Interpretation of Dynamic Data Plots for Troubleshooting and Resolving Vibration in Large Rotating Machinery

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Abstract. Vibration in large turbine-generator rotor trains can be diagnosed by recognizing the specific correlated patterns visible in data presented in Bode, polar (Nyquist), shaft centerline, shaft orbit, and frequency spectrum plots, among others. Properly interpreting these data plots requires the recognition of true rotor behavior when affected by various faults. By the authors' experience, the most prevalent faults seen in practice are the direct and indirect effects of distributed mass eccentricity within individual rotors or across an assembled rotor train. The presence of mass eccentricity leads to various effects of inertia during machine operation (often torque or load dependent), which create the principal graphical data signatures to look for and utilize in diagnostic assessment. The graphical signatures that result from the presence of imposed forces "fighting against" a rotor's natural inertia are readily and independently recognizable, even when in combination with additional contributors of dynamic vibration such as that caused by resonant responses within critical speed regions, rubs, and bearing/fluid instability. This paper describes and presents typical data signatures in the various types of plots, including examples of shaft bows or other unresolved distributed mass eccentricity on individual rotors, plus examples of rotor misalignment on assembled multi-rotor machines caused from bent or off-square coupling faces and radially offset, misaligned bearings. Furthermore, understanding the most prevalent root causes underlying the diagnostics leads to the recognition of proactive shop strategies to prevent these faults and to substantially improve the probability of a successful startup following a turbine or generator rotor service outage.

Keywords: Diagnostics, Eccentricity, Inertia, Troubleshooting, Vibration.

1 Introduction

When troubleshooting dynamic vibration problems on large turbine-generator units, a diagnostician's primary source of insight into machine behavior comes from collected sensor data. Real-time dynamic motion is indicated by proximity probes on the rotor journals and seismic probes on the bearings/pedestals. While these data signals are crucial to observe in real time to immediately warn of the presence of a vibration problem, the raw amplitude and phase data is more useful for diagnostic purposes only after the data is converted and suitably presented in standard graphical formats. There are a few key rules of interpretation based on the recognition of some fundamental principles of physics. To make the most sense of any data plots, it is crucial to first recognize the primary root cause(s) of rotor vibration, and their effects on the dynamic behavior of rotors in a rotor train.

1.1 The Importance of Mass Eccentricity and Symmetry

From troubleshooting hundreds of real-life cases of excessive rotor vibration, the most commonly found root cause is unresolved mass eccentricity that was unknowingly introduced into the turbine-generator rotor train. Mass eccentricity should be looked at from the point of view of rotor symmetry, with respect to recognizing natural behavior following the conservation of angular momentum. The fundamental point to recognize is that every object will naturally and freely tend to spin about its true center of mass (centroidal rotation) upon the application of any torque. This is driven by the inertia of the particles comprising the object upon initiating torque-driven circular motion. The only way to prevent this natural centroidal rotation is when some form of constraint force is imposed on the rotating object to hold it in some forced, non-centroidal rotation, which forces the true center of mass to laterally translate or whirl around the forced axis of rotation.

As a result of this imposed rotational asymmetry, a portion of the total energy provided from input torque to accelerate the rotor mass is converted into what is ultimately observed as "vibration" (more accurately, the radial deflection and lateral translation of the journal axis of the rotor). This energy is manifested in the centrifugal force generated by the inertia of any portion of rotating mass that is net-asymmetric, with respect to the forced axis of rotation/spin. This force is what is typically recognized as "unbalance" during rotation. In combination with all centrifugal force is the reaction force(s) from the point(s) of constraint maintaining this forced manner of rotation.

With regard to turbine-generator rotor vibration, a great majority of vibration problems arise when the intended designed and installed position and orientation of the rotor (centered concentrically about its journals and couplings) does not match where the rotor naturally wants to go when it is accelerated by torque, given the presence of mass eccentricity (either inherent to a rotor or induced by misalignment to other rotors). The amount of parasitic force (or power, when taken over time) acting to drive the rotor vibration is directly related to the amount of force required to hold the rotor in its intended operating position when it does not naturally tend toward doing so

(Newton's third law of action-reaction). When mass eccentricity is present, the physical constraints that maintain a horizontal spinning rotor in its intended but "unnatural" state are the journals held by gravity, and the bolted couplings (and connected mass) of adjacent rotors. The power lost into these constraints is "stolen" from the energy in the input torque and converted into unwanted points of dissipation, namely material stress, vibration of support structures, and into potential energy deflecting/bending the "spring" of the rotor as its mean mass axis laterally translates (whirls) about the journal axis, particularly at the system's fundamental harmonic frequency mode, within the first critical speed region.

It is important to note that the goal of resolving rotor vibration should center on restoring the natural tendency of rotation (about the center of mass axis) to coincide with the designed, concentric geometric rotational axis centered at the rotor's journals/couplings, to create the natural dynamic behavior of an ideally concentric rotor. In so doing, most other dynamic problems (rotor resonant responses, bearing instabilities, etc.) also disappear, as they will no longer have a driving force.

In the more traditional rotordynamic perspective, the natural assumption is that a rotor is always rotating about its journal axis at any speed, and any mass asymmetry that creates vibration is to be counted as unbalance relative to this axis of rotation. As such, most measured excessive displacement amplitudes are considered as targets to be resolved with forces from balancing weights. This can create problems when the actual source of mass asymmetry in a continuous rotor or rotor train comes from misalignment, off-square couplings and rotor bows, or when the unwanted vibration response is being observed above the first system critical speed. Additionally, placing weights does not necessarily allow achieving a true "balanced" state, defined as the sum of all forces ***and*** the sum of all moments being zero. When a true balanced condition is not achieved, residual axial moments act to bend or distort the rotor, create residual motion and forces, and often create a modified "natural" self-alignment orientation (within bearing and seal clearances) that still does not correspond to the journal axis of the rotor. These rotor conditions are identifiable within vibration data plots.

