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Attachment to
MPR Letter dated
April 13, 2006

Vibration Acceptance Criteria for ERV Valve Actuators

Revision 1

Prepared by: Hans P. Giesecke
Hans Giesecke

Reviewed by: ____________
Robert Coward

Approved by: ____________
Robert Coward

QUALITY ASSURANCE DOCUMENT

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

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Contents

Background

The main steam piping near the reactor at both Quad Cities units has been experiencing significant vibration for some time. This vibration has resulted in damage to the actuators on the Electromatic Relief Valves (ERVs), among other components. The vibration increased significantly (by about a factor of 2) when the plant power level was increased to Extended Power Uprate (EPU) conditions. Apparently, the dominant part of the vibration is being generated by the response of the piping to several acoustic modes that exist inside the piping. These acoustic modes are most likely standing pressure waves that form in the standpipes that attach the ERVs, Safety Relief Valves (SRVs), and Target Rock valve to the main steam piping. Most likely, the acoustic response is excited by turbulence in the main steam flow at the connecting tees to the standpipes. Since the turbulence spectrum typically increases in both amplitude and frequency with increasing steam flow, the increase in vibration at EPU power levels is significantly higher than at pre-EPU power levels.

Due to damage to the ERV actuators that was discovered at Quad Cities Unit 1 during Q1F51, Exelon decided to perform vibration testing of an ERV at the Wyle Laboratories in February 2004. These tests are described in Reference 1 and Figure 3-5 in Reference 1 shows the initial test configuration, which included an ERV with the attached pilot valve and the pilot valve actuator. The testing resulted in modifications to the actuators to "harden" them against wear by employing better materials for parts where wear was observed. The modified actuators were installed in both units at Quad Cities and after operating an extended period of time at pre-EPU levels, they were inspected and no damage was observed. However, after several months of operation at EPU power levels, significant damage was discovered on some of the actuators and all actuators had some wear in the bushings and guide posts. Figure 1 is a picture of an actuator similar to the modified actuators that were installed following the February 2004 Wyle testing. The damage to these actuators was concentrated in the guide post-spring-bushing interface due to the hardened spring getting caught between the post and the bushing, causing considerable wear to all three components due to vibratory motion. Damage also occurred to the pivot plate that opens the cutoff switch and limits the current to the solenoid when the pilot valve is open. If this switch fails to operate, the solenoid will fail in about 30 seconds due to excessive current, making the valve inoperable.

Based on the observed damage, in January 2006 a beveled washer was inserted below the bushings to protect the post and bushing from the spring (see Figure 2) and the plants were restarted and operated at pre-EPU power levels while a permanent solution to the problem was being investigated. This permanent solution concentrated on the installation of acoustic side branches (ASBs) on the standpipes that support the valves to mitigate the acoustic modes in conjunction with modifications to the ERV actuators to make them considerably more resistant to the damage mechanisms that were being observed.

Figure 3 shows a picture of the newly modified actuator. Vibration testing of the new actuator was conducted during February and March of 2006 to test modifications and improve confidence in the new actuator for installation in the plant. The test included testing of alternative designs such as the design that was damaged during operation in the plant to provide a comparison of relative damage levels during the testing. This report evaluates the new design and provides an estimate of the vibration levels that are considered acceptable for operation in the plant.

Simplified Acceptance Criteria for Modified Actuator

The vibration limits specified by the Vibration Acceptance Criteria described below are based on the testing that was conducted during February and March of 2006. The acceptance criteria considers a broad band random vibration base with a single frequency sine dwell from an acoustic mode such as was measured in the plant during EPU operating conditions. The Vibration Acceptance Criteria is sufficiently high to envelop the vibration levels experienced by ERV 3D from Quad Cities Unit 2, the actuator with the highest vibration as measured during operation at EPU. Since there is some uncertainty associated with the effect that the ASBs may have on the acoustic mode, the testing included excitation of the actuator at various frequencies, including excitation close to resonance, so that a Vibration Acceptance Criteria could be established that considered all reasonable possibilities for the frequency of the acoustic mode. However, at lower frequencies a strong acoustic mode could damage the actuator due to the higher displacements associated with lower frequency at the same acceleration magnitude. Therefore, the acceptable acceleration levels had to be reduced at lower frequencies. In effect, at frequencies below 100 Hz, the Acceptance Criteria must be limited by displacement amplitude whereas at high frequencies, the Acceptance Criteria must be limited by acceleration amplitude.

To account for both these phenomena, an acceptance criteria was devised that limits the vibration at the base of the actuator where it attaches to the yoke of the pilot valve to about ½ the levels tested based on the integrated spectrum RMS values as follows:

- The integrated RMS value for the spectrum of the measured displacement should be less than 2.2 mils RMS. Due to the difficulty of obtaining accurate acceleration values at low frequency, the displacement RMS value should be based on the frequency interval between 30 and 200 Hz.

- The integrated RMS value for the spectrum of the measured acceleration should be less than 2 g RMS. To be consistent with the displacement value, the frequency range of integration should also be between 30 Hz and 200 Hz.

- A horizontal value that is the square root of the sum of the squares (SRSS) of the two orthogonal horizontal components should be obtained and compared to the acceptance criteria value. The vertical component should be compared directly to the acceptance criteria value.

Based on the test data, the actuator will be able to withstand vibration levels up to the acceptance criteria limits for at least one full operating cycle (24 months) without incurring an unacceptable amount of damage that would prevent its operation. The vibration levels should be limited by both the acceleration and displacement values. The acceleration value of 2 g RMS will limit the amplitude of the acoustic mode at frequencies above 100 Hz whereas the displacement value of 2.2 mils RMS will ensure that the acoustic mode, if its frequency is less than 100 Hz, will not be strong enough to damage the actuator. The basis for the acceptance criteria is discussed in the rest of this report.

Vibration Test Basis

Vibration testing for the ERV actuator was performed at the Quanta Laboratories in Santa Clara, California in two phases during February and March of 2006. The tests are described in Reference 2.

Measured Vibrations at EPU Power Levels

The acoustic modes generate single frequency responses that are tuned to each specific standpipe and they exist over a frequency range of about 130 to 160 Hertz, depending on the particular standpipe involved. Vibration measurements obtained at Quad Cities during operation at EPU indicate that, in general, the vibration levels on Unit 2 are higher then those on Unit 1. In particular, the valve with the highest vibration is ERV 3D on Unit 2 and its vibration is dominated by an acoustic mode with a frequency of about 151 Hz. Figure 4 provides a plot of all the power spectral densities of the measured locations on the Unit 2 ERVs. The plot is provided in a linear scale to accentuate the dominance of the acoustic modes. The acceleration vibration spectra for ERV 3D on Unit 2 are plotted separately in Figure 5 and the ones for all the other valves on Unit 2 are plotted in Figure 6 to demonstrate that the 3D valve is subjected to the highest vibration levels. In fact, when the spectra from Unit 2 ERV 3E is removed from Figure 6, the remaining responses for valves 3B and 3C are much smaller. Consequently, the vibration spectrum levels from ERV 3D on Quad Cities Unit 2 were used to develop the "base test spectrum". However, the frequency inputs for the base spectrum were expanded to account for the different frequencies of the various acoustic modes as well as for any impact that the ASBs may have if they are installed on the standpipes of the valves.

