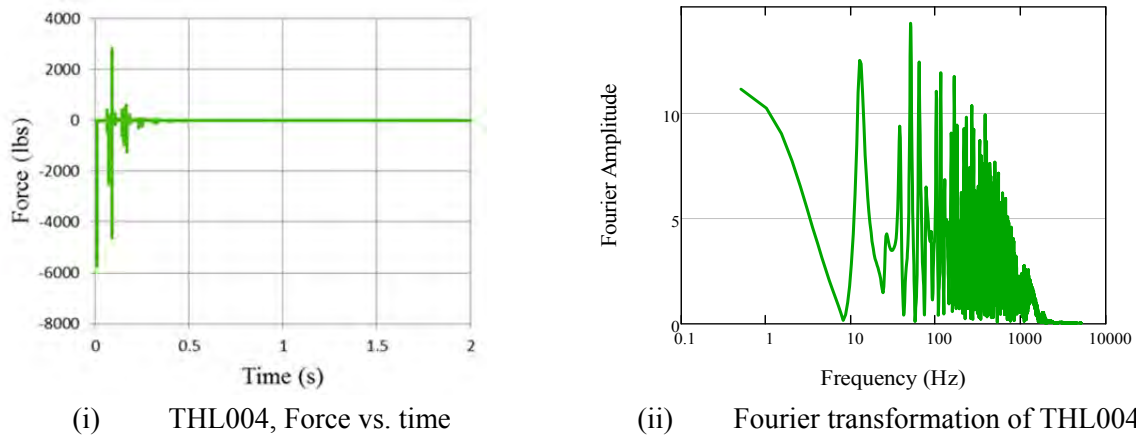


Figure 2. An example HD load (THL004) and the frequency content of the same load by using the Fourier transformation.

## BACKGROUND

### *Fragility Test Data*

As part of developing the basis for a frequency cutoff, the paper also examines existing fragility data to determine if this data would be applicable relative to the reported equipment type and characteristics of the input motion by means of acceleration levels and frequency content.

Fragility tests performed in the early 1970s as part of the SAFEGUARD program (referenced in NUREG/CR-3558), demonstrate acceptable reliability of power and process equipment installed in a hardened radar installation. The SAFEGUARD program accomplished selected fragility and shock environment qualification tests on generic equipment. This program developed about 300 test reports, of which 63 were directly applicable to equipment that were later used in the Seismic Safety Margin Research Program (SSMRP) to assess nuclear power plant equipment fragilities.

Most of the SAFEGUARD tests were single-axis, superimposed sine beat pulse tests that were selected to fit prescribed acceleration spectra. Equipment was excited to levels of more than 15g's while their functions were monitored. Consequently, the tests were true measures of fragility. The prescribed spectra reflect significant spectral accelerations in the high frequency range (in excess of 150 Hz) typical of blast loading.

Gaberson and Eubanks (1982) presented a simplified procedure to implement the SAFEGUARD data in design. In this procedure, if a test spectrum which the equipment has survived ("the hardness curve") is greater than the environmental spectrum ("the severity curve") for all frequencies above the fundamental frequency of the equipment; the equipment would survive the environmental spectrum. A subsequent pilot study by Electrical Power Research Institute (EPRI) examined the transfer of the military hardness assessment to the nuclear industry for evaluating the operability of equipment in the high frequency range of 20 to 100 Hz (EPRI NP-3946, 1985). The pilot study concluded that additional effort was needed to put the data and procedures in a form utilities could use more easily for evaluating a wider range of nuclear plant equipment under plant specific high frequency loads.

The HD loads are similar in frequency content and displacements to the shock loads used in the SAFEGUARD program. Therefore, the fragility information is useful, at least in a qualitative sense. However, because of differences in the type of equipment, the fragility information needs to be supplemented by representative quantitative test and analysis data to support the basis for the high frequency response of in-line valves subjected to HD loads.

