Validation of Motor-Operated Valve Frequency Domain Analysis Techniques

Stan Hale
CRANE Nuclear, Inc.

Abstract

Frequency domain analysis of motor current data is used to identify Motor Operated Valve (MOV) degradations that are not normally detected by conventional time domain signature analysis. CRANE MOVATS and a group of utility customers conducted validation tests of MOV frequency domain analysis techniques during the second half of 1997 using Motor Current Signature Analysis (MCSA) technology licensed from Oak Ridge National Laboratory (ORNL). This paper will describe the validation process, analysis results and reveal the insights gained from frequency domain analysis of MOV motor current data.

A calculational model that predicts the fundamental frequencies of rotating MOV components was developed for the validation program and field use. A key validation objective was to develop an understanding of how the frequency of predicted events varied in both position (frequency) and magnitude for identical actuators and how these differences affected overall performance.

Two groups of identical actuators were assembled for the lab portion of the validation testing. Five Limitorque SMB−00 and five SMB−0 actuators were configured to be identical to typical actuators in use at the Vermont Yankee and Farley Nuclear Plants. Motors, gearing, bolt tightness, gaskets, shimming and supply voltages were changed in a controlled manner in order to assess the potential effect on the baselines. Various degradations were induced and direct torque and thrust signals were monitored during the entire process in order to capture potential correlations.

Introduction

"At-the-valve" diagnostic testing played a key role in improving the setup and performance of safety-related MOVs. As a result of ongoing programs, safety-related MOVs will be subjected to periodic preventive maintenance activities on a regular basis and are not expected to undergo rapid change due to degradation. Likewise, costly "at-the-valve" testing to detect an occasional mechanical degradation or program weakness does not appear to be the best use of nuclear maintenance and engineering resources. As an alternative, time domain analysis of motor torque developed from data acquired at the motor control center (MCC) provides an effective margin assessment tool and frequency domain overlays reveal subtle changes in gearbox or motor performance that may be precursors to more significant problems.[1]

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[1] Use of time domain motor torque data to assess MOV margin has been discussed in previously published engineering reports and technical papers. This paper does not provide additional information on use of MOV motor torque technologies. Instead, this paper focuses exclusively on MOV frequency domain analysis.

Nuclear plants that plan adoption of MCC based diagnostics for periodic verification of MOV capability should employ both time domain and frequency domain signature analysis tools. Time domain analysis has been used in MOV diagnostics for many years and is well understood by MOV diagnostic technicians. Though frequency domain analysis has been the cornerstone of vibration analysis, the industry's knowledge of MOV frequency response is extremely limited.

The ORNL MCSA technology employs Fast Fourier Transform (FFT) analysis of electric supply current signature data in order to characterize motor and actuator frequency response. FFT analysis tools that are built into the CRANE MOVATS MC² TM software provide the critical frequency analysis capability needed for MOV diagnostics. The frequency distribution identifies changes in actuator performance that are not normally detected with conventional time domain analysis tools. FFT analysis of current data is also used to detect degradation that may reduce the output torque capability of the motor.

CRANE MOVATS organized an extensive test program to validate frequency analysis techniques. This test program was the result of a customer request to identify and validate a frequency domain analysis approach that would support MOV periodic verification activities. A customer working group was formed and began work in May 1997. The working group identified MOV operating characteristics or degradations that should be visible in the frequency domain, an analysis method and the type of validation tests required to use this information in a periodic verification program. CRANE MOVATS contributed the engineering resources for development of the calculational models required to predict the MOV operating characteristics (frequencies) defined by the working group.

CRANE MOVATS sponsored the test program activities by providing the test equipment and test specimens, compiling the analysis results and developing the final report. The customers/users participated in most of the actual testing, supported analysis activities and contributed field test data. Customer/user participation was encouraged because it ensured development of industry expertise and a critical reference base for this evolving technology. Customer participation also helped complete this portion of the validation effort before users completed development of Generic Letter 96-05 periodic verification programs.

Basic Frequency Analysis

In its simplest form, MOV frequency analysis is the graphical representation of repetitive events over some unit of time often expressed as revolutions per minute (RPM) or cycles per second. For example, Figure 1 is a time domain graph of an AC current waveform. Using time domain analysis tools the analyst can determine that the period is consistently .0167 seconds from peak to peak or 60 peaks per second. When converted to the frequency domain using the FFT algorithm (see Figure 2) the graphical presentation identifies one sinusoidal waveform occurring at 60 cycles per second.

MOV technicians perform simplified frequency analysis when counting the number of peaks per minute in a time based signature in order to identify the RPM associated with a cyclic load. In Figure 3, a gross cyclic load that is consistent with the rotating speed of the drive sleeve is easy to identify in a motor current signature without complex FFT tools.

The 60 cycle instantaneous motor current signal also carries a wealth of previously unseen frequency domain information. Small perturbations of the current signal occurring at frequencies consistent with the speed of rotating MOV components can be detected by FFT analysis. The most obvious MOV frequencies include motor speed, slip pole, rotor bar, stator slot, motor pinion, worm gear tooth mesh and a number of frequencies from the many actuator bearings.

The highest observable frequency is limited to one half of the sample rate used during data acquisition and is often referred to as the Nyquist frequency. The Nyquist frequency becomes important when events occur at greater than one half the sample rate. Stator slot, rotor bar pass frequency and, at times, motor pinion tooth mesh frequency occur at greater than 1000 Hz. In order for the FFT to reveal these frequencies the sample rate must be greater than twice the expected frequency.

When significant events occur at frequencies greater than the Nyquist frequency they may be folded back into the frequency spectra as aliases. Alias frequencies complicate the analysis process and should be avoided by using higher sample rates.

Figure 4 is one example of a frequency domain analysis of MOV motor current calculated from data acquired at two thousand samples per second. The horizontal axis defines frequency in Hz and the vertical axis defines amplitude. Each peak along the frequency axis is a product of the actuator rotating components or the electrical signal.

The data acquisition sample rate also determines the frequency resolution of the FFT. Figure 5 illustrates the difference in resolution obtained with data sampled at two thousand samples per second versus six thousand samples per second. The slower sample rate identified one large peak that was further refined to reflect a peak with small sidebands at the higher sample rate.

