ASIA TURBOMACHINERY & PUMP SYMPOSIUM
SINGAPORE | 22 – 25 FEBRUARY 2016
MARINA BAY SANDS

SHOP ROTORDYNAMIC TESTING – OPTIONS, OBJECTIVES, BENEFITS AND PRACTICES

Dr. John A. Kocur, Jr.
ExxonMobil Research & Engineering
3225 Gallows Road, Rm 2A-0601
Fairfax, VA 22037-0001
(703) 846-2275
john.a.kocur@exxonmobil.com

Dr. C. Hunter Cloud
BRG Machinery Consulting LLC
3241 Plank Road
North Garden, VA 22959
(434) 293-9926
chcloud@BRGmachinery.com

John A. Kocur, Jr. is a Machinery Engineer in the Equipment Engineering Division at ExxonMobil Research & Engineering in Fairfax, Virginia. He has worked in the turbomachinery field for 30 years. In his current capacity, he provides support to the downstream, upstream and chemical business lines within ExxonMobil with expertise on vibrations, rotor/thermo dynamics, failure analysis and health monitoring of rotating equipment. Prior to joining EMRE, he held the position of Manager of Product Engineering and Testing at Siemens Demag Delaval Turbomachinery. There Dr. Kocur directed the development, research, design, engineering and testing of the compressor and steam turbine product lines. He has also held positions with Pratt & Whitney and Amoco Corporation.

Dr. Kocur received his BSME (1978), MSME (1982) and Ph.D. (1991) from the University of Virginia and an MBA (1981) from Tulane University. *He has authored papers on rotor instability and bearing dynamics, lectured on hydrostatic bearings, has sat as a committee chairman for NASA Lewis and is a member of ASME. Currently, he holds positions within API as 617 chair, 684 chair and member of Subcommittee on Mechanical Equipment (SOME).*

C. Hunter Cloud is President of BRG Machinery Consulting, LLC, in Charlottesville, Virginia, a company providing a diverse range of rotating machinery technical services. He began his career with Mobil Research and Development Corporation in Princeton, NJ, as a turbomachinery specialist responsible for application engineering, commissioning, and troubleshooting for production, refining and chemical facilities. During his 11 years at Mobil, he worked on numerous projects, including several offshore gas injection platforms in Nigeria as well as serving as reliability manager at a large US refinery.

Dr. Cloud received his B.S. (Mechanical Engineering, 1991) and Ph.D. (Mechanical and Aerospace Engineering, 2007) from the University of Virginia. He is a member of ASME, the Vibration Institute, the API 684 rotordynamics task force, and the advisory committee for the Texas A&M Turbomachinery Symposium.

ABSTRACT

Understanding the lateral rotordynamic behavior is critical in determining the reliability/operability of rotating equipment. Whether examining a centrifugal pump or compressor, steam or gas turbine, motor or generator, rotating machinery share the same need to accurately predict and measure dynamic behavior. Industrial specifications determining fit for purpose rely on the accuracy of rotordynamic predictions where direct measurement is impractical or otherwise impossible in an industrial setting. Testing to confirm rotordynamic prediction and behavior provides both the purchaser and vendor the confidence that the design will meet project expectations.

Rotordynamic shop testing has several options available to the project during acceptance tests at the vendor's shop. These options include mechanical run, string and full load/Type 1 testing as well as verification testing to validate unbalance response and stability predictions. Such testing has numerous advantages; the most important being the avoidance of production disruptions involved with testing at the job site. Each test option has associated costs as well as limitations as to what lateral vibration characteristics are revealed. Understanding these factors is vital to efficiently mitigate the risks associated with the purchased equipment.

Applying best practices and an understanding of the industrial (API) test requirements are needed to derive the maximum benefit of each test option. The best practices not only involve the test procedure but also the associated analytical methods used to post process the measurement information. Whether performing a simple mechanical run test or more complex stability verification during ASME Type I testing, ensuring that a logical, repeatable and proven methodology is followed produces reliable evidence to confirm the rotordynamic model and lateral vibration performance. The rationale behind the API test requirements provides an understanding of why that test is being performed and its correct application to the dynamic behavior.

Test options can be separated into two categories; tests that reveal portions of the dynamic behavior of the equipment to specific operating conditions and those used to verify the analytical predictions of that behavior. API mechanical, string and Type I (or full load) tests reveal the rotordynamic behavior of the equipment to a given set of conditions. These are used specifically to determine acceptability of the design. Unbalance and stability verification testing is used to confirm (or provide confidence in) the rotordynamic model. Confidence in the model permits extrapolation of the design (vendor) and operation (purchaser) beyond the machine's as-built and specific shop test conditions.

INTRODUCTION

Demands on turbomachinery continue to push designs beyond experience limits in terms of speed, power, size, pressure development and flow rate all the while demanding higher reliability and operability. To meet the stringent objectives of the application, almost absolute knowledge of the behavior of the machine is necessary. To aide in this understanding, advanced analytical methods have been developed in parallel with rigorous testing practices.

Shop acceptance testing has long been used as the "final" check of the turbomachinery design and is required for all special purpose equipment. With the advancement of computational methods, failure rates during testing have been greatly reduced. However, experience has taught us analytical methods alone are insufficient to guarantee the "right the first time" philosophy that many reliability systems employ. Whether used to provide data to baseline prototype equipment, to benchmark extensions of current experience limits, or for verification of proven practices, testing remains an integral part of all reliability systems.

Understanding the dynamic behavior is critical in determining the reliability/operability of rotating equipment. Whether designing a pump, compressor, turbine, motor or generator, all rotating machinery share the same need to accurately predict and measure dynamic behavior. Literature is swamped with failures that resulted from both inadequate design and testing methodologies. While the potential for failure originates at the design stage, testing represents the final step to identify that potential. The importance of efficiently employing both cannot be understated [1].

Turbomachinery is dominated by two classes of dynamic behavior; rotordynamics and fluid dynamics. As their names imply, each focuses on a specific dynamic behavior; rotordynamics on the rotating system's vibration and fluid dynamics on the mechanical interaction with the working fluid. As our depth of understanding increases, the more interrelated these behaviors become. Decisions made in the design of one can impact the other with sometimes disastrous effects. Nowhere is this more evident than centrifugal equipment, especially compression. For this reason, this tutorial will focus on testing the rotordynamic behavior of centrifugal compressors. However, most of the principles and practices are applicable across all types of turbomachinery.

Rotordynamic behavior testing involves both direct measurement and inference. Direct measurement of the vibration is typically made at the journal locations. Internal vibration levels at other critical locations can only be inferred from these measurements using the rotordynamic predictions. Thus, industrial specifications determining fit for purpose rely on the accuracy of rotordynamic predictions where direct measurement is impractical or otherwise impossible in an industrial setting. Testing to confirm rotordynamic prediction and behavior provides both the purchaser and vendor the confidence that the design will meet project expectations.

Rotordynamic shop testing has several options available to the project during acceptance tests at the vendor's shop. These options include tests to demonstrate operating behavior at a specific condition (mechanical run, string and full load/Type 1 testing) and verification testing to validate unbalance response and stability predictions. Such testing has numerous advantages; the most important being the avoidance of production disruptions at the job site. Each test option has associated costs as well as limitations as to what lateral vibration characteristics are revealed. Understanding these factors is vital to efficiently mitigate the risks associated with the purchased equipment.

Application of best practices and an understanding of the industrial (API) test requirements are needed to derive the maximum benefit of each test option. The best practice not only involves the test procedure but also the associated analytical methods used to post process the measurement information. Whether performing a simple mechanical run test or more complex stability verification during ASME Type I testing, ensuring that a logical, repeatable and proven methodology is followed produces reliable evidence to confirm the rotordynamic model and lateral vibration performance.

Test options can be separated into two categories; tests that reveal portions of the dynamic behavior of the equipment to specific operating conditions (Vibration Demonstration Tests) and those used to verify the analytical predictions of that behavior (Design Verification Tests). API mechanical run, string and Type I (or full load/full pressure) tests reveal the rotordynamic behavior of the equipment for a given set of conditions. These are used specifically to determine acceptability of the design in a pass/fail mode. Unbalance response and stability verification testing is used to confirm (or provide confidence in) the rotordynamic model and analysis. Confidence in the model permits extrapolation of the design (vendor) and operation (purchaser) beyond the machine's as-built and specific shop test conditions.

The tutorial will cover the following aspects of rotordynamic testing:

- Decision to test: Why and on What Basis?
- Rotordynamic Testing
  - Options
  - Objectives
  - Preparation
  - Information / Knowledge Gain

- Benefits

- Recommended Practices
    - Vibration Demonstration Tests
    - Design Verification Tests

DECISION TO TEST: WHY AND ON WHAT BASIS

The need for rotordynamic testing stems from several objectives; to prove the behavior of the machine, to test the accuracy of the vendor's predictions and to identify problems before the machine is put into operation. The decision to test and which objectives to pursue result from several factors related to the application. These factors are:

- Risk-consequence analysis considering the following:
    - Impact on operations given failure or performance deficiencies
    - Outage length due to location
    - Experience with similar services
- Safety, Health and Environment (SHE) impact of the project
- Technology application experience (vendor or user)
    - Prototype equipment
    - New application of the technology
    - New arrangement
- Variability of process conditions
- Rotordynamic analysis results

These factors involve the project risks and consequences and should be known at the beginning of the project when the testing decisions are made. Detailed machine design and analysis (which may act to mitigate some of these risks) are not performed until latter stages of the project. Since one goal of the testing may involve verification of the analysis, basing the test decision on the project's risks and consequences makes sense.

Approaching the project from a risk-consequence analysis establishes a logical framework upon which to make decisions [2]. The framework provides a basis for efficiency and enables the correct mitigation activities to be performed. A typical risk matrix is shown in Figure 1.

The risk matrix should be used to address specific concerns of the machinery. For rotordynamics, a possible outcome should be weighed with the factors driving risk and those driving the consequence. (Consequence here is defined as being entirely economic.) Examples of problematic outcomes to consider are:

- Instability (high subsynchronous vibration)
    - Risk: Instability drivers (gas density, speed, ΔP, critical speed location), operating characteristics, deposition plugging seal cavities [1]
    - Scenario: Internal rubbing, bearing damage, component failure
    - Consequence: Downtime impact for repair or reconfiguration (extensive), impact of operation restrictions
- High synchronous vibration levels
    - Risk: Unbalance sensitivity, high amplification factors, critical speed encroachment, mode involved, thermal instability
    - Scenario: Failure of casing attachments, bearing failure, internal rubbing
    - Consequence: Downtime for repairs, reconfiguration costs
- Internal rubbing
    - Risk: Sensitivity of rotor to unbalance, high amplification factor, component involved (blade, laby seal, etc.), unbalance due to erosion/corrosion aspects of working fluid
    - Scenario: Efficiency loss due to expanded seal/tip clearances, blade failure
    - Consequence: Impact of performance loss on operation, downtime impact for repair

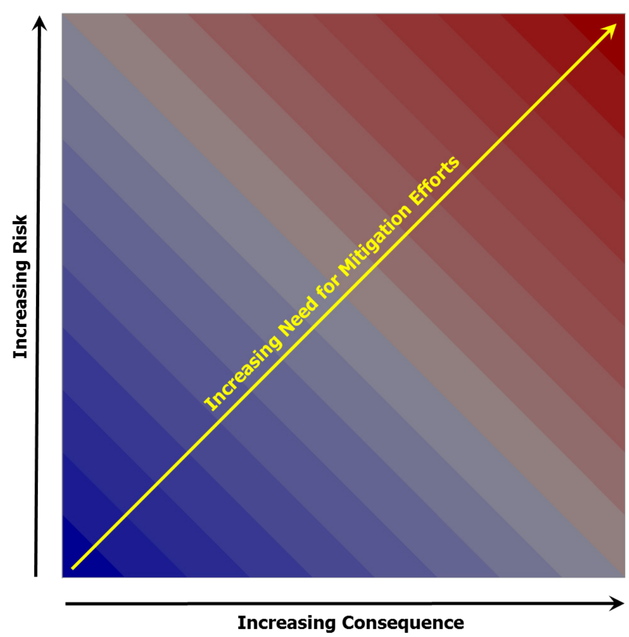

Figure 1) Typical Risk Matrix

[1] K.E. Atkins and R. X. Perez [4] discuss quantification of instability risk using failure rate data.

The user must also consider the safety, health and environmental impact of possible failures that can be attributed to the machinery in question. The impacts can result in the release of gas from component failure due to high vibrations, failure to meet regulatory requirements due to unplanned outage or injury due to parts release during failure. The risks of each can be determined through a failure mode analysis [3] that incorporates the rotordynamic contributions to the identified failure modes.