### ***Regulatory and Industry Position***

Nuclear Regulatory Commission's (NRC) Regulatory Guide (RG) 1.100 (2009) states that "... the seismic qualifications of some safety related equipment were (previously) performed using IEEE 344 type tests with intentional high-frequency content to account for simultaneous BWR hydrodynamic loads. However, the frequency content of the input for such tests needs to have enough energy to be compatible with the amplified region of the Required Response Spectrum (RRS) at high frequencies. Therefore, past experience data may only be used when it is evaluated in accordance with Annex B of IEEE 344, "Frequency Content and Stationarity." It lists a number of techniques that may be used to show the adequacy of the frequency content and stationarity. Accordingly, the Earthquake Experience Based Qualification, discussed in ASME QME-1 Article QR-A7400 (2007) and the Experience methods of IEEE 344 (1987) are not acceptable methods for seismic qualification for **AP1000**<sup>®</sup> plant safety-related equipment.

IEEE 344 (2004) recommends that the strength and function of Class IE electrical equipment be demonstrated by analysis and by testing. According to IEEE 344 (2004), Section 8.1.6.1, the RRS for qualification by testing should reflect the combination of the operating basis earthquake (OBE) or safe shutdown earthquake (SSE) loads and the HD loads. The RRS for testing may be obtained by combinations such as SRSS.

More recently, the EPRI studies on high frequency ground motion (EPRI 2007a and 2007b) also conclude that high frequency (>10Hz) spectral accelerations result in negligible additional response stress in typical nuclear plant Structures, Systems, and Components (SSCs). However, from the functionality or operability perspective, components such as relays, switches, contactors, and other Electrical and Instrumentation and Control (I&C) devices are considered as vibration sensitive and may potentially be affected by high frequency excitation. EPRI (2007b) identifies the following component types as vibration sensitive:

- Electro-mechanical relays (e.g., control relays, time delay relays, protective relays)
- Electro-mechanical contactors (e.g., Motor Control Center (MCC) starters)
- Circuit breakers (e.g., molded case and power breakers – low and medium voltage)
- Auxiliary contacts (e.g., for Molded Case Circuit Breakers (MCCBs), fused disconnects, contactors/starters)
- Control switches (e.g., benchboard, panel, operator switches)
- Transfer switches (e.g., low and medium voltage switches with instrumentation)
- Process switches and sensors (e.g., pressure/differential pressure, temperature, level, limit/position, and flow)
- Potentiometers
- Digital/solid-state devices (mounting and connections only)

In order to assure that potentially vibration vulnerable components are avoided, EPRI (2007b) recommends that safety-related applications of these components be identified, evaluated, and their acceptability determined by screening using either existing qualification data, circuit evaluation for high frequency sensitivity, or high frequency testing. It should also be noted that, the frequency range considered for screening is only between 25 and 50 Hz for high frequency ground motion.

### *Valve Acceptance Criteria*

Seismic qualification of **AP1000**<sup>®</sup> plant valve assemblies are primarily performed by analysis. For active valves, the seismic qualification analysis is supplemented by functional qualification testing.

The valve seismic qualification analysis is based on static equivalent seismic loads associated with SSE accelerations of at least 6g applied along three orthogonal axes (two horizontal and one vertical) of the valve. The acceptance criteria ensure that the valve design meets the requirements of ASME B&PV Code, Section III (2010) for pressure boundary and structural integrity.

While the above analysis demonstrates structural integrity, operability of the active valves under seismic conditions is demonstrated by: 1) static deflection testing, and 2) shake table testing in accordance with IEEE 344 (1987), seismic qualification of operators. Test programs are designed to demonstrate that the active valve performance meets or exceeds the requirements of the valve design specifications. Static and dynamic test results may be compared to assess load sensitive behavior of electric motor actuated valves.

Static deflection testing of active valves demonstrates that the valve, when subjected to the required seismic loading, will not bind or interfere with itself. The load used in the static deflection test is taken to be the resultant (calculated by taking the square root of the sum of the squares) of the tri-axial static equivalent loads of 6g, plus the deadweight of the valve, and a 10 percent margin.

Qualification of active valves for dynamic loading with shake table testing in accordance with IEEE 344 (1987) and IEEE 382 (1996) takes into account the dynamic characteristics of the equipment, such as frequency and mode shapes and are, therefore, more representative of seismic and any other vibratory loads the valves may be subjected to. The shake table tests demonstrate that active power operated valve assemblies meet the functional requirements in accordance with ASME QME-1 (2007) Appendix QR-A for a bounding acceleration input. This bounding acceleration is defined with the Required Input Motion (RIM) curve.

