48TH TURBOMACHINERY & 35TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 9-12, 2019
GEORGE R. BROWN CONVENTION CENTER

# METHODS TO DETERMINE AND SPECIFY ROTODYNAMIC PUMP DYNAMIC ANALYSIS

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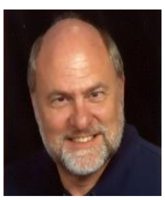

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## ABSTRACT

Vibration caused by resonance is an industry problem for new and retrofit applications that persists due to lack of specification and upfront analysis. To limit the chances of resonant vibration, dynamic analysis of the structure and rotating assembly to evaluate structural, rotor lateral and rotor torsional frequencies is done when the pump installation "warrants" it. However, dynamic analyses take time, require expertise, and cost money: It is not always clear when a pump is purchased if the installation warrants the expense of analysis. Furthermore, the purchaser may not know what type and levels of dynamic analysis should be specified. This results in poor specification, missed specification or specification of analysis when it is not needed. *ANSI/HI 9.6.8 Rotodynamic Pumps – Guidelines for Dynamics of Pumping Machinery* is the first American National Standard to cover this topic; it provides methods to evaluate the risk and uncertainty of a pump installation. A standard specification template is provided to aid the user. This tutorial will address the issue of resonance, review the importance of the guideline, how to apply the guideline, and risk factors, levels of analysis and methodology. Case studies are also provided.


## INTRODUCTION

Historically, a large number of installed vibration problems are due to resonance. Resonance is the amplification of a forced vibration due to interaction with a natural frequency. Dynamic perturbations may result in the excitation of resonance in any installation. With the trend towards increasing use of variable speed drives within pumping systems, avoiding resonance has become increasingly more difficult. It is therefore important to ensure that the potential problems caused by high vibration are properly addressed and mitigated during the pump system design phase.

Dynamic analysis is the evaluation of forces and their frequency, the pump system's natural frequencies, and the consequences of their interactions. The development of the analytical tools and techniques used in dynamic analyses to identify these issues has also dramatically increased. However, it is not always clear which tools are available and how to use them in various applications across various markets and diverse products. Equally, the range of preventive measures remains quite diverse, ranging from simple to complex. The associated expense can be small to relatively large when compared to the cost of the equipment. In all cases it is better to avoid a problem than to fix it after the fact. Understanding these challenges, the pump industry embarked on a nearly 10 year process to develop an American National Standard (ANS). That standard provides a guideline that defines the types and levels of analysis, methods to determine the appropriate analyses, and standard specification language; that a non-expert can use to receive appropriate upfront analysis.

The guideline published in 2014, *ANSI/HI 9.6.8 Rotodynamic Pumps – Guidelines for Dynamics of Pumping Machinery* is the first ANS to cover this topic, and to provide sample specifications for pump dynamic analyses. After using the guidelines to determine the type and level of analyses, one only needs to copy and paste the applicable sample specifications provided, and, provide the desired margins of separation in the spaces provided. Of particular note is unique guidance and sample specifications on vertical pump motors that are of particular importance in dynamic analysis. This guideline describes and recommends the means to appropriately evaluate pumping machinery construction attributes and relevant site characteristics. Those are then used to determine the effects of dynamic performance on equipment life and reliability. The standard describes and recommends various levels of detailed evaluation and validation that are commensurate with the degree of equipment uncertainty and application risk.

This tutorial will provide information on the relevant terms and concepts of dynamics, and, the challenge of designing pumping systems to avoid resonance. It provides an overview of the ANSI/HI guideline, sample specifications, and examples. The user is cautioned however, that this tutorial does not substitute for the complete text of the ANS. The content reproduced in this tutorial from the ANSI/HI 9.6.8 guideline is done with permission, courtesy of the Hydraulic Institute, pumps.org.

## SCOPE

The scope of ANSI/HI 9.6.8 are Rotodyanmic pumps as outlined in *ANSI/HI 14.1-14.2 Rotodynamic Pumps for Nomenclature and Definitions*. Figure 1 shows the general types of rotodynamic pumps within scope. However, the ANSI/HI 9.6.8 guideline will be most applicable to the OH, BB, and VS pumps and less applicable to the RT and CP types due to their respective designs, sizes, uses and costs.

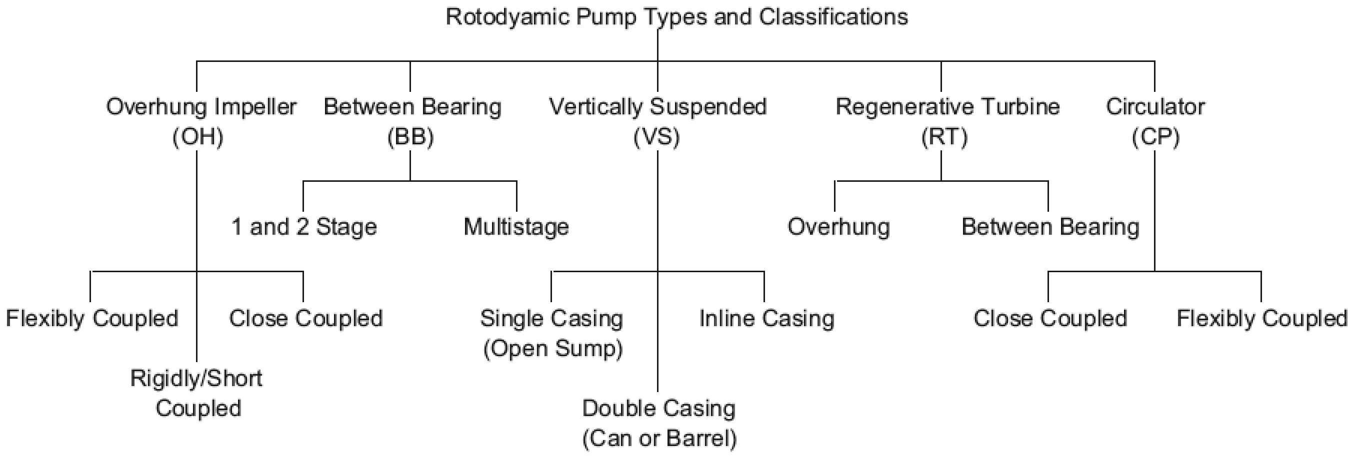


Figure 1 – Rotodynamic Pump Clasifications

Evaluation scenarios the guideline is applicable include the following:

- New equipment prior to field installation
- Existing equipment condition assessment in the field
- Existing equipment undergoing field modifications
- Exiting equipment undergoing field rerate
- Dynamic problem field troubleshooting

Evaluations may include drive systems, ancillary equipment, and the effects of local foundation and piping systems, as appropriate.

Dynamic response is the result of excitation forces sourced in the pump or its system, amplified or attenuated by natural frequency resonances (poles) or antiresonances (zeros), as well as by how well the pump is supported by its foundation. The influence and control of the excitation forces, natural frequencies, and foundation support are considered.

Determination of pumping equipment natural frequencies is important. If a natural frequency is numerically close to an operating speed or other excitation frequency, a state of resonance could exist. A natural frequency response to an excitation force can be greatly amplified, with the resulting stresses and deflections possibly causing premature equipment failure.

Typical excitation forces can be described in terms of the resulting amplitude at a given discrete frequency (e.g., vane passing frequency) or a frequency span, and can be associated with either mechanical or hydraulic causes. Both mechanical and hydraulic excitations are within the scope, in terms of prediction as well as effects, with certain limitations.

To provide practical guidance useful across a broad range of types, sizes, and applications of pumping machinery, three levels of analysis, categorized by methodology, are discussed:

1) Simple calculations that may be performed using a hand calculator using first-order equations.
2) Methods employing basic mass elastic modeling using commercially available software tools, such as finite element analysis (FEA) programs.
3) Computational methods involving multiple specialty programs and complex methodologies

For each level of analysis:

- Terms are defined
- Guidelines are provided concerning when a certain analysis level should be applied
- Guidelines are provided concerning when a certain analysis level should not be applied or trusted
- Limitations are presented concerning the various analysis methodologies
- Recommendations are made about what should be done (although not necessarily all the details regarding how to do it) and what each analysis comprises
- Suggestions are provided concerning what equipment or components should be analyzed
- Lists are provided of typical design and technical information needed to perform the analysis
- An appendix is provided with a sample specification
- A checklist of information is suggested for inclusion in the analysis for reviewing purposes
- Recommendations are made for verification methods that may be utilized, where applicable

- Recommendations are made for minimum standards to be applied concerning numerical analysis, such as FEA, as a function of analysis level
- Suggestions are made concerning how the analysis would fit into the timetable of a project and how it would interface with the other tasks to be performed within the entire scope of the work to be accomplished

The following analyses are beyond the scope of this document:
- Analyses involving complex time-dependent excitation due to fluid phenomenon or external sources of vibration.
- Seismic analysis considerations involving the motion of the mounting base of a structure, e.g., as in an earthquake.
- Analyses involving certain considerations of hydraulic excitations, such as:
  - Acoustic/pressure pulsation analysis (e.g., acoustic resonance of the piping system or network)
  - Analyses involving computational fluid dynamics (CFD) methods, such as
    - Conducting rotor or structure fluid interaction calculations as a function of flow
    - Quantifying the effect of vortex shedding on a vertical pump casing
- Determining the effect on rotor excitation of pressure pulsations generated by an impeller vane passing a volute tongue.
- Rotor stability analysis is outside the scope of the analyses conducted as standard procedure as presented in this document due to the limited exposure of most pumps to rotor-stability-related issues. With respect to this document, a rotor stability analysis may be considered as an optional analysis. A rotor-stability analysis may be advisable in applications involving boiler feed pumps, charge pumps, high-energy density pumps, and very high-speed pumps. For information regarding stability analysis, refer to American Petroleum Institute (API) 610 and RP684.

Vibration level shop and field acceptance criteria based on test results are excluded from this guideline. Refer to 9.6.4 *Rotodynamic Pumps for Vibration Measurements and Allowable Values* or other applicable standard.