Experience plays an important role in determining the extent of testing to perform. Experience in this case relates to both the vendor and user. Obviously, prototype machinery or equipment that extends the experience limits of the vendor should be tested thoroughly. What may be overlooked is the experience of the user with that equipment. First application of technology within a user company may benefit from additional testing at the vendor's shop. The testing can be used to better understand the dynamics and what conditions or operating nuances may affect the rotordynamic behavior. Finally, prototype components within the machinery may require additional component testing to determine their impact on the rotordynamic behavior of the machinery.

Finally, the results of the rotordynamics analysis should be used to weigh the decision to test and which test to select. Machines shown sensitive to destabilizing forces or those predicted to have low stability levels may benefit from stability verification tests. Similarly, a rotor with high amplification factors may influence the decision to perform unbalance response verification testing. Verification testing, as noted earlier, is intended to prove the accuracy of the predictive tools used to model the rotor behavior and thus assess the acceptability of the design. The use of a proven rotordynamic analysis is an effective and efficient mitigation strategy towards reducing risk.

ROTORDYNAMIC TESTING

Options

Fundamentally, there are two types of testing options available for lateral rotordynamic assessment. The first type, referred to as vibration demonstration tests, demonstrates the behavior of the as-built machine and/or train for a specific set of operating conditions. Generally, only the site specific instrumentation is used to measure lateral vibration behavior. The tests do not address the accuracy of the model or tools used to predict the rotordynamic behavior, nor do they attempt to estimate or determine the robustness of the design. Those that fall into this category are:

- Mechanical Run – an example is the API required mechanical test [5]
- String – API 617 refers to this as a complete unit test [6]
- Full load/Full pressure – Referred to as a Type I test by ASME [7]

Many important aspects of the rotordynamic behavior cannot be practically measured (i.e. internal vibration levels, separation margin to modes above operating speed) nor can every possible operating condition or combination of assembly /machining tolerances be tested. To fully understand the acceptability of the design and the rotordynamic behavior, we must rely on the accuracy of the rotordynamic analysis. Consequently, tests to determine the accuracy of those predictions were developed. These additional tests were developed to provide more than a pass/fail test, which is essentially what the vibration demonstration tests are. They were implemented to determine the robustness of the design. This is particularly useful when operating conditions are widely variable or when design experience limits are exceeded. The second type of testing option is referred to as design verification tests with the two most prevalent being:

- Unbalance response – This test would include the more invasive testing required by API [8]
- Stability – Not currently specified by API standards.

Verification testing also has the options of where these tests can be performed. In either case, the tests may be performed in a high speed balance bunker, during the mechanical test or during the full load testing. Each option adds additional considerations in what can be measured, dynamic effects included in the test and what portion of the dynamic behavior is analyzed.

Test Preparation

Following the decision to perform a rotordynamic test, the user should decide which specifications to apply to the test. The specification should describe the objectives and requirements of the test. For several of the test options described above, API standards have described a specific test procedure to be followed. The API standard details the procedure, objective and requirements for the test. For the more specialized testing, the user will need to develop their own test specification. This can be done with the assistance of the vendor or by consulting industry specialists.

Test Objectives

The objectives of the tests performed should be discussed at the initial stages of the project. Agreement on the test objectives will assist in the determination of what equipment is needed, measurements to be taken, and conditions to be run. Generally speaking, the objectives of each test are listed below. Other specialty objectives can be added, but the ones listed below would form the basis for each test.

Vibration Demonstration Tests

Mechanical Run Test

The mechanical run test as required by API is primarily a vibration level check. Measured at the probes located at the journal location, vibration levels are checked against the specified limit for both overall and non-synchronous components. General mechanical performance is also examined including bearing temperatures, close clearance rubbing and seal performance typically up to maximum continuous speed (MCS). Supercritical behavior is examined by determination of the amplification factor and separation margin of typically the 1st critical speed. (Obviously, the behavior of modes above MCS remains undetermined.)

String Test

The string test is not much different than the mechanical test mentioned previously. As the name implies, the string test is performed with all or a major portion of the machinery train connected (typically everything but the driver.) The objectives of this test are also similar to the mechanical test of a single body in that vibration levels are checked against limits, mechanical performance is examined, and supercritical behavior is analyzed. However, in this case, these are determined for the coupled train configuration. The string test is run to measure the coupled body dynamic behavior (when rigid couplings are used in the train) or the coupling spacer dynamics (for couplings with unusually long or heavy spacer tubes.)

Full Load / Full Pressure Test

Full load / full pressure tests are rarely performed based on rotordynamic justifications only, mainly due to the costs involved. However, the Type I test does permit vibration level checks at operating conditions, stall determination, impact of internal loading and deflections on dynamic behavior of individual components (seals and bearings) and a binary check for stability (yes or no). Range testing is rarely done during the Type I test. Typically gas properties are held constant, clearances are left at as-built values and alternate configurations are not considered. Thus, while some aspects of rotordynamic behavior are tested, margins and robustness are left unchecked. The machines undergoing these tests leave those factors to analytical studies whose accuracy may remain unchecked. Keep in mind, prediction of the stability condition (stable vs. unstable) of a particular machine is the first step in developing a good analytical tool. However, it is not the only step. As designs extend the operating or design experience, it becomes necessary to predict the stability threshold, separation margins and overall optimization of the design correctly, thus the need for verification testing.

Design Verification Tests

Unbalance Response

The likely first attempt to publish a verification test in an industry standard was the unbalance response verification test published by API 617. The objective was simply to verify the unbalance response prediction accuracy of the vendor's rotordynamic analysis with regards to the machine's unbalance sensitivity within the operating speed range and the location of the critical speeds (below trip speed.) The verification test analyzes both the predictor tool and model employed. A methodology was refined over several editions of API 617 within the limitations of the mechanical shop test. Alternatively, performing this test in the balance bunker has gained acceptance with the increase in at-speed balancing of rotors and the freedom it permits in terms of weight placement and additional measurement points.

Stability

Several methods have been developed analogous to the API unbalance response verification test with the objective to verify the stability predictions of centrifugal compressors. As with the unbalance response, the intent is to measure more than just "is the compressor stable" but "how stable is it." The measurements are then compared against the analytical predictions to determine accuracy. Pettinato *et al.* [9] presented a methodology employing this test during mechanical and performance testing (as required by API 617).

Test Information / Knowledge Gain

To determine the extent of testing to perform, one needs to understand the information or knowledge gained of the rotordynamic behavior of the body or train. While similar information can be obtained from several of the tests, the costs associated with each determine the overall efficiency of obtaining the necessary information to mitigate risks identified in the risk matrix. As before, the tests are separated into vibration demonstration and verification testing. Vibration demonstration testing confirms the machinery can meet project specifications for a *given operating and as-built condition.* No attempt is made to confirm the accuracy of the analytical prediction beyond confirmation that the specification has been met. Since the accuracy of the analytical prediction remains largely in question, inferred information from the analytical method should also be questioned.

Vibration Demonstration tests

Mechanical Run

Mechanical testing provides information related to the critical speed location and some indication of that modes behavior. The modal information is limited to only those modes located below the maximum test speed achieved (trip speed in most cases.) Typically, this is only the 1st critical speed. The modal behavior information is restricted to the amplification factor which is sketchy at best. The amplification factor can be highly sensitive to the acceleration/deceleration rate and whose magnitude is not restricted by API. The test also validates the balance procedure's effectiveness in meeting the project vibration limit specification.

For certain low risk applications, this amount of information is sufficient. What isn't tested however can be significant. For example, subcritical motor applications (incorrectly termed stiff shaft) operate below the 1st critical. This critical speed can have high amplification factors and can be damaging if the separation margin is lost. Performing only a mechanical test will tell the user whether the mode is on or below the operating speed. The amount of separation remains untested and can only be inferred from the unverified analytical predictions. Thus, the impact of support stiffness loss on the location of the mode in the field (due to structural or bearing clearance changes) can have significant risk associated with it.

String Test

String testing will provide the same type of rotordynamic information as obtained with the mechanical testing but at the higher costs of assembling the entire (or high speed portion) of the train. Information again limited to the critical speed location and some indication of the modal behavior. Ignoring other reasons to perform string testing (e.g. fit checks for trains being sent to remote portions of the globe), rotordynamic justification for the test should be limited to the information supplied by the test, namely, dynamic behavior of the coupling spacer(s) and rigid coupling effects on the dynamic behavior of individual bodies. As noted in API 684 [10], a train lateral analysis should be requested for unusually long or heavy coupling spacers or when rigid couplings are used. In this case, the correct boundary conditions at the shaft ends are obtained when the train is modeled (train in this case refers to the bodies on either side of the coupling(s) in question.)

Testing of this train configuration should mimic the analytical model to verify the behavior in question. With a rigid coupling, the rotordynamic behavior of the bodies rigidly coupled will be affected. For unusually long or heavy coupling spacers, the dynamics of the spacer can only be accurately modeled/tested with the hubs attached to the shafts.

Full Load / Full Pressure (FLFP) Test

Type I tests provide a stable vs. unstable behavior indication of the rotor to specific test conditions. The test also reveals the change in lateral behavior of the measured modes (typically the 1st critical speed as with the mechanical test) due to internal loading of the compressor and application of gas pressures and densities close to the design values at the seals. The latter introduces seal dynamics into the testing that is only achieved during the FLFP test. This is important for both stability and synchronous behavior especially for machines incorporating hole pattern or honeycomb seals. The presence of subsynchronous vibration due to phenomena such as stall, surge or whirl may also be revealed during the FLFP test.

Design Verification Tests

Unbalance Response

The unbalance response verification test (URVT) provides a measurement of how well the analytical predictions match the vibration produced from a known unbalance. This in turn adds confidence to the internal deflection, separation margin and unbalance sensitivity calculations made from that analysis. On the shop floor, the unbalance weight addition is typically limited to the coupling. Some turbines and overhung machines have the ability to add internal weights or weights to the overhung impeller. Optionally, the test can be performed in the vendor's balance bunker. The bunker permits more freedom in terms of weight placement and measurement of shaft deflection at points other than the job's proximity probe location (mid-span, for example.) Of course, the analytical model needs to reflect the setup in the bunker; bunker bearings if used and support stiffness of the bunker pedestals. Since the intent is verify the accuracy of the analytical predictions, these differences in configuration should not affect reaching that conclusion.

Stability

The stability verification (SVT) test provides confidence in the analytical predictions regarding the stability level and position relative to the stability threshold. As with the unbalance response verification, options are available regarding the platform or test configuration to perform this test. The results of each platform can be summarized below regarding the accuracy of the analytic method to predict:

- Balance bunker – Rotor, bearings, and pedestal support impact on stability (bunker bearings if used) at various speeds.
- Mechanical test (Vacuum) – Rotor, job bearings and casing support impact on "baseline" stability (or basic log decrement reflecting the bearings and rotor only) at various speeds.
- Performance test – Rotor, job bearings, casing support and reduced aerodynamic and seal behavior impact on stability for a limited speed range.
- Type I test – Stability level and margin at nearly the same operating conditions as expected in the field. The range of gas conditions, inlet and discharge pressures and flow rates may be limited as a result of the test setup.

Test Benefits

The benefits derived from the rotordynamic testing can be identified for the two groups of testing. Vibration demonstration testing provides the purchaser the following:

- Demonstration that vibration levels and critical speed separation margins (for those under the maximum test speed) specifications have been met – All tests
- A stable vs. unstable check is made for a specific test condition – Type I test
- Proof of effectiveness of the balance procedure in meeting vibration level specifications – All Tests
- Non-synchronous vibration levels examined – Type I test, to some degree all tests

Design verification extends those benefits to:

- Determining accuracy of unbalance response calculation with regards to the unbalance sensitivity – Unbalance verification (shop floor)
- Determining predictive accuracy for mid-span unbalances and deflections – Unbalance verification (at-speed balance bunker)
- Reassurance that mid-span deflections are within operating clearances for close clearance locations – Unbalance verification (higher for at-speed balance bunker)
- Verification of stability level prediction for system with no excitations (baseline stability) – Stability verification (at-speed balance bunker and during the mechanical run test)
- Reassurance that rotordynamic model of shaft and bearings is accurate – URVT and SVT
- Robustness and optimization of machinery design – URVT and SVT
- Correctness of the stability model of impellers, annular seals and other destabilizing mechanisms as well as the effectiveness of any components which are

intended to reduce the destabilizing effects (shunt bypass, swirl brakes, damper seals) – Stability verification (PTC Type I and Type II testing)

The at-speed balance bunker can extend the benefits of the URVT by permitting weight placement at and measurement of locations that are more sensitive to unbalance and the negative effects of high vibrations. For between bearing machinery, critical close clearance locations that impact performance are located at or near the mid-span where the first critical speed has its peak deflection. In addition, unbalance creation is more likely to occur at the mid-span due to deposits, erosion or corrosion and is more likely to excite the 1st critical speed. Unbalance placement at the coupling (typically the only readily accessible location during the mechanical shop test) does not excite the 1st critical speed significantly and provides minimal information or assessment of the prediction accuracy of this mode.