1.2 What is Really Happening Within the Rotor Train?

A critical factor in any diagnostics connected to mass eccentricity is to define whether the observed data is related to the speed region below or above the peak of the fundamental system harmonic frequency or resonance (first critical speed). As a rotor is accelerated under torque, a second, crucial effect to recognize is the natural switch in the center of rotation as the rotor passes its fundamental system critical speed region. Perhaps a more descriptively appropriate term than "center of rotation" is "center of lateral translation", or "center of whirling orbit". In plotted sensor data shown in X-Y axes, this refers to the shaft centerline position as seen in a radial measuring plane. Below the first system critical speed, the center of rotation of an eccentric (horizontal) rotor is governed and maintained by the force of gravity, which forces rotation about the rotor journals as they are held against the bearings. Above the peak of the critical speed, however, the forces generated by the inertia of any eccentric mass become large enough to overcome the forced, non-centroidal rotation, and instead the rotor begins to rotate (in a whirling or orbiting motion) about its actual center of mass axis. This corresponds to what is typically referred to as rotor self-balancing at the end of the critical speed frequency region (where the measured phase angle has shifted 180 degrees).

An easier way to recognize this behavior is to look at a vertical machine, in which the longitudinal spring constant of the rotor is essentially zero (no effect of gravity). Therefore, there is no active constraint acting on the rotor, and the rotor self-balances and self-aligns to its mean mass axis immediately upon the first application of torque to accelerate the rotor from a state of rest, and there is no first critical response seen (e.g. tops, vertical centrifuges, hydro turbines, etc.). By contrast, in a horizontal machine, gravity creates a constraint on the rotor(s), holding the journals against the bearings. The effective stiffness is that of the rotor-bearing system, including the longitudinal stiffness based on the material and geometry of the rotor in series with the springs of the bearings and pedestals. Both vertical and horizontal machines ultimately follow the same behavior, but horizontal rotors must progress through a more convoluted interaction of forces acting on the rotor body "spring", before reaching a self-balancing and self-centering condition. All examples in this paper focus solely on horizontal machines.

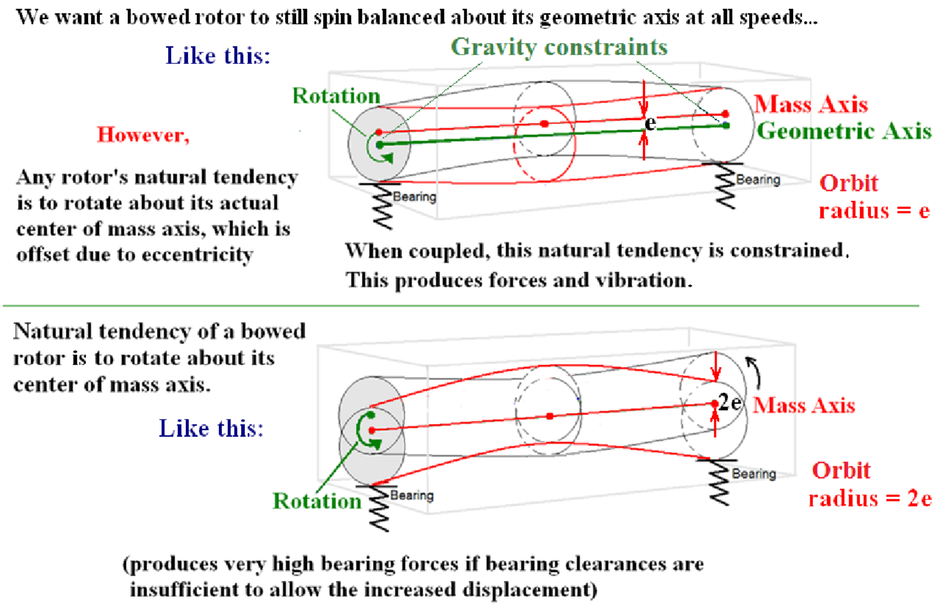

Fig.1. The natural behavior tendency of bowed rotors (top) and (bottom) the first critical speed.

Also of note, the previously described behavior applies both to an individual rotor and an assembled rotor train as a whole. A multi-rotor train can have eccentricity induced through numerous additional sources: bowed/eccentric individual rotor(s), off-square or bent couplings that were forced/pulled together, a “crank” in the rotor train from an off-center journal or coupling, or vertical and/or horizontal misalignment in bearing positions. By recognizing the natural tendency of rotation of an eccentric/bowed rotor, and the tendency to switch its center of "whirling" rotation after passing the fundamental critical speed region, various patterns in plotted vibration data can be identified, as these potential sources of axially distributed eccentricities provide unique data signatures. In practice, overlooking certain unique data signatures can lead to various problems due to misinterpretation of data, leading to unsuitable proposed solutions.

In addition to or in combination with the dynamic effects from mass eccentricities on a rotor driven (spinning) in circular motion are sometimes seen the more familiar contributors of dynamic vibration, such as resonant responses within critical speed regions, subsynchronous whirling, subsynchronous bearing/fluid-related instability, and rubs. These contributors reveal themselves independently within the data plots, and are most often secondary effects to the fundamental root cause of axially distributed mass eccentricities and inertia.