Sample size also effects the frequency resolution of the FFT. Figures 6 and 7 are FFT analysis results for the same MOV motor current data taken at six thousand samples per second. In Figure 6 only the first 2048 data points were used in the FFT calculation versus 65,536 data points in Figure 7.

It should be noted that even though the higher sample size of Figure 7 defined many more frequencies the large peak in the center of Figure 6 corresponding to the motor speed is visible at approximately the same frequency. In this example, the FFT algorithm was able to resolve the motor speed frequency within one third of a second or less than ten motor revolutions.

Actuator Frequency Model Description

A model that identifies the fundamental frequencies of the typical Limitorque actuator was needed for the validation testing and field use. Detailed Limitorque actuator drawings were reviewed in order to identify the rotating components that should contribute to the motor current frequency spectrum. At least twenty-nine fundamental frequencies are possible based on internal components alone. Sidebands and harmonics of the fundamental frequencies, the sixty cycle carrier, line voltage noise and aliasing also contribute to the frequency spectrum.

There are two motor bearings (typically identical) contributing four fundamental frequencies in the typical Limitorque motor. If the inboard and outboard bearings are different, up to eight frequencies would be possible. Information on the motor bearings was not available prior to the testing. Therefore, each motor was disassembled in order to identify the bearing model numbers. The same basic bearing specifications were found in five, ten, fifteen and twenty-five FtLb motors. However, a range of manufactures were discovered each with subtle differences that can affect the frequency spectrum. Additional motor frequencies that should be anticipated include motor speed, slip pole, stator slot and rotor bar pass frequency. The number of rotor bars and stator slots were also verified during disassembly.

There are four primary bearings contributing sixteen fundamental frequencies in the typical Limitorque actuator gearbox. The Limitorque part number and cross reference to the industry standard model number was obtained and used to calculate the various bearing frequencies. Table A identifies the contributing components and number of frequencies calculated by the MOV frequency model. Once component specifications are entered, the model requires an accurate input of motor speed in order to perform the necessary calculations. Motor speed is obtained from the FFT analysis.

Motor and Actuator Selection Process

Detailed MOV database printouts from the Farley and Vermont Yankee nuclear plants were reviewed in order to identify common actuator gearbox configurations in use at the two plants. Both plants have Limitorque SMB−00 actuators with 72:1 ratios. These actuators have a 45:1 worm gear ratio, a 25 tooth motor pinion gear and a 40 tooth worm shaft gear. Both have four pole 1800 RPM motors installed and Vermont Yankee has some with 1900 RPM DC motors.

Both plants also have SMB−0 actuators with a 46.25:1 ratio. These actuators have a 37:1 worm gear ratio, a 32 tooth motor pinion and a 40 tooth worm shaft gear. The motors at Vermont Yankee are four pole 1800 RPM AC and 1900 RPM DC. The motors at Farley are all two pole 3400 RPM.

Table A MOV Frequencies

Motor	9 Frequencies	Gear Box	20 Frequencies
Motor Speed	1	Worm Shaft	1
Slip Pole	1	Worm Shaft Bearing	4
Stator Slot	1	Spring Pack LNB	4
Rotor Bar	1	Limit Switch Gear	1
Motor Bearings	4	Worm Gear Tooth Mesh	1
Motor Pinion	1	Drive Sleeve	1
		Upper Thrust Bearing	4
		Lower Thrust Bearing	4

There are also a number of SMB−00 actuators at Farley with a 31.9:1 ratio and SMB−0 actuators with a 39.1:1 ratio.

All of these actuator configurations, including some of the identical plant MOVs, were tested as part of this test program.

Test Plan Description

One potential approach to MOV frequency analysis, trending and acceptance criteria is to establish a baseline when the actuator is in a known good condition. This condition usually follows baseline testing. Future MCC test results (motor torque margin and frequency domain signatures) should be overlaid in order to assess change.

The baseline and overlay approach presents two problems. Identical MOVs with acceptable baselines may produce different frequency domain signatures and MOV operating characteristics will change slightly over time without affecting operability. A methodology is needed in order to assess baseline frequency response and minor changes, yet not be so conservative that it needlessly increases the at-the-valve testing requirements. The test plan was developed with these problems in mind.

The first phase of test plan activities required dynamometer tests of all program motors. Torque versus speed, torque versus power and frequency domain characteristics under load were graphed and overlaid in order to identify candidates for the actuator baseline tests. The actuator baseline performance tests were performed during the second phase of test plan activities. Two groups of identically configured actuators were tested at various steady state loads. The frequency analysis results from these tests were compared to the predicted frequencies of the calculational model. Various alterations/degradations were introduced in the third phase of test program activities. This data was used to assess the sensitivity of the frequency analysis results.

The test program was performed under the CRANE MOVATS 10CFR50 Appendix B nuclear quality assurance program. All instruments used during the test program were calibrated before and after use. A detailed validation test plan, VTP−16.0, was developed and approved based on QA program requirements. The lead test engineers that supervised the test activities and performed the data storage and transfer functions met the qualification requirements of CRANE MOVATS QAP 2.0.

Motor Dynamometer Test Plan Description

A group of 15 FtLb, four pole, 1800 RPM motors with identical nameplate specifications were tested on a CRANE MOVATS precision dynamometer in order to identify five motors with nearly identical performance. Torque-speed overlays and FFT characteristics under load were used to identify the representative motors and bad motors were identified for future degradation testing. A motor was considered "good" if it produced greater than 120% of its nameplate torque without reaching the "knee" or showing a sharp downturn of its torque-speed capability.

Each motor used in the test program was performance tested on the motor dynamometer in order to verify this capability and identify potential motor degradation. Motor current and voltage data was captured with the Universal™ Diagnostic System (UDS) and MC² ™ systems during the tests. Motor torque and speed data from the dynamometer was captured with the UDS. Motor winding resistance was monitored between tests and test sequences modified as appropriate to minimize motor heating and potential winding damage. All tests were performed at 100 and 110% voltage which is typical of voltages available during actual plant testing.