The vast majority of rotodynamic pump applications involve standard products produced from established designs. All necessary engineering has been completed, substantial operational experience exists for the pump in essentially identical geometry configuration and similar application, and the pump is essentially ready for use. These products may include pumps that are mass-produced and many that are manufactured individually to order. For these products, the manufacturer has completed appropriate validations during the process of design and development prior to introduction. These products do not normally require further dynamic analysis. The user need only install the pumping equipment in accordance with HI Standards and the manufacturer’s instructions to attain satisfactory life and service.

There also exist a number of pump applications where an additional engineering effort by the manufacturer may be required prior to the use of the product. This engineering effort may be related to any of the following:

- Substantially new pump design
- Significant pump modification
- Adaptation of a standard design to a non-typical application that involves a different physical arrangement, mounting, drive, or operating condition, such as speed or rate of flow as a percentage of best efficiency point (BEP)

For these applications, the manufacturer has primary responsibility for pump and contract equipment dynamic considerations. The manufacturer will perform any necessary analysis as part of the design effort, within the context that the design is based on a typical application.

For non-typical applications, or with regard to system issues (eg piping or support structure), the manufacturer can offer guidance. Adequate information must be provided by the user and the user’s representatives.

For some applications, the dynamic characteristics of the pumping equipment are significantly influenced by the installation, including piping systems, mounting, or drive system. In these cases, some type of dynamic analysis may be needed to assess the equipment as it will be installed. This analysis is best completed during the pump design phase. Structure, piping, and drive system of the installation may influence the dynamic characteristics.  Those could lead to reliability problems once the equipment is installed; perhaps requiring extensive troubleshooting and expensive in-place modifications to attain an acceptable situation. The contractual responsibility and cost for the pre-installation evaluation of the pump (with associated equipment) and system interaction is normally carried by the user and the user’s representatives.

There are three main types of dynamic analyses for pumps and pump trains:
1) rotor lateral,
2) rotor torsional, and
3) structural.

Later sections of this tutorial discuss the need for any particular analysis, and the level of its detail as it relates to the pump type, application, potential costs associated with equipment startup problems, and other factors.

**KEY TERMS AND CONEPTS**

As described in the introduction, when application dictates, the appropriate dynamic analysis should be conducted to mitigate the risk of resonance prior to installation or to remediate a resonance after installation. The goal is that installed vibration complies with the allowable levels in ANSI/HI 9.6.4 or other applicable standard.

Some key terms and concepts are defined in this section with some select discussion:

- **Dynamic analysis** - the evaluation of forces and their frequency compared to the pump system's natural frequencies and the consequences of their interactions.
- **Resonance** - the amplification of a forced vibration due to interaction with a natural frequency.
- **Natural Frequency** - frequency at which a rotor and its support system can theoretically execute free vibration indefinitely without the need for any external energy input, after being excited. If excited at this frequency, it will vibrate indefinitely when damping is not present.

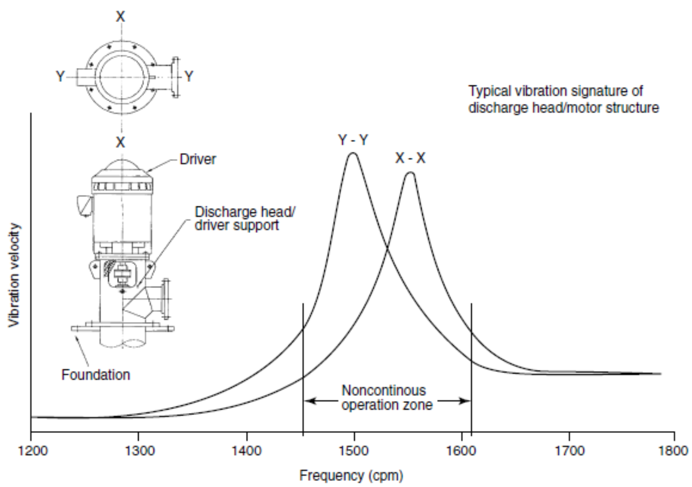


Figure 2 – Example of vertically suspended pump (above ground) discharge head natural frequency

- **Critical Speed** - shaft rotational speed at which the rotor/bearing/support system is in a state of resonance with a natural frequency with an amplification factor greater than 3.33
- **Excitation frequency (forcing frequency)** – Forces sourced in the pump system that occur at a frequency (i.e. imbalance 1XRPM)

 

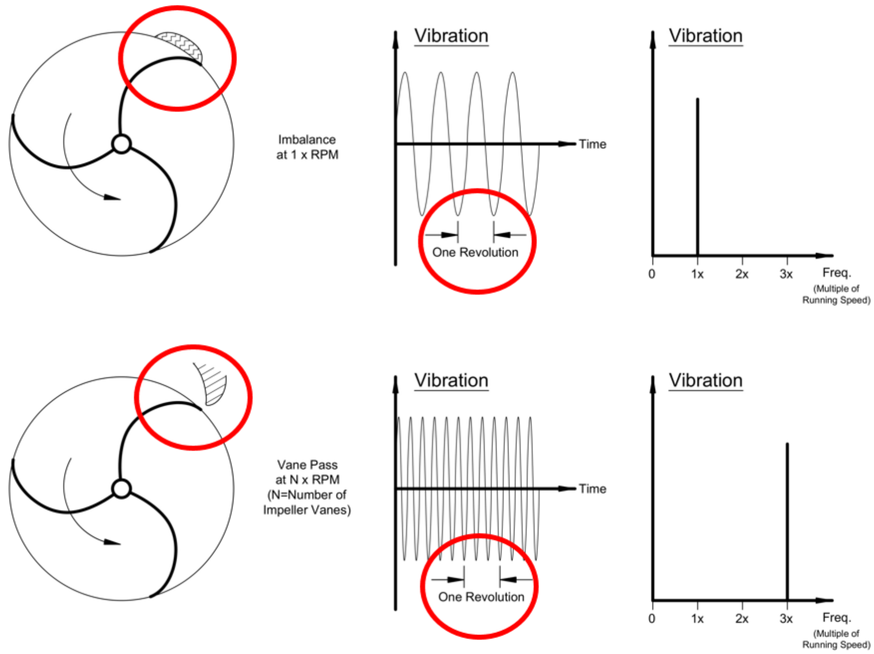


Figure 3 – Forcing frequency examples (imbalance and vane passing)

- **Amplification factor** - Ratio of the response level to a given excitation force applied at the resonant frequency, to the response that would be obtained if the excitation force was applied statically.
- **Damping** - Any effect that tends to reduce the amplitude of oscillations in an oscillatory system, typically through either viscous or Coulomb frictional energy dissipation. Damping is a key consideration for an analyst if operation near a natural frequency is unavoidable. Also, note on fig. 4 that amplification is directly correlated to the proximity of the frequencies.
- **Separation Margin** - Margin, defined as a percentage of the operating speed, between a critical speed that is outside of the operating speed range and the operating speed or other specified excitation frequency closest to it.

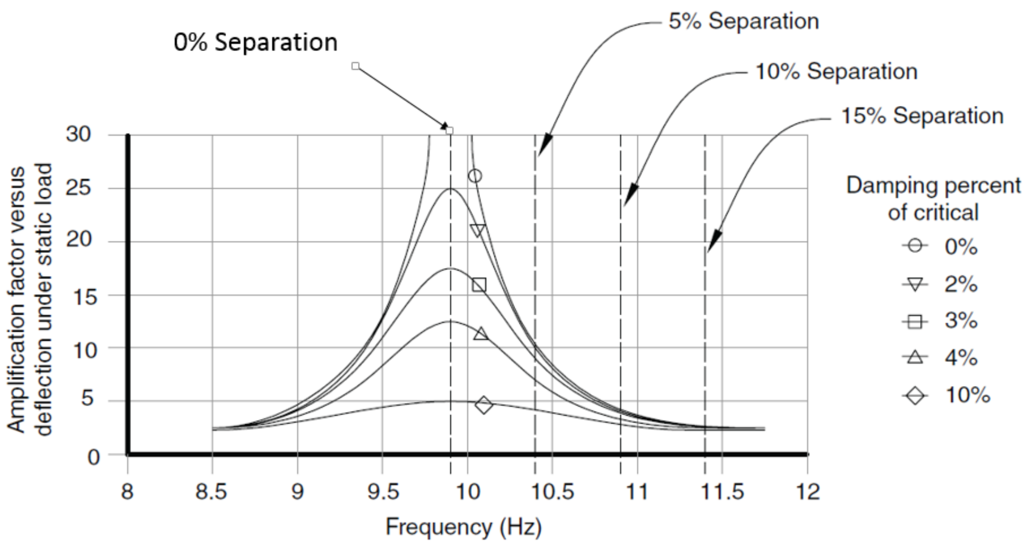


Figure 4 – Amplification chart showing effect of damping and separation margin

The main issue that occurs, which this tutorial is addressing is to avoid amplification of the forced vibration when it aligns with a natural frequency. When a forcing frequency is amplified by a natural frequency the rotor or structure is in resonance. Shown in Figure 4 is the amplification factor with respect to damping and the separation margin of the forcing frequency to the natural frequency. Damping is defined above and is a very important concept in controlling vibration when resonance cannot be avoided. At 10% separation almost no amplification occurs. As a goal it is advisable to strive for a 10% in field separation margin.

- **Campbell Diagram** - A plot of natural frequencies and excitation frequencies as a function of running speed. The intersections of the excitation and natural frequency lines provide approximate locations of resonance. In Figure 5, red circles indicate when the excitation frequency intersects the natural frequency, and yellow circles indicate where the specified separation margin intersects the natural frequency. As noted in the discussion of separation margin above, it is desirable to achieve a separation margin in the field of 10% to limit amplification. A tool to aid in this is the Campbell diagram. This is a plot of forcing frequencies and natural frequencies vs. speed. Here an analyst has identified three natural frequencies of interest (horizontal lines) and plotted 1X, 2X and 3X forcing frequencies as a function of speed. Note each of these has a design separation margin of 15%. The extra 5% separation margin in the analysis is to account for uncertainty in the calculations. This is a variable speed application and note that it becomes difficult to eliminate all resonant conditions due to the expanded forcing frequency range. Specifying separation margins that are too large will result in a pump system that cannot meet the specification.