The forces produced in a rotor's tendency to achieve a "natural" state of rotation about its mean mass axis create the dynamic motion commonly referred to as rotor "rigid modes", comprising the lateral/translational mode and the rocking/pivot mode. In the pure lateral rigid mode, the mass axis is parallel to the rotor but radially offset, and precesses around the axis of rotation. In the pivot mode, the mass eccentricities are axially asymmetric, causing the mass axis to be axially skewed to the axis of rotation. These rigid modes should not be confused with the correlated "first critical" and "second critical" resonant modal responses. The resonant responses (and phase angle shifts) provide ready clues about the amount and location of mass eccentricity, though the total measured rotor response is most often a combination of both rigid mode responses (seen across all speeds) and modal resonant responses (seen in the critical speed regions).

For any resonant modal response to appear in a constrained continuous body, there must be an excitation force applied, and excitation forces arise under the condition of forced, non-centroidal rotation. Resonant excitation of a rotor should be viewed more in the manner akin to the motion of a plucked guitar string (on an otherwise unconstrained rotor free in space), and should be identified separately from whirling/orbiting that is induced from asymmetric inertia on a constrained eccentric rotor. In measured rotor vibration data, a resonant response will always be accompanied by a phase angle shift, whether with or without a corresponding peak in displacement amplitude. If a rotor is ideally concentric, it will pass through its fundamental critical speed region without any notable resonant amplitude response. When a trend is observed of rotor displacement amplitude increasing with speed, but without a phase angle shift, it is not resonant excitation, but is purely whirl driven by centrifugal force from unresolved mass eccentricity (and standard balancing rules do not successfully apply).

Rubs often appear when a rotor train self-aligns and self-centers once above the first critical speed of one or more rotors, shifting the spatial orientation and alignment of the spinning rotor train, bringing rotating components into contact with stationary ones. Bearing instability likewise arises in many cases from the self-alignment tendency of a rotor to its mean mass axis above the critical speed, if for example journal loading on a bearing becomes reduced. Bearing instability can also be induced by momentum from an asymmetric tangential force (from variation of torque or pulsating torque) overcoming the stabilizing forces in the bearing fluid, forcing the particular journal into additional whirling. In the frequency domain, this may generate observable subsynchronous components. When such whirling of a rotor in hydrodynamic bearings occurs when the rotor is spinning at a frequency twice the "first critical speed", it can excite the rotor's natural resonance, and the whirling may be seen at single subsynchronous frequency with large amplitudes, better known as "oil whip".

2 Interpreting Sensor Data

To start, a brief review of the data signal types may be beneficial. It is assumed that shaft motion is being measured with proximity (eddy current) probes using a DC "gap" voltage, with an AC component superimposed proportional to the shaft motion. The DC data represents the static shaft centerline position, or the orbit center when whirling (laterally translating) during operation. This DC data presented in shaft centerline plots provides invaluable insight, especially in verifying rotor train alignment, and often proves to be the most important data assessed for diagnostic analysis. The AC signal component of each probe represents a linear measure of active sinusoidal motion within each revolution, with the measurement simulating the oscillating motion of a linear spring viewed from a stationary reference point. In reality, the rotor vibration is not oscillatory like a flexing spring, but is a lateral translation or orbital motion, while the "measured" shaft orbital motion is a mathematical creation combining the sinusoidal AC signals from two (typically) orthogonal sensors. A secondary signal from a phase reference mark on the shaft indicates the rotational velocity of the shaft, while the AC signals provide a detailed radial deflection vector of the lateral translation of the shaft. By observing the phase reference signal and deflection vector in time, a phase angle relation is obtained between the centrifugal force vector originating in the rotor (rotating frame) and the deflection vector observed from the sensor in an inertial frame. The same signals viewed in the frequency domain display the full frequency spectra of the observed kinetic motion.

When rotor motion and vibration is measured by sensors, all data is referenced to common global coordinates that correspond to stationary sensors identifying the position of the rotor (its shaft centerline, also inferred mathematically by combining the signal from two typically orthogonal sensors in a common radial plane). Any sensor can only see and measure the rotor surface, and it is up to the diagnostician to recognize the manner of rotation of a rotor at a given speed. The same apparent amplitude and phase measurement can vary greatly in interpretation (and in appropriate solution) whether the sensor is observing the rotor elastically bending/deflecting under the influence of centrifugal force, or under resonance, or whether measured "motion" represents a high spot (runout, or eccentricity) passing under the sensor, or some combination of all three. (To define, "eccentricity" is the evaluated 1x radial distance of the center-of- mass axis from the journal/rotational axis in any radial measuring plane. The maximum "runout" vector is the maximum distance from the journal/rotational axis to the highest point on the surface of the measured location.)

For diagnostics, the amplitude and phase measurements should also be correlated to bearing seismic vibration as an indication of transmitted force into the journal supports as a further indication of the amount of eccentricity present. When an eccentric rotor is held and forced in non-centroidal rotation, the resulting reactive centrifugal force will be equally present regardless of the motion of the rotor, as it represents a conversion of power from input torque into a combination of kinetic energy (dynamic rotor whirling, support vibration), potential energy (the bent "spring" of the whirling rotor) and stress and force transmitted into and absorbed by the supports. Viewing high or low vibration displacement amplitudes alone can be meaningless without correlating to the total force present, taking seismic vibration forces into account.