Typically, URVT on the shop floor is limited to the coupling location. Journal probe vibration levels produced by adding a 40W/N weight to the coupling is less than 0.25 mils (6.35 μm) at MCS. Performing the verification test in the bunker can permit measurement of the mid-span and journal locations to a variety of applied unbalances. Direct measurement of the relative displacements along the rotor allows for closer scrutiny of the predicted shaft and bearing dynamics as they relate to the amount of modal bending and damping at the critical speed. It should be noted that the intent of the test is to determine the vendor's ability to predict the unbalance response behavior of a model containing a shaft, bearings and support structure. The results may not be indicative of the actual behavior of the job machine and relies on accurate modeling of the bunker pedestal dynamics. However, the assurance gained from verification of the predictive method in the bunker should carry over to the job rotordynamic predictions.

Vibration demonstration tests can reveal the presence of instabilities. However, even if no problematic nonsynchronous vibrations are observed, the machine's actual stability could be very small (say less than the API design minimum requirement of log decrement of 0.1) and close to the stability threshold. This state of blindness with respect to the machine's actual stability, as well as the significant uncertainties remaining in current modeling tools for stability prediction [11], can only be mitigated by directly measuring the machine's log decrement through a SVT.

Unlike the URVT, a SVT requires additional hardware in the form on a temporarily, mounted shaker. While this is an added complication, a SVT is often a cost and technically effective alternative to the very expensive Type I test. Such Type 1 testing is unlikely to receive project approval unless it is specified very early on during FEED. SVT can provide a valuable stability assessment of the machine during the other operations/tests that are often typically specified, namely, at-speed balancing, MRT or PTC 10 Type II performance tests.

When done in conjunction with high speed balance or the MRT, an SVT can measure the machine's basic log decrement. Figure 2 shows the results of such testing versus speed for a particular test rig mounted on tilting pad journal bearings [12]. Measurement of the basic log decrement provides evidence as to whether or not the machine's rotor/bearing/support system has the robustness to counteract the destabilizing mechanisms that it will experience in the field. As shown in Figure 2, such basic log decrement testing also provides the opportunity to assess competing tilting pad journal bearing models, a topic of much debate in the industry [11, 12].

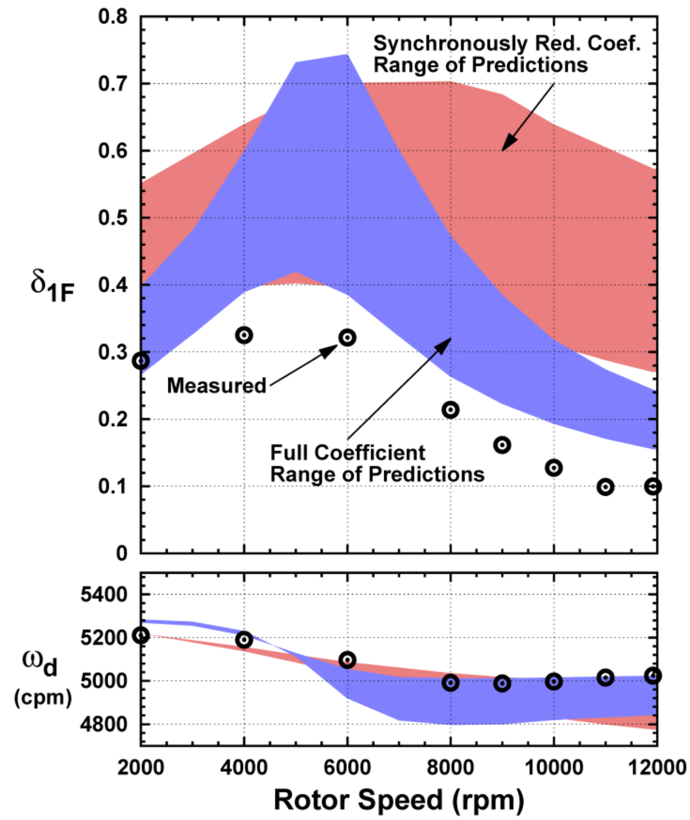

Figure 2) Base stability measurements versus speed

RECOMMENDED PRACTICES OF ROTORDYNAMIC TESTING

This section will focus on the rotordynamic testing of centrifugal compressors due to the complex dynamic behavior of this class of turbomachinery. Concepts developed in this section can be applied to all types of machinery as they all share certain aspects of rotating behavior. In addition, this section will focus on rotordynamics only. The tests listed below, especially the non-verification testing, are also used to prove other aspects of the machine's acceptability. These aspects will not be discussed.

Vibration Demonstration Tests

Mechanical Run Test

Mechanical run test of the rotating equipment should be viewed as a minimal test to determine the rotordynamic acceptability and should be considered for equipment that is designed-for-purpose in contrast to equipment selected from a catalog.

Test Procedure

For most types of equipment, API standards include specific procedures to follow during a mechanical test. Important test factors related to the rotordynamic behavior of the equipment are:

- Test speeds/duration
- Lube oil parameters (temperature, flow rate and viscosity)
- Rotor/support configuration

Test Speeds/Duration

Operation of the compressor during the mechanical test should include a warm up portion where the rotor speed is incremented in 10 minute intervals carefully avoiding exclusions zones of critical speeds and blade natural frequencies. Following the warm up, operation at trip speed is specified for 15 minutes followed by a non-interrupted 4 hour run at maximum continuous speed. A coastdown from trip follows the 4 hour run.

The warm up portion is included to ensure that the case and rotor are given time to thermally expand gradually to avoid creating unintended interferences between the two leading to rub damage. The warm up portion also permits examination of the rotor behavior at increasing speed intervals. Thus, faults can be detected at less energetic stages potentially avoiding rotor/stator damage and project delays. Trip speed is included to ensure that vibration levels (and overall operation) are acceptable at this speed. The 4 hour run portion is used to set the thermal conditions of the system. Bearing temperatures and vibration levels are important factors to watch during the test. Stable levels of each parameter need to be reached during the test. If any parameter shows signs of continual movement (increase or decrease), the test should be extended until stable levels are achieved. If not, the test should be rejected. Following the 4 hour run, a coastdown from trip speed is performed. The coastdown is used to determine the overall and synchronous behavior of the rotor/support system. This data will also be used as the baseline for verification testing, if performed.

Operation at trip speed during the mechanical run test is important for several reasons. First, running to trip speed increases centrifugal forces on the rotor which may relax interference fits and permit the rotor's static shape to change. This may alter the balance state of the rotor. Second, reaching trip speed is necessary for trip testing in the field. It is convenient to reach these speeds during testing to identify any potential problems. Finally, trip speed operation can help identify any critical speeds occurring just above maximum continuous speed that would otherwise not be seen on the test stand.

Additional shutdown/startup transient operation can be added at the beginning of the 4 hour test. When compared against the coastdown following the 4 hour test, thermal transient behavior of the rotor can be examined and any changes to the balance state of the rotor can be identified. This may prove useful in diagnosing Morton's Effect, clearance closure of the radial bearings or fit relaxation due to operation at trip speed. Speed versus time for a typical mechanical test is shown on Figure 3.

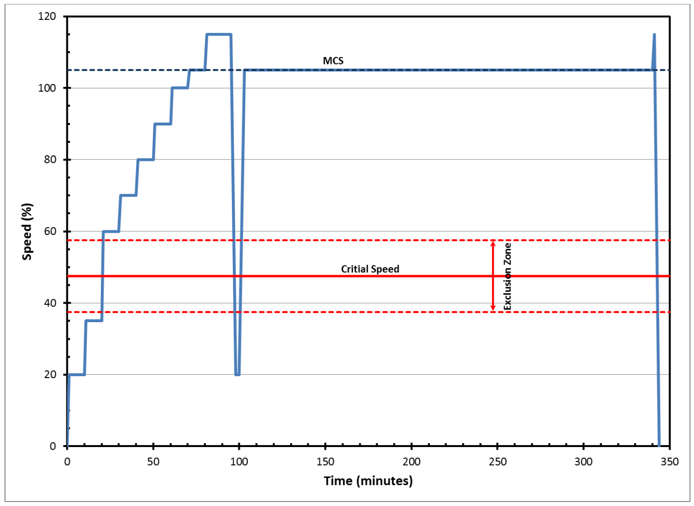

Figure 3) Speed vs. Time for a Typical Mechanical Test of a Centrifugal Compressor

Lube Oil Parameters

Lube oil parameters should mimic that used for the intended application. Bearing flow, lube oil inlet temperature and viscosity should be within the specified operating ranges set by the vendor for field operation. After stabilization during the 4 hour run, lube oil inlet temperature can be varied from the minimum to maximum specified range to examine the effect on vibration levels and bearing operation. Operation at the range limits should be held until steady state conditions are achieved.

Rotor/Support Configuration

The rotor configuration should be as close to the operating condition as possible. The major concern centers around the overhung weight associated with the coupling. Often mechanical testing is done at partial rated power levels. Smaller shop drivers need not have, nor in some cases could accommodate, the larger couplings of the job. It is essential to closely mimic the overhung weight of the job ½ coupling. This may require that a simulator be added to the drive assembly to match the overhung moment. The opposite may be true for vendor's smaller casings where the test coupling's overhung moment is larger than the job coupling. Rotordynamic predictions should be used to determine the impact of using the heavier overhung weights on the 2nd and 3rd critical speeds.

Note: For the cases where rotordynamic predictions are used to study the impact of configuration changes, it is strongly recommended that verification testing be performed. The increased reliance on the rotordynamic model to predict behavior that may not be fully tested necessitates that some verification of the rotordynamic prediction be undertaken.

Test Requirements

Focusing on the rotordynamic behavior, the requirements for the mechanical test involve the overall and non-synchronous vibration levels, the separation margins of identified critical speeds, % change of vibration from MCS to trip speed, rubbing at close clearance locations and bearing behavior in terms of temperature. API is frequently used to set these limits. Overall vibration limit per API 617 [13] is defined as:

$$V_l(mils\ pk-pk) = min\left(\sqrt{\frac{12000}{N}}, 1.0\right)$$

$$V_l(\mu m\ pk-pk) = min\left(25\sqrt{\frac{12000}{N}}, 25\right)$$

Where;

N = maximum continuous speed (rpm)

Additional requirements may include limits on the amplification factor of critical speeds and time-dependent vibration. API 684 [14] provides information on amplification factors, separation margin requirements and factors influencing vibrations beyond speed. Additional requirements as to the thermal related transient maximum behavior may be imposed if the Morton Effect phenomenon is likely.

Test Deliverables

The deliverables of a typical mechanical test are:

- Electronic recording of vibration and static data (i.e. bearing temperatures)
- Overall vibration vs. time & speed (tabular or plot)
- Bode plot - synchronous vibration (mag & phase) vs. speed
- Vibration vs. frequency (at each speed and points during the 4 hour run)
- Statement of acceptability

Several of the deliverables require vibration recording and analysis software/hardware be present during testing. Discussions with the vendor should be held prior to testing to determine what the capabilities of the test facility are and if any additional equipment is needed. Native files from the data collection system, representing the baseline machine behavior, may also prove useful for follow-up diagnostic work or to aide field troubleshooting. In both cases, additional information may be desired that was not examined or plotted during the shop testing.

A bode plot is shown on Figure 4. For the X probe, the first critical speed is easily identified at 5197 rpm with a peak response of 0.72 mils (18.3 μm) 0-pk. The amplification factor can be calculated by multiplying the peak response by 0.7071 (half power point – red dashed line) and locating the speeds associated with that magnitude (green dashed lines). For this example, the amplification factor is $\approx \frac{5197}{(5570-4800)} = 6.7$. This would require a separation margin of $17 * \left(1 - \frac{1}{6.7-1.5}\right) =$ 13.7% of minimum speed [15]. The amplification factor may also be calculated from the slope of the phase angle in radians, θ, at the peak response, N_c. This takes the form: $AF_{phase} = \frac{N_c}{2} * \frac{\Delta\theta}{\Delta N}\Big|_{N_c} \approx \frac{5197}{2} * \frac{4.712}{1900} = 6.4$.

The peak vibration with regards to the API vibration limit is taken in the operating speed range from minimum to MCS. For a range of 7000 to 10000 rpm, our example case would report a maximum overall vibration level of 0.5 mils (12.5 μm) pk-pk. (For this example, the overall vibration is assumed to equal the synchronous vibration.)