As is seen in Figures 4 through 6, the measured vibration is essentially composed of monochromatic accelerations at about 138 Hz, 151 Hz, and 161 Hz with the dominant response at 151 Hz at an amplitude of about 2 g RMS. The amplitude of the associated displacement is less than one mil.

In-Plant Experience

The modified actuators that were installed in Unit 1 operated at EPU vibration levels for over 5 months and those that were installed in Unit 2 operated at EPU vibration levels for over 6 months. Prior to operation at EPU power levels, the Unit 2 actuators operated for over a year at pre-EPU vibration levels, after which they were inspected. No significant damage was identified at that time.

However, after the operation at EPU power levels, some actuators had significant damage. In particular, the actuator on ERV 3D had significant damage to the posts, the springs, and the cutoff switch and was not considered capable of operating. The actuator on ERV 3E also had significant damage to the rods but was still operational. The actuators on ERV 3B and ERV 3C had some damage to the posts but were still functional. The damage level was comparable to the measured vibration level since 3D had the highest measured vibration, 3E had the second highest, and 3B and 3C had relatively low levels of vibration.

Enhancements Made to Actuator Design

The new actuator design (see Figure 3) incorporated a significant number of improvements, many of which were implemented as a result of lessons learned during the testing. The new actuator provides substantially more support for the guide posts. This was accomplished by mounting the posts to a thicker support plate at the bottom in place of the angle brackets on the original and including a frame that is tied to that plate and supports the tops of the guide posts. In addition, the springs are of a larger diameter and are split into two parts. The ends of the spring were mounted on the outside of both the bushings and a spacer that was added between the two spring parts on each post so the springs would remain separated from the posts. This prevents the springs from getting caught between the post and the bushing and limits the wear that will occur to the springs, the bushings, and the posts. Clearance between the top bushing and the post must be loose enough to prevent binding due to clearances between the plunger and the frame that allow the plunger to displace at the bottom but must be small enough to limit impact or fretting wear when the posts vibrate. The new actuator design also includes a silicone isolation pad that prevents much of the high frequency vibration from reaching the posts. In addition, the pivot plate that actuates the cutoff switch to limit current through the solenoids when the pilot valve is open was improved by tightening clearances to avoid rattling, adding bushings for the pivot pins, and modifying the material for the pivot pins and bushings to improve wear characteristics. Finally, the brackets used to attach the actuator components to its base were reinforced by gussets to help redistribute stresses over a larger region and procedures were implemented to ensure there is even loading on the heads of the bolts that secure the solenoid frame to the base plate of the actuator. The gussets were added halfway through the first series of testing due to cracking in the brackets and indicated, by no additional damage through the most severe testing period, that they were effective in eliminating the cracking problem. The modifications to the pivot plate were implemented due to knowledge gained during the first series of testing.

During the second series of tests, it was discovered that the clearance for the upper bushings on the guide rods was too tight, resulting in binding of the bushing on the guide post. The clearance was increased from 2.5 mils to 15 mils and the leading edge of the bushing was chamfered to reduce the susceptibility for binding. In addition, it was discovered that the bolts that hold the frame on the base plate were not making full contact around their circumference, resulting in significant bending moment on the head of the bolt and the failure of several bolts early in the second round of testing. This oversight was corrected by modifying the production procedure to ensure good contact between the head of the bolt and the bearing surface by counter-boring the contact surface to obtain a good fit. Since the prototype actuator had experienced a significant amount of testing at higher vibration levels without any baseplate bolt failures, and since the new actuator on which the contact was modified completed the testing without any additional bolt failures, it was apparent that the cause of the bolt failures was correctly identified.

Description of Vibration Tests

The purpose of the vibration testing was to demonstrate a high confidence level that the modified ERV actuator could operate for a full fuel cycle without being damaged such that it would no longer function to operate the ERV. An alternate purpose was to obtain a vibration spectrum envelope below which only limited damage would be expected so that the envelope could be used for comparison to measured plant vibration spectra. If the plant vibration spectra are below this envelope, the actuator would be expected to perform adequately in response to the applied vibratory loading.

In order to provide confidence in the results of the tests, it was decided to test the original actuator in conjunction to the others, thereby providing a reference point for comparison. All actuators were mounted on the same shaker table simultaneously to ensure they all saw the same vibration input.

The results of the testing are covered in Reference 2 and are only described in general here.

Development of Test Plan

The test plan for these tests utilized a conservative "base test spectrum" that envelops the measured spectra from all the ERV and pilot valve spectra that were obtained from actual plant measurements while the plants were operating at EPU power levels. Including the spectra from the pilot valves in the base spectrum allowed the actuators to be vibration tested independently of the ERV since the applied spectrum was representative of what was applied directly to the actuator in the plant. In order to account for changes in the natural frequency of the acoustic modes that may result from installation of the ASBs, the frequency spectrum was broadened to include any possible actuator resonance between the range of 90 Hz and 160 Hz.

Once the "base test spectrum" was defined, the base spectrum was multiplied by a scale factor to provide a means for adjusting the level of vibration in a controlled manner. The multiplier to that base spectrum could be used to slowly increase the overall vibration levels and ultimately to accelerate the wear mechanism so that an equivalent amount of wear could be generated in a short term test as would be expected to occur in the plant over a full fuel cycle. The inclusion of the original actuators on the table was used to demonstrate that sufficient excitation levels were being applied to the actuators to represent long term plant conditions. If the original actuator displayed damage similar to what occurred in the plant, it would indicate that the alternative actuators were subjected to realistic accelerated vibration levels during the testing.

First Series of Tests

A test spectrum was designed to envelop any spectrum that would be experienced in the plant. This spectrum included a broadband component of 0.3 g RMS and a series of swept sine inputs that ensured any resonant frequencies of the actuators were being excited. There were 4 swept sine inputs, at 2 g RMS each, distributed above 100 Hertz and three additional swept sine inputs below 100 Hz scaled in amplitude to maintain the average vibration displacement amplitude to the same level that would result for a 100 Hz sine component. The low frequency swept sines have the ability to damage the actuators due to their increased vibration displacement amplitude.

Since there was very little low frequency input in the plant spectra, it was considered possible that too much low frequency input would result in unrealistic damage to the actuators and invalidate the test results. Details are provided in Reference 2.

The test spectrum was multiplied by a scale factor so that different levels of excitation could be applied to the actuators in a controlled manner. The testing utilized a gradually increasing multiplier with detailed inspections conducted prior to each time the multiplier was increased so that the level and characteristics of the vibration that would cause damage to occur could be defined. In this manner, the damage that was being incurred by each actuator on the test table could be quantified with respect to the severity of the load.

Prior to age-testing of the actuators, the actuators were subjected to sine sweeps to identify their natural frequencies and modes. Each actuator was tested individually with tri-axial accelerometers attached at various locations such as on the top of the guide posts, the plunger, and the support plate. The sine sweeps were conducted at two levels of input and the results indicated that the natural frequencies were quite sensitive to input level. In order to account for changes in the natural frequencies due to input level or changes in the frequency of the acoustic modes that may result from installation of the ASBs, the frequency spectrum was broadened to include any possible actuator resonance between the range of 90 Hz and 160 Hz.
In the first series of tests, the applied spectra were increased slowly, starting at levels that were 1/8 of the base spectrum and finishing at levels that were more than twice the base spectrum..