Furthermore, all of these interpretations vary depending on if the collected data is from rotor speeds below or above the first system critical speed. Below the first critical speed peak, all observed motion is from centrifugal forces from "unbalance" relative to the journal axis of the rotor. Above the first critical speed (in most cases, with exceptions being situations of continued constraint imposed from adjacent rotors), the observed motion is from residual centrifugal forces (and corresponding axial moments) relative to the mean mass axis of the rotor, while also interacting with a new "virtual" point of constraint at the axial center of mass of the continuous rotor.

Above the first critical, any axially asymmetric bias to the overall eccentricity distribution will create an "out-of-phase" or "couple" motion, with the pivot centered about a nodal point at this axial center of mass. In this state, each half of the rotor acts effectively as a cantilever pinned at this axial center of mass, supported with some freedom of motion at each rotor end by the oil film and bearings. In a manner similar to the rotor as a whole when below the first critical speed, at these higher speeds each half of the rotor acts as its own "rigid mode", driven by centrifugal force. When the mass axis is non-parallel and skewed relative to rotational (journal) axis, this creates the appearance of out-of-phase motion as observed by the sensors. Any increasing journal displacement amplitudes measured at super-critical speeds while maintaining a constant phase angle are indicative of an unresolved axial bias in mass eccentricities. This indicates that the inherent longitudinal mean mass axis of the rotor does not pass through the axial center of gravity of the "rigid" rotor.

(As an important aside, it should be noted that balancing weights calculated or placed based on data collected at super-critical speeds (commonly a weight-couple) will act to bend the ends of a flexible rotor into alignment with its mean mass axis, distorting the intrinsic shape of the rotor and introducing internal cyclic bending moments, while not resolving the unwanted tendency of the rotor to spatially self-align according to its eccentricity distribution. Instead, all weights should be calculated using the response only up to the first critical speed peak, while the rotor is whirling in a gravity-dominated condition, and then if necessary to reduce the rotor's rocking mode, these same weights should be axially redistributed and biased while maintaining same phase angle. This will shift the mean mass axis to pass through the rotor's axial center of gravity, and ideally should bring the mass axis into full coincidence with the journal axis.

3 Constructing and Evaluating Data Plots

Note that the sample plots to be presented in this paper are initially taken from real collected data from turbine-generator rotors in power plants, but are simplified and specifically reconstructed to isolate the particular phenomena being presented and discussed. In practice, it is not at all uncommon to observe multiple "symptom signatures" in a single plot.
To best distinguish the initial root cause producing observed vibration, the raw measured vibration data should be collected from the complete speed range of the machine, starting from complete standstill, ideally with lift oil off (on cold machines). Taking a rotor off gear to standstill can be risky on hot machines, and must be well coordinated with plant operators to immediately resume turning gear operation to avoid rotor bowing. The combined DC data points from standstill, to lift-oil on, to turning gear should be reviewed on a shaft centerline plot. At standstill, it can be assumed that the rotor journals all rest on the bottom center of their respective bearings. When the rotor train is lifted on oil, any misalignment present may be initially seen as the “spring” of the rotor train self-straightens. Note, this misalignment could arise from misaligned bearing positions supporting a “naturally straight” rotor train, or from an angularly misaligned rotor train placed with proper bearing alignment (such as when off-square couplings are pulled and bolted), or both. Next, when turning gear torque is applied, the inertia of the mass of the rotors will tend to further self-straighten the rotor train, revealing any horizontal bearing misalignment as a sideways shift in the shaft centerline plot [Plot 1]. Because large rotors are generally flexible, even after a self-straightening shift is observed upon coming on gear, if the bearing misalignment is large enough there often remains a residual static bow in the rotor train. This provides a source of eccentricity, the affects of which can be seen in all other data plots at higher speeds.

If an individual rotor is highly bowed and sufficiently rigid, the journal path may trace a circle or flat ellipse on the shaft centerline plot even at turning gear speed [Plot 2]. For this reason (among others), it is very strongly recommended to ***not*** use runout subtraction when performing vibration data analysis, or even for shop rotor balancing. Runout vector subtraction was intended to "clean" the dynamic signals from journal surface irregularities, but if the runout vector is an indication of eccentricity, it serves to remove the best indication of mass eccentricity, which is often the root cause of an observed vibration problem.

In a Bode plot, the most important identifier of the presence of unresolved mass eccentricity is the pattern of amplitude slope with proportion to speed, in combination with the presence or lack of radial phase angle shift. If the amplitude at low speed is already non-zero, this is indicative of an inherent rotor bow or mass eccentricity, and is also often proportional to the rigidity of the rotor. A more flexible bowed rotor may absorb the “wobble” from a bow into internal cyclic bending, leaving the motion at the journals reduced, though sometimes with an amplified horizontal motion. This is particularly observable on two-pole generator rotors, and is commonly referred to as a "secondary critical" or “gravity critical”.

