44th Turbomachinery 31st Pump SYMPOSIA | GEORGE R. BROWN CONVENTION CENTER HOUSTON, TX | SEPT. 14 - 17, 2015

Tips for Troubleshooting with the Operating Deflection Shape (ODS) Technique

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ABSTRACT

An Operating Deflection Shape (ODS) is a valuable tool used for visualizing vibration associated with reciprocating and rotating machinery, piping, and structures. The results from an ODS analysis can help identify areas of high vibration, and help determine the best course of action. This is particularly important for engineers who make decisions about reliability and safety for systems subject to vibration.

The purpose of this tutorial is to demonstrate how using ODS measurements can help identify possible cause(s) of equipment vibration such as: bolted joint looseness, clamp or support looseness, insufficient foundation stiffness, etc. This tutorial covers the differences between an ODS and a modal analysis, the required data acquisition equipment, procedure for data collection, and concludes with case studies. Tips for developing more useful ODS models are shared throughout.

INTRODUCTION

There are several commercially available software packages for evaluating machinery and structural vibration. These programs are designed to allow users to measure and diagnose machinery vibration. Unfortunately, knowledge of the software alone is insufficient. Some knowledge of vibration and data measurement techniques is also required to obtain meaningful results.

In many cases the differences between a system's mechanical mode shapes, calculated using modal analysis, and a system's forced response (ODS) are not clearly understood. In other cases, poor data acquisition techniques are used to collect data or poor signal processing methods are used on good data.

In this tutorial, a high-level review of modal analysis is given, followed by a brief description and benefits for using the ODS technique for troubleshooting vibration issues. The required data acquisition equipment will be outlined and suggestions will be provided on how to properly collect ODS measurements. Lastly, case studies where the ODS technique is used to identify root cause(s) for excessive vibration are shared.

MODAL ANALYSIS

In order to better distinguish the differences between the ODS technique and modal analysis, a high-level review of modal analysis theory is presented. Natural frequencies and mode shapes are *inherent* properties of the structure and will not change unless physical properties change (mass, stiffness, and damping, or boundary conditions). Therefore, the natural frequencies and mode shapes of a structure do not depend on the amount or location of its excitation.

Single Degree of Freedom (SDOF)

The equation of motion (EOM) for free vibration of a mass-spring-dampener single degree of freedom (SDOF) system is described by the following equation:

$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = 0 \qquad (1)$$

where m is the mass, c is the damping coefficient, and k is the spring stiffness. In Equation (1), $m\ddot{x}(t)$ is the inertial force, $c\dot{x}(t)$ is the viscous damping force, and $kx(t)$ is the restoring force. Figure 1 illustrates an unforced SDOF system.

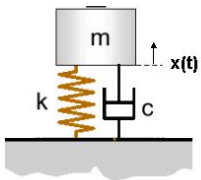

Figure 1. SDOF System (Courtesy of Guillaume)

For the frequency response analysis, the following form solution is assumed by Ewins (2000),

$$x(t) = Xe^{st}$$

where s is complex to accommodate both the amplitude and phase information. Now the EOM is

$$(ms^2 + cs + k)Xe^{st} = 0 \qquad (2)$$

with which we obtain the condition that must be satisfied for a solution to exist.

$$ms^2 + cs + k = 0$$

Solving for the roots leads to

$$s_{1,2} = \frac{-c \pm \sqrt{c^2 - 4mk}}{2m} = -\frac{c}{2m} \pm \frac{\sqrt{c^2 - 4mk}}{2m}$$

$$s_{1,2} = -\overline{\omega}_0\zeta \pm i\overline{\omega}_0\sqrt{1-\zeta^2} \qquad (3)$$

where

$$\overline{\omega}_0{}^2 = \frac{k}{m}; \quad \zeta = \frac{c}{c_0} = \frac{c}{2\sqrt{km}}$$

This implies a modal solution of the form:

$$x(t) = Xe^{-\overline{\omega}_0\zeta t}e^{i\overline{\omega}_0\sqrt{1-\zeta^2}t} \qquad (4)$$

From this exercise, three useful parameters are obtained: the system's natural frequency ($\overline{\omega}_0$), damping ratio (ζ), and the trivial eigenvector (mode shape). Defining these parameters is the primary goal of modal analysis.

Multiple Degree of Freedom (MDOF)

Since very few problems seen in the field can be described using a SDOF model, a multiple degree of freedom (MDOF) model is more frequently used. A general EOM for a MDOF system is described by the following equation

$$[M]\{\ddot{x}(t)\} + [C]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(t)\} \qquad (5)$$

where [ ] and { } denote square matrices and vectors, respectively. Figure 2 shows an example of a MDOF system.

