Vibration Certification Case Studies
Vertical Pump Machinery
Controlled with Variable Frequency Drives
Executive Summary

The use of variable frequency drives, or variable frequency controllers, in the pumping industry has become very common place over the last five years. Pump manufacturers and pump machinery operators have been plagued with the operation of these machines due to vibration issues made very complex by variable speed operation. This document presents eight case histories of vertical pump machinery that have had difficult certification processes due to the variable speed operation. The objective of the document is to alert operators, machinery suppliers, contractors, and pump station designers of the potential complications and some remedies that REAI or REAI clients have implemented. Additionally, the vibration analysis technologies and the vibration test technologies have evolved such that the operation of variable speed machinery can be successfully implemented at the certification phase when proper steps are implemented early in the design phase of the pump stations and the pumping machinery and followed up at the equipment startup certification phase.

Design analysis of machinery using finite element analysis has been implemented for over 30 years and has been integrated into the personal computer workstation for over 15 years. By appropriate finite element models the machinery can be constructed to minimize dynamic problems associated with variable speed operation. Even so, there are situations where the technology hand off from pump station system designers to pump machinery manufacturers and pump machinery operators has resulted in severe startup vibration problems. By the examples presented herein the reader can become aware of the principle issues and some remedial actions.

Machinery vibration analysis has witnessed an explosion of electronic devices designed to be used in predictive maintenance of machinery over the last two decades. These devices started out as single channel data collectors and then dual channel data collectors. These machinery vibration analyzers (MVAs) can gather tremendous volumes of data and they can manipulate the data through semi-automated reporting systems for predictive maintenance vibration surveys. As such, these instruments are especially useful for simplifying the startup vibration certification process of new machinery.

For well over forty years startups of large machinery have included the use of tape recorders and signal analyzers. The instruments were quite large and expensive and were not of use in the startup of vertical pump machinery due to cost ratio of the machinery and the instrumentation. Over the last five years or so the microelectronic technology industry has provided multi-channel digital signal recorder/analyzers (DSRAs) for use in startup vibration certification. A modern 24 channel digital signal recorder/analyzer will easily fit in a shoe box. Some of these devices can be daisy chained to well over one hundred channels for aerospace and turbo machinery startup tests. No special evaluation of these analyzers is provided herein, although they are employed in all the examples presented.

For vertical pump machinery over 500 HP the use of a multi-channel digital signal recorder/analyzer can dramatically reduce the data acquisition time. On very large machines in excess of 2000 HP these multi-channel signal analyzers can act like medical CAT Scan devices for a complete evaluation of the machinery health at relatively low cost and time expended compared to the cost of operation of the machinery. The multi-channel digital signal analyzer can be used to perform resonance evaluations through the operating range without performing complex bump test procedures. Essentially the variable speed operation and pump system transients excite the resonances and the digital signal analyzer captures the complete event. The effectiveness of multi-channel digital signal recorder/analyzers for vertical pump machinery vibration certification is well established by the case studies presented herein.
CASE I: Deep Set VTP– Pump Station Weakness

SUMMARY
This case presents 1250 HP VFD Motors on 70 foot deep set vertical turbine pumps at a water treatment plant reservoir installed on a concrete intake island designed specifically for the purpose. The unique properties of the pump system include sub grade (10 ft) discharge heads with six pumps on the island coupled together in two discharge headers. To avoid thermal expansion issues the plant designers provided vertical supports for the discharge headers, but no lateral supports. The lateral pipe support loads were essentially ignored or were assumed to be within the capacity of the discharge nozzle to handle and no flexible pipe couplings were employed at the discharge nozzle. Since the discharge nozzle was below grade that meant that the lateral support for each header was provided through the three pump columns and from friction on the vertical supports. Friction is unreliable in transient flow conditions.

To complicate matters further the system designers solved the limited space problem on the island by using the pump column interface at the motor supports as the ingress and egress points for the below grade components including large check valves, header pipes, and the discharge nozzle for each location. Had the machinery been operated as fixed speed machinery the vibration problems would have been limited, or non-existent. However, with operation from 840 RPM to 1190 RPM the speed range crossed many critical speeds and these critical speeds were dependent upon the water level in the reservoir that had a variance of 55 feet. After nearly five years and several failed attempts to remedy the vibration issues the owners came back to the pump manufacturer for a resolution.