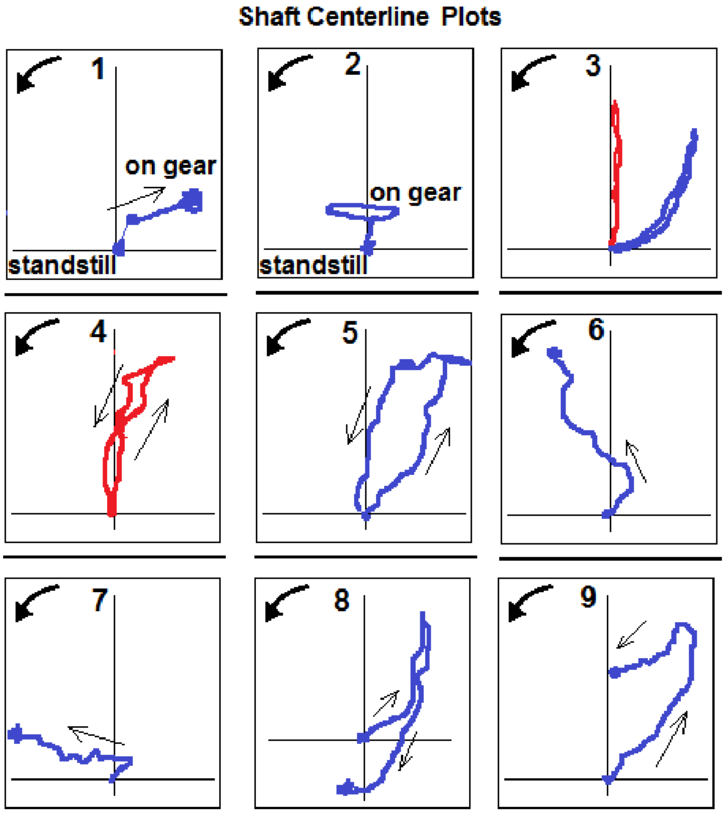

Fig.2. Representation of commonly seen shaft centerline plots, including common faults. The two paths in Plot 3 represent the "ideal" for tilting-pad and elliptical bearings, respectively.

As a flexible rotor train is accelerated, the rate of radial deflection amplitude increase, in conjunction with the phase angle change indicates how much of the observed vibration amplitude is from a static bow, and how much is from additional induced elastic bending or from resonant excitation. In the ideal case, the amplitude should remain flat or with a very small response through the full speed range, while passing through a phase angle shift at any critical speeds [Plot 19]. A non-zero amplitude beginning from low speed and remaining flat, and without any phase angle shift suggests some rotor runout or a bow is present, or there is a possible “crank” in the overall rotor train due to an eccentric journal or bent or non-concentric coupling [Plot 20, 22].

An initial increase in amplitude without notable change in phase suggests a pure rotor bow or eccentric mass that is being driven into increasing whirling by centrifugal force, below the start of the first critical speed region [Plot 24]. This is the condition where the rotor is being held in forced non-centroidal rotation. If the amplitude continues to grow while the phase angle begins to shift, this indicates a dynamic/resonant component being added to the whirling motion [Plot 23]. If the total phase angle shift is notably less than 180 degrees, this indicates that the static runout and whirl is dominant over any resonant elastic bending [Plot 23]. If the phase angle shifts through 180 degrees with minimal change in displacement amplitude, it suggests that the particular rotor is well-balanced, or that the phase angle shift is being dictated by a neighboring rotor passing through its critical speed region [Plot 25, lower speed portion]. Likewise, a Bode plot "hump" in displacement amplitude without a phase shift can also come from the response of a neighboring rotor. Such situations are readily assessed by reviewing the particular speed range of amplitude and phase responses on all rotors' Bode plots. In cases like this it is very helpful to a diagnostician to model the rotors by rotordynamics-specific FEM for identification of each rotor's inherent fundamental resonance.

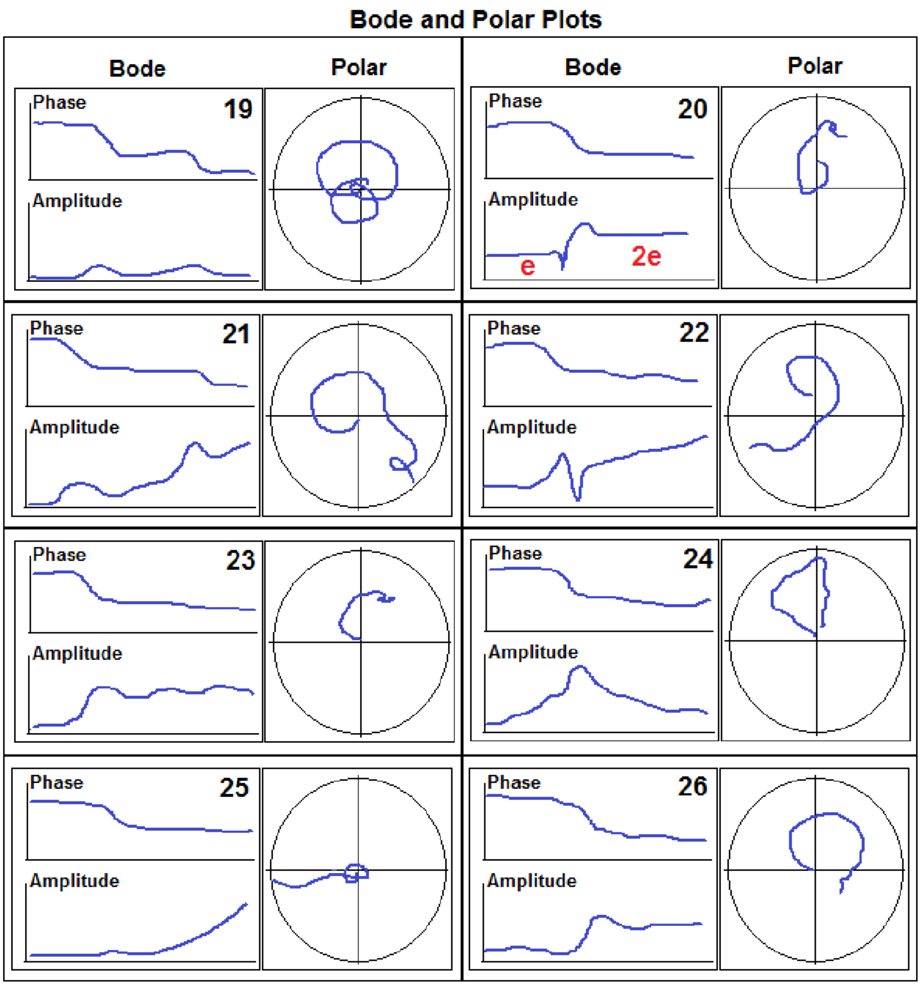

Fig.3. A representation of commonly seen Bode and Polar plots indicative of mass eccentricity. Plot 19 represents the idealized Bode and polar plot for a rotor with minimal eccentricity.