**AP1000**<sup>®</sup> plant Valve Design Specifications requires that the Service Level D load conditions for valve qualification combine the acceleration inputs due to all dynamic loads (pipe break or valve opening) with the seismic input acceleration using the square root sum of squares (SRSS) method. Equipment Qualification Methodology for Valves commits to qualifying the valves to a total input acceleration of 6.6g over a broad frequency range as reasonably bounding the site specific demand. RIM testing from 2 Hz to 32 Hz is performed using a series of single-frequency sine-beat tests at one-third octave interval test

frequencies. For Hard Rock High Frequency (HRHF) potentially sensitive safety-related equipment, seismic RIM testing is extended to 64 Hz. RIM testing from 32 Hz to 64 Hz is performed using a series of single-frequency sine-beat tests at one-sixth octave interval test frequencies.

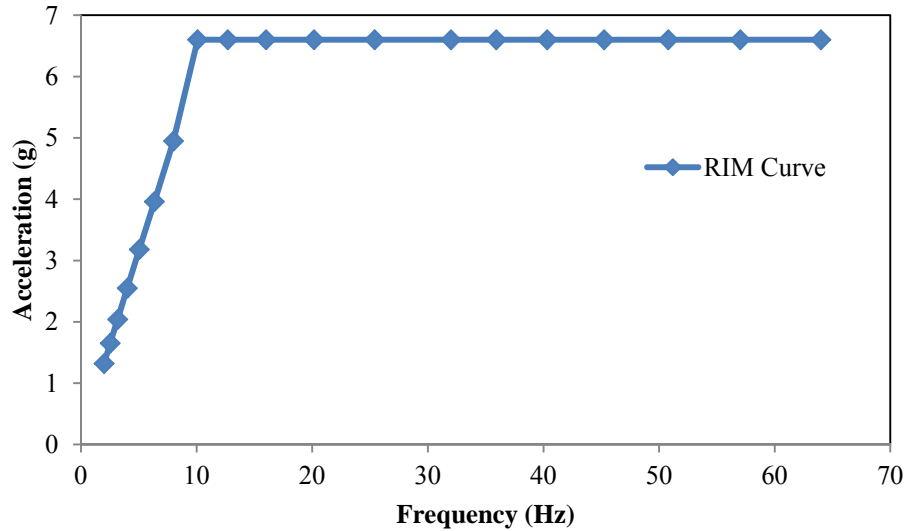


Figure 3. Required input motion (RIM) curve.

### BASIS FOR HIGH FREQUENCY CUTOFF

The basis for a high-frequency cutoff for the qualification analysis of in-line valves is developed. The implementation of the cutoff implies that the response of the in-line valves in modes above this cutoff may be ignored. The basis presented here examines the response of the representative piping system that affects structural integrity and operability, and the parameters that represent this response such as displacements and strain energy imparted to the in-line components. The following sections explain the basis for the high frequency cutoff, procedure implemented to test the cutoff and quantitative studies performed on the selected segment of the piping.

#### *Strain Energy Basis for High Frequency Cutoff*

The modal methods of dynamic analysis are based on the analogy between a vibration mode of a multi-degree system and the equivalent single-degree system. This analogy requires that the kinetic energy (KE), internal strain energy, and work done by all external forces for the equivalent single-degree system are at all times equal to the same quantities for the complete multi-degree system when vibrating in this normal mode alone (Biggs, 1970). Therefore, the sum of the KE in the individual modes can be used to represent the sum of the internal strain energies. Because it is related to the response quantities used in design, the strain energy (or its surrogate, the KE) may be used as basis to assess the cutoff ( $f_{co}$ ).

In general, for a specific value of a cutoff frequency ( $f_{co}$ ), the KE in active modes, i.e., the modes below the cutoff is:

$$KE_{co} = 1/2 \sum_{r=1}^m M_r S_{vr}^2 \quad (1)$$

The KE in modes above cutoff is approximated by:

$$KE_{rbm} = 1/2 [(1 - \sum_{r=1}^m p m_r) \sum_{i=1}^N (M_i)] S_{vco}^2 \quad (2)$$

and the total KE is thus approximated by:

$$KE_{total} \approx 1/2 [(\sum_{i=1}^N M_i)(1 - \sum_{r=1}^m pm_r) S_{vco}^2] \quad (3)$$