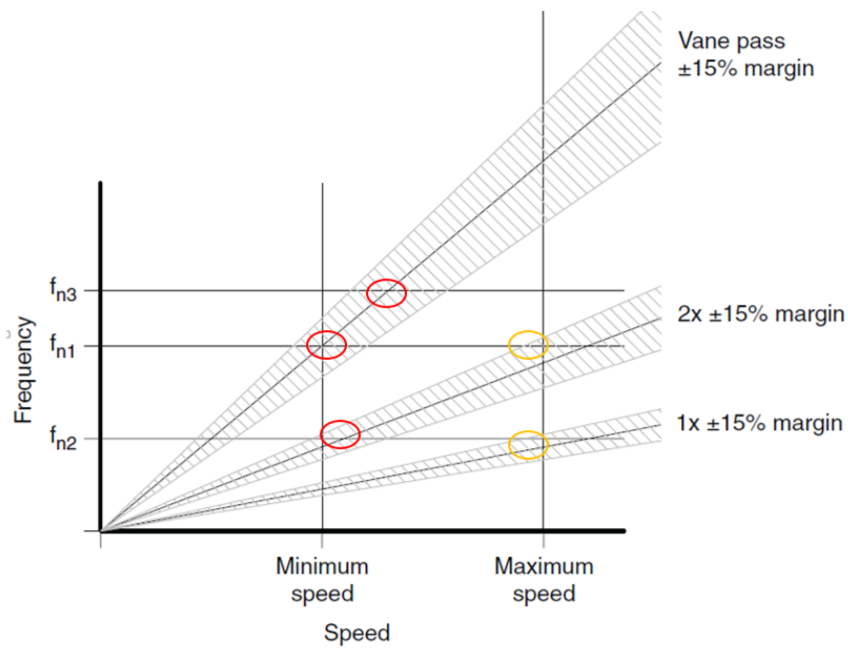


Figure 5 – Campbell diagram inclusive of specified separation margin

- **Lateral Vibration -** vibration any plane that is perpendicular to the axis of rotation
- **Torsional Vibration -** vibration in a manner that twists about the axis of rotation
- **Forced response analysis** – an analysis that predicts the behavior of the pump system due the forced vibration.
- **Lateral rotordynamic analysis -** an analysis of the shaft lateral (bending) dynamics, whether performed manually or computer-aided with software. As a minimum, it consists of determining the shaft lateral bending natural frequencies near considered excitation force frequencies (such as 1 × rpm) and the associated mode shapes. A more detailed analysis may include a forced response bending stress analysis and/or a rotordynamic stability analysis.
- **Lateral structural analysis -** an analysis of the lateral (bending) dynamics on the non-rotating components of the pump, whether performed manually or computer-aided with software. As a minimum, it consists of determining the stationary structure lateral bending and/or shear or twisting natural frequencies, near considered excitation force frequencies (such as 1 × rpm) and the associated mode shapes. A more detailed analysis may include a forced response bending stress analysis of the structure, as well as its nearby attached piping and the portion of the foundation affecting the motion of any particular natural frequency being predicted. Insight into the latter may be based on experience, field test data, or iterative analysis (e.g., trial solutions where the foundation stiffness is reasonably varied, and any effects on the natural frequency in question are noted).

 

- **Torsional rotordynamic analysis -** an analysis of the shaft torsional (twisting) dynamics, whether performed manually or computer-aided with software. As a minimum, it consists of determining the shaft torsional natural frequencies near considered excitation force frequencies and the associated mode shapes. A more detailed analysis may include a steady state forced response torsion torsional stress analysis and/or a transient start-up torsional stress analysis.
- **Transient start-up evaluation -** The calculation of temporary vibration response during system start-up, shutdown, or other transient events, such as a shock load to the system.

## TYPES AND LEVELS OF DYNAMIC ANALYSIS

### Types of Analysis

The three main types of dynamic analyses for pumps and pump trains (rotor lateral, rotor torsional, and structural) are performed to reduce the risk of vibration and reliability problems. Each has optional analyses associated with it, such as forced response or transient start-up evaluation, or in the case of rotor lateral analysis, a rotordynamic stability evaluation. The need for any particular analysis and the level of its detail depends on the pump type, application, potential costs associated with equipment start-up problems, and other factors. In general, a forced response analysis is only considered if a critical speed analysis predicts that a natural frequency resonance is expected, subject to only low to moderate damping. Similarly, a rotordynamic stability analysis (any requirement for which is beyond the scope of this document) is contemplated only for specialty pumps, typically in API or aerospace service, or very high power density pumps.

Some important definitions in the preceding section include natural frequency, resonance, excitation frequency, amplification factor, critical speed, and damping. Natural frequencies (an actual pump structure or rotor has multiple natural frequencies in the range of typical excitations) are those frequencies at which an object, once excited to move, will continue to vibrate once the excitation is removed. When a guitar string is plucked, it vibrates at its resonant frequencies, the cycles per second of which can be modified by adjusting the tension in the string (tuning). Unless energy is removed, for example by damping such as occurs in an automotive shock absorber, no new energy is required to continue vibration at that frequency. The level at which the object will vibrate will depend on the strength of the initial excitation. If the excitation is applied periodically, so that the force peaks at a certain excitation frequency, vibration will occur with or without significant effect from these natural frequencies. If the excitation frequency is close to a natural frequency, then the frequency is said to be a resonant frequency, and the vibrating structure is said to be in resonance. The level of vibration that occurs in this resonance depends on how close the exciting frequency is to the natural frequency, the excitation force level, how much leverage this force (operating at a specific location and direction) can exercise on the structure given the natural frequency's vibrating pattern or mode shape, and how strongly the damping is of vibration mode's natural frequency. Damping is determined not only by the energy-absorbing properties of certain rotor system components (bearings and seals, static structural joints, and hysteretic internal friction in materials), but also by the mode shape. Mode shape in the case of excitation forces, can enhance or detract from the leverage that these components can exercise as the pump rotor or structure vibrates. An example of low damping is a basketball being bounced on hardwood. The damping increases considerably if the ball is bounced on a rug.

Common excitation force frequencies at or above running speed include
- pump running speed ("synchronous") frequency,
- two times (2×) running speed frequency ( as caused by misalignment at the coupling or bearing or by the strong second harmonic inherent in a universal joint), and,
- vane pass (number of impeller vanes or blades times running speed),

as discussed in Texas A&M Pump Symposium as well as Vibration Institute tutorials). Refer to bibliography.

Hydraulically induced flow problems caused by fluid whirl in bearings or annular seals (such as thrust balance devices) or off-design pump operation also can excite vibration responses in the pump or pump system. Those are sub-synchronous, or below running speed, and occasionally super-synchronous, or above running speed. Cavitation can cause excitation forces and resulting vibration where most of the energy is concentrated in the very high-frequency (often ultrasonic) range.

#### Structural analysis (stationary):

To provide reasonable assurance that structural natural frequencies will not be close enough to typical excitations to become significantly resonant, structural analysis should be performed on the relevant nonrotating portions of the pump and attached components that add significant mass and/or stiffness.

A structural natural frequency analysis is often applied to evaluate vertically mounted pump structures. Of particular interest for such structures is the reed frequency. The deliverable is a tabulation of the natural frequencies and mode shapes within the range of excitation frequencies (typically up to impeller vane pass frequency or 2× electric line frequency for a motor-driven pump, whichever is higher).

#### Rotor lateral analysis:

In OH or BB pumps, a lateral rotordynamic analysis is performed to assess the potential for high vibration, associated degradation, and eventually equipment failure after installation. While a typical field vibration analysis can identify an overall problem, waterfall plots, operating deflection shape (ODS), or experimental modal analysis type vibration analysis methods are usually required to pinpoint the problem source and resolve it.

In the case of a VS pump, the type of computational lateral analysis used for an OH or BB pump is often not useful for VS pumps because the bearings and the interaction with the relatively low-stiffness support structure are quite flexible and nonlinear in the reaction to load, and the stiffness and damping can be difficult to accurately predict due to a lack of a well-defined hydrodynamic fluid film being formed. (An example of guidance regarding damping may be found in API RP684.) However, to determine if the bearing span is sized properly, original equipment manufacturers (OEMs) typically perform a basic natural frequency manual calculation assuming the line shaft is a simply supported beam with pin supports at the line-shaft bearings. Fortunately, the line shafting of VS pumps is designed with long L-over-D and high-clearance, line-shaft bearings, such that the line shafting tends to act as a violin string as the musician's fingers move to touch different locations. In this manner, each rotor natural frequency is a constantly "moving target," making significant resonance very unlikely. Therefore, a more practical use of vibration prediction resources is typically the vertical pump's structural natural frequency, particularly the aboveground "reed frequency" mentioned in the section above.

Lateral critical speed analysis for OH or BB pumps is performed for the combined pump rotating components (shaft, impellers, sleeves, coupling half, etc.) up to the first flexible shaft coupling (including the half-coupling), or for the entire pump-driver rotor system if a rigid coupling is used. Both undamped and damped analysis can be performed, but because of the strong effects of seal damping in most centrifugal pumps, the latter is the most relevant for most pumping applications, particularly for pumps of more than two stages. Undamped lateral analyses for BB pumps should be performed only for "legacy analysis comparison" purposes.

**Torsional analysis:**

A torsional analysis is only relevant when performed on the complete train (pump, driver, couplings, gears, etc.) and is applicable to both VS and OH or BB pumps. Torsional vibration is fluctuating angular motion that causes twisting in couplings and shafting systems of rotating machinery. The torsional analysis:

1) simulates how the pump, motor, coupling, and gear (if included) operate dynamically in twist excitation when run together as a system, and
2) identifies torsional natural frequencies (as opposed to natural frequencies that vibrate in the axial or lateral directions).