At speeds above the first critical speed, an up-sloping amplitude trend without any notable phase angle shift is an immediate indication of axial asymmetry in eccentricity distribution generating the rocking/pivot mode [Plot 21, 25]. Sometimes an added amplitude response with a phase angle shift is seen superimposed on the up-sloping trend, which is indicative of a resonant response at the second critical speed [Plot 21].

Another common indication of eccentricity is when the amplitude above the first critical speed region is roughly double the measure below the first critical, or simply remains elevated but somewhat flat [Plots 20, 22, 23, 26]. This is a function of the rotor switching its rotation (whirling) axis through the critical speed region. At lower speeds, the sensors measure the direct amount of eccentricity of the mass axis rotating around the journal axis, indicating amplitude equal to the eccentricity, e. When the rotor is sufficiently flexible, a reduced journal amplitude may be measured as the rotor may primarily "wobble" or flex at its midplane, absorbing most of the corresponding force. At higher speeds, the sensors see the "whole rotor" whirling about the mass axis while still spinning synchronously, such that the sensors measure the intrinsic runout as well as the measure of eccentricity, indicating an amplitude of 2e.

There is sometimes also seen a brief dip in amplitude before or after the critical speed peak. This dip corresponds to a brief moment of rotor self-balancing in transition between rotation axes. If before the critical, it indicates that the resonant response is dominant [Plot 20], and if it is seen after the critical, it indicates that the eccentricity and centrifugal force driven rigid mode is dominant [Plot 22].

In a corresponding Polar plot, the most important identifier of the presence of unresolved mass eccentricity is the presence of any straight-line section of data in the plot. A straight line in a Polar plot can occur across changing speed or at constant speed, and can be at a constant phase angle (pointing "outward), or appear across phase angles. A change in amplitude at constant speed and constant phase means that eccentricity is growing (likely a bow, and possibly thermally sensitive), increasing measured amplitude [Plot 21, 22, 25]. When the straight line is visible on a Polar plot across a phase angle range, this indicates that the overall mass axis alignment is changing, driven by the self-aligning tendency of mean mass axis [Plot 21, 22, 24, 26]. This is often verified with a concurrent shift in the shaft centerline position at one or more journals. Additionally, in a multi-rotor train, a rotor with a greater mass (often the generator, or an LP turbine) will pull lighter rotors (usually at an outboard bearing, often an HP turbine or exciter) into its own line of "natural" alignment based on the eccentricity or bearing alignment of the heavy rotor. This will be visible as a straight line on a polar plot in combination with a shift in shaft centerline position.

Shaft centerline plots through the complete speed range provide the best indication of rotor train misalignment relative to the static alignment of the bearing centers. For a given direction of shaft rotation, in standard hydrodynamic bearings, the shaft centerline path should move in the same direction as rotation, as the journal rides on the oil film up the side of the bearing until reaching a "pseudo static" equilibrium elevation angle [Plot 3, blue]. For tilting-pad bearings, the path should be entirely vertical [Plot 3, red]. If unexpected horizontal movement is seen, this suggests that the bearings are out of position horizontally [Plot 4, 6, 7]. Such horizontal movement can also point to misalignment as the cause of rubs, or in severe cases, as the cause of wiped bearings [Plot 7]. Horizontal misalignment can also appear on Bode plots and Polar plots in the same manner as that of general mass eccentricity or a bow of an individual rotor. Further confirmation can be seen in Orbit plots. If the natural self-alignment of the mean mass axis a rotor train is restricted due to misaligned bearings, there will often be one or two bearings acting as a “pivot point”, absorbing most of the constraint force and corresponding with a very small, compressed, shaft orbit (and often an elevated bearing metal temperature) [Plot 16].

Shaft centerline paths also indicate the presence of mass eccentricity when hysteresis is present, with the magnitude of hysteresis often corresponding to the magnitude of eccentricity. When an eccentric rotor is accelerated with torque, it tends to self-align to its mass axis only after passing through the first critical speed region. The total rotor train will also tend to further straighten in proportion to applied torque, including through to maximum unit MW load under maximum torque. If initial misalignment is present, there can be notable added shaft centerline motion at constant speed through the load range as the rotor train mass axis continues to further self-align and straighten with increasing torque and inertia [Plot 5]. When torque is subsequently released, and particularly as the rotor freewheels down in speed with no torque, it tends to maintain the self-alignment orientation that it achieved while being driven with torque, and will often maintain this orientation until passing below the first critical speed region. This creates a visible “loop” in the shaft centerline path [Plot 5].

On some occasions, the vertical shaft centerline position can be noticeably different for an equivalent speed during startup and shutdown. This can be caused by relative thermal expansion between the sensor mounting location and the rotor shaft, or by excessive pedestal thermal expansion raising a single bearing (such as from a steam leak during operation). When the rotor appears to remain elevated after a shutdown, there are likely thermal causes creating case distortion, or excessive pedestal growth [Plot 9]. This can cause changed operating alignment, and can point to a source of observed rubs. When the rotor appears to drop vertically even below its cold starting position, this is usually due to a relative shift of the sensors being lifted upward with the case or bearing housing (or whatever component they are mounted on) due to thermal expansion [Plot 8].