The system Kinetic Energy, ( $KE_{max}$ ), is another form of (1), and derived using all modes in the modal combination as:

$$KE_{max} = 1/2 [(\sum_{i=1}^N M_i)(\sum_{r=1}^N pm_r S_{vr}^2)] \quad (4)$$

where,

$S_{vco}$  = Spectral velocity at cutoff frequency, inches per second (in/sec) =  $S_{aco} / 2\pi f_{co}$

$S_{aco}$  = Spectral acceleration at cutoff frequency, inches per second squared (in/sec<sup>2</sup>)

$S_{vr}$  = Spectral velocity for r-th mode, in per second (in/sec) =  $S_{ar} / 2\pi f_r$

$S_{ar}$  = Spectral acceleration for r-th mode, in per second squared (in/sec<sup>2</sup>)

$f_r$  = r-th Mode frequency, Hz

$M_r$  = Effective Modal Mass for r-th mode, pounds which can also be described by the equation:

$$M_r = \Gamma_r \sum_{i=1}^N (M_i \phi_{ir}) \quad (5)$$

$M_i$  = Mass at i-th node, pounds

$\Gamma_r$  = Modal Participation Factor for r-th mode

$\{\phi_i\}$  = eigenvector matrix for mode r, with N modal shape coefficients,  $\phi_{ir}$  for  $i = 1$  to N

$pm_r$  = Mass Participation Ratio for r-th mode

$$pm_r = \Gamma_r \sum_{i=1}^N (M_i \phi_{ir}) / \sum_{i=1}^N (M_i) \quad (6)$$

$m$  = number of active modes

$N$  = total number of degrees of freedom

The KE associated with the above combination of modal responses is as follows:

$$KE = 1/2 \sum M_i S v_i^2 \quad (7)$$

summed over all modes.

Based on the above discussion, a cutoff frequency is assessed from a comparison of the strain energy calculated from (1) and (7).

### ***Procedure Implemented to Test High Frequency Cutoff for Valve Response***

For selected valves, the procedure first evaluates the KE in modes below the cutoff frequency (active modes) and reports the ratio of the KE in the active modes to the total system KE based on all modes up to the maximum frequency (500 modes up to 3500 Hz). Second, it performs time history analysis for HD loads and develops the response in terms of maximum displacements. It compares the “total solution” to the “cutoff solution.” The “total solution” represents the response resulting from consideration of 500 modes (up to 3500 Hz), while the “cutoff” solution is the response based on the truncated time history analysis.

The following steps summarize the procedure for the high frequency cutoff, while next section elaborates on it and provides the quantitative results.

1. Perform a modal superposition time history analysis of the structural model and extract the velocity time history for all and truncated modes at the mass point of each valve.
2. Calculate:
  - a. The maximum kinetic energy,  $KE_{max}$ , assuming all N modes to be active at the mass point of each valve.
  - b. The kinetic energy  $KE_{co}$ , for modes below cutoff at the mass point of each valve.
3. Compare and report the computed kinetic energies,  $KE_{co}$ ,  $KE_{max}$ .
4. Evaluate the maximum displacements at the mass point of each valve assuming 500 modes up 3500 Hz
5. Evaluate the maximum displacements at the mass point of each valve up to truncated time history analysis solution.
6. Compare representative displacement quantities for the full analysis and the cutoff analysis to assess the  $f_{co}$ .

## QUANTITATIVE STUDIES

Figure 4-i presents the displacement ratio in the direction of maximum displacement (governing direction) for various in-line valves included in the FE model obtained using various frequency cutoffs to the respective resultant displacement from the full solution considering 500 modes. Figure 4-i shows the results for cutoff frequencies of 33Hz, 50Hz, 64Hz, 80Hz and 100Hz. Figure 4-ii present the respective KE ratios corresponding to Figure 4-i.

Figure 4 illustrates that about 90% of the displacements and kinetic energies are accumulated in modes less than 64Hz in the direction of the major or dominant response. However, when the resultant response (SRSS in X, Y and Z directions) is considered, about 80% of the resultant displacements, and 70% of the total KE are accumulated in modes up to 64Hz. This is reflected in the smaller mass participation of the modes below 64 Hz in the minor directions. For example, Figure 5 which shows the mass participation vs. cutoff frequencies, shows mass participation in modes below 64 Hz in the Z direction is very low compared to that of resulting from the full solution using 500 modes. However, the low mass participation in Z direction doesn't affect the valve integrity or functionality (stem operability) for this type of loading as the valve reaches the maximum displacement in dominant direction (X or Y Direction). In addition, the total Z direction response is small compared to the dominant response, the relatively small contributions to the predicted displacements and KE below 64 Hz in this direction will not affect the overall stresses.