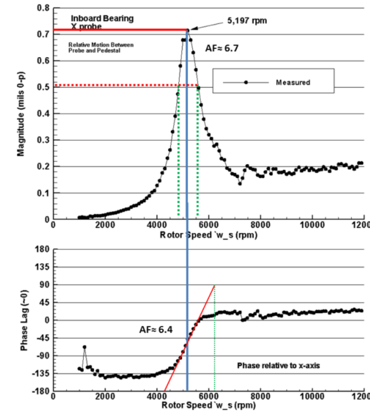

Figure 4) Bode Plot of Inboard Probe

Rotordynamics Modeling

The rotordynamic predictions for the rotor should be available prior to the test. Since the model analysis itself will not be tested, the results are reviewed for problem areas with regards to the analysis and test requirements. Of concern:

- Does the rotor meet sensitivity requirements? Per API 617 [16], an unbalance application of twice the residual (2 * 4W/N) should not produce probe levels above the vibration limit. Sensitive rotors may not meet the vibration limits on test regardless of the quality of the balance correction.
- Are there separation margin problems especially with the 2nd and 3rd critical speeds? To properly model and predict the location of modes above the 1st critical speed, users should verify that the overhung weight for the analysis matches that used for the test. Synchronous vibration at higher speeds should be carefully examined for an indication of critical speed encroachment. Figure 5 shows the impact of a potential mode just above 12,000 rpm on vibration magnitude and phase (circled on plot).

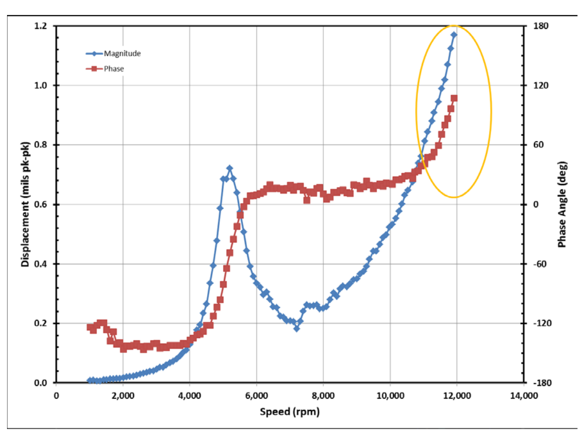

Figure 5) Impact on Synchronous Vibration of Critical Speed Located Above Operating Range

Test Knowledge/Verification

The knowledge gained from the mechanical test is basically a pass/fail determination of the rotordynamic behavior. The vibration levels and separation margins (typically only concerning the 1st critical speed) are checked to show that they meet required levels. In addition, the presence of rubbing at close clearance locations (determined during the posttest inspection) is used as a pass/fail performance of the test. The presence of non-synchronous vibration components may also be an indicator of other mechanical faults (i.e., instability, mechanical looseness).

The rotordynamic predictions are not truly verified in a mechanical test. In fact, the shaft vibration normally associated with the quality of the rotordynamic behavior is more closely related to the quality of the rotor balance performed. Additional checks of the prediction accuracy can be made by incorporating other test requirements not required by API 617 7th Edition:

- Limit the discrepancy between predicted vs. measured critical speed location
    - 1st critical speed is likely the only mode to be identified.
    - Provides a check of the shaft bending and support stiffness (stiffness is the primary factor in critical speed location).
- Limit the discrepancy between predicted vs. measured amplification factor (AF)
    - Amplification factor is an indicator of the modal damping present and may vary end-to-end and from X to Y probe. Care should be taken to compare the same probe from predicted to measured.
    - The AF can vary across measurement locations due to runout and residual excitations in the shaft not fully taken into account in the predictions.

Beyond what is measured during the test, the rotordynamic analysis is relied upon to predict the internal deflections of the rotor (marginally verified by the inspection for rubbing), sensitivity to unbalance (not checked) and stability (not checked).

String Test

With regards to rotordynamics, a string test is needed to accurately determine behavior for the following configurations:

1. A rigid coupling is used in the train – Individual body dynamics are affected by rigidly coupling them together. The rotordynamic model and test configuration should be performed with both bodies coupled.
2. There exists a long or heavy coupling spacer in the train – In this case, the dynamics of the coupling are of concern. To model the boundary conditions at the coupling hubs, the coupling should be tested installed in the train.

Test Procedure

A procedure similar to that of the Mechanical Test as described above is sufficient. For configuration #1, the full four test should be run. For configuration #2, an abbreviated test can be used if the individual bodies were already mechanically run tested. (Unlike #1, individual body dynamics are not altered when tested coupled with a long or heavy spacer tube versus using a moment simulator of equal overhung moment.)

Test Requirements

For configuration #1, the requirements should be identical to the Mechanical Test described above. For configuration #2, additional requirements concerning the location of the coupling natural frequency should be included incorporating vibration measurements made at either end of the coupling. Peak responses at both of these locations could be an indication of a coupling natural frequency interference.

Test Deliverables

No difference from the Mechanical Test.

Rotordynamic Modeling

In either case, the rotordynamic model used for the behavior predictions should include the coupling and train bodies on either side. For example, a train comprising a turbine rigidly coupled to a generator, should be modeled in its entirety. Similarly, a power turbine driving a reinjection compressor coupled with a long spacer should all be modeled as three separate rotors (PT, spacer and compressor) coupled together by the flexible elements modeled as shaft elements or lateral spring elements. Figure 6 presents the rotordynamic for a rigidly coupled steam turbine / generator train.

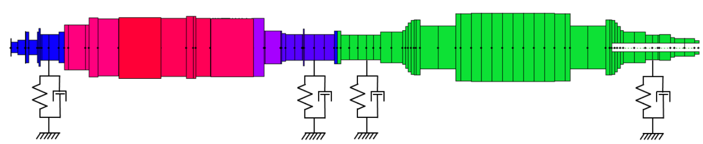

Figure 6) Rotordynamic Model for a Steam Turbine / Generator Train

Test Knowledge/Verification

More knowledge of the coupled train behavior is obtained with the string test. However, the same statements regarding design verification can be made as with the Mechanical Test.

Full Load / Full Pressure

Due to the relative cost of performing a full load / full pressure (FLFP) test, this option is typically limited to centrifugal compressors meeting all of the following criteria:

- Loss of production produces an unacceptable financial impact
- High pressure applications
- Limited vendor experience with the design or application
- Rotordynamic analysis reveals increased risks or concerns with the rotor stability

Test Procedure

For the FLFP test, the procedure should be developed in concert with the rotordynamic analysis. The FLFP test is intended to study the stability of the centrifugal pressure under similar conditions to the field. By applying load and pressure, destabilizing forces of the seals and impeller/shroud interactions are introduced including seal clearance changes due to internal deflections as a result of reaching full pressure. Operating points during the FLFP test should be determined, in part, by the rotordynamic analysis and reflect operating points of minimum stability. These could represent operation at MCS near surge (highest differential pressure) or, in some cases, partial speed towards stonewall. Since this test as normally carried out is a "pass/fail" (e.g. the rotor is stable or not), test conditions should match as close as possible to the field conditions. The parameters of importance include gas MW, power, suction and discharge pressure and temperature, speed and mass flow. If the exact gas composition cannot be tested, some of these parameters will have to be compromised. The rotordynamic model should be used to determine an appropriate combination of factors to produce the maximum instability drivers or minimum log decrement. Miranda and de Noronha [17] propose modifications to the FLFP ASME Type I test to better assess the stability of centrifugal compression equipment. The modifications were intended to create conditions to submit the rotor to instability mechanisms as near as possible to the design conditions rather than reproduce similarity for performance evaluation. The conditions were developed with the aid of the rotordynamic stability predictions.

As with the Mechanical Test, the FLFP test should consist of a warm up portion where the speed is increased gradually to permit stabilization and examination of the behavior at lower speeds. This is followed by an extended run at MCS to ensure thermal equilibrium of the entire machine is achieved. During this test phase, it is recommended that the operating curve at MCS is explored from the surge control line to the end of curve (stonewall.) This operation may include four to five operating points and may include other speeds as highlighted by the rotordynamic analysis. (Note: Other factors may dictate operation at other points as required, i.e. defining the surge line vs. speed, rated point defined at partial speed.)

Factors such as lube oil conditions and rotor assembly are expected to meet the field configuration and specified operating ranges. When practical, the lube oil operating range should be explored during the FLFP test. Lube oil inlet conditions impact the dynamic behavior of bearings. As a critical factor in determining the rotordynamic behavior, it is important to vary these factors over the allowable ranges during testing.

Test Requirements

Test requirements for the FLFP test are defined by agreement between the vendor and purchaser prior to the test and should be done at the contract stage. Holding the overall vibration limit to the level specified for the mechanical run test is impractical due to the additional rotor forces present during the FLFP test. These include aerodynamic forces of the impellers, stator-rotor interactions, seals forces and power transmission forces. However, raising the limit to the vendor recommended trip setting does not leave margin for off-design operation in the field nor deterioration of the balance state from erosion or deposits. An agreed level should take into account both factors and fall somewhere in between.

Other requirements for the rotor and case vibration may include:

- Components of non-synchronous vibration to be less than 20% of the vibration limit or 0.2 mils (5 µm) p-p, whichever is less
- No stall related vibration components
- No instability related vibration (associated with re-excitation of the 1st natural frequency)
- Maximum housing vibration less than 0.1 in/sec (2.5 mm/sec)
- Limitations regarding thermal instability (Morton Effect) vibrations

Test Deliverables

Deliverables are similar to the mechanical run test and should include data for all purchased components tested (as with the string test.) Increased emphasis is placed on the FFT plots of shaft vibration during the test as this is the best indicator of instability, stall, whirl and other phenomena that produce non-synchronous vibrations. Performance data should be recorded during the test to confirm the input used to predict the seal and impeller dynamic behavior and aide further stability analysis if needed.

Rotordynamic Modeling

When FLFP testing is selected, the rotordynamic model should be expanded as necessary to conform to the Level II stability analysis requirements of API 617 [18]. Given the cost, effort and reasons to perform FLFP testing, a Level I stability analysis is insufficient to predict the behavior accurately. The Level II model will reflect changes in the stability level to MW, gas pressures and temperatures, seal clearances and rotor speed to the best of the vendor's or purchaser's analytic capabilities. The Level I model uses an empirical relationship that either estimates these effects or doesn't take them into account at all when calculating the destabilizing forces present in the machine.

Test Knowledge / Verification

The FLFP test will reveal the presence of instability, stall or whirl for a prescribed set of operating conditions for the specific machine's as-built conditions. The test is pass/fail as no measurement of the stability level or margin is included in the test as described. Rotor stability at different gas compositions, other clearances within the tolerance range or other operating points is determined by rotordynamic predictions. The ability to operate successfully at these alternate points, which cannot be tested under all combinations, depends on the stability margin (not measured by FLFP test) of the machine.

Design Verification Testing

Such verification testing should be considered for the following types of equipment:

- Special purpose (as defined by API)
- Loss of production produces an unacceptable financial impact
- Services or applications with a history of bad actors (vibration related)
- Critical service (as defined by the user)

In addition, verification testing of either the unbalance response or stability should be considered when the rotordynamic analysis demonstrates concerns or higher risks associated with the application. The verification testing can be used to mitigate those risks when applied correctly.

Unbalance Response Verification Test (URVT)

Test Procedure

The URVT test is basically a comparison of measured versus predicted vibration levels for the application of a known unbalance. The test is routinely performed following the four hour mechanical run test. For compressors, the verification weight is applied to the coupling flange. This is the only practical location available. For other machinery (e.g., steam turbines, overhung single stage compressors), other locations may be available. Steam turbines may have field accessible balance planes and overhung compressors an impeller checknut with a balance weight placement provision. The measured response of the machine with the verification weight is compared against the analytical prediction using the same weight and location. While this is not a complete check of the analytic unbalance response accuracy, as it only compares the model's prediction at the probe locations for one weight placement, it is an important first step in ensuring the accuracy of the model.

It was recognized early on that an important aspect of the URVT test was to compensate for the residual unbalance in the machine. This residual unbalance creates the synchronous vibration witnessed during the mechanical test run and is present before and after the verification weight is applied to the rotor. The residual unbalance left in a rotor after balancing (either low or at-speed balancing) is uncharacterized and, therefore, cannot be modeled. Thus, the analytical model will have only the verification weight as an excitation source for the response.

The initial attempt by API 617 6th Edition to compensate for the residual unbalance was to apply a significant verification weight to the coupling to raise the response to the vibration limit. The implication was that the majority of the response would be due to the verification weight placement. This had two important drawbacks (besides being analytically incorrect): First, the amount of unbalance weight at the coupling needed to raise the response to the vibration limit could reach unsafe levels. Coupling flanges are not designed with the intent of adding unbalance weight. Large rotating forces applied to the coupling had a chance of failing the flange or, more probably, failing the mechanism used to hold the weight in place. Second, the method relied on larger vibration limits <2.0 mils (50 μm) p-p versus the current <1.0 mil (25 μm) p-p and good balance correction practices to limit synchronous response on the test stand below 0.5 mil (12.5 μm) p-p. For this situation, 75% of the response would be attributable to the verification weight. At the time, this was better than no test at all.