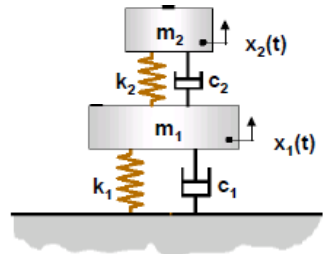

Figure 2. MDOF System (Courtesy of Guillaume)

Using a similar approach as in the SDOF case, we can write the free vibration EOM directly as

$$([M]s^2 + [C]s + [K])\{X\}e^{st} = \{0\} \qquad (6)$$

which constitutes a complex eigenproblem. From this point, the eigenvalue matrix $[\overline{\omega}_N{}^2]$ containing the system's natural frequencies with their corresponding eigenvectors [Ψ], i.e. mode shapes, can be obtained. Note that the free vibration eigenvectors (or mode shapes) only describe the relative motion between DOF of a structure when *no external forcing* is applied to the system.

Conducting a modal analysis on a machine or structure is useful in identifying its natural frequencies (or resonances); however, modal analysis of the unforced system cannot tell us how this machine or structure behaves while in operation.

OPERATING DEFLECTION SHAPE

The ODS technique can help answer two of the most commonly asked questions when a machine or structure seems to exhibit excessive vibration as McHargue and Richardson (1993) describe:

1) **How is the equipment or structure deflecting (or separating)** under a particular loading condition?

2) **How much** is the equipment or structure actually moving at certain points?

These questions can only be answered if a forced vibration response condition is considered. The vibration response of the machine or structure depends on the *amount* and *location* of its excitation source(s). Therefore a machine or structure's ODS depends on its excitation source(s). Only if the amount and location of all the excitation forces are known can the mode shapes, from the modal analysis, answer the questions above. Note that the ODS may resemble the mode shapes if the excitation is near the natural frequency of the structure.

However, what if all the excitation sources cannot be identified or are too complex? Then, direct measurement of the ODS is the only way to answer the proposed questions.

Frequency Domain ODS vs Time Domain ODS

Rotating machinery operating at a constant speed typically exhibits vibration at the frequency of rotation, as well as integer multiples of this frequency. Depending upon the type of machinery, other orders may also be present. For instance, a four-cycle engine will also exhibit vibration at half-orders of running speed and multiples of this frequency. Gearboxes will generate vibration at multiples of gear meshing frequency. At constant operating conditions, the vibration will be periodic, and the vibration at any point on the structure can be characterized by a vibration amplitude and phase (relative to a suitable phase reference) at each order of running speed. Frequency domain ODS measurements can be taken on such machinery by holding operating conditions constant, and sequentially measuring response values at various locations on the machinery. This has the advantage of not requiring a large number of response transducers.

Machinery may also exhibit non-periodic vibration. Machine start-ups or shut-downs, changing operating conditions, flow turbulence, etc., can all result in non-periodic or transient vibration. ODS measurements of transient vibration can be generated using time-domain measurements. This technique typically requires a larger number of sensors, so that response measurements at several points on the structure can be measured simultaneously. The ODS techniques discussed in this paper will be limited to frequency domain measurements.

DATA ACQUISITION

Software and Hardware

The first step in developing a useful ODS model is to obtain appropriate software that has ODS capabilities, like that shown in Figure 3. The market is saturated with options. Some software packages allow you to create a model within the program or to import a model saved in a particular format.

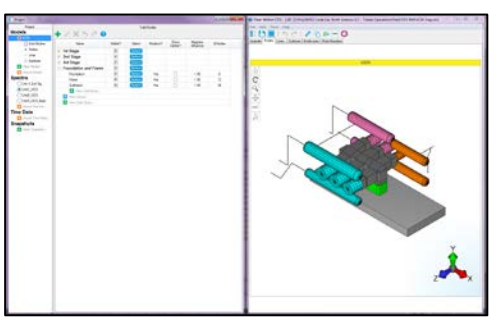

Figure 3. Example of ODS software graphical user interface (GUI).

Next, a multichannel data acquisitions (DAQ) system is necessary. The purpose of the DAQ system is to convert measured analog time-varying signals (voltages) to digital format suitable for further processing by computer software. Many DAQ packages come with spectrum analyzers (Fast Fourier Transform (FFT) analyzers) built-in to the hardware or software. These capabilities will help facilitate the data collection and signal processing procedures.

Transducers

A minimum of two channels are required to collect data for a frequency domain ODS model. One channel is used for a *phase* reference signal and the other channel(s) is used as the measurement transducer. A phase signal can come from a variety of sources: a laser or optical pickup, a pressure transducer, stationary accelerometer, etc. If rotating equipment is involved, the most common phase reference is the once-per-revolution *keyphasor*. The remaining channel(s) are used to measure the response at selected test point locations. Figure 4 shows phase reference transducers.