The vibration problem was resolved by designing, fabricating, and installing a massive sole plate and motor stand well beyond normal design criteria for such pumps so that the pump driver could be properly affixed to the pump station while spanning an opening greater is size than the driver. This opening was the primary vibration issue and it was completely outside the guidelines of the Hydraulic Institute. To resolve the lack of lateral restraints at the lower grade discharge header special one of a kind radial braces to each pump column were designed, fabricated and installed. These braces provided lateral stability to the lower 60 feet of the cantilevered pump column assembly. These remedies are well beyond the normal situation for vibration certification and the case represents a worst case scenario for inadequate pump station to pump machinery mechanical interfaces.

DISCUSSION
Figure A1 provides an excerpt of ANSI HI 9.6.4-2000 for Vertical Turbine Pumps Allowable Pump Field Vibration Values applied to the deep set pump illustrated alongside the standard. The first point to note is the small opening in the mounting flange and sole plate under the discharge head in Figure 9.6.4.13 of the standard. The standard specifically identifies flange loads and the need for special attention by system designers for potential resonance conditions (not shown in Figure A1). There is a 6’ by 4’ opening in the structure at EL 711 under the driver. Also note the vertical supports for the discharge header at EL 697. These two conditions represent weaknesses in the pump station design. Resolution of those weaknesses at the design stage does not occur because there is an implied engineering boundary condition where the civil engineers for the plant and the mechanical engineers for the pump supplier take separate responsibility. It is not until actual fabrication and operation with excess vibration that such conditions are considered. The facility civil engineers are concerned with structural and seismic loads. The machinery mechanical engineers tend to rely on the HI Standards and assume that a “rigid” mounting location exists. In this situation, the “rigidity” of the boundary condition was severely compromised.
Figure A2 illustrates the EL 711 installation of the 1250 HP drivers on motor stands with sole plates. The motor stands and sole plates meet the HI Standards for practically all installations. The sub grade discharge heads in Figure A3 have no supports for lateral bending loads from the sixty feet of pump column below the floor at EL 697. And the concrete pedestals supporting the members of the discharge header are totally inadequate for lateral loading. Concrete is excellent for compression loads. But, the very heavy pump column cantilevered from the discharge nozzle will introduce dynamic forces at these concrete pedestals that will fracture the pedestals. During prior attempts to reduce the vibration at the discharge header steel pipe supports were installed because the concrete pedestals were fractured within a short period of time.

Figure A4 and Figure A5 illustrate the mechanical interface problem. The view above the EL 711 grade in Figure A4 shows a relatively large opening required to drop the sub grade discharge head through the sole plate. The box beams were installed during futile attempts to stiffen the sole plate by others. The vibration troubleshooters did not appreciate the stiffness requirements for the interface, nor the relative stiffness of the three inch thick sole plate compared to the 3/8” wall of the box beams. When we look up to the sole plate from the lower level we can see the 4’x6’ opening that the machinery flange and sole plate had to span. Compare the sub base plate opening in the concrete to the opening in Figure 9.6.4.13 in Figure A1. Steel is very rigid in compression and tension and is substantially more flexible in bending. The overlap of the sole plate was only five inches at all sides. Structurally the machinery was mounted on a diaphragm.

Impact testing using an instrumented hammer resulted in the inertance (g/lb) functions in Figure A6 and A7 for lateral impact and response measurements at the top of the motor. These response functions show that the motor reed mode resonances were close to the top of the speed range at 1132.5 CPM and 1237.5 CPM. These resonances would limit the pumps due to vibration trips when operating near full speed.

Figure A8 illustrates an impact response measurement made at EL 697 at the flange of the center pump column in Figure A3. At least eight clear resonances of the pump column can be identified. The one at 1102.5 CPM is within the upper region of the operating range. This resonance would be excited by normal pump operation and it would shake the discharge header severely enough to crack the vertical concrete pedestals.

To resolve the vibration issue, REAI performed a Finite Element Analysis (FEA) of the pump installation with the assumption that the six anchor bolts at EL 711 and 100 inches of discharge pipe anchored at its end were the only boundary conditions. The quality of the six anchor bolts simulation is presented in Figure A9. Fully grounding the edge of the sole plate at its perimeter had significant effect on the 1st Reed X-Y inline resonance but less effect on the 1st Reed Z-Y crossline resonance, as also shown in Figure A9. The bump test data presented in the frequency response function is correlated to the FEA results in Figure A9 with variable accuracy. The bump test data indicates that a full edge grounding of the sole plate would be an over estimate of the boundary condition.