The full load torque-speed tests were performed at one FtLb increments in both directions. The dynamometer ramp test program was set to disconnect at 120% of the motor's nameplate torque rating. Steady state load tests were then performed in steps equivalent to 25, 50 and in some cases 75% of nameplate torque for the purpose of evaluating steady state frequency characteristics at these loads. A minimum of ten seconds of data acquired at two thousand samples per second with the UDS and the same ten seconds at six thousand samples per second with the MC^2 TM was obtained at each load. This steady state dynamometer data was used to establish motor specific frequency domain characteristics without interfering gearbox frequencies.

Motor Dynamometer Results

Seven four pole, 1800 RPM, 15 FtLb motors passed the initial performance screen. The five motors with the closest performance characteristics were chosen for the actuator effects testing. In Figure 8, motors 1, 2, 3 and 5 produced nearly identical performance curves. Though motor 4 ran at a slightly lower speed it was also selected.

The clockwise (CW) and counter clockwise (CCW) results were overlaid to assess differences in the torque-speed relationship due to direction (see Figure 9). CW versus CCW speed differences produce different frequency spectra for open versus close valve operation at load. This means that for an identical running load in both directions the event frequencies will be shifted up or down depending on which direction the motor's torque versus speed relationship is strongest. Motors 1, 2 and 5 performed exactly the same in both directions. Motors 3 and 4 indicated minor differences between the CW versus CCW direction.

Dynamometer tests were also performed at nameplate and 110% voltage. Differences in the torque-speed relationship due to changes in available voltage may affect frequency domain signature characteristics. These tests were performed because available bus voltages may vary depending on plant operations and electrical grid configuration.

The bus voltage increase strengthened the motors and flattened the torque-speed curves (see Figure 10). At low loads the effect is negligible. Since packing load and internal running load should require less than 25% of the motor's nameplate capability, typical voltage variations are not expected to mislead the frequency analysis.

The phase A instantaneous motor current was used to generate frequency domain signatures for each motor at each load level. Figure 11 provides an overlay of the motor speed frequency for each motor at zero load. The large peak at 29.95 Hz corresponds to a motor speed of 1797 RPM. All five motors ran at 1797 RPM when uncoupled from any load. The fact that all five motors ran at the same speed at zero load was encouraging. However, the different amplitudes of the motor speed peak was an early indication of potential acceptance criteria complications.

As expected these AC MOV motors lose speed under load. In Figure 12 the frequency data is used to determine motor speed at the minimum dynamometer load of 2 FtLbs. In this particular example the motor speed frequency was determined from the upper sideband of 120 Hz. All five motors slowed to 1764 RPM under two FtLbs of load.

Figure 13 further illustrates the motor's response to load. The observed loss of speed is proportional to torque under these steady state loading conditions. Another load indicator visible in Figure 13 is the slip pole frequency which shows up at the low end of the frequency spectrum. Where motor speed moves to the left or slows under load, slip pole frequency moves right or increases. The slip pole frequency is also more sensitive and thus moves more in relation to load. Slip pole and other motor frequencies can be used to verify that the proper motor speed has been selected since these frequencies have a direct relationship to speed.

The slip pole frequency is also a good indicator of motor degradation. High slip pole amplitude followed by several successive harmonics are indicators of motor electrical or mechanical imbalance. Electrical imbalance is a common symptom associated with rotor bar damage.

The MOV frequency model developed as part of this program is only as good as the motor speed input. If the motor speed value is incorrect the fundamental frequencies will be miscalculated. Therefore, the actual motor speed recorded during all steady state dynamometer tests and its corresponding frequency data were compiled in order to assess the accuracy of the motor speed calculation. The difference between the digital tachometer and calculated motor speed is illustrated in Figure 14. In all cases the maximum error was less than .66% of actual. These errors are well within the tolerance of the digital tachometer and a strong argument can be made that the frequency data is more accurate. In fact the predominate error associated with frequency derived motor speed comes from the sample rate used during data acquisition. Data acquisition sample rate related errors result from the FFT's ability to resolve frequencies.

Actuator Test Plan Description

Five identical Limitorque SMB−00 and five identical SMB−0 actuators were assembled for this program. The SMB−00 actuators were configured with a 72:1 ratio and the SMB−0 actuators with a 46.25:1 ratio. The 5 "good" motors were installed on five of each actuator model for baseline testing on the actuator effects test stand (AETS). The AETS is a hydraulic device that simulates actuator load profiles. A hydraulic cylinder is attached to the valve stem and the load controlled by restricting the flow rate from the back side of the cylinder during actuator operation. The resulting thrust reaction simulates valve loading.

The initial baseline tests were performed with a 1500 lb simulated packing load. The simulated packing load was increased to 25 and 50% of the motor's nameplate torque rating. These tests were also run at 100 and 110% voltage for each load level.

The frequency results for all tests were compared to the actuator frequency model for each actuator configuration tested. The primary purpose of this step was to verify that baseline performance for all 5 identical actuators was similar and the frequency spectrum was correctly predicted by the frequency model.

Actuator Baseline Results

It became obvious after very few tests that it was not possible to assemble five different actuators and control test conditions with enough precision to produce identical FFT results. Though key events still show up in the correct general area, tolerances within the actuator, lubricant distribution (both internally and on the stem threads) and small changes in packing load equate to differences in the frequency spectra.

The position of each frequency may be shifted slightly by changes in motor speed and the amplitude of peaks can be different without gearbox contributions. Because of these complications, and the amplitude inconsistencies seen among identical motors on the dynamometer, each MOV test was analyzed "stand-alone" and compared to the frequency model generated from its exact motor speed data.

In Figure 15 all low frequency peaks were predicted by the frequency model. This is not always the case. From time to time unexplainable frequencies show up that could be the result of aliasing or noise. Unexpected frequencies tend to distract an inexperienced analyst and should be minimized by using higher data acquisition sample rates. Defining what could be a harmonic of a lower frequency, an alias signal or noise can be a futile and meaningless exercise. This is where the frequency model helps. The frequencies of important components are predicted ahead of time and the analyst is focused in on these locations. Whether the component frequency shows up or not is not always critical. For example new bearings rarely show up in the frequency spectra. Conversely, the presence of peaks at certain bearing frequencies may warrant further attention.