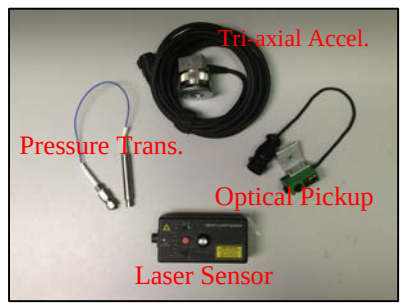

Figure 4. Phase reference transducers.

A common transducer for measuring vibration is a piezoelectric accelerometer. Displacement units are typically preferred in ODS. Recall that an accelerometer produces a voltage proportional to acceleration (g's); thus, analog or digital double integration is required to obtain displacement units.

Although ODS data can successfully be collected with a single axis accelerometer, it is easier to use a tri-axial accelerometer. ODS measurements on carbon steel structures can be facilitated with magnetically mounted accelerometers, though this may limit the maximum frequency that can be reliably measured to a few hundred Hz. To utilize a tri-axial accelerometer for ODS, a minimum of 4-channel data acquisition is required.

Signal Processing Tips

In order to convert the output of an analog transducer to a digital signal, it must be digitally sampled. Typically, the analog signal is sampled many times per second. At each sampling point, the analog value of the signal is converted to a discrete numerical (digital) value. The *sampling rate* (f_s) is the rate at which an analog signal is digitized. The relationship

between sampling rate and time between samples (Δt) is shown below (Beckwith el al., 2007):

$$f_s = \frac{1}{\Delta t} \tag{7}$$

It is important to ensure that the sampling rate is high enough, or *aliasing* can occur. Aliasing causes high frequency content in the analog signal to appear at a lower (aliased) frequency in the sampled signal, and occurs when an analog signal contains frequency components higher than the Nyquist frequency. The Nyquist frequency is equal to half the sample rate of the digitized signal. If the maximum frequency present in the analog signal is f_{max}, then the sample rate must be greater than twice this value or aliasing will be present in the digitized data:

$$f_s > 2f_{max}$$

Fortunately, most modern data acquisition systems are equipped with antialiasing filters in hardware. An antialiasing filter automatically filters out frequencies of the analog signal that are higher in frequency than the Nyquist frequency, which avoids aliasing in the digitized data.

When performing FFT calculations of measured data, it is also important to understand the relationship between frequency range, frequency resolution, and sampling period of the FFT. An FFT converts a time domain signal to frequency domain. The FFT consists of a number of bins. Each bin holds the amplitude and phase of the transformed time domain signal at a particular frequency. The bins are evenly spaced, with a spacing of Δf, which is the *frequency resolution*.

The f*requency range* is the maximum frequency of the FFT, which is equal to ½*f_s. *Frequency resolution* can be calculated with the equation below:

$$\Delta f = \frac{f_s}{N_p} = \frac{1}{N_p \Delta t} \tag{8}$$

where N_p is the number of points in the FFT. The *sampling period* or *time window* (T) required to obtain a number of points is defined below:

$$T = N_p \Delta t = \frac{1}{\Delta f} \tag{9}$$

It is important that sufficient frequency resolution is used to "broadly" separate the excitation frequencies if multiple sources exist. As an example, if N_p = 800 points, collected with f_s = 200 Hz, then Δf = 200/800 = 0.25 Hz. The sampling period T in this case would be 4 seconds.

A common difficulty in performing ODS measurements is choosing the appropriate frequency range and sample rate. The sample rate needs to be high enough to capture the highest frequency of interest. The sampling time needs to be long enough to obtain sufficient frequency resolution. For example, suppose we have a 300 revolutions per minute (RPM) motor driven reciprocating compressor. The frequency of 1× running speed is 300/60 = 5 Hz. We would expect the compressor vibration to be at 5 Hz, 10 Hz, 15 Hz, etc. We might expect most of the vibration to be from 1× to 10× running speed. Thus, the maximum frequency of interest would be 50 Hz, and 100 Hz might be a reasonable frequency range to allow some margin. It is also important to have a separation between each harmonic of *at least* 4 bins. Since the separation of each harmonic is 5 Hz, the frequency resolution should be at most ¼ of this value (1.25 Hz) in order to adequately resolve each order of running speed. To be conservative, a frequency range of about half this value (0.5 Hz) might be reasonable. Now that we know the frequency range (100 Hz) and frequency resolution (0.5 Hz), we can calculate the sample period $T = 1/\Delta f$ = 1/0.5 = 2 seconds.