The key questions to answer via analysis are "Will there be any significant level of vibration, as compared to the specified vibration acceptance criteria, caused by a resonant frequency issue?" and "Will any torsional vibration, resonant or otherwise, lead to alternating stress, at peak torsional deformation locations on the shaft, which is sufficient to cause high cycle or even low cycle fatigue?" Identified problems or resonances are resolved by changing the mass or inertia of components in the system (impellers, couplings, flywheels) or by changing the torsional stiffness of low-stiffness components, for example, by changing shaft diameters or changing the coupling type in the specification.

**Analysis Levels**

There are three levels of complexity for such analyses, as defined in Table 1. In addition to defining the applicable types of analysis, ANSI/HI 9.6.8 introduces the three levels of analysis for each type of analysis. Table 1 summarizes what is include in each level of analysis, and provides a picture of the expected inaccuracies. As the level of analysis is increased, the expected inaccuracies decrease allowing lower separation margins to be specified in the design phase. The ANSI/HI 9.6.8 guideline recommends specifying a 25% separation margin for level 1 analyses, and 15% separation margin for level 2 and level 3 analysis, but requires the user to specify their desired separation margin.

Table 1 – Analysis complexity levels

| Analysis Complexity | Includes | Expected Inaccuracies |
|---|---|---|
| Level 1<br>Simple Calculations | 1. Simple calculations using standard equations | Highest |
| Level 2<br>Intermediate Methods | 1. Lateral rotordynamic analyses<br>1.1 Undamped natural frequency analysis - OH or BB pumps with one or two stages<br>1.2 First dry critical speed calculation - VS pumps<br>1.3 Damped natural frequency analysis - OH or BB pumps with more than two stages<br>2. Torsional rotordynamic analyses<br>2.1 Undamped torsional natural frequency analysis<br>3. Structural analyses<br>3.1 Shaft deflection analysis (static structural FEA) - OH and BB pumps<br>3.2 Pump baseplate or support frame, natural frequency analysis (modal FEA) - OH and BB pumps<br>3.3 Pump structural natural frequency analysis (modal FEA) VS (and vertically mounted OH and BB pump types) | |
| Level 3<br>Advanced Methods | 1. Lateral rotordynamic analysis<br>1.1 Forced response analysis OH and BB pumps<br>1.2 Vertically suspended pump rotordynamics addressed as a combined rotor and structural analysis<br>2. Torsional rotordynamic analyses<br>2.1 Transient forced response analysis of driver-coupling-pump shaft-trains (driver: electrical motors, typically induction [⇔ asynchronous] motors). Also trains operated through gears and fluid-couplings.<br>3. Structural analyses<br>3.1 Forced response analysis of driver and baseplate or support frame - OH and BB pumps<br>3.2 Forced response analysis of VS pump including flexible elements<br>3.3 Forced response analysis of pump, driver, and support structure – OH , BB, and VS pumps | Lowest |

**Level 1 - Simple calculations that may be performed using a hand calculator using standard equations**

Level 1 calculations involve significantly simplifying assumptions. Depending on the type of analysis being performed, these assumptions can introduce significant errors. Although useful for initial sizing and refinement of the design, such methods are usually too coarse for the results to be used to qualify a final design against a particular standard or program.

For example, natural frequency calculations for rotors often assume infinite bearing stiffness and dry-running rotors. Another example would be the calculation of structural resonance in vertical pumps, where foundations and bolted joints are assumed to be infinitely stiff.

Depending upon the design, installation, and operating speed of the pump, these assumptions may render the calculation completely invalid. It is therefore necessary for the designer to confirm the expected level of error attendant when simple calculations are used. Refer to the relevant section in the standard for a more detailed discussion of the assumptions typical for each analysis type.
One way to reduce the error level (albeit at the expense of increased complexity) is to introduce additional coefficients into the equations based upon experimental or operational data. However, this assumes that the measured coefficients can be successfully isolated from the data. It also assumes the measured coefficients remain constant for different sizes and designs of pumps.

**Level 2 - Intermediate methods involving basic mass elastic modeling using commercially available software tools, such as finite element analysis**

These methods are the most predominant that the industry uses for rotordynamic analysis and are used very commonly for structural analysis. As indicated above, there are exceptions where calculations can be performed using spreadsheets with embedded simple textbook type equations. However, even these calculations by necessity are often hybrids, applying coefficients derived from experimental data or subroutines that apply some form of a numeric method.

 

The main assumption in the finite element analysis method (FEA) is that a continuous structure can be approximated by describing it as an assembly of discrete elements, each with a number of boundary points that are referred to as nodes. Typically for rotor and structural dynamic analysis, element mass, stiffness, and damping matrices are generated first and then assembled into global system matrices. Dynamic analysis of the produced model gives the modal properties: the natural frequencies and corresponding eigenvectors and modal damping. The modal solution can subsequently be used to calculate forced vibration response levels for the structure under study.

In some instances, the dividing line between a level 2, or a level 3 analysis can be difficult to determine. However, the following should be used a guide to determine when an analysis is intermediate (level 2).

Typically, an intermediate analysis will include one or more of the following elements that are not included (except in very simplified form) in a level 1 analysis:

- Damped lateral rotordynamic analyses (eigenvalue calculation and unbalance forced response) of OH or BB pumps (and designs of VS pumps when experience has shown that stiff support structures may be assumed), using a specialist rotordynamic program.
- Radial and axial hydraulic loads are included in the analysis.
- Annular seal dynamic coefficients, impeller-casing interaction coefficients, and, static and dynamic sleeve bearing coefficients; all calculated by programs specifically designed for these types of calculations.
- Nonlinear iteration analysis (for all sleeve bearings), to determine the exact static position of the rotor within these bearings.
- Undamped torsional rotordynamic analyses of simple shaft trains comprising a pump, coupling, and driver (eigenvalue calculation and generic forced response analyses). Either a specialist rotordynamics program or a general-purpose FEA program may be used for the analysis.
- Linear structural analyses using general-purpose FEA programs (static deflection analyses and eigenvalue calculations). The models under investigation are typically limited to well-defined structures resting on stiff support structures (e.g., horizontal pump baseplates or skids bolted onto concrete foundation).

**Level 3 - Complex computational methods involving multiple specialty programs and complex methodologies**

The limitations discussed for an intermediate (level 2) analysis also apply here, only to a greater extent. The following also apply:
- Advanced analysis (level 3) is performed by specialists experienced in resolving the attendant complexities.
- Every level of analysis requires that the engineer understand the assumptions and limitations of the methods used, but this is true in particular for level 3. A level 3 analysis will include one or more of the following elements:
- Damped lateral rotordynamic analyses (eigenvalue calculation and forced response) of vertical pumps (and designs of OH or BB pumps mounted on a flexible support structure), applying specialist rotordynamic programs. The influence of the structure is considered by means of a combined structure–rotor analysis or by considering the *dynamic* bearing support stiffness coefficients calculated in a preceding structural harmonic response analysis.
- Radial and axial hydraulic loads together with loads due to mechanical/hydraulic imbalance are included in the analysis. The worst combination of these loads is used as determined by experience or a sensitivity/parameter study.
- Annular seal dynamic coefficients, impeller-casing interaction coefficients, and static and dynamic sleeve bearing coefficients calculated by programs specifically designed for these types of calculations.
- Nonlinear iteration analysis (for all sleeve bearings), to determine the equilibrium static position of the rotor within these bearings.
- Undamped torsional rotordynamic analyses of shaft trains (complex forced response analyses like motor startup). Complex shaft trains containing gearboxes or gas/diesel engine drivers. Analysis is made using a specialist rotordynamics program.
- Complex structural analyses using general-purpose FEA programs (static deflection analyses, thermal analyses, and eigenvalue calculations). Typical examples are analyses involving significant geometric nonlinearities (bolted connections, gaskets) and/or material nonlinearities (high temperature) as well as any type of thermal transient.

## WHEN A DYNAMIC ANALYSIS SHOULD BE CONSIDERED

Any need for dynamic analysis should be evaluated considering the level of proven field experience of the equipment available for any given configuration, and the consequences of the pump failing. The vendor and user should agree on which types of analysis to be performed (Lateral, torsional, and/or structural analyses are three identifiable and normally separable deliverables), and level of the analysis. In all cases it is the user's prerogative to specify additional tests, validations, and/or analyses to further mitigate risk.

## Market considerations

Historically, dynamic analysis trends have developed within the various pump application markets because of the types and characteristics of equipment typically used, and, as a result of past experiences. Table 1 provides proven experience trends by market. As can be seen from Table 2, many markets have proven designs or designs that must met certain design specifications. In these cases additional dynamic analysis is likely not required. However, markets that have custom designs that are not proven tend to require additional dynamic analysis.

Table 2 Market considerations for dynamic analysis

| Market | Proven Experience Trends |
|---|---|
| Municipal Water and Wastewater | Applications with smaller pumps may be proven in the configuration considered, whereas larger pumps often use custom configuration that tend to be not proven |
| Building Trades and HVAC | Tend to be proven. |
| Electric Power Industry | Often use custom configurations that tend to be not proven. |
| Petroleum (Oil & Gas including API, Pipeline and Water Injection) | Often use custom configurations that tend to be not proven. |
| Chemical Industry | Tend to be proven. |
| Pulp and Paper | Tend to be proven. |
| Slurry | Tend to be proven. |
| General Industry | Tend to be proven. |
| Drainage & Dewatering | Tend to be proven, but large systems may involve variable-fill piping networks and unique supports that are not proven. |
| Irrigation | Tend to be proven. |
| Fire | Tend to be proven but fire pump industry standards apply |
| Flood Control | Often use custom configurations that tend to be not proven. |
| Large Water Transport | Often use custom configurations that tend to be not proven. |

## Quantifying Risk and Uncertainty

The purpose of performing any degree of dynamic analysis is to provide some appropriate level of validation prior to the manufacture or installation of the pump package into its intended operating environment. Validation may take several forms in this context. These include validation by:

- proven example,
- actual physical tests on full-size equipment,
- actual physical tests on similar models (or scale models) of equipment, and
- analytical evaluations on pump, pump support systems, drive systems, and/or local foundations and piping systems.

The appropriate level of evaluation will increase with the degree of risk and the level of uncertainty.