Damage assessments indicated that the low frequency vibration may have been causing excess damage to the actuator covers relative to the damage to the posts that was observed in the plant. Since the plant spectra did not have very much low frequency input and the broadband component of the test spectrum enveloped the plant spectrum at those frequencies, it was decided to limit the sine wave part of the spectrum to a single frequency sine dwell in conjunction with the random broadband input to better represent the measured plant conditions. After 20 hours of testing with this new spectrum with a multiplier of 2.0, significant damage occurred to the original actuator that was similar, though less severe, than that experienced by the most highly damaged actuator in the plant. The spring on one side of the original actuator broke and became trapped between the bushing and the guide rod, doing significant damage to the guide rod and resulting in the plunger not being operable. On the other hand, the other two actuators had much less damage with the Modified GE actuator only experiencing a slight amount of wear, more like polishing, between the guide posts and the bushings.

Additional vibration testing resulted in the spring on the original actuator wearing all the way through the bushing and coming out the top of the actuator cover. Since there were observation holes in the cover above the guide post, the spring became free and no additional damage due to the spring-guidepost-bushing interaction occurred on this post. However, the spring would have remained trapped inside the cover during operation in the plant and might have caused additional wear. Also, every inspection of the original actuator freed up the spring from inside the bushing, interrupting the accelerated damage rate that would have resulted in the plant. The new actuator again had only slight wear.

After it had been demonstrated that a spectrum similar to the one measured in the plant would result in damage to the original actuator that was similar to that observed in the plant, the frequency of the sine dwell was moved to excite the natural frequency of the new actuator. A multiplier of 2.0 was still being applied. Despite significant testing with a spectrum that was specifically designed to test the new actuator, only about 2 mils of wear was measured on the guide posts of the new actuator. However, a weak spot associated with the cut-off switch that limits the current to the solenoid after the valve has opened was identified. The tests indicated that wear around the pins about which the plate rotates would likely be the failure mode of this actuator if the vibration becomes too large.

Second Series of Tests

Due to excessive wear in the pins that secure the pivot plate for the cutout switch, the pins were replaced with a harder material and a bushing was embedded into the frame to provide better wear resistance for this part of the actuator. As a result of this modification, a second series of tests were conducted at the Quanta Labs in Santa Clara. This second series included three actuators; the original design, the prototype that was tested previously, and a production actuator that met the criteria necessary for installation in the plant. The second series of testing identified several additional problems that were resolved as a result of the testing. In addition, the broad band component of the base spectrum was increased from 0.3 g RMS to 0.4 g RMS to provide more margin for the acceptance criteria.

Basis for Acceptance Criteria for Modified Actuator

The testing demonstrated that the design improvements of the modified actuator worked well.

- Support to the posts was effective in reducing post vibration, eliminating distortion of the posts, and maintaining a uniform configuration that allowed the plunger to operate.

- The new bushing and spring design was very effective in eliminating the spring as a source of the damage by preventing the springs from becoming trapped between the bushing and the support rods where they would induce significant wear of the posts.

- The isolation damper at the base of the actuator was effective in isolating the internals of the actuator from high frequency vibration. However, the angle brackets that tie the actuator to its base developed cracks and had to be reinforced by gussets during the test. The gussets experienced enough high level vibration during the latter part of the testing to demonstrate they would not crack during normal operation.

- The pivot plate set screw for the cut-off switch that limits current to the solenoid had not been modified at the time the first series of testing was conducted and remained the weak part of the newly modified actuator design. This deficiency led to the last three actuation tests to be inconclusive during the first series of tests since sparks were generated at the cut-off switch when the plunger hit the pivot plate. The pivot pins continued to experience significant wear. If the clearance around these pins becomes too large, the plate will no longer rotate to open the switch. The design was modified for the second series of tests to harden the pins and their mating surface in order to reduce wear in this region. The results indicated the modification was very successful.

Since the spectrum the actuators were subjected to had a broadband component and specific sine dwells that were tuned to excite at the actuators natural frequencies, the frequency content of the excitation was significantly more severe than that which is anticipated in the plant. Based on the test data, a multiplier of 2.0 resulted in damage to the old actuator that was similar to that seen in the plant. Therefore, the multiplier of 2.0 for the test period was comparable to extended operations in the plant with an effective multiplier of 1.0. Since the new actuator experienced little damage during the second series of testing, it should be acceptable for vibration levels equivalent to the "base test spectrum" with a multiplier of 1.0. The equivalence of short term testing with a multiplier of 2.0 to an extended operating cycle at a multiplier of 1.0 is also indicated by the relative damage that occurred to the actuators after operating in the plant at pre-EPU levels compared to operating at EPU levels, particularly when the impact of the spring being caught inside the bushing is removed from consideration.

As a result of the above analysis, we consider a reasonable limit for the vibration spectrum of the new Quad Cities ERV actuators consists of the following:

- Broadband acceleration input at the base of the actuator must be less than 0.4 g RMS which is equivalent to an average amplitude of 0.00089 g^2/Hz. The integrated vibration input should be less than the 0.4 g RMS value. Individual responses slightly in excess of the 0.00089 g^2/Hz value are considered acceptable as long as the integrated value of the broadband component remains below 0.4 g RMS.

- In addition to the broadband excitation, a monochromatic frequency due to the acoustic mode can be tolerated if the amplitude of that frequency is less than 2.0 g RMS at 100 Hz to 200 Hz. Below 100 Hz, the displacement amplitude of the monochromatic vibration must remain below the 100 Hz 2 g RMS value. Figure 7 provides a plot of the vibration limit as a function of frequency for the monochromatic input at the base of the actuator resulting from excitation by the acoustic mode. This graph provides the maximum acceleration of a single sine wave at a specific frequency as a function of that frequency. The graph does not represent a spectrum with multiple frequency inputs. Numeric values of the acceptable limit are listed as a function of frequency in Table 1. The broadband background spectrum of 0.4 g RMS can be tolerated in addition to this monochromatic frequency excitation.

The vibration limits specified consider a broad band random vibration base with a single frequency sine dwell from the acoustic mode. The frequency of the acoustic mode is not specified but its amplitude must be contained by the envelope provided in column 2 of Table 1 and shown in Figure 7. Table 1 also provides the RMS value of the complete spectrum for both the acceleration and the displacement. For simplicity in application, a conservative criteria based on the full spectrum can be implemented by limiting the RMS value of the acceleration to less than 2 g RMS while simultaneously limiting the RMS value of the displacement to less than 2.2 mils RMS. In this manner, the vibration amplitude is limited by the displacement for frequencies below 100 Hz while limiting the acceleration for frequencies above 100 Hz.

These vibration limits are based on the measured vibration of the table on which the actuators were tested. Therefore, the acceptance criteria must be compared to measured values from the plant that are located on the pilot valve yoke or the actuator base plate where it attaches to the yoke.

References

1. SIA Report SIR-04-023, Rev. 0, "Quad Cities ERV Vibration Testing Assessment", dated February, 2004

2. 06Q4568-DR-005, "Quad Cities ERV Pilot Valve Actuator Vibration Test Report", Rev. 1, dated April 11, 2006.