Figure 6 shows the power spectral density plots of the acceleration response at the largest valve to HD loads. Figure 7 on the other hand shows the power spectral density of the design basis earthquake floor response spectra at the piping system supports. It is observed from comparing Figure 6 and 7 that the power spectral amplitudes associated with the HD input are at least two orders of magnitude less than the corresponding seismic input even well into the high frequency range. The displacements, kinetic energies and PSD comparisons presented above provide the basis to conclude that a cutoff frequency of 64Hz captures the significant response of valves in system to HD loads.

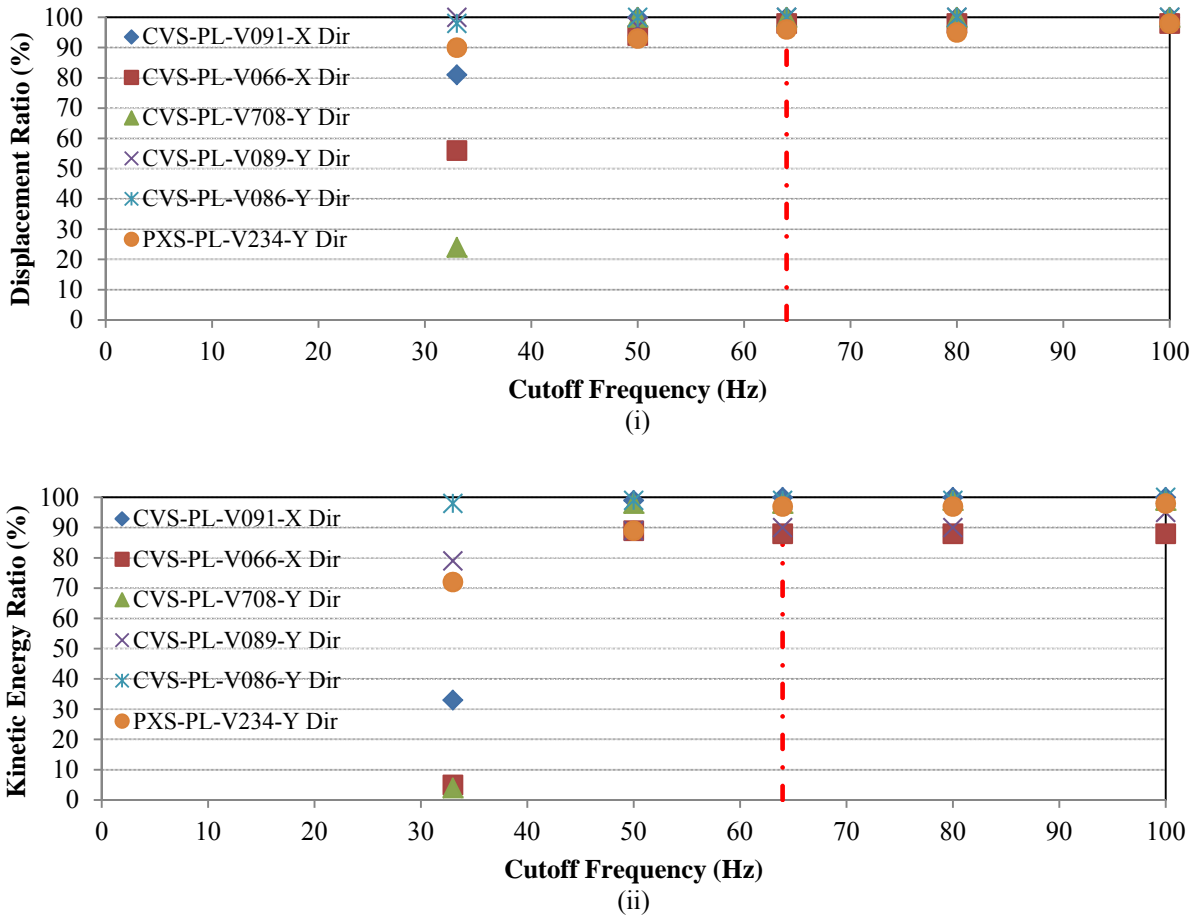


Figure 4. Summary of (i) displacement and (ii) KE ratios (in dominant displacement and KE directions for corresponding valve) vs. different cutoff frequencies.