Risk is normally defined by the user, and uncertainty is normally defined by the vendor. However, these roles may be mutual, combined, or reversed where one party has additional knowledge to bring. It therefore becomes incumbent (necessary as a duty) for both parties to reach a decision on when there is need for dynamic analysis.

Within ANSI/HI 9.6.8, tables 9.6.8.3 is a decision matrix, that provides a methodology to assess the uncertainty “U” as step 1 and in provides a methodology to assess the risk “R” as step 2. In step 3, the product of “R” and “U” produce a “RUN” number. The “RUN” number is an indicator of the combined effects of risk and uncertainty on the basis of pump design, system complexity, and pump size. As such, its value may be used as a guide in determining the appropriate analysis level in step 4 of the process.

### Step 1 - Determination of uncertainty “U”

The uncertainty number value “U” increases with increased uncertainty as related to the various application factors characteristic of the pump and the associated equipment. It may relate to the level of confidence in the reliability of the equipment to perform the intended service or to avoid any type of catastrophic failure based on knowledge of that equipment. Higher uncertainty indicates reduced confidence and therefore a greater need for analytical evaluation to "know" the equipment. An excerpt of Figure 9.6.8.3.1 is presented in this tutorial as Table 3. It quantifies the uncertainty values for OH & BB pumps with rigid rotor designs.

Table 3 – Uncertainty table for OH & BB pumps with rigid rotor designs (excerpt from Figure 9.6.8.3.1)

| Pump Type | Lateral Rotordynamic Analysis | Torsional Rotordynamic Analysis | Structural Dynamic Analysis |
|---|---|---|---|
| Types OH & BB Pumps with Rigid Rotor Designs | Maximum speed > 3800 rpm, U = 2<br>Fly wheel driven, U = 2<br>Drive shaft driven, U = 2<br>Variable speed driven, U = 2<br>Power > 30 kW (40 bhp) and < 375 kW (500 bhp), U = 1<br>Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2<br>Power > 750 kW (1000 bhp), U = 3<br>No. of vanes = 3 or fewer, U = 3 | Trains with three or more elements, U = 1<br>Synchronous motor driven, U = 2<br>Fly wheel driven, U = 2<br>Drive shaft driven, U = 2<br>Internal combustion engine driven, U = 2<br>Variable speed driven, U = 3<br>Power > 30 kW (40 bhp) and < 375 kW (500 bhp), U = 1<br>Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2<br>Power > 1000 bhp, U = 3 | Flexible foundations, U = 1<br>Variable speed driven, U = 3<br>Power > 30 kW (40 bhp) and < 375 kW (500 bhp), U = 1<br>Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2<br>Power > 750 kW (1000 bhp), U = 3<br>No. of vanes = 3 or fewer, U = 3<br>NOTE: For vertically oriented OH & BB pump types, use type VS pump values. |
| | System Configuration Total U ______ (Sum) | System Configuration Total U ______ (Sum) | System Configuration Total U ______ (Sum) |

**Step 2 - Determination of Risk "R"**

Similarly, the risk number value "R" indicated in step 2 of Table 9.6.8.3 increases with increased risk as related to the application. It may relate to the likeliness or unlikeliness of a dynamic issue based on previous design history and proven experience. And, the consequences resulting from the unavailability of the equipment to perform the intended service in field or resulting from a catastrophic failure on any part of the equipment. Higher risk indicates a greater need for assessment or analytical evaluation. An excerpt from Figure 9.6.8.3 is presented in this tutorial as Table 4 that helps quantifies the risk number.

Table 4 – Quantification of risk

| Step 2 - Determine and enter risk value "R" from suggested values below. | | |
|---|---|---|
| RISK NUMBER, R | Unknown, new design with no field experience. | 20 |
| | Significant modifications to standard product or similar design - no experience in field. | 10 |
| | Minor modifications to standard product or similar design proven in field. | 4 |
| | Identical or standard product, proven field history. | 2 |

**Step 3 - Determination of the "RUN" score**

Step 3 of the process is to multiply the "R" and "U" values to determine the RUN score. As shown in Table 5 this is done for each analysis type. The product calculated here,

Table 5 – Calculation of "RUN"

| Step 3 - Multiply the "R" values from step 2 times the risk value "U" selected in step 1 for each type of analysis. These are the "RUN" values. | | |
|---|---|---|
| Lateral | Torsional | Structural |
| | | |
| Products of R x U, or RUN numbers | | |

**Step 4 – Determination of analysis level based on RUN**

Table 6 illustrates the final step in the process in determining the appropirate analysis level. Based on the RUN, the level of analysis ranges from none require to level 3 plus validaiton of the analysis.

Table 6 – Recommended analysis level based on the “RUN”

| Step 4 - Using the calculated "RUN" value from step 3 for each analysis type (lateral, torsional, or structural), determine the suggested level of analysis for each type of analysis from the guidelines below. | |
|---|---|
| RUN value from step 3 | Suggested level of analysis |
| ≤ 15 | None Required |
| > 15, ≤ 20 | Level 1 |
| > 20, ≤ 50 | Level 2 |
| > 50, ≤ 160 | Level 3 |
| > 160 | Level 3 +Validation* |

**Worked example of “RUN” methodology**

The example provided, illustrates how to evaluate the need for dynamic analysis and the appropriate levels for a pump system rerate following the four steps presented above.

**Existing situation**

- The worked example presented in this section is for a vertically suspended (VS3) cooling water circulation pump.
  - Two pump operation is required to meet demand when the system was originally design for a single pump
  - Pumps are operating at approximately 80% of the best efficiency point (BEP)
  - Pumps require repair more often than is normal

**Rerate situation**

- Rerate the pump to meet new design point so that a single pump can meet demand
  - New impeller and suction bell
  - Increased head and capacity
  - Increased power requires larger 2000 hp synchronous motor

**Step 1 - Determine uncertainty “U” for rerate**

The first step, is to determine the uncertainty value for the rerate situation for all three types of dynamic analysis. Table 7 is an excerpt from Figure 9.6.8.3.1 in ANSI/HI 9.6.8. The excerpt relates to vertically suspended pumps. This table provides “U” values for each analysis case.

- The lateral analysis has a U of 3 due to the power of the driver, with no other criteria contributing.
- The torsional analysis has a U of 5 due to the power of the driver and the synchronous design of the driver
- The structural analysis has a U of 7 due to the driver being supported by the pump, the flexible foundation and the power of the driver.

Regarding the foundation rigidity that is a consideration for the structural analysis uncertainty, ANSI/HI 9.6.8 provides methods to determine if the foundation is flexible or rigid in section 9.6.8.3.1. If the foundation rigidity cannot be determined or verified, then a flexible foundation should be assumed. The U values determined in Table 7 will be carried forward and used in step 3.

Table 7 – Determination of uncertainty for rerate

| Pump Type | Lateral Rotordynamic Analysis | Torsional Rotordynamic Analysis | Structural Dynamic Analysis |
|---|---|---|---|
| VS Pumps | Maximum speed > 3800 rpm, U = 2 | Trains with three or more elements, U = 1 | Drivers supported separately, U = 1 |
| | Specific gravity < 0.7, U = 2 | Synchronous motor driven, U = 2 | Drivers supported by pump, U = 2 |
| | Fly wheel driven, U = 2 | Fly wheel driven, U = 2 | Flexible foundations, U = 2 |
| | Drive shaft driven, U = 2 | Drive shaft driven, U = 2 | Variable speed driven, U = 3 |
| | Variable speed driven, U = 2 | Internal combustion engine driven, U = 2 | Power > 30 kW (40 bhp) and < 375 kW (500 bhp), U = 1 |
| | Power > 30 kW (40 bhp) and < 375 kW (500 bhp) U=1 | Variable speed driven, U = 3 | Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2 |
| | Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2 | Power > 30 kW (40 bhp) and < 375 kW (500 bhp), U = 1 | Power > 750 kW (1000 bhp), U = 3 |
| | Power > 750 kW (1000 bhp), U = 3 | Power > 375 kW (500 bhp) and < 750 kW (1000 bhp), U = 2 | No. of vanes = 3 or fewer, U = 3 |
| | Number of Vanes ≤ 3, U = 3 | Power > 1000 bhp, U = 3 | |
| | | | |
| | System Configuration Total U = 3 (Sum) | System Configuration Total U = 5 (Sum) | System Configuration Total U = 7 (Sum) |

**Step 2 - Determine risk "R" for rerate**

Step 2 of the process is to determine the risk associated with the rerate.
*After review with manufacturer it was determined that a 2000 hp synchronous motor had never been proven on this discharge head, and, the pump impeller & suction bell were a new design; however, they have been proven on similar designs. For this reason the risk was determined to be a significant modification to a standard product or similar design with no field experience.

Table 8 – Determination of risk for rerate

| Step 2 - Determine and enter risk value "R" from suggested values below | | | Enter selected "R" Value |
|---|---|---|---|
| RISK NUMBER "R" | Unknown, new design with no field experience | 20 | 10* |
| | Significant modification to standard product or similar design - no field experience | 10 | |
| | Minor modification to standard product or similar design with proven field experience | 4 | |
| | Identical or standard product, with proven field history | 2 | |

**Step 3 & Step 4 - Determination of "RUN" and recommended analysis levels**

The final two steps in the process are to multiply the risk and uncertainty to determine the RUN value as shown in Table 9. Finally, Table 10 shows the recommended level for each analysis based on the RUN value. In this rerate example level 2 analysis is recommend for the lateral and torsional cases and a level 3 analysis is recommended for the structural case.