Table 1. Acceptable Response Magnitude at Base of Actuator

Frequency Hertz	Acoustic Mode g RMS	Broadband g RMS	Total g RMS	Total RMS Displacement
20	0.08	0.4	0.4079	0.0022
30	0.18	0.4	0.4386	0.0022
40	0.32	0.4	0.5123	0.0022
50	0.50	0.4	0.6403	0.0022
60	0.72	0.4	0.8236	0.0022
70	0.98	0.4	1.0585	0.0022
80	1.28	0.4	1.3410	0.0022
90	1.62	0.4	1.6687	0.0022
100	2.00	0.4	2.0396	0.0022
110	2.00	0.4	2.0396	0.0019
120	2.00	0.4	2.0396	0.0017
130	2.00	0.4	2.0396	0.0016
140	2.00	0.4	2.0396	0.0015
150	2.00	0.4	2.0396	0.0014
160	2.00	0.4	2.0396	0.0014
170	2.00	0.4	2.0396	0.0013
180	2.00	0.4	2.0396	0.0012
190	2.00	0.4	2.0396	0.0012
200	2.00	0.4	2.0396	0.0012

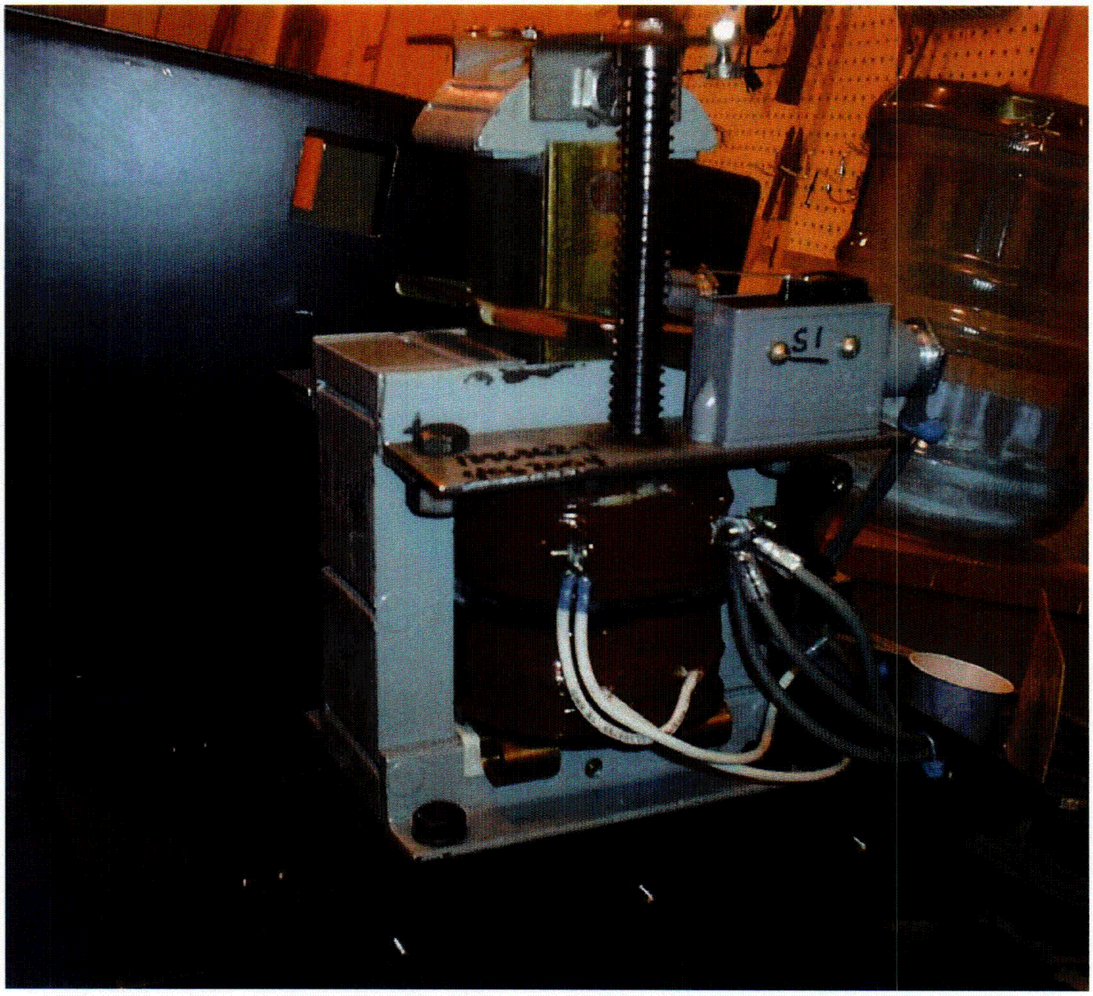

Figure 1. ERV Actuator Similar to the One that was Damaged in the Plant (No Cover)