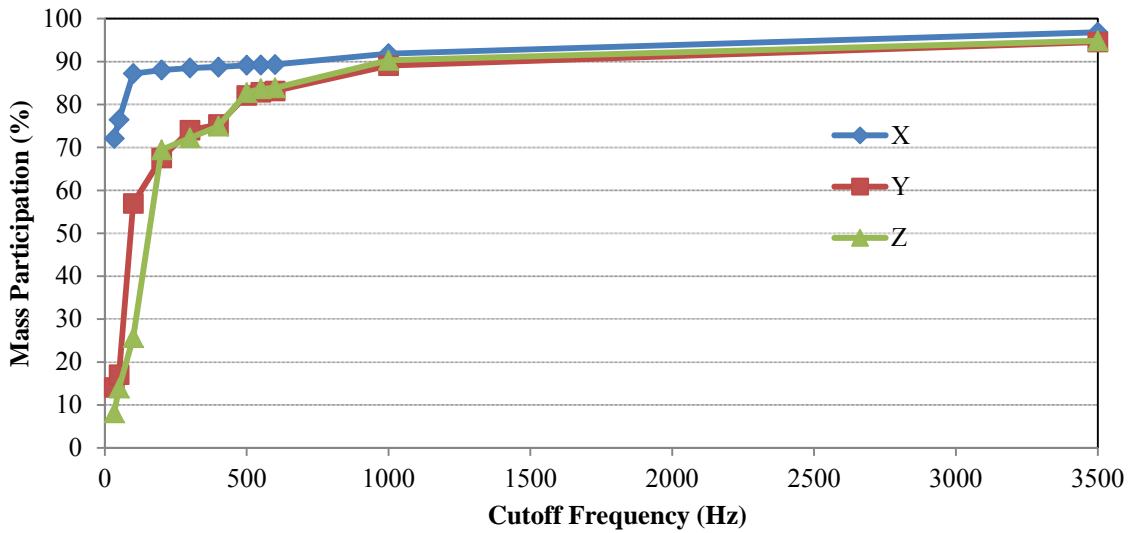


Figure 5. Modal mass participation in each direction for different cutoff frequencies.



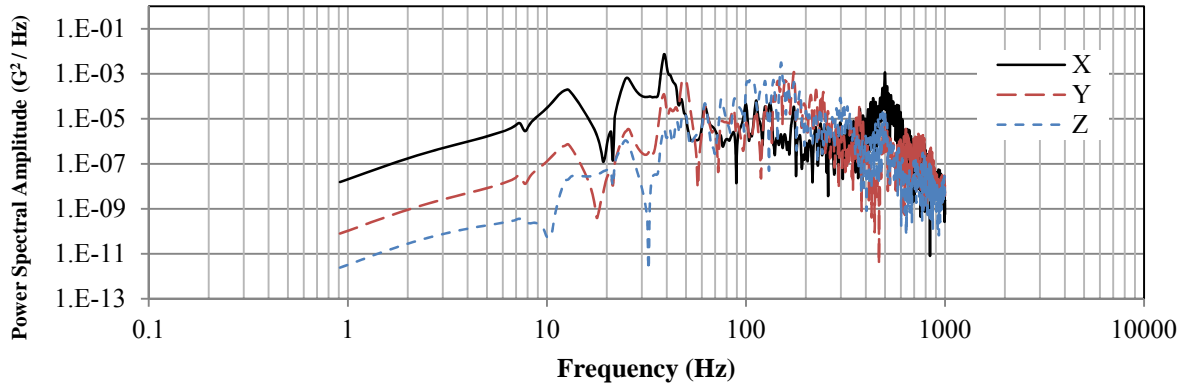


Figure 6. Power spectral density of acceleration response of valve CVS-PL-V091 to HD loads.