Table 9 – "RUN" Calculation

| Step 3 - Multiply the "R" value in step 2 by the "U" value in step 1 | | |
|---|---|---|
| Lateral | Torsional | Structural |
| 3 X 10 = 30 | 5 X 10 = 50 | 7 X 10 = 70 |
| Products of R X U, or RUN numbers | | |

Table 10 – Determination of analysis level

| Step 4 - Determine Level of analysis based on the calculated RUN value | |
|---|---|
| RUN value | Recommended Analysis Level |
| ≤ 15 | None |
| > 15 but ≤ 20 | Level 1 |
| > 20 but ≤ 50 | Level 2 |
| > 50 but ≤ 160 | Level 3 |
| > 160 | Level 3 + Validation |

**SAMPLE SPECIFICAITONS**

ANSI/HI 9.6.8 provides sample specification language in its Appendix E that can be used to ensure the guideline's recommended analyses are received. A user, after deciding what type and level of analyses (lateral, torsional, or structural) are needed, then need only copy and paste the applicable sample specifications provided in the appendices. The user must provide the desired margins of separation in the spaces provided. These provide a simple way for specifiers to obtain the correct analyses with minimal confusion and without having to be an expert. Sample specifications are provided specific to vertical pump/motor structures as well in Appendix F.

Presented in this section is separation margin and sample specifications for the level 3 structural dynamic analysis, for the rerated vertical pump example presented above.

**Separation margin**

ANSI/HI 9.6.8 provides recommendations for separation margin; however, the sample specification in ANSI/HI 9.6.8 require the user to input their desired separation margin analysis being specified.

A separation margin of 10% obtained in the field conditions is typically satisfactory to avoid unacceptable vibration response amplification (applicable to all modes). This is illustrated in the previously discussed Figure 4, which shows that not much benefit is indicated by a margin that exceeds 10%.

However, a higher separation margin by calculation is recommended, typically 25% for level 1 and 15% for levels 2 and 3 (applicable to all modes).

Higher margins (by analysis) may be specified; however, it is cautioned that such margins may not be necessary. Furthermore, using higher separation margins may incur the need for additional analytical studies and design measures with subsequent increases in lead time and cost of equipment.

In common cases involving variable speeds with multiple excitation sources, successful solutions by analysis that comply with the required margin may not be obtainable. This is a situation that is amplified by the use of higher margins. The Campbell diagram in the previously discussed Figure 5 illustrates the difficulty in obtaining a satisfactory separation margin in such an application with a specified margin of only 15%.

**Level 3, structural analysis, VS pumps, driver baseplate structural system specification language**

For the rerate scenario the following specification language from ANSI/HI 9.6.8 can be used. The result should be an analysis per the guideline recommendations that considers all the required inputs.

> Sample specification language: To determine the potential for a critical structural natural frequency occurring within the normal operating speed range of the pump, a level 3 structural dynamic analysis shall be performed in accordance with ANSI/HI 9.6.8 *Rotodynamic Pumps Guideline for Dynamics of Pumping Machinery*, Table 9.6.8.4.
>
> The structural analysis shall include foundation mass and stiffness within a radial distance (measured from the center of the pump base) at least equal to the height of the top of the motor. This is relative to the level of attachment of the baseplate to the floor and piping details. They are important to modal mass and stiffness (including enclosed fluid) within a spherical zone of radius (relative to the centerline) equal to twice the height of the top of the motor relative to the level of attachment of the baseplate to the floor.
>
> The pump structure shall be subject to a natural frequency analysis (modal FEA) in accordance with Section 9.6.8.6.2.3.3. The pump should be considered complete with driver, baseplate, and structural system. The minimum frequency separation margin obtained by analysis shall be ± _____ (to be completed by specifier). Pumps shall also be subject to a forced response analysis in accordance with Section 9.6.8.7.2.3.2. The forced response analysis of the structure shall demonstrate compliance with the vibration acceptance criteria as per ANSI/HI 9.6.4 *Rotodynamic Pumps for Vibration Measurements and Allowable Values*.

Of particular importance to the structural analysis of a vertical pump is the reed frequency of the vertical motor. Appendix F of the guideline provides sample specification language for the procurement of the vertical motor. The purpose of this sample specification is to help ensure that stated vertical motor reed critical frequency values and related motor properties are reasonably accurate for motors used in vertical pump/motor structures. Such values must be validated for use in dynamic analyses performed for such structures in order to avoid resonance. This information is intended to supplement the sample specifications information provided for the vertical structural analysis.

Sample specification language for vertical motors:

1. In the proposal phase, the expected reed critical frequency (RCF), mass, and location of the center of gravity of the motor shall be determined by the vendor and provided to the purchaser as part of the tendering documentation.
2. The accuracy of this information shall be understood to be within ±10% of the values that would be obtained by verification methods. In the case of RCF, this would be an impact test in either of two perpendicular planes when the motor is rigidly attached to a rigid foundation mass at least 10× as stiff and 10× as massive as the motor itself. The ±10% range of variance shall be used in the structural dynamic analysis, as applicable.
3. After manufacture of the motor and before shipment, the motor RCF, mass, and center of gravity properties shall be verified at the motor manufacturer's facility. In the case of RCF, this involves an impact test in two perpendicular planes, to be denoted relative to the conduit box location.
4. The report provided to the purchaser before shipment shall include:
    a. the impact test results (as-built RCF) in both directions (to be denoted relative to the conduit box location),
    b. the as-built mass of the motor, and
    c. the as-built center of gravity location shall be provided to the purchaser of the motor.
5. If the as-built values of reed critical frequency, mass, and center of gravity location are outside of the allowed tolerances, then the vendor shall inform the purchaser of the motor and ensure that the matter is resolved with the purchaser, before shipment of the motor.

## CASE STUDY OF DYNAMIC ANALYSIS PER ANSI/HI 9.6.8 SPECIFICATIONS AND RESULTS

### System to be analyzed

Dewatering pump station with five (5) parallel submersible pumps at design phase and prior to final pump procurement.

### Purpose of analysis

To verify that the pump station design would not contribute to any structural vibration resonances of the five deep submersible pumps, nor would any downstream piping and valve assemblies have harmful resonances. Provide recommendations in order to reduce any likely resonance conditions.

### Specified analysis

Level III structural analysis per Appendix E of ANSI/HI 9.6.8, including the support structure, representative pumps, check valve assemblies, and additionally the submersible pumps' discharge piping.

1. Specified separation margin: 15%
2. Acceptance: Achieve specified separation margins from 1x and vane pass running speed excitations.
3. Forced response analysis: If separation margins are not met via modifications, predict vibration levels using assumed levels of damping and excitations.

### Analysis results per ANSI/HI 9.6.8 and recommendations

Preliminary results for the original spacing of the pipe supports predicted potential interferences of 1x pump RPM with the 1$^{st}$ bending modes of the discharge piping. It also predicted interferences of vane pass frequency with the 2$^{nd}$ bending modes of the discharge piping. Therefore, multiple iterations were made to the spacing of all pipe supports to dial in the ideal natural frequencies of the piping to be at least ±15% from 1x pump RPM and VPF taking into consideration 3 distinct water levels in the station. Once these modifications were made, the FEA results revealed that most of the vibration modes of the pump structure and piping system fell outside of any zones which would cause interference. However, the first bending mode of the piping guide rail was still predicted to fall within ±15% of the operating frequency of the pump. Additionally, it was discovered that the 3$^{rd}$ bending mode of the discharge piping interfered with the Vane Pass Frequency (VPF) of the pump, and both of these required further modifications. A summary of figures from the analysis are shown in Figures 6 through 12.

### Benefits of specifying dynamic analysis per ANSI/HI 9.6.8

The results from the analysis performed at the dewatering plant revealed many piping support deficiencies which needed modification to avoid interference with potential excitations from the pumping equipment. Special care had to be taken to select proper spacing of the discharge piping so that the 1$^{st}$ and 2$^{nd}$ bending modes avoided 1x RPM and VPF excitations over a range of water levels. Additional supports to support the guide rails also were recommended. In order to prevent interference from the 3rd bending mode of the discharge piping, it was recommended that an additional piece of piping be used to limit possible vibration. This additional pipe section was shown to reduce the effective length of the pipe, raising the natural frequency to above the interference zone. Since all the modes of interest were shown to have sufficient separation margin from the considered excitations, additional forced response analysis was not necessary. Without this extensive upfront analysis being performed per ANSI/HI 9.6.8, the likelihood of encountering vibration issues was high, and if the recommendations are implemented, the risk of having response issues should be greatly reduced.