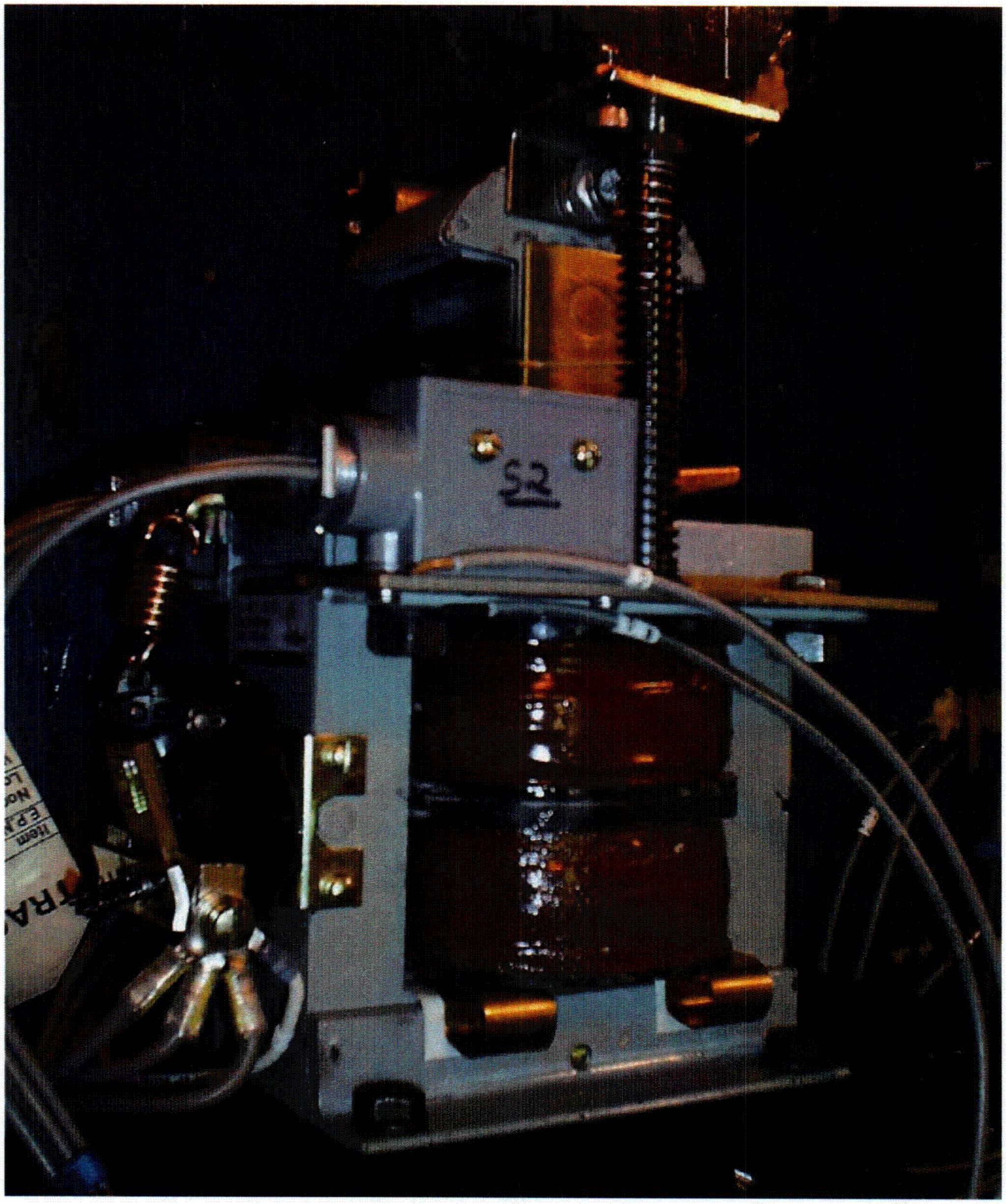

Figure 2. Modification to Post-Spring-Bushing Interface Using a Beveled Washer

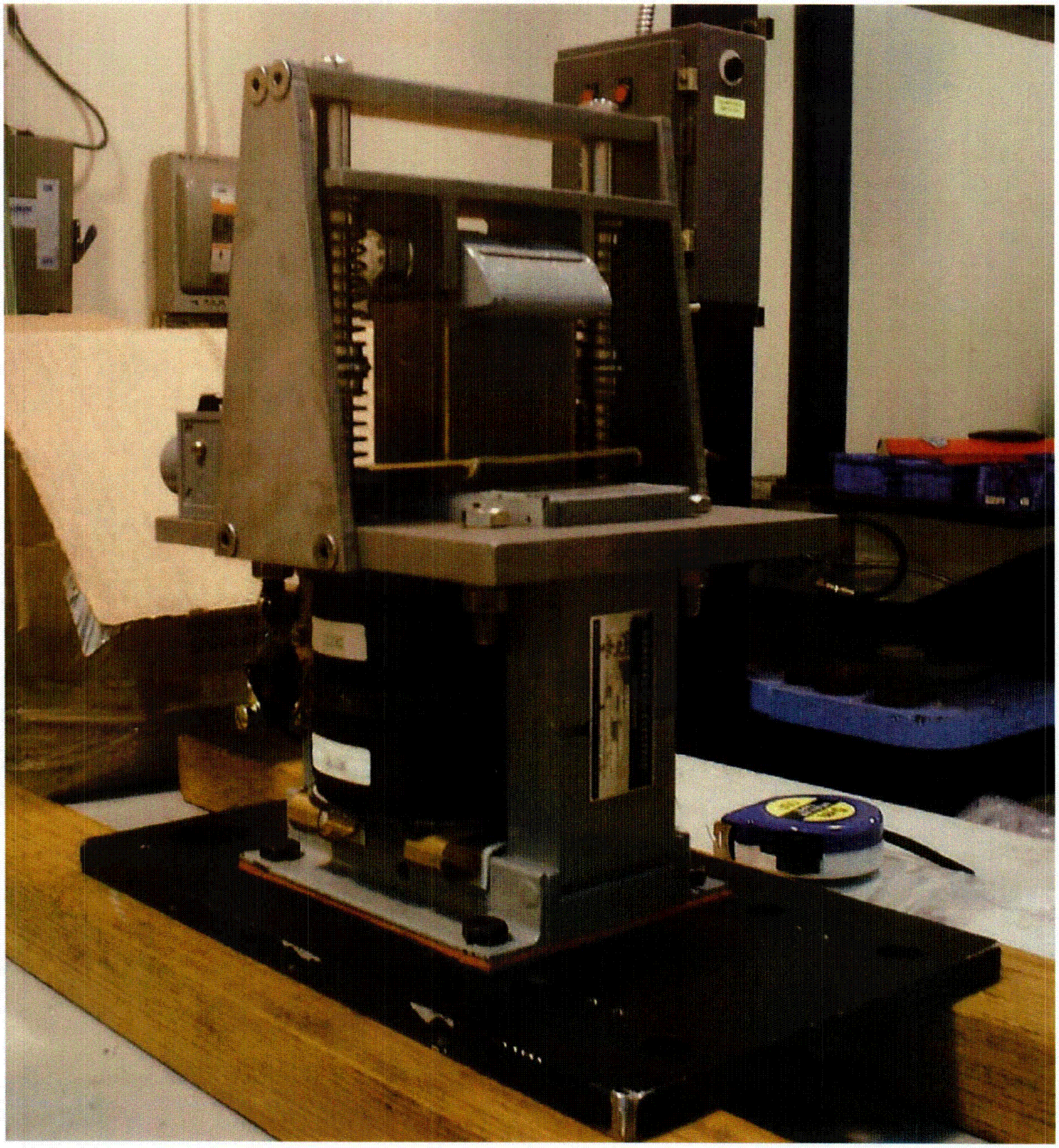

Figure 3. New Actuator Design for Testing

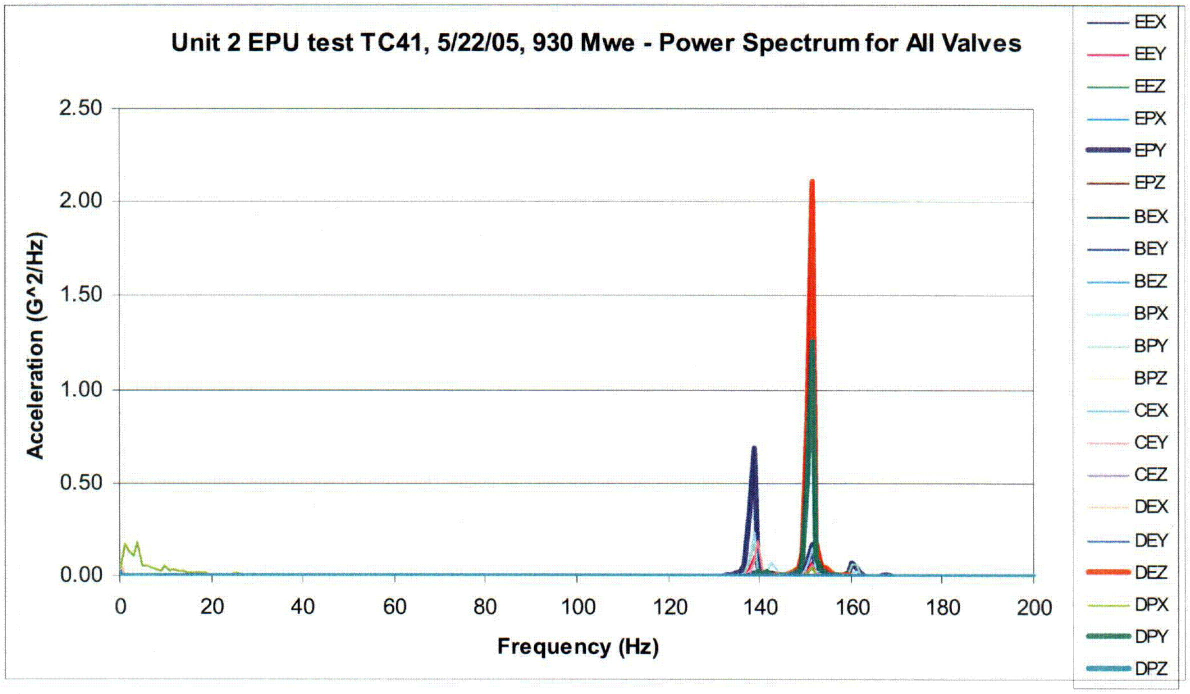

Figure 4. EPU Vibration Spectra for all Unit 2 ERVs

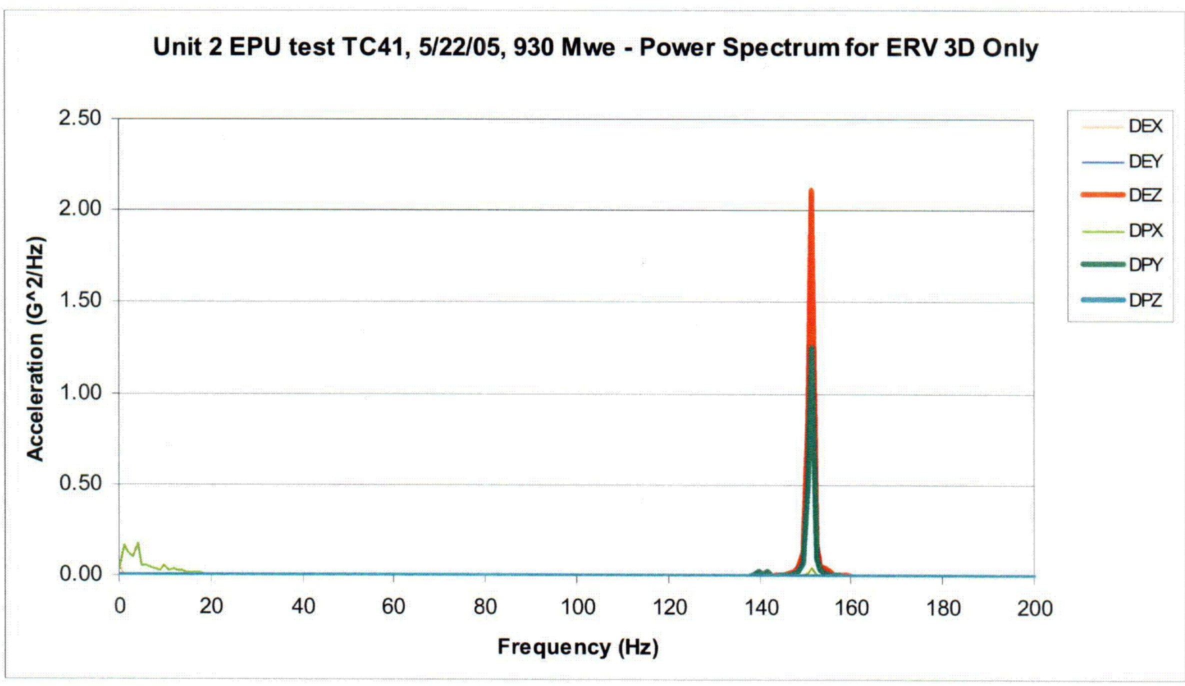

Figure 5. EPU Vibration Spectra for ERV 3D Only

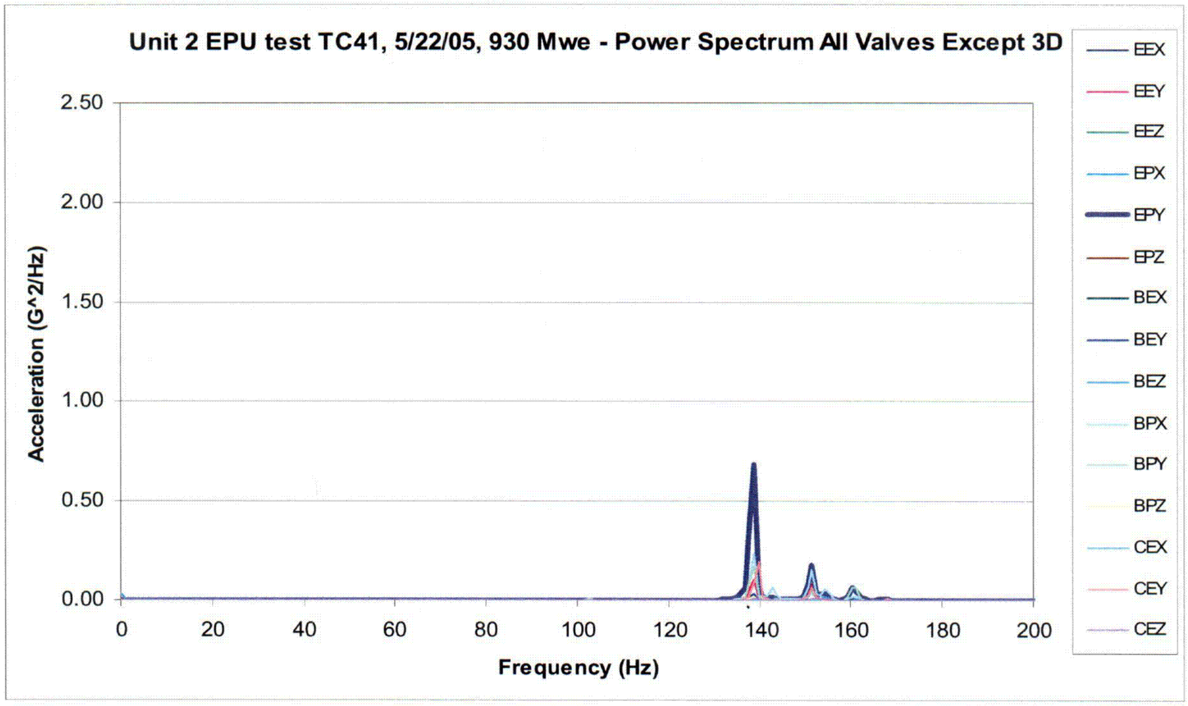

Figure 6. EPU Vibration Spectra for all Unit 2 ERVs Except 3D

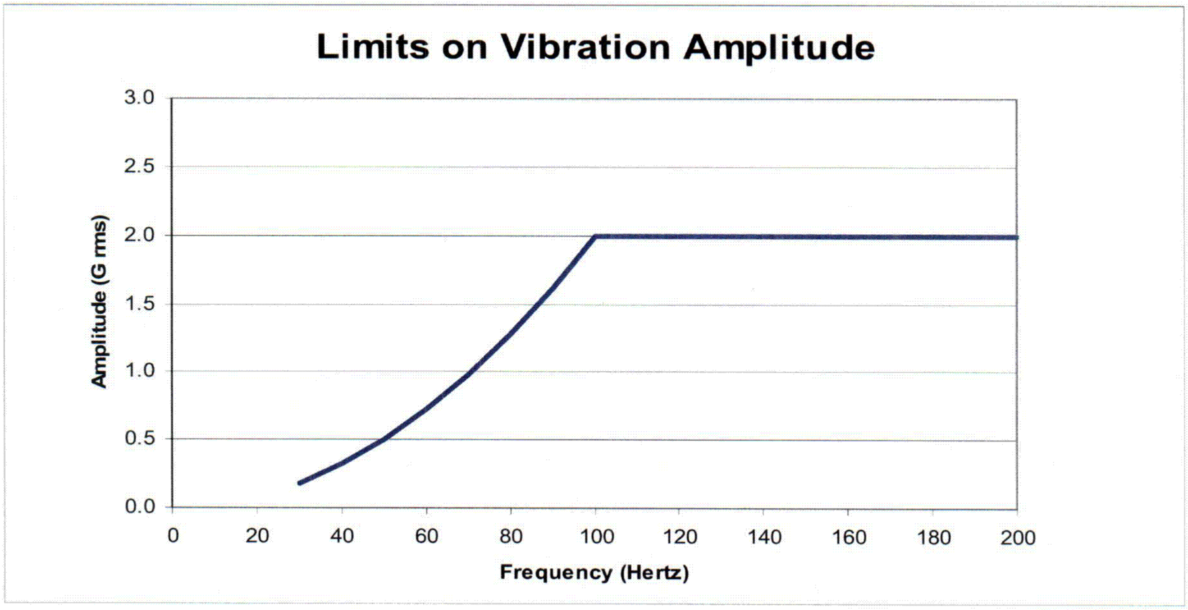

Figure 7. Acceptable Vibration Magnitude at the base of the New Actuator due to the Acoustic Mode as a Function of the Acoustic Mode Frequency

Table 1 summarizes the actuators tested.
Table 2 summarizes the tests performed.
Table 3 summarizes all the events that occurred during testing and their resolutions.

TABLE 1. List of Actuators Shake Table Tested

	Actuator Designation	Description
1st Test Series (ACT-1 to ACT-10)	#1	The design used at the Quad Cities Nuclear Plant prior to January 2006. Newly refurbished and assembled in accordance with the plant procedures used prior to January 2006.
	#2	The design used at the Quad Cities Nuclear Plant from January 2006 to April 2006. Refurbished and assembled in accordance with the plant procedures updated in January 2006. Similar to #1 with the addition of a tight tolerance washer placed on each guide post between the spring and the bushing.
	#3	Prototype of the new GE design
	TR	Target Rock valve and actuator considered as an alternative.
2nd Test Series (ACT-11 to ACT-18)	#1A	Same as #1. Newly refurbished and assembled in accordance with the procedures used prior to January 2006.
	#3A	Refurbished #3: new guide posts, bushing and springs; same solenoid frame with the gussets added during the 1st test series; same cover with the stiffeners added during the 1st test series; new hardened steel (Nitronic 50) pivot plate pins and bushings.