Windowing is another important concept. The FFT is a very useful tool for analyzing data. However, the calculations used to generate the FFT assume that the data is exactly periodic, containing an exact integer number of cycles of measured vibration. In general, this is not the case. For instance, if we have a machine running at 1,150 RPM, and the FFT length is 1 second, then the FFT will contain 19.17 revolutions of the machine. To be truly periodic, the FFT would have to contain exactly 19 or 20 or some other integer number of machine rotations. This non-periodic data can introduce errors into the FFT calculation. Windowing is used to account for this problem.

There are many different types of windows available (Hanning, Hamming, Blackman, Flat-Top, Uniform, etc.). Each of these windows has different applications and characteristics. The basic tradeoff for each different window type is frequency resolution versus amplitude accuracy. A window with poor frequency resolution tends to smear the amplitudes across multiple bins, so that the peaks are not as sharp. A window with poor amplitude accuracy may significantly distort the amplitude of the data in each bin. When performing frequency domain ODS measurements, a good choice is a Flat-Top window. This window has poor frequency resolution, but excellent amplitude accuracy. The poor frequency resolution can be compensated for by using a greater number of bins. Hence, the recommended minimum of 4 bins separating each harmonic that was mentioned earlier.

TAKING MEASUREMENTS

After the measuring hardware has been setup and the DAQ software parameters have been chosen, *data collection* is the next step in creating a useful ODS model for troubleshooting machines or structures.

Data Measurement Tips

During data collection, operating conditions (operating speed, process pressures, flows, etc.) must remain essentially

constant while collecting data at all measurement locations. Varying speeds or process conditions during ODS measurements will cause discrepancies in the model animation.

When collecting vibration data on a machine or structure, it is crucial that the difference between a *local coordinate system* (LCS), e.g. the accelerometer using an x,y,z coordinate system, and a *global coordinate system* (GCS), e.g. the ODS geometry using an X,Y,Z coordinate system, be clearly understood. Consider Figure 5 which shows two test point locations, TP-10 and TP-20, where vibration data were collected. Note that depending on the geometry of the equipment cylindrical coordinates can be a better alternative to the Cartesian coordinate system. Although the tri-axial accelerometer's LCS was rotated 180 degrees about the y-axis between points, the orientations were recorded in terms of the GCS (X,Y,Z). It is important the ODS model test point locations also be in terms of the GCS to avoid animating an incorrect deflection shape.

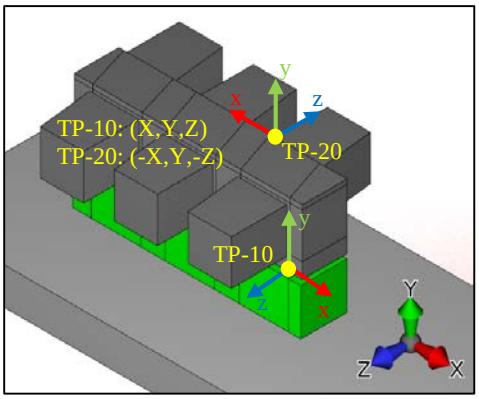

Figure 5. Global and local coordinates.

Reciprocating Machines - Suction/Discharge Bottles

Reciprocating machines typically have cylindrical bottles mounted on the suction and discharge side of the cylinders, as shown in Figure 6. These vessels are used to control pulsation induced shaking forces from traveling into the process piping.

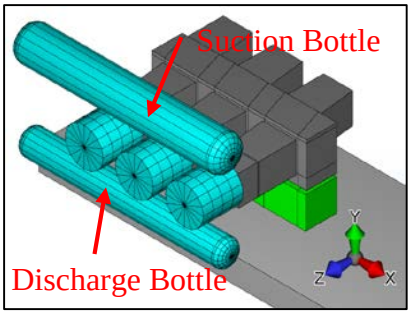

Figure 6. Suction & discharge filter bottles in ODS model

In order to capture the translational and rotational motion of the bottles, a minimum of two test point locations, at each end of the bottle, should be measured. At each end, the test point locations should be positioned at least 90 degrees from each other. Using a cylindrical LCS could be used for the bottles; especially if animating the shell wall modes of the bottle is desired.

Reciprocating Machines - Cylinders

The cylinder, as shown in Figure 7, is another component that makes up the compressor-manifold system. As with the pulsation bottles, a minimum of two test point locations, at each end, positioned at least 90 degrees from each other are needed.