One of the more promising products of the validation program was the consistent change in motor frequencies due to load. The peak at the far right in Figure 16 is the rotor bar frequency at zero load. At 1500 pounds of packing load or 2 FtLbs of motor torque the frequency dropped from 1318 Hz to around 1305 Hz. The peak at approximately 1264 Hz is the result of 3.75 FtLbs of motor torque.

A change in the position of these frequencies indicates higher or lower load on the motor depending on the direction of the change. Changes in packing load, stem factor and actuator efficiency would change the load on the motor.

Modified Configuration and Degradation Tests

One of the five SMB–00s and one of the five SMB–0s were selected for modified configuration and degradation testing. A number of alterations or degradations were induced in each actuator and tests performed to identify potential changes in the frequency domain signature characteristics. Each retest was performed at the normal 1500 lb packing load.

Alterations included changing the upper-bearing housing cover position by installing an extra gasket, removing all gaskets, loosening two adjacent upper housing cover bolts, then loosening the remaining bolts. The correct bolt torque and gasket thickness were re-established and tests performed to verify that the baseline frequency response had been re-achieved before changing motors and gearing.

A 3400 RPM motor was installed and tests run with the original gear ratios. The pinion ratio was changed and tests run with the 3400 RPM motor. The original motor was reinstalled and tested with the new ratios. The original gear set was reinstalled and teeth were broken from the pinion and wormshaft gears. Frequency data for all of the above alterations or degradations were analyzed as well as the corresponding time domain data.

Modified Configuration Results

As expected, gear ratio and motor speed changes moved the position of predicted frequencies. In Figure 17 the gear ratio was changed from 72:1 to 31.9:1. Though motor speed remained in the same general area, the worm gear tooth mesh frequency moved up to the predicted location.

The worm gear tooth mesh frequency was also more pronounced with less side-banding at the faster gear ratio. The sidebands on the slower 72:1 gear ratio were equivalent to drive sleeve rotation. The time domain data for these two ratios revealed that though slightly more motor torque was required with the higher speed gears, actuator efficiency was 10–15% better. Similar results have been observed during other industry test programs. Though not specifically a part of this test program, this trend continued and supports the contention that there is a relationship between motor and gear speed and actuator efficiency. The evolving Limitorque actuator efficiency questions may warrant additional investigation in this area.

Degradation Results

The degradations used during testing were close simulations of degradations or errors that can be accidentally induced during the maintenance process. Actuators with extra upper housing cover gaskets or no gasket at all are found from time to time. Incorrectly tightened or loose upper housing cover bolts are the most likely mechanical degradation in today's environment.

Figure 18 identifies the change in the frequency spectra when the gasket is missing and the drive sleeve is heavily loaded due to the torque on the upper housing cover bolts. In this example the motor torque required during running load increased slightly and sidebands increased around worm gear tooth mesh frequency and its first harmonic. The sidebands are equal to the drive sleeve rotation frequency. The time domain data confirmed the increased motor torque and a drop in actuator efficiency.

The extra gasket and loose bolt data tended to quiet the frequency spectrum in the open direction since the drive sleeve is being pulled down by the stem nut which is pulling up on the stem. The close direction data produced more noise and sidebands as expected. In all cases the actuators tended to be most efficient when the worm gear tooth mesh peak was well pronounced with little to no sidebanding.

Field Data

Field data acquired during prior outages at the Vermont Yankee and Farley nuclear plants was gathered, analyzed and compared to the lab results. These comparisons were complicated by the slower one thousand Hz sample rate used during the field testing. The Nyquist frequency for the field test data was limited to 500 Hz and as a consequence higher frequencies were folded back into the frequency spectrum as alias frequencies. Otherwise, the field data looked very similar to the lab data and the predicted actuator frequencies were found in the proper locations.

Zero load and steady state dynamometer data from Vermont Yankee was compared to similar steady state load data for the 15 FtLb, four pole 1800 RPM motors. Figure 19 indicates that the zero load frequency results, particularly the speed frequency for the Vermont Yankee motor was identical to the five test program motors.

This particular motor also revealed a large peak at the motor bearing outer race frequency. In Figure 19 the peak appears at 42 Hz. However, the real frequency is 78 Hz. The 42 Hz position is actually a lower sideband of 120 Hz. This was verified by finding the corresponding upper sideband at 198 Hz.

This motor's output torque capability was verified to be 120% of nameplate during the same dynamometer test that revealed the higher amplitude 78 Hz bearing frequency. Additional tests on this motor are planned during the 1998 refueling outage to verify suspected alias frequencies and to trend changes in motor performance that may affect output capability.

Several of the field motors revealed higher amplitudes at the motor bearing outer race frequency. These bearings are sealed and not normally lubricated during maintenance which may lead to degradation over time.

Dynamometer steady state load data from Vermont Yankee was also compared to the lab results. Figure 20 indicates that the plant motor responded similarly to load when compared to the five test program motors. In this example a 15 FtLb, four pole, 1800 RPM motor from Vermont Yankee was overlaid with dynamometer data from test program motor 2 at a similar load.

Plant actuator data was also compared to the AETS baseline data. Figure 21 is an overlay comparison of field data from Vermont Yankee and two test program SMB−00 actuators with the 72:1 ratio.

Figure 22 is an overlay comparison of field data from Farley and a test program SMB−0 actuator with the 46.25:1 ratio and a 3400 RPM Motor. Figure 23 is an overlay comparison of field data from Farley and a test program SMB−00 actuator with the 31.9:1 ratio and 3400 RPM motor. In both Farley examples the fundamental frequencies calculated by the frequency model are easy to locate. There are also minor differences in the appearance of the frequency spectrum but nothing indicative of motor or gearbox degradation.

Conclusions

The commercial nuclear power industry has expended considerable resources over the past decade improving the performance of safety-related MOVs. The current concern is how to verify that MOV margin established through costly "at-the-valve" test programs does not erode. Increases in the MOV thrust or torque requirement due to age is one degradation mechanism that can erode margin. Increases in thrust requirements due to valve aging may continue to be assessed through limited "at-the-valve" differential pressure test results.