If individual shaft centerline paths appear typical, but there is a very noticeable vertical discrepancy in the shaft centerline path between neighboring bearings, this can indicate vertical bearing misalignment and an improper rotor train catenary. Vertical misalignment can be verified by a combination of reviewing bearing metal temperatures (hot for a high, overloaded bearing, and low for an unloaded bearing), and orbit shapes (very flat or compressed if overloaded, or very round for unloaded [Plot 12, 13]).

If bearing misalignment is indicated, a question becomes whether the rotor couplings are within tolerances and the rotor train itself is straight with only the bearings misaligned, or whether the couplings are bent or off-square creating an angularly misaligned rotor train within properly aligned bearings (or sometimes, both problems are present). A verification of the source(s) of misalignment can be done with proper coupling gap and rim evaluation [1].Generally, an angularly misaligned rotor train with bad couplings will create a signature of whirling motion from eccentricity, and will often present elevated amplitudes on a Bode plot at lower speeds at the coupling location responsible, though the shaft centerline paths may appear “correct”, albeit with hysteresis present. In a case of purely misaligned bearing position, the Bode and Polar plots may appear “correct”, but there will be excessive or “incorrect” horizontal shaft centerline shift. In both cases, to resolve this question conclusively with dynamic data alone is very difficult. It is strongly recommended to include into the analysis a review of 16-point coupling rim and gap measurements from the installation alignment[1]. The full evaluation of coupling alignment data is a complete study of its own, and is unfortunately too involved to be detailed here. In addition, fully evaluated shop runout data is also very important to include into such analysis, especially measures of all coupling faces and rims[1].

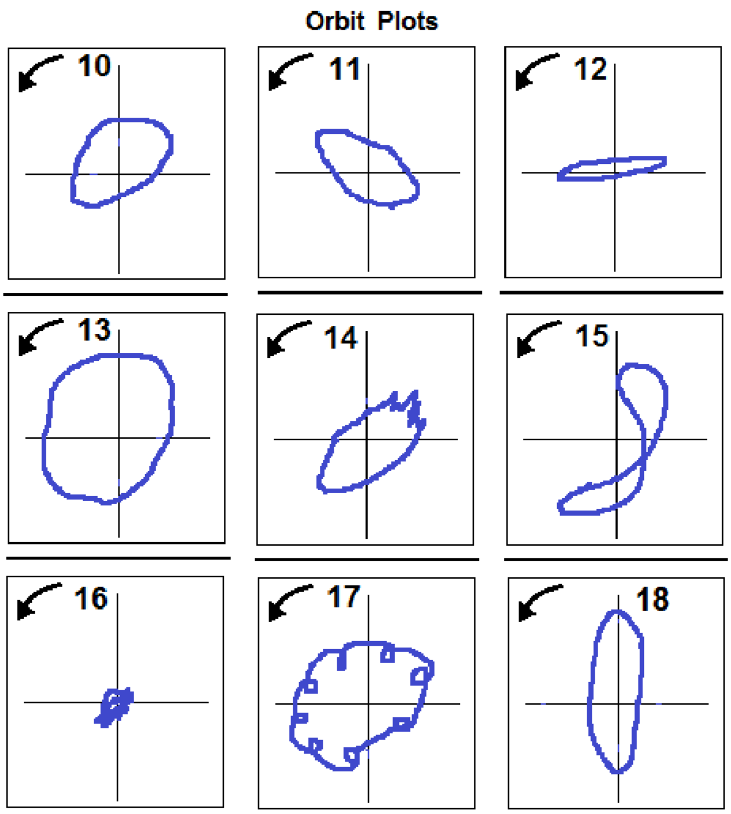

Fig.4. Representation of commonly seen shaft orbit plots, including common faults.

Shaft Orbit plots also provide valuable clues to assess the cause(s) of observed vibration and to indicate the presence of unresolved eccentricity, especially with regard to the effects of rotor train misalignment. In an ideal case with proper gravity loading in a hydrodynamic bearing, an orbit should be moderately elliptical, with the orbit inclined to match the direction of rotation [Plot 10]. When an orbit is inclined in the "wrong" direction, this is a sign that the journal is being pulled out of position by the effect of inertia, typically from an adjacent rotor [Plot 11]. This could arise from a heavier rotor governing the motion of a misaligned lighter rotor, or from self-orientation of an eccentric rotor above the first critical speed range. Vertical misalignment creating high bearing preload from the force of gravity can be indicated by a flat orbit with a high ratio of horizontal to vertical shaft deflection vectors [Plot 12]. The larger horizontal deflection vector (a direction uninfluenced by the force of gravity) is only affected by the unconstrained "spring" of the rotor. A bearing that is too low and unloaded can sometimes present a very vertical orbit, as the ratio of rotor spring stiffness and horizontal support stiffness is reduced [Plot 18].

This particular orbit shape is a precursor of the "Morton effect" when in some cases bearing contact occurs on only one specific consistent side of the journal within every revolution, resulting in localized heating at the area of contact. This steady asymmetric heating can create a thermal shaft bow over a period of minutes to hours, increasing rotor eccentricity and leading to ever-increasing whirling amplitude. During a subsequent shutdown, the corresponding shaft centerline plot will often show notable hysteresis in such cases. When thermally-sensitive bows appear, the measured displacement amplitudes during shutdown will usually be appreciably increased, especially while passing critical speed regions, creating a visible hysteresis in the corresponding Bode and Polar plots as well.