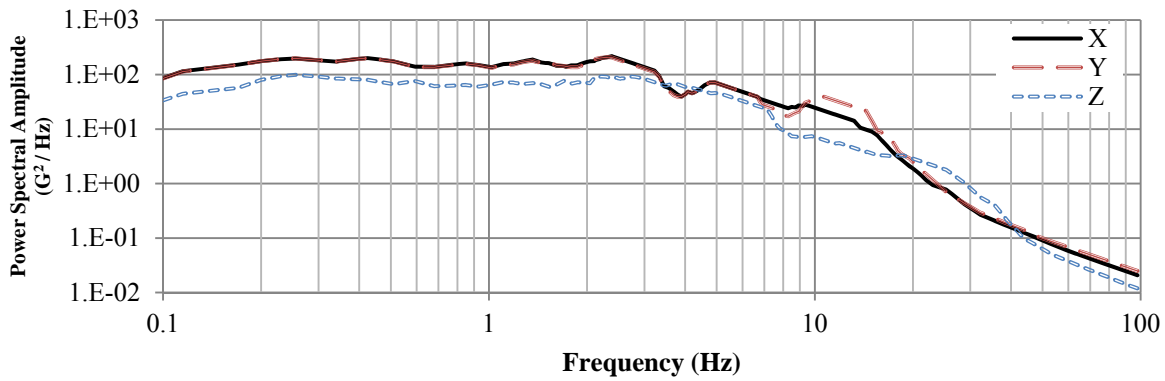


Figure 7. Power spectral density associated with the seismic (SSE) floor response spectrum loading.

## CONCLUSIONS

- HD loads generate system response that has high frequency content with accelerations that may exceed the qualification limits. These accelerations typically occur at locations with a small mass and have a very low impulse. The accelerations are observed at relatively high frequencies and the response is controlled by displacements which are found to be small.
- For the given piping system (or similar systems with similar response and transient characteristics), the valve is shown to have small displacements and the kinetic energy (KE).
- Power spectral density plots of the valves verify that the energy content under HD loads is much lower than that of seismic response.
- Because the response displacements and the KE are relatively small, structural integrity of valve body and pipes are not a concern under HD loads
- A cutoff frequency of 64 Hz is sufficient to capture the displacements, and KE content of the valve body and its response is controlled by the piping system.
- The findings of this study are in agreement with EPRI. High frequency motions are non-damaging.
- The performance of supporting electrical components subjected to high frequency pulse loading needs to be evaluated further.

## ACKNOWLEDGEMENTS

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## REFERENCES

- ASME QME-1, 2007, "Qualification of Active Mechanical Equipment Used in Nuclear Power Plants," The American Society of Mechanical Engineers.
- ASME 2010, Boiler and Pressure Vessel (B&PV) Code, Section III, American Society of Mechanical Engineers.
- Biggs, John M., 1970, "Section 3.7 Modal Analysis of Multi-Degree Systems," Introduction to Structural Dynamics, Mc-Graw Hill, New York.
- Cover, L. E.; M. P.Bohn, R. D. Campbell, and D. A. Wesley, 1985, "Handbook of Nuclear Power Plant Seismic Fragilities," Seismic Safety Margins Research Program, NUREG/CR-3558, University of California Radiation Laboratory (UCRL)-53455, Lawrence Livermore National Laboratory, Livermore, CA.
- Electric Power Research Institute (EPRI) 1015108, 2007a, "Program on Technology Innovation: The Effects of High Frequency Ground Motion on Structures, Components, and Equipment in Nuclear Power Plants"
- Electric Power Research Institute (EPRI) 1015109, 2007b, "Program on Technology Innovation: Seismic Screening of Components Sensitive to High-Frequency Vibratory Motions".
- Gaberson, H.A. and R.A. Eubanks, 1982, "Simplified Shock Design for Installation of Equipment," Naval Civil Engineering Laboratory, Port Hueneme, CA.
- Institute of Electrical and Electronics Engineers (IEEE), 1987, "IEEE Recommended Practice for Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations," Institute of Electrical and Electronics Engineers, Inc., New York, NY.
- Institute of Electrical and Electronics Engineers (IEEE), 2004, "IEEE Recommended Practice for Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations," IEEE-344, IEEE Power Engineering Society, New York, NY.
- United States Nuclear Regulatory Commission (USNRC), 2009, "Seismic Qualification of Electrical and Active Mechanical Equipment and Functional Qualification of Active Mechanical Equipment for Nuclear Power Plants," RG 1.100, Revision 3, USNRC Washington, DC, September 2009.

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