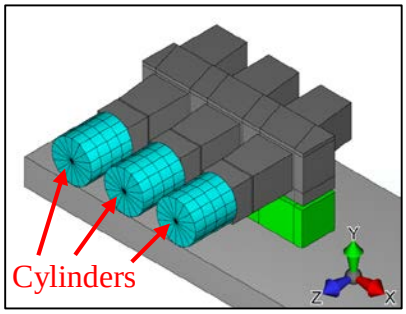

Figure 7. Cylinders in ODS model

Reciprocating Machines - Crosshead Guide (XHG) & Distance Piece (DP)

Modern reciprocating machines can have many configurations with regards to the crosshead guide (XHG) and distance piece (DP). Some machines have integral XHG and DP, others may have no DP, or maybe multiple DP's. It all depends on the OEM and the service. Figure 8 shows a typical setup for a XHG and DP.

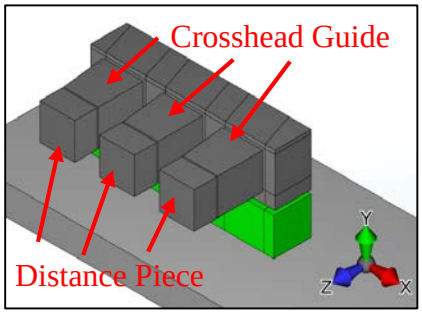

Figure 8. Crosshead Guide & Distance Piece in ODS Model

With this equipment, vibration data should be collected across the bolted joints, e.g. cylinder-DP bolted joint and XHG-DP bolted joint. At least two measurement points are needed on each side of the bolted joint. Separation between the components may indicate overload conditions and/or broken bolts.

Reciprocating Machines - Compressor Frame & Subframe

Compressor frames come in many configurations depending on the OEM and service. The frame may or may not have some kind of subframe. These frames are typically bolted down on either a concrete foundation or transportable skid, as shown in Figure 9.

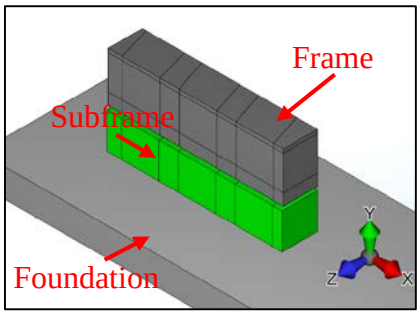

Figure 9. Frame/subframe/foundation in ODS model

Often, the most useful measurements for an ODS are those that are near the bolted connections between these components, e.g. frame-subframe bolted connections and subframe-foundation bolted connections. If bolted joints are scarce, then a few measurements should be taken between hold-down bolts. The ODS animation can indicate if bolt looseness exists between components.

Rotating Machines

Compared to reciprocating machines, identifying locations to acquire vibration data on rotating equipment is less standardized and is handled on a case-by-case basis. Examples of rotating equipment are: centrifugal compressors, gas turbines, electric motors, gearboxes, centrifugal pumps, etc. An example of a vertical pump assembly is shown in Figure 10.

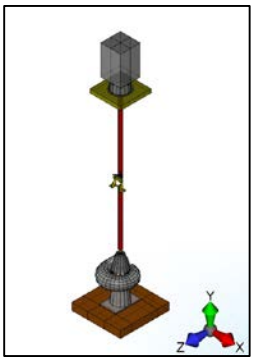

Figure 10. Rotating equipment example in ODS model

Structures - Foundations & Skids

Taking measurements on the foundation and/or skid the equipment is mounted on can also help identify potential root causes for excessive vibration. Measurement should be focused near the bolted connections between the foundation/skid and the equipment. These measurements may indicate if insufficient foundation or skid stiffness is to blame for the experienced vibration. Similarly, Atkins and Price explored using vibration measurements to identify deficiencies in foundation and bolt systems.

Piping & Supports

The location and number of vibration measurements needed on piping and/or supports directly depends on what type of vibration phenomenon is being observed. At times a 3-dimensional ODS model is required with several vibration readings about the circumference and length of the structure; especially, if obtaining shell wall modes is the objective. At other times, a 1-dimensional representation will suffice and vibration data at only one location about the circumference is required. Figure 11 shows an example an ODS model with 3D piping and an I-beam support.

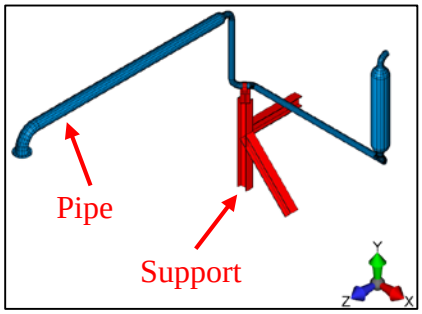

Figure 11. Piping & Support in ODS model

ANIMATING THE ODS MODEL

Once amplitude and phase data are recorded at all relevant test point locations, the ODS model can be animated. Rotating and reciprocating machinery produce excitation (and resulting vibration) at multiples of operating speed. Figure 12 represents a typical overlay of vibration spectra at all measurement locations on an engine driven reciprocating compressor. Note that vibration is predominantly at integer harmonics of operating speed. However, vibration at half-orders is also apparent. The ODS software should have the capability to select which harmonic is animated (i.e. 1×, 1.5×, 2×, etc.), as noted by the shaded band below.