In 1997, Nicholas et al. [19] defined an improved procedure to better correlate test stand vibrations to the analytic predictions. This methodology was subsequently adopted by API 617 7th Edition and is explained in API 684 [20]. Their method took advantage of vibration diagnostic equipment that permitted vector subtraction of recorded databases. The procedure can be summarized as:

1. Record the probe synchronous readings during coastdown from trip speed following the four hour mechanical run test – This represents the baseline vibration of the rotor
2. Add the verification weight to the rotor – The method is general enough to accommodate weight placement anywhere on the rotor
3. Bring the rotor back to MCS and achieve steady state conditions (i.e. constant bearing temperature, vibration magnitude and phase) – Attempt to reproduce the operating condition of the machine at the conclusion of the four hour test in step #1. The sampling frequency and speed increment should be identical to that used in Step #1.
4. Record the probe synchronous readings during coastdown from trip speed following Step #3 – This represents the combined vibration of the rotor (verification weight and residual unbalance)
5. Vectorially subtract the synchronous vibration database taken in Step #1 from that recorded in Step #4 – The resulting data represents the response due to the verification weight placement
6. Compare the resultant data in Step #5 to the analytical predictions (incorporating the range of bearing clearances and oil inlet temperatures) – This is the test for accuracy of the unbalance response predictions

Graphically (and using vector math) the method can be described as:

- A baseline response reading, $\overrightarrow{V_1}$, (known) is taken at one speed and one probe and is attributed to the residual unbalance, $\overrightarrow{U_r}$, in the rotor (unknown).
- The subsequent response, $\overrightarrow{V_2}$, (known) taken after the addition of the verification weight, $\overrightarrow{U_v}$, (known) to the rotor is recorded. The net unbalance present in the rotor at this stage can be described as $\overrightarrow{U_r + U_v}$.

- Performing the vector subtraction of the 1st reading from the second yields the response, $\overrightarrow{V_s} = \overrightarrow{V_2} - \overrightarrow{V_1}$. The accompanying vector math with the unbalance state of the rotor, $\overrightarrow{U_2} - \overrightarrow{U_1} = \overrightarrow{U_r + U_v} - \overrightarrow{U_r} = \overrightarrow{U_v}$, demonstrates that the resultant response, $\overrightarrow{V_s}$, is due only to the verification weight added to the rotor.

This is shown on Figure 7.

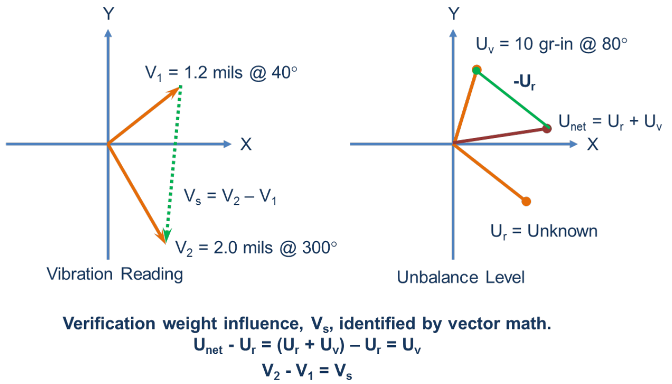

Figure 7) Determination of Rotor Response due to Verification Weight

All other aspects of the URVT should conform to the requirements of the mechanical run test in terms of oil inlet temperatures, speeds and rotor support.

Operating Speed Balance Bunker Option

An option exists of where to perform the URVT. Operating speed balance bunkers (also referred to as at-speed or high speed balance bunkers) can be used and has advantages over the test floor. The advantages are associated with access to the rotor that running in the bunker presents. The operating speed balance bunker permits alternate locations for the verification weight (i.e. mid-span, quarter-span), use of multiple weights and additional shaft vibration readings. The accessibility of the bunker permits additional verification of the rotordynamic predictions, i.e. mid-span response relative to journal locations for mid-span and quarter-span weight placements.

There is one important complexity when considering performing the URVT test in the balance bunker; the rotor support stiffness in the bunker will be different than the actual machine. Often the bunker bearings, while similar, are not identical to the job bearings. Different clearances, L/D ratio and pad arc length and the existence of a levitation pocket will alter the stiffness and damping coefficients. The support stiffness of the pedestals will also be different than the job machine. Bunker pedestals are intended to be flexible to permit adequate velocity measurements used in most bunkers' balance software. The stiffness is a function of frequency and inaccurate representation will produce significant differences between predictions and measurements. This can lead to the erroneous conclusion that the model of the job compressor (not using bunker bearings and pedestal stiffness) also suffers from the same inaccuracy. Table 1 presents the advantages of each option for URVT.

URVT Vehicle	
Shop Floor (Mechanical Run Test)	Operating Speed Balance Bunker
+ Test performed with the job configuration	+ Increased flexibility for weight placement
	+ Additional measurement points possible
	+ Additional verification of the rotordynamic analysis

Table 1) Comparing Advantages of Each Test Vehicle

- As required by API 617 7th Edition, predictions for the close clearance locations and unbalance sensitivity should be adjusted as dictated by the verification testing – For example, should the verification test show the rotordynamic analysis is under predicting by a factor of two, all response levels calculated for the lateral analysis should then by multiplied by two and sensitivity and close clearance requirements rechecked for compliance.

Depending on the vehicle, additional checks regarding critical speed shape, amplification factor and modal response can be performed.

Test Deliverables

The deliverables for the URVT should include:

- Rotordynamic analytical response for the setup, weight placement and magnitude and measurement locations used in the test. The response should be carried out over the specified range of clearance and oil inlet temperature to produce the largest variations in radial bearing dynamic coefficients. The analysis should identify the critical speed location (response peaks) and agreed upon unbalance response versus the variation in bearing dynamic coefficients for the verification weight.
- Measured synchronous vibration data should be plotted for the following conditions:
    - Coastdown following the 4 hour test plotted for the four proximity probe locations – This represents the baseline data for the machine (vibration as a result of the residual unbalance and forces in the rotor.)
    - Coastdown following placement of the verification weight and warmup at MCS (if needed) – This represents the vibration data for the residual + verification weight.
- Vector subtraction of baseline data from the verification + residual data – This will be the rotor's measured response to the verification weight.
- Comparison of the measured peak response speed (typically the 1st critical speed) to the predicted range – Discrepancies outside the permitted range (5% for API 617) should be addressed by correcting the rotor model (i.e., addressing the values used for support

stiffness, bearing stiffness and/or shaft bending stiffness).

- Comparison of the measured response magnitude versus the predicted value at the required speed(s) – If predicted value is less than the measured value, then the corrected analytical predictions based on the factor of measured/predicted is supplied. Close clearance and sensitivity requirements should be reviewed for compliance.
- Other analytical vs. measured requirements as specified in the test procedure.

Two examples of the URVT are provided to illustrate the procedure. Both are taken from ref. [19]. The first example is an 8 stage compressor. Figure 8 plots the predicted response for a verification weight placed at the coupling location (40W/N magnitude) of the compressor for one drive end probe.

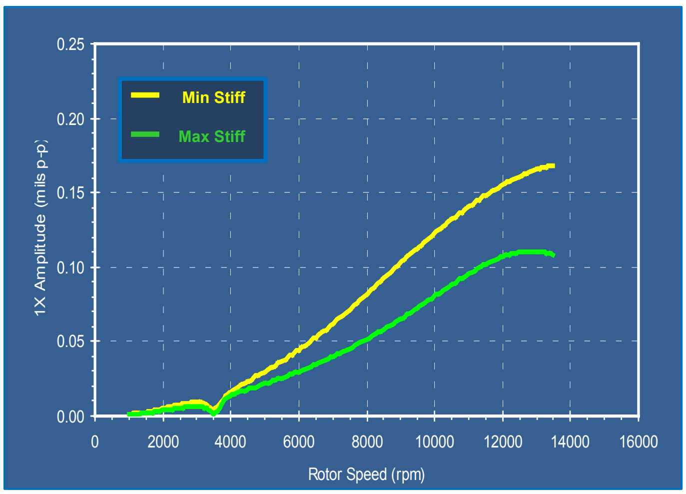

Figure 8) Response Plot of Coupling Verification Weight for Min and Max Bearing Stiffness

The corresponding probe response for the baseline and verification weight added runs are plotted on Figure 9). A drive-end probe was selected since the verification weight (identical to that used in the analysis) was added to the coupling and the drive-end would have the largest response. The shop floor during the mechanical test run was used for the verification test.

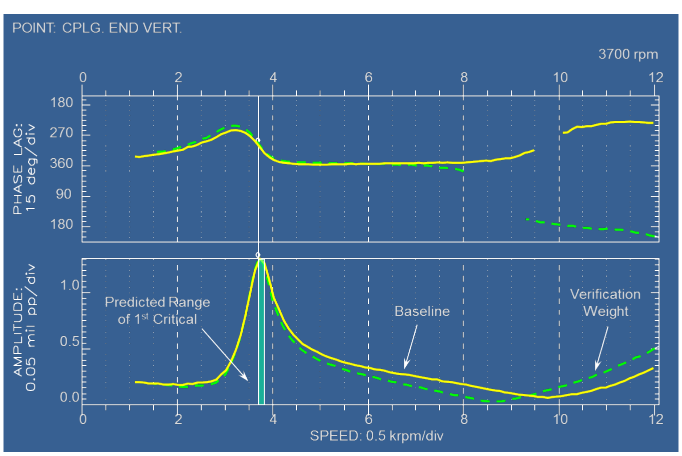

Figure 9) Measured Probe Synchronous Response for 8 Stage Compressor

Notice that measured probe response with the verification weight does not resemble the predicted response, Fig. 8. This is due to the residual unbalance response in the rotor. Performing the vector subtraction of the two data files isolates the response due solely to the verification weight. This is compared against the predicted response on Fig 10.

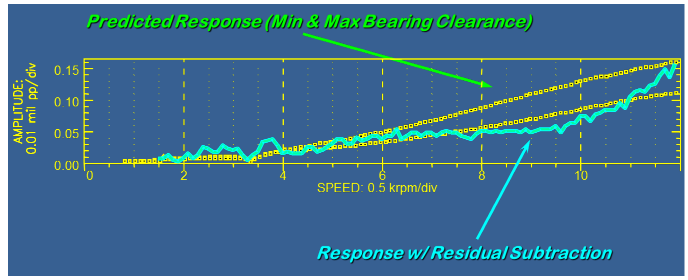

Figure 10) 8 Stage Compressor – Measured vs. Predicted Verification Response

Figure 10 illustrates the need to remove the effect of residual unbalance due to the relative insensitivity of the rotor to coupling unbalance. Even at low amplitudes, ≈0.00015", the measured response falls within the maximum levels calculated from the rotordynamic response especially at MCS where the requirement was enforced. In addition, the peak response (1st critical speed) falls within the range predicted by the analysis, Figure 8. For this example, no correction of the lateral analysis was needed.

The measured response of a three stage compressor's baseline and verification run is shown on Figure 11. In the second example, the measured response to the coupling verification weight is shown to exceed the predicted value at MCS by 25%, Figure 12. The measured peak response of the 1st critical speed does occur within the range predicted by the analysis, Figure 11. For this case, the lateral analysis unbalance response magnitudes at the close clearance locations and for 8W/N unbalance should be increased by a factor of 1.25. The rub limits and sensitivity requirements should be rechecked for compliance.

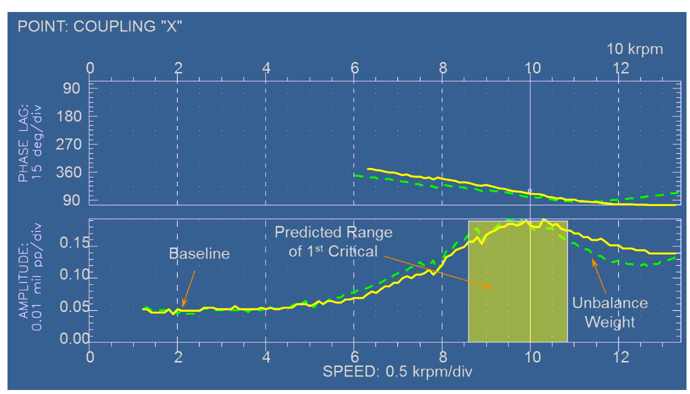

Figure 11) Measured Probe Response for 3 Stage Compressor

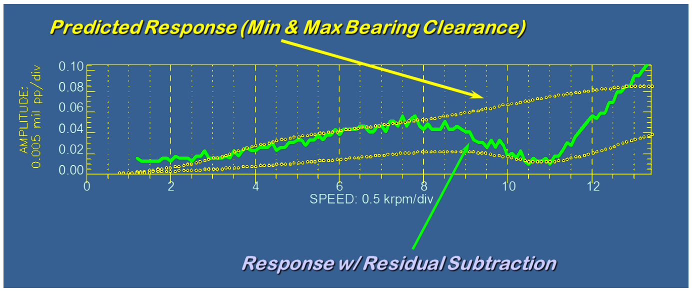

Figure 12) 3 Stage Compressor – Measured vs. Predicted Verification Response

Rotordynamic Modeling

As previously mentioned, the rotordynamic model for URVT needs to match the configuration of the test setup. If a balance bunker is used, the support stiffness of the model should include the dynamic effects of the bunker bearings and pedestals. The rotor model should also reflect the ½ weight of the test coupling (or bunker drive assembly) in its correct c.g. location. The predictions should cover the minimum to maximum bearing stiffness (as required by API 617 7th Edition) based on the specified ranges for the radial bearing clearances and oil inlet temperatures. Finally, the analysis should specify the predicted ranges for the 1st critical speed and verification response magnitudes (additional information as necessary to determine compliance with other requirements.)