A decrease in actuator output capability is another effect of degradation that erodes margin. Changes that affect actuator output capability include stem factor degradation, gearbox lubricant or mechanical degradation and motor degradation. Evidence of these degradations, including increases or decreases in packing load can be detected and evaluated with MCC diagnostic technology.

The specific conclusions of this test program that support the usefulness of frequency domain analysis include:

- The motor's rotating frequency is an accurate indicator of motor speed
- Motor speed is a function of load
- The MOV model accurately predicts the frequency of the key rotating components
- FFT results are repeatable provided sample rates and sample size do not change.
- FFT results are most sensitive to mechanical change

In addition, the motor current sample rate used during data acquisition should be greater than twice the highest expected frequency event and the largest repeatable sample size should be used for trending frequency domain overlays.

Frequency analysis results also confirm or provide a second verification of the motor torque margin analysis. When the motor torque analysis indicates that output capability has degraded, frequency analysis should identify a corresponding decrease in motor speed and similar change in other motor related frequencies. Used together, motor torque margin and frequency domain analysis results provide a strong technical basis for extending the frequency of "at-the-valve" testing for the purpose of assessing the capability of safety-related MOVs.

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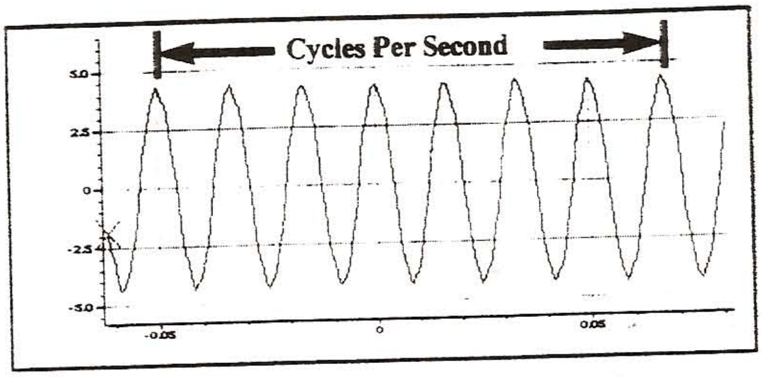