Another unusual orbit shape attribute that occasionally appears is that of multiple loops (sometimes quite subtle) corresponding in number to the number of coupling bolts [Plot 17]. This can appear in situations of angular misalignment where off-square coupling faces are pulled and bolted together, creating a static induced bow in a flexible rotor. Within each revolution, the stiffness of the coupling is rapidly increased and decreased with the passage of each bolt, resulting in a small "wobble" of variable stress in coupling bolts and across the coupling.

Shaft orbits can also indicate secondary faults that ultimately originate from the effects of eccentricity. Rubs can often be created as a side-effect of bearing misalignment under torque-driven inertial self-centering or straightening of an angularly or vertically misaligned rotor train. If a rub creates a consistent single area of impact within a shaft revolution, there is often a flat, "choppy" side visible in the plotted direct orbit [Plot 14]. When the component sensor sine wave is independently visible, there is usually a corresponding "cut-off" or choppiness of the upper or lower peaks of the sensor's waveform. A rub can also create a distorted, flat, or even afigure-8-like orbit shape if sufficient horizontal force is applied to a bearing under large self-aligning inertial forces [Plot 15].

4 Applications to Shop Practice

By recognizing the fundamental importance of mass eccentricity as a root cause of observed turbine-generator vibration and other faults causing damages, it becomes apparent that identifying and resolving eccentricity during a shop service outage is of utmost importance. The most important factor is preventing the introduction of unresolved or unknown eccentricities into an assembled rotor train in the field. There are three key steps that should be incorporated into every rotor outage plan, which when properly adhered to can prevent the great majority of potential vibration problems upon unit restart. [1]

There are two mandatory steps that should be performed in the service shop. The first is to acquire complete rotor runout (TIR) data and properly evaluate it to derive axially distributed 1xRev eccentricities, and to assure that journals and couplings comply to manufacturer machining tolerances. For the second step, if the evaluated axially distributed eccentricities are found to exceed permissible limits, then the rotor should be balanced by the appropriate balancing method. [2]

The key element in properly balancing a flexible rotor with mass eccentricity is to axially distribute balance weights to approximately mirror the inherent eccentricity distribution, thereby shifting the mean mass axis into coincidence with the designed journal/rotation axis of the rotor. In so doing, the overall balance condition must meet the criteria of the sum of all forces and sum of all moments equaling zero. In an eccentric or bowed rotor, there should be no intent to “straighten” the rotor by “unbending” or distorting it. The optimum condition for an eccentric rotor is to spin about its geometric axis at all speeds maintaining any eccentric or bowed shape it might have without distortion, even if a sensor might read this as “high amplitude”. By restoring the “natural” tendency of the rotor to spin about its journal axis as its effective center of mass, all other secondary vibration effects can be avoided.

The third key step to apply to eliminate rotor train eccentricity (though not specifically a “shop” practice) is to evaluate the 16-point coupling alignment data during rotor train assembly to ensure optimal rotor alignment. A pre-condition should have been the assurance from final shop TIR evaluation that all couplings are within proper tolerances, since the coupling gap and rim measurements are typically used to determine bearing positioning. Clearly, unknown or unmeasured defects in coupling perpendicularity and concentricity will lead to non-ideal bearing alignment, and introduce eccentricity into the rotor train. Coupling side-to-side face gaps must be maintained as close to "zero" as possible, with a maximum 0.001" tolerance. Bearings, especially those lightly loaded by gravity, must be assembled in the pedestals with "zero" vertical clearance (pinched). To help verify the alignment condition, prior to coupling the rotors, 16-point adjacent-coupling face readings (feeler gauge measurements), as well as rim readings (dial indicator), should be evaluated. It is not enough to only look at the averages, as is commonly done. For a true assessment, the individual gap readings should be compared as the rotors are rotated. If a gap remains open on a single side of the unit during a full rotation, there is likely a misalignment in bearing position. If the measured open gap itself rotates around with the rotors, then likely there is a defect in one or both coupling faces that should be considered.

5 Conclusion

There are clearly many more variants and subtleties to all the representative data plots presented here, though it is hoped that these few primary patterns can helpfully assist in recognizing the signs of unresolved mass eccentricity when diagnosing excessive rotor train vibration problems. When analyzing data plots, the first thing that an analyst must note is if the data is taken on a rotor in a balancing facility, where all eccentricities are inherent internally to the rotor, or if the rotor is assembled in an operating machine, and the effects from eccentricities could be additionally coupled from other causes external to the rotor. From the eccentricity and inertia-based perspective described here, the most basic questions to ask in diagnostics when reviewing vibration data plots are, “Where does the rotor train in its present condition want to naturally align and rotate?”, and “What is preventing the rotor train from achieving this condition?”. The majority of other vibration symptoms, including resonances and “high criticals”, rubs, bearing instabilities and high bearing temperatures originate in the rotor train being constrained to rotate in an “unnatural” manner relative to the mass eccentricities present. It is oversimplified in practice to attribute such symptoms purely to equivalent “unbalance”, and to react to such problems by balancing without deeper investigation. Even longer-term cumulative damages such as free-standing blade fatigue cracks and broken lashing wires, generator rotor shorts, insulation wear and dusting, and even the origination of off-square coupling faces (due to micro-fretting stress relief), all originate from the same root cause of unresolved mass eccentricities while operating with corresponding constraint forces. Strong diagnostic success can be had by learning to recognize the effects and displayed symptoms of the various types of distributed mass eccentricities that can be present within a rotor train.

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