Summarizing the aspects of the rotordynamic modeling critical to this testing:

- Support stiffness to represent the test vehicle
    - Design configuration for URVT performed during the mechanical run test on the shop floor
    - Reflecting the bearings and pedestals of the operating speed balance bunker
- Verification weight placement and magnitude
- Rotor vibration at the probes or other locations as used during the URVT
    - Include support motion if probes are mounted on bunker pedestals

Test Knowledge / Verification

The verification test provides a determination of the accuracy of the rotordynamic predictions in a quantitative fashion. Beyond the pass/fail observational nature of the vibration demonstration tests, the URVT demonstrates the predictive versus measured response discrepancy to a known weight placed on the rotor. Performed correctly, the operating speed bunker can provide additional accuracy assurances regarding the modal response (i.e., mid-span versus journal response ratio) and rotor sensitivity to locations other than the coupling[2].

Stability Verification Testing (SVT)

SVT Methodology

Currently, there are no industry standards in place to guide the end-user and OEM as to what methodologies and practices should be used to accurately perform these SVTs. This tutorial will attempt to provide some guidance in this area, specifically, in the three fundamental elements of the testing process: nonsynchronous excitation design, measurement process, and damping ratio estimation. Since this last element dictates much of the methodology requirements for the other two, our discussion will begin with some guidelines regarding damping ratio estimation.

SVT Damping Ratio Estimation

It should be noted as to why the term "damping ratio estimation" has been chosen instead of "log decrement measurement." This choice is meant to emphasize several aspects:

1. The fundamental modal parameter of interest is the damping ratio.
2. Estimation of the damping ratio involves a post-processing analysis of the measurement data. The resultant damping ratio is not measured directly, it is only estimated.
3. Logarithmic decrement is its own estimation technique for damping ratio.

Damping ratio estimation is a subset of modal parameter identification. Since this identification occurs while the machine is operating, such testing is often called operational modal analysis which is a subset of the more general field of system identification. Identification of other modal parameters, such as natural frequency and mode shapes, has historically received much greater attention. This is likely due to their importance with respect to resonance problems, a predominant concern in every field whereas stability is not as much so.

As mentioned earlier, the estimation involves a post-processing analysis of the measurement data. While there are a multitude of estimation techniques available and continually emerging, each technique basically involves a curve fitting of the data. Some are very simple and deal with just a single response channel, while others curve-fit multiple channels' data at once. These techniques are being developed by various communities, such as controls and speech processing, with each technique designed to utilize time or frequency domain data.

It is beyond the scope of this tutorial to discuss all the aspects of damping ratio estimation. However, it is important that machinery engineers understand some of the peculiar challenges and the recommended practices. When conducting a SVT on a turbomachine, the estimation process faces several challenges when dealing with the measurement data:

- The data is typically from only a few vibration response locations.

[2] Note: For some well balanced and well damped rotors, the 1st critical speed may not be identifiable on the test stand as a clear peak in the vibration data and will not be appreciably excited by a verification weight placed at the coupling. Performing the test in the balance bunker enables mid-span weight placement and greater excitation of the 1st critical speed for between bearing machines. All options regarding the use of the balance bunker for URVT (e.g., mid-span probes, alternate unbalance weight locations) should be specified and agreed during contract development with the vendor.

- The data contains additional response from the significant presence of immeasurable, internal excitations within the machine such as unbalance, misalignment, aerodynamics, etc.
- The data likely contains contribution from multiple modes, not just the mode of interest.
- The data is from a system whose dynamic characteristics are not like those for static, nonrotating structures.

Each of the above challenges influences what techniques can be used for accurate stability estimation. Some techniques rely on a large number of sensors, meaning they are not appropriate here because response measurements are typically only available at a few restricted locations on a machine. Other techniques assume the system being tested is a static structure whose dynamic properties are symmetric or self-adjoint. Techniques relying on such an assumption are not appropriate for a turbomachine where fluids within the machine create non-symmetric cross-coupling in the system [21,22].

Perhaps the biggest estimation challenge is the fact that the measurements from a rotor's SVT often consists of significant contributions from several modes, more than just solely the mode of interest. Many structures possess very close natural frequencies which can be difficult to identify through the estimation process. Sharing this high modal density characteristic, rotor systems possess pairs of sister modes, one forward whirling and one backward that are often in close proximity and share similar modal characteristics.

The easiest way to observe the close proximity of these sister mode pairs is through a Campbell diagram. Figure 13 shows the Campbell diagram for a multistage centrifugal compressor. Typical for between-bearing machines, the first mode sisters remain very close in frequency as speed increases, where the backward mode's frequency is slightly less than the forward mode. The second mode sisters become more and more separated with speed, indicating the stronger influence of gyroscopics and support anisotropy for these sisters. The influence of gyroscopics and support anisotropy is highly dependent on the overall rotor system characteristics. For example, unlike the between-bearing compressor, the first sister modes for an overhung compressor will be very sensitive to speed due to the overhung impeller's strong gyroscopic moment.

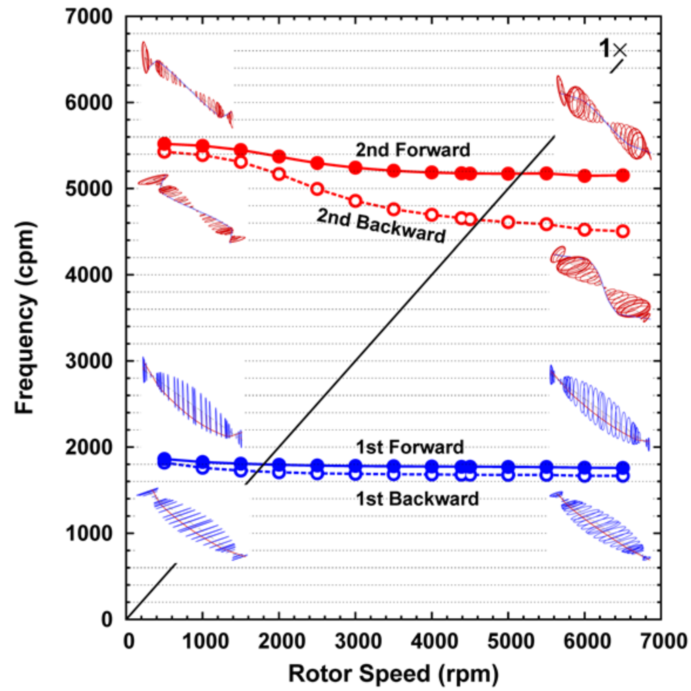

Figure 13) Example Campbell Diagram

To date, many previous SVT investigators have relied on the assumption that only the mode of interest, typically the first forward mode, is participating in the response. It is easy to understand why this assumption has historically been applied by the industry. First, the measurements often only show one peak in the frequency response functions. Second, the nonsynchronous excitation was applied in the forward circular direction with the intent of only exciting the first forward whirling mode. And finally, single degree of freedom (SDOF) damping estimation techniques, such as amplification factor, phase slope or mechanical log decrement, involve convenient and simple formulas that are familiar to every vibration specialist.

Unfortunately, even when forward circular excitation is applied and only one peak is present in the measurements, the sister backward mode can be excited, reducing the accuracy of the SDOF damping estimators. Figure 14 illustrates this behavior for a simulated rotor system where the individual modes' response can be distinguished. Even though the forward circular excitation is being applied and only a single response peak is measured, the backward mode is excited. Because two modes are responding, instead of just one, the peak is distorted and, in this case, broadened, from that due to just a single mode.

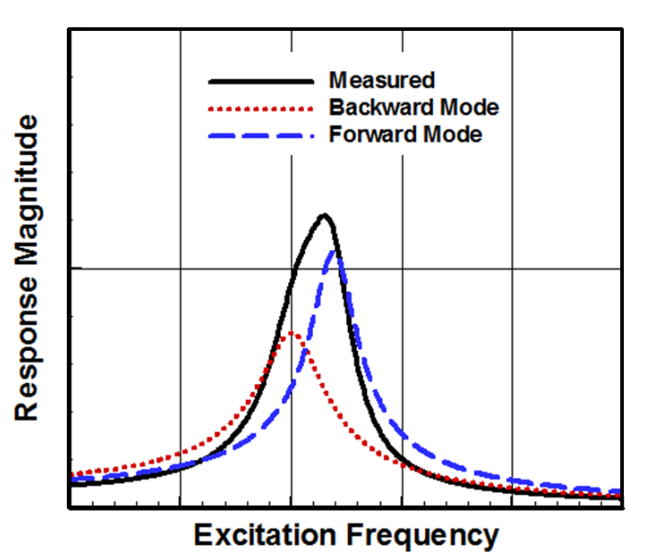

Figure 14) Modal Participation from Nonsynchronous Forward, Circular Excitation [23]

While the methodologies associated with SVT's damping estimation are still evolving, there are several ones, long considered good practice in the system identification community, that have emerged as recommended practices:

- Multiple degree of freedom (MDOF) techniques provide the most reliably accurate estimates of damping ratio without relying on any assumptions about a particular machine's rotordynamic characteristics. To date, there are two MDOF techniques which have demonstrated reliable accuracy for stability verification testing: multiple output backward autoregression for time domain response measurements, and the prediction error method for frequency response function measurements.

- To resolve the closely spaced sister modes of rotors, MDOF techniques should be applied to multiple output (MO) data sets consisting of response from multiple locations along the rotor.

- Any candidate estimation technique should be validated using simulated excitations and measurements that follow the methodology planned for shop testing but are taken from a simulated rotor system where the eigenvalues and stability are known. This validation should assess the potential for asymmetry in the bearing system and various levels of stability. While a simple rotor system can be used for such validation, the most preferable choice is to use the model of the machine to be tested.

Recently, some OEMs have been relying on ambient excitations in conjunction with output-only identification techniques for stability verification testing [37]. This approach, known as operational modal analysis (OMA), has been applied in some communities where either measurement or application of an excitation mechanism is difficult or impossible. It is very attractive for a machine's SVT because no excitation device needs to be installed.

There are a variety of powerful output-only identification techniques available for OMA. Some rely on relatively long recordings of vibration data for their estimation, while others, such as MOBAR, were developed to handle vibration data over a short duration.

While OMA is a promising approach, its strengths and weaknesses with respect to stability verification testing have yet to be fully identified. The amount of ambient excitation present and the "color" of its frequency content, e.g., white, pink, etc., can vary significantly within each machine and test stand operating conditions (vacuum, FLFP, etc.). These ambient excitation aspects, combined with variabilities in the rotordynamic nature of different machines (wide variations in stability and proximity of sister modes), all affect the signal-to-noise ratio (SNR) and, thus, the reliability of damping ratio estimation accuracy of the mode of interest, 1F.

Because of the variability and evolving nature of OEMs' adoption of estimation techniques, some end-user companies will independently verify the SVT measurements using their own trusted methodologies. This practice is directly analogous to end-users' independent verification of calculations for rotordynamic predictions and aerodynamic performance testing.

SVT Nonsynchronous Excitation

Our SVT methodology discussion will next focus on aspects regarding the nonsynchronous excitation that is necessary to measure a mode's stability. The design of both the excitation device itself, and the signals that it applies to the machine, are important elements of the SVT's success.

Unless the subject machine is supported by radial active magnetic bearings, the nonsynchronous excitation must originate by some temporary means. The earliest forms of excitation were from impacts made directly on the rotor [24,25,26]. More recently, shakers were mounted externally on machines' bearing housings [4,27] with an example shown in Figure 15.

Typically installed on the non-drive, outboard of the machine, electromagnetic shakers have emerged as the most popular device for nonsynchronous excitation. Two methodologies have been used to temporarily attach the shaker's laminated rotor assembly, a bolted extension has been used by several investigations [28,29,30] while others have mounted a tapered sleeve [9,31]. Figures 16 and 17 illustrate examples of these two attachment designs.