Figure 1

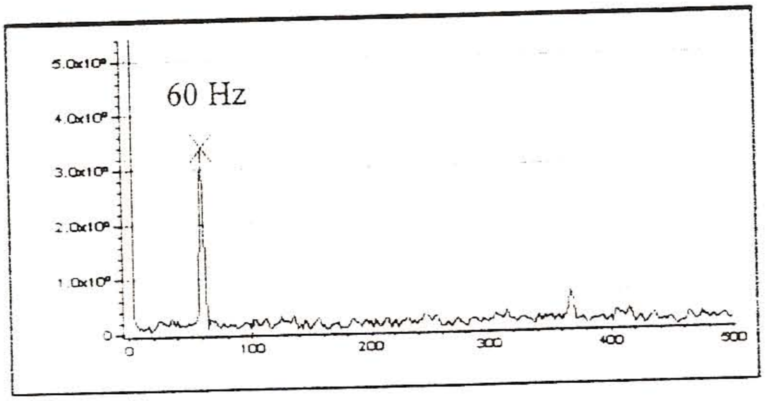

Figure 2

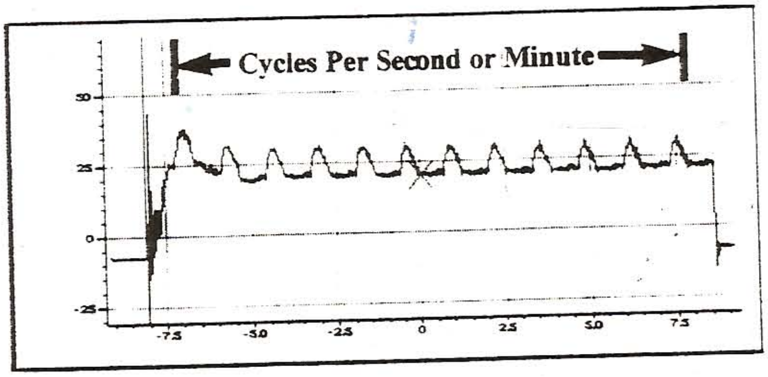

Figure 3

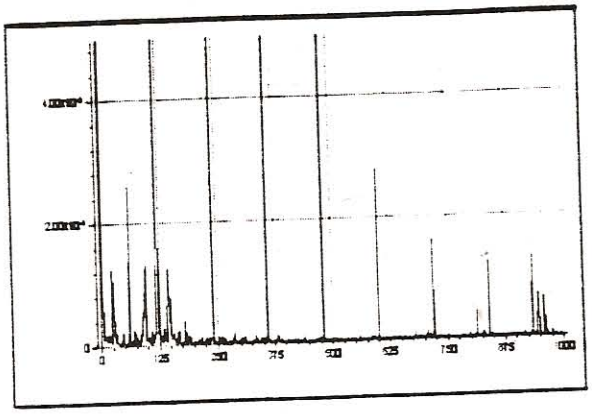

Figure 4

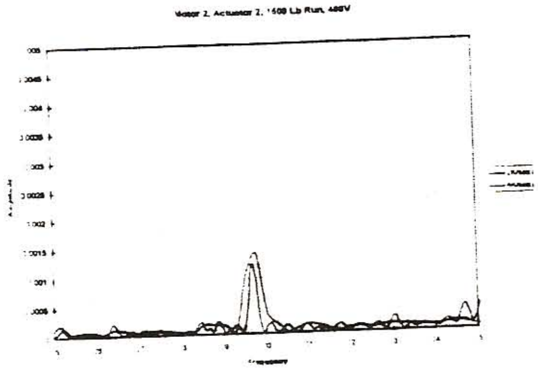

Figure 5

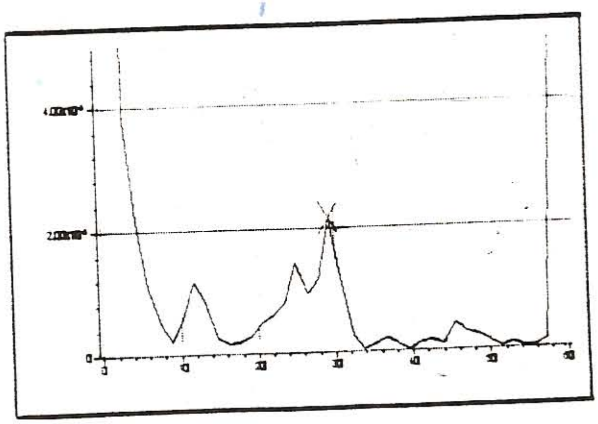

Figure 6

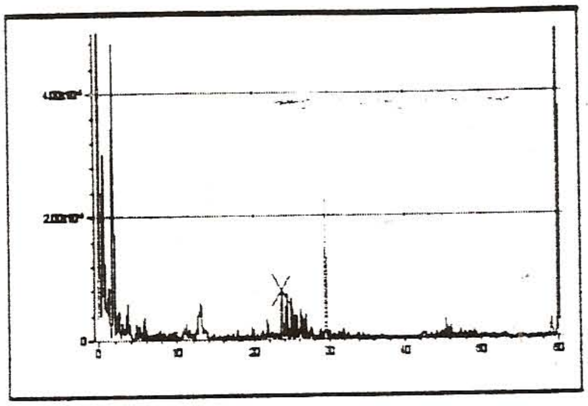

Figure 7

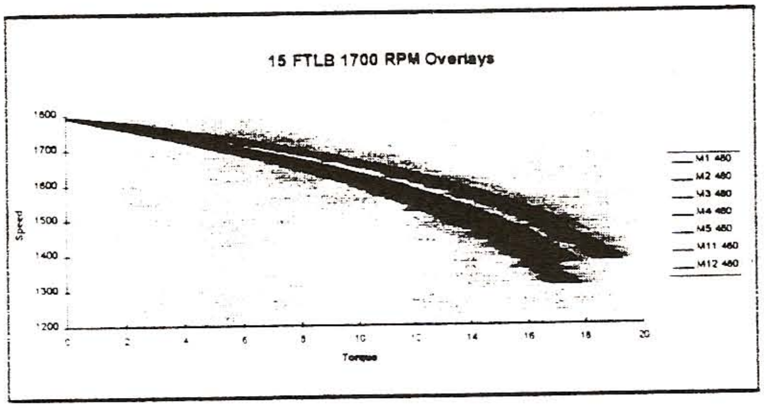

Figure 8