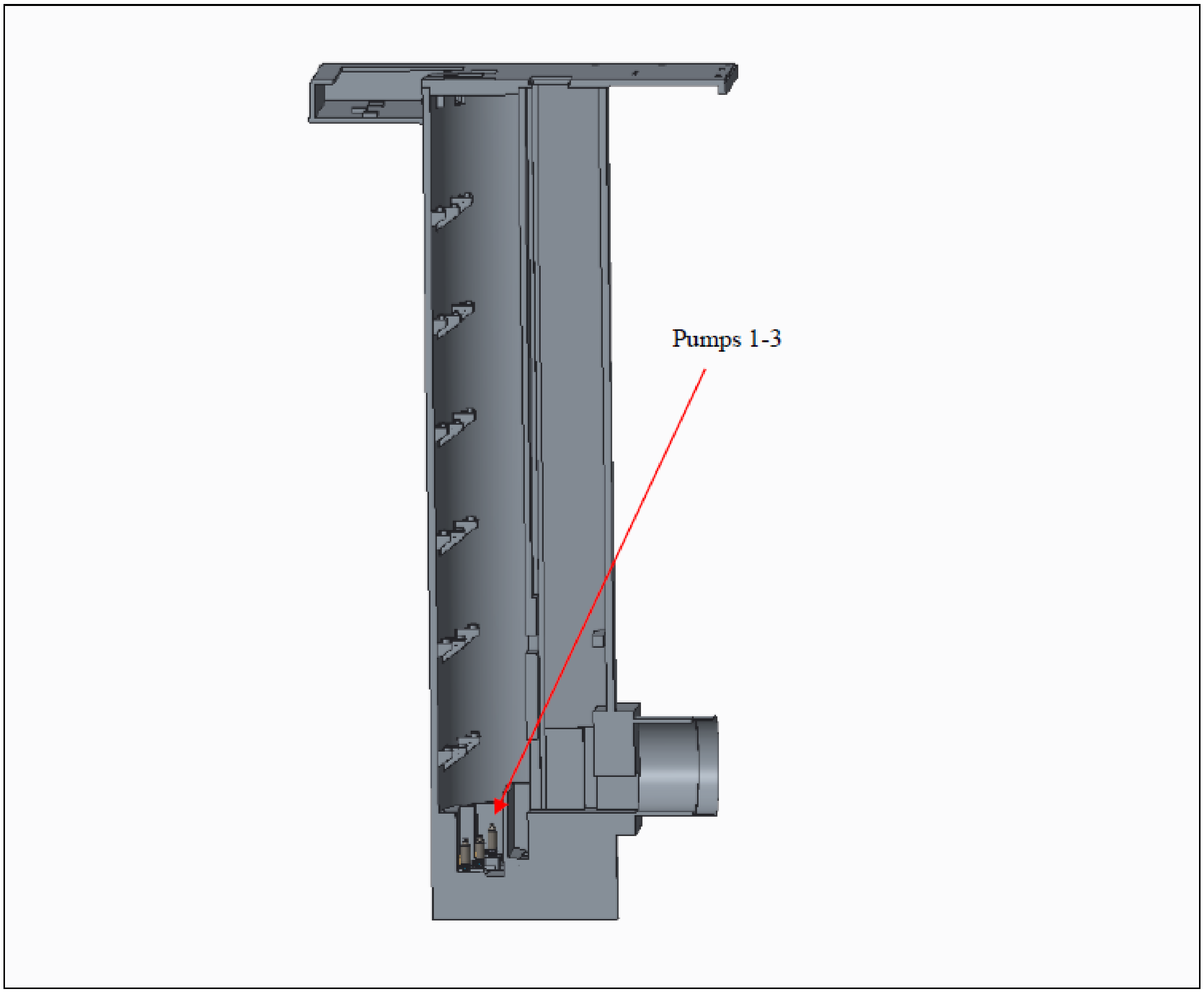


**Figure 6:** Finite element analysis half-symmetry solid model of the entire pump station.

**Note:** Discharge piping removed from view for easier visualization

 

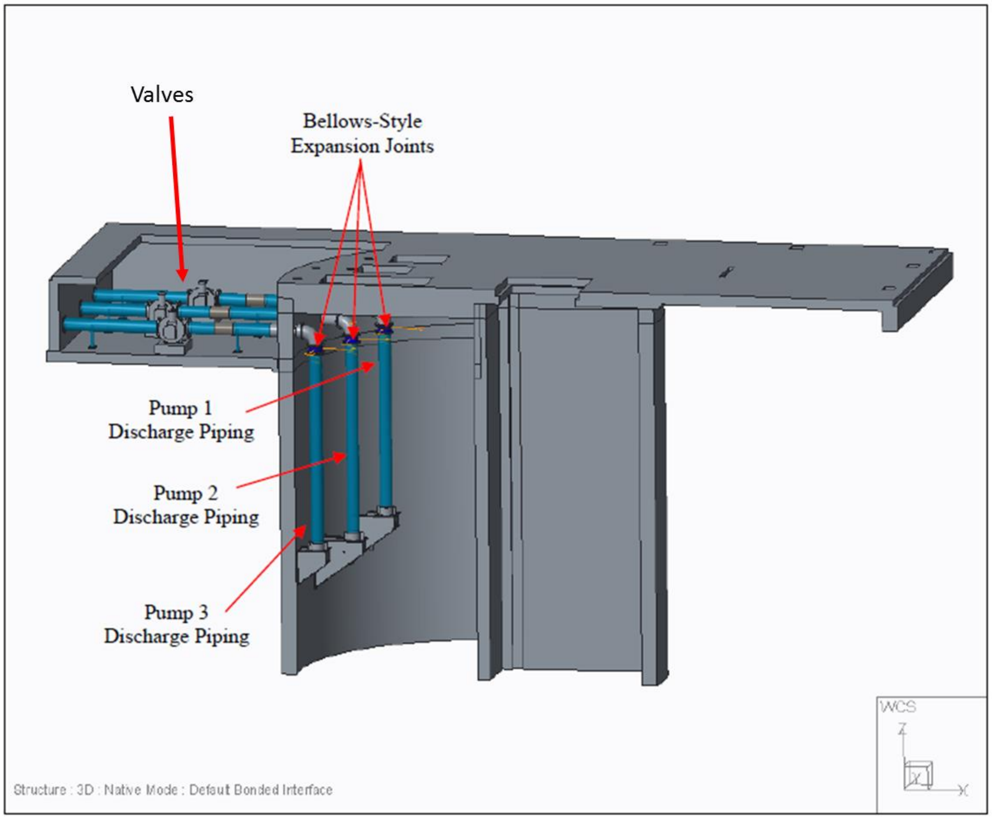


**Figure 7:** Finite element analysis solid model of the upper portion of the pumping system.

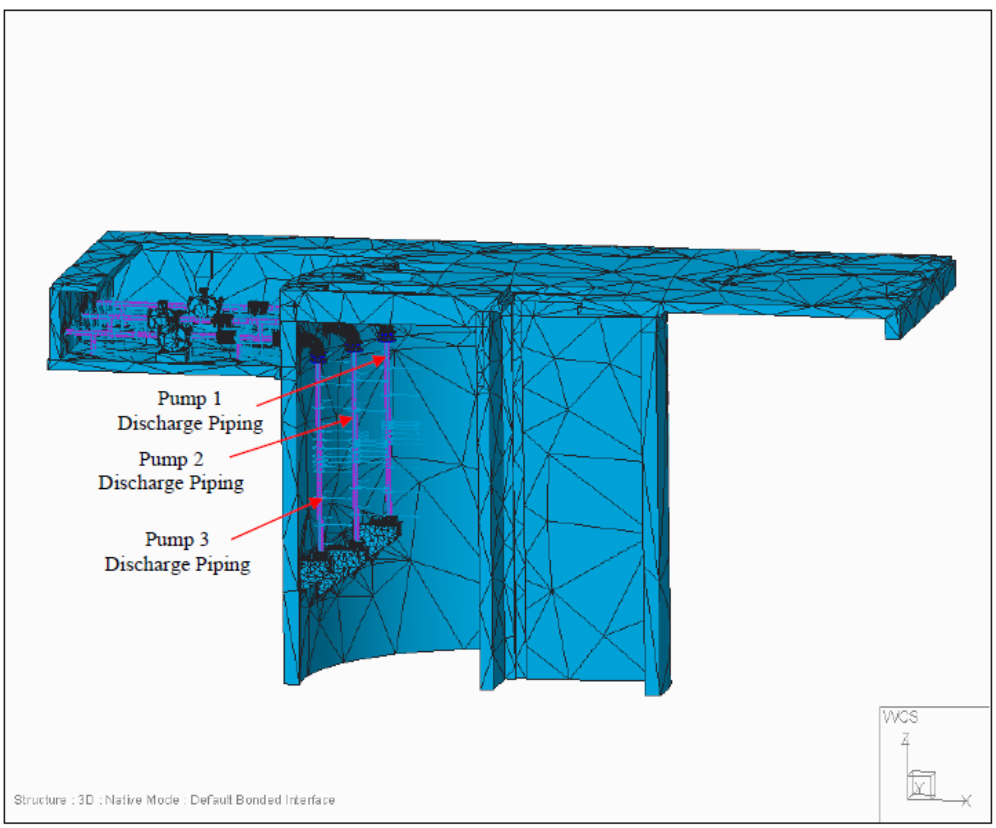

**Figure 8:** Mesh of the solid model in order to predict natural frequencies of the upper portion of the pumping system.