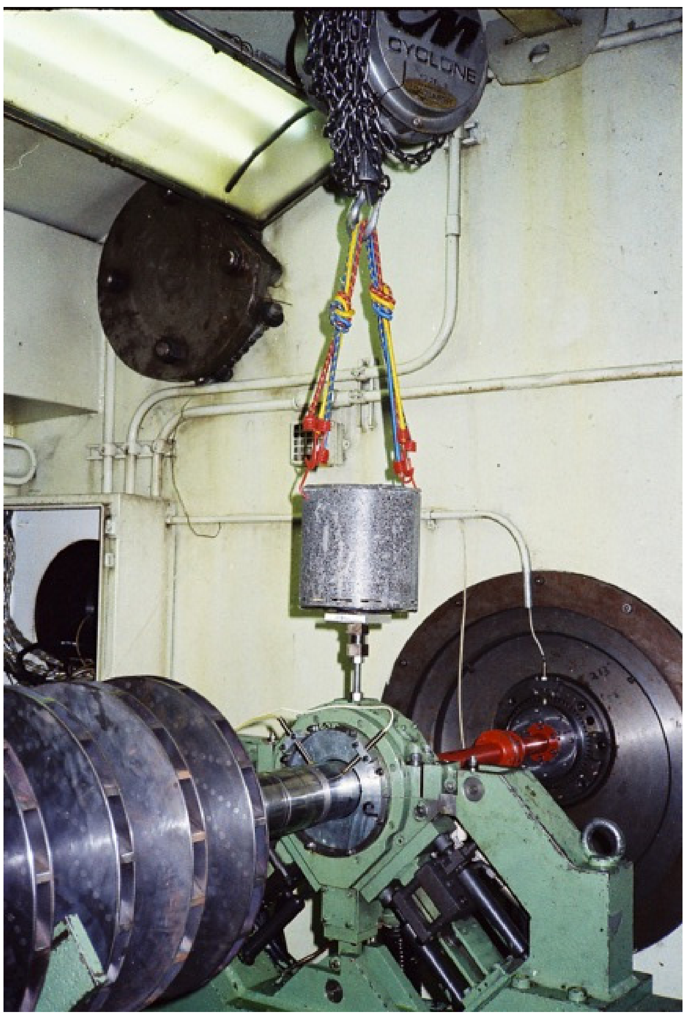

Figure 15) Shaker Mounted on Bearing Pedestal in Balance Bunker [4]

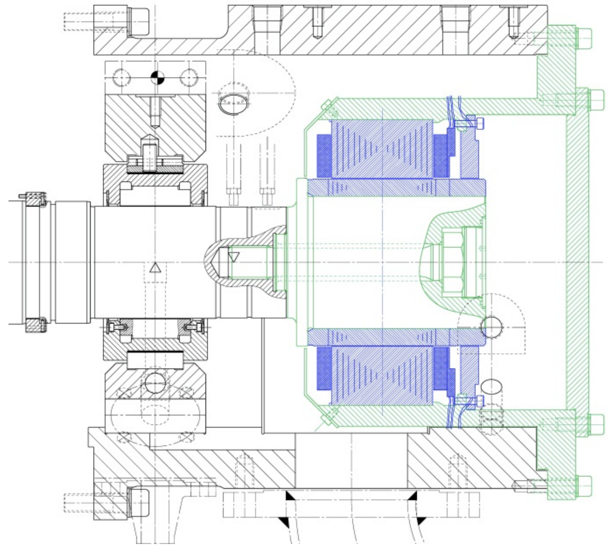
Figure 16) Shaker with Bolted Shaft Extension [30]

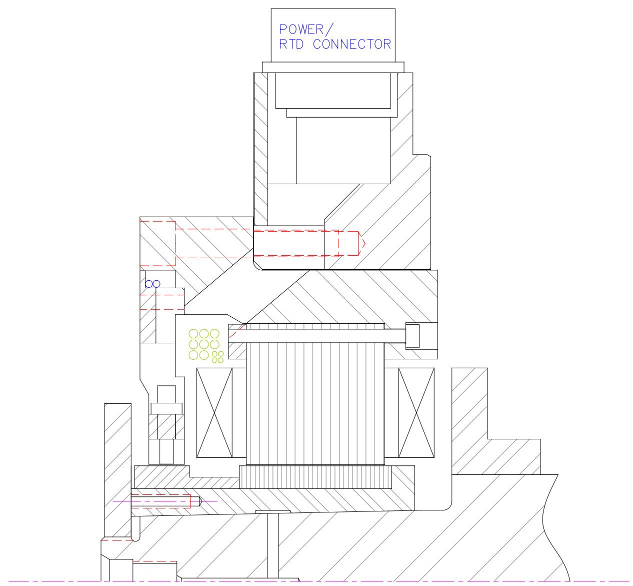

Figure 17) Shaker with Sleeve Mounted on Tapered Shaft End [9]

There are several key objectives that are important to the successful design of the excitation device:

- Design for ease of installation and removal during shop testing,
- Minimize alteration of the machine's balance state and rotordynamics when the device is installed,
- Provide sufficient force capacity and bandwidth to excite the mode of interest when the machine is operating at the test conditions.

When utilizing an electromagnetic shaker, its laminated sleeve or shaft extension can add sufficient weight and inertia to significantly alter the machine's baseline dynamics. Rotordynamic calculations should be conducted during the shaker's design process to examine the impact on the machine's rotordynamics, in particular, to ensure minimal impact on the mode of interest's stability and frequency. If it is undesirable to alter the test speeds, one may have to accept lower than desired separation margins for other critical speeds. While it is vital to protect the machine from damage during the testing, it should also be recognized by all parties that typical vibration acceptance criteria are not applicable during a SVT. Regardless of the exciter's impact on the rotordynamics, it is a good practice to design the shaker's rotor assembly with a balance correction plane as well as check balance the rotor assembly with the shaker installed prior to testing. Low synchronous vibrations help improve the quality of the SVT measurement data and the resulting damping estimation.

Finally, the frequency content and direction of excitation must be determined. Both must provide the type of measurement data needed for the damping estimation technique originally chosen. If time domain estimation techniques are employed, the best signal-to-noise ratio (SNR) can be obtained through a blocking test. A type of tuned-sinusoidal method [32], blocking testing effectively tries to isolate a mode by exciting at its natural frequency and in its predominant direction. Several investigators have successfully applied this excitation method [9,27,33] for stability verification testing of rotor systems. Direction of the blocking excitation, forward or backward whirling or along one axis, can be chosen to best excite the mode of interest.

If a frequency domain estimation technique is chosen, the required measurement data consists of frequency response functions (FRF) across the frequency range containing the mode of interest. Calculated using correlation functions that consider noise in the system, a measured FRF has units of response (displacement, velocity or acceleration) divided by force. The frequency range is spanned by the excitation device using stepped sine, chirp or pseudorandom signals, with the final choice determined by the desired SNR and testing time. Stepped sine is generally considered to have the best SNR, while other frequency signals can provide faster measurement times.

The direction of the applied excitation must be considered when the FRFs are being calculated during the measurement process. When exciting in only one direction, such as along the machine's horizontal splitline or along one proximity probe's axis, the FRFs are easily calculated according to single input, multiple output (SIMO) procedures. Contrary to popular thinking, such SIMO testing along only a single axis is sufficient to excite the first forward whirling mode and can provide accurate damping ratio estimates when used with an appropriate MDOF frequency domain technique.

Figure 18 presents the FRFs from a SIMO test conducted on a simulated rotor system with known stability. In this case, horizontal excitation is applied at the inboard bearing and four FRFs are obtained, one for each of the four bearing probes. Noise has been added to the measurements to simulate real world conditions. Table 2 compares the actual stability levels of the first sister modes with that obtained using the SIMO measurement data and a MDOF estimation technique. For either horizontal (SI_xMO) or vertical (SI_yMO) forcing, excellent accuracy is achieved not only for the primary mode of interest (1F) but also its sister backward mode (1B). Vertical forcing provides a slightly more accurate damping estimate for the 1F mode because the mode shape is more vertically oriented for this particular machine. Such performance has given investigators the confidence to use single input excitation on industrial machines [9].

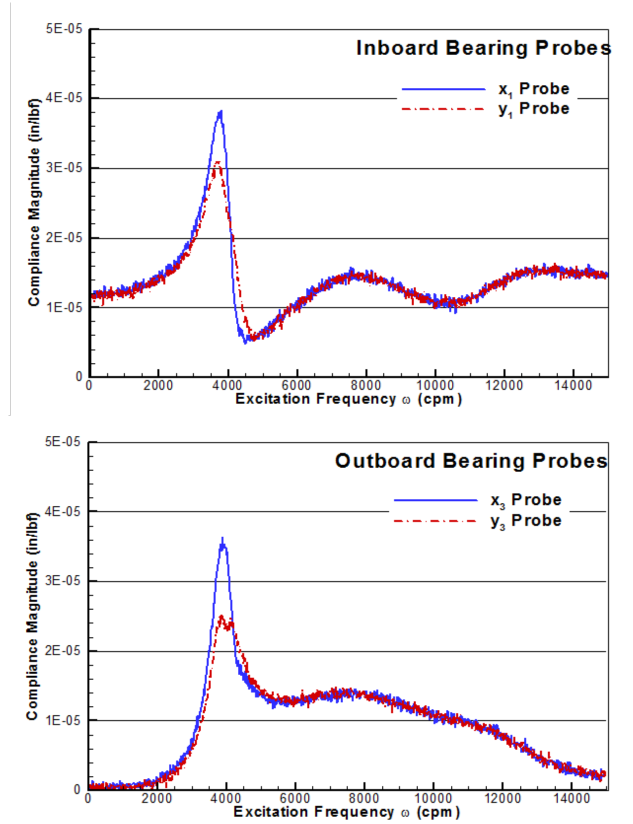

Figure 18) Simulated Rotor FRF Measurements from Excitation in only Horizontal Direction

Mode	1F		1B	
Parameter	ζ (%)	ω_d (cpm)	ζ (%)	ω_d (cpm)
Actual	4.62	4110	8.56	3809
SI_xMO	4.54	4111	8.39	3805
SI_yMO	4.62	4111	9.02	3799

Table 2) Modal Parameters Obtained from SIMO FRF Measurements of a Simulated Rotor System [33]

As mentioned earlier, it is a popular choice to excite the rotor in a forward circular direction. However, it is often not recognized that exciting in a forward circular or any elliptical fashion essentially means that a multiple input, multiple outputs (MIMO) test is being conducted. Whenever the excitation is not along a single axis, but takes any form of elliptical shape, two forcing inputs are required creating a multiple input situation. When multiple input testing is preferred, such as when using forward circular excitation, MDOF frequency domain estimation techniques cannot be applied unless the measured FRFs are calculated from force and response data from two MIMO tests: one with forward circular input, and the other backward circular input. Ewins [34], Maia [32] and Pintelon [35] provide details on the practical and theoretical aspects of such multiple input testing.

SVT Measurement Process

Fair correlation with the predictions requires careful attention during the measurement process. This means that the operating parameters of the machine (speed, inlet pressure, etc.) must be held within that assumed for the modeling predictions. Oil supply temperature has proven to be an operating parameter to closely monitor and control especially when trying to reconcile the difference between measurements and predictions [9]. Without proper control of the bias currents, an electromagnetic shaker can impose a static force on the shaft. Therefore, the operating position of the journal bearings should be also closely monitored. A significant shift in journal operating position when the shaker is energized will alter the bearings' stiffness and damping characteristics, and thus, the measured stability of the machine.

When a frequency domain estimation technique is employed, accurate measurement of the FRFs is vital to the success of the SVT. Often overlooked by the turbomachinery industry, there are two recommended practices originating from the modal testing community that should be adhered to with respect to these FRF measurements:

- Calculation of any frequency response function requires measurement of the response outputs' *and* the forcing inputs' magnitude and phase.
- The coherence of each FRF should be examined to ensure the quality of the measurement around the mode of interest's frequency.

It is often assumed that, once the current amplitudes are set in its control system, an electromagnetic shaker will produce a force of constant amplitude and phase across the whole frequency range. However, like other exciter devices where "force drop-out" can occur, electromagnetic shakers can be susceptible to irregularities in the amount of force that is actually applied to the rotor when traversing a natural frequency. A highly amplified natural frequency can create large vibration amplitudes at the shaker, resulting in significant differences between the actual currents being applied to the shaker and those specified by the control system.

Therefore, like any force transducer used in modal testing, it is recommended that the electromagnetic shaker be calibrated to determine the relationship between the measured parameters (such as flux or coil currents and rotor displacements) and the forces applied to the rotor. Without measurement of the shaker's force during testing, the FRFs' amplitudes at the mode of interest could be distorted, resulting in poor stability estimates.

SVT Rotordynamic Modeling

In addition to the issues discussed for URVT, there are several additional modeling aspects that needed to be considered for stability verification testing. As mentioned earlier, rotordynamic calculations should be conducted during the shaker's design process to examine its impact on the machine's behavior. This examination should include not only the impact on the mode of interest's stability and frequency, but also the impact on other critical speeds and their separation margins.

