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 work station 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 is 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 IV: Vertical Can Pump System Resonance

This case involves three 60 HP vertical turbine can pumps installed on a common header. The machinery exhibited a 1xRPM critical resonance within the operating range. During the certification process the peak vibration amplitude of the machine at the critical speed was missed due to the limited speeds chosen for certification. The test specifications and standards provide for guidance in ensuring that the vibration at any critical speed is determined. However, in light of the multi-channel data acquisition system capabilities it is feasible to acquire a complete operating range transition to ensure that frequencies of excess vibration are not missed. The added complexity of the multi-channel analyzer is probably not warranted for a 60 HP can pump, but the case represents a guideline for using the technology.

DISCUSSION

Figure D1 identifies the ANSI/HI 9.6.44 Figure 9.6.4.14 vibration criterion for short set vertical turbine pumps. Note that the figure indicates measurement locations at the top of the discharge head in directions parallel (Y) and perpendicular (X) to the discharge nozzle. The text of the standard identifies the locations more specifically than the figure, which has the primary purpose of identifying the orientation of the vibration probes.

9.6.4.3.2 Location of vibration probes
The vibration probes should be located on both bearing housings of horizontal split case or double case pumps, near the upper or outer bearing of end suction pumps and near the upper motor bearing of vertical pumps with close-coupled motors.

For the certification process REAI acquires vibration readings at all bearing locations, and in general, at the suction and discharge nozzles. Some operator specifications include measurement locations on piping, as well.

Figure D2 illustrates the installation of the vertical turbine pumps for this case study. The project involved replacement of three pumps. Two new pumps were installed at the time of the photograph. It is important to note that the branches of the discharge header system merge in the distance. The header assembly is supported by vertical pipe stands but no horizontal supports for the headers are present. This places a mass load on the discharge head consisting of the local discharge pipe and the fluid in the discharge pipe.

Figure D3 illustrates an impact response in the direction perpendicular to the discharge pipe for the middle pump of Figure D2. Phase was not acquired for the impact response. The acceleration spectrum indicates strong resonances at 964 CPM and 1549 CPM.

The data presented in Figure D4 on 14-Jun-07 shows elevated vibration at 968 RPM, which was essentially the critical speed. The vibration amplitude was 0.17 in/sec pk (0.12 in/sec rms) and was below the allowable vibration limit of 0.28 in/sec pk (0.2 in/sec rms) for the standard as shown in Figure D1.

Due to mechanical problems with the shaft setting the pump was removed and the third pump was installed in the middle position. Tests were repeated on 24-Jul-07 that indicated overall vibration amplitude of 0.60 in/sec pk (0.42 in/sec rms) at 994 RPM, Figure D5. In the second installation the resonance frequency was 2.6% higher and the resonance response was 214% greater.

During the second test the operator was asked to slowly change the speed until the maximum vibration was occurring. Then the vibration was recorded. It is very likely that the same procedure would have revealed excess vibration during the first test. However, due to time and operational constraints the operator was
given a specific speed that corresponded to the resonance peak at 946 CPM in Figure D3. When a single or dual channel machinery vibration analyzer is used for measuring vibration at resonance it is not uncommon that the peak vibration is not acquired unless special attention is paid to the measurement to sweep the speed through the resonance peak frequency, as was done on 24-July-07.

The ANSI/HI 9.6.4 standard makes the following recommendations regarding critical speed encroachment, or margins of safety, and methods of testing.

9.6.4.2.1 Lateral Critical Speed (Shafts)
Allowable margins of safety between operating speeds and calculated critical speeds vary within the pump industry depending on the service and the complexity of the analysis. Commonly used margins of safety as determined by analysis are about 15% to 25%, however, these margins may be impractical in certain applications, such as those having multiple excitation sources and operating at variable speed.

With respect to difficult applications, margins of safety between operating speeds and actual (not calculated) critical speeds can be less than the 15% to 25% range and allow successful installations to be obtained, depending on the levels of excitation and damping present.

The method of calculating critical speed, the extent of the analysis, and the margins of safety to be used should be agreed upon by the purchaser and the manufacturer.

9.6.4.2.6 Structure dynamic analysis (optional)
The method of calculating structure natural frequency, the extent of the analysis, and the margins of safety to be used should be agreed upon by the purchaser and the manufacturer.

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.

The resonant frequency may be obtained approximately by impact testing. In this technique, the structure is excited by striking with a soft hammer. This will cause the structure to vibrate at its resonant frequency. A single channel vibration analyzer may be used to measure the frequency of this vibration. A more elaborate and expensive impact test, when justified, may be performed using a multi-channel analyzer with an instrumented hammer or shaker to obtain all the modal characteristics (natural frequency, mode shape, and damping) of the structure and to ensure that the measured response is caused by the forced input, and not effected by background noise or electrical interference.

Structures are not generally symmetrical and may therefore have different natural frequencies in different axes. Any axis of significant mechanical dissymmetry should be checked. The axis parallel to the pump discharge piping and the axis perpendicular to the piping are axes commonly having significant mechanical dissymmetry for pump structures, which should be checked.
CASE IV CONCLUSION

The test procedure utilizing a dual channel machinery vibration analyzer captured the resonance condition as per the standard. However, as the data indicates, it is feasible to miss the peak amplitude of vibration when the non-operating bump test data is used to establish the operating speed for resonance evaluation. The operation of the machine can result in changing of the static resonance frequency due to gyroscopic stiffening and other dynamic effects. The standard speaks of the implementation of a multi-channel analyzer to acquire elaborate modal characteristics. This is rarely required when the fundamental reed frequency is the encroaching resonance. However, a multi-channel digital recorder/analysis system can acquire vibration for all changes of speed and allow for exact pin pointing of the resonance peak amplitude when the speed changes are slow through any critical speed. This process is not generally warranted for smaller machinery such as the one depicted in this case study. For this case the speed was modified by 1% steps while monitoring the vibration amplitude when operating conditions allowed for it on the second certification test. Once the speed for maximum vibration amplitude was established the full set of data was acquired. During the first certification test the water level and operating conditions limited the search for the peak resonance condition. If a recording analyzer was used a smooth transition from minimum speed to full speed would have captured data for the resonance condition. Post data analysis would have identified the maximum vibration for certification evaluation.
American National Standard for Centrifugal and Vertical Pumps for Vibration Measurements and Allowable Values
HYDRAULIC INSTITUTE  www.pumps.org

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, 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 9.6.4.14 — Vertical turbine, short set pumps, assembled for shipment by the manufacturer
Figure D2: Pump Station 1 Looking East

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Figure D3: Vertical Turbine Pump Crossline Impact Response Spectrum and Signal, 50% Load, 576 RPM
Figure D4: Vertical Turbine Pump RAS 2, Motor Top Vibration

Vertical Turbine Pump RAS 2
Motor Top Vibration Summary: 14-Jun-07 14:02 - 14:56

- 0.4 - 40 N/REV, IPSpk
- 0.9 - 1.1 N/REV, IPSpk
- 1.8 - 2.2 N/REV, IPSpk
- 0.4 - 40 N/REV, Milis p-p

Figure D5: Vertical Turbine Pump RAS 2, Motor Top Vibration

Vertical Turbine Pump RAS 2
Motor Top Vibration Summary: 24-Jul-07 09:43 - 10:20

- 0.4 - 40 N/REV, IPSpk
- 0.9 - 1.1 N/REV, IPSpk
- 1.8 - 2.2 N/REV, IPSpk
- 0.4 - 40 N/REV, Milis p-p