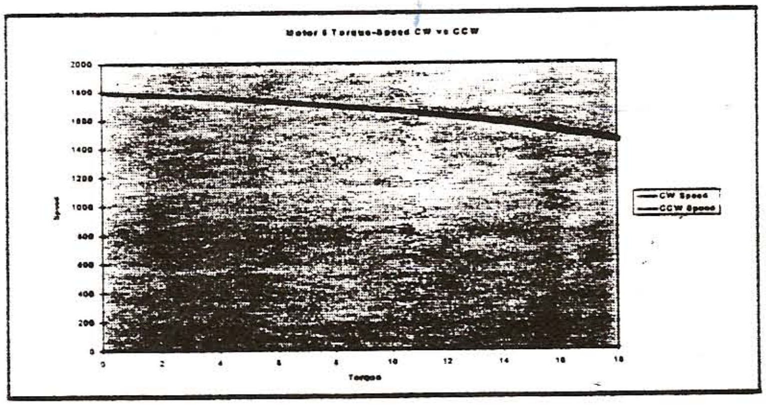

Figure 9

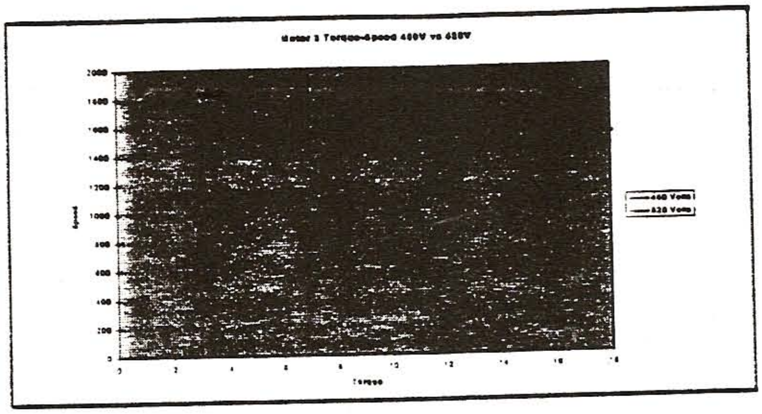

Figure 10

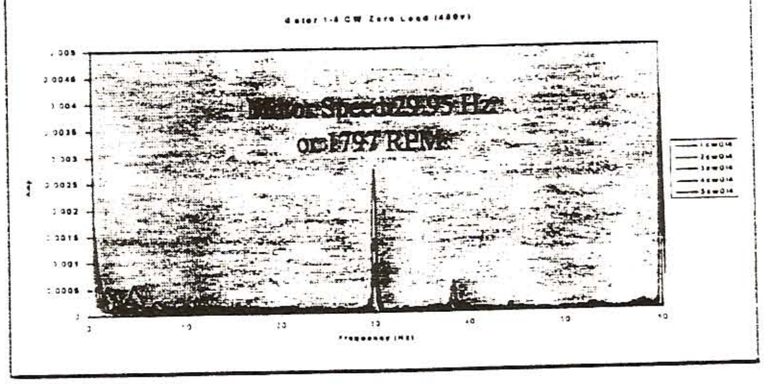

Figure 11

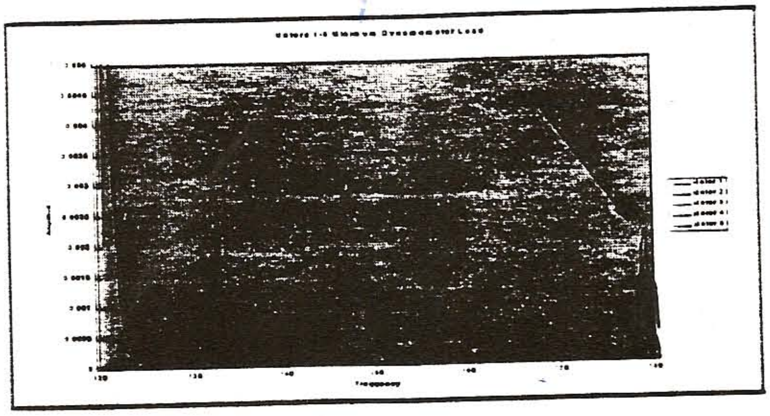

Figure 12

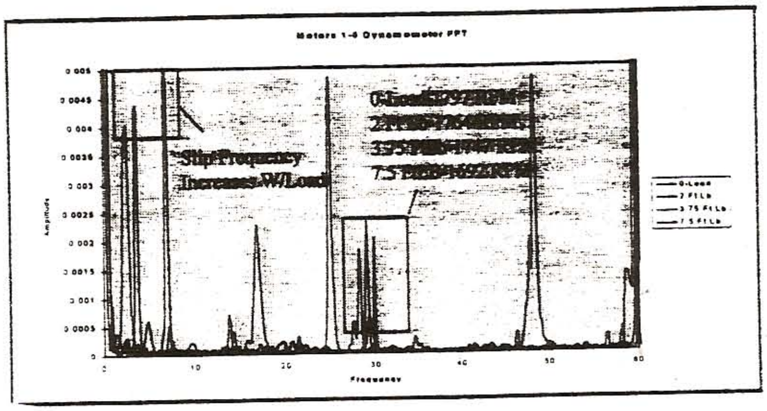

Figure 13

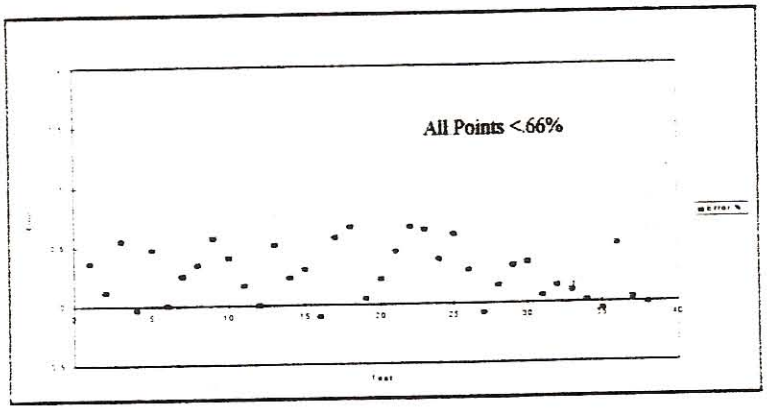

Figure 14

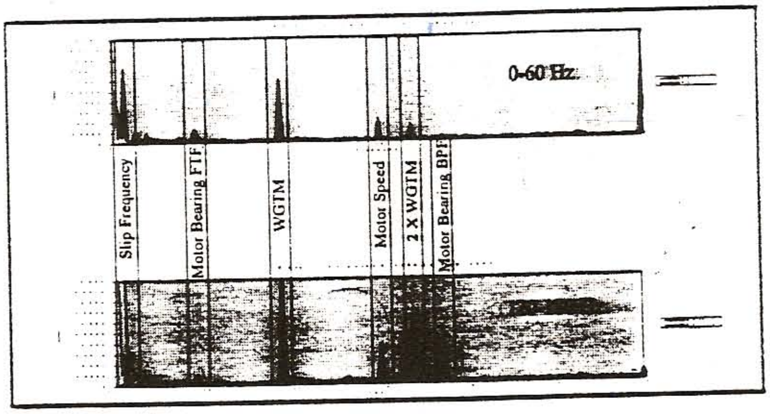

Figure 15

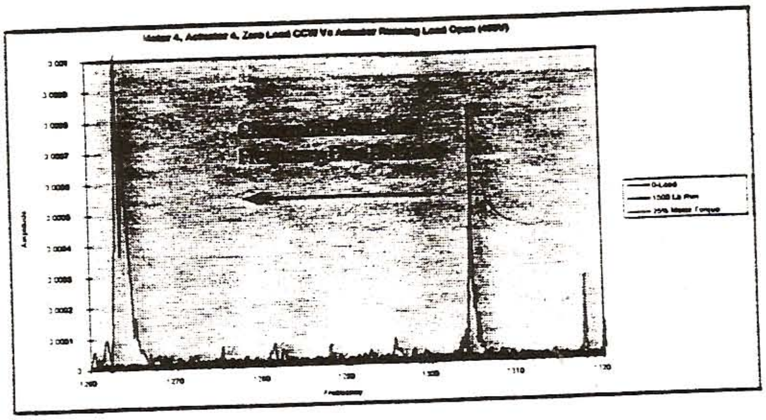