When tilting pad journal bearings are employed, it is heavily debated whether the bearings should be represented using their full coefficients which allow whirl frequency dependent stiffness and damping characteristics, or using the traditional, whirl frequency independent model of synchronously reduced coefficients [11,12]. Current API standards require the use of synchronously reduced coefficients for predictions. However, it is recommended that measurements should be correlated with predictions using both representations to help shed light on this ongoing debate. Comparing the correlations of the two representations is especially important for base stability testing when using bearings with center pivots and low (<0.4) pad preloads.

If an SVT is conducted as part of a performance test (*e.g.* PTC 10 type I or II), the measurements should be correlated with API Level II stability predictions that include the dynamics created by the machine's internal seals and other components. Each manufacturer has its own methodology for how these internal dynamics are modeled and analyzed. As required by API Level II stability analysis procedures, this methodology should be explained and documented by the manufacturer.

SVT Deliverables

- For each test condition, the predicted range of stability (log decrement and damped natural frequency) for the machine with, and without, the presence of the shaker device.
- Description of the damping ratio estimation technique employed
- Measured operating data for each stability test condition, such as:
  - Speed
  - Inlet and discharge pressure and temperatures
  - Molecular weight
  - Oil supply pressure, temperature, and flow rate
- When stability estimates are obtained from outputs' time transient data,
  - Sampling frequency
  - Number of transients events recorded
  - For each output location and operating condition, plot showing the measured time transient data
- When stability estimates are obtained from frequency response functions,
  - Description and records of the calibration of the input force measurement
  - Number of averages taken, window type and overlap percentage
  - Frequency resolution
  - For each input/output location and operating condition,
    - Plot showing the final measured FRF (magnitude and phase) and the identified FRF from the estimation technique

- Plot showing the final measured coherence associated with the measured FRF

- Comparison of the measured stability (log decrement and damped natural frequency) versus the predicted range for each tested operating condition
- Depending on the specifics of the acceptance criteria, the resulting stability predictions from the corrected analytical model should presented to determine acceptability of the design.

SVT Requirements

Currently, there are no industrial standards in place for guidance on what acceptance criteria should be applied for this type of design verification testing. Individual OEMs and end-users are developing their own acceptance criteria in the meantime. It is recommended that the criteria should have a similar two-step evaluation process as that standardized for the URVT:

1. How well does the original rotordynamic model and analysis predict the measured stability results?
2. If the model has poor accuracy with respect to the measured stability results, does the machine still have acceptable rotordynamic performance over the full range of design/operation *after* its model has been corrected based on the test results?

When assessing the accuracy the stability measurements and predictions, there is a key difference with the correlations done during an URVT: the measured vibrational response should not be under-predicted for the URVT, while the measured log decrement should not be over-predicted in the SVT.

The exact methodology used to correct the model should be agreed upon, prior to testing, by the purchaser and OEM. Once again, this is another area where manufacturers and end-users are developing their own methodologies. Pettinato *et al.* [9] applied two methods to correct the model and determine acceptability of a particular centrifugal compressor design. One method applied a bias shift using the base stability measurements, while the second method applied a slope correction based on measurements that included aerodynamic excitations within the machine.

SVT Knowledge / Verification

Analogous to the knowledge obtained from the URVT, the stability verification test assesses the rotordynamic predictions' accuracy to help verify a design's stability characteristics beyond the pass/fail nature of the vibration demonstration tests. No nonsynchronous vibrations may be observed during a full load, full pressure test. However, the machine's stability level (log decrement) and margin away from instability remain in question. For the selected test conditions, the SVT yields some insights by providing a measurement of the stability level and, using this measurement in conjunction with the predictions, an estimate of the machine's stability threshold.

Since we cannot test every situation, we must rely on the accuracy of the model to design for these other situations. The SVT provides at least some verification concerning the reliability of the model's stability predictions.

CONCLUSIONS

Rotordynamic testing is an effective and efficient tool, when applied appropriately, to mitigate machinery risks. The tutorial presented the basis for determining the machinery application risks. Test options available to address those risks, test procedures, preparation for the test, knowledge gained from each test and the testing benefits were discussed. The difference between vibration demonstration and design verification testing was highlighted. The principle difference being that vibration demonstration tests assess the acceptability of rotordynamic behavior in a pass/fail mode while design verification testing is used to confirm the rotordynamic predictions. This verification provides confidence in extrapolating the design (by the vendor) and operation (by the purchaser) beyond the machine's as-built and specific shop test conditions. The recommended practices of performing the test options for vibration demonstration and verification testing concluded the tutorial.

NOMENCLATURE

Variables

δ	= Log decrement (dim)
ζ	= Damping ratio (%)
ω_d	= Damped natural frequency
$\overrightarrow{U_1}$	= Baseline unbalance (M-L)
$\overrightarrow{U_2}$	= Unbalance with verification weight (M-L)
$\overrightarrow{U_r}$	= Residual unbalance (M-L)
$\overrightarrow{U_v}$	= Verification weight unbalance (M-L)
$\overrightarrow{V_1}$	= Measured baseline vibration (L, L/T or L/T^2)
$\overrightarrow{V_2}$	= Measured vibration with verification weight (L, L/T or L/T^2)
$\overrightarrow{V_s}$	= Vibration due to verification weight alone (L, L/T or L/T^2)

Acronyms

1B	= First backward rotor mode
1F	= First forward rotor mode
AF	= Amplification factor
FEED	= Front-end engineering design
FLFP	= Full load, full pressure
FRF	= Frequency response function
MCS	= Maximum continuous speed
MDOF	= Multiple degree of freedom
MIMO	= Multiple input, multiple output
MOBAR	= Multiple output backward autoregression
MRT	= Mechanical run test
OEM	= Original equipment manufacturer
OMA	= Operational modal analysis
PEM	= Prediction error method
PTC	= Power Test Code (ASME)
SDOF	= Single degree of freedom
SIMO	= Single input, multiple output
SVT	= Stability verification test
URVT	= Unbalance response verification test

REFERENCES

1. C. Jackson, 1998, "Shop Testing – Is It Worth It?," *Orbit Magazine*, June, pp. 10-15.
2. J. Heagney, 2011, *Fundamentals of Project Management*, AMACON.
3. C. Carlson, 2012, *Effective FMEA's: Achieving Safe, Reliable, and Economical Products and Processes Using Failure Mode and Effects Analysis*, John Wiley & Sons, Inc.
4. K. E. Atkins and R. X. Perez, 1992, "Assessing Rotor Stability Using Practical Test Procedures," *Proceedings of the Twenty-First Turbomachinery Symposium*, Texas A&M University, pp. 151-158.
5. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 2, Paragraph 6.3.1.
6. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraph 6.3.3.2.
7. *Performance Test Code on Compressors and Expanders*, ASME, PTC 10-1997, Paragraph 3.2.1, p. 11.
8. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraph 4.8.3 & 4.8.4.
9. B. C. Pettinato, C. H. Cloud and R. S. Campos, 2010, "Shop Acceptance Testing of Compressor Rotordynamic Stability and Theoretical Correlation," *Proceedings of the 39th Turbomachinery Symposium*, Texas A&M University, pp. 31-42.
10. *API Standard Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals and Rotor Balancing*, API Recommended Practice 684, 2nd Edition, August 2005, Paragraph 2.3.8.
11. J. A. Kocur, J. C. Nicholas and C. C. Lee, 2007, "Surveying Tilting Pad Journal Bearing and Gas Labyrinth Seal Coefficients and Their Effects on Rotor Stability," *Proceedings of the Thirty-Sixth Turbomachinery Symposium*, Texas A&M University, pp. 1-10.
12. C. H. Cloud, E. H. Maslen and L. E. Barrett, 2012, "Rotor Stability Estimation with Competing Tilting Pad Bearing Models," *Mechanical Systems and Signal Processing*, Vol. 29, May, pp. 90-106.
13. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraph 4.8.8.8.
14. *API Standard Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals and Rotor Balancing*, API Recommended Practice 684 2nd Edition, August 2005, Sec 2.7.3, pp. 2-63 to 2-68 and Sec. 3.5.2, pp. 3-38 to 3-43.
15. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraph 4.8.2.9.
16. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraph 4.8.2.10.
17. M. A. Miranda and R. F. de Noronha, 2007, "ASME PTC 10 Modified Test for Mechanical Assessment of Centrifugal Compressors," *Proceedings of the Thirty-Sixth Turbomachinery Symposium*, Texas A&M University, pp. 87-93.
18. *Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services*, API Standard 617 8th Edition, September 2014, Part 1, Paragraphs 4.8.5 & 4.8.6.
19. J. C. Nicholas, S. L. Edney, J. A. Kocur and J. F. Hustak, 1997, "Subtracting Residual Unbalance for Improved Test Stand Vibration Correlation," *Proceedings of the Twenty-Sixth Turbomachinery Symposium*, Texas A&M University, pp. 7-18.
20. *API Standard Paragraphs Rotordynamic Tutorial: Lateral Critical Speeds, Unbalance Response, Stability, Train Torsionals and Rotor Balancing*, API Recommended Practice 684 2nd Edition, August 2005, Sec 2.9, pp. 2-97 to 2-104.
21. Nordmann, R., 1982, "Modal parameter identification and sensitivity analysis in rotating machinery," *Proceedings of Conference on Rotordynamic Problems in Power Plants*, IFToMM, pp. 95-102.
22. Nordmann, R., 1984, "Identification of modal parameters of an elastic rotor with oil film bearings," *ASME Journal of Vibration, Acoustics, Stress , and Reliability in Design*, Vol. 106, pp. 107-112.
23. Cloud, C. H., Maslen, E. H., and Barrett, L. E., 2009, "Damping ratio estimation techniques for rotordynamic stability measurements," *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 131, No. 1, January, pp. 012504.
24. Newkirk, B. L., 1924, "Shaft Whipping," *General Electric Review*, Vol. 27, March, pp. 169-178.
25. Newkirk, B. L. and Lewis, J. F., 1956, "Oil film whirl—An investigation of disturbances due to oil films in journal bearings," *Transactions of the ASME*, Vol. 78, pp. 21-27.
26. Hagg, A. C., 1946, "The influence of oil-film journal bearings on the stability of rotating machines," *ASME Transactions, Journal of Applied Mechanics*, Vol. 68, pp. A211-A220.

27. Kanki, H., Fujii, H., Hizume, A., Ichimura, T., and Yamamoto, T., 1986, "Solving nonsynchronous vibration problems of large rotating machineries by exciting test in actual operating condition," *Proceedings of the International Conference on Rotordynamics*, IFToMM, pp. 221-225.

28. Baumann, U., 1999, "Rotordynamic stability tests on high-pressure radial compressors," *Proceedings of the Twenty-Eighth Turbomachinery Symposium*, Texas A&M University, pp. 115-122.

29. Moore, J. J., Walker, S. T., and Kuzdzal, M. J., 2002, "Rotordynamic stability measurement during full-load full-pressure testing of a 6000 psi re-injection centrifugal compressor," *Proceedings of the Thirty-First Turbomachinery Symposium*, Texas A&M University, pp. 29-38.

30. Bidaut, Y., Baumann, U., and Al-Harthy, S. M. H., 2009, "Rotordynamic stability of a 9500 psi reinjection centrifugal compressor equipped with a hole pattern seal—measurement versus prediction taking into account the operational boundary conditions," *Proceedings of the Thirty-Eighth Turbomachinery Symposium*, Texas A&M University, pp. 251-259.

31. Takahashi, N., Magara, Y., Narita, M., and Miura, H., 2012, "Rotordynamic evaluation of centrifugal compressor using electromagnetic exciter," *ASME Journal of Engineering for Gas Turbines and Power*, Vol. 134, pp. 032505.

32. Maia, N. M. M. and Silva, J. M. M., eds., 1997, *Theoretical and Experimental Modal Analysis*, Research Studies Press Ltd..

33. Cloud, C. H., 2007, *Stability of Rotors Supported by Tilting Pad Journal Bearings*, PhD dissertation, University of Virginia.

34. Ewins, D. J., 2000, *Modal Testing: Theory, Practice and Application*, Research Studies Press Ltd., 2nd Edition.

35. Pintelon, R. and Schoukens, J., 2001, *System Identification: A Frequency Domain Approach*, IEEE Press.

36. de Jongh, F., 2008, "The Synchronous Rotor Instability Phenomenon – Morton Effect," *Proceedings of the Thirty-Seventh Turbomachinery Symposium*, Texas A&M University, pp. 159-167.

37. Guglielmo, A., Mitaritonna, N., Catanzaro, M. and Libraschi, M., 2014, "Full Load Stability Test on LNG Compressor", *Proceedings of the ASME Turbo Expo 2014*, Dusseldorf, Germany, No. GT2014-25353.

ACKNOWLEDGEMENTS

The authors wish to thank ExxonMobil for its' support and Jim Byrne, Minhui He, Eric Maslen and José Vázquez from BRG for their helpful suggestions.