Figure A10 presents the first six resonances of the pump assembly in the plane perpendicular to the discharge pipe. The 4th Z-Y Column Mode and the 1st Z-Y Reed Mode are within the operating range and are identified by the “Encroachment” indication. Both of these modes involve substantial lateral movement at the top of the motor. The critical speeds are 75.4% and 89.8% of full speed (FS). If we assume that the resonance produces excess vibration within 5% of the critical speed, then the pump would have excess crossline lateral vibration at the top of the motor in the 70-80% and 84-95% speed region. The usable portion of the 840-1190 RPM (70-100%) operating region would be negligible.
Figure A11 presents the first five resonances of the pump assembly in the plane parallel to the discharge pipe. The 1st X-Y Reed Mode and the 4th X-Y Column Mode are within the operating range and are identified by the “Encroachment” indication. Only the 1st X-Y Reed Mode involves substantial lateral movement at the top of the motor. The 4th X-Y Column Mode is restrained by the 100 inches of simulated discharge pipe. The critical speeds are 78.2% and 100.1% of full speed (FS). The pump would have excess inline lateral vibration at the top of the motor in the 73-83% and 95-100% speed region. The combination of the two sets of modes means that a usable portion of the 840-1190 RPM (70-100%) operating region would only exist between 83% and 95%. These are very extreme conditions even for a VFD controlled machine.

The interference diagrams in Figure A12 and A13 identify the critical speeds for the High Water and Low Water conditions of the reservoir. Since there are multiple pump impellers with more than five vanes on the impellers the encroachments of the 2xRPM energy are not critical. However, the 1xRPM encroachments are critical because they can be driven by residual imbalance and dynamic imbalance caused by synchronous whirl eccentricity.

At least a dozen alternatives we evaluated using the FEA model. Figure A14 presents the final design modification that was chosen. A heavier, and thus, stiffer motor stand and sole plate were designed for the EL 711 interface. The four braces at EL 697 are not unique to pump installations, although they are unique for the location on the pump column. These braces were designed to eliminate the dynamic loads at the cantilevered pump column from being transmitted to the discharge header components.

Figure A15 provides an idea just how extreme the solution had to be to control the motor 1st Reed Modes. The original 2.5” think motor stand flange was increased to 5” thick. The sub base or sole plate thickness was increased from 2.88” thickness to 4.5” thickness. A thicker sole plate could not be employed because stuffing box geometry and the limited grout thickness were restrictions. The graphic impression that is most important from a system design engineering perspective is the limited grout area and subsequently small concrete load path for the machinery. A total of nine and a half inches of steel was required to raise the 1st Reed Modes sufficiently above the maximum speed to ensure lack of encroachment.

Figure A16 indicates that the 4th Z-Y Column mode was increased to 108.5% FS from the 75.4% FS condition of the original design. The mode shape for the 4th Z-Y Column mode indicates that it is not strongly coupled to motor later motion. This lack of coupling is due to the thickness of the two plates at EL 711 and to the braces at EL 696. Figure A16 indicates that the 1st Z-Y Reed mode was increased to 112.7% FS from the 89.8% FS condition of the original design.

Figure A17 indicates that the 4th X-Y Column mode was increased to 110.4% FS from the 100.1% FS condition of the original design. The mode shape for the 4th Z-Y Column mode indicates that it is not strongly coupled to motor later motion. This lack of coupling is also due primarily to the thickness of the two plates at EL 711 and to the braces at EL 696 as well as the discharge pipe at EL 700. Figure A17 indicates that the 1st X-Y Reed mode was increased to 117% FS from the 78.2% FS condition of the original design.

The design modifications were guided by the goal of eliminating encroachment within the speed range. Figure A18 and A19 show that it was not feasible to achieve the ANSI/HI 9.6.4 Para. 9.6.4.2.6 Structure Dynamic Analysis (optional) goals of 10% or 25% encroachment margins.
ANSI/HI 9.6.4 Para. 9.6.4.2.6 Structure Dynamic Analysis (optional)

Margins of safety between structure natural frequency and operating speed obtained by calculation vary within the industry. Typical margins are 25%, particularly in structures involving multiple components and/or multiple bolted joints; however some application-specific industry specifications allow margins as low as 10%. Margins of safety on the order of 25% may be impractical in certain applications, such as those having several closely-spaced natural frequencies operating at variable speed, with multiple excitation sources.