	#4	Production version of the new GE design. Essentially the same as #3A, including the gusset plates, cover with stiffeners, and hardened steel pivot plate pins and bushings.

TABLE 2. List of Shake Table Tests

Series	Test Designation	Date	Direction	Duration (hh:mm)	Input
1st Test Series (Actuators #1, #2, #3, TR)	ACT-1	2/10/06	H H V	10:00 10:00 10:00	Swept Sine Set, Scale Factor = 0.125 Swept Sine Set, Scale Factor = 0.25 (same)
	ACT-2	2/11/06	V H	7:57 10:00	Swept Sine Set, Scale Factor = 0.50 (same)
	ACT-3	2/12/06	H V	10:00 8:33	Swept Sine Set, Scale Factor = 1.00 (same)
	ACT-4	2/13/06	V H	8:00 9:00	Swept Sine Set, Scale Factor = 1.50 (same)
	ACT-5	2/14/06	H	13:11	Swept Sine Set, Scale Factor = 2.00
	ACT-6	2/15/06	H H	1:00 12:00	Swept Sine Set, Scale Factor = 2.50 0.6g rms 20- 200 Hz + 4g rms @ 151 Hz
	ACT-7	2/16/06	V	14:07	0.6g rms 20-200 Hz + 4g rms @ 151 Hz
	ACT-8	2/17/06	H	19:33	0.6g rms 20-200 Hz + 4g rms @ 151 Hz
	ACT-9	2/18/06	H V	8:46 9:20	0.6g rms 20-200 Hz + 2g rms @ 63 Hz 0.6g rms 20-200 Hz + 2g rms @ 70 Hz
	ACT-10	2/19/06	V H	10:00 9:45	0.6g rms 20-200 Hz + 4g rms @ 118 Hz 0.6g rms 20-200 Hz + 4g rms @ 80 Hz
2nd Test Series (Actuators #1A, #3A, #4)	ACT-11	3/21/06	H	12:00	0.8g rms 20-200 Hz + 4g rms @ 151 Hz
	ACT-12	3/24/06	H V	12:00 19:30	(same) (same)
	ACT-13	3/25/06	H	20:00	(same)
	ACT-14	3/26/06	V	20:00	(same)
	ACT-15	3/27/06	H	20:00	0.8g rms 20-200 Hz + 3g rms @ 85 Hz, 120 Hz
	ACT-16	3/28/06	V	19:30	(same)
	ACT-17	3/29/06	H	18:46	(same)
	ACT-18	3/30/06	V	20:00	(same)

Swept Sine Set, Scale Factor = 1

From	To	Sweep Time	Amplitude
20 Hz	50 Hz	5 min	0.25g rms
40 Hz	80 Hz	5 min	0.72g rms
60 Hz	100 Hz	5 min	1.3g rms
95 Hz	125 Hz	5 min	2.0g rms
120 Hz	150 Hz	5 min	2.0g rms
145 Hz	175 Hz	5 min	2.0g rms