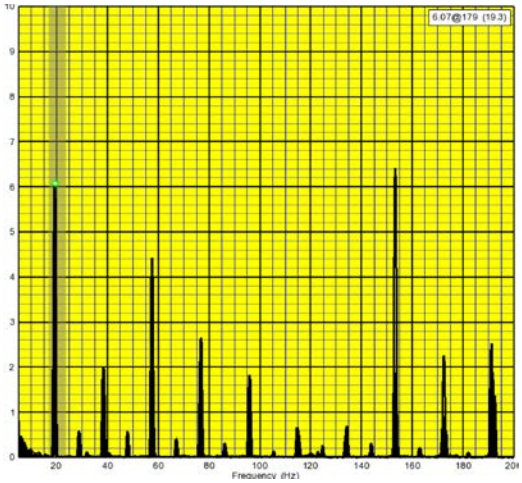

Figure 12. Typical Vibration Spectra

Various types of rotating and reciprocating equipment will exhibit different vibration behaviors which could be discovered using the ODS technique. Below are a few examples of what to look for in the animation based on different types of equipment.

Reciprocating Machines

Reciprocating machines can be prone to bottle vibration, cylinder vibration, cylinder–crosshead guide separation, insufficient foundation stiffness, etc., if not designed according to API guidelines. Excessive manifold (suction/discharge bottles and cylinder) vibration is often due to mechanical resonance, which is a coincidence of a MNF and an excitation frequency. Note that impact testing can confirm the MNF for the component(s) and an ODS alone cannot definitively determine if high vibration is due to mechanical resonance. However, an ODS at discrete frequencies can resemble typical response mode shapes of bottles and cylinders. Typical 'shapes' to look for include:

- Cylinder Low Mode (in-phase)
- Cylinder Resonance Mode (out-of-phase)
- Suction Bottle Cantilever Mode
- Discharge Bottle Cantilever Mode
- Suction Riser Mode

Excessive cylinder vibration can also be due to insufficient preload of cylinder and crosshead guide bolts or broken bolts. ODS animation can help identify this by showing separation at these bolted joints.

Excessive frame vibration can be due to insufficient hold down or skid deflection. The ODS can show separation between the frame and the skid or the skid and foundation.

Rotating Machines

Vibration in rotating machines often manifests itself at the bearing housings as this is typically where permanent vibration transducers are mounted. Excessive vibration can be caused by:

- Bearing housing resonances
- Structural resonances of the entire system (equipment on pedestals and skid)
- Misalignment
- Unbalance
- Instability

ODS measurements can help identify which of these is the primary culprit.

CASE STUDIES

Case Study 1 – Cylinder Low Mode

Figure 13 shows an engine driven, one-stage, two-throw reciprocating compressor used for natural gas service that operated at 1,150 RPM. Since its commissioning in the late 1990's this unit has had numerous failures and modifications.

As shown in the figure, a cylinder low mode for a two-throw reciprocating machine was identified. The blue wire frame shows the equipment undeflected. This deformation is characterized by the horizontal in-phase movement of the cylinders, bottles, and attached piping. Installing or improving cylinder and/or DP supports could help remedy this problem. See Figure A1 in Appendix A for more detailed animation.

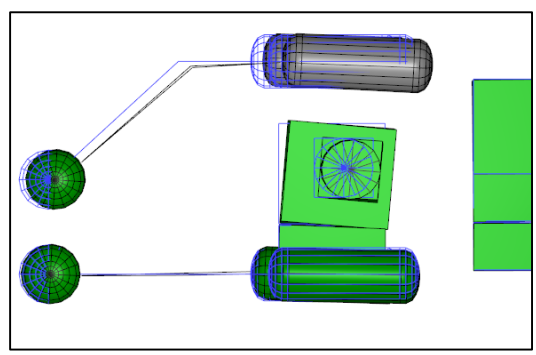

Figure 13. Cylinder Low Mode

Case Study 2 – Suction Bottle Cantilever Mode

Figure 14 shows one cylinder of a multi-stage, three-throw reciprocating compressor used for gas recovery that operates at 300 RPM. This unit had experienced excessive piping vibration since its repurposing in 1999. The ODS revealed that the piping vibration was causing a cantilever mode type behavior of the suction bottle. This deformation is described by the suction bottle "rolling" about the suction nozzle. This mode is a function of the bottle diameter, nozzle diameter, and any re-pad geometry. Since the unit could not be shut down, an impact test could not be performed to identify the MNFs of the suction bottle. Redesigning the bottle or adding brace-back supports to the compressor frame are typical options to fix this suction

bottle problem. In this particular case, improving the piping supports helped reduce the cantilever mode type behavior of the bottle. See Figure A2 in Appendix A for more detailed animation.