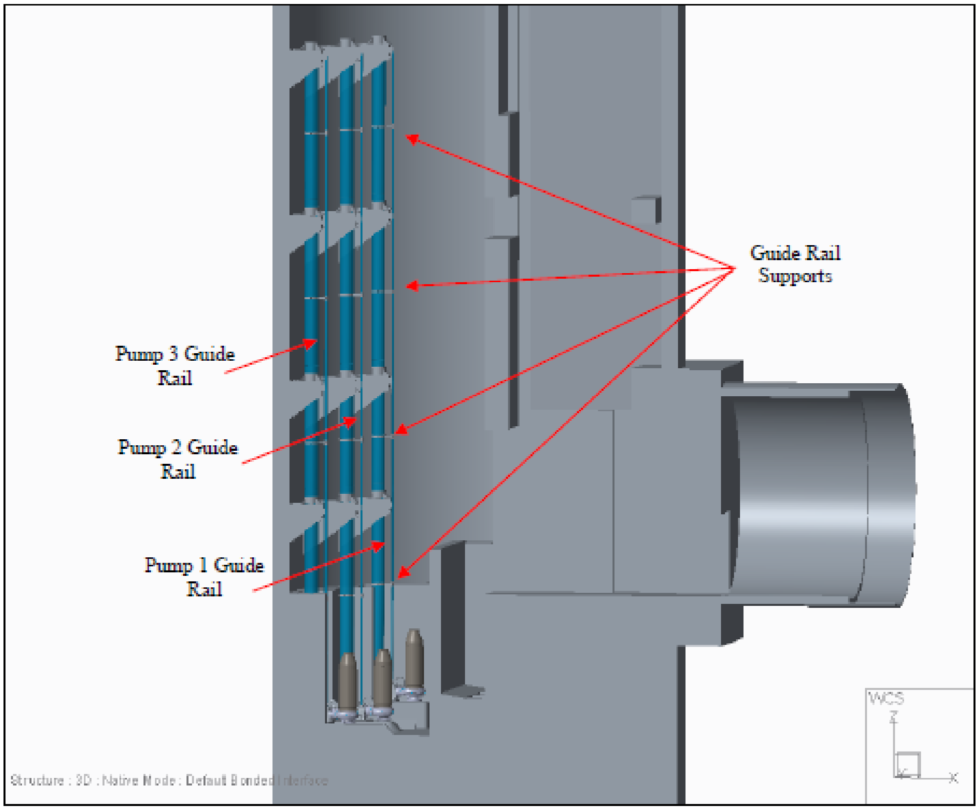


**Figure 9:** Finite element analysis solid model of the lower portion of the pumping system.

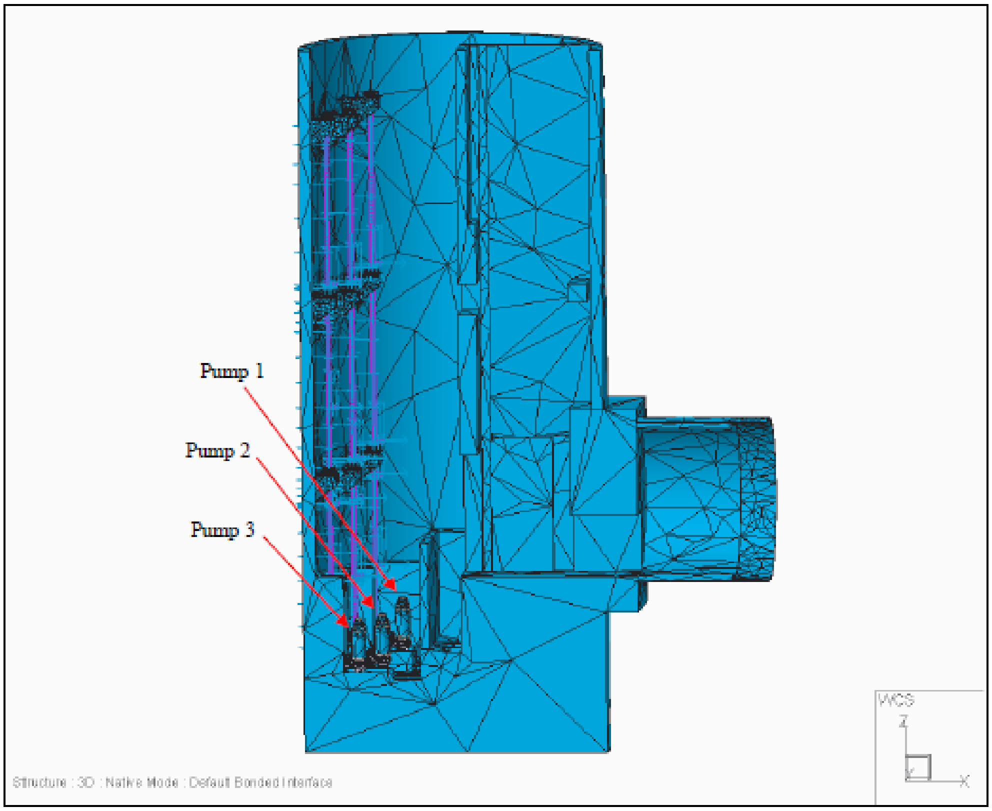


**Figure 10:** Mesh of the solid model in order to predict natural frequencies of the lower portion of the pumping system.

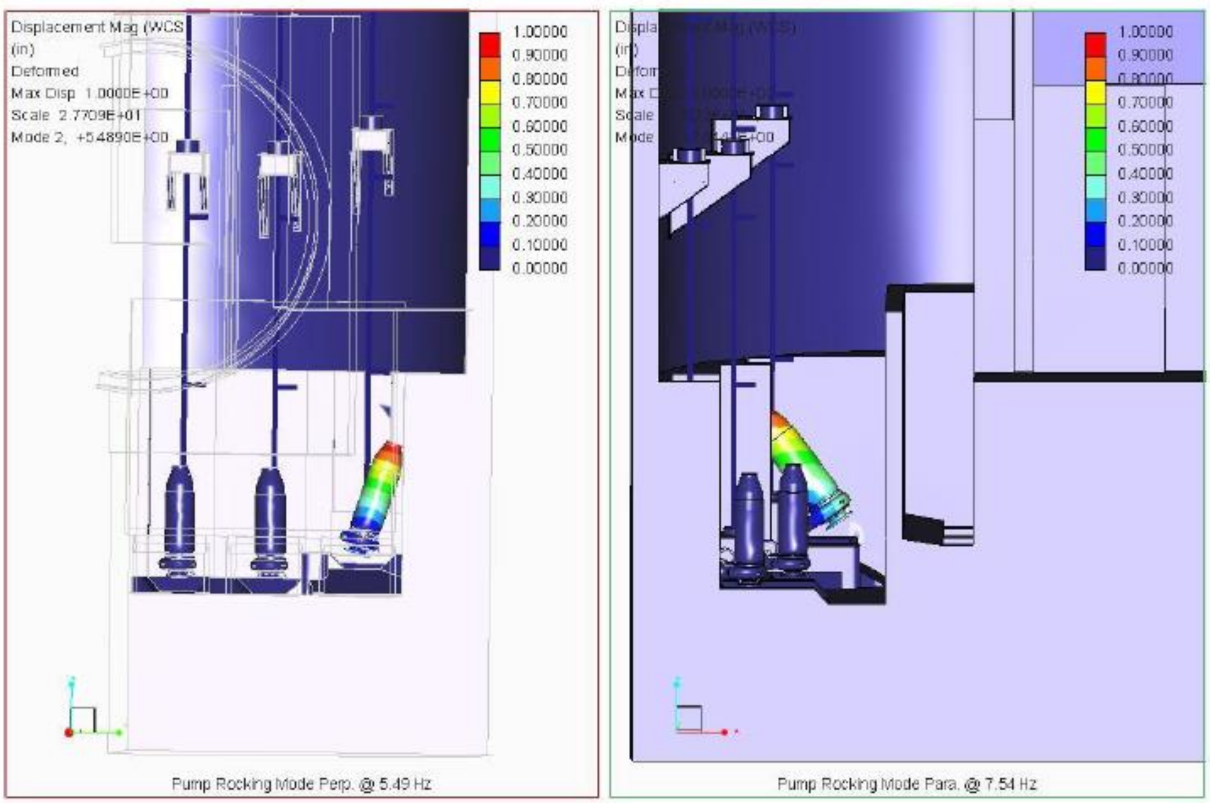


**Figure 11**: Plot of the mode shape of the pump casing at 5.5 Hz during normal operation in current configuration.

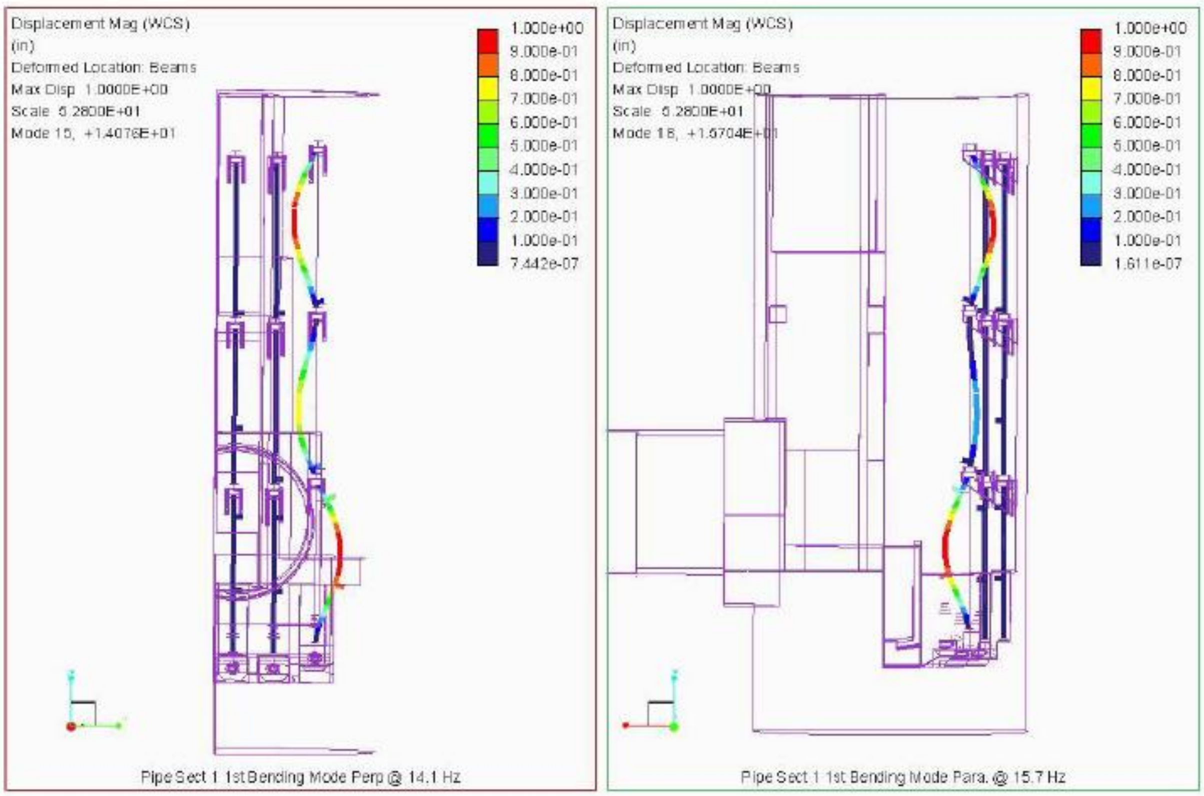


**Figure12:** Discharge piping displacement shape of the first bending mode in the perpendicular and parallel direction to the pump discharge.

**CONCLUSIONS**

The limitation and avoidance of undesirable pump vibration requires an evaluation of dynamic effects at the design phase. When dynamic effects are not considered, resonant vibration and failures may often occur. Addressing a vibration problem after installation is stressful for all involved parties and costs are many times more than performing the proper analysis upfront. The ANSI/HI 9.6.8 guidelines provide the user with tools by which to determine the type of dynamic analysis that is appropriate, the level of detail of that analysis, and most importantly, an easy use of specification language to allow a non-expert to correctly specify and receive the desired analysis.

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**ACKNOWLEDGEMENTS**

Acknowledgments are given to the Hydraulic Institute committee that developed ANSI/HI 9.6.8 over a ten year period, with special thanks to the committee chair and steering committee member - Jack Claxton, the committee vice-chair and steering committee member - Mick Cropper, steering committee member – Bill Marscher, and steering committee member – John Anspach.  Additionally we thank the Hydraulic Institute for use of the ANSI/HI 9.6.8 guideline materials in this tutorial, as well as Mechanical Solutions Inc. for providing the case study of the guidelines use.

 