170 Hz	200 Hz	5 min	2.0g rms
20 Hz	200 Hz	Broad Band	0.3g rms
		Total	4.3g rms

TABLE 3. List of Shake Table Test Events and their Resolutions

<table>
<tr><th>Actuator</th><th>Event(s)</th><th>Resolution</th></tr>
<tr><td colspan="3">Plunger Jamming</td></tr>
<tr><td>#1</td><td>After ACT-6, -9, -10
One or both guidepost springs were wedged between the guidepost and the bushing, preventing the plunger from moving.</td><td rowspan="2">#3A was redesigned precisely to address this issue. The redesign was successful because (1) the design does not allow the spring to catch between the bushing and the guidepost, eliminating both the jamming and the wear caused by the spring rubbing on the guidepost, and (2) the design does not allow the spring to come into contact with the guidepost at locations away from the bushing, eliminating any fretting wear on the guideposts caused by spring impacts.

The testing demonstrated that the new design has eliminated the wear mechanisms that damaged the original designs.</td></tr>
<tr><td>#1A</td><td>After ACT-11, -14, -15, -16, -17, 18
One or both guidepost springs were wedged between the guidepost and the bushing, preventing the plunger from moving.</td></tr>
<tr><td>#3A</td><td>After ACT-11, -13
The upper guidepost bushings bound on the guideposts, preventing the plunger from moving.</td><td rowspan="2">The original diametric clearance between the guideposts and both the upper and lower bushings on #3A and #4 was 0.002" to 0.004". After ACT-11 the upper bushings on #4 were modified so that the nominal clearance was 0.014", the bottom lip was chamfered to 0.020" clearance, and both the upper and bottom lips were rounded to a radius of 0.020". The lower bushings were not modified. The bushings on #3A were left as-is, to compare its performance to #4 with the modified upper bushings.