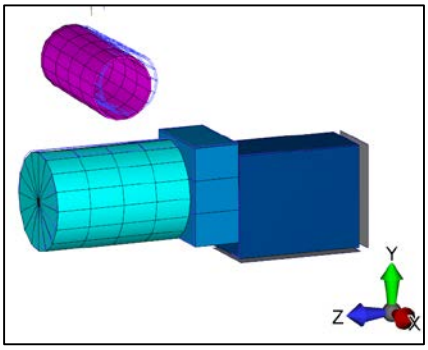

Figure 14. Suction Bottle Cantilever Mode

Case Study 3 – Vertical Pump Support Whirling

Figure 15 shows an enclosed impeller vertical pump for waste water treatment which operates at 885 RPM. High vibration was experienced on the pump, driveshaft, and driveshaft pillow block bearing supports. The ODS identified a "whirling" mode on the vertical pump's support. Minor vibration was present at the foundation; thus, adding large gussets between the volute and base plate/foundation could increase stiffness. See Figure A3 in Appendix A for more detailed animation.

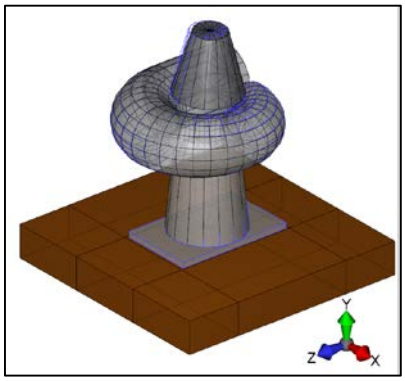

Figure 15. Vertical Pump

Case Study 4 – Exciter Bearing Housing

Figure 16 shows the exciter-end of a generator operating at 3,600 RPM. The generator's rotor had recently been rebuilt and upon reinstallation and startup began to experience excessive vibration on the exciter housing. The ODS showed vibration characterized by the horizontal in-phase deflection of the exciter, bearing housing, and duct. Impact testing showed the exciter housing to have a natural frequency close to the running speed of the generator; thus, adding a support to the exciter would increase the stiffness of the overhung exciter and reduce the level of deflection. See Figure A4 in Appendix A for more detailed animation.

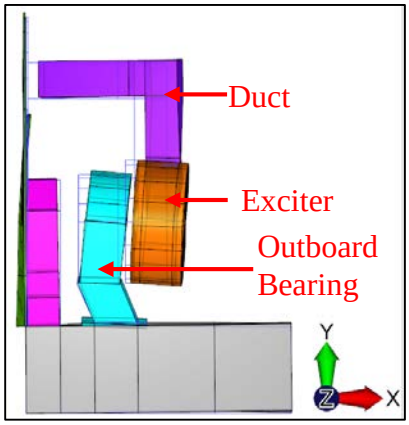

Figure 16. Exciter and Bearing Housing

Case Study 5 - Regenerator

In Figure 17, an out-of-phase rocking was identified about the support skirt of the vessel. The attached piping moves in-phase with the lower portion of the regenerator. Normally, these structural modes are at very low frequencies, e.g. less than 5 Hz. See Figure A5 in Appendix A for more detailed animation.

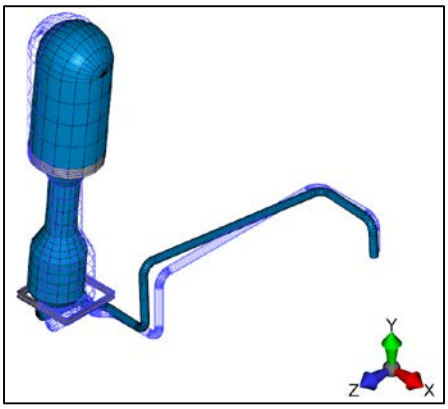

Figure 17. Regenerator

Case Study 6 – Bearing Housing & Catwalk Vibration

In Figure 18, a bearing housing attached to a vertical shaft for a vertical pump (not shown) rocks back-and-forth causing the attached catwalk to experience vibration issues. See Figure A3 in Appendix A for more detailed animation.