In some applications, low margins of safety between operating speed and actual (not calculated) natural frequency can result in successful operation, depending on the levels of excitation and damping present.

After installation of the modifications and correction of misalignment and imbalance conditions the vibration was reduced from greater than 1.0 in/sec to less than 0.03 in/sec at the top of the motor. It was also determined that the reed mode resonances were within 10 CPM of the FEA model. The 0.7% accuracy was achievable because the original condition was evaluated using bump test response functions. Numerical resonance analyses without field verification require the 25% margin to ensure a 10% encroachment margin.

CASE I CONCLUSION

This case is a horror story for pump suppliers and owners. It demonstrates what happens when the commodities mentality in the low bidder competitive pump industry is guided by pump station designers whose primary intent is to efficiently move fluids. Vibration caused by energy transfer between machinery and the fluid is largely ignored by the system designers, while the pump suppliers attempt to design by codes and standards which are not understood by the system designers. It is not so much that either party is to blame, because in 95% of the cases there is no vibration problem. But, as this case indicates, when we mount a long vertical beam on a diaphragm we should be ready to identify serious vibration issues.
Figure A1: Hydraulic Institute Standard for Vertical Turbine Pumps Allowable Pump Field Vibration Values and WTP Pump Station Elevations for Pump Centerline and Discharge Header

American National Standard for Centrifugal and Vertical Pumps for Vibration Measurements and Allowable Values
HYDRAULIC INSTITUTE www.pumps.org

ANSI/HI 9.6.4-2000

9.6.4.4 Allowable pump field vibration values

The vibration values shown in Figures 9.6.4.4 through 9.6.4.14 are for unfiltered RMS velocity readings. These values assume the following conditions:

- Operation under steady state conditions at the rated speed ± 10%
- Pump must be installed so that shaft alignment and flange loads are kept in accordance with the manufacturers' recommendations
- Vibration level recorded is the maximum of measurements taken in each of three planes: vertical, horizontal or axial; measured as shown in Figures 9.6.4.4 to 9.6.4.14

These vibration values are to be used as a general acceptance guide with the understanding that vibration levels in excess of these values may be acceptable by mutual agreement if they show no continued increase with time and there is no indication of damage, such as an increase in bearing clearance or noise level.

Figure 9.6.4.13 — Vertical turbine, mixed flow and propeller type
Figure A2: Vertical Pump Motors 4 – 5 – 4, West Side, EL 711

Figure A3: Vertical Pump Columns 4 – 5 – 6, West Side EL 697
Figure A4: Rectangular Hole for Installation of Pump 5 with Under Grade Discharge Head

Figure A5: Under Side of Pump 5 Mounting Pad
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Figure A6: Pump 5 Motor Top Inline Frequency Response Function, Pump 4 Running

Figure A7: Pump 5 Motor Top Crossline Frequency Response Function, Pump 4 Running
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Figure A8: Pump 4 Header Bottom Flange Crossline Frequency Response Function, Pump 4 Running

WTP BUMP TEST RESULTS
FRF 39: COL 4 XL, Column 4, 4C1 4CF3CN/4CF3CN

Figure A9: Resonance Frequencies, Bump Test Versus Finite Element Analysis, CPM

Correlation of Finite Element Models to Bump Tests at WTP

Figure A10: Perpendicular Mode Shapes for Finite Element Model of Original Design, High Water

Figure A11: Parallel Mode Shapes for Finite Element Model of Original Design, High Water
Figure A12: Interference Diagram, Finite Element Model of Original Design, High Water

Interference Diagram Original Design High Water, WTP Pump Units 4 & 5

Figure A13: Interference Diagram, Finite Element Model of Original Design, Low Water

Interference Diagram Original Design Low Water Level, WTP Pump Units 4 & 5
Figure A14: Motor Stand, Sub Base, and Stabilizer Modifications
Figure A15: Motor Stand and Sub Base Components
Figure A16: First Six Perpendicular Modes with Stabilizers, Finite Element Model of New Design, High Water

Figure A17: First Five Parallel Modes with Stabilizers, Finite Element Model of New Design, High Water
Figure A18: Interference Diagram, Finite Element Model of New Design with Stabilizers, High Water

Interference Diagram with Column Stabilizers High Water, WTP Pump Units 4 & 5

Figure A19: Interference Diagram, Finite Element Model of New Design with Stabilizers, Low Water

Interference Diagram with Column Stabilizers, Low Water, WTP Pump Units 4 & 5