After modification, #4 actuated smoothly for the remainder of the tests. #3A bound again after ACT-13, but did not bind thereafter. #3A was "stickier" than #4: to manually depress the plunger, a force of about 10 lbs was required to initiate movement of #3A, while #4 required a force of less then < 1 lb.

The tight tolerance upper bushings were susceptible to binding, although this problem appears to dissipate after a wear-in period. Increasing the tolerance eliminated the problem and resulted in smoother operation of the plunger. This modification was incorporated in all of the production actuators.</td></tr>
<tr><td>#4</td><td>After ACT-11
The upper guidepost bushings bound on the guideposts, preventing the plunger from moving.</td></tr>
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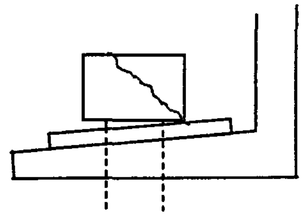
Pivot Plate & Contact Switch		
#1	<u>After ACT-10</u> The pivot pins had worn to the point were neither manual or electrical actuation of the pivot plate would lift the contact switch.	After ACT-10, it was clear that pivot pin wear was a problem - the pins would wear to the point that the pivot plate would no longer lift the contact switch (which could cause the solenoid to burn out.) This occurred on both the original designs (#1, #2) and the new design (#3). The original pivot pins were fabricated from mild steel. For the production actuators, the pivot pins were fabricated using hardened steel (Nitronic 50) and a matching bushing was pressed into the solenoid frame. For the 2nd test series #1A had the original mild steel pins, #3A and #4 had the hardened steel pins and bushings. After ACT-16, the #1A pivot plate could not lift the contact switch, and by the end of the testing, the pivot pins' OD had worn from 0.184" to < 0.100" (one of the pins broke). At the end of the testing the pivot pins and bushings on both #3A and #4 showed no measurable wear, and the pivot plates lifted the contact switches fully. *The testing demonstrated that the hardened steel pivot pins and bushings in the new design eliminated the pivot pin wear that prevented the pivot plate from lifting the contact switch.*
#2	<u>After ACT-7</u> Manual actuation of the pivot plate would not lift the contact switch, but electrical actuation would. <u>After ACT-10</u> Neither manual or electrical actuation of the pivot plate would lift the contact switch.	
#3	<u>After ACT-5</u> Manual actuation of the pivot plate would not lift the contact switch, but electrical actuation would. <u>After ACT-10</u> Neither manual or electrical actuation of the pivot plate would lift the contact switch.	
#1A	<u>After ACT-16</u> Neither manual or electrical actuation of the pivot plate would lift the contact switch. The pivot pin ODs were originally 0.184". By the end of ACT-18, they had worn to < 0.100", and one pin had broken.	
Limit Switch Condition & Operation		
#1	ACT-2: S2 non-functional ACT-4: S1 non-functional & cover off ACT-5: S2 cover off	Per plant procedures, Loctite was not applied to the #1 or #1A mounting screws. This may have caused the switches to loosen slightly and vibrate more, causing the switches to fail relatively early in the testing. Per plant procedures, Loctite was applied to the #2 mounting screws. The switches stayed functional longer. Loctite was applied to the #3, #3A and #4 mounting screws, and the mounting plate is thicker and stiffer than the #1, #1A, #2 plates. The switches remained functional and in good condition through all of the tests. *The testing demonstrated that the new design is not susceptible to the limit switch problems seen on the original design.*
#2	ACT-5: S1 mounting screw broke; switch fell off. ACT-8: S2 non-functional	
#1A	ACT-15: S1 non-functional ACT-17: S1 cover off ACT-18: S2 non-functional	
#3, #3A, #4	Switches functioned properly through all tests	
Actuator Covers		

#1, #2, #3	The bolt holes in the actuator covers starting elongating and fracturing during ACT-3 and ACT-4. Reinforcement was added after ACT-4. The reinforcement stopped the holes from fracturing, but several bolts sheared in subsequent tests.	The covers are attached to the actuator with six 1/4" socket head cap screws. The holes are at the bottom of the cover, with only about 1/4" of material between the bolt hole and the edge of the cover. During the 1st test series, some of the bolts had broken, and others had caused segments of the cover between the bolt hole and the edge to break off, rendering the bolts ineffective. 1/4" thick stiffeners plates were welded over the bolt holes. This prevented any further damage to the covers, but several bolts still sheared in subsequent tests. The stiffeners were included on all three covers in the 2nd test series. In addition, each bolt was completely hand threaded into the holes to ensure that the threads were not stripped, and then torqued to 14 ft-lbs with a calibrated torque wrench (installation of the cover bolts was not controlled in the 1st test series.) The #1 and #4 covers functioned without incident. On the #3A cover, one bolt broke, in the same location, in tests ACT-15 and ACT-17. This cover was the same cover used in the first test series for #3. The bolt hole in question suffered significant damage during the first series (before the stiffeners were put on), losing a fair amount of material on the original box around the bolt hole. As a result, despite the stiffener, there was not complete metal-to-metal contact between the cover and the base plate at this bolt hole, resulting in a local bending stresses on this bolt. *The testing demonstrated that the stiffeners together with the controlled bolt installation used in the 2nd test series eliminated the problem with the cover bolts.*
#3A	One bolt broke, in the same location, in tests ACT-15 and ACT-17.	
Other Issues		
#3	<u>During ACT-4</u> During ACT-4, Dresser #3 developed longitudinal cracks in the crotch of the steel angles connecting the solenoid frame to the base plate. The cracks went completely through the thickness of the angle.	The cracks in #3 were weld repaired and gussets were added. The same solenoid frame (with the gussets) was used in the 2nd test series, and the gussets were incorporated #4 and all the new production actuators. With the gussets installed, the cracking did not re-occur, even though the actuator was subsequently subjected to higher vibration levels. This cracking did not occur on #1, #1A, #2, none of which have gussets. The cracking is attributed to two factors: (1) the new designs have a heavier guide post support structure, and (2) the new designs have a rubber isolation pad between the bottom of the frame and the base plate. Both factors increased the bending stress on the angles. The gusset plates reduce these stresses. *The testing subsequent to installing the gussets demonstrated their effectiveness. All production actuators have the gussets.*

#4	**<u>During ACT-16</u>** The bolt heads broke on three of the four bolts that attach the solenoid frame to the base plate.	The bolt heads broke along the red line shown in the following sketch.  As the lower leg of the angle thickens towards corner, the bolt head did not rest flush on the washer, resulting in an eccentric point load on the bolt head. Vertical accelerations caused the bolt head to fail in shear (diagonal tension). #4 was shipped back to the manufacturer, where the lower leg of the angle was machined (about 0.020" of material was removed) so that the washer laid flat and the bolt head made full contact with the washer. The bolt heads did not fail in any of the subsequent tests (note that ACT-17, 18 was a repeat of ACT-15, 18). *This change was incorporated in all of the production actuators.*
TR	**<u>During ACT-3</u>** All the electrical leads in the Target Rock actuator broke. **<u>During ACT-4</u>** The support cylinder connecting the actuator to the valve body broke.	The Target Rock actuator was more lightly damped than the Dresser actuators, resulting in higher accelerations. Accelerations of 100g rms were measured on the cover prior to the support cylinder breaking. *As a result, the Target Rock actuator was judged inappropriate for a potentially high vibration environment, and removed from consideration.*