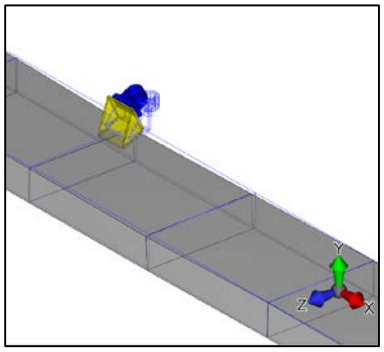

Figure 18. Bearing Housing and Catwalk

Case Study 7 – Forced Draft Fan

Figure 19 shows a forced draft (FD) fan that vibrated excessively in the radial direction at 1× operating speed after replacing the roller bearings. A second set of bearings were installed but vibration remained high. Several balance attempts were made but 1× vibration remained high. ODS showed that bearings, pedestal, and foundation were all participating in mode of vibration at 1× operating speed. No looseness was identified at bearing housings or pedestals in ODS. Additional testing identified a pedestal/foundation resonance near 1× operating speed.

Balancing, using conventional methods, while operating at a resonant condition is difficult because phase angles tend to vary. Knowing the unit was operating near resonance, a four-run balance method (which does not use phase) was utilized to improve balance condition and reduce vibration at 1× operating speed. See Figure A6 in Appendix A for more detailed animation.

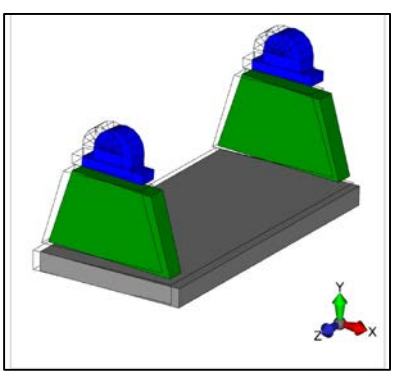

Figure 19. FD Fan Bearings and Pedestal

CONCLUSIONS

As shown in this paper, the ODS technique is a valuable tool engineers can use to troubleshoot and identify the sources(s) for excessive vibration on reciprocating machines, rotating equipment and structures. However, as with any tool, it is important to understand how to properly use it. It is imperative that the engineering professional(s) have a thorough understanding of vibration theory, data acquisition techniques, and signal processing techniques. Used properly, the results from an ODS analysis can narrow the scope of the best course of action, minimize down time, and lead to increased production.

NOMENCLATURE

API = American Petroleum Institute
DAQ = Data acquisition
DOF = Degree of freedom
DP = Distance Piece
EOM = Equation of motion
FD = Forced Draft
FFT = Fast Fourier transform
GCS = Global Coordinate System
GUI = Graphical user interface
LCS = Local Coordinate System
MDOF = Multiple degree of freedom
MNF = Mechanical natural frequency
ODS = Operating Deflection Shape
OEM = Original Equipment Manufacturer
RPM = Revolutions Per Minute
SDOF = Single degree of freedom
XHG = Crosshead Guide

Appendix A

Model / Unit 1 ODS Data / 19.25 Hz

Figure A1. Cylinder Low Mode Animation

Model / K504 ODS 300rpm / 14.750 Hz

Figure A2. Suction Bottle Cantilever Mode Animation

Model / All / 29.50 Hz

Figure A3. Vertical Pump, Bearing Housing and Catwalk Animation

Model / july 22 with skid (displacement units) (1x)

Figure A4. Exciter Bearing Housing Animation

Model / SS ODS 70000 Shims Removed / 1.8125 Hz

Figure A5. Regenerator Animation

FD Fan / 29 Hz

Figure A6. Forced Draft Fan Foundation and Bearings Animation

REFERENCES

Atkins, K. E. and Price S. M., 1992, "Evaluation of Reciprocating Compressor Foundations Using Vibration Measurement," PCRC 7th Annual Reciprocating Machinery Conference, Denver, Colorado.

Beckwith, T. G., Marangoni, R. D., and Lienhard V, J. H., 2007, *Mechanical Measurements*, Pearson Prentice Hall.

Ewins, D. J., 2000, *Modal Testing: Theory, Practice and Application*, Hertfordshire: Research Studies Press.

Guillaume, P., "Modal Analysis", Vrije Universiteit Brussel, Department of Mechanical Engineering [Online] Available: http://mech.vub.ac.be/avrg/

McHargue, P. L. and Richardson, M. H., 1993, "Operating Deflection Shapes from Time Versus Frequency Domain Measurements," 11th IMAC Conference.

ACKNOWLEDGEMENTS

We would like to acknowledge the entire Engineering Dynamics, Inc. Senior Staff for their feedback and support.