Figure 16

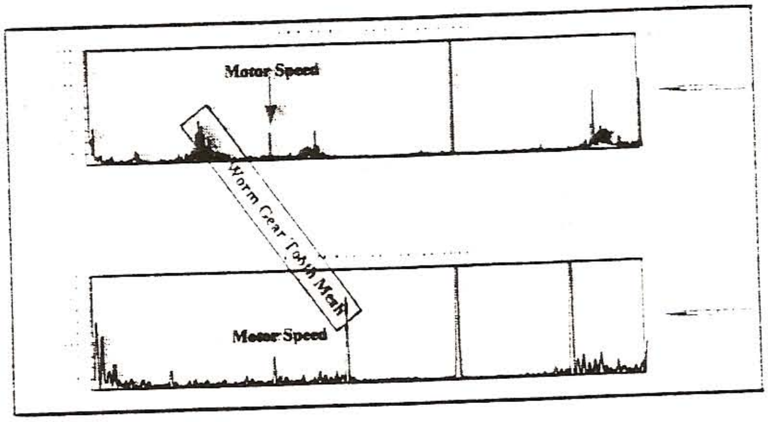

Figure 17

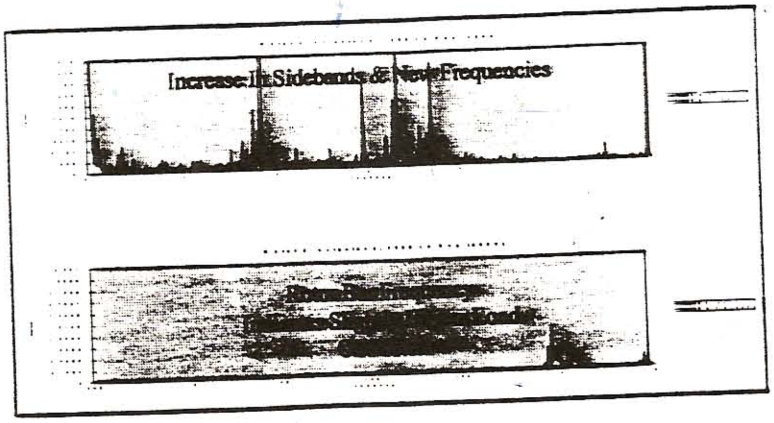

Figure 18

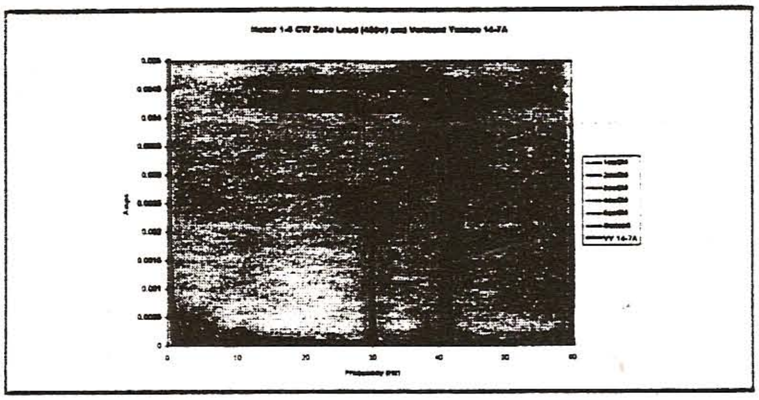

Figure 19

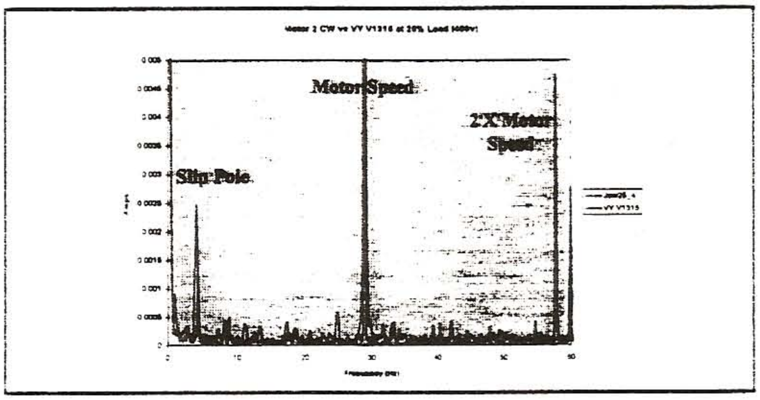

Figure 20

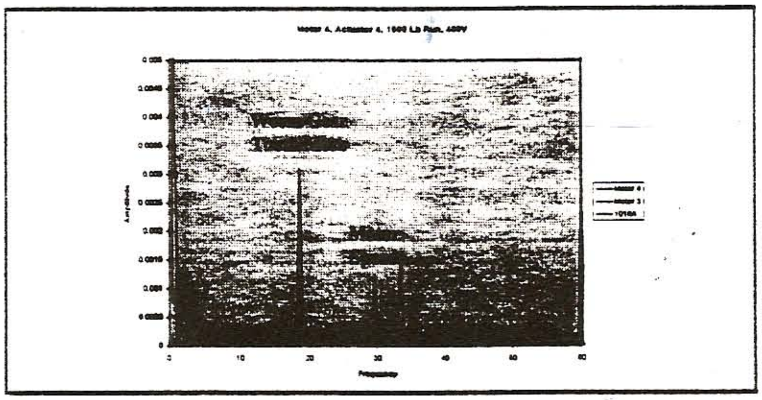

Figure 21

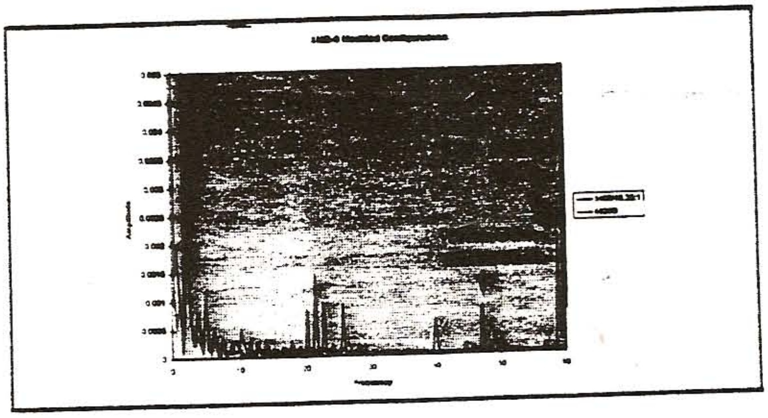

Figure 22

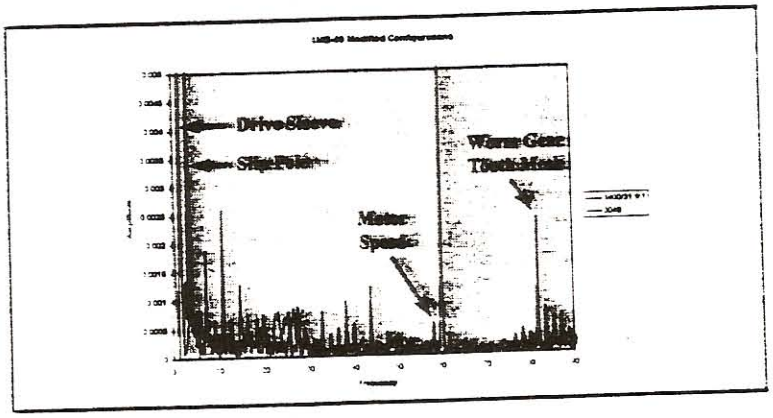

Figure 23